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Shaft Alignment Handbook

Third Edition



John Piotrowski

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Dedication

To my wife, Bobbie Jo, who has provided the inspiration and motivation for me to write this book and who has sacrificed thousands of hours of her time to allow me the chance to learn and record this information

To my children, Tracy, Paula, and Peter who have proofread much of this information and provide me with an outsiders view of this material and

To my parents, Joseph and Magdalena, who have taught me that any honorable path will be difficult, but with hard work, discipline, and inspiration . . . anything is possible.

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Preface

Many of the material conveniences taken for granted in today's society have been made possible by the numerous rotating machinery systems located in every part of the world. Virtually everything we use or consume has somehow been produced or touched in some way by rotating equipment. The multitude of industrial facilities that generate our electricity, extract and deliver our fossil fuels, manufacture our chemicals, produce the food we eat, provide our transportation, furnish the clothing we wear, mine and refine metals, manufacture the paper and printing products used for books, require millions of pieces of rotating machinery to create all the products we use.

It makes good sense to keep the motors, pumps, gears, turbines, fans, diesels, and compressors running for long periods of time to prevent financial losses due to decreased production and overhaul costs. Repair or replacement of this equipment is expensive and the loss of revenue when this machinery is is not in operation can spell the difference between continued prosperity or financial disaster for any company. Keeping these machines running requires a thorough understanding their design and operating envelope, careful attention during their installation and overhauls, the faculty to prevent or predict imminent failures, and the expertise to modify and enhance existing hardware to extend its operating lifespan.

In the past 30 years, easily half of the rotating equipment problems I have experienced had something to do with misaligned shafts. Additionally, operating rotating equipment under misalignment conditions can be dangerous. I have seen a coupling burst apart on a 500 hp, 3600 rpm process pump that literally sheared a 10 in. pipe in half and coupling pieces landed 400 yards away from its point of origin. Keep in mind that rotor speeds above 100,000 rpm and drivers pushing 60,000+ hp are now commonplace.

With all of the rotating machinery in existence, you would think that shaft misalignment is well understood and that everyone who is involved with installing, maintaining, and operating this equipment is well versed in preventing this. When in fact, just the opposite is true. The information contained in this book is not taught in any high schools or junior colleges, is not a required course for mechanical engineers, is not discussed in business schools, and is typically not taught in trade schools for mechanics, millwrights, pipe fitters, or electricians.

Over the past 100 years, hundreds of technical books and articles have been published on rotor balancing, flexible coupling design, vibration analysis, structural dynamics, and industrial productivity. The first technical article on shaft alignment, on the other hand, was not published until after World War II and not until the late 1960s did anyone begin paying attention to this.

I feel very fortunate to have been granted the opportunity to work in a wide array of different industries and the hundreds of different plant sites after studies at the University of Pittsburgh in 1974 as a bright eyed, eager, fledgling mechanical engineer. Back then, I had no idea how many talented, patient people I would have the pleasure working with over the next 32 years. It wasn't clear, at least for the first few years working in industry, what exactly it was that I was destined to do or where I could make a contribution.

My first experience with rotating machinery came working at a automotive tire plant. The tires were cured using hot water circulated through a closed loop piping system. The water was heated to 400°F and the pumps delivered it to the supply headers at 400 psi. Once you got the system up, everything seemed to work fine (generally). The problems usually appeared when the plant was shut down every Friday night at midnight and then started back up

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Sunday night at midnight. As the circulating water was heating up, the tremendous change in temperature from ambient to 400°F wrecked havoc on the piping and the pumps, causing many seal and bearing failures on both the pumps and the motors driving them. It was not uncommon to overhaul at least one of the pump drive systems every month.

On one particular occasion, after rebuilding one of the pumps, I was working in the pump room on a control valve and a mechanic was working on finishing up the installation of the rebuilt pump. I noticed that he had a magnetic base attached to the motor shaft and a dial indicator touching the pump shaft. He would rotate the motor shaft so it would sweep around the perimeter of the pump shaft looking at what happened to the indicator as he rotated. I had never seen anyone do that before. I went over to ask if I could watch what he was doing. He glanced up and stared at me for about 30 seconds not saying a word. OK, I got the picture. I left him alone and went back to work setting up the control valve.

I was there for about three hours and every so often, would glance over to see what he was doing. He would sweep the indicator around the pump shaft, stop, loosen the motor bolts, add some shim stock, tighten the bolts, and sweep the indicator around the pump shaft again. He would then loosen the motor bolts, push the motor sideways, tighten the bolts, and sweep the indicator around the pump shaft again. He would then loosen the motor bolts, remove some shim stock, tighten the bolts, and sweep the indicator around the pump shaft again. Then he would then loosen the motor bolts, push the motor sideways in the opposite direction from the last sideways move, tighten the bolts, and sweep the indicator around the pump shaft again. After I saw this back and forth up and down iteration four times, it dawned on me that he was guessing. This went on for three hours and the level of his frustration began to show. I left for a few minutes and when I came back, he was gone, with the coupling guard in place. I have no idea if the alignment was correct. I thought to myself "Thank goodness I'll never have to do that." Six weeks later the bearings in the pump and motor failed. No one bothered to figure out what happened or why the bearings failed. It happened so often and for so long that the consensus was "that's the nature of these beasts."

In 1977, I was interviewed for a job at an ammonia plant in Ohio. They were looking for a rotating equipment engineer. I had become somewhat proficient at instrumentation and attempted to convince them that's why they needed to hire me. The engineer interviewing me indicated that they already had an instrumentation engineer and he was doing a fine job, what they really needed was a rotating equipment engineer. He asked "Do you have any experience with rotating machinery?" "I did order parts and oversee the rebuilding of some hot water circulating pumps at the plant I am currently employed at," I explained. "Well then, we would like to offer you a job," he said. "You are the eighth engineer we've interviewed and you are the first one who has some experience with rotating machinery."

I took the job, expecting to convince them that I was their new instrumentation engineer but they had other plans. I still remember my first day. My supervisor gave me a plant tour. The ammonia plant had four major compressor trains and the acrylonytrile plant had three. "There are over 1500 pieces of rotating machinery in the plant ranging from 2 to 32,000 hp and speeds ranging from 150 to 126,000 rpm and they are now your responsibility," he said to me. I got a lump in my throat and wondered what I had gotten myself into. To say the least, I was overwhelmed. Within the first three months the entire plant was shutdown twice due to rotating equipment failures one involving a 32,000 hp, 11,000 rpm steam turbine driving three barrel compressors that tripped off line due to high vibration. All of the major compressor trains had permanently installed proximity probe vibration sensors that were fed to the control room, which displayed the overall vibration levels and thrust positions. For those of you old enough to remember, vibration spectrum analyzers and laser alignment systems did not exist in 1977. We did eventually purchase a swept frequency analyzer and an oscilloscope. The alignment tooling consisted of custom Xmas tree brackets we made and

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some hand built alignment brackets for the machinery under 1000 hp. We also had an optical tilting level and a hp jig transit that no one knew how to use.

I was also tasked with the job of maintenance training. I was asked to train the maintenance personnel in rotating machinery basics, pump overhaul, mechanical seals, basic vibration, and shaft alignment. Reference material and books on these subjects were few or nonexistent. I had never taught anyone anything before in a formal classroom setting and I had no visual aids or training equipment besides an overhead projector, a screen, a flip chart, and a training room. So I began preparing overhead slides with hand drawn sketches in pen and ink. For those of you old enough to remember, personal computers and software did not exist then so you either printed words by hand or if you wanted to be fancy, you could use a typewriter. That was the beginning of the contents of this book. As a farewell gift, my coworkers gave me a 35 mm camera, which was used to take most of the photographs in this book.

I left for another position in 1979 for a job at an aircraft engine plant where I managed a high-pressure high flow air compressor facility. I became involved with the Vibration Institute and we formed a local chapter in the Cincinnati, Ohio area. We had meetings every month where a guest speaker was invited to give presentations on vibration and other topics. We would solicit the attendees for future topics and one of the requests was for someone to do a presentation on shaft alignment. The other board members suggested that I do a presentation on alignment since they knew I had some material from the maintenance training I had done at the chemical plant. After I gave the presentation, which went much longer than the one hour allotted for the discussion due to questions from the audience, several people came up to me suggesting that I write a book on shaft alignment. I laughed. You must be kidding! Me? Write a book? I shrugged it off as a nice compliment and didn't think anything about it until another presentation I gave a few months later when I was asked again. So, around 1981, I started writing trying to expand on the material I already had and writing down the things I was still learning. Three years passed and bit by bit I generated six chapters, working on the draft copy at night and for a little bit early in the morning before I went to work. I bought a typewriter transferring my hand written text and gluing my pen and ink drawing to the pages. I made five photocopies and sent them out to five publishers. I got five rejection letters. The letters were cordial and short. The general consensus in the letters was "We don't think this material would be of any help or interest to anyone." I put the manuscript into the closet and forgot about it. What a waste of time I thought. About a year later I gave another presentation at the annual meeting of the Vibration Institute and the director of the institute came up after the presentation and suggested that I write a book on alignment. I mentioned that I had and that it was rejected. He suggested that I contact a publisher called Marcel Dekker, Inc. who had approached him for anyone he might know who would be interested in writing engineering books. I resurrected the manuscript from the closet and submitted it one more time.

The first edition of the *Shaft Alignment Handbook* was published in 1986 and had 278 pages. The second edition was published in 1996 and had 14 chapters and 579 pages. The contents of this edition have been rearranged based on feedback from thousands of people who read the first two editions. Additional information has been included in Chapter 2 on detecting misalignment on operating rotating machinery. Chapter 3 on foundations, baseplates, and piping strain has been expanded considerably and addresses just about every type of interface between the rotating machines and how they are attached to building, structures, or Earth. Chapter 6 through Chapter 16 reflect suggestions and improvements from readers of the previous two editions. Rather than have one huge chapter that not only discusses the myriad of both mechanical and electronic alignment measurement tools, but also covers how to perform the five basic alignment measurement methods, the five basic alignment measurement methods are now covered in separate chapters (Chapter 10 through Chapter 14) as well

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as a chapter solely dedicated to electro-optical tools (Chapter 15). For those who have mastered alignment graphing and modeling, it becomes apparent that there are many similarities between the different modeling methods for each of the alignment techniques described in Chapter 10 through Chapter 14. It was suggested that a chapter be added that covers the basic concepts of alignment graphing and modeling prior to covering the specific modeling procedure for each of the five alignment methods, which is the intent of Chapter 8. Chapter 16, which discusses movement of rotating machinery from off-line to running conditions, has been expanded considerably and includes some very interesting information from equipment that has been studied in the field since the publication of the second edition. The chapter on aligning multiple element drive trains has also been expanded to include new information learned about these complex drive systems. Chapter 18 on alignment of belt/sheave driven machinery now includes information on the tooling and techniques developed since the second edition. Chapter 19 and Chapter 20 on bore and parallel alignment are brand new to this edition. Information on specific types of rotating machinery covered in Chapter 21 has had a significant amount of new and expanded information, including hydroelectric generators and alignment of rudders on ships.

There are several people who have made valuable contributions to this book as indicated by the multitude of references at the end of each chapter. None of these people had to do any research in this area nor did they have to write down what they learned for the benefit of everyone else who works in this area . . . but they did and I, for one, am thankful for that. It is very unsettling to be responsible for machinery that is typically the heart and soul of your operation and realize that you don't know enough about this equipment to keep it running satisfactorily. I have made many mistakes over the years trying to learn about rotating machinery and I wouldn't want you to go through what I had to, so this book is an opportunity to share with you what I have learned.

I have had the opportunity to work with many people in industry over the years and can't remember anyone not wanting to know how to align machinery properly or anyone who purposely wanted to damage machinery because it was operating in a misaligned condition. There are several people who have expended a tremendous effort to get their equipment alignment correct and very few of the people they work with have a clue what they had to go through to accomplish that goal. They often do it with no acknowledgment when it's done right and no increase in pay. In my humble opinion, there is no better feeling than walking by a piece of machinery that you aligned 10 years ago and know that equipment is still running great. These people do it right because it makes them proud of the quality of workmanship they perform and although they might not be able to take that to the bank, they can certainly take that to their grave when they reflect back on their accomplishments.

The primary reasons why machinery is misaligned is lack of proper training, improper tools to do the job, and that people are not given enough time to do it right. This book will help with the training and the tools; the rest is entirely up to you.

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1 Introduction to Shaft Alignment

1.1 BENEFITS OF GOOD MACHINERY ALIGNMENT

Industry worldwide is losing billions of dollars a year due to misalignment of machinery. The heart and soul of virtually every industrial operation pivots on keeping rotating machinery in good working order. Countless processes are dependent on the successful operation of rotating machines that produce electric power, fuels, paper, steel, glass, pharmaceuticals, the food we eat, the clothes we wear, the buildings we live and work in, and the vehicles that transport us across the surface of the Earth. Just about everything you see around has somehow been influenced by rotating machinery of some kind.

The primary objective of accurate alignment is to increase the operating life span of rotating machinery. To achieve this goal, machinery components that are most likely to fail must operate well within their design limits. As the parts that are most likely to fail are the bearings, seals, coupling, and shafts, the accurately aligned machinery will reduce excessive axial and radial forces on the bearings to insure longer bearing life and rotor stability under dynamic operating conditions. Precise alignment will reduce the possibility of shaft failure from cyclic fatigue; it will minimize the amount of wear in the coupling components, alleviate the amount of shaft bending from the point of power transmission in the coupling to the coupling end bearing, and it will maintain proper internal rotor clearances.

In a nutshell, accurate alignment will do nothing, but the good things and the key part of making this happen centers on the people who are responsible for installing, troubleshooting, maintaining, and operating this machinery.

1.2 CONSEQUENCES OF DEFECTIVE ALIGNMENT

Despite popular belief, misalignment can disguise itself very well on industrial rotating machinery. What we witness are the secondary effects of misalignment as it slowly damages the machinery over long periods of time. Some of the common symptoms of misalignment are as follows:

1. Premature bearing, seal, shaft, or coupling failures.
2. Elevated temperatures at or near the bearings or high discharge oil temperatures.
3. Excessive amount of lubricant leakage at the bearing seals.
4. Certain types of flexible couplings will exhibit higher than normal temperatures when running or will be hot immediately after the unit is shut down. If the coupling is an elastomeric type, look for rubber powder inside the coupling shroud.
5. Similar pieces of equipment seem to have a longer operating life.
6. Unusually high number of coupling failures or they wear quickly.
7. The shafts are breaking (or cracking) at or close to the inboard bearings or coupling hubs.

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FIGURE 1.1 Disk coupling failure due to shaft misalignment.

8. Excessive amounts of grease (or oil) on the inside of the coupling guard.
9. Loose foundation bolts, typically caused by a "soft foot" condition, are exacerbated by misalignment.
10. Loose or broken coupling bolts. This is frequently due to improperly torquing the coupling bolts and aggravated by a misalignment condition.

If you are expecting to see a statement on how misalignment increases the vibration levels in machinery, it is recommended that you thoroughly review the information in Chapter 2.

Without a doubt, the greatest loss of revenue attributable to misalignment is due to loss of production. If a piece of machinery fails, then whatever it was producing stops and so does the revenue it was making, which then leads to the second largest amount of financial loss. Now that the machine broke, new parts have to be purchased and someone has to be paid to repair or completely replace the defective components or the entire unit. As production is lost, there is a tendency to rush the installation process, frequently sacrificing the time needed to perform an accurate alignment of the machinery. And the degradation process starts anew.

A large percentage of industrial plants do not understand how bad their misalignment problems are. Conservatively more than half of all the equipment operating today exceeds 4 mils/in. of misalignment when it is running. Figure 1.2 shows data from an alignment survey taken during a 1 week period of time. Bear in mind that acceptable misalignment deviation for rotating machinery is 1 mils/in. (that is the first tick mark on the *y*-axis).

Disappointingly the vast majority of plant sites cannot produce the alignment records for every piece of rotating machinery they operate. Even in facilities where a good preventive and predictive or condition-based maintenance (CBM) program exists, typically there are 100 times more data collected on vibration, temperature, oil analysis, and motor current than on machinery alignment. The vast majority of people who measure vibration

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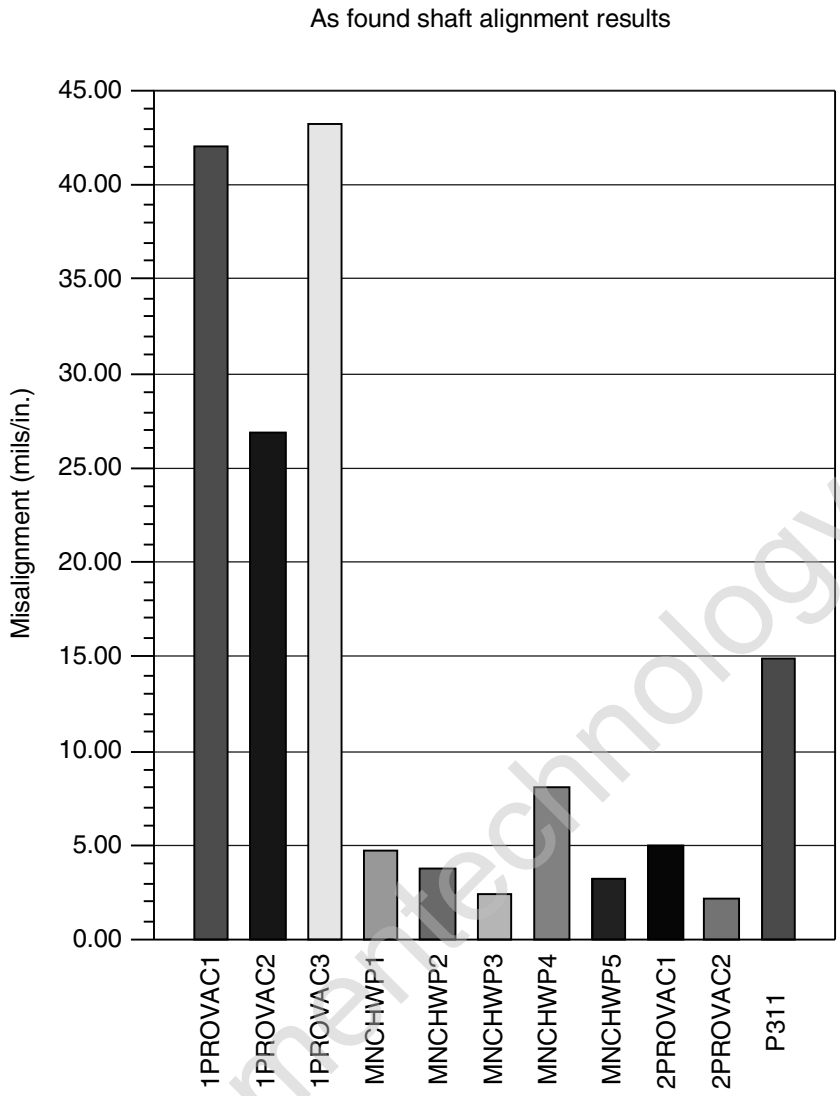


FIGURE 1.2 Alignment survey taken on 11 drive systems during a 1 week period of time showing the as-found alignment condition.

and other data have received incorrect technical information about what symptoms occur on misaligned machinery, frequently resulting in an incorrect analysis of the problem (refer to Chapter 2 for more info). Additionally, the same people are usually not the ones doing the alignment work, so communication gets scrambled between the people identifying the problem, the people assigning the work, and the people actually doing the work. Countless times I have been told by a mechanic that they were reprimanded for sloppy workmanship because vibration levels on a machine stayed the same or increased after they completed realignment. Surprisingly to many, this is quite normal. For a detailed explanation for why this occurs, see Chapter 2.

1.2.1 What Happens to Rotating Machinery When It Is Misaligned a Little Bit, or Moderately, or Even Severely?

Figure 1.3 illustrates what will happen to rotating machinery when it is subjected to moderate to severe misalignment conditions. When two shafts are subjected to a slight misalignment condition (2–5 mils/in.), the coupling connecting these two shafts together flexes internally to accept the misalignment condition. Keep in mind that as the shafts rotate the internal parts of the coupling are continually having to move around, or bend in one direction then the other,

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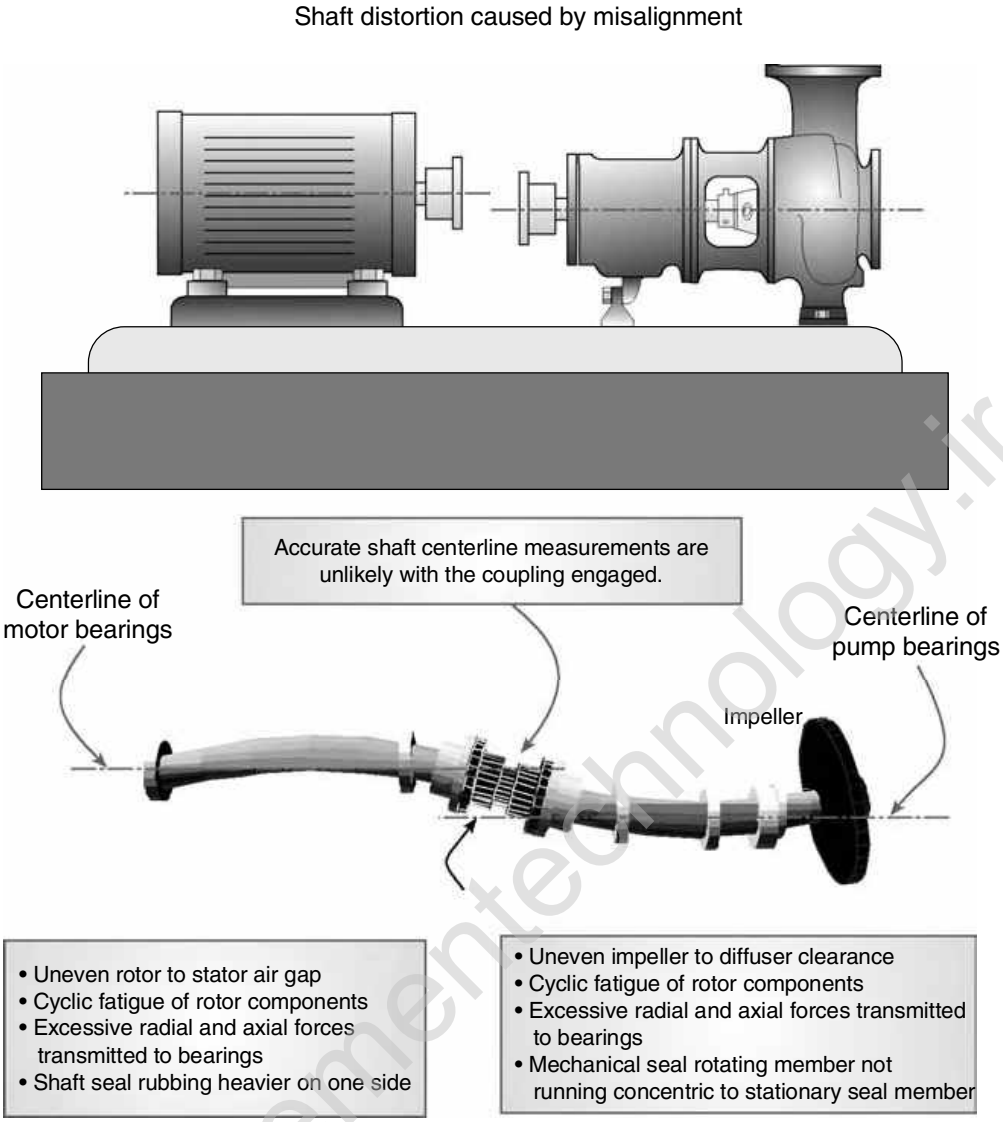


FIGURE 1.3 Problems caused by shaft misalignment.

or stretch then compress. If the cyclic movement, bending, or distortion is not severe, the coupling may survive for some time.

Most people forget that the majority of the industrial machinery runs 24 h a day, 7 d a week for months or years without stopping. If your automobile engine were operated at 3600 rpm (about 80 mph for most vehicles) without stopping, in 1 year your vehicle would have traveled 700,800 mi. That is over 28 times around the circumference of the Earth. I wonder what the tires would look like if the front end of the vehicle were out of alignment?

As the misalignment becomes greater (5–10 mils/in.), and the coupling approaches its maximum allowable misalignment condition, the movement of the internal parts begins to interfere with other parts that do not want to move, the bending gets worse, and the distortion starts to tear things apart on a microscopic level. Heat begins to build up and the shafts and bearings of the machinery begin to realize that something is beginning to tug on them and that something is seriously wrong.

When the misalignment gets into the severe range (above 10 mils/in.), the coupling has exhausted its maximum possible operating capacity and now the shafts start bending elastically in an attempt to accommodate the misalignment imparting high radial loads on the

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bearings. We are in big trouble now since the lubricating film in the bearings is getting squeezed thinner, running the risk of metal-to-metal contact. High stress concentration areas on the shafts near the inboard bearings and coupling hubs begin to experience cyclic fatigue. The oil seal clearances begin to open up, allowing the lubricant to leak out of the bearing chamber. Lubricant in the coupling starts to get squirted out. Tremendous amounts of heat are generated in certain types of flexible couplings. The bearings are getting hotter from the excessive forces. Somebody better get ready to call the ambulance because this is not going to last much longer under these circumstances! Boom! There it went! Oh my! Even though it is two o'clock in the morning, we need to call someone in here to get this back up and running. Should not take more than 30 min to get this fixed, right?

Albeit, the last misalignment condition shown here is exaggerated but this is exactly what happens. The rotating machinery shafts will undergo elastic bending when vertical or lateral loads are transferred from shaft to shaft.

It is fully understood that flexible couplings do just what they are designed to do; they flex to accommodate slight misalignment. But the shafts are flexible also, and as the misalignment becomes more severe, the more the shafts begin to flex. Keep in mind that the shafts are not permanently bent, they are just elastically bending as they undergo rotation.

Notice also that the pump shaft in this example is exerting a downward force on the inboard motor bearing as it tries to bring the motor shaft in line with its centerline of rotation. Conversely, the motor shaft is exerting an upward force on the inboard pump bearing as it tries to bring the pump shaft in line with its centerline of rotation. If the forces from shaft to shaft are great enough, the force vector on the outboard bearing of the motor may be in the upward direction and downward on the outboard bearing on the pump. This explains why misaligned machinery may not vibrate excessively due to the fact that these forces are acting in one direction at each bearing. Forces from imbalanced rotors, for instance, will change their direction as the "heavy spot" is continually moving around as the shaft rotates, thus causing vibration (i.e., motion) to occur. Shaft misalignment forces do not move around; they act in one direction only.

The chart in Figure 1.4 illustrates the estimated time to failure of a typical piece of rotating equipment based on varying alignment conditions. The term "failure" here implies a degradation of any critical component of the machine such as the seals, bearings, coupling, or rotors. The data in this graph were compiled from a large number of case histories where misalignment was found to be the root cause of the machinery failure.

1.3 FOUR BASIC INGREDIENTS NECESSARY TO INSURE ALIGNMENT SUCCESS

If machinery alignment is so important and we are wasting billions of dollars due to the fact that it is not done correctly, why has not this problem been eradicated? Perhaps the best way to illustrate the root cause of the problem is to relay what I hear from the people in industry and show you what they have to say about this. Every so often I receive correspondence that goes something like this:

John:

How long does it take to convince seemingly intelligent trades people of the importance of proper alignment and what steps can be taken to help lead them to this belief? We are using an S&M system (i.e., laser alignment) to perform our alignment jobs. Our intent when purchasing this system was to educate the trades people on the use of it, then give them part ownership, in hopes of encouraging more usage. Their acceptance was, and is good to the concept of proper alignment, as long as we (predictive maintenance) will come out and do the work for them. When left on their own, the way it came off is the way it will go on. Any suggestions?

Sincerely

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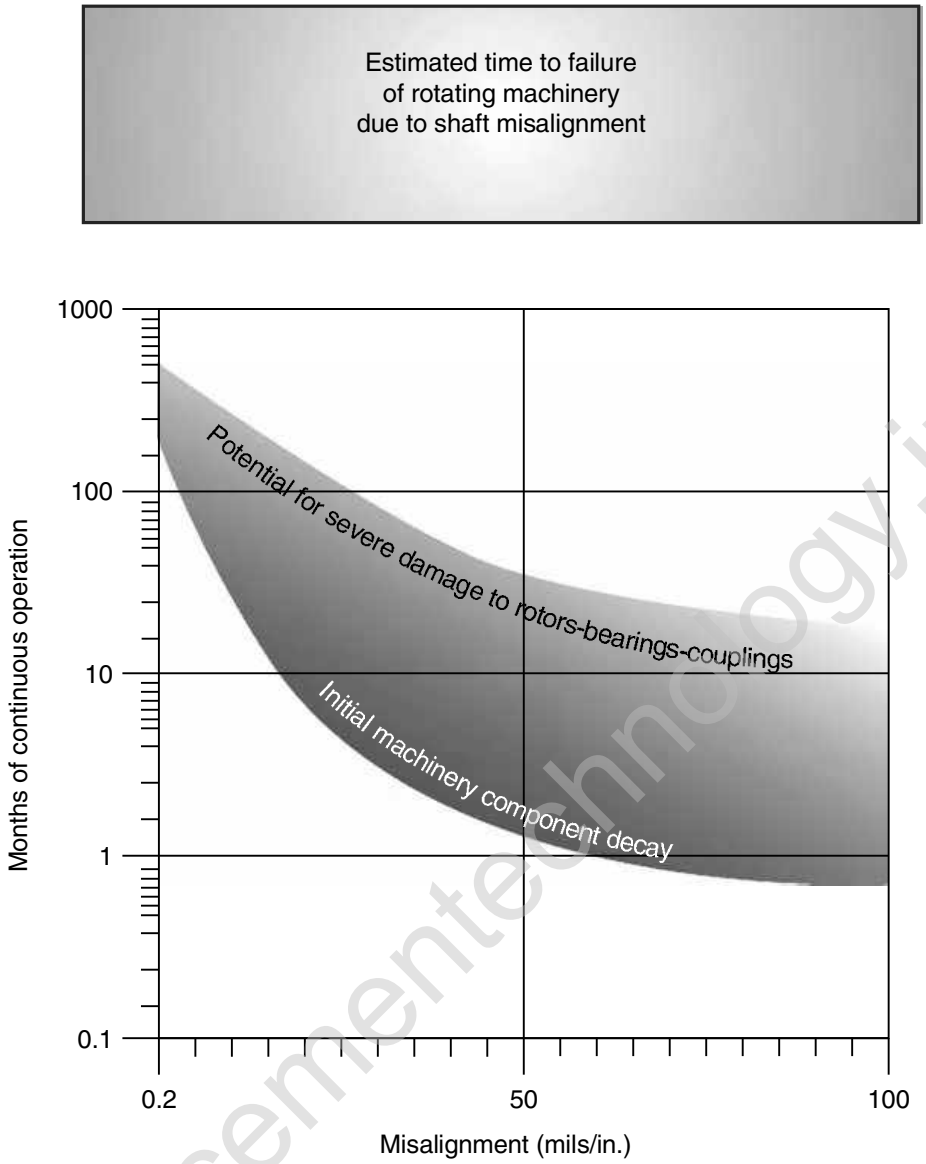


FIGURE 1.4 Statistical information on survivability of rotating machinery subjected to various misalignment conditions.

Name withheld by request

My reply to the above letter:

Your question concerning the length of time it takes to convince trades people about the importance of shaft alignment is most difficult to answer. Perhaps the best way to respond would be to relate observations I have made over the years and also to tell you what I hear from the trades people themselves. Their concerns are most valid and in many cases cut to the root of the problem.

The philosophical foundation of machinery alignment (and perhaps a wide variety of other assignments) are successfully implemented only if the following four ingredients are employed. I refer to it as the "T-T-T-I solution" (Training, Tools, Time, and Inspiration).

1. *Training*

I firmly believe that many chores that need to get completed don't simply because people do not realize the significance of the assignment and its long-term implications. It is indeed

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possible to perform a difficult task with no formal training but there is a greater chance of success if a knowledgeable individual instructs someone why and how the job should be done. For most people it is best to cover the "why" aspect before the "how" process. An explanation of "why" will invariably provide the trainee the incentive to reduce the pain that could ensue if the task is not undertaken in an orderly fashion.

Concerning instructing personnel in machinery alignment, here are key issues that need to be discussed in any training situation:

- Who is responsible for machinery alignment?
- What consequences will occur if the alignment is not acceptable?
- What rewards will I garner if the alignment is done successfully?
- What is involved in the overall requirements of this job and what role do I play in the larger scheme of things?
- For the tasks that I am not responsible for, how do I interface with others who are participating in the overall job to inform them that I have completed my duties or that I am having some problems?

Shaft alignment should be a major concern to every conscientious manager, engineer, foreman, and trades person. All of these job descriptions in a typical industrial organization have a role to play.

The tasks of the trades personnel (mechanics, electricians, pipe fitters) are to perform the preliminary alignment steps, measure the position of the shafts, determine and perform the proper moves on the machinery to achieve acceptable alignment tolerances, and communicate the as-found and final alignment results or problems encountered during the alignment job to their supervisor.

The foreman or unit manager's task is to assign the right people to do the alignment job, insure the necessary and appropriate tools are available and in working order, provide adequate time to complete the job, answer any questions the trades personnel may have, provide guidance for potential problems that might occur during the alignment job, coordinate and communicate these problems to engineers and managers for resolution, and keep records of what was done.

The job of vibration technicians and vibration engineers is to identify problems with the machinery, one of which is shaft misalignment. Are you 100% sure you know what this really looks like? If you're only looking for 1 or 2 times running speed vibration components in the spectrum with higher axial vibration levels than radial vibration levels and a 180 degree phase shift across the coupling.... you are in for a big surprise, because misalignment doesn't always show up that way. Do you know that vibration can actually decrease if you misalign equipment? If you don't understand why this can happen, you need some good training, not just what you've heard others say.

Some of the roles of a maintenance or plant engineer may be to purchase rotating machinery, design foundations, specify installation procedures, buy couplings, design and oversee piping installation, and insure the equipment is aligned properly when it's installed or rebuilt. Engineers frequently are responsible for specifying and procuring the types of measuring tools used to determine the positions of the machinery shafts, provide the technical expertise and tooling to measure off-line to running (OL2R) machinery movement, design piping/duct support mechanisms to minimize induced stress in machine casings or coordinate piping/ducting rework, review new methods and techniques that could be used on the rotating machinery in their plant, analyze failures of rotating machinery to determine if the root cause can be traced to misalignment, listen and respond to any and all problems that were reported to them by the trades personnel and the foremen, provide training to the trades

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personnel, foremen, and managers, and work side-by-side with the trades personnel if necessary to fully understand what actually occurs in an alignment job to determine if more efficient means can be found to improve the alignment process or accuracy.

The responsibility of plant and/or engineering managers is to provide the funds necessary to procure the tools needed to accomplish the job, to insure that the personnel have been given proper training to understand how to do alignment, and to provide due credit to the individuals who have done the alignment job properly.

Proper training must also provide information on runout detection and correction, soft foot correction, measurement instrument basics, rotating machinery design basics, foundation and baseplate repair techniques, bearing information, coupling information, machinery movement tools, alternative machinery movement solutions, OL2R machinery movement measurement (and recognition of when it is and is not needed), finding and relieving piping stresses, and record keeping. We also cannot expect training to just be given to trades people only. Rarely will I see management personnel or engineers or foremen attend a hands-on alignment training class. The ones who do attend get an opportunity to see first hand what's involved in the process and they finally get a full understanding of why alignment jobs can't be done properly in 45 minutes. Also some of the alignment tasks should probably be done by technicians and engineers. I don't feel that it is the responsibility of trades people to specify a new coupling design, or to use a proximity probe alignment bar system to measure OL2R machinery movement, or to maintain maintenance record files, or to procure the right kind of tools to do the job right. Management and engineering must get involved to assist in the successful completion of an alignment task. How can they direct what is to be done if they don't understand what's involved in the overall process?

2. *Tools*

The people performing the alignment job have to be given the proper tools to do the job correctly. Purchasing an alignment measurement system is a good start but that is not the only tool one needs to perform the job. An alignment measurement system does one task ... measures the amount of misalignment that exists. The alignment measurement system does not correct the problem. There are a myriad of other related issues that need to be addressed before the actual alignment corrections begin. What tools have been procured to verify that people aren't aligning bent shafts or improperly bored coupling hubs? What tools are available to control the position of the machines when correcting the misalignment? What tools do the people have to insure they are aligning reliable pieces of rotating machinery? Also, if the alignment measurement system contains a software program that suggests how to correct the misalignment, it must inform the operator to make an intelligent move and offer that person other options if a suggested corrective action does not make sense. For example, the software program may tell the operator to move a machine 120 mils sideways to correct a lateral misalignment condition. However if the unit becomes "bolt bound" before the move is accomplished, and the alignment measuring system does not have an alternative suggestion to get that person out of the jam it proposed, the software program is flawed and may significantly increase the amount of time needed to achieve satisfactory alignment.

3. *Time*

Undoubtedly the number one complaint I hear from people in industry is that they are not given enough time to perform all the necessary tasks required for successful alignment. The reason for this is due to a lack of understanding on the part of operations, production, and management personnel who have not received training on alignment. If many foremen or supervisors have no idea of how to do an alignment job, how can they tell a trades person that it should only take 45 minutes to do it right? What authority should an operations manager

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have to tell the maintenance department that a machine should be put back on line even if it means that its not installed properly? That operations manager isn't going to be down there fixing the machine again in two months some early Sunday morning.

4. *Inspiration*

This is perhaps the most important ingredient and the one that seems to be lacking in many industrial sites. The answer to this is as complex as the human personality itself. I guess to begin, I'll ask these questions. What incentives does the trades person have to do the job right? Do they feel that they have some ownership for the equipment they work on? Are they constantly instructed on how to fix a problem and how much time they should take to fix it? Have they ever been told that there is not enough time to do it right, that operations needs the equipment right now and that compromises will have to be made? Have they been denied all of the tools to do a job properly? Are they really doing shaft alignment properly and the machinery is being shoved back out of alignment after a brief period of time due to excessive piping strain or unstable foundations? Do the trades people have a clear understanding about how accurate the alignment has to be? Has anyone asked them if they like the alignment measurement system that they were given without choice? Have they ever been congratulated for doing a job correctly? Do they get to make decisions on how to do a job, when to do it, and how long it should take? If someone prefers to do alignment with dial indicators, are they given that option and then given training on how to use other types of measurement tools? Can they go back into the maintenance records on every piece of machinery in their plant and find information on . . . how accurate the alignment was done the last time, when it was done the last time, who did the alignment last time, how many shims and what thickness of what material are under all of the machinery feet, what soft foot corrections were made, was there any piping strain present, if any of the bolts were undercut, how long the machinery has been there, were any special tools needed to do the alignment, was there any runout on the shafts or the coupling hubs, how much "free-play" existed in the bearings, what is the shaft to shaft distance supposed to be, what type of coupling could be substituted if the one that's there gets damaged and there are no replacements in stores, what type of lubricant should be used in the coupling and how much should be added, etc.? I could go on about this for quite some time and I've only mentioned a few items that seem to go overlooked in many organizations. Indeed, some of the blame for lackluster maintenance correction methods falls on the shoulders of the trades people, but definitely not all of it.

Sincerely . . . yours truly

Not even one of my shorter letters is by any stretch of the imagination. Here is another typical one:

John:

I work for XYZ Chemical (name changed by request) in a Midwestern state in the U.S.; I'm the engineering superintendent. In addition to my site engineering responsibilities, I also have responsibility for the maintenance department.

Our plant is a union facility. We have approximately 30 folks in maintenance, of which 15 of them work on pumps, seals, etc. People come into the maintenance department based on their seniority. Their maintenance training is gained by working with other maintenance personnel, learning from onsite vendor seminars, and from their maintenance supervisor.

I have been here at this facility for 4 years now. During the past 4 years, I have seen a lot of turnover in the maintenance department, due to retirements, etc. As you can see, I do not have an easy job keeping people in their positions and also providing them with the necessary training. It seems to me that any training we do here goes in one ear and out the other.

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I have conducted training with my maintenance personnel in seal repair, pump repair, laser alignment, etc. but we don't seem to get any better at it. As you say in your article, we operate in the "Breakdown or Run-to-Failure Maintenance Mode."

I would like to solicit your opinion on what steps I can take to get our maintenance department out of this mode and move into the "Preventive or Time Based Maintenance Mode." I know this takes desire and a want to attitude, I have it, but they don't. Do you have any recommendations that I can use to make this shift? Would appreciate your ideas/comments.

Regards,

Name withheld by request

My reply to the above letter:

Due to the opportunities I've had in being able to visit and work at a wide variety of plant sites in virtually every industry, I'd like to reassure you that the problems you are facing in trying to get the maintenance personnel to get out of the breakdown/run to failure mode and into something that is more effective is virtually a universal problem. Not that this is any consolation to you, however. My recommendation is to move toward a Pro-Active/Prevention Maintenance approach rather than a Preventive or Time Based approach. I know that might be quite a leap but the benefits would be justified.

I'm not quite sure where to start. I remember an Engineering Manager at a plant I worked at once telling me, "We need to hire psychologists and sociologists and stop hiring engineers." As an engineer myself, I was crestfallen to hear him say this but there is some substance to his remark.

Over the past 26 years in industry, I have seen many changes occur yet many things have stayed the same and in some cases, gotten worse. Here are some observations, comments, and ideas that hopefully may address your concerns and desires.

1. *There must be at least one person in any organization who is willing to spearhead the movement to elevate the quality of maintenance work being done*

The buzzwords (Reliability Engineering, Pro-Active, Prevention, Predictive, Maintenance etc.) used to describe this effort are irrelevant. What is important is to have a leader who has devised a clear plan on what needs to be done and mechanisms to ensure that they will be carried out. At a minimum, this person needs to coordinate, and in many cases perform, the following efforts:

- Assume that there could be a problem with every piece of equipment in the facility and devise a process to find every detrimental condition.
- Identifying existing problems with the equipment and determining the severity of the problem.
- Illustrate evidence to support the validity of the problem.
- Recommend a course of action to correct the problem.
- Investigate the cause of the problem or unexpected failure by performing root cause failure analysis.
- Suggest and incorporate changes to insure the problem does not occur again or at least not as often.
- Communicate all of the above to the production and maintenance personnel involved and have a way for others to communicate what they discovered during the repair/improvement process.
- Set standards for the quality of workmanship to be performed during the repair process.
- Financially justify the work that was performed.
- Become an evangelist who supports the reasons for the program.

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Every company that is serious about minimizing failures and increasing productivity must be able to say that this person or that person is doing each of the things listed above. Who is doing all of this in your company? Where is it being done? What is missing? Why should we do it in the first place?

2. *When a piece of equipment needs repair, the maintenance department should be in control, not the production department*

Everyone knows that disruption of service or production is uncomfortable and there is a strong pus to "Get this up and running as quickly as possible." Production's job is to get a quality product out the back door as efficiently as possible and they are generally very good at that. In many cases the operations personnel do not know the intricacies of the machine or process that makes the product and are therefore not qualified to decide what needs to be done or how quickly it should happen. It is important for them to understand that it is not immediately evident what transpired to cause the failure and troubleshooting can take more time than the correction phase of a repair. They also need to know that if the cause of the failure is not determined, there is a good possibility that it will occur again and again. During the time the repairs are being made, frequently twiddling their thumbs, rather than continually asking "Uh, when do you think you'll be done?" they should be working on reviewing what transpired prior to the failure to determine if the mishap was process related and communicate their findings to the maintenance or engineering department for review. Sometimes we get very lucky and things happen quickly. More often we have to remove several thorn bushes before we can actually find the path. The maintenance people I work with are very talented but don't ask us to perform miracles.

3. *We have a great education system in this country but it does not prepare our work force for the specific tasks required by industry*

After four years in engineering school it took only a month working in industry for me to realize that I did not know what I was doing. I was taught how to go about solving a problem, not how to solve very specific problems. I probably studied harder for the next 16 years in industry than I did the previous 16 years when I was in a formal education setting. The one thing that I realized is that pretending that I knew everything alienated me from those who did know something. Working closely with the maintenance and production personnel on a daily basis, watching what they were doing, asking them questions, being a technical resource for them, having the courage to try something new, and learning from each failure was a key part of my continuing education. Thank goodness every company I worked for provided some way for me to expand my knowledge base. As far as I'm concerned, I'm still in school and so is everyone else.

I think that there should be a list of skills generated for every job in the industry. Each list should have minimum requirements to perform the job and additional skills that should or could be acquired over time. I believe that those who acquire the additional skills should be paid more for these additional abilities. I also think that there should be some way to reward those who utilize these abilities to their fullest extent. In my opinion, the last year of a maintenance persons time in a facility should be spent teaching others what they have learned prior to retirement. I also think that any retiree should be granted the opportunity to come back as a consultant/trainer on a part time basis and be paid well for such efforts.

4. *There should be a mechanism to reward people for what they have done correctly and a process for inferior work to be done over by the people who did the slipshod work initially*

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This is somewhat related to what was mentioned above concerning those who utilize these abilities to their fullest extent. I have noticed that some people do higher quality work than others. What is disappointing is that they don't seem to be rewarded for their efforts. Nor do I see those who do poor quality work held responsible for their shortcomings. In my humble opinion, if a piece of machinery fails within three months after someone worked on it and the failure was due to shoddy workmanship, that person should be required to work on the same machine again and continue working on it until it is done correctly. If that person is doing poor quality work because they don't know how to do the task, they should admit this and be given proper training on how to do it correctly.

5. *It is important for people to feel that their contribution is valuable to their company*

One minute for what someone did right, one minute for what they did wrong, and one minute for what they are supposed to be doing. What I call the "Well Done!.... Not that, please!.... Which way do we go?" approach.

We need to abrogate the industrial caste system. Everyone's job is important.

In maintenance, the personnel need to be shown the results of higher quality workmanship and what benefits it reaps them and their company. There are two prime motivational tools: fear and inspiration. Both work but inspiration works better. If fear is the motivator, when the tyrant leaves, the subjects will become idle or vengeful. If someone is told they are doing well, they have a tendency to continue when no one is looking.

6. *Industry is not preparing itself for the long haul*

It's what I call "the under 40 void." What percentage of your maintenance department is between the ages of 20 and 40? What percentage of the population is between the ages of 20 and 40? Get the drift?

Can we expect qualified maintenance people to appear from thin air when people retire? If we don't groom the individuals ourselves, who will? For those who decide that the work force needs to be culled, their names should be at the top of the list for allowing the flock to get out of control.

7. *Installation of new equipment is frequently not being done correctly*

Just as you noticed that a maintenance organization can get into the breakdown or run to failure mode there is an equally disturbing and similar practice with construction work. A large percentage of machinery is being installed wrong or shoddily. The reason that this has gone unnoticed is because poorly installed equipment can actually run, it just doesn't run for long periods of time before failures begin to occur. Too often the quality of workmanship in construction projects is ignored by the end user. Usually the end user does not know enough about good installation practices to insure the equipment is installed correctly and no checks are made during the construction process to verify the quality of workmanship.

8. *Any training should be quickly reinforced with jobs that reflect the material covered in the training session*

Several years ago, after conducting a two day basic shaft alignment training course at a food plant, the company requested that I stay for another day for on the job training with the students who attended the course. I arrived there early the next morning and all of us met back in the training room to discuss what we were going to be working on that day. We split

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the group up into two person teams and the maintenance supervisor handed me a list of machinery that was available for us to work on and we began deciding which team was going to work on which piece of machinery. After each team selected the machine they wanted to work on, I asked if anyone had any questions before we went out and began work on the equipment. One of the more outspoken students from the class raised his hand as asked, ''Hey Teach, the information you went through over the past few days in the shaft alignment course was very interesting but now that we have to work on real pieces of machinery, what are we really going to be doing?'' I was puzzled at his question but after a brief pause I looked at him and said, ''The work that you were doing on the demonstrators in the training course is exactly what I would like you do to do with 'real' machines in your plant.'' For a moment it appeared that he got the Uh-oh, I-should-have-listened-better-look on his face but he said nothing and just stared in what appeared to me to be disbelief.

As we left, the group of us went to each machine we decided to work on, discussed what needed to be done to get started, then that team stayed there and began work while the rest of us moved on to the next machine until the entire group was dispersed and working on their respective equipment. While this was happening, I noticed that many of the students, some of whom had several years of experience in industry, when arriving at their machine would ask, ''Do you want us to do the same process on our machine?'' ''Yes, please,'' I would answer.

After an hour or so, each team was at their machine and I began to loop back to the first group that we dropped off to see how they were doing. As I approached I noticed that they were busily working on the machine and as I walked up one of the mechanics noticed I arrived and with a big smile on his face said, ''Hey Teach, look at what we found here!'' showing me some problem that had not been observed before. ''I bet this is what has been causing some of the problems with this thing,'' he said. ''Now that we found this, do you think we need to do all of the other things you talked about too?''. ''Great job, and yes, please do all of the other things we discussed. You never know what other problems you're going to find until all of this is checked and corrected,'' I exclaimed. Throughout the day, I would go back to each team one by one to see how they were doing and answer any questions they had. There was a recurring theme all day long with every team. Things that had never been checked before on the machinery were disclosing why certain machines were performing a certain way or why failures were occurring more frequently than expected. Some of the problems that were discovered were corrected on the spot, others required new parts to be purchased, and other issues that were revealed needed some major work or design changes.

At the end of the day, all of us reconvened back at the training room to discuss what went on that day. It was bedlam. You couldn't hear yourself think! Everyone was telling each other what they discovered, what problems they corrected, and what else needed to be done. There seemed to be a contest going on to determine who found the worst problem, each team trying to convince the others that they did the best work or found the biggest problem. The maintenance supervisor looked at me and said, ''I've never witnessed anything like this before with these guys.''

After things quieted down, the same outspoken student who asked ''What are we really going to do?'' before work got started, raised his hand again. ''Hey Teach,'' he said, ''I learned more today that in the entire 14 years I've been working here. This stuff is really great. I would have never believed what you told us would work, it just sounded too good to be true!''

9. *Be explicit about what needs to be done and nurture the desire to exceed the basic requirements*

''It broke and you need to fix it'' is not a satisfactory maintenance work request. Despite my best attempts to perform consistently high quality workmanship, I have the tendency to

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skip or forget items that should be done. I don't like having an itemized list of things to do on a job, but it sure beats the sinking feeling I get after assembling a piece of machinery only to find three parts lying on the work bench that were supposed to be installed inside the machine I just "finished."

At an electric generating plant where I worked recently, every maintenance job had a written "work packet" describing the tasks to be done. At the beginning of every shift all of the participants got together for 15 to 30 minutes and the step by step instructions were discussed by the front line supervisor and the maintenance personnel. After the meeting was done and everyone understood what needed to be done, we left and went out to the job site and started working. As each step was completed, it was checked off the list. When problems were encountered, the front line supervisor, an engineer, a manufacturer's representative, or outside consultant(s) were contacted to assist in rectifying the concern. Near the end of the shift, the completed items were discussed with the next crew who had a 30 minute overlap time with the personnel on that shift to discuss what was done and what yet needed to be done. When the job was finished, a complete review of the work was conducted by the front line supervisor(s) with the maintenance personnel, engineers, and the individual who wrote the work packet. If things were missing or wrong with the work packet, it was changed. If things went badly, suggestions were made to improve the process. Often the mishaps were due to a lack of knowledge, experience, or skill and the people realized that training was needed and then the appropriate training was immediately scheduled. If things went well, accolades were distributed to the appropriate personnel which boosted morale. For me, it was a very enjoyable process and the time flew by. If it feels like you've worked for eight hours instead of eight minutes every work day, something needs to change.

One final note. This work was initiated not because there was a failure on this pump but because there was higher than desired axial vibration at the outboard bearing of the pump and the rotating machinery engineers wanted to determine the cause of the excessive vibration. This pump was a safety backup pump for two other pumps and had logged six hours of operation in the last twelve months. In fact, the six hours were test runs that were conducted to insure proper operation of the unit. The level of vibration prior to the work was not threatening, but it was near a predetermined alarm limit they set and it bothered the people enough to investigate the cause. In the first of the two different visits I made to the plant site to investigate the source of the vibration, we decided to include some additional checks not mentioned in the work packet. When we suggested that the additional work be done, there was some resistance from the mechanics who did not want to deviate from the scripted plan but we explained our concerns to them and made it understood that we might not find anything and we were willing to "take the blame" for slowing the work down. I remember three of them staring at me when I said that and the group leader exclaimed "No, we will do what you ask and still try to get all the work done. You're here with us and we will accept any blame together." These additional checks led to the discovery of a locked up gear coupling. The shaft alignment was checked with their laser alignment system and it was slightly out of their specifications but well within traditional alignment tolerances. The gear coupling was replaced in a few weeks and another test run confirmed that there was no change in the vibration.

They could have quit right then, and explained the vibration as the "nature of the beast," but they didn't. Another work packet was generated for further investigation. On the next visit the thrust of the work was to improve the alignment to near perfect conditions to see if that would reduce the vibration. One of the first tasks was to use the laser alignment system to check for a soft foot condition. The check was made and the laser indicated that a soft foot condition was not present. As they began to move onto the next step, I asked them if we could again deviate from the work plan and conduct another soft foot check by loosening all of the

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motor foot bolts and attempt to slide a feeler gauge between each of the motor feet and the frame contact points. Again, they resisted and again they agreed after hearing my plan despite the fact that "We never had to do anything like this before" exclaimed one of the mechanics. When one of the mechanics slid a 30 mil feeler gauge under the inside edge at one of the inboard feet everyone stopped what they were doing as he said, "Uh, you better take a look at this because something is not right here." That was the understatement of the day. As it turned out there was a soft foot condition at all four feet of the motor which took the remainder of the shift to correct. Not a single person flipped out because we didn't get through the eight steps scheduled for our shift. No one from production asked why the machine wasn't ready for the scheduled post realignment test run. Instead the conversation was centered around the fact that the laser alignment system was unreliable in determining a soft foot condition and that some major changes were needed in the alignment procedures and that everyone needed additional training in alignment. The final test run showed that the axial vibration of the pump had decreased substantially. Without the resolve of all of the people at all levels from the Plant Manager to the apprentice mechanic in this particular facility, this work would have never been done. That is the kind of commitment that is needed to elevate an organization to superior maintenance excellence. It takes just one or two key personnel to choke this process.

Sincerely . . . yours truly

Sorry, I can get carried away responding to these types of letters. I suppose I could show you dozens of other letters like this but I am sure you get the idea. It seems somewhat disheartening to read about the plight of these individuals and situations they convey. I get the sense that these problems are quite commonplace and I applaud the honesty of these people to be forthright in describing their situation.

1.4 EIGHT BASIC STEPS TO ALIGN MACHINERY

Below are the eight basic steps involved in aligning rotating machinery. The "Tasks" section summarizes the items that should be done in each step and the "Time requirements" section attempts to give you an idea how much time each step will take to perform. If you do each step in the process, your chance of aligning successfully is pretty good. Decide to skip a few steps here and there and you are apt to run into some trouble. It is your choice.

Step 1

Tasks: Get the tools you need and train the personnel who will be performing the alignment

Purchase or fabricate the necessary tools and measuring devices. Insure that the people involved in the alignment process have been adequately trained on various alignment procedures and techniques, how to care for delicate measuring instruments and how to use them, what tools should be used to reposition the machinery, whether a machine is really ready to be aligned and operated or whether it should be removed and rebuilt, when a baseplate or foundation has deteriorated to the point where repairs are needed or corrections should be made, correcting problems that exist between the underside of the machine case and the points of contact on the baseplate, how to check for static and dynamic piping stress, what the desired off-line machinery positions should be, how to measure OL2R machinery movement, what the alignment tolerance is for the machine they are working on, and how to keep records on what was done during the alignment job for future reference.

Time requirements: The information contained in this book is not taught in any K-12 educational system, so it is unrealistic to expect a high school or college graduate, lacking such training, to know how to perform an alignment job. The burden of training personnel to

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understand and apply this body of knowledge falls on the employer and the companies who own and operate their machinery or on dependable outside personnel if such maintenance is contracted rather than done by in-house forces. For someone who has no experience in machinery alignment, it is recommended that somewhere between 24 and 40 h of classroom training is needed to educate the person on this information and give the person opportunity to practice alignment on simulation training devices rather than production machinery. After the classroom training has been completed, it is suggested that the individual who attended the course performs an alignment job on a process drive system within a short period of time. Someone who is experienced in aligning machinery should oversee the trainee and enforce the material taught in the course with on the job training. Since there are a wide variety of different alignment techniques taught in this book, one field alignment job is not going to be sufficient to learn every method. The progress of each individual should be tracked as they become more knowledgeable in this field of endeavor. On the average, it usually takes 2–5 y for someone to become proficient at machinery alignment assuming they encounter alignment tasks about once a month.

Step 2

Tasks: Obtain relevant information on the equipment being aligned

Are there any special tools needed to measure the alignment or reposition the machines? Do the machines move from off-line to running conditions? If so, how much and do you have to purposely misalign them so they move into alignment when they are running? Procure any historical information on the drive system to prevent yourself from having to discover problems that have already been detected and corrected.

Time requirements: Gathering information about each drive system in your plant can take a considerable amount of time and effort. Also getting the machinery nameplate data, coupling information, bearing information, shaft diameter information, ideal shaft-to-shaft distances, OL2R machinery movement data, recommended and alternative alignment measurement methods, historical records on the drive system, runout data, current soft foot shims, current final shim packs, shim sizes and material, final alignment measurements (e.g., the last alignment readings obtained and maybe a complete history of "as-found" and "final" readings), bolt torques, wrench sizes for foot and coupling bolts, piping stress tests, current and historical vibration data, normal bearing and coupling temperatures, number of operating hours since the last work was performed, type of lubrication for the bearings and coupling, how much lubrication should be installed, how often the lubricant should be refreshed, correct direction of rotation, safety tag lockout procedures, and an alignment task checklist.

Where would you find this information on all the rotating machinery in your plant? Is it readily available for everyone to use and can you get to it quickly? Who is responsible for entering and updating the information? Should it be kept in a printed book and kept in a library or should it be put on a computer database program and available over a network, or both? To be honest, I do not know of many facilities that have anything close to having this information in printed book form or on a computer network. Most often, the people who go out to align a piece of machinery approach the job as if they are doing it for the first time even though the drive system may have been in operation for decades. No records, no information, no help, no clue. Why do we have to start from square one every time we do this?

Step 3

Tasks: Work safe

Before you begin working on any machinery remember safety first. Properly tag and lock out the equipment and inform the proper people that you are working on the machine. Obtain any required work permits and perform flammable gas tests in the work area.

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Time requirements: I cannot emphasize the need for safety enough here. There are times when I have been negligent about properly safety tagging the equipment before I start working on it. Rotating machinery can hurt or even kill you or your coworkers. Please be extra careful when handling this machinery.

Probably the best credo is LO TO TO (Lock Out, Tag Out, Try Out). Insure that you have identified the right equipment that you will be working on by talking to the operations personnel. It may be a good idea to walk out to the equipment with someone from operations and point at the machine so there is no mistake which one you are talking about. Has the person indicated to you where the disconnection hardware is and the correct procedure for locking it out? Is the machine running now? If it is and you shut it down, what effect will that have on the process? Ask if the equipment can be started remotely by a computerized control system that may bypass local switches near the drive system. If it can, how do you prevent that from happening while you are working on it?

If you are working on an electric motor, where is the local disconnect switch? Is it labeled correctly? Once you disarm (Lock Out) the disconnect, and hang your safety tag on it (Tag Out), then try to start it (Try Out). Won't start? Good, you did it right.

If you are working on a centrifugal pump, is there a possibility of the pump rotating backward if the suction and discharge valves are open? If you are working on a fan, is there a possibility of the fan rotating if the dampers won't close all the way? In other words, think about all the possible things that could go wrong and stop them from happening.

Depending on the complexity of the drive system and its controls, Lock Out, Tag Out procedures will take anywhere from 30 min to several hours to complete. It may end up being the best time you ever spend, unless of course you prefer to spend your time in a hospital or eternity someplace else.

Step 4

Tasks: Conduct any preliminary checks before starting the alignment

Perform bearing clearance or looseness checks, measure shaft and coupling hub runout, inspect the coupling for any damage or worn components, find and correct any problems with the foundation or baseplate, find and correct any soft foot conditions, and find and eliminate any excessive piping, conduit, or ductwork stresses on the machines.

Time requirements: This is perhaps the most overlooked step in the alignment process and possibly the most important. The vast majority of people who align rotating machinery skip this step. Then they start having problems later on in the ensuing steps only to discover there is one or more than one thing in here that was causing them their grief. There is an entire chapter devoted to the items listed above so I will not dwell on this here. It is not unusual to spend anywhere from 2 h to 2 weeks finding and correcting these problems depending on what you find and how committed you are to fixing them.

Step 5

Tasks: Measure the amount of misalignment

First, rough align the machinery and check that all of the foot bolts are tight. Then measure the shaft positions using accurate measurement sensors (± 0.001 in. or better) such as dial indicators, laser detectors, proximity probes, optical encoders, charge couple devices, or some other types of precise sensors. From these data, determine if the machinery is within acceptable alignment tolerances.

Time requirements: Regardless of the type of measuring instrument and associated tooling you use, it is going to take somewhere between 15 min and 1 h to mount the fixtures on the shafts, set up the instruments, rotate the shafts to capture a set of measurements, determine

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your misalignment conditions, and calculate your alignment deviation. This typically is not where the majority of time is spent in the alignment job so do not be in a rush. Make sure you get accurate measurements since how you plan on correcting the misalignment problem depends on how accurate your measurements are in this step.

Step 6

Tasks: Correct the misalignment condition

If the machinery is not within adequate alignment tolerances, first, determine the current positions of the centerlines of rotation of all the machinery; then, observe any movement restrictions imposed on the machines or control points; next, decide which way and how much the machinery needs to be moved; and finally, go ahead and physically reposition the machines in the vertical, lateral, and axial directions. After you have made a move, be sure to check the alignment as described in Step 5 to determine if the machines really moved the way you hoped they did. When the final desired alignment tolerance has been satisfied, record the final alignment position for future reference, the orientation of the soft foot shim corrections, and the final shim packs used to adjust the height of the machinery. If lateral and axial jackscrews exist, touch each jackscrew against the side of the machine case, then draw them out so there is a 10–20 mils gap between the machine case and the tip of the jackscrew, lock the jackscrew in that position, and make sure the foot bolts are secured.

Time requirements: The amount of time spent to correct the misalignment is often directly related to the size of the machinery that you are working on. For a 100 hp or smaller drive system, it may take you anywhere from 1 to 3 h to correct the misalignment. If you are attempting to align a 50 t steam turbine and a 75 t generator, it may take you 12 h or more to install just one set of shims and move the equipment sideways to correct the problem. That is not to say that 100 hp or smaller equipment is going to be a cakewalk. If you change the position of a machine, you need to go back to Step 5 and measure the shaft-to-shaft alignment again to insure the move was successful. Besides Step 4 (the preliminary steps), here is typically where the majority of time is spent on an alignment job.

Step 7

Tasks: Get ready to operate the machinery

Make sure the drive system is ready to run before you remove the safety tags. Make sure the correct type and amount of lubricant is in the bearings of all the machinery in the drive system. Insure all the foot bolts are tight. If you replaced an electric motor, you may want to temporarily remove the disconnect Lock Out tag and insure the motor is rotating in the correct direction. If you were working on a steam or gas turbine, you may want to check overspeed trip settings. With the driver disconnected from the driven unit, now might be a good time to measure vibration levels on the bearings (as well as any other performance parameters). After the driver ''solo'' test runs are complete, install the Lock Out tag again. Make sure the shaft end to shaft end distance is correct for the coupling. Assuming the coupling was inspected for damaged components during the alignment job, replace any defective parts, reassemble the coupling, and check for rotational freedom of the entire drive train if possible. Make sure any connections (piping, wiring or conduit, ductwork, etc.) to the machinery is secure. Install the coupling guard and make any final checks on the drive train prior to removing the safety tags.

Time requirements: Depending on what items you are going to do, this step will usually take anywhere from 30 min to 3 h to perform assuming weather is not a factor.

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Step 8

Tasks: Start the drive system and insure it is operating satisfactorily

Operate the unit at normal conditions checking and recording vibration levels, bearing and coupling temperatures, bearing loads, flow rates, suction and discharge pressures, current drawn, and other pertinent operating parameters.

Time requirements: Assuming you have healthy machinery, the balance characteristics of the rotors are good, the lubrication is the right kind and it is getting where it needs to be, and the operational parameters are normal, now is a great time to capture "baseline data" on the equipment for future comparison. Baseline data describe the point where the mechanical and operational performance is at the best it can be. The type of information that may be collected might include vibration, temperature, pressure, flow, and power consumption, to name a few. Over time, as the components begin to wear, these parameters may change. To extract the greatest amount of performance from the drive system, it is important to know what a healthy machine looks like and at what point should the equipment be shut down for corrective actions to prevent a catastrophic failure from occurring. Depending on what data you want to collect, this step may take anywhere from 30 min to 2 h to perform.

1.5 HOW QUALIFIED ARE YOU TO DETECT AND CORRECT MACHINERY MISALIGNMENT?

Purchasing an alignment measurement system does not necessarily qualify you as an alignment expert. What is the best method to measure misalignment under different circumstances? What effect do excessive soft foot, runout, and piping stress have on rotating machinery? What are the different levels of testing that can be performed at your plant to determine the skill level of the people responsible for alignment? What sort of training is required to become proficient in this area? The effectiveness of written or oral exams, simulated misalignment test, and actual on the job skills testing on process machinery will henceforth be discussed.

1.6 WHY SHOULD PEOPLE BE TESTED ON THEIR ALIGNMENT SKILLS?

Perhaps several true life examples will illustrate the need for certification and qualification testing in machinery alignment.

- A maintenance technician at a chemical plant was asked to align a motor and a pump with a newly purchased laser shaft alignment system. Shaft position measurements were captured with the instrument and the alignment corrections required to align the motor (assigned as the movable machine) to the pump indicated that the outboard end of the motor had to be lowered 85 mil and the inboard end of the motor had to be lowered 37 mil and there was no shim stock under any of the motor feet. After completely removing the motor, the technician began grinding the baseplate away. The motor was placed back on the base and shaft position measurements were captured again. The technician then added shims under the motor since too much metal had been ground away and several side-to-side moves were made to bring the equipment into alignment.
- A company was in the process of installing several large air compressors to expand the capacity of the compressed air system. Requests for bids were sent out to several general contractors to install the 11,000 hp motors, gearboxes, and compressors. Very detailed specifications were sent to each of the general contractors including very specific instructions on installation of the foundations, sole plates, correcting soft foot conditions, rough

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alignment procedures, final cold alignment procedures, and hot alignment procedures. The general contractor was told to subcontract the alignment work to companies who specialize in machinery alignment. When the alignment specifications were sent to the alignment subcontractors, several of the contractors did not understand many of the detailed specifications for the required work but submitted bids anyway believing that no one would check their work. The contract awards were dispensed and toward the completion of the project the company discovered that the alignment work was not performed to the written specifications and withheld payment to the contractors who performed the work (or lack thereof).

- A company, which was in the process of becoming ISO compliant, requested information on who administers certification testing for their maintenance personnel for shaft alignment. Several of their people had been certified in vibration analysis and thermography and they wanted to show documentation that their personnel were adept at not only finding problems but also adept at fixing them.
- A petroleum company decided to sell one of their facilities and several of the prospective buyers were interested in retaining as many employees as possible. However they only wanted to hire the people who were adequately trained and were certified to do certain tasks. When asked to provide information on task certification of their employees, the petroleum company was unable to provide the information.
- A steel company was having some problems with a fairly complex, multiple element drive train where misalignment was found to be the root cause of the failures. No one in the plant knew how to align the drive system. An alignment service company was contacted and the technician said that they could align the drive system in less than 4 h. When the work was actually done, the job took several days to complete.
- An electric utility company had experienced several failures on a critical pump. The in-house maintenance personnel had been using a laser shaft alignment system to measure the positions of the shafts. The pump was driven by a variable speed hydraulic clutch and the manufacturer of the clutch stated in the instruction manual that the clutch would rise upward 15 mils once it attained normal operating conditions. The maintenance personnel henceforth set the clutch 15 mils lower than the pump shaft assuming that the pump would not move from off-line to running conditions. A survey was conducted that showed that the pump shaft rose upward far more than the clutch did forcing the unit to run under severe misalignment conditions.

Most of us are keenly aware that purchasing this equipment does not mean that we know how to use it. Buying a vibration analyzer does not make one a vibration analyst, nor does purchasing an infrared imaging camera qualify someone as knowledgeable in thermography, or a scalpel magically transform someone into a surgeon. The learning curve for this equipment and the knowledge to use it properly is long and steep. Determining what buttons to push, how to operate the software program that came with the instrument as well as learn how to use a computer can consume several months of time. Once you have figured out how to operate the equipment and begin to collect data, the next task is to ascertain what the data is telling you, so some sort of judgment on the condition of your machinery can be made. This part can take years to become proficient at as the equipment being monitored at your plant does not necessarily follow generic equipment deterioration patterns, and proper condition assessment can only be realized with experimentation and experience. During its infancy, many CBM programs have disappointed people because one or two bad calls were made, assessing incorrectly the condition of the machinery. People can get upset after disassembling a machine to find that nothing or very little was wrong, when it was made to sound like the machine was about to catastrophically fail.

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FIGURE 1.5 Excessively worn gear coupling due to operation under moderate misalignment conditions.

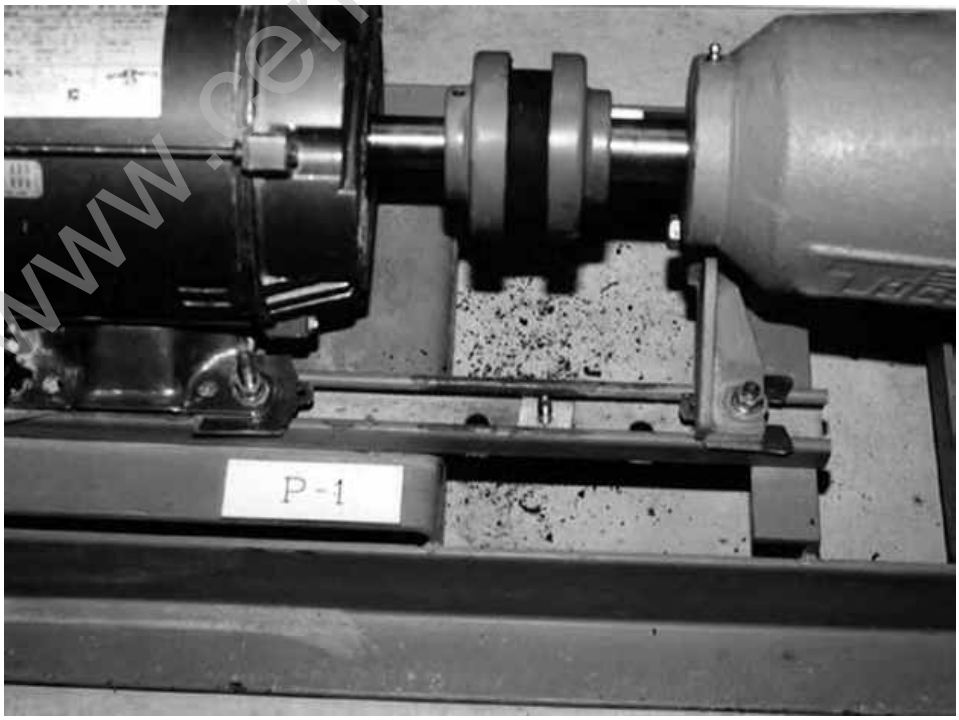


FIGURE 1.6 Rubber dust under coupling on a misaligned motor and pump after running intermittently for 2 months.

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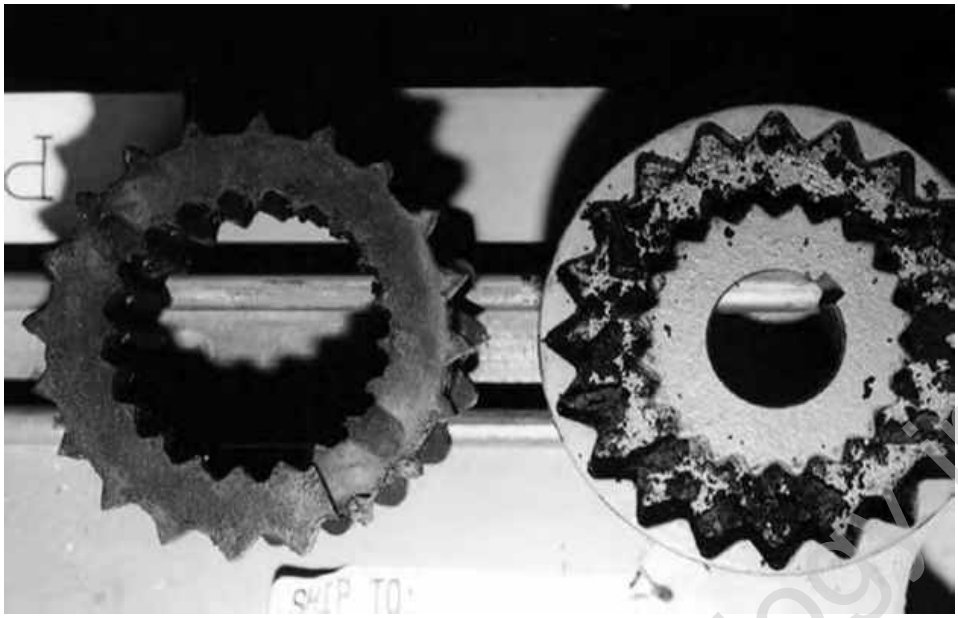


FIGURE 1.7 Coupling hub and rubber insert from drive system shown in Figure 1.6.



FIGURE 1.8 Worn gear coupling hub.

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So it became apparent that these condition-based and predictive maintenance programs only worked effectively after the people attended some training courses. Some people learned a lot, some learned a little, and some had their minds occupied elsewhere during the training courses. Those who worked hard and learned a lot started wondering how they could prove that they knew more than those who did not. Certification testing appeared to provide the best proof of a person's skills in vibration analysis and infrared thermography. Some people may disagree with the need for testing, but it has become a way of life and many companies are requiring their employees to become certified in these nondestructive testing methods. In fact, this made people begin thinking about certification for several other kinds of tasks performed in the industrial work place. If certification is desirable for methods to detect problems with rotating machinery, then certification should also be desirable for correcting problems with rotating machinery such as balancing, shaft alignment, and tribology, for example. How qualified are people in finding and correcting misalignment conditions on rotating machinery? As more companies are turning to task-based training and task-based testing, what is the best way to determine if employees and contractors are suitably qualified to do machinery alignment?

1.7 EXPERIENCE EVALUATION FOR MACHINERY ALIGNMENT

Before qualification testing begins, you should have a fairly good idea of how knowledgeable you, your coworkers, or the contractors you hire are in the field of machinery alignment already, so unnecessary training is not done. Assessing what you and your coworkers know about shaft alignment can be done by soliciting the employees through an experience evaluation form that queries their knowledge and experience on specific types of machinery and specific types of tasks performed on this machinery. The next step is to test each individual on the tasks that they said they were adept at to determine if they are truly capable of performing that task and if not, what supplementary training would be required to elevate their level of proficiency. In summary, what do they think they know, what do they actually know, and what do they need training in?

1.8 WHO NEEDS TO BE TRAINED AND QUALIFIED IN SHAFT ALIGNMENT?

Many organizations feel that the full responsibility for machinery alignment rests solely in the hands of the trades people (e.g., mechanics, millwrights, pipe fitters, electricians). But should it be the responsibility of a trades personnel to:

1. Pick training courses they feel they need and send themselves to the courses?
2. Research all of the available types of shaft alignment measurement systems and purchase a system that best fits the needs of their organization?
3. Tell a contractor that they are not installing new rotating machinery correctly?
4. Hire some new people or a contractor to help with the work overload?
5. Rebuild a piece of rotating machinery due to excessive runout conditions?
6. Determine that a rotating machinery foundation or baseplate has to be removed and reinstalled if it has deteriorated excessively or been installed improperly?
7. Redesign and rework improperly installed piping that is putting an excessive strain on the rotating machinery it is attached to?
8. Purchase and install piping supports or design a custom piping anchor on a computer-aided design system, purchase the materials, and install the anchor?
9. Select a new flexible coupling design to replace one that does not work well or fails often?

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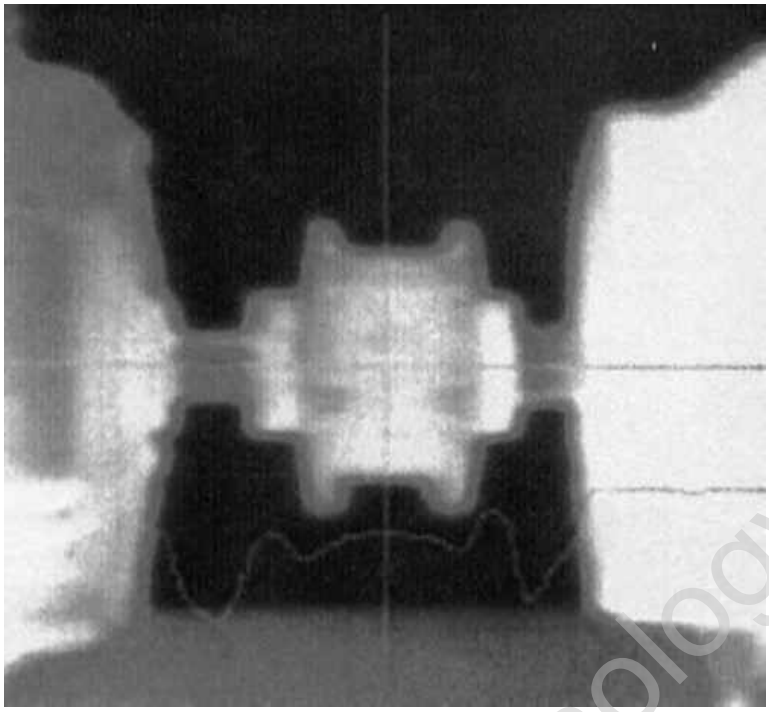


FIGURE 1.9 Infrared thermal image of misaligned metal ribbon coupling operating under a misalignment condition. (Courtesy of Infraspection Institute, Shelburne, VT.)

10. Pick a pump to be the "movable" machine and leave the motor as the "stationary" machine?
11. Issue work orders to check the alignment of all the rotating machinery every year?
12. Shut a machine down based on vibration and temperature data that indicates a misalignment or a soft foot condition?
13. Determine which machinery might need to have the hot alignment checked, select an OL2R machinery movement measurement technique, install the equipment on the machinery, measure and analyze the data, and alter the cold alignment position based on the data collected?
14. Maintain records of the alignment work that was performed and save it in the equipment files or computer database?
15. Install *X–Y* proximity probes on a machine supported in sliding-type bearings to analyze the Lissajous orbit for signs of running misalignment?

I would not be surprised if a lot of managers, engineers, front line supervisors, vibration technicians, and trades people felt that some, if not several, of the above items are not their responsibility (let alone what some of them mean). If the trades people are not responsible for many of the above items, then who should be?

Machinery alignment training should be for managers (who cannot understand why alignment takes longer than 45 min), engineers (who were never taught this in school and are rarely allowed to do hands-on work), technicians (who are asked to identify it through CBM programs but do not know how to correct it), front line supervisors (who need to help the trades people if they get stuck or need a special tool for the work), and trades people (who are the ones ultimately responsible for the final outcome) to give them the minimum working knowledge to achieve accurate alignment and to let them know what is involved in the overall alignment process. As it is apparent that trades personnel should not be accountable for every facet of shaft alignment, several people need to be trained and qualified in whatever tasks they

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are responsible for. Engineering and maintenance managers, rotating equipment and maintenance engineers, maintenance technicians, vibration specialists, foremen and front line supervisors, as well as the trades personnel all should be trained and qualified to do their respective tasks.

1.9 ASSESSING A PERSON'S KNOWLEDGE AND EXPERIENCE LEVEL IN SHAFT ALIGNMENT

Field experience evaluation forms similar to those shown in Figure 1.10 through Figure 1.14 could be used for your in-house personnel or for contractors working at your plant site.

Rotating machinery and shaft alignment field experience evaluation

Part 1. Background information.

Name: _____
Company: _____
Address: _____
City: _____ ST: _____ Zip: _____
Phone: _____ FAX: _____
E-Mail: _____

Current job title or responsibility

Number of years — Job description

_____ Plant manager
_____ Engineering manager
_____ Maintenance manager
_____ Engineer
_____ Front line supervisor/foreman/planner
_____ Technician
_____ Mechanic
_____ Millwright
_____ Pipefitter
_____ Electrician
_____ Electronic repair/instrumentation
_____ Other (please list) _____

Past job titles or responsibilities

Number of years — Job description

_____ Plant manager
_____ Engineering manager
_____ Maintenance manager
_____ Engineer
_____ Front line supervisor/foreman/planner
_____ Technician
_____ Mechanic
_____ Millwright
_____ Pipefitter
_____ Electrician
_____ Electronic repair/instrumentation
_____ Other (please list) _____

FIGURE 1.10 Shaft alignment experience evaluation form—Part 1.

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Rotating machinery and shaft alignment field experience evaluation

Part 2. Rotating machinery experience.

You have been working on the following types of industrial rotating machinery for __X__ number of years.

Number of years	Machinery type
_____	Synchronous motors
_____	Induction motors
_____	Steam turbines
_____	Gas turbines
_____	Water turbines
_____	Internal combustion engines
_____	Up to 50 hp
_____	50 to 200 hp
_____	200 to 1000 hp
_____	1000 to 5000 hp
_____	5000+ hp
_____	Single stage centrifugal pumps
_____	Multiple stage centrifugal pumps
_____	Centrifugal air compressors
_____	Reciprocating air compressors
_____	Screw or sliding vane air compressors
_____	Gearboxes
_____	Fans
_____	Blowers
_____	AC generators
_____	DC generators
_____	Cooling tower fan drives
_____	Multiple element drive trains
_____	Other machinery (please list)

_____
_____
_____

FIGURE 1.11 Shaft alignment experience evaluation form—Part 2.

Figure 1.10 (Part 1) identifies the person and what his or her job titles are (or have been). Figure 1.11 (Part 2) gives an indication of what type of rotating machinery a person has worked on in the past. Figure 1.12 (Part 3) gives some idea of the training a person has received in different alignment measurement tools and techniques. Figure 1.13 (Part 4 and Part 5) lists the amount of actual field experience a person has in measuring shaft alignment when the equipment is off-line and the amount of actual field experience a person has in measuring off-line to running machinery movement.

1.10 ALIGNMENT QUALIFICATION OR CERTIFICATION TESTING

The experience evaluation forms can be used to determine what training is required for personnel installing, maintaining, or aligning rotating machinery. However anyone could fill out the form and say they were proficient and experienced working with every type of rotating machine and alignment method listed but how can they verify what they say they know? Written or oral examinations are one mechanism to verify a person's knowledge level for each item in the experience evaluation forms. There could be one comprehensive test

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Rotating machinery and shaft alignment field experience evaluation

Part 3. Shaft alignment training.

I have received training in the following shaft alignment methods, procedures, or tasks. Formal training means you attended a training course, on the job training (OTJT) means you where shown this information by a coworker or you were self taught.

Formal	OTJT	Shaft alignment method(s), procedures, tasks
_____	_____	Foundation and baseplate/soleplate installation and repair
_____	_____	Soft foot detection and correction
_____	_____	Excessive runout detection and correction
_____	_____	Excessive piping strain detection and correction
_____	_____	Straightedge, eyeball, feeler gauge
_____	_____	Face–rim indicator
_____	_____	Reverse indicator
_____	_____	Shaft to coupling spool
_____	_____	Double radial
_____	_____	Face–face
_____	_____	Laser/detector system(s)—model(s)_____
_____	_____	Calculating growth using the thermal strain equation
_____	_____	Inside micrometer/tooling ball/angle measurement devices
_____	_____	Proximity probes with water cooled stands
_____	_____	Optical alignment tooling
_____	_____	Alignment bars and proximity probes (aka Dodd/Dynalign bars)
_____	_____	Laser/detector system(s)—model(s)_____
_____	_____	Ball-rod-tubing connector system
_____	_____	Vernier–Strobe method
_____	_____	Instrumented coupling system
_____	_____	Shaft alignment software programs
_____	_____	Other (please list)

_____
_____

FIGURE 1.12 Shaft alignment experience evaluation form—Part 3.

administered that encompasses every facet of machinery alignment or there could be a series of tests given for discrete blocks of information. By breaking down the overall body of information into separate blocks, the people with little or no experience could be tested incrementally as their level of knowledge grew.

A Basic or Level 1 examination would test the individual's knowledge on the following blocks of information:

1. The detrimental consequences of poor alignment on rotating machinery
2. Detecting misalignment on running rotating machinery (i.e., vibration, infrared methods)
3. Use and care of measuring tools and instruments (feeler gauges, dial indicators, optical encoders, laser-detector system, proximity probes, etc.)
4. Finding and correcting excessive runout conditions
5. Finding and correcting soft foot problems
6. Finding and correcting excessive piping strain
7. Foundation and baseplate design, installation, and care
8. Concrete and grouting installation
9. Alignment tolerances

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Rotating machinery and shaft alignment field experience evaluation

Part 4. Shaft alignment field experience.

I have performed the following off-line shaft alignment methods for __X__ number of years.

Number of years	Shaft alignment methods you have used
_____	None
_____	Eyeball
_____	Straightedge, eyeball, feeler gauge
_____	Face–rim indicator
_____	Reverse indicator
_____	Shaft to coupling spool
_____	Double radial
_____	Face–face
_____	Optical alignment tooling
_____	Laser/detector system(s)—model(s) _____
_____	Other (please list)

_____
_____

Part 5. Off-line to running (OL2R) machinery movement measurement and alignment compensation methods hot and cold alignment) field experience.

I have performed the following OL2R methods for __X__ number of years.

Number of years	OL2R machinery movement measurement methods you have used
_____	Calculating machine case thermal expansion (thermal strain equation)
_____	Inside micrometer/tooling ball/angle measurement devices
_____	Proximity probes with water cooled stands
_____	Optical alignment tooling
_____	Alignment bars & proximity probes (aka Dodd bars/Dynalign system)
_____	Laser/detector system(s)—model(s) _____
_____	Ball-rod-tubing connector system
_____	Vernier–Strobe method
_____	Instrumented coupling system
_____	Other (please list)

_____
_____

FIGURE 1.13 Shaft alignment experience evaluation form—Part 4 and Part 5.

10. Rigid and flexible coupling design, installation, and care
11. How to perform the reverse indicator method
12. Basic mathematical- or graphical-modeling principles for realignment
13. How to determine effective alignment corrections using the reverse indicator technique
14. Keeping records of alignment work

An Intermediate or Level 2 examination would test the individual's knowledge on the following blocks of information:

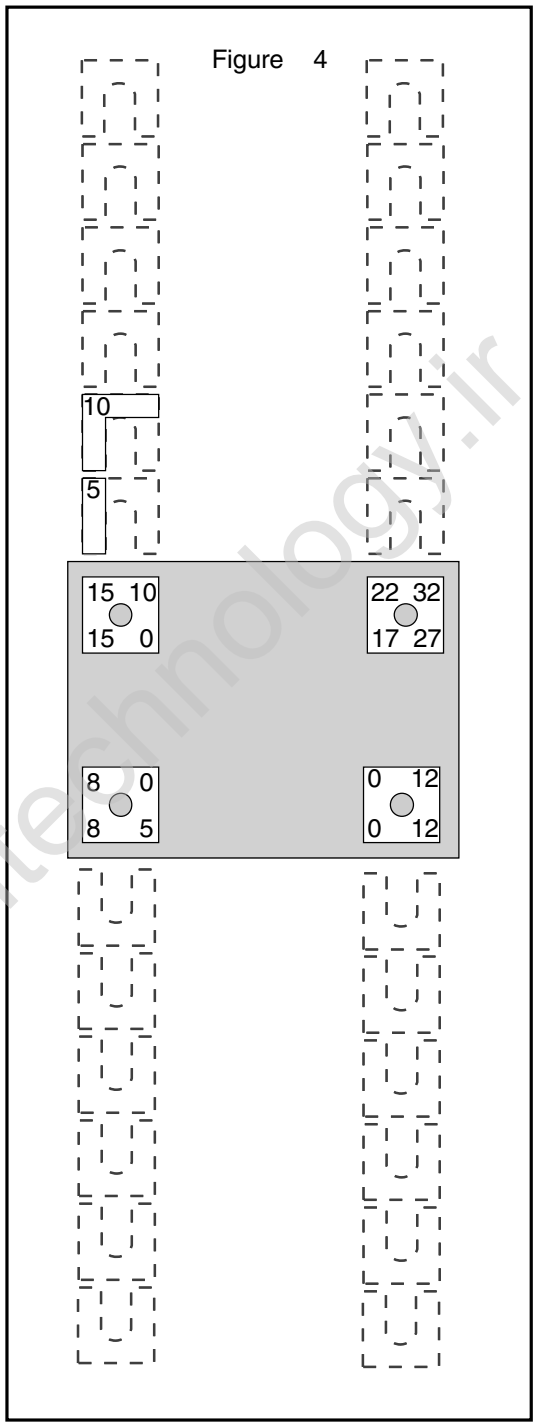
1. How to perform and determine effective alignment corrections using the face and rim technique.
2. How to perform and determine effective alignment corrections using the shaft to coupling spool technique.

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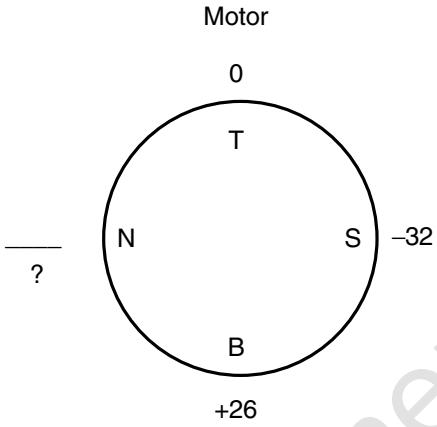
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Basci / Level 1 Certification Test

15. Show how you would correct the soft foot conditions found on an electric motor as shown in Figure 4. Draw in the shape of the shim(s) and the shim thickness to correct the soft foot conditions at the other three feet. Standard shim thicknesses are 1, 2, 3,4, 5, 10, 15, 20, 25, 50, 100, 125 mils. (6 pts)



16. Fill in the missing dial indicator reading. (2 pts)



17. A motor and pump running at 3600 rpm has a maximum misalignment deviation of 1.9 mils per inch. Is that an acceptable amount of misalignment? (2 pts)

a. yes
b. no

FIGURE 1.14 Sample question from a Level 1 shaft alignment certification exam.

3. How to perform and determine effective alignment corrections using the double radial technique.
4. How to perform and determine effective alignment corrections using the face–face technique.
5. How to generate mathematical- or graphical-modeling principles for all of the above methods.

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An advanced or Level 3 examination would test the individual's knowledge on the following blocks of information:

1. How to align multiple element drive trains.
2. How to align right angle drives.
3. The four general categories for measuring OL2R machinery movement (hot and cold alignment)
4. Calculating machine case thermal expansion
5. Inside micrometer-tooling ball–angle measurement methods
6. Proximity probes with water cooled stands technique
7. Using optical alignment tooling for OL2R machinery movement
8. Alignment bars with proximity probes OL2R method
9. Using laser-detector systems to measure OL2R machinery movement
10. Using the ball-rod-tubing connector system to measure OL2R machinery movement
11. Using the Vernier–Strobe system to measure OL2R machinery movement.
12. Mathematical- or graphical-modeling principles for all of the above methods
13. How to compensate for OL2R machinery movement when aligning rotating machinery

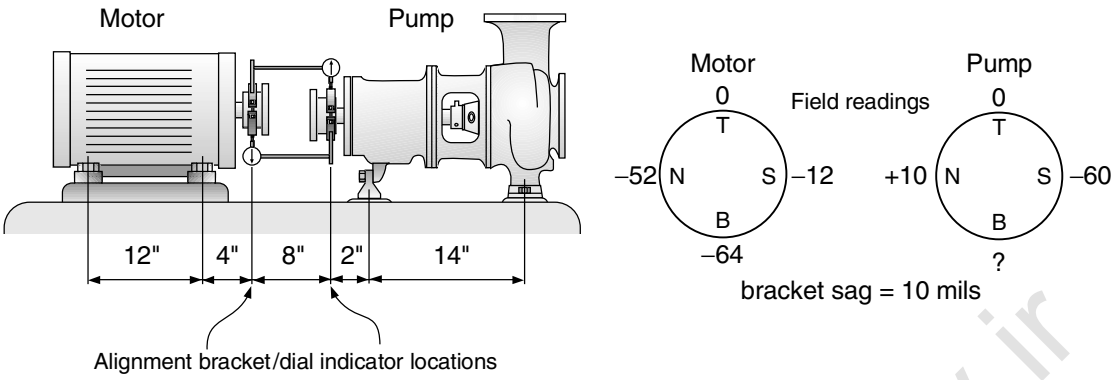
Figure 1.14 and Figure 1.15 show sample problems that might appear on a Level 1 exam. Written or oral exams can test a person's knowledge on a subject but is somewhat inadequate in determining a persons skill level in performing specific tasks. Machinery alignment is a hands-on task. Someone can understand how the reverse indicator alignment method works but not be able to perform the method correctly. Perhaps the most effective means of verifying a person's knowledge and skill level is to have the person perform the task on a simulation machine or directly on an operating rotating equipment drive system at the plant site. However using process machinery as a test platform may not be possible. What if the equipment cannot be shut down for a scheduled test time, or what if the person being tested misaligns the machine by mistake and no one checks to see that it is correct? By having simulation equipment available, testing can occur at virtually any time without impacting production or maintenance schedules. For accurate skills assessment, it is important to insure that the test equipment simulate real life conditions.

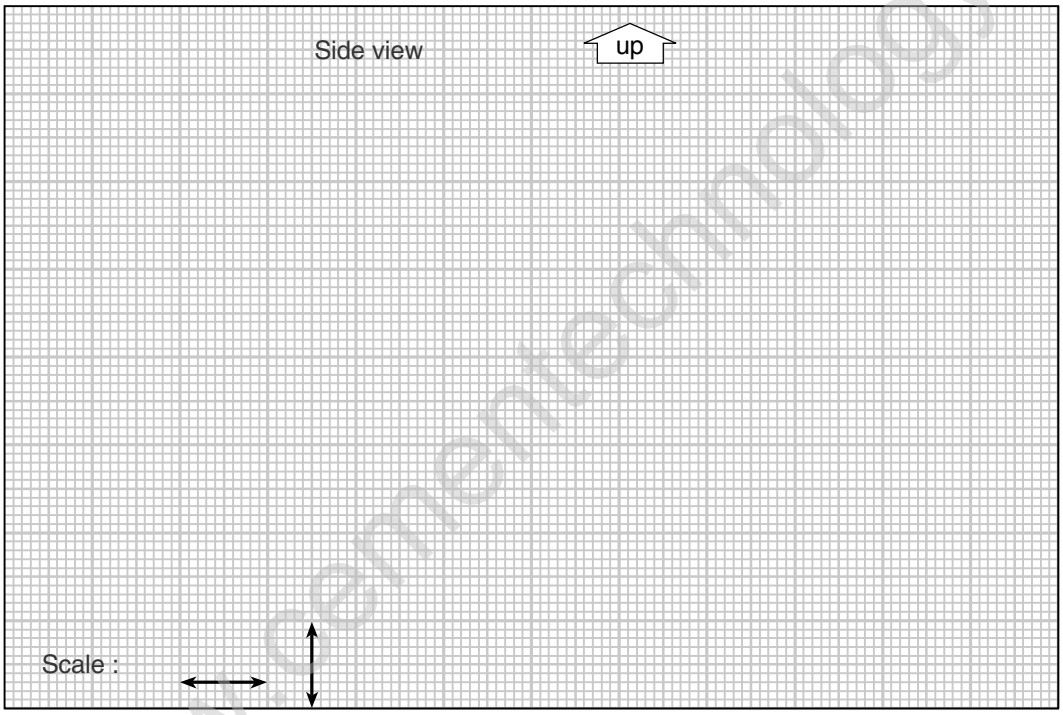
In today's society, specialized jobs in many different areas require some sort of certification to verify that an individual is proficient in that task. In the health, education, and finance industries, for example, doctors, nurses, teachers, and accountants must show that they are certified in those areas before they are even considered for employment. In industry, welders must be certified before they can work on pressure vessels and heating, ventilation, and air conditioning (HVAC) technicians must be certified to work with Freon refrigerants. You cannot even drive a car or fly a plane without a license. The need for qualification or certification testing in specialized tasks such as vibration analysis, thermography, and shaft alignment is justifiable. Establishing the requirements for qualification or certification can be accomplished by appraising the experience level of personnel through an evaluation form that addresses all of the aspects of the task. Skill requirements for each individual can then be assessed and appropriate training can be administered. Written or oral exams and task simulation tests can then be conducted to determine the true proficiency of the individual. Although most of us do not like to be tested on what we know (or think we know), appraising our skill level is beneficial to the companies or customers we all work for. Certification, qualification, and licensing examinations can be used as an effective tool in verifying an individual's level of proficiency in any specific task and a viable mechanism to let someone know where he or she needs additional training.

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20. Plot the position of the motor and pump shafts in the side view (up/down) direction from the following reverse indicator readings. (10 pts)





21. How would you correct the misalignment condition in the example shown above assuming there are no shims under any of the motor or pump feet?. (10 pts)

FIGURE 1.15 Sample question from a Level 1 shaft alignment certification exam.

1.11 PERIODIC ALIGNMENT CHECKS

Typically, it is assumed that once a piece of machinery has been aligned, it stays that way forever. What a surprise it is for many to learn that is not always true. Just as you should check the alignment on your personal vehicle periodically (typically recommended around 25,000 miles), the alignment of rotating machinery should also be checked periodically.

Here are some recommended guidelines. For newly installed machinery, the alignment should be checked between 500 and 2000 h of intermittent operation or 1–3 months of continuous operation. If there was no apparent shift in the alignment position then the next check should be made between 4500 and 9000 h of intermittent operation or 6 months to

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1 year of continuous operation. If no apparent shift occurred during that period of time then checks should be made every 2–3 years.

If a moderate shift in alignment occurred at any time, then the equipment should be aligned within acceptable tolerances. If a radical shift occurred, then additional investigation should be conducted to determine what is causing the shift to occur. The most common causes of

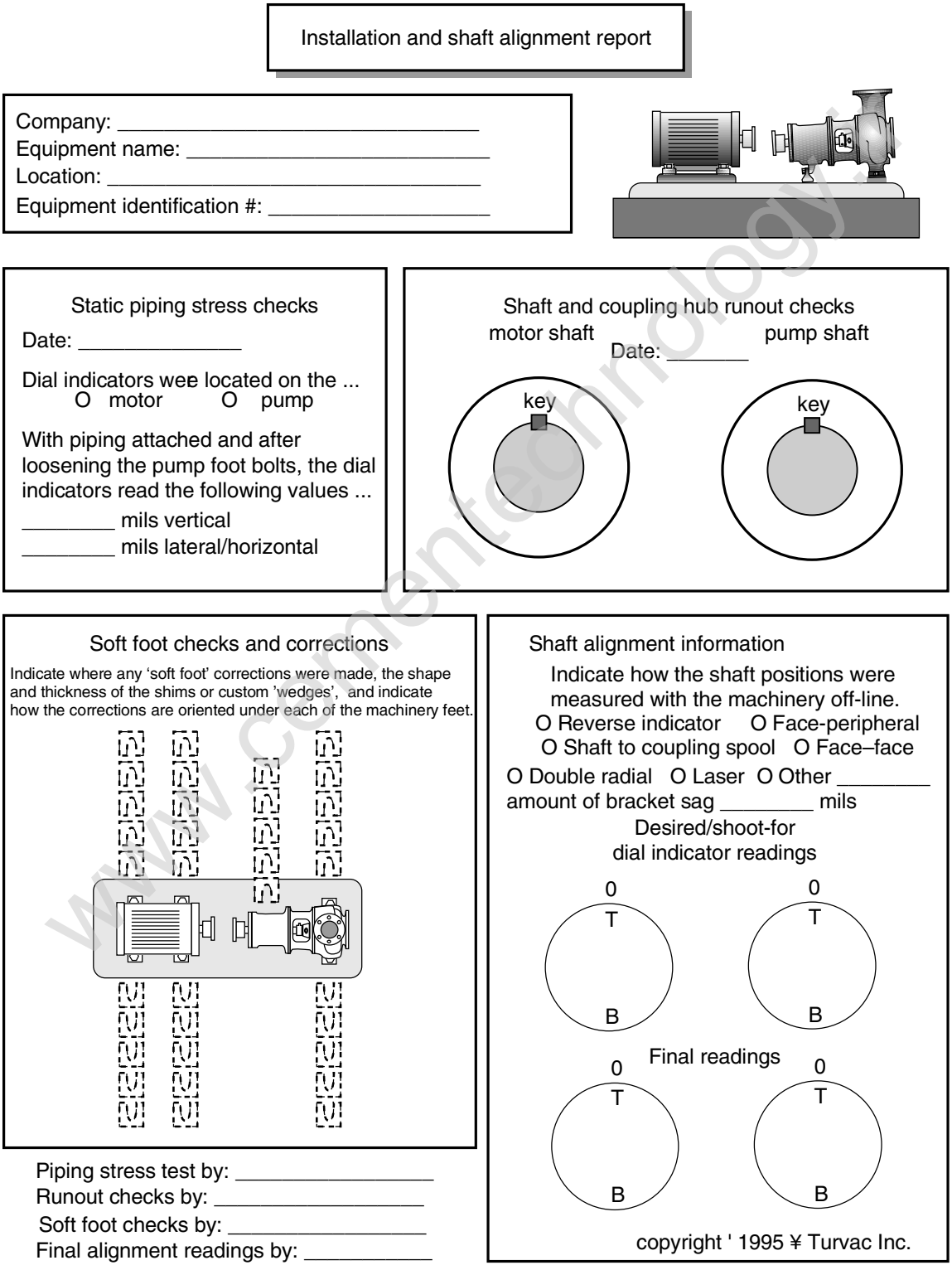


FIGURE 1.16 Sample alignment recording form.

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alignment shifts can be traced back to unstable foundations or baseplates, piping stress (both static and operational stresses), and inconsistent OL2R machinery movement. These problems are not easy to find and equally difficult to fix. All of the effort expended getting rotating machinery within acceptable alignment tolerances is worthless if the machinery cannot maintain its position over long periods of time.

1.12 ALIGNMENT RECORD KEEPING

I was once told, "If something is important enough to measure, then it's important enough to write it down and save it." Keeping historical records of the soft foot conditions, runout on the shafts and coupling hubs, piping stress checks, as-found and final off-line shaft alignment position measurements, and machinery dimensions can take some time and effort to compile and record but there is a good chance that this information might save you or one of your coworkers as much and possibly more time than it took to generate the records in the first place. Figure 1.16 illustrates an example of what type of information might be kept in the historical record file for a motor and pump.

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Guidebook for Maintenance Proficiency Testing, Electric Power Research Institute, December 1989, EPRI NP-6679.

Handbook for Evaluating the Proficiency of Maintenance Personnel, Electric Power Research Institute, March 1988, EPRI NP-5710.

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2 Detecting Misalignment on Rotating Machinery

It has long been known that one of the principal causes of damage to rotating machinery can be directly attributed to misalignment conditions. The primary cause of this damage can be traced back to excessive forces impinging on the rotors, bearings, seals, and housings. As there seems to be a direct link between how long the rotating machinery will operate and the presence of undesirable forces on the machinery, it seems prudent to reduce or eliminate as many of these forces as possible.

The capacity to detect and then correct misalignment on rotating machinery is directly related to the knowledge, skills, and desire of the personnel responsible for the equipment. If it is understood that accurate alignment is paramount for the long-term survival of rotating machinery, then it is somewhat of a dilemma trying to explain why misalignment continues to be a major source of damage to rotating machinery. The principal reason for this can be traced back to the fact that there are major philosophical differences on how people operate and maintain their equipment. For a moment, let us examine the different approaches organizations apply when operating and maintaining their industrial machinery.

2.1 THE FOUR MAINTENANCE PHILOSOPHIES

If you work long enough in the industry, you may get an opportunity to observe all the different "styles" of maintenance. How maintenance organizations operate their equipments usually fall into four different categories:

1. Breakdown or run-to-failure maintenance
2. Preventive or time-based maintenance
3. Predictive or condition-based maintenance
4. Proactive or prevention maintenance

2.1.1 Breakdown or Run-to-Failure Maintenance

The basic philosophy of this type of maintenance is to allow the machinery to run to failure and only repair or replace the damaged equipment when obvious problems occur. Studies have shown that the costs to operate in this fashion are about $18 per horsepower per year. The advantage of this approach is that it works well if equipment shutdowns do not affect production and if labor and material costs do not matter. Where would this apply?

The disadvantages are that the maintenance department perpetually operates in unplanned and "crisis management" maintenance activities with unexpected production interruptions and the plant must have a high inventory of spare parts to react quickly. Without a doubt, it is the most inefficient way to maintain a facility. Futile attempts are made to reduce costs by

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purchasing cheap parts and hiring cheap labor, further aggravating the problem. Frequently the personnel are overworked and the department is understaffed, and the workers arrive at work each day to be confronted with a long list of unfinished work and a half dozen new emergency jobs that occurred while they were at home in the evening. It is not uncommon to send someone out to work on an emergency job first thing in the morning and by ten o'clock, half way through the job, stop their progress and send them on a new higher priority emergency job.

Despite the wonders of modern life in the new millennium, I see many places like this. I hope you are not in one of them.

2.1.2 Preventive or Time-Based Maintenance

This maintenance philosophy consists of scheduling maintenance activities at predetermined time intervals where you repair or replace damaged equipment before obvious problems occur. If such maintenance is carried out correctly, studies have shown that the costs to operate in this fashion are about $13 per horsepower per year. The advantage of this approach is that it works well for equipment that does not run continuously and the personnel have enough knowledge, skill, and time to perform the preventive maintenance work.

The disadvantages are that the scheduled maintenance may be done too early or too late. It is quite possible that there may be a drop in production due to unnecessary maintenance work. In many cases there is a possibility of diminished performance through incorrect repair methods. I have witnessed perfectly good machines disassembled, good parts removed and discarded, and then new parts improperly installed. For some, squirting grease into bearings every month is their idea of a preventive maintenance program.

2.1.3 Predictive or Condition-Based Maintenance

This philosophy of maintenance consists of scheduling maintenance activities only if and when mechanical or operational conditions warrant by periodically monitoring the machinery for excessive vibration, temperature, lubrication degradation, or observing any other unhealthy trends that occur over time. When the condition gets to a predetermined unacceptable level then the equipment is shut down to repair or replace damaged components in the equipment to prevent a more costly failure from occurring. In other words, "do not fix what is not broke." If this kind of maintenance is performed correctly, studies have shown that the costs to operate in this fashion are about $9 per horsepower per year. The advantage of this approach is that it works very well if personnel have enough knowledge, skill, and time to perform the predictive maintenance work. The repairs to equipment can be scheduled in an orderly fashion and it allows some lead time to purchase materials for the necessary repairs, reducing the need for a high parts inventory. As maintenance work is only performed when it is needed, there is a likely increase in production capacity.

The disadvantages are that maintenance work may actually increase if the personnel improperly assess the level of degradation in the equipment. Observing the unhealthy trends in vibration, temperature, or lubrication requires the facility to procure equipment to monitor these parameters and provide training to in-house personnel. The alternative is to outsource this work to a knowledgeable contractor to perform predictive or condition-based duties. If an organization had been running in the breakdown or run-to-failure mode or the preventive maintenance style, the production and maintenance management must conform to this new philosophy, which can be problematic if the maintenance department is not allowed to purchase the necessary equipment, provide adequate training to the people to learn the new techniques, not given the time to collect the data, or not permitted to shutdown the machinery when problems are identified.

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2.1.4 Proactive or Prevention Maintenance

This maintenance philosophy utilizes all of the predictive or preventive maintenance techniques discussed above in concert with root cause failure analysis to not only detect and pinpoint the precise problems that occur but also to insure that advanced installation and repair techniques are performed including potential equipment redesign or modification to avoid or eliminate problems from occurring. If this kind of maintenance is done correctly, studies have shown that the costs to operate in this fashion are about $6 per horsepower per year. The advantages of this approach are that it works extremely well if personnel have enough knowledge, skill, and time to perform all of the required activities. As in the prediction-based program, repairs to equipment can be scheduled in an orderly fashion but then additional efforts are made to provide improvements to reduce or eliminate potential problems from repetitively occurring. Again, repairs to equipment can be scheduled in an orderly fashion and it allows lead time to purchase materials for the necessary repairs, reducing the need for a high parts inventory. As maintenance work is only performed when it is needed, and extra efforts are put forth to thoroughly investigate the cause of the failure and then determine ways to improve the reliability of the machinery, there can be a substantial increase in production capacity.

The disadvantages are that this requires extremely knowledgeable employees in preventive, predictive, and prevention or proactive maintenance practices or to outsource this work to a knowledgeable contractor who works closely with the maintenance personnel in the root cause failure analysis phase and then assists in the repairs or design modifications. This also requires procurement of equipment and properly training personnel to perform these duties. If an organization had been running in the breakdown or run-to-failure mode or the preventive maintenance style, the production and maintenance management must conform to this new philosophy, which again can be problematic if the maintenance department is not allowed to purchase the necessary equipment, provide adequate training to the people to learn the new techniques, not given the time to collect the data, not permitted to shutdown the machinery when problems are identified, not given the time and resources to conduct the failure analysis, and then do not modify the component or procedure to increase the reliability.

2.1.5 Industrial Maintenance Philosophy Survey Results

Over the past few decades, surveys have shown the level of participation in each of the above-mentioned maintenance approaches or styles as shown in Figure 2.1. Despite the proven benefits of predictive or condition-based maintenance and proactive or prevention maintenance programs, 50% of the industries surveyed still allow their machinery to run to failure.

2.2 TYPES OF FORCES THAT OCCUR ON ROTATING MACHINERY

As mentioned earlier, the majority of failures in rotating machinery are caused by undesirable forces. Machinery shafts want to rotate freely with little or no external forces impinging on the rotors, bearings, couplings, and seals. When moderate or excessive static or dynamic forces are present, the components begin to slowly degrade, eventually leading to mechanical failure. The performance of the equipment relating to its designed operating conditions such as output horsepower, discharge pressure, flow, speed, etc., may all be optimum, but if excessive amounts of force and vibration are present, it is definitely not going to be running for very long.

There are two basic forces that act on the rotating machinery: static forces and dynamic forces as shown in Figure 2.2. Static forces are forces that always act in one direction.

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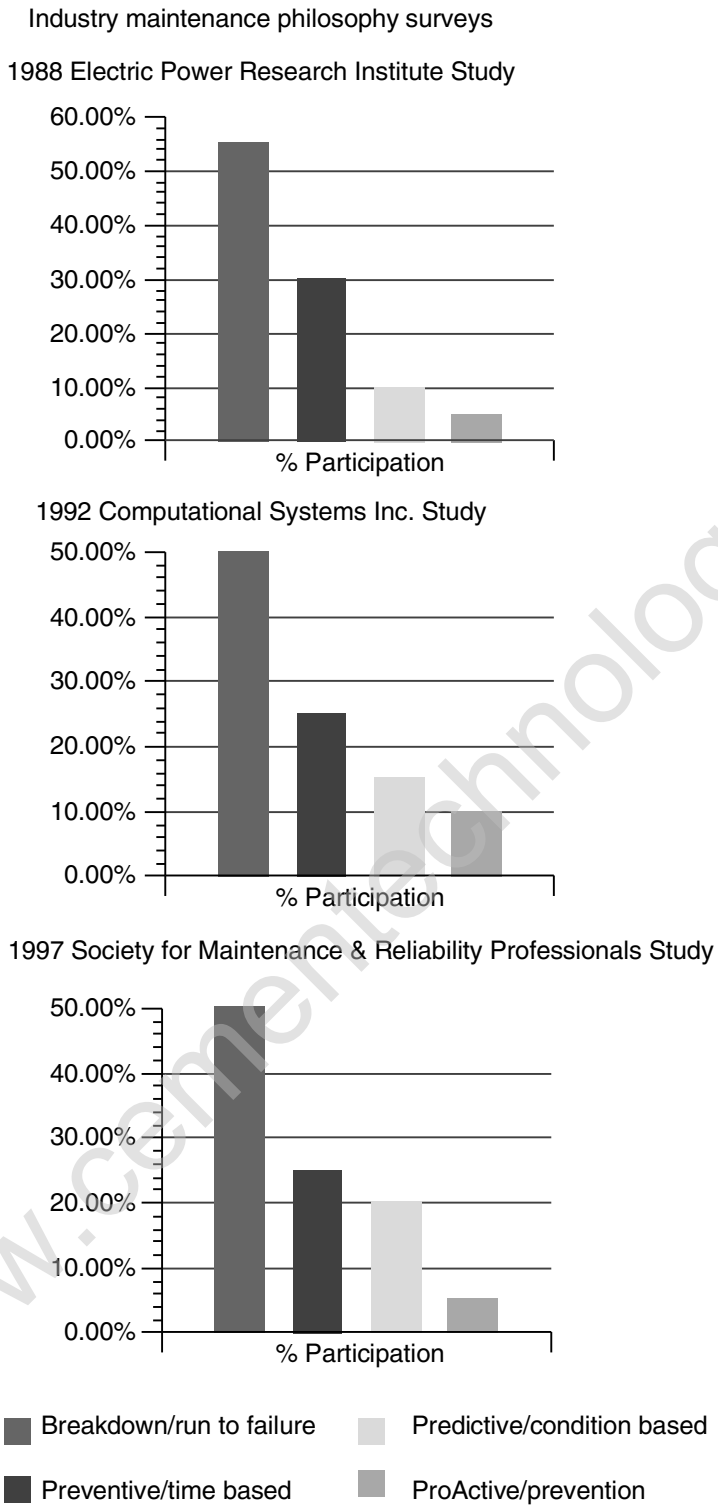


FIGURE 2.1 Industry survey showing how maintenance departments classify their maintenance approach.

Dynamic forces are forces that change their direction or periodically fluctuate for some reason. It is very important to understand that only dynamic forces can cause vibration to occur in machinery.

We will examine two different analysis methods commonly used in the industry to determine if rotating machinery is misaligned—vibration analysis and infrared thermography. Many of the generalizations made by vibration and infrared thermographic equipment vendors concerning detection of misalignment are not always true.

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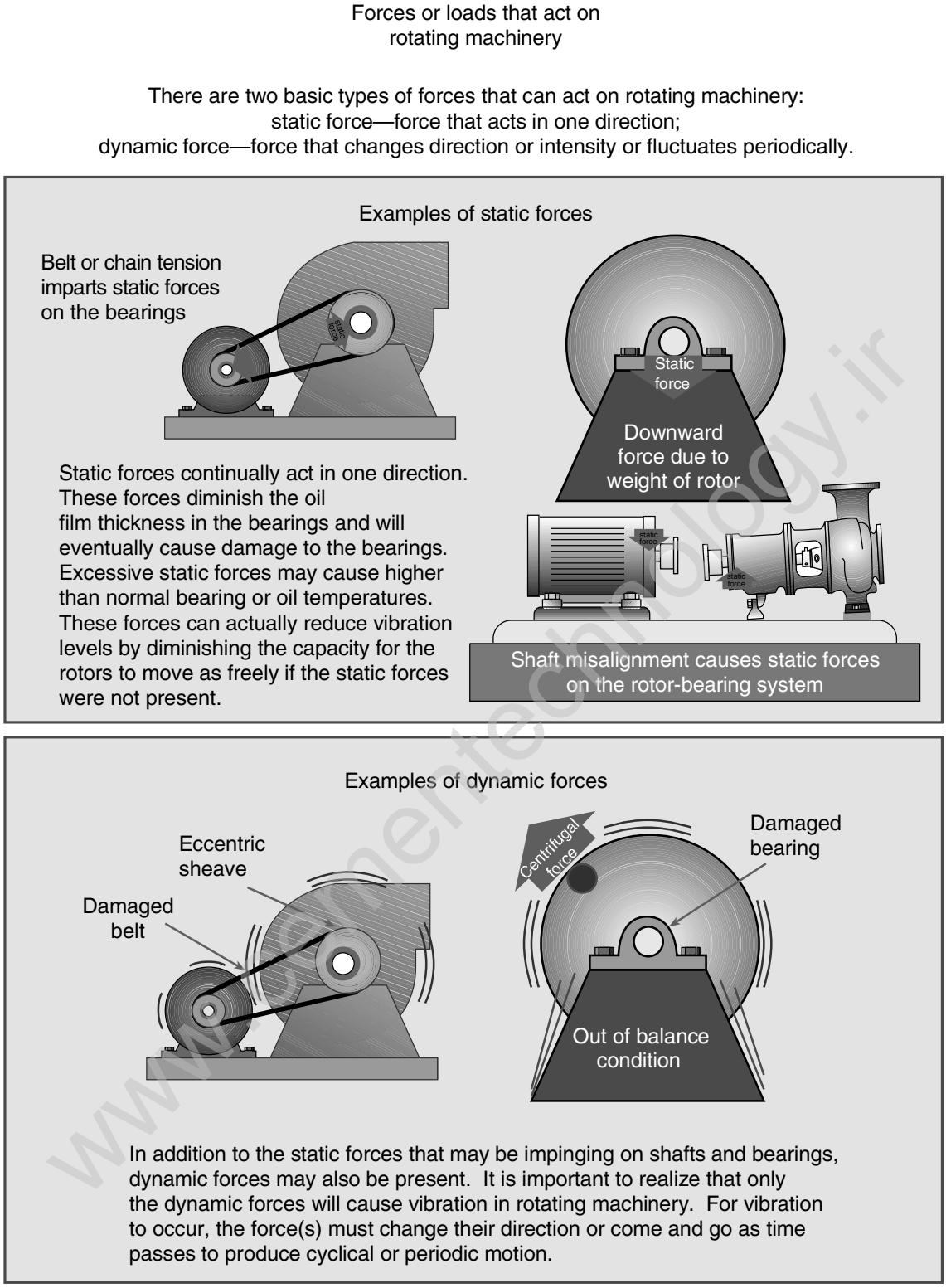


FIGURE 2.2 Forces in rotating machinery.

2.2.1 How Mass, Stiffness, and Damping Affect the Vibration Response of Machinery

One of the most overlooked aspects of vibration analysis is the effect of mass, stiffness, and damping of rotating machinery. As force is the major factor for stress and eventually cause damage to rotating machinery, it is important to understand what effect the mass, stiffness, and damping of this equipment has on the level of vibration that is measured.

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Mass is frequently, but incorrectly, referred to as the weight of an object. More precisely, it is the amount of material present in an object. Newton's second law of motion states that

$$F = ma$$

or

$$\text{force} = \text{mass} \times \text{acceleration}$$

At the Earth's surface, our planet exerts a gravitational pull of 32.174 ft/s^2 or 386.088 in./s^2 and is designated as 1 g. On the Moon, the gravitational pull is much less so you may "weigh" less but your mass remains the same. The accepted unit for mass is a slug [1 lbf (one pound of force) = 0.03108 slug]. Therefore 1 lbm (one pound of mass) = 0.00259 lbf s^2/in.).

Stiffness refers to the flexibility, rigidity, or "springiness" of an object. Stiffness is expressed in lbf/in. or how much force does it take to stretch or compress an object 1 in. in length.

Damping refers to the resistance to motion. Frequently rubber vibration isolators or dampers are installed between machinery and the floor of a building to prevent the transmission of vibration in the machine to the floor. The accepted unit for damping is lbf s/in.

The equation below describes force as a function of time:

$$F(t) = (K \times \text{displacement}) + (C \times \text{velocity}) + (M \times \text{acceleration})$$

where K represents the stiffness (lbf/in.), C the damping (lbf s/in.), and M the mass (lbf s^2/in.).

All of this may seem very boring but it explains why vibration analysis is so very difficult. Remember, vibration sensors measure motion, not force. As vibration amplitude is often used to describe the severity of a problem, the dynamic forces that generate the amount of motion of an object will change if the mass, stiffness, or damping is altered.

2.2.2 How Vibration Is Measured

There are three types of vibration sensors: accelerometers as shown in Figure 2.3, seismometers (velocity pickups) as shown in Figure 2.4, and proximity probes as shown in Figure 2.5. Accelerometers and seismometers are typically used to take casing–bearing cap measurements. Proximity probes are permanently mounted to the bearing housing and are typically used to measure motion of a shaft supported in a sliding type bearing to observe shaft motion with respect to the machine case. Proximity probes can also be used to measure axial position of a shaft, commonly referred to as thrust probes, and they can be used to capture rotational timing information as explained below. Figure 2.6 shows the useful frequency range of each of these sensors. The typical units of measure for acceleration is g s; for velocity, inches or millimeters per second (in./s or mm/s); and for displacement mils (1 mil = 0.001 in.) or millimeters. You can convert from one unit to another as shown in Figure 2.7.

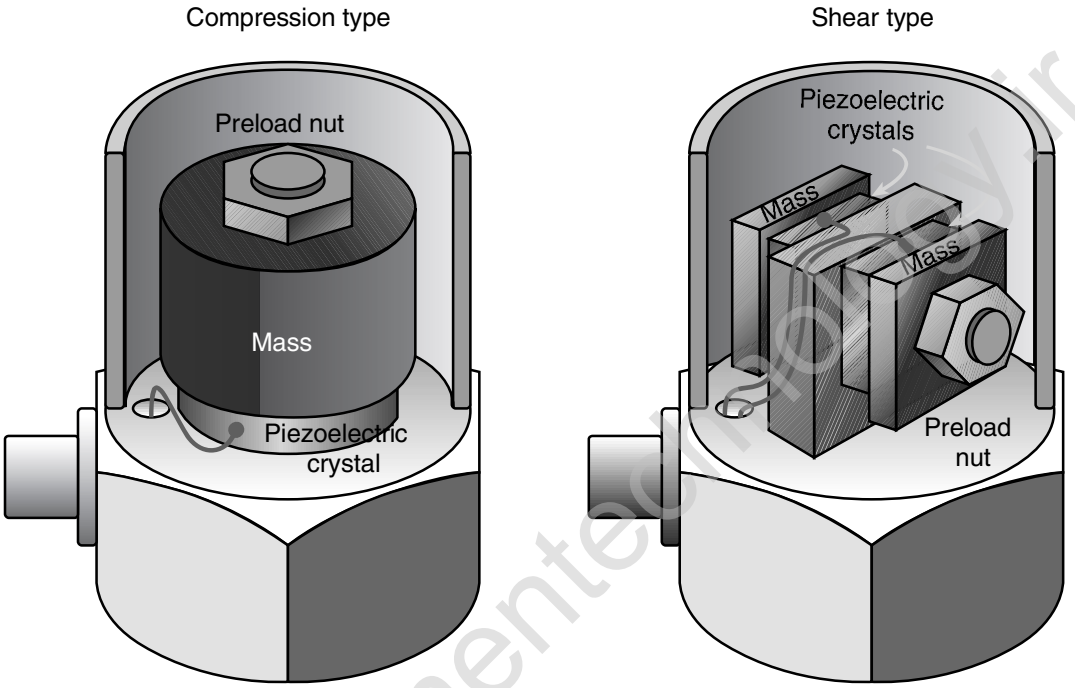
On each two bearing machines, a total of five vibration readings are typically measured: four radial vibration measurements and one axial vibration measurement. A radial vibration measurement is taken at a 90° angle to the centerline of rotation of the shaft and an axial vibration reading is taken in line with the centerline of rotation. Radial vibration measurements taken from the outside of the machine case or bearing housing with accelerometers or seismometers on machinery where the shafts are in a horizontal orientation are usually taken at the top (i.e., twelve o'clock) and side (i.e., three o'clock or nine o'clock) positions at both the inboard (coupling end) and outboard bearings as shown in Figure 2.8. The axial vibration measurement is taken at either the inboard or outboard end of the machine. The accelerometer

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Accelerometers

Accelerometers are perhaps the most popular vibration sensor today for capturing bearing cap or casing measurements. The piezoelectric material used to generate the signal in these sensors is usually quartz or a ceramic called lead zirconate titanate (PZT). This type of material produces anelectrical charge if force is applied to it. There are several different configurations but they are typically manufactured to be compression or shear type of sensors. The piezoelectric material must be placed under a strain (preloaded force) in order to measure either tension orcompression forces. Permanently mounting these sensors must be done carefully as to prevent distortion of the housing.



Advantages

¥ Wide frequency response (up to 3,000,000 cpm).
¥ Lightweight and physically small.
¥ Good ambient temperature range. Special sensors can be placed in environments up to 1000°F.
¥ Moderate pricing.

Disadvantages

¥ Sensitive to stresses induced when permanently mounted.
¥ Requires a power supply and amplification circuitry (some sensors have amplification circuits built into the sensor housing).

FIGURE 2.3 Operating principles of accelerometers.

is held in position by hand, or with a magnetic base, or with a quick release connector, or it may be permanently attached to the housing using epoxy or a threaded rod drilled and tapped into the machine case.

Another useful piece of information that can be gathered to assist in understanding the vibration on machinery is a rotational timing reference signal commonly known as the "phase angle." There are four different ways to gather phase angle data as shown in Figure 2.9:

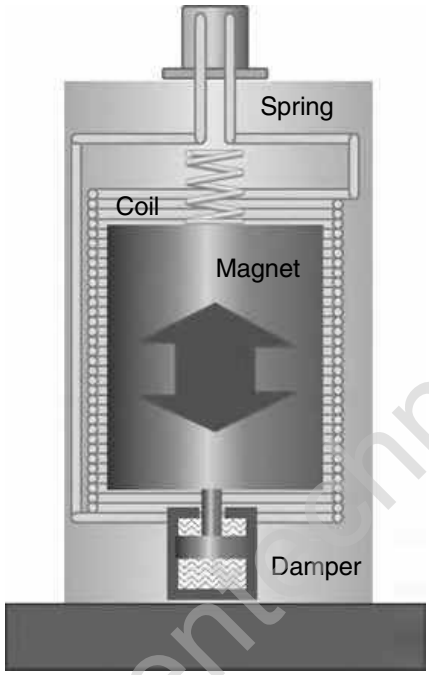
1. A strobe light viewing a reference mark on the rotating shaft with a stationary protractor positioned around the circumference of the shaft

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Velocity sensors

Velocity sensors are also known as seismic probes. Effectively they are linear voltage generators. As the permanent magnet moves back and forth inside the housing, the magnetic flux lines cut through the windings of the coil surrounding the magnet producing an alternating current. The faster the magnet moves, more flux lines are cut through the windings and the voltage increases. These probes are excellent for measuring casing or bearing housing movement. They have also been employed with shaft riders that physically touch the shaft as it



Advantages

¥ Strong signal generated by coil which can be transmitted over fairly long distances (up to 100 ft).
¥ Self-generated signal requires no special power supply or circuitry.
¥ Rugged construction can be mounted virtually in any position.
¥ Frequency response is good to 300,000 cpm.

Disadvantages

¥ Physically large and heavy compared to proximity probes and accelerometers.
¥ Output signal decays exponentially below 600 cpm.
¥ Relativey expensive and not manufactured by many companies.

FIGURE 2.4 Operating principles of seismometers.

2. A strobe light viewing a rotating protractor marked on the circumference or end of a shaft with a stationary reference mark positioned around the circumference of the shaft
3. A photo tachometer, which consists of a piece of reflective tape attached to the rotating shaft and a fiber optic cable with one set of fibers shining a light onto the shaft and another set of fibers sensing the piece of reflective tape as it passes by
4. A proximity probe positioned over a keyway or hole in the rotating shaft

Bear in mind that the phase angle signal is somewhat meaningless unless the phase angle sensor (strobe light, phototach, or prox probe) is controlled (triggered) by the same analyzer that is also used to measure the vibration signal. Also be cognizant of the fact that the phase

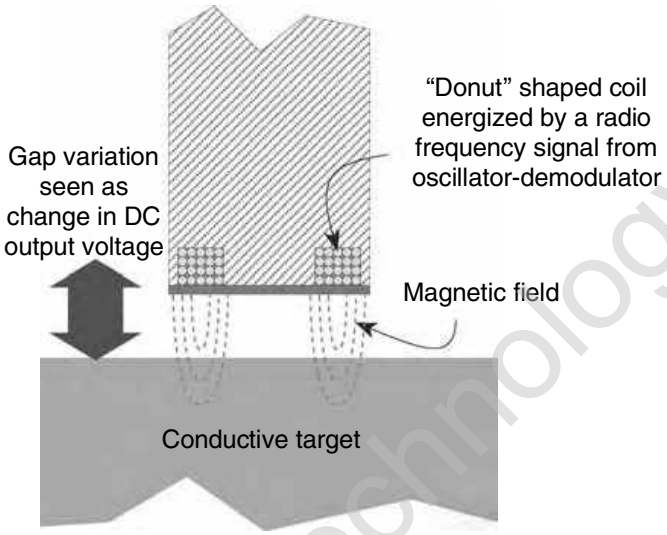
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Proximity probes

Proximity probes can effectively be considered as noncontacting, electronic dial indicators. The measure displacement or distance directly. A small pancake coil located in the tip of the probe induces an electromagnetic field. When an electrically conductive surface (e.g., a steel shaft) is placed near the magnetic field, eddy currents are induced in the surface. As the probe tip is moved toward the surface, the circuit voltage drops; as the tip is moved away from the surface, the circuit voltage increases. These probes are excellent for use in monitoring shaft vibration where the rotor is supported in sliding type bearings where the AC component of the signal measures radial vibration. These probes are also used to monitor the axial/thrust position of a rotor to ascertain wear on a thrust bearing by monitoring the DC component of the signal.

Typical probe sensitivity is 100 or 200 mv/mil



Advantages

¥ Small in physical size and lightweight. Probes diameters typically used for monitoring radial vibration range between 1/4 to 3/8 and can be made in just about any length. Special probes can accurately measure up to 1/2 of distance.
¥ Relatively inexpensive at around $400.00 for probe and oscillator-demodulator (aka proximitor). Power supplies (usually either –18 or –24 VDC) and visual readout devices add significantly to the cost.
¥ Measurements are virtually unaffected by lubricants or gases that pass between the probe tip and the shaft surface.
¥ Senses actual shaft motion relative to bearing housing and when used in an *X–Y* configuration (i.e., two probes set 90° apart) with a dual channel analyzer can display orbital shaft motion inside the bearing.
¥ Frequency response ranges from 0 to 300,000 cpm making it very usable for rotating machinery applications.
¥ Proximity probes can also be used to measure distance and are frequently used to monitor sliding type thrust bearing wear physical position of a shaft inside the bearing, casing movement or expansion measuring off-line to running machinery movement surveys for hot and cold shaft alignment studies. Probes are also used to measure once-per-rev positions
(usually at the keyway) to provide phase angle reference.
¥ Satisfactory service in environments up to 250°F. Higher temperatures can cause a shift in the DC voltage and linearity loss at extremes of range.
¥ The output signal from the oscillator-demodulator can be transmitted over a significant distance (up to 1000 ft).

Disadvantages

¥ Probes are usually permanently attached to bearing housings requiring special mounting fixtures and careful placement.
¥ Surface imperfections in the shaft (i.e., scratches, grinding marks) will also be observed by the probe and can significantly affect the signal making it confusing as to how much of the overall signal is actual shaft motion and how much is due to surface imperfections.
¥ Damaged shaft journal areas that have been reworked by chrome plating are particularly troublesome as the probe will ignore the chrome plated surface and see the rough surface beneath the chrome.
¥ Requires a power supply and impedance matched circuitry (i.e., you may not be able to mix different probes, cables, and oscillator-demodulators).
¥ Susceptible to induced voltages if sensor wiring is run in conduct with other higher voltage wires (e.g., 110 VAC, 60 Hz).

FIGURE 2.5 Operating principles of proximity probes.

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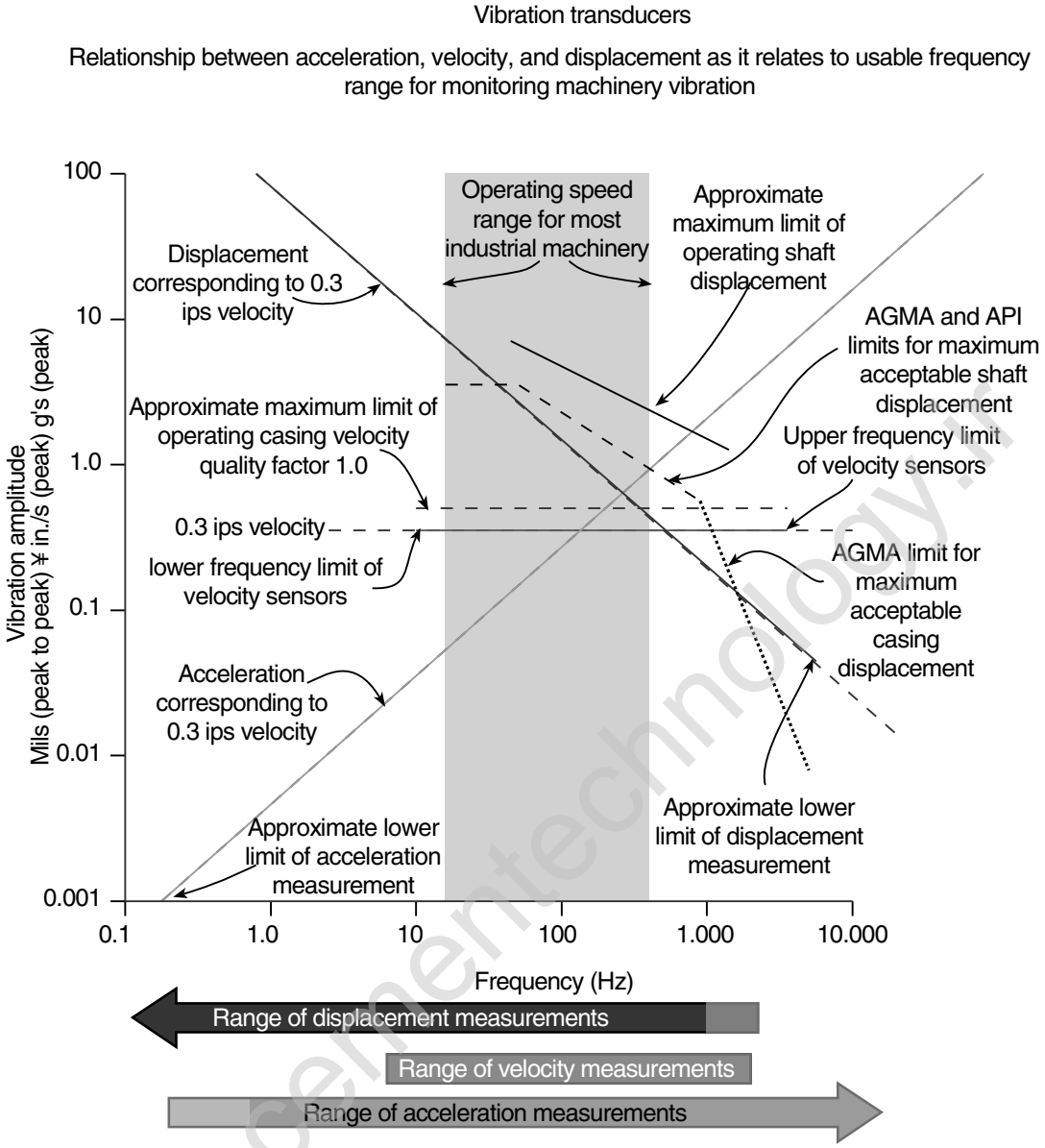


FIGURE 2.6 Frequency range of accelerometers, seismometers, and proximity probes.

angle data is directly related to the running speed of the machine and is only useful if the majority of the vibration is occurring at running speed. Not all of the vibration that exists in rotating machinery occurs at the machine's running speed as explained below.

The two most commonly used parameters in vibration analysis are amplitude and frequency. Amplitude, as shown in Figure 2.10, is an indicator of the severity of vibration. Frequency, as shown in Figure 2.11, is an indicator of the source of the vibration. If overall vibration amplitude levels are relatively high at any of the sensor locations, a more detailed survey should be taken with a spectrum (fast Fourier transform, FFT) analyzer to aid in determining the vibration signature of the machinery.

2.2.3 Time and Frequency Domain Vibration Information

Vibration information is typically displayed in two different ways: in the time domain and in the frequency (spectral) domain as shown in Figure 2.12. Actual examples of time and frequency domain information are shown in Figure 2.13.

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Converting from one vibration unit to another

Finding acceleration (*g* s) if frequency (cpm) and velocity (in./s) or displacement (mils) is known . . .

acceleration = 0.0000000142 × frequency^2 × displacement [A=0.0000000142 F^2 D]

acceleration = 0.00027144 × velocity × frequency [A=0.00027 VF]

Finding velocity (in./s) if frequency (cpm) and acceleration (*g* s) or displacement (mils) is known . . .

velocity = 0.00005236 × frequency × displacement [V=0.00005236 FD]

velocity = 3684 × (acceleration/frequency) [V=3684 (G/F)]

Finding displacement (mils) if frequency (cpm) and acceleration (*g* s) or velocity (in./s) is known . . .

displacement = 19108.3 × (velocity/frequency) [D=19108.3 (V/F)]

displacement = 70400000 × (acceleration/(frequency)^2) [D=70400000 (G/F^2)]

Vibration parameter	*Symbol*	*Common units of measurement*
Frequency	F	Cycles per minute (cpm)
Displacement	D	Mils peak to peak
Velocity	V	in./s peak
Acceleration	A	G peak

Example A

F = 600 cpm

D = 15 mils peak to peak

$A = 0.0000000142\ F^2\ D$

$A = (0.0000000142) \times (600)^2 \times 10$

A = 0.0767 *g* s

Example B

F = 600 cpm

V= 0.5 in./s. peak

$A = 0.00027\ VF$

$A = 0.00027 \times 0.5 \times 600$

A= 0.0814 *g* s

Example C

F = 600,000 cpm

A = 50 *g* s

$V = 3684\ (G/F)$

$V = 3684\ (30/600{,}000)$

V = 0.307 in./s peak

Example D

F = 600,000 cpm

A = 50 *g* s

$D = 70400000\ (G/F^2)$

$D = 70400000\ (50/600{,}000^2)$

D = 0.0098 mils peak to peak

Notice in example A how a considerable amount of displacement (15 mils) at 600 cpm has a small corresponding amount of acceleration (0.0767 *g* s). Also notice that a tremendous amount of acceleration (50 *g* s) at 600,000 cpm produces an extremely small amount of displacement (0.0098 mils peak to peak). Finally notice how the velocity in examples B (0.5 in./s peak) and C (0.307 in./s peak) would be of much concern regardless of the frequency (600 cpm in B and 600,000 cpm in C). That is why velocity is typically chosen as the unit of choice for measuring casing vibration on machinery, it is somewhat frequency independent. By reviewing the equation for force, notice that at low frequencies, displacement and stiffness are the predominant factors but at high frequencies, acceleration and mass are the predominant factors. Velocity across a much broader frequency range represents the damaging effects of dynamic force better than displacement or acceleration.

FIGURE 2.7 Converting one vibration unit to another.

Discerning what a vibration sensor is telling you happens to be one of the most difficult tasks the machinery diagnostician is faced with. However with the use of an FFT signal analyzer, vibration "signatures" can be taken that split the complex overall vibration signal and enable one to look at various frequencies of the sensor output. Many rules of thumb have emerged that attempt to classify specific machinery problems with specific types of vibration signatures. The experienced vibration analyst quickly learns that these rules of thumb are to be used as guidelines on arriving at the source of the problem. Quite often, more than one problem exist on

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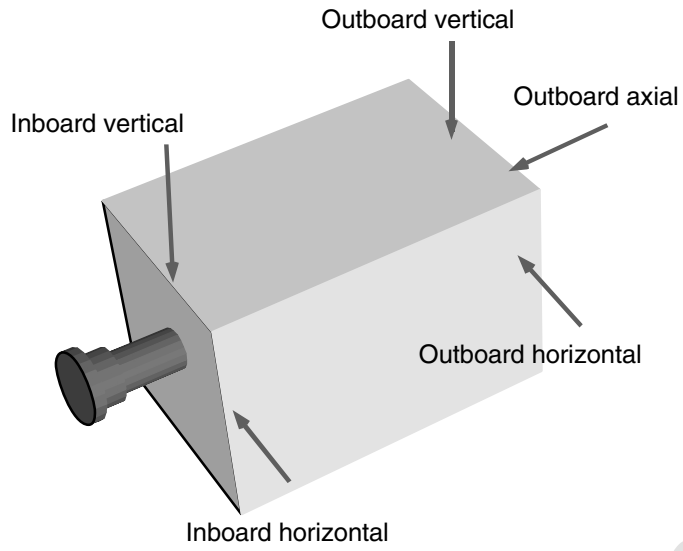


FIGURE 2.8 The area/position in which the vibration measurements on bearing caps of rotating machinery are typically gathered.

a piece of rotating machinery such as a combination of imbalance, misalignment, and damaged bearings that will all appear on the vibration spectrum.

If the source of the problem is misalignment, what will the vibration spectrum look like? Does the signal look the same for all different types of machinery with different kinds of couplings, bearings, and rotors?

2.2.4 Using Vibration Analysis to Detect Misalignment

One goal of this chapter is to examine the types of vibration signatures misaligned rotating equipment exhibit and the forcing mechanisms involved to generate this signal. Many people who work in the field of vibration analysis feel that shaft misalignment can be detected by the following symptoms:

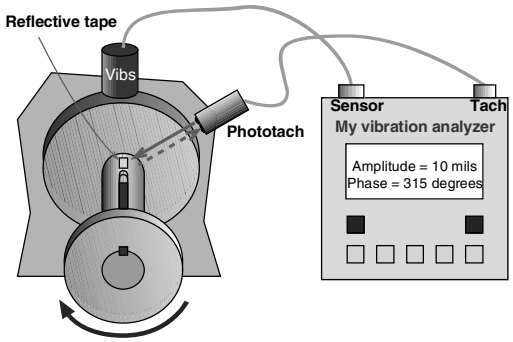
- High one or two times running speed frequency components
- High axial vibration levels
- A 180° phase shift will occur across the coupling

The above symptoms can happen during shaft misalignment, but not always. There are four general statements that can be made from a number of controlled tests where rotating machinery was purposely misaligned and also from many field observations that have been made on equipment that was operating under misalignment conditions.

1. You cannot detect the severity of misalignment using vibration analysis. In other words, there is no relation between the amount of misalignment and the level or amplitude of vibration.
2. The vibration signature of misaligned rotating machinery will be different with different flexible coupling designs. For example, a misaligned gear coupling will not show the same vibration pattern as a misaligned rubber tire-type coupling.
3. Misalignment vibration characteristics of machinery rotors supported in sliding type bearings are typically different than the vibration characteristics of machinery rotors supported in antifriction type bearings.

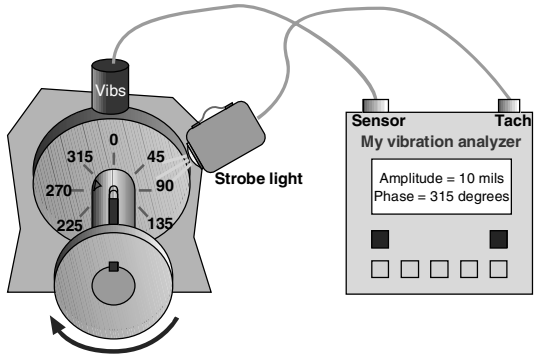
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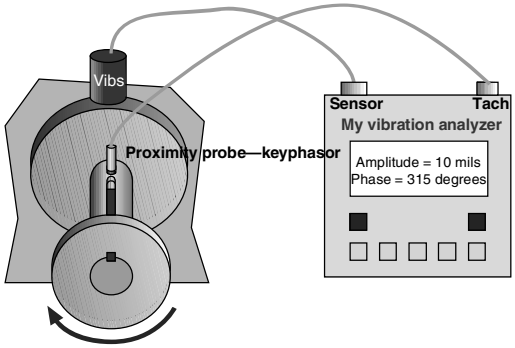
Phototachometer and reflective tape

A piece of reflective tape is placed on the rotating shaft. The analyzer shines a beam of light through a fiber optic cable that is aimed at the shaft where the reflective tape was placed. Another fiber optic cable observes the tape as it goes by. The analyzer measures the period of time between each pulse from the phototach and divides by 360 to determine the amount of time required for one degree of rotation. As the voltage signal from the vibration sensor goes from a positive value to a negative value, the analyzer measures the period of time from the voltage spike seen by the phototach and the peroid of time from the positive to negative crossover point from the vibration sensor then divides by the period of time required for one degree of rotation.



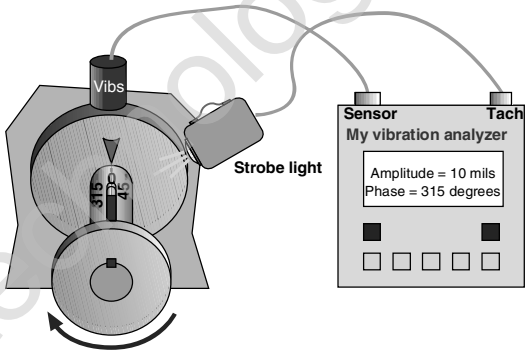
Stationary protractor—rotating reference mark

A reference mark is placed on the rotating shaft and angles are positioned on a stationary part of the machine case near the shaft where the reference mark was placed. As the voltage signal from the vibration sensor goes from a positive value to a negative value the analyzer flashes the strobe light.



Proximity probe and keyway (aka keyphasor)

A proximity probe is positioned over a keyway, a key, or a hole in the shaft. The analyzer measures the period of time between each pulse from the keyphasor and divides by 360 to determine the amount of time required for one degree of rotation. As the voltage signal from the vibration sensor goes from a positive value to a negative value, the analyzer rmeasures the period of time from the voltage spike seen by the keyphasor and the peroid of time from the positive to negative crossover point from the vibration sensor then divides by the period of time required for one degree of rotation.



Stationary Reference Mark - Rotating Protractor

A reference mark is positioned on a stationary part of the machine case near the shaft where angles are marked around the circumference of the shaft. As the voltage signa from the vibration sensor goes from a positive value to a negative value the analyzer flashes the strobe light.

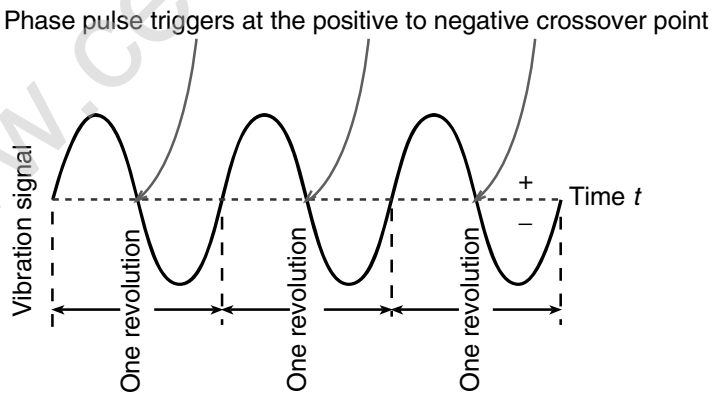


FIGURE 2.9 The four different methods to capture phase angle information.

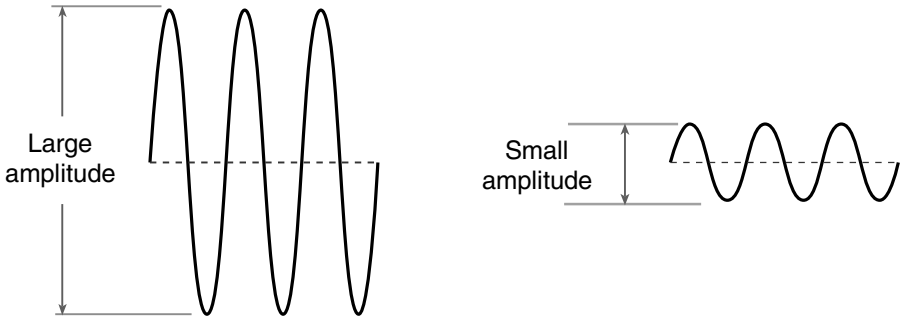
4. Vibration signatures from misaligned flexible couplings frequently have components at multiples of running speed. As previously mentioned, phase angle information only makes sense if the primary vibration is occurring at running speed. If the majority of vibration is principally occurring at frequencies other than running speed, phase angle data is somewhat meaningless.

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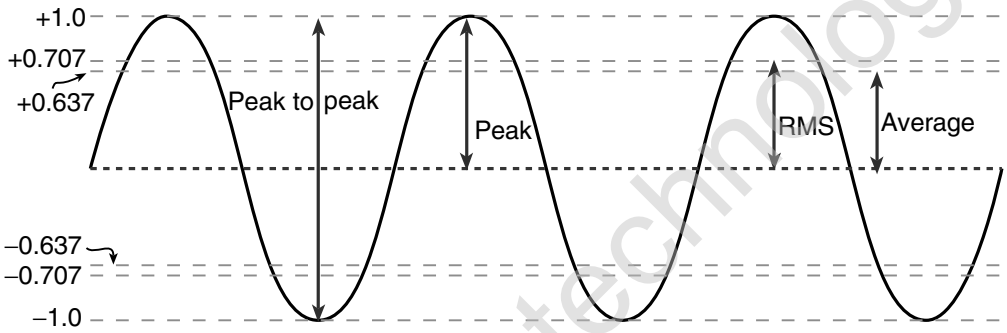
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Amplitude

Vibration amplitude is defined as the strength or size of the signal. In vibration analysis, it is usually the predominant factor in determining the severity of a problem.



Vibration amplitude is expressed in four different ways: peak to peak, peak, RMS (root mean squared), and average.



In the United States, vibration displacement amplitude when using proximity probes is usually expressed in peak-to-peak values, velocity amplitude is usually expressed in peak values when using velocity sensors or accelerometers (integrated to read in velocity), and acceleration amplitude is also usually expressed in peak values. In Europe, velocity and acceleration are usually expressed in RMS values.

If you are comparing vibration amplitude measurements on machinery taken at different time intervals (e.g., once a month) to determine if the vibration is changing, or if you are comparing your measurement to another persons measurements, it is important to insure that the amplitude values remain the same.

FIGURE 2.10 Vibration amplitude.

2.2.5 Relationship between Vibration Amplitude and Misalignment Severity

If two pieces of rotating machinery are coupled together and misaligned by 5 mils/in. and vibration readings are taken on the bearing caps and then the unit is shutdown and misaligned to 10 mils/in. (double the initial amount), started back up and vibration readings taken on the bearing caps again, the overall vibration amplitudes measured will not be twice the amount compared to the first set of data collected at misalignment conditions of 5 mils/in.

Increasing misalignment may actually decrease vibration levels. Conversely, if a drive train that has been running misaligned for a period of time was shutdown and realigned more accurately, the vibration levels may increase after it is restarted. Many drive trains may be slightly to moderately misaligned and vibration analysis will not be able to detect the misalignment condition. Surprising facts, but true! To verify these statements, let us review some experimental tests and actual field data taken on misaligned rotating machinery.

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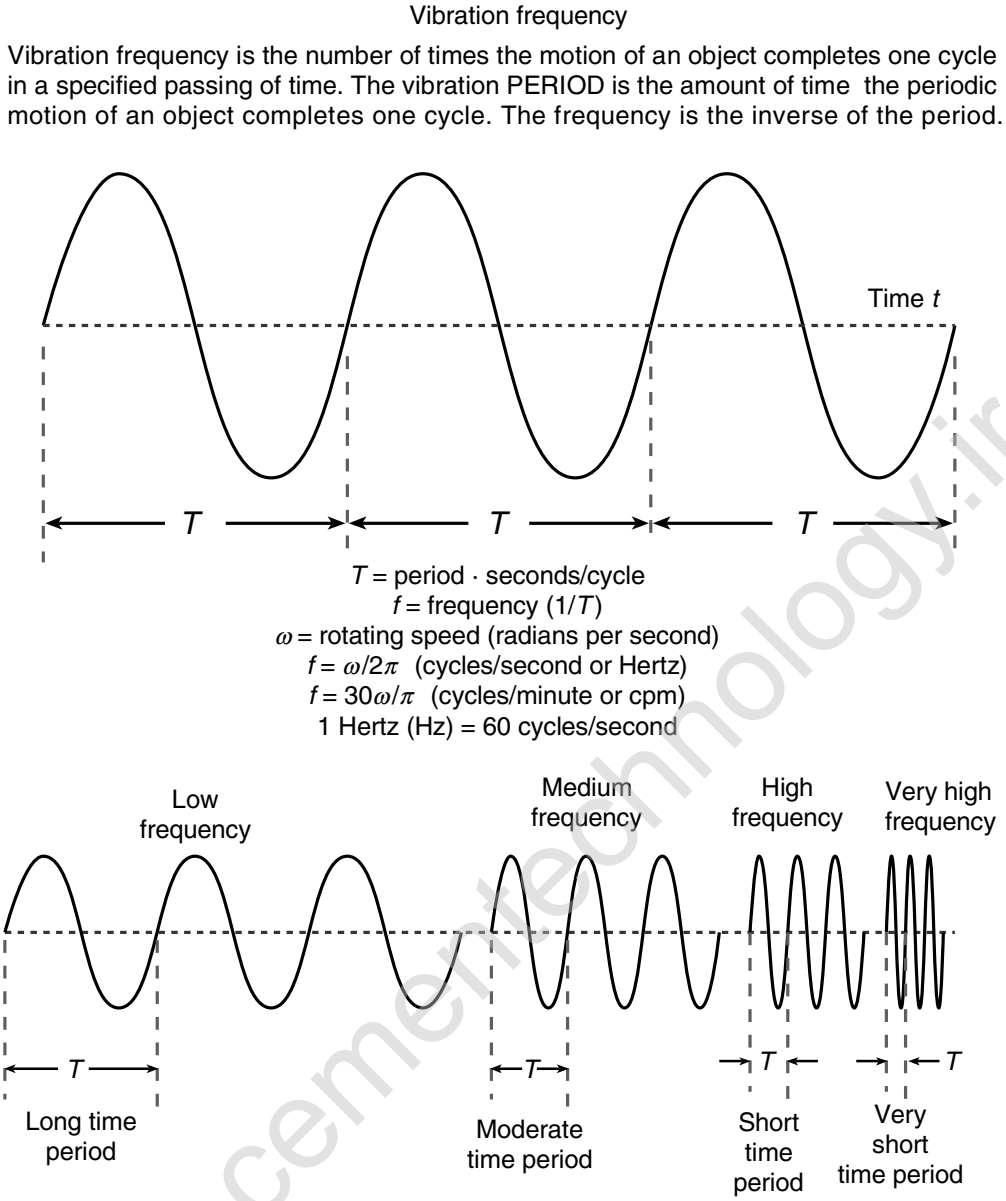


FIGURE 2.11 Vibration frequency.

2.2.6 Vibration Results from a Controlled Misalignment Test on a Training Demonstrator

Figure 2.14 shows a 1/2 hp, 1775 rpm motor driving a center shaft with two balance disks coupled to another outer shaft with one overhung balance disk. A steel bar was attached to both bearing housings that support the center shaft. Two proximity probes were positioned on this bar to monitor the distance between the two balance disks and the probe tips to measure the amount of shaft distortion that occurred on the center shaft. The three shafts (motor, center, and outer) were connected together using flexible disk-type couplings. The unit was near perfectly aligned, operated at running speed, and overall vibration measurements taken on all six bearings. The demonstrator was then shutdown, the bolts holding the middle shaft to the frame were loosened, and the middle shaft was misaligned 31 mils sideways, locked in that position, started back up, and vibration, proximity probe gap readings, and amperage measurements were taken. The demonstrator was again shutdown, the bolts holding the middle shaft to the frame were loosened, and the middle shaft was

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Periodic vibration and the Fourier transform

The vibration that comes from rotating machinery is typically not simple, harmonic vibration but can be comprised of several varying forces emanating at different frequencies. When two or more forces occurring at different frequencies develop on rotating machinery, they add together to produce a more complex waveform. Around 1790, Jean Baptiste Fourier, a French physicist and mathematician, discovered a mathematical relationship that showed any complex waveform comprised of several different harmonic components emanating at different frequencies can be resolved into their distinct frequencies. This mathematical relationship is known as the Fourier series or Fourier transform. Today, with the use of computers, this mathematical formula can be done at a very fast rate of speed aptly named as the fast Fourier transform or FFT.

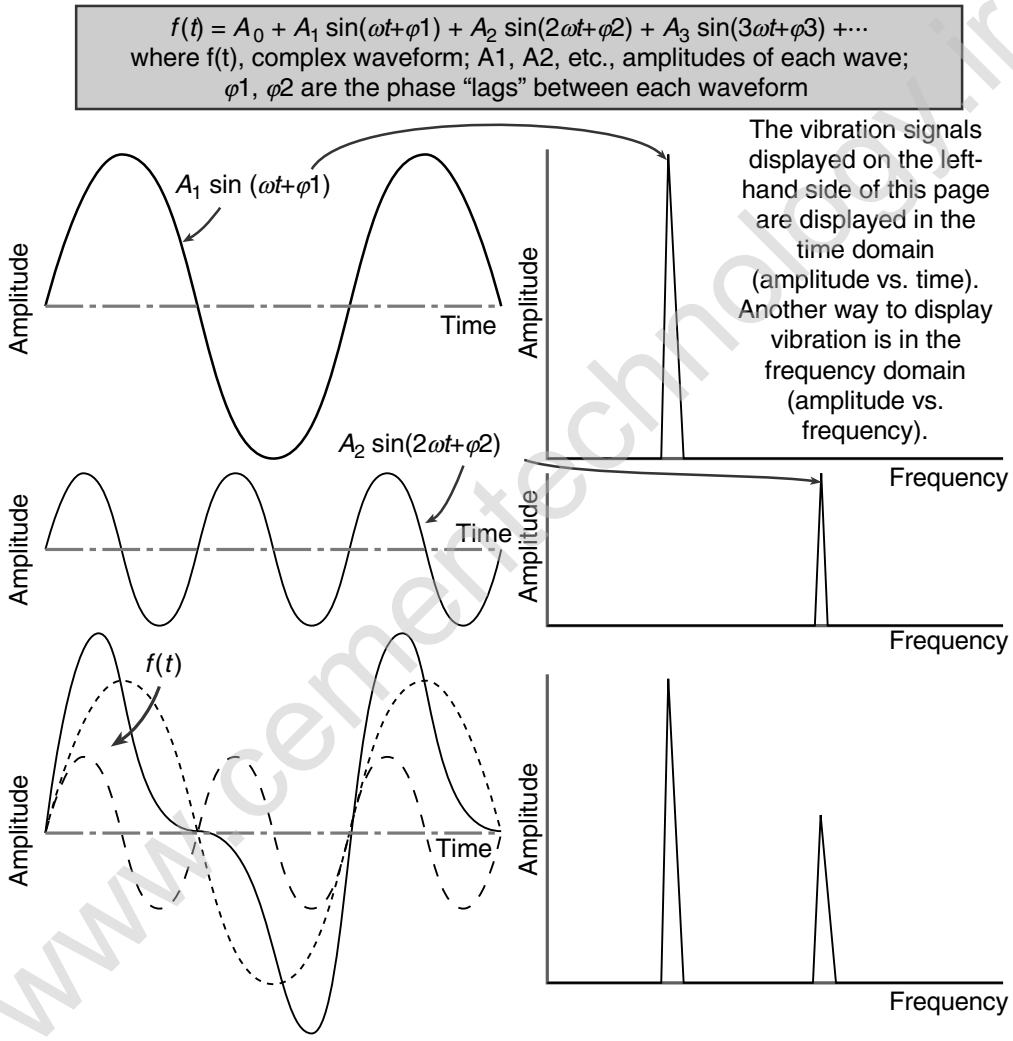


FIGURE 2.12 Spectral information and the Fourier transform.

misaligned another 31 mils sideways (now a total of 62 mils sideways), locked in that position, started back up, and vibration measurements were taken for a final time.

Figure 2.15 shows the results of the vibration measurements, the proximity probe gap readings, and the amperage draw.

Notice that in virtually every case, the vibration levels dropped as the misalignment increased. The amperage draw showed very little change across each of the runs. However notice that the distance between the proximity probes and the balance disks increased just

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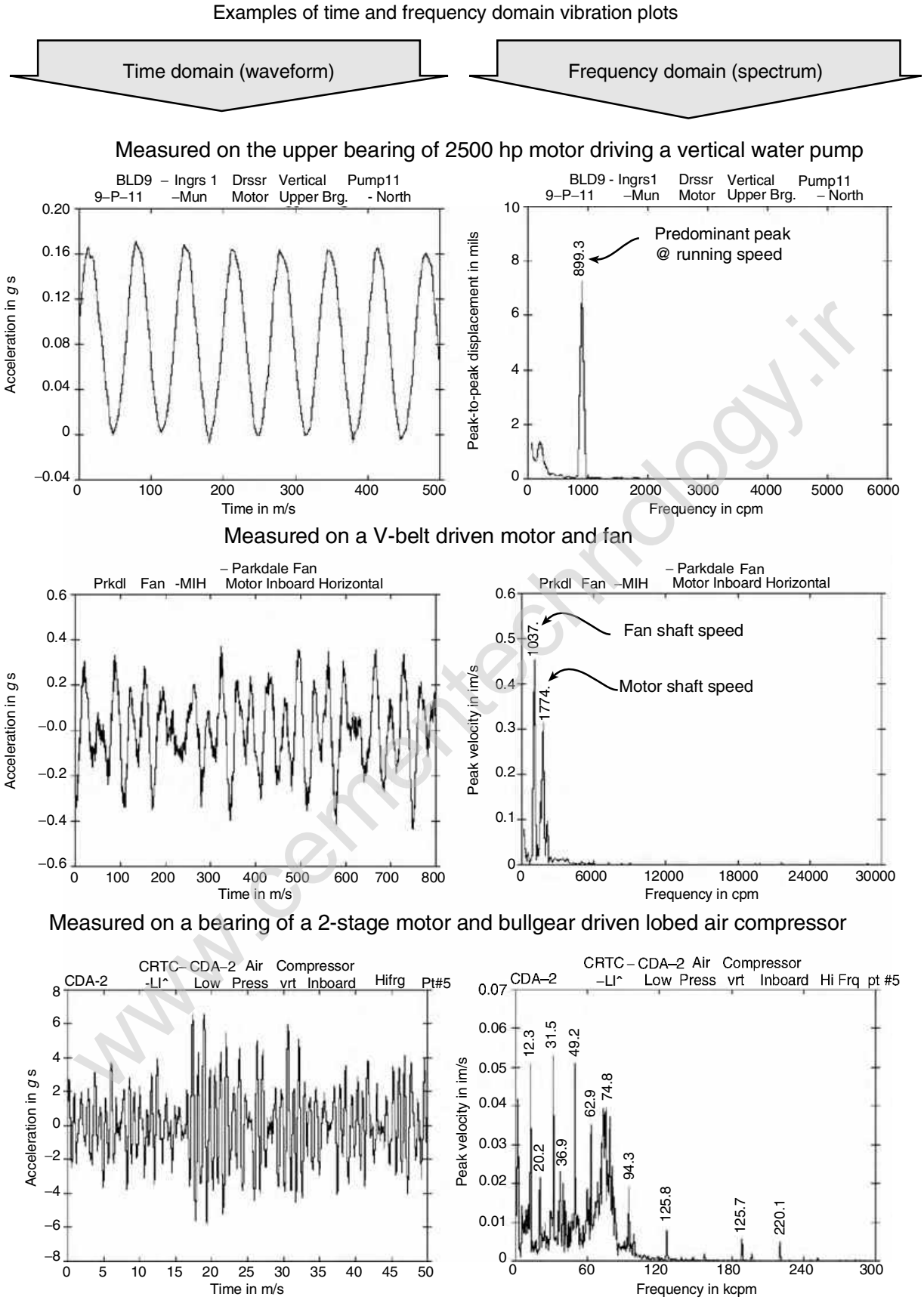


FIGURE 2.13 Actual time and frequency plots.

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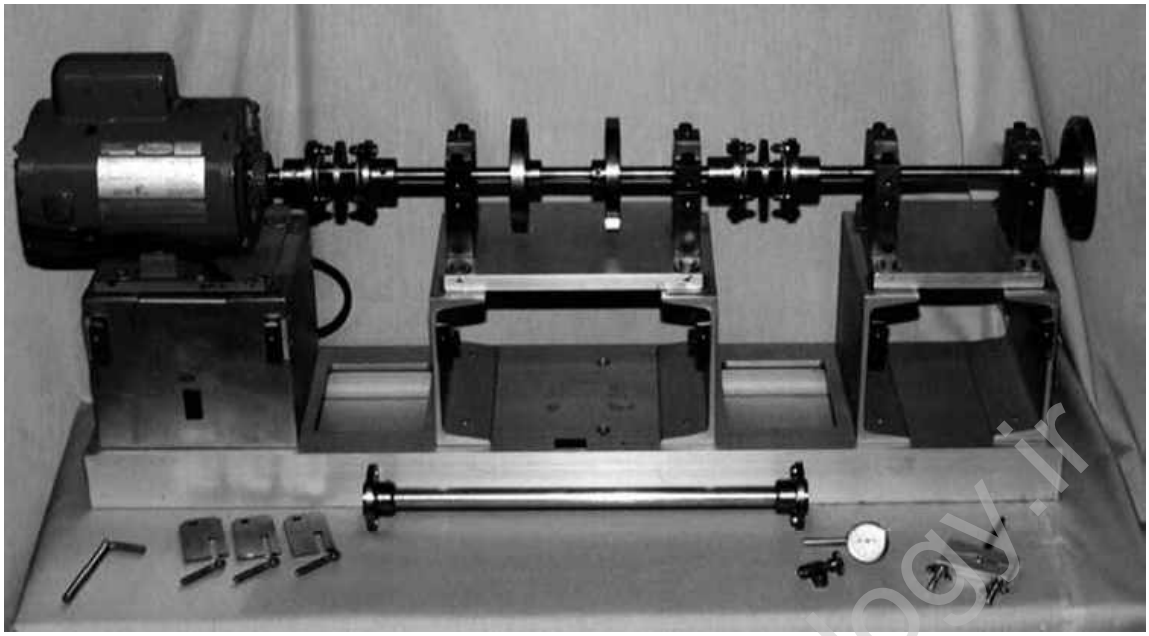


FIGURE 2.14 Training demonstrator used for misalignment vibration test.

about proportionally with the amount of misalignment. This verifies that elastic bending of the shafts does indeed occur similar to what was shown in Figure 1.3.

2.2.7 Vibration Results from a Controlled Misalignment Test on a Motor and Pump

Figure 2.16 shows a 60 hp, 1775 rpm motor and circulating water pump, connected by a metal ribbon-type coupling as shown in Figure 2.17 that was purposely misaligned by 21 and 36 mils sideways (laterally) and 55 and 65 mils vertically. A total of seven test runs were made under various alignment conditions. The vibration analysis equipment used for the test is shown in Figure 2.18. Vibration data was taken at five points on each unit using a hand-held vibration meter with an accelerometer sensor to record overall readings on the bearing housings. A seismometer with a magnetic base was also attached at each sensor location and the signals were fed into an analyzer and an X–Y plotter to record the vibration signature. As the motor housing was aluminum construction, ¼ in. thick carbon steel plates were epoxied to the motor in the horizontal and vertical planes at both bearings and another plate was attached on the inboard end bell to capture the axial vibration levels.

The reverse-indicator shaft alignment method was used to determine the relative shaft positions for all test runs. The nomenclature used to identify each run indicates the maximum amount of motor shaft deviation with respect to the pump shaft at the point of grid contact in the coupling. In Run #2 M2W, for instance, the maximum amount of shaft deviation occurred in the lateral (i.e., side-to-side) position of the shafts. The motor coupling hub contact point is 0.002 in. to the west of the contact point on the pump coupling hub and is so designated as M2W or Motor-2 mils-west.

Run #1 (Motor uncoupled and running solo). The first run was conducted with the motor uncoupled to determine whether the motor had an unbalance condition, damaged bearings, or some other problem that could affect the vibration response when coupled to the pump.

Run #2 M2W (Motor-2 mils-west). The pump and motor were initially aligned well within acceptable alignment tolerances.

Run #3 M21W (Motor-21 mils-west). The motor was positioned 0.021 in. to the west with no shims being added or removed from the pump or the motor from the second run.

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Misalignment vibration, rotor distortion,
and power consumption study

A motor, center shaft, and outer shaft demonstrator model was used for the misalignment study. The drive train was initially aligned within 1/2 mil/in. misalignment, started, and vibration, motoramperage, and proximity probe gap data was collected. The unit was stopped and the center shaft was purposely misaligned by 31 mils sideways, locked in that position, started, and data was collected. Again, the unit was stopped and the center shaft was purposely misaligned by 62 mils sideways, locked in that position, started, and data was collected. Below is the recorded data.

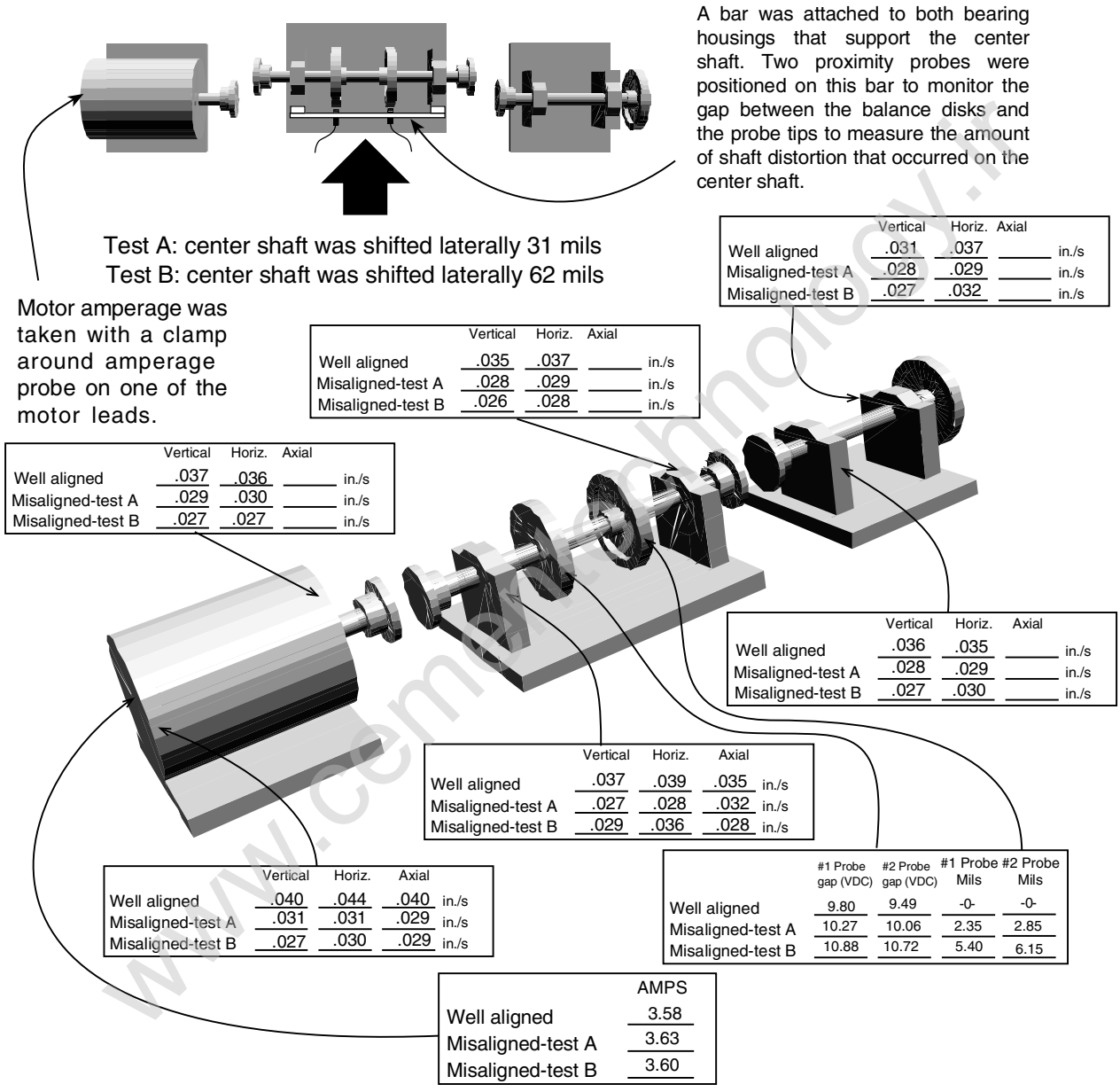


FIGURE 2.15 Results of misalignment vibration test on demonstrator.

Run #4 M36W (Motor-36 mils-west). The attempt was made in this run to slide the motor further to the west. The motor however became bolt bound and was unable to be moved any further sideways.

Run #5 M65H (Motor-65 mils-high). The motor is now positioned well from side to side, but is 0.065 in. higher than the pump shaft centerline.

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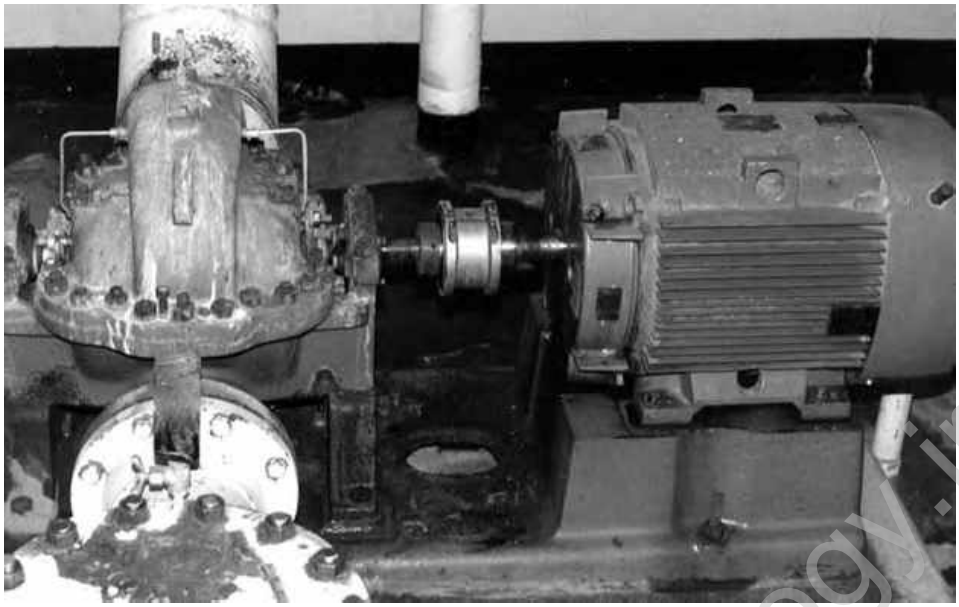


FIGURE 2.16 Pump and motor used for misalignment vibration test.

Run #6 M55L (Motor-55 mils-low). In this test run, the motor shaft centerline is set considerably below the pump shaft centerline while still maintaining good side-to-side alignment.

Run #7 M6W (Motor-6 mils-west). The pump and motor were again aligned well within acceptable alignment tolerances (similar to Run #2 M2W) to determine if the vibration response at the bearings would repeat.



FIGURE 2.17 Metal ribbon coupling connecting pump and motor shafts.

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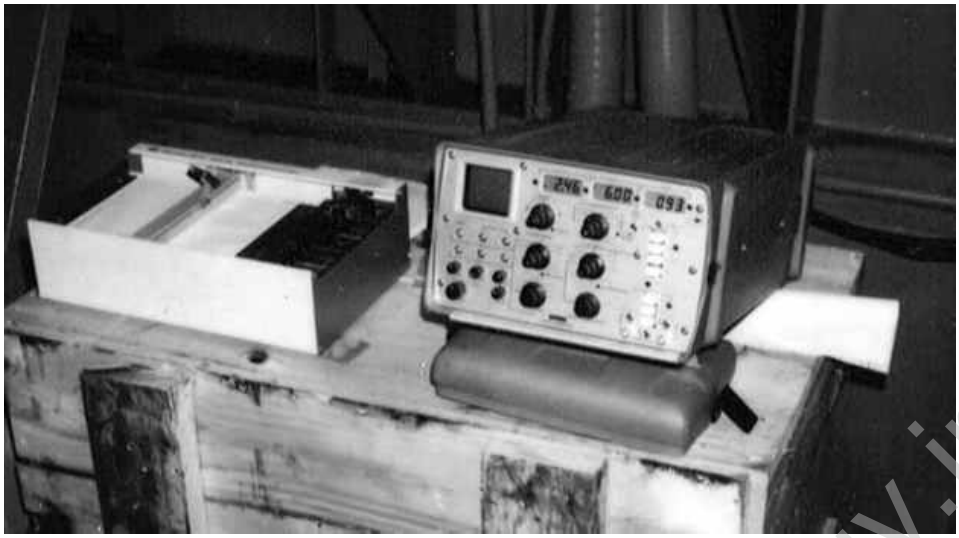


FIGURE 2.18 Vibration equipment used for the test.

The overall vibration results of the seven test runs are shown in Figure 2.19. Figure 2.20 through Figure 2.29 show the vibration spectral data collected at each measurement point during these induced misalignment conditions and were arranged to observe the vibration signature changes during each test run at each of the sensor locations on the motor and the pump.

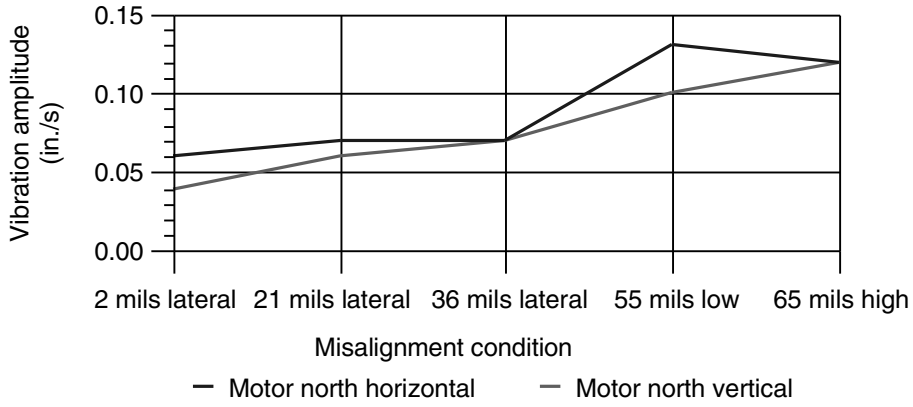
Notice in every trend that the overall vibration only slightly increased, and in some cases, slightly decreased, as the equipment was misaligned by 21 and 36 mils. In some cases, when the equipment was at the greatest misalignment condition (65 mils), the vibration decreased from the levels seen at 55 mils.

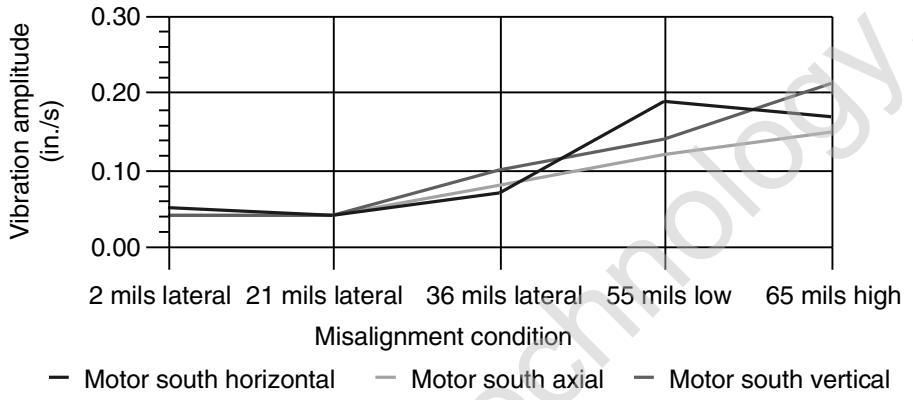
Below are some observations and comments on the vibration data collected:

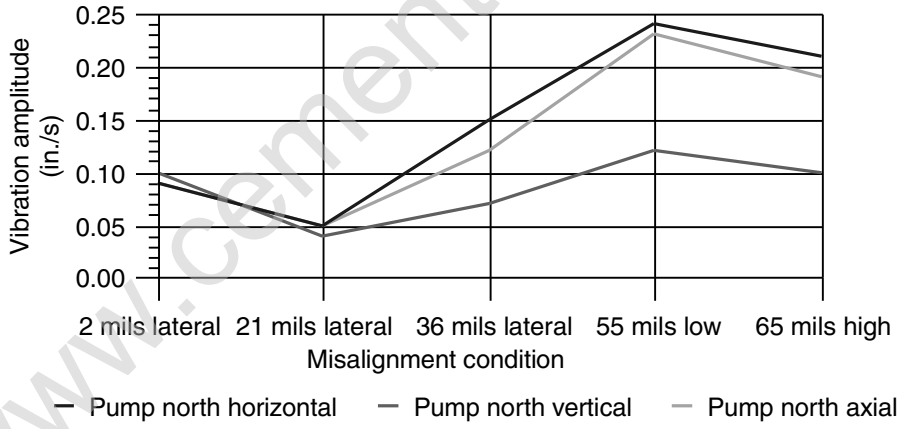
- The largest change in overall vibration levels occurred at the inboard bearings of both the pump and the motor.
- The axial vibration level on the motor attained its highest value when its shaft was higher (above) the pump shaft. Likewise the axial vibration level on the pump attained its highest value when its shaft was higher than the motor shaft.
- The highest horizontal amplitude readings on both the pump and the motor occurred when the motor shaft was low with respect to the pump shaft centerline (M55L) even though the amount of misalignment was not as severe as in the M65H case.
- The pump bearings increased in amplitude when the motor shaft was higher than the pump shaft (from Run #5 to Run #6) whereas the motor bearings decreased or stayed the same.
- The outboard bearing of the pump experienced a greater increase in overall vibration levels compared to the outboard bearing of the motor.
- The pump experienced a decrease in overall vibration levels on both bearings when the motor was misaligned laterally (notice Run #2 and Run #3).
- The horizontal and axial amplitude levels of the inboard pump bearing increased as the amount of the misalignment increased. The vertical levels however changed only slightly during all the different shaft configurations.
- The overall vibration levels on the pump and motor taken during Run #7 were, for the most part, the same as the levels taken during Run #2 where the alignment conditions

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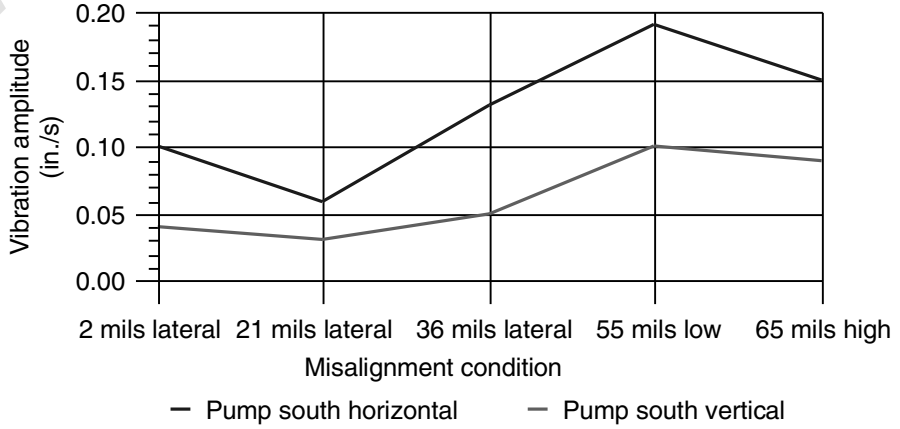


FIGURE 2.19 Overall vibration amplitude levels on bearing housings.

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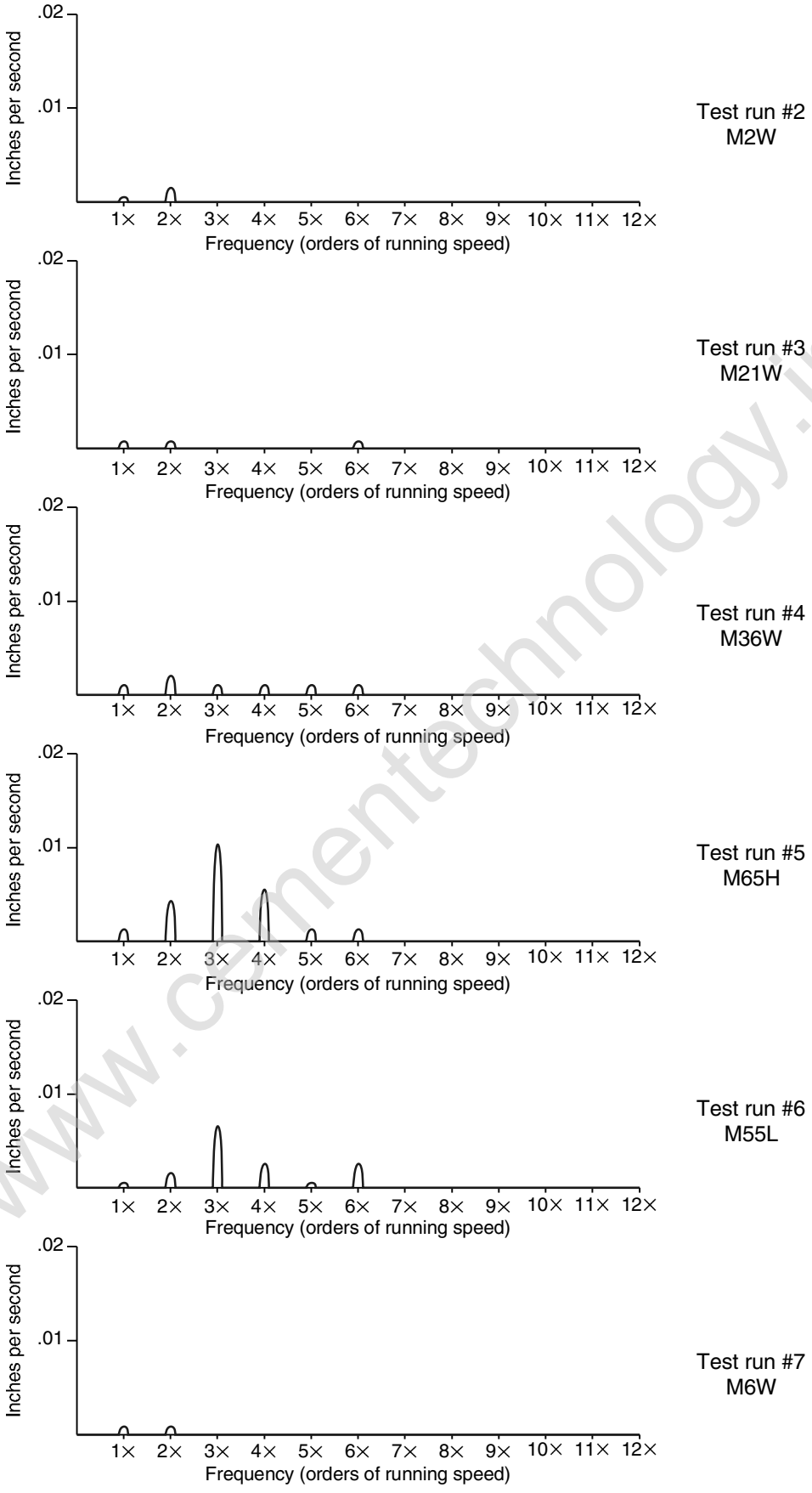


FIGURE 2.20 Outboard MOTOR, horizontal direction.

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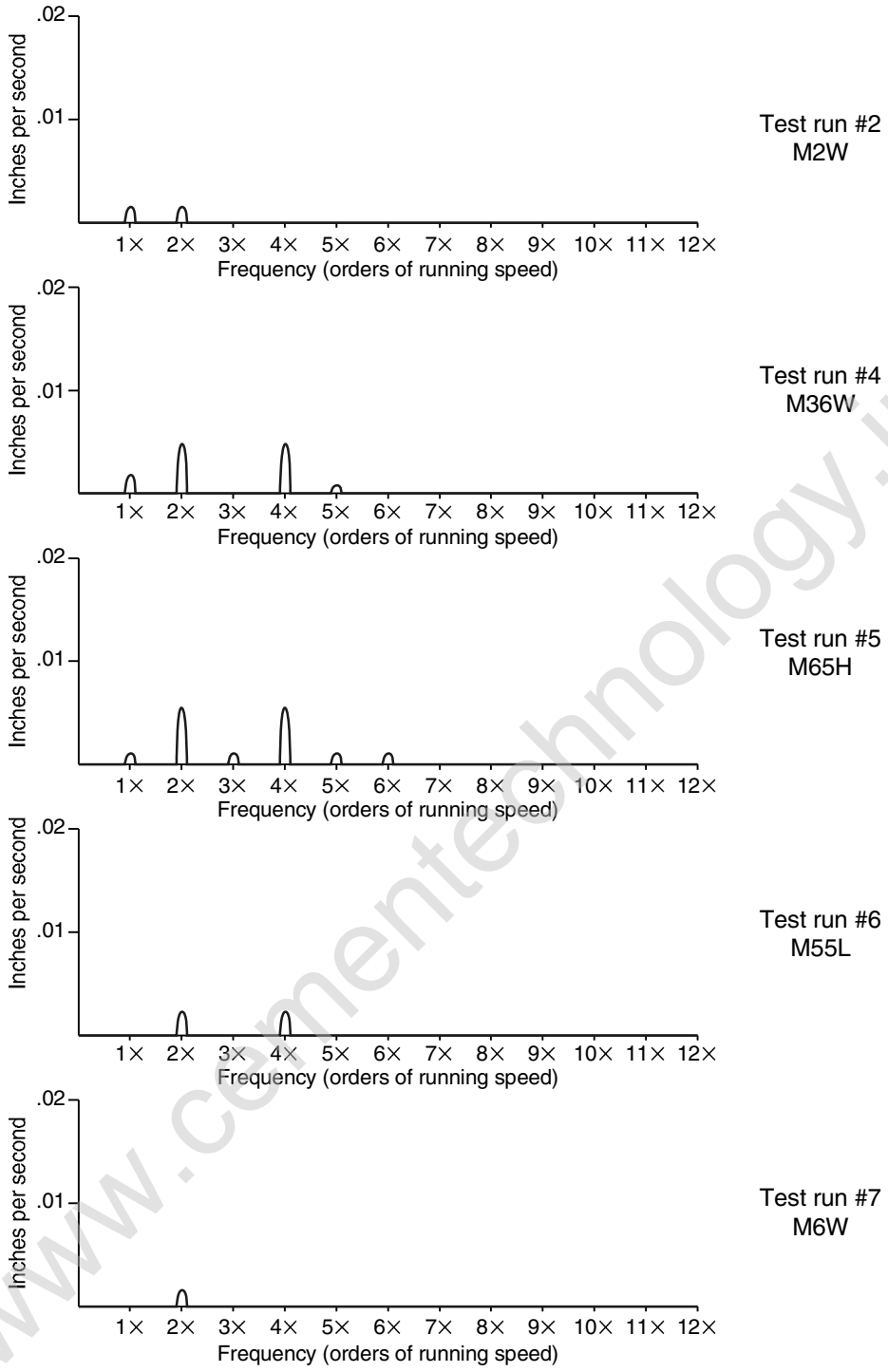


FIGURE 2.21 Outboard MOTOR, vertical direction.

were nearly identical verifying that the vibration was due to misalignment and not other factors.

- The vibration signatures of the pump and motor displayed multiples of running speed with the predominant frequencies occurring at 2×, 3×, 4×, 5×, 6×, 7×, and 8× (i.e., higher order harmonics). The once-per-rev amplitude levels were not affected by any of the various shaft alignment positions regardless of the severity of shaft misalignment.
- Because the once-per-rev amplitude levels were not affected by any misalignment conditions, and in fact decreased in many instances when the misalignment increased, the

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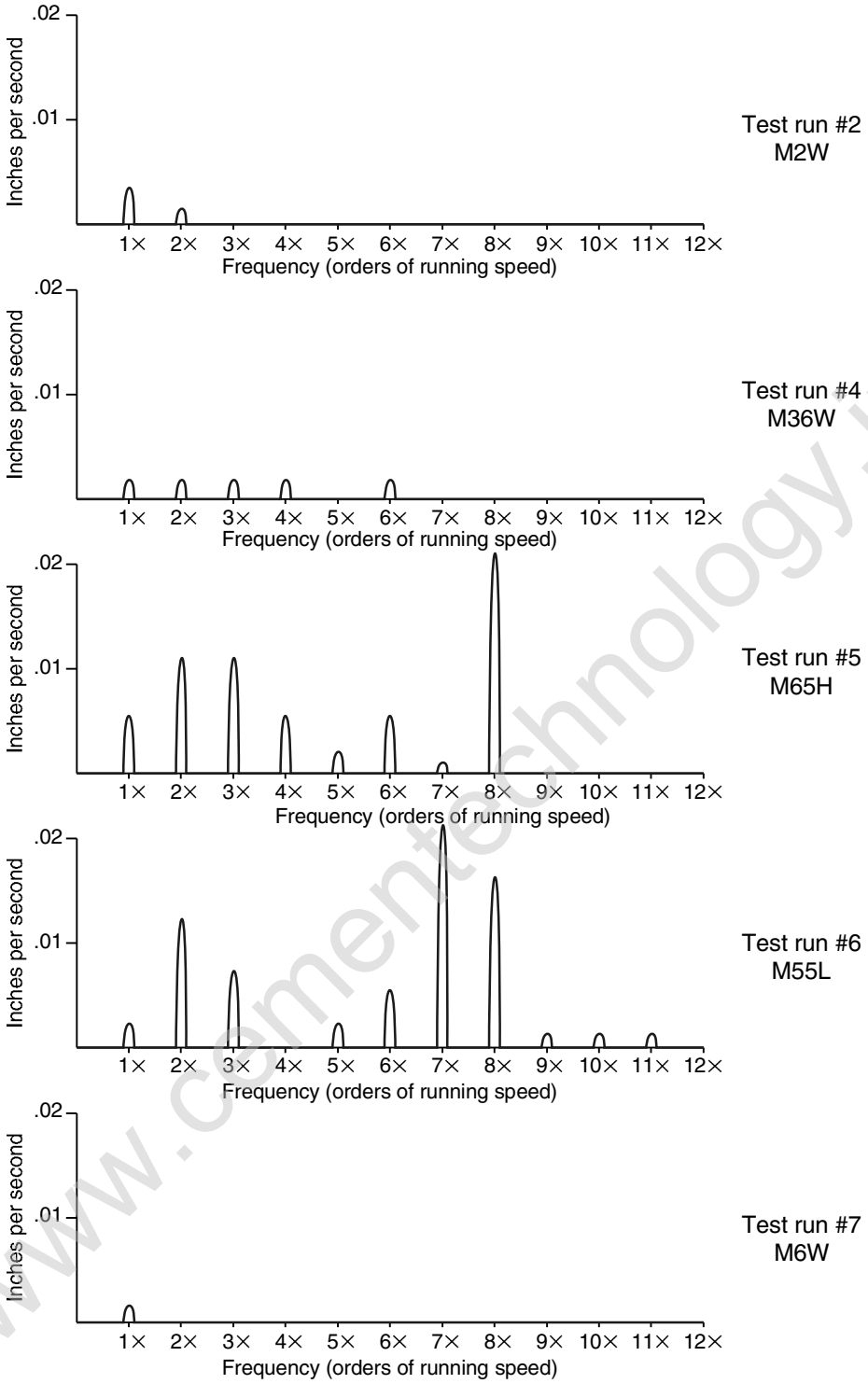


FIGURE 2.22 Inboard MOTOR, horizontal direction.

attempts at taking phase angle data when tracking the once-per-rev signal was inconclusive as the phase angle would continually drift.

- The highest peaks occurred at the maximum misalignment conditions (M65H and M55L) on the inboard sensor locations on the motor.
- The 3× peaks on the motor were higher in amplitude in the horizontal directions than in the vertical direction.
- The 2× and 4× peaks on the motor were more dominant in the vertical and axial directions than in the horizontal direction.

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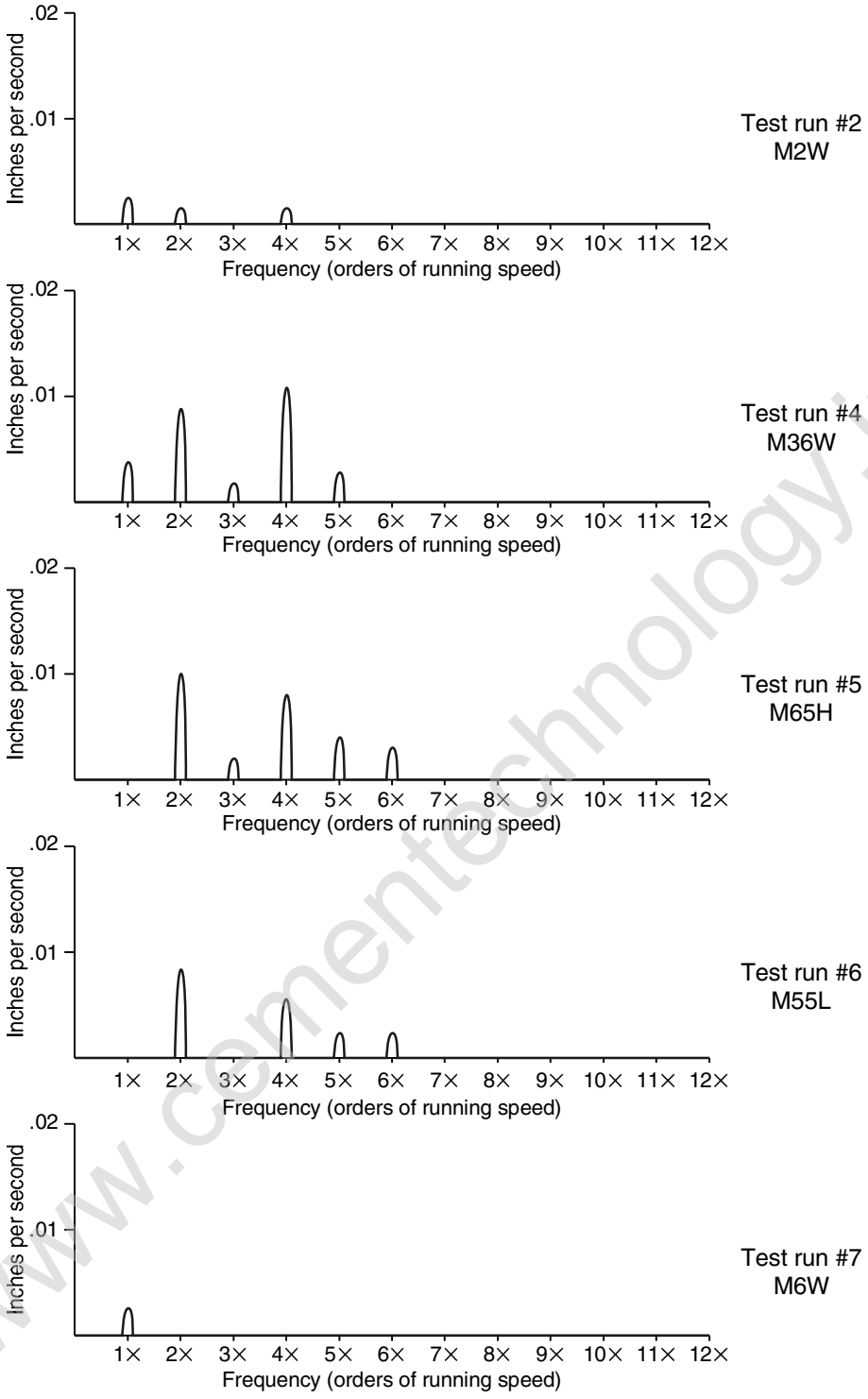


FIGURE 2.23 Inboard MOTOR, vertical direction.

- The 2×, 4×, and 6× peaks prevailed in the pump bearings.
- Higher multiples of running speed occurred on the pump from 40 to 100 kcpm particularly during the vertical misalignment runs.
- The twice, fourth, and eighth running speed frequencies are a result of the S-shaped grid as it traverses from its maximum tilted and pivoted positions twice each revolution on both the coupling hubs. The third and sixth running speed multiples occur as the metal grid in the coupling changes its position during each revolution of the shafts. The

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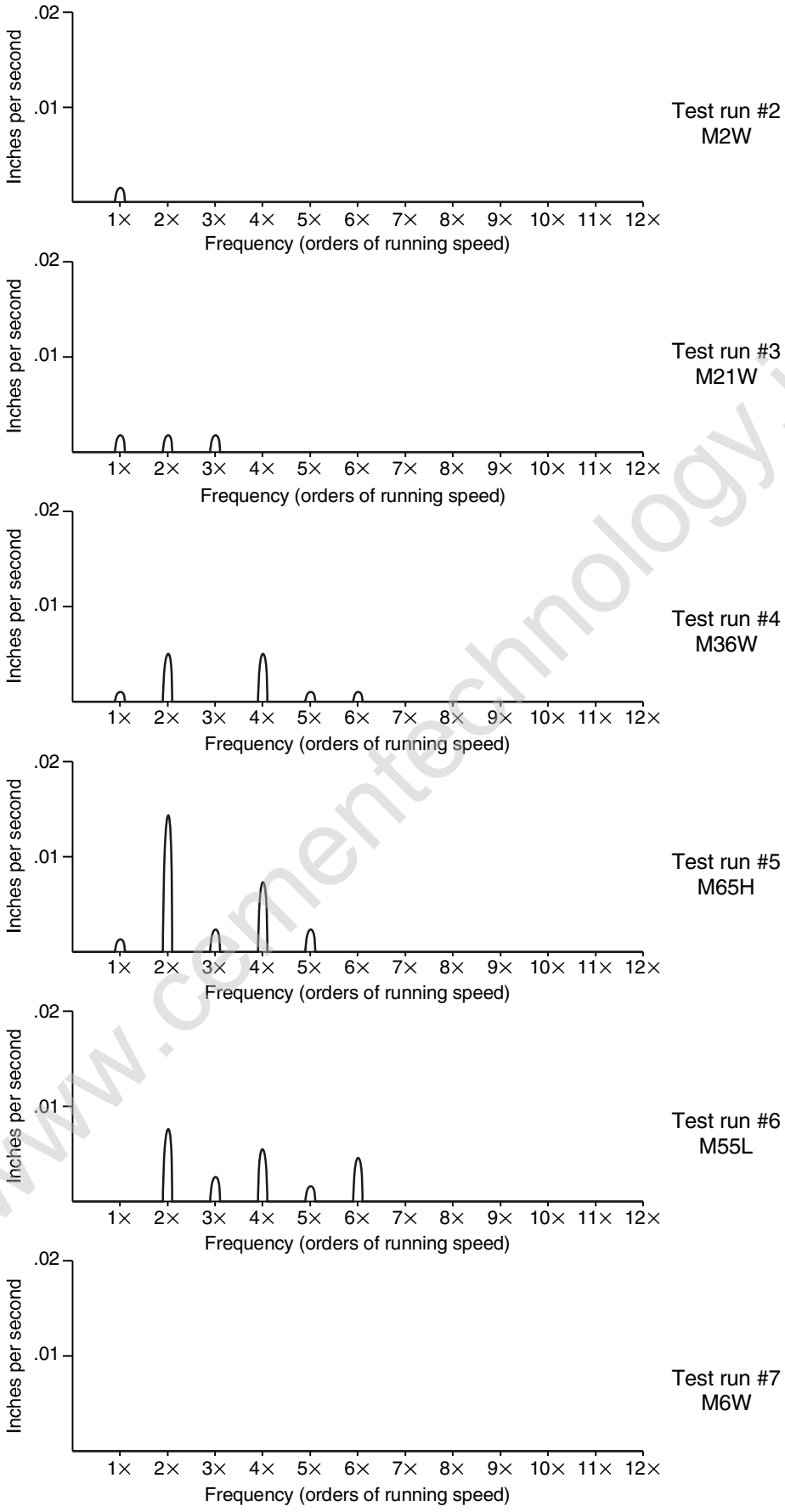


FIGURE 2.24 Inboard MOTOR, axial direction.

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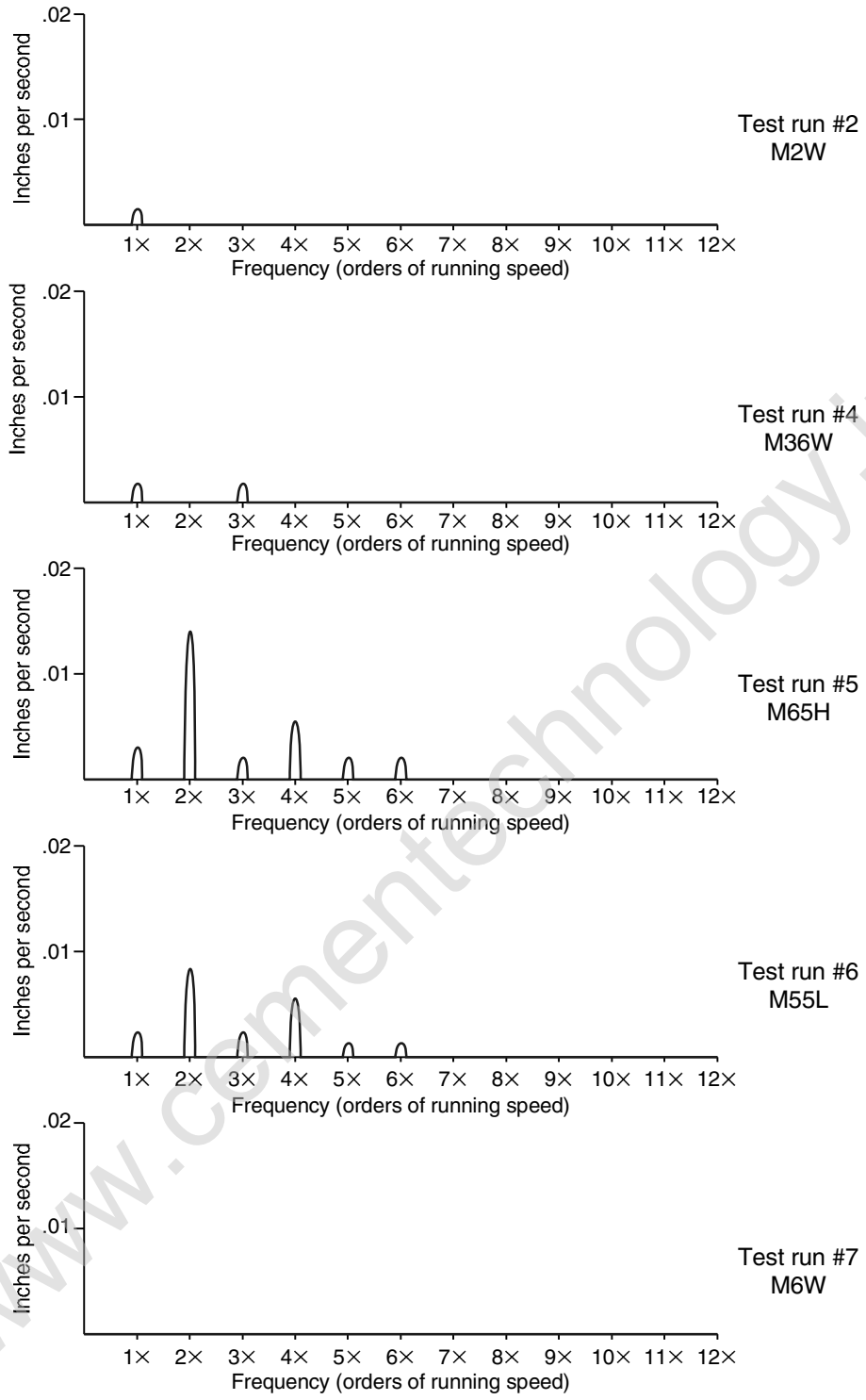


FIGURE 2.25 Inboard PUMP, horizontal direction.

maximum amount of rotational force occurs when the grid is in the tilted position where bending occurs across the thickness of the grid member.

- The seven times running speed peak that occurred in the horizontal direction on the inboard motor bearing during the M55L run and the five times running speed peak that occurred in the axial direction on the pump are, as yet, not completely understood as to the source of the forcing mechanism involved. The higher multiples appear to be caused by overloading the antifriction bearings.

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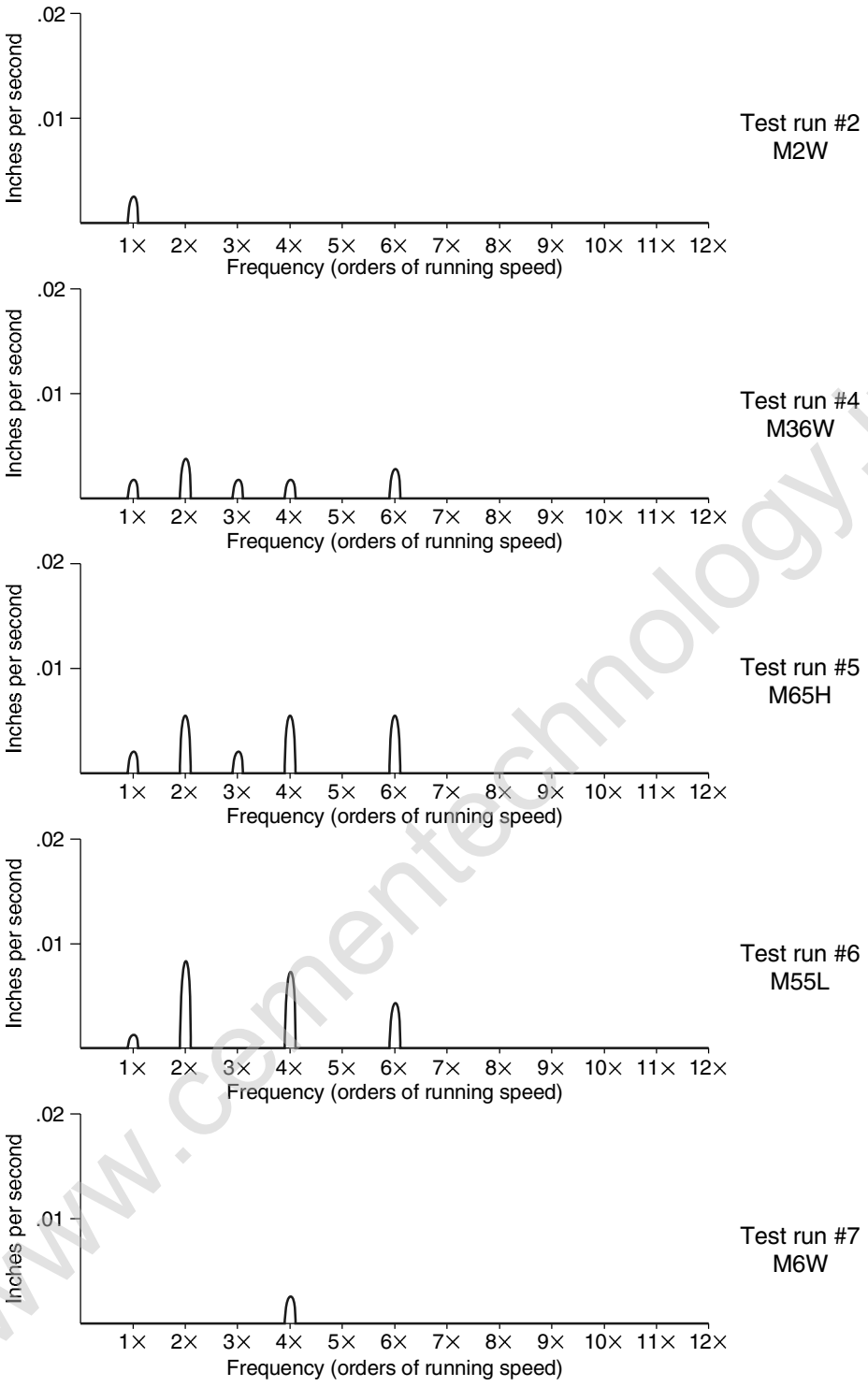


FIGURE 2.26 Inboard PUMP, vertical direction.

2.2.8 Before and After Vibration Results Found on a Misaligned Motor and Pump

This case history shows actual alignment and vibration data from a drive system that had been operating under a misalignment condition. Vibration information was collected before shutdown and realignment, the unit was then aligned properly, started back up, and vibration

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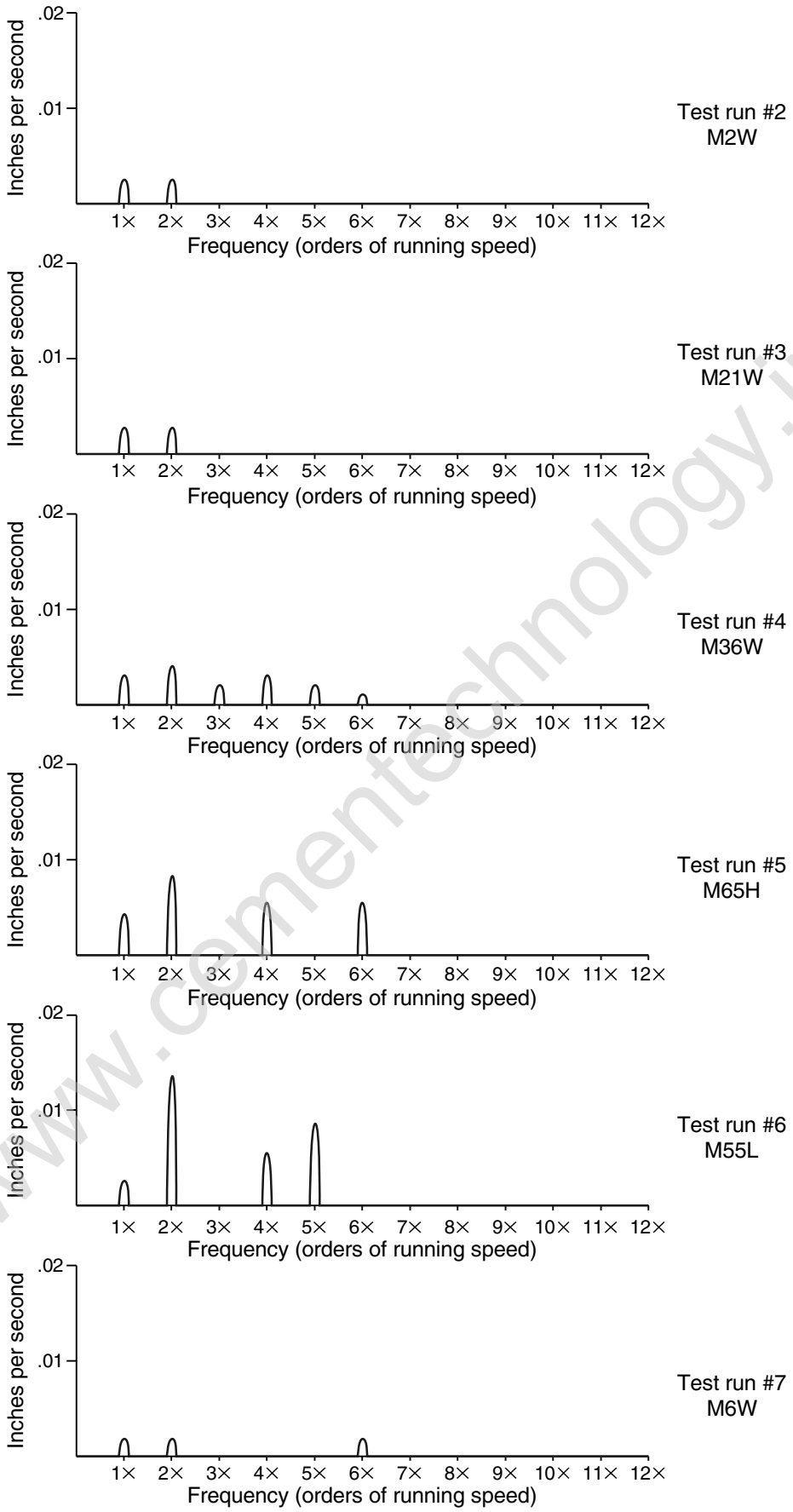


FIGURE 2.27 Inboard PUMP, axial direction.

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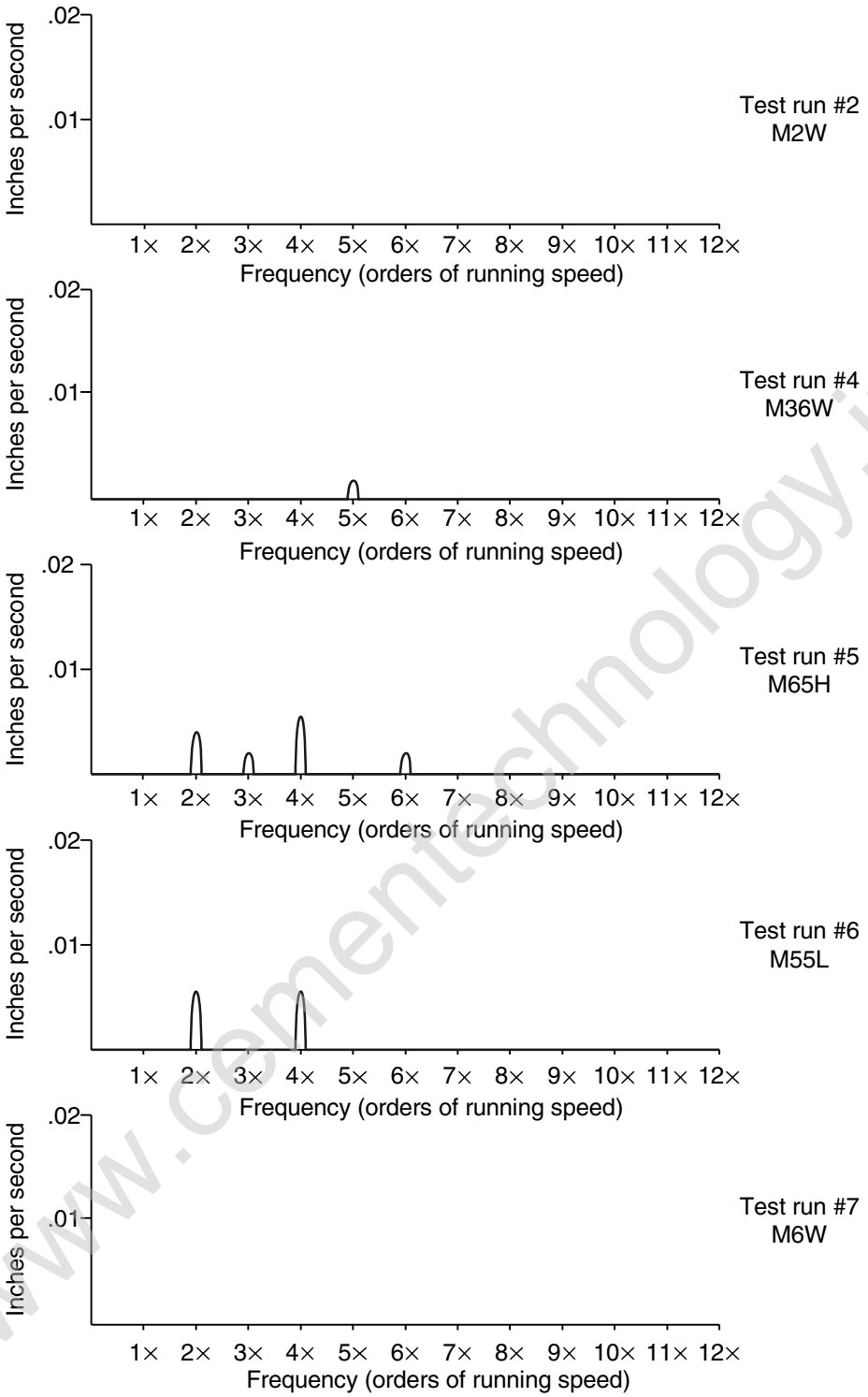


FIGURE 2.28 Outboard PUMP, horizontal direction.

data taken again. Figure 2.30 shows the as-found and final alignment-data. Figure 2.31 shows the before and after radial vibration spectral data on the motor. Figure 2.32 shows the before and after radial vibration spectral data on the pump. Figure 2.33 shows the before and after axial vibration spectral data on both the motor and the pump. Notice that the radial and axial vibrations on the motor increased and the vibration on the pump decreased after the misalignment was corrected.

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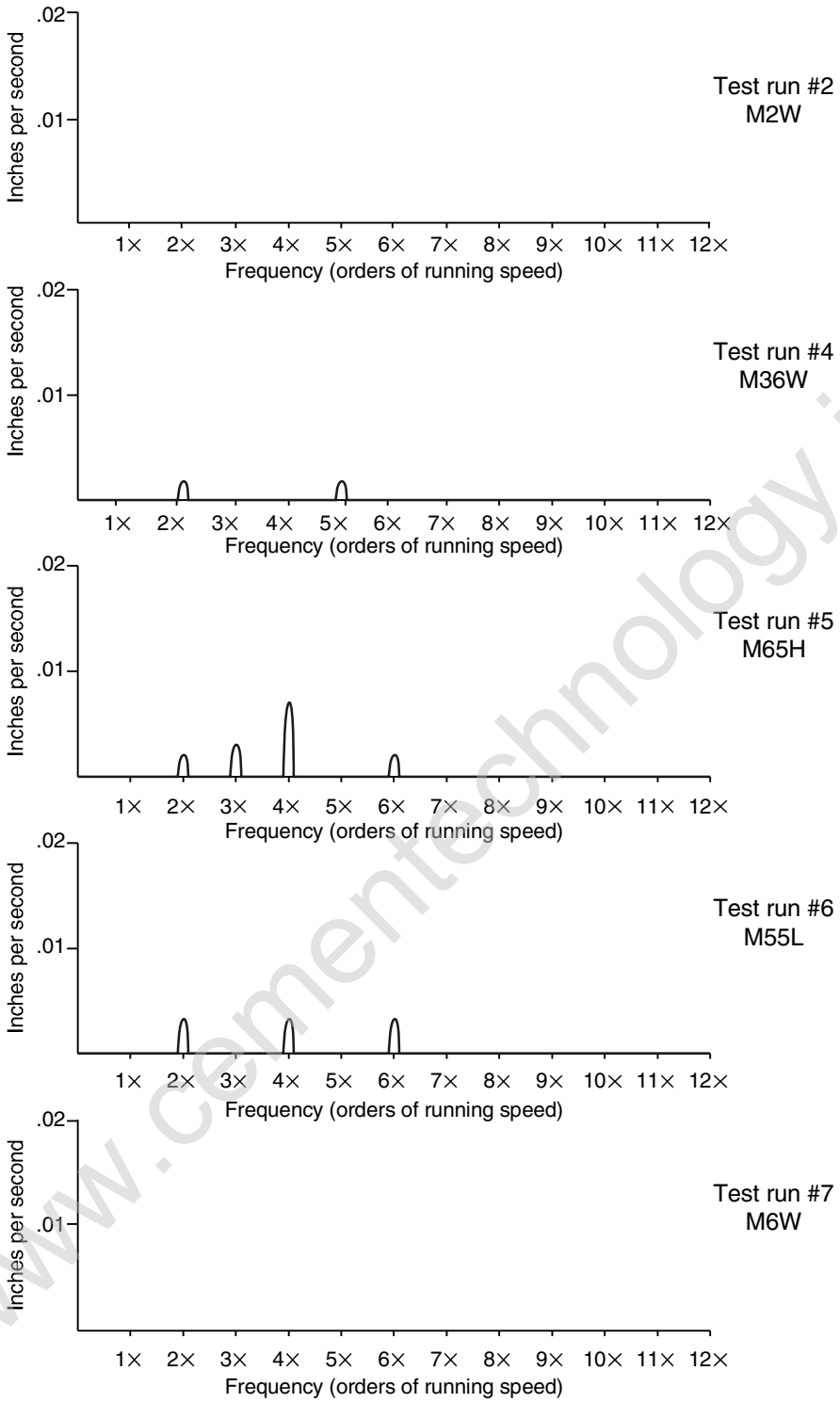


FIGURE 2.29 Outboard PUMP, vertical direction.

2.2.9 Why Vibration Levels Often Decrease with Increasing Misalignment

As illustrated in Figure 2.2, rotating machinery shafts are exposed to two types of forces. Static forces that act in one direction and dynamic forces that change their direction. Static forces are also called preloads. Preloads on shafts and bearings are caused from many of the following sources:

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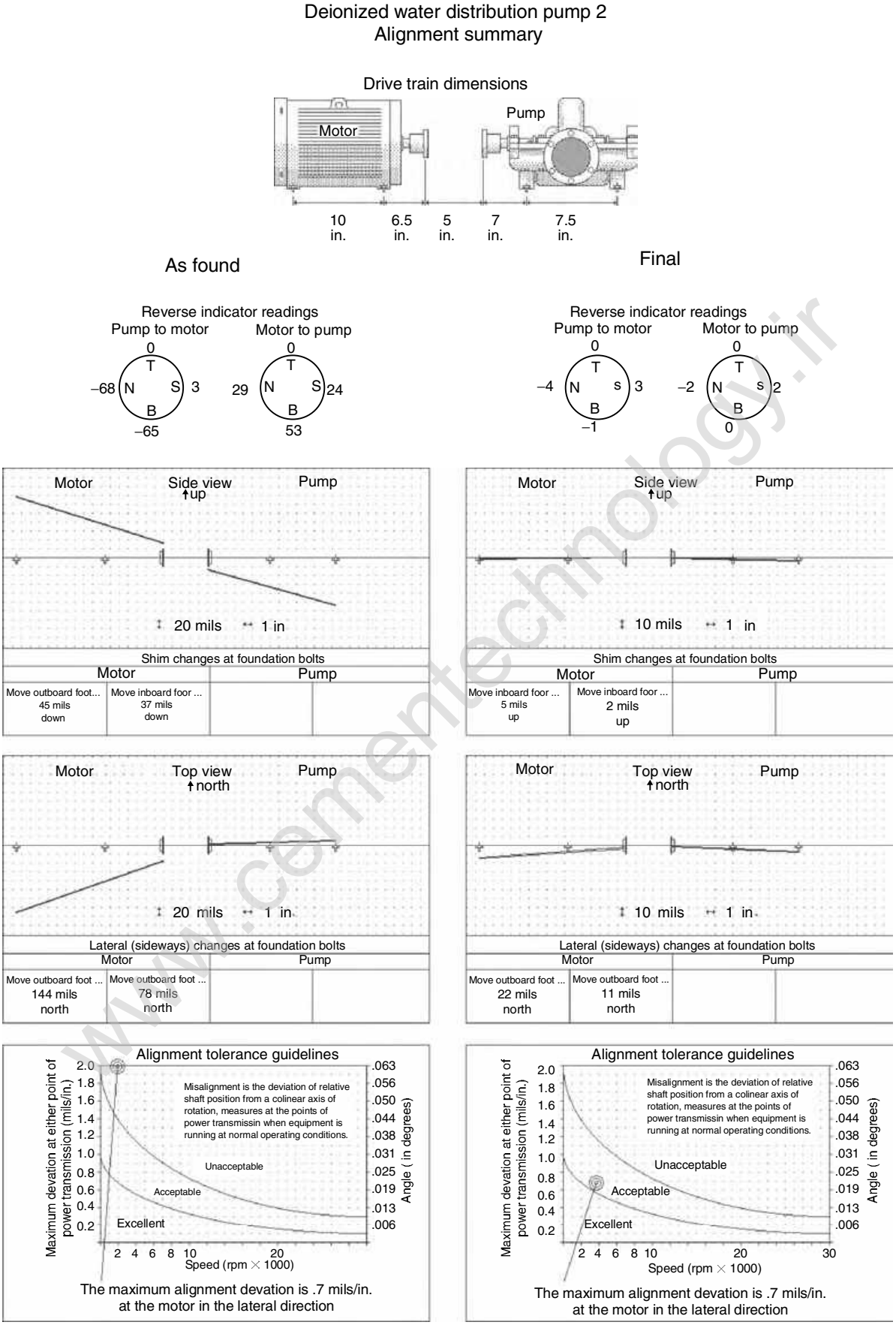


FIGURE 2.30 As-found and final alignment data on motor and pump.

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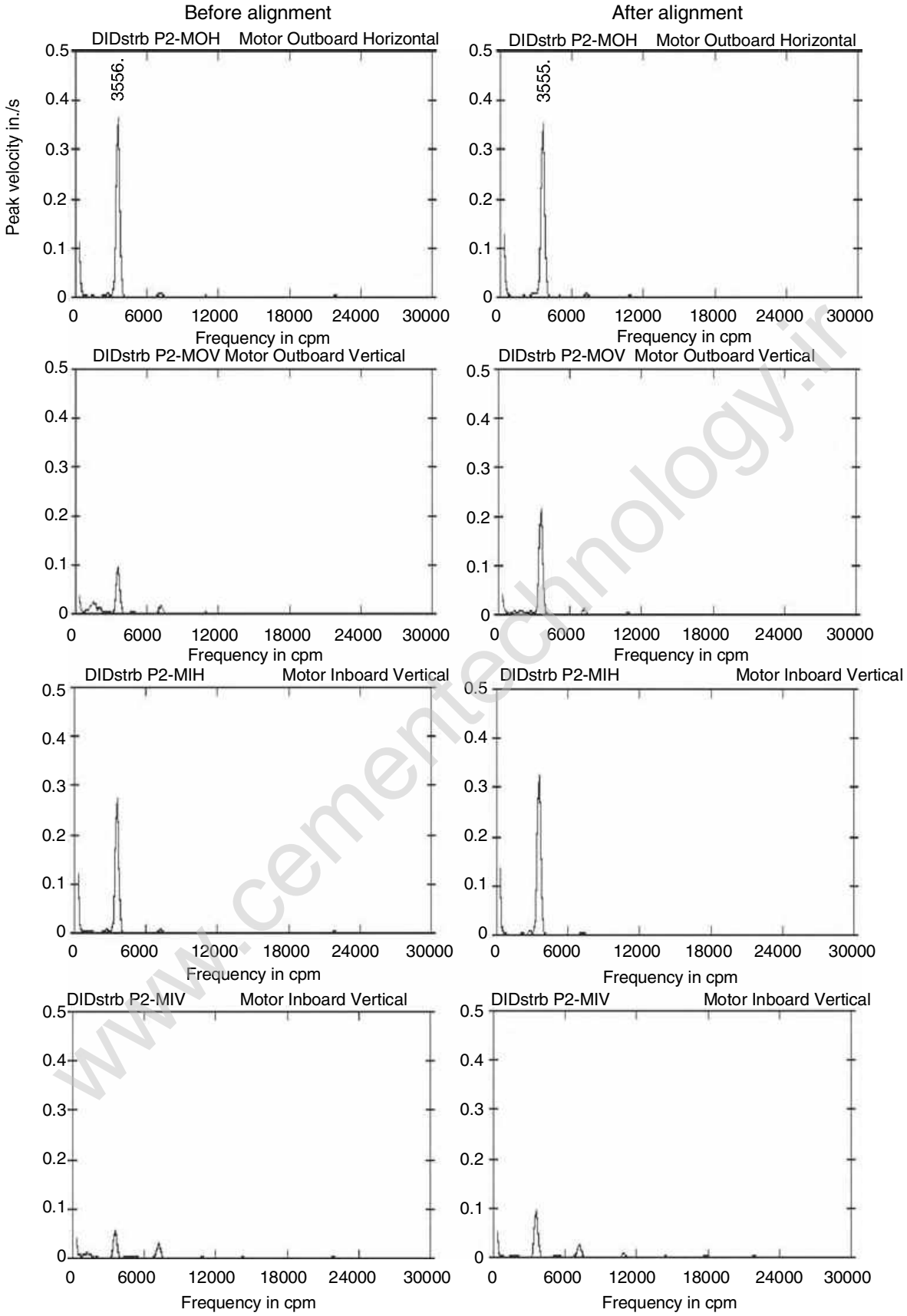


FIGURE 2.31 Before and after radial vibration data on motor.

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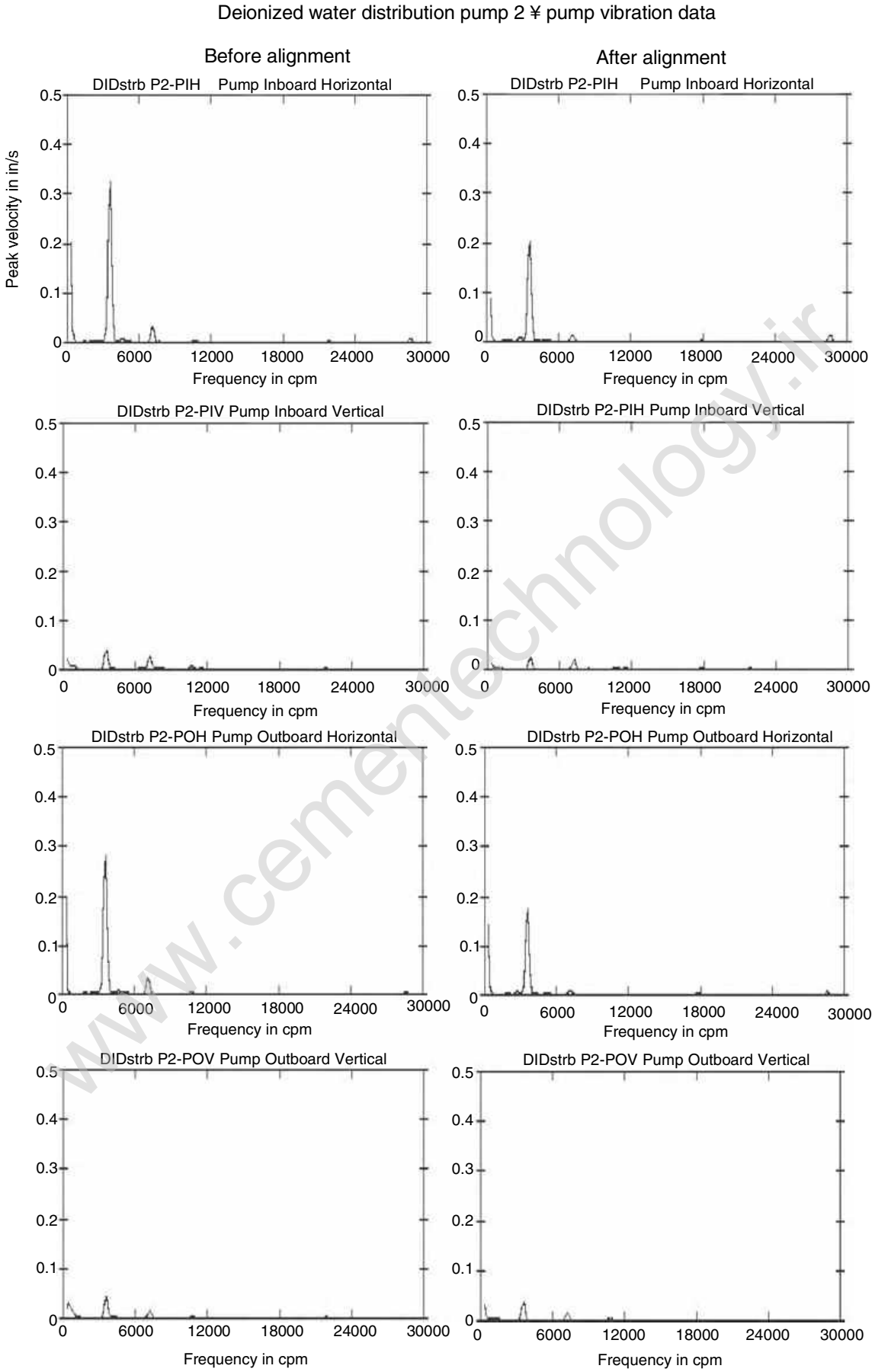


FIGURE 2.32 Before and after radial vibration data on pump.

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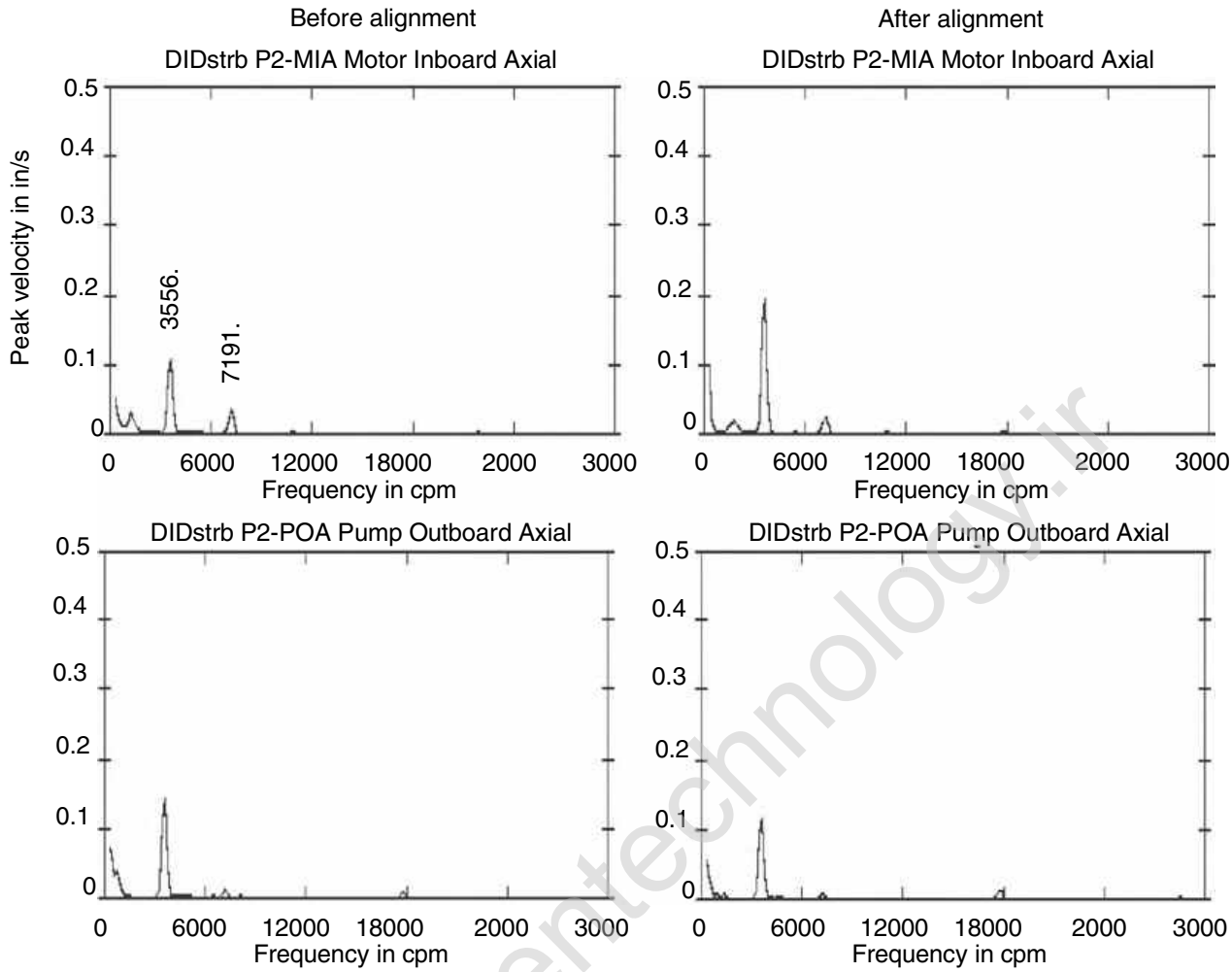


FIGURE 2.33 Before and after axial vibration data on motor and pump.

- Gravitational force
- V-belt or chain tension
- Shaft misalignment
- Some types of hydraulic or aerodynamic loads

Dynamic loads on shafts and bearings are caused by some of the following sources (not a complete list by any means):

- Out of balance condition (i.e., the center of mass is not coincident with the centerline of rotation)
- Eccentric rotor components or bent shafts (another form of unbalance)
- Damaged antifriction bearings
- Intermittent, period rubs
- Gear tooth contact
- Pump or compressor impeller blades passing by a stationary object
- Electromagnetic forces

Simply stated, vibration is motion. Vibratory motion in machinery is caused by forces that change their direction. For example, a rotor that is out of balance and is not

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rotating, does not vibrate. As soon as the imbalanced rotor begins to spin, it also begins to vibrate. This occurs because the ''heavy spot'' is changing its position, causing the (centrifugal) force to change its direction. The rotor/bearing/support system, being elastic, consequentially begins to flex or move as these alternating forces begin to act on the machine.

Another detectable vibration pattern exists in gears and is commonly referred to as gear mesh. Gear mesh can be detected as forces increase or subside as each tooth comes in contact with another. Other types of mechanical or electrical problems that can be detected through vibration analysis can be traced back to the fact that forces are somehow changing their direction.

On the other hand, when two or more shafts are connected together by some flexible or rigid element where the centerlines of each machine are not collinear, the forces transferred from shaft to shaft are acting in one direction only. These forces do not change their direction, as an imbalance condition does. If a motor shaft is higher than a pump shaft by 50 mils, the motor shaft is trying to pull the pump shaft upward to come in line with the motor shaft position. Conversely, the pump shaft is trying to pull the motor shaft downward to come in line with the pump shaft position. The misalignment forces will begin to bend the shafts, not flutter them around like the tail of a fish.

Static forces caused by misalignment act in one direction only, which is quite different than the dynamic forces that generate vibration. Under this pretense, how could misalignment ever cause vibration to occur? If anything, misalignment should diminish the capacity for motion to occur in a rotor/bearing/support system.

2.2.10 Known Vibration Spectral Signatures of Misaligned Flexible Couplings

Despite the fact that shaft misalignment may decrease the amount of vibration in rotating machinery, vibration can and does occur due to this condition. As previously mentioned, it has been observed that the vibration spectral pattern of misaligned rotating machinery will frequently be different depending on the type of flexible coupling connecting the two shaft together.

Figure 2.34 through Figure 2.39 show vibration patterns that have been observed on misaligned rotating machinery with different types of flexible couplings. Notice that the vibration peaks are occurring at running speed (1X) or multiples of running speed (2X, 3X, 4X, etc.).

2.2.11 Vibration Characteristics of Misaligned Machinery Supported in Sliding Type Bearings

The vibration spectral patterns in Figure 2.34 through Figure 2.39 were seen on rotating machinery supported in rolling element type bearings. Frequently a different pattern emerges on machinery supported in sliding type bearings as shown in Figure 2.40.

2.2.12 Using Infrared Thermography to Detect Misalignment

A very interesting study was performed by two maintenance technicians from a bottling company in 1991. The test was conducted by coupling a 10 hp motor to a 7200 W electric generator. A specific flexible coupling was installed between the motor and the generator; the unit was then accurately aligned and then started up. Vibration, ultrasound, and thermal

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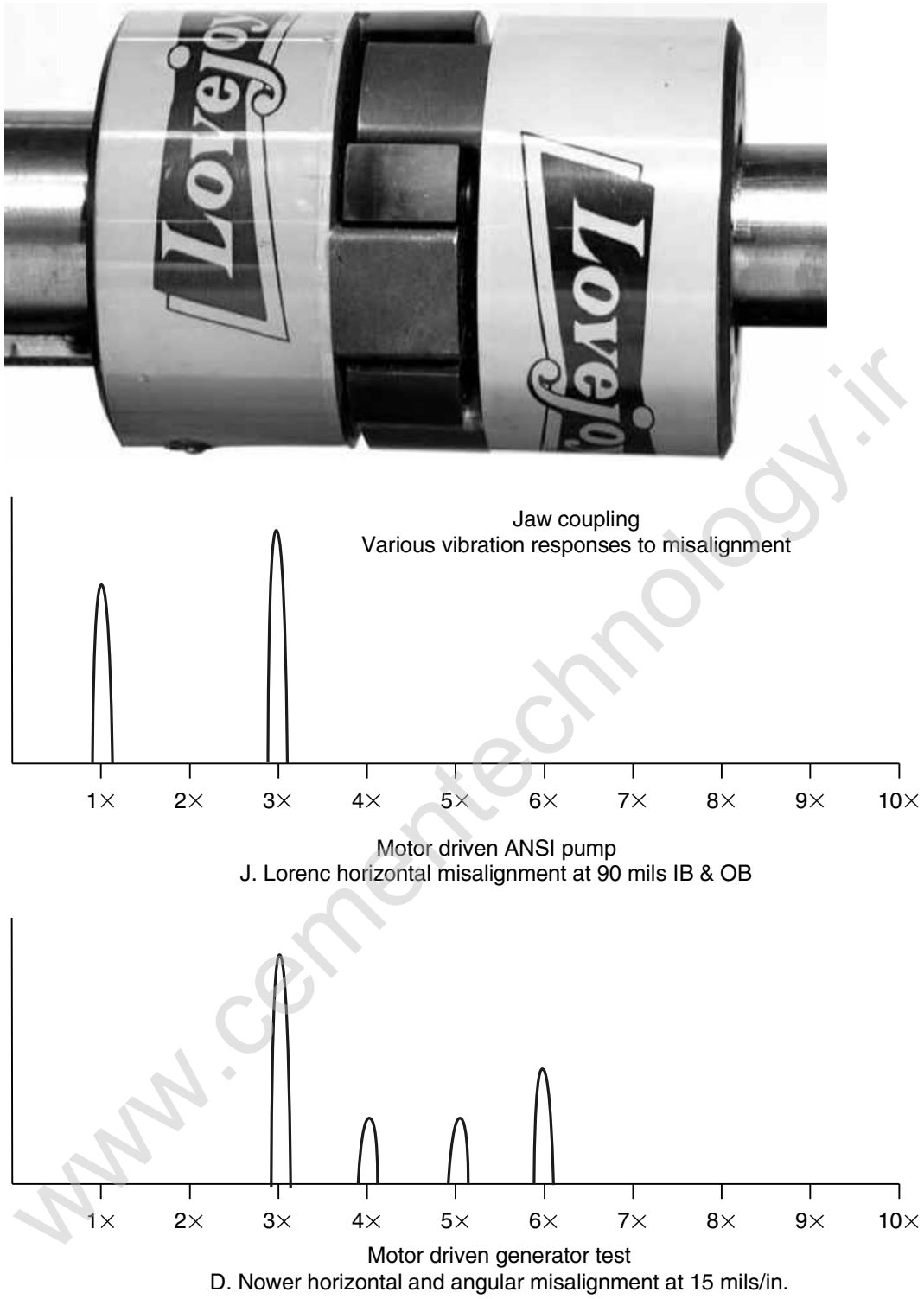


FIGURE 2.34 Observed vibration patterns on misaligned jaw-type couplings. (Courtesy of Lovejoy, Downers Grove, IL. With permission.)

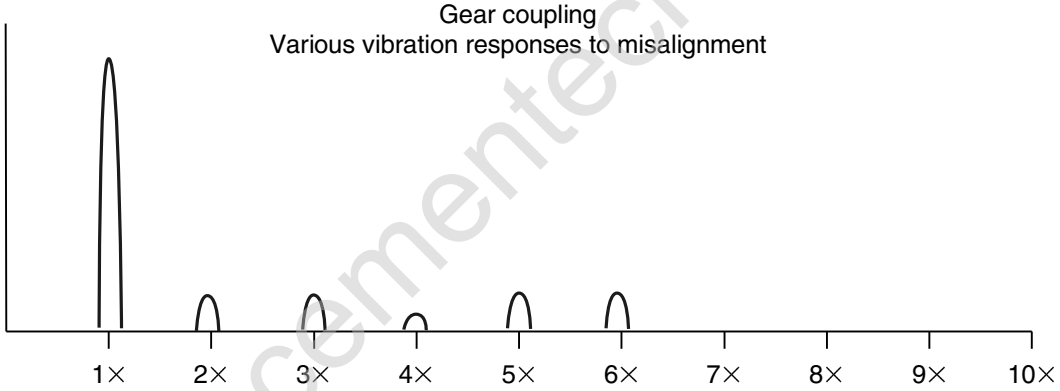
imaging data was then collected after 10 min run time. The unit was then shutdown, 10 mils of shims were placed under all 4 ft of the motor, the drive system started back up and the data was collected again. This was repeated several times with an additional 10 mils of shims installed under the motor feet each time. After the motor and generator drive was misaligned

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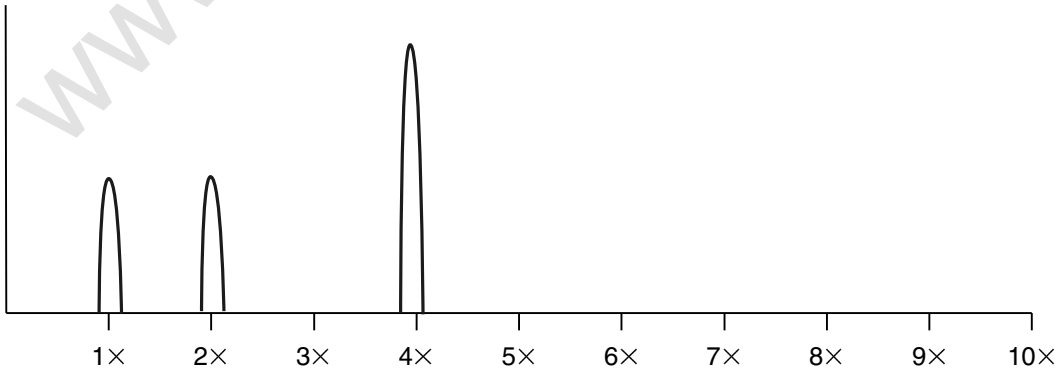
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Gear coupling
Various vibration responses to misalignment



Motor driven ANSI pump
J. Lorenc horizontal misalignment at 30 mils IB & OB



Gas/power turbine driven compressor
J. Piotrowski horizontal misalignment at 65 mils IB & OB

FIGURE 2.35 Observed vibration patterns on misaligned gear type couplings. (Courtesy of Rexmord Coupling Group, Milwaukee, WI. With permission.)

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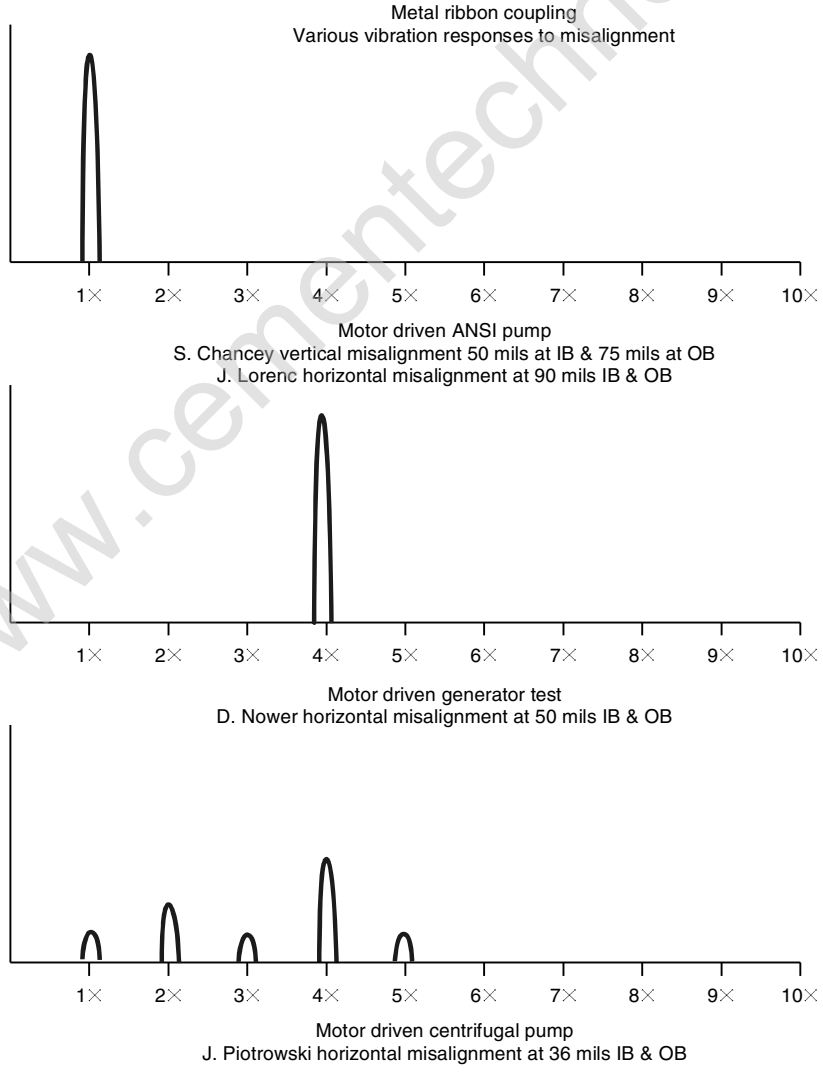


FIGURE 2.36 Observed vibration patterns on misaligned metal ribbon-type couplings. (Courtesy of Rexmord Coupling Group, Milwaukee, IL. With permission.)

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30–40 mils, the flexible coupling being tested was removed, a different flexible coupling design was then installed, the shims were removed from the motor to get back to near perfect alignment, and the process was repeated.

Figure 2.41 through Figure 2.46 show the results of the six different flexible couplings that were tested. Notice that as the misalignment increased, so too did the temperature of the coupling or of the flexing element. The increase in temperature is somewhat linear as illustrated in the temperature graphs with each coupling tested. Disappointingly, however, the vibration and ultrasound data was never published with the infrared data.

In addition, there must be a word of caution here because it is very tempting to make generalizations from this data. Not every flexible or rigid coupling will increase in temperature when subjected to misalignment conditions. The flexible couplings used in this test were mechanically flexible couplings (the chain and metal ribbon types) or elastomeric types.

In mechanically flexible couplings the heat is generated as the metal grid slides back and forth across the tooth slots in the coupling hubs or as the chain rollers slide across the sprocket teeth as the coupling elements attempt to accept the misalignment condition. In the elastomeric couplings, the elastomer is heated through some sliding friction but primarily by shear and compression forces as these coupling elements attempt to accept their misalignment conditions.

What would have happened if a flexible disk or diaphragm type coupling was also tested? Flexible disk or diaphragm couplings accept misalignment conditions by elastically bending the two disk packs or diaphragms and virtually no heat will be generated by the flexure of metal disks as these types of couplings attempt to accommodate any misalignment conditions.

2.2.13 Power Loss due to Shaft Misalignment

It has been widely publicized that shaft misalignment will cause the driver to work harder and therefore take more energy or power to run the drive system. However, a study conducted by the University of Tennessee in 1997 where both 50 and 60 hp motors were purposely misaligned to dynamometers using four different types of couplings and subjecting each coupling to 15 misalignment conditions came to the following conclusions: "The results of these tests show no significant correlation between misalignment and changes in efficiency when the tested couplings were operated within the manufacturer's recommended range. Power consumption and power output remained constant regardless of the alignment condition."

2.2.14 The Most Effective Way to Determine if Misalignment Exists

After years of study, one invariable conclusion can be made. Misalignment disguises itself very well on the operating rotating machinery. There are no easy or inexpensive ways to determine if rotating machinery is misaligned while it is running. The most effective way to determine if a misalignment condition exists is to shut the drive system down, safety tag and lock out the machinery, remove the coupling guard, and employ one of the alignment measurement methods described in Chapter 7 to see if a misalignment condition is present. Even if the alignment looks good when you do an off-line check, running misalignment may occur. So it is suggested that you also review Chapter 9, which discusses off-line to running machinery movement.

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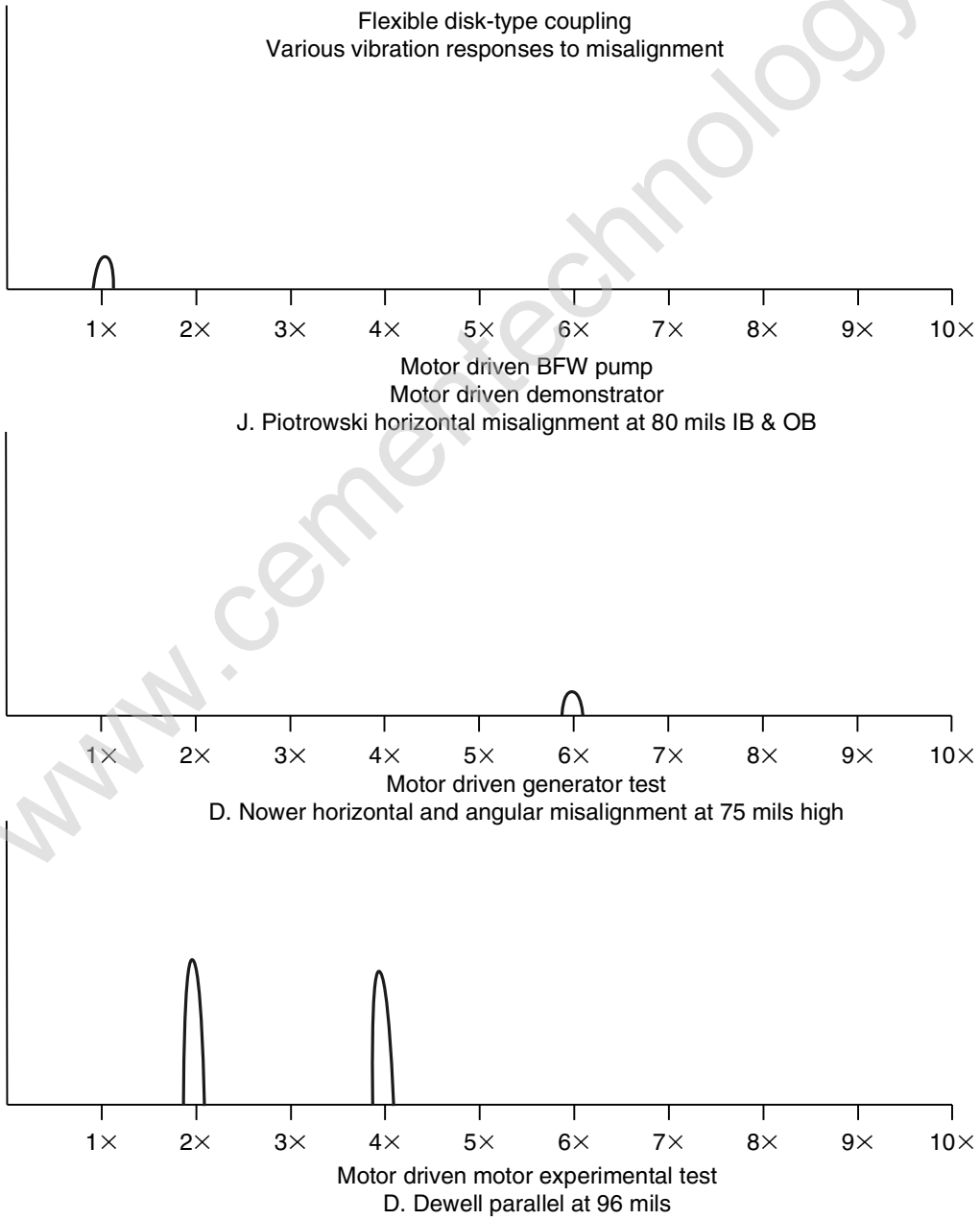
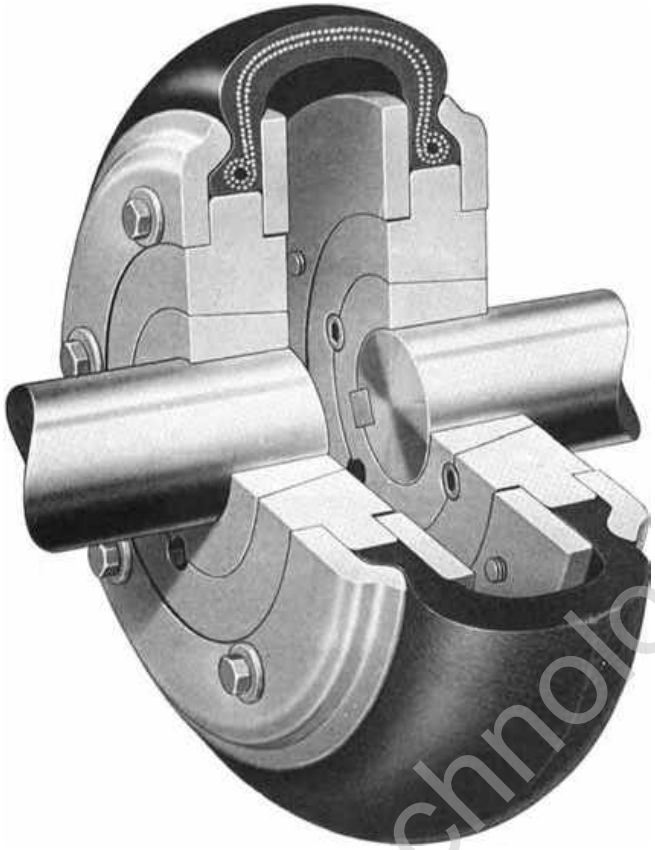


FIGURE 2.37 Observed vibration patterns on misaligned flexible disk-type couplings. (Courtesy of Thomas Rexnord, Warren, PA. With permission.)

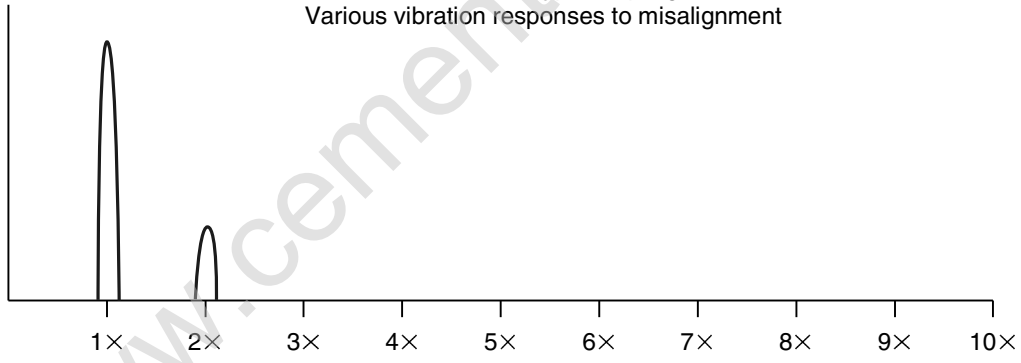
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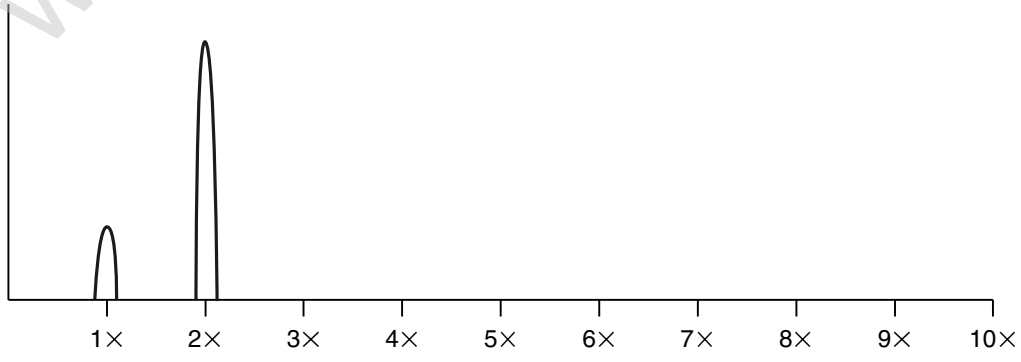
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Rubber tire-type coupling
Various vibration responses to misalignment



Motor driven ANSI pump
J. Lorenc horizontal misalignment at 90 mils IB & OB
J. Piotrowski horizontal misalignment at 80 mils IB & OB



Motor driven generator test
D. Nower horizontal and angular misalignment at 75 mils high

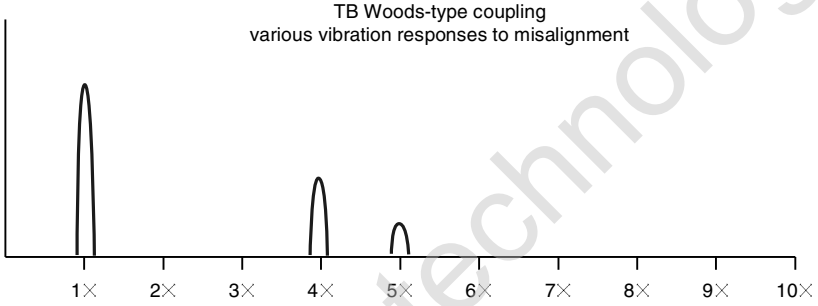
FIGURE 2.38 Observed vibration patterns on misaligned flexible disk-type couplings. (Courtesy of Dodge-Reliance Electric, Cleveland, OH. With permission.)

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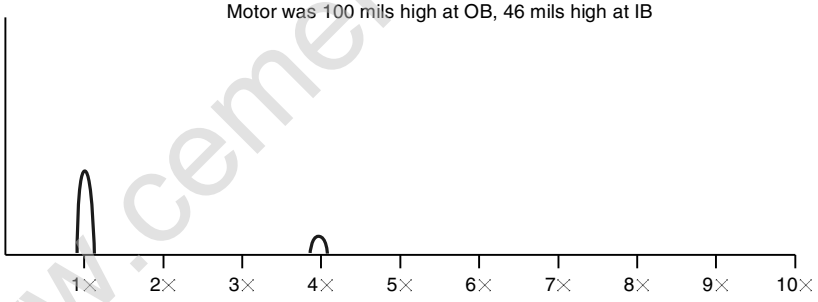
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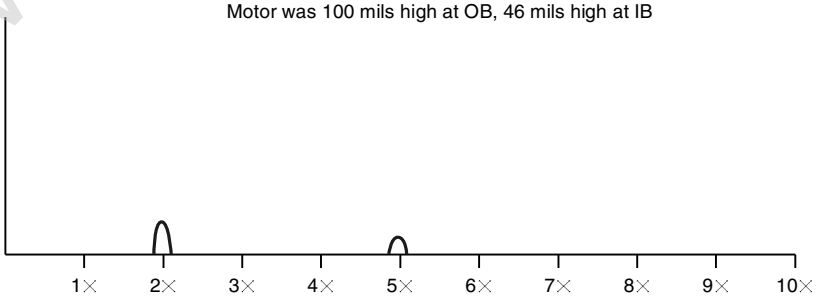
TB Woods-type coupling
various vibration responses to misalignment



Motor driven pump—Motor OB Hrz
vertical misalignment
Motor was 100 mils high at OB, 46 mils high at IB



Motor driven pump—Motor IB Hrz
vertical misalignment
Motor was 100 mils high at OB, 46 mils high at IB



Motor driven pump—Pump IB Hrz
vertical misalignment
Motor was 100 mils high at OB, 46 mils high at IB

FIGURE 2.39 Observed vibration patterns on misaligned flexible disk-type couplings. (Courtesy of T. B. Woods and Sons, Chambersburg, PA. With permission.)

(*continued*)

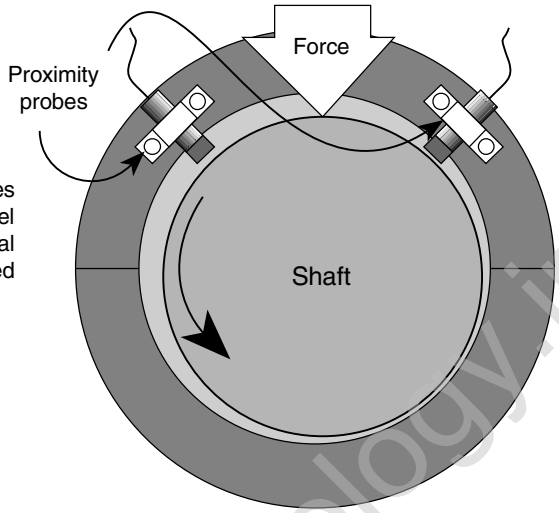
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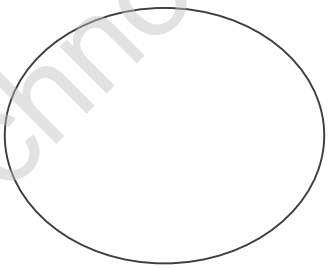
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Sliding type bearing

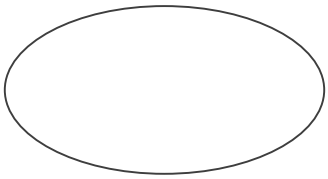
When the signals from two proximity probes are combined together in a two channel oscilloscope or vibration analyzer, the orbital motion of the shaft can be observed (called a Lissajous pattern).



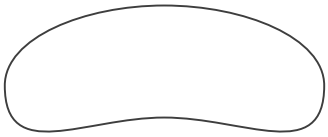
A typical shaft orbit in a sliding type bearing with no external forces applied to the shaft is shown to the right. Even if a pure imbalance condition existed causing an even radial force, the orbital pattern would be elliptical due to the different horizontal and vertical stiffnesses of the machine case.



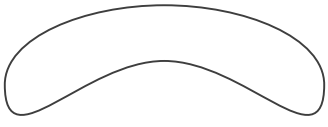
If a downward force from shaft misalignment is now applied to the rotor/bearing system, the elliptical orbit begins to "flatten out". The static misalignment force is limiting the amount of shaft movement in the vertical direction.



If the force from misalignment increase the orbit continues to flatten and distort.



As the force begins to steadily increase, the orbit begins to take a pickle shape.



When the force is great enough, the orbit changes shape to a figure "8", hence a 2× running speed vibration component appears.

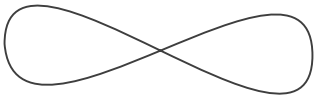


FIGURE 2.40 Observed vibration orbital patterns on rotors supported in sliding type bearings.

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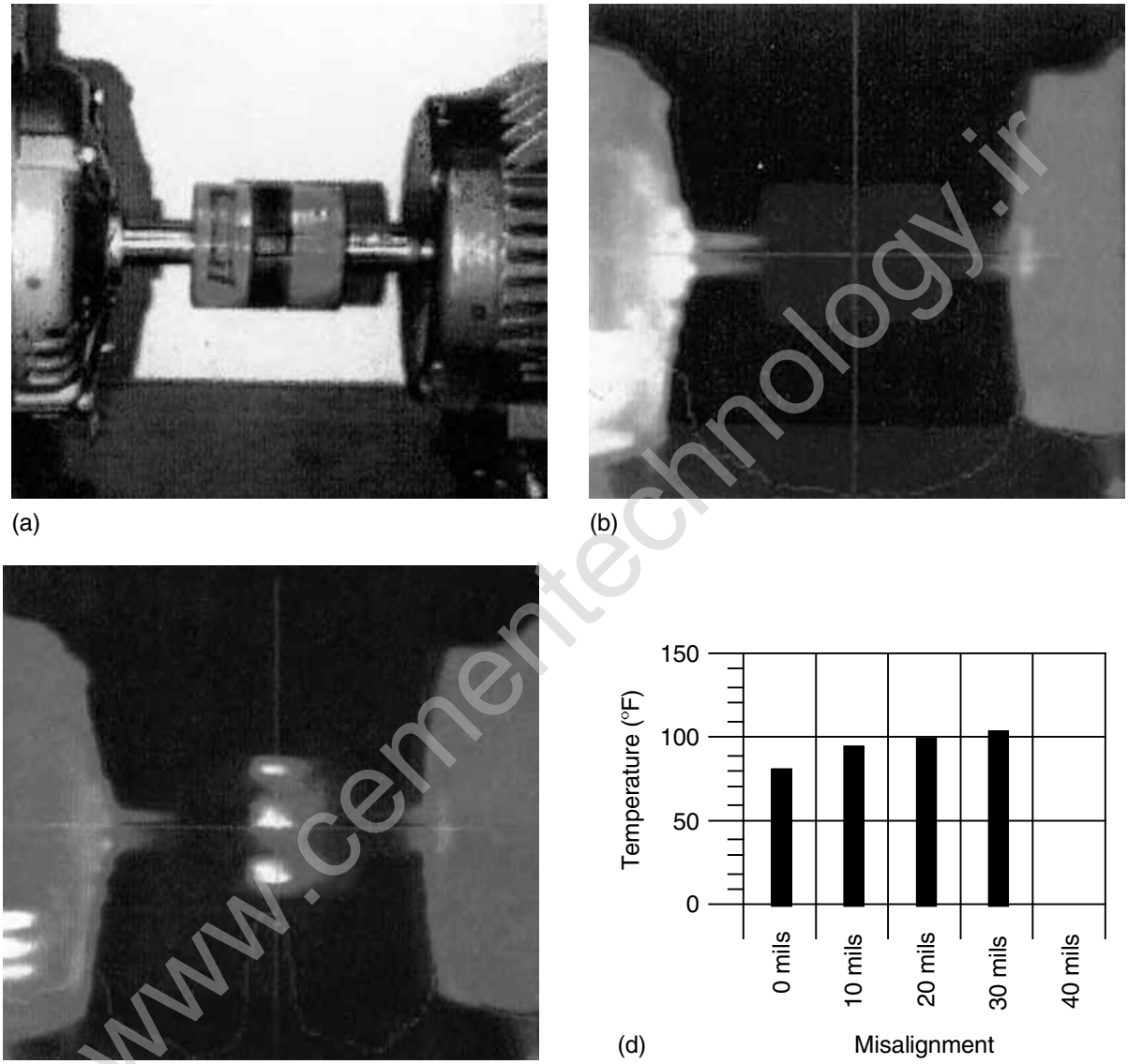


FIGURE 2.41 Observed temperature patterns on misaligned jaw-type coupling. (a) A photograph of the coupling, (b) an infrared image of the coupling running under good alignment conditions, (c) an infrared image of the coupling running with the worst misalignment condition (d) temperature of coupling at each 10 mil misalignment condition. (Photos and data courtesy of Infraspection Institute, Shelburne, VT.)

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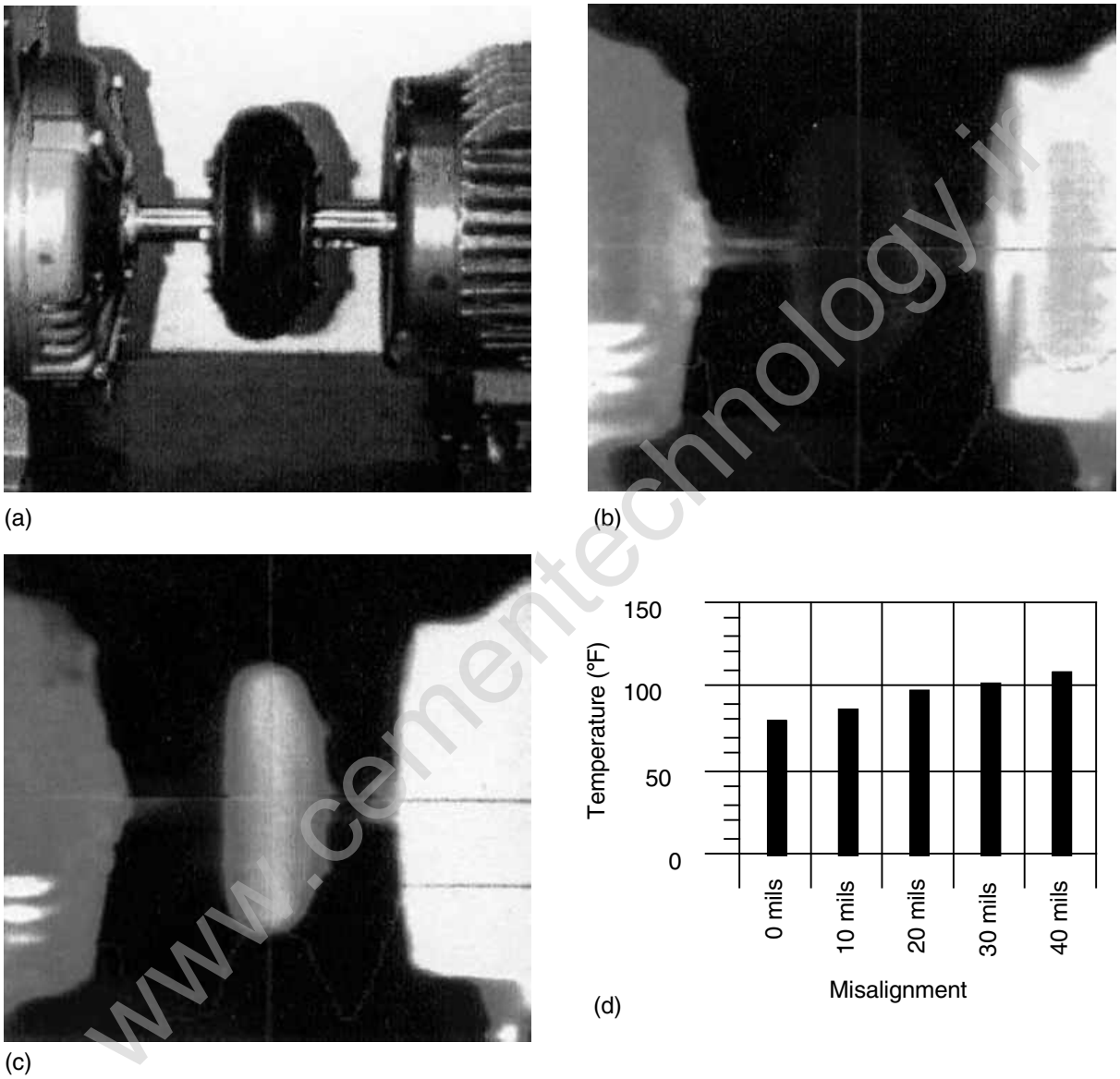


FIGURE 2.42 Observed temperature patterns on misaligned rubber tire-type coupling. Upper right photo shows infrared image of coupling running under good alignment conditions. Lower right photo shows coupling running under "worst case" misalignment condition indicated by rightmost bar on temperature graph. (Photos and data courtesy of Infraspection Institute, Shelburne, VT.)

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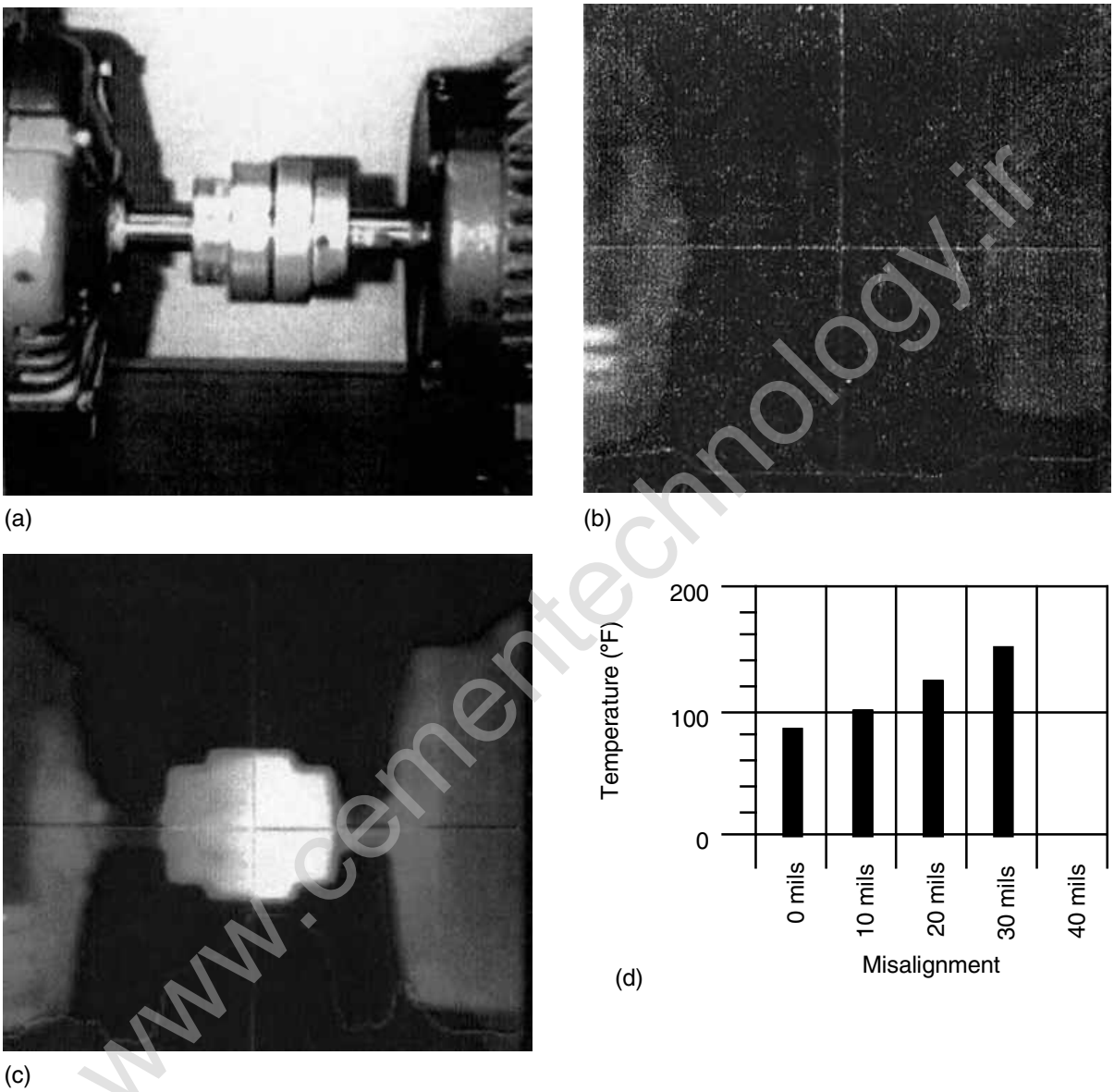


FIGURE 2.43 Observed temperature patterns on misaligned rubber insert type coupling. (a) A photograph of the coupling, (b) an infrared image of the coupling running under good alignment conditions, (c) an infrared image of the coupling running with the worst misalignment condition (d) temperature of coupling at each 10 mil misalignment condition. (Photos and data courtesy of Infraspection Institute, Shelburne, VT.)

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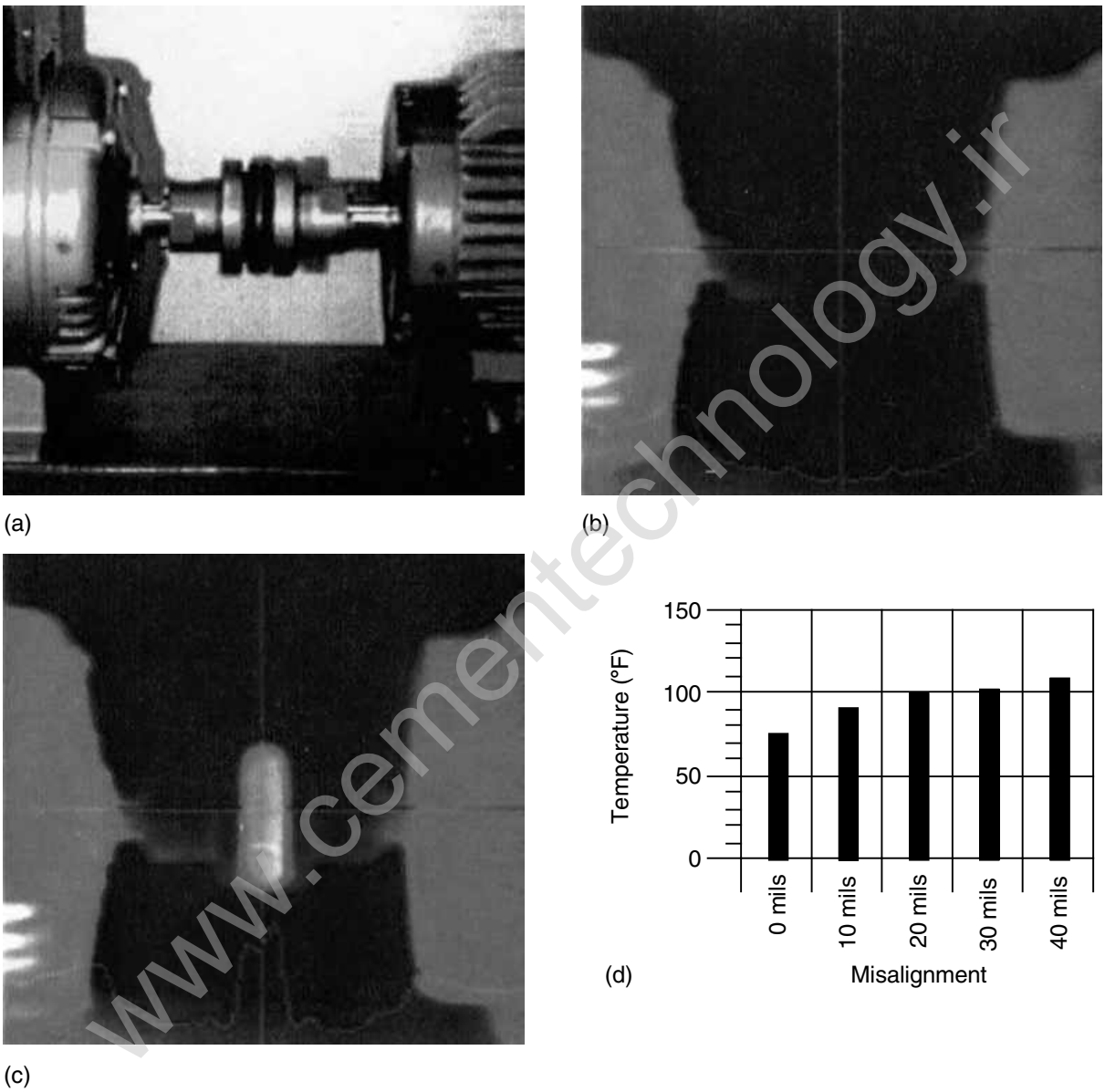


FIGURE 2.44 Observed temperature patterns on misaligned rubber "gear" type coupling. (a) A photograph of the coupling, (b) an infrared image of the coupling running under good alignment conditions, (c) an infrared image of the coupling running with the worst misalignment condition (d) temperature of coupling at each 10 mil misalignment condition. (Photos and data courtesy of Infraspection Institute, Shelburne, VT.)

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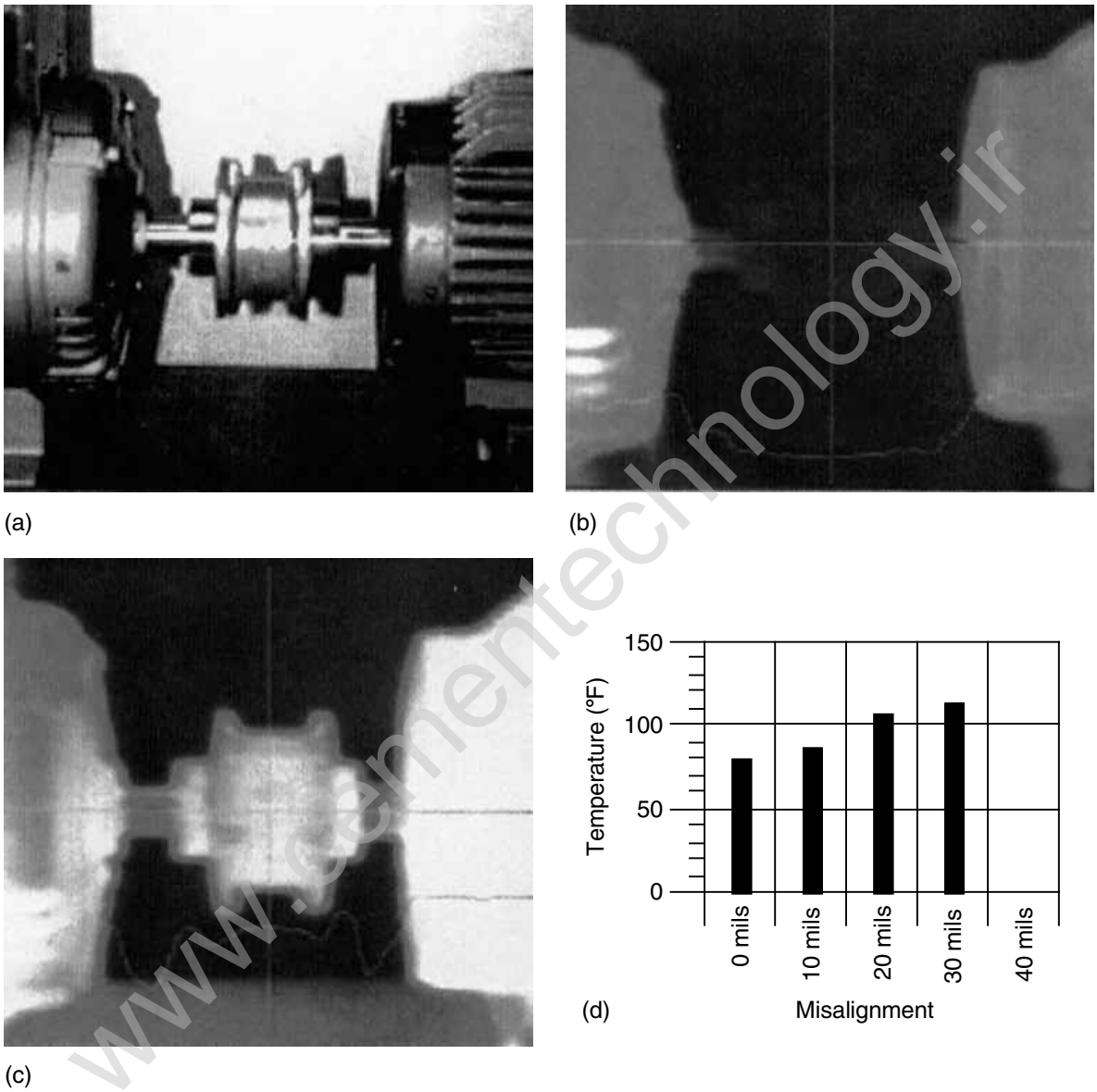


FIGURE 2.45 Observed temperature patterns on misaligned metal ribbon-type coupling. (a) A photograph of the coupling, (b) an infrared image of the coupling running under good alignment conditions, (c) an infrared image of the coupling running with the worst misalignment condition (d) temperature of coupling at each 10 mil misalignment condition. (Photos and data courtesy of Infraspection Institute, Shelburne, VT.)

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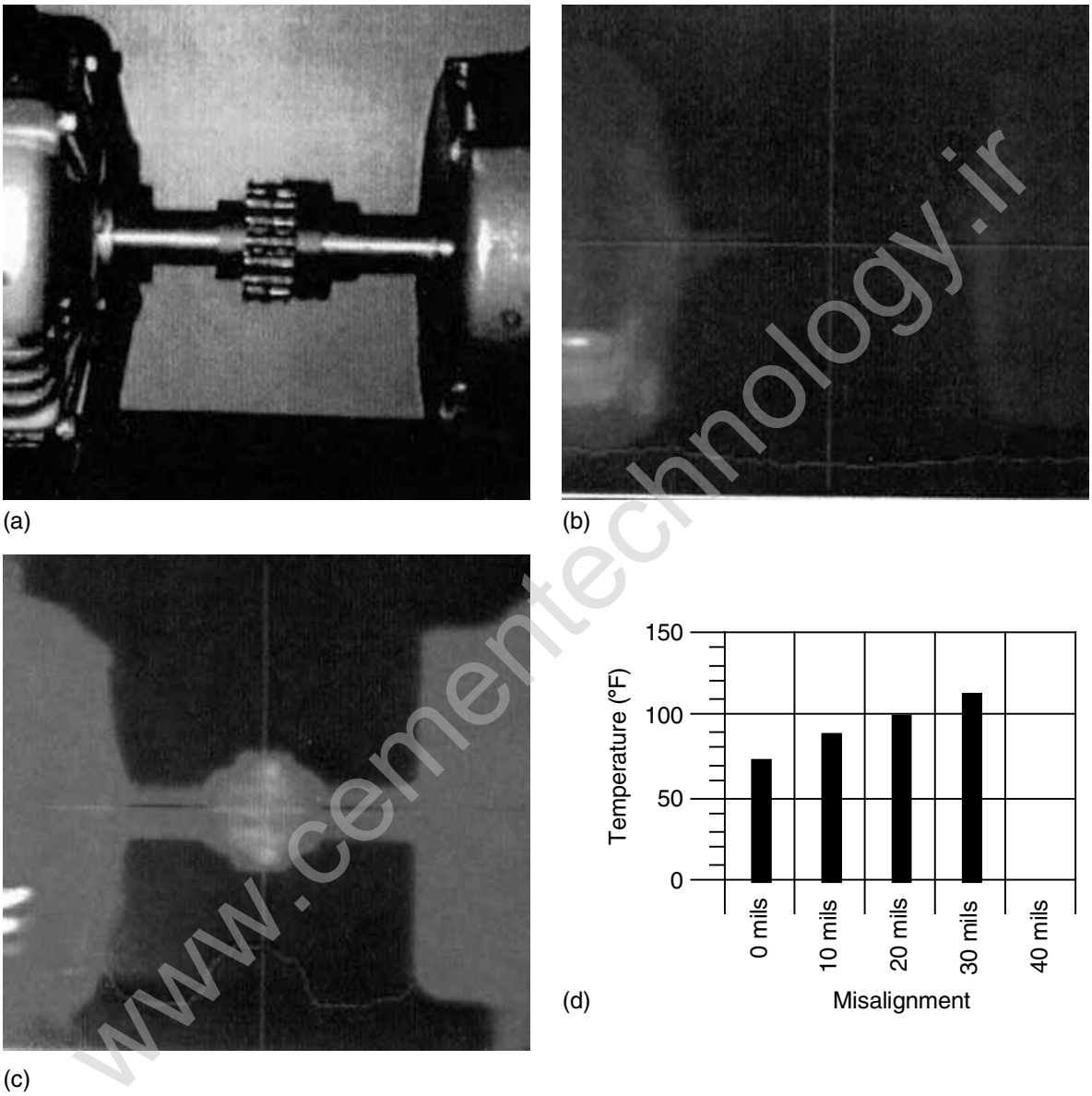


FIGURE 2.46 Observed temperature patterns on misaligned chain type coupling. (a) A photograph of the coupling, (b) an infrared image of the coupling running under good alignment conditions, (c) an infrared image of the coupling running with the worst misalignment condition (d) temperature of coupling at each 10 mil misalignment condition. (Photos and data courtesy of Infraspection Institute, Shelburne, VT.)

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Xu, M. and Marangoni, R.D., Vibration analysis of a motor—flexible coupling—rotor system subject to misalignment and unbalance, Part I: Theoretical model and analysis, *Journal of Sound and Vibration* (1994a), 176(5), 663–679.

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3 Foundations, Baseplates, Installation, and Piping Strain

Every rotating machinery drive system requires some kind of supporting structure to hold it in position. Imagine for a moment, the design concerns for machinery that is located near a river or a lake, on top of an underground aquifer, near a busy highway, in the middle of a swamp, operating on seagoing vessels, offshore oil and gas platforms, or on the 18th floor of an office complex. The foundations and support structures not only have to bear the weight of the machinery but also have to be designed to maintain a stable position if the machinery begins to vibrate. Frequently alignment problems can be traced back to design, installation, or deterioration problems with the foundation, base or soleplate, or the machine housings themselves. It is going to be not only difficult to obtain accurate alignment initially but also equally difficult to maintain satisfactory alignment over long periods of time if the machinery is sitting on unstable or improperly designed foundations and frames.

Not all, but a large percentage of rotating machinery sits on or is somehow attached to the ground. When selecting a site for rotating machinery, civil engineers must be concerned with the soil conditions and stability of the ground where the machinery is to be located. To a great extent, the Earth will act as a giant shock absorber for any motion that occurs in the machinery and also act as the main support for the equipment. What is the earthbound rotating machinery sitting on—bedrock or sand? It is also common to find rotating machinery in the upper floors of a building or on the roof. Is the frame attached to beams or columns and what isolates the frame from the building?

All types of rotating machinery will exhibit some level of vibration during its operation so design engineers need to be concerned about how much vibration (or noise) can or will be transmitted through the structure to the surrounding environment. Foundations, structures, and machine casings can be rigorously designed and checked utilizing computer-aided design and engineering techniques before fabrication ever begins. The field of structural dynamics and finite element analysis has provided the means to calculate structural mode shapes and system resonances of complex structures to insure that frequencies from the attached or adjacent machinery do not match the natural frequency of the structure itself. However this technology cannot easily remedy all the equipment installed before these analysis tools were available and many of us are saddled with equipment sitting on poorly designed or constructed bases that are cracked or warped or static piping strain that was not corrected during the installation or that has increased from the foundation settling over a period of time or from movement of the piping supports.

Over moderate to long periods of time soils, foundations, and structures will gradually shift due to a wide variety of factors. Temperature changes from season to season, compaction of soils underneath foundations, swelling of base soils from water or freezing are some of the more common causes of shifting to occur. It is unreasonable to assume that alignment conditions will not change over time and periodic alignment checks should be performed.

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It is important for the personnel who maintain rotating machinery to have a basic understanding of how machinery should be supported and what problems to look for in their foundations, baseplates, and frames to insure long-term alignment stability in their machinery.

In addition to the machinery to ground or structure interface, attention must also be directed to any physical attachments to the machinery such as piping, conduit, or ductwork. It is desirable to insure that these attachments produce the minimum amount of force on the machinery to also insure good stability. This chapter will hopefully provide the reader with the basic foundation design principles and some techniques to check equipment in the field to determine if problems exist with the foundation and frame, or the interface between the machinery and the foundation, or piping and conduit attached to the machine itself.

3.1 VARYING COMPOSITION OF EARTH'S SURFACE LAYER

The best place to start this discussion is at the bottom of things. All of us realize that there is a major difference in stability as we walk along a sandy beach and then step onto a large rock outcropping. Different soil conditions produce different amounts of firmness. Since rotating machinery could potentially be placed anywhere on the planet, the soil conditions at that location need to be examined to determine the stability of the ground. For new installations or where foundations have shifted radically, it may be a good idea to have boring tests conducted on soils where rotating machinery foundations will be installed. Table 3.1 shows safe bearing load ranges of typical soils. The recommended maximum soil load from a combination of both static and dynamic forces from the foundation and attached machinery should not exceed 75% of the allowable soil bearing capacity as shown in Table 3.1.

3.2 HOW DO WE HOLD THIS EQUIPMENT IN PLACE?

I suppose someone has attempted to sit a motor and a pump on the ground, connected by the shafts together with a coupling, and started the drive system up without bolting anything down. My guess is that they quickly discovered that the machines started moving around a little bit after start up, then began moving around a lot, and finally disengaged from each other hopefully without sustaining any damage to either of the machines. Maybe they tried it again and quite likely had the same results. I am sure they finally came to the conclusion that this

TABLE 3.1
Soil Composition

	Bearing Capacities of Soils: Safe Bearing Capacity	
Type of Soil	**t/ft^2**	**MPa**
Hard rock (e.g., granite, trap, etc.)	25–100	2.4–9.56
Shale and other medium rock (blasting for removal)	10–15	0.96–1.43
Hardpan, cemented sand and gravel, soft rock (difficult to chisel or pick)	5–10	0.48–0.96
Compact sand and gravel, hard clay (chiseling required for removal)	4–5	0.38–0.58
Loose medium and coarse sand medium clay (removal by shovel)	2–4	0.20–0.38
Fine loose sand	1–2	0.10–0.20

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was not going to work for long periods of time and decided to "hold the machines" in their starting position somehow. How are we going to do this exactly? What should we attach them to? How about some wood? No, better yet, something like metal or rock, something that is strong.

Our rotating equipment needs to be attached to something that will hopefully hold it in a stable position for long periods of time. I have seen just about every possible configuration you can imagine. Even the scenario mentioned above. The most successful installations require that the machinery be attached to a stable platform that enables us to detach one or more of the machines from its platform in the event that we want to work on it at another location. Classically we attach and detach our equipment with threaded joints (i.e., bolts and nuts). You could, I suppose, glue or weld the machines to their platform, and it would just be a little more difficult to detach them later on.

The devices that we have successfully attached our machinery to are baseplates, soleplates, or frames. There are advantages and disadvantages to each choice. The baseplates, soleplates, or frames are then attached to a larger structure, like a building, ship, aircraft and automotive chassis, or Earth. There are many inventive ways of attaching rotating machinery to transportation mechanisms (e.g., boats, motorcycles, airplanes), and design engineers are still coming up with better solutions for these types of machinery-to-structure interface systems. Our discussion here will concentrate on industrial machinery.

The vast majority of rotating machinery is either held in position by a rigid foundation (monolithic), attached to a concrete floor, installed on an inertia block, or held in position on a frame. There are advantages and disadvantages to each design. There are also good ways and poor ways to design and install each of these methods to keep our machinery aligned and prevent them from bouncing all over the place when they are running. In summary, machines are attached to intermediary supports (i.e., baseplates, soleplates, and frames) that are then attached to structures (i.e., buildings, floors, foundations). Figure 3.1 shows a typical rigid



FIGURE 3.1 Rigid foundation for induced draft fan.

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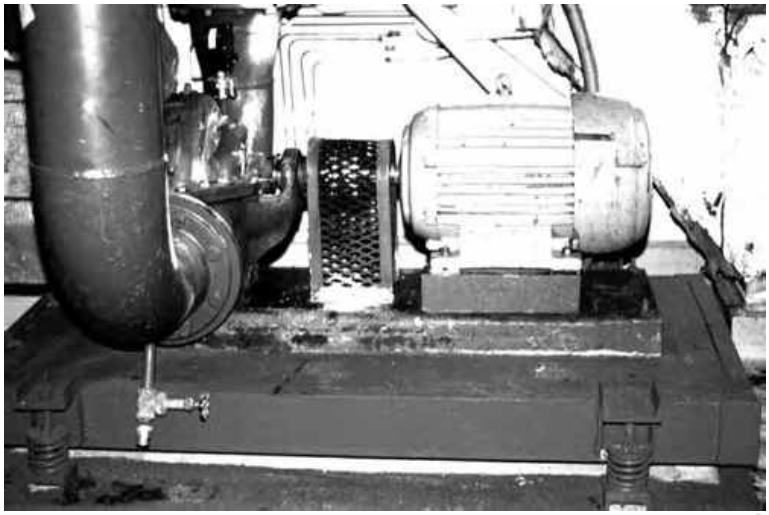


FIGURE 3.2 Spring isolated inertia block with motor and pump.

foundation design, Figure 3.2 shows a typica inertial block (aka floating) design, and Figure 3.3 shows a typical frame design.

3.2.1 Baseplates

Baseplates are typically either cast or fabricated as shown in Figure 3.4 and Figure 3.5. A fabricated baseplate is made using structural steel such as I-beams, channel iron, angle, structural tubing, or plate, cutting it into sections, and then welding the sections together. It is not uncommon to replace structural steel with solid plate to increase the stiffness of the base similar to Figure 3.6.

3.2.1.1 Advantages

1. Most commonly used design for industrial rotating machinery
2. Provides excellent attachment to concrete foundations and inertia blocks assuming the anchor bolts were installed properly and that the grout provides good bonding
3. Can be flipped upside down and grout poured into the cavity before final installation

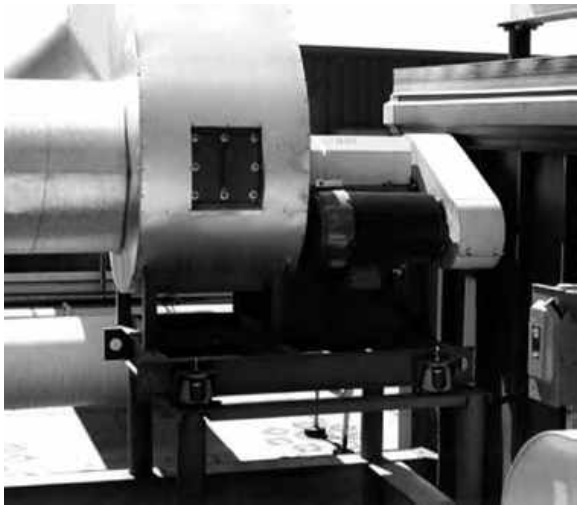


FIGURE 3.3 Frame supporting a belt drive fan.

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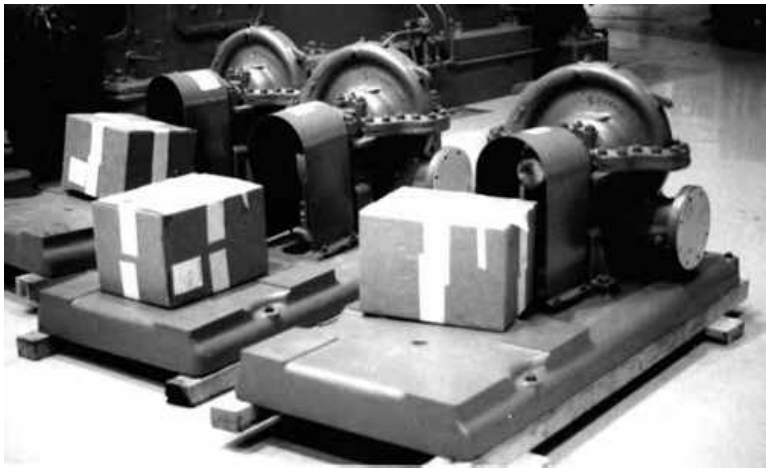


FIGURE 3.4 Cast baseplate.



FIGURE 3.5 Fabricated baseplate.



FIGURE 3.6 Weak structural steel was replaced with solid plate on this baseplate.

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4. Machinery can be placed onto the baseplate prior to installation and roughly aligned in the lateral and axial directions to insure that the foot bolt locations are drilled and tapped accurately to hopefully prevent a bolt bound condition or incorrect shaft end to shaft end spacing
5. Equipment mounting surfaces can be machined flat, parallel, and coplanar prior to installation
6. Some designs include permanent or removable jackscrews for positioning the machinery in the lateral and axial directions

3.2.1.2 Disadvantages

1. Usually more expensive than using soleplates or frames
2. Equipment mounting surfaces are frequently found not to be flat, parallel, and coplanar prior to installation
3. Difficult to pour grout so it bonds to at least 80% of the underside of the baseplate
4. Possibility of thermally distorting baseplate using epoxy grouts if pour is thicker than 4 in.
5. Frequently installed with no grout

3.2.2 Soleplates

Soleplates are effective machinery-mounting surfaces that are not physically connected together. Figure 3.7 shows a soleplate being prepared for grouting on a medium-sized fan housing. They are typically fabricated from carbon steel and there are usually two or more soleplates required per concrete foundation or inertia block. Correct installation is more



FIGURE 3.7 Soleplate being prepared for grouting.

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FIGURE 3.8 Optical jig transit being used to level a soleplate.

tedious than baseplates due to the care required to insure that the individual soleplates are in level and in the same plane. On larger machinery where the soleplates can be six or more feet apart, using machinist levels is not going to work effectively and either optical or laser alignment tooling is recommended to get the plates level and in the same plane. Ideally the soleplates should be level to 1 mils/ft (1 mils = 0.001 in.), and there should not be a deviation of more than 5 mils at any point on all soleplates from being coplanar. Figure 3.8 shows an optical jig transit used to level and plane the soleplates shown in Figure 3.7.

3.2.2.1 Advantages

1. Works best for large machinery where a contiguous baseplate would be too heavy or cumbersome
2. Somewhat easier to properly grout to concrete foundation or inertia block

3.2.2.2 Disadvantages

1. More difficult to insure that surfaces of soleplates are flat, coplanar, and parallel

3.2.3 Frames

Frames are typically constructed from structural steel such as channel iron, I-beams, angle iron, or structural tubing and are often custom made for each application. The frames are then attached to a larger structure such as a building frame, floor, or concrete foundation. They are classically not as rigid as equipment mounted to baseplates or soleplates and frequently will exhibit higher level of vibration amplitude. However, it is common to provide vibration isolation from the structure or floor with springs or damping isolators (e.g., rubber

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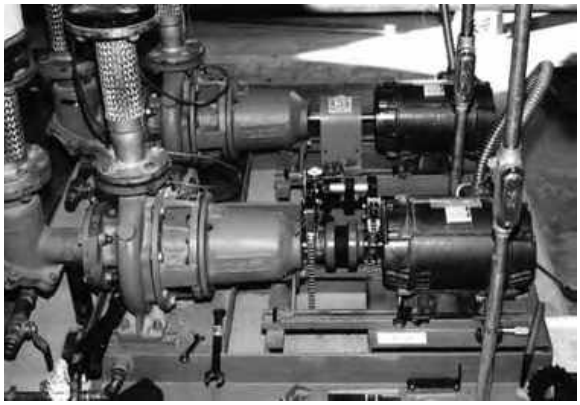


FIGURE 3.9 Motors and pumps sitting on structural steel frames. The unistrut used for the motors is not recommended.

mounting isolators). Due to the fact that most frames are welded construction, the surfaces that the machinery attaches to are often not coplanar or parallel to each other. Figure 3.9 through Figure 3.11 show a variety of rotating machinery mounted on frames.

3.2.3.1 Advantages

1. Most practical design for machinery that cannot be attached to Earth or building structures
2. Used in situations where excessive floor loads are exceeded with concrete construction
3. Easier to fabricate and install than rigid foundations or inertia blocks

3.2.3.2 Disadvantages

1. Due to the low frame-to-machinery weight ratio, vibration levels are typically higher than equipment located on rigid foundations or inertia blocks
2. Subject to more rapid deterioration from environment
3. Difficult to insure flatness of machinery-mounting surfaces during construction
4. Excitation of structural natural frequency more prevalent with this design

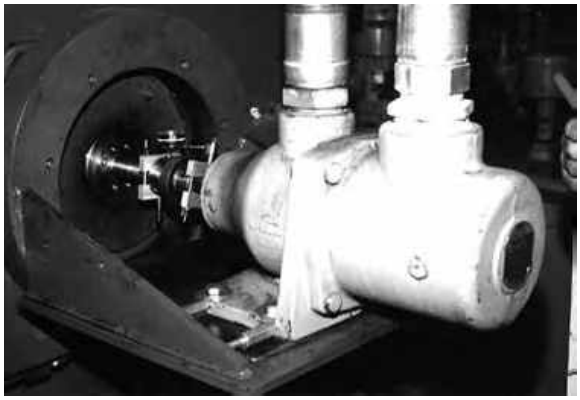


FIGURE 3.10 Main lube oil pump coupled to outboard end of motor sitting on a fabricated frame bolted to the motor's end bell.

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FIGURE 3.11 Series of water bearings held in place on a dredge frame.

3.2.4 Monolithic Rigid Foundations

Rigid foundations are typically found at the ground level. The basic design of a rigid foundation is shown in Figure 3.12. Their sole purpose is to provide an extremely stable platform for the rotating machinery with no intention of supporting any other object but the machinery that is placed on it except perhaps piping, ductwork, or conduit that attaches to the machines in the drive system. Effectively, the rigid foundation consists of a poured reinforced concrete block with anchor bolts that have been imbedded in the concrete.

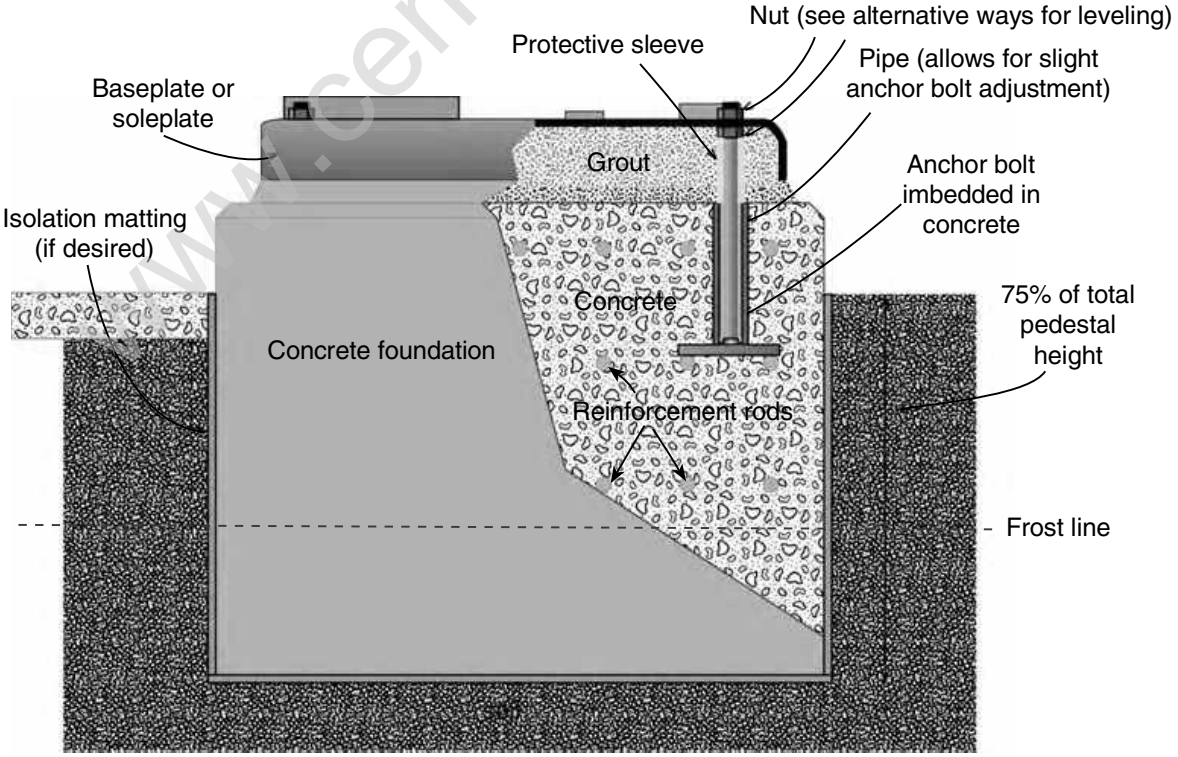


FIGURE 3.12 (**See color insert following page 322.**) Section view of a typical rigid foundation.

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Reinforcement rods should be spaced no more than 12 in. apart, using a minimum rod size of 1/2 in. (12.7 mm). The concrete should be rated at a compressive strength of 4000 psi for 28 d. Once the concrete has set to at least 50% cure (typically 7 d for most concrete) the baseplate or soleplates are set into a level and coplanar condition slightly above the top of the concrete (usually 1–2 in.). The baseplate is then grouted to the concrete foundation as illustrated in Figure 3.12.

Here are some basic design "rules of thumb" for concrete foundations:

1. Whenever possible, mount every machine in the drive system on the same foundation.
2. The mass of the foundation should be three to five times the mass of centrifugal machinery it supports and five to eight times the mass of reciprocating machinery it supports.
3. The width of the foundation should be 1.5 times the distance from ground level to the centerline of rotation.
4. Use baseplates or soleplates to attach the machinery to the concrete foundation.

3.2.4.1 Advantages

1. Provides a stable platform to attach rotating machinery using the surrounding soil to absorb motion or vibration
2. Ability to design foundation mass to effectively absorb any vibration from attached machinery and isolate residual motion by segregating the foundation block with vibration absorptive material preventing transmission of vibration to surrounding area

3.2.4.2 Disadvantages

1. If located outdoors, eventually degradation of foundation imminent especially if located in geographical area where climatic conditions change radically throughout the year
2. For machinery with attached, unsupported piping or ductwork, extreme forces from improper fits can occur causing damage to machinery
3. Potential settling of foundation causing instability and potential transmission of forces from attached piping

3.2.4.3 Tips for Designing Good Foundations

1. Insure that the natural frequency of the foundation–structure–soil system does not match any running machinery frequencies or harmonics (such as 0.43×, 1×, 2×, 3×, 4×, etc.) with the highest priority being placed on staying +20% away from the operating speed of the machinery sitting on the foundation being considered. Also watch for potential problems where running speeds of any machinery nearby the proposed foundation might match the natural frequency of the system being installed.
2. In case the calculated natural frequency of the structure does not match the actual structure when built, design in some provisions for "tuning" the structure after erection has been completed such as extension of the mat, boots around vertical support columns, attachments to adjacent foundations, etc.
3. Minimize the height of the centerline of rotation from the baseplate.
4. Rotating equipment that will experience large amounts of thermal or dynamic movement from off-line to running conditions should be spaced far enough apart to insure that the maximum allowable misalignment tolerance is not exceeded when the shafts

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are located in the off-line position. Refer to Chapter 16 for more details on off-line to running machinery movement.
5. Design the foundation and structure to provide proper clearances for piping and maintenance work to be done on the machinery, and provisions for alignment of the machine elements.
6. Install removable jackscrew devices on the baseplate for moving (i.e., aligning) equipment in all three directions: vertically, laterally, and axially. If jackscrews will not be used, provide sufficient clearance between baseplate and rotating equipment for insertion of hydraulic jacks for lifting equipment during shim installation or removal.
7. Provide vibration joints or air gaps between the machinery foundation and the surrounding building structure to prevent transmission of vibration.
8. If possible, provide centrally located, fixed anchor points at both the inboard and outboard ends on each baseplate in a drive train to allow for lateral thermal plate expansion. Insure there is sufficient clearance on the casing foot bolt holes to allow for this expansion to occur without binding against the foot bolts themselves.

3.2.4.4 Tips for Installing Foundations and Rotating Machinery

1. Select a contractor having experience in installing rotating machinery baseplates and foundations or provide any necessary information to the contractor on compaction of base soils, amount and design of steel reinforcement, preparing concrete joints during construction, grouting methods, etc.
2. If the concrete for the entire foundation is not poured all at once, be sure to chip away the top 1/4 in. to 1/2 in. of concrete, remove debris, keep wet for several hours (or days if possible), allow surface to dry and immediately apply cement paste before continuing with an application of mortar (1–6 in.) and then the remainder of the concrete. If not done, the existing concrete may extract the water from the freshly poured concrete too quickly and proper hydration (curing) of the new concrete will not occur.
3. Use concrete vibrators to eliminate air pockets from forming during the pouring process but do not over vibrate, causing the larger concrete particles to settle toward the bottom of the pour.
4. Check for baseplate distortion prior to installing the baseplate. Optical alignment or laser tooling equipment can be used to measure this. Mounting pads should be machined flat and not exceed 2 mils difference across each individual pad (i.e., machinery foot contact point). If there is more than one pad that each individual machine will come into contact with, insure that those pads are coplanar within 5 mils. Insure that the contact points for each machine are parallel to the contact points for every other machine on that baseplate within 10 mils/ft. If the baseplate is slightly distorted it may be possible to stress relieve by oven baking or vibratory shakers. If the distortion is excessive, the contact surfaces may have to be machined flat, coplanar, and parallel.
5. Sandblast the underside of the baseplate. If sandblasting is unreasonable, grind at least 90% of the surface to bare metal. If cement-based grout is going to be used, coat with inorganic zinc silicate primer as per coating manufacturers specifications to prevent corrosion and provide good bonding to cement-based grout. The primer should not exceed 5 mils in coating thickness. If epoxy-based grout is going to be used, do not coat with primer and grout within 48 h of sandblasting to insure that excessive oxidation does not occur to the sandblasted surface.
6. Insure that the baseplate has leveling jackscrews at each of the anchor bolt locations. Try not to use wedges to level the baseplate. If jackscrews were not provided, weld 3/4 in. or 1 in. fine threaded nuts to the outside perimeter of the baseplate near the anchor bolts

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to use with jackscrews for precise leveling. Optical or laser alignment equipment should be used to check levelness particularly for medium and large machinery drive systems. A machinist level could be used on smaller baseplates but additional precautions need to be taken to insure that all of the mounting points for each machine are coplanar (i.e., it is possible to have two level surfaces not in the same plane).

7. For large baseplates with two or more bulkheads, grout one bulkhead section at a time. Apply grout through a 4–6 in. diameter hole centrally located in each section. Provide at least 1 in. diameter vent holes near the corners of each section. Allow a minimum of 48 h cure time before setting rotating equipment onto base.
8. Protect the foundation from any radiant heat generated from machinery, steam, or hot process piping by insulation or heat shields where possible.

3.2.5 Baseplates Attached to Concrete Floors

Similar in design to the monolithic rigid foundation, foundations attached to concrete floors require a slightly different approach in their design and installation. To a certain extent, the concrete floor now acts as the "foundation." There are three different approaches to attaching machinery to concrete floors:

1. A baseplate that is grouted by traditional methods to a concrete floor or a raised concrete pad as shown in Figure 3.13.
2. A baseplate that is pregrouted prior to installation and is then bonded to a concrete floor or a raised concrete pad as shown in Figure 3.14.
3. A solid metal baseplate is bonded to a concrete floor or a raised concrete pad that is also bonded to the concrete floor as shown in Figure 3.15.

If the machinery is going to be mounted to a floor at ground level, holes (usually 4–6 in. in diameter) should be cored through the concrete floor for the anchor bolts. The top surface of the floor should then be chipped away (i.e., scarified), a form built, reinforcement rods set in place, anchor bolts positioned, and a raised concrete block poured.

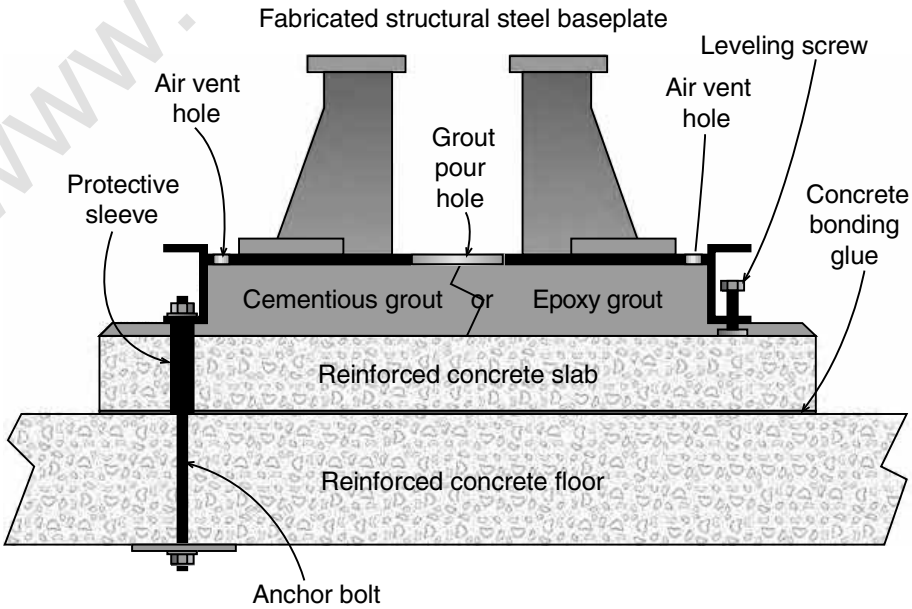


FIGURE 3.13 A baseplate that is grouted to a concrete floor or pad using traditional grouting methods.

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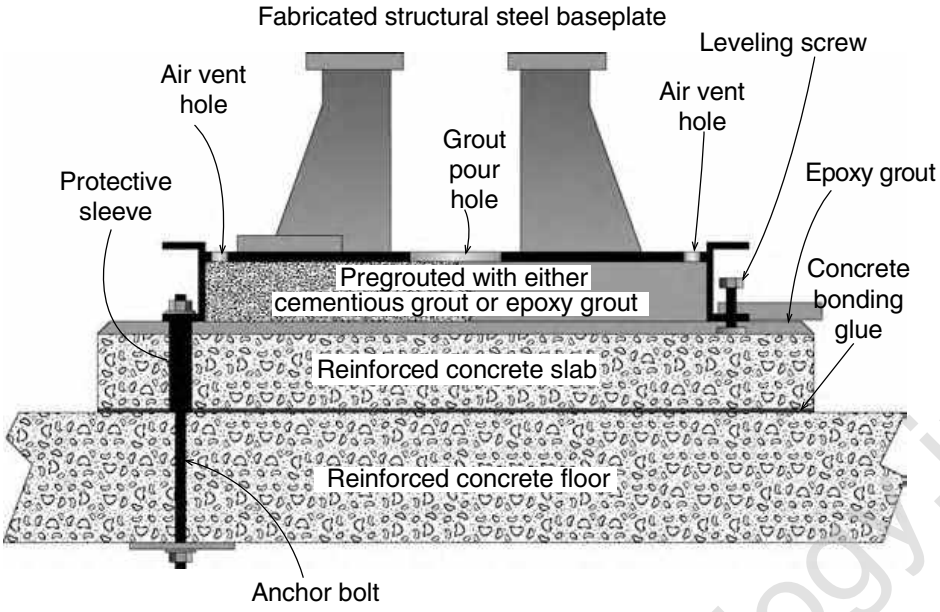


FIGURE 3.14 A pregrouted baseplate bonded to a concrete floor or pad.

3.2.5.1 Advantages

1. Provides a stable platform to attach rotating machinery using a large concrete floor mass
2. Installation costs slightly less than cutting out a section of the floor, digging a hole, building a frame, pouring concrete for a monolithic foundation

3.2.5.2 Disadvantages

1. If located outdoors, eventually degradation of foundation imminent especially if located in geographical area where climatic conditions change radically throughout the year

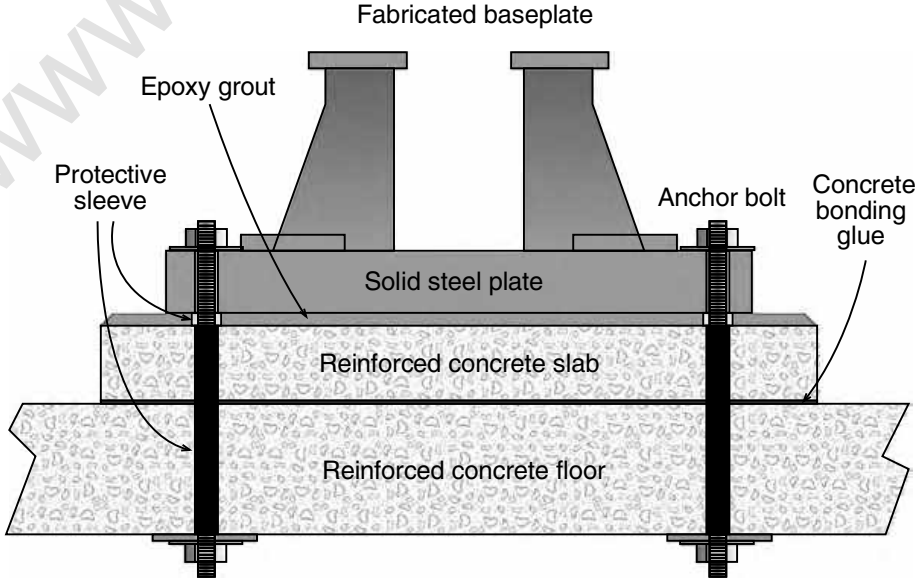


FIGURE 3.15 Solid metal baseplate bonded to a concrete floor or pad.

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2. Possibility of anchor bolts pulling out, loosening, or breaking if proper precautions are not taken during the installation of the anchor bolts
3. Possibility of baseplate (or soleplates) and the concrete slab loosening and degrading rapidly if care is not taken to properly bond the baseplate to the concrete slab to the concrete floor
4. For machinery with attached, unsupported piping or ductwork, extreme forces from improper fits can occur causing damage to the machinery
5. Limited ability to isolate any vibration in the machinery from the surrounding environment
6. Possibility of absorbing vibration from other machinery in the immediate vicinity

3.2.6 Anchor Bolts

Figure 3.16 shows various anchor bolt designs. Anchor bolts are imbedded in the concrete and serve as the device that secures the baseplate or soleplates to the concrete mass. The best designs incorporate a sleeve that allows the bolt to stretch properly when tightened and also allows for some minor positional changes if the anchor bolts do not index to the holes in the baseplate or soleplate as shown in the bottom two diagrams in Figure 3.16. The anchor bolts should be at least four anchor bolt diameters from the outside edge of the concrete, should be of sufficient size and strength (ASTM A36 or ASTM A575-M1020), the anchor bolt should be able to resist chemical attack or oxidation, the washer should conform to ANSI B18.22.1, lock washer should not be used, and nuts should be heavy hex, full size, and conform to ANSI B18.2.2.

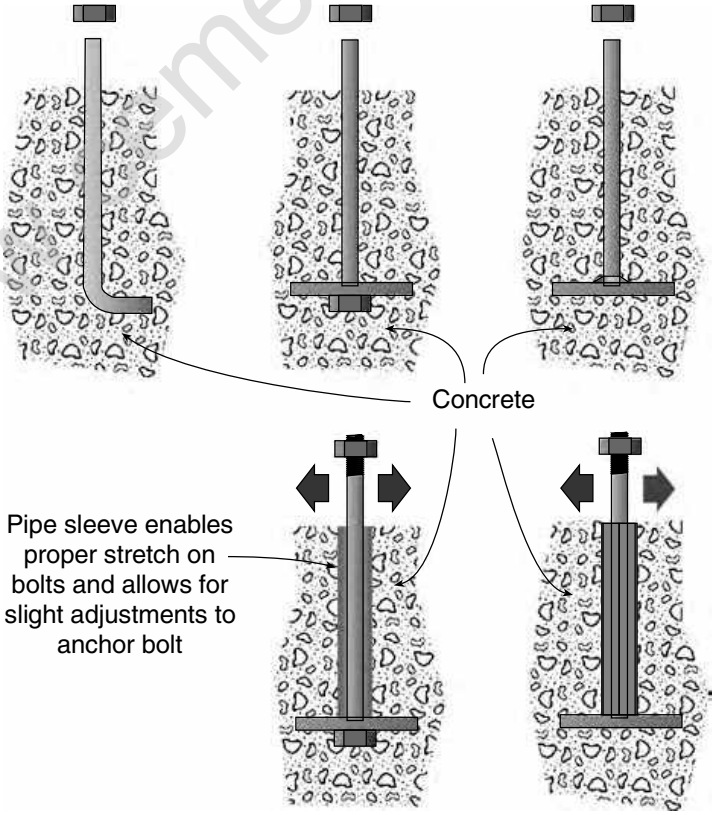


FIGURE 3.16 Various anchor bolt designs. (Courtesy of Unisorb, Jackson, Michigan. With permission.)

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If the machinery is going to be mounted on a floor above ground level where there is access from underneath the floor surface, holes should be drilled completely through the floor for the anchor bolts. Whether the machinery is going to be set onto a monolithic foundation at ground level, or onto a floor, properly positioning the anchor bolts is of extreme importance. To insure that the anchor bolts maintain their desired position, it is a good idea to set the baseplate onto a wooden template, mark where the anchor bolts will be, drill holes into the template, and then place the template on top of the wooden form to assist in correctly positioning and holding the anchor bolts in position when the concrete is poured.

3.2.7 Inertia Blocks

Also known as concrete "slabs," machinery "pallets," or equipment "skids," these foundations are typically constructed from structural steel, such as channel iron, I-beams, angle iron, or structural tubing, and concrete is poured into the interior of the structural skeleton (Figure 3.17). They can be rigidly attached to the building structure or floor or more commonly, be isolated from the structure or floor with springs or damping isolators (e.g., rubber mounting isolators). The inertia block could hold a single drive system or several drive systems. The usual purpose of this design is to isolate any vibration emanating from the machinery to the floor or building where it is attached.

3.2.7.1 Advantages

1. If concrete slab and baseplate act as a single unit with sufficient stiffness, this design provides a stable platform to attach rotating machinery, allowing the whole assembly to move in the event outside forces such as piping strain are bearing on unit
2. Ability to somewhat isolate any vibration from attached machinery to surrounding structure or other machinery in nearby area

3.2.7.2 Disadvantages

1. Slightly more difficult to construct, install, and maintain than rigid foundations
2. If excessive amount of vibration exists on machinery for prolonged periods of time, potential damage may occur to the machinery or attached piping



FIGURE 3.17 Inertia block for motor—fan drive. (Courtesy of Unisorb, Jackson, Michigan. With permission.)

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3. Potential degradation of support springs or isolators
4. Frequently more difficult to align machinery and keep aligned for long periods

3.2.8 Cement, Concrete, and Grout Basics

Since rigid foundations and inertia block design incorporate concrete or other types of pourable liquid to solid media, it is important to have a rudimentary understanding of these basic building materials. Concrete is typically a mixture of inert materials and cement. Grout can be cement based or epoxy based. Cement-based grout is typically a mixture of sand and cement. Epoxy-based grout can be pure epoxy consisting of a resin and a hardener (curing agent) or it can be mixed with inert material such as sand, steel shot (small round steel balls), fly ash, etc.

The inert materials in concrete are typically stone and sand but a wide variety of other materials can be used. The word "cement" is from the Latin verb "to cut" and originally referred to stone cuttings used in lime mortar. Lime consists of CaO (60%–67%), silica (SiO_2, 17%–25%), alumina (Al_2O_2, 3%–8%), and small amounts of iron oxide, magnesia, alkali oxides, and sulfuric anhydride. Cements may be naturally occurring (lime) or manufactured by crushing anhydrous calcium silicate–bearing rock into powder and then heated to around 1500°F. Manufactured cement is often called portland cement. There are six basic types of cement set forth in ASTM specification C150-61, shown in Table 3.2.

The cement, typically limestone, clay, or shale, acts as a glue to bond the inert materials together by mixing water with the cement and the aggregates. When the water migrates through the mixture and eventually evaporates, the cement and aggregates chemically bond together by hydration and hydrolysis to form a continuous block. The ratio of water and cement is critical to proper curing insuring that adequate strength is attained. Too much water will cause the paste to be too thin and will be weak when hardened. An U.S. engineer, Duff Abrams, developed the water cement ratio law in the 1920s and it is still widely used today. The proportion of a typical concrete mixture is shown in Table 3.3.

Compressive strengths of concrete can range from 1000 to 10,000 psi with a density of around 150 lb/ft^3. A compressive strength for concrete typically used in foundations for machinery is between 3000 and 5000 psi.

A "slump test" is used to determine the consistency of concrete. A standard slump cone is filled with concrete, smoothed off at the top of the cone, and then the cone is lifted vertically clearing the top of the concrete pour allowing the concrete in the cone to slump downward. The measured distance in inches from the original to the final level of the concrete mass is then observed. Concrete slump values for concrete used in machinery foundations should range from 3 to 5 in.

TABLE 3.2
Types of Cements

Type	Name	Description
1	Normal	General purpose
2	Modified	Low heat of hydration (curing) desired
3	High early strength	High strength required at an early age
4	Low heat of hydration	Typically used in dams to reduce cracking and shrinkage
5	Sulfate resistant	Used when exposed to soils with a high alkali content
6	Air entrained	Used when severe frost action present

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TABLE 3.3
Component Ratios of Low- and High-Strength Concrete

	Low Strength (%)	High Strength (%)
Water	15	20
Cement	7	14
Aggregates	78	66

Proper curing of the concrete requires that the water remain in the mixture for an acceptable period of time to insure that the chemical reaction of the cement is completed. Spraying small amounts of water on the concrete mass, laying wet burlap or plastics sheeting over top of the mass will insure that the rate of water loss is minimal. Pouring concrete in extremely hot temperatures (90°F–120°F) may cause the water to evaporate too quickly. Pouring concrete in extremely cold temperatures (below 32°F) will cause the water to expand when frozen and produce a very porous mixture with diminished compressive strength. Temperatures down to 25°F may be acceptable since curing of the concrete mixture is slightly exothermic but the mass should be insulated to entrain any heat during cure. The complex chemical reaction that occurs in the concrete takes place over several months of time. Concrete compressive strengths typically attain 70%–80% of their final value 6–8 d after the initial pour.

3.2.9 Reinforced Concrete

Concrete is ten times stronger in compression than in tension and must therefore be reinforced by imbedding steel reinforcement rods in the concrete mixture during the pour to prevent cracking when subjected to tensile loads. The amount of reinforcement in a foundation varies and should be taken into careful consideration during the design phase. Prestressed concrete is made by placing the reinforcement rods or cables in tension prior to pouring the concrete mixture. The amount of reinforcement rods in concrete foundations should be approximately 18% of the cross-sectional area of the concrete. Reinforcement rod sizes have been standardized and are approximately the diameter of the bar in 1/8 in. increments (i.e., a #4 bar is 1/2 in. in diameter). The maximum recommended rod spacing should not exceed 12 in. and there should be at least 2–3 in. of concrete covering the outer reinforcement rods to prevent corrosion of the rods.

3.2.10 Grouting

Grouting is typically used as the final binding medium between the machinery baseplate or soleplates and the concrete foundation. There are two basic classes of grout, cement based and epoxy based, as shown in Figure 3.18 and Figure 3.19. Proper mixing of the grout is essential to obtain the necessary strength. Be sure to carefully read the mixing instruction from the manufacturer when using any product.

3.2.10.1 Traditional Grouting Methods

Once the baseplate or soleplates are in position, a wooden form is then built on the upper surface of the concrete foundation to contain the grout as it is poured into the cavity. Grouting can be done in either a one-step or a two-step procedure. The most common method is to use a two-step grouting process as illustrated in Figure 3.20. One of the problems

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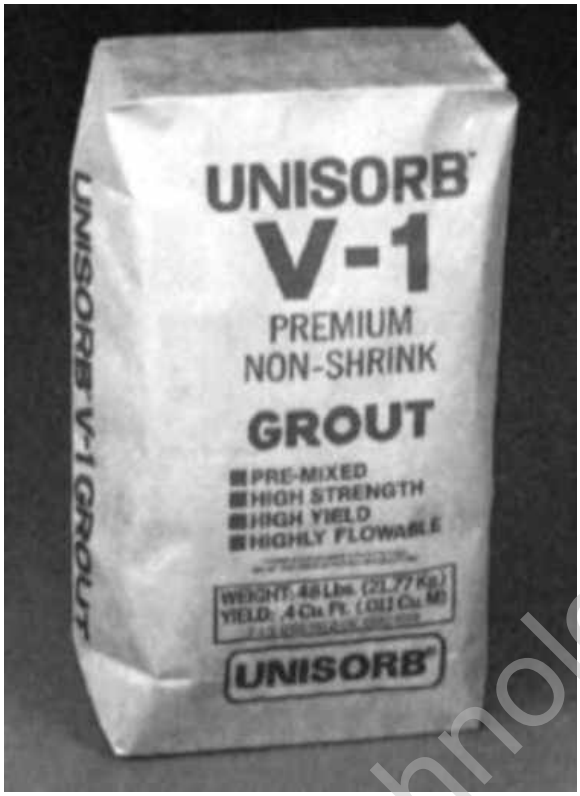


FIGURE 3.18 Cement-based grout. (Courtesy of Unisorb, Jackson, Michigan.)

with a two-step procedure is that a poor bond could occur between the mating surface of the first grout pour and the second grout pour if cement-based grouts are used. The single-pour method shown in Figure 3.21 requires a little more time and effort in building a form that will prevent the grout from oozing out under the baseplate but produces a continuous block of



FIGURE 3.19 Epoxy-based grout. (Courtesy of Unisorb, Jackson, Michigan.)

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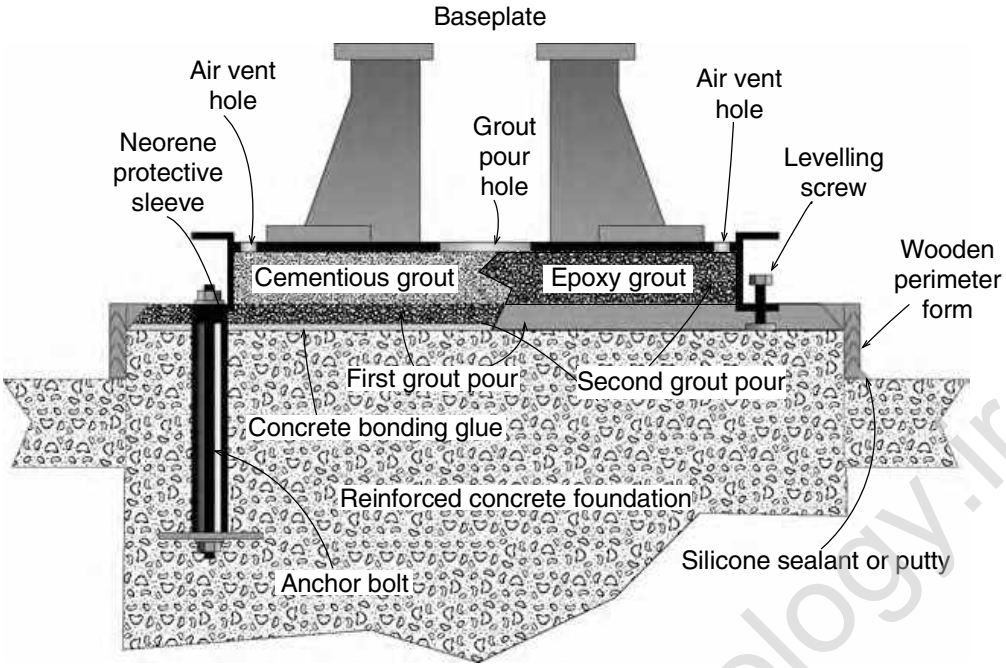


FIGURE 3.20 Two-step grouting process. (*Note:* If epoxy grout is used, liberally coat the surfaces of the wooden form with paste wax.)

grout. Cement-based grouts do not bond well to wood, which is a good thing when dismantling the form after the grout has hardened. Bear in mind that cement-based grouts do not bond too well to metal either, like the underside of the baseplate or soleplate. That is why it is suggested that the underside of the baseplate or soleplate be coated with a zinc-based primer, which will bond to the cement better than bare or rusty metal will. However, if the zinc-based primer applied too thickly, it can delaminate from the metal. The goal is to completely fill the

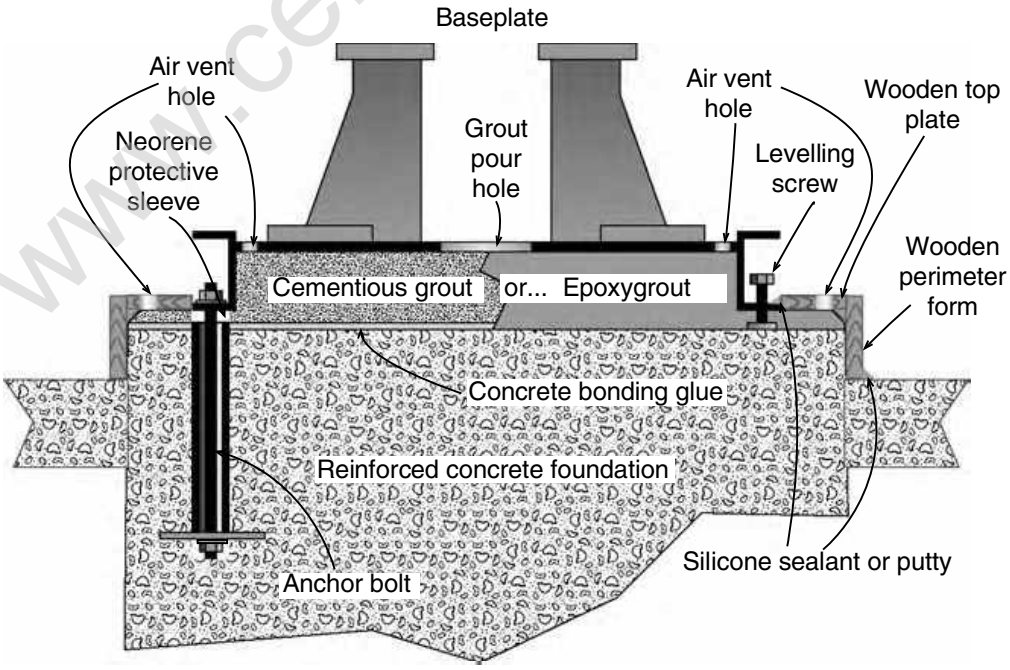


FIGURE 3.21 One-step grouting process. (*Note:* If epoxy grout is used, liberally coat the surfaces of the wooden form with paste wax.)

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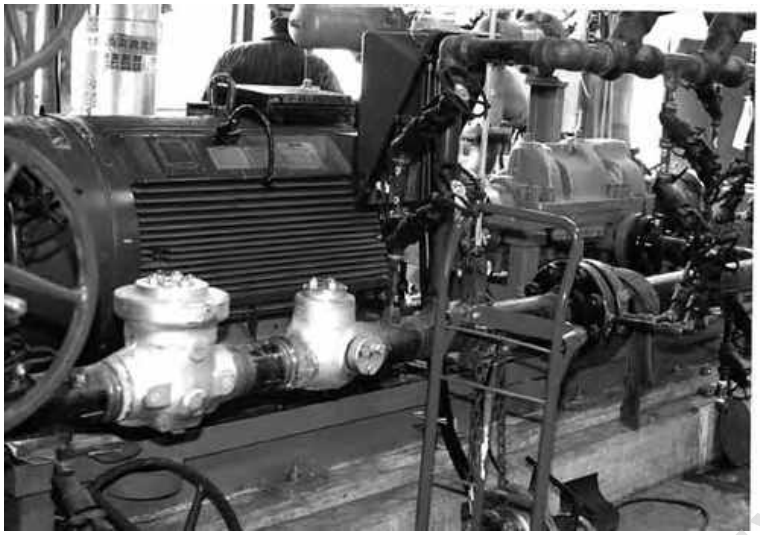


FIGURE 3.22 Cement-based grout used for motor and pump baseplate.

cavity between the baseplate or soleplate, with no air pockets, and solid bonding between the baseplate or soleplate and the concrete foundation. Figure 3.22 shows a motor and multistage pump, which was grouted with cement-based grout.

3.2.10.2 Suggested Grouting Procedure

1. *Planning:* Prepare a materials list of all the required components (grout, wood, bracing, pump, hose, water, mixing tools, vibrators, etc.). Plan an adequate amount of time to perform the job. Instruct the personnel on the task at hand. Are there enough vent holes in the base or frame for venting trapped air? Has the concrete foundation cured completely? Is the machinery base in the position you want it in and is it leveled and not warped? Will the base lift up when grout is pumped under it?
2. *Machinery base*, frame: and concrete preparationInsure that all contact surfaces on the undersides of the machinery base or frame are clean, rust free, and oil free. If possible, metal surfaces should be sand blasted and primed if you are using a cement-based grout. The concrete surface should also be clean, dust free, and oil free. If you are using a cement-based grout and do not plan on applying a concrete bonding glue to the top surface of the foundation, the concrete surface should be soaked with water for at least 24 h prior to grout placement to insure dry concrete does not extract the water in the grout mix at an excessive rate preventing proper cure. Prior to pouring grout, remove any puddles of water.
3. *Building the form*: Construct a form (typically wood) around the perimeter of the machinery base or soleplate to be grouted (Figure 3.23). Insure that there is adequate clearance between the machinery base or soleplate and the form to allow for placement of grout and access for pumping or pushing the grout completely under the base. Build the form with a number of pouring points around the perimeter. Insure that there are numerous vent holes of adequate size (at least 1 in. diameter) to discharge trapped air during the pour. If you are using epoxy-based grouts, insure that there are at least two to three coats of paste wax on all of the wooden form surfaces that will be exposed to the grout so the form does not permanently bond to the grout. Insure that the forms are adequately anchored. It is suggested that the baseplate or soleplates be protected with plastic sheeting or masking tape.
4. *Mix the grout*: Carefully follow the manufacturer's recommended mixing instructions.

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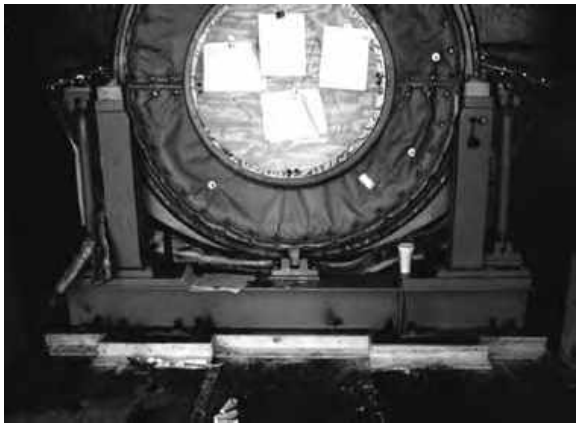


FIGURE 3.23 Wooden form for grouting soleplates.

5. *Pour the grout*: Insure that the grout is flowing under all areas of the form removing entrapped air at all points. Vibrators can be used with most of the cement-based grouts but not on epoxy grouts.
6. Allow adequate time for the grout to cure.
7. Remove forms.
8. Torque the anchor bolts to their final value after the grout has cured completely.
9. Tap the top surface of the baseplate with a hammer to detect any voids that may have occurred. Voids will have a distinctive hollow sound. There should be at least 80% adhesive surface contact. Voids larger than 3 in. in diameter should have epoxy injected into them. Define the perimeter of the void, drill, and tap a hole on one side of the void, and install a grease fitting. Drill a hole on the other side of the void perimeter to enable air to escape when injecting the void with an epoxy-filled grease gun or epoxy pump. Figure 3.24 shows epoxy grout being injected into voids on the baseplate.

3.2.11 Pregrouted and Solid Metal Baseplates

Getting the grout to bond to the underside of a cast or fabricated baseplate is quite a difficult task since we are fighting against gravity. Cement-based grouts do not want to bond to metal very well anyway and the chances of getting air pockets between the underside of the baseplate and the top of the grout is quite likely even if adequate vent holes are provided. Since gravity is the problem here, why not turn things around and make it the solution?



FIGURE 3.24 Injecting epoxy-based grout into voids in baseplate.

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Prior to installing the baseplate, just flip the baseplate upside down and pour the grout into it so good bonding is achieved. That is the concept behind pregrouted baseplates. This sounds simple to do but several precautions must be addressed.

The underside of the baseplate must be clean and oil free. The best way to do this is sand blasting the surface. Baseplates that have been designed to pour the grout from above will need to have both the grout pour holes and vent holes plugged. Dowels will have to be installed where the anchor bolts will go through the grout and the baseplate. With the baseplate flipped upside down, adequate supports need to be provided to prevent the baseplate from distorting due to the weight of the grout. A cement-based grout can be used but there is still the bonding issue to the underside of the baseplate. To get the cement-based grout to bond, an epoxy adhesive should be applied to the underside of the base prior to pouring the cement-based grout. Epoxy-based grouts could also be used and since flow ability is no longer an issue, an aggregate (e.g., pea gravel) can be added to the epoxy grout to reduce the cost and the shrinkage problem. If a cement-based grout is used, it should be kept wet and covered for at least 3 d to allow for proper curing. At that time, an epoxy resin can be applied to the surface of the grout to prevent contamination and water evaporation.

Once the grout has cured, the mounting surfaces on the baseplate can then be checked for parallelism and coplanar surfaces. In the event that the baseplate is distorted, the mounting surfaces can then be machined prior to installation. The baseplate is then set into its foundation and grouted into its final position using the one-step grouting procedure described in Figure 3.21.

An alternative to pregrouted baseplates is to use a solid metal baseplate. Although not many rotating machinery baseplates are constructed this way, the ones that have seem to work very well. For smaller machinery (i.e., 75 hp or smaller), 1.5 to 2 in. thick plates are recommended. For medium size machinery (i.e., 75 to 1000 hp), 3 to 4 in. thick plates are recommended. Indeed, the solid plate is heavier than a cast or fabricated baseplate (unless it is been pregrouted, then there is not much difference in weight). Compared to the expense of fabricating a baseplate from structural steel, sandblasting, painting, pregrouting, and then possibly having to machine the contact surfaces prior to installation, solid plates are frequently less labor intensive and cheaper to install. Instead of having jackscrews at each anchor bolt, sacrificial leveling devices could be installed at each anchor bolt as shown in Figure 3.25.

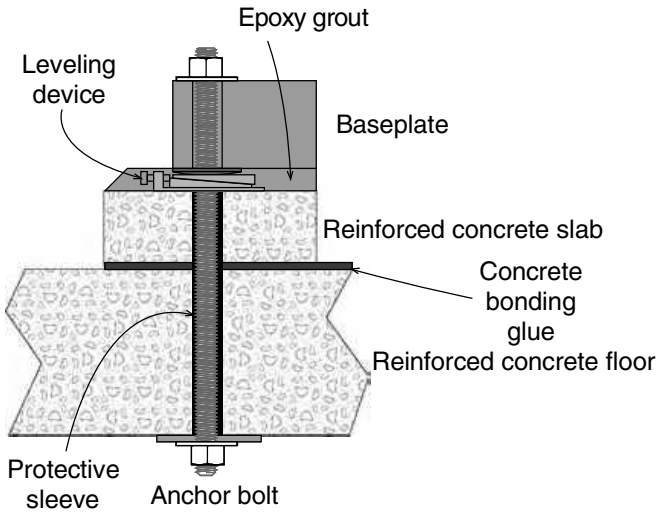


FIGURE 3.25 Section view of solid baseplate with leveling device at anchor bolts.

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FIGURE 3.26 Scarifying the top surface of a concrete foundation.

3.2.12 Case History of Installing a Baseplate Using Epoxy-Based Grout

Figure 3.26 shows the concrete foundation for a pump and turbine drive system. The top 1/8 in. to 1/4 in. of the concrete was chipped away (scarify) so the epoxy would bond to the aggregate, not the sand and portland cement that floated to the top during final floating of the concrete. Figure 3.27 shows a close-up of one of the anchor bolts.

Figure 3.28 shows the baseplate and the rigging used to lift and move the baseplate into position. The baseplate was flipped over on its side and the paint on the underside of the top plate was ground off so the epoxy grout could bond to the metal, not the paint as shown in Figure 3.29. Notice that there are several structural steel cross members placed under each of the pump and turbine mounting feet locations. After the paint was removed, any paint dust and metal chips were blown off and the underside of the baseplate was wiped clean with a rag.

Figure 3.30 shows a bead of silicone sealant applied to the anchor bolts to prevent the epoxy from flowing into the anchor bolt sleeve imbedded in the concrete. A neoprene sleeve is installed over each anchor bolt to prevent the epoxy grout from adhering to the anchor bolt as shown in Figure 3.31, Figure 3.32 shows a protective covering put onto the anchor bolt threads and Figure 3.33 shows the concrete foundation ready to have the baseplate set onto it.

Prior to setting the baseplate, final preparations were made to the baseplate. Figure 3.34 shows the top surfaces of each foot being cleaned with ScotchBrite to remove any rust. A fine flat file was then used to remove any burrs on the top surface and used to bevel the outside



FIGURE 3.27 Close-up of one of the anchor bolts.

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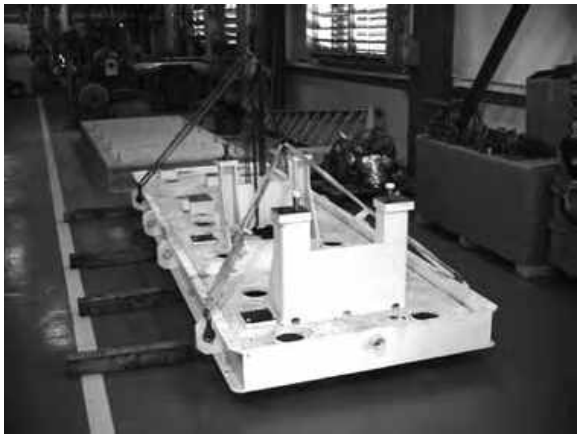


FIGURE 3.28 Rigging used to lift and move the baseplate into position.



FIGURE 3.29 Paint on underside of baseplate removed to insure good bonding with epoxy.



FIGURE 3.30 Sealant applied to anchor bolt to prevent grout from entering the protective sleeve.

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FIGURE 3.31 Neoprene sleeve to protect threads on anchor bolt.



FIGURE 3.32 Protecting the threads.

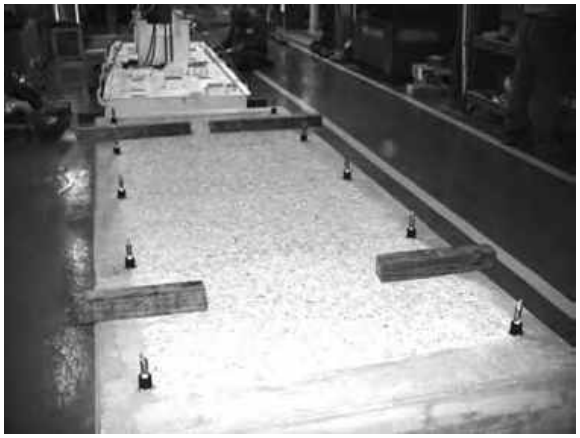


FIGURE 3.33 Ready to set baseplate onto concrete foundation.

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FIGURE 3.34 Cleaning the top surfaces of each mounting point.

edges as shown in Figure 3.35. The flat file was then used to bevel each edge of the mounting pads as shown in Figure 3.36. A tap was ran through each of the jackscrew holes and an antiseize compound applied to the jackscrew threads as shown in Figure 3.37. Any dirt and dust was then blown off the top of the baseplate and the surface wiped clean with a rag.

The baseplate was then lifted and carefully positioned over the anchor bolts as shown in Figure 3.38. Each jackscrew was set onto stainless steel disks with a "V" coned into the top of each disk for the jackscrew to set into as shown in Figure 3.39.

Figure 3.40 shows a top view drawing of the baseplate indicating how each foot pad was labeled. A precision machinists level was then used to roughly level the baseplate, then an optical jig transit was set up and elevations were taken on all eight mounting pads as shown in Figure 3.41 through Figure 3.44. The line of sight of the jig transit was set slightly above the tops of the pump foot pads so readings could be taken on the 10 in. optical scale targets located there. A 20 in. scale target was then used on the turbine foot pads since there was a 17 in. offset in elevation between the pump foot pads and the turbine foot pads. We decided to set the "shoot for" elevation on the pump pads at 0.800 in. and the turbine foot pads at 17.800 in.

We were unable to level the baseplate by adjusting only the jackscrews. The greatest amount of difficultly in positioning the foot pads at the "shoot for" elevation occurred between the pump pads (B and G) and the turbine pads (C and H). To position the pads at these locations, we had to distort the baseplate by alternately tightening down on an anchor



FIGURE 3.35 Removing burrs with a flat file.

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FIGURE 3.36 Beveling the edges.



FIGURE 3.37 Cleaning the threads of the jackscrews.

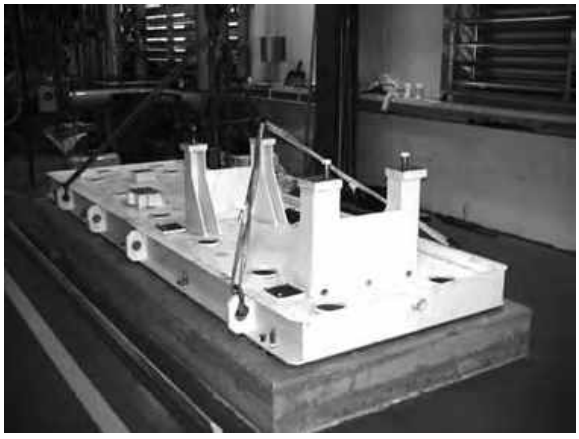


FIGURE 3.38 Setting the baseplate.

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FIGURE 3.39 Close-up of jackscrew and anchor bolt.

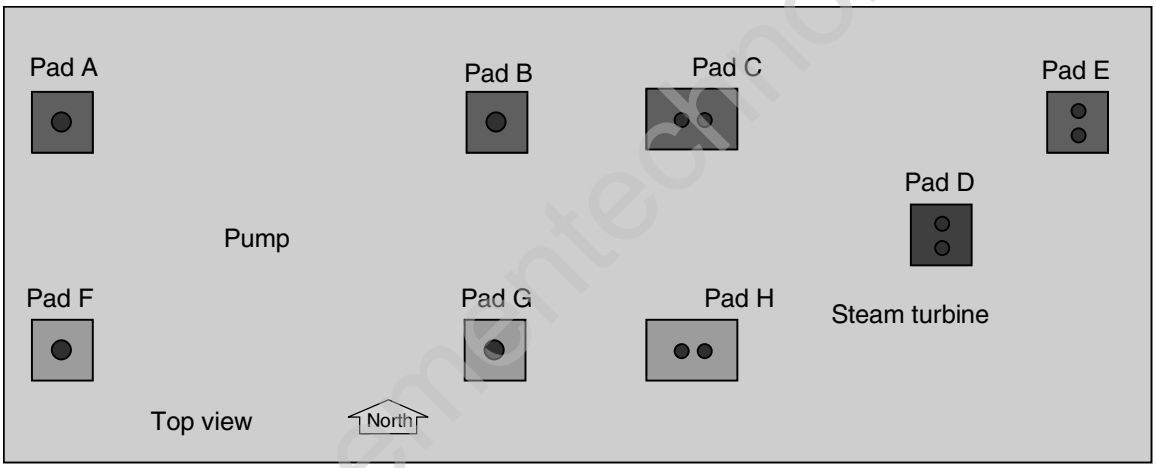


FIGURE 3.40 Foot pad identification plot.



FIGURE 3.41 Leveling the jig transit.

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FIGURE 3.42 Double checking elevations with two independent observers.



FIGURE 3.43 Jig transit observing scale targets.

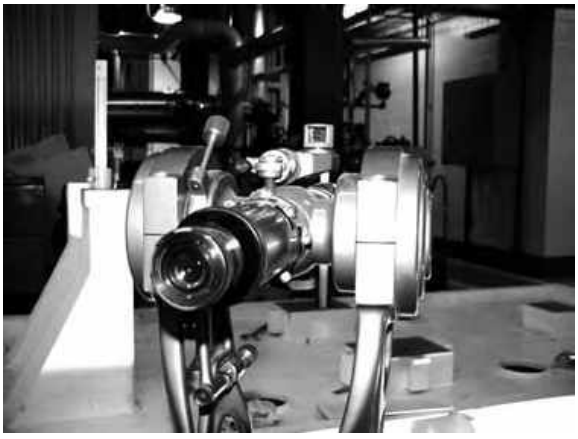


FIGURE 3.44 Observers view from jig transit.

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bolt (i.e., lowering a point) then tightening down an adjacent jackscrew (i.e., to raise a point). At foot pads G and H, the anchor bolts and jackscrews had to be tightened at or slightly over the desired torque values to distort the baseplate enough to achieve the desired elevation. There was no jackscrew or anchor bolt located at pad D to warp the baseplate to get that pad into the desired level plane. Attempts were made to get pad D into the plane using the jackscrews and anchor bolts on the perimeter of the frame but to no avail.

Once we got close to the desired elevation (i.e., around 5 mils), we noticed that began chasing our tail. Since we only had one 20 in. optical scale target to use at the turbine foot pads, we had to move the scale target from foot pad to foot pad to check the elevation. After moving the optical scale target from pad H to pad D and then back to pad H again, we noticed that the elevations were different without making any adjustments to the anchor bolts or jackscrews. At that point, we decided to take an elevation at one corner of pad H and then move the scale target to another corner of pad H and take the elevation there. The elevations at the two corners of pad H were not the same. We then realized that the foot pads were welded to the baseplate at an angle (i.e., tilted) explaining why there was an apparent elevation change when the scale target was moved from one foot pad to another. Depending on where the scale target was placed on the surface of the foot pad when it was moved, if the scale target was not placed in exactly the same spot, the elevation would be different.

We then adjusted the anchor bolts and jackscrews to place the center of each pad as close as possible to the desired elevation. All of the anchor bolt nuts were then firmly tightened to secure the baseplate in its position. We then measured the elevations at the four corners of every foot pad.

Figure 3.45 shows the data collected and the surface profile of the baseplate. Notice that none of the foot pads are in the same plane. Pad D is low by an average of 35 mils with respect to the other 3 ft of the turbine.

According to API Spec 686 (*Recommended Practices for Machinery Installation and Installation Design*, Chapter 5):

> 3.9.4 Baseplate Installation and Leveling
>
> Section 3.9.4.4—The equipment mounting surfaces are to be leveled longitudinally and transversely to within 200 micrometers per meter (0.002 inches per foot) for API 610 pumps and to within 400 micrometers per meter (0.005 inches per foot) for general purpose equipment and ASME pumps.

After careful examination of the data in Figure 3.45, the baseplate profile does not seem to meet this specification. The data was forwarded to the pump and baseplate manufacturer for their review. They seemed to be somewhat surprised that none of the foot pads were perfectly in level but particularly alarmed that Pad D was, on the average, 35 mils lower than the other in 7 pads. Approval was given to go ahead with the grout pour with the understanding that pad D would have to be shimmed.

Once the baseplate was set in this position, checks were made to determine the final elevation of the pump and turbine centerlines of rotation. According to the drawings, the floor elevation at the baseplate was 65 ft. By using the optical transit and a tape measure, it was observed that the floor elevation varied ± 1 in. depending on where the measurement was taken around the baseplate. The final centerline of rotation elevation was determined to 68 ft 5.75 $\pm$ 1 in. (desired elevation according to drawing was 68ft 6.75in).

The calculations for the volume of the baseplate (164 in. long, 60 in. wide, and 7 in. deep) were 68,880 in.^{3} = 40 ft^3 = 1.47 yd^3. The epoxy grout is sold as a unit, which consists of one can of resin, one bottle of hardener, and four bags of filler or aggregate. Each unit yields

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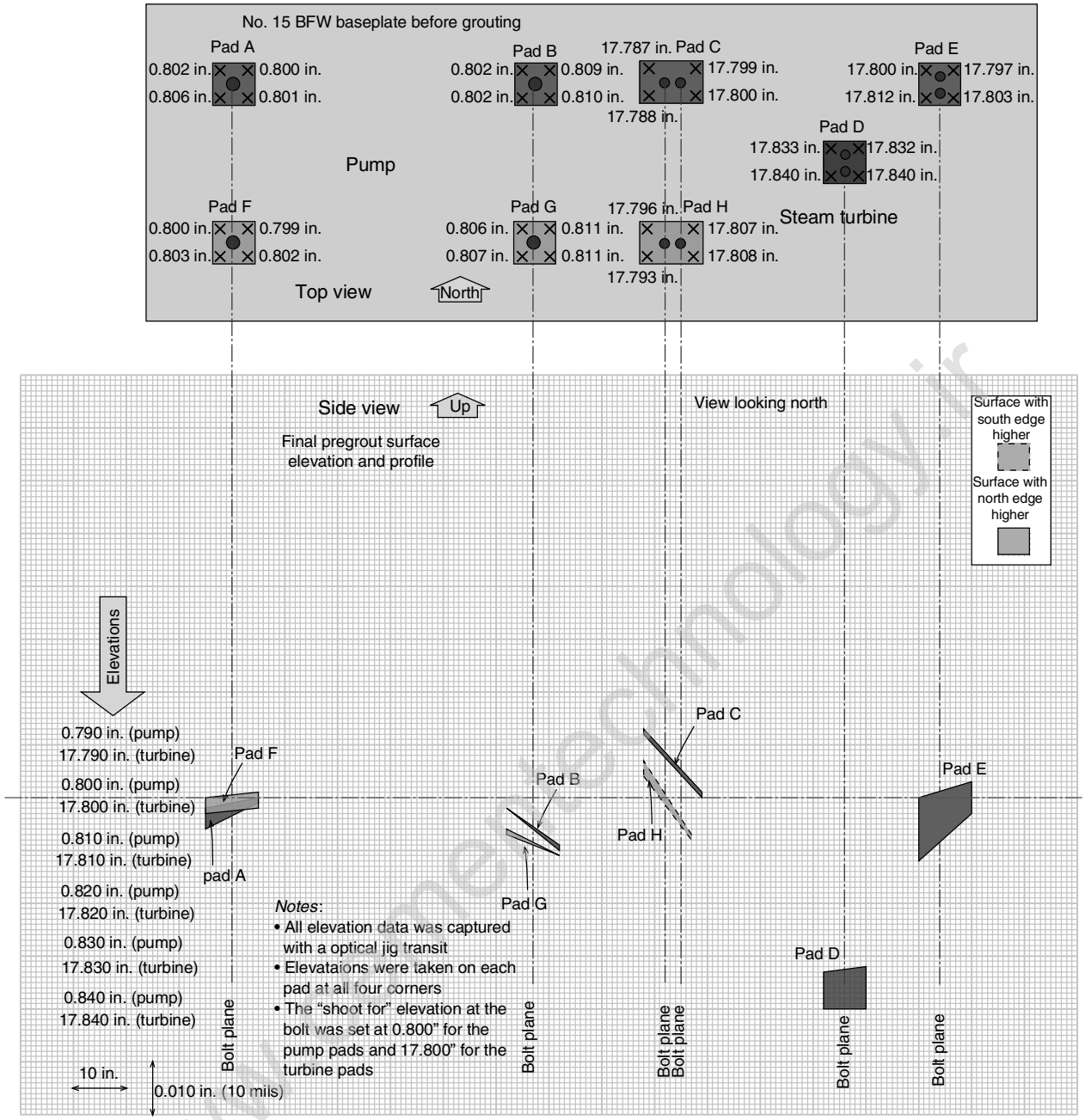


FIGURE 3.45 (**See color insert following page 322.**) Baseplate elevation profile prior to grouting.

1.73 ft^3. Therefore we will need 23 units. Each unit cost is $215.95 for a total material cost of $4966.85.

Final preparations were made prior to the pour. An inventory of the grout material was checked and it was discovered that we were 4 gal short of the hardener. The baseplate and a wide area around the foundation were covered with clear plastic sheeting to protect the baseplate and the floor as shown in Figure 3.46.

After carefully reading the epoxy grout instructions and the Grouting Checklists in the API 686 specs, a meeting was held with the contractor prior to beginning the grout pour to insure that everyone knew what tasks each had and the fact that the grout would begin to cure somewhere between 60 and 120 minutes after the pour was started based on the air temperature that day (80°F–90°F).

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FIGURE 3.46 Protecting the baseplate and surrounding area with plastic covering.

The four bottles of hardener arrived and the pour began in earnest at 1245 h. The procedure was to blend the liquids (epoxy and hardener) for 3 min and then slowly add each bag of aggregate to the mixture as shown in Figure 3.47. Once the fourth bag of aggregate was added, another 2 min of mixing was suggested before the grout should be poured. An electric drill with a mixing blade was used to mix the contents. When the first batch was mixed together, it became apparent that the mixture was very viscous (almost like peanut butter). The drill motor quickly became overloaded and the windings began to overheat and smoke. Another drill motor was at hand so the drills were swapped out and by the second batch, it too became overheated. A larger drill motor was quickly found to handle the mixing with one batch mixed at a time. After adding all the contents, the barrel weighed around 250 lb, a little too heavy for two people to lift and pour directly into the baseplate.

Because of the viscosity of the grout, the most effective way to pour was by hand, scooping the grout out of the mixing barrel and then pouring it into a hole as shown in Figure 3.48. After the barrel got half empty, it was light enough to be lifted by two people. We then would fill up a 5 gal bucket and use it to pour while another one or two people would continue to scoop it out with the smaller buckets (which were made with thick plastic 1 gal laundry detergent containers). Thankfully, two barrels were available where one crew would pour the grout whereas another crew would mix the next batch. Initially, each batch took 15 min from the time the mix was started to the time the barrel was empty. After a quick calculation, it was



FIGURE 3.47 Mixing the grout contents.

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FIGURE 3.48 Pouring the grout.

conveyed to the work crew that it would take 4 h to finish the pour, far in excess of the cure time so additional personnel were called in to speed up the process as shown in Figure 3.49.

About the time the fifth batch was poured, we noticed that the wooden form around the baseplate started to leak as shown in Figure 3.50. Thankfully, we had the foresight to have some duct sealant at hand in the event that this happened and the leaks were plugged as shown in Figure 3.51. It then took approximately 6 min for each batch with bodies scrambling around feverishly mixing and pouring the grout. At 1530 h the final batch pour was made as shown in Figure 3.52.

The top plate of the baseplate was designed with a slope toward the pump side so in the event of a water leak, the water could drain off the base to prevent rust from damaging the baseplate. Knowing this, we started the pour at the low end of the baseplate. Once the entire baseplate was filled, the grout began to flow and seek its own level and we noticed that the grout began to swell higher in the pour holes at the low end of the baseplate and drop down at the high end as shown in Figure 3.53. As seen in Figure 3.54, we then topped off the pour holes at the high end hoping that the epoxy would begin to harden at the low end where the pour first began to prevent it from overflowing onto the top of the baseplate. The grout indeed did begin to slowly harden and, in fact, became somewhat like a soft putty so it was decided to begin carving off the grout that had leaked out around the form as shown in Figure 3.55 and Figure 3.56.



FIGURE 3.49 More people needed.

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FIGURE 3.50 Grout leak around form.



FIGURE 3.51 Plugging the leak with duct sealant.



FIGURE 3.52 Finishing the pour.

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FIGURE 3.53 Grout began swelling at low end.



FIGURE 3.54 Hand packing the grout.



FIGURE 3.55 Scraping off the excess grout while still in the putty stage.

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FIGURE 3.56 Scraping off the excess grout while still in the putty stage.

The majority of the work crew then cleaned up and left. Two of us decided to stick around to smooth off the grout at the pour holes as the grout began to harden, which began to quickly accelerate about 1 h after the last pour had been made. Since the chemical reaction of the epoxy grout is exothermic, the baseplate began to get warm, then somewhat hot. We also began to notice that the epoxy began swelling out of all of the pour holes apparently from expansion of the grout during hardening. Surface temperatures of the baseplate were taken with an infrared pyrometer with temperatures ranging from 127°F to 139°F. The epoxy began to harden very quickly around 1700 h, so it was decided to remove the protective plastic sheeting from the top of the baseplate before the epoxy hardened completely. The epoxy had also oozed out of the vent holes and by this time we had to chisel them off as shown in Figure 3.57. Figure 3.58 and Figure 3.59 show the grout pour holes after the epoxy had cured. A fan was placed to begin cooling off the baseplate overnight.

The next morning it was decided to take another set of optical alignment measurements on the 8-ft pads to see if the baseplate had stayed in the same position prior to the addition of the grout. Figure 3.60 shows the jig transit and optical scale target on the foot pads. The transit was precision leveled and the line of sight was adjusted to buck back into the same elevation plane by observing the adhesive backed target placed on a nearby building column when the first set of measurements were taken as shown in Figure 3.61.



FIGURE 3.57 Chiseling off the excess grout.

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FIGURE 3.58 Epoxy at grout pour hole.



FIGURE 3.59 Baseplate after clean up.

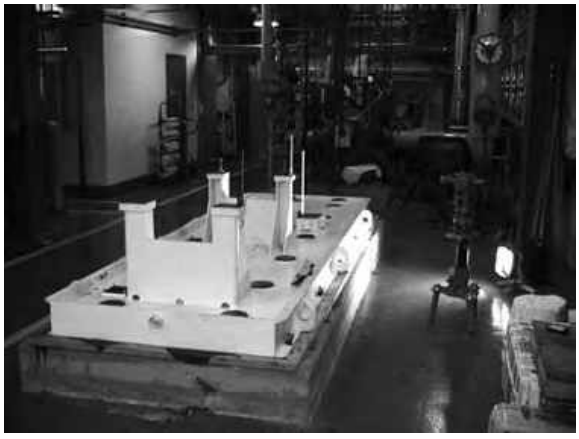


FIGURE 3.60 Jig transit set up to observe final elevations on foot pads.

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FIGURE 3.61 Reference target on building column used to buck in to same elevation.

Figure 3.62 shows the elevation data and the baseplate profiles before and after the grout had hardened. Figure 3.63 shows the pump to be installed and Figure 3.64 the turbine to be installed onto the baseplate.

As shown in Figure 3.62, the baseplate had distorted after the grout had been poured. Notice that pad E did not change its position very much after the pour had been made. All of the other pads changed their position with the pads in the center of the baseplate now much lower than either end. The baseplate bowed downwards more in the center, a little at the west end, and virtually none at the east end.

The following conclusions can be made:

1. The top surfaces of the four pump foot pads were, and still are not in the same plane.
2. The top surfaces of the four turbine foot pads were, and still are not in the same plane.
3. An 18 in. long precision machinists level is unable to span across two of the pump or turbine foot pads to check for longitudinal and transverse levelness.
4. If a precision machinists level would have been used, the leveling process would have gone on forever. Depending on which pad the machinists level was placed on and what direction it was placed in, the baseplate would have to be re-leveled for each pad. Since the pads are sloped differently, once one pad was precisely leveled, when the machinists level was moved to another pad, it would be out of level.
5. Every effort was made to position the pump foot pads and the turbine foot pads in an averaged level, coplanar, and parallel condition prior to grouting. This was achieved on seven out of the eight foot pads. This could not have been achieved by the leveling jackscrews alone. In several cases, the anchor bolt nuts had to be tightened to bend the baseplate downward to achieve the desired elevation at certain foot pads. There were no jackscrews or anchor bolts located at pad D to distort the baseplate at that position. All anchor bolt nuts were tightened and the adjacent jackscrews tightened to hold the baseplate in its pregrouted position.
6. Every effort was made to follow the installation guidelines set forth by the equipment manufacturers, the grout manufacturer, and the procedures set by API Recommended Practice 686. The objective was to completely fill the baseplate so the grout would bond to the top of the concrete foundation and the underside of the baseplate.

By carefully studying the before and after baseplate profiles in Figure 3.62, it becomes obvious that the baseplate changed its shape after the epoxy grout had cured. Our initial

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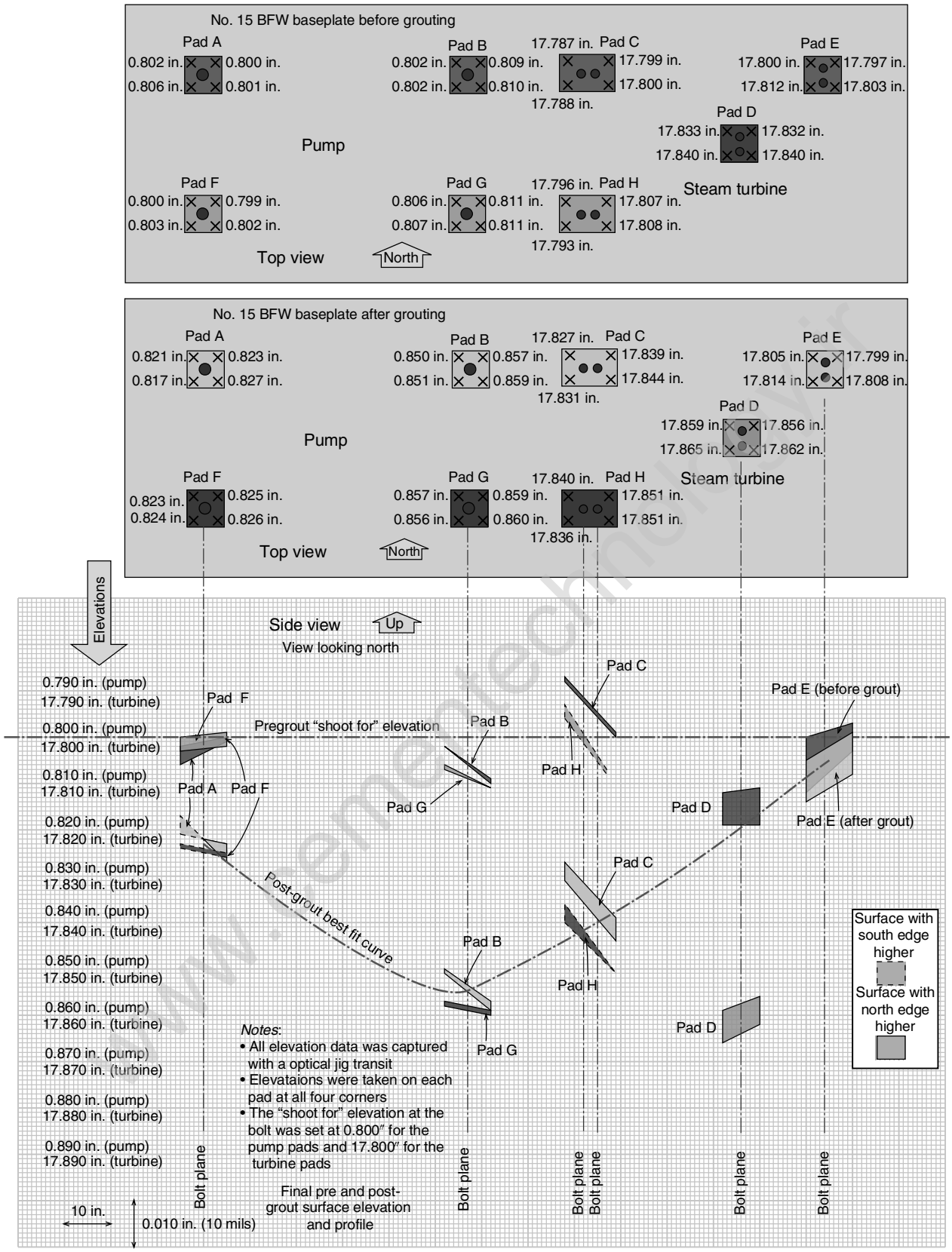


FIGURE 3.62 (**See color insert following page 322.**) Elevation data and profiles before and after grout pour.

thoughts were that the baseplate may have moved upward due to the temperature increase from the exothermic reaction of the epoxy. Instead, the opposite happened and it was not a linear move. Upon curing, epoxy grout shrinks. Once the baseplate was filled and the epoxy

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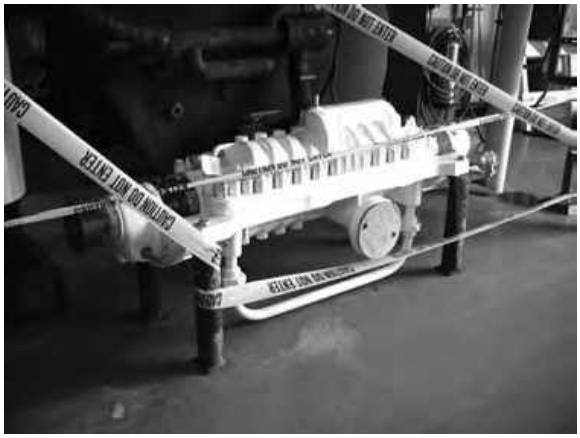


FIGURE 3.63 Pump being installed on baseplate.

bonded to the underside of the top surface of the baseplate, the grout shrunk and bowed the 1/2 in. thick top surface plate downward in the middle despite the fact that there were several structural steel cross members in the baseplate design (see Figure 3.29).

Discussions took place on how to fix the out of level and noncoplanar surfaces of the foot pads now that the baseplate was grouted. Suggestions were forwarded to field machine all of the foot pads to make them level and coplanar. If this was to be done, optical alignment equipment should be available to assist in periodically measuring the surfaces that would be machined to achieve level, parallel, and coplanar foot pad surfaces. This however would be a waste of time and money. Getting the foot pads flat and in the same plane assumes that the surfaces on the underside of the pump and turbine are flat and in the same plane. Is this true? No data was taken to verify this despite the comments of one of the equipment manufacturers: "This couldn't possibly happen." If you look at the photograph of the turbine in Figure 3.64, you will notice that the turbine supports that will touch pads D and E are L-shaped plates that are axially bolted to the lower turbine casting. If these bolts are loosened in the casting, it is possible that these support plates could be moved due to any clearance between the shank of the bolts and the holes cut into the support plates.

It was decided to set the turbine and pump onto the baseplate without machining and check for any soft foot conditions using the procedures described in Chapter 5. Figure 3.65 shows the soft foot map when the turbine and pump were set onto the base. Assuming the undersides of the pump and turbine feet were flat and in the same plane, there should have been very little

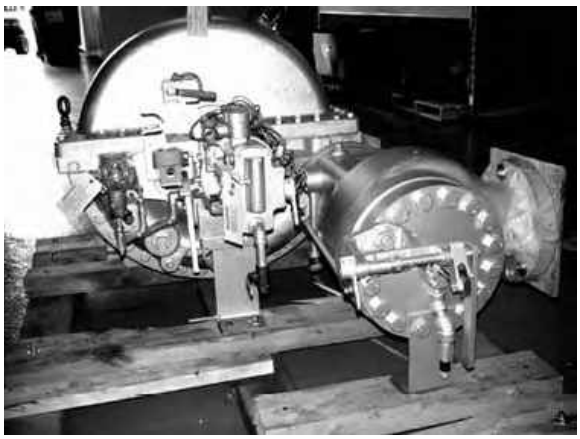


FIGURE 3.64 Turbine being installed on baseplate.

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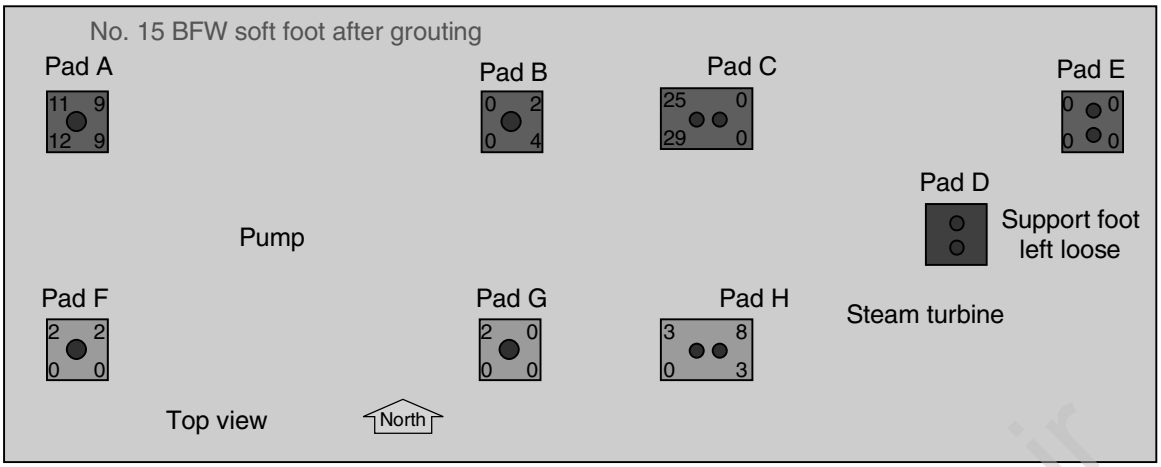


FIGURE 3.65 Soft foot map of pump and turbine.

(if any) soft foot problems on the pump (pads A, B, F, G). Now look at the soft foot map in Figure 3.65 and observe that the gaps at pad A indicate that the pump foot was not making contact there. Therefore the undersides of the pump feet were not in the same plane. A similar study of the turbine foot pad profile (pads C, D, E, H) and the soft foot map will illustrate that the underside of the turbine feet was also not in the same plane.

I have had the opportunity (i.e., been allowed) to use optical alignment equipment a total of four times measuring the four corners of all the foot pads and have seen similar conditions on every baseplate checked this way. I also know that very few baseplate installations are done with this rigorous of a measurement process and that carpenters levels, not machinists levels, are frequently used and that very few people verify that a baseplate is indeed in level after the installers say it was. I am also not sure how often a baseplate was checked for levelness after the grout was poured.

It should now become obvious that this may indeed occur very frequently and that foot pads quite likely have a tilt and or twist condition and that the surfaces are not coplanar and that when the machinery is placed onto the uneven, twisted, tilted foot pads, that a flat piece of shim stock will not correct a complex, wedge-shaped gap that will occur. It is difficult, but not impossible to fix this. Chapter 5 will discuss the procedure for doing this.

Finally, what effect will a baseplate that is not in level within 2 mils/ft and 5 mils across the entire baseplate have on the successful operation of the machine? If a drive system has a 50 mil slope across the entire baseplate, will the thrust bearings not be able to accept this slight axial force from gravity? I do not prescribe installing baseplate with that radical a slope, but attempting to achieve the tolerances set forth by manufacturers and professional organization seems to be unachievable in the real world. Many may think that what was observed during this particular installation does not occur very frequently, when in fact, it is probably quite common. Perhaps some rethinking needs to be done in soleplate and baseplate installation specifications.

3.3 PROBLEMS TO LOOK FOR IN YOUR FOUNDATIONS AND BASEPLATES

A complete visual inspection should be made at least once a year of all rotating equipment foundations, baseplates, piping, etc. Many of these problems are quite obvious as shown in Figure 3.66 through Figure 3.71.

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FIGURE 3.66 Fan frame mistakenly designed with no soleplates imbedded in concrete resulting in no contact between underside of frame and top of concrete in the inertia block.



FIGURE 3.67 During a torque check on an anchor bolt, it was discovered that the anchor bolt had sheared and an attempt was made to tack weld the anchor bolt so no one would notice.



FIGURE 3.68 During a torque check on a steam turbine sway bar, the bolts threaded into the underside of the outboard bearing were found to be loose. Channel iron was used for the sway bar and the wedge-shaped washers would loosen after a short period of time from the vibration of the machine. The bolts were tightened, the vibration on the outboard bearing would be acceptable for a period of time but as the bolts began to loosen again, the vibration would steadily increase to unacceptable levels.

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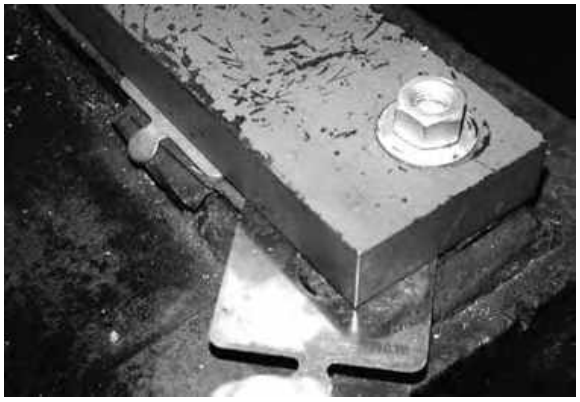


FIGURE 3.69 Inspection of the grout under the soleplate revealed that the grout had begun to deteriorate. Apparently this was observed prior to this and precut shim stock was used to attempt to correct the gap between the top of the grout and the underside of the soleplate, which is not recommended. Also notice that the top of the anchor bolt is not fully engaged in the nut.

3.3.1 Piping, Ductwork, and Conduit Strain

Piping strain is a monumental problem in industry, difficult to detect on installed piping, and time consuming to correct. The widely held design philosophy seems to be that piping should be loosely constrained so it can move and grow wherever it wants. Many people are surprised to learn that the vast majority of piping failures have occurred from cyclic fatigue, not from tension, compression, or shear failures. Most of the piping supports in existence were installed by pipe fitters who were just supporting the pipe before all the connections were made. Similar strain problems can also occur on ductwork for fans and conduit for electric motors. Excessive piping, ductwork, or conduit forces can:

1. Distort the machine case upsetting internal clearances between moving and stationary parts of the machine
2. Cause the machine case to shift its position over a short (or long) period of time disrupting the alignment condition
3. Cause the hold down bolts to loosen or shear along with the shim packs and dowel pins if used



FIGURE 3.70 A misalignment condition between the motor and water pump caused the packing to leak. Over time, the spraying water severely oxidized the pump casing and baseplate.

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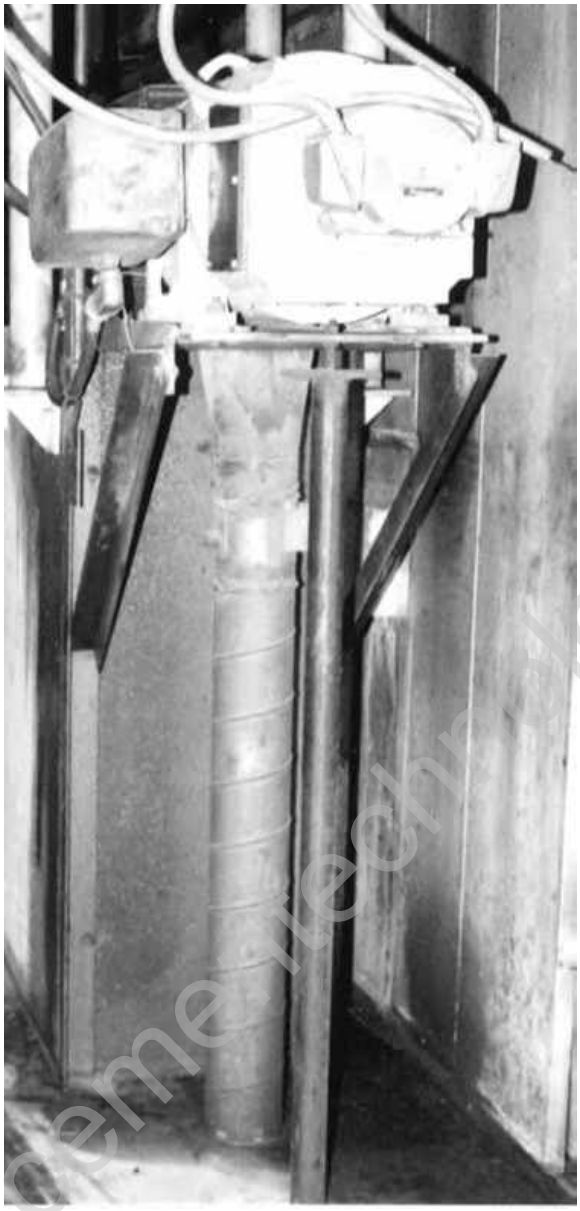


FIGURE 3.71 A poorly designed frame fabricated from plate steel vibrated excessively and eventually the angle iron support gussets broke loose from the concrete support holding the gearbox in position putting the unit into a severe misalignment condition. The bearings failed on both machines and rather than remove the concrete pedestal and provide a complete support for the gear and the motor, a pipe and jackscrew were installed, which eventually failed after a short period of time.

Static piping forces that result from improper fits cannot be detected by simple visual inspection after the piping has been attached to a pump, compressor, or turbine. Looking at a spring hanger and seeing that the spring is compressed does not indicate that the load is within acceptable limits. Also, spring hangers can only support piping loads in one direction. What if there are other forces acting in directions other than that through the axis of the spring?

Even if expansion joints or flexible hose sections are included in the piping, these devices can only accept forces in one or two planes of motion. Installing flexible piping sections may be just an excuse for someone to do a poor piping installation. Furthermore, flex hose is more susceptible to failure than rigid pipe.

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Forces from the expansion or contraction of piping attached to rotating equipment carrying fluids whose temperatures are above or below the temperature of the pipe when no fluid is moving can be enormous and frequently cause drastic movement in the turbo machinery from excessive forces at the connection points. The flanges and connections on pumps, compressors, fans, etc., were never meant to bear the weight or strain of piping and ductwork. They are fluid connection points. The piping must have adequate support mechanisms that bear the weight and strain of the piping in the vertical, lateral, and axial directions. A good piping design engineer should never view a pump or a compressor flange as an anchor point for the piping.

I know that this has never happened at your plant, but I have seen pipe fitters attach a 20 t chain fall around one end of a pipe and the other end around an I-beam, pull the pipe into place, install and tighten the flange bolts, and then remove the chain fall. Some of the piping in industry is so poorly installed that the pipe fitter has to stand out of the way when the pipes are disconnected from machinery for fear of getting hit by the pipe when it springs away from the connection.

Figure 3.72 shows an adjustable piping anchor support. If the piping misalignment is too excessive and you are not willing to fit the pipe properly, you may want to consider using supports similar to this at the suction and discharge pipes on the pumps and compressors to prevent the pipe from forcing the machinery out of alignment. If the piping strain is excessive and there are no supports to hold the poorly fit pipe in place, there is no guarantee that the equipment will stay aligned for long periods of time even if you do a great job aligning the rotating machinery.

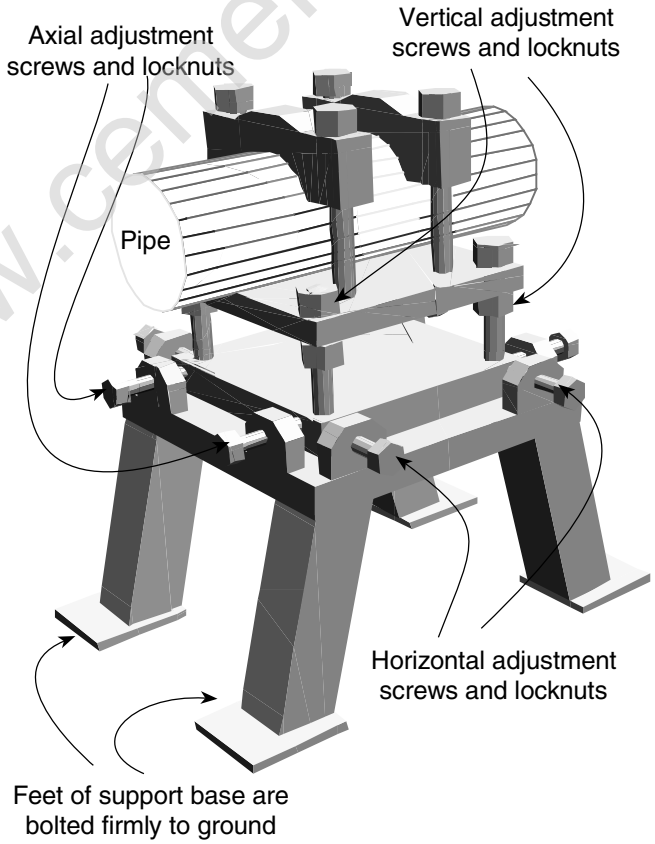


FIGURE 3.72 Adjustable piping support.

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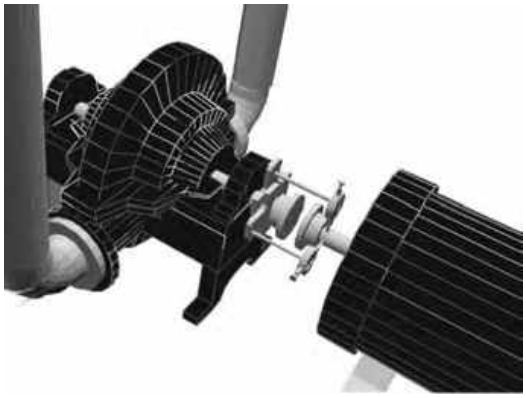


FIGURE 3.73 Excessive piping stress test. (Align the machinery and then attach brackets or clamps to one shaft and mount dial indicators in the vertical and horizontal position against the other shaft. Set the indicators at zero, loosen the foot bolts holding the piped machine in place, and monitor the indicators for any movement. Ideally less than 2 mils (0.002 in.) of movement should occur.)

3.4 CHECKING FOR EXCESSIVE STATIC PIPING FORCES ON ROTATING EQUIPMENT

Since a majority of rotating equipment is used to transfer liquids or gases, the connecting piping will undoubtedly have an effect on the machinery and could potentially be another source of machinery movement due to thermal expansion of the piping, reactionary forces from the movement of the liquid in the piping itself, static weight of the piping, or piping that has not been installed properly causing tension or compression at the piping–machine interface.

The forces that cause machinery to move from improper installation of piping can be checked by using dial indicators to monitor both the horizontal and vertical movement of the machine case. By placing indicators at each corner of the machine element, loosening all the foundation bolts, and observing the amount of movement shown on the indicators, any undesirable forces acting on the machine can be determined. If more than 2 mils of movement is noticed, it may be possible to reposition the other elements in the drive train without modifying the piping to eliminate this problem.

This movement can also be checked with a shaft alignment bracket attached to one shaft with dial indicators positioned at the twelve o'clock and three o'clock positions, on the adjacent shaft as shown in Figure 3.73 or it could be check using a magnetic base and dial indicator set up to observe motion of the machine case itself as the foot bolts are loosened with the piping attached as shown in Figure 3.74. Shaft or casing movement exceeding 2 mils on any dial indicator is unacceptable after all the foot bolts have been loosened.

3.5 VISUAL INSPECTION CHECKLIST

On an annual basis (at least), the following inspection should be made on all rotating machinery at your plant site:

1. Cracked concrete bases or support columns
2. Cracks propagating at concrete joints
3. Water seeping between baseplate and concrete foundation that could freeze and damage the structure
4. Loose foundation bolts
5. Shim packs that worked loose
6. Rusty shims

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FIGURE 3.74 Alternate piping stress check using magnetic base and indicator to observe case movement when foot bolts are loosened with piping attached.

7. Loose or sheared dowel pins
8. Paint on shims
9. Properly positioned piping hangars that carry the weight of the piping
10. Piping expansion joints that move freely to accept thermal or hydraulic movement
11. Loose piping flange bolts

3.6 HOW LONG WILL ROTATING MACHINERY STAYED ACCURATELY ALIGNED?

As discussed in Chapter 1, periodic alignment checks will indicate if shifting is occurring. Disappointingly, very few long-term studies of the alignment shifting of rotating machinery in industry have been performed. It is logical to conclude that the shaft alignment will change if there is a shift in the position of the foundation. This shifting can occur very slowly as the base soils begin to compress from the weight and vibration transmitted from the machinery above. It can also occur very rapidly from radiant or conductive heat transfer from the rotating equipment itself heating the soleplate, concrete, and attached structure. There are documented case histories where drive trains were aligned well within acceptable alignment tolerances and after a 4–6 h run, moved considerably out of alignment. Many people assume that when rotating equipment is aligned when it is installed or rebuilt, the alignment will stay stable forever.

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4 Flexible and Rigid Couplings

One of the most important components of any drive system is the device connecting the rotating shafts together known as coupling. As it is nearly impossible to maintain perfectly collinear centerlines of rotation between two or more shafts, flexible couplings are designed to provide a certain degree of yielding to allow for initial or running shaft misalignment. There is a wide assortment of flexible coupling designs, each available in a variety of sizes to suit specific service conditions.

The design engineer invariably asks why are there so many types and is one type better than any other? Simply put, there is no perfect way to connect rotating shafts (so far!). As you progress through this chapter, you will find that perhaps two or three different coupling types will fit the requirements for your drive system. One coupling being better than another is a relative term. If two or more coupling types satisfy the selection criteria and provide long, trouble-free service, they are equal, not better. The ultimate challenge is for you to accurately align shafts, not find a coupling that can accept gross amounts of misalignment to compensate for your ineptitude.

The pursuit to effectively connect two rotating shafts dates back to the beginning of the industrial era where leather straps and bushings or lengths of rope intertwined between pins were the medium used to compensate for shaft misalignment. Several flexible coupling designs emerged immediately after the introduction of the automobile from 1900 to 1920. As shaft speeds increased, coupling designs were continually refined to accept the new demands placed on them. As industrial competition became more severe, equipment downtime became a major concern, and industry became increasingly more interested in their coupling failures in an effort to prolong their operating lifespan.

Patents for diaphragm couplings date back to the 1890s but did not become widely used until just recently as diaphragm design, material, and construction vastly improved. O-ring type seals, and crowning of gear teeth in gear couplings came about during World War II. The awareness and concern for coupling and rotating machinery problems are reflected by the increase in technical information generated since the mid-1950s. Coupling designs will continually be refined in the coming years with the ultimate goal of designing the "perfect" coupling.

4.1 COUPLING AND SHAFT MISALIGNMENT TOLERANCES—WHAT IS THE DIFFERENCE?

It is important for the person selecting the coupling not to be confused by the term "allowable misalignment" in a coupling. The coupling manufacturers will often quote information on allowable misalignment for the coupling and not necessarily the equipment it is coupled to. These tolerances seem to lull the user into a sense of complacency, leading one to believe that accurate shaft alignment is not necessary as "... the coupling can take care of any misalignment" (famous last words).

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It is imperative that you can differentiate between coupling tolerances and alignment tolerances. Coupling misalignment tolerances quoted by flexible coupling manufacturers typically specify the mechanical or fatigue limits of the coupling or components of the coupling. These misalignment tolerances are frequently excessive compared to the misalignment tolerances specified in Chapter 5, which deal with the rotating drive system as a whole. The misalignment tolerance guide shown in Figure 5.4 is concerned with the survivability of not only the coupling, but also the shafts, seals, and bearings of the machinery over long periods of time.

4.2 THE ROLE OF THE FLEXIBLE COUPLING

Exactly what is a coupling supposed to do? If a "perfect" coupling were to exist, what would its design features include?

- Allow limited amounts of parallel and angular misalignment
- Transmit power
- Insure no loss of lubricant in grease packed couplings despite misalignment
- Be easy to install and disassemble
- Accept torsional shock and dampen torsional vibration
- Minimize lateral loads on bearings from misalignment
- Allow for axial movement of shafts (end float) even under misaligned shaft conditions without transferring thrust loads from one machine element to another or, in some cases, limit the amount of end float to allow running at a motors magnetic center but prevent end thrust in sleeve bearings not designed to tolerate this
- Stay rigidly attached to the shaft without damaging or fretting the shaft
- Withstand temperatures from exposure to environment or from heat generated by friction in the coupling itself
- Have ability to run under misaligned conditions (sometimes severe) when equipment is initially started to allow for equipment to eventually assume its running position
- Provide failure warning and overload protection to prevent coupling from bursting or flying apart
- Produce minimum unbalance forces
- Have a minimal effect on changing system critical speeds
- Be of materials capable of long life in the environment in which installed (e.g., do not use austentite stainless steel disks in coupling installed on acid pumps)

4.3 WHAT TO CONSIDER WHEN SPECIFYING A FLEXIBLE COUPLING

Although some of the items listed below may not apply to your specific design criteria when specifying a flexible coupling for a rotating equipment drive system, it is a good idea to be aware of all of these items when selecting the correct coupling for the job.

- Normal horsepower and speed
- Maximum horsepower/torque being transmitted at maximum speed (often expressed as hp/rpm)
- Misalignment capacity parallel, angular, and combinations of both parallel and angular.
- Can the coupling accept the required amount of "cold" offset of the shafts without failure during startup?
- Torsional flexibility
- Service factor

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- Temperature range limits
- How is the coupling attached to the shafts?
- Size and number of keyways
- Type and amount of lubricant (if used)
- Type and design of lubricant seals
- Actual axial end float on rotors
- Allowable axial float of shafts
- Actual axial thermal growth or shrinkage of rotors
- Type of environment coupling will be exposed to
- Will coupling be subjected to radial or axial vibration from the equipment?
- Diameter of shafts and distance between shafts
- Type of shaft ends (straight bore, tapered, threaded, etc.)
- Starting and running torque requirements
- Are the running torques cyclic or steady state?
- Where is a failure likely to occur and what will happen?
- Noise and windage generated by the coupling
- Cost and availability of spare parts
- Lateral and axial resonance's of the coupling
- Coupling guard specifications for size, noise, and windage control
- Installation procedure
- Moments of inertia
- Heat generated from misalignment, windage, friction
- Intermittent high starting torque in some cases

4.4 TYPES OF FLEXIBLE COUPLINGS

The couplings found in this chapter show some of the commonly used couplings in industry today but in no way reflect every type, size, or manufacturer. The information presented for each coupling concerning capacity, maximum speeds, shaft bore diameters, and shaft-to-shaft distances are general ranges and do not reflect the maximum or minimum possible values available for each coupling design.

There are four broad categories of flexible couplings:

1. Miniature
2. Mechanically flexible
3. Elastomeric
4. Metallic membrane/disk

Misalignment capacities will not be given for a variety of reasons:

1. Manufacturers of similar couplings do not agree or publish identical values for angular or parallel misalignment.
2. Manufacturers rarely specify if the maximum values for angular misalignment and parallel misalignment are stated separate or a combination of the angular and parallel values stated.
3. It is the intent of this book to provide the reader with the ability to obtain alignment accuracies well within the limits of any flexible coupling design.

Coupling manufacturers assume that the user will operate the coupling within their stated maximum misalignment values. If your rotating equipment or coupling has failed due to

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excessive misalignment, it is your fault. Good luck trying to get the coupling manufacturer to pay for the damages!

This does not infer that all couplings accept the same maximum misalignment amounts or that these allowable values should not influence the selection of a coupling. Always consult with your coupling vendor or manufacturer about your specific coupling needs. If you are not getting the satisfaction you feel, you need to properly select a coupling, consult a variety of manufacturers (or end users) to comment on design selection or problem identification and elimination.

Although there are a variety of coupling designs that accommodate fractional horsepower devices such as servomechanisms, this chapter will primarily show flexible couplings used on high horsepower, high-speed turbomachinery. However, to give the reader an idea of the design differences between the fractional and higher horsepower couplings, Figure 4.1 illustrates a few of the fractional horsepower designs.



(a)



(b)

FIGURE 4.1 Miniature flexible couplings. (Courtesy of Guardian Industries, Michigan, IN (b). With permission.)

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4.4.1 Mechanically Flexible Coupling Designs

4.4.1.1 Chain Couplings

The chain coupling is basically two identical sprockets with hardened teeth connected by double width roller or "silent" type chain. Packed grease lubrication is primarily used with this type of construction necessitating a sealed sprocket cover. A detachable pin or master link allows for removal of the chain. Clearances and flexing of the rollers and sprocket allow for misalignment and limited torsional flexibility.

- Capacity: to 1000 hp at 1800 rpm (roller), 3000 hp at 1800 rpm (silent).
- Maximum recommended speed: up to 5000 rpm.
- Shaft bores: up to 8 in.
- Shaft spacing: determined by chain width, generally 1/8 to 1/4 in.
- Special designs and considerations: Wear generally occurs in sprocket teeth due to excessive misalignment or lack of lubrication. Torsional flexibility limited by yielding of chain.

Advantages:

- Easy to disassemble and reassemble
- Fewer number of parts

Disadvantages:

- Speed limited due to difficulties in maintaining balancing requirements
- Requires lubrication
- Limited allowable axial displacement

4.4.1.2 Gear Couplings

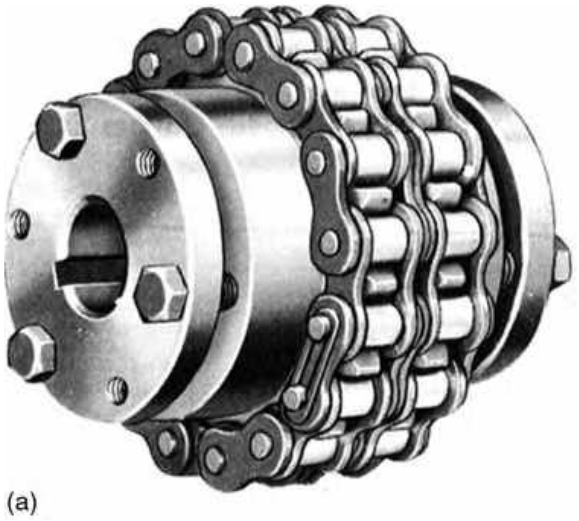
The gear coupling consists of two hubs with external gear teeth that are attached to the shafts. A hub cover or sleeve with internal gear teeth engages with the shaft hubs to provide the transmission of power. Gear tooth clearances and tooth profiles allow misalignment between shafts. Lubrication of the gear teeth is required and various designs allow for grease or oil as the lubricant.

Gear couplings can also employ a spacer or spool piece in the event that greater separation is needed or desired between shaft ends. The coupling hub with the external gear teeth and its mating sleeve can be located on each shaft with the spool piece employing rigid flanges at both ends or the coupling hubs with the external gear teeth and their mating sleeves can be located on the spool piece with rigid flanged hubs on each shaft end as shown in Figure 4.8.

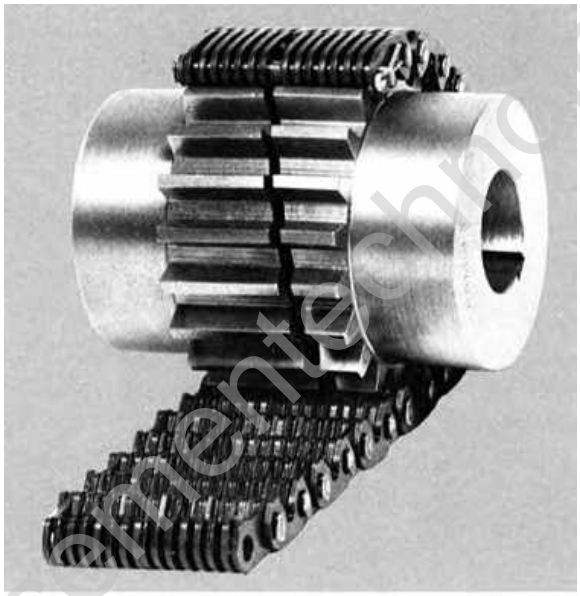
- Capacity: up to 70,000 hp.
- Maximum recommended speed: up to 50,000 rpm
- Shaft bores: up to 30 in.
- Shaft spacing: up to 200 in.

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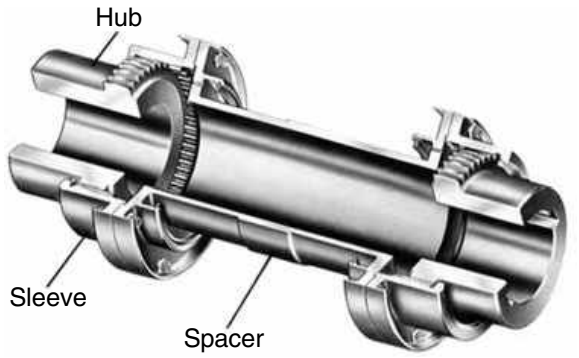
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(a)



(b)



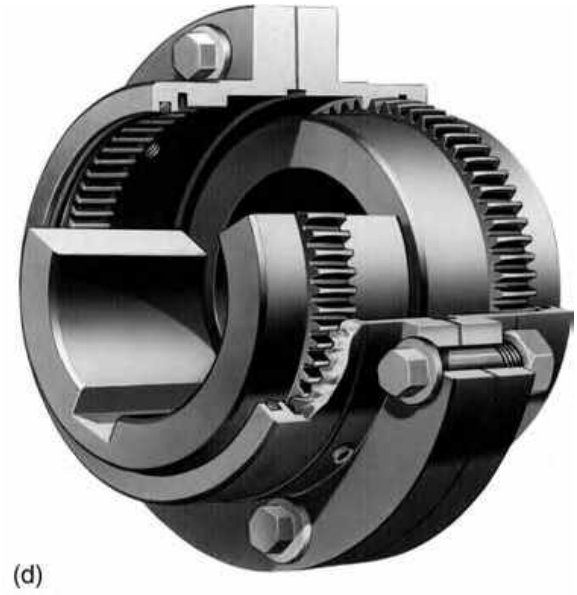
(c)

FIGURE 4.2 Mechanically flexible couplings. [Courtesy of (a) Browning Mfg., Maysville, KY; (b) Ramsey Products, Charlotte, NC; (c) Falk Corporation, Milwaukee, WI. With permission.]

(*continued*)

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(d)



(e)



(f)

FIGURE 4.2 (continued) Mechanically flexible couplings. [Courtesy of (d) Zurn Industries, Erie, PA; (e) Dodge-Reliance Electric, Cleveland, OH; (f) Zurn Industries, Erie, PA. With permission.]

- Special designs and considerations: A considerable amount of attention is paid to the form of the tooth itself and the tooth "profile" has progressively evolved through the years to provide minimum wear to the mating surfaces of the internal and external gear sets.

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To provide good balance characteristics, the tip of the external gear tooth is curved and tightly fits into the mating internal gear hub cover. If the fit is too tight, the coupling will be unable to accept misalignment without damaging the coupling or the rotating equipment. If it is too loose, the excessive clearance will cause an imbalance condition. Obtaining a good fit



(a)



(b)

FIGURE 4.3 Elastomeric flexible couplings. [Courtesy of (a) Falk Corporation, Milwaukee, WI; (b) Browning Mfg., Maysville, KY. With permission.]

(*continued*)

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FIGURE 4.3 (continued) Elastomeric flexible couplings. [Courtesy of (c) KTR Corporation, Michigan, IN; (d) Magnalog Coupling Company, Alpena, MI; (e) Lovejoy, Downess Grove, IL. With permission.] (*continued*)

can become very tricky when the coupling hubs have been thermally or hydraulically expanded and shrunk onto a shaft where an increase in diameter of the external gears will occur. As a rule of thumb, 1 mil per inch of external gear tooth diameter can be used as the clearance.

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FIGURE 4.3 (continued) Elastomeric flexible couplings. [Courtesy of (f) T.B. Wood's and Sons, Chambersburg, PA; (g) Dodge-Reliance Electric, Cleveland, OH. With permission.]

The amount of misalignment in a gear coupling directly affects the wear that will occur in the mating gear teeth. To better understand the motion of a gear coupling, Figure 4.9 shows the two basic positions gear teeth will take in its sleeve.

At a certain point during the rotation of the shaft, the gear tooth is in a tilted position and will completely reverse its tilt angle 180° from that point relative to the coupling sleeve. The gear tooth, at 90° from the tilted position, now assumes a pivoted position, which also reverses 180°. The gear tooth forces are at their maximum when in the tilted position as they supply the rotational transmission of power and as the misalignment increases, fewer teeth will bear the load. Depending on the relative position of the two shafts, each set of gear teeth on each hub may have their maximum tilt and pivot points at different positions with respect to a fixed angular reference location.

Under excessive misalignment conditions, the load will be carried by the ends of the gear tooth flank and eventually cause gouging of the internal gear teeth and "knife edging" of the external gear as the compressive stresses are extremely high forcing any lubricating film out, allowing metal to metal contact to occur.

Another peculiar wear pattern evolves when the gear tooth sliding velocity falls into the 5–8 in./s range and lubrication between the gear teeth diminishes. This type of wear is known as "worm-tracking" where gouges occur generally from the base to the tip of the tooth flank. The formation of this type of wear pattern will occur when little or no lubrication occurs at

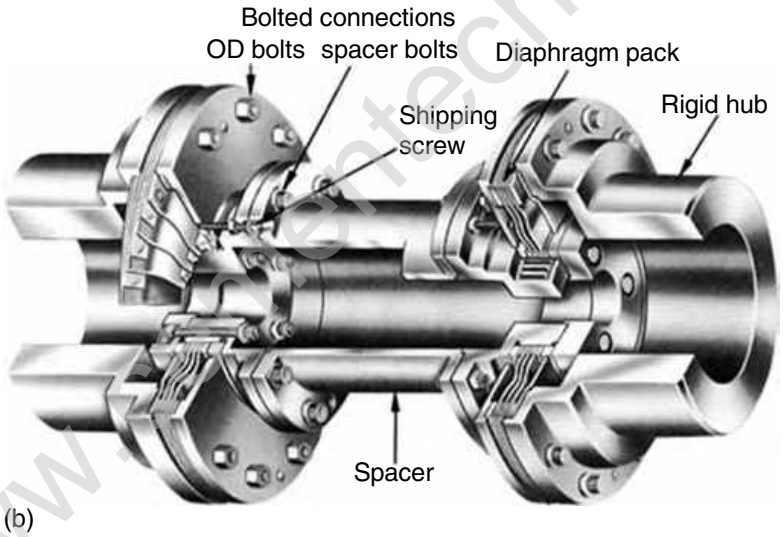
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the points of contact between teeth and the metal-to-metal contact fuse welds a small portion of the tooth flanks. As rotation continues, cracks begin to form at the outer edges of the weld and eventually propagate until the two welded pieces separate entirely from the mating external and internal gear teeth.



(a)



(b)



(c)

FIGURE 4.4 Disc and diaphragm flexible couplings. [Courtesy of (a) Coupling Corporation of America, Jacobus, PA; (b) Zurn Industries, Erie, PA; (c) Dana Industrial Power Transmission, San Marcos, TX. With permission.]

(*continued*)

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FIGURE 4.4 (continued) Disc and diaphragm flexible couplings. [Courtesy of (d) Thomas Rexmord, Warren, PA; (e) Dodge-Reliance Electric, Cleveland, OH; (f) Falk Corporation, Milwaukee, WI. With permission.]

Advantages:

- Allows freedom of axial movement
- Capable of high speeds
- Low overhung weight
- Good balance characteristics with proper fits and curved tooth tip profile
- Long history of successful applications
- High "power density" ratio (i.e., it can transmit large amounts of torque in a small package)

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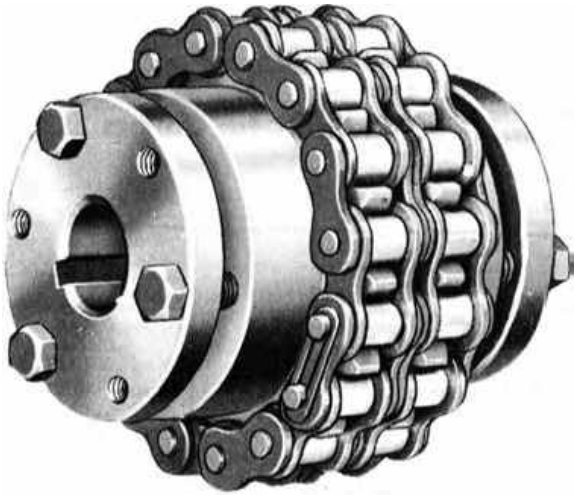


FIGURE 4.5 Chain type flexible coupling. (Courtesy Browning Mfg., Maysville, KY. With permission.)



FIGURE 4.6 Gear type flexible coupling. (Courtesy of Falk Corporation, Milwaukee, WI.)



FIGURE 4.7 Gear type flexible coupling. (Courtesy of KTR Corporation, Michigan, IN. With permission.)

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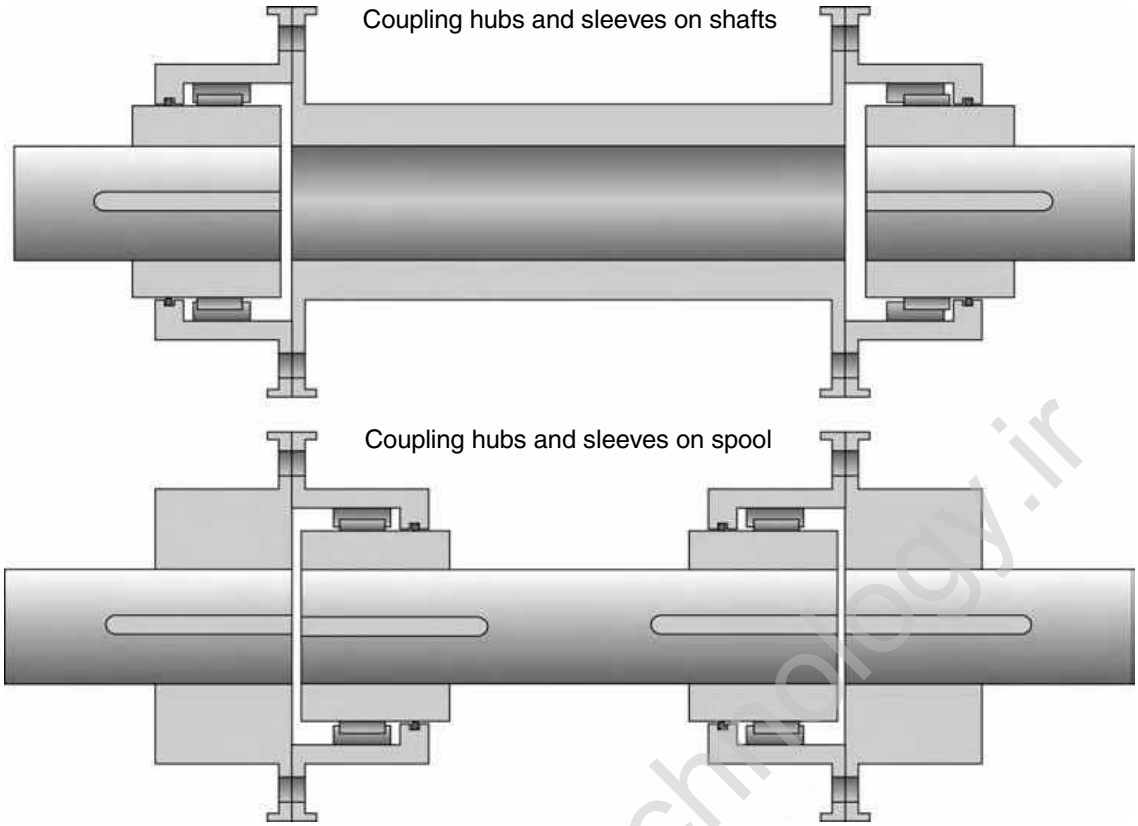


FIGURE 4.8 Two different variations of gear couplings employing a spool piece.

Disadvantages:

- Requires lubrication
- Temperature limitation due to lubricant
- Difficult to calculate reaction forces and moments of turbomachinery rotors when using these couplings since the values for the coefficient of friction between the gear teeth vary considerably

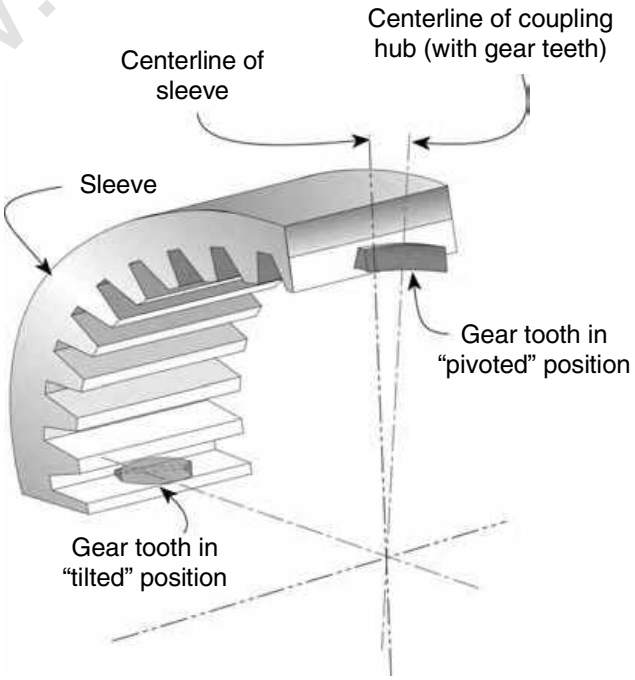


FIGURE 4.9 Tilted and pivoted positions of the gear teeth in its sleeve.

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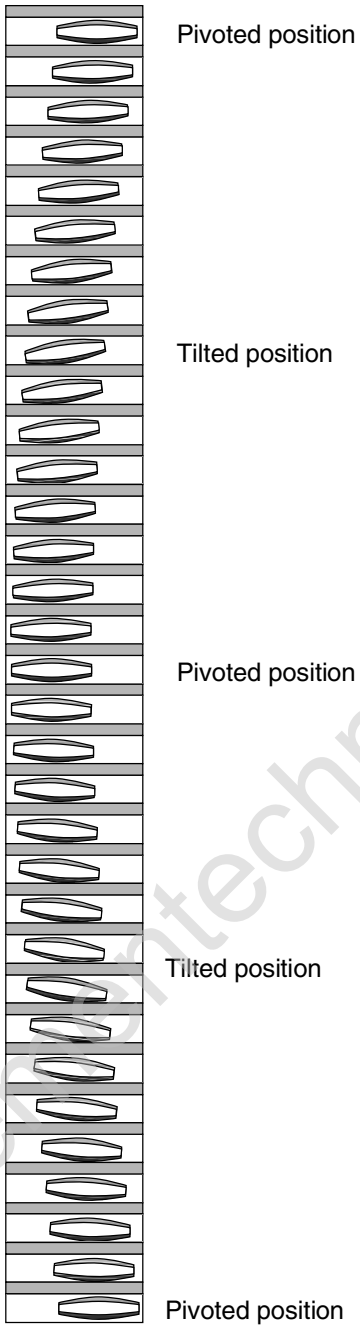


FIGURE 4.10 Gear tooth tracking pattern when subjected to misalignment conditions.

4.4.1.3 Metal Ribbon Couplings

The metal ribbon coupling was introduced around 1919 by Bibby Co. Metal ribbon couplings consist of two hubs with axial "grooves" on the outer diameter of the hub where a continuous S-shaped grid meshes into the grooves. Misalignment and axial movement is achieved by flexing and sliding of the grid member in specially tapered hub "teeth."

- Capacity: up to 70,000 hp/100 rpm
- Maximum recommended speed: to 6000 rpm
- Shaft bores: to 20 in.
- Shaft spacing: to 12 in.

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FIGURE 4.11 Metal ribbon type coupling. (Courtesy of Falk Corporation, Milwaukee, WI. With permission.)

- Special designs and considerations: Grid fabricated from hardened, high-strength steel. Close coupled hubs with removable spacer available.

Advantages:

- Easy to assemble and disassemble
- Long history of successful applications
- Torsionally soft

Disadvantages:

- Requires lubrication
- Temperature limited
- Speed limited

4.4.1.4 Universal Joint Couplings

Perhaps the oldest flexible coupling in existence is the universal joint coupling. This coupling is also known as the Cardan or Hooke joint. The basic design consists of U-shaped shaft ends with a hole drilled through each "U" to accept a "+" shaped cross.

If one universal joint is used to connect two shafts together, then only pure angular misalignment can exist where the centerlines of rotation intersect at the center of the "+" shaped cross. For a flexible coupling to accept both parallel and angular misalignment, there

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FIGURE 4.12 Universal joint. (Courtesy of Zurn Industries, Erie, PA. With permission.)

must be two flexing points. Therefore most universal joint couplings have two cross/yolk assemblies as illustrated in Figure 4.12 and Figure 4.13.

When one universal joint is used it is important to recognize that variations in angular velocity will occur between the two connected shafts often referred to as the "Cardan error." Sinusoidal motion will occur in the axial and torsional directions, producing axial vibration and torsional (i.e., twisting) vibration particularly if the torque is transmitted and the rotational speed is high.

When two universal joints are used it is important to recognize that sinusoidal motion will also occur in the axial and torsional directions if the "entrance" and "exit" angles are not the same as shown in Figure 4.13. When these angles are the same in both planes, perfect kinematic balance exists across the coupling, canceling the torsional and thrust variance.

4.4.1.5 Flexible Link

The flexible link coupling utilizes a series of cross laced, metallic links with one end of each link attached to a disc mounted on the driven shaft and the other end of each link attached to a disc mounted on the driver shaft. The links are matched in pairs so that when one is in tension, the other is in compression. Misalignment and axial displacement is accomplished by a flexing action in the series of cross links.

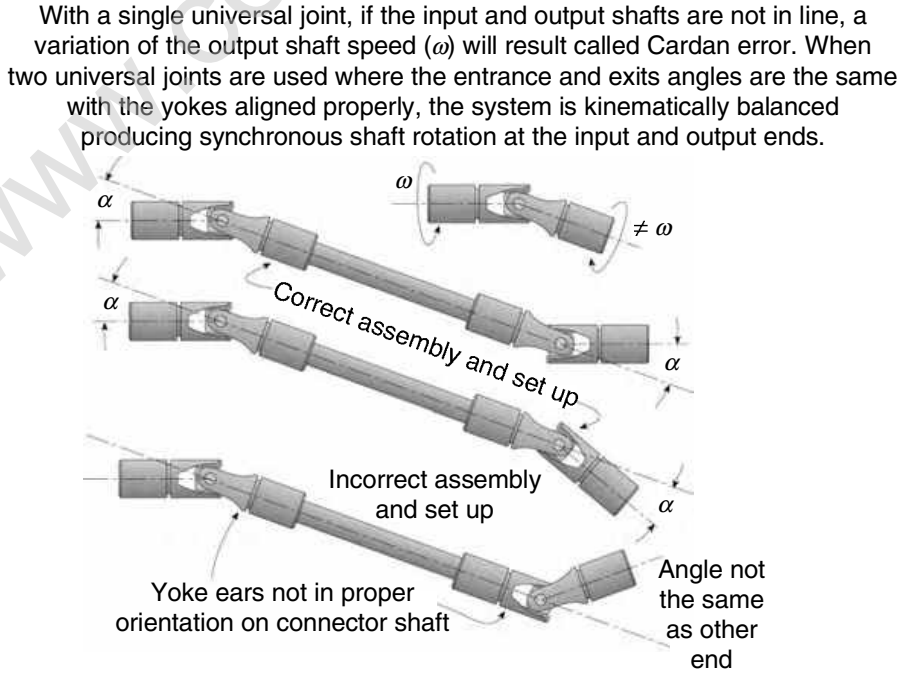


FIGURE 4.13 Universal joint basics.

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FIGURE 4.14 Flexible link coupling. (Courtesy of Eaton Corporation, Airflex Division, Cleveland, OH under license from Dr. Ing. Geislinger & Company, Salzburg, Austria. With permission.)

- Capacity: up to 1100 hp/100 rpm.
- Maximum recommended speed: to 1800 rpm.
- Shaft bores: up to 20 in.
- Shaft spacing: close coupled or 100 mm spacer with certain designs.



FIGURE 4.15 Leaf spring coupling. (Courtesy of Eaton Corporation, Airflex Division, Cleveland, OH. With permission.)

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FIGURE 4.16 Pin drive type coupling. (Courtesy of David Brown Gear Industries, Agincourt, Ontario, Canada. With permission.)

- Special designs and considerations: An axial "fixation" device can be installed to prevent any axial movement if desired. Different designs can accommodate unidirectional or bidirectional rotation.

Advantages:

- No lubrication required

Disadvantages:

- Limited axial movement
- Limited misalignment capabilities (can accept pure angular misalignment only)

4.4.1.6 Leaf Spring

This coupling employs a series of radially positioned sets of leaf springs attached to an outer drive member and indexed into axial grooves in the inner drive member. The chamber around each spring set is filled with oil. When the spring pack is deflected, damping occurs as the oil flows from one side of the spring pack to the other.

- Capacity: up to 15,000 hp/100 rpm
- Maximum recommended speed: 3600 rpm
- Shaft bores: up to 12 in.
- Shaft spacing: up to 40 in.
- Special designs and considerations: Designed primarily for diesel and reciprocating machines. Capable of transmitting shock torque values substantially higher than other couplings until springs reach their maximum allowable angular movement where the radial stiffness increases substantially. Various spring stiffnesses can be installed in each size coupling to properly match the torsional requirements to the drive system.

Advantages:

- Torsionally soft with good damping characteristics
- Freedom of axial shaft movement

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Disadvantages:

- Requires lubricant for damping
- Temperature limitations due to lubricant
- Torsional characteristics change drastically with loss of oil

4.4.1.7 Pin Drive

A series of metal pins with leaf springs are placed near the outer diameter where they engage into a series of holes bored into both shaft hubs. Some pin designs consist of a pack of flat springs with cylindrical keepers at each end that act as the flexing element in the coupling design. The spring sets can swivel in the pin connection to allow movement across the width of the spring set.

- Capacity: up to 3800 hp at 100 rpm
- Maximum recommended speed: to 4000 rpm
- Shaft bores: to 13 in.
- Shaft spacing: close coupled (1⁄8 to 1⁄2 in.).
- Special designs and considerations: Drive pins can be fabricated to accommodate various torsional flexibility requirements and are indexed into oil impregnated bronze bushings in the coupling hubs.

Advantages:

- Can accommodate up to 1/2 in. of axial displacement
- No lubrication required

Disadvantages:

- Limited offset misalignment capability

4.4.1.8 Elastomeric Couplings

A wide variety of design variations that employ an elastomeric medium to transmit torque and accommodate misalignment as illustrated in Figure 4.17. Most of these couplings are torsionally "soft" to absorb high starting torques or shock loads.

- Capacity: up to 67,000 hp/100 rpm but varies widely with design
- Maximum recommended speed: approximately 5000 rpm (varies widely with design)
- Shaft bores: up to 30 in.
- Shaft spacing: up to 100 in. (varies widely)
- Special designs and considerations: A considerable amount of inventiveness and ingenuity has been applied to this type of coupling design through the years as evidenced by the large array of design variations. The elastomeric medium is generally natural or synthetic rubber, urethane, nylon, teflon, or oil-impregnated bronze. As the elastomer is markedly softer than the hubs and solid-driving elements (wedges, pins, jaws, etc.), wear is minimal and replacement of the elastomer itself is all that is usually needed for periodic servicing.

Advantages:

- Minimal wear in coupling
- Acts as vibration damper and isolator

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FIGURE 4.17 Elastomeric couplings. [Courtesy of (a and b) Lovejoy Corporation, Downers Grove, IL; (c) Holset Engineering Company, Cincinnati, OH; (d) T.B. Wood's Sons, Chambersburg, PA. With permission.]

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FIGURE 4.17 (continued)



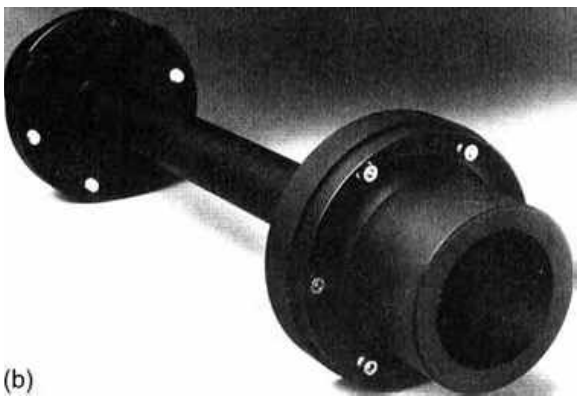


FIGURE 4.18 Contoured diaphragm coupling: (a) cutaway view of the flexible disk at one end of the coupling; (b) entire coupling. (Courtesy of Kopflex Corporation, Baltimore, MD. With permission.)

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- Acts as electrical shaft current insulator in some designs
- Torsionally "soft"
- Accepts some axial movement and dampens axial vibration
- No lubrication required

Disadvantages:

- Speed limited due to distortion of elastomer from high centrifugal forces, causing imbalance
- Deterioration of elastomer from temperature, oxidation of rubber, corrosive attack from undesirable environment
- Potential safety hazard if elastomeric member releases from drive elements
- Some designs may cause undesirable axial forces
- Heat generated from cyclic flexing of elastomer

4.4.2 Metallic Membrane/Disk-Type Coupling Designs

4.4.2.1 Diaphragm Couplings

Transmission of power occurs through two flexible metal diaphragms, each bolted to the outer rim of the shaft hubs and connected via a spacer tube. Misalignment and axial displacement is accomplished by flexing of the diaphragm members.

- Capacity: up to 30,000 hp
- Maximum recommended speed: up to 30,000 rpm
- Shaft bores: up to 7 in.
- Shaft spacing: 2 to 200 in.
- Special designs and considerations: Metal diaphragm couplings are a highly reliable drive component when operated within their rated conditions. Exceeding the maximum allowable angular or parallel misalignment values or axial spacing will eventually result in disc failure. As the diaphragm is, in effect a spring, considerations must be given to the axial spring rate and vibration characteristics to insure that the diaphragm coupling natural frequency does not match rotating speeds or harmonics in the drive system.

Advantages:

- Excellent balance characteristics
- No lubrication required
- Low coupling weight and bending forces on shafts when operated within alignment limits
- Accepts high temperature environment

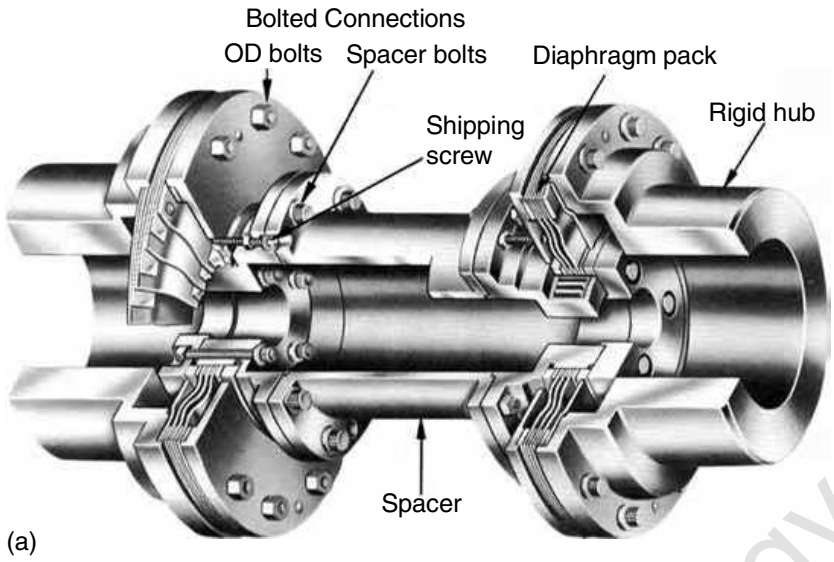
Disadvantages:

- Limited axial displacement and oscillation
- Proper shaft spacing requirements are generally more stringent than other coupling types
- Excessive misalignment will transmit high loads to shafting

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(a)



(b)



(c)

FIGURE 4.19 Flexible disk couplings. [Courtesy of (a) Zurn Industries, Erie, PA; (b) Thomas-Rexnord, Warren, PA; (c) Schmidt Couplings, Cincinnati, OH. With permission.]

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FIGURE 4.20 Rigid coupling. (Courtesy of Browning Mfg., Maysville, KY. With permission.)

4.4.2.2 Flexible Disc Couplings

The flexible disc coupling is very similar in design principles to the diaphragm coupling with the exception that multiple, thinner discs or a noncircular flexing member is used as the flexing element instead of circular, contoured diaphragm elements.

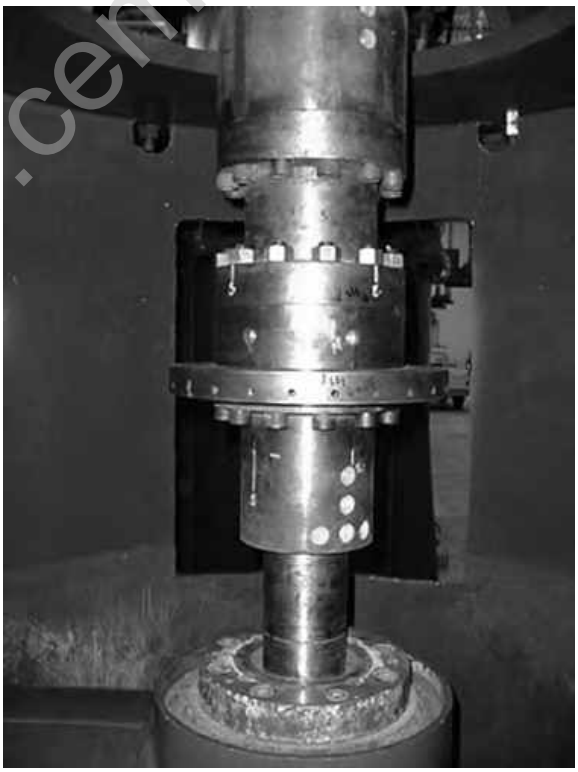


FIGURE 4.21 Rigid coupling on vertical pump.

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- Capacity: up to 65,000 hp/100 rpm
- Maximum recommended speed: up to 30,000 rpm
- Shaft bores: to 12 in.
- Shaft spacing: to 200 in.
- Special designs and considerations: It is important to note that two disc packs (or diaphragms) are needed to accommodate parallel misalignment whereas a single disc can only handle pure angular misalignment. Convolutions in the discs provide a linear stiffness vs. deflection characteristics as opposed to flat disc profiles. Once again, coupling axial resonance information must be known to prevent problems where a match may occur with machinery running speeds, higher order harmonics, or subsynchronous forcing mechanisms (oil whirl, looseness of bearing housings, clearance induced whirls, etc.).

Advantages and disadvantages:

- Same as diaphragm couplings

4.5 RIGID COUPLING DESIGN

Well before flexible couplings came into existence, rigid couplings were used to connect two (or more) shafts together. Although flexible couplings are used on the vast majority of rotating machinery drive systems today, rigid couplings still have their place and are frequently used on systems where very little misalignment occurs and in situations, where high horsepowers are transmitted from shaft to shaft, or in vertical pump applications where one of the drive motor bearings is carrying the weight (thrust) of the armature and pump rotors.

It is important to recognize that when two shafts are connected together with rigid couplings, the two separate shafts have effectively become one continuous shaft. Therefore the ''misalignment'' tolerances for rigid couplings are the same tolerances that apply for acceptable runout conditions on a single shaft as discussed in Chapter 6. There are two classic rigid coupling alignment techniques—the 16-point and 20-point methods and they are discussed in Chapter 7 but other techniques have also been successfully employed on rigid couplings. It is highly recommended that rigid couplings be disconnected when taking alignment measurements between the two shafts as explained in Chapter 1 and Chapter 2.

4.6 FLEXIBLE COUPLING LUBRICATION

The majority of mechanically flexible couplings require lubrication. There are basically two methods used to lubricate couplings: single charge or continuous feed. Greases are generally used in single charge lubricated couplings and the type is generally specified by the coupling manufacturer.

Problems that can occur in greased packed couplings are:

1. Loss of lubricant from leakage at: lube seals, shaft keyways, mating flange faces, or lubricant filler plugs.
2. Excessive heat generated in the coupling from an insufficient amount of lubricant, excessive misalignment, or poor heat dissipation inside the coupling shroud, which reduces viscosity and accelerates oxidation.
3. Improper lubricant.
4. The centrifugal forces generated in the coupling can be high enough to separate greases into oils and soaps.

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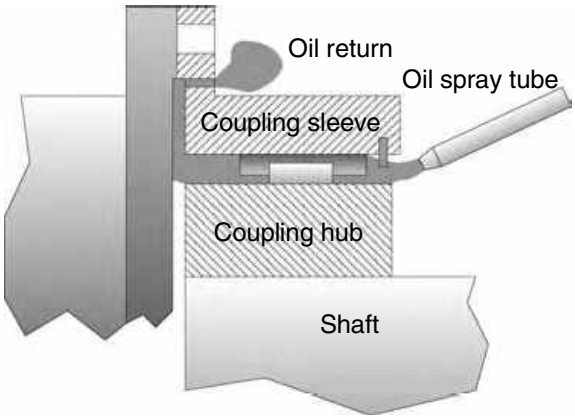


FIGURE 4.22 Continuous oil feed systems for a gear coupling.

As soaps have a higher specific gravity than oil, it will eventually collect where the force is the highest (namely where the gear teeth are located), causing a buildup of sludge.

Periodically inspect the inside of the coupling guard and directly under the coupling to see if any leakage is occurring. If so, do not continue to add more grease because the oil usually leaks out and the soaps continue to build up. Thoroughly clean the coupling, replace the seals and gaskets, and replenish with the correct kind and amount of lubricant, preferably a "coupling grease" in which the oil and soap have the same specific gravity so they do not separate under centrifugal force.

Continuous feed lubrication systems generally use the same lube oil as the bearings and spray tubes are positioned to inject a directed stream of oil into the coupling as shown in Figure 4.22.

In addition to supplying lubricant to the coupling, a continuous supply of oil acts as an excellent heat transfer agent, maintaining a relatively stable temperature in the coupling. However contaminants in the oil, particularly water (which often condenses in lube oil tanks) or corrosive process gases carried over from the inboard oil seals on compressors, can damage the coupling in time. Stainless steel lube oil piping, condensate and particulate matter removal with lube oil centrifuges, 5–10 μm filters, and entrained gas venting systems will alleviate many of these problems.

4.7 COUPLING INSTALLATION

Once a flexible coupling has been selected for a specific service, the next important step is proper installation. It is quite easy to destroy an expensive coupling assembly in short order due to sloppy shaft fits, incorrect key dimensions, improperly measured shaft diameters, storage in a corrosive environment, and so on. After the coupling has been uncrated, the following steps should be performed before installation is even attempted.

1. Insure that the correct type of coupling was ordered and all the parts are with it (bolts, spacer spool, hubs, cover, gaskets, etc.). Clean any and all protective coatings from the surfaces. Thoroughly inspect each component for damage.
2. Physically measure all the dimensions against the coupling drawing and parts listings paying particular attention to coupling hub bores, keyway dimensions, and spool length.

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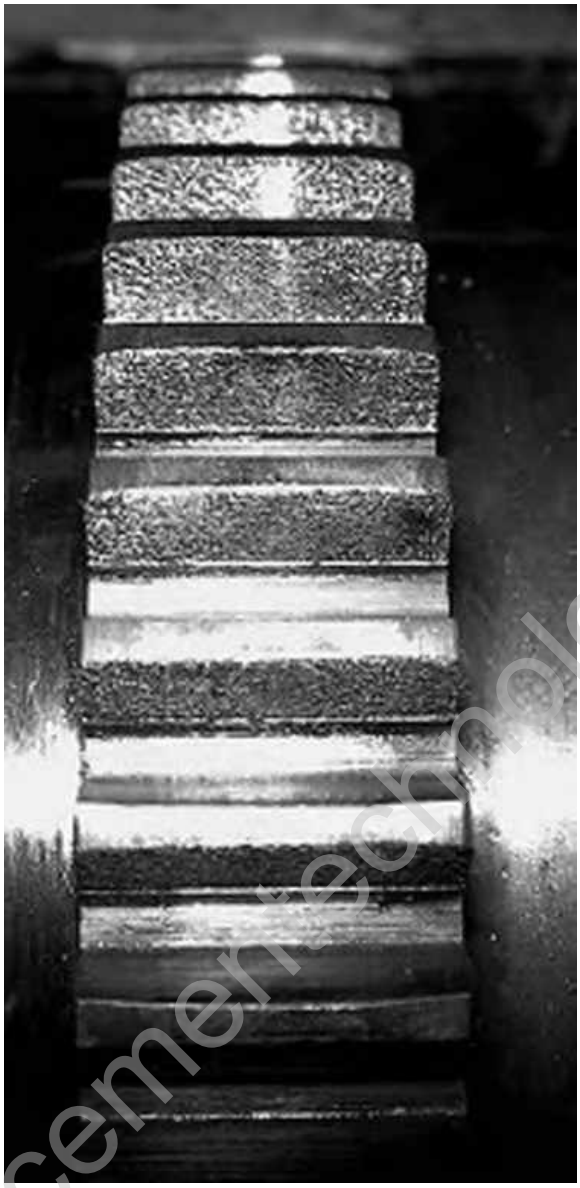


FIGURE 4.23 Pitting across entire surface of gear teeth from operating with no lubricant.

3. Measure the shafts where the coupling is going to be installed. Measure the outside diameters of each shaft, the keyway widths and depths, and the distance between the shafts, insuring that each shaft has been placed in its normal axial position during operation as shown in Figure 4.24 through Figure 4.26.
4. If possible, assemble the entire coupling before it is placed on the shaft checking for proper gear tooth clearances, elastomeric member fits, bolt hole diameter fits, and clearances.

4.8 COUPLING HUB ATTACHMENT METHODS

There are a variety of methods employed to attach the coupling hubs to a shaft, each one having its advantages and disadvantages. Recommended guidelines for installing these various shafts to coupling hub fits are outlined ahead and should be followed to insure a proper fit to prevent slippage or unwanted shaft fretting. Shaft fretting occurs when a coupling hub is

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FIGURE 4.24 Measuring bore of coupling hub with a snap gauge.

loose on its shaft and the oscillatory rocking motions of the hub cause pitting on the mating surfaces of the shaft and the coupling hub.

Coupling hub to shaft fits are classified as follows:

- Straight bore, sliding clearance with keyways
- Straight bore, interference fit with keyways
- Splined shaft with end lock nut
- Tapered bore, interference fit with keyways
- Keyless taper bore
- Locking taper cone

4.9 KEYS AND KEYWAYS

A large percentage of shafts employ one or more keys to prevent the coupling hub from rotating on the shaft as the rotational force is applied. The American National Standards



FIGURE 4.25 Using an outside micrometer to measure the snap gauge.

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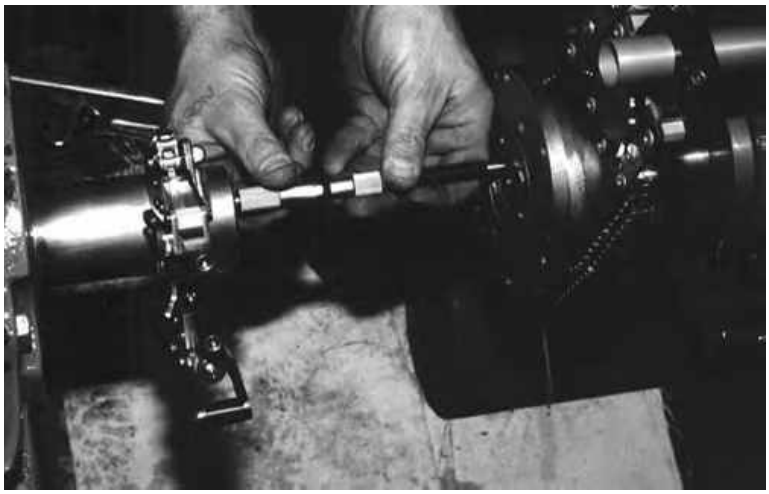


FIGURE 4.26 Using an inside micrometer to measure the distance between the shaft ends.

Institute (ANSI) has set up design guidelines for proper shaft sizes to key sizes and these are shown in Table 4.1.

4.9.1 Types of Keys

There are three different classes of key fits:

- Class 1—side and top clearance, relatively loose fit, only applies to parallel keys (see Table 4.2)

TABLE 4.1
Key and Keyway Sizes for Various Shaft Diameters

Nominal Shaft Diameter		Nominal Key Size Height (H)			Nominal Keyseat Depth (H/2)	
Over	To (inclusive)	Width (W)	Square	Rectangular	Square	Rectangular
0.31250	0.43750	0.09375	0.09375	—	0.04688	—
0.43750	0.56250	0.12500	0.12500	0.09375	0.06250	0.04688
0.56250	0.87500	0.18750	0.18750	0.12500	0.09375	0.06250
0.87500	1.25000	0.25000	0.25000	0.18750	0.12500	0.09375
1.25000	1.37500	0.31250	0.31250	0.25000	0.15625	0.12500
1.37500	1.75000	0.37500	0.37500	0.25000	0.18750	0.12500
1.75000	2.25000	0.50000	0.50000	0.37500	0.25000	0.18750
2.25000	2.75000	0.62500	0.62500	0.43750	0.31250	0.21875
2.75000	3.25000	0.75000	0.75000	0.50000	0.37500	0.25000
3.25000	3.75000	0.87500	0.87500	0.62500	0.43750	0.31250
3.75000	4.50000	1.00000	1.00000	0.75000	0.50000	0.37500
4.50000	5.50000	1.25000	1.25000	0.87500	0.62500	0.43750
5.50000	6.50000	1.50000	1.50000	1.00000	0.75000	0.50000
6.50000	7.50000	1.75000	1.75000	1.50000	0.87500	0.75000
7.50000	9.00000	2.00000	2.00000	1.50000	1.00000	0.75000
9.00000	11.00000	2.50000	2.50000	1.75000	1.25000	0.87500

All dimensions are given in inches. Square keys preferred for shaft dimensions above line ... rectangular keys below.

Source: *Machinery's Handbook*, 21st ed., Industrial Press, New York, NY, 1980.

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TABLE 4.2
Key Fits

Key Width		Side Fit Width Tolerance			Top and Bottom Fit Depth Tolerance			
Over	To (inclusive)	Key	Keyseat	Fit Range	Key	Shaft Keyseat	Hub Keyseat	Fit Range
Type of Key square								
—	0.500	+0 –2	+2 –0	4 CL 0	+0 –2	+0 –15	+10 –0	32 CL 5 CL
0.500	0.750	+0 –2	+3 –0	5 CL 0	+0 –2	+0 –15	+10 –0	32 CL 5 CL
0.750	1.000	+0 –3	+3 –0	6 CL 0	+0 –3	+0 –15	+10 –0	33 CL 5 CL
1.000	1.500	+0 –3	+4 –0	7 CL 0	+0 –3	+0 –15	+10 –0	33 CL 5 CL
1.500	2.500	+0 –4	+4 –0	8 CL 0	+0 –4	+0 –15	+10 –0	34 CL 5 CL
2.500	3.500	+0 –6	+4 –0	10 CL 0	+0 –6	+0 –15	+10 –0	34 CL 5 CL
Type of Key rectangular								
—	0.500	+0 –3	+2 –0	5 CL 0	+0 –3	+0 –15	+10 –0	33 CL 5 CL
0.500	0.750	+0 –3	+3 –0	6 CL 0	+0 –3	+0 –15	+10 –0	33 CL 5 CL
0.750	1.000	+0 –4	+3 –0	7 CL 0	+0 –4	+0 –15	+10 –0	34 CL 5 CL
1.000	1.500	+0 –4	+4 –0	8 CL 0	+0 –4	+0 –15	+10 –0	34 CL 5 CL
1.500	3.000	+0 –5	+4 –0	9 CL 0	+0 –5	+0 –15	+10 –0	35 CL 5 CL
3.000	4.000	+0 –6	+4 –0	10 CL 0	+0 –6	+0 –15	+10 –0	36 CL 5 CL
4.000	6.000	+0 –8	+4 –0	12 CL 0	+0 –8	+0 –15	+10 –0	38 CL 5 CL
6.000	7.000	+0 –13	+4 –0	17 CL 0	+0 –13	+0 –15	+10 –0	43 CL 5 CL

Key dimensions are in inches. Fit dimensions are in mils (1 mil = 0.001in.)

- Class 2—minimum to possible interference, relatively tight fit
- Class 3—interference with degree of interference not standardized

Shafts with two keyways can present another type of problem from improper machining of the coupling hub or shaft keyways as illustrated in Figure 4.28. If the offset gap is larger than 10% of the total key width, it is recommended that the shaft or coupling hub be reworked for improved indexing of the keys in their mating keyways.

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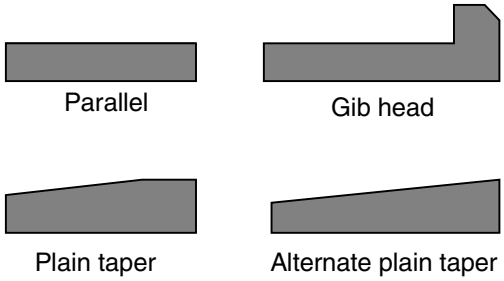


FIGURE 4.27 Types of keys.

4.9.2 Straight Bore—Sliding Clearance with Keyways

This method of shaft and coupling hub fit is used extensively in the industry and provides the easiest and the quickest installation of the coupling hub. However shaft fretting is likely to occur with this sort of arrangement due to the clearance between the coupling hub and the shaft. The best arrangement is to have a light press fit (i.e., 0 to 1 mil of clearance). To prevent the coupling hub from sliding axially along the shaft, set screws are usually locked against the key as shown in Figure 4.29. A light press fit often requires that the coupling hub be heated slightly (around 200°F–250°F) to thermally expand the hub to prevent from beating the hub onto the shaft. To remove the hub, a mechanical puller is usually required. If the clearance is excessive (e.g., 2 mils or more), when the set screw is tightened, the coupling hub is drawn off center, causing a runout condition. Some precautions to consider when installing a straight bore with sliding clearance coupling hub:

- Insure clearance does not exceed 1 mil for shaft diameters up to 6 in.
- Remove any burrs; clean components carefully before installation.
- If hub sticks part way on, remove it and find the problem. Do not attempt to drive it on further with a hammer.
- Install keys before the coupling is placed on the shaft.

4.9.3 Straight Bore—Interference Fit with Keyways

To insure that a coupling hub can be removed once it is "shrunk" onto a shaft, proper interference fits must be adhered to. The general guideline for straight bore interference fits is found in Table 4.4.

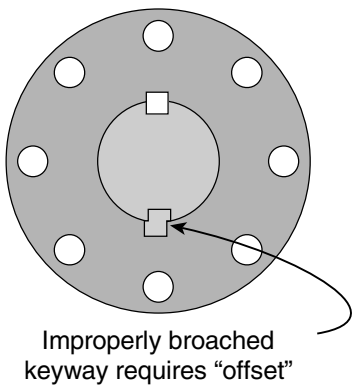


FIGURE 4.28 Offset key.

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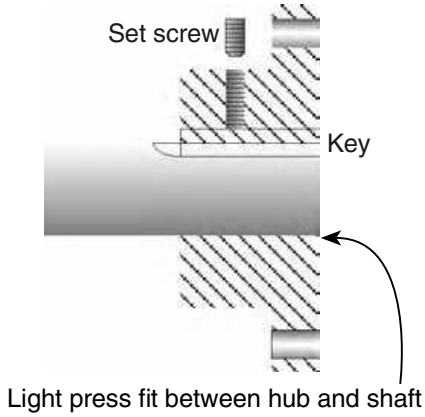


FIGURE 4.29 Straight bore with clearance fit.

The coupling is installed by heating in a clean oil bath or oven to approximately 200°F–250°F and in some cases cooling the shaft simultaneously with a dry ice pack. Do not exceed 300°F or use any direct heat such as propane or oxyacetylene torches to expand the hub as the material properties of the shaft can be altered by direct, high-temperature concentration. Once the interference fit has been determined by measuring the shaft diameter and coupling hub bores, the temperature increase needed to expand the coupling hub to exceed the shaft diameter by 2 mils (to allow for a slide on fit) is found in Equation 4.1.

$$\Delta T = \frac{i}{\alpha(d - 0.002)} \tag{4.1}$$

where T is the rise in coupling hub temperature from ambient (°F), i is the interference fit (in mils), α is the coefficient of thermal expansion (in./in. °F), and d is the coupling hub bore diameter (in.).

TABLE 4.3
Set Screw to Key Size Directory

Nominal Shaft Diameter		Key Width–Set Screw Sizes	
Over	To (inclusive)	Key Width	Set Screw Diameter
0.31250	0.43750	0.09375	No. 10
0.43750	0.56250	0.12500	No. 10
0.56250	0.87500	0.18750	0.25000
0.87500	1.25000	0.25000	0.31250
1.25000	1.37500	0.31250	0.37500
1.37500	1.75000	0.37500	0.37500
1.75000	2.25000	0.50000	0.50000
2.25000	2.75000	0.62500	0.50000
2.75000	3.25000	0.75000	0.62500
3.25000	3.75000	0.87500	0.75000
3.75000	4.50000	1.00000	0.75000
4.50000	5.50000	1.25000	0.87500
5.50000	6.50000	1.50000	1.00000

All dimensions are given in inches.

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TABLE 4.4
Guidelines for Shrink Fits on Shafts

Shaft Diameter (in.)	Interference Fit (in.)
1/2 to 2	0.0005 to 0.0015 in.
2 to 6	0.005 to 0.0020 in.
6 and up	0.0001 to 0.00035 inches per inch of shaft diameter

The removal of the coupling hub is accomplished by pulling the hub off the shaft with an acceptable puller mechanism and at times cooling the shaft with a dry ice pack. Shrink fit coupling hubs should always have fine threaded puller holes (preferably four) in the end of the coupling as shown in Figure 4.31.

Bearing-type pullers that "push" the hub off from the backside are not recommended as there is a great possibility that the puller can twist or pitch slightly preventing a straight axial draw on the hub. For larger shaft diameters with tight interference fits, it may be necessary to apply gentle heating to the coupling for removal.

4.9.4 Splined Shaft with End Lock Nut or Locking Plate

A splined shaft and coupling arrangement is shown in Figure 4.32. There should be a slight interference fit (0.0005 in.) to prevent backlash or rocking of the hub on the shaft.

4.9.5 Tapered Bore—Interference Fit with Keyways

Tapered shaft ends are generally used where high torques and speeds are experienced on rotating machinery, necessitating a tight coupling hub to shaft fit up. The shaft end is tapered to provide an easier job of removing the coupling hub.

The degree of taper on a shaft end is usually expressed in terms of its slope (inches per foot). The amount of interference fit is expressed in inches per inch of shaft diameter. The general rule for interference fits for this type of shaft arrangement is 1 mil per inch of shaft diameter. The distance a coupling hub must travel axially along a shaft past the point where the hub is just touching the shaft at ambient temperatures is found in the following equation:

$$\mathrm{HT} = \frac{12I}{\mathrm{ST}} \tag{4.2}$$

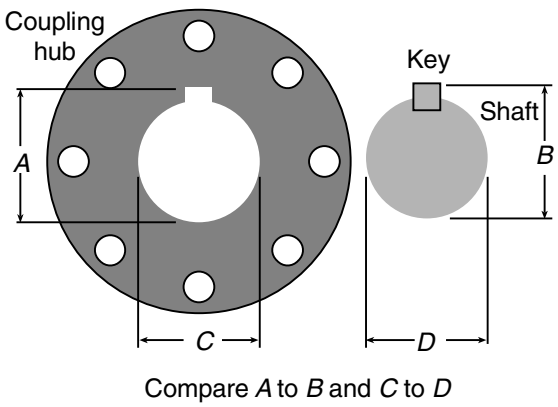


FIGURE 4.30 Measuring the shaft and coupling hub for proper fits.

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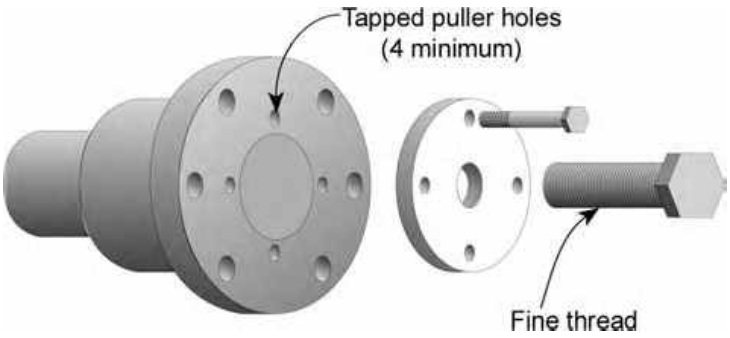


FIGURE 4.31 Coupling hub puller.

where HT is the distance the coupling hub must travel to provide an interference fit equal I (mils), *I* is the interference fit (mils), and ST is shaft taper (in./ft).

The procedure for mounting a tapered coupling hub with keys:

1. Mount bracket firmly to coupling hub and slide hub onto shaft end to lightly seat the hub against the shaft. Insure all surfaces are clean.
2. Measure hub travel gap HT with feeler gauges and lock nut down against bar. Use Equation 3.2 to determine the correct axial travel needed to obtain the required interference fit onto the shaft.
3. Remove the coupling hub and puller assembly and place in an oven or hot, clean oil bath to desired differential temperature. Refer to Equation 4.1 to determine the required temperature rise to expand the coupling hub.
4. Set key in keyway and insure all contact surfaces are clean and burr free.
5. Carefully slide the heated coupling hub onto the shaft until the center measurement bolt touches the shaft end and hold in place until hub has cooled sufficiently.

4.9.6 Coupling Hub to Shaft Surface Contact

One extremely important and often overlooked consideration when working with tapered shafts and coupling hubs is the amount of surface contact between the shaft and hub. Due to

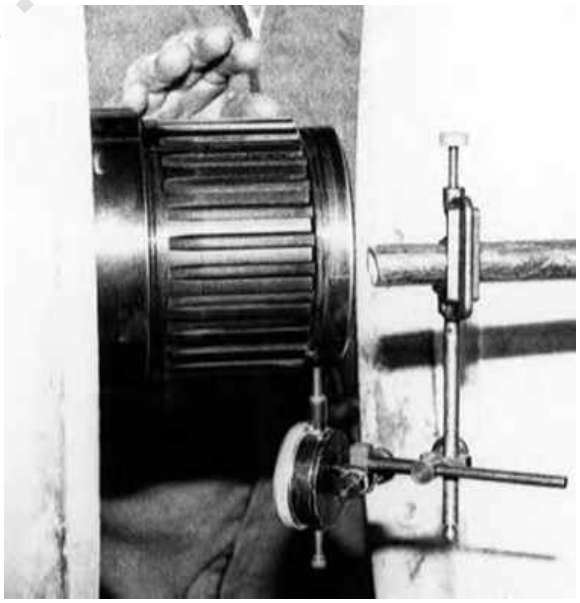


FIGURE 4.32 Splined shaft end.

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slight machining inaccuracies, coupling hubs may not fully contact the shaft resulting in a poor fit when the hub is shrunk or pressed on in final assembly.

To check the surface contact, apply a thin coat of Prussian blue paste to the inner bore of the coupling hub with your finger or a soft cloth. Slide the coupling hub axially over the tapered shaft end until contact is made and rotate the coupling hub about 15° to transfer the paste to the shaft. Draw the coupling hub off and observe the amount of Prussian blue paste that transferred from the hub to the shaft (not how much blue came off the inside bore of the coupling hub). If there is not at least 80% contact, the fit is not acceptable. If the bore discrepancies are slight, it is possible to lap the surfaces with a fine grit lapping compound. Apply the compound around the entire surface contact area of the tapered shaft end, lightly pushing the coupling hub up the taper and rotating the coupling hub alternately clockwise and counterclockwise through a 45° arc. Check the surface contact after 10 or 12 lapping rotations. Continue until surface contact is acceptable. However if a "ridge" begins to develop on the shaft taper before good surface contact is made, start making preparations for machining of the shaft and the coupling hub. It is better to bite the bullet now than try to heat the hub and put it on only to find out that it does not go on all the way or to pick up the pieces of a split coupling hub after the unit ran for a short period of time.

4.9.7 Keyless Taper Bores

After working with shafts having keyways to prevent slippage of the coupling hub on the shaft, it seems very unnerving to consider attaching coupling hubs to shafts with no keys. Keyless shaft fits are quite reliable and installing hubs by hydraulic expansion methods proves to be fairly easy if installation and removal steps are carefully adhered to. As the interference fits are usually "tighter" than found on straight bores or tapered and keyed systems, determining a proper interference fit will be reviewed.

4.9.8 Proper Interference Fit for Hydraulically Installed Coupling Hubs

The purposes of interference fits are twofold:

1. Prevent fretting corrosion that occurs from small amounts of movement between the shaft and the coupling hub during rotation.
2. Prevent the hub from slipping on the shaft when the maximum amount of torque is experienced during a start up or during high running loads.

For rotating shafts, the relation between torque, horsepower, and speed can be expressed as

$$T = \frac{63{,}000P}{n} \tag{4.3}$$

where P is the horsepower, T is the torque (in lbs), and n is the shaft speed (rpm).

The maximum amount of shearing stress in a rotating shaft occurs in the outer fibers (i.e., the fibers at the outside diameter) and is expressed as

$$\tau_{\max} = \frac{Tr}{J} = \frac{16T}{\pi d^3} \tag{4.4}$$

where $\tau_{\max}$ is the maximum shear stress (lb/in.), T is the torque (in lbs), r is the radius (in.), d is the diameter (in.), and J is the polar moment of inertia, $J = \pi r^4/2 = \pi d^4/32$.

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TABLE 4.5
Allowable Torsional Stresses for Shafts

AISI #	τ_{max} Allowable Torsional Stress (psi)
1040	5000
4140	10000
4340	11000

The accepted "safe allowable" torsional stress for the three commonly used types of carbon steel for shafting can be found in Table 4.5.

Therefore the torsional holding requirement for applied torques is expressed as

$$T = \frac{\tau_{\max} \pi d^3}{16} \tag{4.5}$$

The amount of torque needed to cause a press fit hub to slip on its shaft is given by

$$T = \frac{\mu \pi p L d^2}{2} \tag{4.6}$$

The amount of contact pressure between a shaft and a coupling hub is related to the amount of interference and the outside diameters of the shaft and the coupling hub and is expressed as

$$p = \frac{IE(\mathrm{DH}^2 - \mathrm{DS}^2)}{2(\mathrm{DH}^2)(\mathrm{DS})} \tag{4.7}$$

where p is the contact pressure (lbs/in.^{2}), I is the interference fit (mils), E is the modulus of elasticity (30×10^6 lb/in. for carbon steel), DH is the outside diameter of coupling hub (in.), and DS is the outside diameter of shaft (in.).

As the shaft is tapered, dimension DS should be taken on the largest bore diameter on the coupling hub where the contact pressure will be at its minimum value as shown in Figure 4.34.

Therefore to find the proper interference fit between a coupling hub and a tapered shaft to prevent slippage from occurring:

1. Determine the maximum allowable torque value for the shaft diameter and the shaft material.
2. Determine the contact pressure needed to prevent slippage from occurring based on the maximum allowable torque value found in step 1.
3. Calculate the required interference fit (solve for p in Equation 4.6 and i in Equation 4.7).

4.9.9 Installation of Keyless Coupling Hubs Using Hydraulic Expansion

Installing keyless taper hubs requires some special hydraulic expander and pusher arrangements to install or remove the coupling hub onto the shaft end. Figure 4.35 shows the general arrangement used to expand and push the hub onto the shaft.

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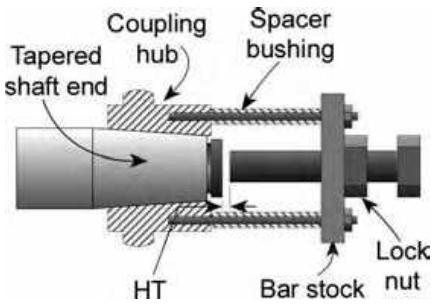


FIGURE 4.33 Measuring tool for proper interference fits on tapered shaft ends.

The procedure for installation of coupling hub using hydraulic expander and pusher assembly:

1. Check for percentage of surface contact between coupling hub and shaft (must have 80% contact or better).
2. Insure all mating surfaces are clean and that oil passageways are open and clean.
3. Install "O" rings and backup rings in coupling hub and shaft insuring that backup ring is on "outside" of "O" ring with respect to hydraulic oil pressure. Lightly oil the "O" rings with hydraulic fluid. Place coupling hub (and hub cover) onto shaft.
4. Install expander pump supply line to shaft end. Install the pusher piston assembly onto the end of the shaft insuring that the piston is drawn back as far as possible. Hook up the expander pump and begin pumping hydraulic oil through supply line to bleed any air from expansion ports and expansion groove in shaft. Once the oil has begun to seep through the coupling hub ends, lightly push the coupling hub against the shaft taper and begin to pump oil into the pusher piston assembly to seat the piston against the coupling hub.
5. Place a dial indicator against the backside of the coupling hub and zero the indicator.
6. Start applying pressure to the pusher piston assembly forcing the hub up the taper (approximately 2000 to 4000 psig).
7. Slowly increase the pressure on the expander pump supply line until the coupling hub begins to move. The hydraulic pressure on the pusher piston assembly will begin to drop off as the hub begins to move. Maintain sufficient pressure on the pusher piston to continue to drive the hub onto the shaft. If the pusher piston pressure drops off considerably when the expansion process is underway, there is a great potential for the "O" rings to "blow out" the ends of the coupling hub immediately seizing the hub to the shaft.

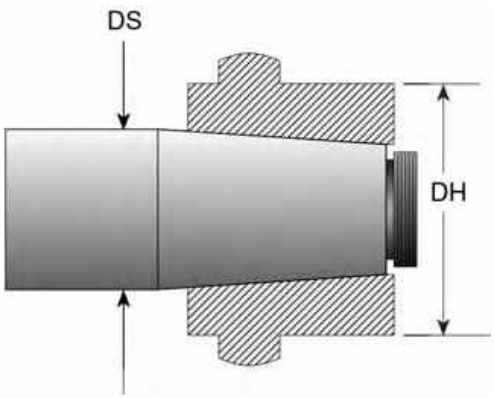


FIGURE 4.34 Shaft and coupling hub outside diameter measurement locations.

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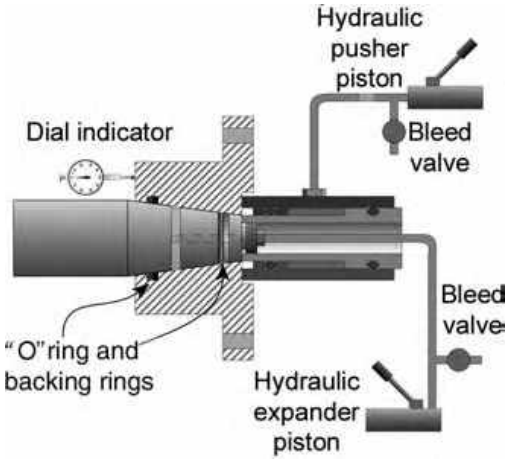


FIGURE 4.35 Hydraulic expander and pusher assembly.

8. Continue forcing the hub up the shaft until the desired amount of hub travel and interference fit is attained. The expansion pressure will have to attain the required holding pressure as defined in Equation 4.7.
9. Once the correct hub travel has been achieved, maintain sufficient hydraulic pressure on the pusher assembly to hold the coupling hub in position and bleed off the pressure in the expansion system. Allow 15–20 min to elapse while bleeding to insure any trapped oil has had a chance to escape before lowering the pusher piston pressure.
10. Remove the pusher and expander assemblies.

The removal of the coupling hub is achieved by reversing the installation process. The key to successful installation is to take your time and not try to push the hub up the shaft end all in one move.

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5 Preliminary Alignment Checks

In Chapter 1, we examined the eight basic steps of aligning rotating machinery. This chapter will cover in detail the tasks identified in step 4, conducting and performing any preliminary checks before starting the alignment. Perhaps the most overlooked step in the process of aligning rotating machinery is this one.

All too often, people who skip this step find themselves having problems in measuring the off-line shaft positions accurately, adding and then removing shim stock several times under the machinery feet, and they frequently find themselves "chasing their tail," trying to reposition the machines laterally several times with marginal or no success. After wasting several hours in their attempt to align the machinery, they realize that something is wrong and they go back to check for many of the problems discussed herein.

In summary, you will be trying to find and correct any problems in the following areas:

- Unstable or deteriorated foundations and base plates
- Damaged or worn components on the rotating machinery (e.g., machine casings, bearings, shafts, seals, couplings)
- Excessive runout conditions (e.g., bent shafts, improperly bored coupling hubs)
- Machine casing to base plate interface problems (e.g., soft foot)
- Excessive piping, ductwork, or conduit forces

Some of the items mentioned above are related very closely to the information given in Chapter 3 on foundations, base plates, sole plates, and piping strain and so it is recommended that you review this chapter. As discussed in Chapter 1, a considerable amount of time can be spent on these preliminary checks and corrections. In fact, I typically spend much more time for performing these tasks it takes to actually align the machinery. Many of the problems may be time consuming, expensive, and difficult to correct. Because of this, there is a great tendency to come up with excuses for not doing it.

5.1 FOUNDATION AND BASE PLATE CHECKS

With respect to the successful long-term operation of the machinery, an outstanding alignment job can quickly deteriorate if the equipment is perched on unstable frames, inertia blocks, or foundations. Chapter 3 discussed about what desired design features should be incorporated into foundations and base plates but these features may not necessarily exist on the machinery that is worked on. Therefore the first place of looking for problems would be in the supporting structure for the machinery. A quick review of Figure 3.19 through Figure 3.25 will show examples of what to check for and correct.

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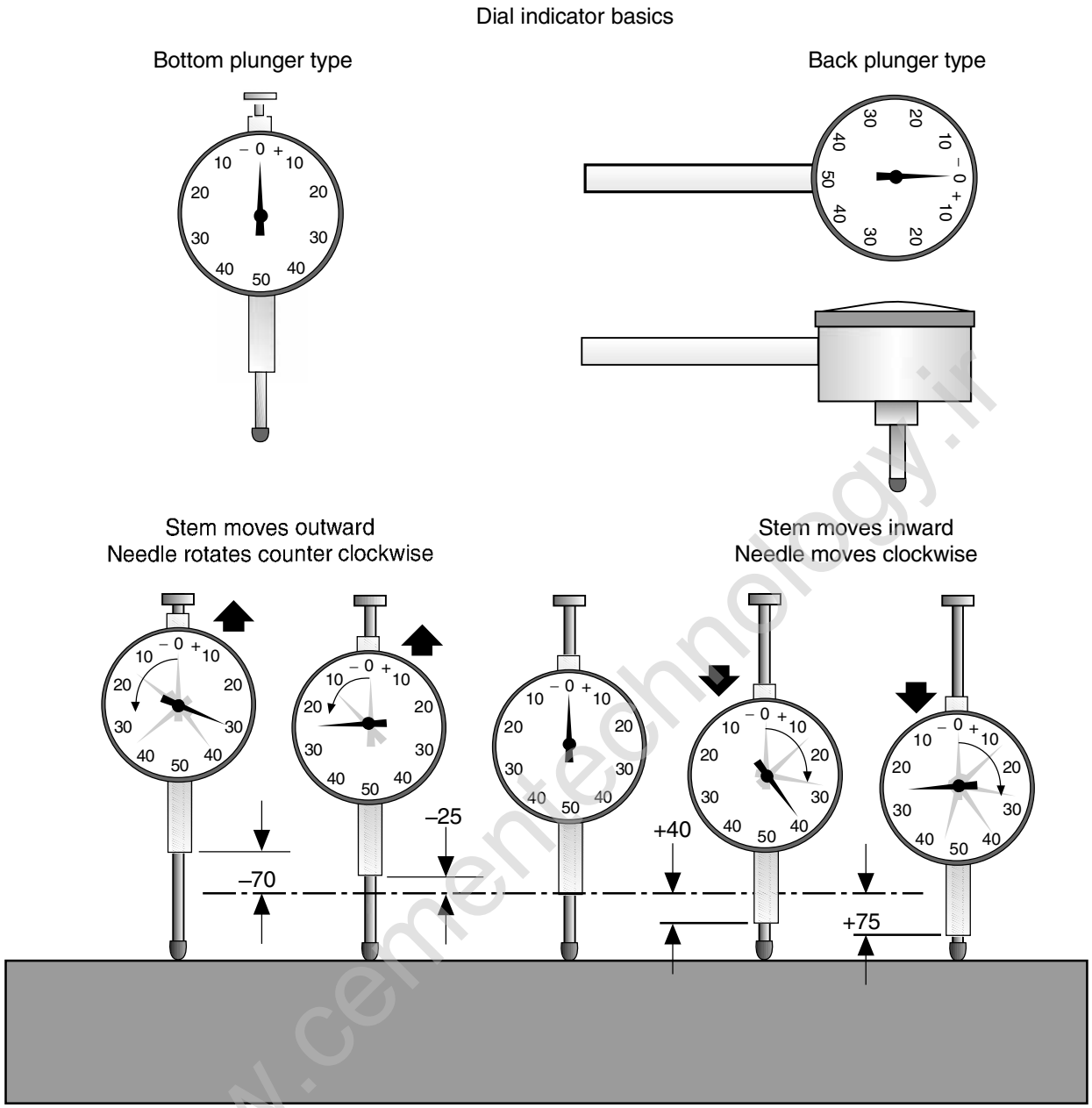


FIGURE 5.1 Dial indicator basic operation.

5.2 DIAL INDICATOR BASICS

Since the use of dial indicators will be discussed frequently in this chapter, Figure 5.1 shows the basic operating principle of this versatile measurement tool. It is highly recommended to get familiar with this device since it will be used for a wide variety of tasks in the overall process of machinery installation, troubleshooting, problem solving, and shaft alignment.

5.3 DAMAGED, WORN, OR IMPROPERLY INSTALLED MACHINERY COMPONENT CHECKS

Every once in a while, you may have the pleasure of installing brand new rotating machinery. If you are in the construction industry that is primarily what you will be doing. However, in a maintenance organization, you will very likely be working with equipment that has been in service for sometime and invariably it is required to find and correct a problem with the

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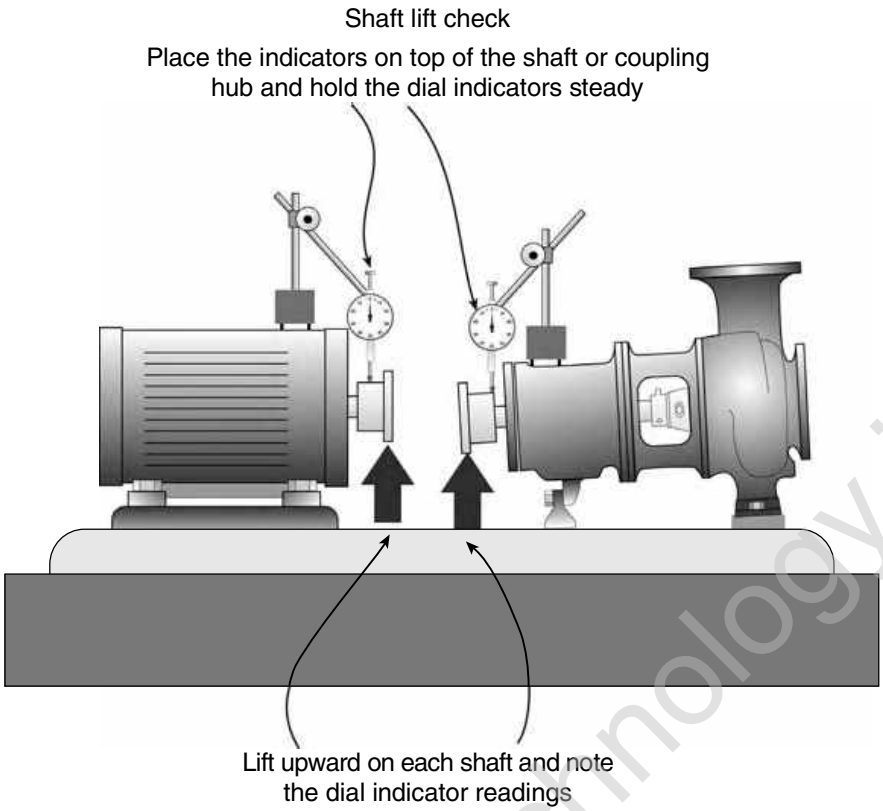


FIGURE 5.2 How to perform a shaft lift check.

equipment. Chapter 1 discussed the four different maintenance philosophies. The ultimate goal of a quality maintenance group is to achieve proactive or prevention maintenance status. The capacity to detect ensuing problems with machinery, stop the damage before it becomes a financial loss to the company, have the capability to quickly detect the problems with the equipment, and engineer the corrective measures to prevent the malady from occurring again is the ultimate goal. Very few people have been able to attain this level of performance. This chapter discusses many of the tasks that make the difference between run-to-failure maintenance and proactive or preventive maintenance.

If machinery has been operating for sometime, the bearings that support the rotor may have sustained some damage and it is suggested that some checks should be made to insure that the bearings are in good working order. One of the simplest tests that can be performed is a shaft "lift check" as shown in Figure 5.2 and Figure 5.3.

Positioning a dial indicator on top of the shaft as close as to get it to the inboard bearing, it is essential to anchor the indicator to a stationary object with a magnetic base or a clamp. Then lifting the shaft upward enough to detect if any motion occurs, but not with so much force as to permanently deform the shaft, can easily happen by using a hydraulic piston, chain hoist, or overhead crane.

If the shaft is supported in rolling element-type bearings as shown in Figure 5.4, the amount of lift on the shaft should be negligible (i.e., 0 to maybe 1 mil). If there is an excess amount of shaft lift with a rolling element bearing, four possible reasons for this is as follows:

1. The inner race of the bearing is loose on the shaft.
2. There is too much clearance between the rolling elements and the inner and outer raceways.
3. The outer race is loose in its housing.
4. A combination of two or more of the items above.

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FIGURE 5.3 Performing a lift check on a pump shaft.

If the inner race is loose on the shaft, the inner race will begin "skidding" on the shaft, eventually damaging the shaft (if it has not already done so). If this condition exists, the machine's running is stopped immediately and the bearing is removed to make a thorough inspection of the shaft, bearing, and bearing housing. The shaft and the bearing have to be replaced.

If there is too much clearance between the rolling elements and the inner and outer raceways, the rollers will begin skidding on the raceways, eventually damaging the bearing (if it has not already done so). If this condition exists, the machine's running is stopped immediately and the bearing is removed to make a thorough inspection of the shaft, bearing, and bearing housing. The shaft and the bearing have to be replaced.

Rolling element bearings began to appear in the early 1900's and are also referred to as antifriction or ball bearings. The bearing consists of an inner race, rolling elements, and an outer race. Sometimes the rolling elements are held in place with a cage assembly. As the shaft turns, a film of lubricant forms between the rolling elements and the raceways. The oil film thickness can range between1 and 3 μm (4 to 12 millionths of an inch) and the oil pressures at the minimum oil film thickness are very high (approximately 40 kpsi). If the oil film breaks down, metal to metal contact between the rolling elements and the raceways can occur causing damage to the bearing. Damage to the rolling elements, raceways, or cage assembly can be detected through vibration analysis.

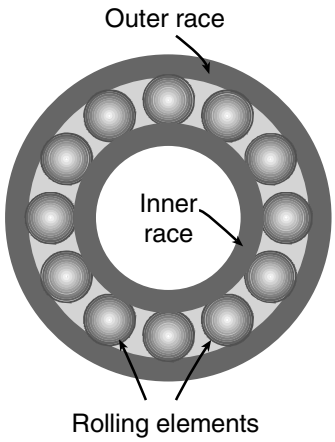


FIGURE 5.4 Rolling element bearing design.

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If there is too much clearance between the outer race and the housing, the outer raceway will begin skidding on inside the housing and eventually damaging the housing (if it has not already done so). If this condition exists, the machine's running is stopped immediately and the bearing is removed to make a thorough inspection of the shaft, bearing, and bearing housing. The bearing housing and machine case have to be replaced.

There are other types of "fixes" possible for items 1 and 3 (i.e., loose inner or outer race) but they are usually not recommended for long-term satisfactory performance. It may be tempting to use epoxy-based adhesives between the raceway and shaft or raceway and housing but that is not the best corrective measure. Sounds like a good idea until you have to remove the bearing at some later date. It may be tempting to sleeve the shaft or the housing but that is also not the best corrective measure. The ability to make a sleeve to achieve the correct interference fits enables to fix the sleeve correctly in the first place. It is recommended to consult the equipment manufacturer for the correct procedure for installing new bearings and the proper type and amount of lubricant to use for that bearing.

If the shaft is supported in sliding-type bearings, the amount of lift on the shaft should be within the acceptable radial bearing clearance range. Figure 5.5 shows the basic operating principle of sliding-type bearings. As noted in Figure 5.5, the "rule of thumb" for radial bearing clearance should be from 3/4 to 2 mils/in. of shaft diameter for oil-lubricated babbit bearings. If the amount of lift is greater than the maximum clearance for that shaft diameter, the bearing should be removed and inspected. With cylindrical sliding-type bearings, another

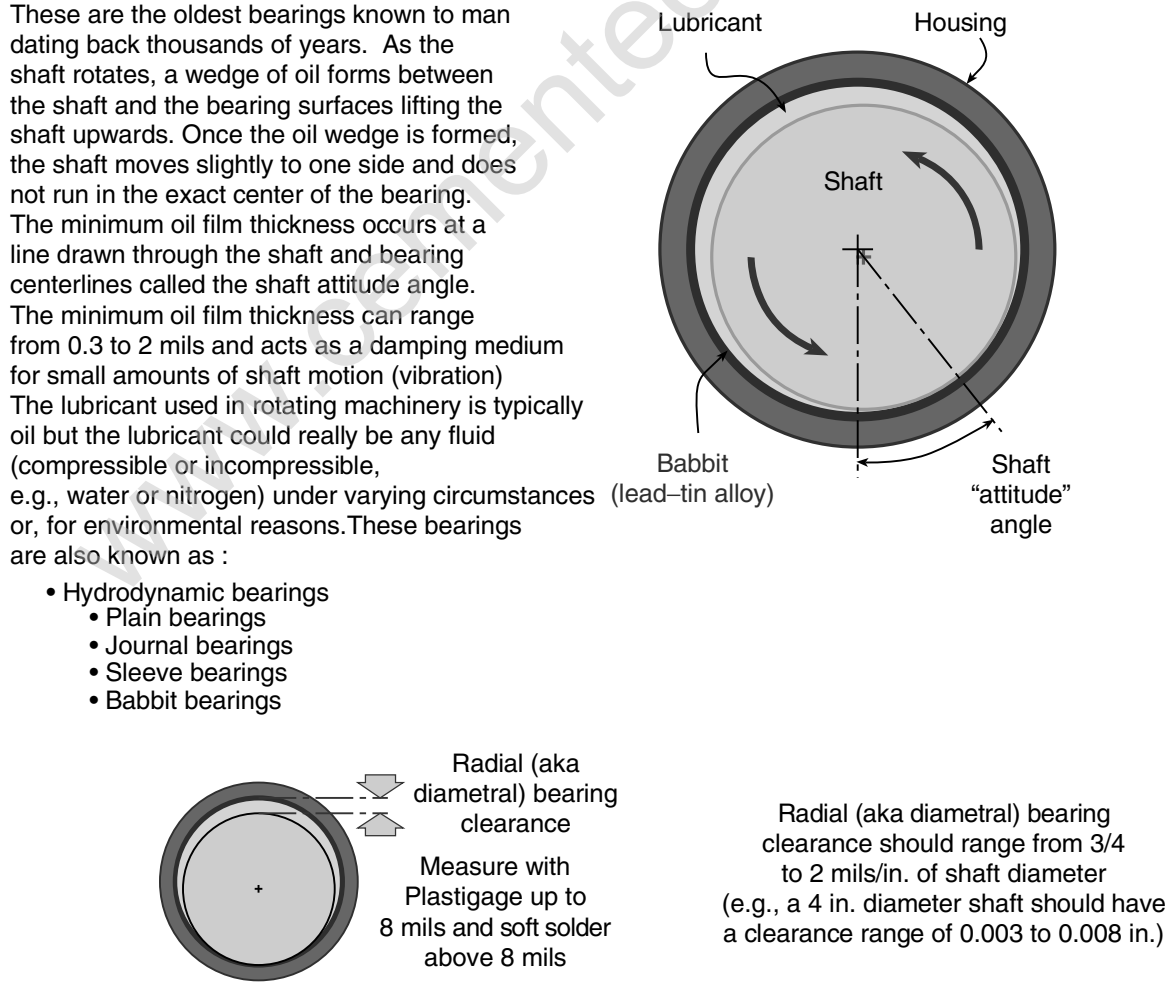


FIGURE 5.5 Sliding bearing design.

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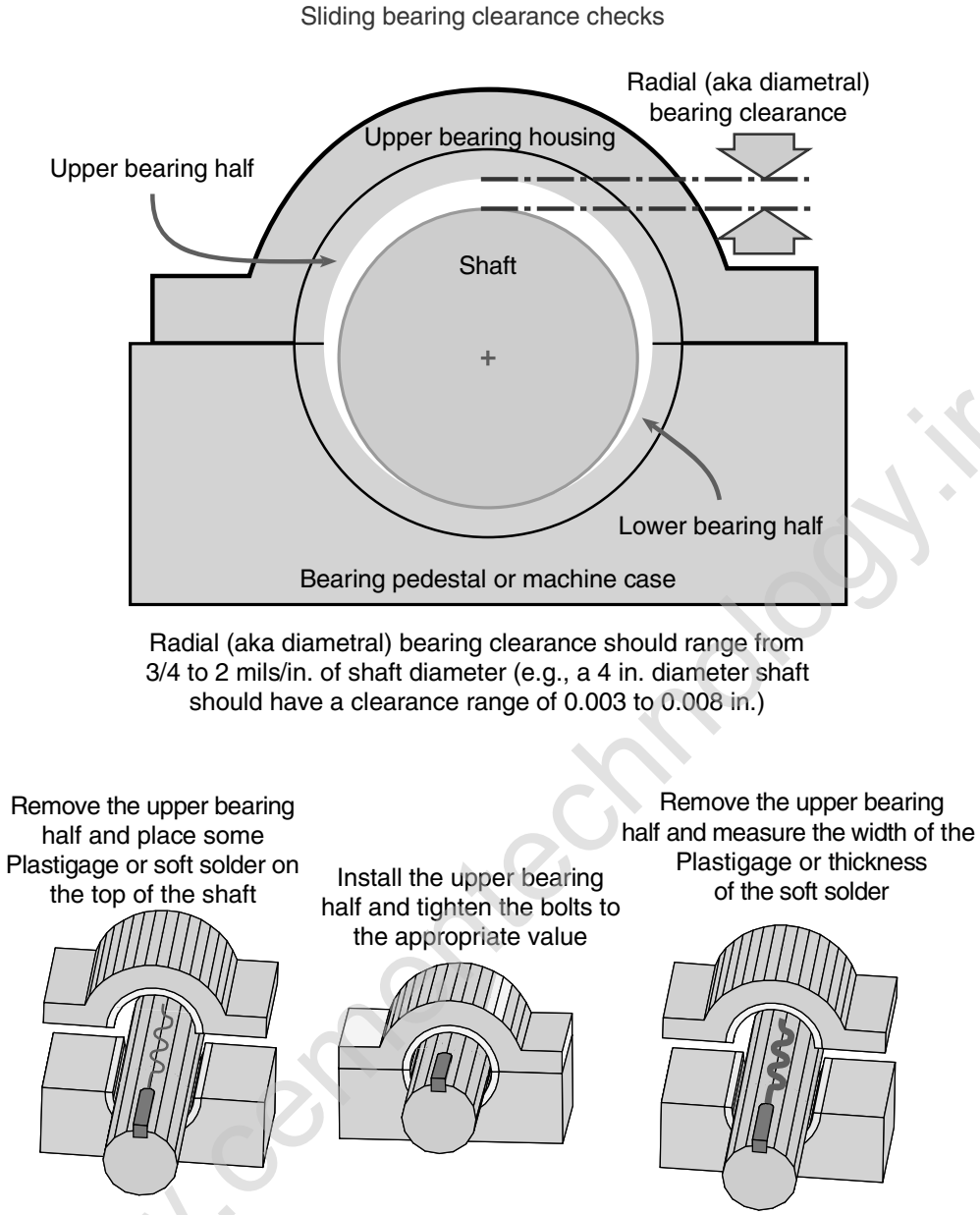


FIGURE 5.6 Sliding bearing clearance checks.

recommended method for checking bearing clearance is given in Figure 5.6. Plastigage or soft solder can be used for the clearance check. In addition to the clearance check, a "tilt and twist" check should be made as shown in Figure 5.7. The tilt and twist checks are performed to determine if the centerline of the bore of the bearing is parallel to the centerline of rotation of the shaft in the up and down (tilt) and side-to-side (twist) direction. An alternative check is a "blue check" where a thin coat of Prussian bluing is applied to the lower half of the bearing. The bearing is then installed into its lower hemisphere, and the shaft is lowered onto the bearing and then lifted to allow the removal of the lower half of the bearing. The bearing is then examined to determine how much of the bluing is transferred to the shaft to insure that there is at least 80% shaft to bearing contact. Figure 5.8 shows a bearing in the process of blue checked. Bearing in mind that blue checking will determine if there is a tilt problem but not necessarily a twist problem.

Some bearings are spherically seated in their housing to hopefully compensate for any tilt and twist conditions. Figure 5.9 shows an arrangement for a large steam turbine bearing

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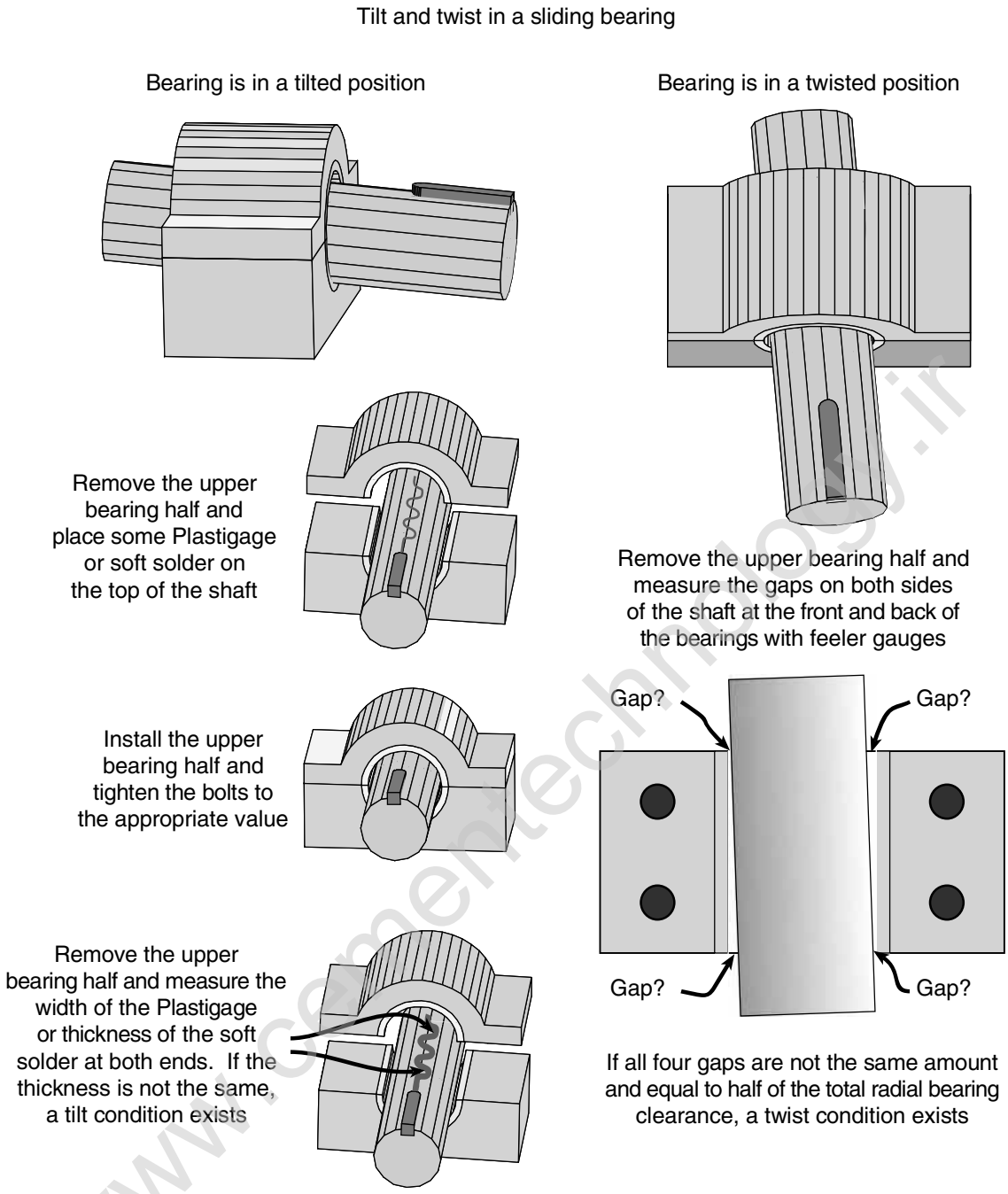


FIGURE 5.7 Finding a tilt and twist problem in a sliding bearings.



FIGURE 5.8 Checking contact on a sliding bearing with bluing.

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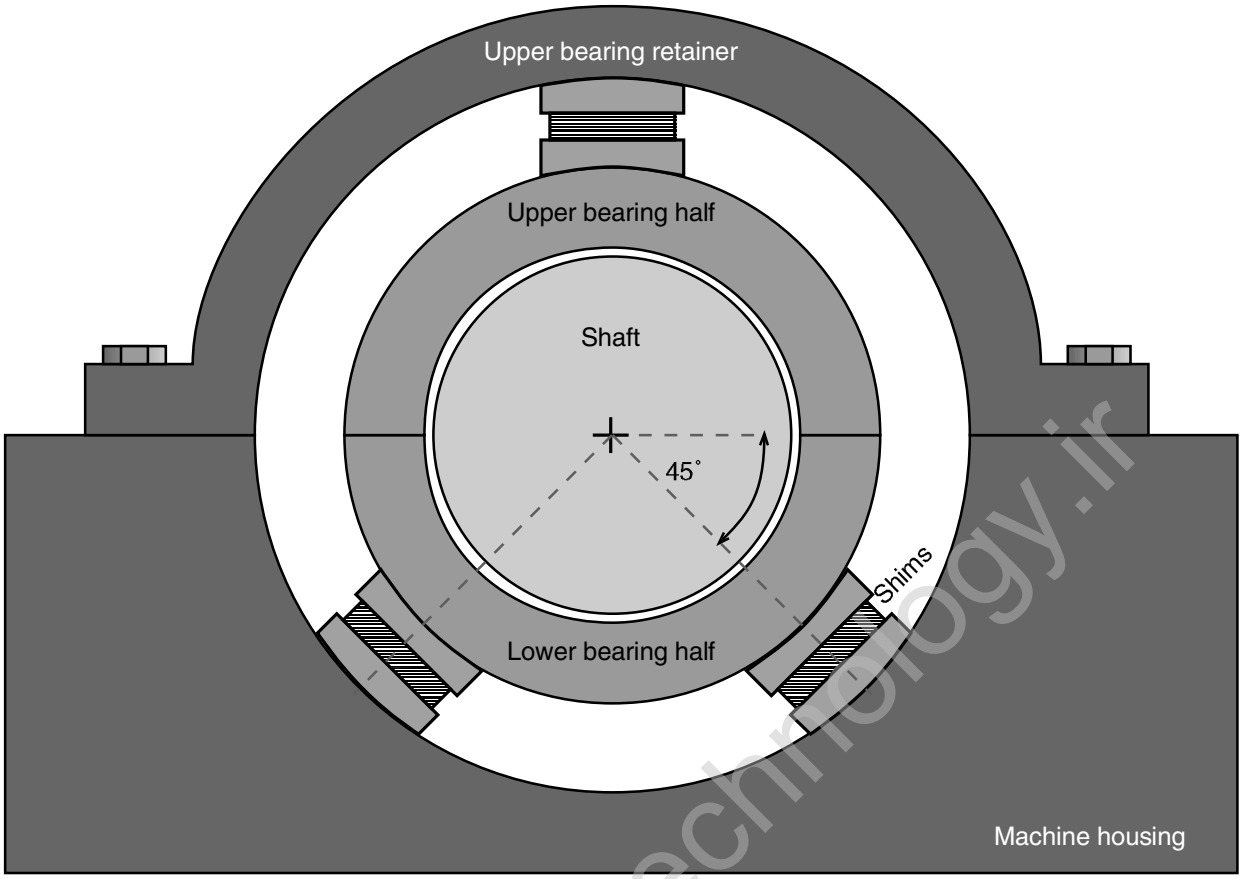


FIGURE 5.9 Spherically seated sliding bearing on adjustable support blocks.

where the bearing assembly is held in position with three support blocks. Shims can be added or removed from each support block to position the bearing in the vertical and lateral directions and to allow for a small amount of clearance to enable the bearing and pads to pivot in the spherically shaped housing.

A shaft supported in sliding-type radial bearings can float axially and therefore requires some device (or force) to maintain its correct axial position. In electric motors supported in sliding-type radial bearings, the electromagnetic force centers the armature in the housing. This is often referred to as magnetic center or "mag center." To find mag center, it is necessary to disconnect the coupling between the motor and what it is driving, and to start the motor up and run it "solo." When the motor has attained its normal operating speed, it is advised to scribe a line with a felt tip pen or soap stone onto the rotating shaft (care should be taken while doing this) near the inboard bearing using the seal housing or another stationary object on the motor as a reference point. The motor is de-energized (i.e., shut down) and to stop the armature from rotating. After properly safety tagging the breaker, the armature is rotated by hand and as it is rotating, the armature in the axial direction is pushed or pulled until the scribed line that was made on the shaft aligns with the selected stationary reference. This is where the armature wants to run under normal operating conditions. This will become important later on during the alignment process to get the correct axial position between the shafts.

Other rotating machines supported in sliding-type radial bearings do not have an electromagnetic force to center the shaft like motors do. So a thrust bearing is used. There are three major components to a thrust bearing:

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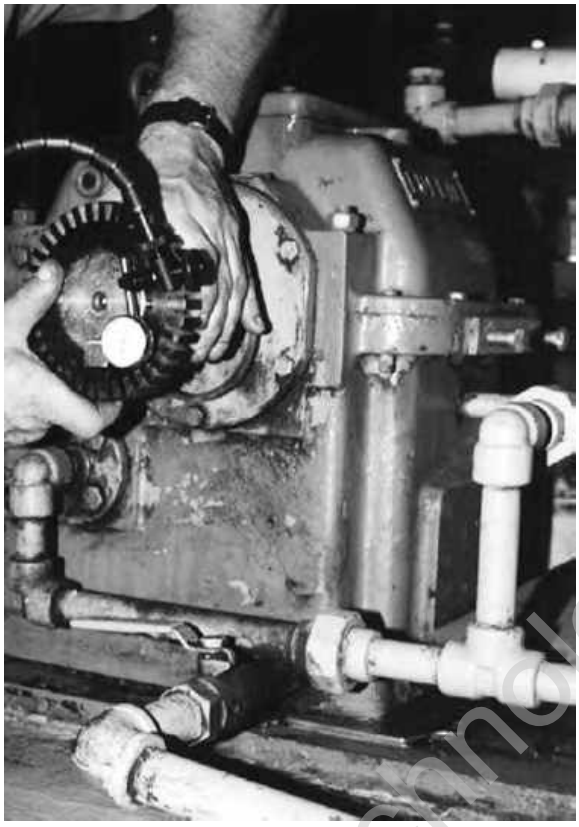


FIGURE 5.10 Performing a thrust-bearing clearance check, step 1.

1. The thrust runner or thrust disk: This is a disk permanently attached to the shaft.
2. An active thrust bearing: This is the bearing that the thrust runner typically seats against while it is operating. A film of lubricant prevents the thrust runner from wearing the thrust bearing out.
3. An inactive thrust bearing: It looks the same as the active thrust bearing and under normal operation the thrust runner never seats against it, since most machinery wants to thrust in one direction only. However, if the shaft wants to move in the opposite direction, this bearing will stop the shaft before it contacts something stationary in the machine.

Some of the most catastrophic failures of machinery have occurred due to a thrust-bearing failure or due to improperly installing and setting the correct thrust-bearing clearance. To check this clearance, it is essential to position a dial indicator on the end of the shaft (or coupling hub) and anchor the indicator to a stationary object with a magnetic base or a clamp. The shaft toward the operator is pulled until it seats against one of the thrust bearings and zero the indicator as shown in Figure 5.10. The shaft away from the operator is pushed until it seats against the other thrust bearing as shown in Figure 5.11. This is repeated for two or three times and the amount of indicator travel each time is observed. Typically, the thrust-bearing clearance is somewhere between 15 and 40 mils but it is recommended to consult the equipment manufacturer for the correct thrust-bearing clearance and the procedure to correct it if it is not within the recommended range.

Figure 5.12 shows a lower half of a tilt pad-type sliding bearing. Notice that there is some evidence of wear in the pads. With tilt pad-type sliding bearings, a mandrel (a cylindrical bar machined to the same outside diameter as the shaft) is used in concert with a dial indicator for the clearance check. This can be done on a table and the procedure is the same as the shaft lift check except that the mandrel is placed in a vertical position, the assembled bearing is slid

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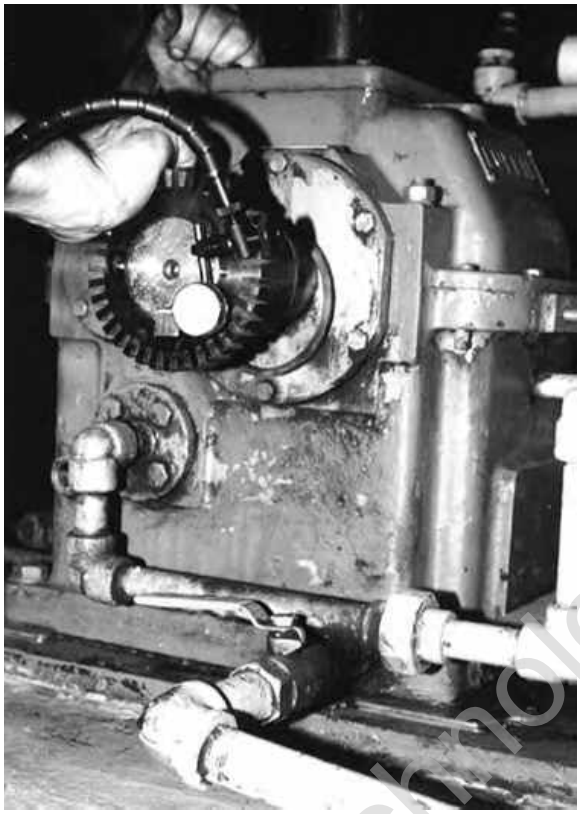


FIGURE 5.11 Performing a thrust-bearing clearance check, step 2.

over the mandrel, and a dial indicator is positioned against the bearing and then anchored to the table. The bearing is then moved toward and away from the dial indicator to measure the clearance.

The radial bearing clearances mentioned above are not for all types of sliding-type bearings. Water-lubricated "cutlass"-type bearings have greater clearances. New cutlass bearings typically have clearance between 15 and 20 mils and maximum clearances typically should not exceed 80 mils. With these types of bearings, clearance checks can be made with feeler gauges at four points around the circumference of the bearing. A cutlass bearing with excessive clearance on a dredge drive shaft is shown in Figure 5.13. Figure 5.14 shows the feeler gauge readings on that bearing, indicating an excessive amount of clearance. Notice that there seems to be a twist problem with this bearing. The condition and fit of bearings is extremely

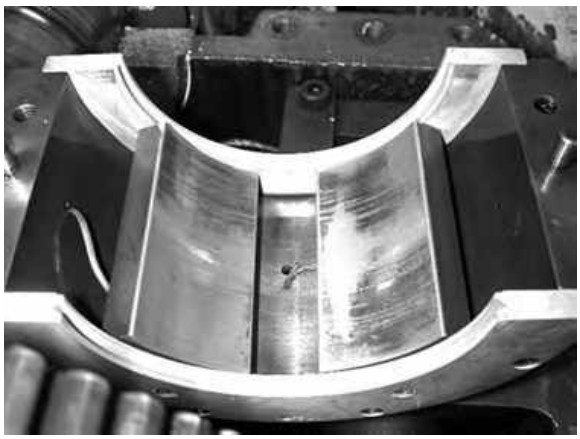


FIGURE 5.12 Lower half of a tilt pad-type sliding bearing.

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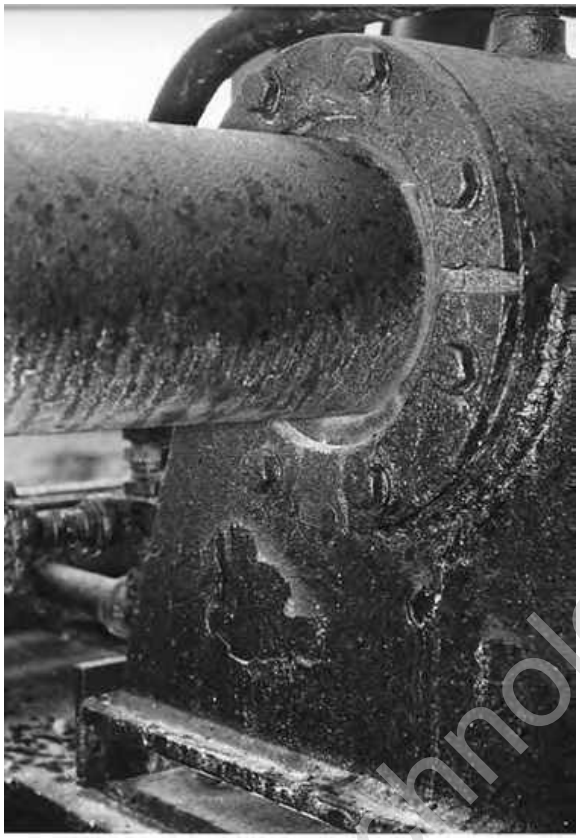


FIGURE 5.13 Cutlass-type water bearing with excessive clearance.

important in rotating machinery and should be one of the first items that should be checked before alignment but there are other components that need to be examined for mechanical integrity.

In a large majority of rotating machinery, some type of fluid or gas is present inside the machine case and unless it is sealed properly, the fluid or gas will leak out. The lubricant in the bearings can also leak out if proper sealing is not achieved. Sensory clues are the first sign of trouble with seals. If one can notice the seeping out of oil from the machine case under the shaft or oil on the base plate, it is a sign that leakage is occurring.

Air or steam leaks frequently can be audibly detected (sound). Frequently high-pressure leaks can be outside the range of human detection and may require leak detection sensors and

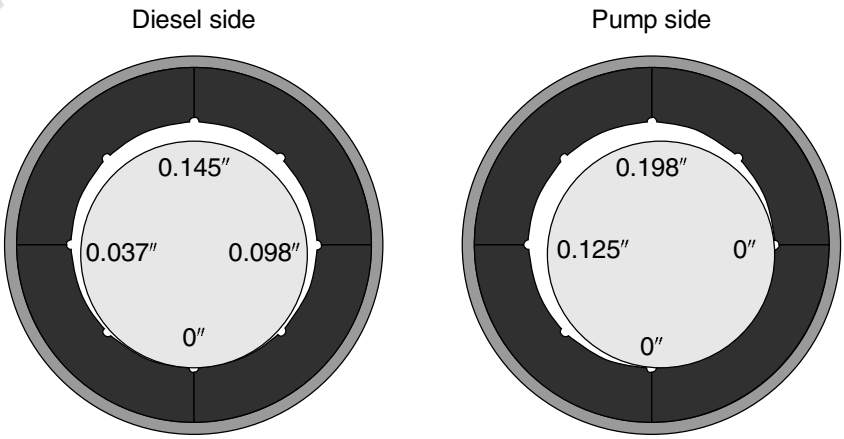


FIGURE 5.14 Measured clearances on above bearing.

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equipment to be located. The typical range for hearing for humans is from 20 to 20,000 Hz (1 Hz = 1 cycle per second).

To contain compressible or incompressible fluids inside a machine case, there are four most commonly used types of shaft seals: labyrinth, lip, mechanical, and packing. Figure 5.15 shows the basic design of each of these seals. Although these seals are shown with an overhung centrifugal pump, they are used on a wide variety of rotating machinery. Figure 5.15 illustrates how to prevent the fluid that is pumped from leaking out along the shaft by employing the two most common sealing methods of either mechanical packing or mechanical seals.

Mechanical packing consists of flexible rings that look like braided rope with a square cross section. The packing rings are inserted into a cylindrical cavity surrounding the shaft called the ''stuffing box'' as shown on the top part of the shaft in Figure 5.15. Frequently three packing rings are inserted. A device called the ''lantern ring'' contains three more packing rings. The packing gland is then bolted to the pump housing to compress the packing rings to provide the seal. The mechanical packing was never meant to provide a perfect seal and some leakage should occur just to keep a film of fluid between the packing rings and the shaft, otherwise the packing will eventually wear away the shaft. Although it is not shown in the diagram, a sacrificial shaft sleeve is often installed in this area so that the packing does not damage the shaft itself. A hole is bored through to the stuffing box area where the lantern ring is located. This is used to inject a fluid (frequently the process fluid itself) into the stuffing box to provide the lubricating film between the packing and the shaft or shaft sleeve.

Mechanical seals consist of a stationary seal ring and a rotating seal ring as shown on the bottom part of the shaft in Figure 5.15. There are a variety of designs incorporating one or two sealing ring sets. The mating faces of the sealing rings are ground flat and have a very

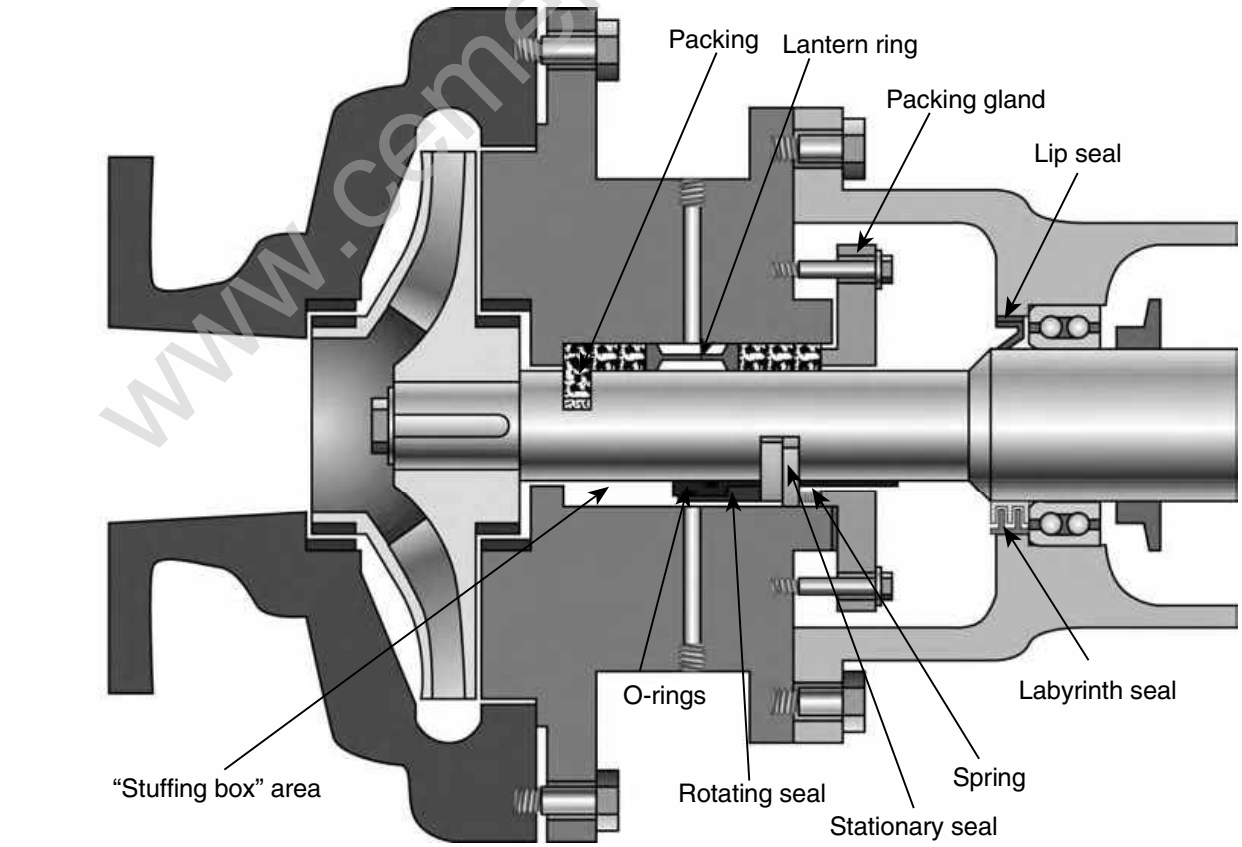


FIGURE 5.15 (**See color insert following page 322.**) Four commonly used seals in rotating machinery.

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smooth surface finish (frequently 4 rms or better). To keep the faces together a spring (or springs) is employed. Again, this needs to be a film of fluid between the rotating and stationary seal faces. The design premise of a mechanical seal is that as the process fluid attempts to traverse across the mating seal faces, by the time it gets to the outside world, the fluid has vaporized. For processes where the vapors could be harmful, double seals are typically used and a nonvolatile "barrier fluid" is injected into the stuffing box area.

To insure successful sealing capabilities, it is important to insure that the shaft is centered in the stuffing box or seal housing. For the pump shown in Figure 5.15, assuming the bolt-hole patterns for the bearing housing and the stuffing box housing were machined concentric to the bearing bores and that the shaft is not bent, the clearance between the outside of the shaft and the bore of the stuffing box should be the same all the way around. Usually this is true, but not always. In other pump designs, however, the bearing housing is not part of concentrically machined housings and may not "automatically" be centered to the bore of the stuffing box. In either case, it may be wise to check the concentricity of the stuffing box.

To determine if the shaft is centered, the distance from the outside surface of the shaft to the inside bore of the seal housing is measured at four points 90° apart. These measurements can be taken in a number of different ways. Feeler gauges, snap gauges, and in some cases, a magnetic base and a dial indicator can be used for the measurements. Figure 5.16 shows a snap gauge that is used to measure the distance between the outside surface of a pump shaft and the bore of the stuffing box where packing is used to seal the water inside the pump. Figure 5.17 shows the initial measurements taken on this pump. The pump shaft is lowered toward the east. On this particular pump, the bearing housing has to be positioned to center the shaft in the stuffing box before aligning the pump to its driver.

In addition to keeping the process fluid from leaking out along the shaft, the bearing lubricant needs to be present in the bearing housing with seals. Figure 5.15 shows two of the



FIGURE 5.16 Snap gauge used to measure stuffing box clearance.

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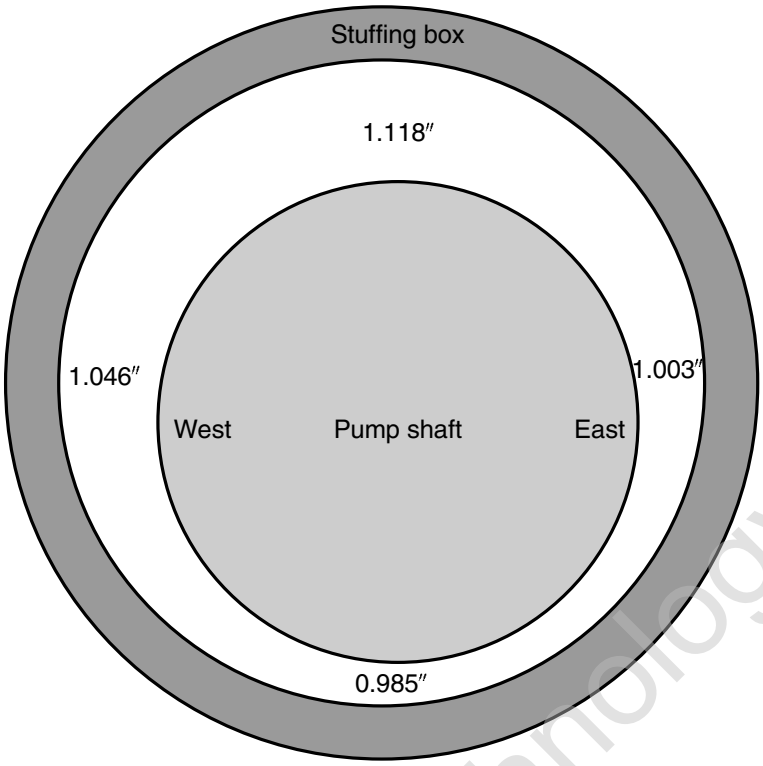


FIGURE 5.17 Stuffing box clearance measured on pump in Figure 5.16.

more commonly used lubricant seals: lip seals and labyrinth seals. Lip seals are frequently made of rubber and can easily be installed backwards if care is not taken. While performing the preliminary checks on a motor and pump drive, it was observed that an oil seal was installed backward on the electric motor as shown in Figure 5.18.

Since rotating machinery is likely to exhibit some vibration during operation, there is a possibility that the integrity of the housing can degrade over moderate to long periods of time. High-stress concentration areas will begin to cyclically fatigue and cracks may begin to form and then propagate. A complete visual inspection of the machine casing and housing may uncover problem areas. Figure 5.19 shows where a crack was found on a machine housing during a visual inspection. Cracks can be very difficult to visually detect without help and die penetrant checks may be warranted.



FIGURE 5.18 Oil seal installed backwards on a motor.

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FIGURE 5.19 Cracked weld on machine housing.

For machinery that has been operating for some period of time, it is also suggested that a visual inspection be made of the coupling (flexible or rigid) for wear or problem areas. Figure 5.20 shows excessively worn teeth on a gear coupling. Elastomeric couplings will degrade rapidly under moderate to severe misalignment conditions. Figure 5.21 shows extreme wear on a new elastomeric coupling which had been subjected to 20+ mils/in. of misalignment that occurred over a period of just 30 weeks of intermittent operation.



FIGURE 5.20 Excessively worn gear coupling.

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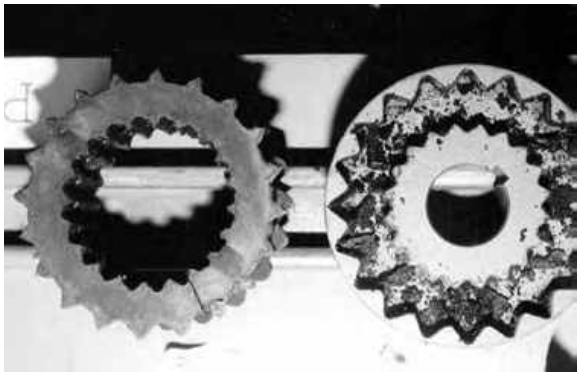


FIGURE 5.21 Excessively worn elastomeric coupling.

5.4 RUNOUT

The term "runout" describes a condition where a rotating object is not concentric or perpendicular with its centerline of rotation. Runout is also referred to as an "off center" or "eccentric" condition and should be one of the first things to check on the machinery that is attempted to be aligned. All rotating machinery shafts, or any device that is attached to a shaft such as coupling hubs, shaft sleeves, impellers, fan blades, armature windings, gears, blades, shrouds, or other types of components rigidly attached to shafts, will exhibit some runout condition. There is runout in just about anything and everything that rotates and it is just a matter of how much runout is present. Some runout can be as small as 10 millionths of an inch or as high as 100 mils or greater.

Runout checks are standard operating procedures when assembling components onto a rotating shaft. First the shaft itself is checked and then each component that is rigidly attached to the shaft is checked for excessive runout conditions. Moderate to excessive runout will cause moderate to excessive vibration in the machinery where the greatest amount of vibration will appear at the running speed of the machine making it appear as an "out-of-balance" condition. In many cases, it is not wise to attach balance correction weights to reduce the vibration. The correction weights may reduce the dynamic forces that are causing the vibration but they will not reduce or remove the eccentricity condition. If an internal rotating part such as a pump impeller has excessive runout, the balance correction weights will reduce the dynamic forces but the eccentric impeller may contact a stationary object inside the pump case, potentially causing a catastrophic failure when the unit is started or later when it reaches operating conditions.

Rotating equipment manufacturers understand this and are very careful to insure that excessive runout does not cause rub conditions. However, the majority of rotating machinery is shipped to its final destination without the power transmission device (e.g., coupling hub or V-belt sheave) attached to the end of its shaft. This is most often installed on the machine at the factory where the equipment will be used in production. All too often, runout checks are not made when these parts are placed at the end of a shaft. The runout problems that occur with the power transmission device (e.g., coupling hub or V-belt sheave) will be discussed in detail.

There are two basic types of runout conditions, radial and face runout. Radial runout quantifies the eccentricity of the outer surface of a shaft or a component rigidly attached to a shaft with respect to the shaft's centerline of rotation. Face runout quantifies the amount of nonperpendicularity that may exist at the end of a shaft or on surfaces of components rigidly attached to a shaft. Runout conditions are typically measured with dial indicators as illustrated in Figure 5.22. Runout checks should also be made at several points along the length of

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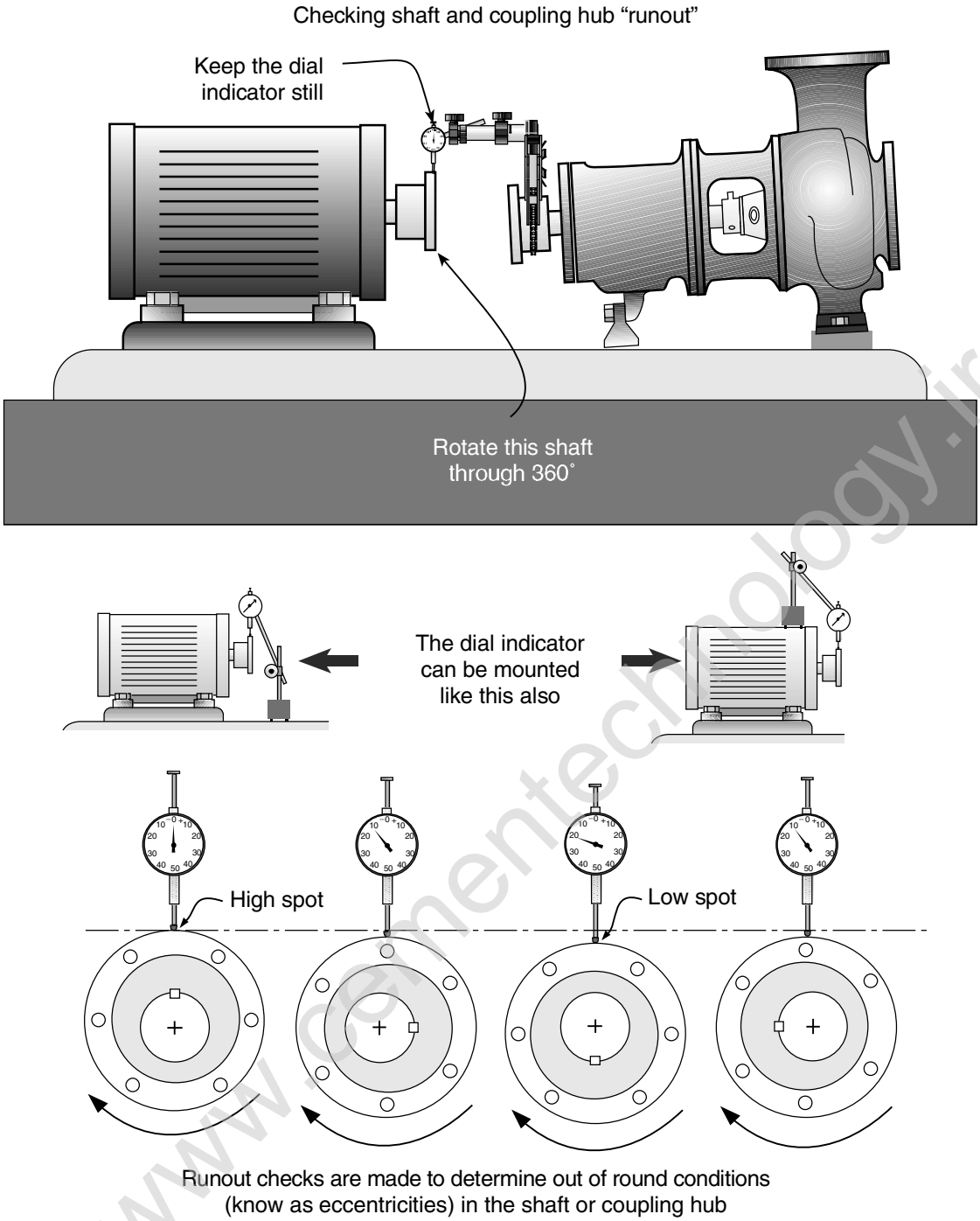


FIGURE 5.22 How to measure runout in a coupling hub.

a rotor. Figure 5.23 illustrates a runout measurement taken on a long drive shaft midway between the two support bearings.

Figure 5.24 and Figure 5.25 show runout checks, made on the coupling hubs, connecting a steam turbine and compressor shaft. Figure 5.26 shows a runout check that is made on another coupling hub for a ''rubber tire''-type coupling. The magnetic base is attached to the shaft on the right while the shaft on the left (where the dial indicator is located) was turned by hand for the check. Figure 5.27 and Figure 5.28 show runout measurement taken on a vertical pump shaft at two different locations along the length of the shaft. There were 12+ mils of runout discovered at the measurement taken in Figure 5.27 and 15+ mils of runout discovered at the measurement taken in Figure 5.28 with the ''high spots'' in the same angular

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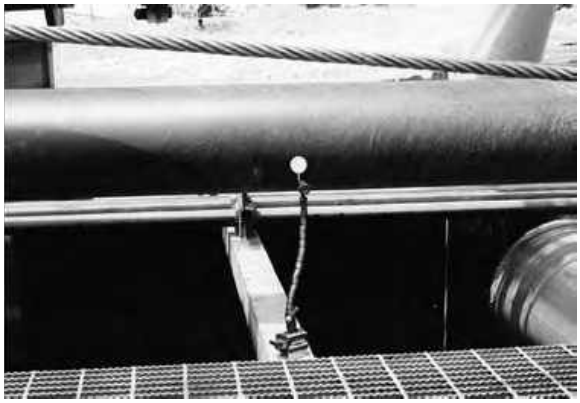


FIGURE 5.23 Measuring runout on a long drive shaft section midway between the support bearings.

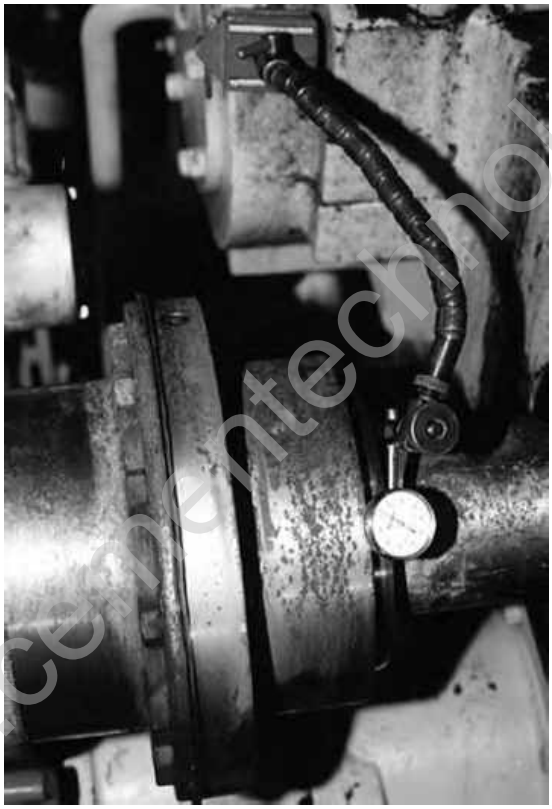


FIGURE 5.24 Measuring runout on the backside of the gear coupling hub attached to a steam turbine.



FIGURE 5.25 Measuring runout on the backside of a gear coupling hub attached to a compressor.

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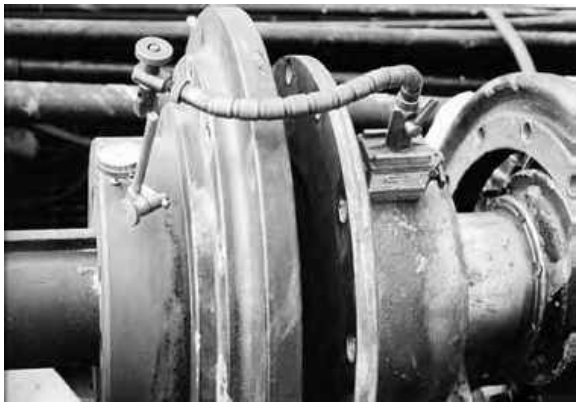


FIGURE 5.26 Measuring runout on rubber time coupling hub.

position. The motor drive shaft (with the motor removed) where the runout measurements were taken was connected to the pump shaft with a threaded coupling. In this vertical pump, the motor armature had a hollow shaft and the upper shaft section, shown in Figure 5.27 and Figure 5.28, was attached to the motor shaft at the upper bearing. Upon further investigation, it was discovered that the lower end of the upper shaft was not machined at a precise 90° angle to the centerline of the shaft. When the upper shaft fully engaged into the threaded coupling, as the ends of the two shafts met, the slight angle on the lower end of the upper shaft caused the shaft to pitch sideways producing the excessive runout condition.

Figure 5.29 and Figure 5.30 show a similar problem with a stub shaft that was bolted to the end of a motor shaft. The stub shaft was used to drive the main oil pump for the motor and compressor drive system. An excessive amount of runout was noticed when the checks were

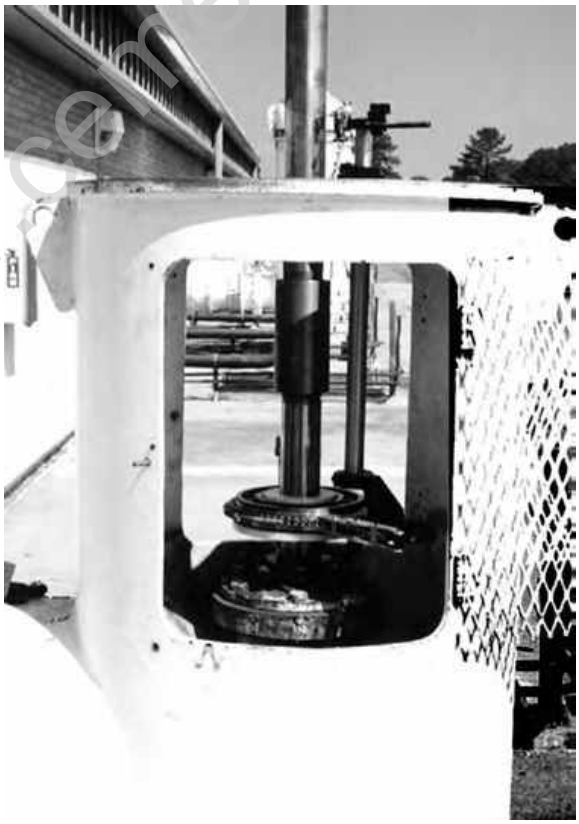


FIGURE 5.27 Measuring runout on a vertical drive shaft above a threaded coupling.

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FIGURE 5.28 Measuring runout on the same drive shaft near the top of the shaft.



FIGURE 5.29 Measuring runout on a stub shaft.

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FIGURE 5.30 Metal shaving found on mating face of stub shaft causing the runout in Figure 5.29.

made as shown in Figure 5.29. When the stub shaft was removed, a small piece of metal shaving was found lodged between the mating surfaces, causing the stub shaft to pitch when the bolts were tightened to the motor shaft as shown in Figure 5.30.

Runout can also occur on belt- and sheave-driven equipment and is a common problem with sheaves. Figure 5.31 and Figure 5.32 show runout measured on the rim and face of a multi-V sheave. This is a very important check to make and can cause a tremendous amount of confusion when attempting to align belt-driven machinery. This will be discussed further in detail later in the book.

How much runout is acceptable? The generally accepted limits for radial runout on rotating machinery are shown in Figure 5.33. Runout on V-belt sheaves will be discussed in Chapter 11.



FIGURE 5.31 Measuring radial runout on a multi-V sheave.

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FIGURE 5.32 Measuring face runout on a multi-V sheave.

Runout checks are frequently made from the inboard bearing to the end of a shaft. At the end of a shaft typically a coupling hub or sheave is mounted. There are three basic types of runout problems that are detected when taking measurements and are shown in Figure 5.34. From the inboard bearing out to the end of the shaft, there are three basic types of eccentricity problems that can occur: a coupling hub whose hole was bored off, center, or overbored and drawn-off center with the set screw, a skew (angle) bored hole in the coupling hub, or a bent shaft.

Measuring runout can be confusing at times. ''High spots'' and ''hills'' are not the same thing. ''Low spots'' and ''valleys'' are not the same thing when it comes under runout definitions. The high spot and low spot should occur 180° apart and these two points describe the axis of asymmetry or eccentricity. Hills, bumps, gouges, flat spots, and valleys describe the surface profile of an object and can occur at any point or perhaps at several points around the outer surface of that object (a coupling hub for example).

Acceptable radial runout limits

Shaft speed	Maximum recommended runout
0–1800 rpm	5 mils (0.005 in.)
1800–3600 rpm	3 mils (0.003 in.)
3600+ rpm	2 mils (0.002 in.)

FIGURE 5.33 Radial runout acceptance guidelines.

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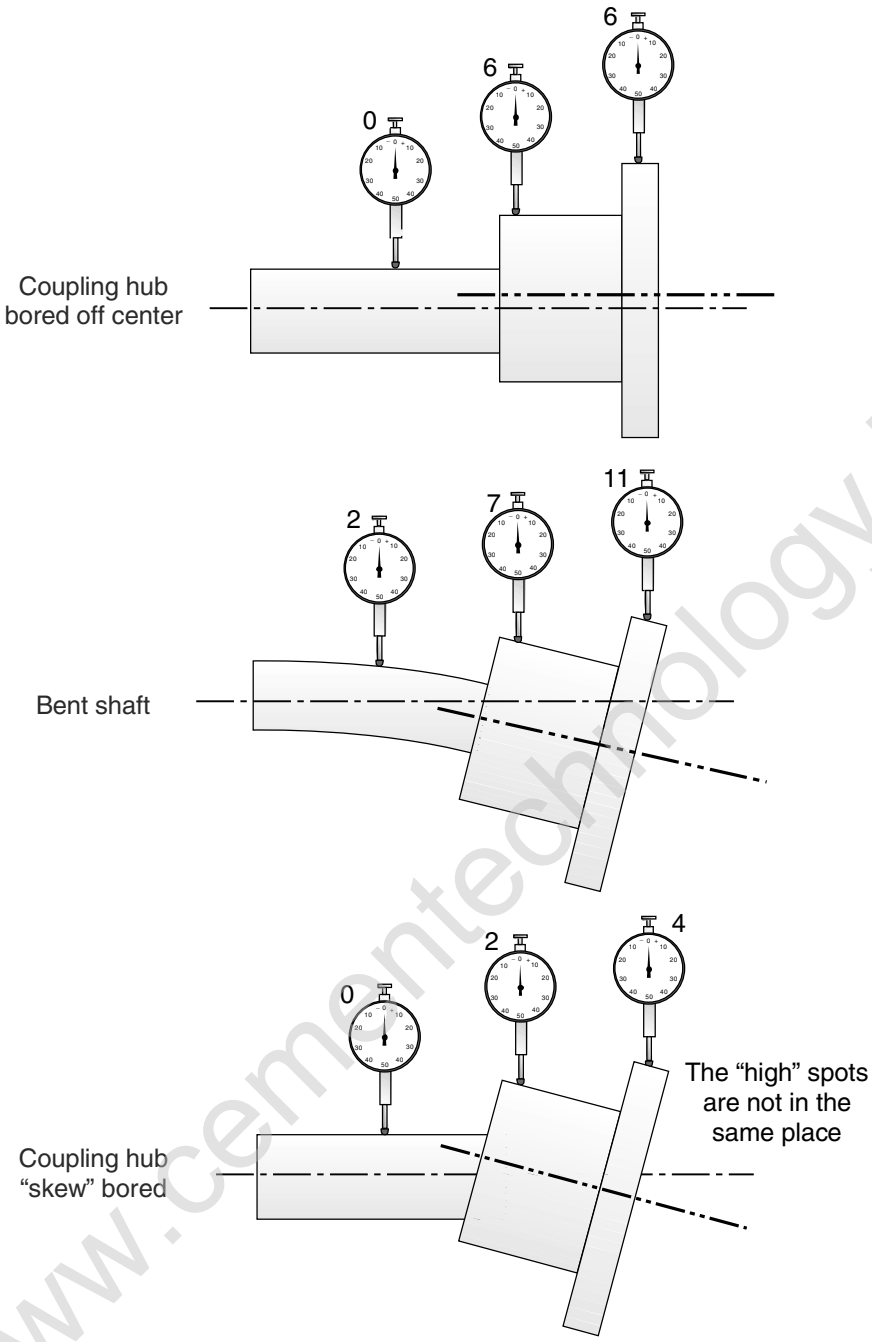


FIGURE 5.34 Three basic types of runout problems with shafts and coupling hubs.

Why such a big issue exists for finding and correcting excessive runout? Without too repetitive, the following segment is repeated from Chapter 5:

> By its purest definition, shaft alignment is when the centerlines of rotation are collinear. This is a very important point in aligning rotating machinery that a vast number of people overlook. It is possible to align the centerlines of rotation of machinery shafts that are bent or that have improperly bored coupling hubs and never know that these eccentricity problems exist.

When aligning rotating machinery shafts, the centerlines of rotation of each shaft are identified and then placed in a collinear axis. Ignoring the possibility of runout and very possible aligning of two bent shafts lead to a huge mistake.

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5.5 MACHINE HOUSING TO BASE PLATE INTERFACE PROBLEMS

One of the most prevalent problems with aligning rotating machinery can be traced to machine case to base plate interface problems. When rotating machinery is set in place on its base, frame, or sole plate, one or more than one of the "feet" are not making good contact at the "foot points" on the frame. This can be attributed to warped or bowed frames, warped or bowed machine cases, improper machining of the equipment feet, improper machining of the base plate, or a combination of a warped or uneven frame and warped or uneven machine case. This problem is commonly referred to as "soft foot." Soft foot generally describes any condition where poor or no surface contact is made between the underside of the machine feet and where they contact the base plate or frame.

Soft foot problems seem to be worse on fabricated base plates as opposed to cast base plates. A fabricated base plate is frequently made from sections of channel iron, angle iron, structural tubing, or I-beams. These pieces are then welded together to construct a machine frame. The chances of making true 45° or 90° cuts on the frame pieces and welding them all together insuring that everything is flat, square, and in the same plane is very unlikely. However, cast base plates are not exempted from this problem either. Even in cast base plates where the base is sand cast and the machinery feet are machined, it is possible that during the installation process the frame was warped when it was placed on the concrete pedestal introducing a soft foot problem.

Why would we bother to worry about this phenomenon? There are several important reasons why this should be corrected:

1. Depending on which sequence the foot bolts are tightened down, the centerline of rotation can be shifted into various positions, causing a considerable amount of frustration when trying to align the machinery.
2. Tightening down any foot bolts that are not making good contact will cause the machine case to warp upsetting critical clearances on critical components such as bearings, shaft seals, mechanical seals, pump wear rings, compressor staging seals, motor armature, or stator air gaps, etc.
3. Over time, residual vibration in the machine will begin to loosen the foot bolts and the shim packs may possibly work out from under the feet with the soft foot condition. With loose foot bolts comes the possibility that the alignment may shift into undesirable positions.
4. Localized high stress areas on the machine case may begin to cyclically fatigue eventually cracking the machine case (Figure 5.19).
5. With loose feet, vibration in the machine will cause fretting corrosion and repetitive impacting of the machine case and mating frame damaging the surfaces.

Figure 5.35 shows the underside of a pump bearing housing that sustained a considerable amount of damage due to a soft foot condition and uneven shim packs installed between the bearing housing and the frame. Figure 5.36 shows a shim plate under a pillow block bearing on a large fan that was not making contact with the underside of the pillow block bearing. After the soft foot condition was corrected, the vibration dropped by 400+%. Figure 5.37 shows shims that had worked loose under a compressor frame where the anchor bolt loosened due to a soft foot condition. Again, the vibration dropped significantly after the soft foot problem was corrected.

People often mistake this problem as analogous to the short leg of a four-legged chair. This metaphor is too simplistic and does not reflect the true nature of what really occurs on the machinery. The feet of a chair typically are making point contact on the floor. If our rotating

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FIGURE 5.35 Underside of pump bearing housing frame that needed resurfacing die to improper contact with its frame.

machinery feet were only making point contact then every soft foot problem could be corrected by using three legs such as found in a tripod.

It is important to recognize that our machinery feet are not making point contact. Instead there are typically four (or more) supposedly flat foot surfaces on the underside of our machine case trying to mate up to four (or more) supposedly flat surfaces on the base plate. Now, the chances of all four surfaces on the undersides of our machine cases are flat and in the same plane and all four surfaces on the base plate are flat and in the same plane are rare indeed. Quite often when we try to mate the underside of a machine foot to the point of contact on the base plate, a nonparallel, very complex, tapered, wedge-shaped gap condition exists that cannot be corrected with a flat piece of shim stock.

In addition, it is most probable that a soft foot condition exists at all of the foot points. Now it is not said that the machinery is levitating in free space, just that the feet are not making good contact on the base plate. A wide variety of different conditions can exist. Machinery can ''rock'' across two diagonal corners or can rock from side to side or end to end. It is not uncommon to see three of the feet ''toe up'' and the fourth foot ''toe down.'' It is possible to have ''edge contact'' at the inboard side of a foot and the outboard edge at that foot exhibit a gap.

There are four basic steps for detecting and correcting a soft foot problem as illustrated in Figure 5.38 through Figure 5.41.

The real trick for correcting soft foot (Figure 5.42) is to insure that contact crosses through the axis of each bolt-hole. Ideally it would be great to achieve full annular contact

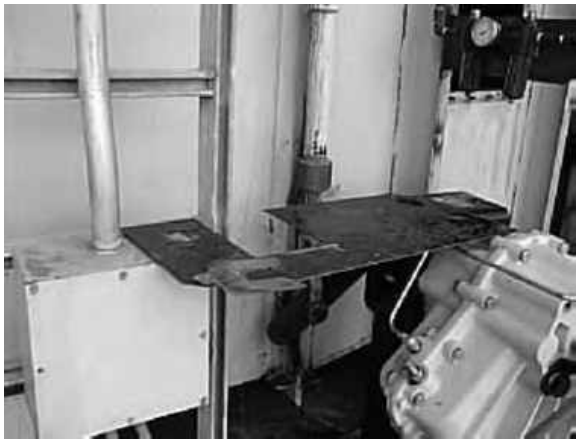


FIGURE 5.36 Shim pack that had worked loose under a fan bearing due to a soft foot condition.

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FIGURE 5.37 Machine frame to sole plate shim pack that had worked loose due to a soft foot condition.

all the way around each bolt-hole but this would require fabricating compound wedge-shaped shims that would tax the patience of many people with this problem. However, there is a small minority of people who are willing to spend time to do this to insure that a quality job is done.

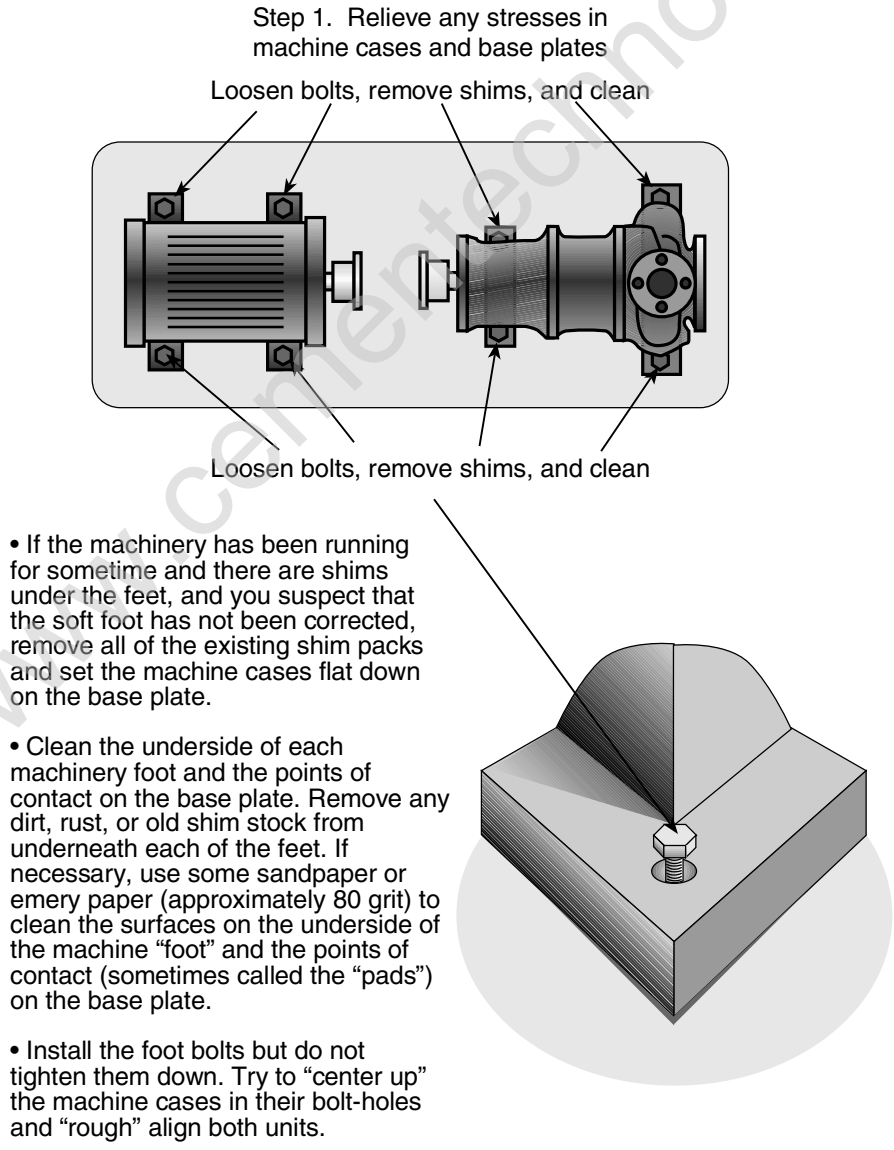


FIGURE 5.38 Soft foot—step 1. Remove existing shims and clean surfaces.

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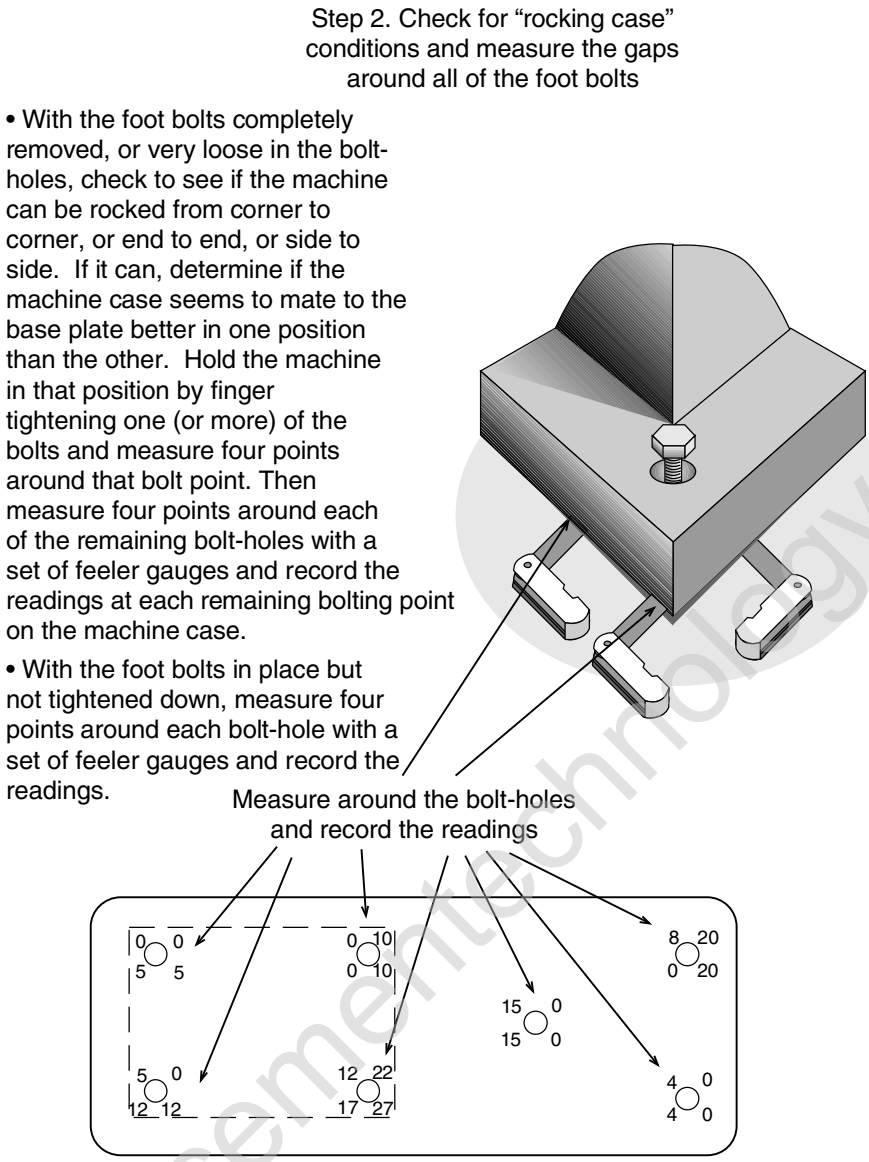


FIGURE 5.39 Soft foot—step 2. Measure gap at four points around each foot bolt with feeler gauges.

5.6 VERIFYING THAT THE SOFT FOOT HAS BEEN ELIMINATED

There are several methods for checking whether a soft foot condition has been eliminated. Due to the complex nature of machine case and frame warpage sometimes, the best method requires that several points on the machine case be monitored for movement. If this is impractical, one point on the machine case can be monitored whereas several bolts are loosened. Another method is to monitor the movement of the machinery shafts while loosening one or more foot bolts (Figure 5.43 through Figure 5.50).

5.6.1 Multiple Bolt–Multiple Indicator Method (Preferred Method)

1. Tighten all of the foot bolts holding the machine in place.
2. Place one dial indicator at each bolt location holding the machine case in place. Anchor the dial indicators to the frame or base and place the dial indicator stems as close as possible to the bolt-holes, insure that the stems are touching the top of the feet, and zero the indicators at mid-range.

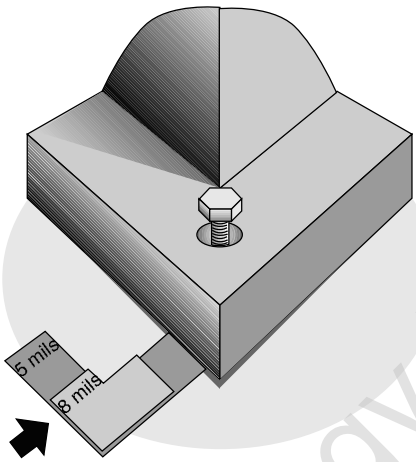
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Step 3. Correcting the soft foot conditions

• Eliminate the soft foot condition under each foot by installing complete U-shaped shims (if you have an even gap at all four points around the bolt-hole) or by constructing a "stair stepped shim wedge" with L-shaped or J-shaped shims or shims strips and installing the special wedges under each foot that needs correction. See shim sections below.

• If you have to build a custom "shim wedge" with L-shaped or J-shaped shims or shim "strips," as much as possible, try to maintain the outline of a complete U-shaped shim when you stack the shim pieces together. Later on, you may have to install additional shims under that foot to change the height or pitch of the machine case when you align the machinery. If the soft foot shim packs are neatly fabricated and stacked together in a U-shape, you can easily remove the soft foot shim pack, place the additional shims on top or underneath the pack and reinstall the entire assembly without disorienting the soft foot shim arrangement.



Note: After you install the shim correction under a foot, it might be helpful to "feel" if the soft foot has been eliminated. To do this, initially finger tighten the bolt, put a wrench on the bolt head and try to tighten it all the way. If the bolt tightens very quickly (e.g., you only have to turn the wrench an 1/8 turn or less) the soft foot is probably corrected. If however, you have to make a 1/4 or 1/2 turn on the wrench and the foot feels "spongy," the soft foot probably still exists and you have to try another shim pack

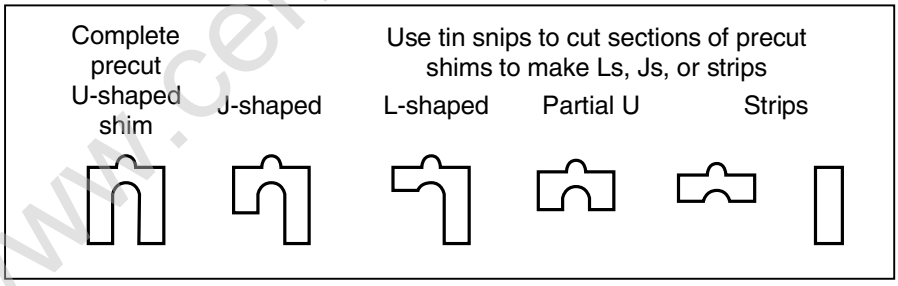


FIGURE 5.40 Soft foot—step 3. Install the correction shims.

3. Loosen the bolt where the worst soft foot condition existed watching the indicator at that foot for any movement. If more than 2–3 mils of movement is detected, there is probably some soft foot still remaining at that foot but do not do anything yet. Leave the bolt loose.
4. Loosen another bolt watching the indicator at that foot for any movement and also watch the indicator at the first foot for any additional movement. If more than 2–3 mils of movement is detected when this bolt was loosened and if more than 2–3 mils of additional movement is detected at the indicator on the first bolt that was loosened, there is probably some warpage occurring across those two bolts but do not do anything yet. Leave both bolts loose.

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Step 4. Verifying that the soft foot has been corrected

Verify that the soft foot or machine case and frame warpage problem has been eliminated by one of the following methods:

- Multiple bolt–multiple indicator method
- Multiple bolt–single indicator method
- Shaft movement method
- Single bolt–single indicator method

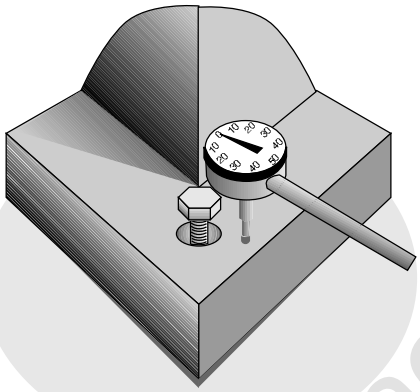


FIGURE 5.41 Soft foot—step 4. Check for lift or movement.

5. Continue loosening each of the remaining bolts, holding the machine case, watching the indicators at every loosened bolt, and observing any additional movement. Carefully watch the indicator since each corner may raise or lower as each bolt is loosened, giving you clues as to whether diagonal frame warpage or warpage along a side is occurring.
6. Once all of the bolts have been loosened, review what you observed when each bolt was loosened. If only one indicator showed more than 2–3 mils of movement, then there is probably a soft foot condition at that foot only. Remove any soft foot shims under that foot and remeasure four points around that bolt-hole with feeler gauges and install a flat shim or shim wedge to correct the observed condition. If more than 2–3 mils of movement were noticed at several bolt locations, then there is probably a soft foot condition at each one of those feet. Remove any soft foot shims under those feet and remeasure four points around those bolt-holes with feeler gauges and install flat shims or shim wedges to correct the observed condition.
7. Repeat the procedure if additional corrections are required.

5.6.2 Multiple Bolt–Single Indicator Method (Second Choice)

1. Tighten all of the foot bolts holding the machine in place.
2. Place one dial indicator at the bolt location where the worst soft foot condition was noticed. Anchor the dial indicator to the frame or base and place the dial indicator stem as close as possible to the bolt-hole, insure that the stem is touching the top of the foot, and zero the indicator at mid-range.
3. Loosen the bolt where the indicator is located, watching the indicator at that foot for any movement. If more than 2–3 mils of movement is detected, there is probably some soft foot still remaining at that foot but do not do anything yet. Leave the bolt loose.
4. Loosen another bolt watching the indicator at the first foot for any movement. If more than 2–3 mils of additional movement is detected when this bolt was loosened, there is

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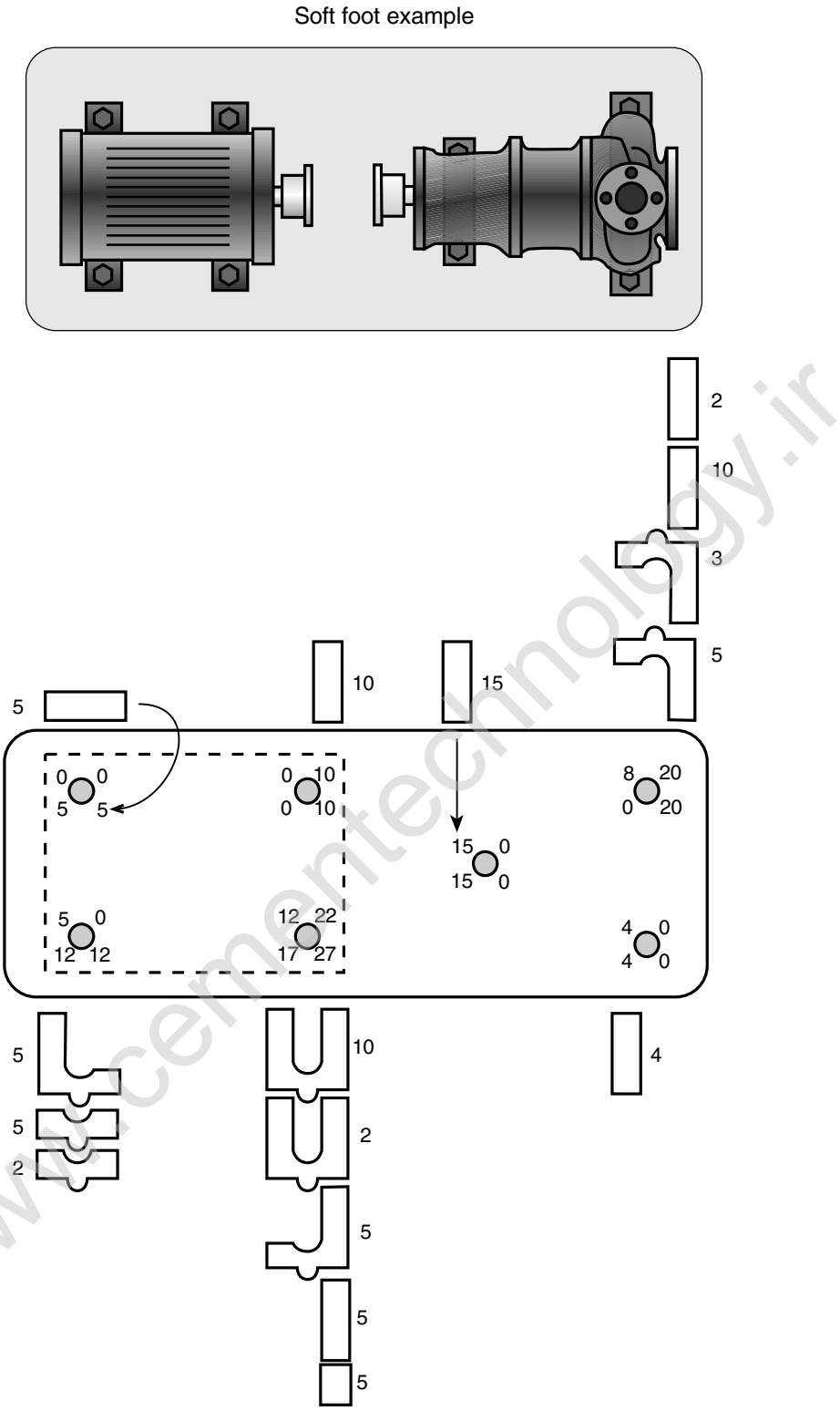


FIGURE 5.42 Soft foot example.

probably some warpage occurring across those two bolts but do not do anything yet. Leave both bolts loose.

5. Continue loosening each of the remaining bolts, holding the machine case, watching the indicator at the first bolt, and observing any additional movement. Carefully watch the indicator since each corner may raise or lower as each bolt is loosened giving clues as to whether diagonal frame warpage or warpage along a side is occurring.

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FIGURE 5.43 Cleaning the underside of the feet on a motor.

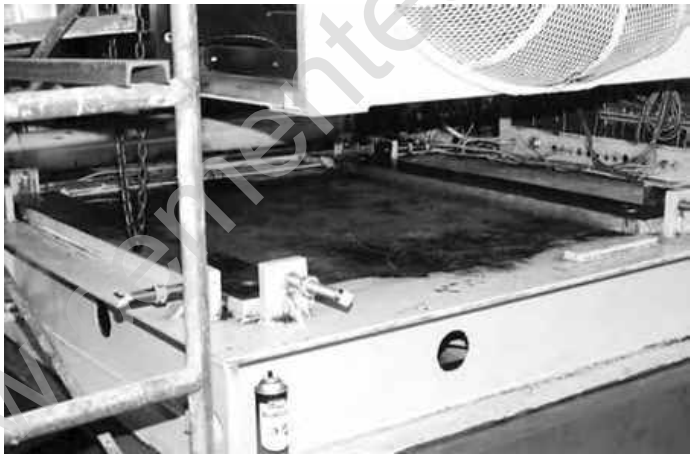


FIGURE 5.44 Sole plates cleaning under motor shown in Figure 5.42.

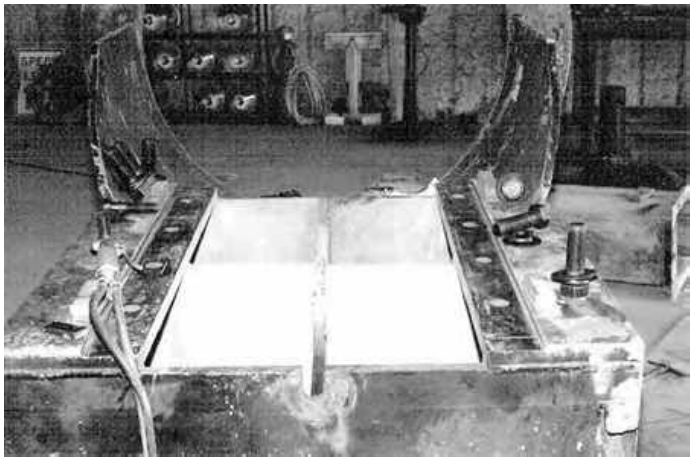


FIGURE 5.45 Sole plates cleaning on pump frame.

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FIGURE 5.46 Cleaning the pump feet using 180 grit emery cloth wrapped around a 1/8 in. thick steel bar and "sawing" back and forth to clean the underside of the machine foot and the base plate at the same time.



FIGURE 5.47 The underside of this motor foot is "hollow." Make sure for using the right size shims to get as much contact as possible here.

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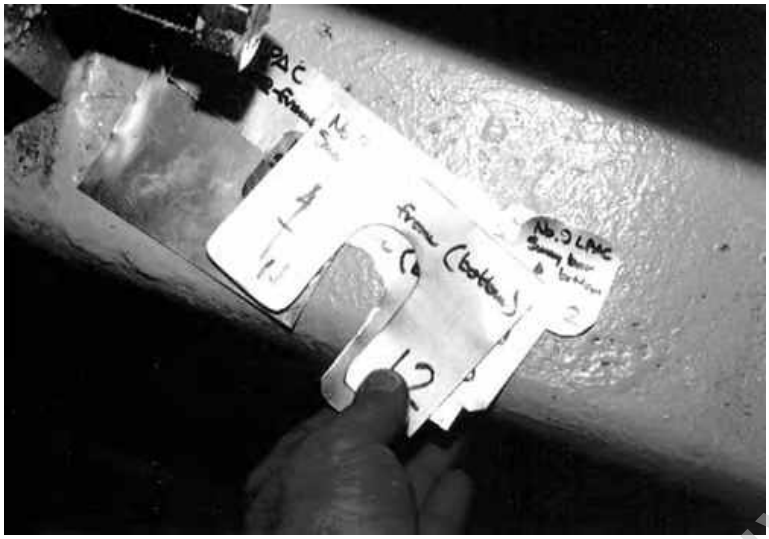


FIGURE 5.48 Soft foot correction shim stack.

6. Once all of the bolts have been loosened, review what you observed when each bolt was loosened. If more than 2–3 mils of movement occurred when just one of the bolts was loosened, then there is probably a soft foot condition at that foot only. Remove any soft foot shims under that foot and remeasure four points around that bolt-hole



FIGURE 5.49 Soft foot correction shims for a motor.

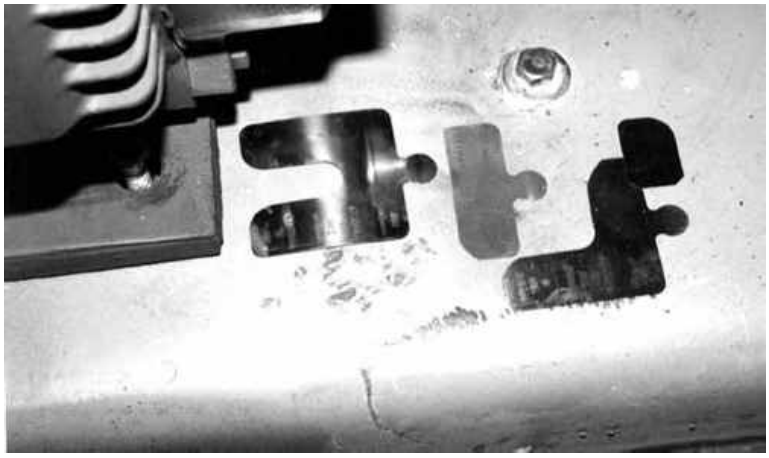


FIGURE 5.50 Soft foot correction shims stack.

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with feeler gauges and install a flat shim or shim wedge to correct the observed condition. If more than 2–3 mils of movement were noticed when several of the bolts were loosened, then there is probably a soft foot condition at each one of those feet. Remove any soft foot shims under those feet and remeasure four points around those bolt-holes with feeler gauges and install flat shims or shim wedges to correct the observed condition.

7. Repeat the procedure if additional corrections are required.

5.6.3 Shaft Movement Method (Third Choice)

1. Tighten all of the foot bolts holding the machine in place.
2. Attach a bracket to one shaft, place a dial indicator on the topside of the other shaft, and zero the indicator at mid-range.
3. Sequentially loosen one-foot bolt at a time observing for any movement at the indicator when each bolt is loosened.
4. If there were more than 2–3 mils of movement when only one of the bolts were loosened, then there is probably a soft foot condition at that foot only. Remove any soft foot shims under that foot and remeasure four points around that bolt-hole with feeler gauges and install a flat shim or shim wedge to correct the observed condition. If more than 2–3 mils of movement were noticed when several of the bolts were loosened, then there is probably a soft foot condition at each one of those feet. Remove any soft foot shims under those feet and remeasure four points around those bolt-holes with feeler gauges and install flat shims or shim wedges to correct the observed condition.
5. Repeat the procedure if additional corrections are required.

5.6.4 Single Bolt–Single Indicator Method (Last Choice)

1. Tighten all of the foot bolts holding the machine in place.
2. Place a dial indicator at one of the feet on the machine case. Anchor the dial indicator to the frame or base and place the dial indicator stem as close as possible to the bolt-hole, insure that the stem is touching the top of the foot, and zero the indicator at mid-range.
3. Loosen the bolt where the indicator is located, watching the indicator at that foot for any movement. If more than 2–3 mils of movement are detected, there is probably some soft foot still remaining at that foot. Remove any soft foot shims under that foot and remeasure four points around that bolt-hole with feeler gauges and install a flat shim or shim wedge to correct the observed condition. Retighten the bolt.
4. Sequentially move the indicator to each one of the feet, loosening that bolt and watching the indicator for any movement. If more than 2–3 mils of movement are detected when each bolt was loosened, there is probably some soft foot still remaining at that foot. Remove any soft foot shims under that foot and remeasure four points around that bolt-hole with feeler gauges and install a flat shim or shim wedge to correct the observed condition. Retighten each bolt.
5. Repeat the procedure if additional corrections are required.

Once the soft foot has been corrected, the shims will stay there for the rest of the alignment process. We may be adding more shims later on to change the height or "pitch" of the machine case but the shims used to correct the soft foot condition will remain in place.

As illustrated in Figure 5.51 through Figure 5.53, a soft foot condition can occur on other machinery components besides the machine case itself, in this instance, it is a pillow block bearing and its mating pedestal. Figure 5.54 shows a turbine bearing with lateral support

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FIGURE 5.51 Soft foot correction shims for a pillow block bearing on a fan.



FIGURE 5.52 Checking for lift with a magnetic base and dial indicator on a machine foot.



FIGURE 5.53 Checking for lift with a magnetic base and dial indicator on a fan frame.

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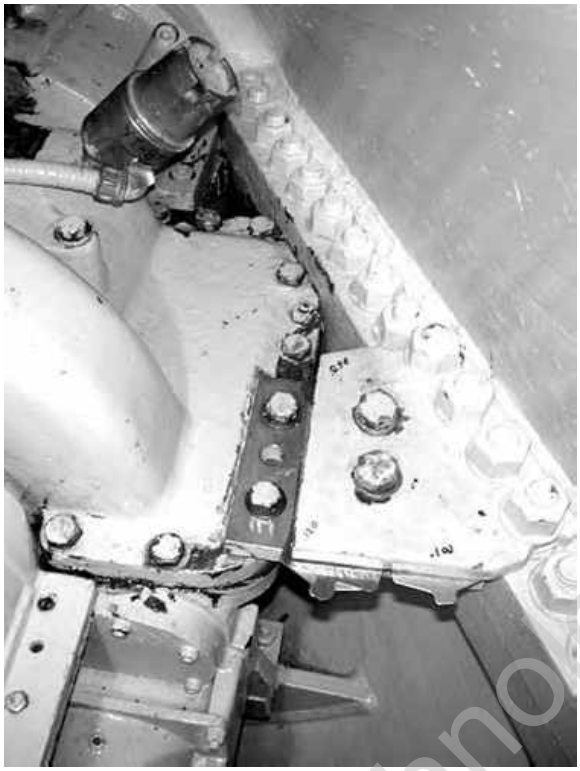


FIGURE 5.54 Soft foot shims installed on turbine bearing lateral support plates.

plates that are bolted to the condenser shell. There was uneven contact between the lateral support plates and the condenser shell flange. Soft foot shims had to be installed here to provide sufficient contact to achieve the desired lateral stiffness.

Figure 5.55 through Figure 5.58 show another pillow block bearing that was not making adequate contact. Figure 5.55 shows the underside of the lower pillow block casting. Notice that it too is "hollow." To determine where the lower casting was not touching the pedestal, 5-mil thick shim strip were placed on the pedestal to elevate the lower casting a known distance. Plastigage was then placed at the areas of contact, the bearing set down and the bolts tightened slightly. After loosening the bolts and removing the lower pillow block casting, the amount of crush was measured using the guide on the Plastigage container sleeve as shown in Figure 5.57. Shims were then installed where the gap was observed to be over 5 mils. Figure 5.58 shows a lift check made on the pillow block to insure the lack of contact problem was corrected.

5.7 OTHER METHODS FOR CORRECTING SOFT FOOT PROBLEMS

For many of the readers who are reading about this problem for the first time, there is a great tendency to disbelieve that this malady actually exists. Be forewarned, this is a time consuming, frustrating process that frequently can consume more time than actually aligning the rotating machinery itself. Despite the fact that two out of three pieces of rotating machinery have a soft foot problem, very few solutions have been forwarded on how to correct this problem easily.

It does seem rather silly to cut U-shaped shims into strips, L-shapes, J-shapes, or shortened U-shapes to correct this problem but precut, U-shaped shims are commonly used in industry to adjust the position of rotating machinery in the process of aligning equipment. But it is not possible to correct a wedge-shaped gap condition with a flat piece of shim stock. Since many

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FIGURE 5.55 Underside of pillow block bearing lower casting.

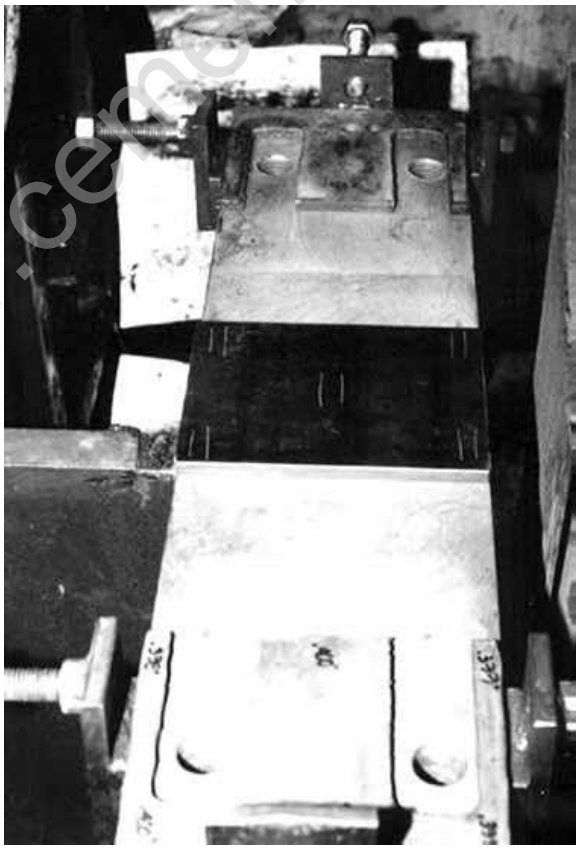


FIGURE 5.56 Installing shims and Plastigage for contact check.

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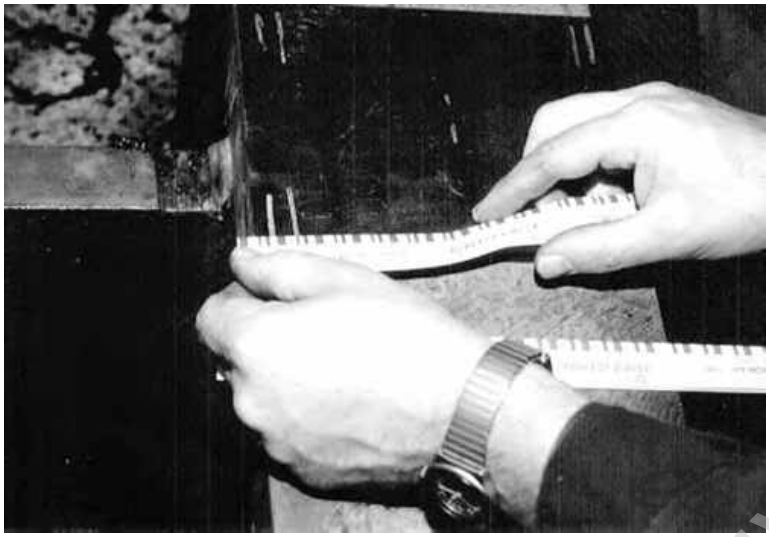


FIGURE 5.57 Measuring the crushed Plastigage.

people only have this precut shim stock available to them, then the only way to construct a wedge is to ''stair-step'' pieces of shims together to construct the wedge that is needed. For people who have a lot of time on their hands, they could actually machine a custom wedge shape shim after they ''mapped'' out the gaps at each foot. People have actually done this.

There have been a few attempts to create a device that automatically corrects for a soft foot condition. Proprietary plastic shims were experimented within the 1990s but they did not seem to meet the requirements satisfactorily. They did provide some damping between the machine and the base plate however.

Before that there were ''peel away'' shim blocks. Thin shim stock was made into a multi-layer sandwich, where several thin shims were bonded together with a thin adhesive layer. People would then peel away as many layers as they needed and could trim each layer to form a wedge if desired. Since nonhardening adhesive was used, applications on machinery that ran hot would begin to debond the layers. One always hoped the adhesive would not flow and squeeze out since it had a thickness to it also.



FIGURE 5.58 Checking for lift after installing the noncontact (i.e., soft foot) correction shims.

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FIGURE 5.59 Foot plane compensator. Underside view showing epoxy channels. (Courtesy of Max Roeder Consulting, Inc., Danville, IN.)

One idea from a colleague suggested that you take two 10-mil thick shims, mix up some epoxy resin and hardener, spread the epoxy on one side of a shim, and then make a shim "sandwich," install it under the foot, and then let it harden. In the event that too much epoxy was applied, the idea suggested that you can put the shim inside a plastic bag so the excess epoxy would flow into the bag and not adhere the machine to its base plate. Then, after the epoxy cured, it is essential to remove the bagged shim, trim away the squeezed out epoxy, remove the plastic bag, and reinstall the shim. I tried that one time and got yelled at for spending too much time "goofing around" and not getting the alignment job done.

Another similar yet more elegant idea was developed by another colleague. These devices were dubbed "foot plane compensators" and are shown in Figure 5.59 and Figure 5.60. The underside of the foot plane compensator has channeling to allow epoxy to flow into the cavity. An O-ring seals the perimeter to prevent the epoxy from flowing out. A special bolt is used to attach the foot plane compensator to the underside of every foot on a machine and then the machine is set down onto its base plate. Tubing is then placed into one of the openings on the side of the foot plane compensator and epoxy is injected into the cavity. Once the epoxy sets and hardens the foot plane compensator to the base plate, the special



FIGURE 5.60 Foot plane compensator. Injecting epoxy into the channels. (Courtesy of Max Roeder Consulting, Inc., Danville, IN.)

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bolt is removed and the original foot bolts are then installed and the final alignment process then continues.

However, there is one problem with all of the above ideas. As mentioned in Figure 5.40, after the soft foot has been corrected, you may very well install additional shims under the machinery feet to correct a misalignment condition. What if, later on during the final alignment process, you find out that you need to add 300 mils of shims under the outboard bolting plane of a machine and 5 mils of shims under the inboard bolting plane? If you raise one end of a machine significantly higher than the other end, will it introduce a soft foot problem. And if the angular pitch is severe enough, it is essential to correct the soft foot problem, just introduced into the machine–base plate interface.

Whatever soft foot correction device or mechanism is invented to automatically correct this problem, there are eight issues (eventually "features" if successful) that need to be addressed by the brave inventor:

1. The vast majority of soft foot problems are nonparallel gap situations.
2. One or more than one machine foot may not be making contact whether parallel or nonparallel conditions exist between the machine and its point of contact on the base or frame. It must therefore be recognized that soft foot is not a surface area problem, but a volume problem.
3. It is possible that a soft foot condition could be introduced when adding more shims under one end of a machine case than the other end when attempting to correct a misalignment condition. Therefore the device has to change its shape to account for an intended angular pitch on the machine casing.
4. Thermal warpage of a machine base or frame can occur during operation that would alter the soft foot condition as observed during the off-line condition.
5. Be "thin" enough to fit under all of the currently installed rotating machinery without having to make major frame, machine case, or piping alterations.
6. Maintain its shape and volume for long periods of time under vibratory forces, extreme pressure from torqued foot bolts, and possibly high temperatures from the machine during operation.
7. Be relatively inexpensive.
8. Easy to install or remove and have little or no maintenance required.

Perhaps someday the solution will come in a material that can alter its shape and volume when an electrical charge is applied to it. Or maybe people are looking at this in the wrong way. The above solutions are macroscopic in approach. Maybe a microscopic approach is needed. Perhaps thousands of tiny wedges or pistons or solenoids can arrange themselves in such a manner to solve the eight issues mentioned above. Nanotechnology may provide the answer.

But before these pie-in-the-sky approaches are explored, the first thing is to educate the people designing, purchasing, installing, and aligning rotating machines that this issue exists. Disappointingly, not enough people are aware that they even have this problem.

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6 Shaft Alignment Measuring Tools

An alignment "expert" is someone who is knowledgeable about the myriad of measuring tools available for shaft alignment and also knows how to perform all five of the shaft positional measurement methods and understands the limitations of them. There are advantages and disadvantages to each one of these methods as discussed in Chapter 10 through Chapter 15. There is no one method or measuring device that will solve every alignment problem that one can possibly encounter on all the various types of rotating machinery drive systems in existence. It is important to understand each one of these techniques so you can select the best measurement method for the alignment situation confronting you. In many cases, two (or more) different techniques could be used to make shaft centerline positional measurements on the same drive system.

Every once in a while, people who capture a set of shaft alignment readings using one of these techniques or measurement tools will run across a situation where the measurements they have taken do not seem to make sense. Knowing how to perform shaft positional measurements, a different way can verify whether the data from the initial technique in doubt are valid. Since the machinery shafts can only be in one position at any point in time, the data from two or more measurement methods should indicate the same shaft positional information. For example, if you have captured a set of readings with a laser alignment system and you do not believe what the system is telling you then take a set of reverse indicator readings. If the two sets of readings agree, then the measurement data are probably correct. If they do not, then it would be wise to determine why there is a discrepancy between the two methods before you continue. If you do not investigate the cause, you may incorrectly position the machinery based on bad measurement data. Therefore, knowing all the methods offers you a choice of which one you would like to do and, if necessary, compare one method to another, or validate one against the other.

Since shaft alignment is primarily concerned with the application of distance measurement, this chapter will begin by covering the wide variety of tools available to measure dimensions. Next, the five currently known shaft alignment measurement techniques commonly employed for rotating machinery shafts connected together with flexible couplings will be shown. Two other shaft alignment techniques used on rotating machinery shafts connected together with rigid couplings are explained. The illustrations for these techniques show utilizing mechanical dial indicators as the measurement device but any measurement device with an accuracy of 1 mil (or better) could be used. It is recommended that you understand each of these basic measurement methods shown in Chapter 10 through Chapter 14 since every alignment measurement system in existence utilizes one or more of these methods regardless of the measurement sensor used to capture the shaft position information.

Keep in mind that this chapter covers one small but important facet of shaft alignment, measuring the relative positions of two rotating machinery shafts. In other words, these

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methods will show you how to find the positions of two shaft centerlines when the machinery is not running (step 5 in Chapter 1). Once you have determined the relative positions of each shaft in a two-element drive train, the next step is to determine if the machinery is within acceptable alignment tolerances (Chapter 9). If the tolerance is not yet acceptable, the machinery positions will have to be altered as discussed in Chapter 8, which discusses a very useful and powerful technique where the data collected from these methods (Chapter 10 through Chapter 15) can be used to construct a visual model of the relative shaft positions to assist you in determining which way and how far you should move the machinery to correct the misalignment condition and eventually achieve acceptable alignment tolerances.

6.1 DIMENSIONAL MEASUREMENT

The task of accurately measuring distance was one of the first problems encountered by man. The job of "rope stretcher" in ancient Egypt was a highly regarded profession and dimensional measurement, technicians today, can be seen using laser interferometers capable of measuring distances down to the submicron level.

It is important for us to understand how all of these measurement tools work, since new tools rarely replace old ones, and they just augment. Despite the introduction of laser shaft alignment measurement systems in the early 1980s, for example, virtually all manufacturers of these systems still include a standard tape measure for the task of measuring the distances between the hold down bolts on machinery casings and where the measurement points are captured on the shafts.

The two common measurement systems in worldwide use today are the English and metric systems. Without going into a lengthy dissertation of English to metric conversions, the easiest one most people can remember is this:

25.4 mm = 1.00 in.

By simply moving the decimal point three places to the left, it becomes obvious that

0.0254 mm = 0.001 in. = 1 mil (one thousandth of an inch)

6.2 CLASSES OF DIMENSIONAL MEASUREMENT TOOLS AND SENSORS

There are two basic classes of dimensional measuring devices that will be covered in this chapter, mechanical and electronic.

In the mechanical class, there are the following devices:

- Tape measures and rulers
- Feeler and taper gauges
- Slide calipers
- Micrometers
- Dial indicators
- Optical alignment tooling

In the electronic class, there are the following devices or systems:

- Proximity probes
- Linear variable differential transformers (LVDT)
- Optical encoders
- Lasers and detectors
- Interferometers
- Charge couple device (CCD)

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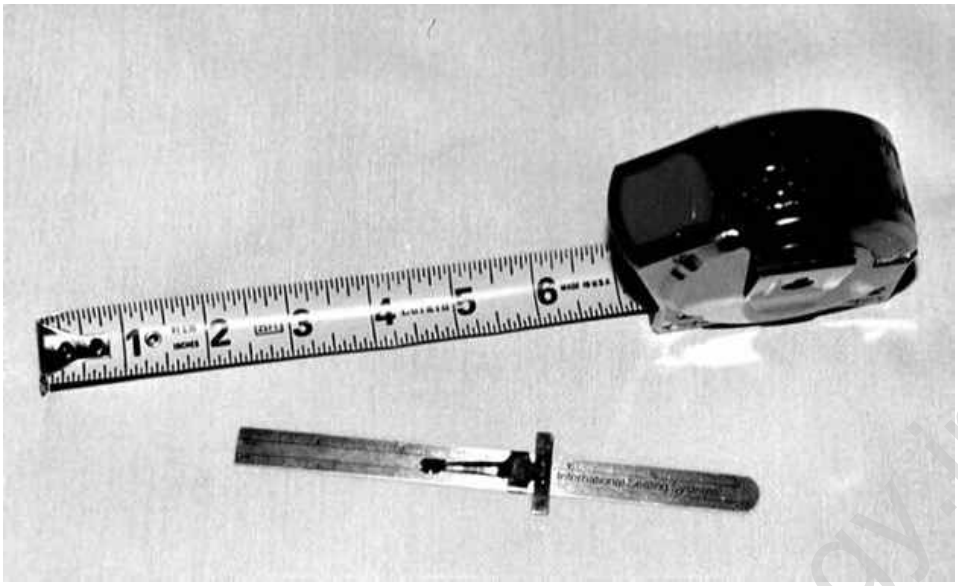


FIGURE 6.1 Standard linear rulers.

Many of these devices are currently used in alignment of rotating machinery. Some could be used but are not currently offered with any available alignment measurement systems or tooling but are covered in the event future systems incorporate them into their design. They are discussed so you can hopefully gain an understanding of how these devices work and what their limitations are. One of the major causes of confusion and inaccuracy when aligning machinery comes from the operators lack of knowledge of the device they are using to measure some important dimension. Undoubtedly you may already be familiar with many of these devices. For the ones that you are not familiar with, take a few moments to review them and see if there is a potential application in your alignment work.

6.2.1 Standard Tape Measures, Rulers, and Straightedges

Perhaps the most common tools used in alignment are standard rulers or tape measures as shown in Figure 6.1. The tape measure is typically used to measure the distances between machinery hold down bolts (commonly referred to as the machinery "feet") and the points of measurement on the shafts or coupling hubs. Graduations on tape measures are usually as small as 1/16 to 1/32 in. (1 mm on metric tapes), which is about the smallest dimensional measurement capable of discerning by the unaided eye. A straightedge is often used to "rough align" the units as shown in Figure 6.2.

6.2.2 Feeler and Taper Gauges

Feeler gauges are simply strips of metal shim stock arranged in a "foldout fan"-type of package design. They are used to measure soft foot gap clearances, closely spaced shaft end to shaft end distances, rolling element to raceway bearing clearances, and a host of similar tasks where fairly precise ($\pm$1 mil) measurements are required.

Taper gauges are precisely fabricated wedges of metal with lines scribed along the length of the wedge that correspond to the thickness of the wedge at each particular scribe line. They are typically used to measure closely spaced shaft end to shaft end distances where accuracy of $\pm$10 mils is required.

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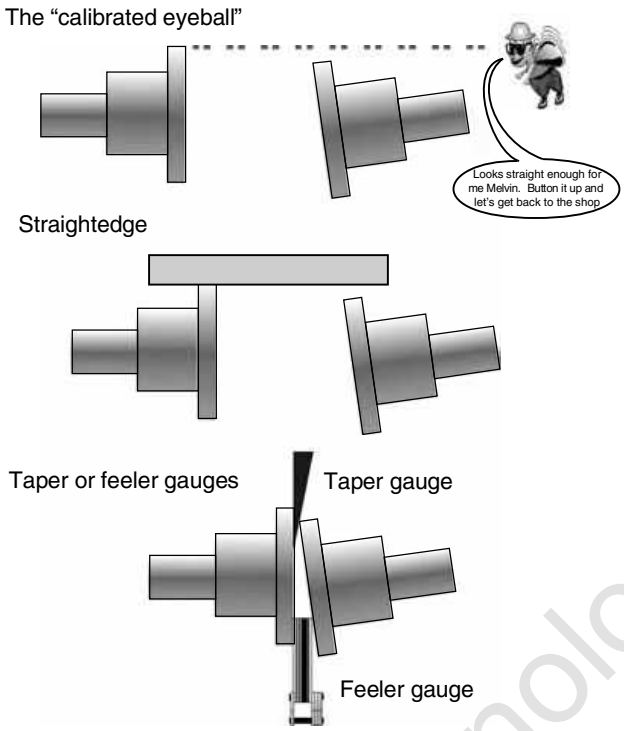


FIGURE 6.2 Rough alignment methods using straightedges, feeler gauges, or taper gauges.

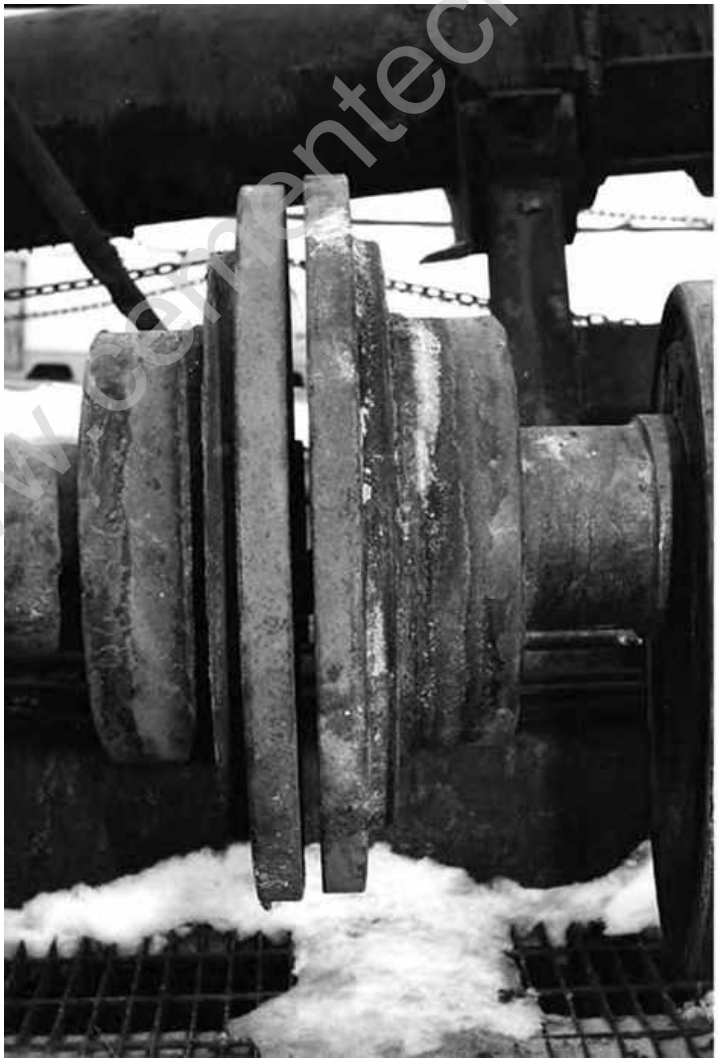


FIGURE 6.3 Misalignment visible by eye.

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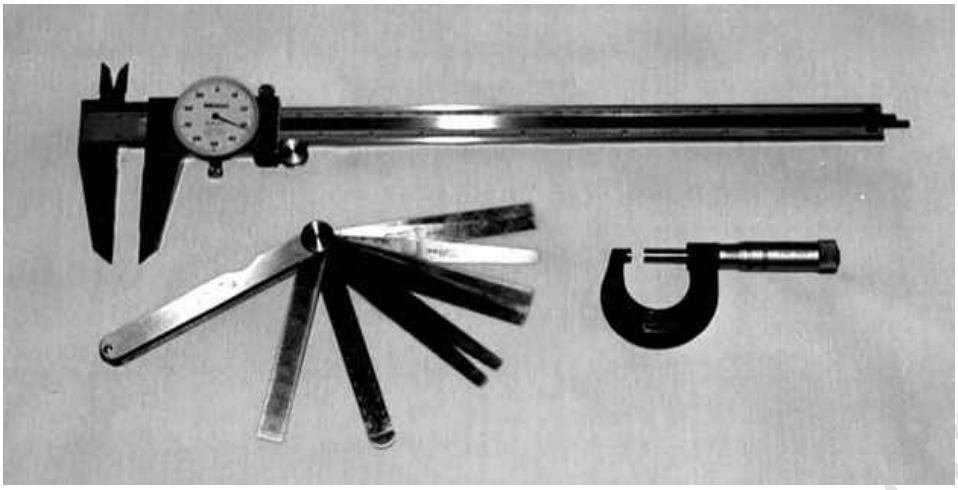


FIGURE 6.4 Feeler gauges, slide caliper, and outside micrometer.

6.2.3 Slide Caliper

The slide caliper has been used to measure distances with an accuracy of 1 mil (0.001 in.) for the last 400 years. It can be used to measure virtually any linear distance such as shim pack thickness, shaft outside diameters, coupling hub hole bores, etc. A very ingenious device has been invented to measure shaft positional changes, whereas machinery is running utilizing miniature slide calipers attached to a flexible coupling that will be reviewed in Chapter 16. The primary scale looks like a standard ruler with divisions marked along the scale at increments of 0.025 in. The secondary, or sliding scale, has a series of 25 equally spaced marks where the distance from the first to the last mark on the sliding scale is 1.250 in. apart. The jaws are positioned to measure a dimension by translating the sliding scale along the length of the primary scale as shown in Figure 6.4. The dimension is then obtained by:

1. Observing where the position of the zero mark on the sliding scale is aligning between two 25-mil division marks on the primary scale. A mental (or written) record of the smaller of the two 25-mil division marks is made.
2. Observing which one of the 25 marks on the secondary scale aligns most evenly with another mark on the primary scale. The value of the aligned pair mark on the secondary scale is added with the recorded 25-mil mark in step 1.

Some modern slide calipers as shown in Figure 6.4 have a dial gauge incorporated into the device. The dial has a range of 100 mils and is attached to the sliding scale via a rack and pinion gear set. This eliminates the need to visually discern which paired lines match exactly (as discussed in step 2 above) and a direct reading can then be made by observing the inch and tenths of an inch mark on the primary scale, and then adding the measurement from the indicator (Figure 6.5). With care and practice, measurement to ± 0.001 in. can be made with either style.

6.2.4 Micrometers

Although the micrometer was originally invented by William Gascoigne in 1639, its use did not become widespread until 150 years later when Henry Maudslay invented a lathe capable of accurately and repeatably cutting threads. That of course brought about the problem of how threads should be cut (number of threads per unit length, thread angles, thread depth,

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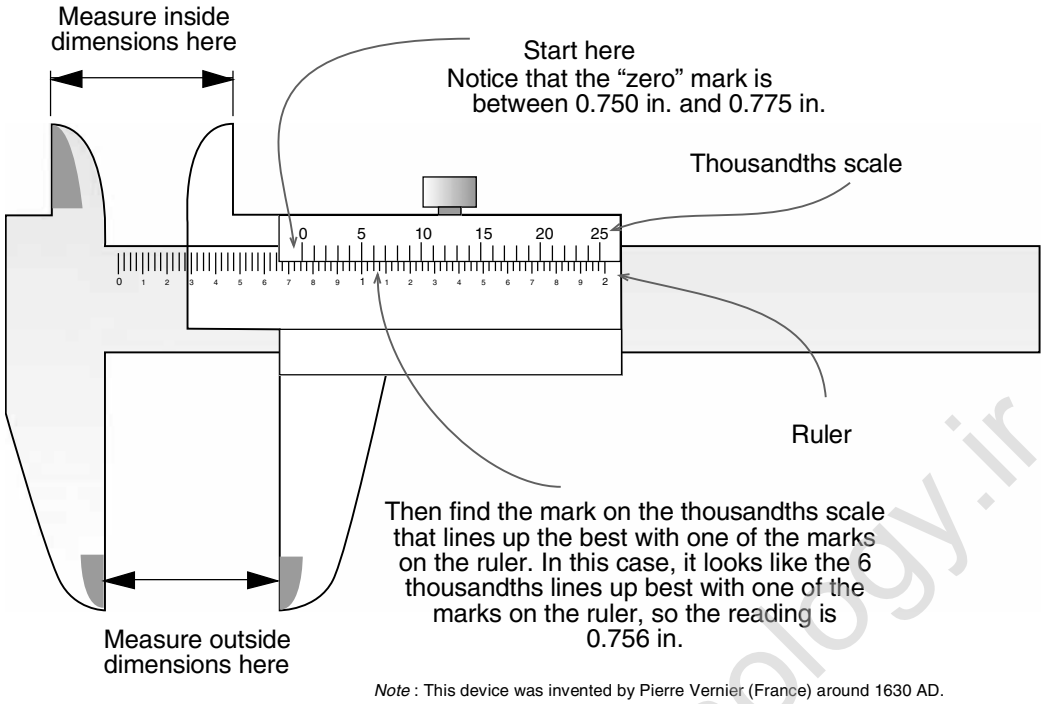


FIGURE 6.5 How to read a slide caliper.

etc.), which forced the emergence of thread standards in the Whitworth system (principally abandoned) and the current English and metric standards.

The micrometer is still in prevalent use today and newer designs have been outfitted with electronic sensors and digital readouts. The micrometer is typically used to measure shaft diameters, hole bores, shim or plate thickness, and is a highly recommended tool for the person performing alignment jobs.

A mechanical outside micrometer consists of a spindle attached to a rotating thimble, which has 25 equally spaced numbered divisions scribed around the perimeter of the thimble for English measurement system as shown in Figure 6.6. When the spindle touches the mechanical stop at the tip of the C-shaped frame, the zero mark on the thimble of the micrometer aligns with the sleeve's stationary scale reference axis. As the thimble is rotated and the spindle begins to move away from the mechanical stop, the precisely cut threads (40 threads/in.) insure that as the drum is rotated exactly one revolution, the spindle has moved 25 mils (1/40th of an inch or 0.025 in.). As the thimble continues to rotate, increasing the distance from the spindle tip to the mechanical stop (anvil), the end of the thimble wheel exposes division marks on the sleeve's stationary scale scribed in 25-mil increments. Once the

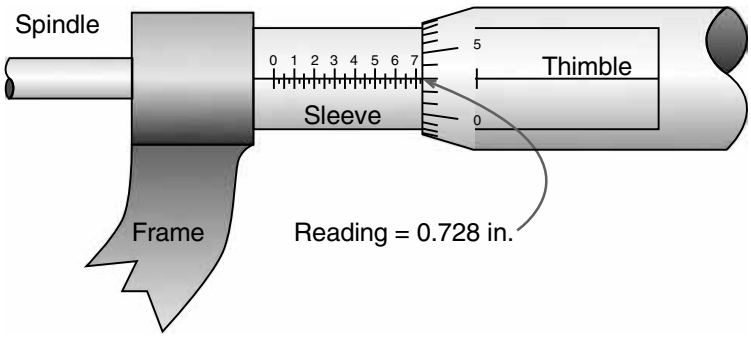


FIGURE 6.6 How to read a micrometer.

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desired distance between the anvil and the spindle is obtained, observe what 25-mil division on the stationary scale has been exposed, then add whatever scribed division on the drum aligns with the reference axis of the stationary scale.

6.2.5 Dial Indicators

The dial indicator came from the work of a nineteenth century watchmaker in New England. John Logan of Waltham, Massachusetts, filed a U.S. patent application on May 15, 1883 for what he termed as "an improvement in gages." Its outward appearance was no different than the dial indicators of today but the pointer (indicator needle) was actuated by an internal mechanism consisting of a watch chain wound around a drum (arbor). The arbor diameter determined the amplification factor of the indicator. Later, Logan developed a rack and pinion assembly that is currently in use today on most mechanical dial indicators.

The full range of applications of this device was not recognized for another 13 years when one of Logan's associates, Frank Randall, another watchmaker from E. Howard Watch Co., Boston, bought the patent rights from Logan in 1896. He then formed a partnership with Francis Stickney and began manufacturing dial indicators for industrial use. A few years later B.C. Ames also began manufacturing dial indicators for general industry.

The German professor Ernst Abbe established the measuring instrument department at the Zeiss Works in 1890 and by 1904 he had developed a number of instruments, which included a dial indicator, for sale to industry. The basic operating principle of dial indicator was discussed in Chapter 5 (see Figure 5.1). The dial indicator is still in prevalent use today and newer designs have been outfitted with electronic sensors and digital readouts.

For the past 50 years, the most common tool that has been used to accurately measure shaft misalignment is the dial indicator as shown in Figures 6.7 through Figures 6.9. There are some undeniable benefits of using a dial indicator for alignment purposes:

- One of the preliminary steps of alignment is to measure runout on shafts and coupling hubs to insure that eccentricity amounts are not excessive. As we have seen in Chapter 5, the dial indicator is the measuring tool typically used for this task and is therefore usually one of the tools that the alignment expert will bring to an alignment job. Since a dial indicator is used to measure runout, why not use it also to measure the shaft centerline positions?
- The operating range of dial indicators far exceeds the range of many other types of sensors used for alignment. Dial indicators with total stem travels of 0.200 in. (5 mm) are traditionally used for alignment but indicators with stem travels of 3 in. or greater could also be used if the misalignment condition is moderate to severe when you first begin to "rough in" the machinery.
- The cost of a dial indicator (around US$70 to US$110) is far less expensive than many of the other sensors used for alignment. You could purchase over 140 dial indicators for the average cost of some other alignment tools currently on the market.
- Since the dial indicator is a mechanically based measurement tool, there is a direct visual indication of the measurement as you watch the needle rotate.
- They are very easy to test for defective operation.
- They are much easier to find and replace in virtually every geographical location on the globe in the event that you damage or lose the indicator.
- Batteries are not needed.
- The rated measurement accuracy is equivalent to the level of correction capability (i.e., shim stock cannot be purchased in thickness less than 1 mil)

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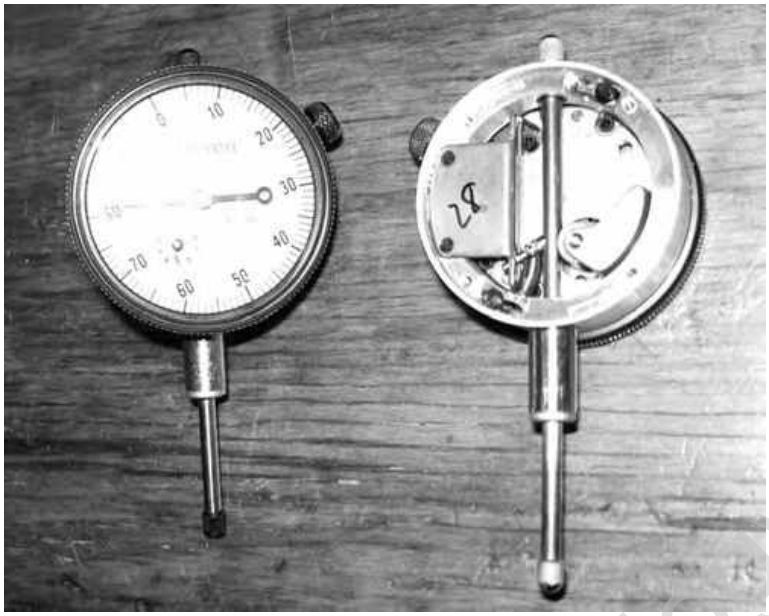


FIGURE 6.7 Dial indicator.

6.2.6 Optical Alignment Tooling

Optical alignment tooling consists of devices that combine low-power telescopes with accurate bubble levels and optical micrometers for use in determining precise elevations (horizontal planes through space) or plumb lines (vertical planes through space). They are not to be confused with theodolite systems that can also measure the angular pitch of the line of sight. They are similar to surveying equipment but with much higher measurement accuracies.

Optical alignment systems are perhaps one of the most versatile tools available for a wide variety of applications such as leveling foundations (e.g., see Figure 3.11), measuring OL2R machinery movement (covered in Chapter 16), checking for roll parallelism in paper and steel

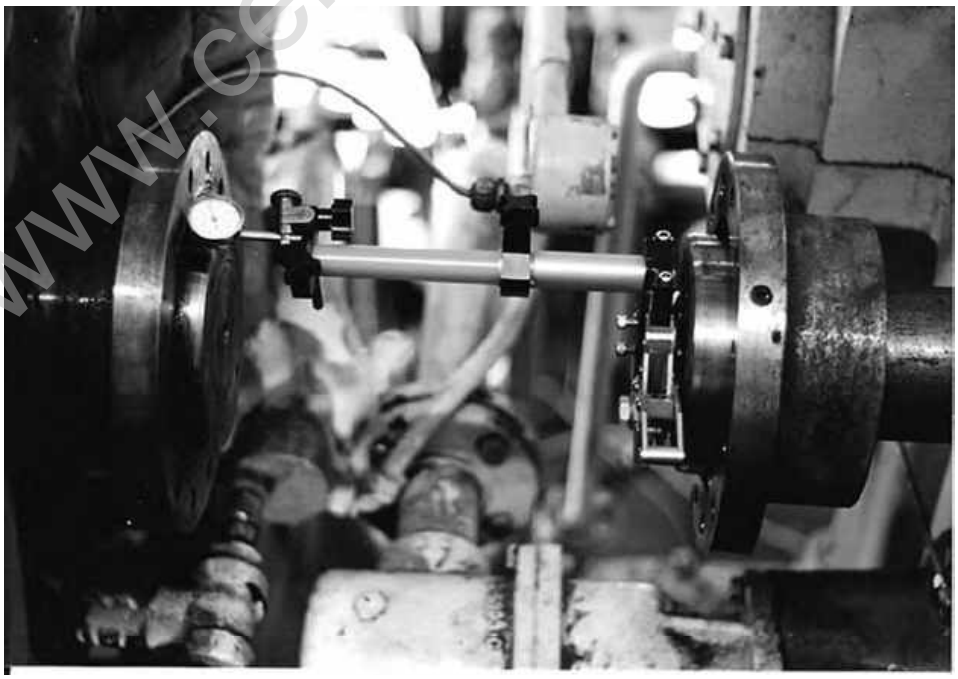


FIGURE 6.8 Dial indicator taking rim measurement on steam turbine shaft with bracket clamped onto end of compressor shaft.

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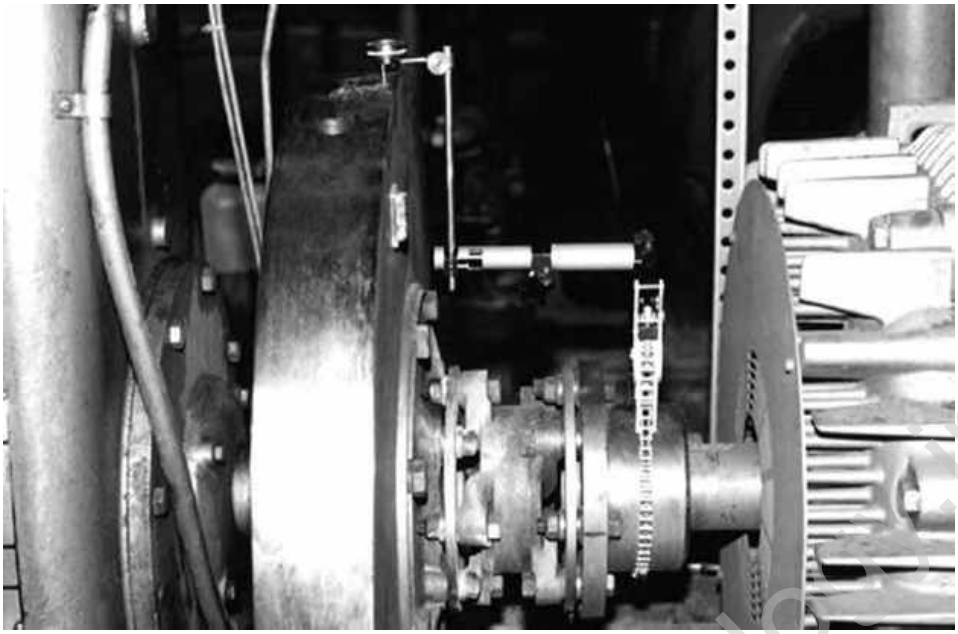


FIGURE 6.9 Dial indicator and bracket arrangement taking rim reading on a large flywheel.

manufacturing plants, aligning bores of cylindrical objects such as bearings or extruders, measuring flatness or surface profiles, checking for squareness on machine tools or frames, and will be discussed in Chapter 19. If you have a considerable amount of rotating machinery in your plant, it is highly recommended that someone examine all the potential applications for this extremely useful and accurate tooling.

Optical tooling levels and jig transits are one of the most versatile measurement systems available to determine rotating equipment movement. Figure 6.10 and Figure 6.11 show the



FIGURE 6.10 Optical tilting level and jig transit.

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FIGURE 6.11 Jig transit. (Courtesy of Brunson Instrument Co., Kansas City, MO. With permission.)

two most widely used optical instruments for machinery alignment. This tooling is extremely useful for leveling foundations, squaring frames, checking roll parallelism, and a plethora of other tasks involved in level, squareness, flatness, vertical straightness, etc.

The detail of a 3 in. scale target is shown in Figure 6.12. Optical scale targets can be purchased in a variety of standard lengths (3, 5, 10, 20, and 40 in.) and metric scales are available. The scale pattern is painted on invar bars to minimize thermal expansion or contraction of the scale target itself. The scale targets are held in position with magnetic base holders as shown in Figure 6.13 and Figure 6.14.

There are generally four sets of paired line sighting marks on the scales for centering of the crosshairs when viewing through the scope as shown in Figure 6.12. An optical micrometer, as shown in Figure 6.15, is attached to the end of the telescope barrel and can be positioned in either the horizontal or vertical direction. The micrometer adjustment wheel is used to align the crosshairs between the paired lines on the targets. When the micrometer wheel is rotated, the crosshair appears to move up and down along the scale target (or side to side

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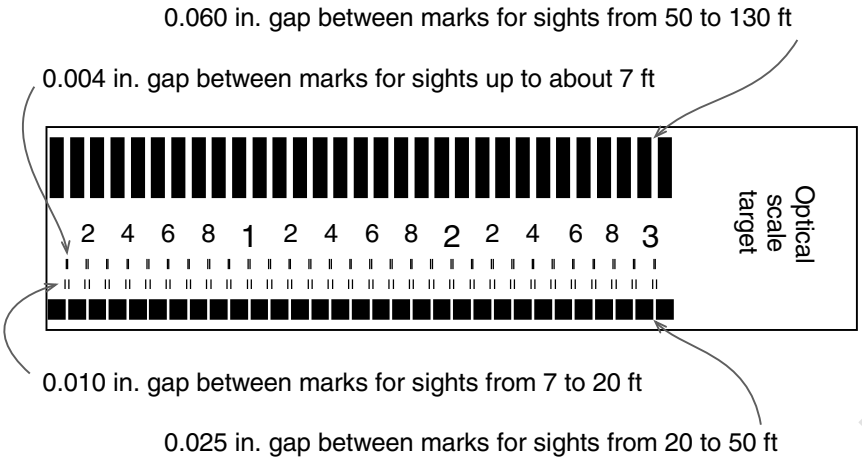


FIGURE 6.12 Three inch optical scale target.

depending on the position of the micrometer). Once the crosshair is lined up between a set of paired lines, a reading is taken based on where the crosshair is sighted on the scale and the position of the optical micrometer. The inch and tenths of an inch reading is visually taken by observing the scale target number where the crosshair aligns between a paired line set, and the hundredth and thousandths of an inch reading is taken on the micrometer drum as shown in Figure 6.16.

The extreme accuracy (one part in 200,000 or 0.001 in. at a distance of 200 in.) of the optical instrument is obtained by accurately leveling the scope using the split coincidence level mounted on the telescope barrel as shown in Figure 6.17.

6.2.7 Optical Parallax

As opposed to binoculars, 35 mm cameras, and microscopes that have one focusing adjustment, the optical scope has two focusing knobs. There is one knob used for obtaining a clear, sharp image of an object (e.g., the scale target) and another adjustment knob that is used to focus the crosshairs (reticle pattern). Since your eye can also change focus, adjust both these knobs so that your eye is relaxed when the object image and the superimposed crosshair image are focused on a target.
Adjusting the focusing knobs:

1. With your eye relaxed, aim at a plain white object at the same distance away as your scale target and adjust the eyepiece until the crosshair image is sharp.
2. Aim at the scale target and adjust the focus of the telescope.
3. Move your eye slightly sideways and then up and down to see if there is an apparent motion between the crosshairs and the target you are sighting. If so, defocus the telescope and adjust the eyepiece to refocus the object. Continue alternately adjusting the telescope focus and the eyepiece to eliminate this apparent motion.

Before using any optical instrument, it is highly recommended that a Peg Test be performed. The Peg Test is a check on the accuracy of the levelness of the instrument. Figure 6.18 shows how to perform the Peg Test.

Figure 6.19 and Figure 6.20 show the basic procedure on how to properly level the instrument. If there is any change in the split coincidence level bubble gap during the final check, go back and perform this level adjustment again. This might take a half an hour to an hour to get this right, but it is time well spent. It is also wise to walk away from the scope for about 30 min to determine if the location of the instrument is stable and to allow some time

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FIGURE 6.13 Scale targets mounted on an electric generator bearing.

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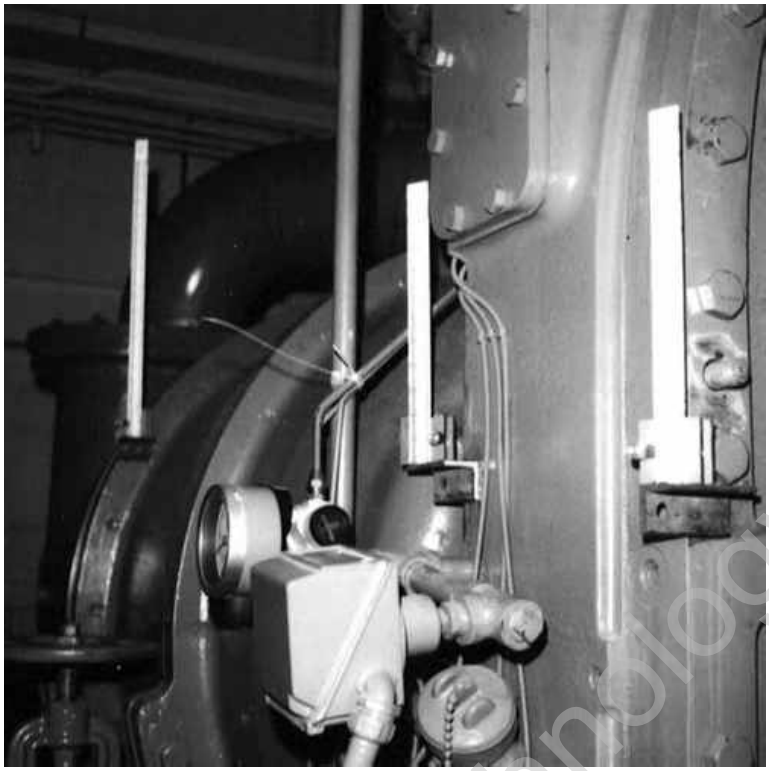


FIGURE 6.14 Scale targets mounted on compressor casing near their centerline of rotation.



FIGURE 6.15 An optical micrometer. (Courtesy of Brunson Instrument Co., Kansas City, MO. With permission.)

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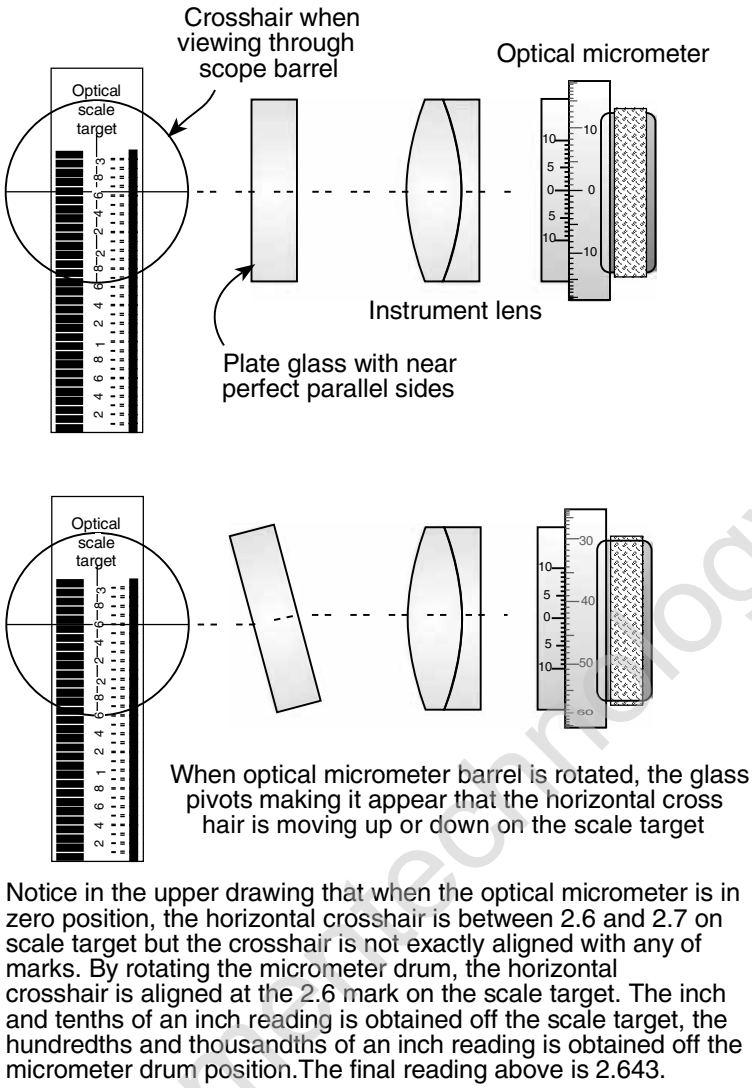


FIGURE 6.16 Principle of an optical micrometer.

for your eyes to uncross. If the split coincidence bubble has shifted during your absence, start looking for problems with the stand or what it is sitting on. Correct the problems and relevel the scope.

I cannot overemphasize the delicacy of this operation and this equipment. It is no place for people in a big hurry with little patience. If you take your time and are careful and attentive when obtaining your readings, the accuracy of this equipment will astonish you.

6.2.8 Proximity Probes

Proximity probes (also known as inductive pickups) as shown in Figure 6.21 and Figure 6.22 are basically noncontacting, electronic dial indicators. They are devices used to measure distance from the tip of the probe to a conductive surface. They are typically used to measure vibration (i.e., shaft motion) or thrust position and are usually permanently mounted to the machine. When used to measure vibration, the alternating current (AC) voltage from the probe is measured. When used to measure distance, the direct current (DC) voltage is measured.

Although the probes have been proposed for use as shaft alignment measuring devices, no company currently offers such a system for sale. Proximity probes can also be used to

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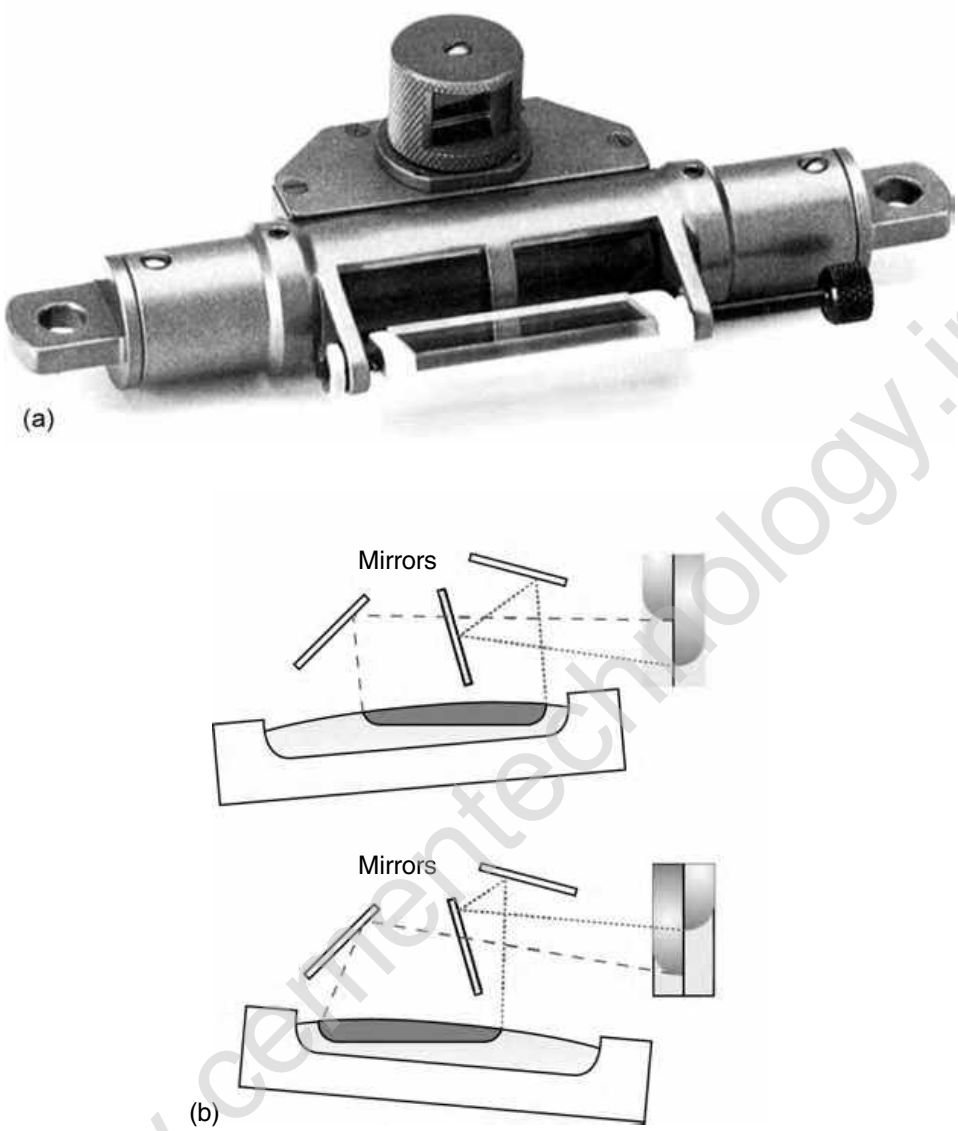


FIGURE 6.17 Principle of the coincidence level. (Courtesy of Brunson Instrument Co., Kansas City, MO. With permission.)

measure OL2R machinery movement in some very innovative ways as explained in Chapter 16. Their primary limitation is the range of useful distance measurement (approximately 50–150 mils) that can be attained with standard probes. Various sensitivities can be attained depending on the construction of the probe. Proximity probes frequently used as vibration sensors have either a 100 or 200 mV/mil sensitivity.

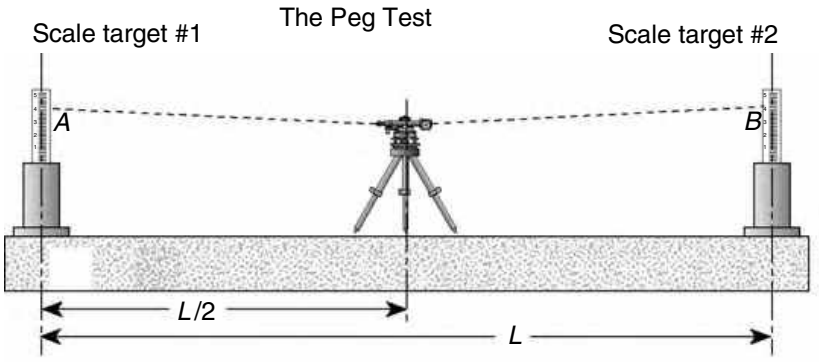
6.2.9 Linear Variable Differential Transformers

These devices are also called variable inductance transducers. They output an AC signal proportional to the position of a core that moves through the center of the transducer as illustrated in Figure 6.23 and Figure 6.24. These devices can attain accuracies of $\pm 1\%$ of full-scale range with stroke ranges available from 20 mils to over 20 in. No current

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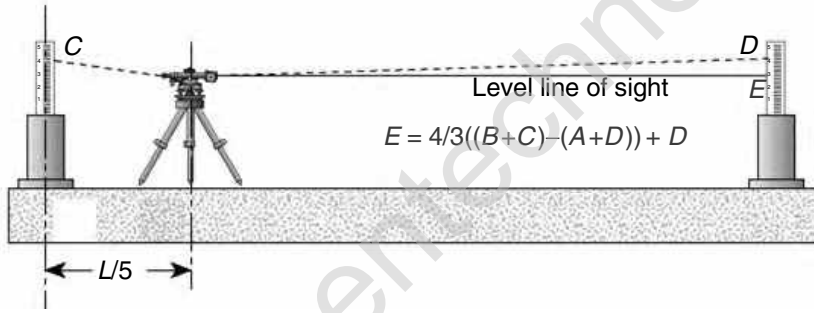
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Before using any optical instrument, it is recommended that the Peg Test be performed to insure measurement accuracy. At 40 ft, the accuracy of the scope is plus or minus 0.0024 in.



1. Set two scales apart by distance *L* (usually 40 ft) on stable platforms. Position the optical telescope or transit exactly half way between both scales. Accurately level the instrument using the split coincidence level.

2. Alternately take four readings on scale target #1 (reading *A*) and scale target #2 (reading *B*). Record and average these readings.



3. Move the scope to the 1/5 *L* position, level the scope, and alternately take four readings on scale target #1 (reading *C*) and scale target #2 (reading *D*). Record and average these readings.

4. If the instrument is calibrated, *A* minus *B* should equal *C* minus *D* (at 32 ft, this should be no more than 0.002 in.).

If the error is greater than that, adjust the split coincidence levels as follows:

A. Set the optical micrometer drum to the hundreths and thousandths value of *E*. For example, if *E* = 4.656, set the micrometer drum to 0.056.

B. Using the tilting screw, tilt the scope barrel to align the horizontal crosshair to the inch and tenths of an inch mark on scale target #2. For example, if *E* = 4.656, align the horizontal crosshair to the paired lines at 4.6. At this point, the split coincidence level will be not be coincident.

C. Adjust the nuts holding the split coincident level to the scope barrel to bring the bubble halves into coincidence.

D. Perform step 1 through step 4 above to verify that the adjustment worked. Should this not be the case, the coincidence level calibration adjustment nuts can be adjusted to position the leveled line of sight to be set at reading *E*.

FIGURE 6.18 Coincidence level calibration test (the Peg Test).

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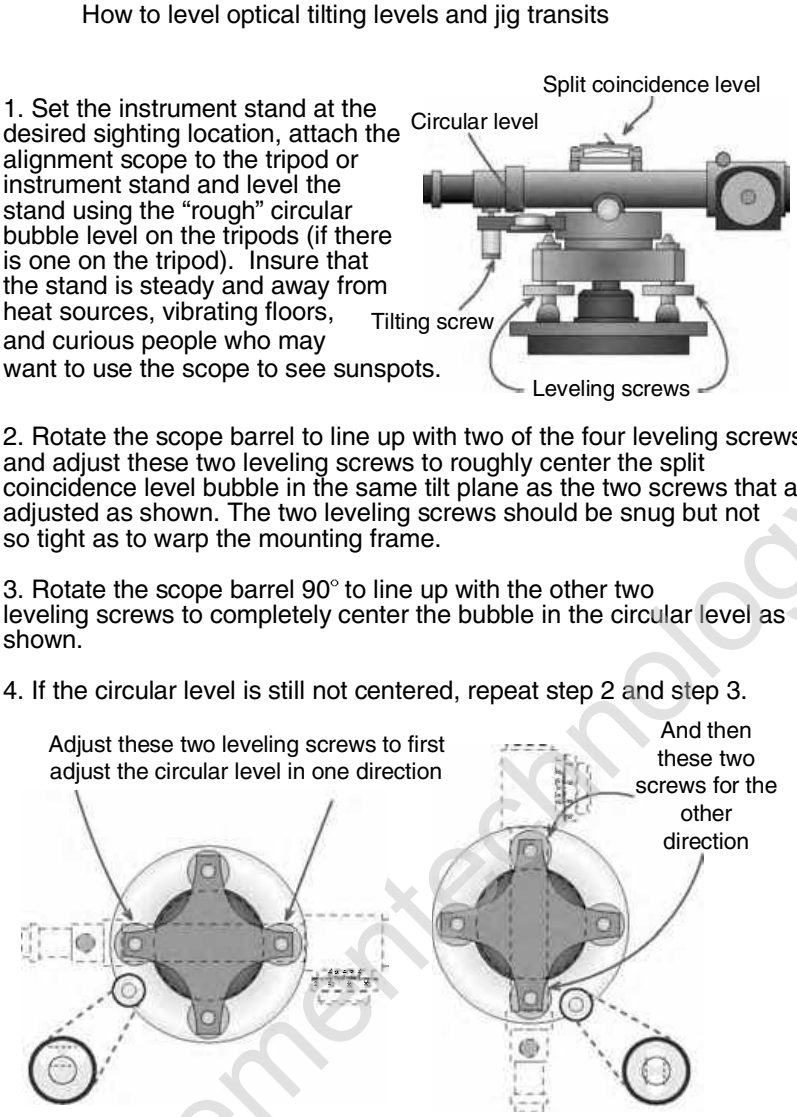


FIGURE 6.19 How to level a tilting level or jig transit, part 1 through part 4.

manufacturer of alignment measurement systems uses this type of transducer for shaft alignment purposes.

6.2.10 Optical Encoders

Optical encoders are essentially pulse counters as shown in Figure 6.25. They are most frequently used to measure shaft speed or shaft position and are therefore sometimes called shaft or rotational encoders. A series of slots are etched on a disk or flat strip. A light source (typically an light-emitting diode, LED) aims at the disk or flat strip and as the disk or strip is moved or rotated, a photodetector on the other side of the disk or strip counts the number of slots that are seen. One manufacturer currently uses this type of sensor for shaft alignment measurement.

6.2.11 Lasers and Detectors

With the advent of the microprocessor chip, the semiconductor junction laser, and silicon photodiodes, new inroads have been forged in the process of measuring small distances that utilize these new electronic devices instead of mechanical measuring instruments. Since the

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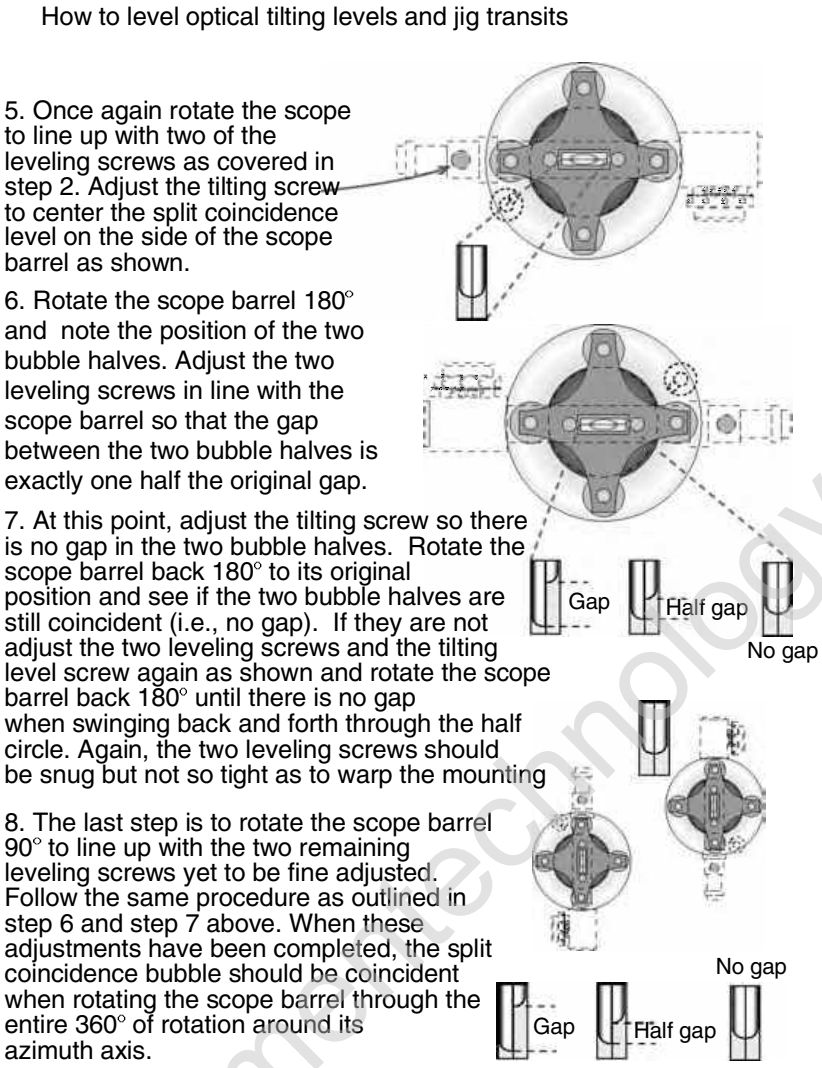


FIGURE 6.20 How to level a tilting level or jig transit, part 5 through part 8.

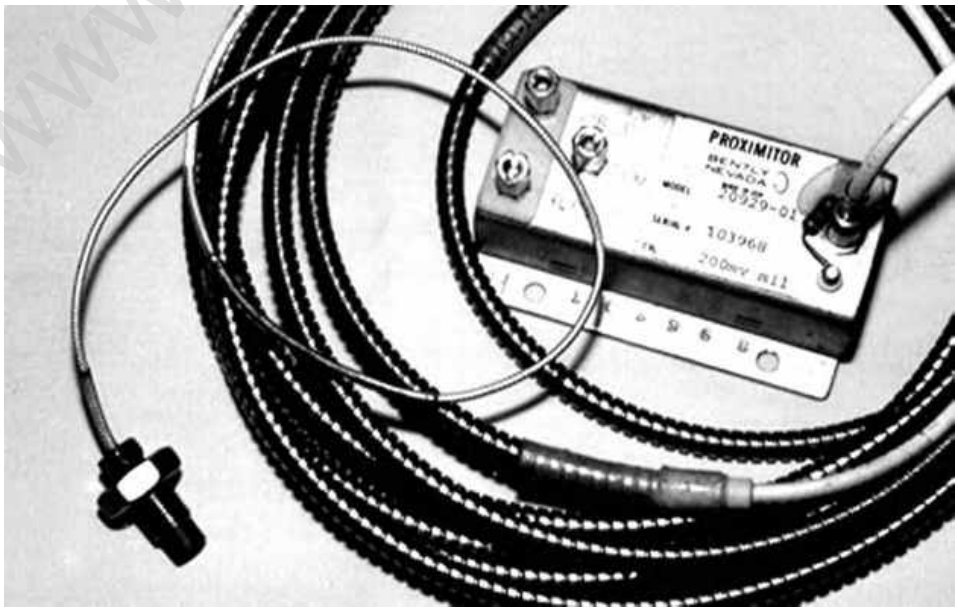


FIGURE 6.21 Proximity probe and oscillator–demodulator.

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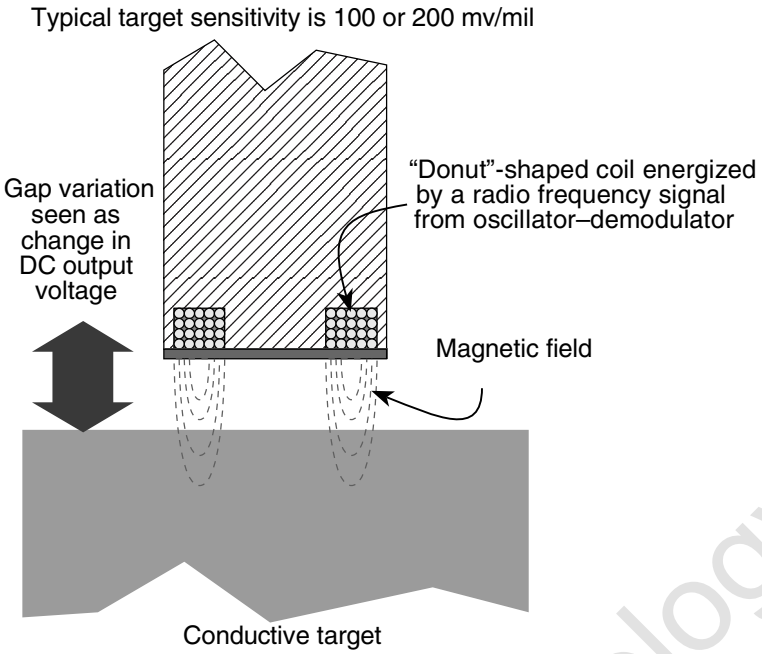


FIGURE 6.22 Basic operation of a proximity probe.

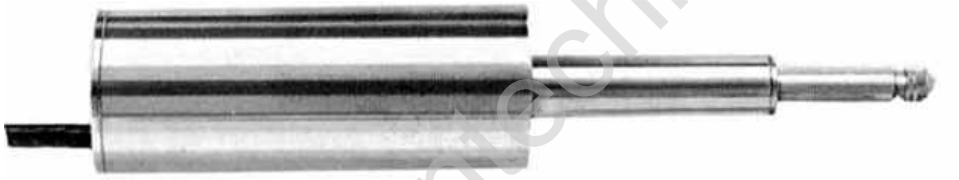


FIGURE 6.23 LVDT sensor.

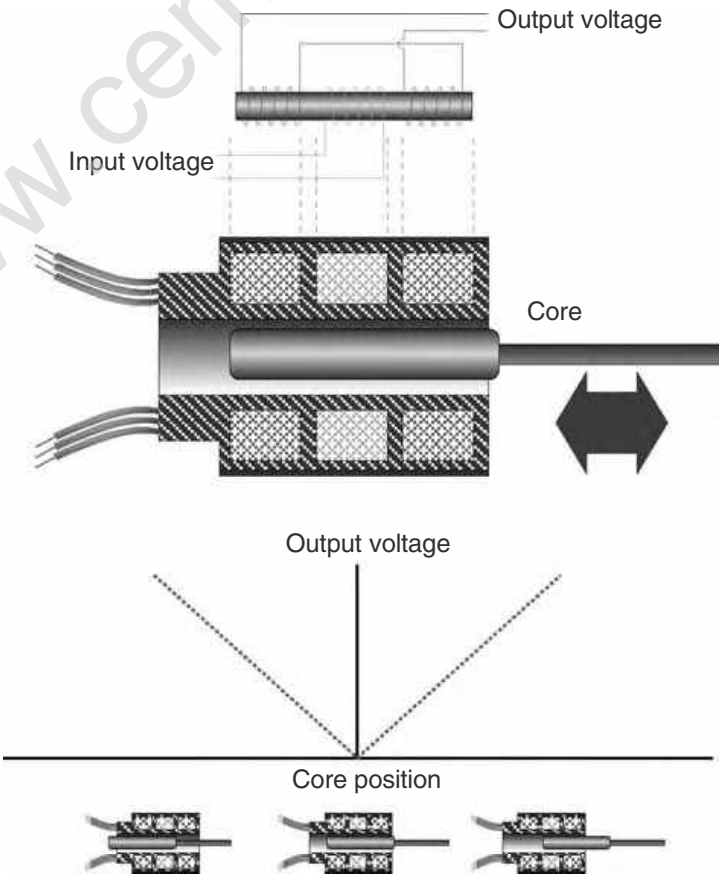


FIGURE 6.24 Basic operation of an LVDT.

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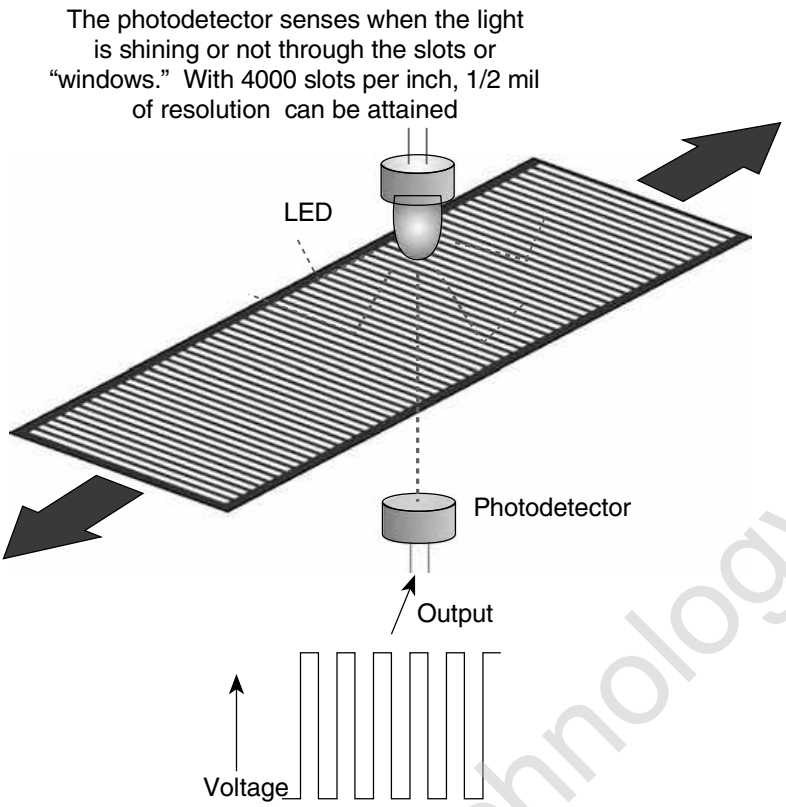


FIGURE 6.25 Basic operation of an optical encoder.

first useable laser shaft alignment measurement system was introduced in Germany in 1984, a host of manufacturers have introduced other laser shaft alignment systems. Since some of the manufacturers have taken slightly different approaches for using lasers and detectors, it will be beneficial to initially discuss some of the basic theory of operation of photonic semiconductors and how they are applied to mechanical measurements.

Useful terms:
Photonics: Field of electronics that involves semiconductor devices that emit and detect light.
Semiconductors: Typically silicon crystal doped (i.e., made impure) with other elements such as phosphorus (n-type due to five electrons in outer shell) or boron (p-type due to three electrons in outer shell). Depending on certain conditions, semiconductors can act as insulators or conductors.
LASER: Acronym for light amplified by stimulated emission of radiation.
LED: Acronym for light-emitting diode. All diodes emit some electromagnetic radiation when forward biased. When the forward current attains a certain level, called the threshold point, lasing action occurs in the semiconductor. Gallium–arsenide–phosphide diodes emit much more radiation than silicon-type diodes and are typically used in semiconductor junction diode lasers.
Photodiode: All diodes respond when subjected to light (electromagnetic radiation). Silicon diodes respond very well to light and are typically used to detect the presence or position of light as it impinges on the surface of the diode.

Figure 6.26 shows the broad frequency range of the electromagnetic spectrum. The human eye can detect but a very small range of frequencies from 400 to 700 nm. Figure 6.27 illustrates the basic operation of semiconductor junction laser diodes. As current is passed through the diode, photons (light) are emitted in the junction region as electrons move from a higher

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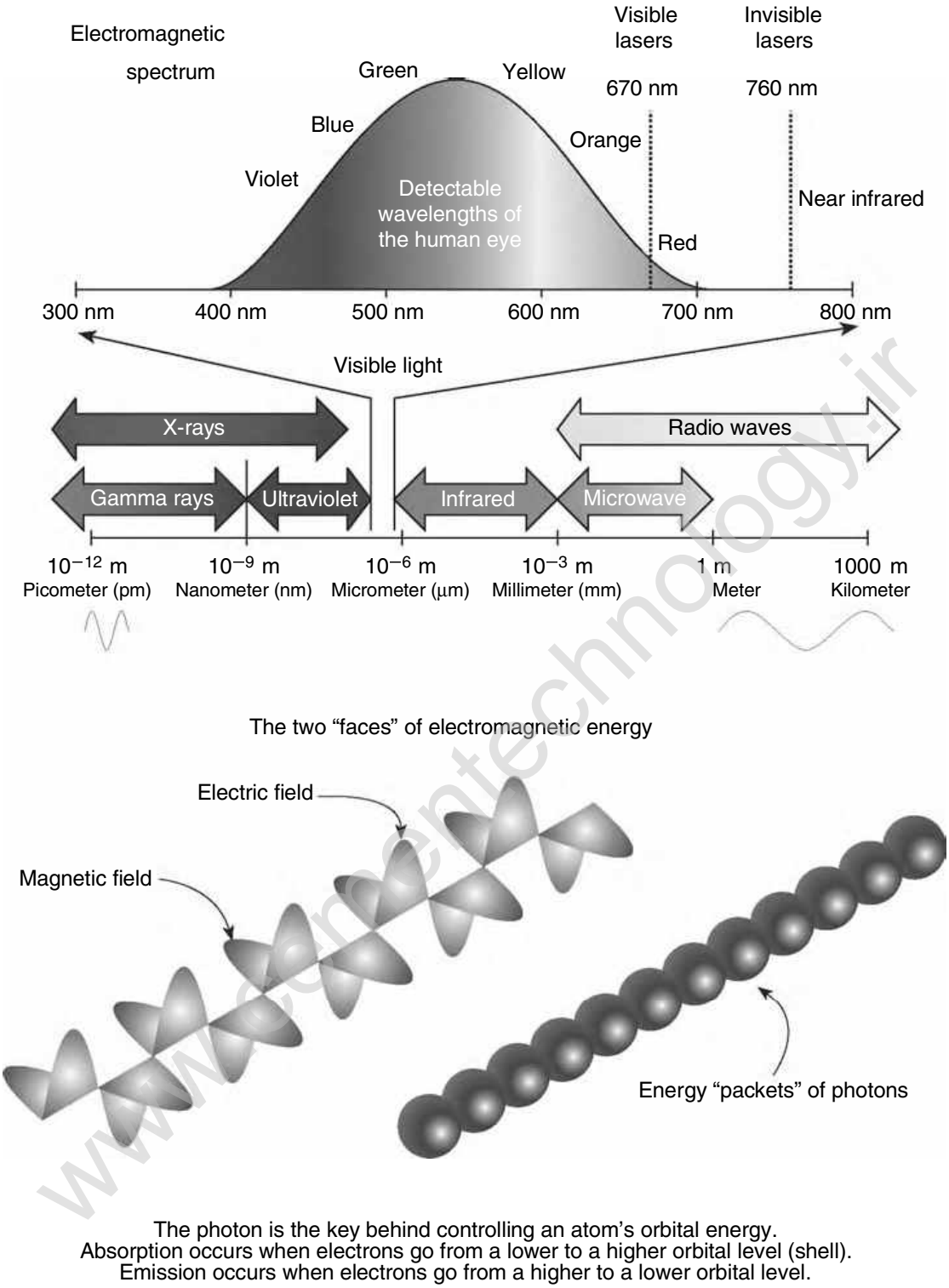


FIGURE 6.26 (**See color insert following page 322.**) The electromagnetic spectrum.

orbital shell to a lower one, giving up energy in the form of quanta (photons) in the process. By altering the chemical composition of the semiconductor, the wavelength of the light emitted from the semiconductor can be shifted to different frequencies.

The first lasers used in shaft alignment measurement systems emitted light at a wavelength of 760 nm, outside the visible range of human sight. The lasers currently used in alignment now emit a red light (670 nm), which is within the visible range of human sight. The beam of light that is emitted from the laser is not a thin strand of light 1 μm in diameter. Instead it is

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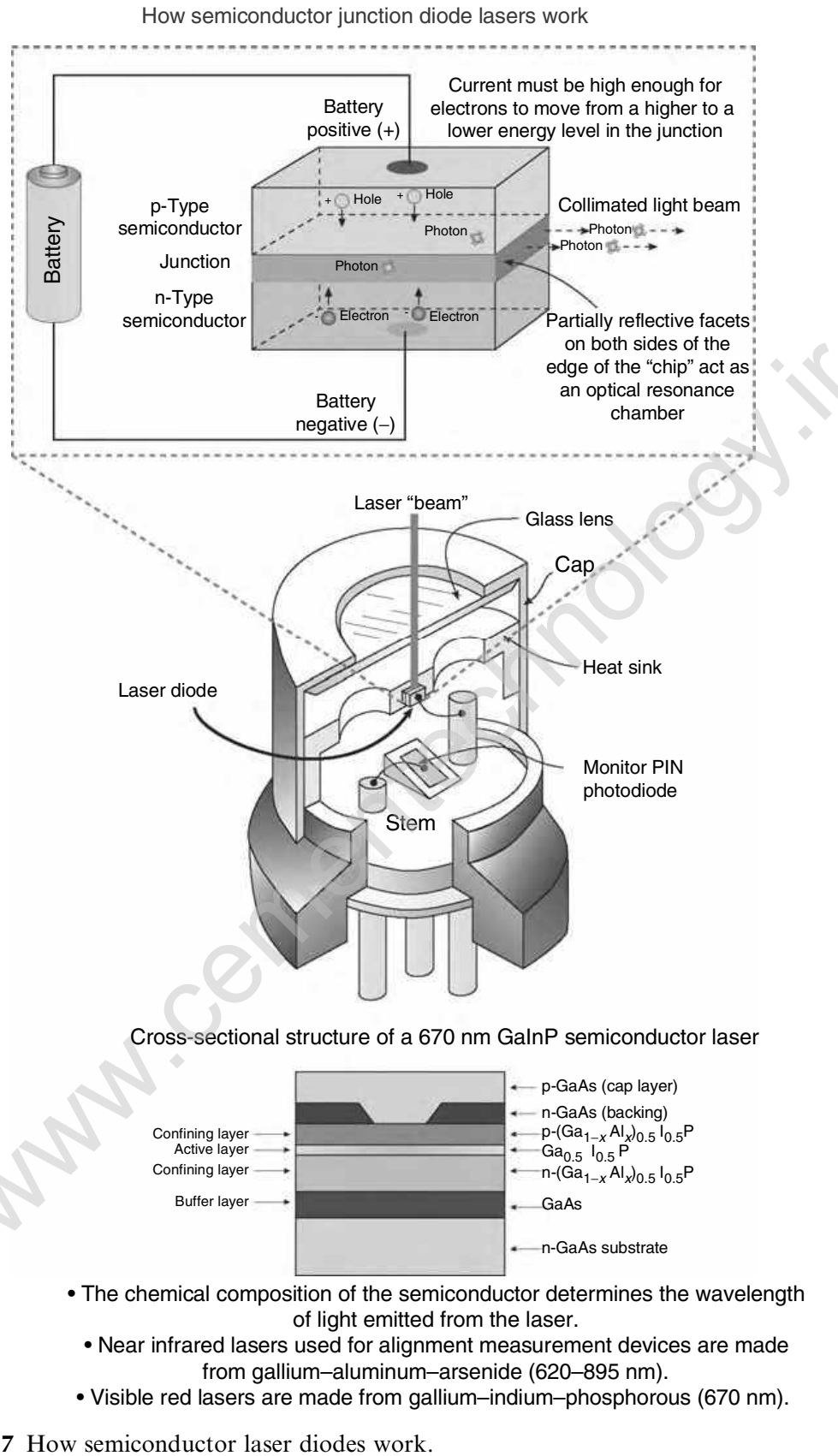


FIGURE 6.27 How semiconductor laser diodes work.

about 1.5 mm (approximately 60 mils) in diameter as it exits the diode and is collimated (i.e., "focused"), since only one side of the diode actually allows the light to exit. After exiting the diode, if the light beam was in a pure vacuum, the beam would stay focused for long distances. However, since there are small molecules of water vapor in the air we breathe,

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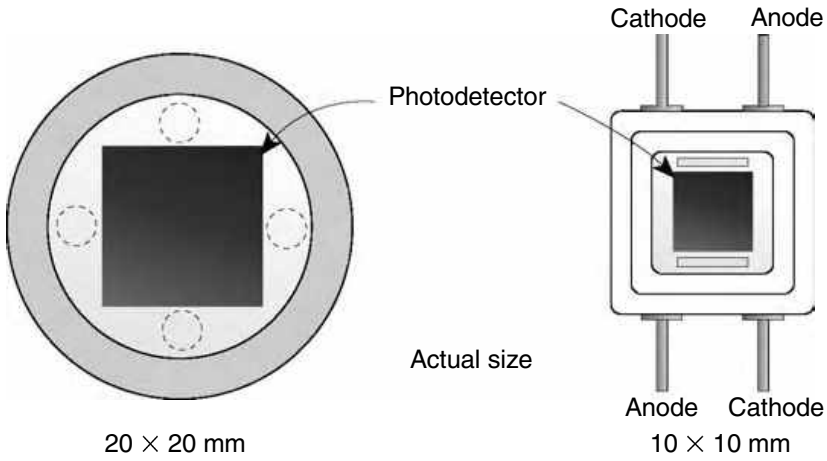


FIGURE 6.28 How photodiodes work.

the light from the laser is diffracted as it passes through each molecule of water vapor diffusing the beam. Typically, the useable distance of laser is somewhat limited to 30 ft due to the diffraction of the beam. Since the laser beam is around 60 mils in diameter as it exits the diode, the measurement accuracy would only be 60 mils (i.e., about 1/16th of an inch) if just the laser beam were solely used as the measurement device. This accuracy is just fine for laser levels when constructing buildings, for example, but since we are looking for accuracies of measurement at 1 mil or better, another device is needed in concert with the laser to attain this measurement precision. That device is the beam detector target.

Laser–detector systems are also semiconductor photodiodes capable of detecting electromagnetic radiation (light) from 350 to 1100 nm. When light within this range of wavelengths strikes the surface of the photodiode, an electrical current is produced as shown in Figure 6.28. Since the laser beam is emitting light at a specific wavelength (e.g., 670 nm), a colored translucent filter is positioned in front of the diode target to hopefully allow only light in the laser's wavelength to enter. Otherwise, the detector could not tell whether the light that was striking its surface was from the laser, overhead building lighting, a flashlight, or the sun.

As shown in Figure 6.29, when light strikes the center of the detector, output currents from each cell are equal. As the beam moves across the surface of the photodiode, a current imbalance occurs, indicating the off-center position of the beam. Most manufacturers of laser–detector shaft alignment systems use 10 × 10 mm detectors (approximately 3/8 sq. in.); a few may use 20 × 20 mm detectors. Some manufacturers of these systems use bicell (unidirectional) or quadrant cell (bidirectional) photodiodes to detect the position of the laser beam. An unidirectional photodiode measures the beam position within the target area from left to right only whereas a bidirectional photodiode (Figure 6.30 and Figure 6.31) measures the beam position in both axes, left to right and top to bottom. Therefore, laser–detector systems measure the distance the laser beam has traversed across the surface of the detector by measuring the electrical current at the beam's starting position and the electrical current at the beam's finishing position.

6.2.12 Charge Couple Devices

The CCD was originally proposed by Boyle and Smith in 1970 as an electrical equivalent to magnetic bubble digital storage devices. The basic principle of their device was to store information in the form of electrical "charge packets" in potential wells created in the semiconductor by the influence of overlying electrodes separated from the semiconductor

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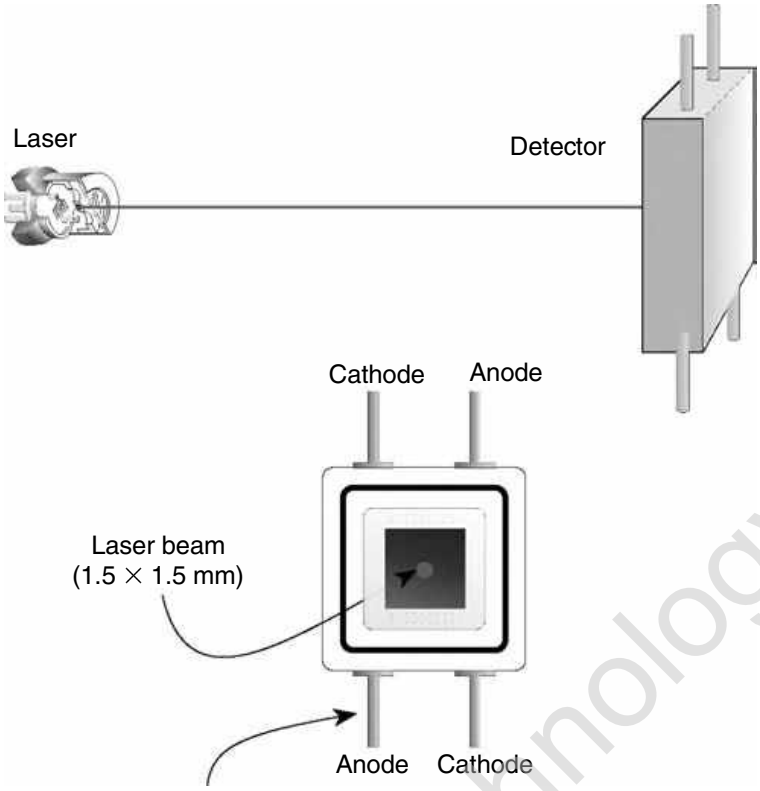


FIGURE 6.29 Laser–photodiode operation.

by a thin-insulating layer. By controlling voltages applied to the electrodes, the potential wells and hence the charge packets could be shifted through the semiconductor (Figure 6.32).

The potential wells are capable of storing variable amounts of charge and can be introduced electrically or optically. Light impinging on the surface of the charge-coupled semiconductor generates charge carriers, which can be collected in the potential wells and afterward clocked out of the structure enabling the CCD to act as an image sensor.

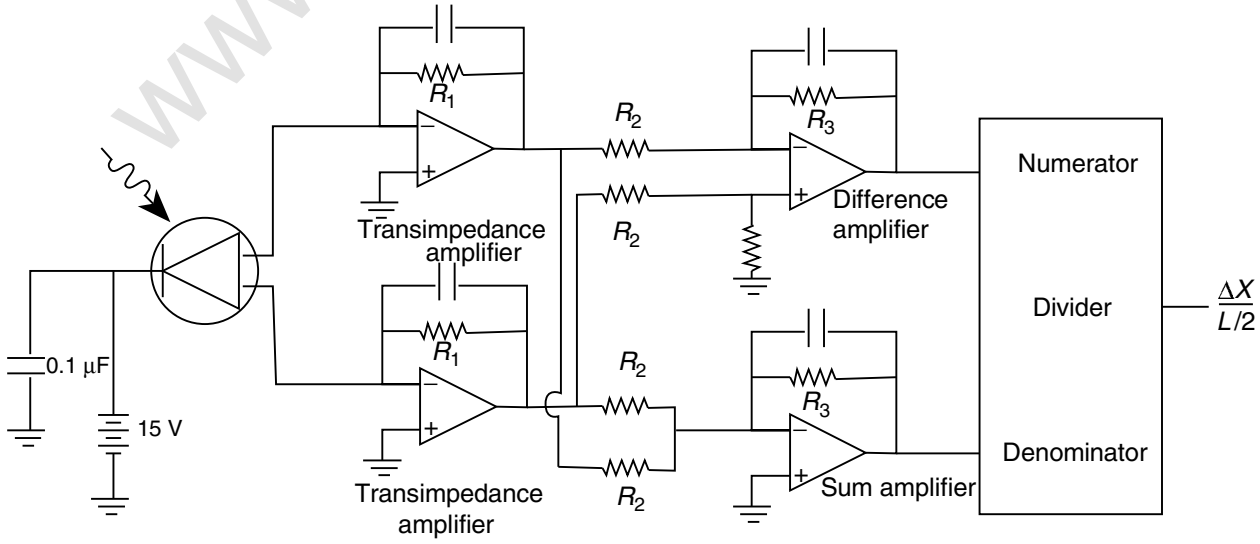


FIGURE 6.30 Typical single axis photodiode circuit.

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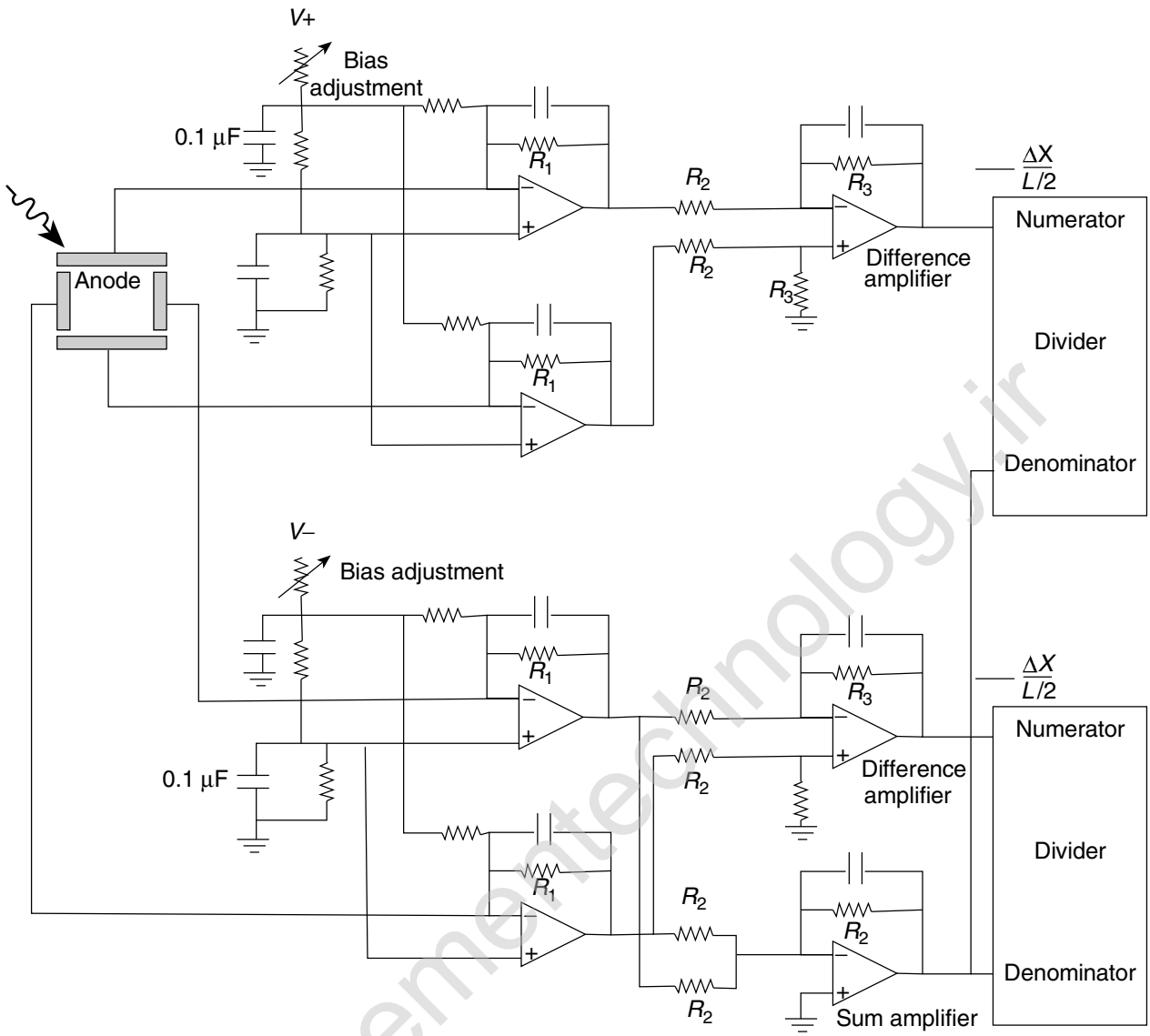


FIGURE 6.31 Typical dual axis photodiode circuit.

A considerable amount of effort was put forth in the 1960s in developing optical imagers that utilized matrices of photodiodes that effectively became undone by the development of the CCD. The rate of progress in CCD design through 1974 was so astonishing that Rodgers demonstrated a 320×512 bit CCD sensor that could be used for 525 line television imaging just 4 years after the CCD was invented. CCDs have found their way into everyday life in video cameras and in high technology fields such as astronomy where large area CCDs capture images in telescopes both in orbit and on Earth.

With the recent pace of introducing electronic measurement sensors in the arena of alignment, it seems odd that no one has incorporated the CCD as a measurement sensor. The only known application of CCDs for use in alignment was presented as a doctoral thesis by Brad Carman and a research project at the University of Calgary (see references).

6.2.13 Interferometers

It is suggested that one has to study Figure 2.10 through Figure 2.12 to get a basic understanding of amplitude and frequency. Although the discussion in Chapter 2 for these figures

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A CCD is a multilayered silicon chip. In one layer, an array of electrodes divides the surface into pixels. Each electrode is connected to leads, which carry a voltage. The image forms on the silicon substrate. Light particles pass through the CCD freeing electrons in the silicon substrate. The voltage applied to the leads draws freed electrons together in special areas in the silicon substrate, called photo sites. The number of gathering electrons at the photo site is dependent on the intensity of the light striking in that area. The CCD transfers captured electrons, one by one, to an analog to digital converter, which assigns each site a digital value corresponding to the number of electrons a site holds. The number of electrons at each site determines how light or dark each pixel in the image is.

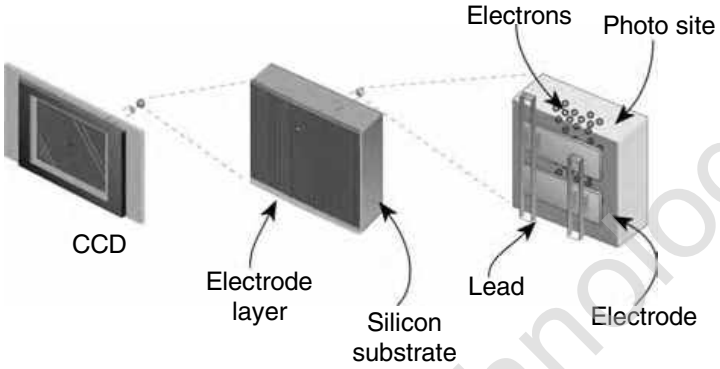


FIGURE 6.32 How a charge-coupled device (CCD) works.

centers around vibration, the same principles can also be applied to sound or light. Also Figure 6.26 explains the electromagnetic spectrum.

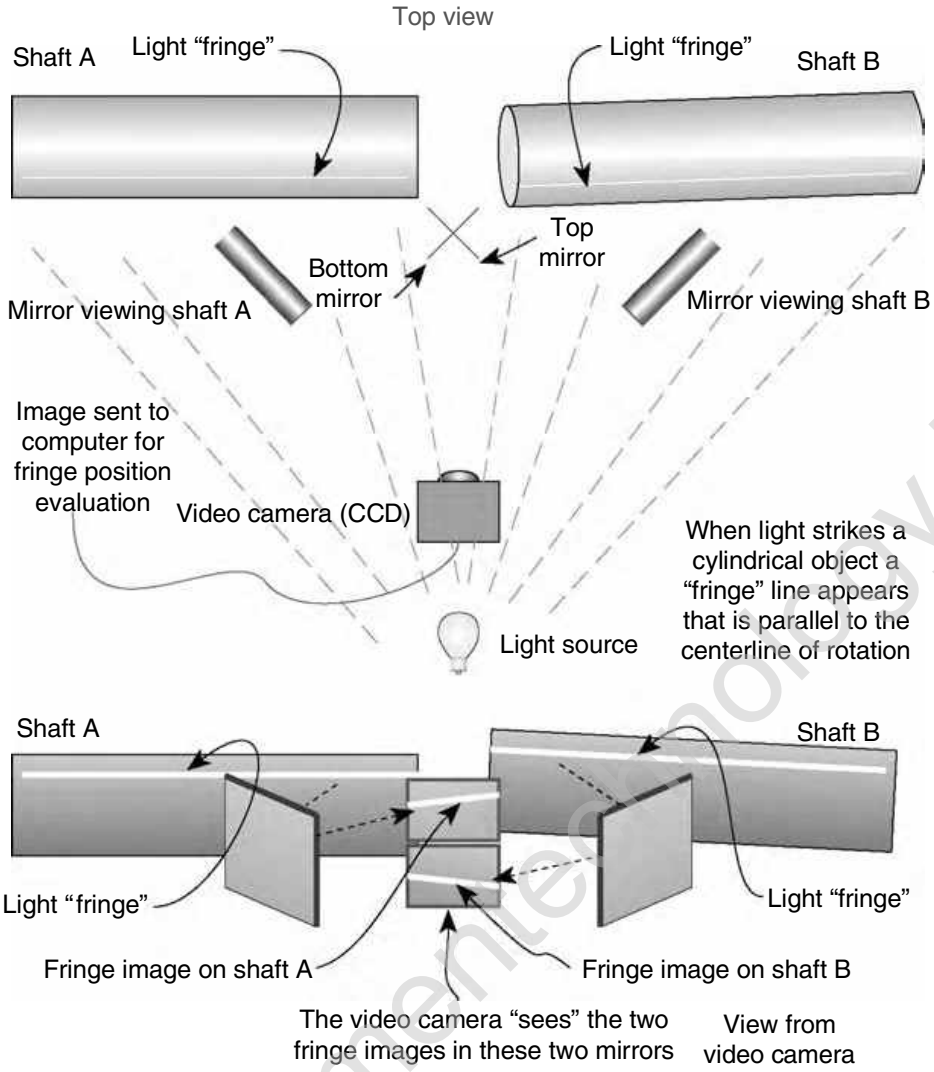
Interferometers are instruments that utilize monochromatic (i.e., single wavelength) beams of light to measure distance by utilizing the principle of interference of waves. When two signals of the same frequency combine and are in phase, the amplitude of the combined signal intensifies. However, when two signals of the same frequency combine and are exactly 180° out of phase, the two signals cancel each other out. This is referred to as constructive or destructive interference and is the basis of the field of interferometry. Since the wavelength of light is very small, small amounts of distance can be measured very accurately with these devices. Linear resolutions of 0.0059 μin. (0.15 nm) and angular resolutions of 0.005 arc seconds can be measured with these systems. Not only can these systems measure distance, but using the Doppler effect, they can also measure the speed of the object. Distance measuring interferometers work on two principles:

1. Homodyne interferometers count fringes. A fringe is defined as one full cycle of light variation, that is, from light to dark and back to light again, a full 360° phase shift in the two signals.
2. Heterodyne interferometers measure the change in optical phase of the known frequency of a reference signal to the known, but different frequency of a measurement signal at defined time intervals.

Although interferometers are not used in the area of shaft alignment, they are frequently used in the field of metrology. Figure 6.34 shows the basic operating principles of a Michelson interferometer.

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Basic procedure

Since the relative mirror positions are unknown during setup, a reference line is obtained by placing a straight cylinder (possessing a continuous straight fringe line) in front of the two mirrors establishing a reference line (and reference fringe positions). An image of the actual shaft fringes is then compared to the reference fringe lines to determine the relative displacements and slopes between the two shafts.

FIGURE 6.33 Using a CCD to detect light fringes on shafts.

6.3 SWEEPING 90° ARCS TWICE TO MEASURE A MISALIGNMENT CONDITION

To determine where a centerline of rotation of a shaft is, a bracket or holding fixture is attached to one of the shafts. A beam attached to that bracket spans across the gap between the two shafts and holds a precision measuring device (e.g., a dial indicator). The measuring device is positioned to observe the other shaft's position by hand rotating the shaft, bracket, and attached measuring device through a full rotation (i.e., 360°) around the surface of the observed shaft. As the sweep is made, the shaft, bracket, and attached measuring device are temporarily stopped at 90° intervals and a measurement is recorded on the observed shaft.

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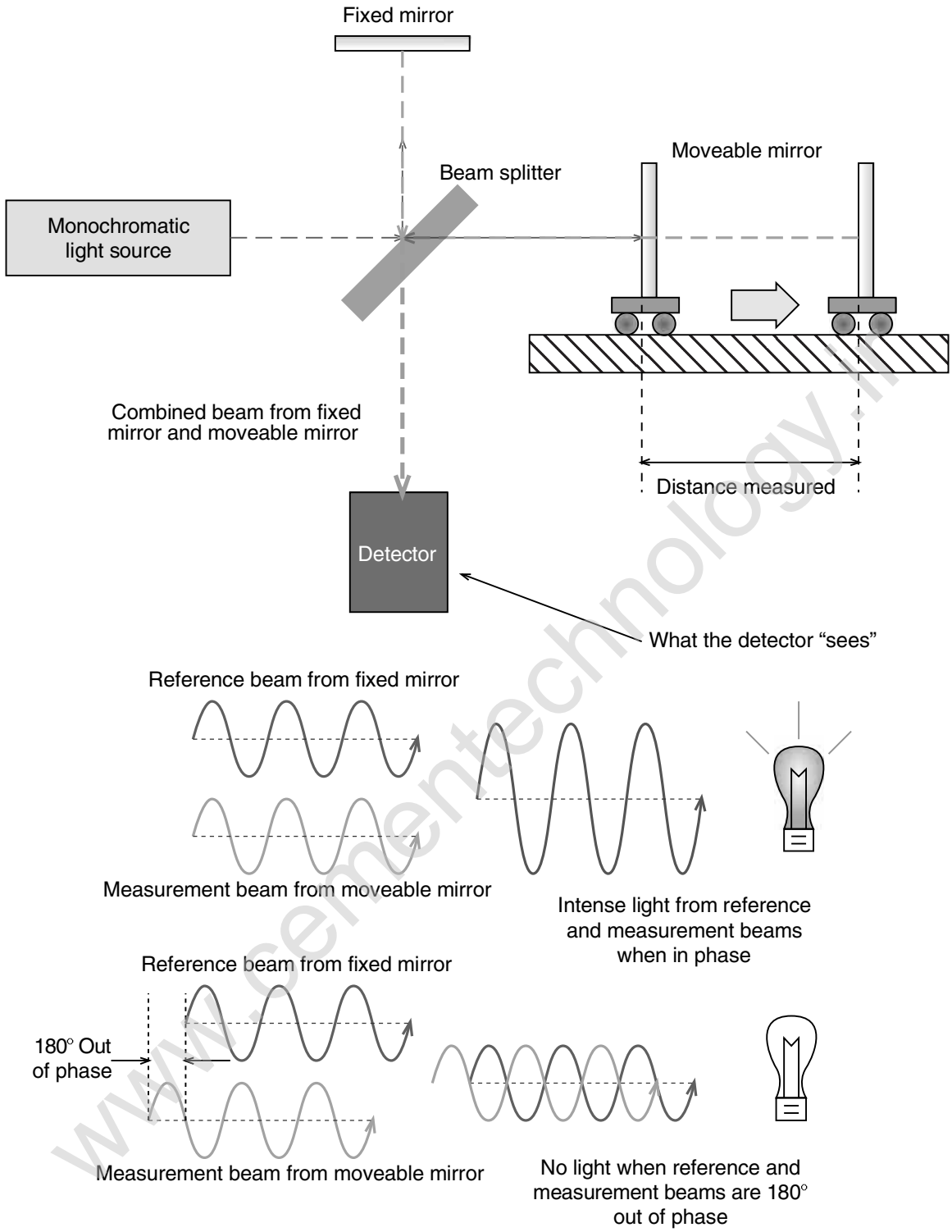


FIGURE 6.34 How an interferometer works.

If the centerline of rotation of the shaft, bracket, and attached measuring device is directly in line with the centerline of rotation of the observed shaft where the measuring device has been placed, there will be no observed displacement in the measurement device. If the centerline of rotation of the shaft, bracket, and attached measuring device is not directly in line with the centerline of rotation of the observed shaft where the measuring device has been placed, there will be observed displacement in the measurement device.

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In horizontally mounted rotating machinery, we are trying to determine if the observed centerline of the shaft appears to be higher or lower than that of the shaft the bracket is attached to. We are also trying to determine if the observed shaft appears to be to the left or right of the shaft the bracket is attached to.

When the measuring device traverses from the top to the bottom of the observed shaft, the amount and the direction of the measured displacement tell us whether the centerline of the observed shaft is higher or lower than the shaft we have attached to and how high it is or how low it is at that plane of measurement. Therefore we are looking for two pieces of information from this measurement, which way is the shaft? (i.e., is it high or low?) and how far off it is? (i.e., how much high or low?). In other words, this is a vector problem. We need both an amount and a direction.

Similarly, when the measuring device traverses from the left side to the right side of the observed shaft, the amount and the direction of the measured displacement tells us whether the observed shaft is to the left or right with respect to the shaft it is attached to and how far to the left or right it is. Since several alignment methods require that these measurements be taken from both shafts, it is recommended that compass directions (i.e., north, south, east, west) be used when recording the side measurements rather than left or right or three o'clock and nine o'clock to eliminate some possible confusion on which way is which. If you are working on a seagoing vessel, then port, starboard, fore, and aft would be appropriate. One popular convention is to stand at the outboard end of the driver looking toward the drive system referencing everything to your left and right from that vantage point. That works fine until you encounter a multiple element drive train where the driver shaft is rotating machinery on both ends of the driver. Where is the outboard end then? Top, bottom, left, and right does not work very well when we are aligning vertically oriented shafts either but compass directions do. Anyway the orientation is designated as just fine as long as you stay consistent.

Figure 6.35 and Figure 6.36 show rim measurements taken at 90° intervals from a pump shaft to a motor shaft. Figure 6.35 shows a dial indicator placed at the top dead center position and plunged down approximately half of its total stem travel. This is the usual starting point of the sweep and it is convenient to zero the indicator at this position. It is



FIGURE 6.35 Dial indicator positioned at twelve o'clock and zeroed.

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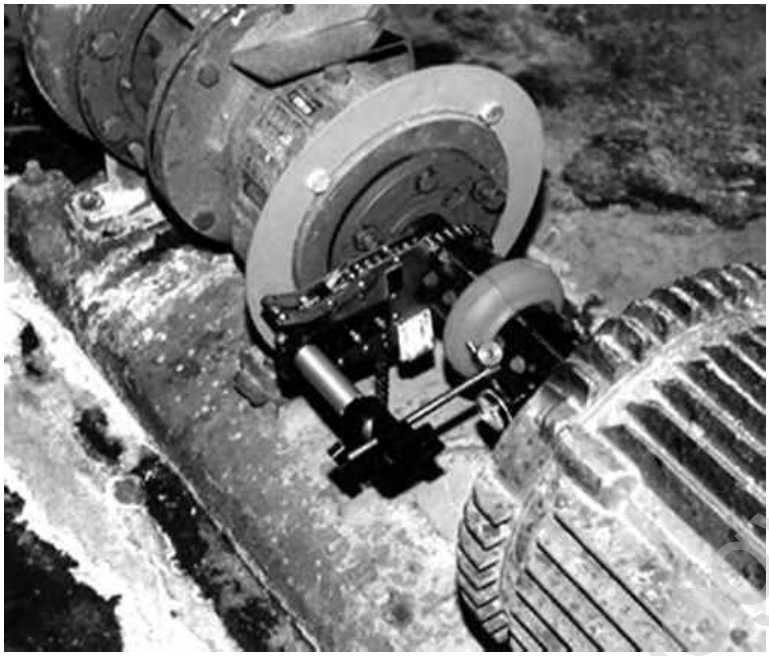


FIGURE 6.36 Pump shaft with bracket and indicator rotated 90° to side of motor shaft, stopped, and indicator reading recorded.

noticed that there is a dual spirit level attached to the pump shaft and one of the two levels has been centered here. This dual spirit level is used to stop the rotation at fairly precise 90° arcs to record the measurements at each 90° location. Figure 6.36 through Figure 6.38 show measurements taken at the remaining three other 90° locations.

The measurement sweep taken from the pump to the motor shown in Figure 6.35 through Figure 6.38 is not an enough information to determine where both shafts are however. Another measurement must be taken to ascertain where the centerlines of each shaft are located.

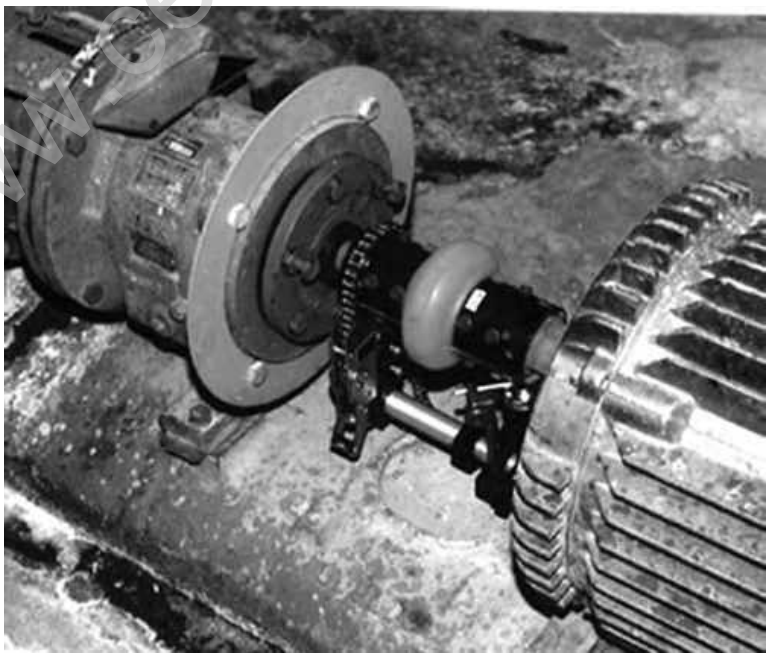


FIGURE 6.37 Pump shaft with bracket and indicator rotated 90° to bottom of motor shaft, stopped, and indicator reading recorded.

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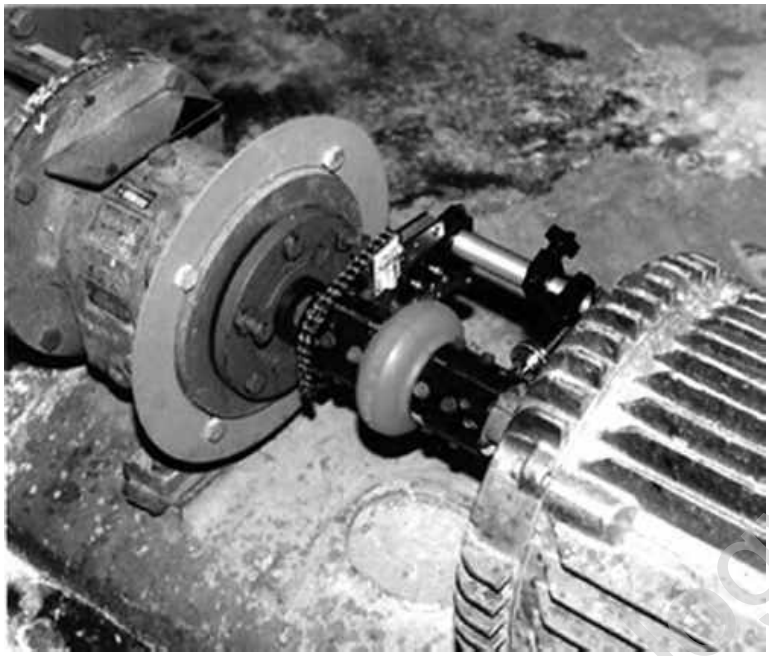


FIGURE 6.38 Pump shaft with bracket and indicator rotated 90° to other side of motor shaft, stopped, and indicator reading recorded.

6.4 WHY MEASUREMENTS ARE TAKEN AT 90° INTERVALS

Invariably the question arises as to why readings are taken at the twelve-, three-, six-, and nine o'clock positions on rotating machinery shafts.

In horizontally mounted rotating equipment, adjustments are made to the machinery cases to align the shafts in two planes, the up and down plane to correct the misalignment in the vertical position and the side-to-side plane to correct the misalignment in the sideways or lateral position. Vertical adjustments made to horizontally mounted rotating machinery casings are based on the top and bottom measurements (i.e., twelve o'clock and six o'clock). Lateral adjustments are made to horizontally mounted rotating machinery casings based on the north and south or east and west measurements (i.e., three o'clock and nine o'clock). In other words, when adjusting the height or pitch of the machinery cases, the side readings (three o'clock and nine o'clock) do not mean anything, only the top and bottom readings indicate the vertical position. Likewise, when adjusting the side-to-side positions of the machinery casings, the top and bottom readings are not regarded, only the side readings are considered. Shaft positional measurements are taken in the planes that define the directions of movement the machinery casings will undergo to correct the misalignment condition.

In vertically oriented rotating machinery, however, it becomes obvious that there will not be any "top" and "bottom" measurements. In this case, one must determine what the planes of movement or translation will be on the machinery cases and capture the measurements in those planes. Examples of aligning vertically oriented shafts will be covered later in the book.

6.5 ROTATING BOTH SHAFTS TO OVERRIDE A RUNOUT CONDITION

Chapter 5 covered information on measuring runout on rotating machinery. A runout condition will affect the accuracy of the alignment measurements and certain precautions need to be taken to insure that the alignment measurement process only finds the centerlines of rotation of the shafts and ignores any runout that may be present. If you have dial

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indicators to measure the positions of the shafts and the dial indicator is positioned on a coupling hub that is bored off center or skew bored or if the shaft is bent, the indicator cannot determine how much of what it is measuring is due to a runout condition and how much is due to a misalignment condition. Can you still align the centerlines of rotation if the surfaces you are capturing readings on are eccentric?

To override any runout problems that exist on coupling hubs or shafts, you must insure that when you observe and record the measurements at each 90° position, the dial indicator is positioned in the same angular location. If possible, just rotate both shafts together or you could do what is called the "lead, then follow" rotation. To do this, scribe a reference mark on the shaft or coupling hub with a marker, pen, or soapstone in the twelve o'clock position, place the tip of the indicator on this mark, and then zero the indicator. If the coupling is disengaged, you may then rotate the bracket and indicator 90° to your first reading position (i.e., three o'clock or nine o'clock) but do not record the measurement until you rotate the shaft where the indicator was taking the measurement position 90° to match your scribed reference mark backup with the tip of the indicator. Figure 6.39 illustrates what will happen if you have a runout condition and you only rotate one of the shafts when taking readings. Keep in mind that taking readings at the same angular position on the shafts will override any

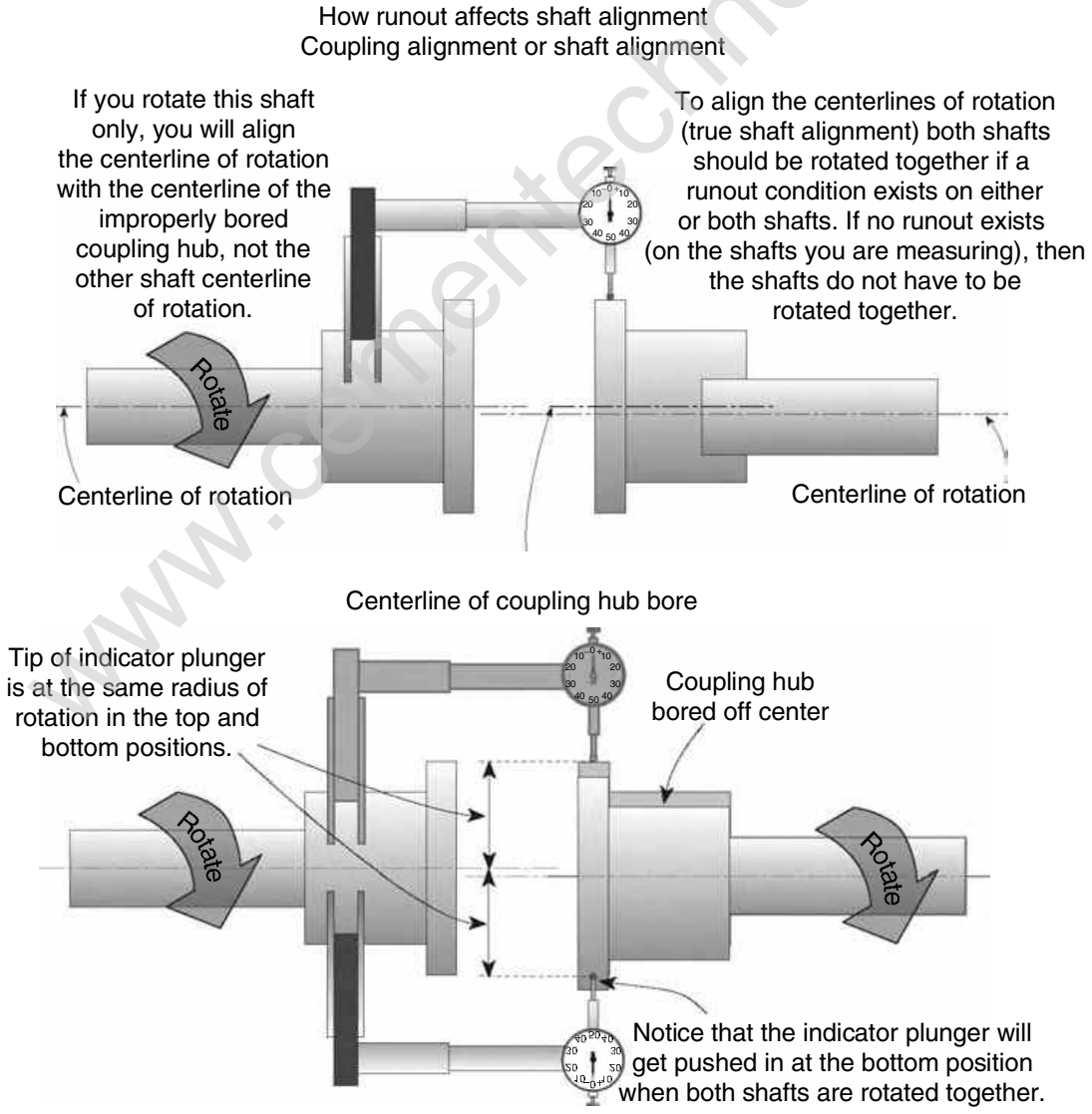


FIGURE 6.39 Why you should rotate both shafts to override a runout condition.

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runout that exists. This does not suggest that runout conditions should be ignored. If excessive amounts of runout exist, they should always be corrected before you ever begin to align the equipment.

6.6 TIPS FOR GETTING GOOD ALIGNMENT MEASUREMENTS

It is important that the measurements we take from shaft to shaft be as accurate as possible since we are going to use these measurements to correct a misalignment condition on the drive system. If we get bad measurements we will get bad results when trying to correct the problem. Figure 6.40 shows some tips for insuring that you are capturing good measurements. The upper drawing effectively embellishes the saying of an experienced carpenter: ''Measure twice, cut once.'' If after two complete sweeps, the measurements at each 90° location do not repeat (within ±2 to 3 mils), something is wrong and you should investigate why the readings are not repeating. There may be something loose on your bracket or fixture, the indicator stem may be sticking, or you may be reading the indicator incorrectly. There is also a possibility that the bearings supporting the shafts are damaged. It may be wise to do a

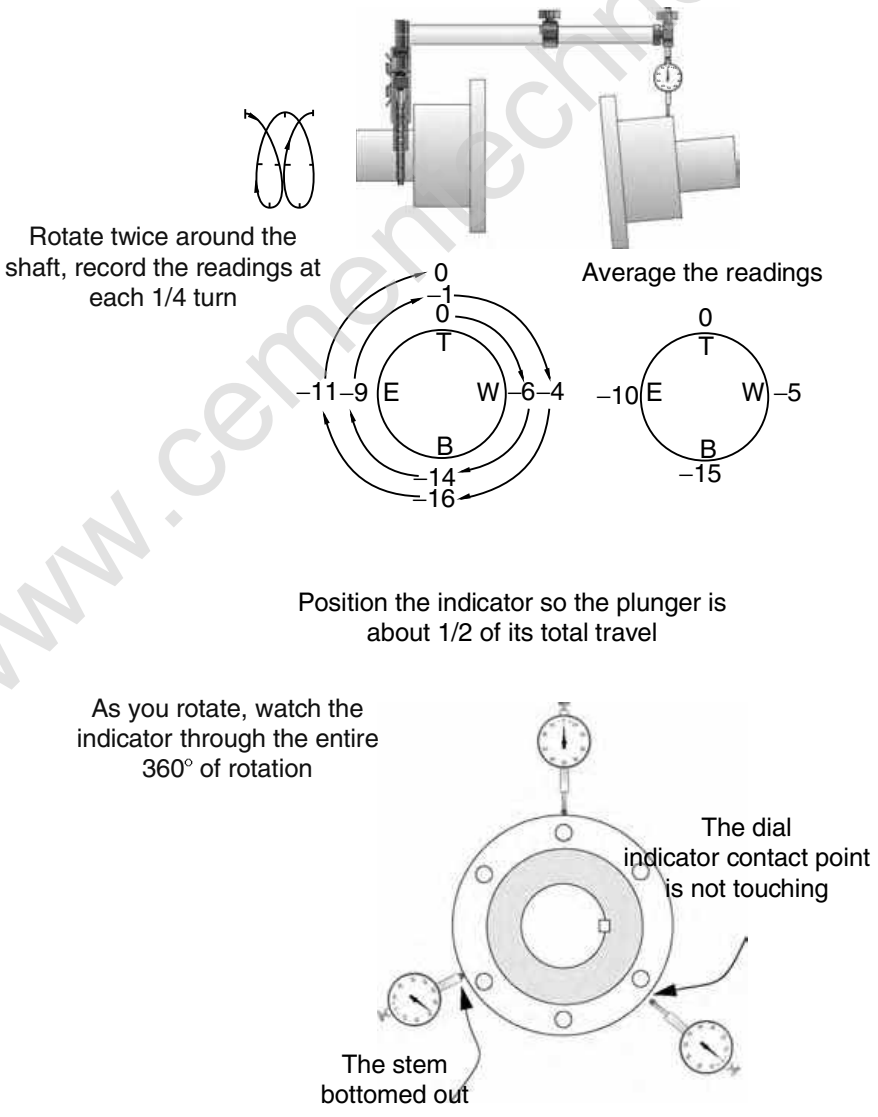


FIGURE 6.40 Suggestions for getting good measurements.

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shaft "lift check" as shown in Figure 5.2. The lower drawing in Figure 6.40 shows that you need to make sure you have not exceeded the range of the measuring device.

6.7 ENGAGED COUPLINGS WILL PRODUCE MEASUREMENT ERRORS

For a moment, refer to Figure 1.3, which shows how, under moderate to severe misalignment conditions, the shafts will start elastically bending. This elastic bending occurs on both rigid and flexible couplings. On rigid couplings, the elastic bending will begin with just small amounts of misalignment. Therefore, shaft alignment measurements should never be taken across an engaged rigid coupling.

In fact, shaft-to-shaft alignment measurements should never be taken with an engaged flexible coupling. There can be a significant amount of misalignment present but the alignment measurement system that is attempting to observe the unstressed true centerlines of rotation of the shafts with the coupling engaged will not be able to determine where the shafts would actually be if the coupling was not bolted together. As the misalignment between shafts becomes more severe, the flexing capability in the coupling diminishes and the shafts begin to elastically bend as shown in Figure 6.41.

Every alignment measurement method and alignment measurement tooling is going to be attached for observing the position of each shaft between the inboard bearings and the ends of each shaft. As you can see in Figure 6.41, the shafts are elastically bending where the brackets and measuring sensors (e.g., the dial indicators or laser–detectors) will be placed, not where the actual shaft centerline would be if the bending stresses were not present.

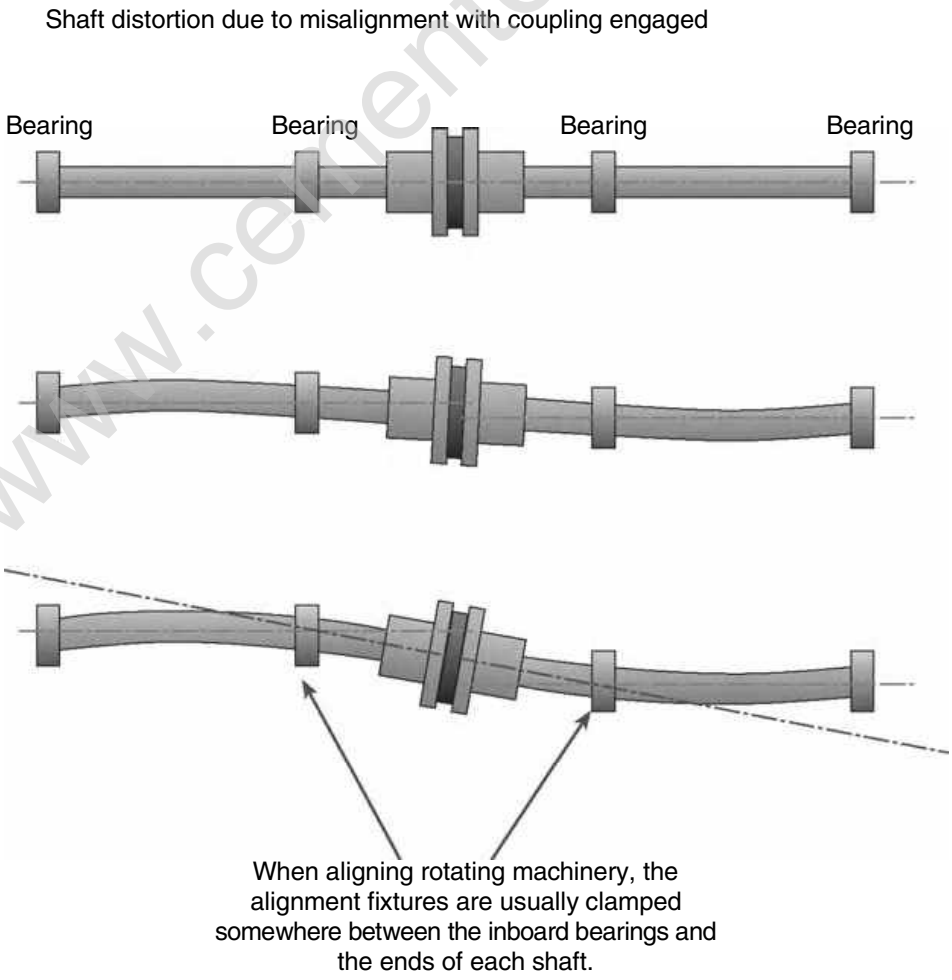


FIGURE 6.41 Elastic bending of shafts under misalignment will produce incorrect movement solutions.

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The amount of elastic shaft bending that occurs under misalignment conditions when the flexible or rigid coupling is engaged varies on the coupling design and flexibility of the coupling. The shearing capacity of rigid couplings is very low and elastic bending will occur with very little misalignment. With flexible couplings, elastic shaft bending will occur with as little as 5 mils/in. of misalignment on some flexible couplings and definitely occurs on all flexible couplings at 10+ mils/in. of misalignment. Determination of your maximum misalignment deviation, expressed in mils per inch, is explained in Chapter 9.

Under these stressed conditions with an engaged coupling, the shaft-to-shaft positional measurements you gather will produce incorrect movement solutions causing you to under- or overcorrect the misalignment. This is one of the major reasons why people put shims under the feet; then after checking the shaft positions, shims had to be taken back out, then put back in on the next iteration, then some taken back out. After three, four, or five iterations up then down, then up again, the elastic bending eases and you eventually begin observing the true, unstressed shaft positions. Not only does this occur in the up and down direction, but also the side-to-side direction.

How do we prevent this from happening? If you know that you have a moderate to severe shaft misalignment condition, either disengage the coupling completely and let each shaft rotate on its bearings in an unstressed condition or at a minimum, loosen the coupling bolts as much as possible to relieve the bending forces across the coupling.

6.8 RIM INDICATOR SETUP VARIATIONS

When capturing rim readings, the dial indicator is usually attached to the end of the span bar with its stem touching the outer surface of the other shaft or it can be positioned on any object that is rigidly attached to the other shaft such as a coupling hub or even another bracket attached to that shaft assuming that the shafts will be rotated together. This classic arrangement is shown in Figure 6.42.

Another way to accomplish the same measurement is to use the span bar as a "target" platform for the indicator which is attached to the other shaft and positioned upside down as shown in Figure 6.43. In either setup, the measurements will produce the same results. The advantage of using the span bar as a target platform for the indicator is to reduce the overhung weight of the indicator when it is attached to the end of the span bar. This will reduce the amount of bracket sag, which will be discussed shortly.

6.9 RIM READINGS INDICATE TWICE THE CENTERLINE OFFSET

When traversing from one side to the other on a shaft, the total amount that the indicator (or any measurement sensor) observes is twice the amount that the centerlines of rotation are actually out. Figure 6.44 shows what happens when observing the shafts from a side view. Figure 6.45 shows what happens when observing the shafts from an axial view.

6.10 VALIDITY RULE

Due to the geometry of taking readings around the circumference of a circular shaft, a pattern emerges which is commonly referred to as the "validity rule." The validity rule states that when the two measurements taken 90° on each side of the "zeroing" point are added together, they will equal the measurement taken 180° from the zeroing point. Examples of the validity rule are shown in Figure 6.46, as you may have noticed, in all of the example readings shown in the various dial indicator techniques covered in the previous sections. The validity rule applies to both radial (i.e., perimeter or circumferential) and face readings.

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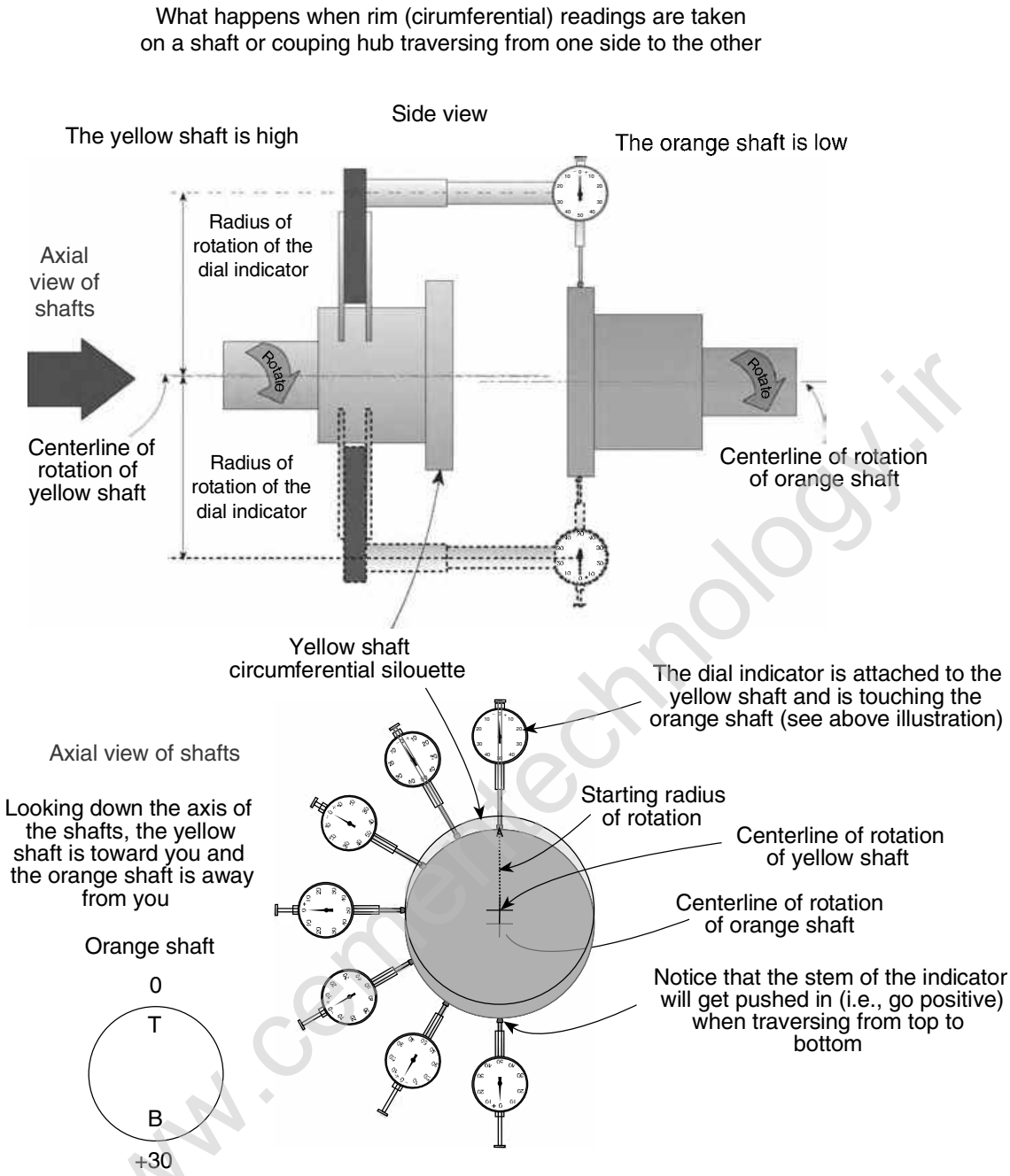


FIGURE 6.42 Taking rim readings on the outside surface of a shaft.

The validity rule is important for two reasons:

1. To insure that you are capturing an accurate set of readings when measuring the off-line positions of machinery shafts. A considerable amount of time will be wasted attempting to reposition machinery based on inaccurate measurements.
2. You do not have to rotate all the way around a shaft to determine the position of its centerline. If you can capture readings through a 180° arc (i.e., three out of four readings), you can determine what the other reading would be without actually having to take the measurement at that position. This comes in very handy when a physical restriction (e.g., lube oil pipe, the base plate, a coupling guard stand, etc.) prevents sweeping through a full circle.

The most common causes for the validity rule not working perfectly each time is shown in Figure 6.47. As a rule of thumb, the two readings that straddle the zero should add together to the reading that is opposite the zero within 10% of the largest number in the reading set.

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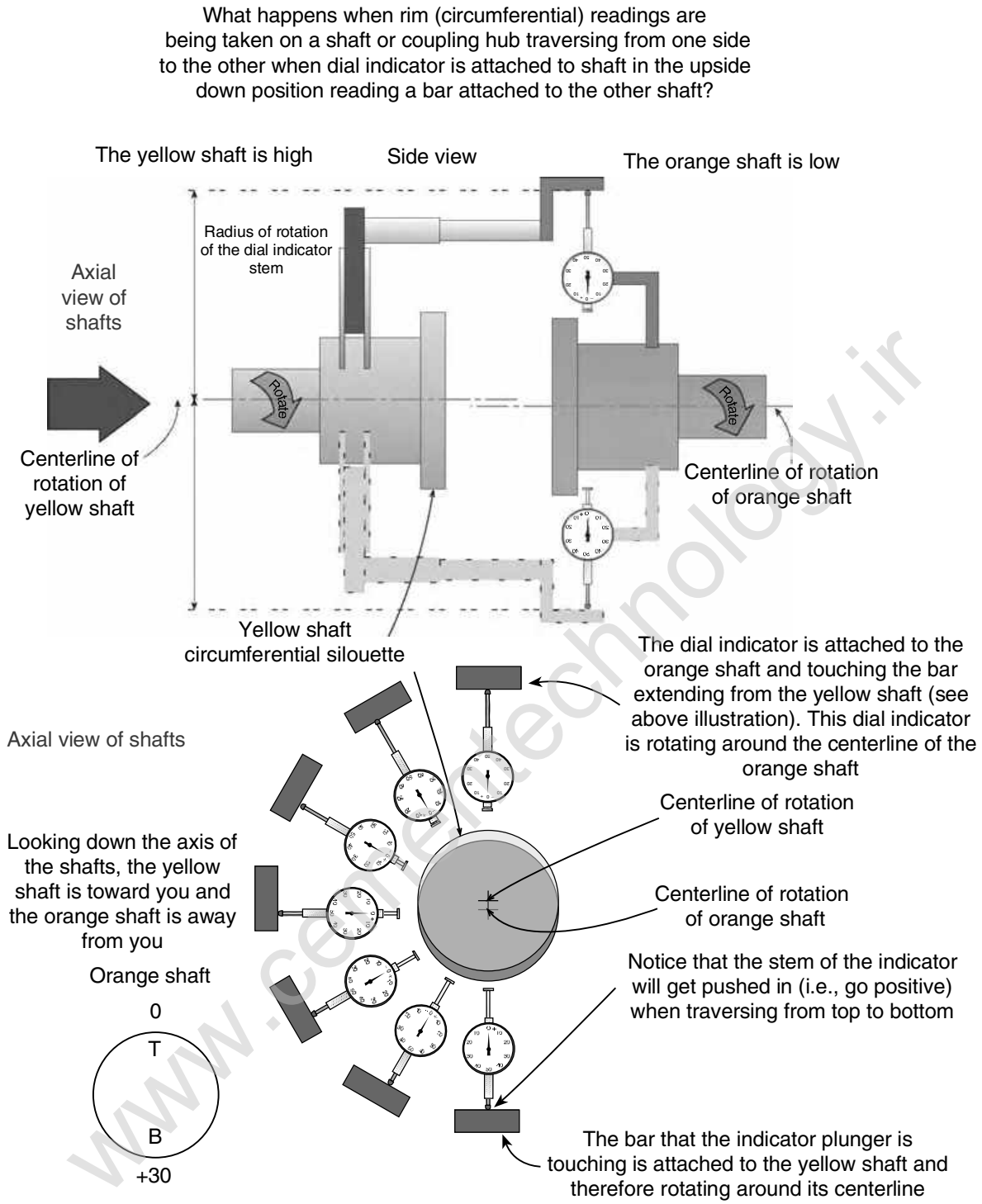


FIGURE 6.43 Taking rim readings with the indicator measuring the position of the span bar.

6.11 PARTIAL ARC MATHEMATICS

What if you cannot sweep through a 180° arc when attempting to capture shaft positional measurements? What if you can only sweep through a 90° arc or only a 30° arc?

It is possible to determine the position of a shaft centerline with readings captured on less than a 180° sweep. These types of readings are called "partial arc" readings. If you know the angular position of the indicator (or any other type of precision distance measurement sensor) and the distance the indicator stem has traveled (amplitude) as it swept through a known angle, you can determine what the distance (amplitude) is at any other angular position.

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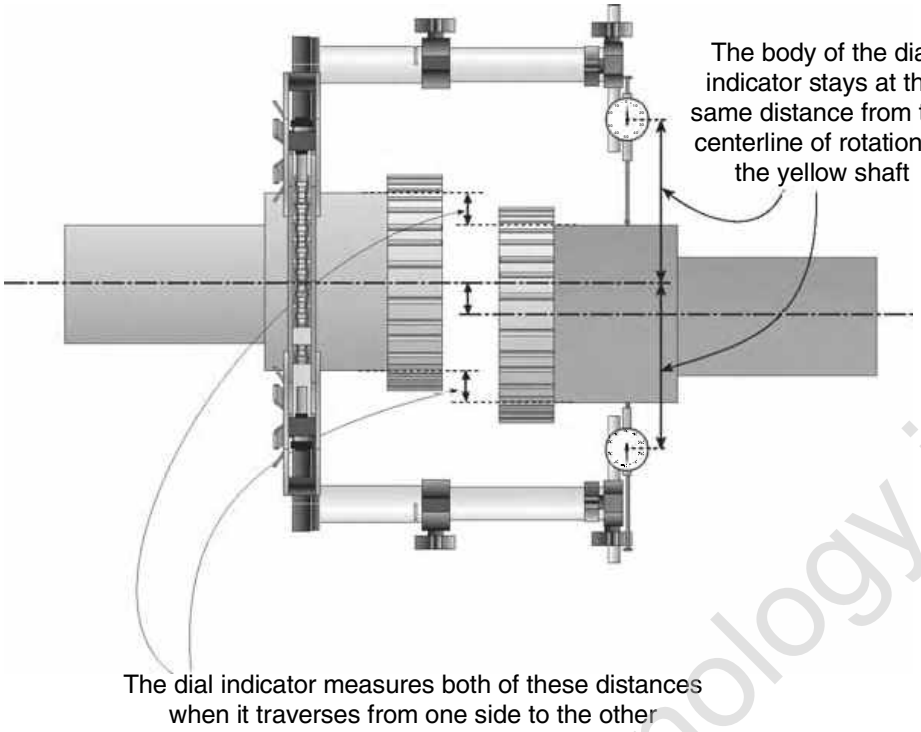


FIGURE 6.44 Side view showing indicator measuring twice the actual offset of the shaft centerlines.

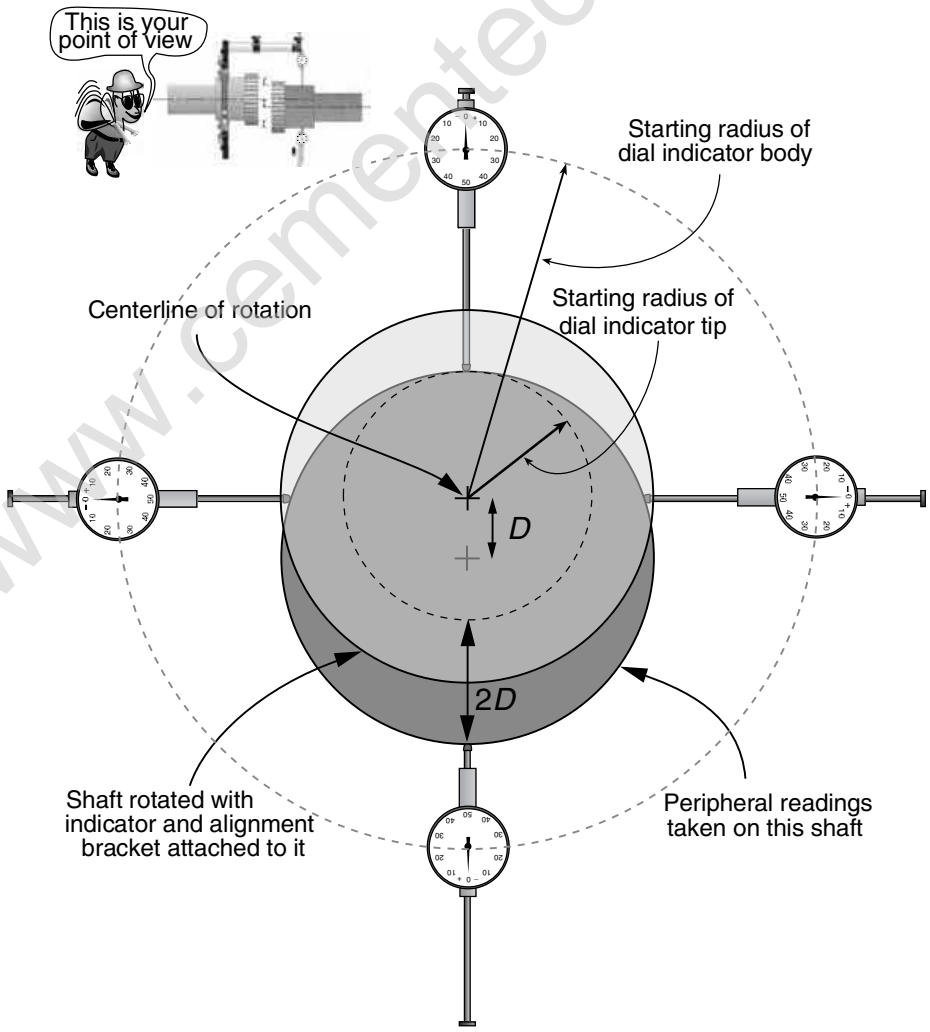


FIGURE 6.45 (**See color insert following page 322.**) End view showing indicator measuring twice the actual offset of the shaft centerlines.

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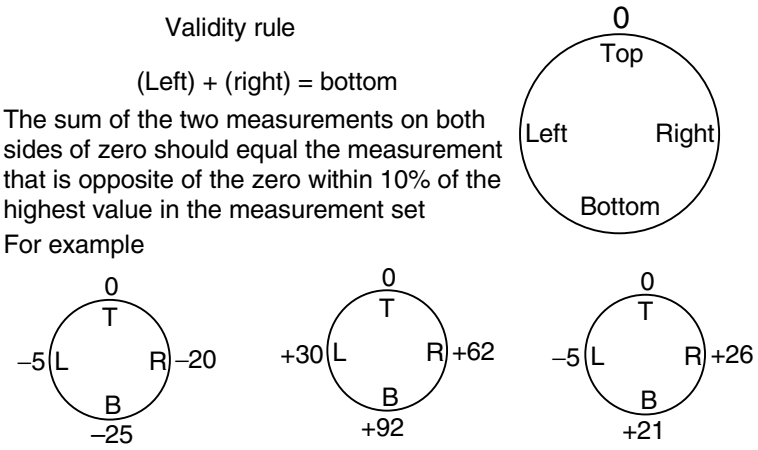


FIGURE 6.46 The validity rule.

For partial arc readings, you need to know the following things:

1. An angular reference point, usually where top dead center is
2. The amount of angular sweep made
3. The amount of distance measured by your sensor at the furthest counterclockwise position, the distance measured midway through the sweep, and the distance measured at the furthest clockwise position

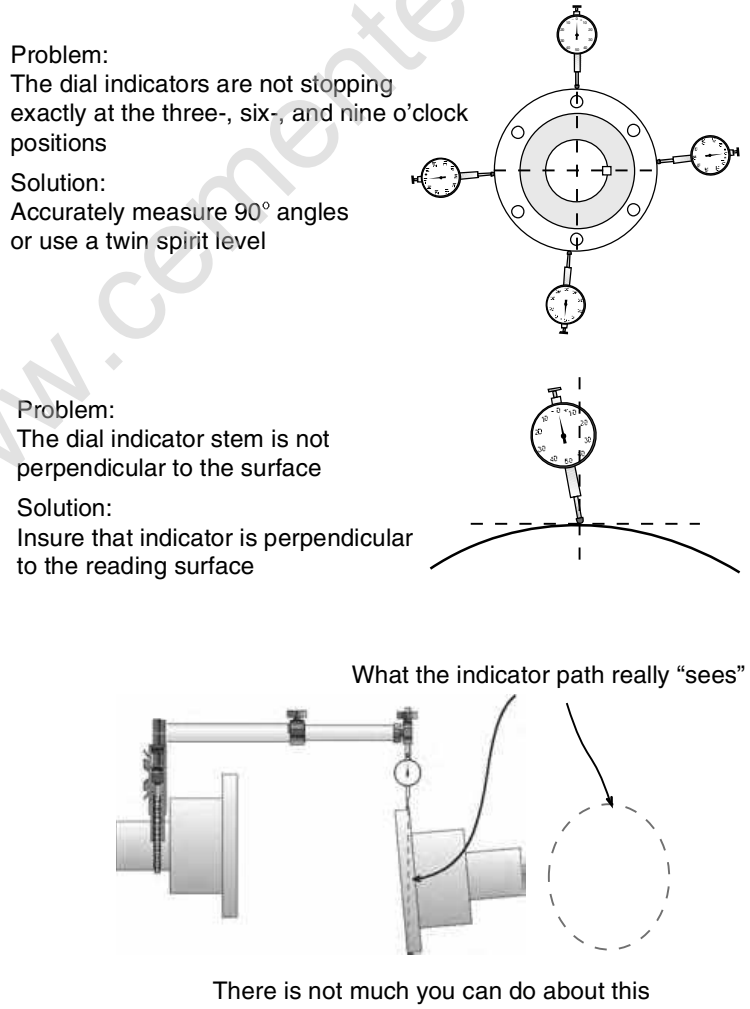
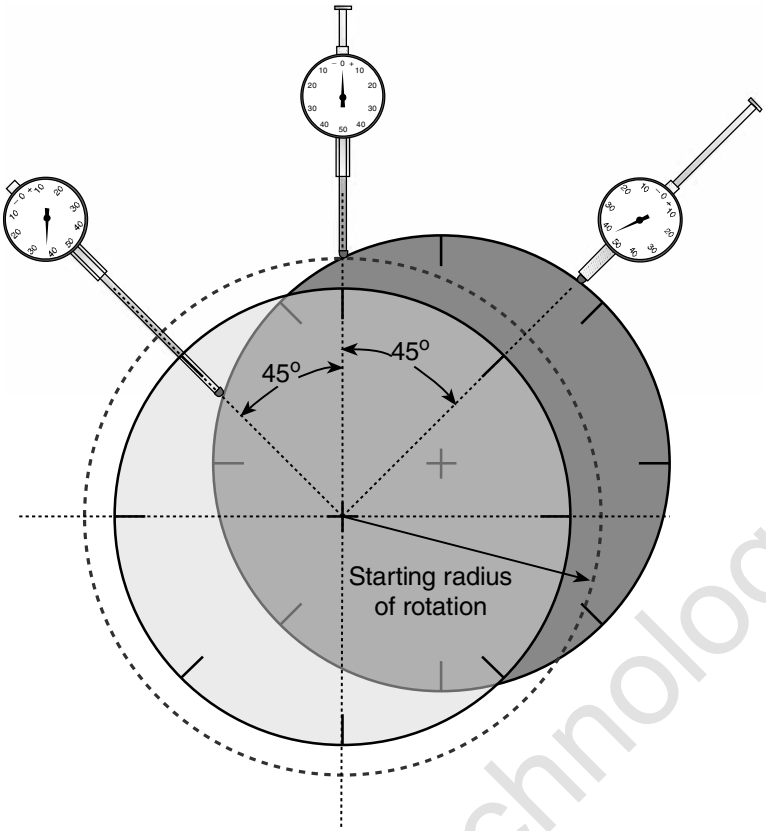


FIGURE 6.47 Why the validity rule does not work perfectly every time.

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Left reading at 90° = 2.414 (left reading at 45°) + right reading at 45°
Right reading at 90° = 2.414 (right reading at 45°) + left reading at 45°
Bottom = 3.416 (left reading at 45° + right reading at 45°)

Example:
Top = 0
Left reading at 45° = –38
Right reading at 45° = +54

Left reading at 90° = 2.414 (–38) + (+54) = (–91.7) + (+54) = –37.7
Right reading at 90° = 2.414 (+54) + (–38) = (130.4) + (–38) = +92.4
Bottom = 3.416 ((–38) + (+54)) = +54.6

FIGURE 6.48 How to determine the side and bottom readings if you can only sweep through a 90° arc.

Figure 6.48 shows how to determine the left, right, and bottom readings if you zero the indicator at twelve o'clock (top) but can only sweep 45° counterclockwise and 45° clockwise.

To get a better picture of what is happening when measurements are swept around a centerline of one shaft with respect to another shaft, let us examine three different scenarios. Figures 6.49 through 6.51 show an axial view looking down the centerline of rotation of two shafts. In Figure 6.49, the two shafts are misaligned in the horizontal direction only. As you can see, by plotting the angular position of the measurement device on the *x*-axis and the dial indicator readings (amplitude) on the *y*-axis, the curve that is generated is a sine wave. The mathematics of taking partial arc readings is based on the premise that the measurements taken around the circumference or face of a shaft define the shape of a perfect sine wave. With pure horizontal misalignment, when the dial indicator is positioned at the top and zeroed, if the shaft is rotated in a clockwise direction, the stem of the indicator gets pushed in (i.e., goes positive). At exactly 90° of rotation, on the right side, the indicator has been pushed in the maximum amount and as rotation continues to the bottom, the indicator begins retracting back outward. When the indicator reaches the bottom position, it is back to zero again. As rotation continues, the indicator stem begins retracting outward (i.e., goes negative).

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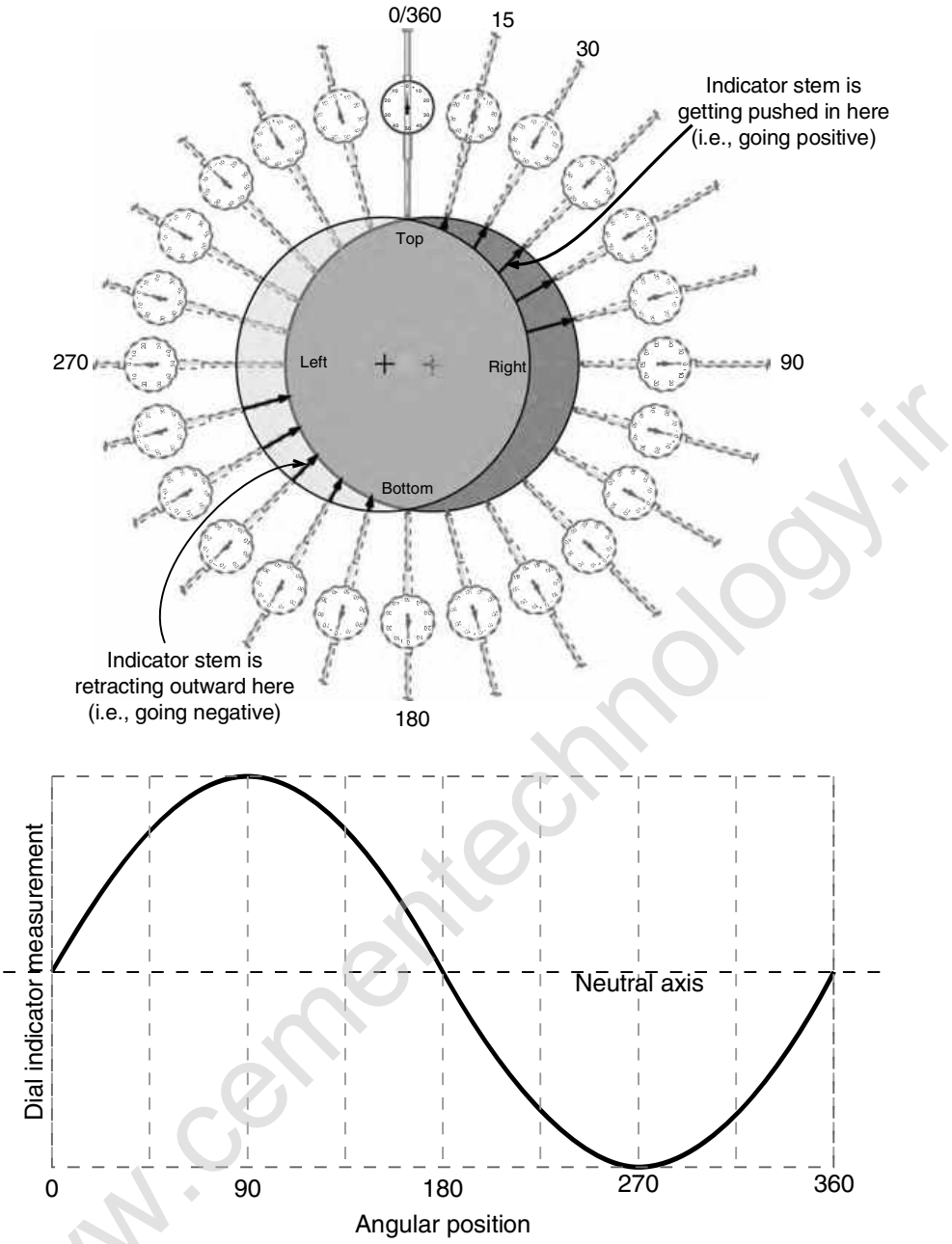


FIGURE 6.49 (**See color insert following page 322.**) Plotting the measurements for a pure horizontal misalignment will produce a sine wave.

At exactly 270° of rotation, the indicator has retracted out to its maximum position. In the final sweep from 270° to 360° position, the indicator gets pushed back from its maximum negative position and returns back to its starting position at zero. The neutral axis of the sine wave is split evenly between the maximum and minimum dial indicator measurements. This is similar to the waveform generated in vibration as shown in Figure 2.10.

In Figure 6.50, the two shafts are misaligned in the vertical direction only. As the indicator traverses around the circumference of the shaft under measure, the stem is continually pushed into its maximum value at the bottom generating a cosine wave. As many of you remember from your math classes, the cosine wave and sine wave are of the same shape except that the cosine wave lags the sine wave by 90°. The neutral axis of the sine–cosine wave is offset from the starting position by half of the total measurement from the top to the bottom of the shaft.

In Figure 6.51, the two shafts are misaligned in both the vertical and horizontal directions. In this particular case, as the indicator rotates in a clockwise direction from the top to the

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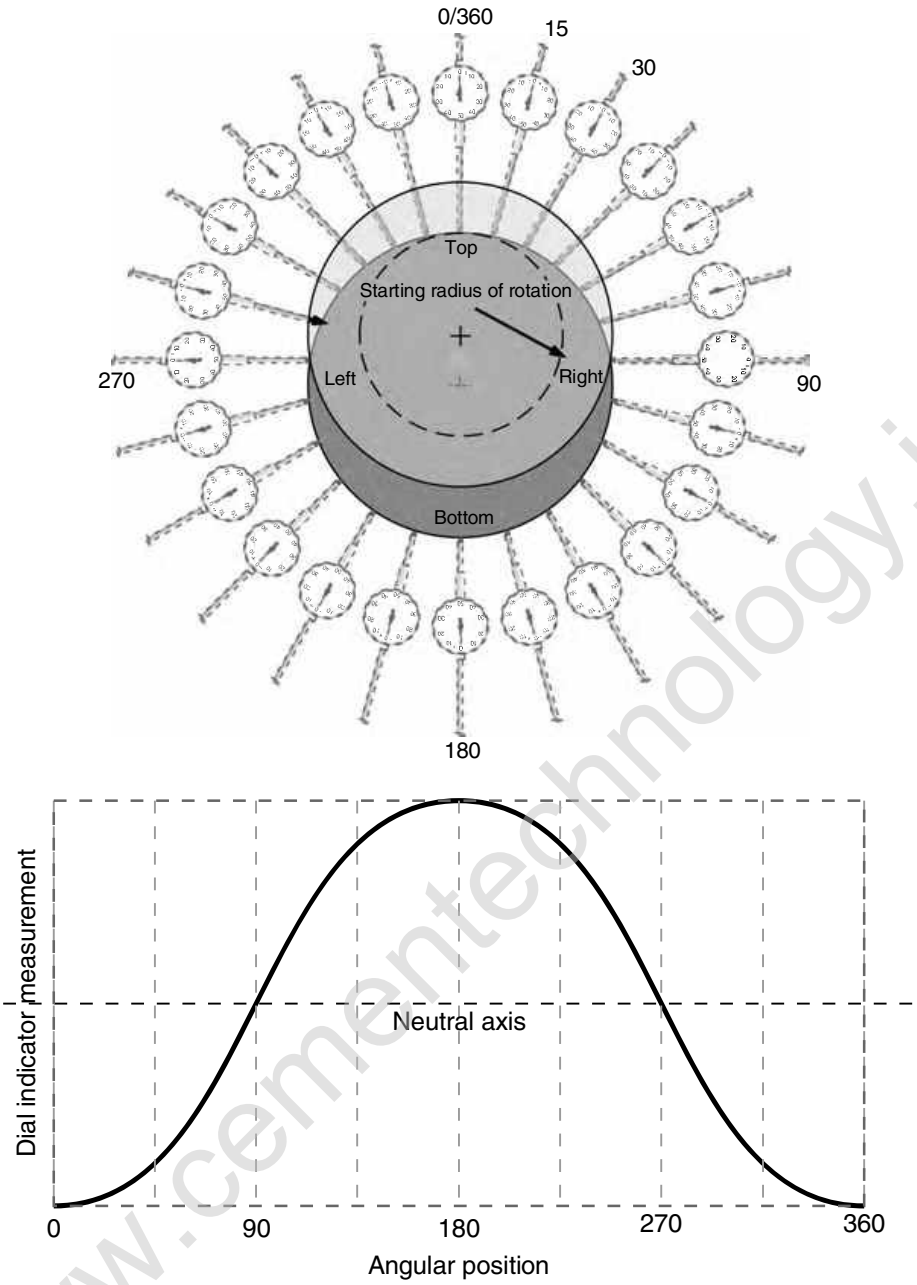


FIGURE 6.50 Plotting the measurements for a pure vertical misalignment will produce a cosine wave.

right side, the stem of the indicator retracts outward (i.e., goes negative) for the first 45° of rotation but then begins to get pushed in and returns back to zero after 90° of rotation. From that point onward, the indicator stem gets pushed in (i.e., goes positive) until it gets to 225° of rotation where it then begins retracting outward and returns back to zero again at the top. Again, the curve that is generated is a sine wave but the neutral axis is offset by a certain amount and there is an angular lag in the waveform.

6.11.1 Partial Arc Measurement Procedure

In Figure 6.48, a dial indicator was zeroed at the top (i.e., twelve o'clock position), rotated 45° counterclockwise from the top where a reading was taken, and then rotated 45° clockwise from the top where another reading was taken. Plug those measurements into the equations

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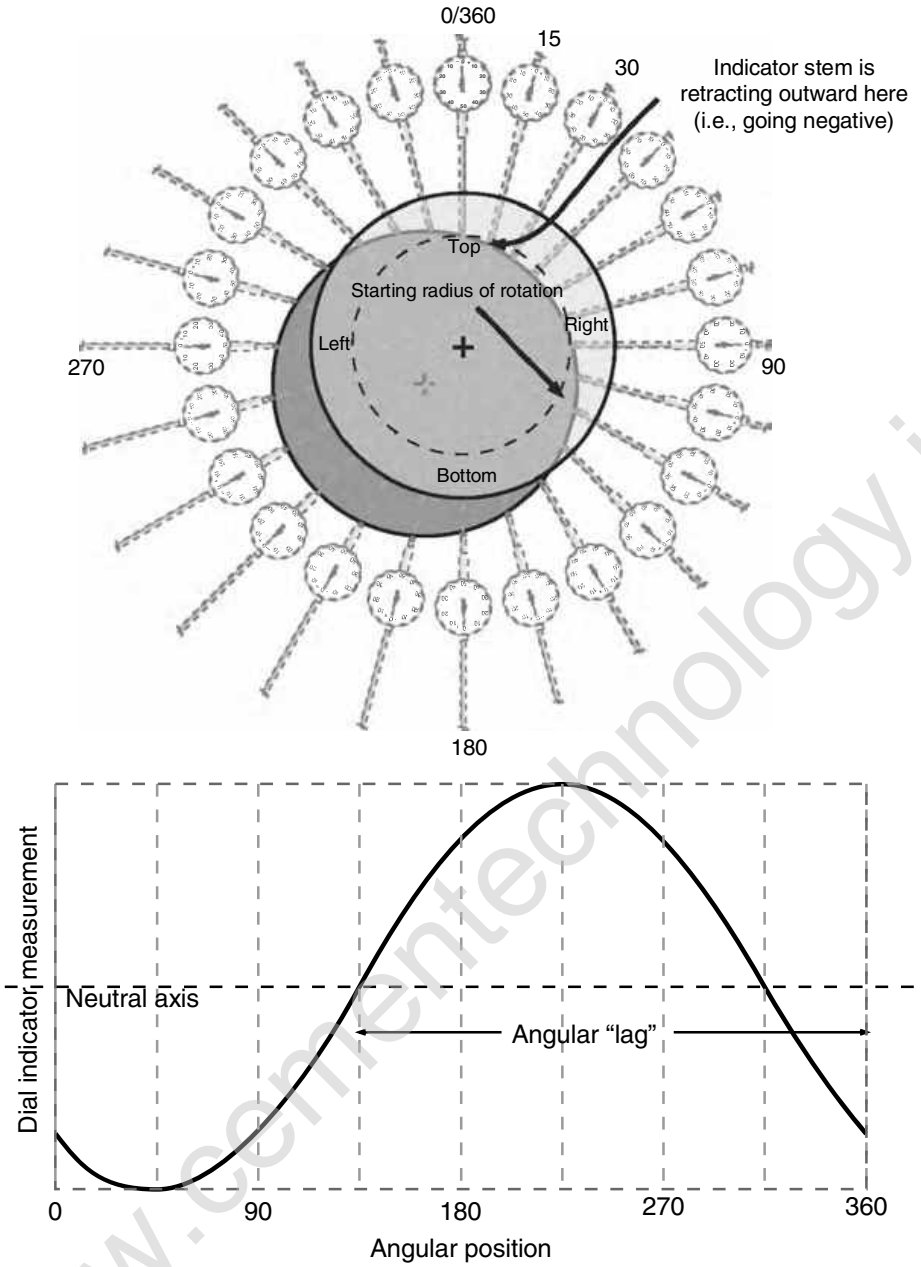


FIGURE 6.51 Plotting the measurements for a misalignment condition where the shafts are misaligned both vertically and horizontally will produce a sine wave with an offset in the neutral axis and an angular lag in the waveform.

and you can then determine the side and bottom readings. Simple enough, but what if you had to zero the indicator on the right side of the shaft and only rotate 30° clockwise?

We know that the curve generated from the measurements around a shaft is a sine wave, but what angular position on the measurement sine wave did the dial indicator start at and what is the amplitude of the sine wave? The slope of a sine curve at any point is equal to the amplitude of the cosine at that point. If you knew what angular position on the sine wave you started at and you knew what the amplitude of the sine wave was, you could then generate the rest of the sine wave and determine what the measurements would be at all the angular positions. But you do not know either one of those things. What you do know is the change in measurements through the arc you swept and the amount of arc that was swept in degrees of rotation. The key to solve this problem is to determine the ratio of the rate of change of slope through the angular sweep of the measurements.

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1. Establish at what angular position on the outer circumference of the shaft you are able to take readings. Are you able to start at the top of the shaft or at some other angular position? With respect to the twelve o'clock position, what is the angular starting point going to be? On the left side? On the bottom? 45° clockwise to the right? Using the twelve o'clock reading as 0°/360°, the right side as 90°, the bottom as 180°, and the left side as 270°, measure and record the starting angular position (STARTANGLE).
2. Determine the angular sweep (ANGULARSWEEP) you are going to (or can) make and divide it into two equal arcs. Capture three measurements across the angular sweep. One in the furthest counterclockwise position (CCWRDG), one in the center of the angular sweep (CNTRDG), and another in the furthest clockwise position (CWRDG).
3. Subtract the center reading from the counterclockwise reading and the clockwise reading from the center reading.
   a. CNTRDG2CCWRDG = CNTRDG−CCWRDG
   b. CWRDG2CNTRDG = CWRDG−CNTRDG
4. Divide each of these reading differences by the angle you swept. Insure the angle is in radians (1° = 0.01745 rad). This will determine the rate of change of slope (ROCOS) through that angle.
   a. ROCOSCNTR2CCW = CNTRDG2CCWRDG/(ANGULARSWEEP/2)
   b. ROCOSCW2CNTR = CWRDG2CNTRDG/(ANGULARSWEEP/2)
5. Divide the rate of change of slope that occurred from the center to the counterclockwise reading by the rate of change of slope that occurred from the clockwise to the center reading. This will determine the ratio of the rate of change of slope (ROROCOS).
   a. MEASUREDROROCOS = ROCOSCNTR2CCW/ROCOSCW2CNTR
6. Generate a table of the sine values from 0° through 360° incrementing each by half the total angular sweep you made. Subtract the each set of sine values from each other and divide each sine value difference by the subsequent one. This will determine the ratio of the rate of change of slope of a sine wave (with an amplitude value of 1) for half the total angular sweep you made. Plot these values onto a graph.
7. To determine the sine wave angular position of the center reading, take the value computed in step 5 and find the corresponding angular sweep from the graph plotted in step 6 that defines the ratio of the rate of change of slope. Determine what quadrant you are in on the sine wave. The starting point of that angular sweep defines the sine wave angular position of the center reading (CNTRDGANGLE).
8. To determine the amplitude of the measurement sine wave, divide the difference between counterclockwise reading and the center reading by the difference between the same sine value difference for the same corresponding angular change of a pure sine wave with an amplitude of 1.
   a. (CWRDG−CNTRDG)/(sin A2−sin A1)
9. Generate the complete measurement sine wave and determine the values at the top, right, bottom, and left sides on the shaft under measure.

6.11.2 Partial Arc Measurement Sample Problem

Let us say we were able to take a 30° sweep (arc of measurement) starting at 15° clockwise from the top and swept our readings in the clockwise direction toward the right side. We zeroed the indicator at 15° position (the counterclockwise reading), 15° later (the center

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reading) the dial indicator measured a +31, then 15° later (the clockwise reading) the dial indicator measured a +55.

1. Establish at what angular position on the outer circumference of the shaft you are able to take readings.
   STARTANGLE = 15
2. Determine the angular sweep (ANGULARSWEEP) you are going to (or can) make and divide it into two equal arcs. Capture three measurements across the angular sweep. One in the furthest counterclockwise position (CCWRDG), one in the center of the angular sweep (CNTRDG), and another in the furthest clockwise position (CWRDG).
   CCWRDG = 0
   CNTRDG = +31
   CWRDG = +55
3. Subtract the center reading from the counterclockwise reading and the clockwise reading from the center reading.
   CNTRDG2CCWRDG = CNTRDG−CCWRDG
   CWRDG2CNTRDG = CWRDG−CNTRDG
   CNTRDG2CCWRDG = (+31) − (0) = +31
   CWRDG2CNTRDG = (+55) − (+31) = +24
4. Divide each of these reading differences by the angle you swept. Insure the angle is in radians (1° = 0.01745 rad). This will determine the rate of change of slope (ROCOS) through that angle (15° = 0.2618 rad).
   ROCOSCNTR2CCW = CNTRDG2CCWRDG/(ANGULARSWEEP/2)
   ROCOSCW2CNTR = CWRDG2CNTRDG/(ANGULARSWEEP/2)
   ROCOSCNTR2CCW = (+31)/(0.2618) = 118.411
   ROCOSCW2CNTR = (+24)/(0.2618) = 91.673
5. Divide the rate of change of slope that occurred from the center to the counterclockwise reading by the rate of change of slope that occurred from the clockwise to the center reading. This will determine the ratio of the rate of change of slope.
   MEASUREDROROCOS = ROCOSCNTR2CCW/ROCOSCW2CNTR
   MEASUREDROROCOS = (118.411)/(91.673) = 1.29166
6. Generate a table of the sine values from 0° through 360° incrementing each by half the total angular sweep you made. In this case, half the angular sweep is 15°. Subtract each set of sine values from each other and divide each sine value difference by the subsequent one. This will determine the ratio of the rate of change of slope of a sine wave (with an amplitude value of 1) for half the total angular sweep you made. Plot these values onto a graph (see Figure 6.52 and Figure 6.53).
7. To determine the sine wave angular position of the center reading, take the value computed in step 5 and find the corresponding angular sweep from the graph plotted in step 6 that defines the ratio of the rate of change of slope. Determine what quadrant you are in on the sine wave. The starting point of that angular sweep defines the sine wave angular position of the center reading.

   The value computed in step 5 is 1.29166. Notice that there are two angular positions that share this value, 45° and 225°. Since the value from the counterclockwise reading to the center reading is increasing (e.g., 0 to +31) and the value from the center reading to the clockwise reading is again increasing (e.g., +31 to +55), we can surmise that we are in quadrant 1 on the sine wave.
   CNTRDGANGLE = 45

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Ratio of the rate of change of slope (ROROCOS)

A1	A2	sin A1	sin A2	sinA2–sinA1	ROROCOS
0	15	0	0.2588190451	0.2588190451	
15	30	0.2588190451	0.5	0.2411809549	1.073132185
30	45	0.5	0.70710678119	0.20710678119	1.1645246646
45	60	0.70710678119	0.86602540378	0.1589186226	1.3032253728
60	75	0.86602540378	0.96592582629	0.0999004225	1.5907702752
75	90	0.96592582629	1	0.03407417371	2.9318516526
90	105	1	0.96592582629	–0.03407417371	–1
105	120	0.96592582629	0.86602540378	–0.0999004225	0.3410813774
120	135	0.86602540378	0.70710678119	–0.1589186226	0.62862627974
135	150	0.70710678119	0.5	0.20710678119	0.76732698798
150	165	0.5	0.2588190451	–0.2411809549	0.85871946761
165	180	0.2588190451	1.2246467991e-16	–0.2588190451	0.93185165258
180	195	1.2246467991e-16	–0.2588190451	–0.2588190451	1
195	210	–0.2588190451	–0.5	–0.2411809549	1.073132185
210	225	–0.5	–0.70710678119	–0.20710678119	1.1645246646
225	240	–0.70710678119	–0.86602540378	–0.1589186226	1.3032253728
240	255	–0.86602540378	–0.96592582629	–0.0999004225	1.5907702752
255	270	–0.96592582629	–1	–0.03407417371	2.9318516526
270	285	–1	–0.96592582629	0.03407417371	–1
285	300	–0.96592582629	–0.86602540378	0.0999004225	0.3410813774
300	315	–0.86602540378	–0.70710678119	0.1589186226	0.62862627974
315	330	–0.70710678119	–0.5	0.20710678119	0.76732698798
330	345	–0.5	–0.2588190451	0.2411809549	0.85871946761
345	360	–0.2588190451	–2.4492935983e-16	0.2588190451	0.93185165258

FIGURE 6.52 Sine value table for 15° increments.

8. To determine the amplitude of the measurement sine wave, divide the difference between the clockwise reading and the center reading by the difference between the same sine value difference for the same corresponding angular change.
   (CWRDG −CNTRDG)/(sin A2−sin A1)
   ((+55) − (+31))/(0.1589186226) = 151.02
9. Generate the complete measurement sine wave and determine the values at the top, right, bottom, and left sides on the shaft under measure.

Remember, we did not start taking our readings on the top of the shaft, we started at 15° clockwise from that point. We now know that the center reading occurred at 45° on the measurement sine wave and we also know that the amplitude was 151 mils (AMPLITUDE). We also know that the angle on the measurement sine wave where we started the sweep is 30° (CCWRDGANGLE) and we also know that the angle on the measurement sine wave where we ended the sweep is 60° (CWRDGANGLE). Figure 6.54 shows the measurement sine wave.

Since we know that the angle on the measurement sine wave where we started the sweep is 30° (CCWRDGANGLE), we can now compute at what angular position on the measurement sine wave where the top, right, bottom, and left side readings will occur.

$$\text{TOPRDGANGLE} = \text{CCWRDGANGLE} - \text{STARTANGLE} = 30 - 15 = 15$$
$$\text{RIGHTRDGANGLE} = \text{TOPRDGANGLE} + 90 = 105$$
$$\text{BOTTOMRDGANGLE} = \text{TOPRDGANGLE} + 180 = 195$$
$$\text{LEFTRDGANGLE} = \text{TOPRDGANGLE} + 270 = 285$$

Since we know that the angle on the measurement sine wave where we started the sweep is not 0°, there will be an offset in the neutral axis of the measurement sine wave that we must compute.

From trigonometry, the equation for a sine wave is

$$y = a \sin x$$

where y is the instantaneous value at angle x, a is the amplitude, and x is the angle.

Since our measurements started at 30°, the instantaneous value at that angle is

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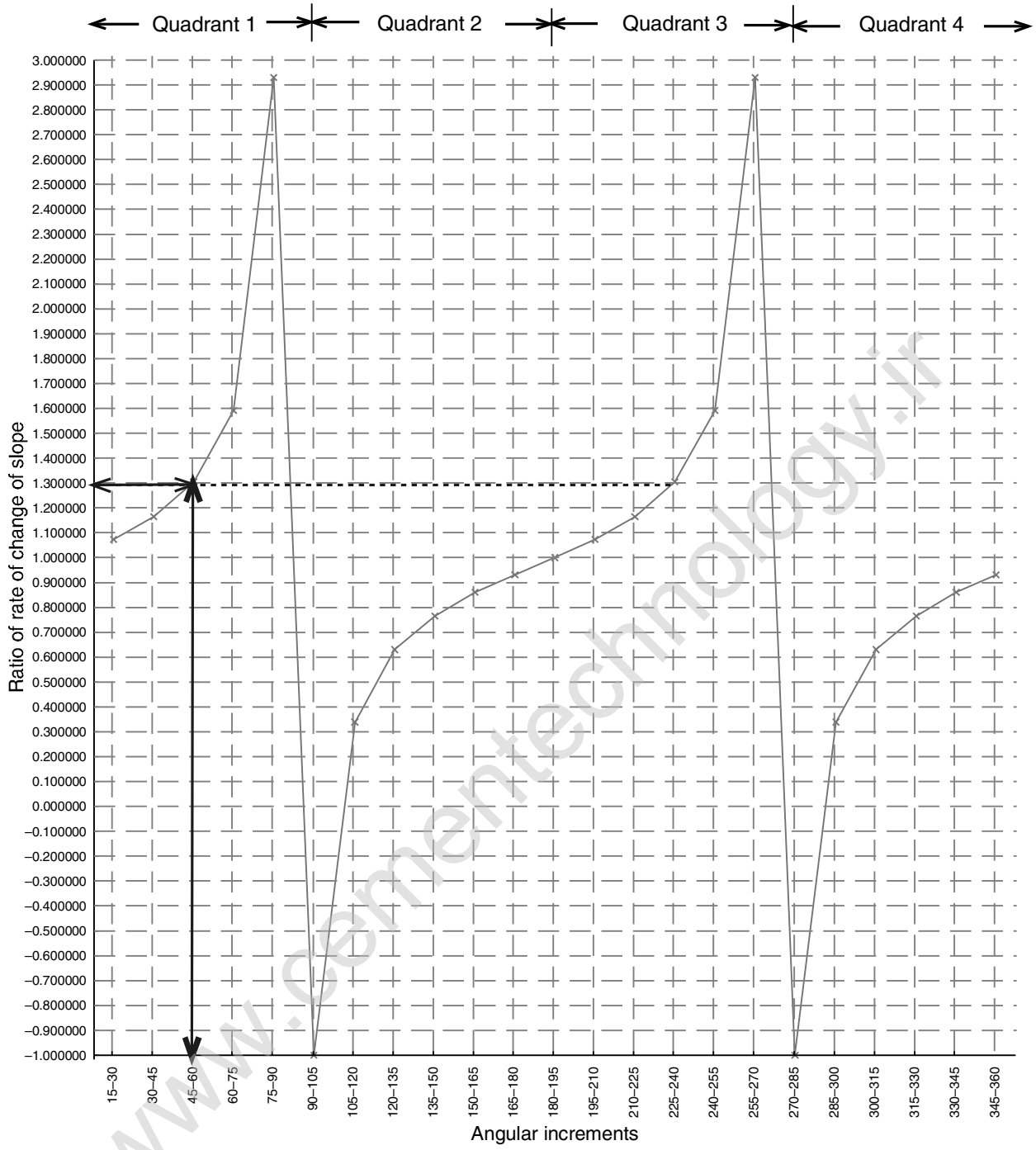


FIGURE 6.53 Plot of the ratio of the rate of change of slope for 15° arcs for a sine wave with an amplitude of 1.

$(151.02) \times (\sin 30) = (151.02) \times (0.5) = 75.51$

This value defines the amount of offset from the neutral axis of the measurement sine wave at our starting angular position (i.e., 30°).

NEUTRALAXISOFFSET = 75.51

We can now compute the top, right, bottom, and left side readings that will occur on the measurement sine wave:

TOPRDG = ((AMPLITUDE) × (sin TOPRDGANGLE)) − NEUTRALAXISOFFSET

TOPRDG = ((151.02) × (sin 15)) − 75.51 = −36.42

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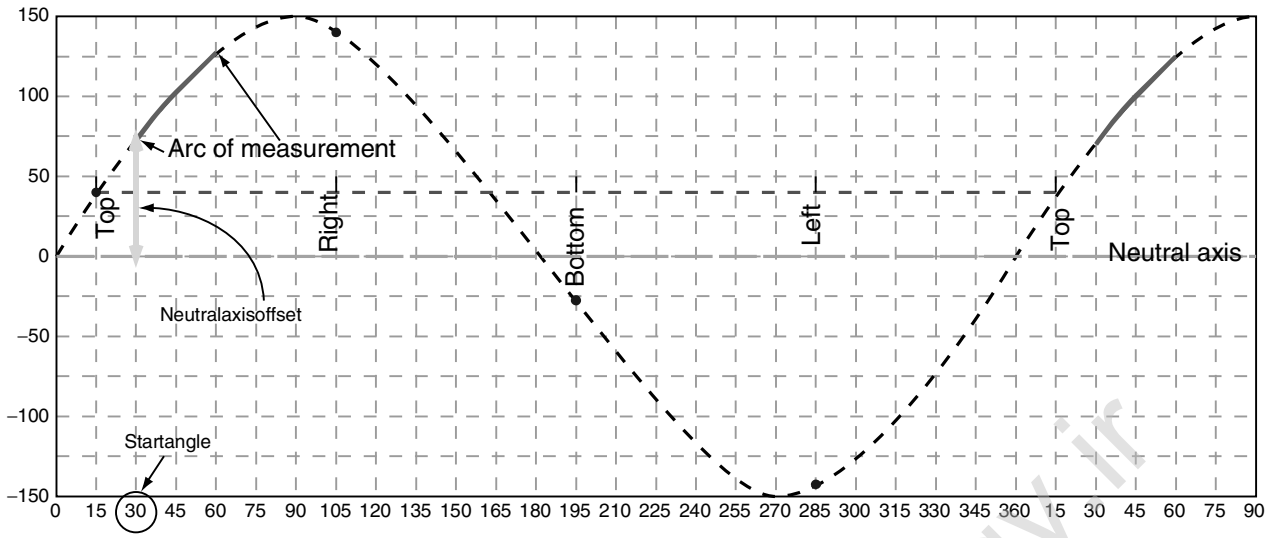


FIGURE 6.54 Measurement sine wave generated for the readings taken through a 30° arc.

RIGHTRDG = ((AMPLITUDE) × (sin RIGHTRDGANGLE)) − NEUTRALAXIS-OFFSET
RIGHTRDG = ((151.02) × (sin 105)) − 75.51 = +70.36
BOTTOMRDG = ((AMPLITUDE) × (sin BOTTOMRDGANGLE)) − NEUTRALAX-ISOFFSET
BOTTOMRDG = ((151.02) × (sin 195)) − 75.51 = −114.59
LEFTRDG = ((AMPLITUDE) × (sin LEFTRDGANGLE)) − NEUTRALAXISOFFSET
LEFTRDG = ((151.02) × (sin 285)) − 75.51 = −221.38

To determine what the measurements would have been if we could have placed the indicator in the top position and zeroed it there, add 36.42 to all the above computed measurements to "normalize" the readings:

TOP (normalized) = (TOPRDG) − (TOPRDG) = (−36.42) − (−36.42) = 0
RIGHT (normalized) = (RIGHTRDG) − (TOPRDG) = (+70.36) − (−36.42) = +106.78
BOTTOM (normalized) = (BOTTOMRDG) − (TOPRDG) = (−114.59) − (−36.42) = −78.17
LEFT (normalized) = (LEFTRDG) − (TOPRDG) = (−36.42) − (−36.42) = −184.96
If we round off the readings:
TOP (normalized) = 0
RIGHT (normalized) = +107
BOTTOM (normalized) = −78
LEFT (normalized) = −185
Does the validity rule work?
(LEFT) + (RIGHT) = BOTTOM
(−185) + (+107) = −78

6.11.3 Pitfalls of Partial Arc Measurements

As shown in Figure 6.49 through Figure 6.51, the plot of distance versus angular position of the dial indicator is a pure sine wave. Or so you think it is. For small amounts of

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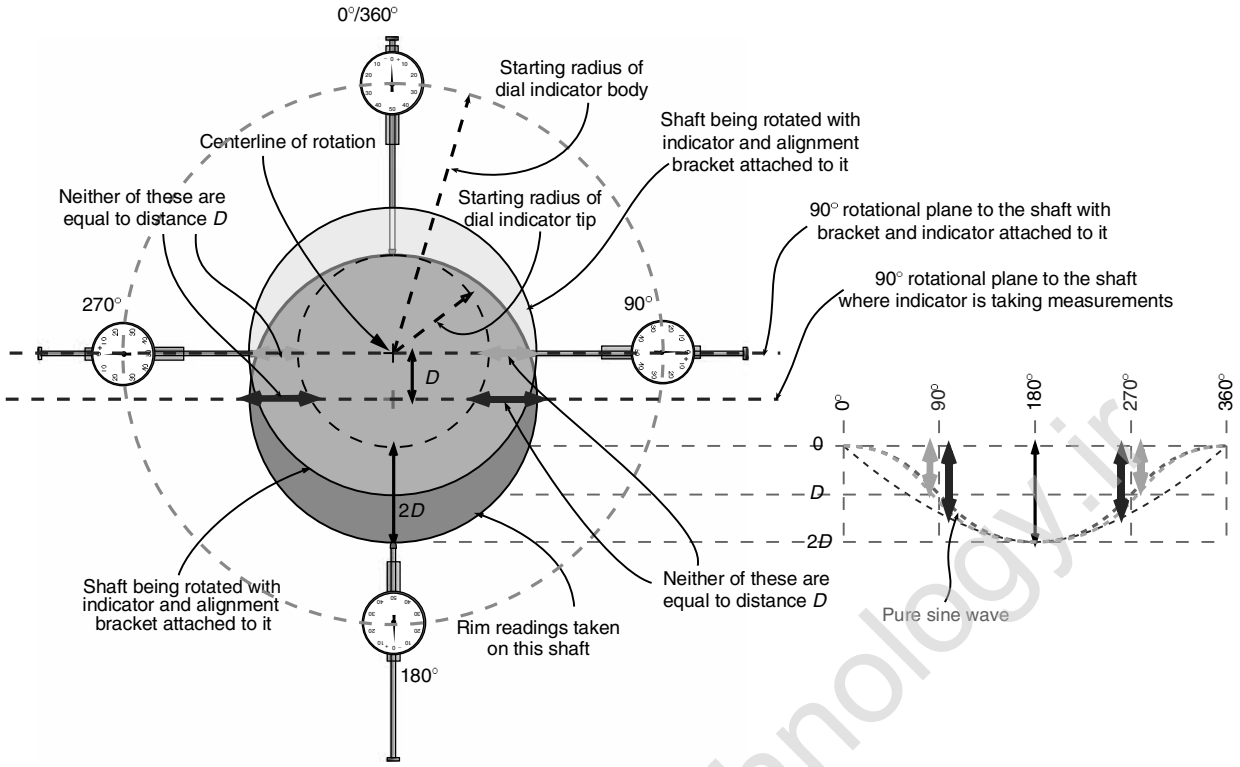


FIGURE 6.55 (**See color insert following page 322.**) Error increases with increasing misalignment.

misalignment, for all practical purposes, it is a pure sine wave. Whenever the amount of misalignment increases, the measurement error increases because the angle swept by the measurement sensor does not match the actual arc of measurement the indicator has observed on the other shaft as shown in Figure 6.55.

Notice in Figure 6.53 when plotting the ratio of the rate of change of slope of a sine wave how the curve radically changes at 90° and 270° positions. If, by sheer coincidence, you happened to be starting at, sweeping through, or stopping at those angular positions when you were capturing your measurements, the calculations will begin to go a little haywire particularly if you are only able to sweep a 5° arc of measurement. Additionally, the accuracy of the dial indicator (or any measurement sensor), the angular measurement made, and the starting angular position is directly related to the accuracy and precision of the calculated results. If you are eyeballing the angular positions and reading the dial indicator imprecisely, you may get large errors in your calculations.

If you absolutely have to do partial arc measurements it is recommended that you try to sweep at least a 45° arc of measurement, carefully measure your angles, and accurately observe and record the measurements taken with the dial indicator.

6.12 BRACKET OR BAR SAG

Whenever mechanical brackets and dial indicators are used to measure shaft positions, bracket sag must be measured and compensated for. The span bar is an overhung cantilever beam that bends under its own weight and the weight of devices attached to the end of the bar from the gravitational pull of the Earth.

Bracket sag is somewhat confusing (and surprising) to people who observe this for the first time. Not only does this phenomenon affect radial or circumferential measurements, but it also affects face measurements. Attempting to align machinery with measurements that have not been compensated for bracket sag will result in incorrect shim changes to fix the

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misalignment condition. This is one of the classic mistakes made by people who align rotating machinery and the importance of measuring and compensating for this cannot be stressed enough.

There are four factors that affect the amount of sag that you will get with any mechanical bracket arrangement:

1. The amount of overhung weight (the weight of the dial indicator at the end of the span bar and the incremental weight of the span bar itself)
2. The span of the bar (i.e., the distance from where you clamp to where the indicator measures)
3. The stiffness of the span bar
4. Clamping force of the bracket to the shaft

Typically, when you go out to align a piece of rotating machinery, there are several things that you do not know until you set your measurement measuring system onto the shafts. You do not know neither what the diameter of the shaft is that you will be clamping onto, nor do you know what the height of the span bar needs to be from the point of contact on the shaft, and nor do you know the distance you will be spanning from shaft to shaft.

Perhaps the best way to approach this, due to all of these variables, is to perform the following procedure when taking alignment readings:

1. Set up the bracket, span bar, and indicator on the machinery aligned.
2. Take a set of shaft-to-shaft readings and record the data. These are referred to as the "field" readings.
3. Remove the bracket, span bar, and indicator setup, carefully without disturbing the span bar length and the arrangement of the bracket and span bar configuration. Use the same dial indicator you used to capture the readings.
4. Find a rigid piece of piping, conduit, or bar stock of sufficient length to clamp the bracket, span bar, and indicator setup onto. Try to select a rigid piece of pipe close to the diameter of the shaft the bracket was clamped to when you measured the shaft-to-shaft positions.
5. Set the indicator in the top position and insure that the stem is loaded partway in its travel and zero the indicator.
6. Hold the assembly in the horizontal position and rotate the entire pipe, bracket, span bar, and dial indicator assembly through 90° arcs and note the readings at each position (particularly at the bottom) and record what you see. These are referred to as the "sag" readings. Usually, the readings on each side are half of the bottom reading and all of the readings have a negative value (typically) (refer to Figure 6.56 through Figure 6.58).
7. Calculate what the readings would be if you would have been using a bracket that had no sag in it. These are referred to as the "compensated" readings. Refer to the example shown in Figure 6.59.

If you are aligning a motor and a pump with a bracket that has 12 mils of sag, what measurements should you get to know that the two shafts are in alignment? It is extremely important to note that when the machinery shafts are in perfect alignment, the dial indicator readings will be exactly what the sag readings are as illustrated in Figure 6.60. In other words, if you are "spinning zeros," the shafts are misaligned. When using mechanical brackets on horizontally mounted shafts, you virtually never want to get a series of zeros when you have done, unless of course you are using a perfect mechanical bracket that has no sag.

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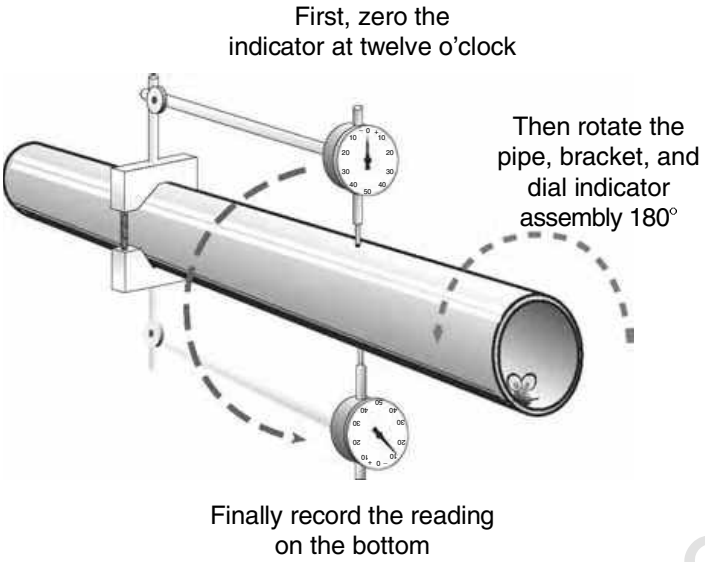


FIGURE 6.56 Bracket sag.

As mentioned previously, if you do not compensate the field readings for bracket sag, and attempt to calculate shim changes to adjust the positions of the machinery for realignment, you will come up with wrong amounts for the shims.

6.13 XMAS TREE BRACKETS AND FACE SAG

Some mechanical bracketing fixtures do not clamp around the circumference of a shaft but bolt directly to the end of a shaft as shown in Figure 6.61. These are often referred to as "Christmas tree" brackets. The brackets are generally very stiff and it is assumed that there is no sag in the fixture. That is not usually true.

To measure the amount of sag in an Xmas tree bracket, you will have to fabricate a sag checking device similar to what is shown in Figure 6.62, which also shows how to check for face sag. If the Xmas tree bracket is very large, it can also be checked by holding the bracket in jaws of a lathe and setting the indicator out on the live center. This works fine assuming

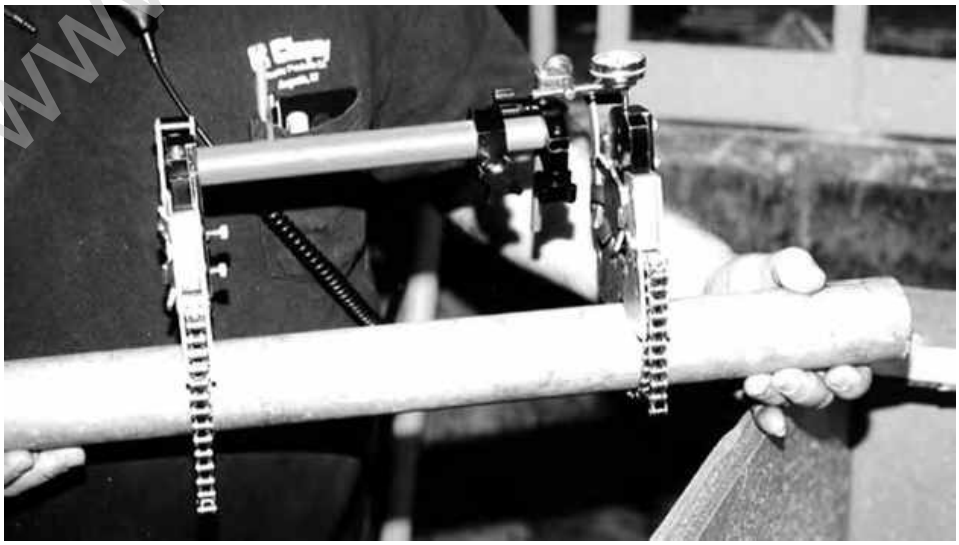


FIGURE 6.57 Initial position of the bracket in the twelve o'clock position where the indicator is zeroed.

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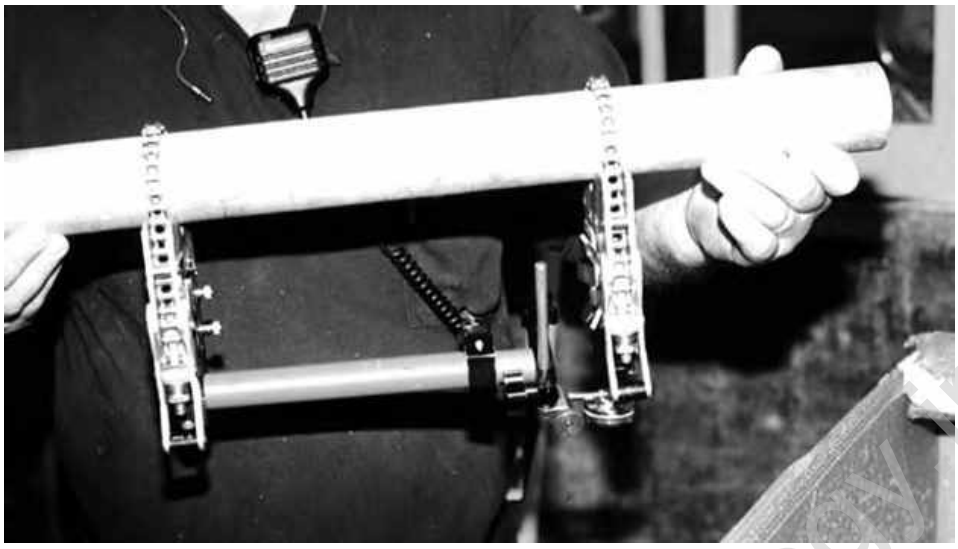


FIGURE 6.58 Observe the amount of sag on the dial indicator as you rotate the assembly to the six o'clock position.

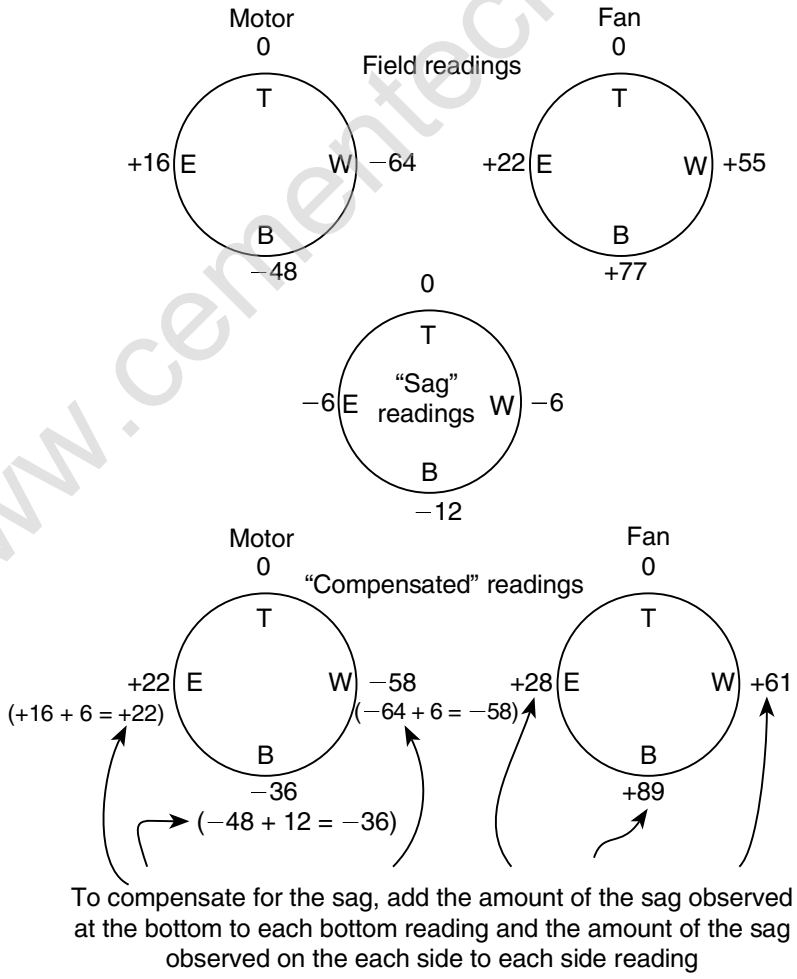


FIGURE 6.59 Mathematically compensating for sag.

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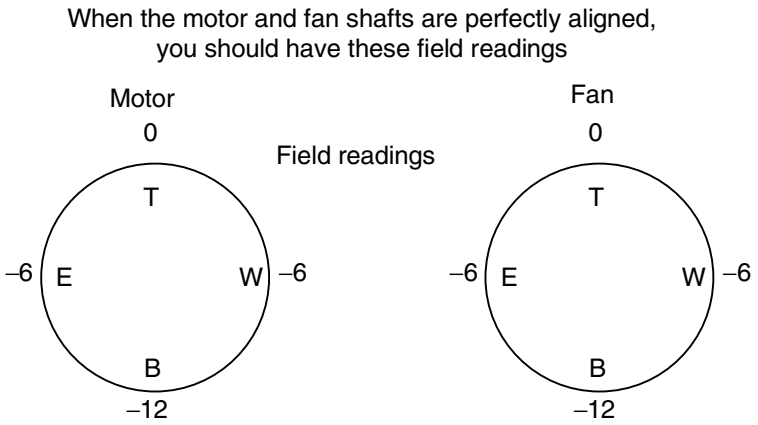


FIGURE 6.60 Ideal "shoot-for" readings (assuming no OL2R movement occurs).

that the centerline of rotation of the spindle and chuck is perfectly in line with the centerline of rotation of the live center.

6.14 ZERO SAG BRACKETS

Is it possible to defy gravity and construct a cantilevered beam that compensates for the elastic bending due to the force of gravity? You bet, within limits. Figure 6.63 shows a bracket design where a second span bar holds a weight so that its position is on the opposite side of the clamping point on the shaft, thereby counterbalancing the overhung weight. Figure 6.64 shows another arrangement for long spans where a different type of counterweight is placed on a coupling spool that "lifts" the span bar. Neither of these prototype "anti-sag" devices are included in any of the dial indicator based alignment measurement systems. Another long span bracket system fabricated from lightweight carbon tubes is shown in Figure 6.65.

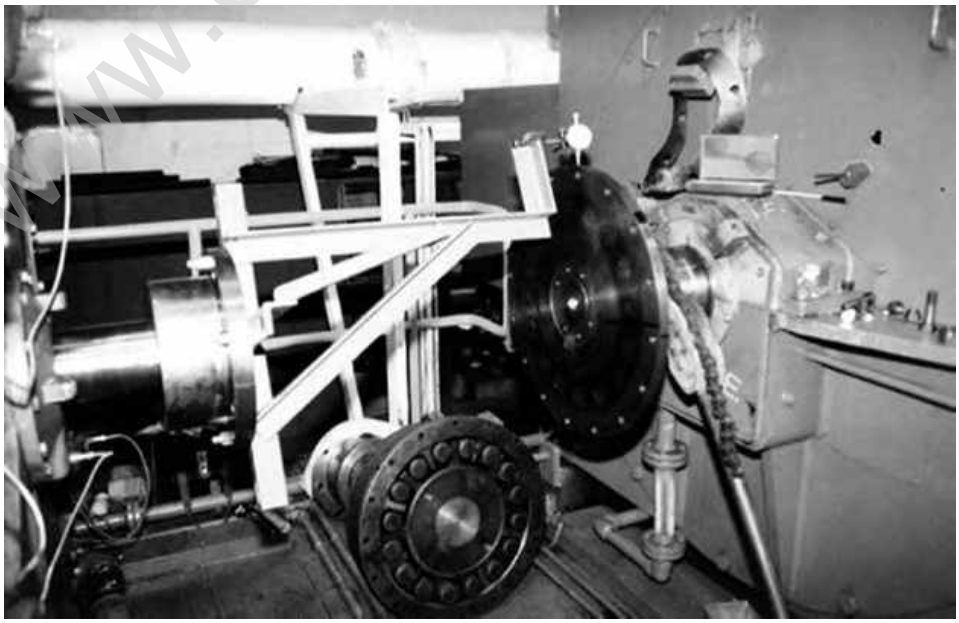


FIGURE 6.61 Xmas tree bracket.

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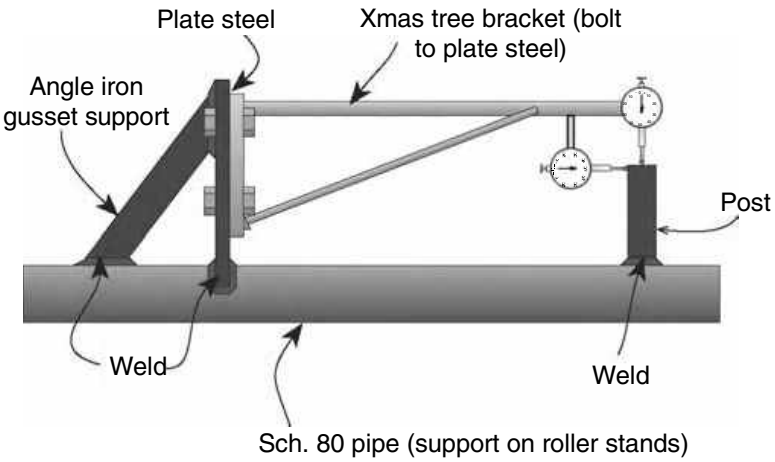


FIGURE 6.62 How to measure sag when using an Xmas tree bracket.

6.15 DIAL INDICATOR SHAFT ALIGNMENT SYSTEM MANUFACTURERS

The ensuing pages cover the currently available dial indicator based shaft alignment systems. There are currently 7 manufacturers offering 19 different models.

A talented person could fabricate his own shaft alignment fixtures, purchase dial indicators, and perform very satisfactory shaft alignment. The Xmas tree brackets shown in this chapter were custom fabricated and work very well. Small to medium size fixtures can be made that clamp around the outside of a shaft or coupling hub using roller chain or hose clamps. The bracket body can be machined out of a block of steel or fabricated with pieces of angle iron. Span bars can be made with tubing or round stock and can be cut to various lengths to suit the users needs. Dial indicator sleeves can be purchased to clamp on various diameters but the ones typically used clamp on 1/4, 5/16, or 3/8 in. rods with metric sizes available also. It may take several hours for some experiments (also known as prototypes) to get what you need. On the other hand, the following tools have already gone through a development cycle and are commercially available at relatively modest prices.

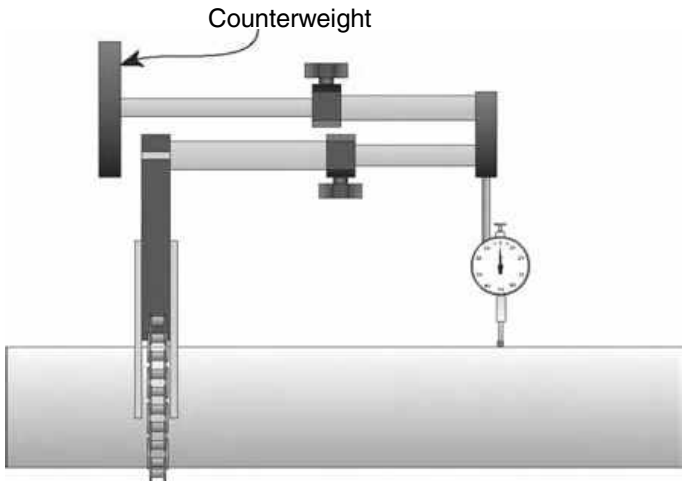


FIGURE 6.63 Turvac Inc. sag compensation bracket design.

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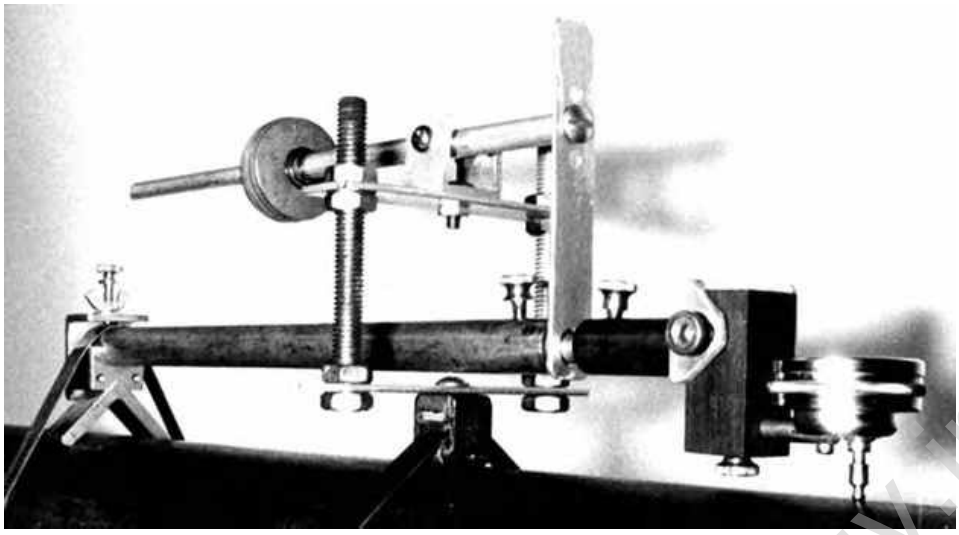


FIGURE 6.64 Sag compensation fixture design. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

6.15.1 Accushim Systems

Accushim offers five different dial indicator based systems as shown in Figure 6.66 through Figure 6.70. The REACT SA-2 system includes a pocket computer with alignment software. Chain, brackets, posts, targets, and rods can be purchased separately to extend the capabilities of the systems.

SK-1 system cost: $425.00
SK-2 system cost: $575.00
REACT HA-3 system cost: $825.00
REACT HA-2 system cost: $1500.00



FIGURE 6.65 Long span carbon tube. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

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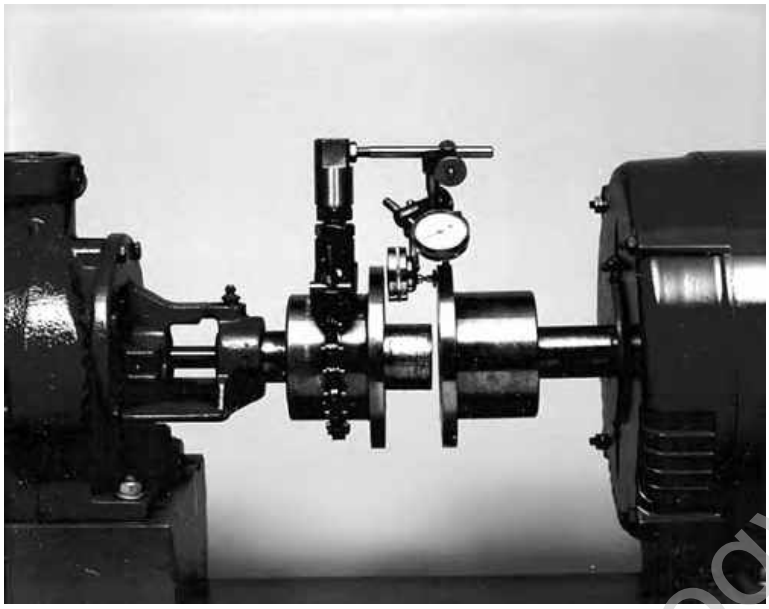


FIGURE 6.66 SK-1 system. (Courtesy of Accushim Inc., Lyons, IL. With permission.)

REACT SA-2 system cost: $2250.00
Manufacturers Web site: http://www.accushim.com/

6.15.2 A-Line Systems

A-Line Mfg. Inc. A-750 system as shown in Figure 6.71 is designed for smaller, hard to fit equipment, and ANSI pumps. The A-1000 system is designed to clamp directly onto the shafts and reach over the coupling as shown in Figure 6.72. The A-2000 model is designed to clamp onto the coupling hub and reach over the hub flange as shown in Figure 6.73. The A-3000 system combines both the A-1000 and A-2000 components (Figure 6.74). An optional Palm OS computer is available with alignment software.

A-750 system cost: $875.00
A-1000 system cost: $1025.00
A-2000 system cost: $1225.00
A-3000 system cost: $1675.00
Palm OS with software cost: $750.00
Manufacturers Web site: http://www.alinemfg.com/

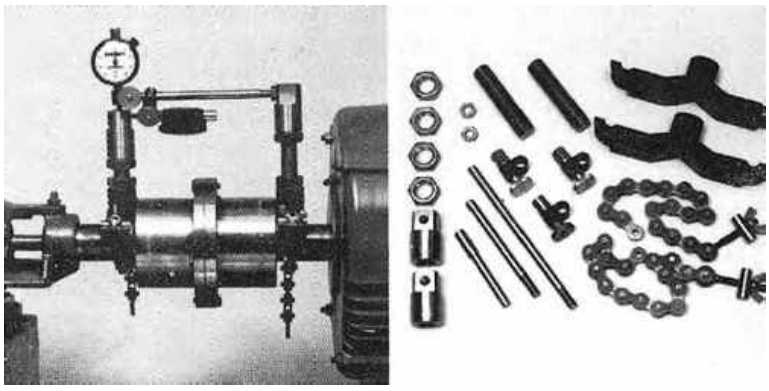


FIGURE 6.67 SK-2 system. (Courtesy of Accushim Inc., Lyons, IL. With permission.)

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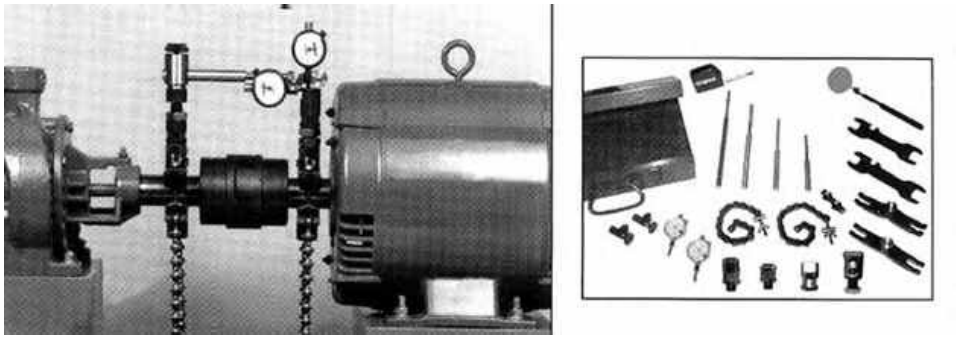


FIGURE 6.68 REACT HA-3 system. (Courtesy of Accushim Inc., Lyons, IL. With permission.)



FIGURE 6.69 REACT HA-2 system. (Courtesy of Accushim Inc., Lyons, IL. With permission.)

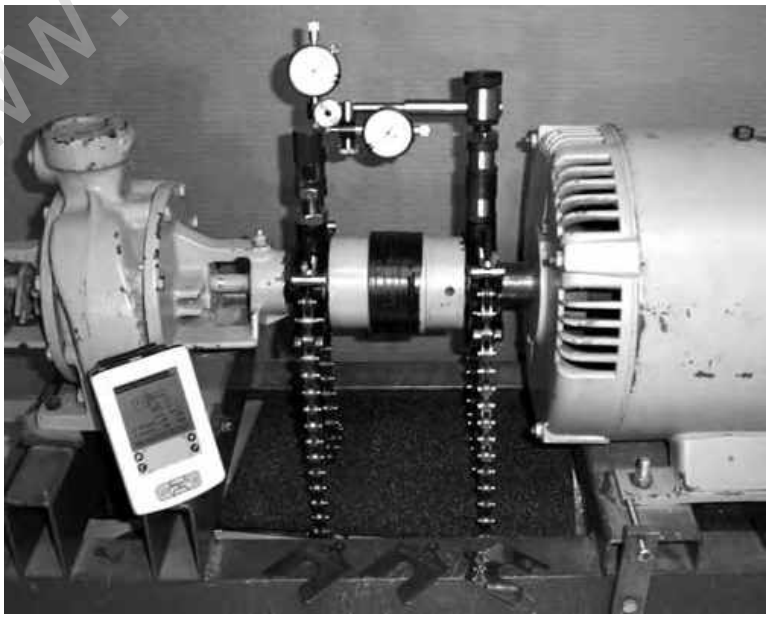


FIGURE 6.70 REACT SA-2 system. (Courtesy of Accushim Inc., Lyons, IL. With permission.)

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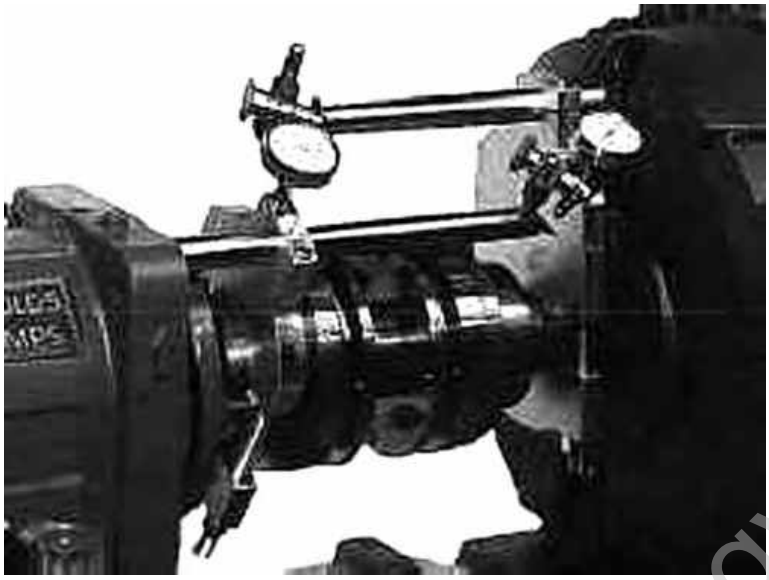


FIGURE 6.71 A-750 system. (Courtesy of A-Line Mfg. Inc., Liberty Hill, TX. With permission.)

6.15.3 Benchmark System

The Benchmark PAS-1 alignment kit is shown in Figure 6.75 and Figure 6.76.
PAS-1 System cost: $1200.00
Manufacturers Web site: http://www.withinspec.com/

6.15.4 Murray & Garig System

If you have not been noticing, there are several references in this book to describe the owner of this company, Malcolm Murray. Malcolm is one of the foremost experts in shaft alignment and holds seven patents, including a patent for several reverse indicator brackets as well as



FIGURE 6.72 A-1000 system. (Courtesy of A-Line Mfg. Inc., Liberty Hill, TX. With permission.)

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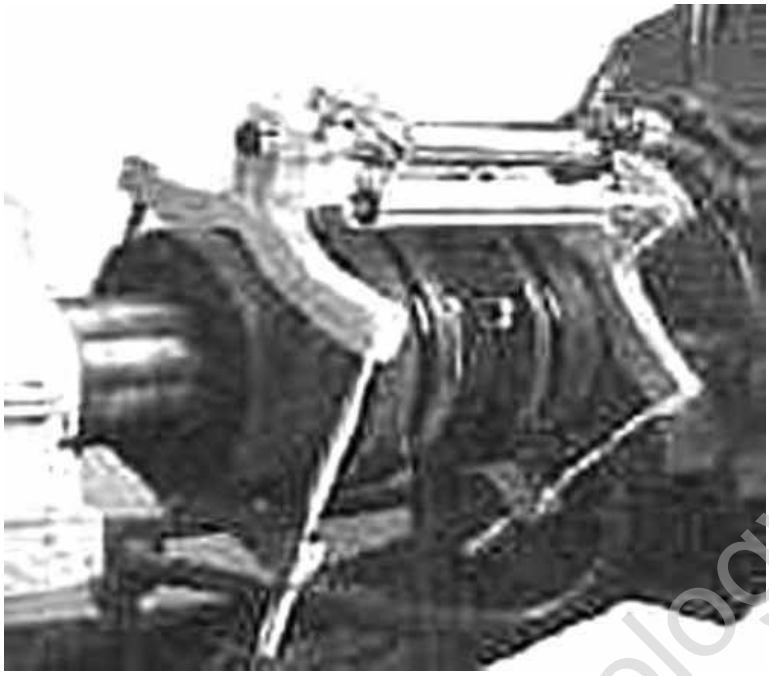


FIGURE 6.73 A-2000 system. (Courtesy of A-Line Mfg. Inc., Liberty Hill, TX. With permission.)

many other machinery alignment products shown in this book. He has been a consultant to many of the manufacturers and users of dial indicator and laser-based shaft alignment measurement systems shown in this book. The Murray & Garig K-5 shaft alignment system is shown in Figure 6.77 and Figure 6.78.

K-5 system cost: $2300.00 to $2900.00 depending on options.

6.15.5 Peterson Alignment Tools Co. Systems

Peterson Alignment Tools Co. offers two different alignment kits, the 20RA and 30RA, as shown in Figure 6.79 and Figure 6.80. Also offered is the Alignment Manager™ 2002



FIGURE 6.74 A-3000 system. (Courtesy of A-Line Mfg. Inc., Liberty Hill, TX. With permission.)

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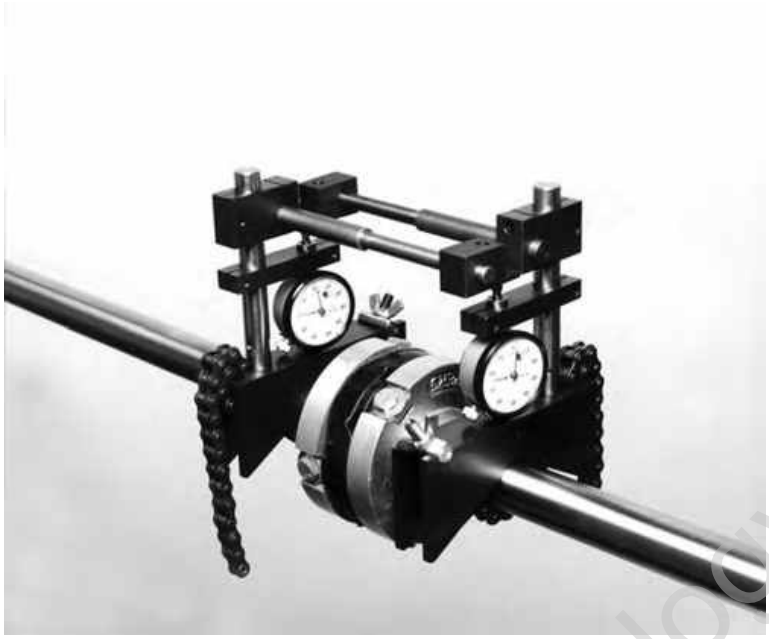


FIGURE 6.75 PAS-1 system set up for reverse indicator method. (Courtesy of Benchmark Maintenance Services Inc., Toronto, Ontario, Canada. With permission.)

software program that operates on Windows™ operating systems that can be purchased separately. Another version (Alignment Manager 3.1a) can be loaded onto a TI-86 scientific calculator.

20RA system cost: $795.00 (includes Alignment Manager 2002 software program)
30RA system cost: $995.00 (includes Alignment Manager 2002 software program)
AMTI-86 scientific calculator cost: $199.00
Alignment Manager 2002 or 3.1a cost: $79.95
Manufacturers Web site: http://www.petersontools.com/

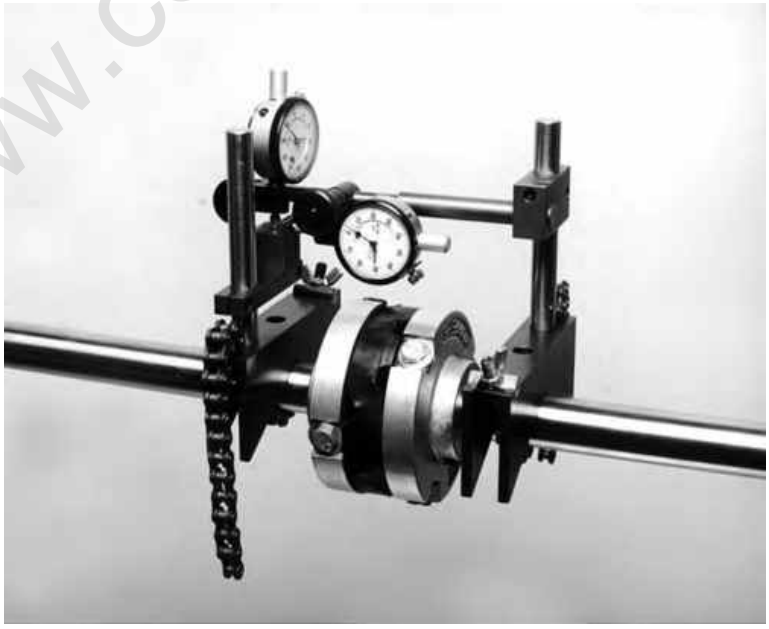


FIGURE 6.76 PAS-1 system set up for face–rim method. (Courtesy of Benchmark Maintenance Services Inc., Toronto, Ontario, Canada. With permission.)

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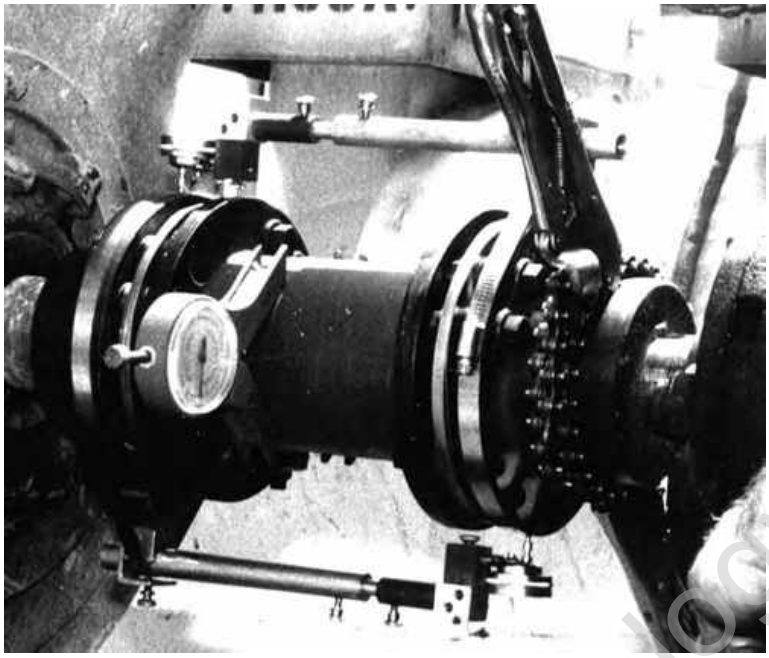


FIGURE 6.77 K-5 system. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

6.15.6 Turvac Inc. Systems

Turvac Inc. offers five different systems. Figure 6.81 shows the miniature bracket that is used for small couplings. Figure 6.82 shows the standard bracket kit. Figure 6.83 is the Journeyman system which includes two standard bracket kits, a magnetic base, and the Shaft Alignment Software Program or Technical reference library (the book *Basic Shaft Alignment*



FIGURE 6.78 K-5 system components. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

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FIGURE 6.79 Peterson 20RA system. (Courtesy of Peterson Alignment Tools Co., Channahon, IL. With permission.)

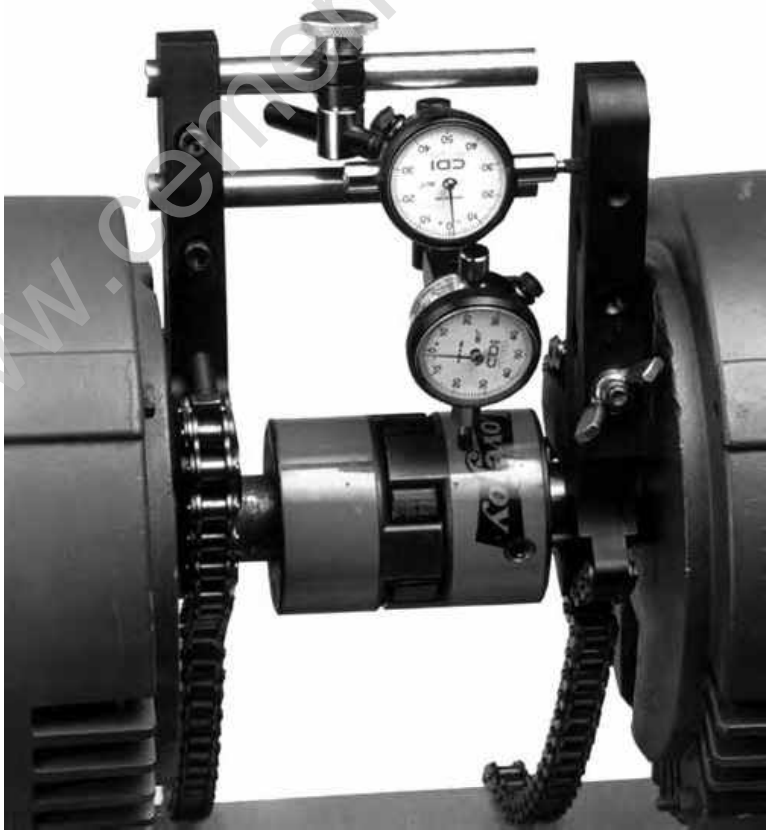


FIGURE 6.80 Peterson 30RA system. (Courtesy of Peterson Alignment Tools Co., Channahon, IL.)

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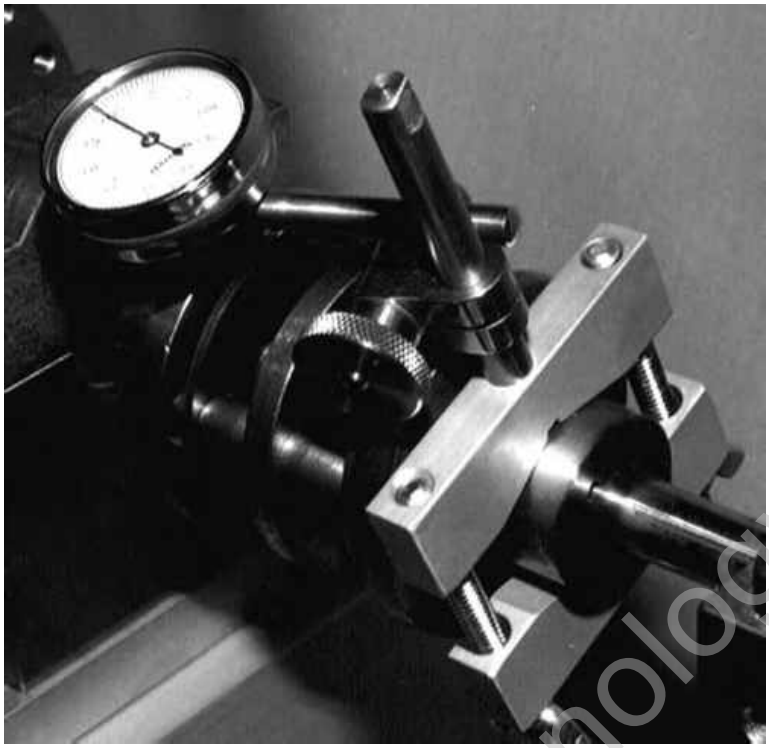


FIGURE 6.81 Miniature bracket. (Courtesy of Turvac Inc.)

Workbook and a manual for the Shaft Alignment Software Program). The software program comes in two versions, one that runs on Windows or disk operating system (DOS)-based computers, and another that runs on Apple Macintosh operating systems. The Journeyman system can be bundled with a Windows (Master system) or Macintosh computer (Professional system) (Figure 6.84).

Miniature bracket cost: $495.00
Standard system cost: $1495.00

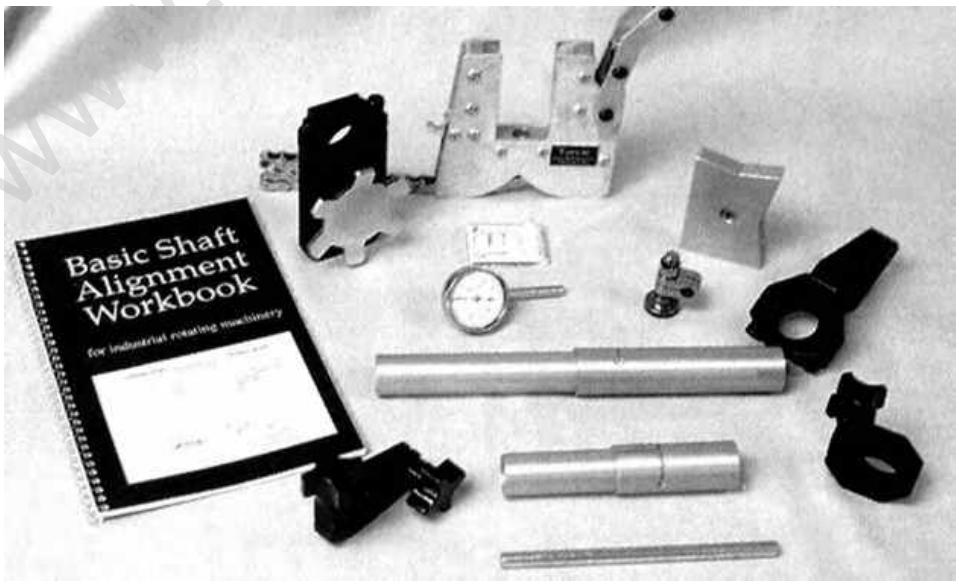


FIGURE 6.82 Standard alignment system. (Courtesy of Turvac Inc.)

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FIGURE 6.83 Journeyman alignment system. (Courtesy of Turvac Inc.)

Journeyman system cost: $3495.00
Master system cost: $5995.00
Professional system cost: $5995.00
Shaft Alignment Software Program and technical reference library cost: $1295.00
Manufacturers Web site: http://www.turvac.com

6.15.7 Update International System

The Update Proaction system is shown in Figure 6.85.

Proaction system cost: $2490.00
Manufacturers Web site: http://www.update-intl.com

6.16 DIAL INDICATOR MANUFACTURERS HARDWARE SPECIFICATIONS

A questionnaire was sent out to all of the above manufacturers concerning the pricing, specifications, and features of their systems. The questions are listed below. A summary of their answers is shown in Figure 6.86 and Figure 6.87.

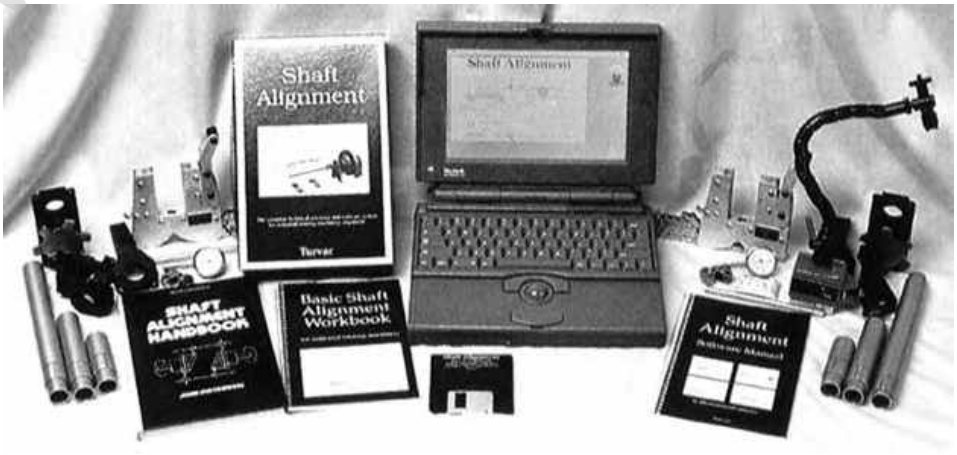


FIGURE 6.84 Master or Professional alignment system. (Courtesy of Turvac Inc.)

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FIGURE 6.85 Update Proaction system. (Courtesy of Update International Inc., Denver, CO. With permission.)

Dial indicator shaft alignment system questionnaire:

1. Are you a manufacturer of a dial indicator shaft alignment system?
   _____ Yes
   _____ No
2. Are you a distributor of a dial indicator shaft alignment system? If so, please list the manufacturer of the dial indicator shaft alignment system. Please indicate the company, address, phone number, contact person, and Web site (if applicable) of the manufacturer. If you are a distributor only, skip the remaining questions.

   Distributor for:
   Model# _____
   Manufacturer name _____
   Address _____
   P.O. Box • Suite _____
   City _____
   State/Province _____
   Zip/Postal Code _____
   Country _____
   Phone# _____
   Fax# _____
   E-mail address _____
   Web site _____

   Questions for the manufacturer of a dial indicator shaft alignment system

1. What are the minimum and maximum shaft diameters that your alignment brackets can be clamped to? If you have different models that can be clamped to different ranges of shaft diameters, please indicate what range applies to each model.
   _____ (inches? millimeters?) minimum shaft diameter
   _____ (inches? millimeters?) maximum shaft diameter
2. What are the minimum and maximum distances your alignment brackets can span from shaft to shaft (or from bracket to bracket)?
   _____ (inches? millimeters?) minimum shaft-to-shaft span
   _____ (inches? millimeters?) maximum shaft-to-shaft span

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Manufacturer system/model	Accushim SK-1	Accushim SK-2	Accushim REACT HA-3	Accushim REACT HA-2	Accushim REACT SA-2	A-Line A-750	A-Line A-1000	A-Line A-2000	A-Line A-3000
System price (US$)	425.00	575.00	825.00	1500.00	2250.00	875.00	1025.00	1225.00	1675.00
Basic system information									
System weight (lb)	10	12	15	32	32	7	17	20	30
Brackets supplied?	One	Two	Two	Two	Two	Yes	Yes	Yes	Yes
Dial indicators supplied?	One	One	Two	Two	Two	Yes	Yes	Yes	Yes
Use on horizontally mounted machines?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Use on vertically mounted machines?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Custom calculator supplied?	No	No	No	No	Yes	Optional	Optional	Optional	Optional
Standard computer supplied?	No	No	No	No	No	No	No	No	No
Alignment software included?	No	No	No	No	Yes	Optional	Optional	Optional	Optional
Warranty period (months)	36	36	36	36	36	12	12	12	12
Purchase parts separately?	Yes	Yes	Yes	Yes	Yes				
Shaft bracket fixtures									
Minimum shaft diameter (in.)	0.375	0.375	0.375	0.375	0.375	0.5	1	3	1
Maximum shaft diameter (in.)	3.5	3.5	5	9.5	9.5	2.88	5	10.5	10.5
Minimum shaft-to-shaft span (in.)	2	2	2	3	3	1.5	1.5	1.5	1.5
Maximum shaft-to-shaft span (in.)	6	6	12	14	14	9	18	18	18
Minimum span bar height (in.)	3	3	3	3	3	2.2	3.5	2.75	3.2
Maximum span bar height (in.)	7	7	9	12	12	2.2	3.5	75	3.2
Brackets attach around shaft circumference?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Brackets attach to face of coupling?	No	No	Yes	Yes	Yes	No	No	No	No
Shafts must be rotated together?	No	No	Yes	Yes	Yes	Yes	Yes	Yes	Yes
How are rotational positions measured?	Level	Level	Level	Level	Level	Twin spirit level	Twin spirit level	Twin spirit level	Twin spirit level
Shaft alignment method capabilities									
Face–rim method?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Reverse indicator method?	No	No	No	Yes	Yes	Yes	Yes	Yes	Yes
Shaft to coupling spool method?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Double radial method?	Yes	Yes	Yes	Yes	Yes	No	No	No	No
Face–face method?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes

FIGURE 6.86 Dial indicator manufacturer pricing, specifications, and features summary.

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Manufacturer system/model	Benchmark PAS-1	Murray & Garig K-5	Peterson 20RA	Peterson 30RA	Turvac Inc. Miniature	Turvac Inc. Standard	Turvac Inc. Journeyman	Turvan Inc. Master	Turvac Inc. Professional	Update Int'l. Proaction
System price (US$)	1200.00	2300.00–$2900.00	795.00	995.00	495.00	1495.00	3495.00	5995.00	5995.00	2490.00
Basic system information										
System weight (lb)	5	24	9	12	2	8	20	25	25	35
Brackets supplied?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Dial indicators supplied?	Yes	Yes	Yes	Yes	One	Yes	Yes	Yes	Yes	Yes
Use on horizontally mounted machines?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Use on vertically mounted machines?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Custom calculator supplied?	Optional	No	Optional	Optional	No	No	No	No	No	Yes
Standard computer supplied?	No	No	No	No	No	No	No	DOS-based laptop Apple	Macintosh	No
Alignment software included?	Optional	No	Optional	Optional	No	No	Yes	Yes	Yes	Yes
Warranty period (months)	12	12	12	12	12	12	12	12	12	12
Purchase parts separately?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	
Shaft bracket fixtures										
Minimum shaft diameter (in.)	0.5	1.5	0.75	0.75	0.13	0.5	0.5	0.5	0.5	0.625
Maximum shaft diameter (in.)	8	48	3.5	7.5	2.75	48	48	48	48	12
Minimum shaft-to-shaft span (in.)	4	1.25	5	5	0.5	0.5	0.5	0.5	0.5	2.5
Maximum shaft-to-shaft span (in.)	10	23	9	15	3	33	33	33	33	23
Minimum span bar height (in.)	3.5	1.75	2.1	1.9	0.5	–0.5	–0.5	–0.5	–0.5	3
Maximum span bar height (in.)	6.25	7	2.1	4.8	2.75	11	11	11	11	12
Brackets attach around shaft circumference?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Brackets attach to face of coupling?	No	No	Yes	Yes	No	Yes	Yes	Yes	Yes	No
Shafts must be rotated together?	No	No	Yes	Yes	No	No	No	No	No	Yes
How are rotational positions measured?	Twin spirit level	Automatic protractor	Eyeball	Eyeball	Twin spirit level	Twin spirit level	Twin spirit level	Twin spirit level	Twin spirit level	Spiral level
Shaft alignment method capabilities										
Face–rim method?	Yes	Yes	Yes	Yes	No	Yes	Yes	Yes	Yes	Yes (addtn'l parts req'd)
Reverse indicator method?	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Shaft to coupling spool method?	Yes	Yes	No	No	Yes	Yes	Yes	Yes	Yes	Yes (addtn'l parts req'd)
Double radial method?	Yes	Yes	No	No	Yes	Yes	Yes	Yes	Yes	Yes (addtn'l parts req'd)
Face–face Method?	Yes	Yes	No	No	No	Yes	Yes	Yes	Yes	Yes (addtn'l parts req'd)

FIGURE 6.87 Dial indicator manufacturer pricing, specifications, and features summary.

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3. What is the distance from the point where the bracket touches the shaft to the center of the span bar? If your bracket system can change the height of the span bar, what is the minimum and maximum distance from the point where the bracket is touching the shaft to the center of the span bar?
   _____ (inches? millimeters?) fixed height
   _____ (inches? millimeters?) minimum height
   _____ (inches? millimeters?) maximum height
4. Can your brackets be attached to the face of coupling hubs rather than clamp around a shaft diameter?
   _____ Yes
   _____ No
5. Can your alignment bracket system be used to perform the reverse indicator technique?
   _____ Yes
   _____ No
6. Can your alignment bracket system be used to perform the face-peripheral (i.e., face–rim) technique?
   _____ Yes
   _____ No
7. Can your alignment bracket system be used to perform the shaft to coupling spool technique?
   _____ Yes
   _____ No
8. Can your alignment bracket system be used to perform the double radial technique?
   _____ Yes
   _____ No
9. Can your alignment bracket system be used to perform the face–face technique?
   _____ Yes
   _____ No
10. Do the shafts have to be rotated together while capturing readings? If so, what amount of rotational backlash between the two shafts can be tolerated before measurement accuracy is sacrificed?
    _____ Yes
    _____ No
    _____ (angular degrees) allowable backlash
11. Do you supply dial indicators with your system? If so, are they standard types of dial indicators that can be purchased directly from the dial indicator manufacturers (e.g., Starrett, Mitutoyo, Central, etc.) in the event that the user breaks or loses a dial indicator?
    _____ Yes, indicators are supplied
    _____ No, indicators not supplied
    Manufacturer and model _____
12. Can other measuring devices such as LVDTs, proximity probes, lasers, or CCDs be used with or on your brackets?
    _____ Yes
    _____ No
13. How do you measure the rotational position of the brackets, indicators, and sensors when capturing readings?
    Angle measuring device used _____

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14. If a user damages or loses a component of the system, can the parts be purchased separately?
    _____ Yes
    _____ No
15. Can the system be used on vertically oriented rotating machinery?
    _____ Yes
    _____ No
16. Can the system be configured to measure English or metric units?
    _____ Yes
    _____ No
17. If your system is patented, please send a copy of the patent (in English). If you have patents in other countries, please indicate where the patents are held and their corresponding patent number or identification.
    _____
    _____
    _____
18. What is your warranty period?
    _____ (days, months, years)
19. What are the recommended calibration intervals? Does the unit have to be sent back to the factory for calibration, and what is the charge for recalibration? Is the calibration you offer traceable back to U.S. or International Standards?
    _____ (days, months, years) recommended calibration interval
    Send back to factory?
    _____ Yes
    _____ No
    Traceable to standards?
    _____ Yes
    _____ No
20. Do you offer equipment for the user to check and adjust calibration? If so, what is the cost of the test equipment?
    _____ Yes
    _____ No
21. If you have introduced new shaft alignment system models, do the people who own your original models have the option to trade in their older models to upgrade to a newer model? If so, what is the cost of the upgrade?
    _____ Yes—upgrade cost _____ (US$)
    _____ No
22. What is the overall weight of the entire system?
    _____ (pounds, kilograms) system weight
23. What is the price (or price range) of the system?
    _____ (US$)

This chapter has shown the wide variety of measurement tools that are used in aligning rotating machinery. Chapter 10 through Chapter 15 will show how these tools are used to capture shaft-to-shaft positional measurements.

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7 Correcting Misalignment

Assuming that we are attempting to align healthy machines (good bearings, seals, shafts, couplings, baseplates, foundations, etc.), our objectives are as follows:

Objective 1: Determine the relative position of two centerlines of rotation of the machinery shafts. If the amount of misalignment is not acceptable decide which way and how far one, or the other, or both of the machines have to move in the *X*, *Y*, and *Z* planes (i.e., up, down, left, right, toward or away) to improve the desired off-line alignment.

Objective 2: Physically move one or the other or both of the machines in the *X*, *Y*, and *Z* planes (i.e., up, down, left, right, toward or away) as shown in Figure 7.1. Go back to objective 1 until you reach acceptable alignment tolerances.

7.1 INSTALLING MACHINERY FOR THE FIRST TIME

If we are installing new machinery for the very first time, there are several issues we need to deal with. Whether the machinery will be placed on a rigid foundation, an inertia block, or a structural steel frame, and the following items should be taken into consideration:

1. If we are installing machinery on a rigid foundation or an inertia block has the concrete and grout been given enough time to cure? Typically concrete- and cement-based grouts will have achieved 50% of their cure after 4–7 d and 80%–90% of their cure after 15–30 d. Therefore it is not recommended to place machinery onto its baseplate the day after the concrete was poured.
2. Will there be any external connections made to the machines such as piping, ductwork, or electrical conduit? If so, have these external connections already been fabricated and put in their final position before placing the machinery onto its baseplate? Hopefully not. The machinery should be set into a rough or final alignment position before making any connections. Remember, the piping flanges on pumps, compressors, and steam turbines are fluid connection points, not piping anchors. Adequate support structures should be provided to support the piping that will eliminate any stresses on the machinery itself.
3. Have any considerations been drilled and tapped into the baseplate or soleplates where the machines will be mounted? If so, have the machines that will actually be mounted to the base been at least roughly aligned in the axial and lateral directions on this base before drilling and tapping the holes insuring that the foot bolts have been placed in the center of the holes? Hopefully so. Drilling and tapping holes in the baseplate or soleplate is more difficult than it looks and requires some effort in locating the holes precisely.

Figure 7.2 shows the specifications from the National Electrical Manufacturers Association (NEMA) for foot pad sizes, the hole diameters to be drilled into the machinery feet, and the bolt diameter to hold that foot firmly to the baseplate. As you can see, there is not much

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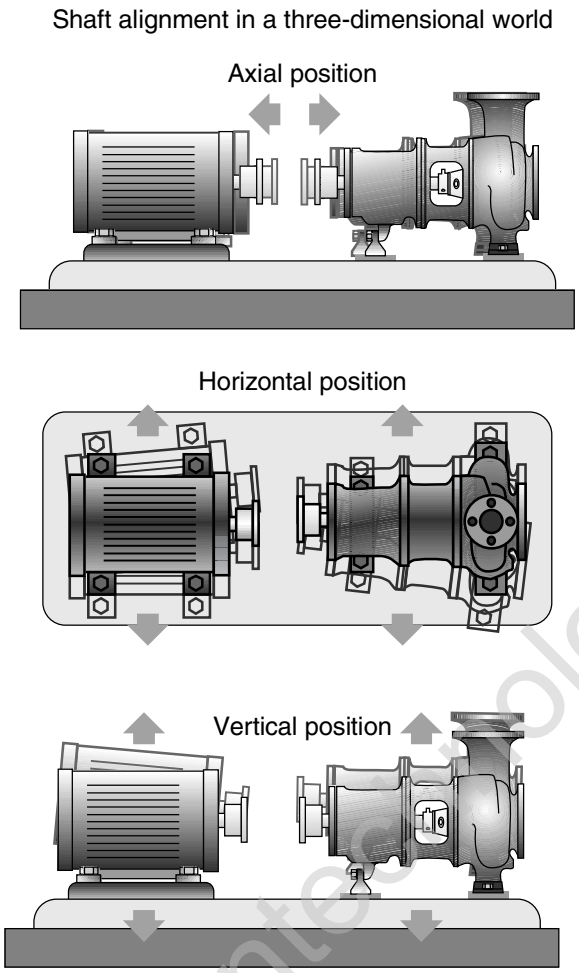


FIGURE 7.1 Moving in the *X*, *Y*, and *Z* planes.

room between the shank of the bolt and the hole drilled into the machine foot. Figure 7.3 shows the bolt hole size, the bolt used for that hole size, and the total diametral clearance between the bolt and the hole. Be aware that if the bolt is placed in the center of the hole the machine can only translate half that amount (i.e., the radial clearance).

7.2 BOLT-BOUND CONDITIONS

Notice that in three of the footprint sizes there are only 78 mils of radial clearance (156 diametral clearance) on each side of the bolt assuming it is centered in the hole. There is not a lot of room to move a machine sideways. In the vertical direction, if you wanted to move a machine up 250 mils (i.e., 1/4 in.), you could indeed install a shim or plate to raise it that high. But if you wanted to move a machine 250 mils sideways, you would become "bolt bound" before the total move was made. The restrictions between the holes drilled in the machinery feet and the bolts holding the feet to the base are the primary reason why more machinery is misaligned in the side-to-side and axial directions than in the vertical direction. In the up direction, you have no limit to how far up it can go. You can add as many shims under the machinery feet (within reason) as you want. However, in the side-to-side or end-to-end direction, you will be limited by the amount of room left between the holes drilled in the machinery feet and the bolts holding the feet to the base. If you do not know how much room there is between the shank of the bolt and the hole in the machine case, loosen the bolt, remove the washer, and thread the bolt in a few turns as shown in Figure 7.4.

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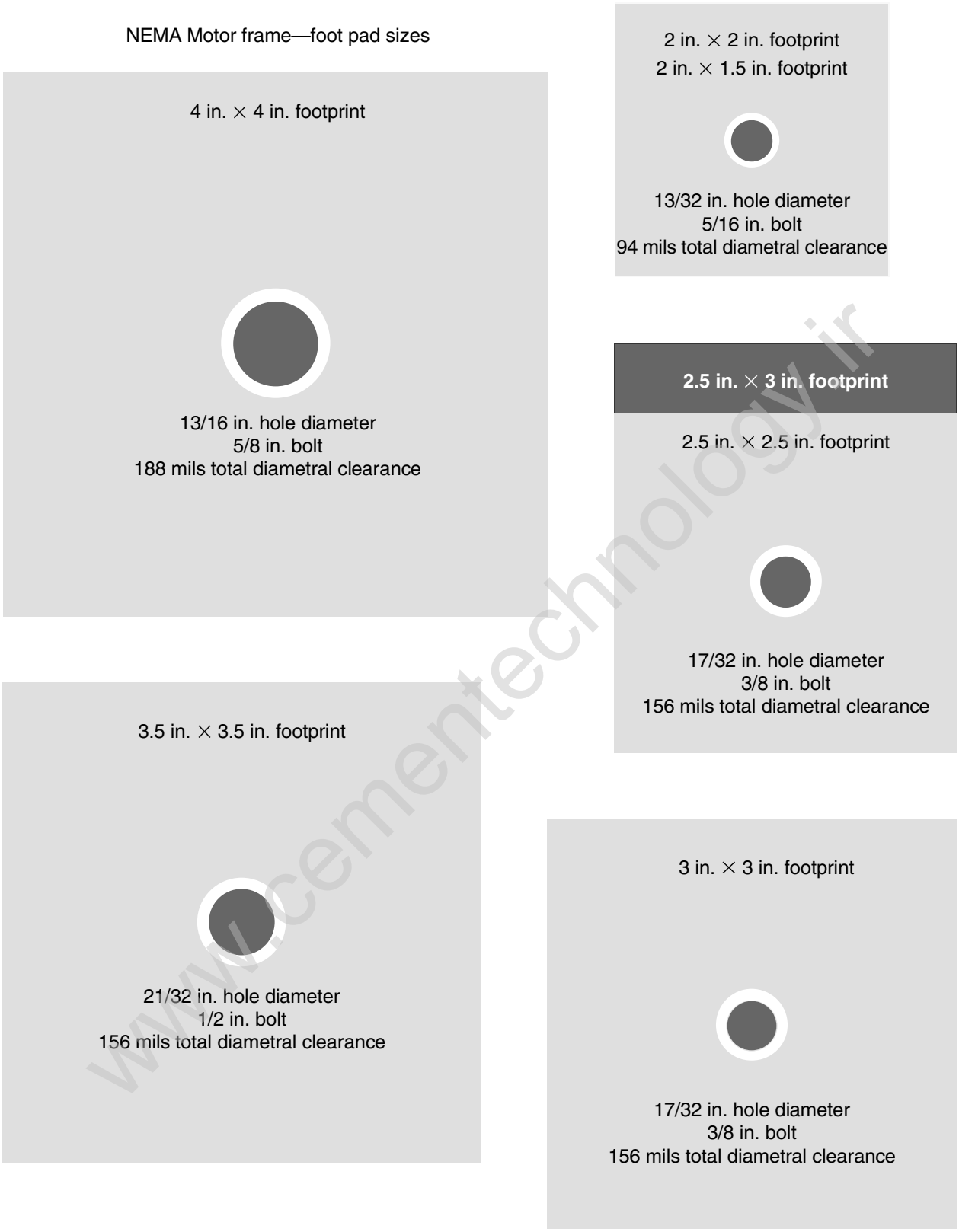


FIGURE 7.2 NEMA foot pad specifications.

Bolt	Diameter	Hole	Hole diameter	Clearance
5/16 in.	0.313	13/32 in.	0.406	0.094
3/8 in.	0.375	17/32 in.	0.531	0.156
1/2 in.	0.5	21/32 in.	0.656	0.156
5/8 in.	0.625	13/16 in.	0.813	0.188

FIGURE 7.3 Bolts, hole diameters, and diametral clearances.

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FIGURE 7.4 Observing bolt clearances at a machine foot.

7.3 LAST RESORT MEASURES FOR BOLT-BOUND CONDITIONS

An undercut bolt has the shank of the bolt cut down to the diameter at the root of the threads of the bolt as shown in Figure 7.5. Undercutting a bolt is one of the last resort measures you would take to achieve alignment. Another last resort measure would be to enlarge or slot the hole in the machine case.

Before you do this, study Chapter 8 and learn how to determine a lateral movement envelope. This could save you a considerable amount of time and unnecessary pain aligning machinery. In the event that you decide that undercutting a bolt is absolutely necessary, do

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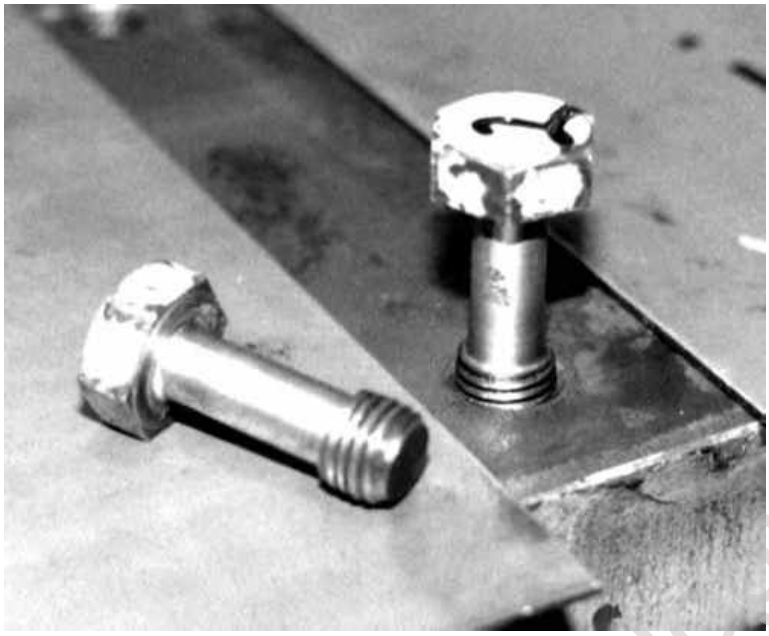


FIGURE 7.5 Undercut bolts.

not cut the shank down below the depth of the threads and use grade 8 bolts. If you decide to enlarge or slot the hole in the machine case, use thicker or hardened washers with the bolt. There are companies that offer undercut bolts as shown in Figure 7.6 and Figure 7.7.



FIGURE 7.6 Undercut bolt and hardened washer kit for bolt sizes from 5/16 to 1/2 in. (Courtesy of Posi Lock Puller, Inc., Cooperstown, ND. With permission.)

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FIGURE 7.7 Undercut bolt and hardened washer kit for bolt sizes from 9/16 to 3/4 in. (Courtesy of Posi Lock Puller, Inc., Cooperstown, ND. With permission.)

7.4 MACHINERY POSITIONING BASICS

We live in a three-dimensional world so correcting shaft misalignment requires a three-dimensional solution. Once we have determined our corrective moves as explained in Chapter 8 and Chapter 10 through Chapter 15, usually the most efficient way to reposition machinery casings is as follows:

1. Roughly adjust the axial position of the machinery insuring that the shaft end to shaft end distance is $\pm$100 mils of the final desired dimension.
2. Make a large move in the up and down direction by adding or removing shim stock between the machinery feet and the baseplate.
3. Make a fairly large move in the lateral (side-to-side) direction on one or all the machines.
4. Make a small (trim) move in the up and down direction by adding or removing shim stock between the machinery feet and the baseplate.
5. Make a small (trim) move in the lateral (side- to -side) direction on one or all the machines.
6. Adjust the axial position of the machinery insuring that the shaft end to shaft end distance is $\pm$10 mils of the final desired dimension.

7.4.1 Axial Spacing

For many applications where rotors are supported in antifriction bearings, the shaft end to shaft end distance measurement is fairly straightforward. For rotors with sliding type radial and thrust bearings, the normal operating position of the rotors must be taken into account. For example, electric motors with armatures supported in sliding type bearings will seek "magnetic center" when the field is applied. Therefore, motors should be run solo to determine where magnetic center is prior to setting the axial spacing. Centrifugal compressors, turbines, and other driven machinery will typically operate against their "active" thrust bearing. During the alignment process, their rotors should be seated against the active thrust bearing prior to setting the axial spacing. Refer to Chapter 5, specifically Figure 5.10 and Figure 5.11 for measuring axial thrust bearing clearance.

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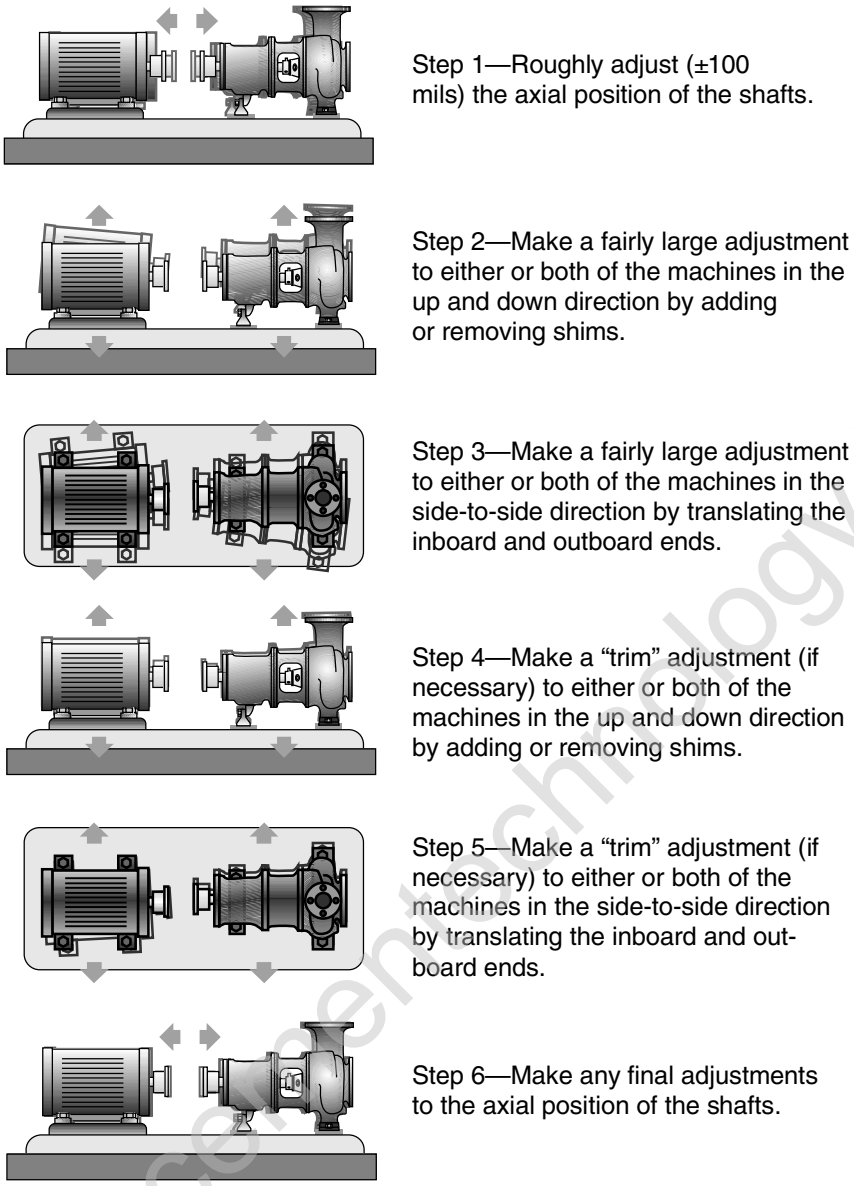


FIGURE 7.8 (**See color insert following page 322.**) Typical sequence of corrective moves.

The axial distance should be measured as close as possible to the centerline of rotation of each shaft unless otherwise specified by the equipment or coupling manufacturer. As shown in Figure 7.4, the first move you should make is an axial move. If you are angularly misaligned, and you measure the axial distance at the outer diameter on the coupling hub on one side, the distance on the other side of the coupling hub will be different. Bear in mind that the first axial move you make is $\pm$100 mils meaning that you could use a standard tape measure or ruler to measure this.

For your final axial distance, if no axial distance tolerance is given, the rule of thumb is to hold this distance to $\pm$0.010 in. of the recommended dimension. The importance of properly maintaining this gap cannot be over stressed as coupling "lockup" conditions occur as often from improper gap distances as from excessive vertical or lateral misalignment. The purpose of the coupling is to transmit rotational force, not thrust forces from one machine to another.

The coupling manufacturer will frequently have guidelines for what the correct spacing should be for their coupling. It is recommended that you obtain this information prior to your alignment job. Figure 7.9 shows information on determining the correct axial spacing for a gear type coupling.

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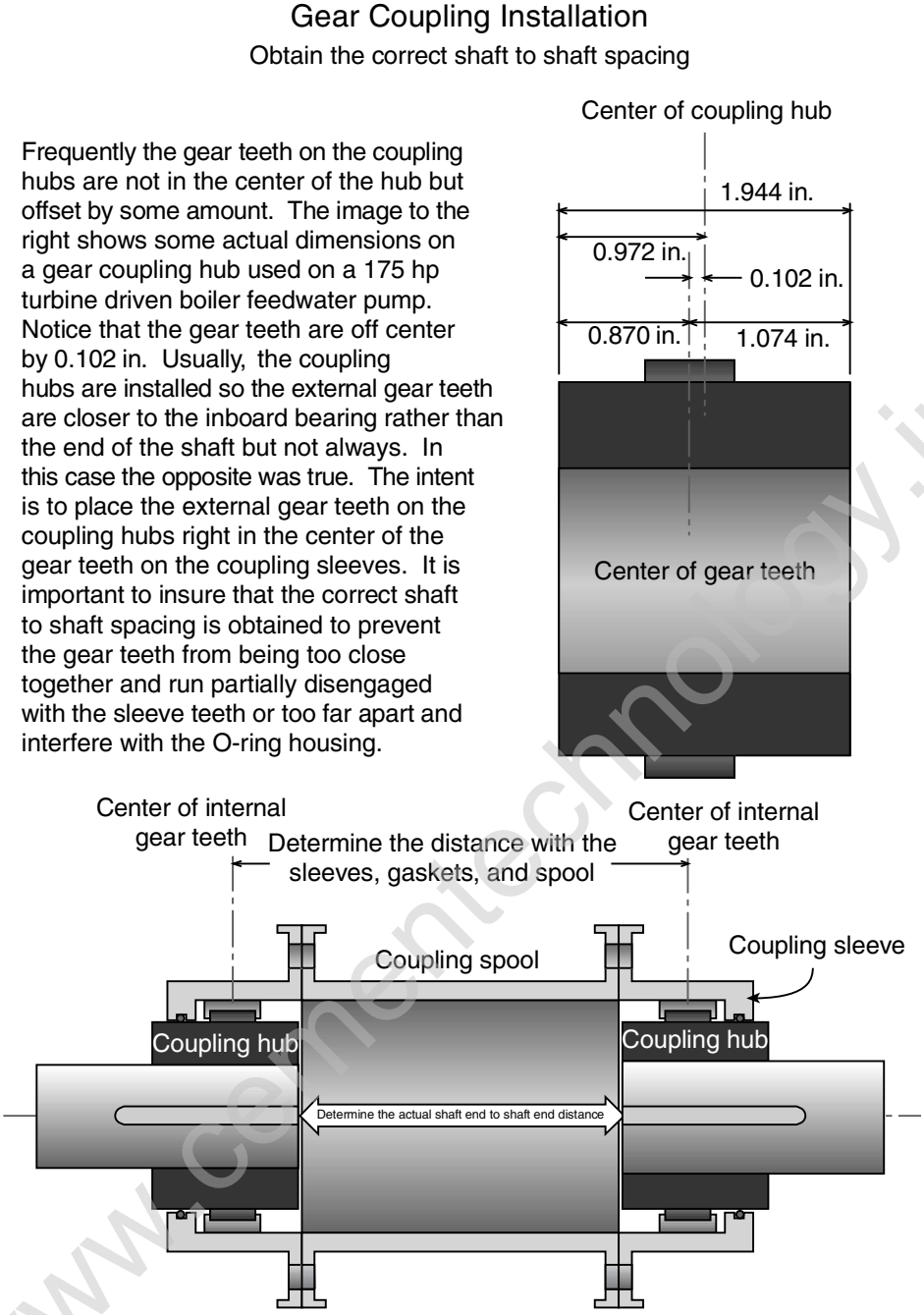


FIGURE 7.9 Determining the correct axial spacing for a gear coupling.

7.4.2 Making Corrections in the Vertical Direction

Well, that is, if you are talking about horizontally mounted machines. The most commonly used device to change the height or pitch of a machine case is shim stock. Shim stock is flat sheets of metal rolled or ground to thickness ranging from 0.001 to 0.125 in. (1 to 125 mils). Shim stock over 125 mils is generally referred to as "spacers" or "plates." Shim stock is commonly available in brass, carbon steel, or stainless steel, but can be made out of virtually any metal (or plastic) if desired. The accuracy of the thickness of shim stock is typically 5% of the rated thickness. For example, a 50 mil shim could range anywhere from 48 to 52 mils. Shim stock is typically sold in widths of 6 in. in lengths of 10 ft.

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FIGURE 7.10 A, B, C, D size shims. (Courtesy of Accushim, Inc., Lyons, IL. With permission.)

Several companies offer precut "U-shaped" shim stock sold in various sizes and thickness as shown in Figure 7.10 and Figure 7.11. These "precut" U-shaped shims are offered in four standard sizes (some manufacturers vary on these dimensions):

A size—2 in. × 2 in. with a 5/8 in. slot
B size—3 in. × 3 in. with a 3/4 in. slot
C size—4 in. × 4 in. with a 1–1/4 in. slot
D size—5 in. × 5 in. or 5 in. × 6 in. with a 1–1/4 in. slot

Precut U-shaped shim stock is typically made in the following shim thickness: 1, 2, 3, 4, 5, 10, 15, 20, 25, 50, 75, 100, and 125 mils thick. Some companies offer wider slots on what they



FIGURE 7.11 A, B, C, D size shims. (Courtesy of Precision Brand Products, Inc., Downers Grove, IL. With permission.)

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refer to as ''oversized slot'' shims. If it is necessary to install a shim/spacer thicker than 125 mils, it is recommended that carbon or stainless steel plate be used and that the plate be surface ground on both sides to insure parallel surfaces within 1 mil over a 6 in. distance. The spacer should have 125 rms surface finish or better after grinding.

The precut U-shaped shims mentioned above are typically made from stainless steel, which is the recommended material to use to prevent oxidation or corrosion that could occur with carbon steel shims. Brass shims work well unless there is ammonia present. Carbon steel is usually acceptable if you have to make a plate over 125 mils thick. The precut U-shaped shim sizes A through D were meant for use on NEMA motor frame sizes 100 through 400 series. These shims can obviously be used on other machines besides motors. These standard shim sizes work on about 70% of the machinery in industry but they do not work on everything, particularly larger machinery, so it is not uncommon to fabricate custom shims to work in many situations. The rule of thumb is to provide at least 80% contact between each machine foot and its point of contact on a baseplate, frame, or soleplate. Often I have found C-sized shims sitting under a machine foot whose footprint is 6 in. × 9 in. Just because the shim slot fits the hold down bolt diameter, does not necessarily mean that is the right shim for the job. Therefore, in many cases, special shims need to be fabricated.

Shim stock can also be purchased in 6 in. wide rolls in 100 in. and 180 in. lengths and can be aluminum, brass, carbon steel, and stainless steel in usually the following thickness: 1, 1.5, 2, 3, 4, 5, 6, 7, 8, 10, 12, 15, 20, 25, and 31 mils. To fabricate shims over these thickness, you need to use sheet metal that can be purchased in the following gauges:

Gauge	Thickness (in.)
28	0.015
26	0.018
24	0.024
22	0.030
20	0.036
19	0.042
18	0.048
16	0.060
14	0.075
13	0.090
12	0.105
11	0.120
10	0.135
8	0.165
7	0.187

In the event that you have to make your own shims from 6 in. wide rolls, a good pair of tin snips can be used to cut the shims. Making a straight cut is not too difficult on shims up to 20 mils thick. It is difficult to do the cutting, if the thickness exceeds 20 mils thick. Cutting the slot where you make a hemispherical cut gets a little tricky particularly with shims over 15 mils thick. I usually use a permanent felt tip marker for the shim pattern and then start trimming away. I would recommend to wear a pair of leather gloves when cutting the shims. After the shim is cut, I usually have a block of wood and a ball-peen hammer nearby to flatten out the edges I cut. Once I have the shim made, I use the felt-tip marker to write down the thickness of the shim on both sides. If you are having to cut several shims, it is easy to forget who's who when you pick up the pile of shims.

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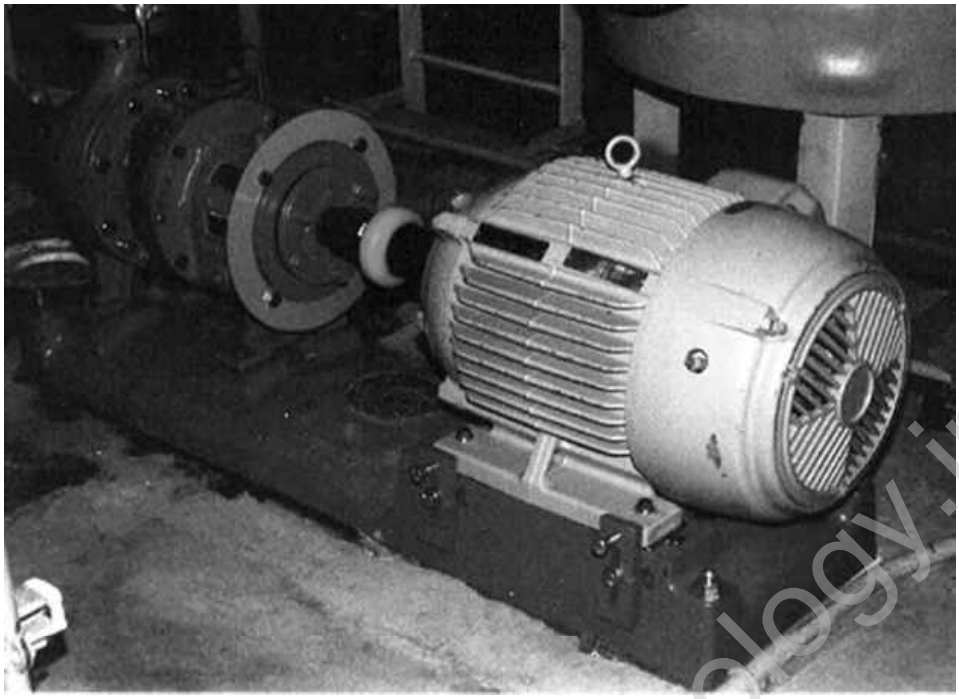


FIGURE 7.12 Removable jackscrew for moving a motor sideways.

As the slot is the toughest thing to cut, you may want to invest in a punch and die set to make a hole first, then you only have to cut straight lines. If you need to make shims over 25 mils thick, you should be using a hydraulic or mechanical shear for the overall shape of the shim and then drill the hole for the slot and use a band saw to make your cuts to the hole. If you have to make a whole bunch of shims, cut the overall shape, stack them together with the thinner shims in between thicker shims, sandwich the entire pack between two pieces of 1/4 in. plate steel, then use an end mill to cut the slot on all of them at the same time.

Some "rules of thumb" to follow for shimming machinery are:

1. Excluding the soft foot shims, you should not have more than five to six individual shims (or plates) under a foot to make an alignment correction shim "pack."
2. The total thickness of the alignment correction shim pack should not exceed half the diameter of the bolt holding that foot down.

7.4.3 Lateral Movement

Permanently mounted jackscrews or whatever devices may be used to slide equipment sideways should be placed as close as possible to the foot points without interfering with tightening or loosening of foundation bolts. A typical jackscrew arrangement is shown in Figure 7.12. Dial indicators mounted on the baseplate that are used to monitor sideways movement should usually be placed on the opposite side of the machine case from where the movement device (e.g., jackscrew) is located to keep the indicator from inadvertently bumping as shown in Figure 7.13. The disadvantage of permanently mounted jackscrews that have been welded to the baseplate or frame is that it will become very difficult to add or remove shim stock as the jackscrew plates are now in the way.

The fact of the matter is that the vast majority of rotating machinery do not have jackscrews installed for positioning the machinery. A useful trick in the field to achieve a lateral move is to use a corner foot bolt as a "pivot" point and move one end of a unit at a time when moving sideways as shown in Figure 7.14.

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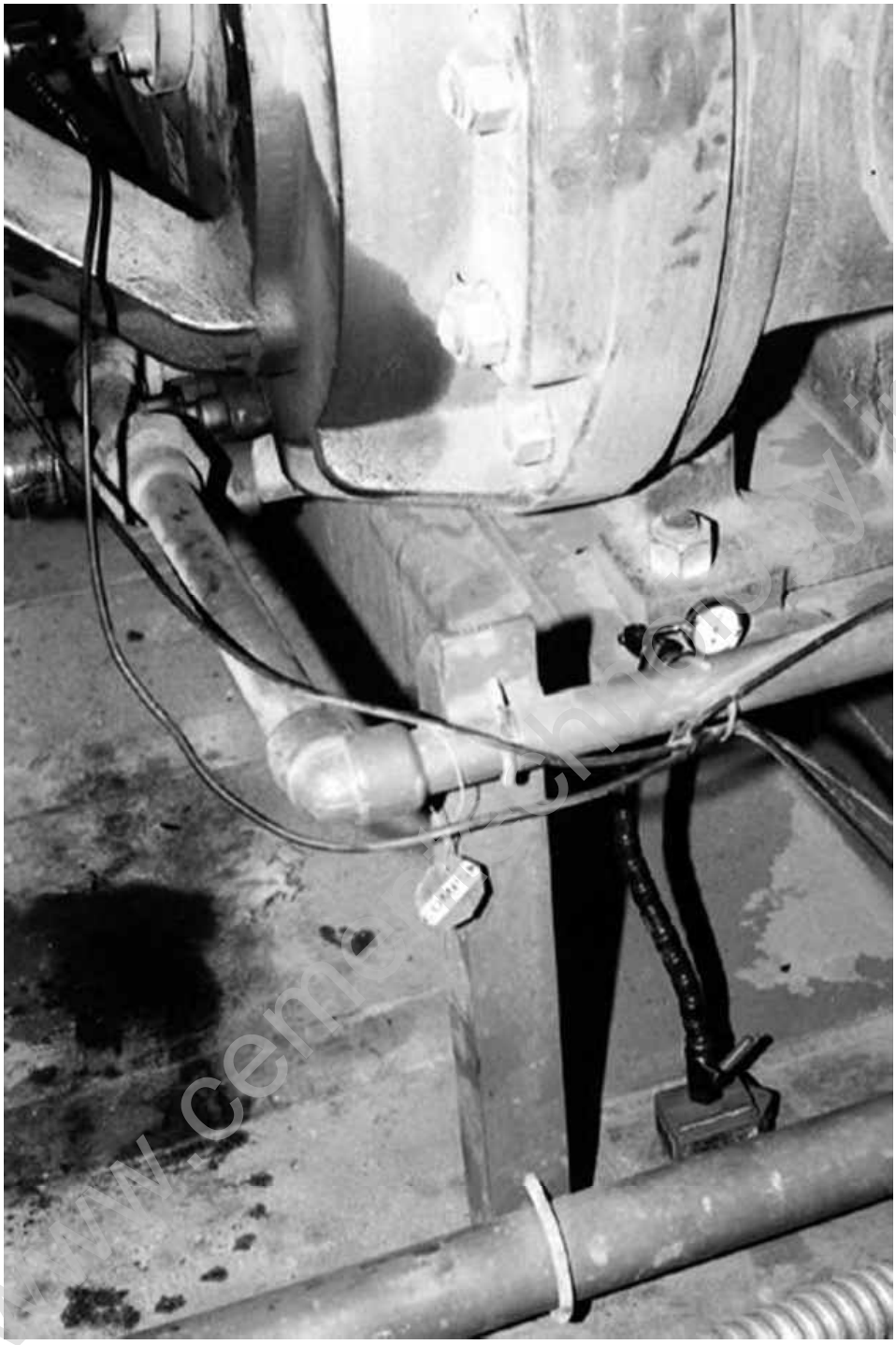


FIGURE 7.13 Monitor the move with an indicator.

Start by tightening one of the inboard bolts, usually on the same side as you will be pushing from. After the outboard end has been moved the desired amount, tighten one of the outboard bolts and loosen the inboard bolt that you used as the pivot point. Monitor the movement of the inboard end either by placing a dial indicator at the side of the machine casing at the inboard foot or by using a dial indicator and bracket arrangement attached to one shaft, zero the indicator on one side of the coupling hub then rotate the dial indicator and bracket arrangement 180° and note the reading as shown in Figure 7.15. Start moving the inboard end in the appropriate direction until the dial indicator on the coupling hub reads one half the original value. Zero the indicator again and rotate the dial indicator and shaft

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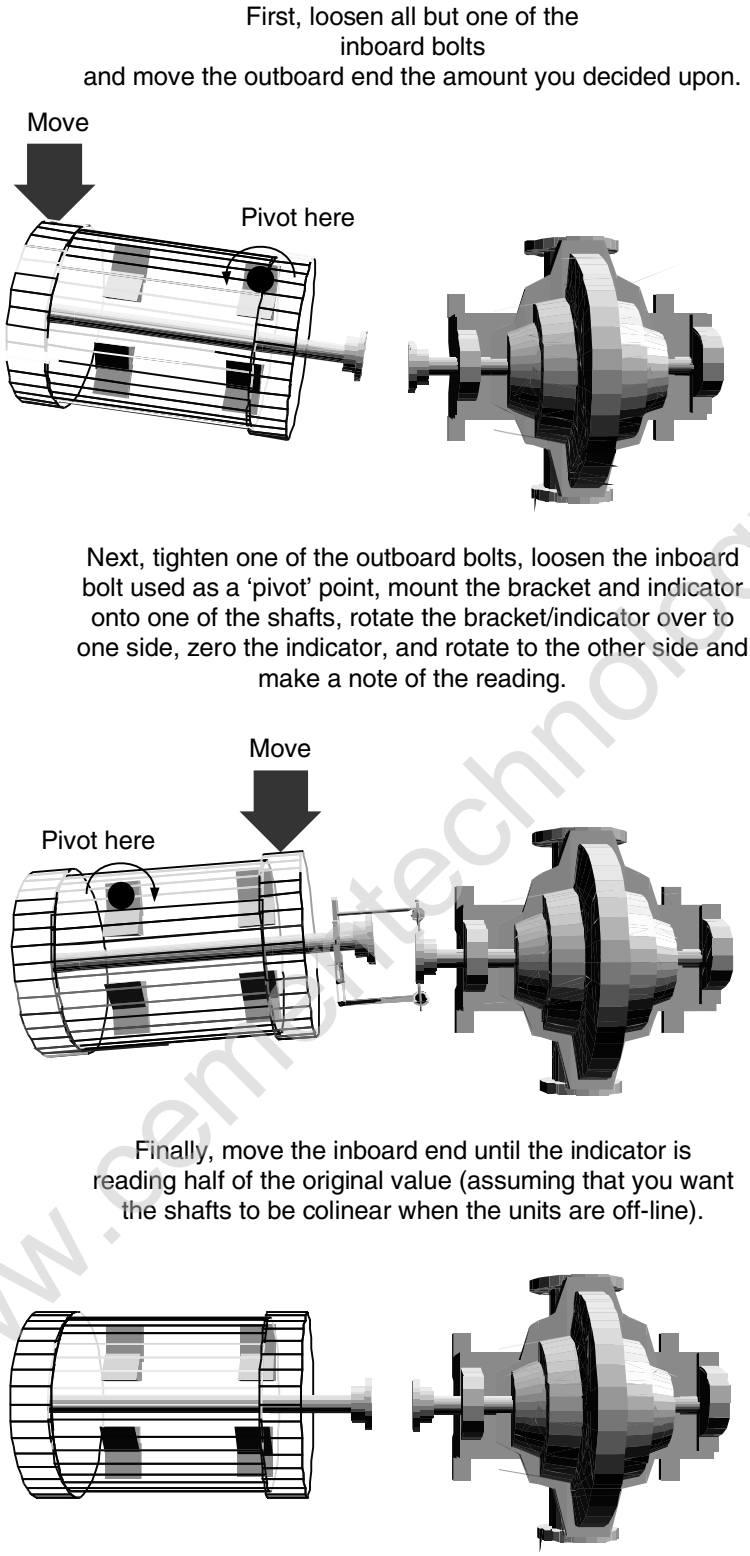


FIGURE 7.14 Pivoting at the outboard then inboard ends.

assembly back 180° to the original zeroing point on the other side of the coupling hub and check the reading on the indicator. This assumes that you do not want to have an intentional lateral offset due to movement from off-line to running(OL2R) conditions (refer to Chapter 16). If necessary, continue moving the inboard end to get the dial indicator to read zero when swinging from side to side on the coupling hub. It is also possible to move both ends simultaneously using indicators and jackscrews at each foot. For equipment with inboard

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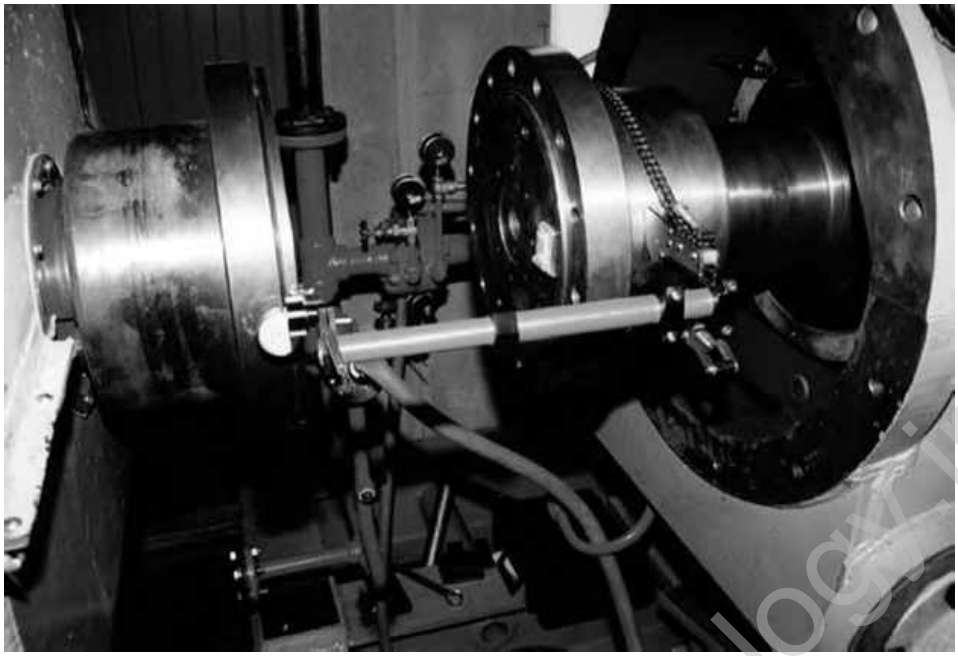


FIGURE 7.15 Monitor the move at the inboard end with a bracket and indicator mounted to the shafts.

to outboard foot distances of 3 ft or less, this seems to work all right with two people on the alignment job. Larger equipment usually requires four or more people to be effective.

On new installations it may be desirable to generate a total sideways movement "map." This will come in handy when calculating the necessary lateral moves to determine whether it is possible to move it as far as the calculation requires. An example of a typical allowable movement map is illustrated in Figure 7.16.

Once the map has been established, place each unit in the center of its allowable sideways and axial travel and begin to take your shaft alignment measurements (refer to Chapter 10 through Chapter 15). These movement maps will prove invaluable when aligning multiple element machine drive trains (refer to Chapter 17).

7.4.4 Vertical Movement

Lifting equipment is markedly more difficult than sliding it sideways so it is desirable to make the minimum number of moves necessary to achieve the correct vertical position. Lifting jackscrews are rarely found at the corners on rotating machinery casings.

If good lateral alignment has been achieved, try to keep as many foot bolts tight or have the jackscrews tightened against the machine element to prevent the unit from moving back out of alignment when shims are added or removed from the feet. Lifting equipment with a couple of foot bolts tightened can be a very delicate and challenging operation and must be performed with extreme caution. The idea is to lift the unit just far enough to slide shims in or out.

7.5 TYPES OF MOVEMENT TOOLS

Hammers are probably the most widely used tool for moving machinery sideways. Even if these are the least desirable method, there are preferred techniques when using hammers to move a unit sideways.

1. Use dead blow or soft faced hammers (plastic or rubber) instead of steel hammers.
2. If soft faced hammers are not available, place a piece of wood or plastic between the hammer and the impact point on the piece of machinery to prevent damaging the case.

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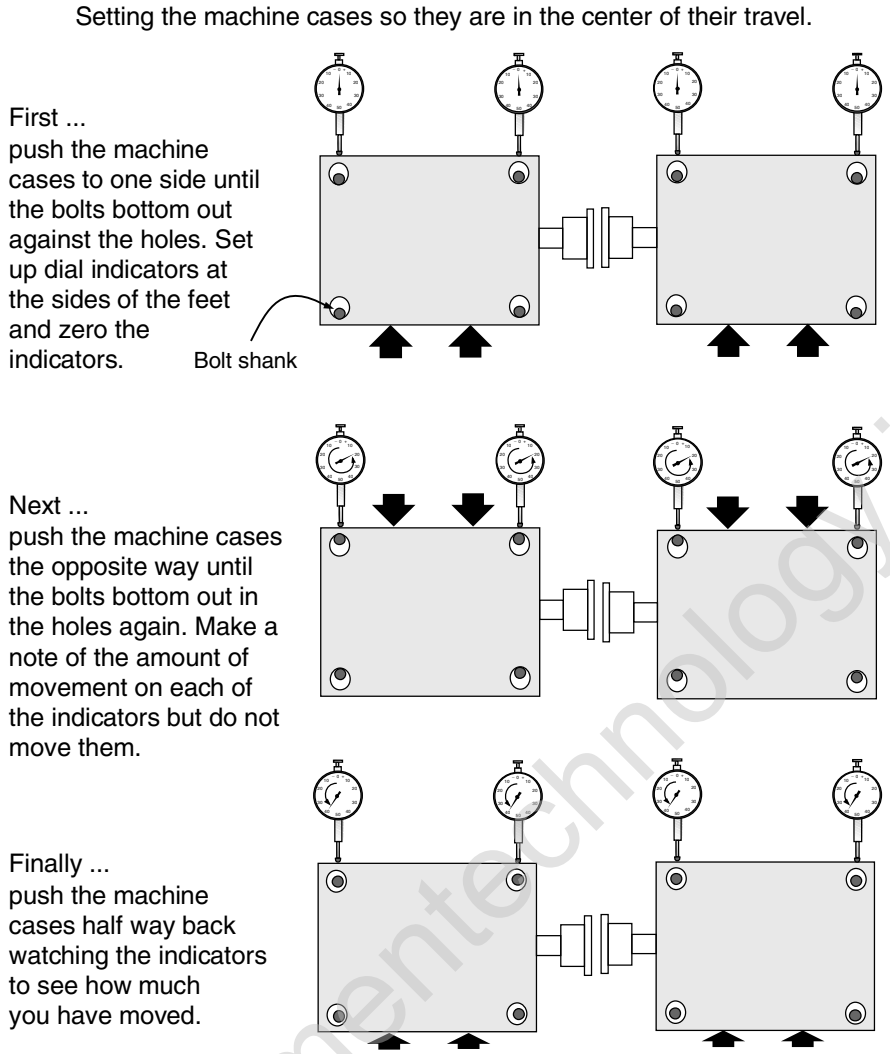


FIGURE 7.16 Placing the machine in the center of its sideways travel.

3. Take easy swings at first then begin increasing the force. With practice, you can develop a feel for how the unit moves at each impact. The more force that is used however, the greater chance there is to jolt the dial indicators that are monitoring the unit movement rendering the readings useless.

7.5.1 Pry and Crowbars and Wedges

Pry and crowbars can be found in every mechanic's toolbox and they invariably end up out at the alignment site just in case they are needed. Consequently, for smaller, light equipment, pry bars end up the most widely used device to lift equipment. A pry bar however, provides very little control in accurately lifting equipment and can slip from its position easily, which can be very painful for your partner who is trying to remove old shims from under the feet with his fingers.

A pry bar can also be used to move the equipment sideways assuming there is a leverage device near the feet. The leverage device, however, usually ends up being a piping, electrical conduit, or a long piece of 2×4 supported against something else on the machinery frame or foundation.

Using hammers and steel wedges to lift equipment is the least desirable method. If there is no alternative, here are some tips when using this technique:

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1. Place the wedge close to the foot that needs to be lifted without interfering with the process of adding or removing shims. The casing may distort enough to get the necessary shims in or out of that foot area without having to lift the entire unit.
2. Apply a thin film of grease or oil to both sides of the wedge.
3. It is fairly easy to install a wedge but it is quite another thing to get it out from under a heavy piece of machinery. Before installation, provide some means for removal of the wedge.

7.5.2 Comealongs and Chain Falls

These devices can be used to both lift and move equipment laterally. The primary problem with this equipment is usually the lack of proper rigging or anchor points for the chain falls or comealongs when moving sideways. There is also the problem of exceeding the capacity of the chain fall when rigged to lift the equipment. The better quality chain falls and comealongs however provide improved control and safety than do hammers and pry bars.

7.5.3 Hydraulic Jacks

There are many types of hydraulic jacks and kits that can readily be purchased at reasonable prices. When rigged properly, hydraulic jacks provide good control and safety when lifting or sliding equipment and are one of the preferred methods for moving rotating machinery. Figure 7.17 shows a custom-made portable hydraulic lifting and lateral positioning system.

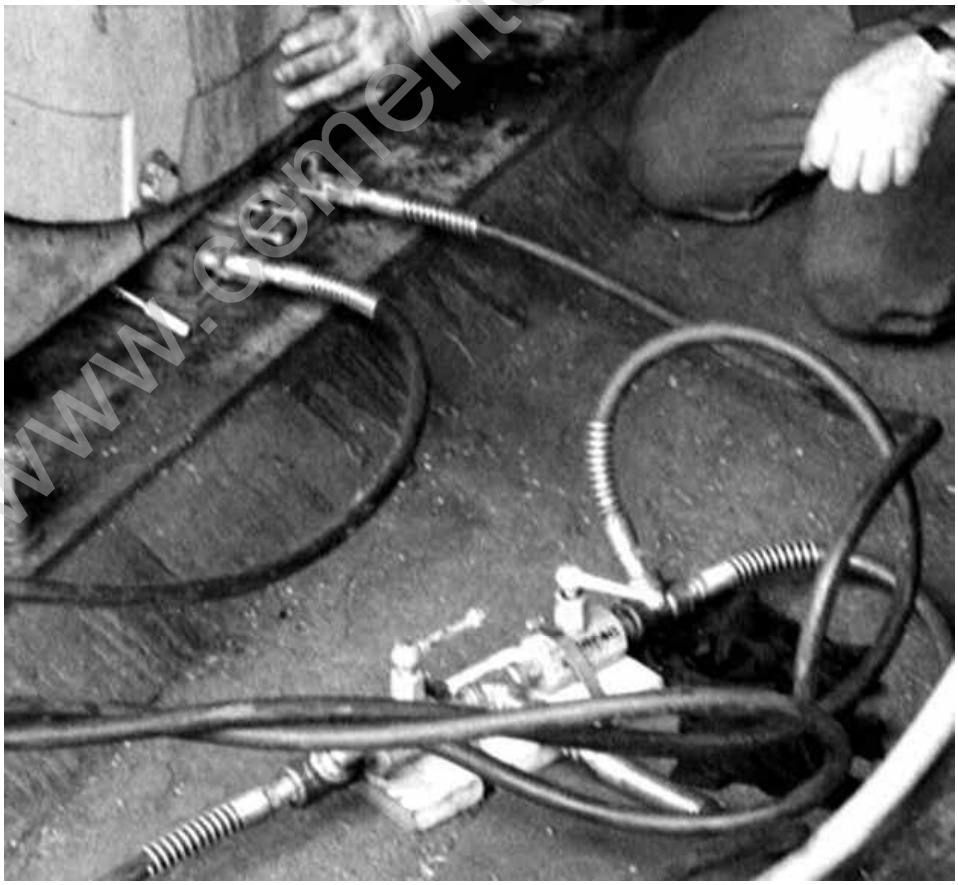


FIGURE 7.17 Hydraulic machinery lifting and lateral positioning device. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

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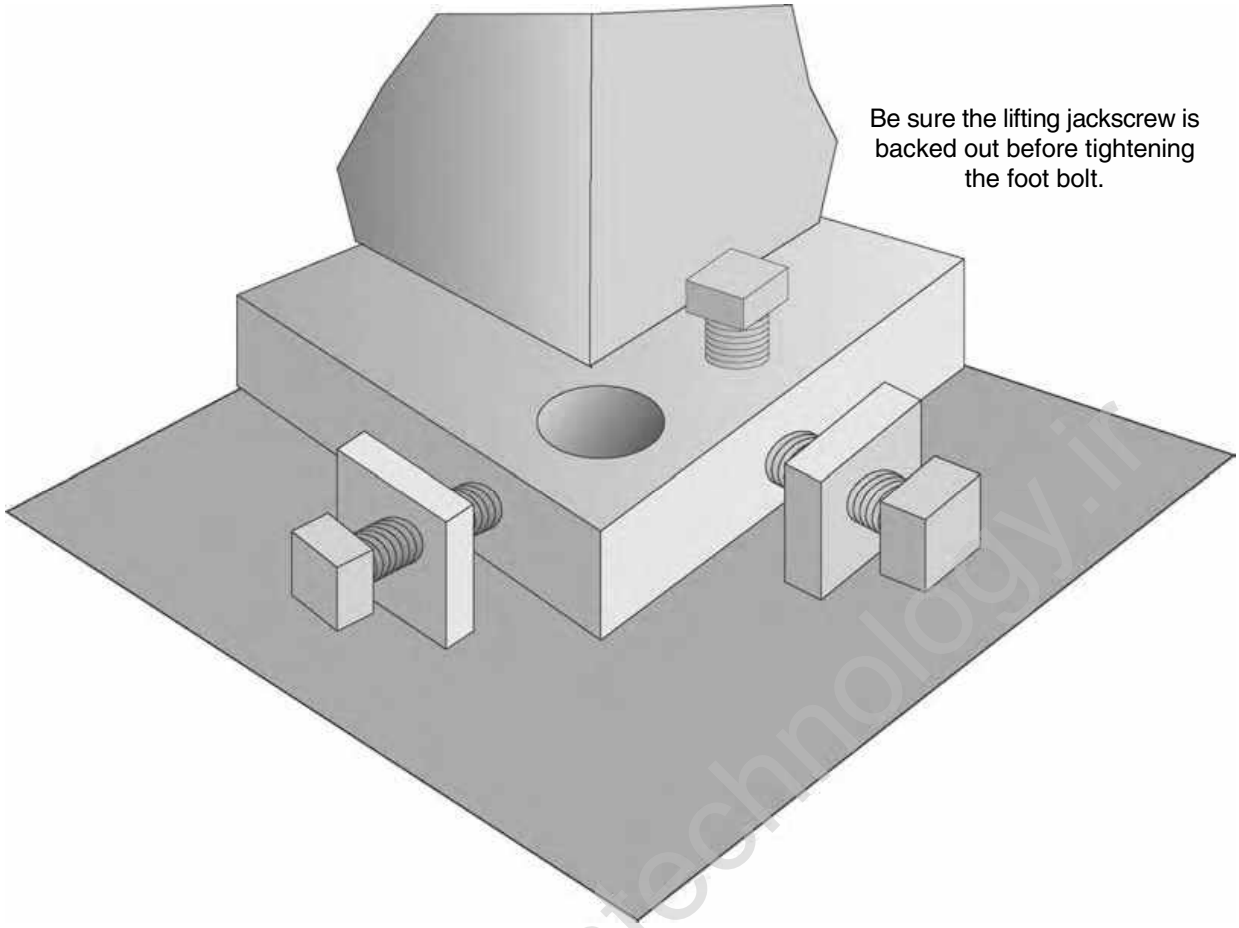


FIGURE 7.18 Permanent jackscrews for vertical, lateral, and axial movement.

7.5.4 Permanent Jackscrews

Although jackscrews are the most preferred method for moving machinery, they are not found frequently in industry mainly because of the cost and effort required to install them. A typical jackscrew arrangement is illustrated in Figure 7.18.

7.5.5 Portable Jackscrews and Machinery Positioners

A considerable amount of imagination has gone into designing these clever devices and could be used in one form or another by most of the industry for machinery alignment applications. If you have more than one of the same type of pump, motor, compressor, etc. it is recommended that drop-in puller devices such as shown in Figure 7.19 through Figure 7.24 be used for your specific application. Portable jack bolt kits are also available as shown in Figure 7.25 through Figure 7.27.

Another device that can be placed between the baseplate/soleplate and machine casing which has the capacity to adjust for nonparallel surface contact with the added feature of height adjustment are machinery positioners shown in Figure 7.28 and Figure 7.29.

7.6 WHAT TO DO WHEN THINGS ARE NOT WORKING

When you are aligning a piece of machinery and everything you try does not seem to be working and in some cases, the alignment seems to be getting worse rather than better. Or perhaps you have improved gross amounts of misalignment but just cannot seem to bring the

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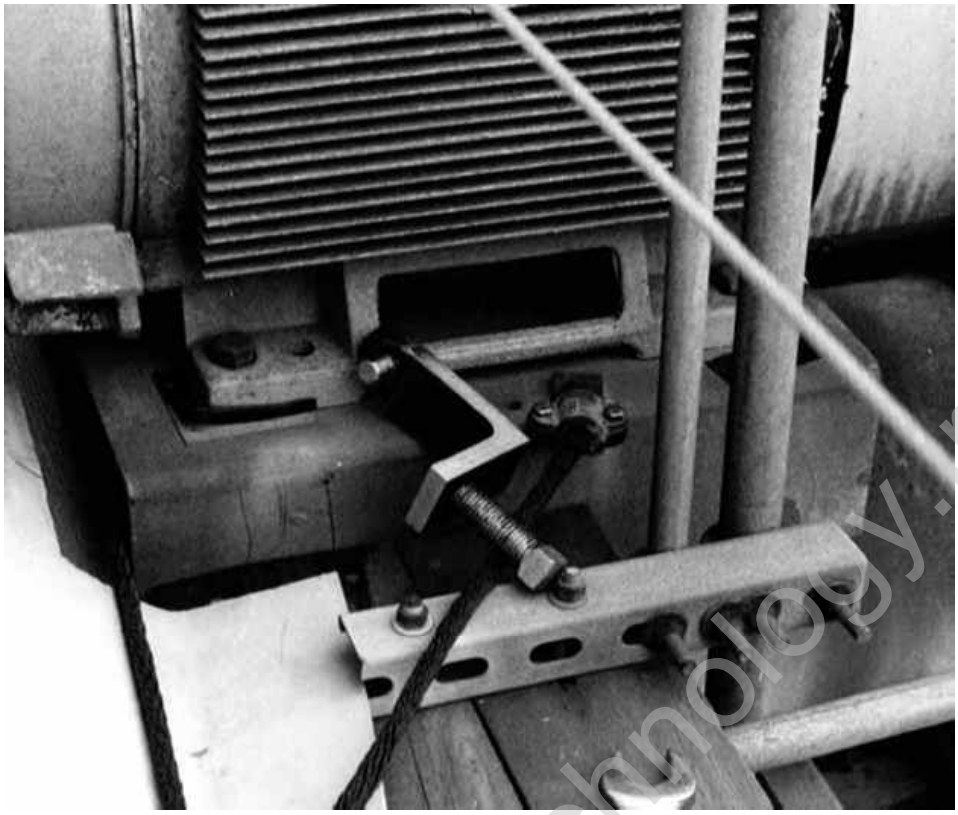


FIGURE 7.19 Drop-in puller device ready to be inserted into bolt hole on machine case.

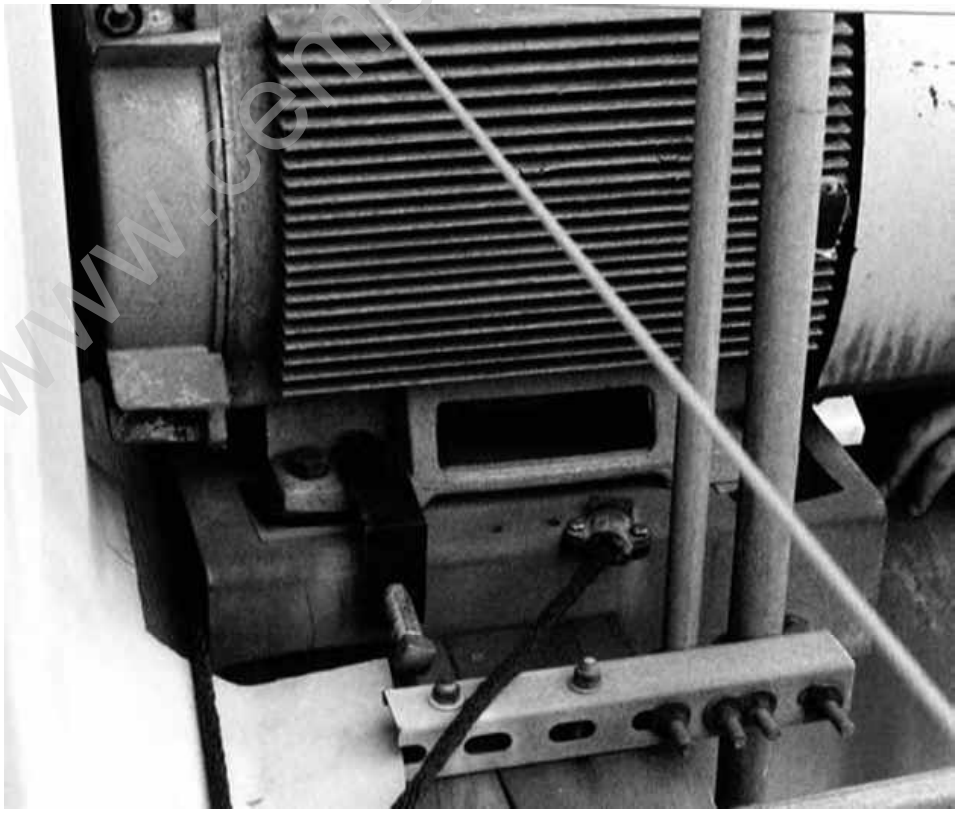


FIGURE 7.20 Drop-in puller device inserted into hole and ready to pull machine sideways.

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FIGURE 7.21 Drop-in puller device ready to be inserted into bolt hole on machine case.

units within 1 mil per inch or better. You keep taking 5–10 mils of shim stock out and then putting it back under the same bolting plane you just removed it from or the machinery seems to keep wanting to move out of alignment sideways when you tighten the bolts down. After muttering obscenities to the equipment and tools you are working with, you look up and notice someone walking by looking at you as if you are deranged. What should you do?

The "It's not working" Alignment Troubleshooting List

1. Stop, and mosey off the job site for a few minutes. Go back to your break area and relax. First of all, you have to realize there is something that you have overlooked. It might be

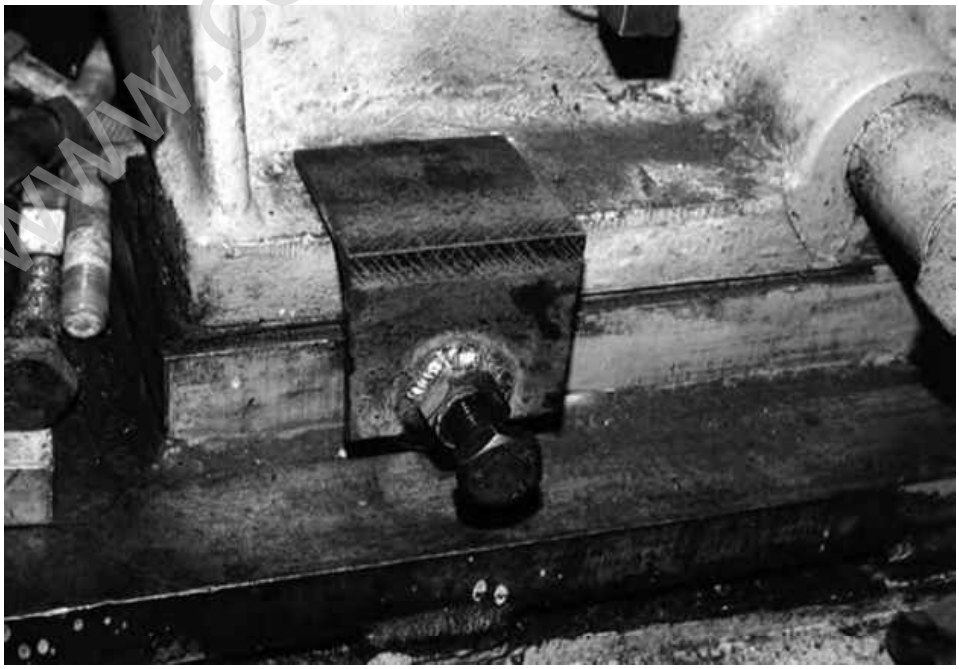


FIGURE 7.22 Drop-in puller device inserted into hole and ready to pull machine sideways.

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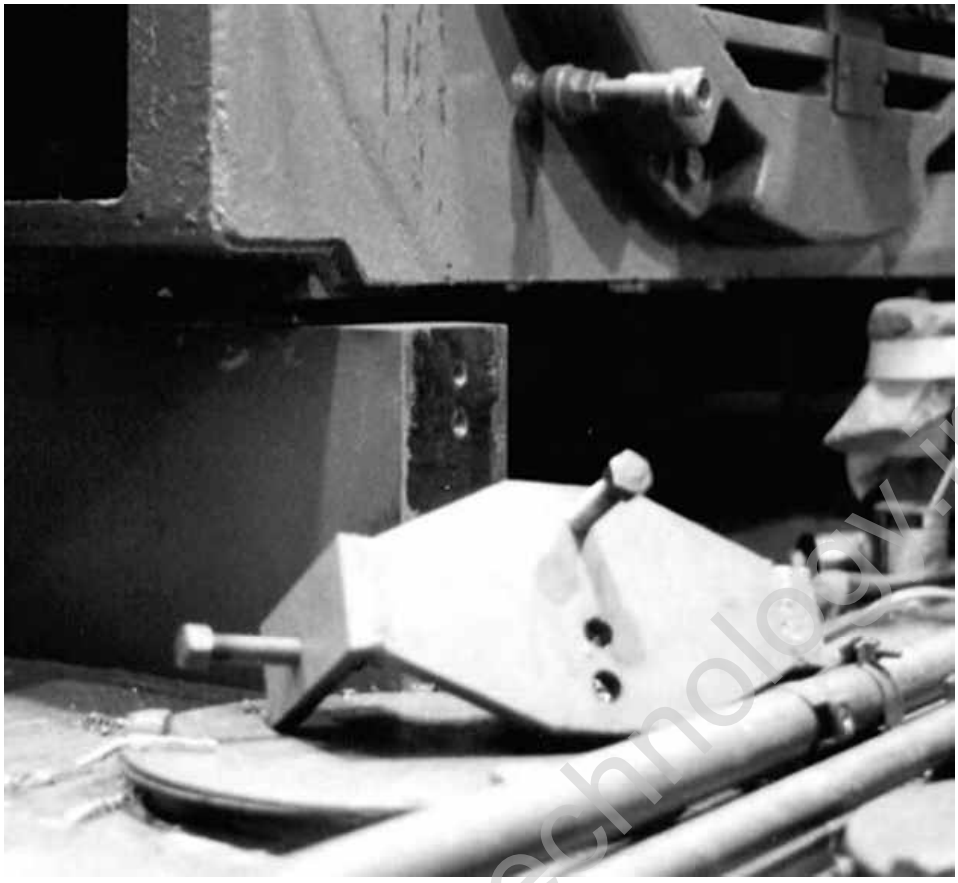


FIGURE 7.23 Bolt-on pusher/puller device ready to be inserted into drilled and tapped holes in frame rail.



FIGURE 7.24 Bolt-on pusher/puller device bolted to frame rail and ready to push or pull in the sideways direction or push in the axial direction.

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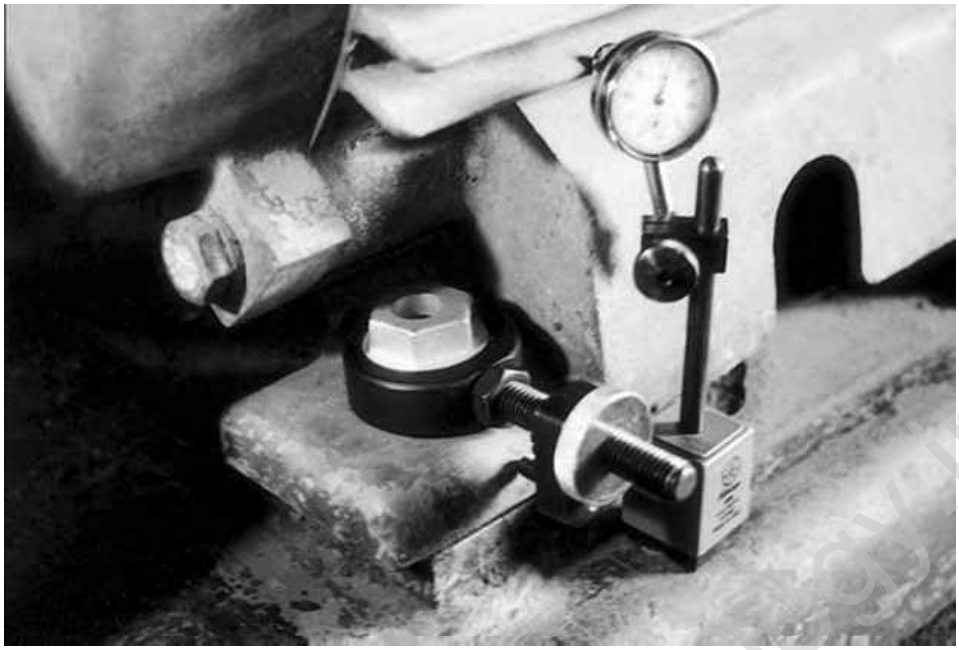


FIGURE 7.25 Portable jack bolt attached to machine case. (Courtesy of Posi Lock Puller, Inc., Cooperstown, ND. With permission.)



FIGURE 7.26 Portable jack bolt kit for bolt sizes 3/8 to 3/4 in. (Courtesy of Posi Lock Puller, Inc., Cooperstown, ND. With permission.)

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FIGURE 7.27 Portable jack bolt kit for bolt sizes 3/8 to 1 1/4 in. (Courtesy of Posi Lock Puller, Inc., Cooperstown, ND. With permission.)

staring you right in the face and you cannot see it through the fog or it could be one or several minor things acting together to cause a larger problem. As you go through these next steps, do not lie or pretend you did these, really go back and check each and every one again (or maybe for the first time). Remember, you might find more than one thing wrong so do not stop if you happen to discover something half way through this list.

2. Go back to Chapter 1 and read Section 1.4 on overall alignment job. You can skip step1 and start concentrating on what is being said from step 2 on. Do not just read the words like you are reading a novel. Stop after each sentence and understand what each



FIGURE 7.28 Standard profile Vibracon machinery positioner. (Courtesy of Machine Support Inc., Virginia Beach, VA. With permission.)

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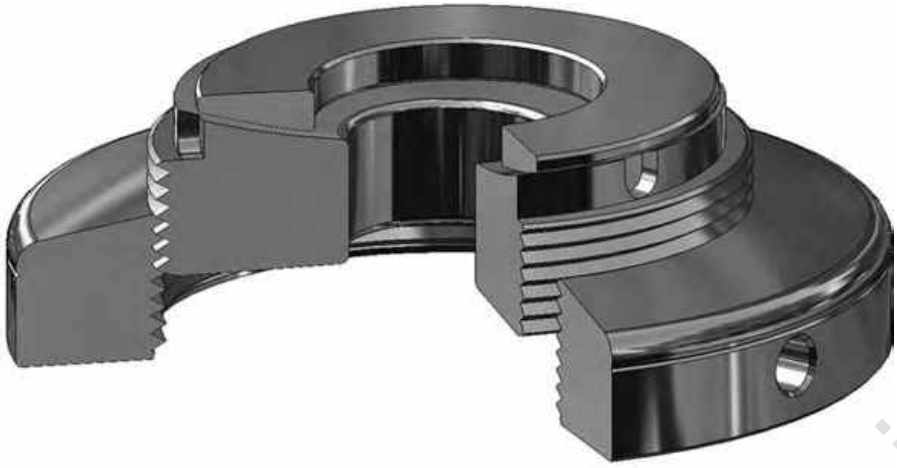


FIGURE 7.29 Low profile Vibracon machinery positioner. (Courtesy of Machine Support Inc., Virginia Beach, VA. With permission.)

sentence means. If you read a sentence and have not done what was suggested, stop, mark your place, and do it now. If you do not understand what is being asked, take a few moments and find where that piece of information is in this book and read through it. After you have completed the missed item, pick up where you left off.

3. Pay particular attention to step 4 (the preliminary checks). Over half of the time someone is having problems aligning the machinery, it is due to something in this step. Again, without trying to be excessively repetitive, do not pretend you did these. Even if you did, go back and check them again. If you find that there is an excessive amount of runout, or soft foot, or sloppy bearing fits, or warped baseplates, or deteriorating foundations, or excessive piping strain, fix the problem. I fully understand what is involved in the simple sounding statement "fix the problem." Some of these might be big, time consuming, expensive, pain-in-the-neck problems to solve. I also understand that in many cases you are going to have to get other people involved and when you explain the problems that you have found, there might be (will probably be) some (a lot of) hesitation on their (and your) part to correct what has been found. It is your choice. If you find something here, and decide not to do anything about it, you can try to go on and do the best job you can but do not expect great results and do not blame your alignment measurement system, your wrenches, the equipment manufacturer, your coupling manufacturer, the alignment graphing or modeling technique, this book, or your barber for what is going to happen. You and people who decided not to do anything should walk up to a mirror, stare straight ahead, and start blaming the person you see in front of you.
4. Whew! Did you get this far yet? OK, you should be feeling a little better by now (assuming you feel good about the previous item). Now, take a good look at your shaft alignment measurement system. If you are using dial indicators, remove them and with your finger, gently push the stem in trying to feel for any rough spots or sticky spots in the stem travel. If you are using a laser system or optical encoder system (or any system with electronic sensors) check if all of the cables are good and that they are making good connections to and from the sensors. Check the batteries. Take a good look at your shaft clamping brackets. Are they firmly affixed to the shafts? Is the base of the bracket sitting in a keyway? Is the span bar tube clamped firmly to the bracket on

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dial indicator type systems? Is the laser and detector housing clamped firmly to the bracket-posts? You may want to remove the system from the machinery . . . get a 20–30 in. long piece of Schedule 40 pipe . . . set the pipe on some "V" blocks or pipe roller stands (or in a vise)—clamp the brackets, indicators/sensors to the fixtures—zero the indicators on top and roll the system to the bottom and watch what happens. On dial indicator systems, does the bracket sag stay the same if you do this several times? On laser systems do the readouts stay at zero all the way around? Now put the alignment measurement system back on the machinery shafts. Check to make sure the foot bolts are all tight and then take several sets of readings. Are the reading sets pretty much the same each time? Are you quite sure you are reading the dial indicator correctly? If there is any doubt, ask someone you believe knows what he or she is doing and have them take a set of readings (without your help) and see what he or she gets. Do the two sets of readings agree? Loosen the shaft brackets, rotate the entire shaft alignment measurement systems 120° around the shaft and take another set of readings. Do the two sets of readings agree? If the flexible coupling is bolted in place, take a set of readings, loosen the coupling bolts up and take another set of readings. Are they the same? If the machine is moderately to severely misaligned, they probably will not be the same. Now why is that? Take a good look at Figure 1.2. When rotating machinery shafts are connected together, even with a flexible coupling, the shafts begin to bend elastically due to the misalignment condition. Now take a good look at where the shaft alignment measurement system is going to be set up on the shafts. Notice that the brackets are going to be attached to shafts that are undergoing bending stresses. If the coupling was not connected, these bending stresses would not be present and the shafts would not be in the positions shown in Figure 1.3, but the shafts would be in a position concentric to the centerline of the bearings that support each shaft. This phenomenon is one of the major reasons why you "chase your tail" when using alignment measurement systems that require you to keep the coupling in place when taking measurements. Disconnecting or loosening coupling bolts is especially important with smaller machines, and with non-spacer couplings. For larger machines with spacer couplings, it is often unnecessary. Look, the only way you are going to be able to determine what moves need to be made to correct the misalignment conditions is to make sure you are getting an accurate set of measurements on where the centerlines of rotation are going to be when the shafts are in an "unstressed" state of affairs. If you work with garbage measurements, you are going to get garbage results.

5. OK, by now you should be sure you are getting really good measurement information on the positions of the machinery shafts when you are off-line. If you are using one of the graphing or modeling methods explained in Chapter 8, go back and check to see if you are graphing the measurements correctly. If there is any doubt, ask someone you believe knows what he or she is doing and have them model the information (again, without your help please) and see what he or she gets. Do the two sets of graphs compare? If you are using a computer of some kind, take the raw readings from the measurement system, and graph the data. Does this compare to what the software program is telling you? If you are using a laser system, remove the laser system, set up some brackets and take some dial indicator readings, graph–model the readings and compare these to what the laser is telling you. Do the results match?
6. You still feel really good about everything so far? Because now comes the ultimate test. Are the machines moving exactly where you want them to go? OK, you said the inboard end of the pump had to be lifted up 27 mils. Did you measure the shim pack thickness with a micrometer before you installed it? Did you install them under the right machine? Did you put them under the right bolt set? Did you put them under both of the inboard

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bolts? Are you pushing shims out that are currently under the feet when you install the new ones? Are you disrupting your soft foot corrections when you install the new shims? By the way, if you just asked what "soft foot" means I will wring your neck if I find you. Did you tighten the bolts back down after the shims were installed? When you try to move the machinery sideways, is the machine case shifting sideways without seeing it when you loosen the foot bolts? Did you set up some dial indicators on the sides of the machine case at the inboard and outboard ends to monitor for this movement? OK, you said the outboard end of the steam turbine had to be moved eastward 86 mils. Did you see if that end could be moved that much before you started beating on it with a hammer? Did you move the outboard end of the pump rather than the outboard end of the steam turbine? Did you move the outboard end to the east or did you really move it to the west? When you tighten the bolts back down, is the machine staying where you located it sideways? How do you know that? Did you set up some dial indicators on the sides of the machine case at the inboard and outboard ends to monitor for this movement? Are you sliding shims out that are currently under the feet when you move the machine sideways?

7. Take another set of readings after each move that you make. If you are using a dial indicator system, are you "spinning zeros" all the way around? You better not be! Unless of course you do not have any bracket sag and you actually want the two shaft centerlines to be exactly in line with each other. If there is some OL2R movement, did you back calculate what the "shoot for" readings were supposed to be? Did you graph the new positions of the shafts and are they in the "desired off-line positions"? Did you determine if you are within acceptable alignment tolerances? Is the shaft to shaft distance correct? Are you trying to stick 1 mil of shims under the feet to get a perfect alignment? If so, button everything up, come back after you ran the machinery for 24 h and see if the shafts are in the exact same position that you left them.
8. Did you run the machinery and measure bearing temperatures, vibration levels, and other pertinent operating data? Did the vibration go up and someone is accusing you of doing a poor alignment job? If so, have them read Chapter 1 and Chapter 2.

7.7 MISALIGNMENT RANTINGS

I frequently wonder how far we have really advanced in the area of machinery alignment.

In the previous chapters we have seen the various types of devices used to measure shaft alignment, covered graphical modeling techniques used to determine which way and how much machinery casings should be moved, and reviewed a host of methods used to measure OL2R machinery movement. A tremendous amount of effort on the part of hundreds of people over a span of decades has been put into the measurement methodology of shaft alignment. Millions of dollars have been invested in developing these three core aspects of alignment—mechanical measurement devices, calculating the required moves on the machinery, and determining how machinery will move from OL2R conditions. Development in these three areas continues at a feverish pace. But is this where the majority of our efforts should be spent in the future?

Take a moment and refer back to Figure 1.5 showing the amount of time typically spent on the overall alignment job. Notice that there are seven basic tasks performed in the alignment process.

1. Safety tag and lockout
2. Preliminary checks
3. Measure shaft positions

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4. Calculations
5. Decide moves
6. Reposition machinery
7. Run and check

If you eliminate the last step (run & check) and examine the amount of time taken to perform each of the other six tasks as illustrated in Figure 11.1, you easily can see that the vast majority of time in virtually every alignment job is spent on just two of the tasks—the preliminary checks and repositioning the machinery.

Something is critically wrong here! The vast majority of research and development work in shaft alignment is spent trying to figure out how to do the following three tasks better and faster:

1. Measure shaft positions
2. Calculations
3. Decide moves

Why are we spending so much time, effort, and money on something that only occupies less than 14% of the overall time performing an alignment job? What difference is it going to make in the overall scheme of things if we reduce the amount of time to do these three tasks (measure shaft positions, calculations, decide moves) from 15 to 5 min if it takes us somewhere between 2 and 4 h to reposition the machinery?

One of the most bizarre sights I have witnessed in industry is to watch someone have an expensive laser shaft alignment measurement system attached to machinery shafts and stand next to the machines with a sledge hammer beating the equipment sideways to get it into alignment.

> Question: What is the number one tool used in industry to lift a piece of machinery to install shims under the feet?
> Answer: A screwdriver or pry bar.

Sounds like a joke. Doesn't it? In a very real sense, it is a joke that isn't very funny. People continue to use wood boards, crowbars, fork lift trucks, chain falls, mallets, pipes, conduit, cranes, and wedges to lift machinery. The lever was first explained by Aristotle in 400 BC? Well my fellow readers, that is about as far as we have progressed.

We have been using flat spacers (e.g., shim stock) to vertically reposition machinery as the Dark Ages (and probably before). There have been no significant advancements made in how to effectively reposition machinery for 50 centuries! Yet today we are using optical encoders, lasers/photodiodes, proximity probes, charge-coupled devices and computers to measure and calculate machinery moves. Of the 140 worldwide patents researched for this book only three patents have been filed that focus on the aspect of repositioning machinery. Something is seriously wrong with this narrow-minded portraiture of our "achievements."

All of the measuring tools and all of the calculators and computers in the world will not move, reposition, and align one single piece of machinery in existence! Dial indicators, laser, encoders and the full gamut of measuring tools do not align machinery, people do!

Judging the quality of an alignment measurement tool by how minute a measurement of a shafts position can be made is fruitless. What good will it do you to have an alignment measurement system that can measure to one-hundred thousandths of an inch (0.00001 in.) if you cannot purchase shim stock that thin?

The true "art" of shaft alignment pivots on your ability to control what you want to happen to the machine casings to bring them into an aligned state. You might be able

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to accurately measure shaft positions within one-thousandth of an inch, but can you control the position of the machinery to that level of precision? If you cannot, all of your valiant efforts during the measurement process are for naught because you cannot achieve the true final goal of properly positioning the machinery to achieve acceptable alignment tolerances.

Accurately repositioning machinery is the ultimate test of your expertise in shaft alignment and it will be your most challenging one. Hard work, sweat, dirty, and sometimes bloody hands, perseverance, discipline, patience, and practice are the core elements of success here.

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8 Alignment Modeling Basics

So far, we have reviewed the nondestructive techniques to find misalignment, studied good baseplate and foundation design principles, looked at the multitude of flexible and rigid couplings, reviewed the critical preliminary steps that should be performed prior to an alignment job, covered a variety of measuring tools that are used in the alignment process, and discussed how to position machinery to correct a misalignment condition. Sounds like we are just about done with the entire subject at this point, but exactly how do we actually measure the positions of the shafts? Once we get the shaft-to-shaft measurements, how do we decide which way and how much of a move is required on the machinery to correct the misalignment in the most efficient manner?

Before we begin discussing the variety of ways to measure and correct misalignment on rotating equipment, this chapter will cover the basic concept behind a methodology called alignment modeling, also known as alignment graphing. Chapter 10 through Chapter 15 will show how to construct these models using the five basic alignment measurement methods. For now, let us focus on the idea of visually representing only the centerlines of rotation of the shafts in their misalignment condition, so we can make effective corrections to the misalignment problems we face.

In the purest definition of shaft alignment, the goal is to find and then align the centerlines of rotation of two (or more) machinery shafts with each other. As far as shaft alignment is concerned, the entire machine can be stripped away, and its rotating shaft can be represented as a straight line similar to what is shown in Figure 8.1.

The alignment model is an accurately scaled, yet exaggerated picture of the misalignment condition of machinery shafts. Additional information can be superimposed on the alignment model to assist in representing movement restrictions, internal clearances, natural elastic bending, and piping or ductwork connection positions. By far, alignment modeling is the most effective way to represent your misalignment condition.

Once the alignment models are constructed, you then select a desired position for the shafts to be in, which is referred to as the ''overlay line'' or ''final desired alignment line.'' Finally you determine the amount and the direction of the corrective moves from the actual positions of the centerlines to the overlay line directly from the alignment model.

The fact that the model is a scaled visual representation of the shaft positions gives you the opportunity to accurately determine a myriad of possible movement solutions to correct the misalignment condition. Alignment modeling methods and the overlay line concept have been in use since the late 1970s, yet the vast majority of people responsible for aligning machinery have never seen or used these techniques.

You may exclaim that you do not need an infinite number of solutions. You just want one that works.

I hear you loud and clear. If you are already aware that there can be an infinite number of possible alignment solutions to any alignment problem, then you may have also realized that a lot of them are going to be difficult, time consuming, and expensive to do. But there are also

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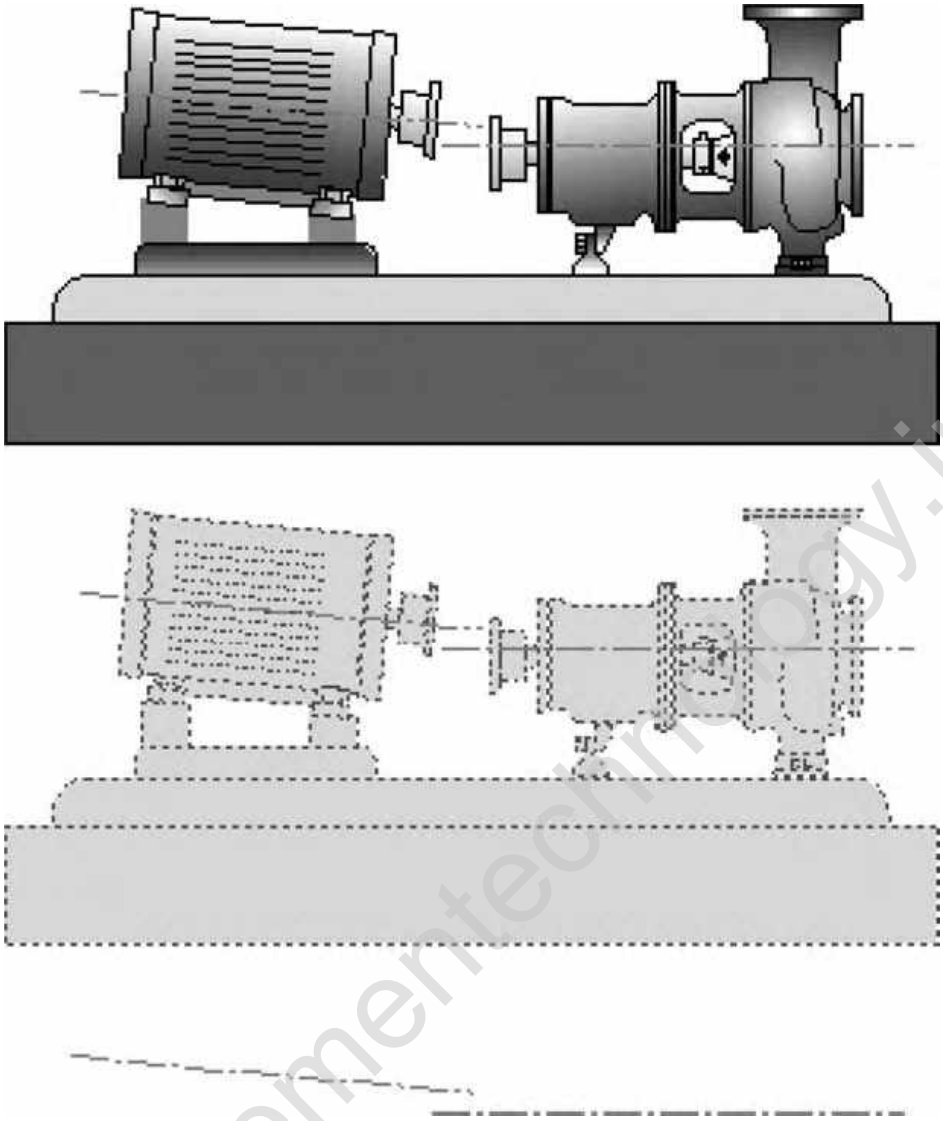


FIGURE 8.1 Representing the centerlines of rotation of machinery shafts as straight lines.

many more that are logical, easy, and inexpensive to do. If your alignment system limits you to a few choices, there is a possibility that you will eventually run into trouble. Honestly, if you do enough alignment jobs, I can virtually guarantee, you will run into a problem with a limited number of solutions. To my knowledge, there is no more effective way to correct misalignment than the methodology explained in this chapter.

Still today, some people who align rotating equipment will do trial-and-error alignment. They install some shim stock under the feet and move the machinery sideways a little bit, take another set of readings, and see if the measurements got any better. This sophisticated technique called guessing will eventually produce frustration, anxiety, and anger if continued for long periods of time. To a certain extent I applaud their effort. At least they made an attempt to improve the misalignment condition; many others do not even try. There happens to be a much better way to determine how to accurately position the machinery instead of guessing. And there happens to be a better way than having a limited computer software program telling you what to do, particularly if there could be a simpler way to solve the misalignment condition.

Even for people who align rotating machinery on a regular basis, it is very difficult to visualize exactly where the centerlines of rotation are by just looking at dial indicator, laser, or optical encoder measurement data. Your goal is to position each machine so that both shafts

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run in the same axis of rotation and you invariably begin to wonder—Is one shaft higher or lower than the other one, is it to the west or is it to the east, and if so, how much?

Alignment models can be as simple or as complex as the drive system itself. If you are trying to align two pieces of machinery such as a motor and a pump, the alignment model can be constructed to show both of those shafts. If you are trying to align an eight-element drive system with a right angle in the drive, the alignment model can be constructed to show every one of the shafts including the right angle turn the drive system makes. This chapter is intended to introduce you to modeling a two-element drive system. More complex drive systems are covered in Chapter 16 and Chapter 17.

8.1 GRAPHING AND MODELING ALIGNMENT TECHNIQUES

Regardless of the device used to measure the positions of the centerlines of rotation (be it dial indicators, optical encoders, lasers, and the like), virtually every alignment measurement system utilizes one (or a slight variation) of the following measurement approaches:

1. Reverse indicator method (Chapter 10)
2. Face and rim method (Chapter 11)
3. Double radial method (Chapter 12)
4. Shaft to coupling spool method (Chapter 13)
5. Face–face method (Chapter 14)

To understand how each of these techniques work, dial indicator readings will be used to illustrate how each method can be graphed or modeled to determine the relative positions of each shaft. All of these techniques can be graphed or modeled by hand. Typically all you need is some graph paper (20 division/in. is a good choice), a straightedge, and a pencil (with an eraser just in case). You do not even really need graph paper; all that is required is a scaled grid or some sort of measurement device like a ruler, but graph paper helps.

8.2 BASIC ALIGNMENT MODELS

The graphical shaft alignment modeling techniques use two different scaling factors. One scaling factor proportions the overall dimensions of the machinery drive system to fit within the boundaries of the graph paper and another different scaling factor is used to exaggerate the misalignment between the machinery shafts.

If we limit our discussion to horizontally mounted rotating machinery drive trains for now, there will be two graphs that need to be drawn. As depicted in Figure 8.2, one graph will show the exaggerated positions of each shaft in the side view illustrating the up and down or vertical positions of the machinery. Another graph will be constructed in the top view that will illustrate the side-to-side or lateral positions of the machinery. Figure 8.3 shows a three-dimensional view of the drive system misalignment.

Keep in mind that the shaft centerline positions shown in the side and the top views are exaggerated to help visualize the misalignment condition. Once the relative positions of the machinery shafts are constructed on the graph, a wide variety of different solutions can be determined to bring the centerlines of rotation in line with each other. The benefit of modeling rotating machinery is to visually represent an exaggerated, but accurately scaled picture of the misalignment condition so you can easily ascertain what positions the machinery could be moved that would make it easy to align the shafts within the boundary conditions imposed by the baseplate or foundation and the allowable lateral restrictions between the machinery casing bolts and the holes drilled in the machine cases (a.k.a. ''bolt bound'' conditions).

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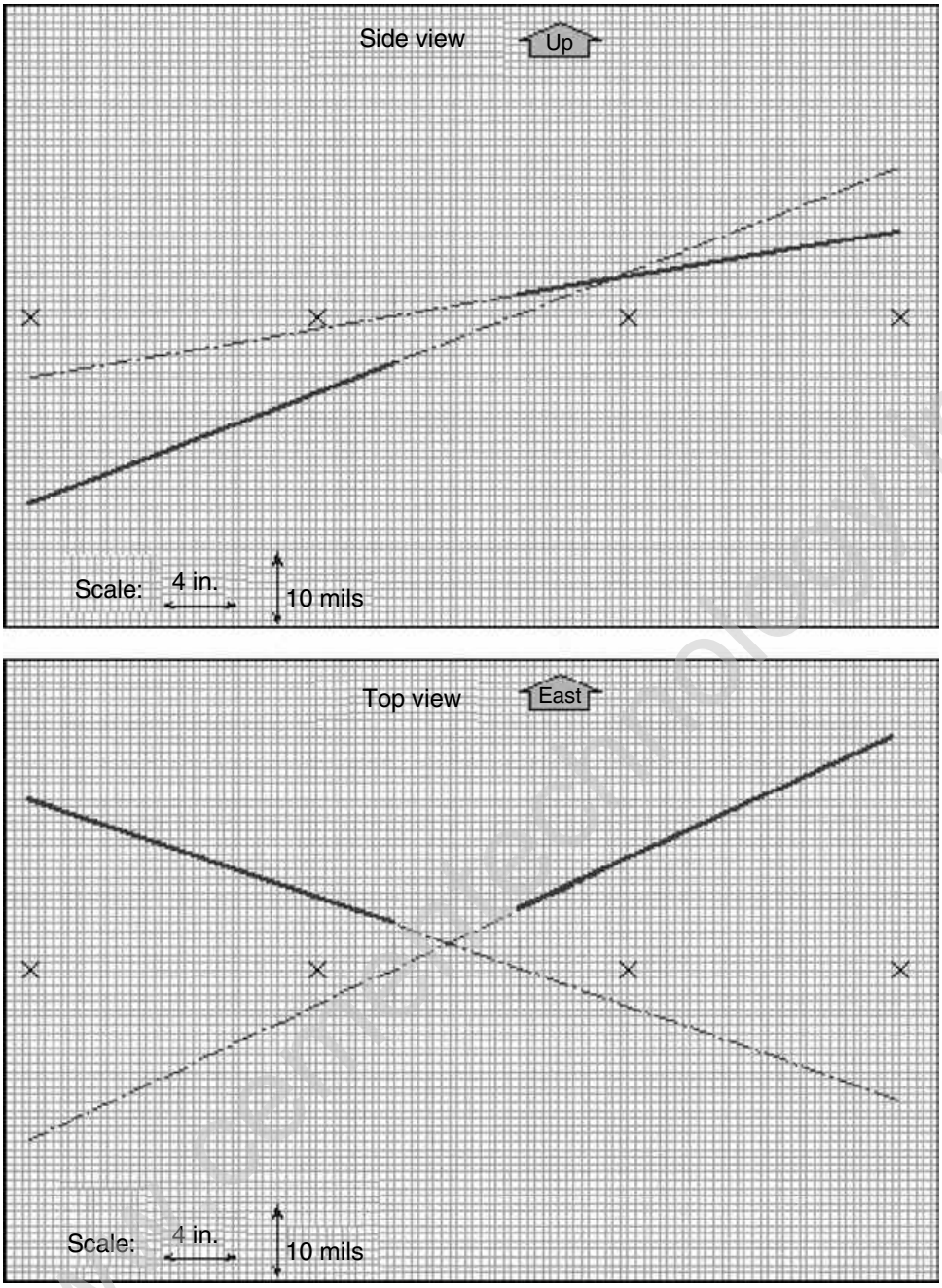


FIGURE 8.2 Side and top view alignment models showing an exaggerated misalignment condition between the two shafts in the vertical direction (side view) and in the lateral direction (top view).

Additionally the modeling technique can include other measurement parameters such as improperly fit piping, air gap clearances between stators and armatures, and fan rotor to shroud clearances, for example. Finally, the graph is a permanent record of the alignment of the machinery and can be kept for future reference.

In summary, this chapter will review the following key steps in correcting the misalignment situation (refer to Step 6 in Chapter 1):

1. Determine the current positions of the centerlines of rotation of all the machinery.
2. Observe any movement restrictions on the machines at the control and adjustment points (usually the machinery feet and hold down bolts).
3. Plot the restrictions on the graph or the model.

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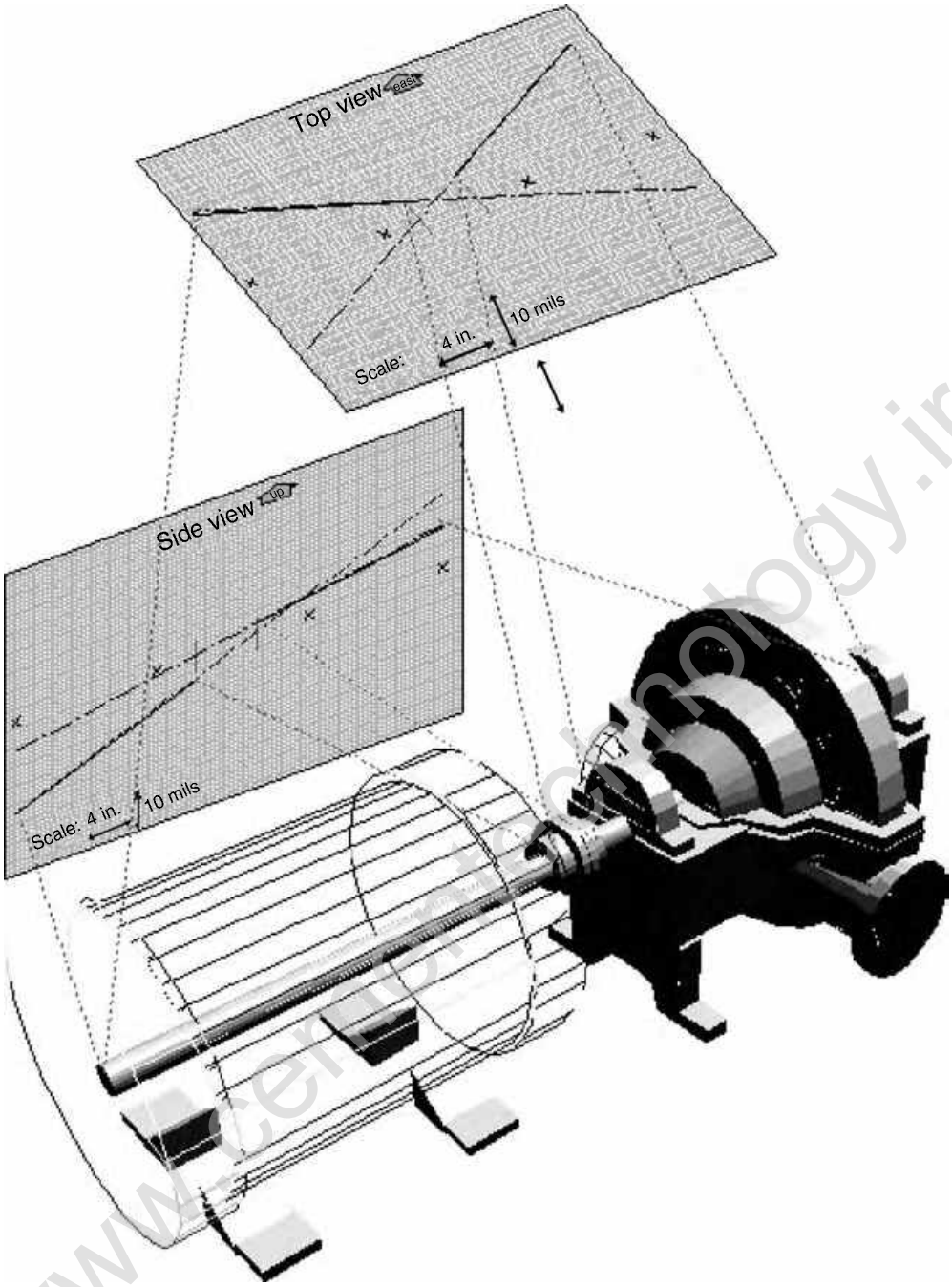


FIGURE 8.3 (**See color insert following page 322.**) Three-dimensional view of the side and top views.

4. Determine the moves for either or both of the machinery casings on the graph or the model that will be feasible to perform.

We will first begin by illustrating the basic principles of how to construct the relative positions of the two centerlines of rotation and then show how you can determine the wide variety of movement options available to you when repositioning misaligned machinery.

8.3 SCALING THE DRIVE SYSTEM ONTO THE ALIGNMENT MODEL

There are several key positions on your drive system where distances need to be measured for scaling onto the graph paper. The most important ones are

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1. Where the foot and the hold down bolts are located on each machine
2. Where the measurements are taken on the machinery shafts

Other critical dimensions that may need to be taken are

1. Where measurements have been taken to observe how the machinery moved from off-line to running conditions (refer to Chapter 16)
2. Where the piping connections are made
3. Where lateral adjustments are made (assuming they are not exactly where the foot bolts are located)
4. Internal clearances between rotating and stationary components in each machine

To begin modeling your alignment problem, the drive system is scaled onto the graph as shown in Figure 8.4 and Figure 8.5. The distances you measure along the length of the drive train should be accurate within 1/4″ if possible or at least with an accuracy of ±1% of the overall length of the drive system.

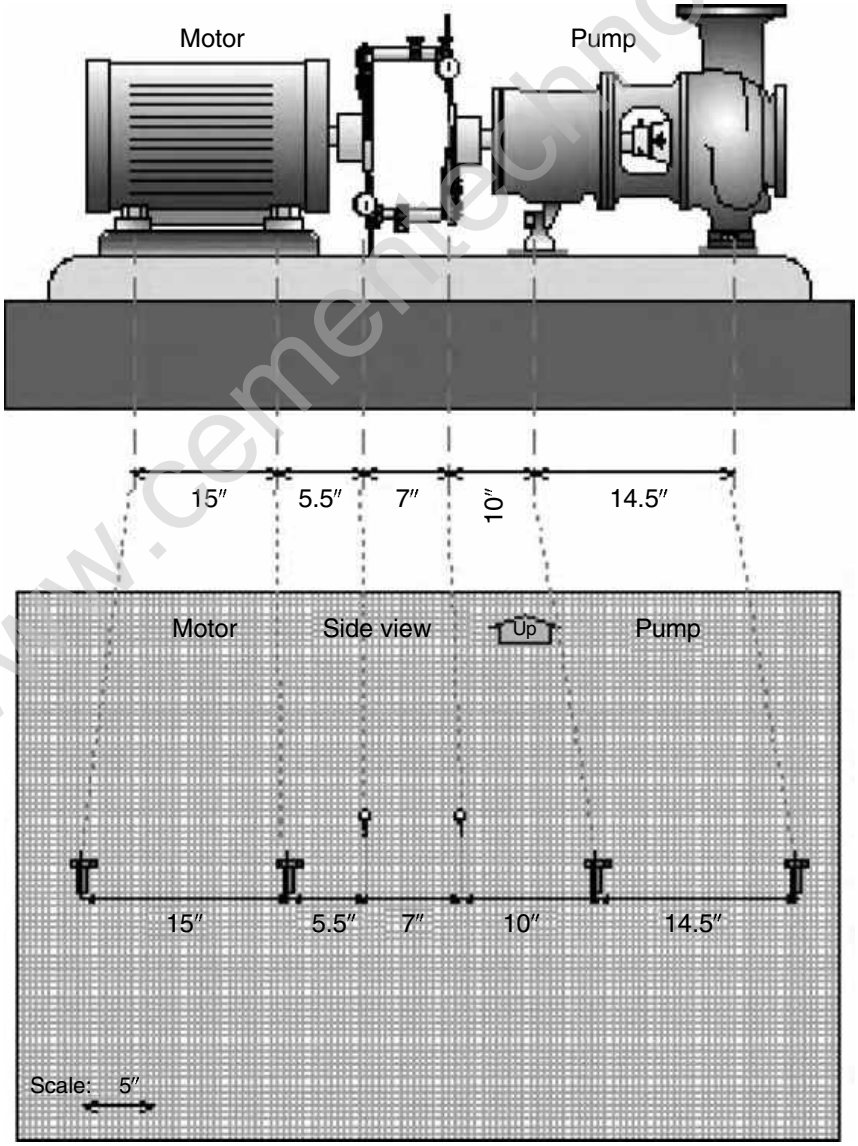


FIGURE 8.4 Scaling the machinery feet and measurement positions onto the graph.

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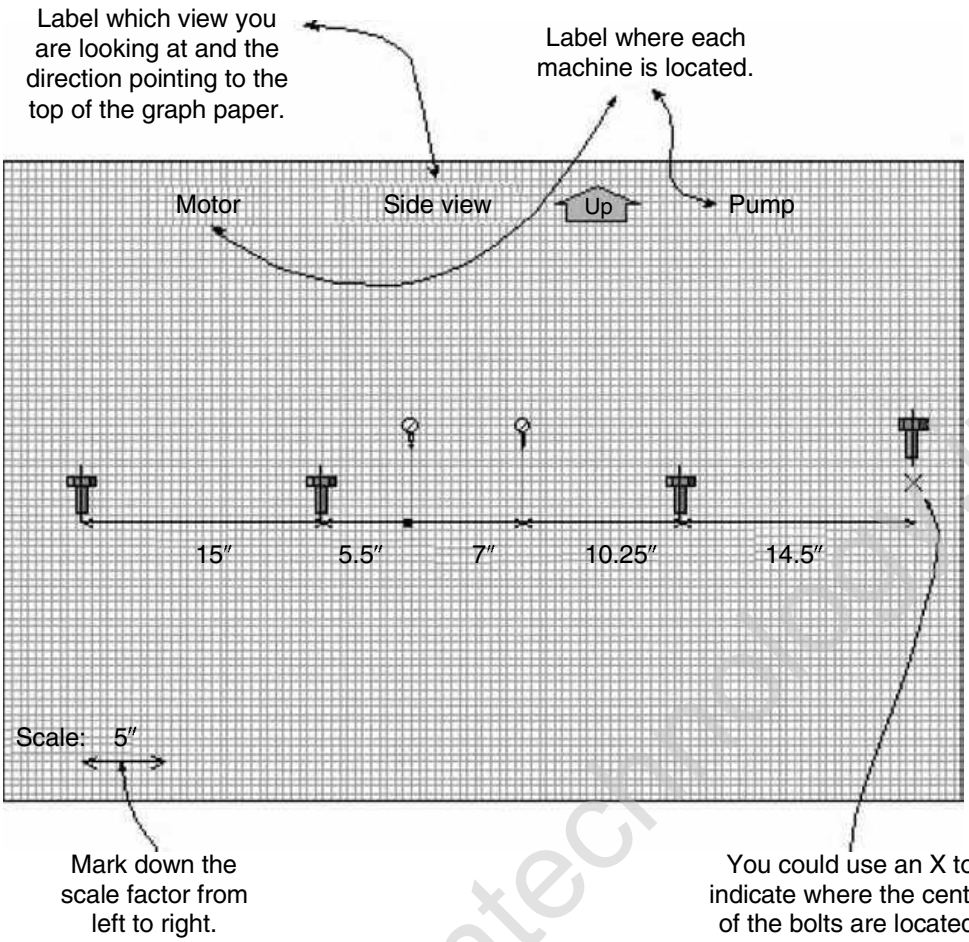


FIGURE 8.5 Scaling the machinery feet and measurement positions onto the graph. Accurately scale off the distances between the inboard and outboard feet of both machines, the distances from the inboard feet of both units to the point where the dial indicator plungers are touching (i.e., taking readings) on both shafts, and the distances between measurement points along the graph centerline from left to right.

8.4 CARDINAL ALIGNMENT GRAPHING AND MODELING RULES

Alignment modeling can be confusing when you first attempt it but there are a few rules that apply when constructing an alignment model using any of the alignment measurement methods:

1. Only plot measurements that have been compensated for bracket sag.
2. Only plot half of a rim dial indicator reading.
3. Positive (+) dial indicator readings means the shaft is "low."
4. Negative (−) dial indicator readings means the shaft is "high."
5. Zero the indicator on the side that is pointing toward the top of the graph paper.
6. Whatever shaft the dial indicator (or any other measuring device) is taking readings on is the shaft that you want to draw on the graph paper.
7. Superimpose your boundary conditions.
8. Select an alignment correction line (a.k.a. overlay line or final desired alignment line) that is possible and easy to do.

8.4.1 Plot Measurements That Have Been Compensated for Bracket Sag

As discussed in Chapter 6, gravity will have an effect on a mechanical bracket when measuring shaft positions on horizontally mounted machinery. Bracket sag typically only

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affects the measurements taken from the top to the bottom of a shaft and will come into play when plotting the shafts in the side view alignment model. Usually the amount of bracket sag is the same on both sides of a shaft and therefore the sags cancel each other out. The readings taken from one side of a shaft to the other side are plotted in the top view alignment model. If you are not sure how to compensate for bracket sag in your measurements, review the section in Chapter 6 and specifically Figure 6.56 and Figure 6.57.

8.4.2 Rim Readings Are Always Twice the Offset Amount

Remember, anytime a rim or circumferential reading is taken, the amount measured from one side to the other side of the shaft (180° of rotation) is twice the amount of the actual distance between the centerlines of rotation at that point. Refer to Figure 6.44 and Figure 6.45 to understand why this happens.

8.4.3 Plus Means "Low" and Minus Means "High"

Where and how to position the rotating machinery shafts on the alignment model will make far more sense if you reason out what the measurement sensor has told you. If you zero a dial indicator at the top of a shaft, sweep to the bottom of that shaft and your indicator has registered a +40 reading, it is low by 20 mils at that point. The sign of the number tells you which way the shaft is, the number tells you how far away it is. This is a vector problem. It has an amount and a direction. The sign tells you the direction, the number tells you the amount.

For example, Figure 8.6 shows a bracket attached to the shaft on the left holding a dial indicator that is measuring the shaft on the right. When the bracket and indicator are rotated to the bottom, the stem of the dial indicator was pushed in as it traversed from the top to the bottom of the shaft on the right. When indicator stems set is pushed in, the needle sweeps in a clockwise direction, producing a positive number. Therefore the shaft on the right is "low" with respect to the shaft on the left.

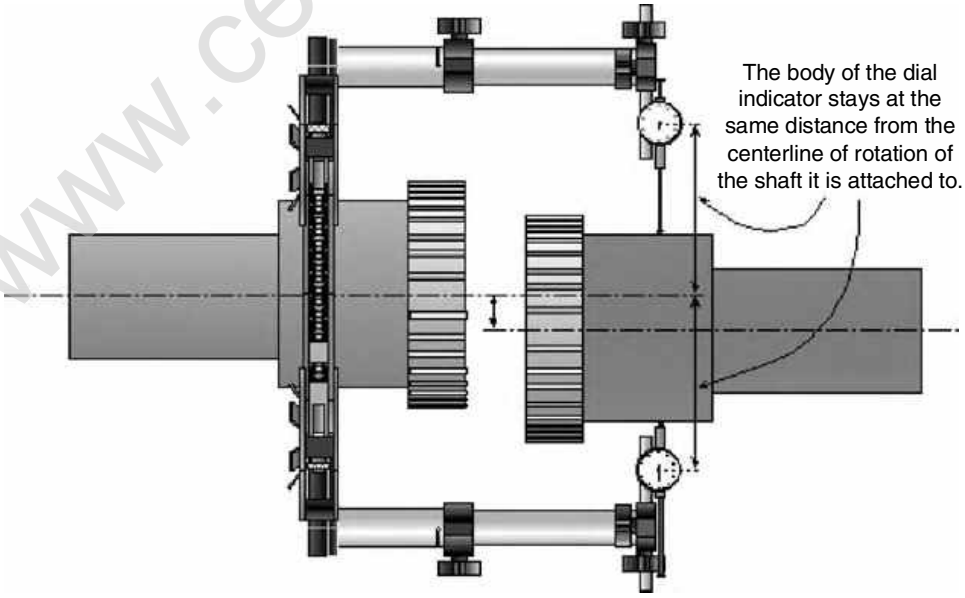


FIGURE 8.6 Positive reading indicates that the shaft you are measuring is "low." The stem of the dial indicator was pushed in as it traversed from the top to the bottom of the shaft on the right. When indicator stems set is pushed in, the needle sweeps in a clockwise direction producing a positive number. Therefore the shaft on the right is "low" with respect to the shaft on the left.

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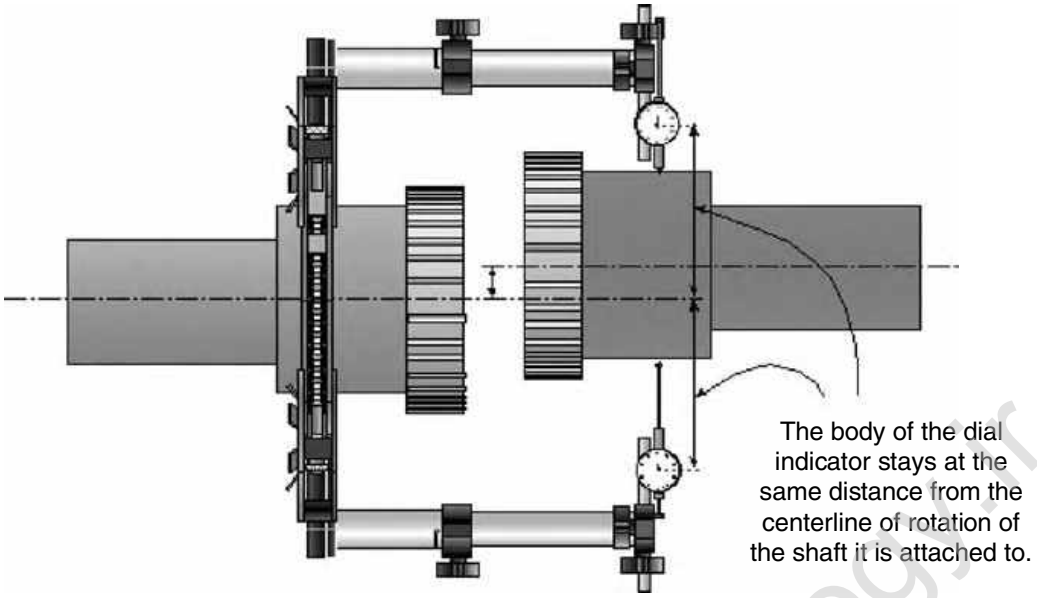


FIGURE 8.7 Negative reading indicates that the shaft you are measuring is "high." The stem of the dial indicator traveled outward as it traversed from the top to the bottom of the shaft on the right. When indicator stems travel outward, the needle sweeps in a counterclockwise direction producing a negative number. Therefore the shaft on the right is "high" with respect to the shaft on the left.

Figure 8.7 shows a bracket attached to the shaft on the left holding a dial indicator that is measuring the shaft on the right. When the bracket and indicator are rotated to the bottom, the stem of the dial indicator traveled outward as it traversed from the top to the bottom of the shaft on the right. When indicator stems travel outward, the needle sweeps in a counterclockwise direction producing a negative number. Therefore the shaft on the right is "high" with respect to the shaft on the left.

The quotation marks around the words "low" and "high" are there for a reason. "High" and "low" are relative terms and only apply if you are viewing horizontally mounted shafts when looking at them in the side view (up and down direction). If for example, you are looking at the shafts in Figure 8.6 from above and the top of the page is north, the shaft on the right in Figure 8.6 would appear to be to the south of the shaft on the right (positive (+) indicator reading). Likewise if you are looking at the shafts in Figure 8.7 from above and the top of the page is north, the shaft on the right in Figure 8.7 would appear to be to the north of the shaft on the right (negative (−) indicator reading).

This sounds very simple but in fact more people have trouble plotting shafts in the top view. Again, it is important to understand what happens to the stem of the indicator as you traverse from one side of a shaft to the other side. Does the stem get pushed in (i.e., go positive) or does it have to travel outward (i.e., go negative)?

8.4.4 Zero the Indicator on the Side That Is Pointing toward the Top of the Graph Paper

In a horizontally mounted drive system, when you are viewing the alignment model in the side view, you will only need to plot the dial indicator measurements you got on the top of the shaft and on the bottom of the shaft. The readings you got on each side (north and south or east and west or left and right) only come into play in the top view.

Classically when people initially set up their alignment measurement system the dial indicator is placed on the top of a shaft in the twelve o'clock position, zero the indicator

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there and sweep through 90° arcs for the other three measurements as shown in Figure 6.35 through Figure 6.38.

8.4.5 Whatever Shaft the Dial Indicator Is Taking Readings on Is the Shaft That You Want to Draw On the Graph Paper

Again, when viewing the alignment model in the side view you want to plot the measurements you got from the top to the bottom of the shaft. Since you typically zero the indicator on the top and sweep to the bottom, you will plot half of the bottom rim reading onto the alignment model. A line representing the centerline of rotation of the pump shaft is drawn from the position where the bracket was attached to the motor shaft through the point where the dial indicator measured the position of the pump shaft as shown in Figure 8.8. Note the scale factor in the lower left corner of the alignment model. Remember, you only plot half of the bottom reading onto the graph. Also remember that whatever shaft the dial indicator is taking readings on is the shaft that you want to draw on the graph paper. In this case, it is the pump shaft.

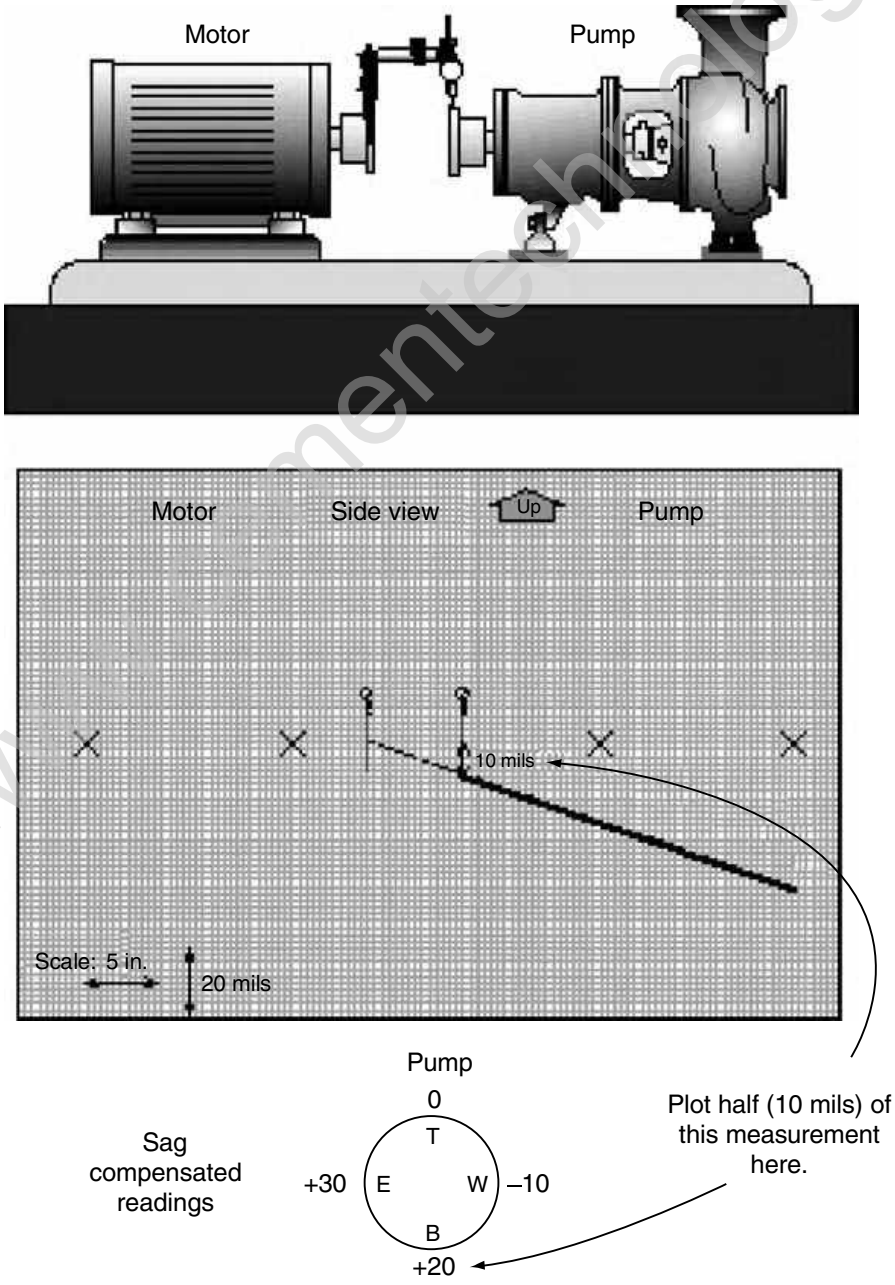


FIGURE 8.8 (**See color insert following page 322.**) Plotting the pump shaft in the side view.

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A line representing the centerline of rotation of the motor shaft is drawn from the position where the bracket was attached to the pump shaft through the point where the dial indicator measured the position of the motor shaft as shown in Figure 8.9. Remember, you only plot half of the bottom reading onto the graph. Also remember that whatever shaft the dial indicator is taking readings on is the shaft that you want to draw on the graph paper. In this case, it is the motor shaft.

Figure 8.9 now shows an exaggerated picture of the misalignment condition of the motor and pump shafts in the up and down direction. But where are the shafts in the side-to-side direction?

When viewing the alignment model in the top view you want to plot the measurements you got from one side of the shaft to the other side of the shaft and here is where a lot of mistakes are classically made. Since you did not zero the indicator on one of the sides, how do you handle the side readings? Real simple, zero the indicator on the side that is pointing toward the top of the graph paper.

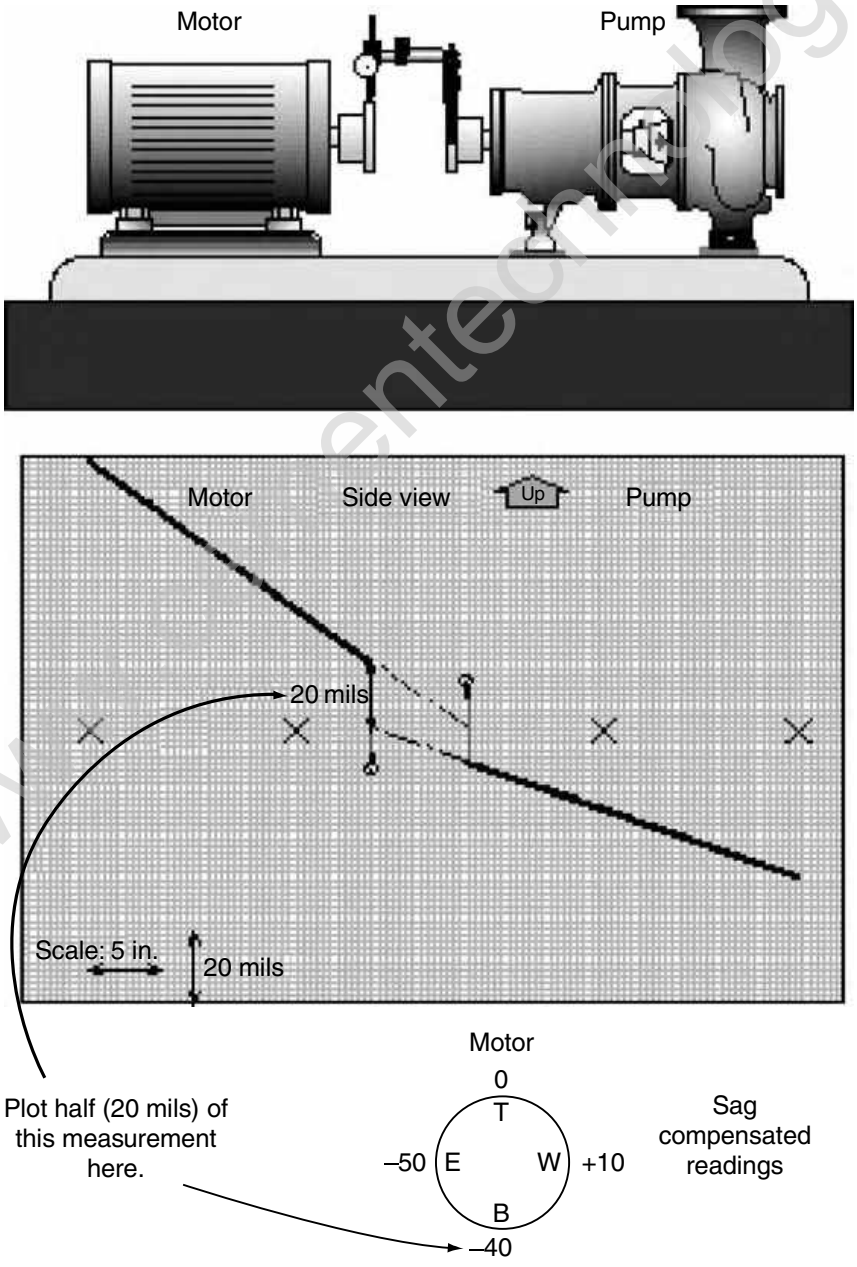


FIGURE 8.9 (**See color insert following page 322.**) Plotting the motor shaft in the side view.

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You have to imagine that you are now looking at your drive system from above. When you are looking at the drive system from the side view with the motor to your left and the pump to your right, which way are you looking (north, south, east, or west)? Getting this direction correct is very important because there is nothing worse than moving your machinery the right amount in the wrong direction.

In our motor and pump drive system we are working on here, let us say that we are looking toward the east as we view the machinery as shown in Figure 8.8 and Figure 8.9. Now that we are going to be viewing our machines from above when modeling the top view, we want to zero the indicator on the side that is pointing toward the top of the graph paper and plot the reading that is on the side that is pointing toward the bottom of the graph paper. In this case, the direction pointing toward the top of the graph paper in the top view is going to be east. Therefore we want to zero the indicator on the east side of each shaft and plot the reading we will obtain on the west side of each shaft.

There are two ways that we can do this. One way is to physically rotate the bracket and indicator over to the east side of each shaft, zero the indicator there, and then rotate the bracket and dial indicator 180° over to the west side and record the dial indicator readings we get there. The other way is to mathematically manipulate the east and west reading we obtained from the complete set of dial indicator readings to zero the east sides. Figure 8.10 shows how to perform this math on the east and west readings.

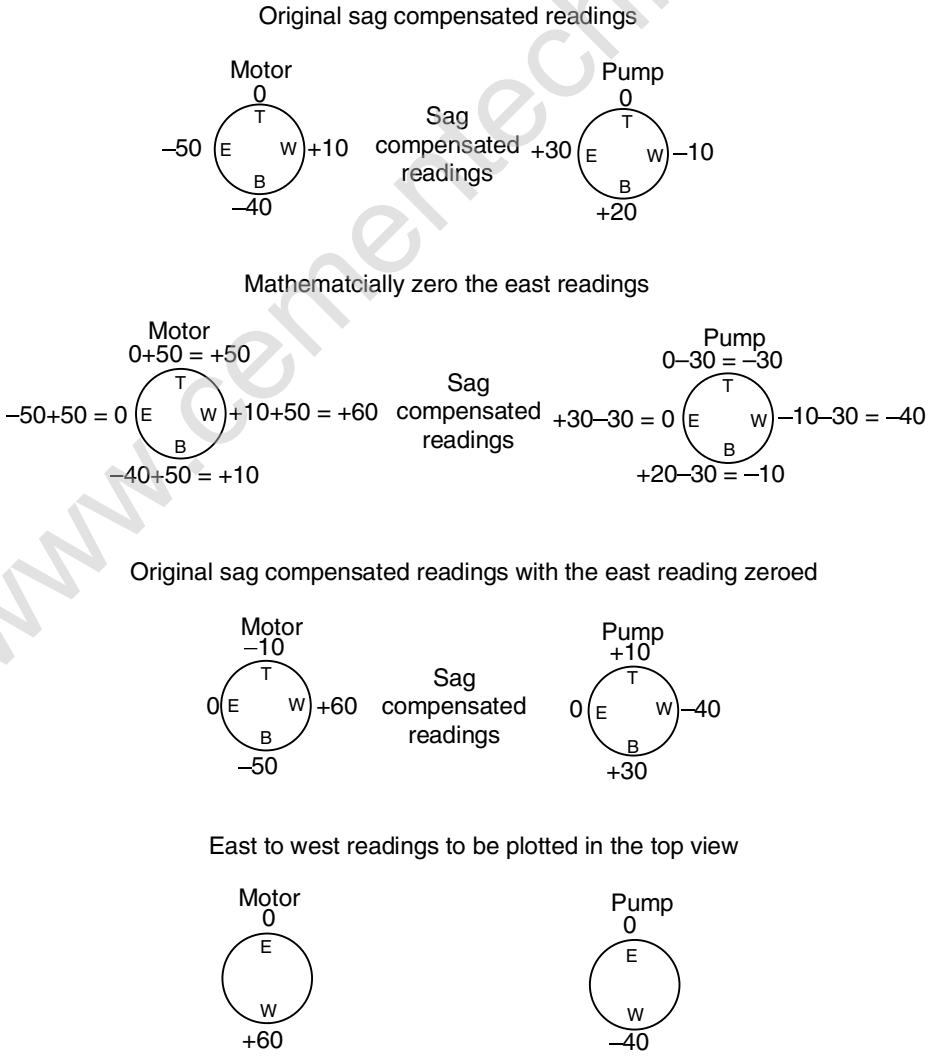


FIGURE 8.10 Zeroing the east readings.

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The original readings with the indicator zeroed on the top and the new readings with the indicator zeroed on the east are telling us the same thing about the misalignment condition between the two shafts. All we did was zero the indicator in a different position and notice that the validity rule still applies whether we zero on top or on the east. Now that we are going to be plotting the shafts in the top view, the top and bottom readings are meaningless, only the east and west readings are important.

Figure 8.11 and Figure 8.12 show how to plot the east and west readings onto the top view alignment model. Notice that the scale factor in the top view is not the same as the scale factor in the side view. They do not have to be the same scale factor in both views but remember what the scale factors are in each view. Without being too repetitive here, remember that you only plot half of the dial indicator reading onto the graph. Also remember that whatever shaft the dial indicator is taking readings on is the shaft that you want to draw on the graph paper. Two of the major graphing mistakes people make are to forget to only plot half of the rim reading and drawing the wrong shaft onto the graph.

The alignment models shown in Figure 8.9 through Figure 8.12 were generated using the reverse indicator method, which is covered in more detail in Chapter 10. The other four alignment

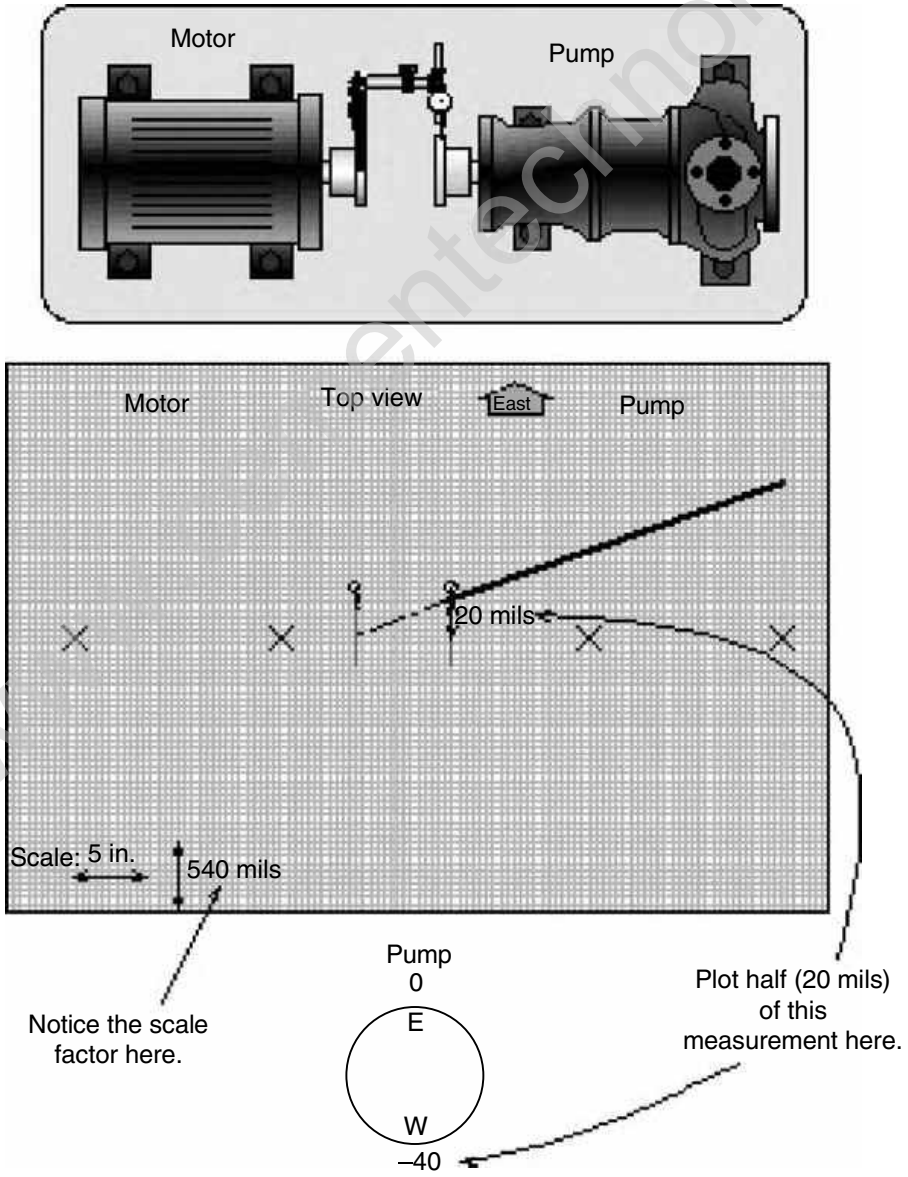


FIGURE 8.11 (**See color insert following page 322.**) Plotting the pump shaft in the top view.

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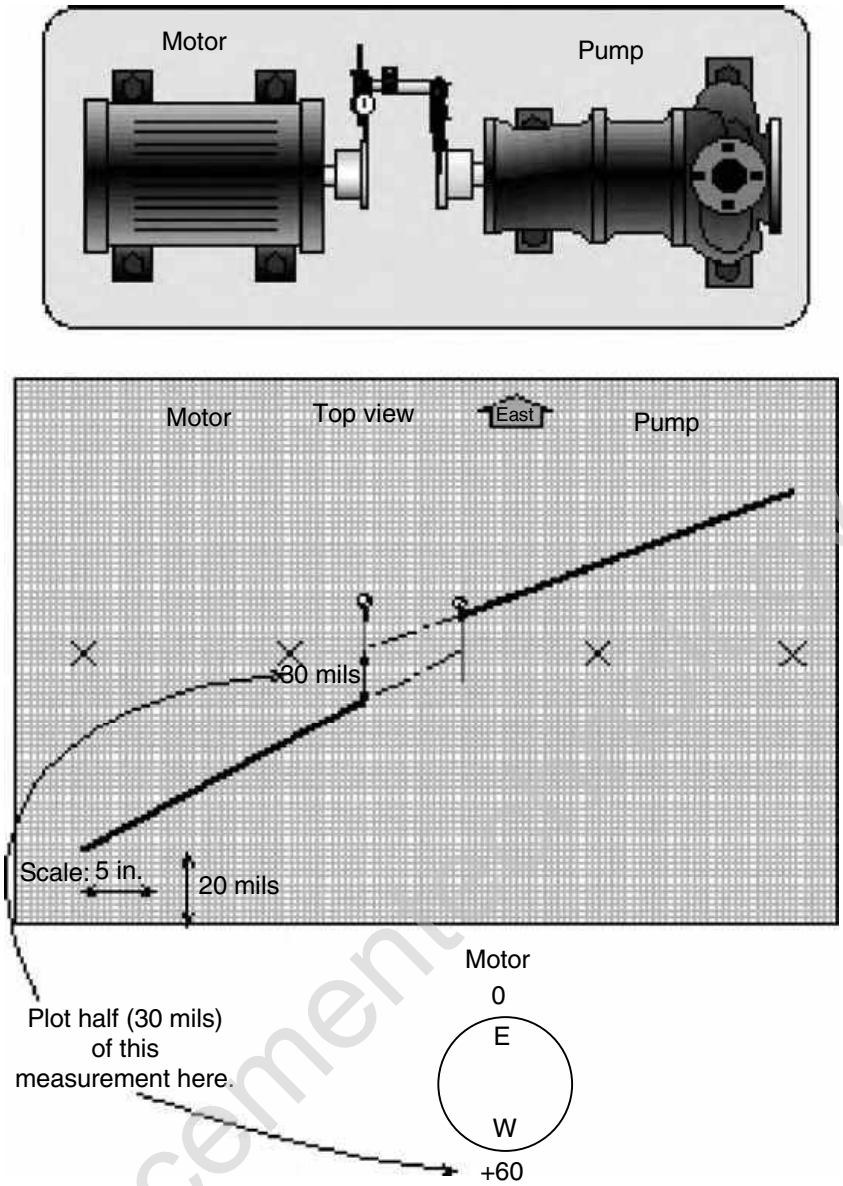


FIGURE 8.12 (**See color insert following page 322.**) Plotting the motor shaft in the top view.

methods (face–rim, double radial, shaft to coupling spool, and face–face) and their associated graphing and modeling techniques will be discussed in Chapter 11 through Chapter 15.

8.4.6 Determining Corrective Moves To Make on One Machine from the Alignment Model

Let us look at another example. Figure 8.13 shows a motor and a fan shaft misalignment condition in the side view. As you can see, the shafts are not in alignment with each other. Now what do we do? The next logical step is to determine the movement restrictions imposed on the machine cases at the control or adjustment points (i.e., where the foot bolts are). Movement restrictions define the boundary condition that help you to make an intelligent decision on what alignment correction would be easy and trouble free to accomplish.

Trouble-free movement solutions? I fully understand that any corrective moves you make on rotating machinery are not going to be trouble free and easy to make. But there are some moves that will be far more difficult to make than others. You really need to have a wide

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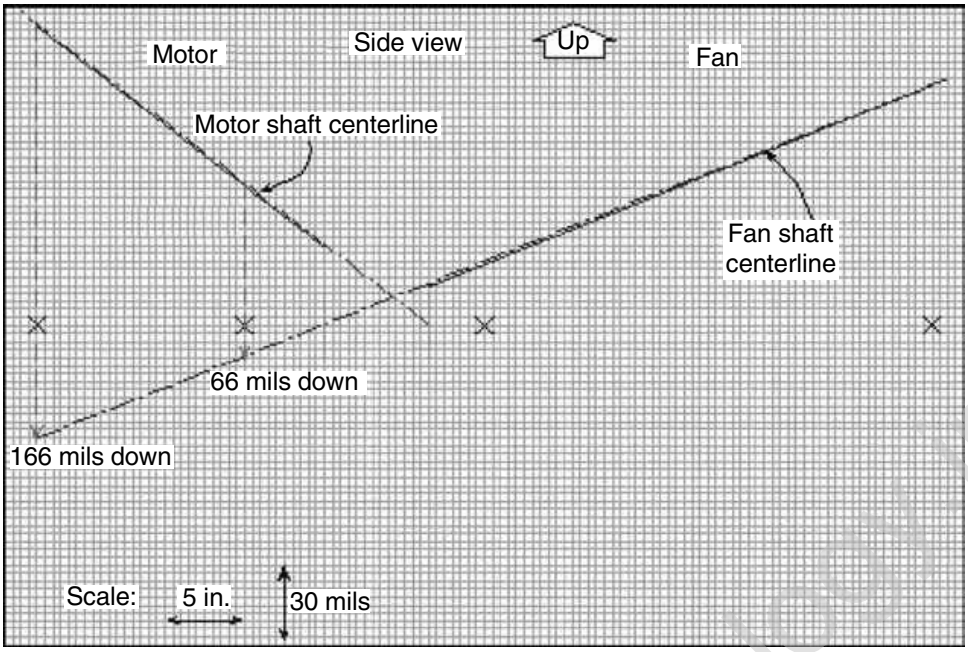


FIGURE 8.13 Movement solutions for the motor only.

variety of options to make the most effective and intelligent alignment correction. Therefore, keep an open, objective mindset when you attempt to fix your alignment problem.

In Figure 8.13, notice that if you wanted to keep the fan in its current position, you would have to move the motor downward at both the inboard and outboard ends. As shown in Figure 8.13, the amount of movement at the outboard bolts is obtained by counting the number of squares (at 3 mils per square with this scale factor) from where the actual motor shaft centerline is at the outboard bolting plane to the extended centerline of rotation of the fan. In this particular case, this is, 166 mils (0.166 in.). The amount of movement at the inboard bolts is obtained by counting the number of squares from where the actual motor shaft centerline is at the inboard bolting plane to the extended centerline of rotation of the fan. In this case, that is, 66 mils (0.066 in.). If there is 166 mils under both outboard bolts and 66 mils under both inboard bolts (that are not soft foot shims) then a good alignment solution would be to remove that amount of shim stock from under the appropriate feet. But what if there are not that many shims under the inboard and outboard feet?

As bizarre as this may sound, I have seen people in a situation like this, remove the motor from the baseplate and grind the baseplate away. Unbelievable, but true. And it is still done somewhere today.

8.4.7 Overlay Line or Final Desired Alignment Line

The final desired alignment line (a.k.a. the overlay line) is a straight line drawn on top of the graph, showing the desired position both shafts should be in to achieve colinearity of centerlines. It should be apparent that if one machine case is stationary, in this case the fan shaft, that machine's centerline of rotation is the final desired alignment line as shown in Figure 8.13.

There is another way to correct the misalignment problem on this motor and fan that will be far less troublesome. Since adjustments are made at the inboard and outboard feet of the machinery, some logical alternative solutions would be to consider using one or more of these feet as pivot points. Both outboard feet or both inboard feet, or the outboard foot of one machine case and the inboard foot of the other machine case could be used as pivot points. By

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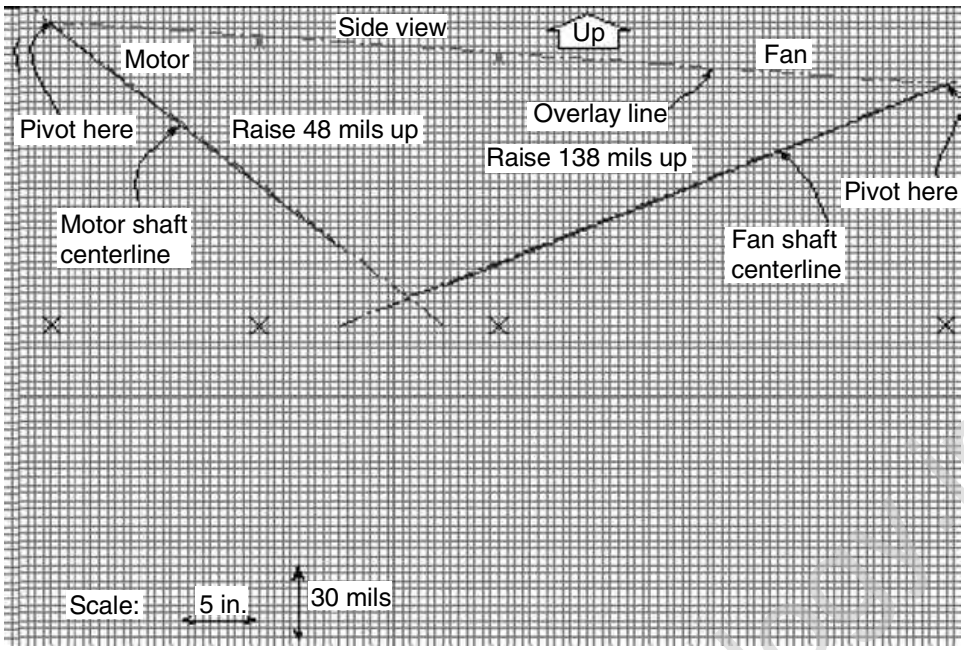


FIGURE 8.14 **(See color insert following page 322.)** Movement solutions for the inboard feet of both the motor and the pump by pivoting at the outboard feet of both machines.

drawing the overlay line through these foot points, shaft alignment can usually be achieved with smaller moves. In real life situations, you will typically have greater success aligning two machine cases a little bit rather than moving one machine case a lot. Figure 8.14 shows using the overlay line to connect the outboard bolting plane of the motor with the outboard bolting plane of the fan. The inboard bolting planes are then moved the amount shown in Figure 8.14 to correct the misalignment condition in the up and down direction. No shims had to be removed and better yet, no baseplates had to be ground away.

8.4.8 Superimpose Your Boundary Conditions, Movement Restrictions, and Allowable Movement Envelope

When viewing the machinery in the up and down direction (side view), the movement restrictions are defined by the amount of movement the machinery can be adjusted in the up and down directions.

How far can machinery casings be moved upward? There is virtually an unlimited amount of movement in the up direction, within reason, that is. Machine cases are typically moved upward by installing shims (i.e., sheet metal of various thicknesses) between the undersides of the machinery feet and the baseplate.

How far can the machinery casings be moved downward? Well, it depends on the amount of shim stock currently under the machinery feet that are not soft foot corrections.

How far can you move a machine down? I don't know. You are going to have to look under the machine to see how much shim stock could be removed from under the machinery feet on every machine in the drive system. Maybe there are 10, 20, or 50 mils of shim stock under the machinery feet that can be removed that are not soft foot corrections that could be taken out. You will have to see what is there. These shims define the "downward movement envelope," or as some people call it, the "basement floor," or as other people call it, the "baseplate restriction point."

Shim stock typically refers to sheet metal thicknesses ranging from 1 mils (0.001 in.) to 125 mils (0.125 in.). There are several companies that manufacture precut, U-shaped shim stock in

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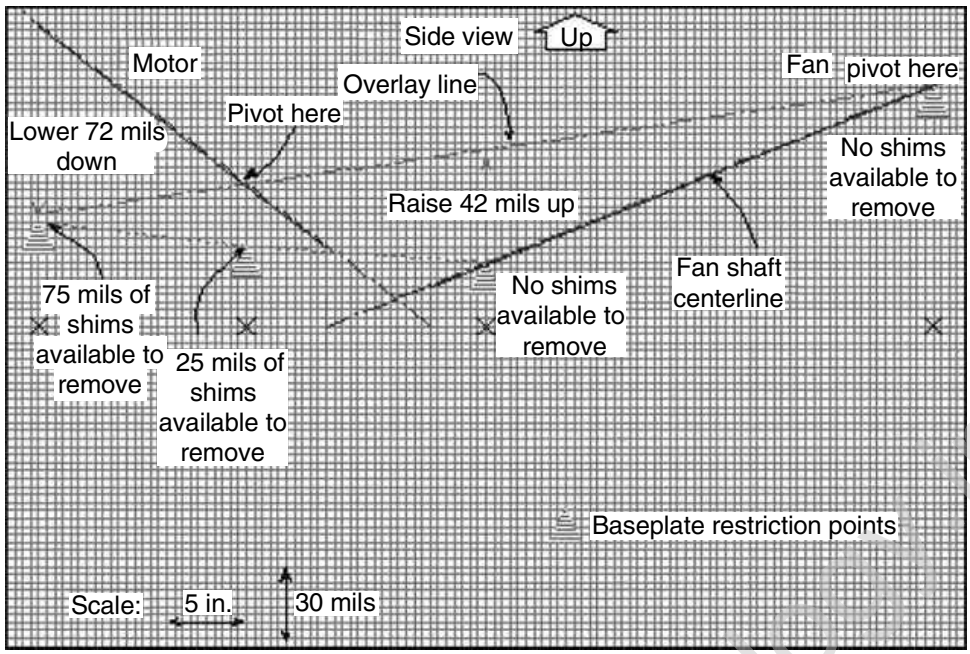


FIGURE 8.15 (**See color insert following page 322.**) Movement solutions using the outboard feet of the fan and the inboard feet of the motor as pivot points.

4 standard sizes and 17 standard thicknesses. Once shim thicknesses get over 125 mils, they are typically referred to as spacers or plates and are custom made from plate steel.

So if you want to move a machine downward and there are no shims under the machinery feet, you are already on the basement floor and that is defined as a downward vertical movement restriction or a baseplate restriction point. Figure 8.15 shows the same motor and fan but now we have observed that there are 75 mils of shim stock under the outboard feet and 25 mils of shims under the inboard feet that are not soft foot corrections that could be removed if we wanted to. By counting down 75 mils from the centerline of the motor shaft and the outboard bolting plane and drawing a baseplate restriction point there, we can now see how far that end can come down without removing metal from the baseplate or machine casing. Similarly, by counting down 25 mils from the centerline of the motor shaft and the inboard bolting plane and drawing a baseplate restriction point there, we can now see how far that end can come down without removing metal. In this particular case, there were no shims under any of the feet of the fan so its baseplate restriction points are positioned directly on the fan centerline at the inboard and outboard ends as shown in Figure 8.15. Now that we know what the lowest points of downward movement could be without removing metal, one possible solution would be to use the outboard feet of the fan and the inboard feet of the motor as pivot points removing 72 mils of shims from under the outboard feet of the motor and installing 42 mils of shims under the inboard feet of the fan as shown in Figure 8.15.

8.4.8.1 Lateral Movement Restrictions

In addition to aligning machinery in the up and down direction, it is also imperative that the machinery be aligned properly side to side. Machinery is aligned side to side by translating the machine case laterally. This sideways movement is typically monitored by setting up dial indicators along the side of the machine case at the inboard and outboard hold down bolts, anchoring the indicators to the frame or baseplate, zeroing the indicators, and then moving the inboard and outboard ends the prescribed amounts. Here is where realignment typically becomes extremely frustrating since there is a limited amount of room between the shanks of the hold down bolts and the holes drilled in the machine case feet.

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If, for example, you wanted to move the outboard end of a machine 120 mils to the south, began moving the outboard end monitoring the move with a dial indicator, and the machine case stopped moving after 50 mils of translation, this would be considered a movement restriction commonly referred to as a "bolt bound" condition. The problem in moving machinery laterally is that there is a limited amount of allowable movement in either direction. The total amount of side-to-side movement at each end of the machine case is referred to as the "lateral movement envelope." To find the allowable lateral movement envelope, remove a bolt from each end of the machine case, look down the hole, and see how much room exists between the shank of the bolt and the hole drilled in the machine case at that foot. If necessary, thread the bolt into the hole a couple of turns, and measure the gaps between the bolt shank and the sides of the hole with feeler or wire gauges.

It is very important for one to recognize that trouble free alignment corrections can only be achieved when the allowable movement envelope is known. Perhaps one of the most important statements that will be made in this chapter is

> When you consider that both machine cases are movable, there are an infinite number of possible ways to align the shafts, some of which fall within the allowable movement envelope.

It seems ridiculous, but many people have ground baseplates or the undersides of machinery feet away because they felt that a machine had to be lowered. When machinery becomes bolt bound when trying to move it sideways, people frequently cut down the shanks of the bolts or grind a hole open more.

There is typically an easier solution. Disappointingly, many of the alignment measurement systems shown in this book force the user to name one machine case stationary and the other one movable which will invariably cause repositioning problems when the machine case has to be moved outside its allowable movement envelope. This may not happen the first time you align a drive system, or the second or third time, but if you align enough machinery, eventually you will not be able to move the movable machine the amount prescribed. Once the centerlines of rotation have been determined and the allowable movement envelope illustrated on the graph, it becomes very apparent what repositioning moves will work easily and which ones will not.

Figure 8.16 shows the top view alignment model of a motor and pump. Not knowing any better, it appears that all you would have to do is move the outboard end of the motor 14 mils to the east and the inboard end of the motor 4 mils to the west. Easy enough. But what if the outboard end of the motor is bolt bound to the east already?

By removing one bolt from the inboard and outboard ends of both the motor and pump, the lateral movement restrictions can be observed. In this case the following restrictions were observed:

1. Outboard end of motor—bolt bound to east and 40 mils of possible movement to the west
2. Inboard end of motor—bolt bound to east and 40 mils of possible movement to the west
3. Inboard end of pump—32 mils of possible movement to the east and 8 mils of possible movement to the west
4. Outboard end of pump—36 mils of possible movement to the east and 4 mils of possible movement to the west

By plotting the eastbound and westbound restriction onto the alignment model, you can now see the easy corridor of movement. One possible solution (out of many) is shown in Figure 8.16.

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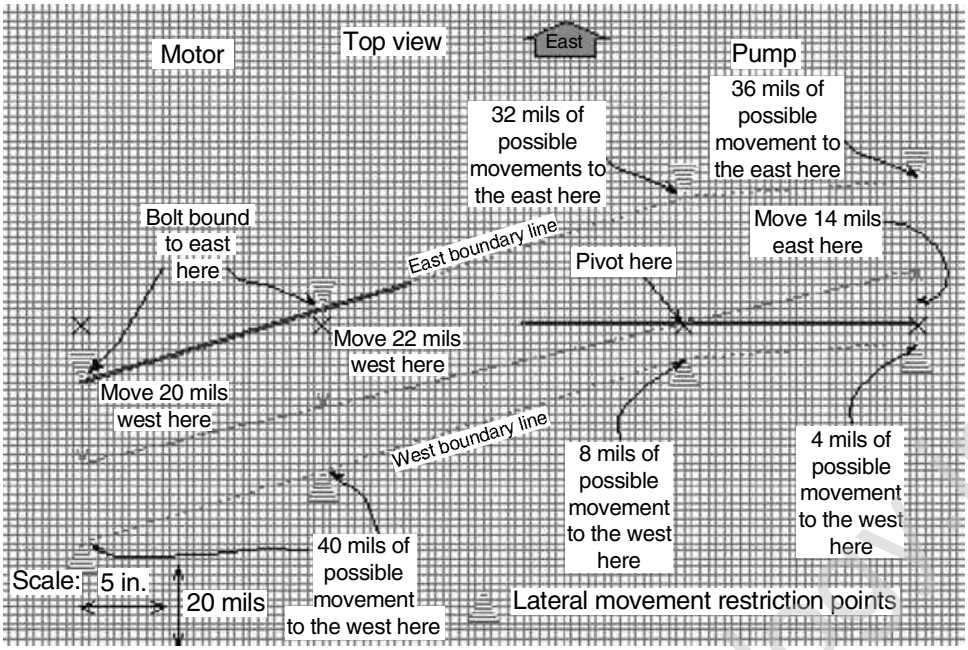


FIGURE 8.16 (**See color insert following page 322.**) Applying lateral movement restrictions to arrive at an easy sideways move within the east and west corridors.

Please, for your own sake, follow these four basic steps to prevent you from wasting hours or days of your time correcting a misalignment condition:

1. Find the positions of every shaft in the drive train by the graphing and modeling techniques shown in this and later chapters.
2. Determine the total allowable movement envelope of all the machine cases in both directions.
3. Plot the restrictions on the graph or model.
4. Select a final desired alignment line or overlay line that fits within the allowable movement envelope (hopefully) and move the machinery to that line.

If you are involved with aligning machinery, by following the four steps above, it is guaranteed that you will save countless hours of wasted time trying to move one machine where it does not really want to go.

8.4.8.2 Where Did the Stationary–Movable Alignment Concept Come From?

I don't know. Every piece of rotating machinery in existence has, at one time or another, been placed there. Mother Earth never gave birth to a machine. They are neither part of the Earth's mantle nor firmly imbedded in bedrock. Every machine is movable, it is just a matter of effort (pain) to reposition it. So why have the vast majority of people who align machinery called one machine stationary and the other machine movable?

The only viable reason that I can come up with is this—in virtually every industry there is an electric motor driving a pump. When you first approach a motor pump arrangement, you immediately notice that the pump has piping attached to it and the only appendage attached to the motor is conduit (usually flexible conduit). From your limited vantage point at this time it would appear to be easier to move the motor because there is no piping attached to it like the pump. You would prefer to just move the motor because it looks easier to move than the pump (and so would I). The assumption is made that the pump will not be moved, no matter what position you find the motor shaft in with respect to the pump shaft.

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But what do you do when you have to align a steam turbine driving a pump? They are both piped; which machine do you call the stationary machine—the pump or the turbine? No matter what your answer is, you are going to have to move one of them and they both have piping attached to their casings.

Piping is no excuse not to move a piece of machinery, particularly in light of what most of us know about how piping is really attached to machinery. For some people, they are afraid to loosen the bolts holding a machinery with piping attached to it because the piping strain is so severe that they fear the machine will shift so far that it will never get back into alignment. So is the problem with the alignment process or the piping fit-up? Refer to Chapter 3 for information on this subject.

If you align enough machinery and insist that one machine will be stationary, eventually you will get exactly what you deserve for your shallow range of thinking.

8.4.8.3 Solving Piping Fit-Up Problems with the Overlay Line

Although we have been showing that the overlay line (a.k.a. final desired alignment line) is drawn through foot bolt points, it is important to see that the overlay line could be drawn anywhere and the machinery shafts moved to that line.

This can be particularly beneficial if there are other considerations that have to be taken into account such as piping fit-up problems. Figure 8.17 shows a motor and pump where the suction pipe is 1/4 in. higher than the suction flange on the pump and there is a 1/4 in. excessive gap at the discharge flanges. Rather than align both shafts, then install an additional

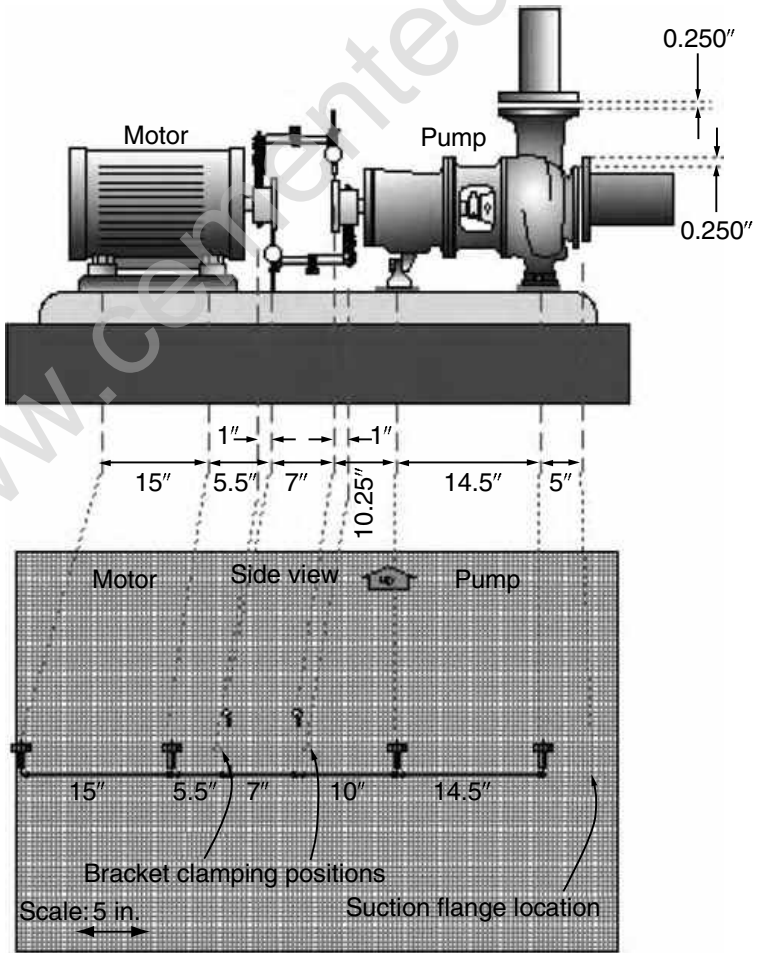


FIGURE 8.17 **(See color insert following page 322.)** Scaling off the dimensions for a motor and pump including the location of the suction flange on the pump.

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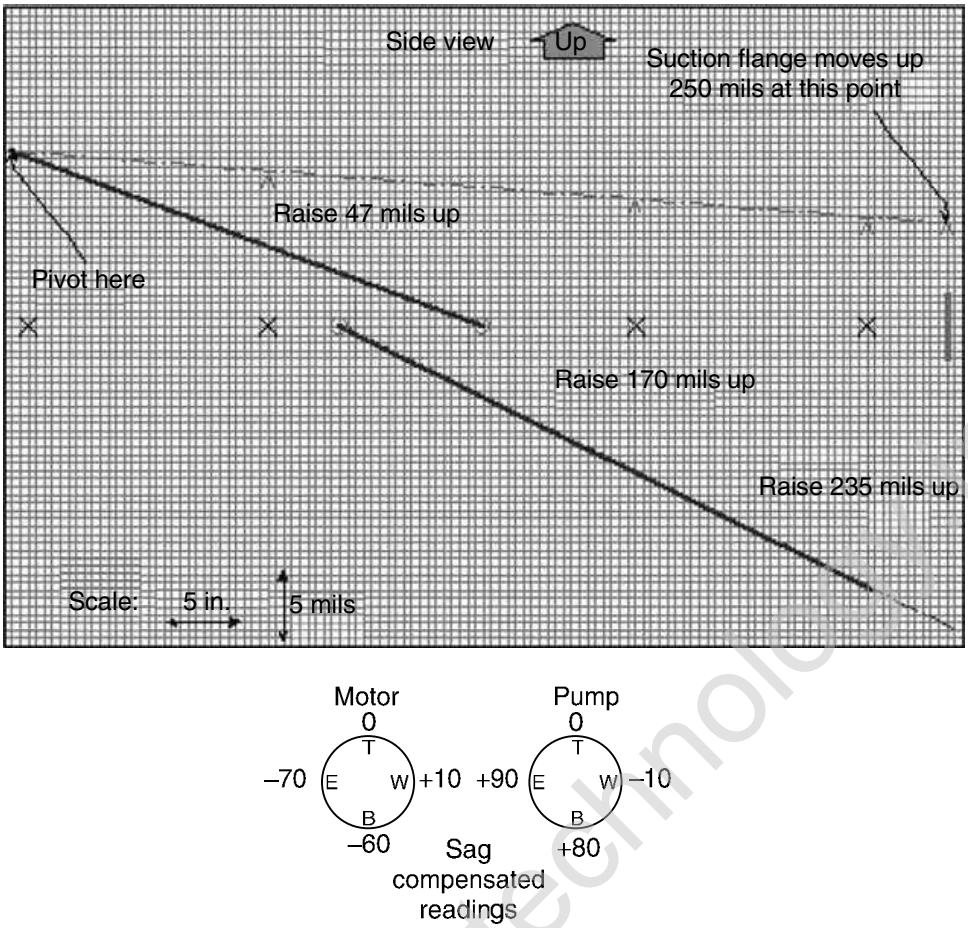


FIGURE 8.18 (**See color insert following page 322.**) Overlay line positioned to correct the piping fit-up problem and align the shafts with one move.

250 mils (1/4 in.) under all the feet, another easier solution exists. Scale off where the suction flange of the pump is onto the alignment model. Extend the centerline of the pump to go out to the suction flange point. Place a mark 250 mils above the pump shaft centerline where the suction flange is located. Construct an overlay line to go from that point to the outboard bolts of the motor as illustrated in Figure 8.18. Then solve for the moves at each bolting plane not only to eliminate the piping fit-up problem but also to align the shafts.

We have reviewed many of the basic concepts behind alignment modeling in this chapter. Determining your maximum misalignment deviation and whether you are within acceptable alignment tolerances will be covered in the next chapter. Specific instruction on how to perform all five-alignment measurement methods and their associated modeling techniques will be covered in Chapter 10 through Chapter 15.

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9 Defining Misalignment: Alignment and Coupling Tolerances

9.1 WHAT EXACTLY IS SHAFT ALIGNMENT?

In very broad terms, shaft misalignment occurs when the centerlines of rotation of two (or more) machinery shafts are not in line with each other. Therefore, in its purest definition, shaft alignment occurs when the centerlines of rotation of two (or more) shafts are collinear when operating at normal conditions. As simple as that may sound, there still exists a considerable amount of confusion to people who are just beginning to study this subject when trying to precisely define the amount of misalignment that may exist between two shafts flexibly or rigidly coupled together.

How do you measure misalignment when there are so many different coupling designs? Where should the misalignment be measured? Is it measured in terms of mils, degrees, millimeters of offset, arcseconds, radians? How accurate does the alignment have to be? When should the alignment be measured, when the machines are off-line or when they are running? Perhaps a commonly asked question needs to be addressed first.

9.2 DOES LEVEL AND ALIGNED MEAN THE SAME THING?

The level and aligned does not mean the same thing. The term "level" is related to Earth's gravitational pull. When an object is in a horizontal state or condition or points along the length of the centerline of an object are at the same altitude, the object is considered to be level. Another way of stating this is that an object is level if the surface of the object is perpendicular to the lines of gravitational force. A level rotating machinery foundation located in Boston would not be parallel to a level rotating machinery foundation located in San Francisco as the Earth's surface is curved. The average diameter of the Earth is 7908.5 miles (7922 miles at the equator and 7895 miles at the poles due to the centrifugal force causing the planet to bulge at the center). When measuring the distances of arc across the Earth's surface, 1° of arc is slightly over 69 miles, 1 min of arc is slightly over 1.15 miles, and one second of arc is slightly over 101.2 ft.

It is possible, although rare, to have a machinery drive train both level and aligned. It is also possible to have a machinery drive train level but not aligned and it is also possible to have a machinery drive train aligned but not level. As shaft alignment deals specifically with the centerlines of rotation of machinery shafts it is possible to have, or not to have, the centerlines of rotation perpendicular to the lines of gravitational force.

Historically, there were a considerable number of patents filed from 1900 to 1950 that seemed to combine (or maybe confuse) the concept of level and aligned. A number of these

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Machinery Type	Minimum Recommended Levelness	Coplanar Surface Deviation
General process machinery supported in antifriction bearings	10 mils per foot	10 mils
General process machinery supported in sleeve bearings (up to 500 hp)	5 mils per foot	5 mils
Process machinery supported in antifriction bearings (500+ hp)	5 mils per foot	5 mils
Process machinery supported in sleeve bearings (500+ hp)	2 mils per foot	5 mils
Machine tools	1 mil per foot	2 mils
Note: 1 mil = 0.001 in.		

FIGURE 9.1 Recommended levelness and coplanar surface deviation for rotating machinery baseplates or soleplates.

alignment devices were used in the paper industry where extremely long ''line shafts'' were installed to drive different parts of a paper machine. These line shafts were constructed with numerous sections of shafting that were connected end to end with rigid couplings and supported by a number of bearing pedestals along the length of the drive system, which could be 300 ft in length or more. Even if a mechanic carefully aligned each section of shafting at each rigid coupling connection along the length of the line shaft and perfectly leveled each shaft section, the centerline of rotation at each end of a 300-ft long line shaft would be out by 0.018 in. due to the curvature of the Earth's surface.

Although level and aligned may not mean the same thing, proper leveling is important as well as having coplanar surfaces. Levelness refers to a line or surface, which is perpendicular to gravity; coplanar surface refers to ''flatness.'' Are the points where the machinery cases contact the baseplate (or soleplates) in the same plane? If not, how much of a deviation is there?

It is very common to see baseplates where the machinery contact surfaces are not in the same plane or several soleplates that contact a single machine case not be in the same plane. This coplanar deviation may be a contributor to a condition commonly referred to as a ''soft foot'' problem and was covered in Chapter 5. Figure 9.1 shows the recommended guidelines for leveling machinery baseplates and coplanar surface deviation. As will be seen, even if the surfaces of a baseplate are perfectly level and perfectly coplanar, a soft foot condition could still exist.

9.3 MEASURING ANGLES

Angular misalignment is shown in Figure 9.2. As the term ''angular'' or ''angle'' is used, it is necessary to briefly describe how angles are measured.

There are 360° in a circle. Each degree can be divided into 60 parts called minutes of arc and each minute of arc can be further divided into 60 parts called seconds of arc. Therefore there are 21,600 min of arc and 1,296,000 s of arc in a circle.

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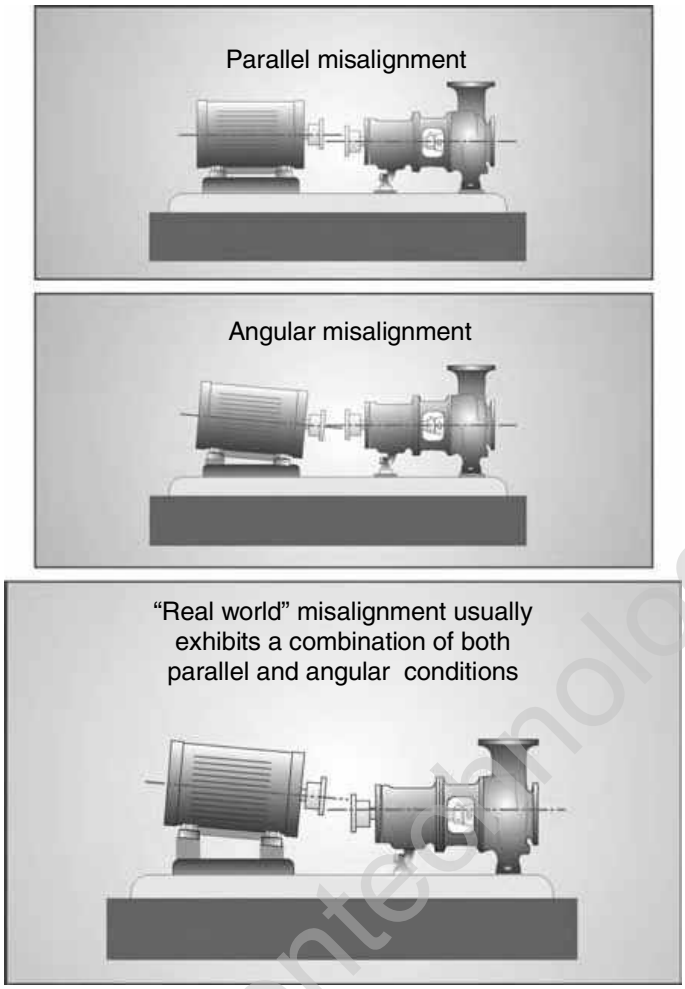


FIGURE 9.2 How shafts can be misaligned.

Another way of expressing circles is by use of radians. All circles are mathematically related by an irrational number called pi (π), which is approximately equal to 3.14159. There are 2π radians in a circle. Therefore one radian is equal to 57.295828°.

Despite the fact that the expression ''angular misalignment'' is used frequently it comes as a surprise to learn that no known shaft alignment measurement system actually uses an angular measurement sensor or device.

9.4 TYPES OF MISALIGNMENT

Shaft misalignment can occur in two basic ways: parallel and angular as shown in Figure 9.2. Actual field conditions usually have a combination of both parallel and angular misalignment so measuring the relationship of the shafts gets to be a little complicated in a three-dimensional world especially when you try to show this relationship on a two-dimensional piece of paper. I find it helpful at times to take a pencil in each hand and position them based on the dial indicator readings to reflect how the shafts of each unit are sitting.

9.5 DEFINITION OF SHAFT MISALIGNMENT

In more precise terms, shaft misalignment is the deviation of relative shaft position from a collinear axis of rotation measured at the flexing points in the coupling when equipment is running at normal operating conditions. To better understand this definition, let us dissect each part of this statement to clearly illustrate what is involved.

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Collinear means in the same line or in the same axis. If two shafts are collinear, then they are aligned. The deviation of relative shaft position accounts for the measured difference between the actual centerline of rotation of one shaft and the projected centerline of rotation of the other shaft.

There are literally dozens of different types of couplings. Rather than have guidelines for each individual coupling, it is important to understand that there is one common design parameter that applies to all flexible couplings:

> For a flexible coupling to accept both parallel and angular misalignment there must be at least two points along the projected shaft axes where the coupling can flex or articulate to accommodate the misalignment condition.

The rotational power from one shaft is transferred over to another shaft through these flexing points. These flexing points are also referred to as flexing planes or points of power transmission. Shaft alignment accuracy should be independent of the type of coupling used and should be expressed as a function of the shaft positions, not the coupling design or the mechanical flexing limits of the coupling. Figure 9.3 illustrates where the flexing points in a variety of different coupling designs are located. I have seen several instances where there is only one flexing point in the coupling and have also seen more than two flexing points in the coupling connecting two shafts together. If only one flexing point is present and there is an offset between the shafts or a combination of an angle and an offset, there will be some very high radial forces transmitted across the coupling into the bearings of the two machines. If there are more than two flexing points, there will be a considerable amount of uncontrolled motion in between the two connected shafts, usually resulting in very high vibration levels in the machinery.

Why should misalignment be measured at the flexing points in the coupling? Simply because that is where the coupling is forced to accommodate the misalignment condition and that is where the action, wear, and power transfer across the coupling is occurring.

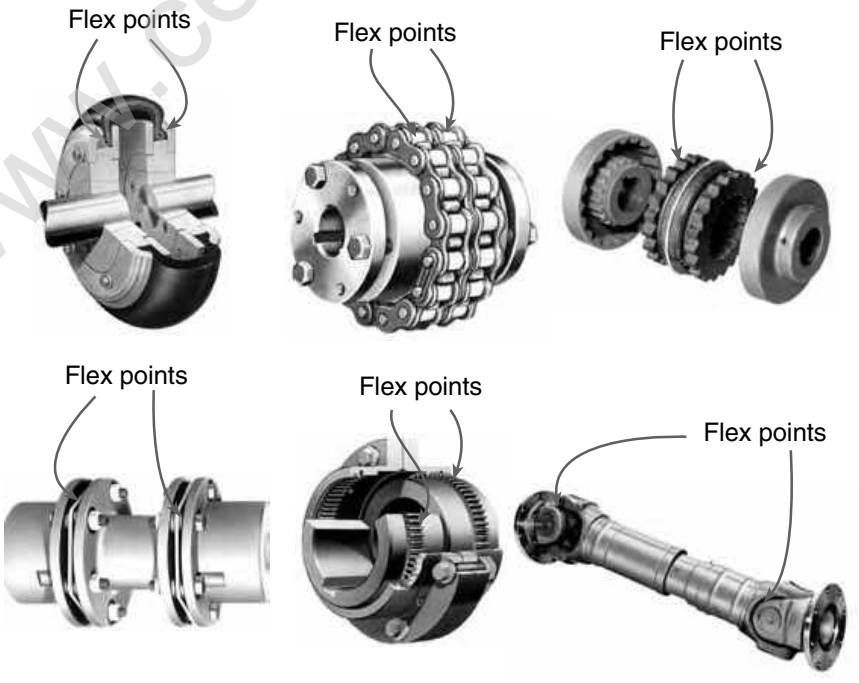


FIGURE 9.3 Flexing point locations in a variety of couplings.

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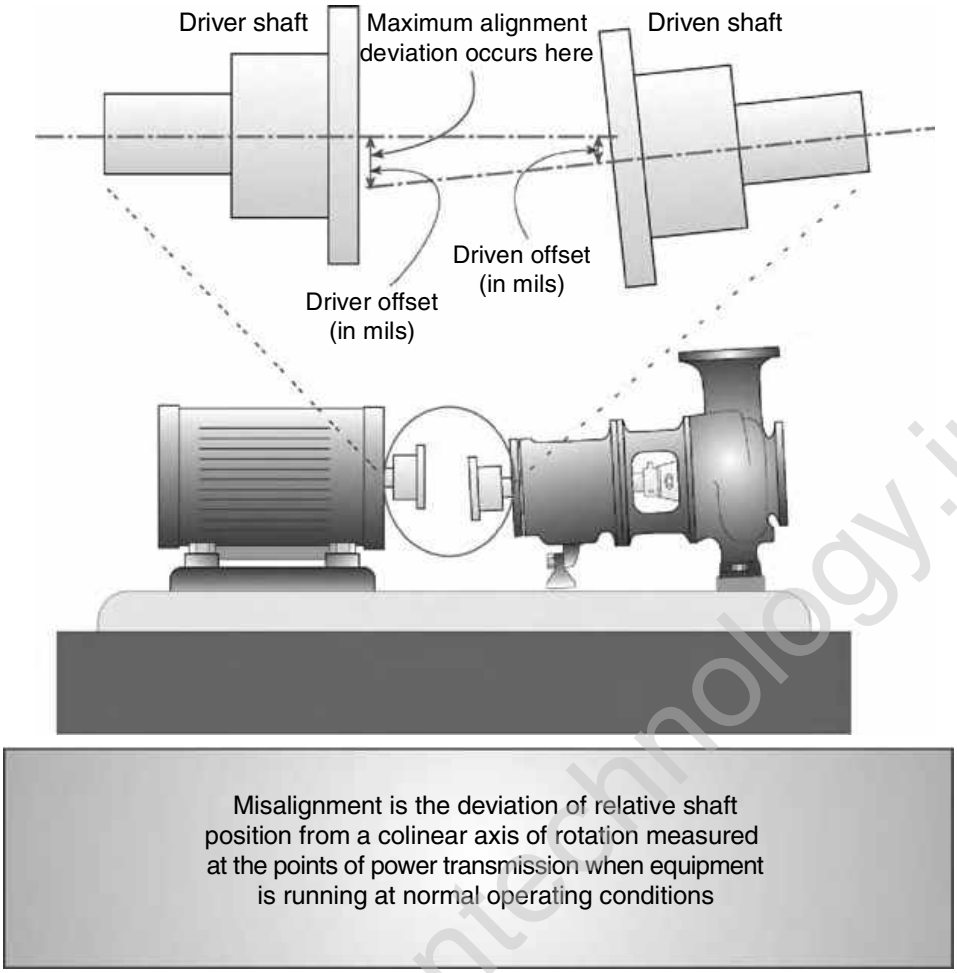


FIGURE 9.4 Definition of shaft misalignment.

Figure 9.4 shows a typical misalignment situation on a motor and a pump. By projecting the axis of rotation of the motor shaft toward the pump shaft (and conversely the pump shaft rotational axis toward the motor shaft) there is a measurable deviation between the projected axes of rotation of each shaft and the actual shaft centerlines of each shaft where the power is transmitted through the coupling from one flexing point to another. As we measure misalignment in two different planes (vertical and horizontal) there will be four deviations that occur at each flexible coupling as shown in Figure 9.5. In a horizontally mounted drive train, two of these deviations occur in the top view describing the amount of lateral (side to side) misalignment. Two more deviations occur when viewing the drive train in the side view, which describes the vertical (up and down) misalignment. The goal of the person doing the alignment is to position the machinery casings such that all of these deviations are below certain tolerance values.

Therefore, there are three factors that affect alignment tolerances of rotating machinery—the speed of the drive train, the maximum deviation at either flexing point or point of power transmission and power reception, and the distance between the flexing points or points of power transmission.

The last part of the definition of shaft misalignment is probably the toughest to achieve and usually the one aspect of alignment that is most often ignored. When rotating equipment is started, the shafts will begin to move to another position. The most common cause of this movement is due to temperature changes that occur in the machinery casings and therefore this movement is commonly referred to as hot and cold alignment. These temperature changes are caused by friction in the bearings or by thermal changes that occur in the process liquids and gases. Movement of machinery may also be caused by process reaction moments

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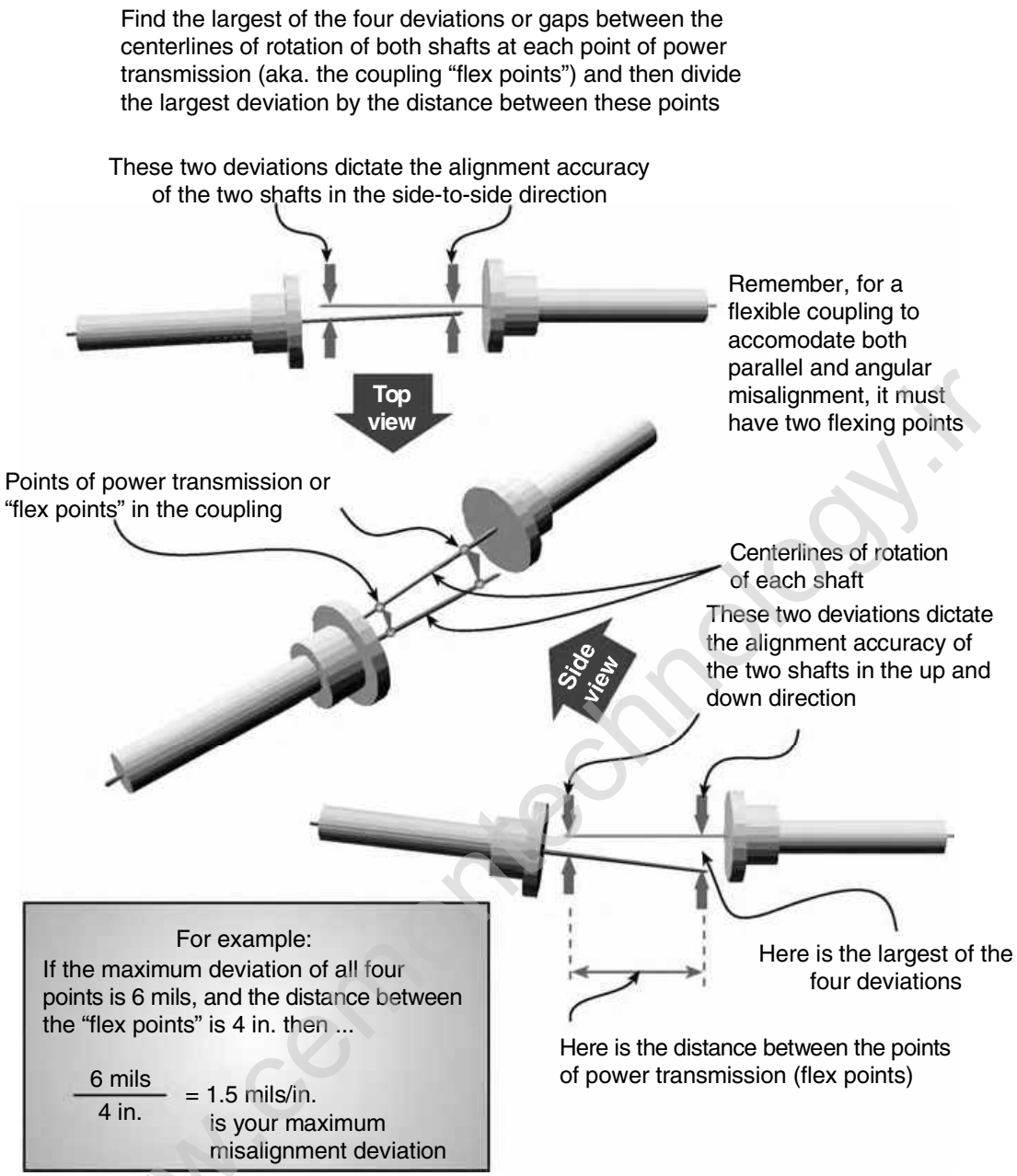


FIGURE 9.5 How to find the maximum misalignment deviation?

in attached piping or counterreactions due to the rotation of the rotor, something similar to the forces you feel when you try to move your arm around with a spinning gyroscope in your hand. Chapter 16 covers this topic in great detail.

Throughout this book misalignment conditions will be specified in mils offset occurring at each point of power transmission, with particular attention being paid to the maximum offset that occurs at either transmission point as determined in Figure 9.5. Figure 9.6 shows a go/no-go guide for acceptable shaft alignment of flexibly connected rotating machinery. Notice that all three factors mentioned above are incorporated in the Misalignment Tolerance Guide. The speed of the machine (in thousands of rpm) is shown on the *X*-axis of the graph. On the *Y*-axis of the graph, the distance between power transmission points (in inches or meters) and the maximum offset (in mils or millimeters) is expressed as a ratio (mils/inch or millimeters/meter).

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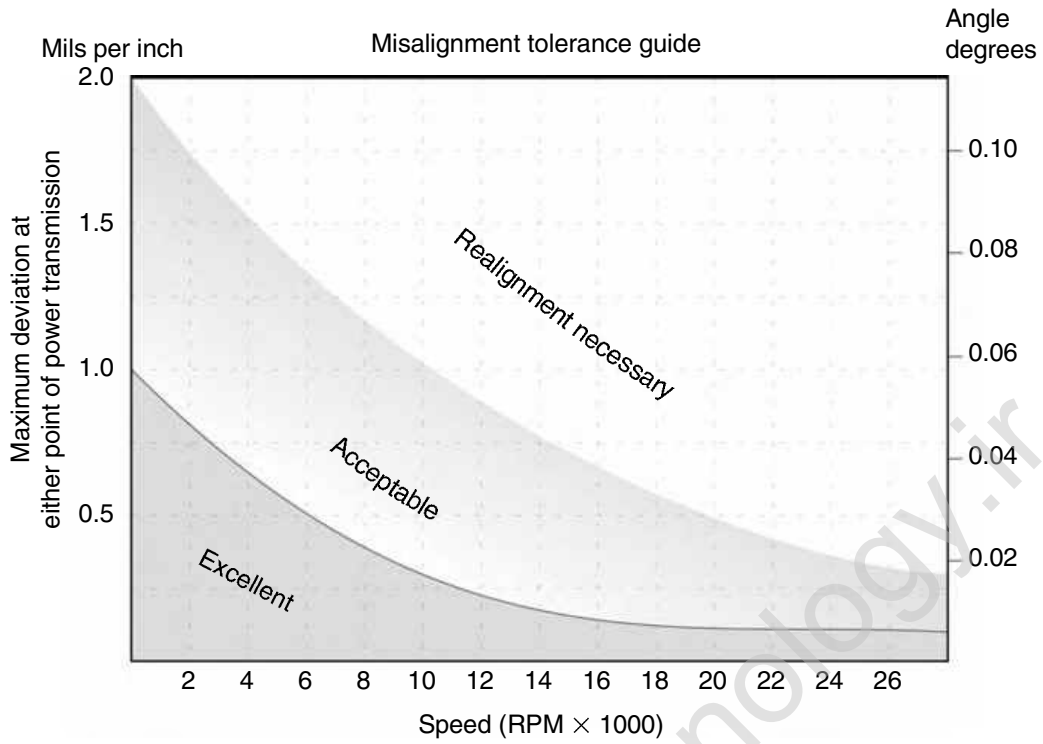


FIGURE 9.6 (**See color insert following page 322.**) Recommended maximum misalignment of flexibly connected rotating machinery.

9.6 CHECKING THE MISALIGNMENT TOLERANCE

The alignment modeling techniques shown in Chapter 8 enable us to visually represent the relative positions of the centerlines of rotation of the two shafts. Once the positions of the shafts have been determined, the first step is to determine whether the amount of misalignment is within tolerance or not.

At any point in time, the machinery shafts are somewhat misaligned side to side and misaligned up and down (or any other coordinates they happen to lie in). The key is to find the largest of the four deviations at the points of power transmission (the flexing points) and divide it by the distance between points of power transmission (the flexing points). Two of these deviations occur in the top view, which will show the amount of lateral (side to side) misalignment, two more deviations occur in the side view, which show the amount of vertical (up/down) misalignment as illustrated in Figure 9.7 and Figure 9.8.

The side view in Figure 9.7 shows a 30 mil deviation at the coupling flex point by the motor and a 15 mil deviation at the coupling flex point by the fan. The top view in Figure 9.8 shows an 18 mil deviation at the coupling flex point by the motor and a 6 mil deviation at the coupling flex point by the fan. Of those four deviations, 30, 15, 18, and 6 mils, the largest deviation is 30 mils. The distance between the flexing points in the coupling is 6.5 in. To determine your maximum misalignment deviation, find the largest of the four deviations, in this case 30 mils, and divide it by the distance between the flexing points in the coupling, in this case 6.5 in. 30 mils divided by 6.5 in. is 4.6 mils per inch. If you refer to the *Alignment Tolerance Guide* (Figure 9.6) you can see that 4.6 mils/in. is not within acceptable alignment tolerances at any shaft speed. Therefore, you have to improve the alignment between those two machines.

The maximum misalignment deviation can be determined directly from the graph. Note that the flexing points do not necessarily occur at the same points where the readings were taken. However, as the misalignment is expressed in mils per inch, whether the deviations are measured where the readings were taken, or at the ends of the shafts, or at the flexing points

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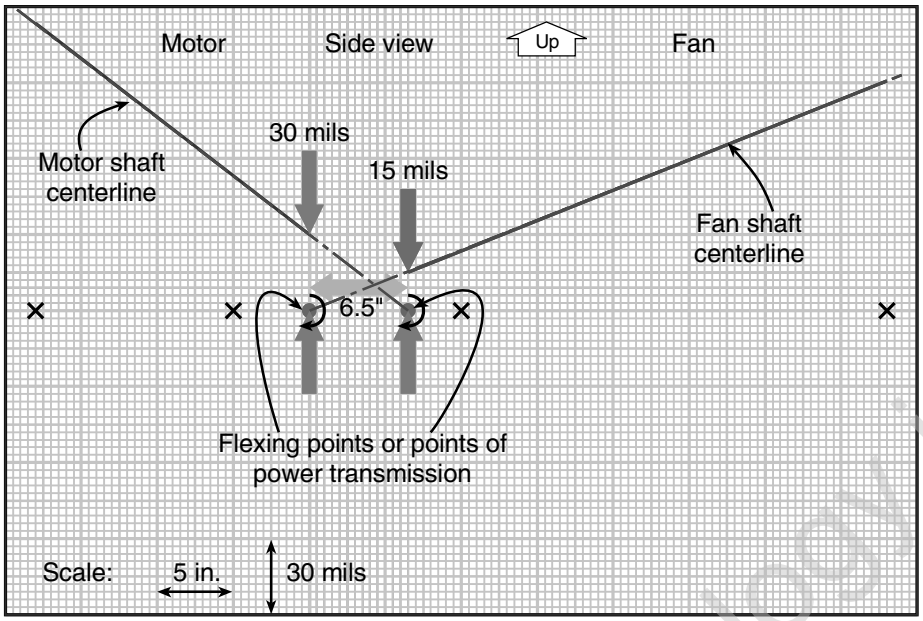


FIGURE 9.7 Misalignment deviations in the side view.

themselves, the misalignment deviation (remember, it is in mils/inch) will always be the same amount (i.e., the deviation might be different at each of the points around the coupling area but so is the distance between them).

9.7 SHAFT VERSUS COUPLING ALIGNMENT

Frequently people use the terms "shaft alignment" and "coupling alignment" interchangeably. Is there really a difference?

Yes, shaft alignment and coupling alignment do not necessarily mean the same thing. Coupling tolerances typically define the maximum misalignment limit of the flexible coupling.

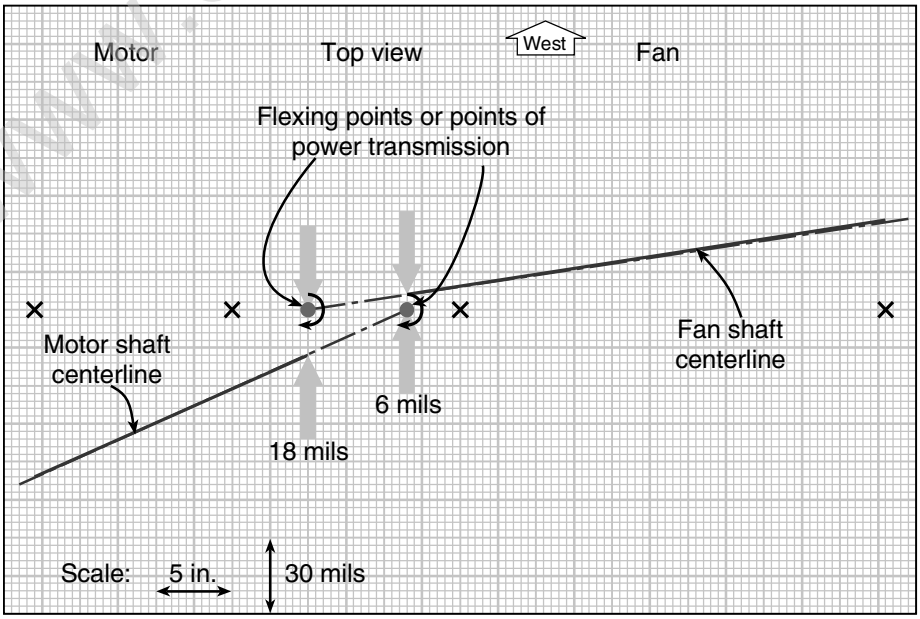


FIGURE 9.8 Misalignment deviations in the top view.

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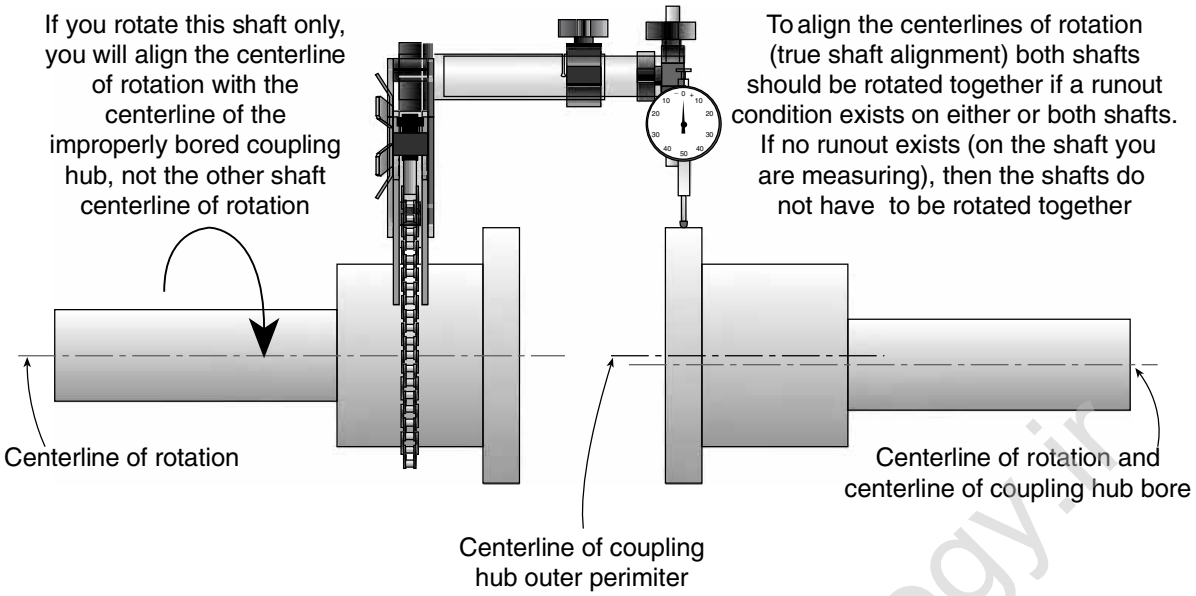


FIGURE 9.9 Aligning a centerline of rotation to the center of an improperly bored coupling.

In other words, at what point will the coupling "lock up" and stop working or begin to sustain rapid deterioration in the coupling. It is not uncommon to see 1/8 in. or more allowable offset advertised by a coupling manufacturer. The coupling may be able to accept this amount of misalignment but can the machinery shafts, bearings, and seals accept this for long periods of time?

There may be other problems with the coupling that can confuse the person performing an alignment job. Notice in Figure 9.9 that the centerline of rotation of the shaft on the left is in line with the centerline of the bore of the coupling hub on the shaft on the right but it is not in line with the centerline of rotation of the shaft on the right.

By its purest definition, shaft alignment occurs when the centerlines of rotation are collinear. This is a very important point in aligning rotating machinery that a vast number of people overlook. It is possible to align the centerlines of rotation of machinery shafts that are bent or that have improperly bored coupling hubs and never know that these eccentricity problems exist. These eccentricities are referred to as runout problems and were discussed in Chapter 5. From the inboard bearing out to the end of the shaft there are three basic types of eccentricity problems that can occur: a coupling hub whose hole was bored off center or overbored and drawn off center with the set screw, a skew (angle) bored hole in the coupling hub, or a bent shaft.

Notice in Figure 9.10 that when a bent shaft is rotated, its centerline of rotation is straight but the shaft itself is not. In this situation what should we try to align the shaft on the right to? The centerline of the bore of the coupling hub at the end of the shaft or the centerline of rotation itself? The correct answer should be: you should never try to align a piece of machinery that has a bent shaft or an improperly bored coupling hub.

9.8 HOW STRAIGHT ARE ROTATING MACHINERY SHAFTS?

The assumption that many people make is that the centerlines of rotation of machinery operate in a perfectly straight line. In vertically oriented shafts this may indeed be true but the vast majority of rotating machinery has shafts that are horizontally mounted and the weight of the shaft and the components attached to the shaft cause the shaft to bow from its own weight. This naturally occurring curvature of a machinery rotor is often referred to as a catenary curve.

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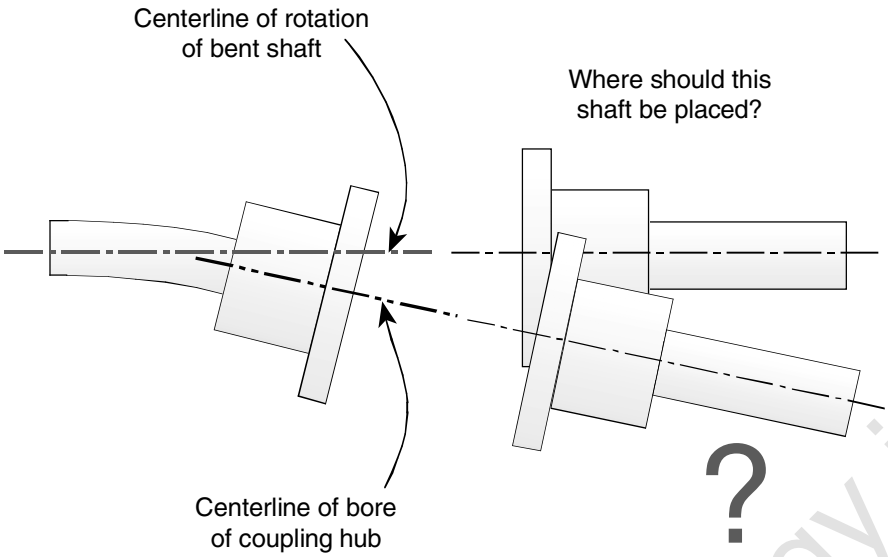


FIGURE 9.10 Problems aligning a straight shaft to a bent one.

Definitions:

Catenary—the parabolic curve assumed by a perfectly flexible inextensible cord of uniform density and cross section suspended from two fixed points.

Catenoid—the surface described by the rotation of a catenary about its axis.

Figure 9.11 shows an exaggerated view of the naturally occurring curvature of center mounted and overhung shafts.

The amount of deflection depends on several factors such as the stiffness of the shaft, the amount of weight between support points, the bearing design, and the distance between support points. For the vast majority of rotating machinery in existence, this catenary bow is negligible and for all practical purposes is ignored. On extremely long drive trains however (e.g., turbine generators in power generating plants and motor-generator sets), this catenary curve must be taken into consideration. Figure 9.12 shows the ideal catenary curve for a 1200 MW turbine generator unit. Notice that the difference in elevation between bearings 5 or 6 compared to bearing 10 is 1.25 in.

When a very long, flexible shaft begins to rotate, the bow tries to straighten out but will never become a perfect straight line. It is important to understand that the axis of rotation of a shaft could very possibly operate on a curved axis of rotation. In situations where two or more pieces of machinery are coupled together where one or more of the shafts are rotating around a catenary-shaped axis of rotation, it is important to align the shafts so they maintain the curved centerline of rotation.

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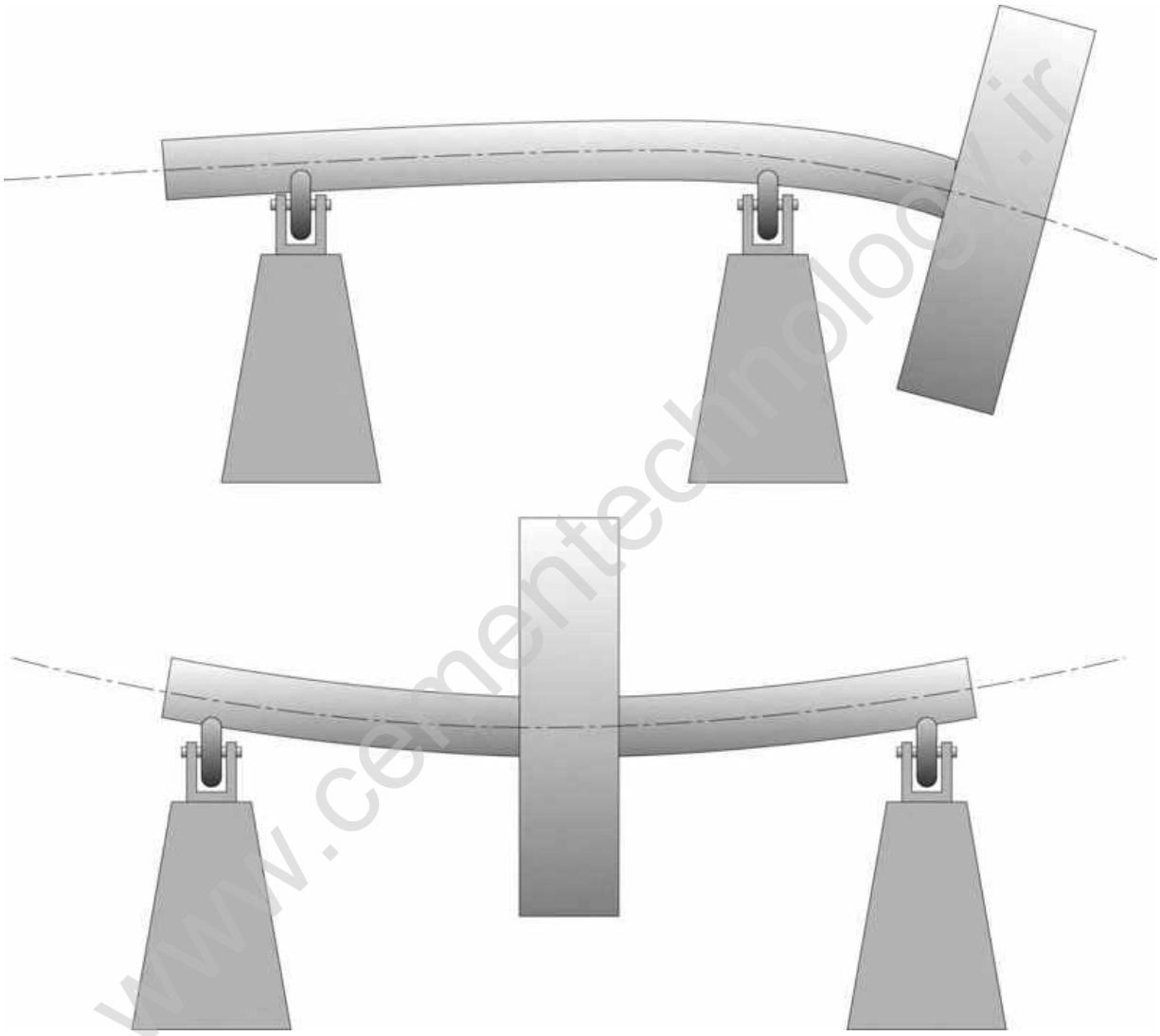


FIGURE 9.11 Natural elastic bending of horizontally mounted shafts.

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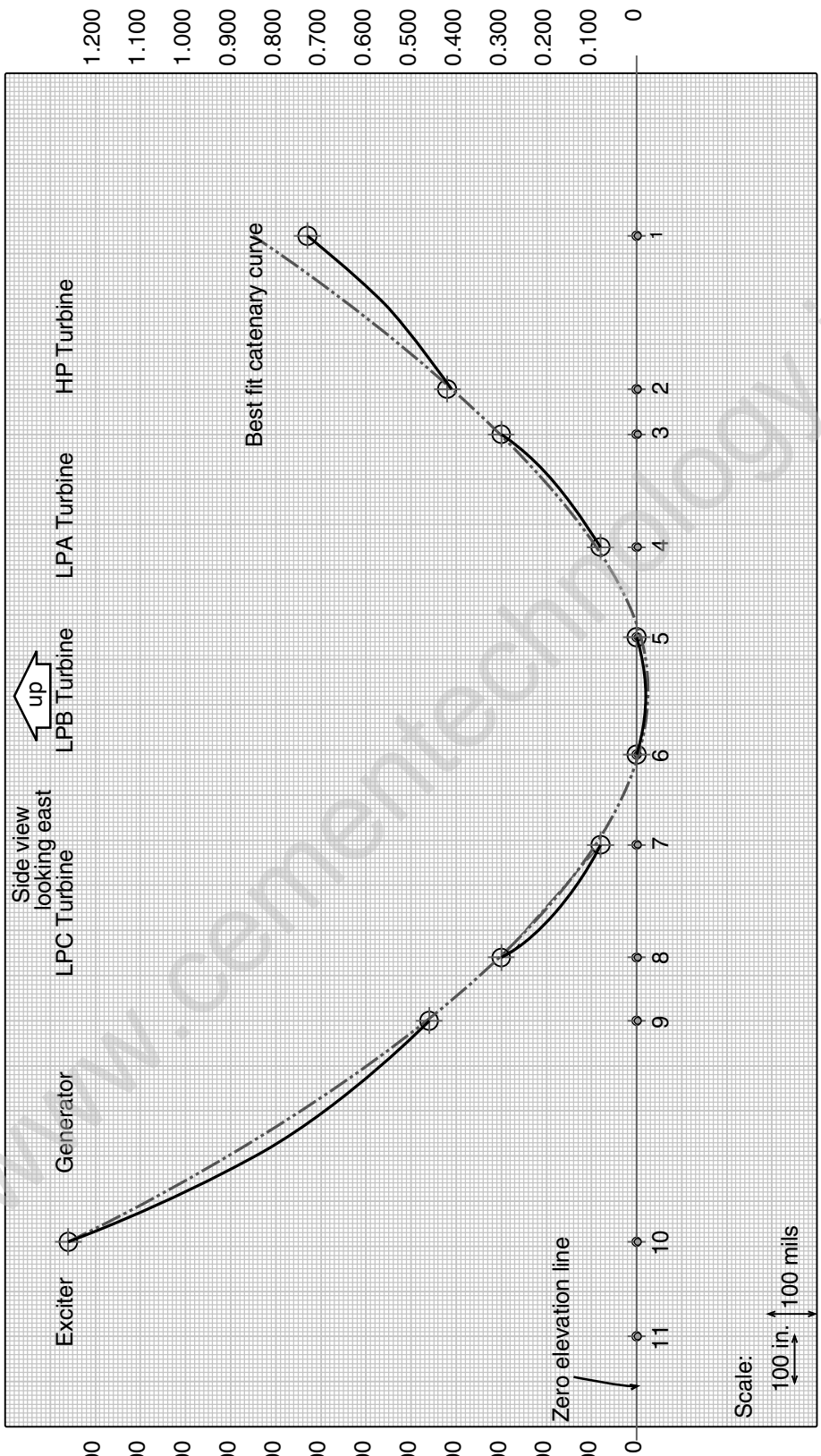


FIGURE 9.12 Catenary curve of a turbine-generator drive system.

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10 Reverse Indicator Method

The reverse indicator method is also often called the indicator reverse method or the double dial method and is shown in Figure 10.1. This method seems to have originated around the mid to late 1950s in the United States and it is not clear who first developed this technique. Clark brothers (now Dresser–Rand) adopted this technique for use in aligning their rotating machinery and Don Cutler (currently working for Thomas-Rexnord) developed the line-to-point graphing method while working for Clark.

The reverse indicator method can be used on 60%–70% of the rotating machinery in existence and is still one of the preferred dial indicator methods for measuring rotating machinery shafts. It is best suited for use when the distances between measuring points on each shaft range from 3 to 30 in. Although Figure 10.1 shows using two brackets and two dial indicators at the same time, there is no reason why one bracket and dial indicator setup could not be used where a set of readings are captured on one shaft first and then reversing the bracket and indicator to capture a set of readings on the other shaft. In fact, it may be wise to use just one bracket at a time to insure that readings are taken correctly and minimize the confusion that could result from trying to observe two indicators simultaneously as illustrated in Figure 10.2. Additionally, how much time does it take to setup two brackets and two indicators versus setting up one bracket and one indicator twice?

Advantages

- Typically more accurate than the face-rim method as the distance from the mounting point of the bracket to the point where the indicators capture the readings on the shafts is usually greater than the distance a face reading can be taken.
- If the machinery is supported in sliding type bearings and the shafts are floating back or forth axially when rotating the shaft-to-capture readings, there is virtually no effect on the accuracy of the readings.

Disadvantages

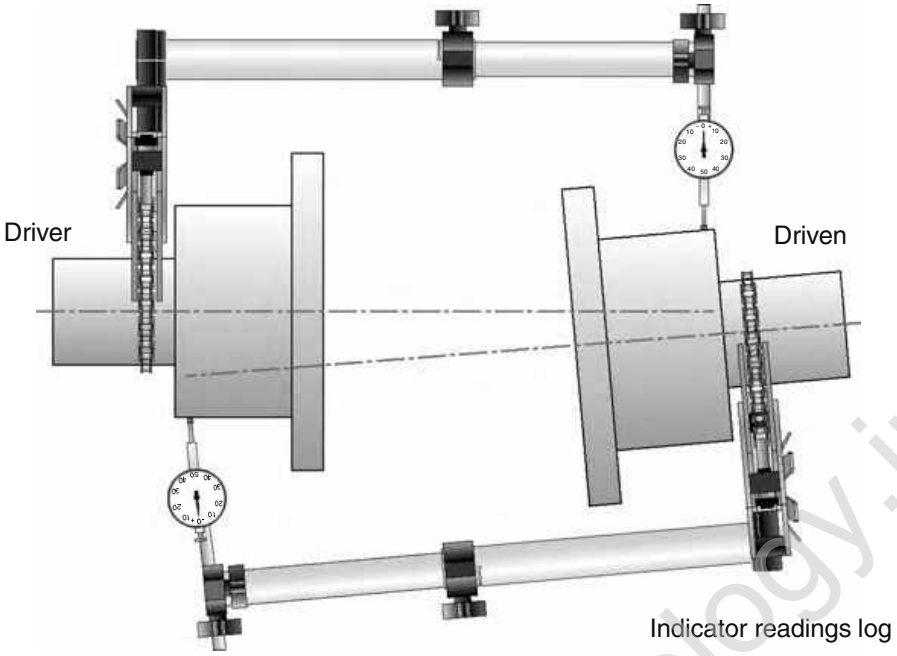
- Both shafts must be rotated.
- Difficult to visualize the positions of the shafts from the dial indicator readings.
- Bracket sag must be measured and compensated for.

10.1 BASIC MATHEMATICAL EQUATIONS FOR THE REVERSE INDICATOR METHOD

There is a mathematical relationship that exists between the size of the machinery (i.e., where the foot bolts are located), where the shaft position measurements are taken, and the shaft

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• Procedure •

1. Attach the alignment bracket(s) firmly to one (both) shaft(s) and position the indicator(s) on the perimeter of the other shaft (or coupling hub).

2. Zero the indicator(s) at the twelve o'clock position.

3. Slowly rotate the shaft and bracket arrangement through 90° intervals stopping at the three, six, and nine o'clock positions. Record each reading (plus or minus).

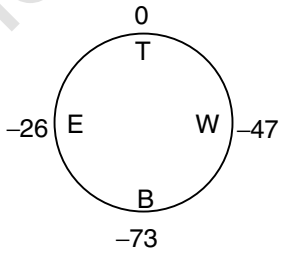
4. Return to the twelve o'clock position to see if the indicator(s) re-zero.

5. Repeat steps 2 through 4 to verify the first set of readings.

6. If one bracket was used, mount the bracket on to the other shaft and repeat steps 1 through 5.

Indicator readings log

Driver



Driven

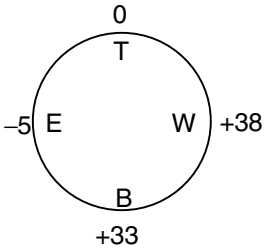


FIGURE 10.1 Reverse indicator method and procedure.

measurements themselves. Once all the measurements have been taken, and the shaft plotted onto the alignment model, the corrective movement solutions can be arrived at.

Figure 10.6 shows the mathematical relationship between the machinery dimensions and the dial indicator readings captured using the reverse indicator technique. The equations will solve for the moves that need to be made to correct the misalignment condition (i.e., bring the shafts into a collinear relationship when off-line) on one or the other machine case. It is an either/or condition. If you decide to keep the driver stationary, you solve for the moves on the driven machine or vice versa. This is often referred to as the "stationary-movable" alignment concept and is not recommended for reasons explained in Chapter 8. Alignment modeling methods and the "overlay line" concept have been in use since the late 1970s.

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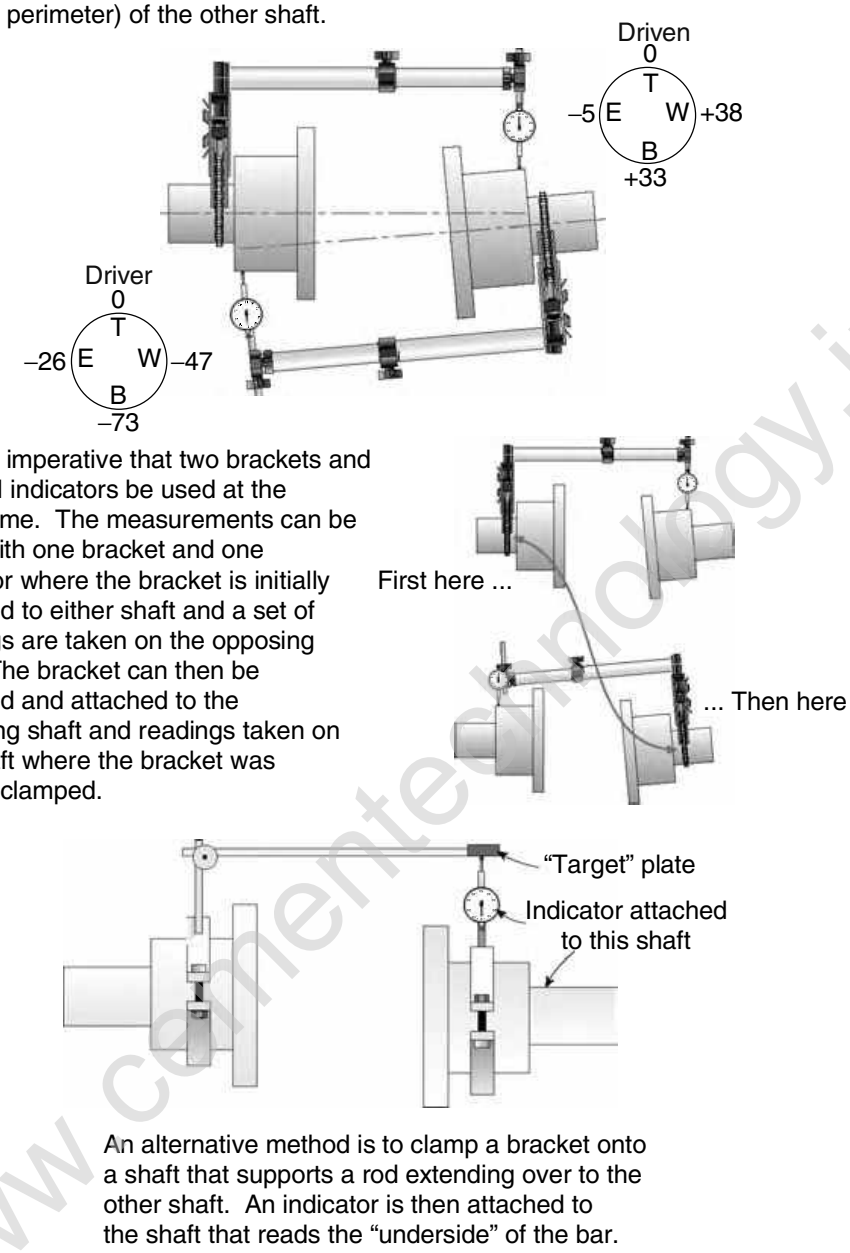


FIGURE 10.2 Reverse indicator setup variations.

10.2 MODELING REVERSE INDICATOR METHOD USING THE "POINT-TO-POINT" TECHNIQUE

Perhaps the easiest modeling technique to learn is the point-to-point reverse indicator modeling method and will therefore be the first one illustrated.

There are eight pieces of information that you need to properly construct the shaft positions using this technique:

1. The distance from the outboard-to-inboard feet (bolting planes) of the first machine.
2. The distance from the inboard bolting plane of the first machine to the point on the shaft where the bracket is being held in place.

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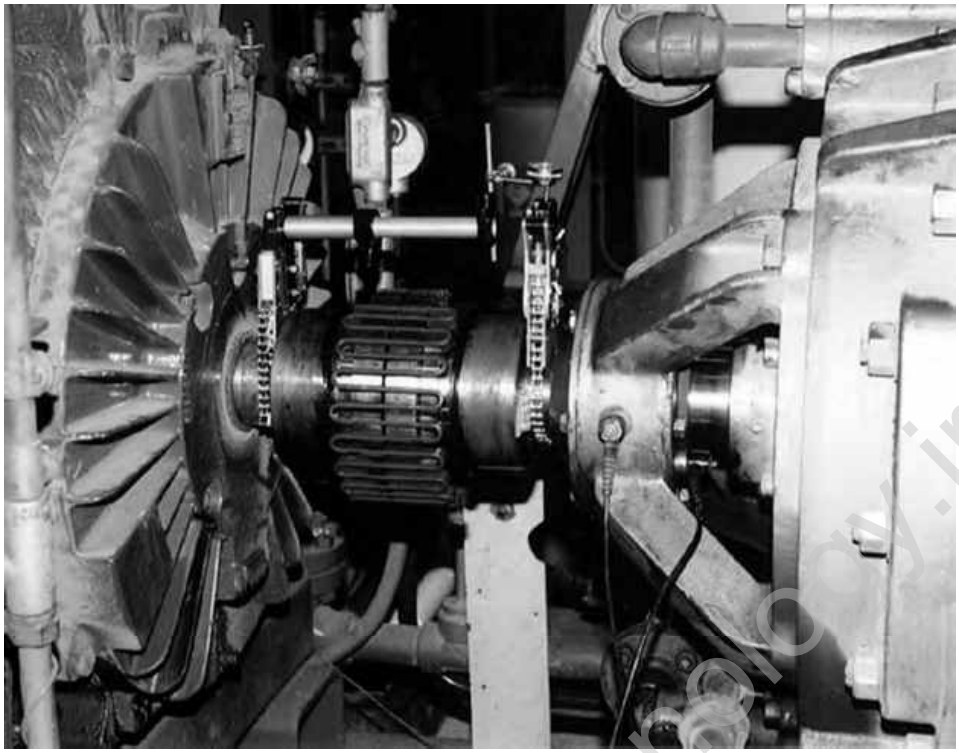


FIGURE 10.3 Reverse indicator technique employed across metal ribbon coupling.

3. The distance from where the bracket is being held in place to the point where the dial indicator is capturing the rim readings on the first machine. Note that this distance could be zero if you are using a symmetrical arrangement where you are clamping and reading at the same points on each shaft.



FIGURE 10.4 Reverse indicator technique employed across small motor and pump drive using miniature brackets.

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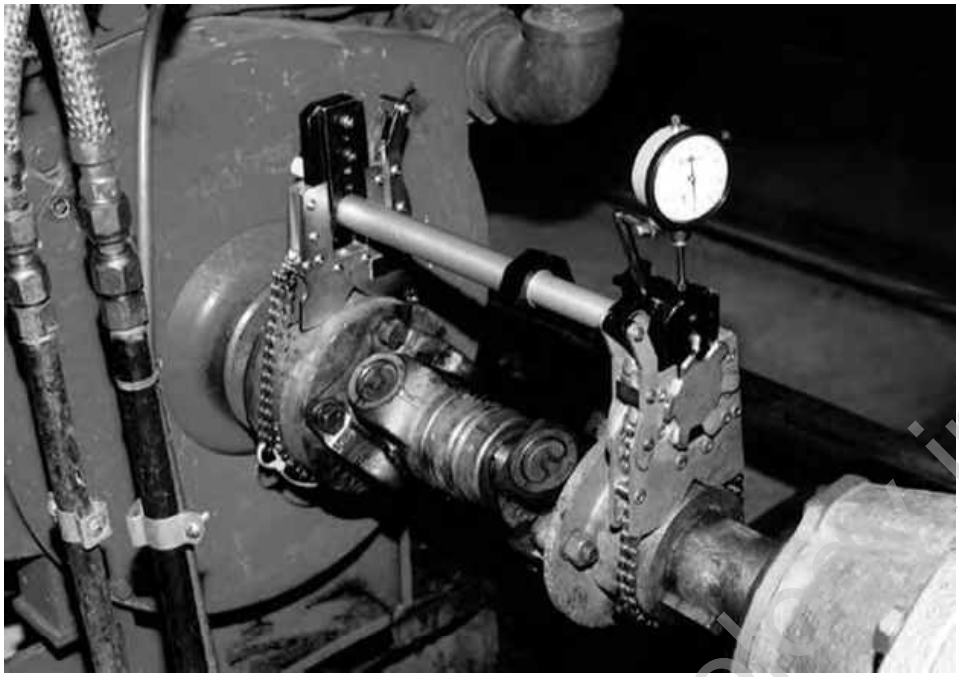


FIGURE 10.5 Reverse indicator technique employed across universal joint.

4. The distance from where the dial indicator is capturing the rim readings on the first machine to the point where the dial indicators are capturing the rim readings on the second machine.

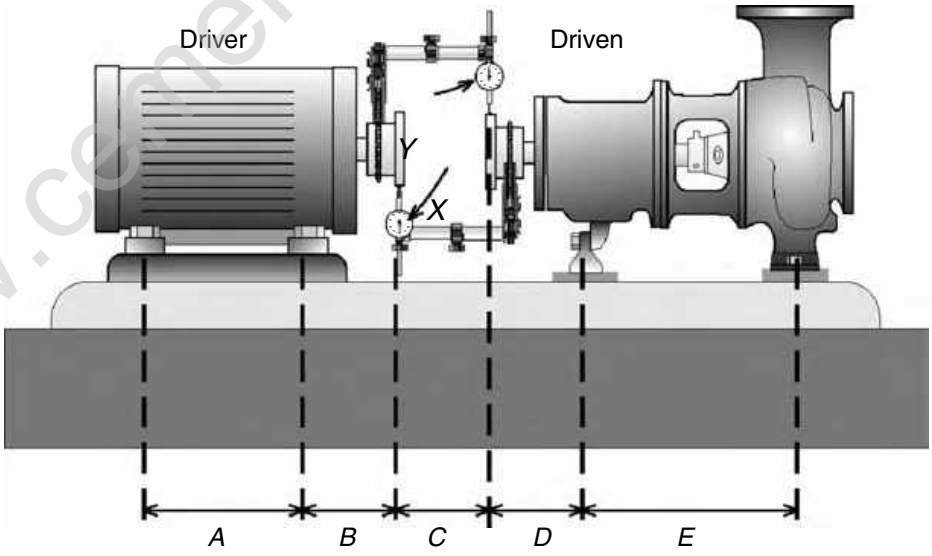
Reverse indicator method mathematics

where:

A, *B*, *C*, *D*, *E* = distances shown (in.)

X = one-half of the driver rim reading difference (from top to bottom or side to side in mils)

Y = one-half of the driven rim reading difference (from top to bottom or side to side in mils)



$$\text{Inboard feet of Driver} = \frac{(B+C)(X+Y)}{C} - (Y)$$

$$\text{Inboard feet of Driven} = \frac{(C+D)(X+Y)}{C} - (X)$$

$$\text{Outboard feet of Driver} = \frac{(A+B+C)(X+Y)}{C} - (Y)$$

$$\text{Outboard feet of Driven} = \frac{(C+D+E)(X+Y)}{C} - (X)$$

FIGURE 10.6 Reverse indicator mathematics for correcting moves on either machine case.

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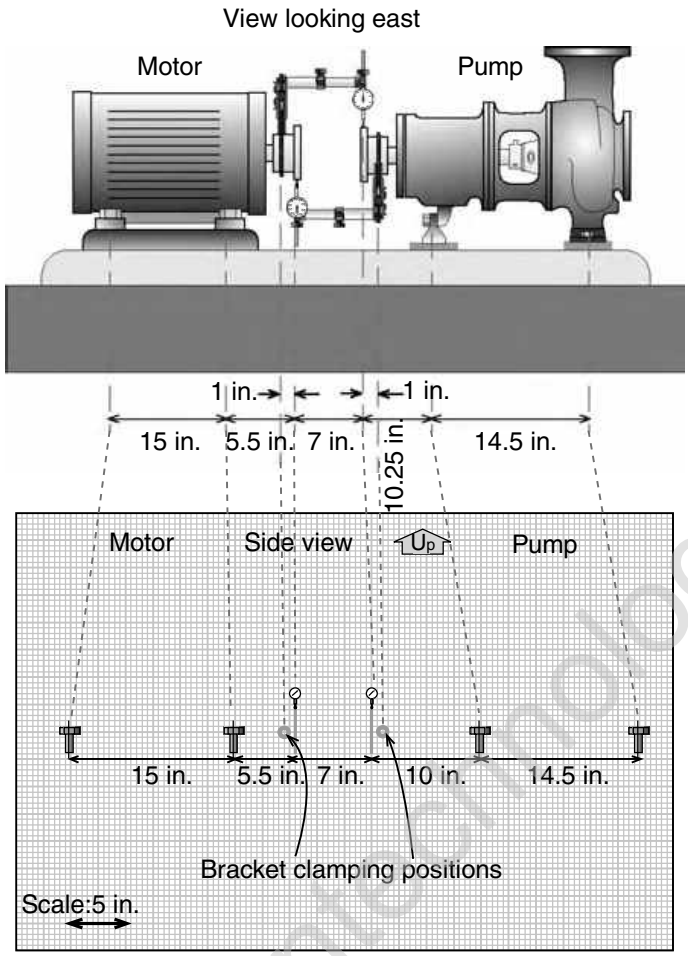


FIGURE 10.7 Dimensional information needed for plotting the reverse indicator readings using the point-to-point plotting method.

5. The distance from where dial indicator is capturing the rim readings on the second machine to the point where the bracket is being held in place. Note that this distance could be zero if you are using a symmetrical arrangement where you are clamping and reading at the same points on each shaft.
6. The distance from where the bracket is being held in place to the inboard bolting plane of the second machine.
7. The distance from the inboard-to-outboard feet (bolting planes) of the second machine.
8. The eight dial indicator readings taken at the top, bottom, and both sides on both shafts after compensating for sag (i.e., what a perfect, ''no sag'' bracket system would have measured).

From the standpoint of geometry, the reverse indicator method measures shaft centerline deviations at two ''slices'' in space at a known distance apart. Figure 10.7 illustrates the core modeling points that we will construct. Accurately scale the distances along the length of the drive train onto the graph centerline as shown in Figure 10.8.

10.3 RIM READINGS ARE ALWAYS TWICE THE OFFSET AMOUNT

Remember, anytime a rim or circumferential reading is taken, the amount measured from one side to the other side of the shaft (180° of rotation) is twice the amount of the actual distance

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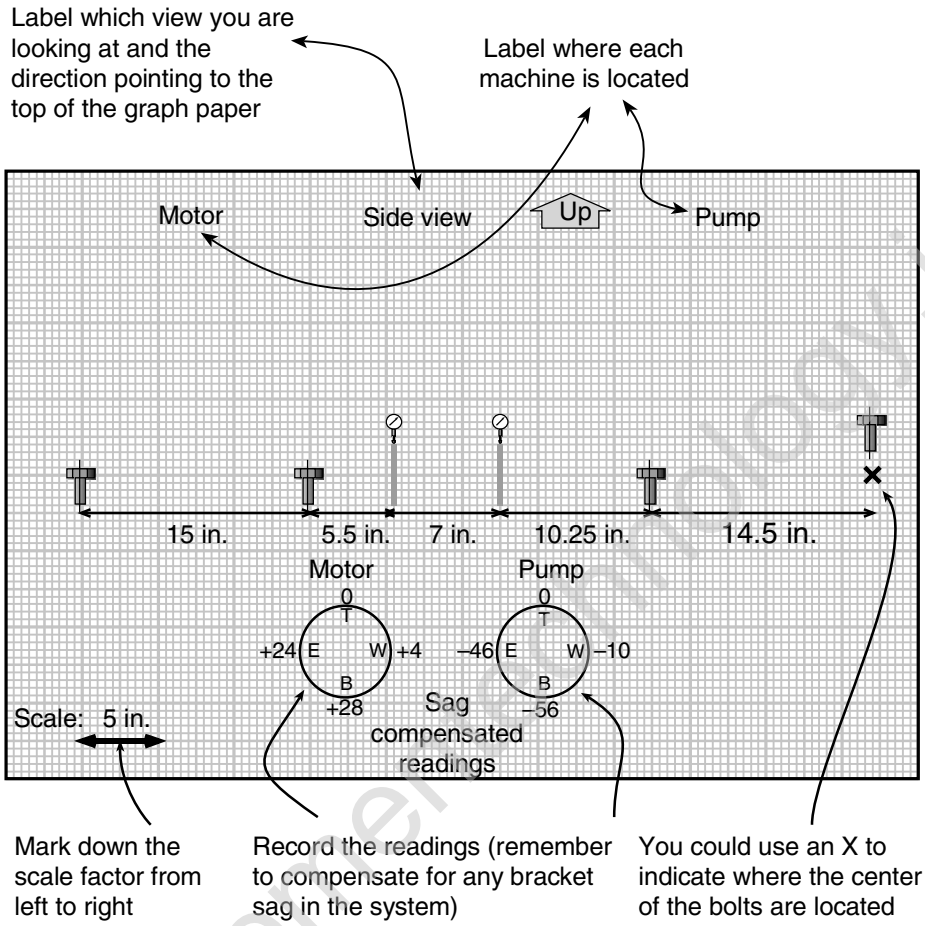


FIGURE 10.8 Scaling the dimensions of the drive system onto the graph.

between the centerlines of rotation at that point. Refer to Figure 6.44 and Figure 6.45 to understand why this happens.

The procedure for plotting the point-to-point reverse indicator technique is as follows:

1. Start with the top to bottom or side-to-side dial indicator readings on the shaft where the largest top to bottom or side-to-side reading occurred (this will help you pick the best scaling factor for the entire graph).
2. At the intersection of the graph centerline and the point where the dial indicator has captured the largest reading, plot a point above or below this intersection one-half of the top to bottom or side-to-side dial indicator reading. If the bottom (or side) reading was negative, place a point half of the bottom (or side) reading from the graph centerline toward the top of the graph. If the bottom (or side) reading was positive, place a point half of the bottom (or side) readings from the graph centerline toward the bottom of the graph (the same as in the point-to-point modeling techniques). Lay a straightedge from the point on the graph centerline where the bracket was held through the point on the graph where the dial indicator captured the

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To select the appropriate up and down scale factor, start with the shaft that had the larger of the two bottom readings. In this example, it is the pump shaft. Pick an up and down scale factor that will keep the entire length of the pump shaft within the boundaries of the graph paper. Usually this scale factor will keep the motor shaft (which has the smaller bottom reading) within the boundaries of the graph paper also.

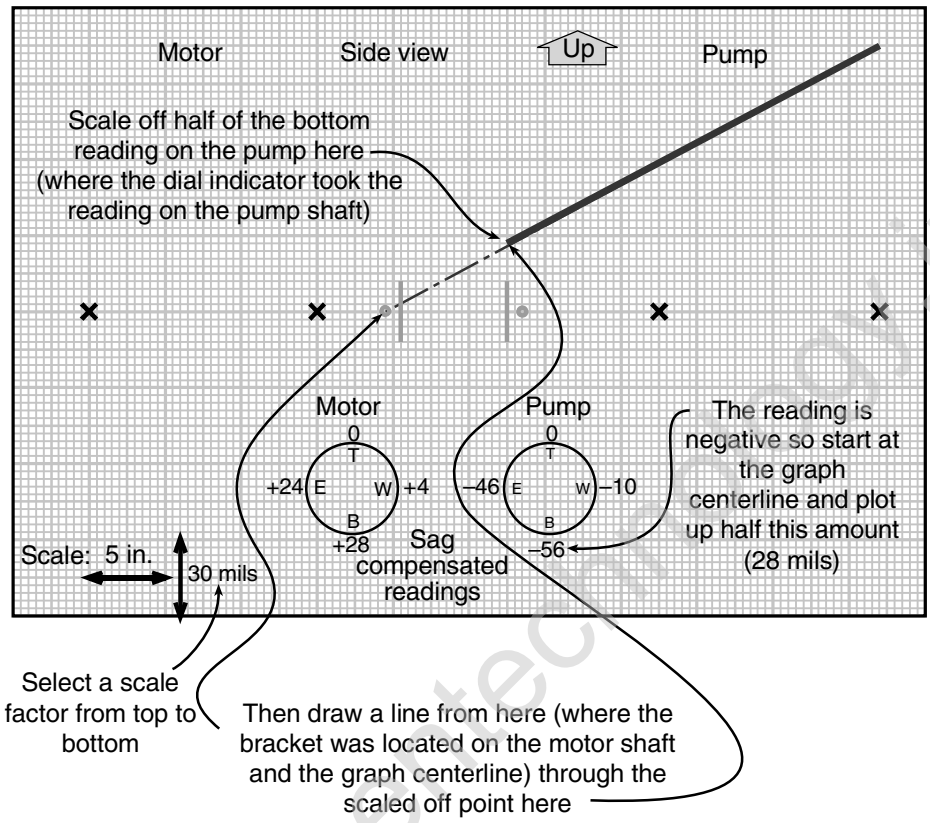


FIGURE 10.9 Plotting a pump shaft onto the side view alignment model.

reading. Draw a line from the point on the graph where the dial indicator captured the reading to the outboard end of that shaft. Remember, whatever shaft the dial indicator has captured the readings on, that is the shaft that will be drawn on the graph paper. Figure 10.9 shows an example of plotting a pump shaft onto the side view alignment model.

3. Next, at the intersection of the graph centerline and the point where the dial indicator has captured the smallest reading, plot a point above or below this intersection one-half of the top to bottom or side-to-side dial indicator reading. If the bottom (or side) reading was negative, place a point half of the bottom (or side) reading from the graph centerline toward the top of the graph. If the bottom (or side) reading was positive, place a point half of the bottom (or side) readings from the graph centerline toward the bottom of the graph (the same as in the point to point modeling techniques). Lay a straightedge from the point on the graph centerline where the bracket was held through the point on the graph where the dial indicator captured the reading. Draw a line from the point on the graph where the dial indicator captured the reading to the outboard end of that shaft. Figure 10.10 shows an example of plotting a motor shaft onto the side view alignment model.

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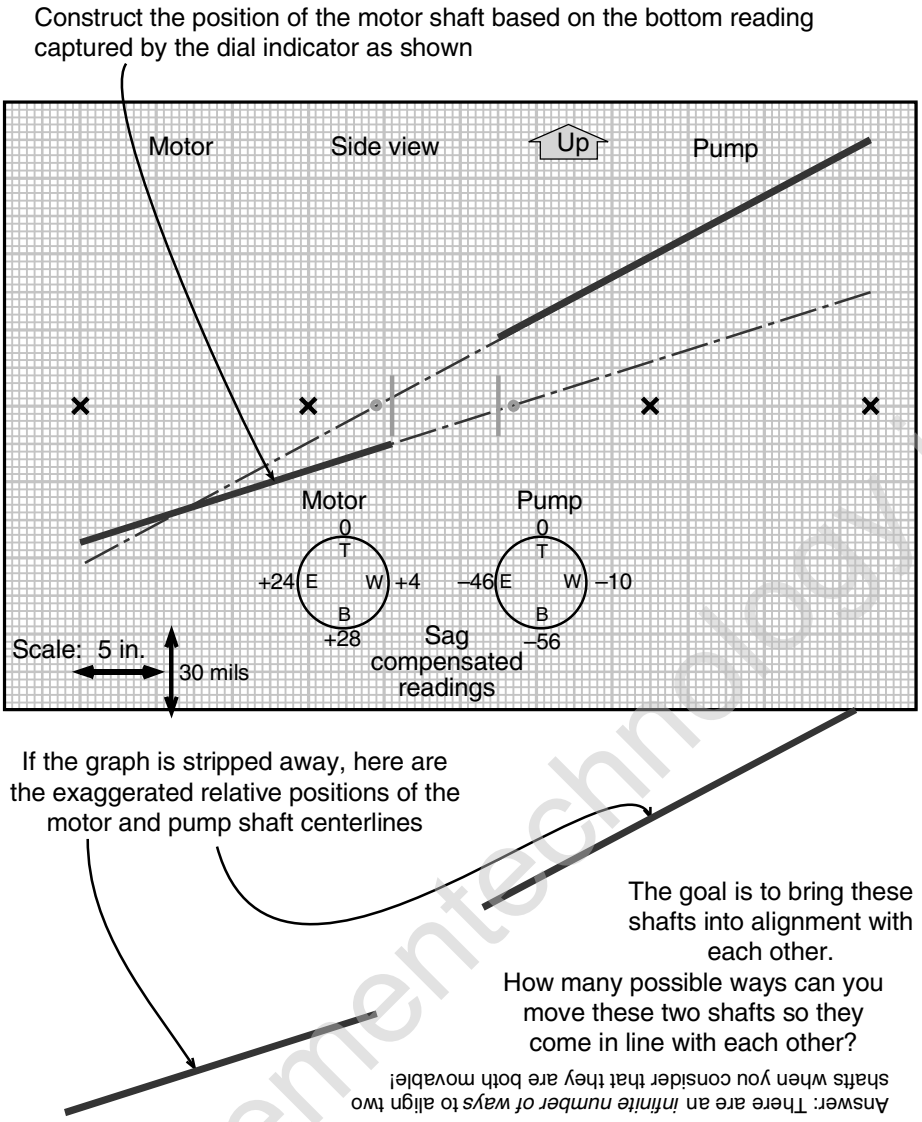


FIGURE 10.10 Plotting a motor shaft onto the side view alignment model.

Notice that there is a consistency to this plotting technique. If the top to bottom or side-to-side dial indicator reading is negative, plot half of the reading toward the top of the graph paper, for either shaft. If the top to bottom or side-to-side dial indicator reading is positive, plot half of the reading toward the bottom of the graph paper, for either shaft.

The process for plotting the shaft in the top view is the same as it is in the side view. As discussed in Chapter 8, one of the cardinal alignment modeling rules is to zero the indicator on the side that is pointing to the top of your graph paper. Refer to Figure 10.7 where it states "view looking east." Therefore, when looking at our drive system from above (i.e., the top view), the direction pointing to the top of the graph paper must be east. Figure 10.11 shows how the pump shaft is plotted in the top view and how the east side readings were zeroed to extract the west side readings on each shaft from the complete set of reverse indicator measurements. Figure 10.12 shows how the motor shaft is plotted in the top view.

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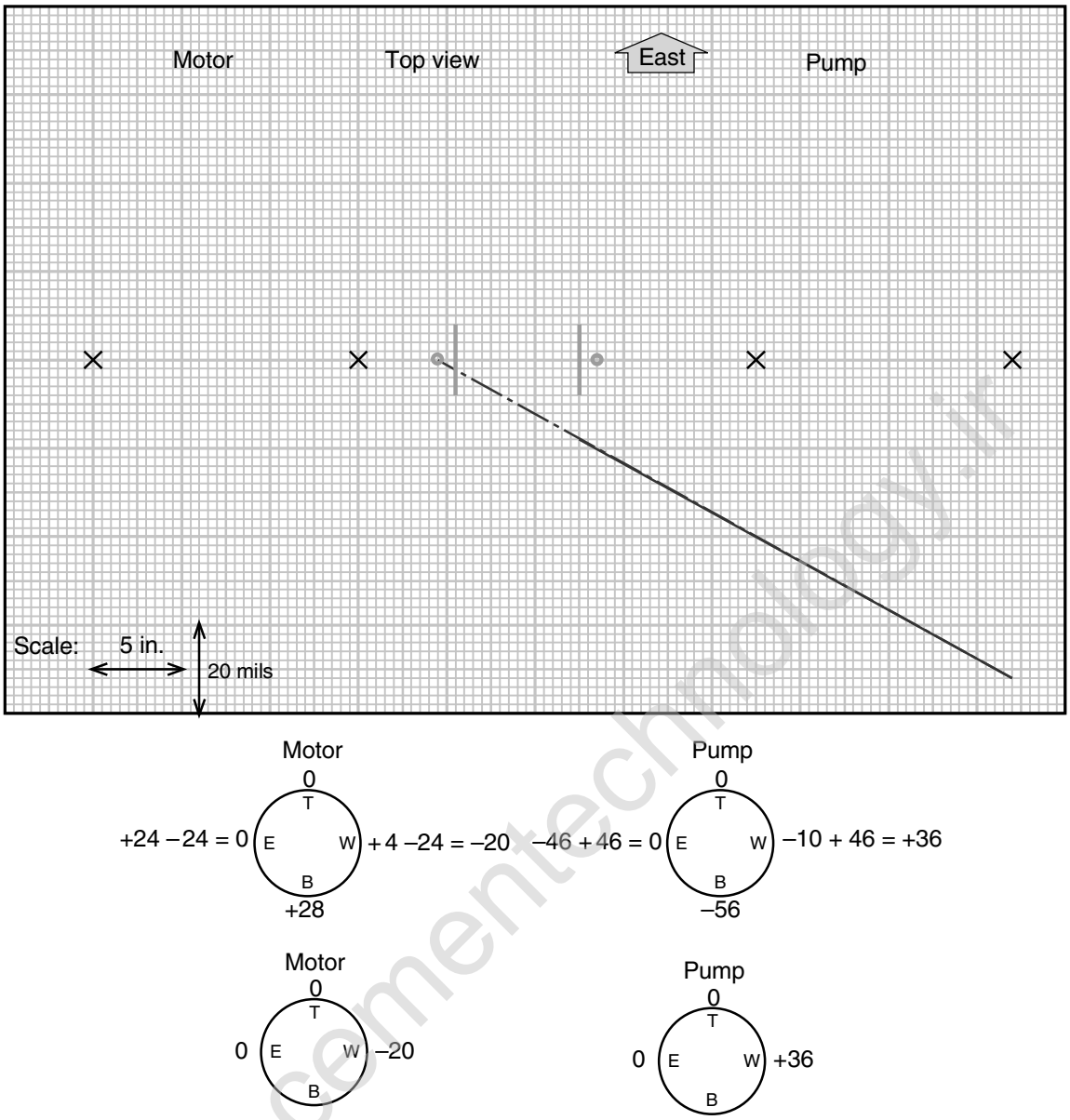


FIGURE 10.11 Plotting a pump shaft onto the top view alignment model.

10.4 MODELING THE REVERSE INDICATOR METHOD USING THE LINE-TO-POINT TECHNIQUE

There is an alternative method to graphing or modeling reverse indicator readings. There are two advantages of this technique as opposed to the point-to-point method:

- Somewhat easier to model multiple element drive trains where reverse indicator readings were captured at two or more flexible couplings.
- Regardless of whether you have an asymmetrical or symmetrical bracket arrangement, the points where the brackets are being clamped to the shaft are not relevant, only the points where the dial indicator readings are being captured are required.

There are six pieces of information that you need to properly construct the shaft positions using this technique.

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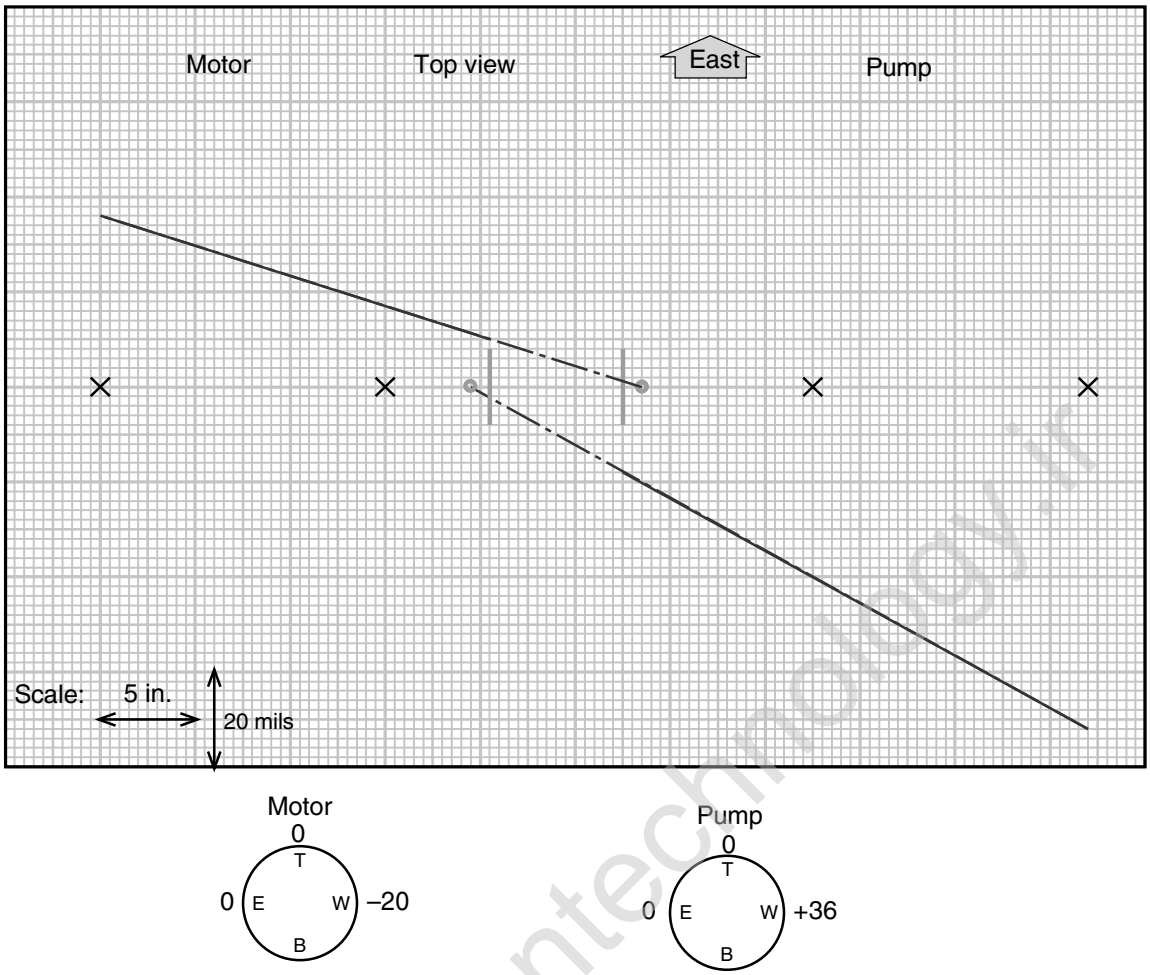


FIGURE 10.12 Plotting a motor shaft onto the top view alignment model.

1. The distance from the outboard-to-inboard feet (bolting planes) of the first machine.
2. The distance from the inboard bolting plane of the first machine to the point on the shaft where the dial indicator is capturing the rim readings on the first machine.
3. The distance from where the dial indicator is capturing the rim readings on the first machine to the point where the dial indicators are capturing the rim readings on the second machine.
4. The distance from where the dial indicator is capturing the rim readings on the second machine to the inboard bolting plane of the second machine.
5. The distance from the inboard-to-outboard feet (bolting planes) of the second machine.
6. The eight dial indicator readings taken at the top, bottom, and both sides on both shafts after compensating for sag (i.e., what a perfect, "no sag" bracket system would have measured).

Accurately scale the distances along the length of the drive train onto the graph centerline as shown in Figure 10.13.

The procedure for plotting the line-to-point reverse indicator technique is as follows:

1. Select one of the two machinery shafts and draw one of those shafts on top of the graph centerline. Figure 10.14 shows an example where the motor shaft was initially placed on

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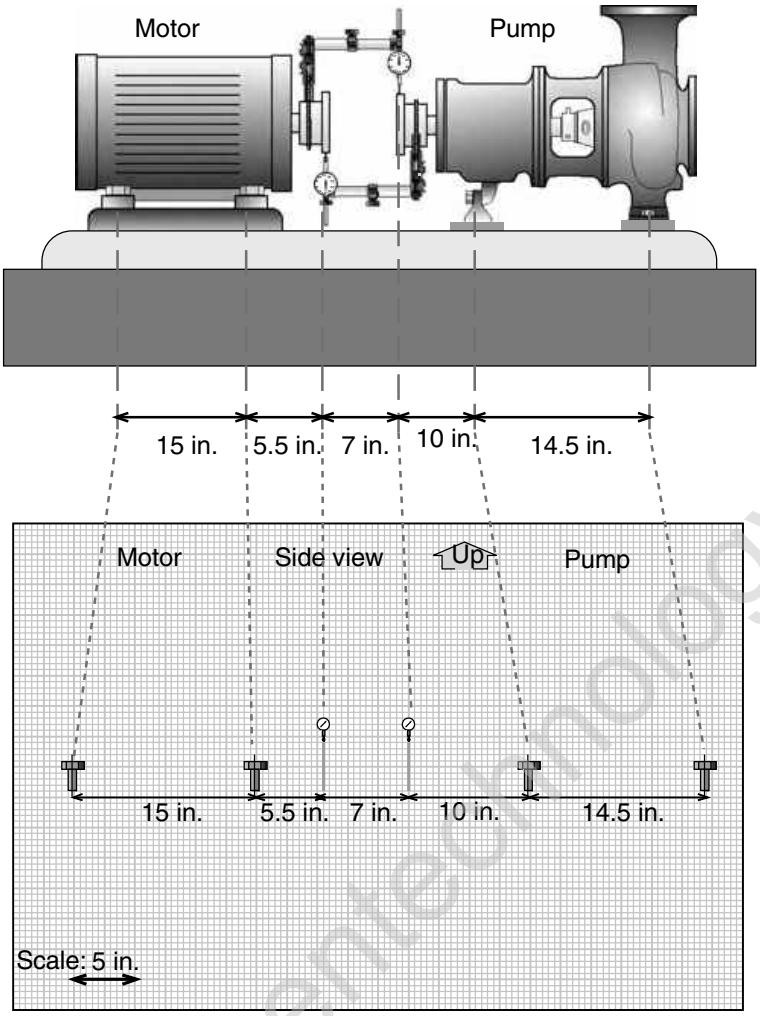


FIGURE 10.13 Dimensional information needed for plotting reverse indicator readings using the line-to-point plotting method.

the graph paper centerline and the pump shaft position was plotted from the reverse indicator measurements. Figure 10.15 shows an example where the pump shaft was initially placed on the graph paper centerline and the motor shaft position was plotted from the same reverse indicator measurements.

2. Start with the top to bottom or side-to-side dial indicator readings on the other shaft (i.e., the one you did not draw on the graph centerline).
3. Plot the other shaft centerline position by starting at the intersection of the graph centerline and the point where the dial indicator was capturing the readings on the other shaft. If the bottom (or side) reading was negative, place a point half of the bottom (or side) readings from the graph centerline toward the top of the graph. If the bottom (or side) reading was positive, place a point half of the bottom (or side) readings from the graph centerline toward the bottom of the graph (the same as in the point-to-point modeling techniques). Do not draw any lines yet!
4. Next, start at the intersection of the graph centerline and the point where the dial indicator was capturing the readings on the shaft that was drawn on top of the graph centerline. If the bottom (or side) reading was negative, place a point half of the bottom (or side) readings from the graph centerline toward the bottom of the graph. If the bottom (or side) reading was positive, place a point half of the bottom (or side) readings

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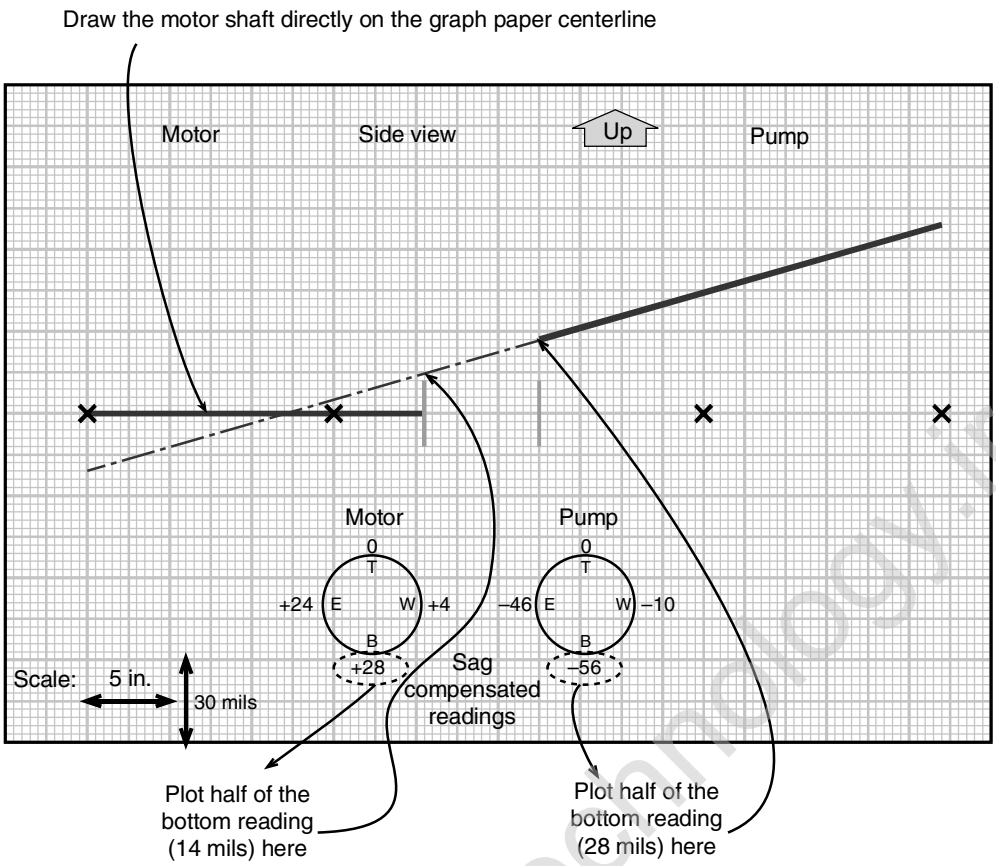


FIGURE 10.14 Side view example where the motor shaft was initially placed on the graph paper centerline and the pump shaft position was then plotted.

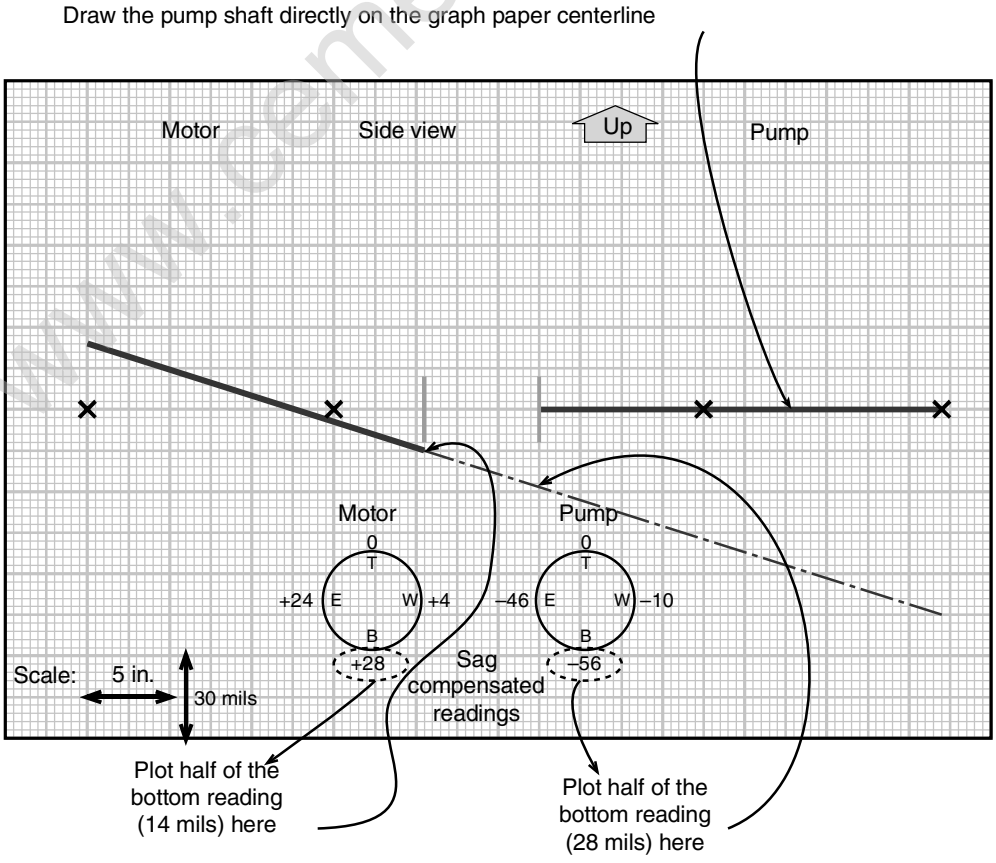


FIGURE 10.15 Side view example where the pump shaft was initially placed on the graph paper centerline and the motor shaft position was then plotted.

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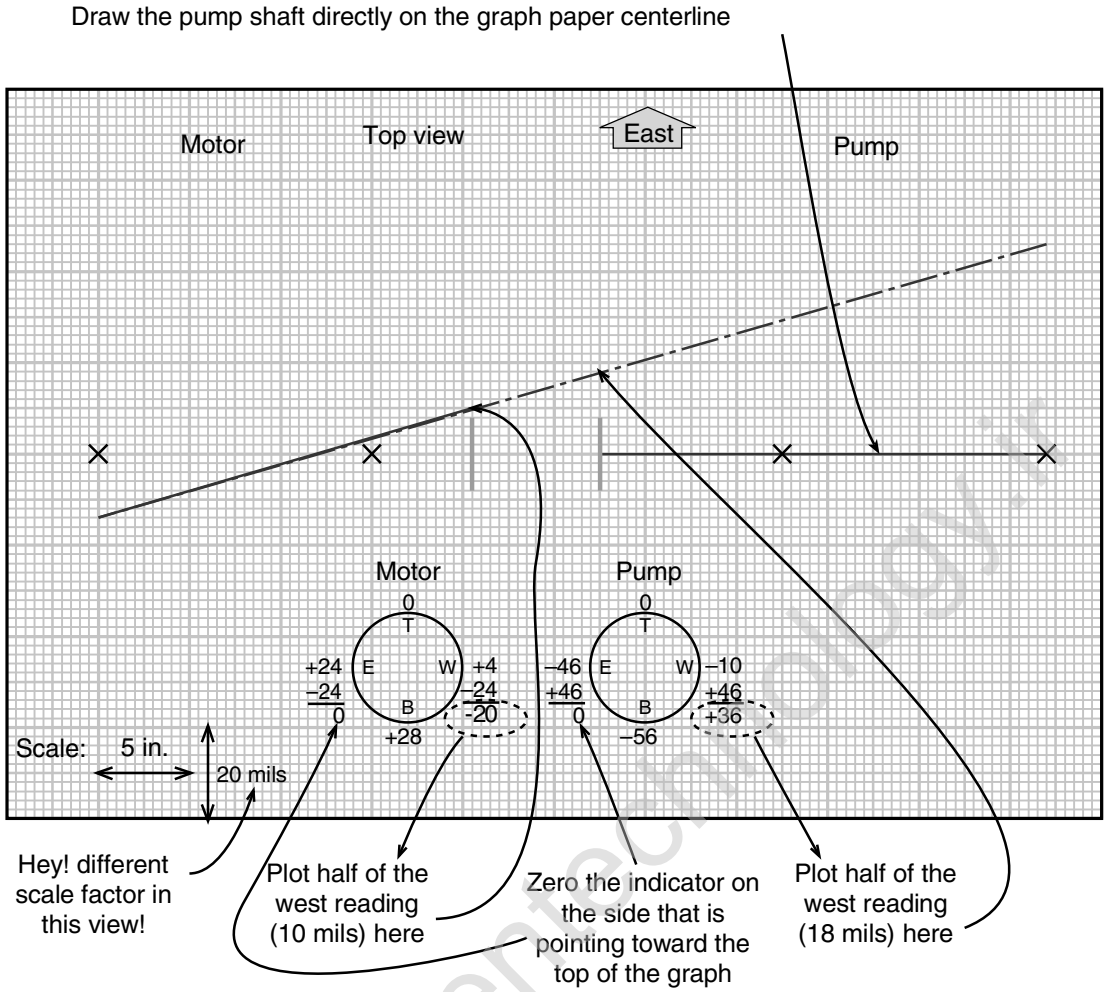


FIGURE 10.16 Top view example where the pump shaft was initially placed on the graph paper centerline and the motor shaft position was then plotted.

from the graph centerline toward the top of the graph (opposite of the point-to-point modeling technique).

5. These two points marked on the graph at the dial indicator reading points define the line of sight (i.e., the centerline of rotation) of the other shaft. Draw a straight line through these two points from the coupling end to the outboard end of the other shaft.

To make your alignment corrections, refer to the Section 8.4.6 and Section 8.4.7.

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11 Face and Rim Methods

Perhaps the first dial indicator technique used to align rotating machinery shafts is the face and rim or face–peripheral method shown in Figure 11.1. It is not entirely clear who initially used this technique or when and where it was employed to align rotating machinery shafts but this method is frequently referred in machinery installation manuals and coupling installation instructions and is still practiced by personnel who align machinery. It is similar to the way machinists center and square work pieces in lathes and mills and undoubtedly came from machining practices during the dawn of the industrial revolution.

As shown in Figure 11.2, the face readings can be taken on either side of the coupling hub (or an object affixed to the other shaft). The accuracy of this method is directly related to the diameter the face readings are taken on as demonstrated in Figure 11.3. The larger the diameter of the face reading sweep, the more accurate this method becomes. Assuming that both shafts can be rotated, the face diameter can be increased by attaching an object to the one shaft and the face indicator placed against that object as shown in Figure 11.4.

Advantages

1. This is a good technique to use in situations where one of the machinery shafts cannot be rotated or it would be difficult to rotate one of the machinery shafts (see also Chapter 12).
2. Many people who use this method understand that the rim (or diametral surface) dial indicator shows centerline offset or parallel misalignment and the face indicator indicates that an angular misalignment condition is present.
3. This is a good method to use when the face readings can be taken on a fairly large diameter (typically 8 in. or greater). This method begins to approach the accuracy of the reverse indicator technique whenever the diameter of the face readings equals or exceeds the span from the bracket location to the point where the rim indicator readings are being captured in the reverse indicator method.

Disadvantages

1. Not as accurate as the reverse indicator method if both shafts can be rotated and particularly if the face measurements are taken on diameters less than 8 in.
2. If the machinery shafts are supported in sliding (plain or sleeve) bearings, it is very easy to axially float the shafts toward or away from each other when rotating the shafts results in bad or inaccurate face readings (see Section 6.10).
3. Bracket sag must be measured and compensated for.

11.1 MATHEMATICAL RELATIONSHIP IN MACHINERY ALIGNMENT

Figure 11.8 shows the mathematical relationship between the machinery dimensions and the dial indicator readings captured using the face–rim method. The equations will solve for the moves that need to be made to correct the misalignment condition (i.e., bring the shafts into a

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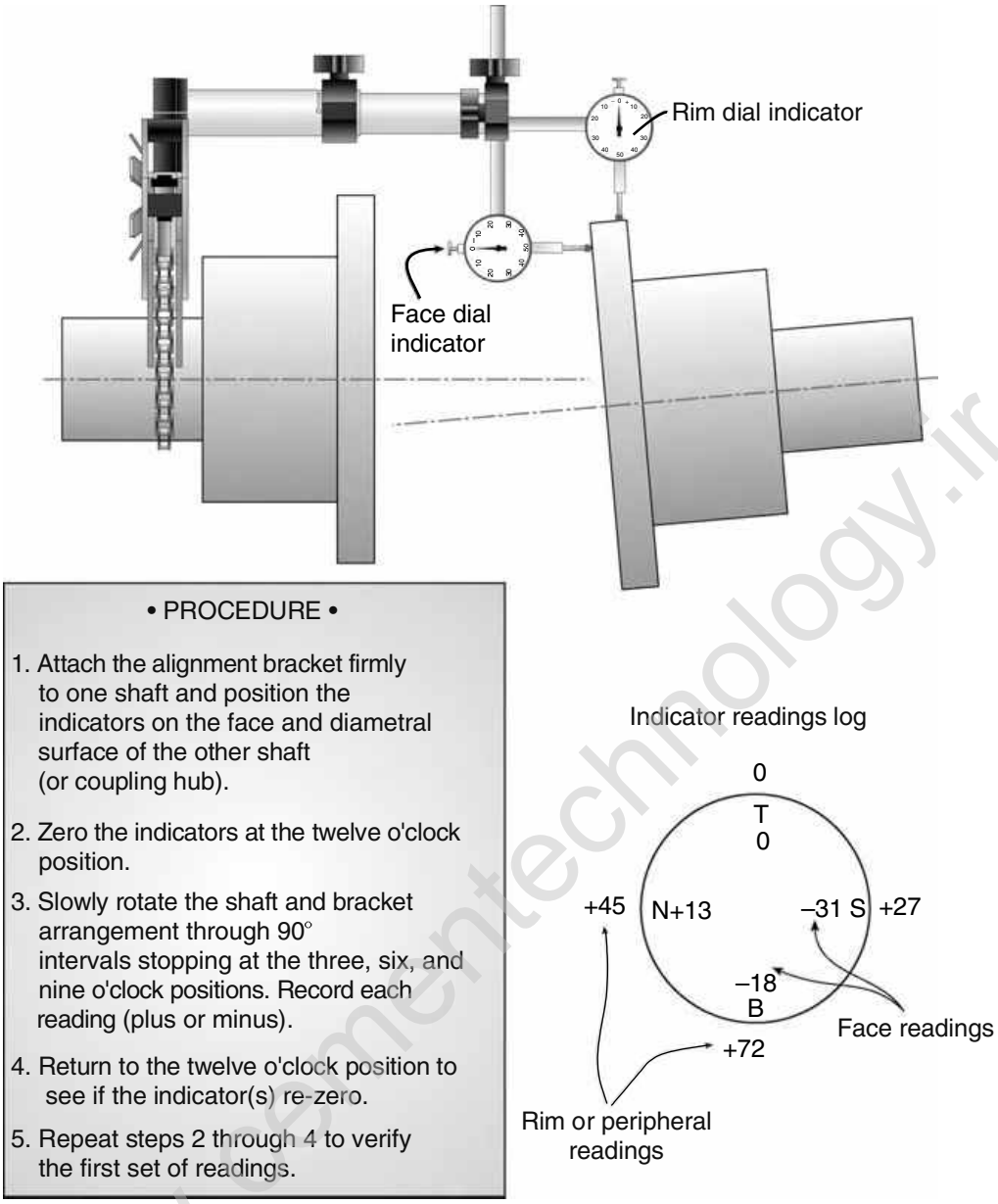


FIGURE 11.1 Face and rim method and procedure.

collinear relationship when off-line) on one or the other machine case. It is an either–or condition. If you decide to keep the driver stationary, you solve for the moves on the driven machine or vice versa.

11.2 SIXTEEN-POINT METHOD

A method similar to the face–rim method called the 16-point method is frequently used on rotating machinery connected together by rigid rather than flexible couplings. The general procedure is illustrated in Figure 11.9.

This method is typically used where one shaft is supported in two bearings and the other shaft is supported in one bearing on the outboard end. The coupling flanges have a recessed (rabbeted) fit. The assumption made when performing this technique is that there is only pure angular alignment present (i.e., no centerline offset) and that the flange faces are

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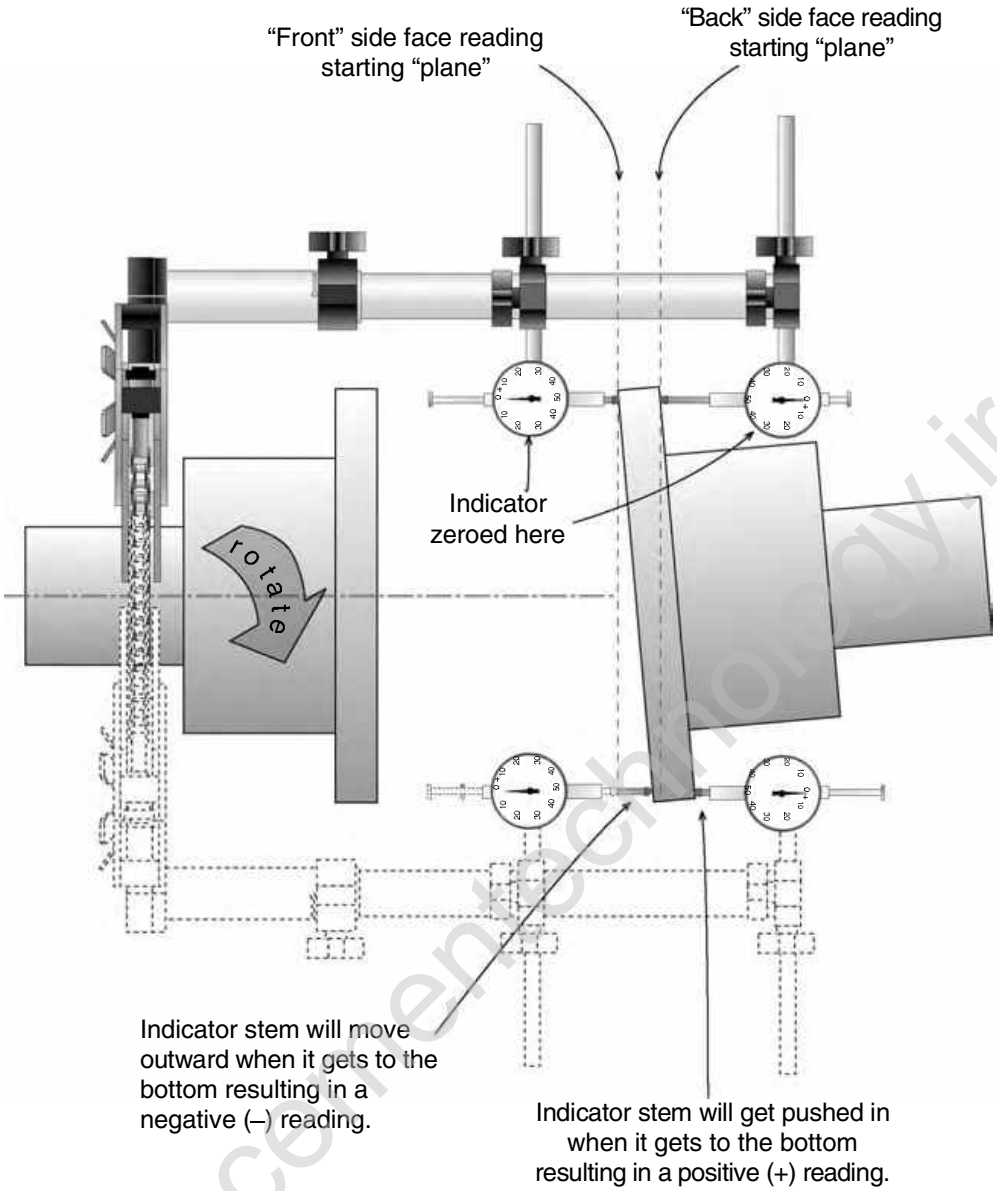


FIGURE 11.2 Face readings can be taken on the front or back sides.

perpendicular to the centerlines of rotation. The flange bolts are loosened, the shafts separated just slightly, insuring that the flange faces are still indexed in the recess, and a series of face readings are taken at four points around the flange faces at the twelve, three, six, and nine o'clock positions. No rim readings are taken.

11.3 TWENTY-POINT METHOD

The 20-point method is also frequently used on rotating machinery connected together by rigid rather than flexible couplings. The general procedure is illustrated in Figure 11.10.

It is typically used where both shafts are supported in two bearings. The flange bolts are loosened, the shafts separated slightly, and a series of face readings are taken at four points around the flange faces at the twelve, three, six, and nine o'clock positions along with a rim (circumferential) reading typically taken with a dial indicator. For all practical purposes this is the face and rim technique explained earlier. Rather than measure the face readings

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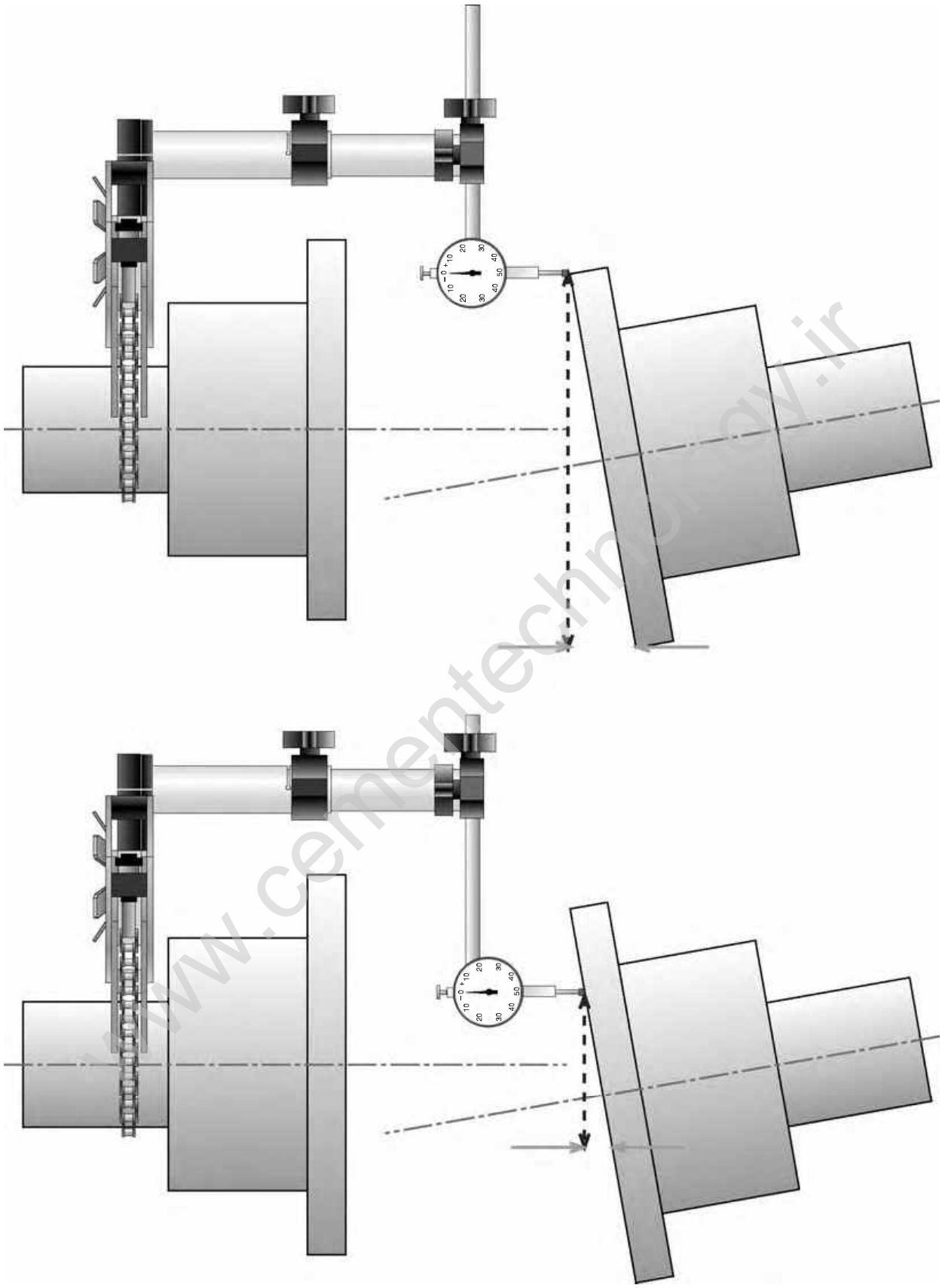


FIGURE 11.3 Taking face readings on different diameters will result in different readings even though the shafts are in the same angular position.

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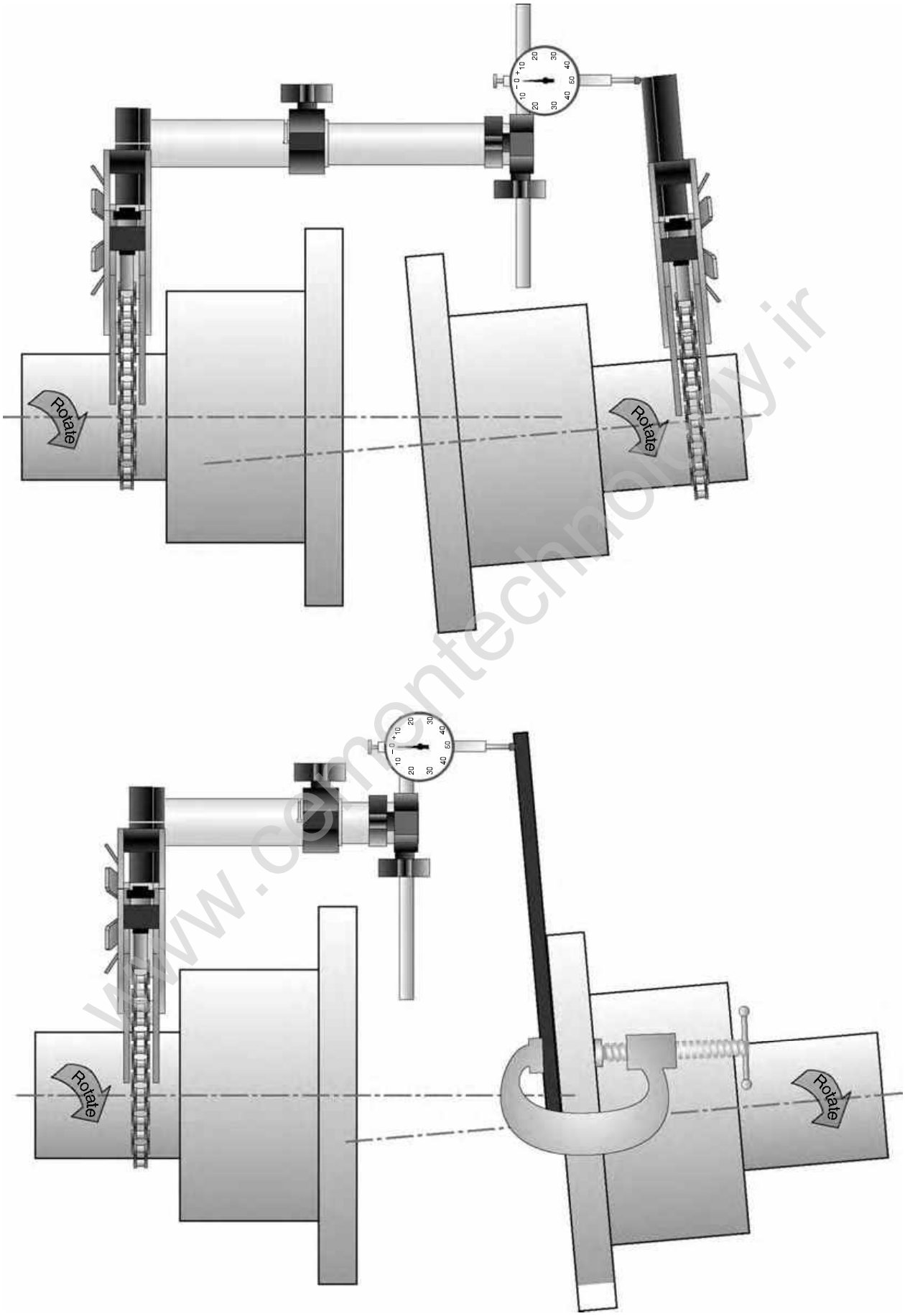


FIGURE 11.4 Face readings can be captured on any surface or device rigidly attached to a shaft (assuming the shafts are rotated together).

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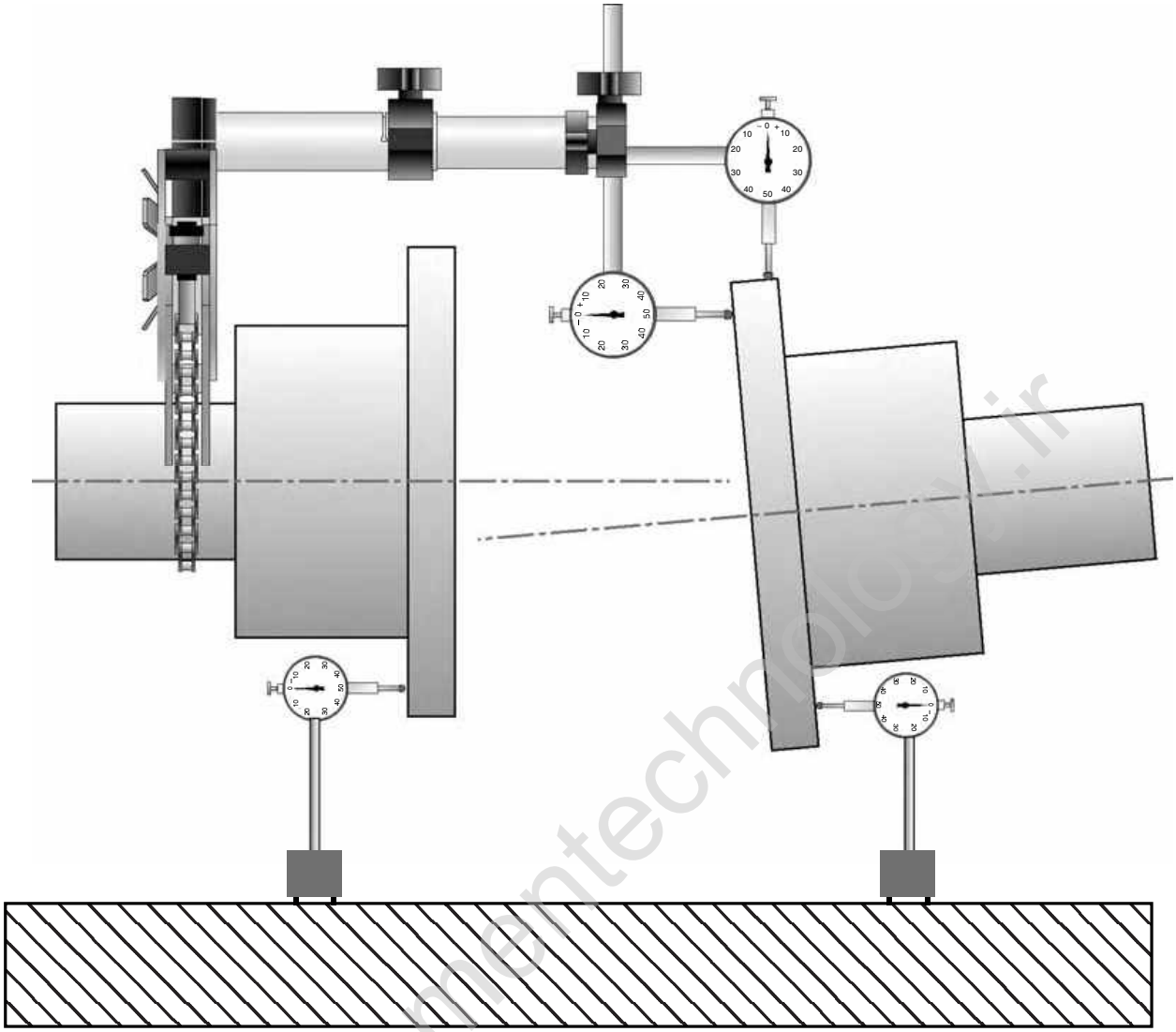


FIGURE 11.5 If the shafts are moving axially during the face measurement sweep, indicators can be positioned to observe the axial movement of each shaft to correct each face measurement.

with a dial indicator, feeler gauge, snap gauge, or an inside micrometer is used to take the face measurements.

11.4 PROBLEMS WITH TAKING FACE READINGS

When performing any method where face readings are taken, measurement inaccuracies and inconsistencies can occur if the shafts that are rotated, move toward or away from each other, during the process of capturing the measurements. This can occur very easily if the shafts are supported in sliding or journal-type bearings.

The first indication that this is occurring is if the dial indicator (or any measurement sensor) does not return to zero after a 360° sweep is made. It is therefore suggested that at least two complete sets of readings are taken to see if there is repeatability in the measurements at each 90° location. If the measurements do not repeat within 1–2 mils after two sweeps are made and you suspect that the shafts are indeed moving toward or away from each other, then you can try one of the following three procedures to improve the accuracy of the measurements.

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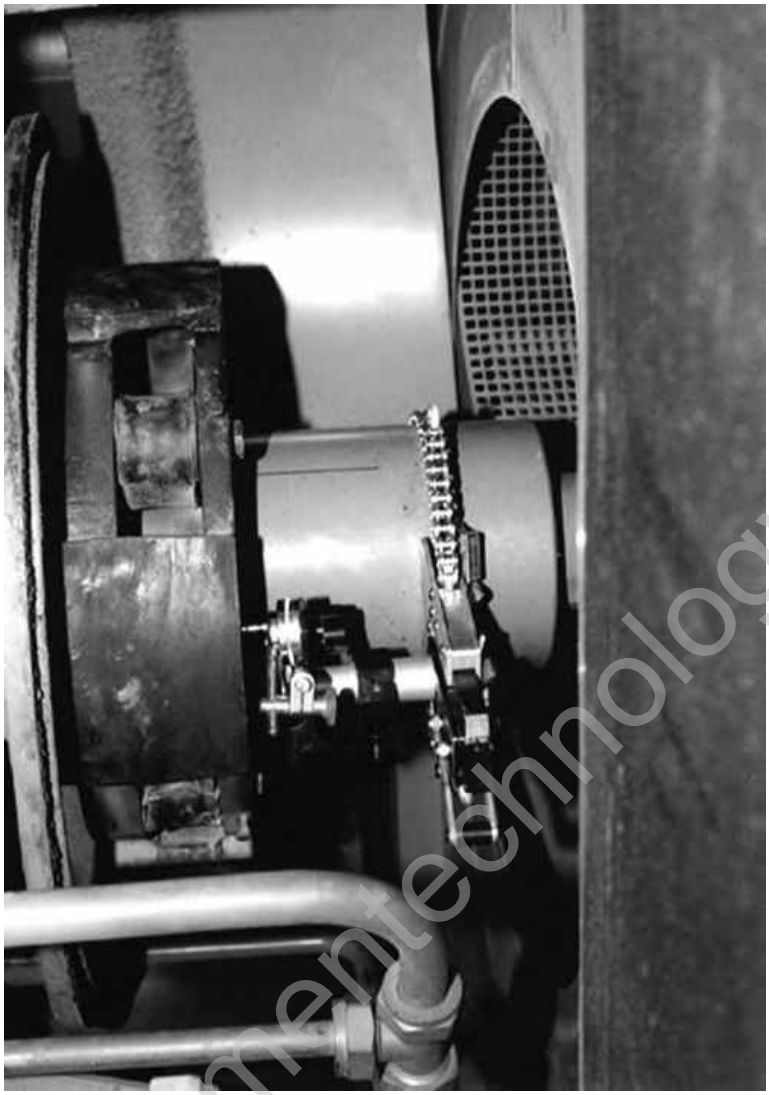


FIGURE 11.6 Face measurements being taken on compressor shaft.

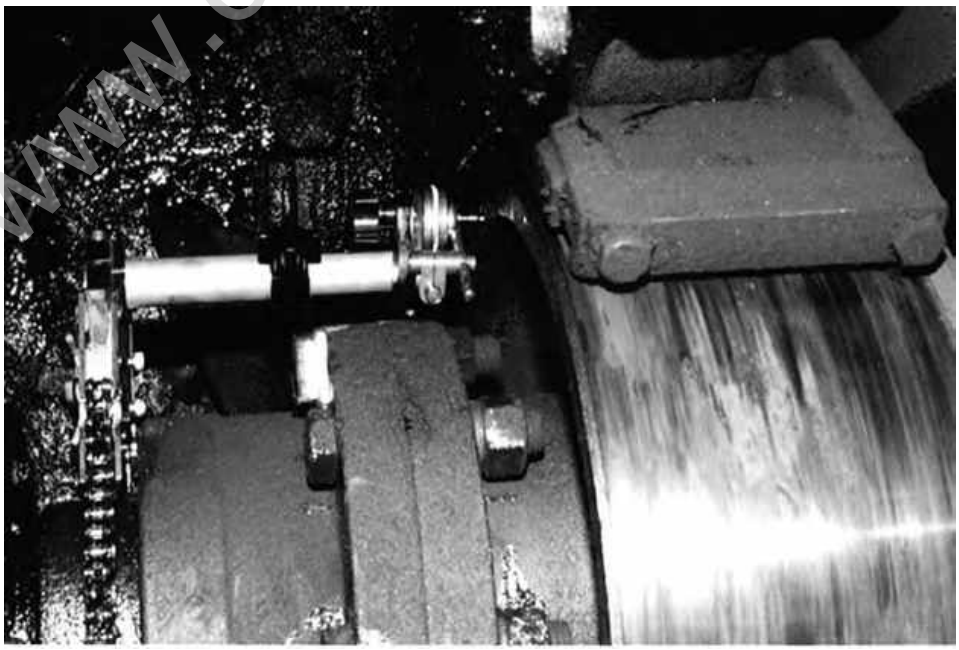


FIGURE 11.7 Face measurement being taken on brake drum.

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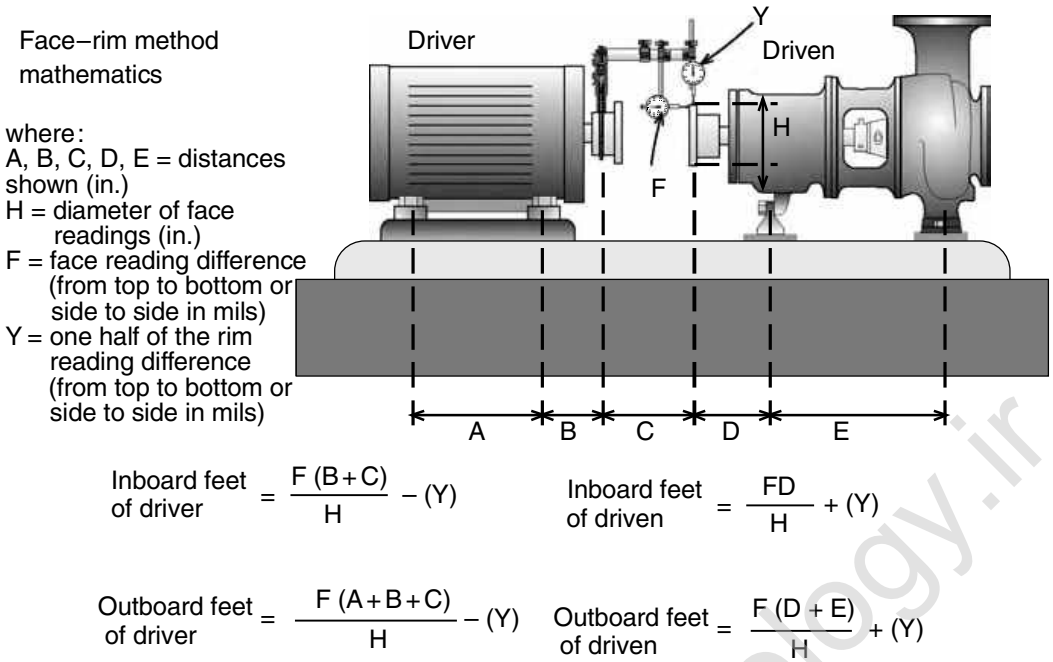


FIGURE 11.8 Face–rim mathematics for correcting moves on either machine case.

11.4.1 Preset the Axial Position

After the measurement fixtures are attached to the shaft and the dial indicator (or whatever measurement sensor is used) is positioned at the twelve o'clock position, before you zero the indicator, either push the shafts apart or draw them together to seat them against their thrust bearings, then zero the indicator. When each 90° rotation is made during the measurement process again, push the shafts apart (or draw them together if that is what you did initially) to seat them against their thrust bearings, then observe and record your measurement.

11.4.2 Compensate for Axial Movement with Stationary Indicators

Figure 11.11 shows an alignment fixture attached to the shafts with an indicator taking a face reading. There are two more indicators attached to magnetic bases (or any stationary reference device) observing for axial movement of each shaft. As the shafts are rotated through their 90° arcs, measurements are observed and recorded on all three indicators. Figure 11.12 shows an example of how to compensate for the axial movement observed.

11.4.3 Compensate for Axial Movement with Rotating Indicators

Figure 11.13 shows an alignment fixture attached to the shafts with two indicators taking face readings 180° apart. During rotation, if the shafts float back or forth, both indicators are affected proportionately. By taking half the algebraic difference between both sets of readings through a 180° rotation, the axial float that occurred will be canceled out. Figure 11.14 shows an example of how to compensate for the axial movement observed.

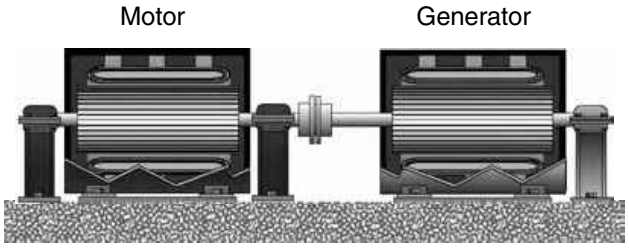
11.5 MODELING THE FACE AND RIM METHOD

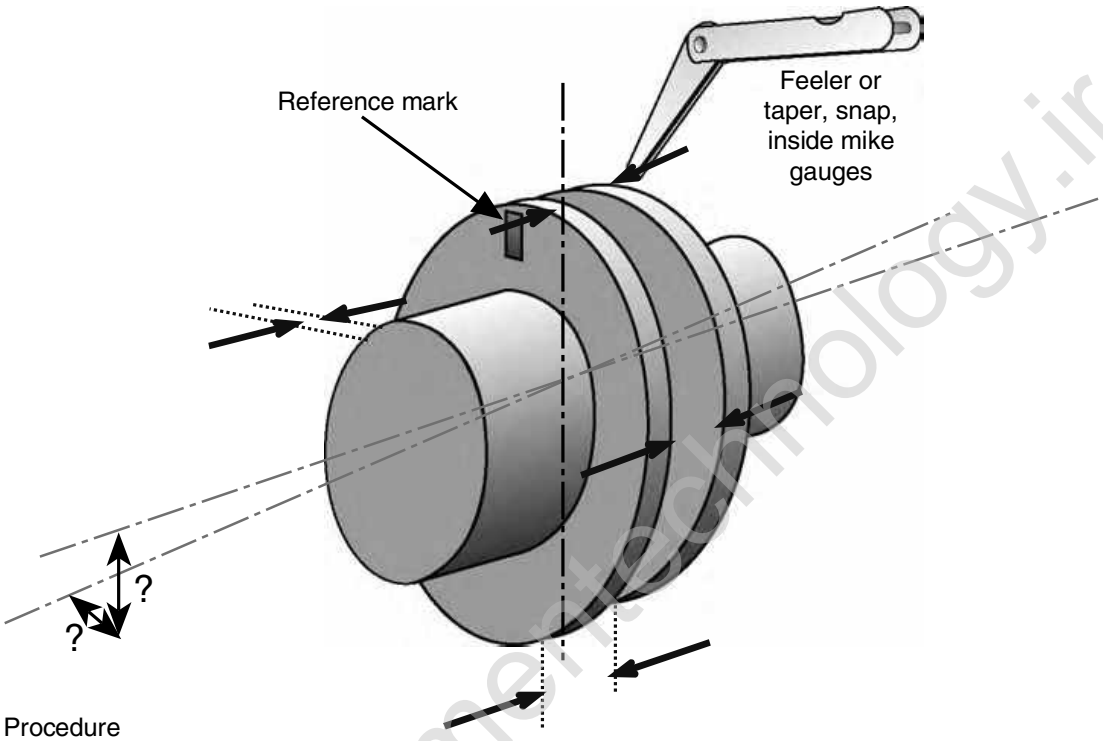
The face and rim method measures an offset and an angle of another shaft's centerline of rotation with respect to the line of sight of a reference shaft. The offset is measured by the rim

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This technique is typically used for rigid couplings with spigot (recessed) fits commonly found on machinery where one rotor is supported in two bearings and the other rotor is supported by one bearing.





Procedure

1. Insure the coupling bolts are loose and there is a slight separation (around 20 mils) between the coupling hub faces to prevent any stress or binding force interaction from one shaft to another.
2. Place a reference mark on one (or both) of the shafts, usually at twelve o'clock.
3. Accurately mark off 90° increments on the coupling hubs from the twelve o'clock reference.
4. Use feeler, or taper gauges capable of measuring to 0.001 in. (1 mil) to measure the gaps between the coupling hub faces at these 90° intervals (i.e., both sides, top and bottom).
5. Measure the diameter of the coupling hubs where the gaps were captured.
6. Record each gap reading and rotate both shafts 90°.
7. Capture another set of readings and rotate the shafts 90° again.
8. Repeat step 7 until the reference mark has returned to its original position at twelve o'clock.

FIGURE 11.9 Sixteen-point method and procedure.

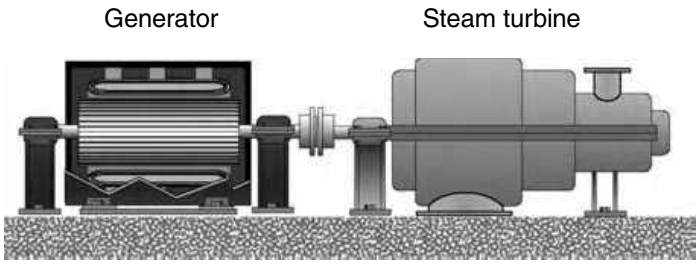
indicator and the angle is measured by the face indicator. Similar to the reverse indicator, line to points modeling method described in Chapter 10, one of the shafts is placed directly on the graph paper centerline as a reference and then the other shaft is positioned based on the dial indicator measurements obtained.

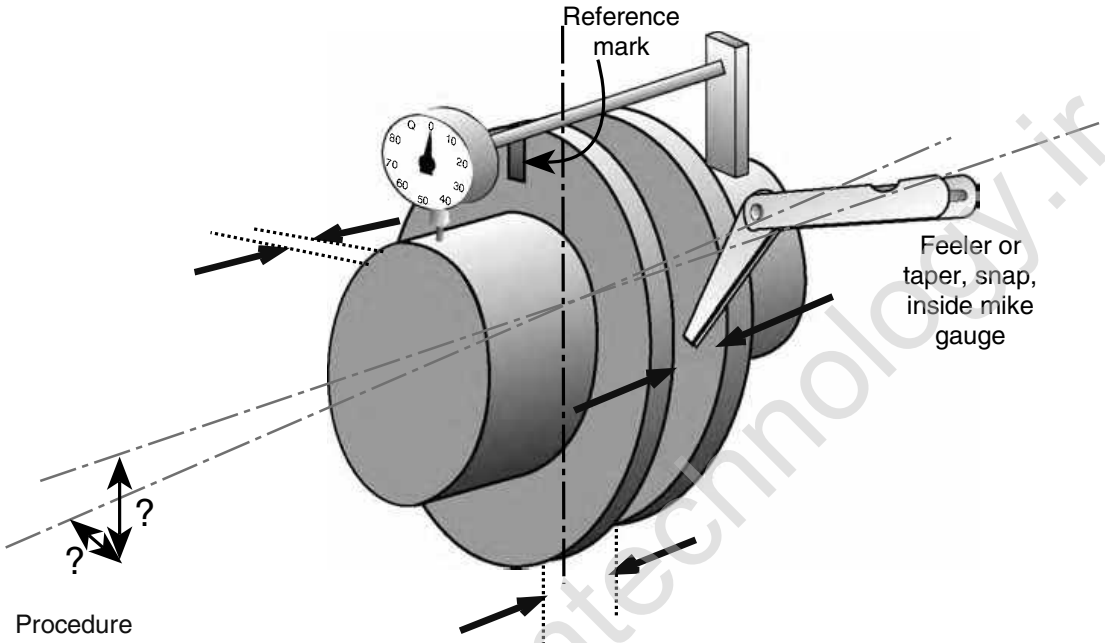
To graph the face–peripheral method you need to have a clear piece of plastic with a "T" inked onto the plastic similar to what is shown in Figure 11.15. The T bar overlay will represent the shaft where the dial indicators are capturing the readings. The shaft that the bracket is clamped to is the reference shaft and therefore will be drawn onto the graph paper centerline.

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This technique is typically used for rigid couplings commonly found on machinery where both rotors are supported in two bearings.





Procedure

1. Insure the coupling bolts are loose and there is a slight separation (around 20 mils) between the coupling hub faces to prevent any stress or binding force interaction from one shaft to another.
2. Place a reference mark on one (or both) of the shafts, usually at twelve o'clock.
3. Accurately mark off 90° increments on the coupling hubs from the twelve o'clock reference.
4. Attach a bracket or fixture to one shaft and span over to the other shaft to place a dial indicator on the diametral surface or rim of the coupling. Zero the indicator at the twelve o'clock position.
5. Use feeler or taper gauges capable of measuring to 0.001 in. (1 mil) to measure the gaps between the coupling hub faces at these 90° intervals (i.e., both sides, top and bottom).
6. Measure the diameter of the coupling hubs where the gaps were captured.
7. Record each gap reading and rotate both shafts 90°.
8. Capture another set of feeler gauge readings and note the reading on the dial indicator that is now on the side of the coupling hub. Rotate the shafts 90° again.
9. Capture another set of feeler gauge readings and note the reading on the dial indicator that is now on the bottom of the coupling hub. Rotate the shafts 90° again.
10. Capture another set of feeler gauge readings and note the reading on the dial indicator that is now on the other side of the coupling hub. Rotate the shafts 90° again returning the reference mark back to twelve o'clock.

FIGURE 11.10 Twenty-point method and procedure.

There are nine pieces of information that you need to properly construct the shaft positions using this technique:

1. Which shaft will the bracket be attached to and on which shaft will the dial indicators be taking readings?
2. The distance from the outboard to inboard feet (bolting planes) of the machine where the bracket is attached.
3. The distance from the inboard bolting plane of the machine where the bracket is attached to the point on the shaft where the bracket is held in place.

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Compensating for axial shaft float when capturing face readings

Why is this important?
Rotating machinery that is supported in sliding type bearings is designed to move somewhat freely in the axial direction. The amount of axial travel is restrained by thrust bearings or by electromagnetic forces. The amount of axial float varies from machine to machine but can be as little as 20 mils (0.020 in.) and as much as a half inch or more such as found on medium to large (i.e., 500 hp+) electric motors. If you plan on using the face–rim alignment measurement method for shaft alignment purposes, you must compensate for any axial movement that occurs during the shaft alignment measurement process.

1. Attach the alignment bracket to either one of the shafts, place a dial indicator at the twelve o'clock position on the other shaft or coupling hub face as shown insuring the dial indicator is at mid-travel on the stem. Anchor a magnetic base (or other stationary fixture) to the machine case (or any stationary object), place a dial against the coupling hub, end of the shaft, or anything attached to the shaft where the indicator can observe any axial displacement during rotation. If both shafts can move in the axial direction, a magnetic base and indicator must be positioned on both shafts as shown. Zero all the indicators and prepare a measurement recording sheet.

2. Rotate both shafts through a 90° rotation. Carefully observe each indicator during rotation noting if the stem is being pushed in (i.e., clockwise needle rotation, aka positive readings) or if it is traveling outward (i.e., counterclockwise needle rotation, aka negative readings). Stop after the 1/4 turn has been achieved and record the measurement on every dial indicator.

3. Again, rotate both shafts through a 90° rotation carefully observe each indicator during rotation noting if the stem is being pushed in or if it is traveling outward. Stop after the 1/4 turn has been achieved and record the measurement on every dial indicator.

4. If possible, again, rotate both shafts through a 90° rotation carefully observe each indicator during rotation, stop after the 1/4 turn has been achieved and record the measurement on every dial indicator. (Also see Section 6.10.)

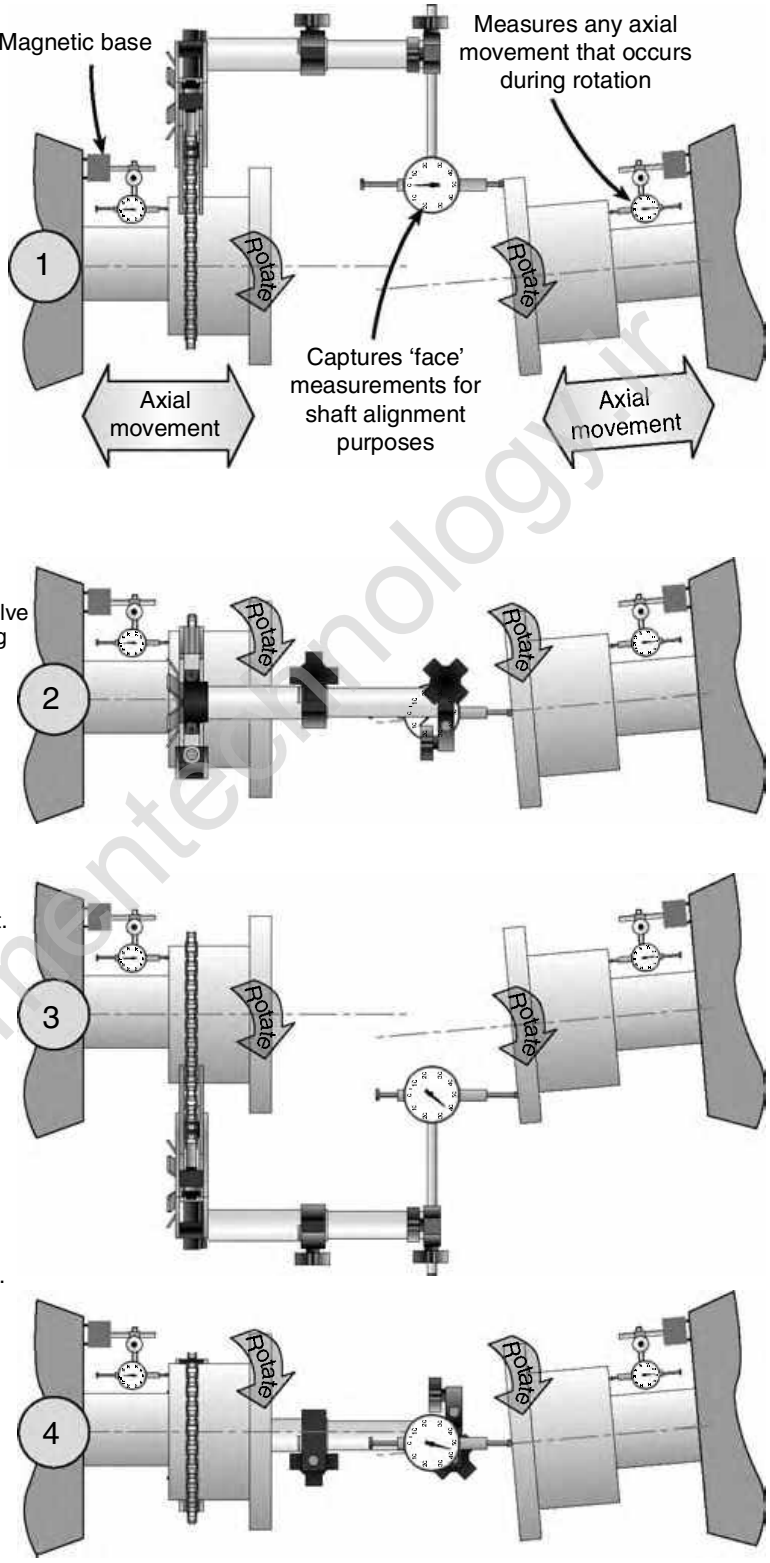


FIGURE 11.11 Compensate for axial movement with stationary indicators.

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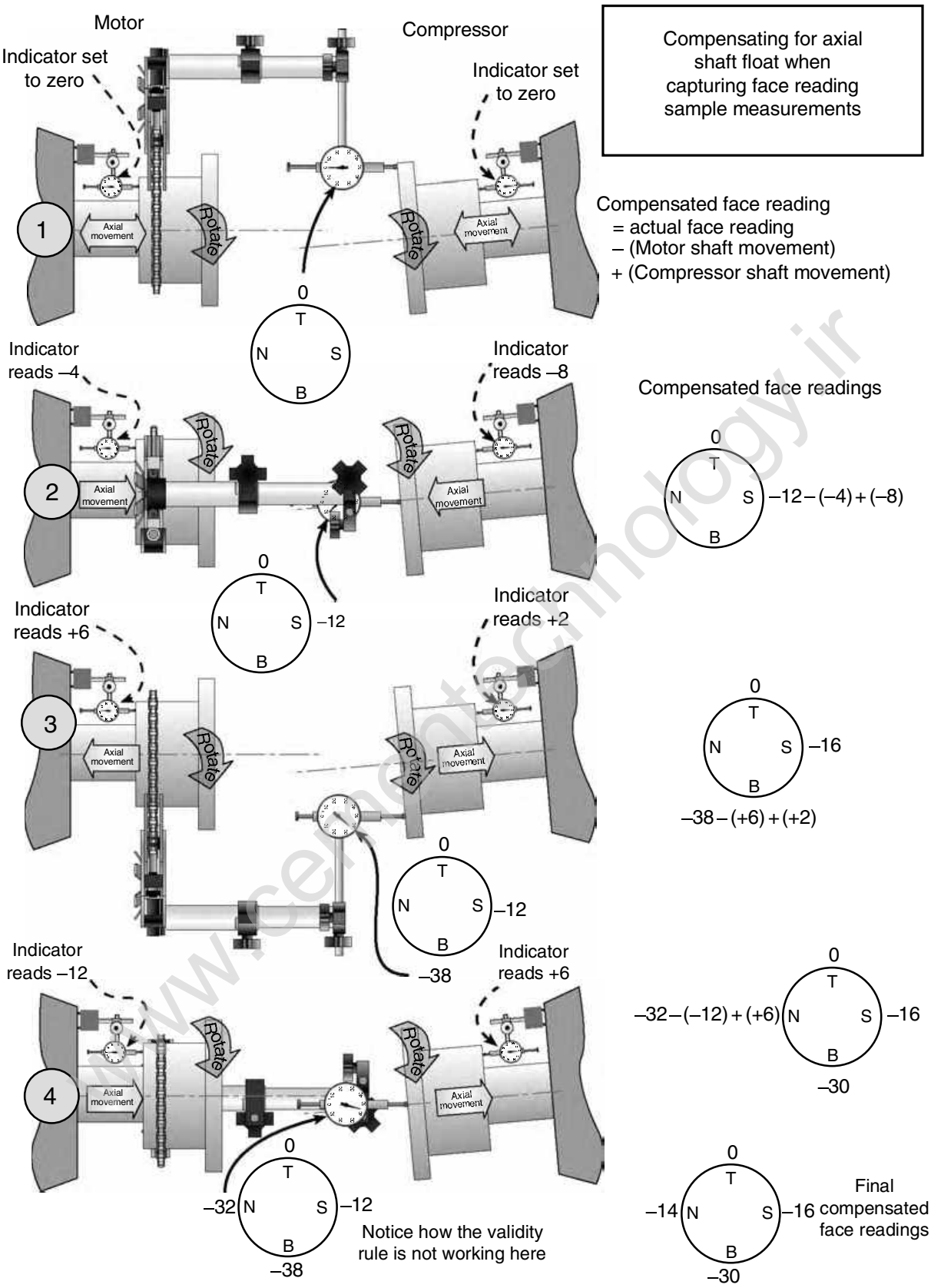


FIGURE 11.12 Example of compensating for axial movement with stationary indicators.

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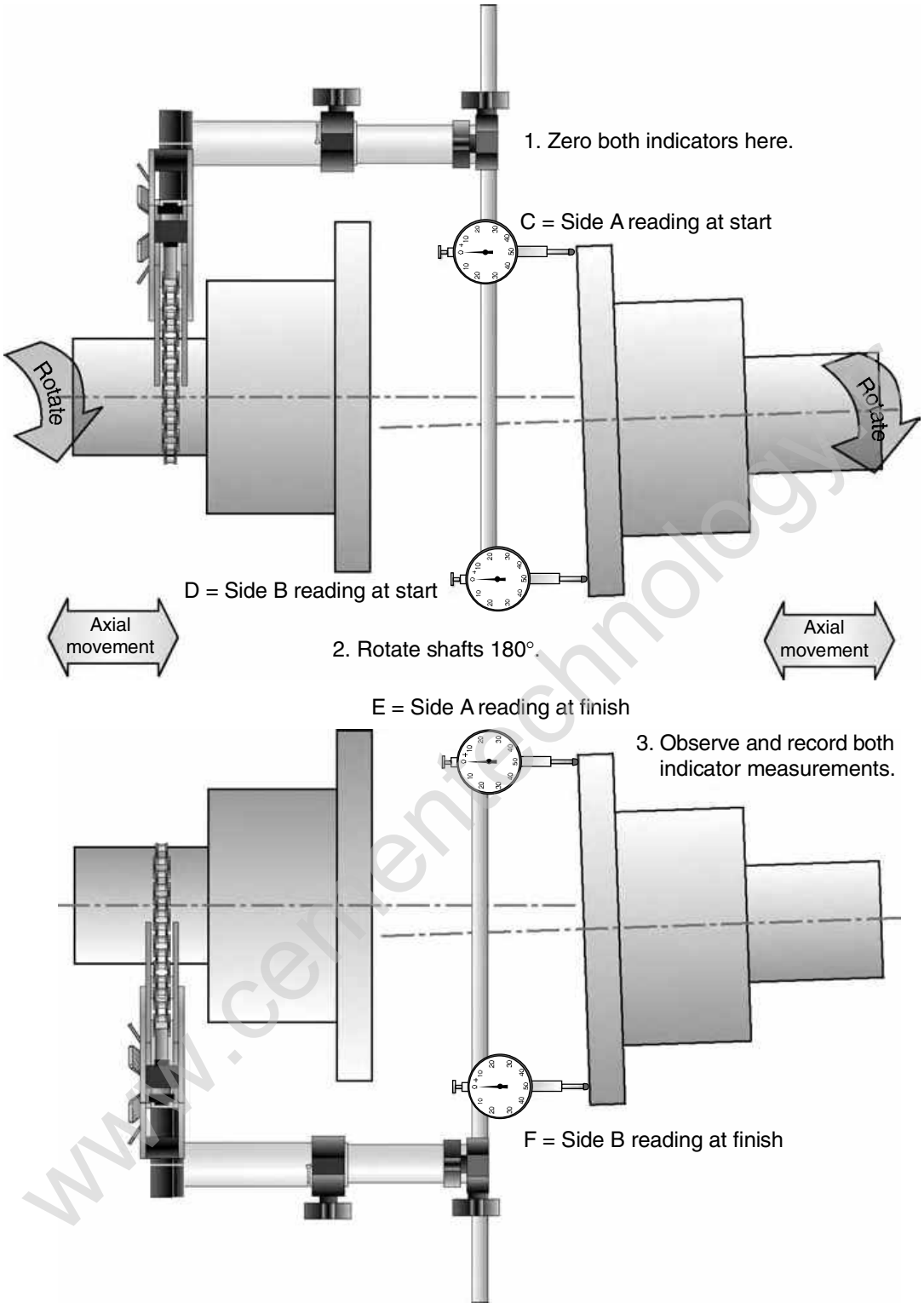
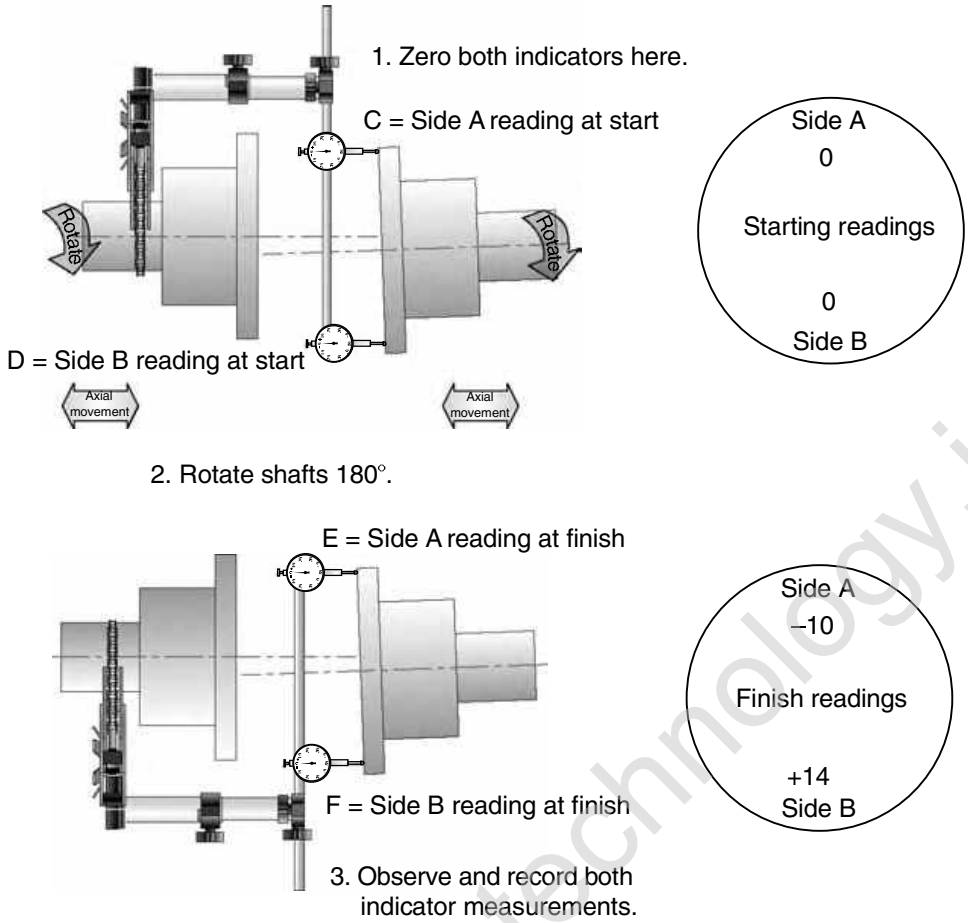


FIGURE 11.13 Compensate for axial movement with rotating indicators.

4. The distance from where the bracket is held in place to the point on the other shaft where the dial indicators are capturing the face and rim readings.
5. The distance from where the dial indicators are capturing the face and rim readings to the inboard bolting plane of that machine.

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(((C) + (F)) + ((D) – (E)))/2 = axial float compensated reading from Side A to Side B

(((0) + (+14)) + ((0) – (–10)))/2 = ((+14) + (+10))/2 = +24/2 = +12

Note: If the readings are taken from top to bottom, readings D and E must be compensated for face sag.

FIGURE 11.14 Example of compensating for axial movement with rotating indicators.

6. The distance from the inboard to outboard feet (bolting planes) of the machine where the dial indicators are capturing the readings.
7. The diameter on which the face readings are being taken.
8. Whether the face readings are being taken on the "front" or "back" side of the coupling hub or face measurement surface. Refer to Figure 11.2.
9. The eight dial indicator readings taken at the top, bottom, and both sides of the rim and face measurement points.

Scale the distances onto a piece of graph paper and scale the diameter of the face reading onto the T bar overlay as shown in Figure 11.16 and Figure 11.17. The top part of the "T" represents the face of the shaft you are taking readings on and the base of the "T" represents the centerline of rotation of the shaft.

In this method, you dual scale the graph. In other words, whatever scale factor you use from left to right to scale the dimensions along the length of the machinery, that same scale factor is used from top to bottom on the graph to scale the diameter the face readings were taken on when you transfer this dimension to the top of the T on the T bar overlay. Likewise,

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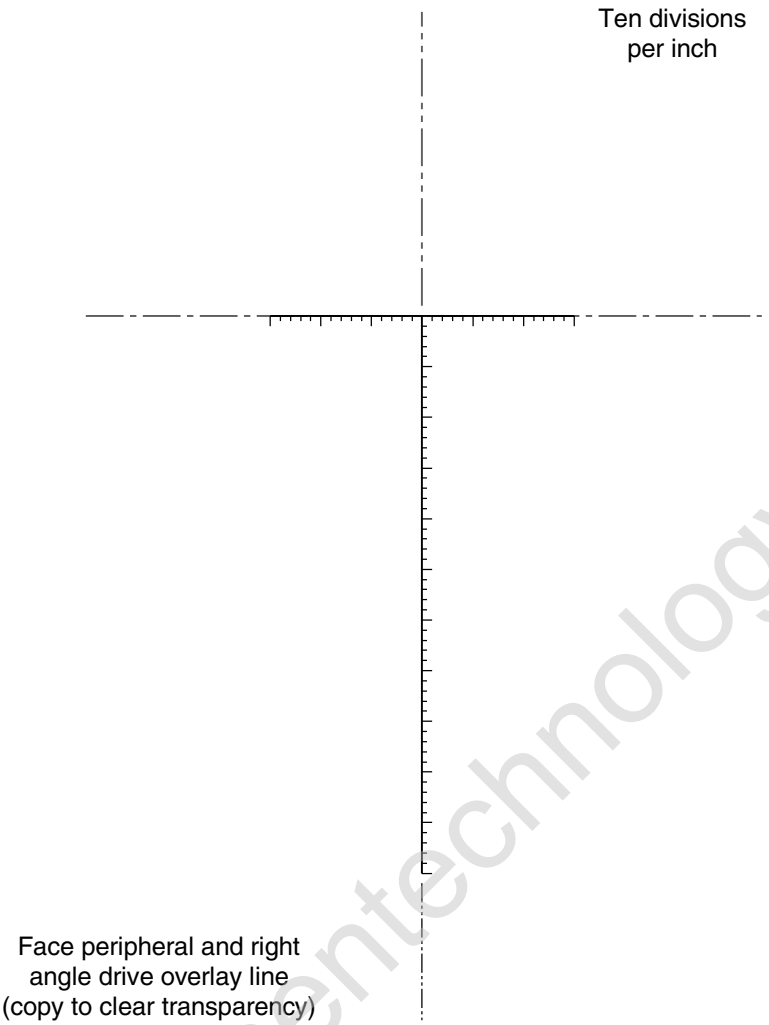


FIGURE 11.15 The T bar overlay (50% scale).

whatever scale factor you select to exaggerate the misalignment condition for the rim readings from top to bottom on the graph, that same scale factor is used from left to right on the graph when pitching or rotating the T bar overlay to reflect the face reading you observed. Insure that you use the same scale factor (inches) for both the machine dimensions and face diameter and the same scale factor (mils) for the rim and face measurements.

The procedure for plotting the face–rim technique is as follows:

1. Draw the shaft where the alignment bracket is attached directly on top of the graph centerline.
2. Next, position the clear T bar overlay to reflect the readings captured on the rim or perimeter of the other shaft. If the bottom (or side) rim reading was negative, slide the T bar toward the top of the graph paper so that the base of the T is one-half of the rim reading from the graph centerline. If the bottom (or side) rim reading was positive, slide the T bar toward the bottom of the graph paper so that the base of the T is one-half of the rim reading from the graph centerline.
3. Pivot the T bar overlay to reflect the face readings captured. There are several ways to accomplish this. You could pivot or rotate the T bar from the upper point on the top of the T bar where the dial indicator was zeroed and move the bottom point. This is

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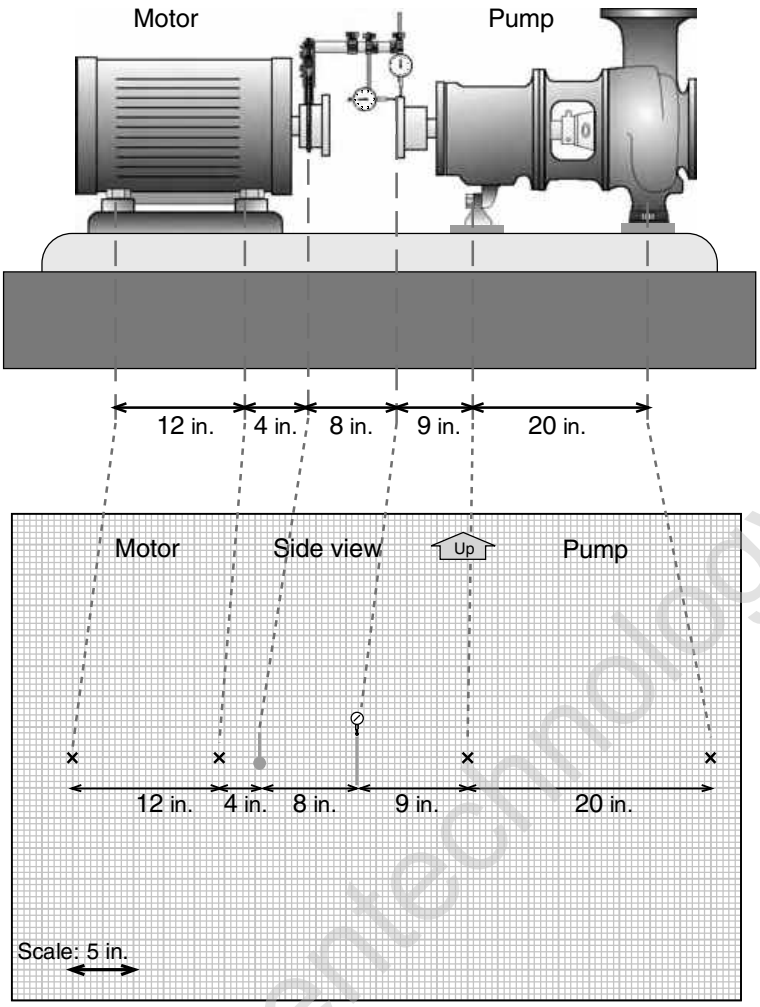


FIGURE 11.16 Dimensional information needed for plotting the face–rim measurements.

referred to as a "top pivot." You could pivot where the base and top of the T intersect and pivot half way at the top or bottom point often referred to as a "center pivot," or you could pivot from the lower point of the T bar and move the top point often referred to as a "bottom pivot."

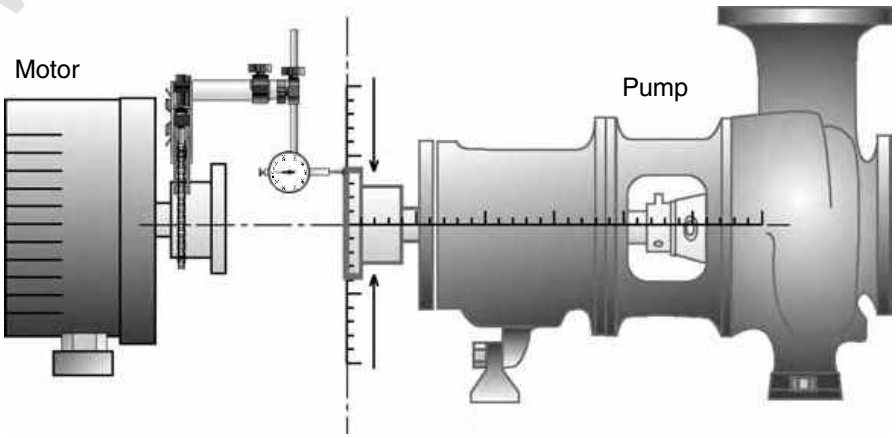


FIGURE 11.17 Scale the diameter of the face readings onto the top of the T bar overlay.

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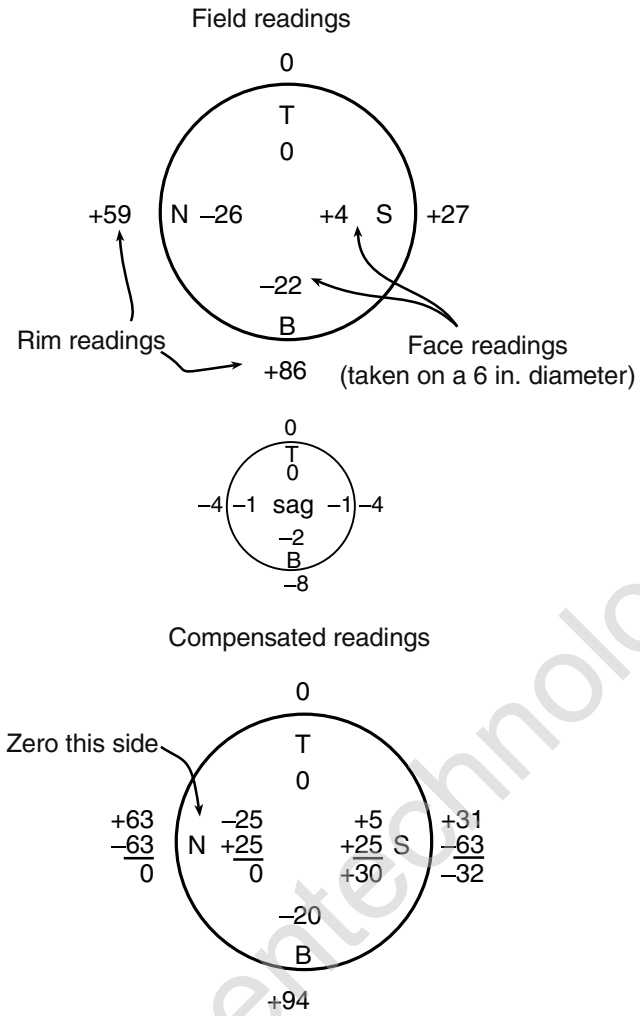


FIGURE 11.18 Face–rim field and compensated measurements for Figure 11.19 and Figure 11.20.

Figure 11.19 and Figure 11.20 show an example of both the side and top view alignment models of a motor and a pump where face–rim readings were taken.

An inexpensive device that uses the T bar overlay principle has been available commercially since 1973 (developed earlier at an oil refinery on the Dutch Caribbean island of Aruba). "The machinery alignment plotting board," shown in Figure 11.21, is an 8.5″ × 11″ laminated plastic graph, with reusable plastic overlay which slides and pivots in a groove for easy positioning. It can be used for face–rim, reverse indicator, and other setups, with any legitimate indicator and bracket configuration. It can also be used for two element move plots (see references).

11.6 ARTIFICIAL FACE SURFACE

In the event that you are unable to rotate one shaft and there is not a good face surface to take measurements on, one idea is to temporarily provide a face reading surface by fabricating a split disk arrangement that can be clamped onto the outer diameter of a shaft and then removed after the alignment is complete. Figure 11.22 shows an arrangement being tested for this purpose.

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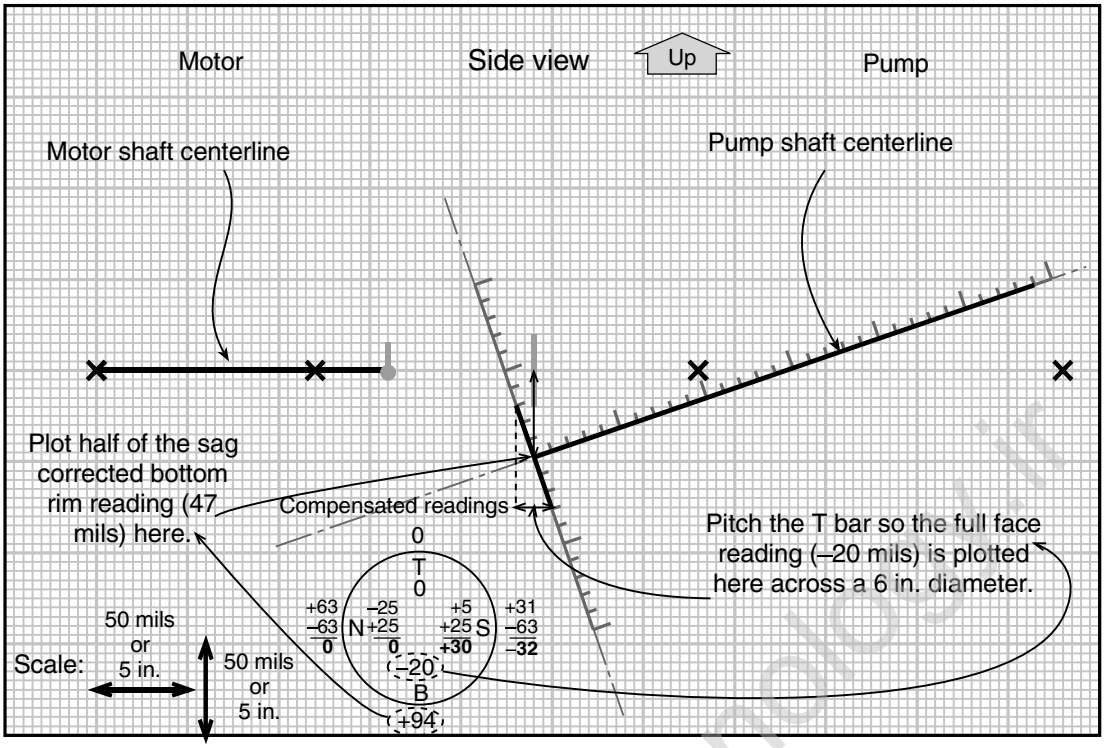


FIGURE 11.19 Face–rim side view example alignment model.

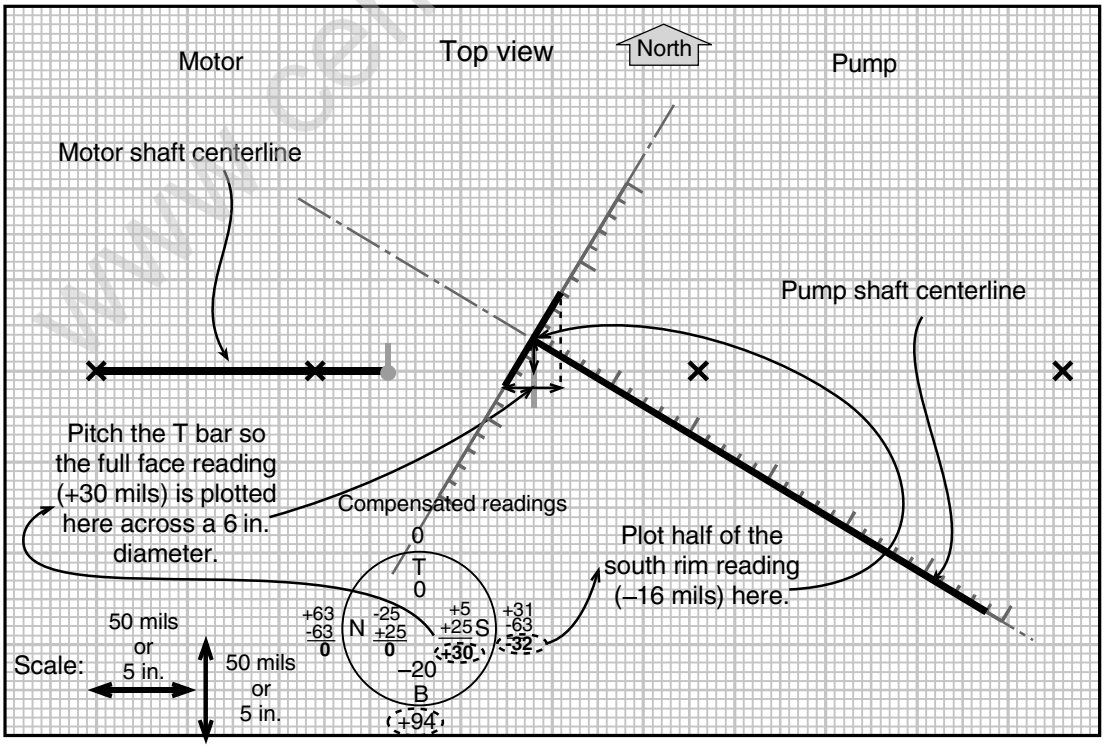


FIGURE 11.20 Face–rim top view example alignment model.

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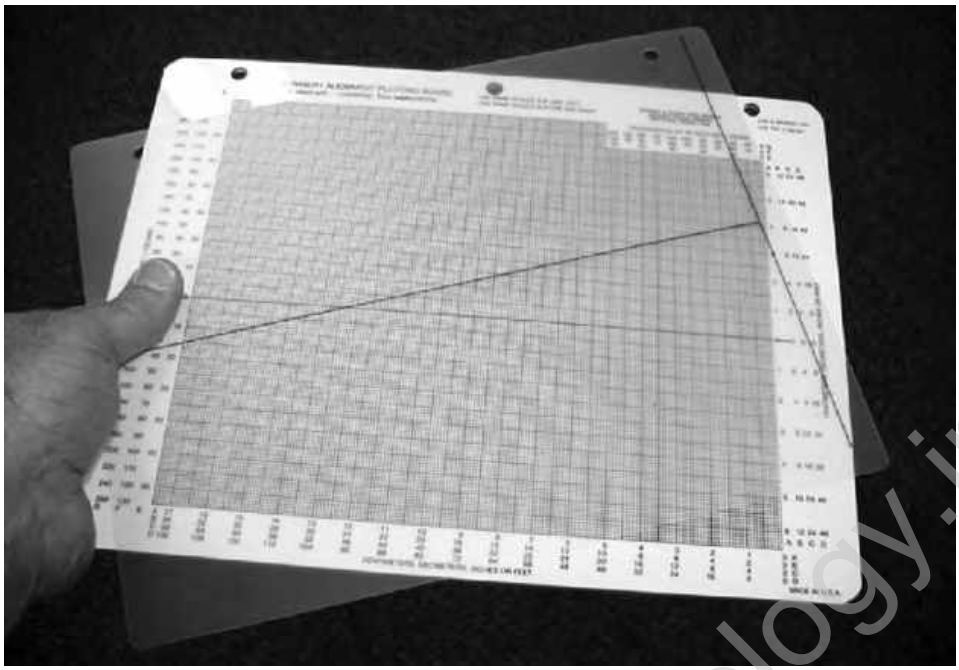


FIGURE 11.21 Murray & Garig Machinery Alignment Plotting Board.

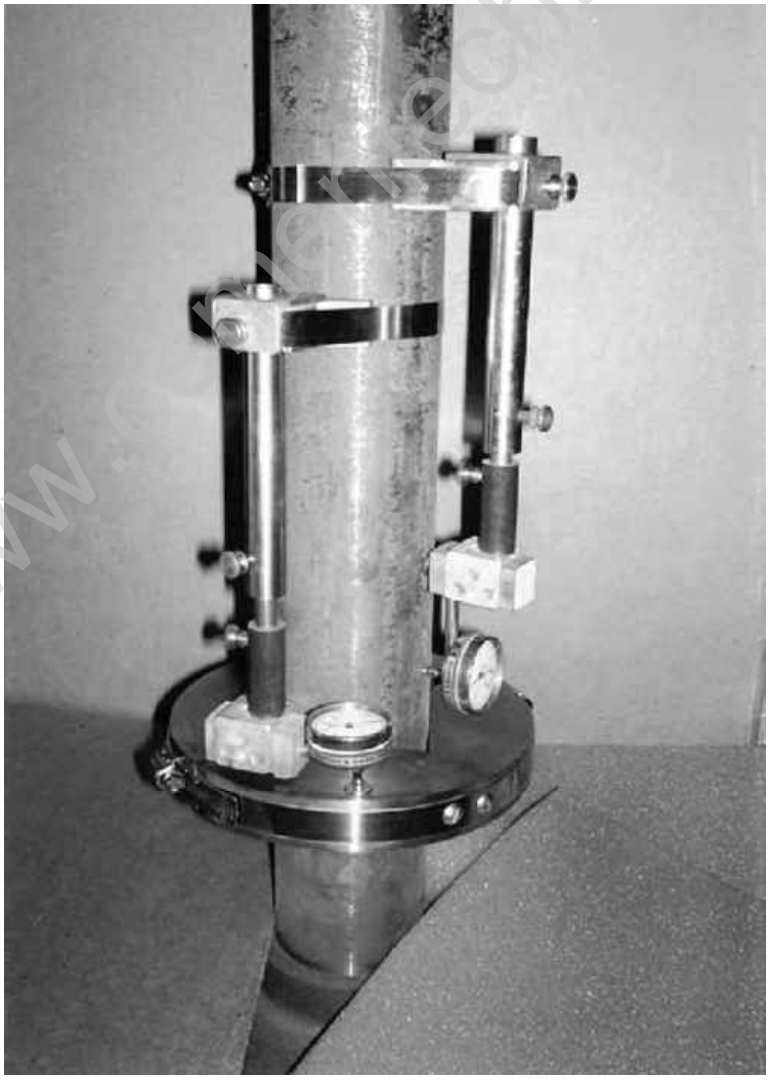


FIGURE 11.22 Artificial face split disk system. (Courtesy of Murray & Garig Tool Works, Baytown, TX.)

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12 Double Radial Method

This relatively unknown method has some distinct advantages compared to the other methods discussed in chapters 10,11,13,14, and 15. The procedure is shown in Figure 12.1. This method should only be used if there is at least a 3 in. or greater separation between the near and far indicator measurement positions. The accuracy of this technique increases as the distance between reading points increases. The disadvantage of this method is that there is usually not enough shaft exposed to be able to spread the indicators far enough apart to merit using the method except for very special circumstances.

Advantages

- This is a good technique to use in situations where one of the machinery shafts cannot be rotated or it would be difficult to rotate one of the machinery shafts.
- A good method to use when the dial indicator readings at the near and far indicator measurement locations can be separated a reasonable distance apart. This method begins to approach the accuracy of the reverse indicator technique when the distance between the two sets of dial indicator readings being captured on one shaft equals or exceeds the span between reading points from shaft to shaft.
- If the machinery is supported in sliding type bearings and the shafts are ''floating'' back or forth axially when rotating the shaft to capture readings, there is virtually no effect on the accuracy of the readings being taken.
- Can be setup to measure inner circular surfaces such as the bore of a barrel.

Disadvantages

- Not enough shaft surface is exposed to spread the readings far enough apart for acceptable accuracy.
- Bracket sag must be measured and compensated for.

Although it has not been mentioned up to this point in the book, any of the alignment measurement methods shown in Chapter 10 through Chapter 15 can be used on shafts oriented in horizontal positions but also on shafts in vertical positions. Figure 12.4 and Figure 12.5 show the double radial method being used on a vertically oriented motor and pump. In this particular case, the motor and pump shafts are connected together using a rigid coupling rather than a flexible one.

For a moment, refer to Figure 1.3 and Figure 6.41, which show how under moderate to severe misalignment conditions, the shafts will start elastically bending. As discussed in Chapter 6, elastic bending occurs on both rigid and flexible couplings. On rigid couplings the elastic bending will begin with just small amounts of misalignment. Therefore, shaft alignment measurements should never be taken across an engaged rigid coupling. On the vertical pump shown in Figure 12.4 and Figure 12.5, the rigid coupling between the motor and pump shafts must be disengaged to relieve any bending stresses due to a misalignment

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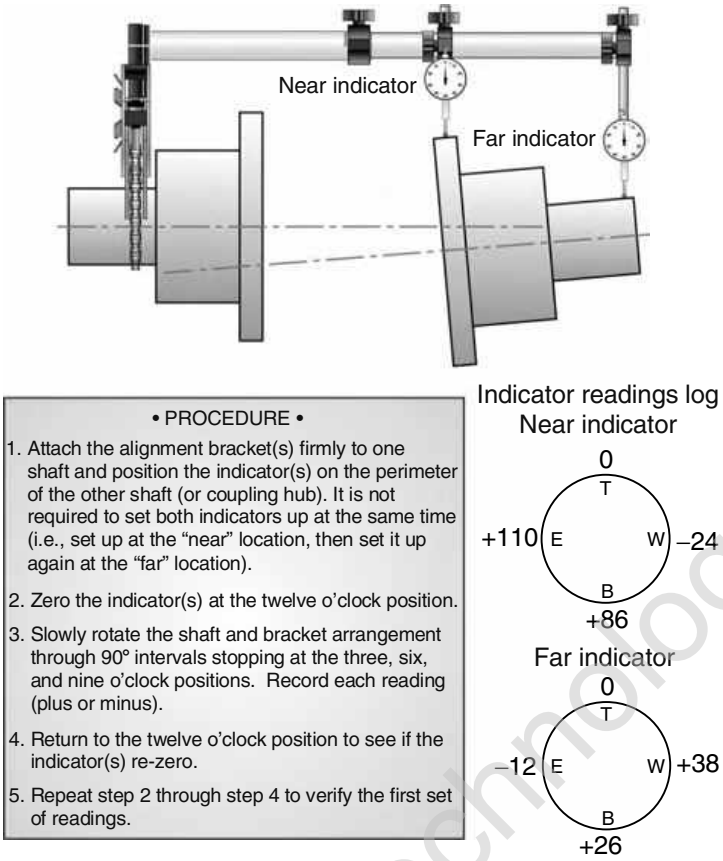


FIGURE 12.1 Double radial method and procedure.

condition. The pump shaft, which is supported by the thrust bearing on top of the motor, drops down and is physically centered in its upper bushing using feeler gauges or wedges. Once this is done, the pump shaft should not be rotated to prevent damaging the impeller from dragging

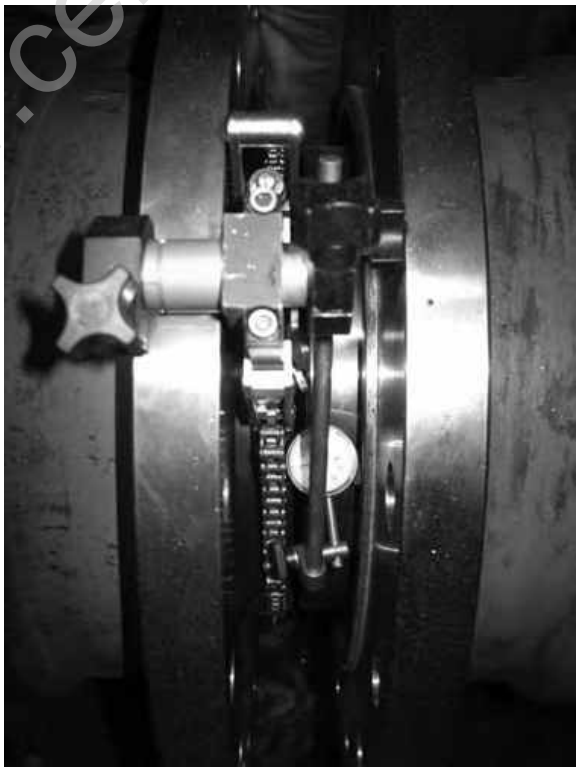


FIGURE 12.2 Double radial method used between an output shaft of a gear, which could be rotated with an indicator measuring the ''near'' position on a gear input shaft that could not be rotated.

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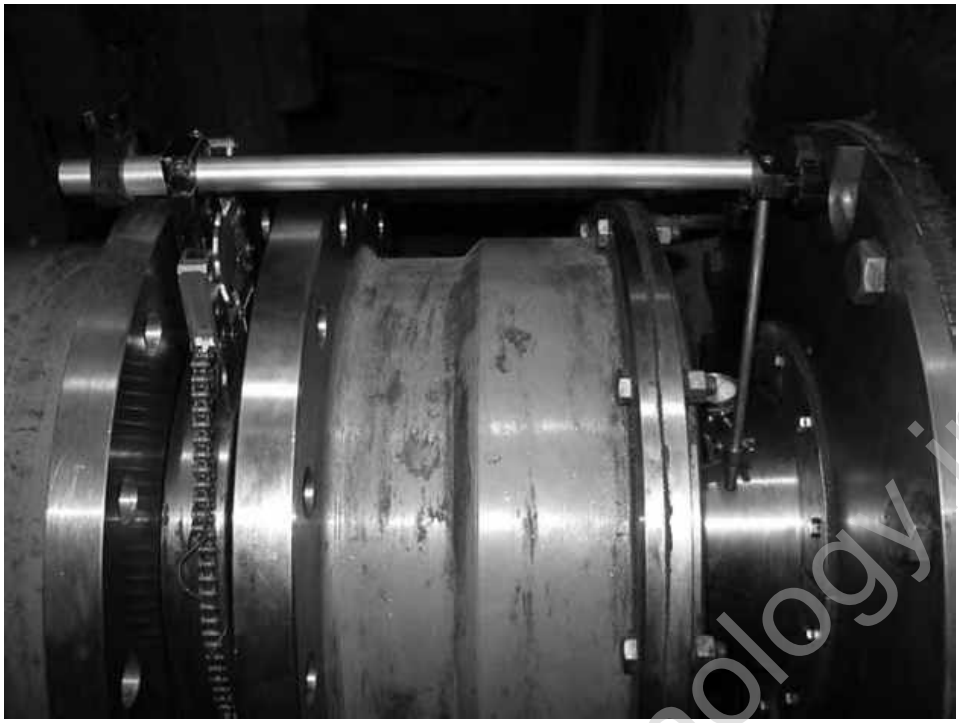


FIGURE 12.3 Double radial method used between an output shaft of a gear, which could be rotated with an indicator measuring the "far" position on a gear input shaft which could not be rotated.

on the pump housing. As the motor shaft can still be rotated and there is a significant distance of pump shaft exposed, the double radial method is a good choice for this alignment situation.

12.1 BASIC MATHEMATICAL EQUATIONS FOR THE DOUBLE RADIAL METHOD

Figure 12.6 shows the mathematical relationship between the machinery dimensions and the dial indicator readings captured using the double radial method. The equations will solve for the moves that need to be made to correct the misalignment condition (i.e., bring the shafts into a collinear relationship when off-line) on one or the other machine case.

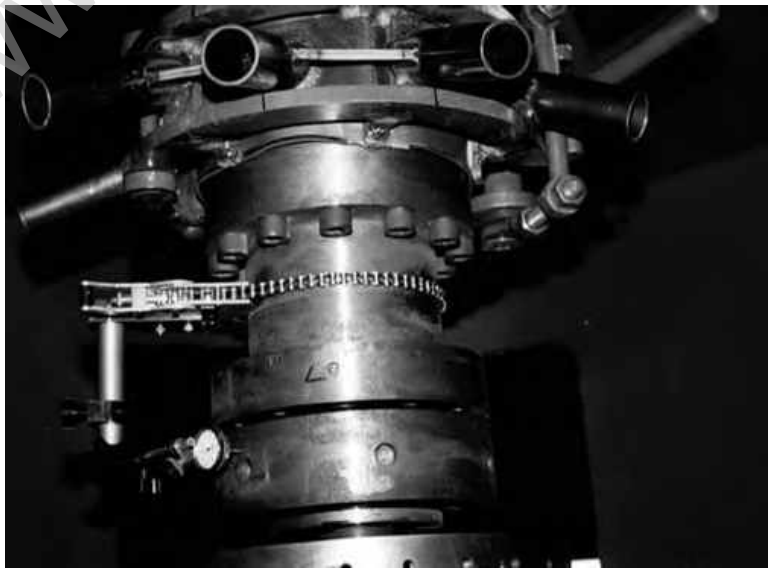


FIGURE 12.4 Double radial method used on vertical motor and pump, with indicator measuring the near position with a dial indicator.

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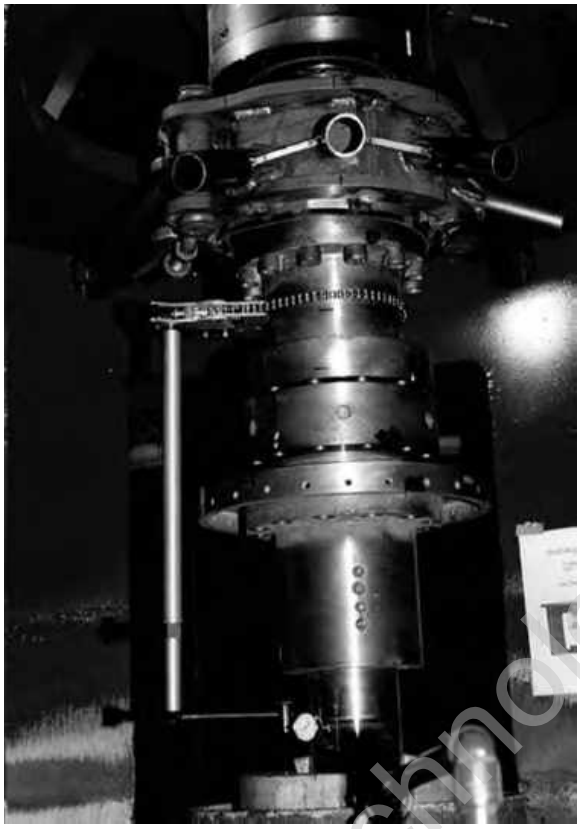
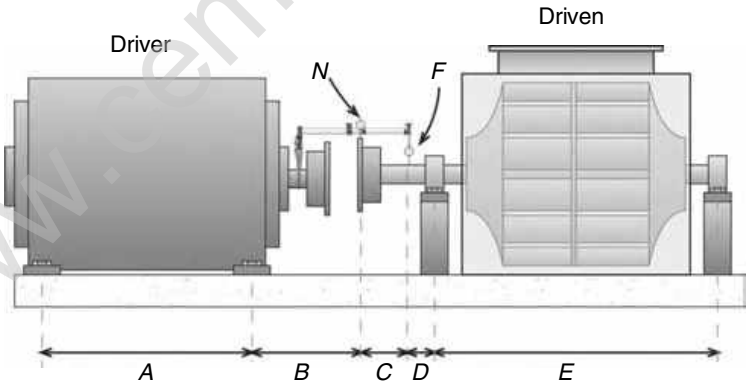


FIGURE 12.5 Double radial method used on vertical motor and pump, with indicator measuring the far position with a dial indicator.



where
A, *B*, *C*, *D*, *E* = distances shown (in.)
N = one half of the near indicator rim reading difference (from top to bottom or side to side in mils)
F = one half of the far indicator rim reading difference (from top to bottom or side to side in mils)
Note: Readings *N* and *F* must be sag compensated readings.

$$\text{Inboard feet of driver} = (B+C)\left[\frac{(F-N)}{C}\right] - (F) \qquad \text{Inboard feet of driven} = (C+D)\left[\frac{(F-N)}{C}\right] + (N)$$

$$\text{Outboard feet of driver} = (A+B+C)\left[\frac{(F-N)}{C}\right] - (F) \qquad \text{Outboard feet of driven} = (C+D+E)\left[\frac{(F-N)}{C}\right] + (N)$$

FIGURE 12.6 Double radial method mathematics for correcting moves on either machine case.

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12.2 MODELING THE DOUBLE RADIAL METHOD

The basic measurement principle of the double radial technique is to capture two (or more if desired) circumferential readings at different points along the length of a shaft.

There are six pieces of information that you need to properly construct the shaft positions using this technique:

1. The distance from the outboard-to-inboard feet (bolting planes) of the first machine
2. The distance from the inboard bolting plane of the first machine to the point on the shaft where the bracket is located on the first machine
3. The distance from where the near dial indicator is capturing the rim readings on the second machine to the point where the far dial indicator is capturing the rim readings on the second machine
4. The distance from where the far dial indicator is capturing the rim readings on the second machine to the inboard bolting plane of the second machine
5. The distance from the inboard-to-outboard feet (bolting planes) of the second machine
6. The eight dial indicator readings taken at the top, bottom, and both sides on both shafts after compensating for sag (i.e., what a perfect, "no sag" bracket system would have measured). Be aware of the fact that there will probably be two different sag amounts at each of the dial indicator locations

Accurately scale the distances along the length of the drive train onto the graph centerline as shown in Figure 12.7.

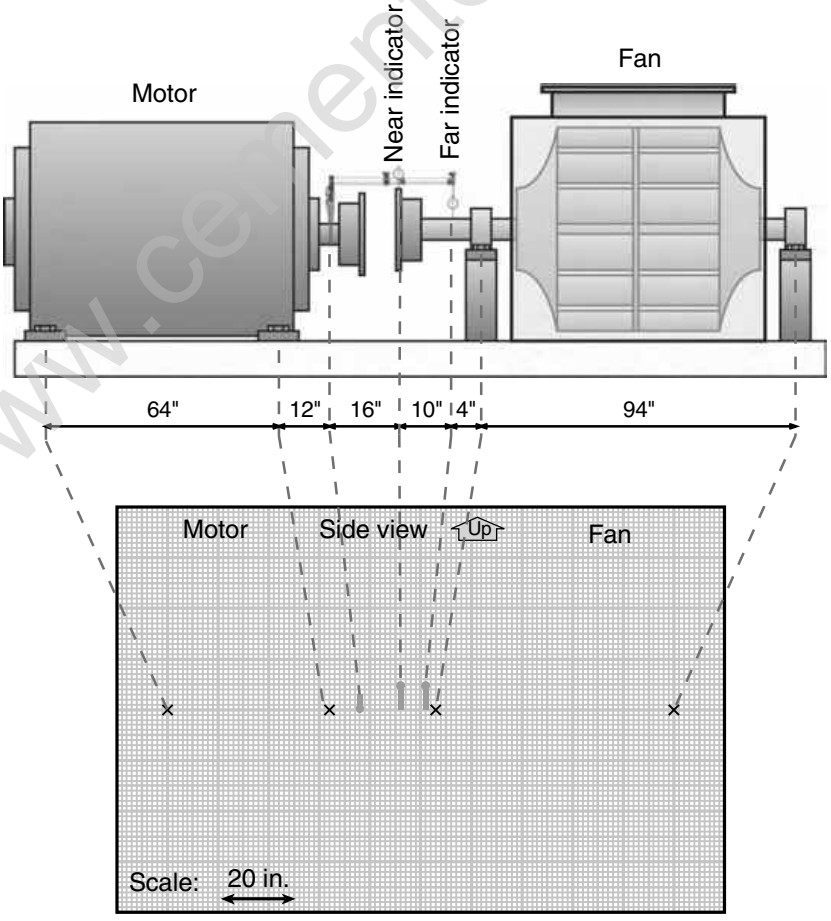


FIGURE 12.7 Dimensional information needed for plotting double radial measurements.

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The procedure for plotting the double radial technique is as follows:

1. Draw the shaft where the bracket is clamped on top of the graph centerline.
2. Start with the top to bottom or side-to-side dial indicator readings on the other shaft (i.e., the one you did not draw on the graph centerline).
3. Plot the other shaft centerline position by starting at the intersection of the graph centerline and the point where the near dial indicator was capturing the readings on the other shaft. If the bottom (or side) reading was negative, place a point half of the bottom (or side) readings from the graph centerline toward the top of the graph. If the bottom (or side) reading was positive, place a point half of the bottom (or side) readings from the graph centerline toward the bottom of the graph (the same as in the point-to-point modeling techniques). Do not draw any lines yet.
4. Next, start at the intersection of the graph centerline and the point where the far dial indicator was capturing the readings on the shaft. If the bottom (or side) reading was negative, place a point half of the bottom (or side) readings from the graph centerline toward the bottom of the graph. If the bottom (or side) reading was positive, place a point half of the bottom (or side) readings from the graph centerline toward the top of the graph (opposite of the point-to-point modeling technique).

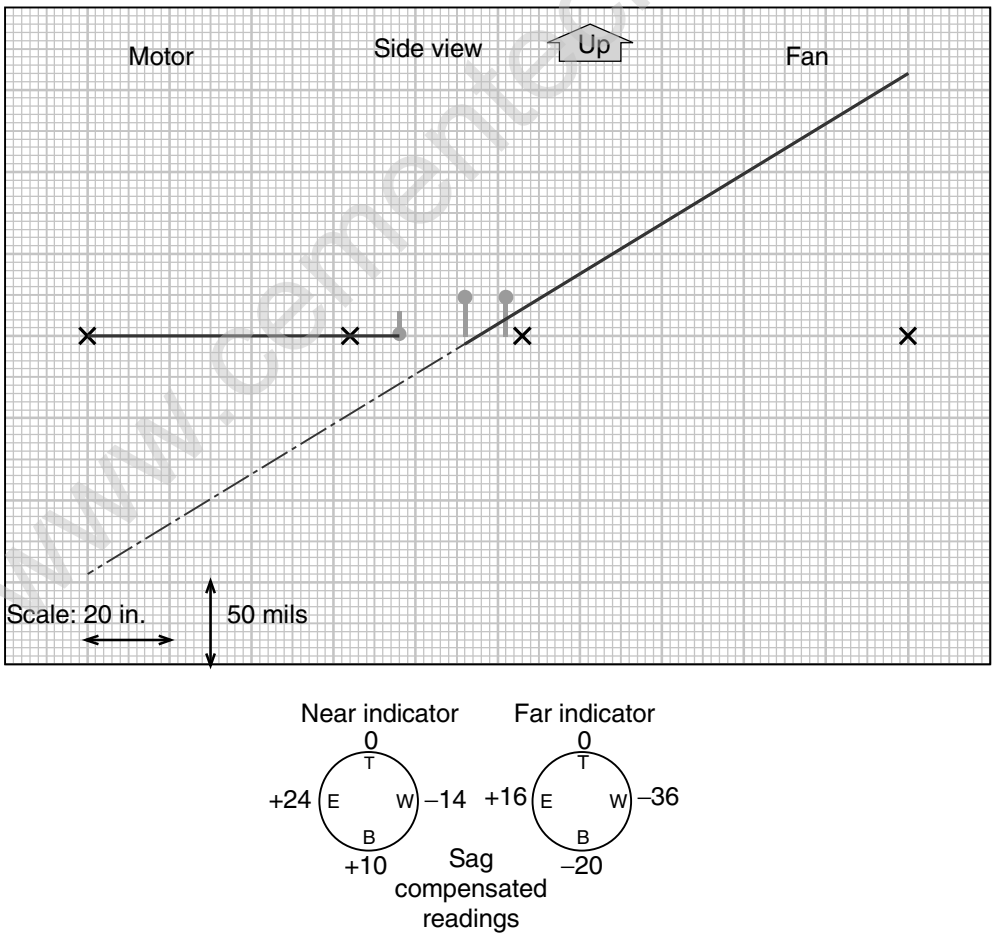


FIGURE 12.8 Double radial side view example alignment model.

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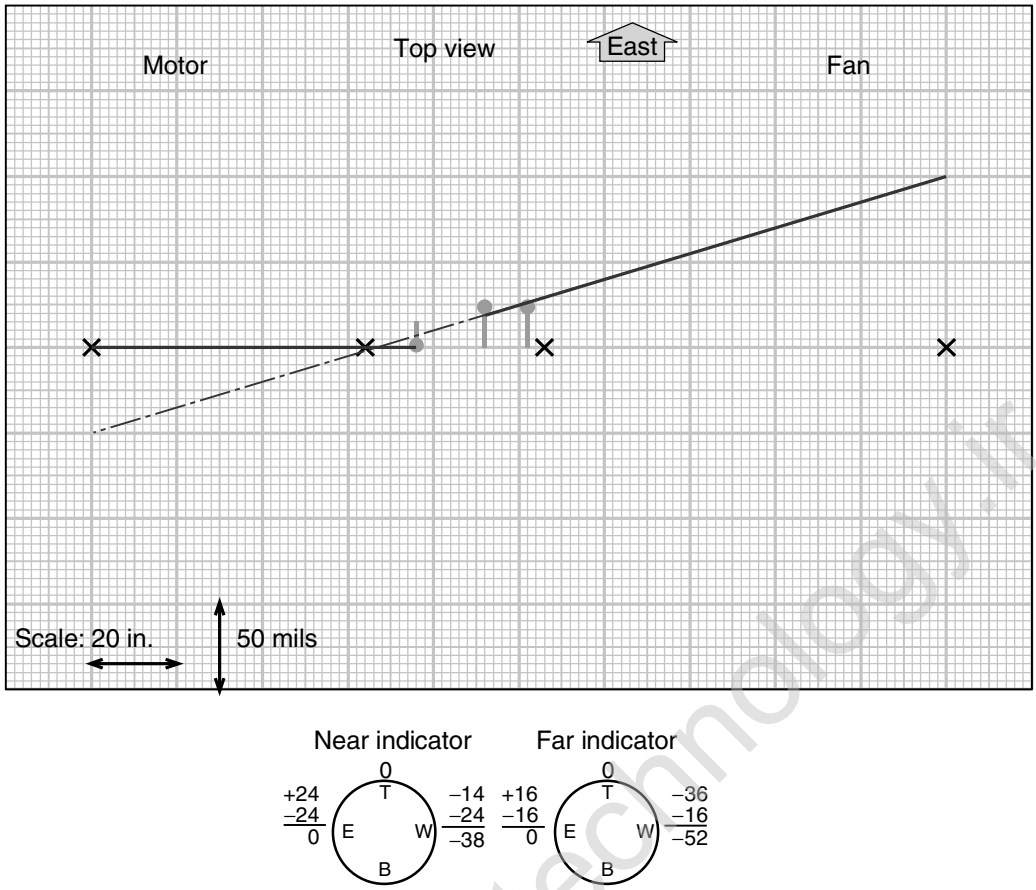


FIGURE 12.9 Double radial top view example alignment model.

5. These two points marked on the graph at the dial indicator reading points define the line of sight (i.e., the centerline of rotation) of the other shaft. Draw a straight line through these two points from the coupling end to the outboard end of the other shaft.

Figure 12.8 and Figure 12.9 show an example of both the side and top view alignment models of a motor and a fan where double radial readings were taken.

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13 Shaft to Coupling Spool Method

There are situations where two pieces of rotating machinery are positioned a considerable distance apart; and trying to employ the alignment techniques shown in chapters 10, 11, or 12 would prove to be cumbersome due to the extreme distances between the ends of each shaft. This technique, shown in Figure 13.1, is recommended for measuring the positions of the shafts when the distance between shaft ends begins to exceed 30–40 in. This technique can be applied to cooling tower fan drive systems, press roll drives in the paper industry with universal joints, dryer can drive sections, vehicular drive shaft systems, gas or power turbine to generators or compressors for example.

For acceptable accuracy when using this technique, the distance from the flexing points at each end of the coupling to the point where the dial indicator is capturing readings should be at least 4 in. A good rule of thumb when setting up the tooling for this technique is to maintain a ratio of at least 1:10 between the flex point-to-reading location distance and the distance between flex points. For example, if the distance between flexing points in the coupling is 120 in., the distance from the coupling flex points to where the indicators are capturing readings should be at least 12 in.

Advantages

- Perhaps the most accurate measurement technique where extreme distances occur between shaft ends.
- Relatively easy to set up and capture readings.

Disadvantages

- Since coupling spool (also known as "jackshaft" or "drive shaft") must be kept in place, both shafts must be rotated together.

13.1 BASIC MATHEMATICAL EQUATIONS FOR THE SHAFT TO COUPLING SPOOL METHOD

The mathematical relationship between the machinery dimensions and the dial indicator readings captured using the shaft to coupling spool method is shown in Figure 13.5. The equations will solve for the moves that need to be made to correct the misalignment condition (i.e., bring the shafts into a collinear relationship when off-line) on one or the other machine case.

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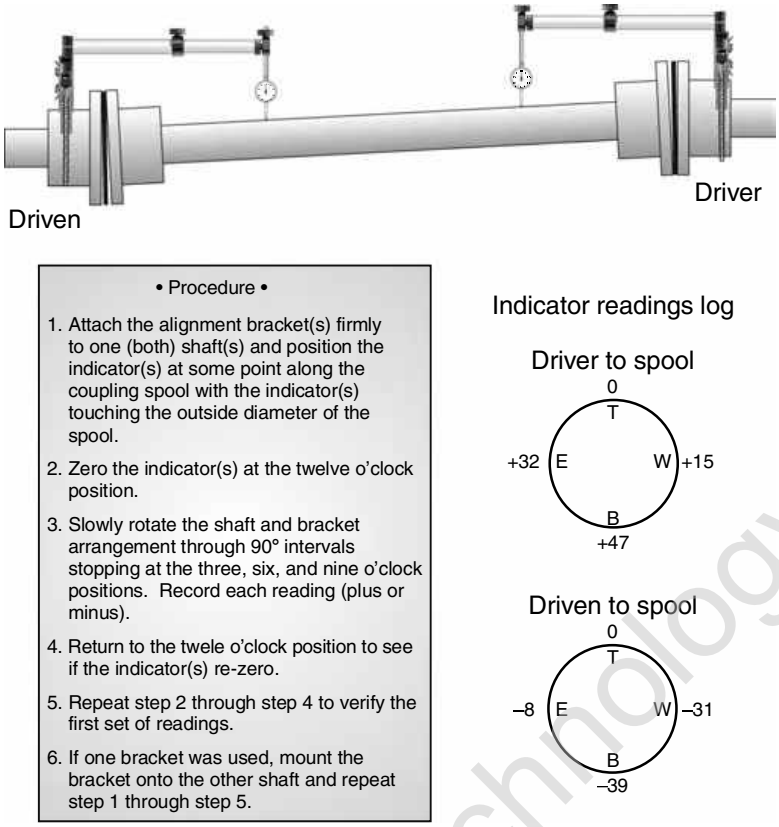


FIGURE 13.1 Shaft to coupling spool method and procedure.

13.2 MODELING THE SHAFT TO COUPLING SPOOL METHOD

The basic measurement principle of the shaft to coupling spool technique lies in the ability to measure the angle between each shaft centerline and the centerline of the coupling spool (jackshaft or drive shaft). Since there is only one flex point at the end of each shaft, near

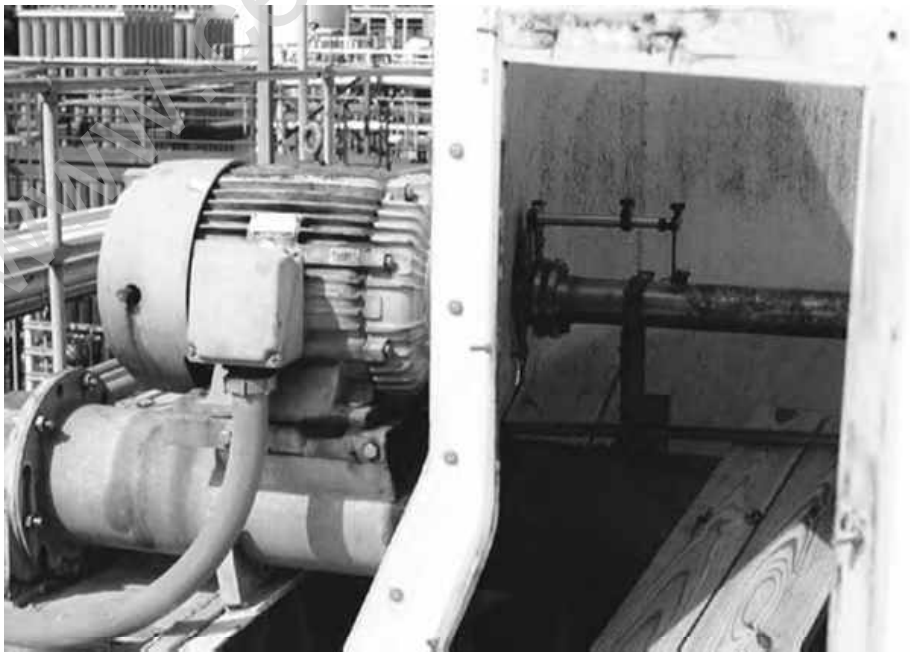


FIGURE 13.2 Shaft to coupling spool method used on cooling tower fan drive with bracket attached to motor shaft and indicator taking reading on spool.

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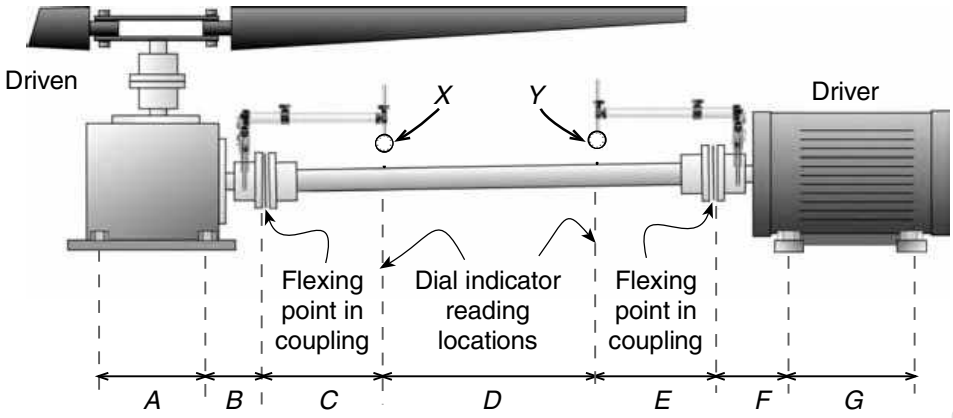
FIGURE 13.3 Shaft to coupling spool method used on cooling tower fan drive with bracket attached to gear input shaft and indicator taking reading on spool.



FIGURE 13.4 Shaft to coupling spool method employed on gear coupling with measurements taken on the spacer (spool).

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where
A, B, C, D, E, F, G = distances shown (in.)
X = one half of the driven to coupling spool dial indicator rim reading difference (from top to bottom or side to side in mils)
Y = one half of the driver to coupling spool dial indicator rim reading difference (from top to bottom or side to side in mils)
Note: Readings X and Y must be sag compensated readings.

$$\text{Inboard feet of driver} = (C+D+E+F)\left[\frac{X}{C}\right] + F\left[\frac{Y}{E}\right]$$

$$\text{Outboard feet of driver} = (C+D+E+F+G)\left[\frac{X}{C}\right] + (F+G)\left[\frac{Y}{E}\right]$$

$$\text{Inboard feet of driven} = (B+C+D+E)\left[\frac{Y}{E}\right] + B\left[\frac{X}{C}\right]$$

$$\text{Outboard feet of driven} = (A+B+C+D+E)\left[\frac{Y}{E}\right] + (A+B)\left[\frac{X}{C}\right]$$

FIGURE 13.5 Shaft to coupling spool method mathematics for correcting moves on either machine case.

perfect angular alignment exists between each shaft and the coupling spool. The coupling spool remains intact (i.e., connected to the shafts) during this procedure.

There are eight pieces of information that you need for properly constructing the shaft positions using this technique:

1. The distance from the outboard to the inboard feet (bolting planes) of the first machine.
2. The distance from the inboard bolting plane of the first machine to the flexing point between the shaft and the coupling spool on the first machine. Note that the point where the bracket is clamped on the shaft is not relevant. A good distance to span past the flex point is anywhere from 3 to 24 in. The greater this distance is, the more accurate this technique becomes.
3. The distance from the flexing point between the shaft and the coupling spool on the first machine to the point where the dial indicator is capturing the rim readings on the coupling spool.
4. The distance from where the dial indicator is capturing the rim readings on the coupling spool near the first machine to the point where the dial indicator is capturing the rim readings on the coupling spool near the second machine.

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5. The distance from where the dial indicator is capturing the rim readings on the coupling spool near the second machine to the flexing point between the shaft and the coupling spool on the second machine. Note that this distance does not have to be the same distance as measured in no. 3 above.
6. The distance from the flexing point between the shaft and the coupling spool on the second machine to the inboard bolting plane of the second machine.
7. The distance from the outboard to the inboard feet (bolting planes) of the second machine.
8. The eight dial indicator readings taken at the top, bottom, and both sides on the coupling spool after compensating for sag (i.e., what a perfect, ''no sag'' bracket system would have measured). Be aware of the fact that there will probably be two different ''sag'' amounts at each of the dial indicator locations if the distances are not the same from the flex points to where the dial indicator readings are taken.

Accurately scale the distances along the length of the drive train onto the graph centerline as shown in Figure 13.6.

The procedure for plotting the shaft to coupling spool technique is as follows:

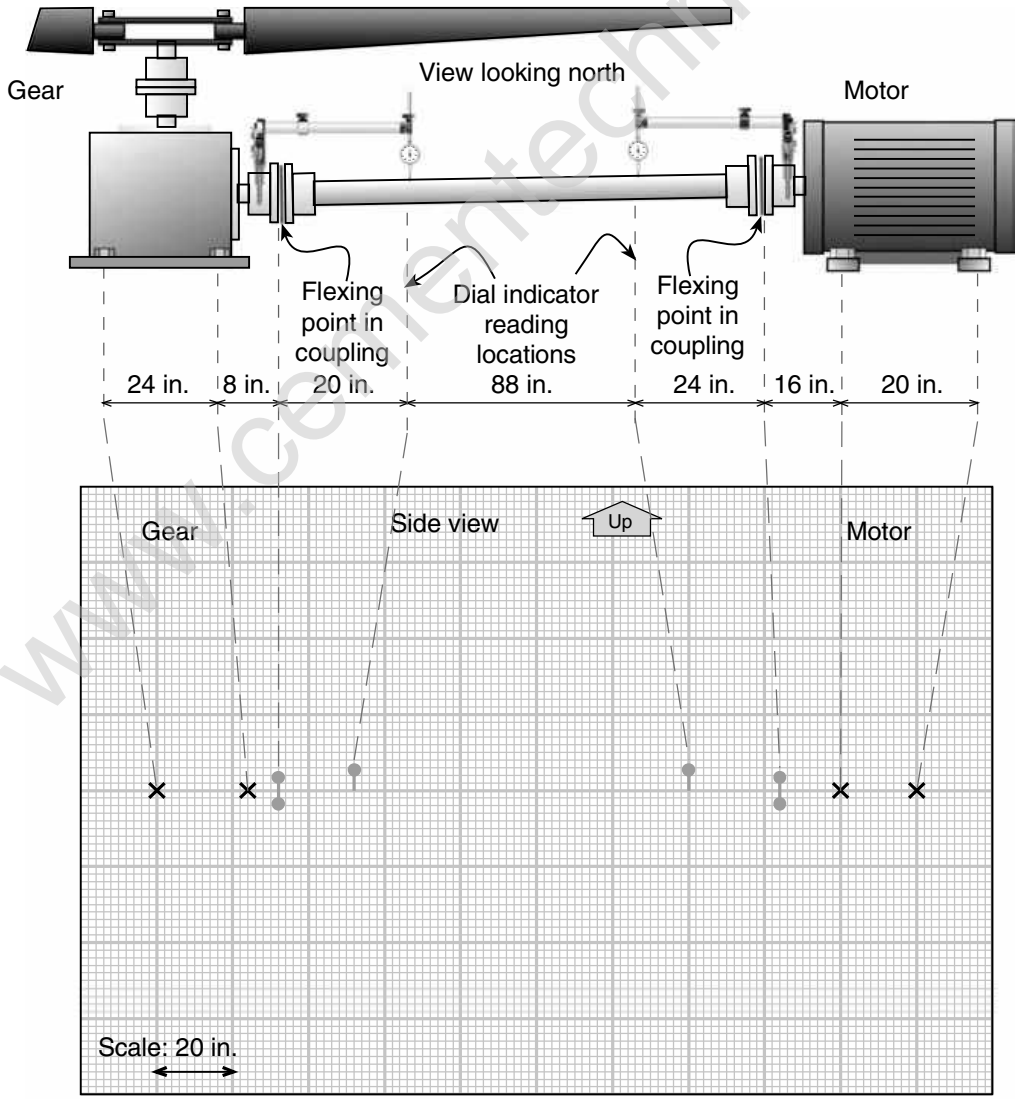


FIGURE 13.6 Dimensional information needed for plotting shaft to coupling spool measurements.

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1. Draw the coupling spool on top of the graph centerline.
2. Start with the top to bottom dial indicator readings (or side to side dial indicator readings if you want to plot the top view) taken from the first machine to the spool piece.
3. At the intersection of the graph centerline and the point where the dial indicator was capturing the readings on the spool piece, place a point half the value of the bottom (or side to side) reading above or below the graph centerline. If the bottom (or side) reading was negative, place a point half the value of the bottom (or side) readings from the graph centerline toward the bottom of the graph. If the bottom (or side) reading was positive, place a point half the value of the bottom (or side) readings from the graph centerline toward the top of the graph. Draw a line through this point and the point where the flexing point and the graph centerline intersect. These two points define the line of sight (i.e., the centerline of rotation) of the first machine.
4. Next, start at the intersection of the graph centerline and the point where the dial indicator was capturing the readings on the spool piece from the bracket attached to the second machine shaft, and place a point half the value of the bottom (or side to side) reading above or below the graph centerline. If the bottom (or side) reading was negative, place a point half the value of the bottom (or side) readings from the graph centerline toward the bottom of the graph. If the bottom (or side) reading was positive,

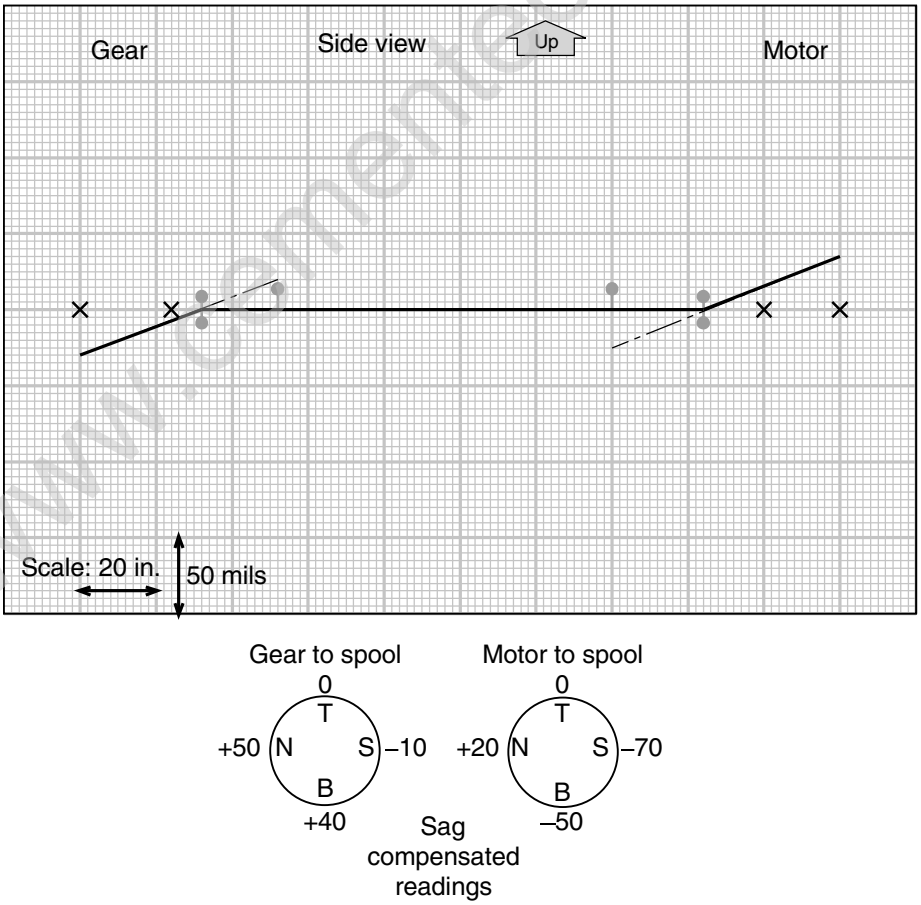


FIGURE 13.7 Shaft to coupling spool side view example alignment model.

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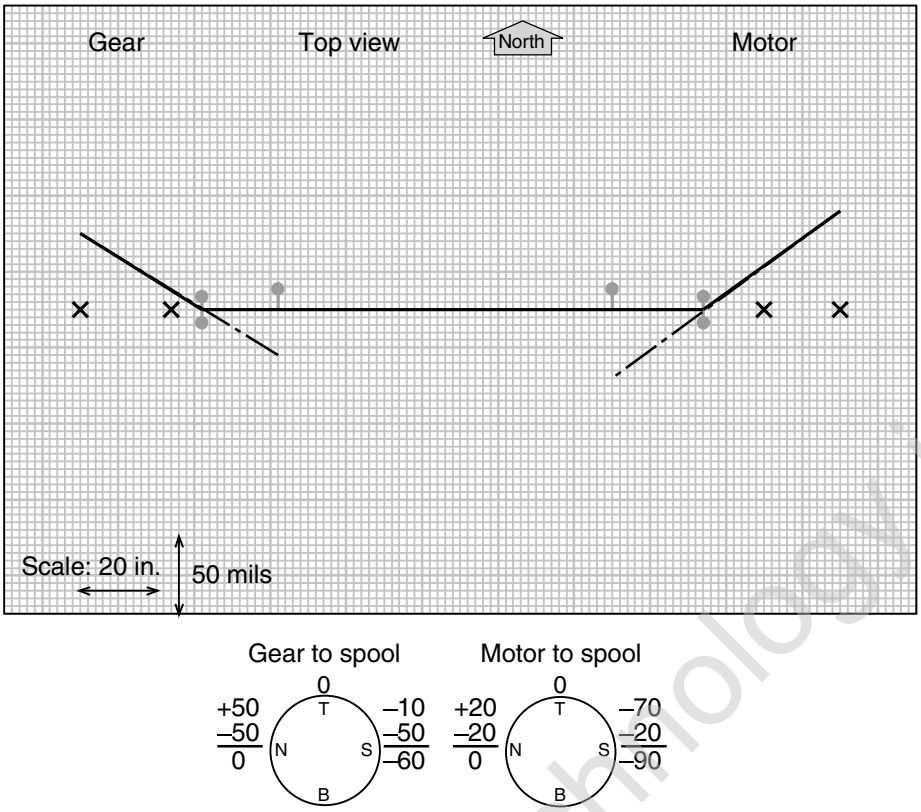


FIGURE 13.8 Shaft to coupling spool top view example alignment model.

place a point half the value of the bottom (or side) readings from the graph centerline toward the top of the graph. Draw a line through this point and the point where the flexing point and the graph centerline intersect. These two points define the line of sight (i.e., the centerline of rotation) of the second machine.

Figure 13.7 and Figure 13.8 show an example of both the side and top view alignment models of a motor and a right angled gearbox where shaft to coupling spool readings were taken.

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14 Face–Face Method

Another method used to measure shaft centerline position data typically for long spans between shaft ends is the face–face technique shown in Figure 14.1. Note that two sets of face readings are taken across each flexing point in the coupling. One of the first shaft-alignment patents filed in the United States embodied this technique, although relatively few people use this method in the field.

Although usually not as accurate as the shaft to coupling spool method, there are occasions where this method must be used instead. The accuracy of this technique increases proportionally with the diameter on which the face readings are measured. The larger the diameter, the more accurate this method becomes. There are some interesting applications of this technique when adapted to measure off-line to running machinery movement as explained in Chapter 16.

Advantages

Better set up if the brackets cannot be attached to the machinery shafts (as in the shaft to coupling spool method) but can be attached to the spool piece.

In the event that the shafts are not to be directly in line with each other but have an intentional centerline offset designed into the drive system, this method is better suited than the shaft to coupling spool method.

Disadvantages

Not as accurate as the shaft to coupling spool method assuming the readings were taken on a relatively small diameter assuming the shafts are to be aligned colinearly.

14.1 BASIC MATHEMATICAL EQUATIONS FOR THE FACE–FACE METHOD

The mathematical relationship between the machinery dimensions and the dial indicator readings captured using the face–face method is shown in Figure 14.2. The equations will solve for the moves that need to be made to correct the misalignment condition (i.e., bring the shafts into a collinear relationship when off-line) on one or the other machine case.

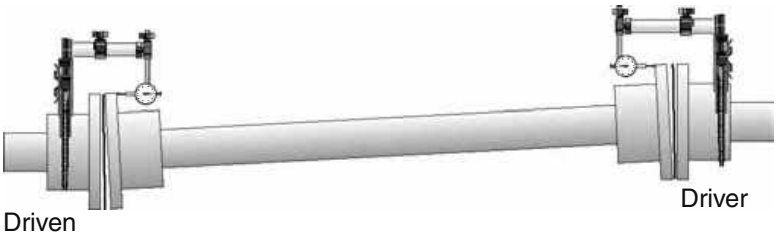
14.2 MODELING THE FACE–FACE METHOD

The face–face coupling spool technique also measures the angle between each shaft centerline and the centerline of the coupling spool (also known as jackshaft or driveshaft) except here the angles are measured across the faces at each flexing point. The coupling spool remains intact (i.e., connected to the shafts) during this procedure.

It is interesting to note that the bracket can be clamped on the shaft or on the coupling spool. Although we are showing the use of brackets and dial indicators to capture the face measurements, it is also possible to take these measurements with feeler gauges at four points

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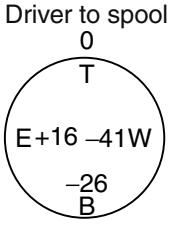


• Procedure •

1. Attach the alignment bracket(s) firmly to the shaft and position the indicator(s) at a point on the "face" of the coupling hub on the spool piece. The bracket could also be attached to the spool piece and the indicator reading the coupling hub face if desired.
2. Zero the indicator(s) at the twelve o'clock position.
3. Slowly rotate the shaft and bracket arrangement through 90° intervals stopping at the three, six, and nine o'clock positions. Record each reading (plus or minus).
4. Return to the twelve o'clock position to see if the indicator(s) re-zero.
5. Repeat step 2 through step 4 to verify the first set of readings.
6. If one bracket was used, mount the bracket onto the other shaft and repeat

Indicator readings log

Driver to spool



Driven to spool

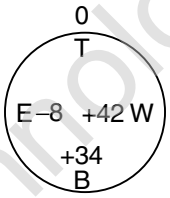


FIGURE 14.1 Face–face method and procedure.

around each coupling face at both ends of the spool piece (see the 16 point method in Chapter 11). The principles applied to the face–rim technique also apply here, in that, the larger the diameter on which the face readings are taken, the more accurate the technique becomes.

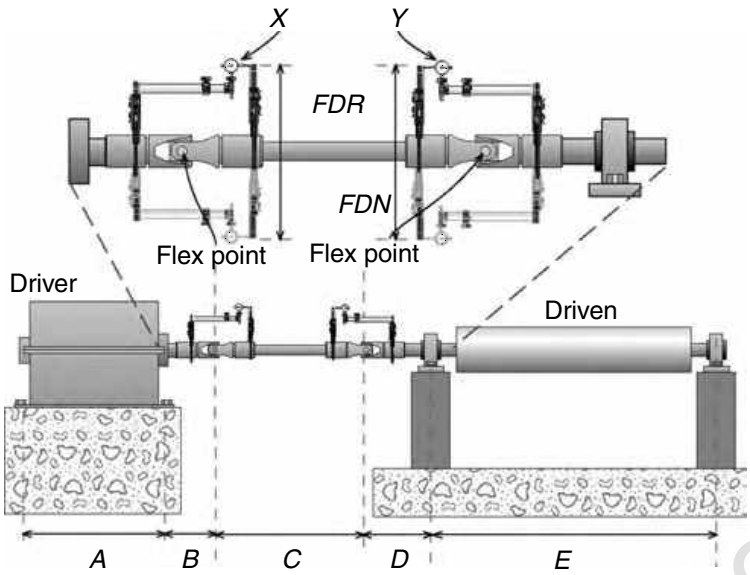
To plot the position of the two shafts and the spool piece, the T bar overlay will be used twice (once at each end of the spool piece or drive shaft). The graph is "dual scaled" similar to the face–rim technique.

There are eight pieces of information that you need for properly constructing the shaft positions using this technique:

1. The distance from the outboard feet to the inboard feet (bolting planes) of the first machine.
2. The distance from the inboard bolting plane of the first machine to the flexing point between the shaft and the coupling spool or drive shaft on the first machine. Note that the point where the bracket is clamped on the shaft (or on the coupling spool) is not relevant. Again, the larger this diameter on which the face readings are taken, the more accurate this technique becomes.
3. The diameter on which the face readings were taken at the flexing point between the first machine and the spool piece or drive shaft.
4. The distance from the flexing point between the shaft and the coupling spool or drive shaft on the first machine to the flexing point between the shaft and the coupling spool or drive shaft on the second machine.
5. The diameter on which the face readings were taken at the flexing point between the second machine and the spool piece or drive shaft.
6. The distance from the flexing point between the shaft and the coupling spool or drive shaft on the second machine to the inboard bolting plane of the second machine.

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where
A, B, C, D, E = distances shown (in.)
FDR = diameter of face readings taken from driver shaft to coupling spool
FDN = diameter of face readings taken from driven shaft to coupling spool
X = face reading difference from driver shaft to coupling spool (from top to bottom or side to side in mils)
Y = face reading difference from driven shaft to coupling spool (from top to bottom or side to side in mils)

$$\text{Inboard feet of driver} = (B+C)\left[\frac{Y}{FDN}\right] - B\left[\frac{X}{FDR}\right]$$

$$\text{Outboard feet of driver} = (A+B+C)\left[\frac{Y}{FDN}\right] - (A+B)\left[\frac{X}{FDR}\right]$$

$$\text{Inboard feet of driven} = (B+C+D)\left[\frac{X}{FDR}\right] + D\left[\frac{Y}{FDN}\right]$$

$$\text{Outboard feet of driven} = (B+C+D+E)\left[\frac{X}{FDR}\right] + (D+E)\left[\frac{Y}{FDN}\right]$$

FIGURE 14.2 Face–face method mathematics for correcting moves on either machine case.

7. The distance from the outboard to the inboard feet (bolting planes) of the second machine.
8. The eight "face" dial indicator readings taken at the top, bottom, and both sides on the coupling spool after compensating for sag (i.e., what a perfect, "no sag" bracket system would have measured). Be aware of the fact that there will probably be two different "sag" amounts at each of the dial indicator locations if the bracket span distances are not the same.

Accurately scale the distances along the length of the drive train onto the graph centerline and the diameters on which the face readings were taken at each flexing point onto the T bar overlays.

The procedure for plotting the face–face technique is as follows:

1. Draw the coupling spool or drive shaft on top of the graph centerline, extending from flexing point to flexing point. It is suggested that you also draw a face diameter line at each end of the coupling spool or drive shaft to aid in pitching the T bars.

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2. Start with the top to bottom face dial indicator readings (or side to side face dial indicator readings if you want to plot the top view) taken across the flex point on the first machine to the spool piece.
3. At the intersection of the graph centerline and the point where the flexing occurs between the first machine shaft and the spool piece, pitch or rotate the T bar overlay to reflect the difference in gap between the top and bottom (or side to side) readings. If the bottom (or side) reading was negative, pitch the T bar clockwise by the full amount of the bottom (or side) readings across the diameter on which the face readings were captured. If the bottom (or side) reading was positive, pitch the T bar counterclockwise by the full amount of the bottom (or side) readings across the diameter on which the face readings were captured. The base of the T represents the centerline of rotation of the first machine. If you are going to use the T bar again at the other flexing point, draw in the position of the shaft so it lines up with the base of the T. The clockwise or counterclockwise pitch of the T bar noted above assumes that you took "front side" face readings similar to the set up shown in Figure 14.3. If you took "back side" face readings, the pitch of the T bar would be opposite the rotation indicated above.

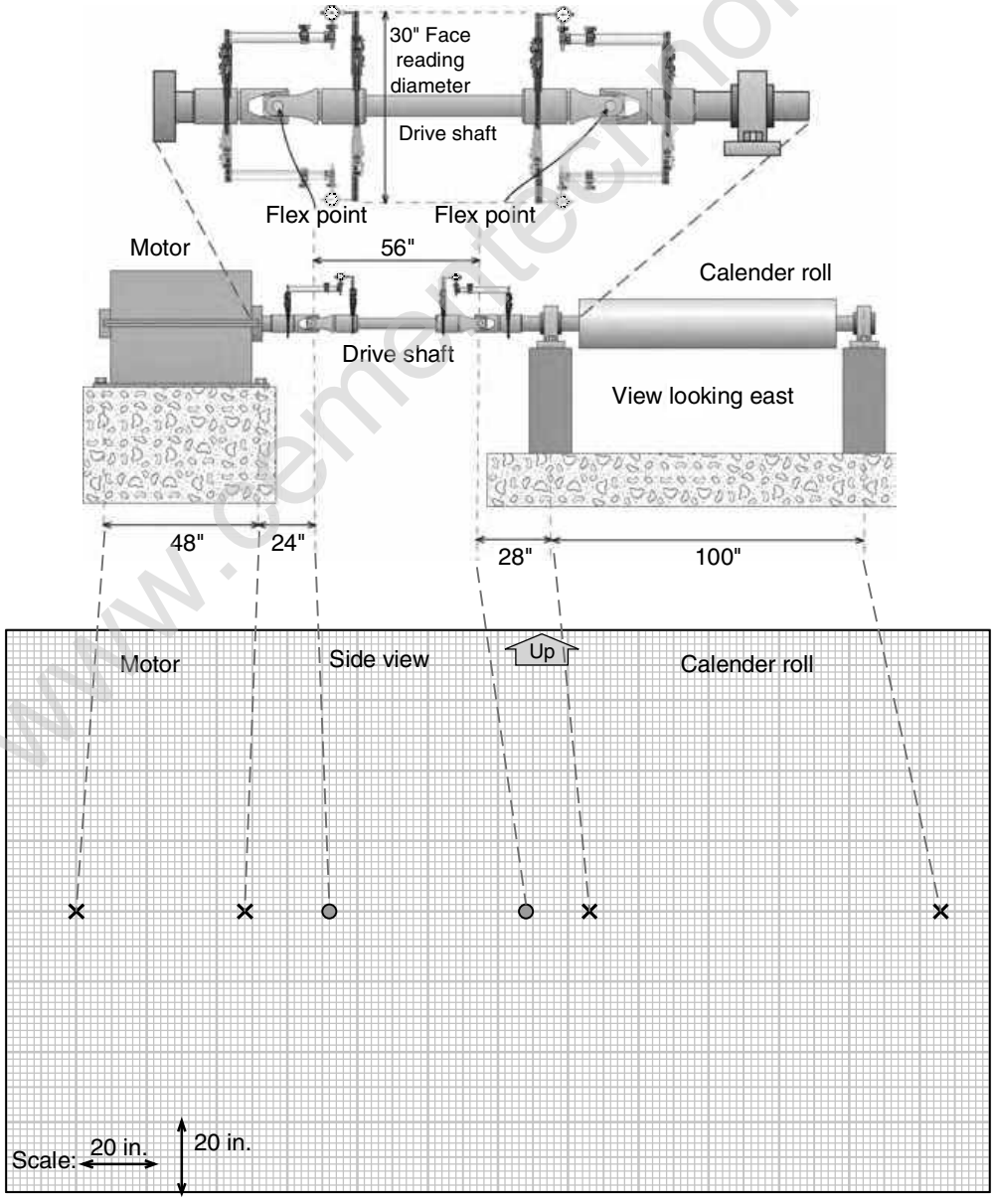


FIGURE 14.3 Dimensional information needed for plotting face–face measurements.

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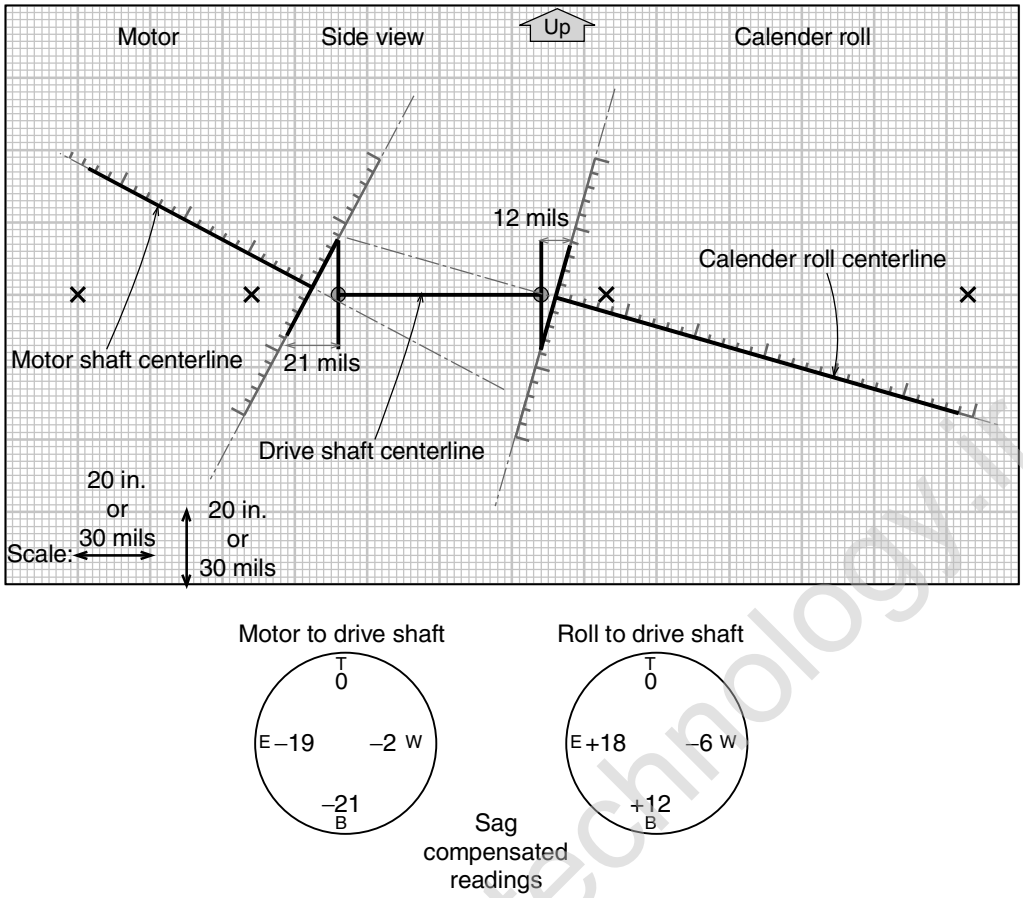


FIGURE 14.4 Face–face side view example alignment model.

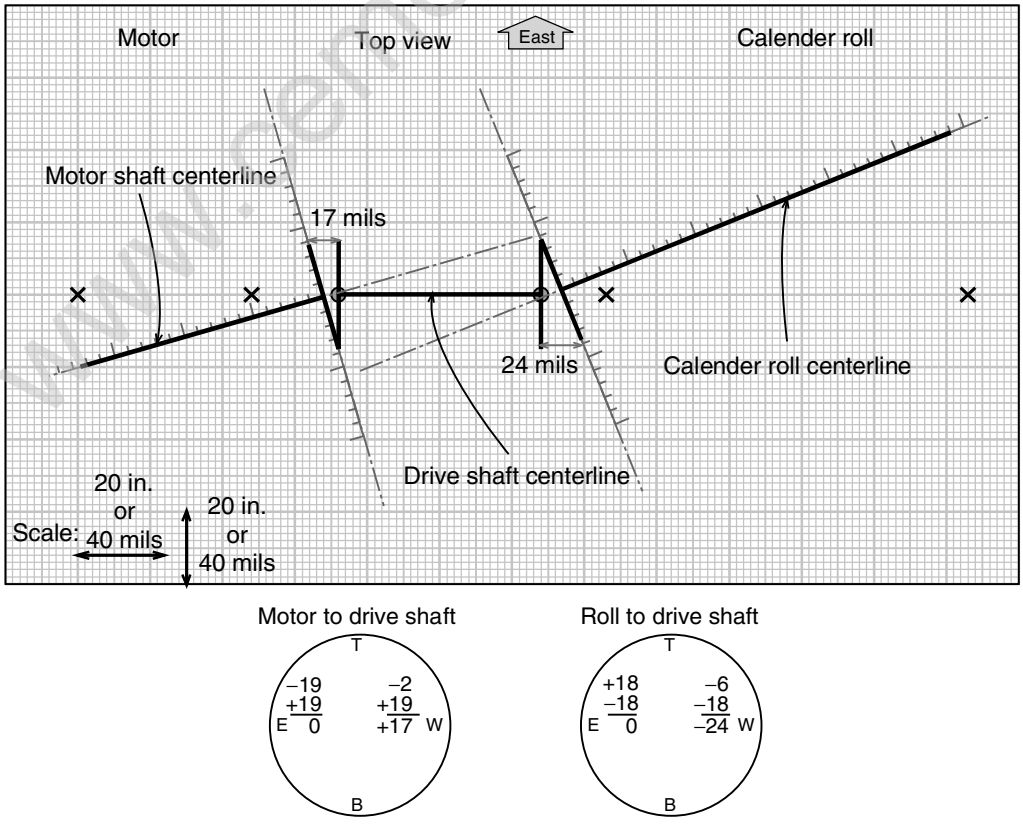


FIGURE 14.5 Face–face top view example alignment model.

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4. Next, start at the intersection of the graph centerline and the point where the flexing occurs between the second machine shaft and the spool piece, pitch or rotate the T bar overlay to reflect the difference in gap between the top and bottom (or side to side) readings. If the bottom (or side) reading was positive, pitch the T bar clockwise by the full amount of the bottom (or side) readings across the diameter on which the face readings were captured. If the bottom (or side) reading was negative, pitch the T bar counterclockwise by the full amount of the bottom (or side) readings across the diameter on which the face readings were captured. The base of the T represents the centerline of rotation of the second machine. Again, draw in the position of the second machine shaft so it lines up with the base of the T. Again, the clockwise or counterclockwise pitch of the T bar noted above assumes that you took front side face readings similar to the set up shown in Figure 14.3. If you took back side face readings, the pitch of the T bar would be opposite the rotation indicated above. Figure 14.4 and Figure 14.5 show an example of both the side and top view alignment models of a motor and a calendar roll where face–face readings were taken.

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15 Electronic and Electro-Optical Shaft Alignment Systems

From relying on the accuracy of our eyesight to using straightedges and feeler gauges and then eventually to shaft clamping brackets and dial indicators, the science of measuring machinery shaft positions has been continually investigated to include new and innovative measuring devices, particularly in light of the technological explosion in electronics. With the advent of the microprocessor chip, silicon photodiodes, and the semiconductor junction diode laser, new inroads have been forged in the process of measuring rotational centerlines that utilize these new electronic devices.

The shaft alignment systems shown in Chapter 6 utilize dial indicators as the shaft position measuring device. This chapter will explore the other currently used types of electronic measuring devices that have been employed to measure shaft alignment.

15.1 OPTICAL ENCODER SYSTEM

One system replaces the dial indicator with an optical encoder (see Figure 6.25). The sensor is, in effect, a mechanically actuated, electronic dial indicator. This system is shown in Figure 15.1.

The MAC10 system is based on the reverse indicator method as explained in Chapter 10. A bracket is attached to each shaft. A bar attached to one of the brackets is then set up to span across the coupling to plunge the encoder stem in half way at the twelve o'clock position. The operator then rotates the shafts to measure one shaft position. A built in inclinometer (device to measure the shaft rotation angle) keeps track of angular position of the assembly. As the stem of the optical encoder is plunged in or retracts outward, the amount of movement observed by the sensor is sent to the operator keypad via cables. Once measurements have been taken on one shaft, the span bar and inclinometer are reversed and the operator again rotates the assembly to capture the measurements on the other shaft. The operator display then solves for the corrective vertical and lateral moves required for the machine designated as the movable machine.

System cost: US$5200.00
System weight: 0.6 lbs (0.3 kg)
Maximum shaft diameter: 4.75 in. (12 cm)
Sensor range: 0.590 in. (15 mm)
Sensor resolution: 0.0004 in. (0.01 mm)
Manufacturers Web site: http://www.spminstrument.com/

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FIGURE 15.1 MAC10 system. (Courtesy of SPM Instruments, Marlborough, CT.)

15.2 LASER–DETECTOR SYSTEMS

The first useable laser shaft alignment measurement system was introduced around 1984. Since then, a few other manufacturers have introduced laser shaft alignment systems. At the time of writing, there are seven companies that currently offer laser shaft alignment measurement systems. The manufacturers of these systems have taken different approaches to use lasers and detectors for measuring the shaft positions. For a review of how lasers and detectors work, refer to Chapter 6 and Figure 6.26 through Figure 6.31.

Figure 15.2 shows the reflected beam principle. A single laser mounted on one-shaft projects its beam to a roof prism, which is mounted on the other shaft. The beam is reflected back 180° and is observed by the photodiode target. The beam is centered in the twelve o'clock position by sliding the roof prism up or down on the support rods and angled horizontally to have the reflected beam strike in the center of the photodiode target.

Figure 15.3 shows the dual beam–dual detector method. One laser–detector assembly is mounted on one shaft with the laser up and the detector down. The other laser–detector assembly is mounted on the other shaft with the laser down and the detector up. Each beam is centered in the opposing photodiode target by pitching each laser up or down and side to side with adjustment screws.

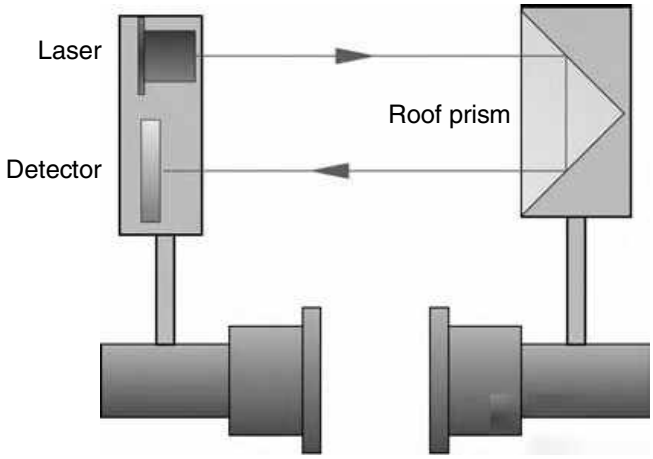


FIGURE 15.2 Single laser–roof prism method.

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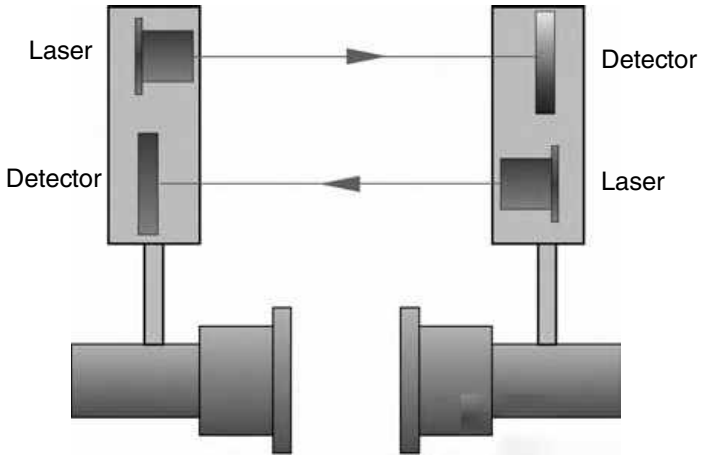


FIGURE 15.3 Dual beam–dual detector method.

Figure 15.4 shows a system where a single laser is projected to a beam splitter where some of the light passes straight through to one photodiode and the rest is reflected at a 90° angle to another photodiode.

Back in 1984 when laser alignment systems were introduced, you had one choice to pick from. Today, there are 7 manufacturers offering 17 different models not including models that have been discontinued but are still used in industry. Which system should you choose? Which tool will meet your current or future needs?

There are four major components to all of the systems: a laser, a detector, tooling to mount the laser and detectors on the shafts, and an operator interface module. Visible lasers are used in the currently offered systems. Some of the systems employ two lasers and two detectors. Some of the systems have a cable that connects from the operator interface module to the lasers and detectors that supply the power needed to fire the lasers and receive the signal from the detectors. Some of the systems have batteries inside the laser or detector enclosures and the data are wirelessly transmitted to the operator interface module via infrared data transmissions. When attached to horizontally mounted shafts, many of the systems also have an angular position sensor that determines the position of the laser or detectors as the shafts are rotated to capture shaft positional measurements.

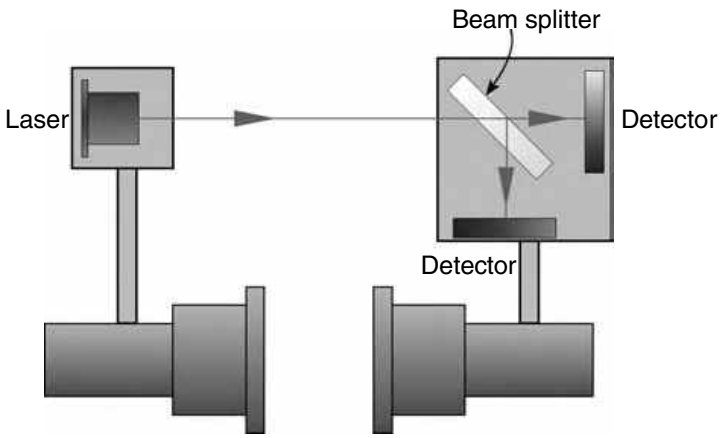


FIGURE 15.4 Single beam–dual detector using beam splitter.

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As laser is a collimated beam of light, you would think that these systems would be great over long distances. That is true in a vacuum such as outer space, but here on Earth we breathe air mixed in with a little bit (sometimes a lot) of water vapor commonly referred to as humidity. Light does strange things when it goes through water or glass. Light will "bend" or diffract so the tight beam that exits the laser diode begins to spread. Light can be bent at precise 90° angles using a pentaprism (five sided cross section) or bounced right back at the laser using a roof prism (triangular cross section). The manufacturers know about the beam dispersion through air so most of the systems have a rated maximum useful distance that is typically around 25 ft.

The accuracy of these alignment systems has very little to do with the laser, it is just a beam of light, typically around 2–3 mm (1/8 in.) as it exits the diode, not 1 μm in diameter as some people suspect. The actual measurement process is accomplished at the photodetector or receiver often referred to as a position sensing detector (PSD) device. As explained in Chapter 6, the PSD measures the distance the laser beam has traversed across the surface of the detector by measuring the electrical current at the beam's starting position and the electrical current at the beam's finishing position, and with many of the current models, many points in between. The photodetector can accurately measure the center of the beam to 1 μm (39 millionths of an inch).

Some manufacturers offer a variety of shaft attachment fixtures that can clamp around the circumference of a shaft, or be held in place on a shaft or coupling hub with permanent magnets, and some that can roll around the outer perimeter of a shaft if it cannot be rotated. To get an idea of the accessories and optional equipment, it is suggested that you contact the manufacturers directly or visit their Web site. There is enough peripheral equipment, hardware and software manuals, and special features in each of these systems to write another book.

The operator interface module has two basic functions. It receives the electrical signals from the PSDs mathematically transforming the information into distance measurements along with information that we give it concerning the machinery we are working on. It then interprets all the data and directs its user through the measurement process and assists in offering information that we will use to correct the misalignment condition. It is not going to align the machinery for us. The software varies from unit to unit and a comparison chart of the features and capabilities of the hardware and software is shown in Figure 15.21 and Figure 15.38 based on the questionnaires sent to each of the manufacturers. By reading what was asked in the questionnaires, you should be able to make an informed decision on what combination of hardware and software features might fulfill your requirements.

15.2.1 Suggestions for Successful Use of Your Laser Alignment System

- Understand how to do alignment another way in case your system fails in the middle of a job or if someone is already using the system for their alignment job.
- Explain to your supervisor or the person who bought the system that the time you spent performing a high-quality alignment job that you had been doing before using this new tool is now not going to get cut in half.
- Read your laser alignment system manuals thoroughly before you start using your tool. It may behoove you to practice on a training simulator before you go out and try to align real machinery.
- Keep the equipment clean. Blobs of grease on the laser or dirt smears on the photodiode lenses will make you look like a goose after you have made four corrective moves and are

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not getting any closer. Keep it clean at the job site and clean it again before you put it back in the carrying case.
- If possible, disengage the coupling or at least loosen up some of the bolts. Never attempt to align an engaged rigid coupling. If you do not understand why, you need to read Chapter 2.
- Measure twice, and align once. After you have all the brackets, lasers and detectors setup on the machinery, sweep at least "two" sets of measurements and compare the results with each other. If the solutions are similar or exactly the same, make the move. If the solutions are different (e.g., more than 5–10 mils), figure out why this is happening (see Chapter 7).
- Ponder several different solutions before you decide what move to make. Many of these systems are finally offering more solutions than just moving the "stationary" machine.
- We are all human. If you dropped the laser or detectors and think they hit really hard, have them checked or calibrated.
- These are tools with software. They are pretty smart, but they will never be as smart or as observant as you are.

Advantages of laser measurement systems:
- Bracket sag does not occur with laser beams but the user must ensure the brackets that hold the lasers and detectors are firmly attached to the shafts.
- Accuracies of ± 3 μm or better.
- Some systems allow for the coupling to be disconnected when capturing readings.
- Systems include an operator keypad or display module that prompts the user through the measurement steps and calculates the moves for one of the machines (some systems can solve for a variety of movement solutions, see Chapter 8).
- Operator does not need to perform mathematical equations or manually plot the shaft positions to solve for some of the corrective moves.
- Some systems are capable of detecting a soft foot condition.
- OL2R data can be entered into the operator interface module on some systems (see Chapter 9).
- Some of the operator interface modules have multiple purpose functions that extend the capabilities of the system.

Disadvantages of laser measurement systems:
- Relatively expensive compared to dial indicator–based systems.
- Range of measurement somewhat limited because most manufacturers use either 10 mm $\times$ 10 mm or 20 mm $\times$ 20 mm detectors. Some systems offer range extension capabilities.
- Most systems are incapable of correcting a soft foot condition.
- Incapable of measuring runout conditions (i.e., it is possible to align bent shafts and not know it).
- Some systems only determine moves for one of the two machines (see stationary–movable concept in Chapter 8).
- Some systems require that the coupling be bolted in place when capturing readings.
- It is recommended that both shafts be rotated during the measurement process.
- Sometimes it is difficult to capture readings in bright sunlight or well-lit areas.
- Accuracy potentially reduced in the presence of excessive steam or heat or if lenses are dirty.
- Calibration suggested on an annual basis.
- Nonlinearity of detectors requires software linearization, which prevents exchanging parts from one system to another.

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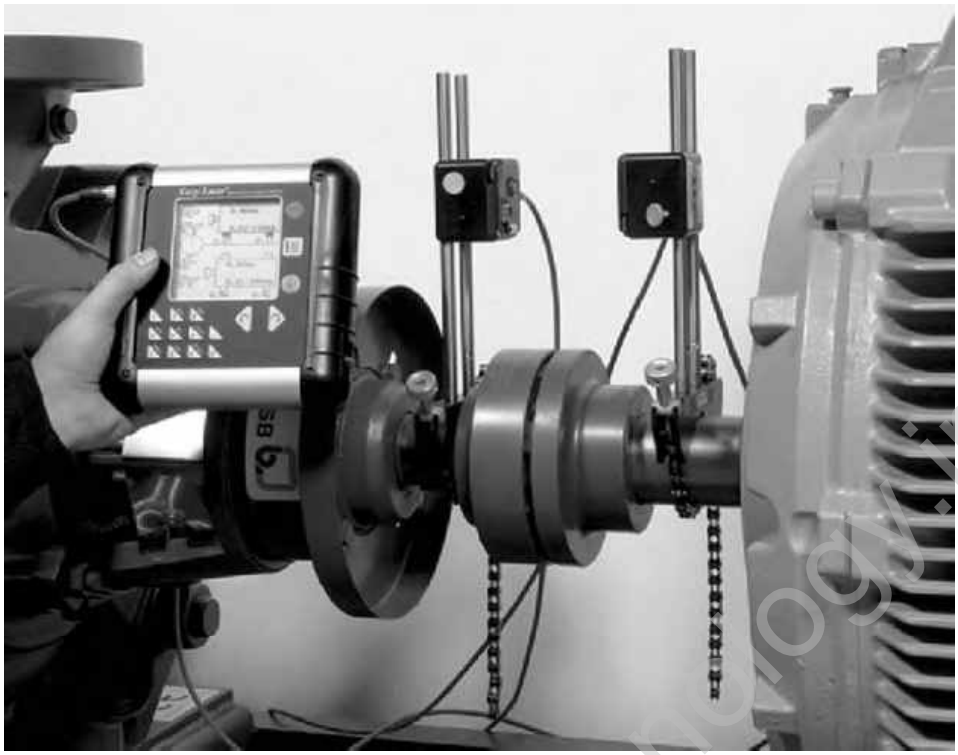


FIGURE 15.5 Easy-Laser system. (Courtesy of Damalini, Molndal, Sweden. With permission.)

15.2.2 Damalini Systems

The Easy-Laser alignment products center around a common operator display that interfaces with a broad array of different lasers and detectors used for a wide variety of alignment tasks that include V-belt—sheave alignment and geometric measurement capabilities. Only the shaft alignment systems will be discussed in this chapter, as the other products will be covered in later chapters.

Figure 15.5 shows the shaft alignment measurement system attached to machinery shafts. This shaft alignment system is based on the reverse indicator method and uses a dual beam–dual detector system as shown in Figure 15.3. A bracket is attached to each shaft and the laser–detector heads are then attached to each bracket. The operator then aims the lasers toward the detector targets on the opposing shaft and centers each beam. Inclinometers are incorporated into the heads to measure the angular position of the shafts. A wireless transmission sends the detector signals to the operator display. The operator then rotates the shafts to capture the shaft positions. Once the measurement process has been completed, the user then obtains the corrective moves for the machine selected as the movable machine.

The model D279 display unit interfaces with the laser–detector heads in the basic D450 model (see Figure 15.6), which does horizontally mounted machinery and has a soft foot function. It includes the D279 display unit loaded with two programs, a leatherette case, two cables, two laser–detector heads, two shaft clamping brackets with chains, two sets of extension rods for holding the laser–detectors on the brackets, a measuring tape, operations manual, and a carrying case for the equipment.

D450 System cost: not supplied
Manufacturer's Web site: http://www.damalini.com/

The D505 model (see Figure 15.7) comprises the same equipment as the D450 model but also includes: 10 total alignment programs, two offset brackets, two magnetic bases, and an

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FIGURE 15.6 D450 system. (Courtesy of Damalini, Molndal, Sweden. With permission.)

EasyLink Windows program with PC cable. It incorporates the EasyTurn features that enable measurement of shaft with only 40° of rotation, functions for entering information on OL2R movement (see Chapter 16), and reference foot locking. Software programs are installed in the display unit for horizontal, vertical or flange-mounted machinery, cardan shaft coupled machinery, and alignment of multiple element machine drive trains (up to ten machines).



FIGURE 15.7 D505/525 system. (Courtesy of Damalini, Molndal, Sweden. With permission.)

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The D525 model has the same equipment as the D450 model but also includes: 20 total alignment programs, two offset brackets, two magnetic bases, and an EasyLink Windows program with PC cable. It too incorporates the EasyTurn features that enables measurement of shaft with only 40° of rotation, functions for entering information on OL2R movement (see Chapter 16), reference foot locking. Twenty software programs are installed in the display unit for horizontal, vertical or flange-mounted machinery, cardan shaft coupled machinery, alignment of multiple element machine drive trains, and programs for geometric measurements should you decide to purchase the hardware for those type of measurements.

D505 system cost: not supplied
D525 system cost: $11500.00

15.2.3 Emerson Process Management System

This shaft alignment system was developed by Computational Systems Inc. in Knoxville, Tennessee. It is based on the reverse indicator method and uses a dual beam–dual detector system as shown in Figure 15.8. A bracket is attached to each shaft and the laser–detector heads are then attached to each bracket. The operator then aims the lasers toward the detector targets on the opposing shaft and centers each beam. Through the touch screen display, the operator is prompted through the alignment process.

CSI 2130 system cost: $12495.00
Manufacturer's Web site: http://www.emersonprocess.com/

15.2.4 Fixturlaser Systems

This shaft alignment system is based on the reverse indicator method and uses a dual beam–dual detector system as shown in Figure 15.3. A bracket is attached to each shaft and the laser–detector heads are then attached to each bracket. The operator then aims the laser



FIGURE 15.8 CSI 2130 system. (Courtesy of Emerson Machinery Health Management business. With permission.)

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FIGURE 15.9 Shaft 30 system. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)

toward the detector targets on the opposing shaft and centers each beam. Using touch screen display, the operator is prompted through the alignment process.

The Shaft 30 system shown in Figure 15.9 is the entry level model allowing the user to align horizontally or vertically mounted machines, conduct a repeatability test, check for soft foot conditions, fix specific bolting planes for base- or bolt-bound conditions (called "Feetlock"), and allows measurements of shaft positions with as little as 60° of rotation. The display also incorporates a user selectable sleep and auto resume mode. The Shaft 100 system shown in Figure 15.10 incorporates all of the features of the Shaft 30 system but includes data communication to a personal computer, OL2R information input (see Chapter 16), and an alignment tolerance table. The Shaft 200 system shown in Figure 15.11 incorporates all of the

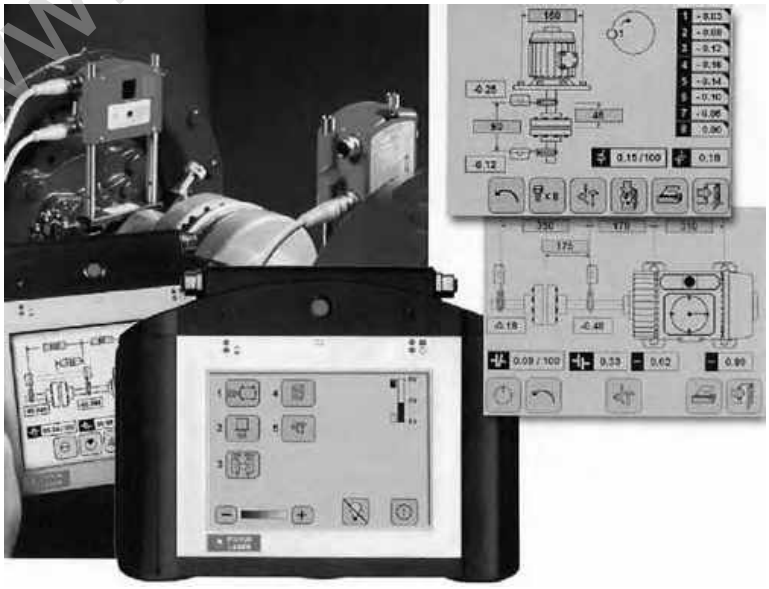


FIGURE 15.10 Shaft 100 system. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)

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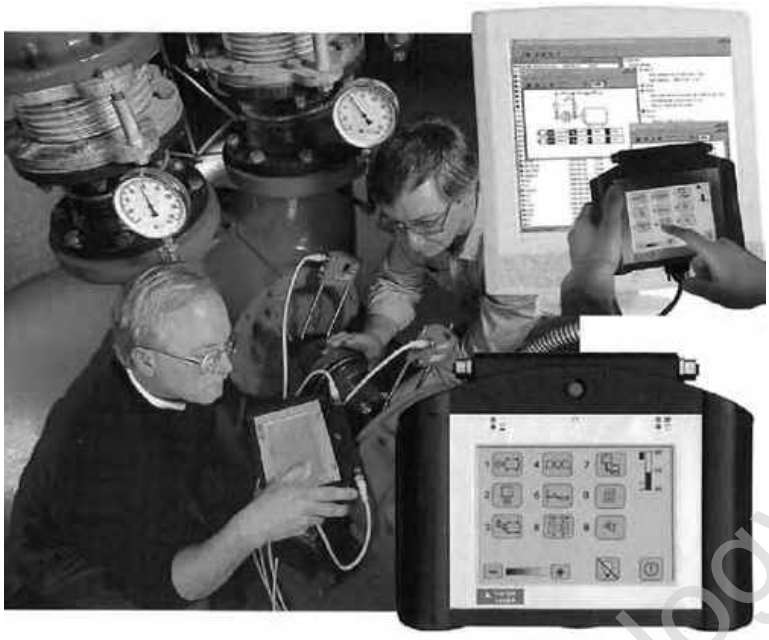


FIGURE 15.11 Shaft 200 system. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)

features of the Shaft 100 system but also includes: alignment of offset mounted equipment (Cardan/U-joint driven machinery), alignment of multiple element drive systems, can be used for measuring as well as entering OL2R movement information, two magnetic bases, and a system printer.

Shaft 30 system cost: $9995.00
Shaft 100 system cost: $15500.00
Shaft 200 system cost: not supplied
Manufacturer's Web site: http://www.fixturlaser.com/

15.2.5 Hamar Systems

The operating principle of the Hamar laser alignment systems is based on a single laser, dual detector using a beam splitter as shown in Figure 15.4. A bracket is attached to each shaft. The laser is attached to one of the brackets and the detector target to the other bracket. The operator then positions the laser and detector assemblies to center the beam in the detector target area.

The Model S-640 system is shown in Figure 15.12. The detector target output is connected to a computer interface module (Model R-358) via a cable, which is then connected to a laptop computer via the serial communications port. An alignment software program supplied with the system is loaded into your laptop computer and the software prompts the user through the alignment process.

The Model S-650 system is shown in Figure 15.13. The detector target output is transmitted via infrared data transmission to a receiver–computer interface module (Model A-908), which is connected to a laptop computer via the serial communications port. An alignment software program and laptop computer are supplied with the system.

Model S-640 system cost: not supplied
Model S-650 system cost: not supplied
Manufacturer's Web site: http://www.hamarlaser.com/

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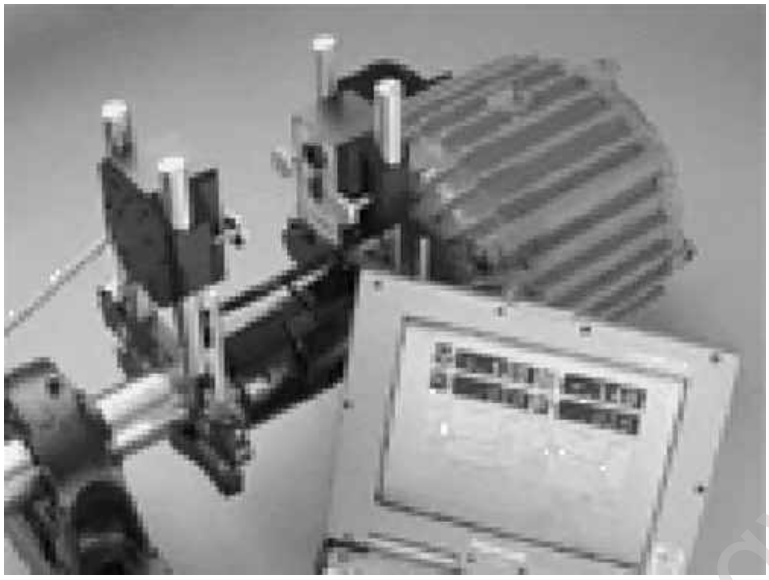


FIGURE 15.12 Model S-640. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)

15.2.6 Prüftechnik Systems

The first useable laser shaft alignment system was introduced by Prüftechnik in 1984. The operating principle of this system is shown in Figure 15.2 where a single laser, mounted on one shaft, is aimed at a roof prism, mounted on the other shaft, which reflects the beam directly back toward its point of origin but offsets the beam a specified distance so the laser



FIGURE 15.13 Model S-650. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)

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FIGURE 15.14 Pocketalign system. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

can strike onto the surface of the photodetector target. After attaching the brackets to each shaft, the operator installs the laser–detector head onto one of the brackets and the roof prism onto the other bracket. Using the display and keypad, the operator enters information about the drive system being aligned and once the laser beam starts operating, the operator adjusts the position of the roof prism to center the returning laser beam onto the photodiode target. The operator then proceeds through the instructions prompted via the display and keypad to capture the shaft positions.

The first Optalign system incorporated an infrared laser (not visible to the naked eye) but as visible lasers became available, they were incorporated into subsequent models.

The Pocketalign system is shown in Figure 15.14. This system utilizes a personal digital assistant (PDA) for the operator and keypad. The PDA operating system requires Microsoft Mobile 2003 and a CF II interface card is required. It uses the same laser–detector head as the Smartalign system. The PDA can be purchased through the manufacturer or you can buy your own. Communication cabling is provided to transfer files from the PDA to a personal computer for file archiving. Continuous sweep measurement mode software can be purchased separately.

The Optalign Plus system as shown in Figure 15.15, now in its fourth generation, has been the workhorse of their product line. The operational software is imbedded into the laser–detector sensor assembly, not the operator keypad and display. It is offered in two basic configurations, the ''Entry Level'' model and the ''All Features'' model. This system is also fully configurable with the exact features and functions the user desires, ''à la carte.'' Intrinsically safe models are also available. The system comes with the laser–detector transducer, the prism, an operator keypad and display, a 6 ft (2 m) cable, brackets for clamping onto shafts up to 8.5 in. in diameter, posts (200 and 600 mm) for attaching the laser–detector and prism to the brackets, a computer cable and adapter, tape measure, manual, cleaning cloth, protective covers, and a carrying case. Optional firmware can be purchased which can

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FIGURE 15.15 Optalign Plus system. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

save either 25 or 99 files, enter both vertical and horizontal target specifications and thermal growth data, vertical shaft alignment, multipoint measurement mode, alignment of spacer shafts, machinery with 6 ft, "static feet" function for bolt-bound or base-bound machines, alignment tolerances, and extended measurement range (known as InfiniRange).

Optalign Plus Entry Level system cost: $6600.00
Optalign Plus Intrinsically Safe Entry Level system cost: $10575.00
Optalign Plus All Features system cost: $13500.00
Optalign Plus Intrinsically Safe All Features system cost: $16875.00
Manufacturer's Web site: http://www.pruftechnik.com/

The Smartalign system is shown in Figure 15.16. The laser–detector transducer visually looks the same as the Optalign Plus where the operational software is actually imbedded in the transducer. In the case of the Smartalign system, the operational software is actually imbedded in the operator keypad and display. The system comes with: the laser–detector transducer, the prism, an operator keypad–display, a 6 ft (2 m) cable, brackets for clamping onto shaft up to 8.5 in. in diameter, 20 different posts of various lengths for attaching the laser–detector and prism to the brackets, "smartREADER®" software, a computer cable and adapter, tape measure, manual, cleaning cloth, protective covers, and a carrying case.

Smartalign system cost: $14800.00

The Rotalign Pro system is a more advanced system than the systems described above and is shown in Figure 15.17. The system comes with an alignment computer, a laser sender, a receiver, a 6 ft (2 m) cable, brackets for clamping onto shaft up to 8.5 in. in diameter,

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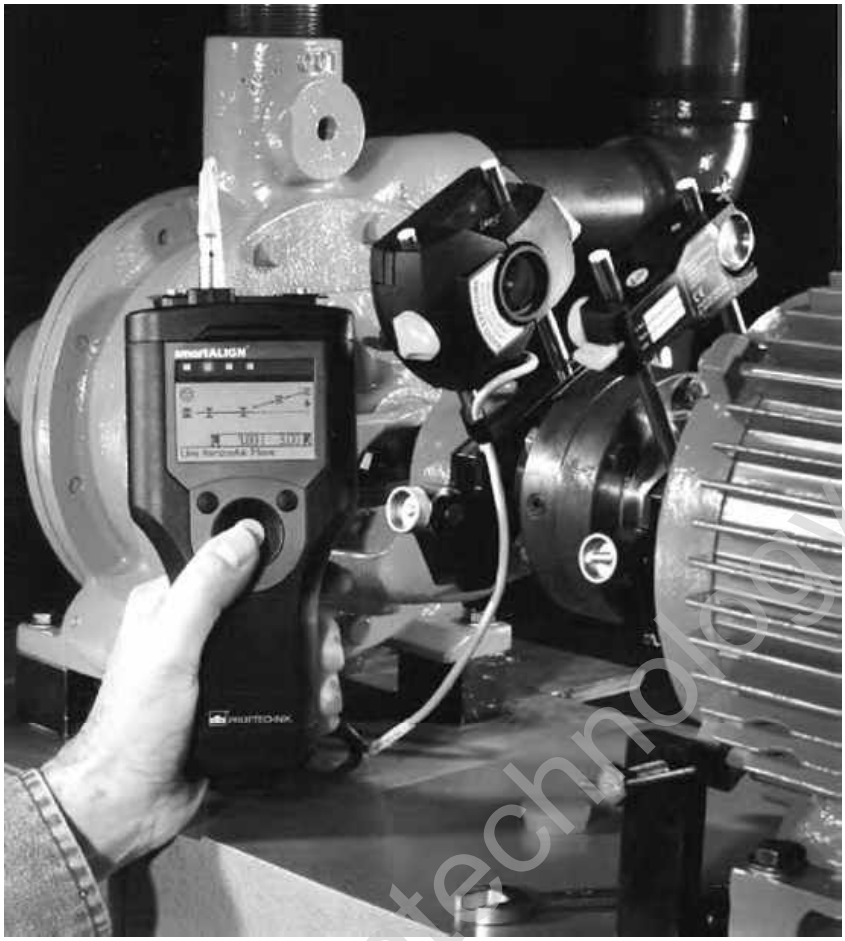


FIGURE 15.16 Smartalign system. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

20 different posts of various lengths for attaching the laser–detector and prism to the brackets, beam adjustment tube, a computer cable and adapter, tape measure, manual, cleaning cloth, protective covers, and a carrying case. The basic software of this multifunction device enables one to align horizontally or vertically mounted machinery, coupled or uncoupled shafts, alignment of multiple element drive systems, soft foot detection, target specifications and thermal growth entry for all machines in the train, infrared data transmission to computer, and U-joint (Cardan shaft) alignment. Additional software packages enables alignment of machinery bores (Boralign) and straightness and flatness (Levalign) measurement.

Rotalign Pro system cost: $18900.00
Rotalign Pro Intrinsically Safe system cost: $23700.00
Boralign add-on package cost: $8500.00
Boralign Long Range upgrade package cost: $6969.00

15.2.7 SPM Instrument Inc. System

The LineLazer system is based on the reverse indicator method and uses a dual beam–dual detector system (See Figure 15.19). The LineLazer is an accessory that interfaces with the Leonova system, which is capable of doing vibration analysis and balancing with the appropriate add-on hardware.

A bracket is attached to each shaft and the laser–detector heads are then attached to each bracket. The operator then aims the laser toward the detector targets on the opposing shaft

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FIGURE 15.17 Rotalign Pro system. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

and centers each beam. Inclinometers are incorporated into the heads to measure the angular position of the shafts. A wireless transmission sends the detector signals to the operator display. The operator then rotates the shafts to capture the shaft positions. Once the measurement process has been completed, the user then obtains the corrective moves for the machine selected as the movable machine.

LineLazer system cost: US$5500.00 (does not include the cost of the Leonova interface system)

Manufacturer's Web site: http://www.spminstrument.com/

15.2.8 Vibralign System

The Shaft Hog system is shown in Figure 15.20 and is based on the reverse indicator method and uses a dual beam–dual detector system. The system comes with: two laser–detector units, cables, display box, shaft clamp brackets, mounting rods, measuring tape, manual (printed and CD-ROM), and a pocket guide.

Shaft Hog system cost: US$4250.00
Manufacturer's Web site: http://www.vibralign.com/

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FIGURE 15.18 Rotalign Ultra system. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

15.3 LASER SYSTEM MANUFACTURERS HARDWARE SPECIFICATIONS

A questionnaire was sent out to all of the above manufacturers concerning the pricing, specifications, and features of their systems. The questions are listed below. A summary of their answers is shown in Figure 15.21 through Figure 15.32.

Laser shaft alignment system questionnaire

1. Are you a manufacturer of a laser shaft alignment system?
   ____ Yes
   ____ No

2. Are you a distributor of a laser shaft alignment system? If so, please list the manufacturer of the laser shaft alignment system. Please indicate the company, address, phone number, contact person, and Web site (if applicable) of the manufacturer. If you are a distributor only, skip the remaining questions.
   Distributor for:
   Model # _____
   Manufacturer name _____
   Address _____
   P.O. Box • Suite _____
   City _____
   State/Province _____

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FIGURE 15.19 Leonova LineLazer system. (Courtesy of SPM Instruments, Marlborough, CT. With permission.)

ZIP/Postal code _____
Country _____
Phone # _____
Fax # _____
E-mail address _____
Web site _____

Questions for the manufacturer of a laser shaft alignment system

3. Describe the basic operation of your laser measurement system. Use drawings or diagrams to enhance your explanation if possible.
4. What is the overall weight of the entire system?
   _____ (pounds, kilograms) system weight
5. What is the price (or price range) of the system in U.S. dollars?
   _____ (U.S. dollars)
6. How is the unit powered?
   _____ internal batteries
   _____ AC power source
   _____ batteries or AC power source

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FIGURE 15.20 V180 system. (Courtesy of Vibralign, Richmond, VA. With permission.)

7. If batteries are used, what sizes are required and how many are needed? Are the batteries rechargeable?
   Battery size and type _____
   _____ Rechargeable
   _____ Not rechargeable
8. What battery life can be expected for continuous use on battery power?
   _____ (minutes, hours) expected battery life
9. Can the unit be plugged into AC electric supplies? Can it work on either 50 or 60 Hz circuits and voltage ranges from 90 to 220 VAC?
   _____ Yes
   _____ No
10. What type of laser is used in your system and how does it work (beam frequency in nm, continuous or pulse? Pulse duration, beam diameter at collimator exit, beam diameter at 1, 3, 10 m, power output, safety precautions, etc.)?
    Type of laser _____
    Beam frequency _____
    _____ continuous
    _____ pulse duration _____
    _____ (inches, millimeters) beam diameter at exit
    _____ (inches, millimeters) beam diameter at 1 m
    _____ (inches, millimeters) beam diameter at 3 m
    _____ (inches, millimeters) beam diameter at 10 m
    _____ (milliwatts) power output
    Safety precautions _____
11. If you are using semiconductor junction diode lasers, which typically produce a low-quality beam that is divergent, elliptical, and astigmatic, what optical correction is incorporated in your system to correct these deficiencies?

    _____
    _____
    _____

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Question Number	Manufacturer System/Model Manufacturer Web Site	Damalini D450 www.damalini.com	Damalini D505 www.damalini.com	Damalini D525 www.damalini.com
5	System price (U.S. dollars)	4,995		11,500
	Basic System Information			
4	Total unit weight (lbs)	27 lbs (12 kg)	27 lbs (12 kg)	27 lbs (12 kg)
3	Alignment operating principle	Reverse indicator	Reverse indicator	Reverse indicator
21	Intrinsically safe/explosion proof models?	No	No	No
16	Shock protection	Internal shock protection	Internal shock protection	Internal shock protection
20	Operating temperature range	32°F–122°F	32°F–122°F	32°F–122°F
22	Environmental protection	FDA approved, 21 CFR 1040. 10&11, SS-EN-60825-1-1994	FDA approved, 21 CFR 1040. 10&11, SS-EN-60825-1-1994	FDA approved, 21 CFR 1040. 10&11, SS-EN-60825-1-1994
	Laser Emitter (*Source*)			
9	Laser type	Visible red diode	Visible red diode	Visible red diode
9	Number of lasers used	2	2	2
9	Wavelength	635–670 nm	635–670 nm	635–670 nm
9	Beam diameter at exit	3 mm	3 mm	3 mn
9	Max. beam power output	1 mW	1 mW	1 mW
9	Laser safety class	Class 2	Class 2	Class 2
	Max. recommended distance			
6	Laser power supplied by …	Cable from D279 display unit	Cable from D279 display unit	Cable from D279 display unit
	Cable length	7 ft (2 m) × 1, 5, 10 m optional	7 ft (2 m) × 1, 5, 10 m optional	7 ft (2 m) × 1, 5, 10 m optional
	Dimensions *W*×*H*×*D* (inches)			
	Laser–Detector (*Receiver*)			
12	Type	Single axis photodiode (two)	Single axis photodiode (two)	Single axis photodiode (two)
12	Detector measurement planes	1	1	1
12	Detector measurement area	10 mm × 10 mm	10 mm × 10 mm	10 mm × 10 mm
12	Number of detectors used	2	2	2
12	Linearity			
12	Accuracy/resolution	1 μm	1 μm	1 μm
12	Optical filtering?	Grey filter	Grey filter	Grey filter
15	Beam position data transfer via …	Cable	Cable	Cable
	Detector power supplied by …	D279 display unit	D279 display unit	D279 display unit
	Battery life at full charge	24–48 h	24–48 h	24–48 h
	Dimensions *W*×*H*×*D* (inches)			
	Interchangeable with other display units?	No	No	No

FIGURE 15.21 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	Damilini D450 www.damalini.com	Damilini D505 www.damalini.com	Damilini D525 www.damalini.com
	Operator Interface			
14	Alignment computer and software included?	Yes	Yes	Yes
14	Dimensions W×H×D (inches)			
	Weight (with battery)			
	Display screen (type and size)			
	Software installation	Pre-installed	Pre-installed	Pre-installed
	Battery type/number requirements	4 C (R14) batteries	4 C (R14) batteries	4 C (R14) batteries
	Battery life at full charge	24–48 h	24–48 h	24–48 h
6	Optional AC power and requirements	90–220 VAC/50–60 Hz	90–220 VAC/50–60 Hz	90–220 VAC/50–60 Hz
30	Enter off-line to running movement?	No	No	No
14	Interface system with standard computers?	Yes, RS232	Yes, RS232	Yes, RS232 serial
	Store alignment jobs?			
18	Turned OFF by mistake?	Start over	Start over	Start over
	Operational Info			
25	Shafts must be rotated together?	No	No	No
26	How are rotational positions measured?	Internal inclinometer and external bubble	Internal inclinometer and external bubble	Internal inclinometer and external bubble
25	Max. allowable rotational backlash			
31	Can be used on vertical shafts?	Yes	Yes	Yes
31	Shaft angular positon measured for vertical			
32	English or metric units?	Yes	Yes	Yes
	Additional Features and Accessories			
	Shaft clamping brackets included?	Yes	Yes	Yes
27	Minimum shaft diameter	0.625″ (15 mm)	0.625″ (15 mm)	0.625″ (15 mm)
27	Maximum shaft diameter	No limit (with extra chain)	No limit (with extra chain)	No limit (with extra chain)
28	Minimum sensor height from shaft surface	0.5″ (10 mm)	0.5″ (10 mm)	0.5″ (10 mm)
28	Maximum sensor height from shaft surface			
30	Adapted for measuring OL2R?	Yes (with optional brackets)	Yes (with optional brackets)	Yes (with optional brackets)
	Calibration			
35	Recommended calibration interval	36 months	36 months	36 months
	Calibration cost	400	400	400
35	Send to factory?	Yes (or authorized dealer)	Yes (or authorized dealer)	Yes (or authorized dealer)
35	Traceable to standards?	Yes	Yes	Yes
36	User can buy calibration equipment?	No	No	No
36	Cost of calibration equipment	N/a	N/a	N/a
34	Warranty period	12 months	12 months	12 months
37	Can user upgrade to new model?	Yes	Yes	Yes
37	Upgrade cost	$2000.00–$5000.00	$2000.00–$5000.00	$2000.00–$5000.00

FIGURE 15.22 Laser system hardware comparison chart

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Question Number	Manufacturer System/Model Manufacturer Web Site	Emerson Process Management UltraSpec 8200	FixturLaser Shaft 30 www.fixturlaser.com	FixturLaserAB Shaft 100 www.fixturlaser.com
5	System price (U.S. dollars)	12,495	9,995	15,500
	Basic System Information			
4	Total unit weight (lbs.)	30 lbs (13.6 kg)	19.8 lbs (9 kg)	26 lbs (11.8 kg)
3	Alignment operating principle	Reverse indicator	Reverse indicator	Reverse indicator
21	Intrinsically safe/explosion proof models?	No	No	No
16	Shock protection	Yes (foam protect in case)	Not answered	Not answered
20	Operating temperature range	15°–120°F	0°C–50°C	0°C–50°C
22	Environmental protection	Sealed detector and laser	IP65	IP65
	Laser Emitter (*Source*)			
9	Laser type	In-Ga-AI-P semiconductor	Visible red diode	Visible red diode
9	Number of lasers used		2	2
9	Wavelength	670 nm	650–670 nm	650–670 nm
9	Beam diameter at exit	2 mm	2 mm	2 mm
9	Max. beam power output	< 1 mW	1 mW	1 mW
9	Laser safety class	Class 2	Class 2	Class 2
	Max. recommended distance	30 ft	21 ft (7 m)	66 ft (20.1 m)
6	Laser power supplied by…	Battery (internal to head)	Cable from display unit	Cable from display unit
	Cable length	N/a	10 ft (3 m)	10 ft (3 m)
	Dimensions $W \times H \times D$ (inches)			
	Laser–Detector (*Receiver*)			
12	Type	Dual axis photodiode	Single axis photodiode (two)	Single axis photodiode (two)
12	Detector measurement planes	2	1	1
12	Detector measurement area	0.4 × 0.4 sq. in. (10 × 10 sq. mm)	10 mm × 10 mm	20 mm × 20 mm
12	Number of detectors used	2	2	2
12	Linearity	99.9% over 90% of active area	±1% × ±0.003 mm	±1% × ±0.003 mm
12	Accuracy/resolution	Not answered	1 micrometer	1 micrometer
12	Optical filtering?	Yes	Yes	Yes
15	Beam position data transfer via…	Cable or infrared beam (no cable)	Cable	Cable
	Detector power supplied by…	Battery (internal to head)	Cable from display unit	Cable from display unit
	Battery life at full charge	4 h	24 h	24 h
	Dimensions $W \times H \times D$ (inches)			
	Interchangeable with other display units?		No	No

FIGURE 15.23 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	Emerson Process Management UltraSpec 8200	FixturLaser Shaft 30 www.fixturlaser.com	FixturLaserAB Shaft 100 www.fixturlaser.com
	Operator Interface			
14	Alignment computer and software included?	Yes	Yes	Yes
14	Dimensions $W \times H \times D$ (inches)			
	Weight (with battery)			
	Display screen (type and size)		VGA	VGA • 6″ × 6″
	Software installation	Downloaded to Keypad/analyzer	Pre-installed	Pre-installed
	Battery type/number requirements	Custom/3	4 “D” (LR20) batteries	4 “D” (LR20) batteries
	Battery life at full charge	4 h continuous/8 h typical	10–20 h	24 h
6	Optional AC power and requirements	90–220 VAC/50–60 Hz (optional)	90–220 VAC/50–60 Hz	90–220 VAC/50–60 Hz
30	Enter OL2R movement?	Yes	No	Yes
14	Interface system with standard computers?	No	No	Yes, RS232 serial
	Store alignment jobs?		Yes, 100	Yes, 128
18	Turned OFF by mistake?		Resume job	Resume job
	Operational Info			
25	Shafts must be rotated together?	No	No	No
26	How are rotational positions measured?	Internal angle sensors	Internal inclinometer (in one unit)	Internal inclinometer (in one unit)
25	Max. allowable rotational backlash	4°	No effect	No effect
31	Can be used on vertical shafts?	Yes	Yes	Yes
31	Shaft angular positon measured for vertical		Yes	Yes
32	English or metric units?		Yes	Yes
	Additional Features and Accessories			
	Shaft clamping brackets included?	Yes	Yes	Yes
27	Minimum shaft diameter	0.0625″	0.625″ (15 mm)	0.625″ (15 mm)
27	Maximum shaft diameter	26″ (with additional chain)	No limit (with extra chain)	No limit (with extra chain)
28	Minimum sensor height from shaft surface	4.375″	1.96″ (50 mm)	1.96″ (50 mm)
28	Maximum sensor height from shaft surface	10.375″	23.6″ (600 mm)	23.6″ (600 mm)
30	Adapted for measuring OL2R?	Yes (special brackets required)	No	No
	Calibration			
35	Recommended calibration interval	12 months	None recommended	None recommended
	Calibration cost		Yes	Yes
35	Send to factory?			
35	Traceable to standards?			
36	User can buy calibration equipment?	Yes	No	No
36	Cost of calibration equipment	7000	Use shim stock to check	Use shim stock to check
34	Warranty period		12 months	12 months
37	Can user upgrade to new model?		Maybe	Maybe
37	Upgrade cost			

FIGURE 15.24 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	FixturLaserAB Shaft 200 www.fixturlaser.com	Hamar S-640 www.hamarlaser.com	Hamar S-650 www.hamarlaser.com
5	System price (U.S. dollars)			11,000 to 14,000
	Basic System Information			
4	Total unit weight (lbs)	24 lbs (11 kg)		25 lbs (11.5 kg)
3	Alignment operating principle	Reverse indicator	Single laser/beam splitter at detector	Single laser/beam splitter at detector
21	Intrinsically safe/explosion proof models?	No		
16	Shock protection	Not answered	Yes	Yes
20	Operating temperature range	0°C–50°C		
22	Environmental protection	IP65	Yes	Yes
	Laser Emitter (*Source*)			
9	Laser type	Visible red diode	Visible red diode	Visible red diode
9	Number of lasers used	2	1	2
9	Wavelength	650–670 nm	670 nm	670 nm
9	Beam diameter at exit	2 mm		
9	Max. beam power output	1 mW	< 1 mW	< 0.9 mW
9	Laser safety class	Class 2	Class 2	Class 2
	Max. recommended distance		20 ft (6.9 m)	30 ft (9.1 m)
6	Laser power supplied by…	Cable from display unit	Rechargeable battery pack (8 h)	Rechargeable battery pack (50 h)
	Cable length	10 ft (3 m)	Not required	Not required
	Dimensions $W \times H \times D$ (inches)		3.25″ × 3.25″ × 2″ (8.25 cm × 8.25 cm × 5 cm)	3.88″ × 3.76″ × 1.68″ (9.86 cm × 9.55 cm × 4.27 cm)
	Laser–Detector (*Receiver*)			
12	Type	Single axis photodiode (two)	4 axis photodiode	2 axis photodiode
12	Detector measurement planes	1	4	5
12	Detector measurement area	20 mm × 20 mm	±0.150″ (3.81 mm) center	10 mm × 33 mm
12	Number of detectors used	2	2	
12	Linearity	±1% × ±0.003 mm		0.01
12	Accuracy/resolution	1 μm	0.00002″ (0.0005 mm)	1 μm
12	Optical filtering?	Yes	Not answered	Yes (optional filtering available)
15	Beam position data transfer via…	Cable	Cable to R-358 computer interface	Infrared transmission
	Detector power supplied by…	Cable from display unit	Cable	0.4 VDC rechargeable battery
				7
	Battery life at full charge	24 h		14 h
	Dimensions $W \times H \times D$ (inches)		3.25″ × 3.25″ × 2″ (8.25 cm × 8.25 cm × 5 cm)	3.88″ × 3.76″ × 2.64″ (9.86 cm × 9.55 cm × 6.71 cm)
	Interchangeable with other display units?	No		Yes

FIGURE 15.25 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	FixturLaserAB Shaft 200 www.fixturlaser.com	Hamar S-640 www.hamarlaser.com	Hamar S-650 www.hamarlaser.com
	Operator Interface			
14	Alignment computer and software included?	Yes	Software, no computer	Yes
14	Dimensions $W \times H \times D$ (inches)		User supplied	User supplied
	Weight (with battery)		User supplied	User supplied
	Display screen (type and size)	VGA • 6″× 6″	User supplied	User supplied
	Software installation	Pre-installed	CD ROM	CD ROM
	Battery type/number requirements	4 "D" (LR20) batteries	User supplied	User supplied
	Battery life at full charge	24 h	User supplied	User supplied
6	Optional AC power and requirements	90–220 VAC/50–60 Hz	Depends on computer used	User supplied
30	Enter OL2R movement?	Yes	Yes	Yes
14	Interface system with standard computers?	Yes, RS232 serial	Windows based computers	Windows based computers
	Store alignment jobs?	Yes, 100	Yes	Yes
18	Turned OFF by mistake?	Resume job		Resume where left off
	Operational Info			
25	Shafts must be rotated together?	No		
26	How are rotational positions measured?	Internal inclinometer (in one unit)	Internal inclinometer (in detector)	Internal accelerometer (in detector)
25	Max. allowable rotational backlash	No effect		
31	Can be used on vertical shafts?	Yes	Yes	Yes
31	Shaft angular positon measured for vertical			
32	English or metric units?	Yes		Yes
	Additional Features and Accessories			
	Shaft clamping brackets included?	Yes	Yes	Yes
27	Minimum shaft diameter	0.625″ (15 mm)	0.375″	0.375″
27	Maximum shaft diameter	No limit (with extra chain)	5″ Std. (12″ optional)	5″ Std. (18″ optional)
28	Minimum sensor height from shaft surface	1.96″ (50 mm)		
28	Maximum sensor height from shaft surface	23.6″ (600 mm)		
30	Adapted for measuring OL2R?	Yes	Yes	Yes
	Calibration			
35	Recommended calibration interval	None recommended		12 months
	Calibration cost			
35	Send to factory?	Yes		
35	Traceable to standards?			
36	User can buy calibration equipment?	No		
36	Cost of calibration equipment	Use shim stock to check		
34	Warranty period	12 months		12 months
37	Can user upgrade to new model?	Maybe		Yes
37	Upgrade cost			Varies

FIGURE 15.26 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	Prüftechnik POCKETALIGN www.pruftechnik.com	Prüftechnik OPTALIGN PLUS www.pruftechnik.com	Prüftechnik smart ALIGN www.pruftechnik.com
5	System price (U.S. dollars)	4,975 to 6,925	6,600 to 16,875	14,800
	Basic System Information			
4	Total unit weight (lbs)	5.8 lbs	16 lbs (7.3 kg)	19.2 lbs (8.7 kg)
3	Alignment operating principle	Reflected beam	Reflected beam	Reflected beam
21	Intrinsically safe/explosion proof models?		Class 1. Div. 1, groups D, C, B, A	Class 1, Div, 1, groups D, C, B, A
16	Shock protection	Yes	Yes	Yes
20	Operating temperature range	32°F–131°F	−4°F–140°F	32°F–140°F
22	Environmental protection	IP67	IP65	IP65
	Laser Emitter (*Source*)			
9	Laser type	Visible and infrared	Visible and infrared	Visible and infrared
9	Number of lasers used	2 (visible & IR for measurement)	2 (visible and IR for measurement)	2 (visible and IR for measurement)
9	Wavelength	675 nm (visible)	675 nm (visible)	675 nm (visible)
9	Beam diameter at exit			
9	Max. beam power output	<1 mW	<1 mW	<1 mW
9	Laser safety class	Class 2; FDA 21CFR 1000 and 1040	Class 2; FDA 21 CFR 1000 and 1040	Class 2; FDA 21 CFR 1000 and 1040
	Max. recommended distance			
6	Laser power supplied by …	Cable from display unit	Cable from display unit	Cable from display unit
	Cable length			
	Dimensions $W \times H \times D$ (inches)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)
	Laser–Detector (*Receiver*)			
12	Type	2 axis photodiode	2 axis photodiode	2 axis photodiode
12	Detector measurement planes	2	2	2
12	Detector measurement area			
12	Number of detectors used			
12	Linearity			
12	Accuracy/resolution	2%/1 μm	2%/1 μm	2%/1 μm
12	Optical filtering?	Yes	Yes	Yes
15	Beam position data transfer via …	Cable from PDA	Cable from display unit	Cable from display unit
	Detector power supplied by …	Cable from PDA	Cable from display unit	Cable from display unit
	Battery life at full charge	25 h	25 h	25 h
	Dimensions $W \times H \times D$ (inches)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)
	Interchangeable with other display units?			

FIGURE 15.27 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	Prüftechnik POCKETALIGN www.pruftechnik.com	Prüftechnik OPTALIGN PLUS www.pruftechnik.com	Prüftechnik smart ALIGN www.pruftechnik.com
	Operator Interface			
14	Alignment computer and software included?	Optional	Yes	Yes
14	Dimensions $W \times H \times D$ (inches)	Depends on PDA selected	5.75″ × 11.5″ × 2.75″	4″ × 8.625″ × 2.2″
	Weight (with battery)	Depends on PDA selected	2.4 lbs (1.2 kg)	2.4 lbs (1.2 kg)
	Display screen (type and size)	Depends on PDA selected	LCD • 3.75″ × 2.875″	LCD • 2.125″ × 1.125″
	Software installation	Downloaded from PC	Pre-installed • downloadable upgrades	Pre-installed • downloadable upgrades
	Battery type/number requirements	Depends on PDA selected	1,9VDC IEC 6LR61 battery	NiMH 7.2 VDC rechargeable
	Battery life at full charge	Depends on PDA selected	25 h	10 h
6	Optional AC power and requirements	Depends on PDA selected	90–220 VAC/50–60 Hz	90–220 VAC/50–60 Hz
30	Enter OL2R movement?		Yes	Yes
14	Interface system with standard computers?		Yes, RS232 serial	Yes, RS232 serial
	Store alignment jobs?		Yes (99)	
18	Turned OFF by mistake?			
	Operational Info			
25	Shafts must be rotated together?		No	No
26	How are rotational positions measured?		Internal inclinometer	Internal inclinometer
25	Max. allowable rotational backlash			
31	Can be used on vertical shafts?		Yes	Yes
31	Shaft angular positon measured for vertical			
32	English or metric units?		Yes	Yes
	Additional Features and Accessories			
	Shaft clamping brackets included?		Yes	Yes
27	Minimum shaft diameter			
27	Maximum shaft diameter			
28	Minimum sensor height from shaft surface			
28	Maximum sensor height from shaft surface			
30	Adapted for measuring OL2R?			
	Calibration			
35	Recommended calibration interval			
	Calibration cost			
35	Send to factory?			
35	Traceable to standards?			
36	User can buy calibration equipment?			
36	Cost of calibration equipment			
34	Warranty period			
37	Can user upgrade to new model?			
37	Upgrade cost			

FIGURE 15.28 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	Prüftechnik ROTALIGN PRO www.pruftechnik.com	Prüftechnik ROTALIGN Ultra www.pruftechnik.com	SPM Instruments Inc. MAC10 www.spminstrument.com
5	System price (U.S. dollars)	18,900 to $23,780	18,900 to $23,780	4,000
	Basic System Information			
4	Total unit weight (lbs.)	24.6 lbs (11.2 kg)	24.6 lbs (11.2 kg)	
3	Alignment operating principle	Reflected beam	Reflected beam	Reverse Indicator
21	Intrinsically safe/explosion proof models?	Class 1, Div. 1, groups D, C, B, A	Class 1, Div. 1, groups D, C, B, A	No
16	Shock protection	Yes	Yes	No
20	Operating temperature range	32°F–113°F	32°F–113°F	32°F–122°F (0°C–50°C)
22	Environmental protection	IP67	IP67	No
	Laser Emitter (*Source*)			
9	Laser type	Visible and infrared	Visible and infrared	N/a
9	Number of lasers used	2 (visible and IR for measurement)	2 (visible and IR for measurement)	N/a
9	Wavelength	675 nm (visible)	675 nm (visible)	N/a
9	Beam diameter at exit			N/a
9	Max. beam power output	<1mW	<1mW	N/a
9	Laser safety class	Class 2; FDA 21 CFR 1000 and 1040	Class 2; FDA 21 CFR 1000 and 1040	N/a
	Max. recommended distance	33 ft (10 m)	33 ft (10 m)	N/a
6	Laser power supplied by …	9VDC • IEC 6LR61 battery	9VDC • IEC 6LR61 battery	N/a
	Cable length			
	Dimensions $W \times H \times D$ (inches)	4.25″ × 2.625″ × 1.875″ (10.5 cm × 6.7 cm × 4.7 cm)	4.25″ × 2.625″ × 1.875″ (10.5 cm × 6.7 cm × 4.7 cm)	
	Laser–Detector (*Receiver*)			
12	Type	2 axis photodiode	2 axis photodiode	Optical encoder
12	Detector measurement planes	2	2	1
12	Detector measurement area	18 mm × 18 mm	18 mm × 18 mm	0.590″ (15 mm)
12	Number of detectors used	2	2	1
12	Linearity			N/a
12	Accuracy/resolution	2%/1 μm	2%/1 μm	0.004″ (0.01 mm)
12	Optical filtering?	Yes	Yes	N/a
15	Beam position data transfer via …	Cable (optional IR transmission)	Cable (optional IR transmission)	N/a
	Detector power supplied by …	Cable from display unit	Cable from display unit	Battery
	Battery life at full charge	25 h	25 h	
	Dimensions $W \times H \times D$ (inches)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)	4.25″ × 2.75″ × 2″ (11 cm × 7 cm × 5 cm)	
	Interchangeable with other display units?			No

FIGURE 15.29 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	Prüftechnik ROTALIGN PRO www.pruftechnik.com	Prüftechnik ROTALIGN Ultra www.pruftechnik.com	SPM Instruments Inc. MAC10 www.spminstrument.com
	Operator Interface			
14	Alignment computer and software included?	Yes	Yes	Yes
14	Dimensions $W \times H \times D$ (inches)	8.25″ × 9.5″ × 2.75″	8.25″ × 9.5″ × 2.75″	145 mm × 90 mm × 32 mm
	Weight (with battery)	3.8 lbs (1.75 kg)	3.8 lbs (1.75 kg)	0.6 lbs
	Display screen (type and size)	LCD • 2.125″ × 1.125″	LCD • 2.125″ × 1.125″	
	Software installation	Pre-installed • downloadable upgrades	Pre-installed • downloadable upgrades	Pre-installed
	Battery type/number requirements	6, “C” IEC LR14 batteries	6, “C” IEC LR14 batteries	1–9 VDC alkaline
	Battery life at full charge	9–50 h	9–50 h	6 h
6	Optional AC power and requirements	90–220 VAC/50–60 Hz	90–20 VAC/50–60 Hz	No
30	Enter off-line to running movement?	Yes	Yes	Yes
14	Interface system with standard computers?	Yes, RS232 serial	Yes, RS232 serial	No
	Store alignment jobs?			No
18	Turned OFF by mistake?			No
	Operational Info			
25	Shafts must be rotated together?	No	No	Yes
26	How are rotational positions measured?	internal inclinometer	Internal inclinometer	Manual
25	Max. allowable rotational backlash		None	
31	Can be used on vertical shafts?	Yes	Yes	No
31	Shaft angular positon measured for vertical			N/a
32	English or metric units?	Yes	Yes	Yes
	Additional Features and Accessories			
	Shaft clamping brackets included?	Yes	Yes	Yes
27	Minimum shaft diameter			
27	Maximum shaft diameter			
28	Minimum sensor height from shaft surface			
28	Maximum sensor height from shaft surface			
30	Adapted for measuring OL2R?			No
	Calibration			
35	Recommended calibration interval			1 year
	Calibration cost			500
35	Send to factory?			Yes
35	Traceable to standards?			Country of origin (Sweden)
36	User can buy calibration equipment?			No
36	Cost of calibration equipment			N/a
34	Warranty period			1 year
37	Can user upgrade to new model?			No
37	Upgrade cost			N/a

FIGURE 15.30 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	SPM Instruments Inc. LineLazer www.spminstrument.com	Vibralign V180 www.vibralign.com
5	System price (U.S. dollars)	5,000 option	4,250
	Basic System Information		
4	Total unit weight (lbs)	20 lbs (9.1 kg)	8 lbs (4 kg)
3	Alignment operating principle	Reverse indicator	Reverse indicator
21	Intrinsically safe explosion proof models?	No	No
16	Shock protection	Yes	Yes
20	Operating temperature range	32°F–122°F (0°C–50°C)	32°F–185°F
22	Environmental protection	Sealed	
	Laser Emitter (*Source*)		
9	Laser type	Visible red diode	Visible red diode
9	Number of lasers used	2	2
9	Wavelength	635–650 nm	675 nm
9	Beam diameter at exit	N/a, its a line laser	
9	Max. beam power output	<1 mW	
9	Laser safety class	Class 2	
	Max. recommended distance	10 ft. (300 cm)	1 m
6	Laser power supplied by …	Rechargeable	Cable from display
	Cable length	Single 3 meter cable	
	Dimensions $W \times H \times D$ (inches)	4″×3.7″×1.7″ (112 mm×93 mm×44 mm)	
	Laser–Detector (*Receiver*)		
12	Type	Single axis PSD	Single axis PSD
12	Detector measurement planes	1	1
12	Detector measurement area	37 mm	10 mm × 10 mm
12	Number of detectors used	2	2
12	Linearity	<2% Deviation	
12	Accuracy/resolution	1 Micrometer	0.0001″
12	Optical filtering?	Yes	Yes
15	Beam position data transfer via …	Wireless (modulated laser), 200 kHz	Cable from display
	Detector power supplied by …	Internal battery	Cable from display
	Battery life at full charge	10 h	
	Dimensions $W \times H \times D$ (inches)	4″×3.7″×1.7″ (112 mm × 93 mm × 44 mm)	
	Interchangeable with other display units?	No	

FIGURE 15.31 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site	SPM Instruments Inc. LineLazer www.spminstrument.com	Vibralign V180 www.vibralign.com
	Operator Interface		
14	Alignment computer and software included?	Yes	Yes
14	Dimensions $W \times H \times D$ (inches)	11.2″ × 4″ × 2.5″ (285 mm × 102 mm × 63 mm)	
	Weight (with battery)	1.3 lbs (600 g)	
	Display screen (type and size)	Touch screen, 3.2″ × 2.4″	77 × 33 mm LCD
	Software installation	Pre-installed	Pre-installed
	Battery type/number requirements	Rechargeable, lithium ion	3 “C” batteries
	Battery life at full charge	10 h	
6	Optional AC power and requirements	90–220 VAC/50–60 Hz.	
30	Enter OL2R movement?	Yes	No
14	Interface system with standard computers?	Yes (in Leonova)	No
	Store alignment jobs?	No	No
18	Turned OFF by mistake?	No	No
	Operational Info		
25	Shafts must be rotated together?	No	
26	How are rotational positions measured?	Internal inclinometers	
25	Max. allowable rotational backlash		
31	Can be used on vertical shafts?	Yes	
31	Shaft angular positon measured for vertical	Yes	
32	English or metric units?	Yes	
	Additional Features and Accessories		
	Shaft clamping brackets included?	Yes	
27	Minimum shaft diameter	0.375″ (1 cm)	
27	Maximum shaft diameter	39.4″ (1 m)	
28	Minimum sensor height from shaft surface	3.9″ (100 mm)	
28	Maximum sensor height from shaft surface	39″ (1000 mm)	
30	Adapted for measuring OL2R?	Yes	
	Calibration		
35	Recommended calibration interval	2 years	
	Calibration cost	500	
35	Send to factory?	Yes	
35	Traceable to standards?	Country of origin (Sweden)	
36	User can buy calibration equipment?	No	
36	Cost of calibration equipment	N/a	
34	Warranty period	1 year	
37	Can user upgrade to new model?	Yes	
37	Upgrade cost	N/a	

FIGURE 15.32 Laser system hardware comparison chart.

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12. What effect does dirt, grease, oil, fingerprints, moisture, etc. have on the laser collimating lens or protective optics? If detrimental to the operation of the system, how should one clean the laser optics?

    _____

    _____

13. What type of detector is used in your system and how does it work (photodiode, CCD, single or dual axis? detector target area, resolution, linearity, accuracy/repeatability, environmental limits, optical filtering, etc.)?
    Detector type _____
    _____ single axis photodiode
    _____ dual axis photodiode
    _____ (millimeters, inches) detector target area
    _____ (microns, inches) beam resolution
    Linearity _____
    Accuracy _____
    Repeatability _____
    Environmental limits (suggested minimum and maximum air temperature, etc.)

    _____

    _____

    _____

    Optical filtering _____
    Others _____

14. What effect does dirt, grease, oil, fingerprints, moisture, etc. have on the detector or protective optics? If detrimental to the operation of the system, how should one clean the detector?

    _____

    _____

    Suggested cleaning procedure

    _____

    _____

15. Can the output from the detector be connected directly to a personal computer (specifically computer using Intel or Motorola processors)? If so, is interface software supplied with the system? Is there a charge for the software?
    _____ Yes, it can be connected to a personal computer
    _____ No, it cannot be connected to a personal computer
    _____ Yes, an interface device is supplied
    _____ No, an interface device is not supplied
    _____ Yes, interface software is supplied with the system
    _____ Yes, interface software is available for _____ (U.S. dollars)
    _____ No, interface software is not available

16. How are the data from the detectors transmitted to the computer–operator interface? Cables or wireless transmission? If cables are used, how long are they? Do the cables have quick disconnect-type fittings? If cables are used, what happens if an operator cannot stop a shaft from rotating once they get it started?
    _____ cable interface
    _____ wireless interface
    _____ cable or wireless interface
    _____ (inches, millimeters) standard cable length

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_____ (inches, millimeters) optional cable length
_____ quick connect fittings
_____ rigid fittings

17. What type of shock protection is incorporated in your system to prevent damage to the laser–detector? What happens if an operator drops the system from 6 ft?

_____

_____

18. Can the user store a shaft alignment job in the computer–operator interface? Can they store more than one alignment job? If, so how many? Can the data stored in the alignment system be transferred to a personal computer? If so, how is this done?
_____ Yes, shaft alignment jobs can be stored
_____ No, shaft alignment jobs cannot be stored
_____ Number of alignment jobs that can be stored internally
_____ Yes, shaft alignment jobs can be uploaded to a personal computer
_____ No, shaft alignment jobs cannot be uploaded to a personal computer

19. If an operator mistakenly pressed the "OFF" button in the middle of an alignment job, what happens?
_____ operator must start over
_____ system returns to last point in process
_____ others _____

20. What type of thermal protection is incorporated in your system to prevent measurement distortion?

_____

_____

21. What environmental limitations are on the system? For example, can it operate at temperatures above 120°F or below 0°F? In outdoor applications, what effect does direct sunlight have on the sensors? If the unit is operated near strong electromagnetic fields (e.g., motor windings, magnetic bases) is there any effect on the electronics that would affect the accuracy of the instrument?
Environmental limits

_____

_____

Effect of direct sunlight

_____

Effect of strong magnetic fields

_____

_____

22. Do you offer an intrinsically safe or explosion proof models or both?
_____ Yes
_____ No

23. What type of environmental protection for the laser–detector is incorporated? Does it comply with U.S. or International environmental protection standards?

_____

_____

_____ Yes
_____ No

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24. If your system includes an operator interface–keypad entry device, describe the basic function of the device.

_____

_____

25. If a customer has more than one of your measurement systems in their possession, can the operator interface modules and laser–detectors be interchanged or are the detectors matched to a specific operator interface? If they can be interchanged, explain why.
_____ Yes
_____ No

_____

_____

26. Do the shafts have to be rotated together while capturing readings? If so, what amount of rotational "backlash" between the two shafts can be tolerated before measurement accuracy is sacrificed? If a rotational sensor is incorporated in your system, what is the minimum amount of angular rotation needed to determine the position of the shafts? What is the minimum recommended rotational angle for good repeatability and accuracy?
_____ Yes, the shafts must be rotated together
_____ No, the shafts do not have to be rotated together
_____ (angular degrees) allowable backlash
_____ Yes, a rotational sensor is incorporated into the system
_____ (angular degrees) minimum suggested rotation
_____ No, the shafts do not have to be rotated together

27. How do you measure the rotational position of the sensors when capturing readings?

_____

_____

28. What are the minimum and maximum shaft diameters the brackets can be clamped to?
_____ (inches/millimeters) minimum shaft diameter
_____ (inches/millimeters) maximum shaft diameter

29. What are the minimum and maximum height range that the laser or detector can be placed from the point of contact on the shaft to the position of the laser or detector?
_____ (inches/millimeters) fixed height
_____ (inches/millimeters) minimum height
_____ (inches/millimeters) maximum height

30. Please complete the shaft alignment software questionnaire to describe the operation of the software that interfaces with your system.
31. Can the sensors be reconfigured to capture OL2R machinery movement? If so, how is this accomplished?
_____ Yes
_____ No

_____

_____

32. Can the system be used on vertically oriented rotating machinery? If so, how is the amount of rotation measured?
_____ Yes
_____ No

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Angular position of system is measured as follows:

_____

_____

33. Can the system be configured to enter English or metric units?
_____ Yes
_____ No

34. If your system is patented, please send a copy of the patent (in English). If you have patents in other countries, please indicate where the patents are held and their corresponding patent numbers or identification.

_____

_____

_____

35. What is your warranty period?
_____ (days, months, years)

36. What are the recommended calibration intervals, does the unit have to be sent back to the factory for calibration, and what is the charge for recalibration? Is the calibration you offer traceable back to U.S. or International Standards?
_____ (days, months, years) recommended calibration interval
Send back to factory?
_____ Yes
_____ No
Traceable to standards?
_____ Yes
_____ No

37. Do you offer equipment for the user to check and adjust calibration? If so, what is the cost of the test equipment?
_____ Yes
_____ No

38. If you have introduced new shaft alignment systems models, do the people who own your original models have the option to "trade in" their older models to upgrade to a newer model? If so, what is the cost of the upgrade?
_____ Yes. If yes, upgrade cost _____ (U.S. dollars)
_____ No

39. What optional equipment or accessories are available for your system?

_____

_____

_____

_____

15.4 LASER SYSTEM MANUFACTURERS SOFTWARE SPECIFICATIONS

A questionnaire was sent out to all of the above manufacturers concerning the pricing, specifications, and features of their systems. The questions are listed below. A summary of their answers is shown in Figure 15.33 through Figure 15.38.

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Damalini D450 www.damalini.com	Damalini D505 www.damalini.com	Damalini D525 www.damalini.com	FixturLaser Shaft 30 www.fixturlaser.com	FixturLaserAB Shaft 100 www.fixturlaser.com
3	Computer for software (standard or custom)				Standard	Standard
4	If custom … suggested computer Mfg.					
4	Custom computer cost					
5	Operating system(s) on custom computer (see list)				Windows (all)	Windows (all)
6	Interface with PC? (yes/no)	Yes	Yes	Yes		
7	Operating system(s) on PC interface computer					
8	Communication interface(s) (see list)					
9	Price of software	Free	Free	Free	Free	Free
10	Release date of program	New 01/05/05	New 01/05/05	New 01/05/05		
11	Minimum RAM					
12	Minimum screen pixel density					
13	Special graphics required?	No	No	No		
14	Mouse/trackball required?	Yes	Yes	Yes		
15	Number of allowable machines in drive train	10	10	10		
16	Number of allowable bolting planes				Infinite	Infinite
17	English or metric input?	Both	Both	Both	Yes	Yes
18	Software help incorporated in software?	Yes	Yes	Yes	Yes	Yes
19	Alignment help incorporated in software?				Yes	Yes
20	Number of detailed machine drawings in library	0	0	0		
21	Generic machine images? (yes/no)	Yes	Yes	Yes	Yes	Yes
22	Edit machine dimensions? (yes/no)	No	No	No	Yes	Yes
23	Selectable side directions? (e.g. north, south, east, west, left, right) (yes/no)	No	No	No	No	No
24	User specified side direction? (yes/no)					
25	Edit side direction? (yes/no)					
26	Selectable flexible coupling design/type? (yes/no)	No	No	No	No	No
27	Input soft foot gap info? (yes/no)	No	No	No	Yes	Yes
28	Calculate and display shim shapes for soft foot correction? (yes/no)	Yes	Yes	Yes	Yes	Yes
29	Input runout info? (yes/no)	No	No	No	Yes	Yes
30	Input symmetric reverse indicator method? (yes/no)				Yes	Yes

FIGURE 15.32 Laser system hardware comparison chart. (*continued*)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Damalini D450 www.damalini.com	Damalini D505 www.damalini.com	Damalini D525 www.damalini.com	FixturLaser Shaft 30 www.fixturlaser.com	FixturLaserAB Shaft 100 www.fixturlaser.com
31	Input asymmetric reverse indicator method? (yes/no)				Yes	Yes
32	Input face–rim dial method? (yes/no)				No	No
33	Front or back side face readings allowable? (yes/no)				No	No
34	Input from shaft to coupling spool dial indicator method?				No	No
35	Input from face–face dial indicator method?				No	No
36	Input from double radial indicator method?				No	No
37	Selectable methods at each coupling for multiple element drive trains?	No	No	No	No	No
38	Input symmetric bracket sag?	No	No	No	Yes	Yes
39	Input asymmetric bracket sag?	No	No	No	Yes	Yes
40	Input shaft centerline readings only through 180° of rotation?	No, 40°	No, 40°	No, 40°	No	No
41	Input shaft centerline readings with less than 180° of rotation?	Yes	Yes	Yes	Yes	Yes
42	Manual input from stationary–movable laser systems?	Yes	Yes	Yes		
43	Manual input from other stationary–movable systems?					
44	Input vertical allowable movement envelope?					

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45	Input lateral movement envelope?					
46	Input off-line to running (OL2R) movement?	No	No	No	Yes	Yes
47	Input both vertical and lateral OL2R data?	No	No	No	Yes	Yes
48	Generic OL2R movement input?	No	No	No		
49	Optical alignment OL2R data input?					
50	Input from inside micrometer/tooling ball OL2R input (triangular set up)?	No	No	No		
51	Input from inside micrometer/tooling ball OL2R input (pyramid set up)?	No	No	No		
52	Calculated thermal expansion input?	Yes	Yes	Yes	Yes	Yes
53	Alignment Bar (Dyn-Align/Dodd) input?					
54	Acculign (Essinger) Bar input?					
55	Laser–Detector OL2R input?					
56	PIBZLT mount system input?					
57	Ball-Rod-Tubing Connector system input?					
58	Vernier-Strobe system input?					
59	Indikon system input?					
60	Calculates "shoot-for" dial indicator readings?	Yes	Yes	Yes		

FIGURE 15.33 Laser system hardware comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Damalini D450 www.damalini.com	Damalini D505 www.damalini.com	Damalini D525 www.damalini.com	FixturLaser Shaft 30 www.fixturlaser.com	FixturLaserAB Shaft 100 www.fixturlaser.com
61	Calculates "target values" for laser systems?	Yes	Yes	Yes	Yes	Yes
62	Automatically shifts shaft positions in side and top views for OL2R data?	No	No	No		
63	Solves correction moves for one machine case?	Yes	Yes	Yes		
64	Solves correction moves for several machine elements?	Yes	Yes	Yes	Yes	Yes
65	Variable position overlay line (aka final desired alignment line)?	Yes	Yes	Yes		
66	Total number of possible alignment correction moves (see list)	Infinite	Infinite	Infinite	Infinite	Infinite
67	Compares existing alignment to alignment tolerance guide?	Yes	Yes	Yes	Yes	Yes
68	Show progress of each move on the alignment tolerance guide?	Yes	Yes	Yes	Yes	Yes
69	Shows graphical side view model of current shaft positions?	Yes	Yes	Yes	Yes	Yes
70	Shows graphical top view model of current shaft positions?	Yes	Yes	Yes	Yes	Yes
71	Show or hide grid background?					
72	Graph or model in color?	No	No	No	No	No
73	Select grid background color?	No	No	No		
74	Select shaft centerline position color?	No	No	No		
75	Shows machinery side view configuration?	Yes	Yes	Yes	Yes	Yes
76	Shows machinery top view configuration?	Yes	Yes	Yes	Yes	Yes
77	Shows soft foot gap information?	Yes	Yes	Yes	Yes	Yes
78	Shows alignment method used?	Yes	Yes	Yes	Yes	Yes
79	Shows alignment data entered?	Yes	Yes	Yes	Yes	Yes

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80	Shows bracket sag amount?	No	No	No		
81	Shows "shoot-for" readings?	Yes	Yes	Yes	Yes	Yes
82	Show desired off-line side view model?					
83	Show desired off-line top view model?					
84	Edit machinery names?	Yes	Yes	Yes	Yes	Yes
85	Edit machinery dimensions?	No	No	No	Yes	Yes
86	Edit dial indicator readings?	No	No	No		
87	Edit bracket sag amount?	No	No	No		
88	Edit stationary—movable data?	No	No	No		
89	Edit OL2R data?	No	No	No		
90	Edit scale factors in side view alignment model?					
91	Edit scale factors in top view alignment model?					
92	Edit number of machines in drive train?	Yes	Yes	Yes	Yes	Yes
93	Aligns right angle drives?	Yes	Yes	Yes	Yes	Yes
94	Aligns C-flanged and vertical machinery?	Yes	Yes	Yes	Yes	Yes
95	Cross calculate alignment methods?	No	No	No		
96	Centering a shaft in a bearing?	No	No	No	Yes	Yes
97	Start a new file?	Yes	Yes	Yes	Yes	Yes
98	Open an old file?	Yes	Yes	Yes	Yes	Yes
99	Close existing file?	Yes	Yes	Yes	Yes	Yes
100	Save existing file?	Yes	Yes	Yes	Yes	Yes
101	Save as renamed file?	No	No	No		
102	Revert to last saved file?	Yes	Yes	Yes		
103	Store each correction move made?	No	No	No	No	No
104	Print data?	Yes	Yes	Yes	Yes	Yes
105	Selectable items for printout? (see list)	Yes	Yes	Yes	All	All
106	Hard copy (printed) software manual included?	No	No	No	Yes	Yes
107	Can the software accept direct input from the measurement sensors?	No	No	No	Yes	Yes
108	What type of sensors can it accept? (see list)				Laser–detector	Laser–detector
109	Patented?	No	No	No		
110	What is your warranty period?	12 months	12 months	12 months	12 months	12 months

FIGURE 15.34 Laser system software comparison chart.

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	FixturLaserAB Shaft 200 www.fixturlaser.com	Hamar S-650 www.hamarlaser.com	Prüftechnik POCKETALIGN www.pruftechnik.com	Prüftechnik OPTALIGN PLUS Series www.pruftechnik.com	Prüftechnik smartALIGN www.pruftechnik.com
3	Computer for software (standard or custom)	Standard	Standard	PDA	Custom computer	Custom computer
4	If custom … suggested computer Mfg.			Prüftechnik alignment	Prüftechnik alignment	Prüftechnik alignment
4	Custom computer cost			-	Within system package price	Within system package price
5	Operating system(s) on custom computer (see list)	Windows (all)	Windows 98, Me, 2000, NT, XP	MS Mobile 2003	-	-
6	Interface with PC? (yes/no)			Yes	Yes	Yes
7	Operating system(s) on PC interface computer			MS Windows 98/Me/NT/ 2000/XP	MS Windows 98/Me/NT/ 2000/XP	MS Windows 98/ Me/NT/2000/XP
8	Communication interface(s) (see list)		Serial	USB/RS232/Bluetooth/WiFi	RS232	RS232
9	Price of software	Free	1495	Within system package price	Within system package price	Within system package price
10	Release date of program		5/25/04	Latest release on Web site	Latest release on Web site	Latest release on Web site
11	Minimum RAM		640K	-	-	-
12	Minimum screen pixel density		640 × 480	320×240	Fixed segment display	128×64
13	Special graphics required?		No	No	No	No
14	Mouse/trackball required?		No	Touch screen	No	No
15	Number of allowable machines in drive train	6		2	6	3
16	Number of allowable bolting planes	Infinite	4	2	3	3
17	English or metric input?	Yes	Yes	English and metric	English and metric	English and metric
18	Software help incorporated in software?	Yes	Yes	Yes	Not required	Not required
19	Alignment help incorporated in software?	Yes	Yes	Yes	Not required	Not required
20	Number of detailed machine drawings in library			1	1	1
21	Generic machine images? (yes/no)	Yes	Yes	Yes	Yes	Yes
22	Edit machine dimensions? (yes/no)	Yes	Yes	Yes	Yes	Yes
23	Selectable side directions? (e.g., north, south, east, west, left, right) (yes/no)	No	Yes	No	No	No
24	User specified side direction (yes/no)		No	No	No	Yes
25	Edit side direction? (yes/no)		No	No	No	Yes
26	Selectable flexible coupling design/ type? (yes/no)	No	Yes	No	Yes	Yes
27	Input soft foot gap info? (yes/no)	Yes	Yes	No	No	No
28	Calculate and display shim shapes for soft foot correction? (yes/no)	Yes	Yes	Yes	Yes	Yes
29	Input runout info? (yes/no)	Yes	No	No	No	No

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30	Input symmetric reverse indicator method? (yes/no)	Yes	No	No	No	No
31	Input asymmetric reverse indicator method? (yes/no)	Yes	No	No	No	No
32	Input face–rim dial method? (yes/no)	No	Yes	No	No	No
33	Front or back side face readings allowable? (yes/no)	No	Yes	No	No	No
34	Input from shaft to coupling spool dial indicator method?	No	No	No	No	No
35	Input from face–face dial indicator method?	No	No	No	No	No
36	Input from double radial dial indicator method?	No	No	No	No	No
37	Selectable methods at each coupling for multiple element drive trains?	No	No	No	Yes	Yes
38	Input symmetric bracket sag?	Yes	No	No	No	Yes
39	Input asymmetric bracket sag?	Yes	No	No	No	No
40	Input shaft centerline readings only through 180° of rotation?	No	Yes	Less than 180° rotation	Less than 180° rotation	Less than 180° rotation
41	Input shaft centerline readings with less than 180° of rotation?	Yes	Yes	Yes	Yes	Yes
42	Manual input from stationary–movable laser systems?		No	No	No	Yes
43	Manual input from other stationary–movable systems?		No	No	No	Yes
44	Input vertical allowable movement envelope?		No	No	No	No
45	Input lateral movement envelope?		No	No	No	No
46	Input off-line to running (OL2R) movement?	Yes	No	No	Yes	Yes
47	Input both vertical and lateral OL2R data?	Yes	No	No	Yes	Yes
48	Generic OL2R movement input?		No	No	No	Yes
49	Optical alignment OL2R data input?		No	No	No	No
50	Input from inside micrometer/tooling ball OL2R input (triangular set up)?		No	No	No	No

FIGURE 15.35 Laser system software comparison chart. (*continued*)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	FixturLaserAB Shaft 200 www.fixturlaser.com	Hamar S-650 www.hamarlaser.com	Prüftechnik POCKETALIGN www.pruftechnik.com	Prüftechnik OPTALIGN PLUS Series www.pruftechnik.com	Prüftechnik smartALIGN www.pruftechnik.com
52	Calculated thermal expansion input?	Yes	No	No	No	No
53	Alignment Bar (Dyn-Align/Dodd) input?		No	No	No	No
54	Acculign (Essinger) Bar input?		No	No	No	No
55	Laser–Detector OL2R input?		No	No	Yes	Yes
56	PIBZLT mount system input?		No	No	No	No
57	Ball-Rod-Tubing Connector system input?		No	No	No	No
58	Vernier-Strobe system input?		No	No	No	No
59	Indikon system input?		No	No	No	No
60	Calculates "shoot-for" dial indicator readings?		No	No	No	Yes

FIGURE 15.35 (contitued)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	FixturLaserAB Shaft 200 www.fixturlaser.com	Hamar S-650 www.hamarlaser.com	Prüftechnik POCKETALIGN www.pruftechnik.com	Prüftechnik OPTALIGN PLUS Series www.pruftechnik.com	Prüftechnik smartALIGN www.pruftechnik.com
61	Calculates "target values" for laser systems?	Yes	No	No	Yes	Yes
62	Automatically shifts shaft positions in side and top views for OL2R data?		No	No	Yes	Yes
63	Solves correction moves for one machine case?		Yes	Yes	Yes	Yes
64	Solves correction moves for several machine elements?	Yes	Yes	No	Yes	Yes
65	Variable position overlay line (aka final desired alignment line)?		Yes	No	Yes	Yes
66	Total number of possible alignment correction moves (see list)	Infinite	Infinite	Infinite	Infinite	Infinite
67	Compares existing alignment to alignment tolerance guide?	Yes	Yes	No	Yes	Infinite
68	Show progress of each move on the alignment tolerance guide?	Yes	Yes	No	Yes	Yes
69	Shows graphical side view model of current shaft positions?	Yes	Yes	Yes	Yes	Yes
70	Shows graphical top view model of current shaft positions?	Yes	Yes	Yes	Yes	Yes
71	Show or hide grid background?		No	No	No	No
72	Graph or model in color?	No	Yes	Yes	Yes	Yes
73	Select grid background color?		No	No	No	No
74	Select shaft centerline position color?		No	No	No	No
75	Shows machinery side view configuration?	Yes	Yes	Yes	Yes	Yes
76	Shows machinery top view configuration?	Yes	Yes	No	No	No
77	Shows soft foot gap information?	Yes	Yes	Yes	Yes	Yes
78	Show alignment method used?	Yes	No	Yes	Yes	Yes
79	Show alignment data entered?	Yes	Yes	Yes	Yes	Yes
80	Show bracket sag amount?		No	No	No	No
81	Shows "shoot-for" readings?	Yes	No	No	Yes	Yes
82	Show desired off-line side view model?		No	No	Yes	Yes
83	Show desired off-line top view model?		No	No	Yes	Yes
84	Edit machinery names?	Yes	Yes	No	Yes	Yes

FIGURE 15.36 Laser system software comparison chart. (*continued*)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	FixturLaserAB Shaft 200 www.fixturlaser.com	Hamar S-650 www.hamarlaser.com	Prüftechnik POCKETALIGN www.pruftechnik.com	Prüftechnik OPTALIGN PLUS Series www.pruftechnik.com	Prüftechnik smartALIGN www.pruftechnik.com
85	Edit machinery dimensions?	Yes	Yes	Yes	Yes	Yes
86	Edit dial indicator readings?		No	No	No	Yes
87	Edit bracket sag amount?		No	No	No	Yes
88	Edit stationary–movable data?		No	No	Yes	Yes
89	Edit OL2R data?		No	No	Yes	Yes
90	Edit scale factors in side view alignment model?		No	Auto scale	Auto scale	Auto scale
91	Edit scale factors in top view alignment model?		No	Auto scale	Auto scale	Auto scale
92	Edit number of machines in drive train?	Yes	Yes	No	Yes	Yes
93	Aligns right angle drives?	Yes	No	No	No	No
94	Aligns C-flanged and vertical machinery?	Yes	Yes	No	Vertical machinery	Vertical machinery
95	Cross calculate alignment methods?		No	No	No	No
96	Centering a shaft in a bearing?	Yes	No	No	No	No
97	Start a new file?	Yes	Yes	Yes	Yes	Yes
98	Open an old file?	Yes	Yes	Yes	Yes	Yes
99	Close existing file?	Yes	Yes	Yes	Yes	Yes
100	Save existing file?	Yes	Yes	Yes	Yes	Yes
101	Save as renamed file?		Yes	Yes	Yes	Yes
102	Revert to last saved file?		Yes	Yes	Yes	Yes
103	Store each correction move made?	No	Yes	Yes	Yes	Yes
104	Print data?	Yes	Yes	Yes	Yes	Yes
105	Selectable items for printout? (see list)	All	Yes	No	Yes	Yes
106	Hard copy (printed) software manual included?	Yes	Yes	No	Yes	Yes
107	Can the software accept direct input from the measurement sensors?	Yes	Yes	Yes	Yes	Yes
108	What type of sensors can it accept? (see list)	Laser–detector	Laser–detector	Laser–detectors	Laser–detectors	Laser–detectors
109	Patented?			Yes	Yes	Yes
110	What is your warranty period?	12 months	12 months	12 months	12 months	12 months

FIGURE 15.36 (continued)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Prüftechnik ROTALIGN Ultra www.pruftechnik.com	Prüftechnik NOVALIGN www.pruftechnik.com	SPM Instruments Inc. www.spminstrument.com
3	Computer for software (standard or custom)	Custom computer	Custom computer	Custom
4	If custom … suggested computer Mfg.	Prüftechnik Alignment	PrÜftechnik Alignment	SPM instrument Inc.
4	Custom computer cost	Within system package price	Within system package price	$5000 option
5	Operating system(s) on custom computer (see list)	—	—	Windows CE
6	Interface with PC? (yes/no)	Yes	Yes	Yes
7	Operating system(s) on PC interface computer	MS Windows 98/Me/NT/ 2000/XP	MS Windows 98/Me/NT/ 2000/XP	Windows 98, Me, 2000, NT, XP
8	Communication interface(s) (see list)	USB	USB/Ethernet	Serial (RS232), USB
9	Price of software	Within system package price	Within system package price	Included
10	Release date of program	Latest release on Web site	Latest release on Web site	August 2004
11	Minimum RAM	—	—	96 Mb
12	Minimum screen pixel density	480 × 320	640 × 480	240 ×320
13	Special graphics required?	No	no	No
14	Mouse/trackball required?	No	no	Touch screen
15	Number of allowable machines in drive train	14	14	2
16	Number of allowable bolting planes	10	10	2
17	English or metric input?	English and Metric	English and Metric	Yes
18	Software help incorporated in software?	Yes	Yes	Yes
19	Alignment help incorporated in software?	Yes	Yes	Yes
20	Number of detailed machine drawings in library	24	24	N/a
21	Generic machine images (yes/no)	Yes	Yes	Yes
22	Edit machine dimensions? (yes/no)	Yes	Yes	Yes
23	Selectable side directions? (e.g. north, south, east, west, left, right) (yes/no)	No	No	Yes
24	User specified side direction (yes/no)	Yes	Yes	Yes
25	Edit side direction? (yes/no)	Yes	Yes	Yes
26	Selectable flexible coupling design/type? (yes/no)	Yes	Yes	Yes
27	Input soft foot gap info? (yes/no)	Yes	Yes	No
28	Calculate and display shim shapes for soft foot correction? (yes/no)	Yes	Yes	No
29	Input runout info? (yes/no)	No	No	No
30	Input symmetric reverse indicator method? (yes/no)	Yes	Yes	No
31	Input asymmetric reverse indicator method? (yes/no)	Yes	Yes	No
32	Input face–rim dial method? (yes/no)	Yes	Yes	No
33	Front or back side face readings allowable? (yes/no)	Yes	Yes	No

FIGURE 15.37 Laser system software comparison chart. (*continued*)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Prüftechnik ROTALIGN Ultra www.pruftechnik.com	Prüftechnik NOVALIGN www.pruftechnik.com	SPM Instruments Inc. www.spminstrument.com
34	Input from shaft to coupling spool dial indicator method?	No	No	No
35	Input from face–face dial indicator method?	No	No	No
36	Input from double radial dial indicator method?	Yes	Yes	No
37	Selectable methods at each coupling for multiple element drive trains?	Yes	Yes	No
38	Input symmetric bracket sag?	Yes	Yes	No
39	Input asymmetric bracket sag?	Yes	Yes	No
40	Input shaft centerline readings only through 180° of rotation?	Less than 180° rotation	Less than 180° rotation	No
41	Input shaft centerline readings with less than 180° of rotation?	Yes	Yes	Yes
42	Manual input from stationary–movable laser systems?	Yes	Yes	Yes
43	Manual input from other stationary–movable systems?	Yes	Yes	Yes
44	Input vertical allowable movement envelope?	No	No	No
45	Input lateral movement envelope?	No	No	No
46	Input off-line to running (OL2R) movement?	Yes	Yes	Yes
47	Input both vertical and lateral OL2R data?	Yes	Yes	Yes
48	Generic OL2R movement input?	No	No	No
49	Optical alignment OL2R data input?	Yes	Yes	No
50	Input from inside micrometer/tooling ball OL2R input (triangular set up)?	No	No	No
51	Input from inside micrometer/tooling ball OL2R input (pyramid set up)?	No	No	No
52	Calculated thermal expansion input?	Yes	Yes	Yes
53	Alignment bar (Dyn-Align/Dodd) input?	No	No	No
54	Acculign (Essinger) Bar input?	No	No	No
55	Laser–detector OL2R input?	Yes	Yes	No
56	PIBZLT mount system input?	No	No	No
57	Ball-Rod-Tubing Connector system input?	No	No	No
58	Vernier-Strobe system input?	No	No	No
59	Indikon system input?	No	No	No
60	Calculates “shoot-for” dial indicator readings?	Yes	Yes	No

FIGURE 15.37 (continued)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Prüftechnik ROTALIGN Ultra www.pruftechnik.com	Prüftechnik NOVALIGN www.pruftechnik.com	SPM Instruments Inc. LineLazer www.spminstrument.com
61	Calculates "target values" for laser systems?	Yes	Yes	No
62	Automatically shifts shaft positions in side and top views for OL2R data?	Yes	Yes	No
63	Solves correction moves for one machine case?	Yes	Yes	Yes
64	Solves correction moves for several machine elements?	Yes	Yes	Yes
65	Variable position overlay line (aka final desired alignment line)?	Yes	Yes	No
66	Total number of possible alignment correction moves (see list)	Infinite	Infinite	Unlimited
67	Compares existing alignment to alignment tolerance guide?	Yes	Yes	Yes
68	Show progress of each move on the alignment tolerance guide?	Yes	Yes	Yes
69	Shows graphical side view model of current shaft positions?	Yes	Yes	Yes
70	Shows graphical top view model of current shaft positions?	Yes	Yes	Yes
71	Show or hide grid background?	No	No	No
72	Graph or model in color?	Yes	Yes	Yes
73	Select grid background color?	No	No	No
74	S2elect shaft centerline position color?	No	No	No
75	Shows machinery side view configuration?	Yes	Yes	Yes
76	Shows machinery top view configuration?	No	No	Yes
77	Shows soft foot gap information?	Yes	Yes	Yes
78	Shows alignment method used?	Yes	Yes	Yes
79	Shows alignment data entered?	Yes	Yes	Yes
80	Shows bracket sag amount?	Yes	Yes	No
81	Shows "shoot-for" readings?	Yes	Yes	No
82	Show desired off-line side view model?	Yes	Yes	No
83	Show desired off-line top view model?	Yes	Yes	No
84	Edit machinery names?	Yes	Yes	Yes
85	Edit machinery dimensions?	Yes	Yes	Yes
86	Edit dial indicator readings?	Yes	Yes	No
87	Edit bracket sag amount?	Yes	Yes	No
88	Edit stationary–movable data?	Yes	Yes	Yes
89	Edit OL2R data?	Yes	Yes	Yes
90	Edit scale factors in side view alignment model?	Auto scale	Auto scale	No
91	Edit scale factors in top view alignment model?	Auto scale	Auto scale	No
92	Edit number of machines in drive train?	Yes	Yes	No
93	Aligns right angle drives?	No	No	No

FIGURE 15.38 Laser system software comparison chart. (*continued*)

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Question Number	Manufacturer System/Model Manufacturer Web Site Question	Prüftechnik ROTALIGN Ultra www.pruftechnik.com	Prüftechnik NOVALIGN www.pruftechnik.com	SPM Instruments Inc. LineLazer www.spminstrument.com
94	Aligns C-flanged and vertical machinery?	Yes	Yes	Yes
95	Cross calculate alignment methods?	Yes	Yes	No
96	Centering a shaft in a bearing?	Yes	Yes	No
97	Start a new file?	Yes	Yes	Yes
98	Open an old file?	Yes	Yes	Yes
99	Close existing file?	Yes	Yes	Yes
100	Save existing file?	Yes	Yes	Yes
101	Save as renamed file?	Yes	Yes	Yes
102	Revert to last saved file?	Yes	Yes	Yes
103	Store each correction move made?	Yes	Yes	Yes
104	Print data?	Yes	Yes	Yes
105	Selectable items for printout? (see list)	Yes	Yes	Vertical and lateral corrections
106	Hard copy (printed) software manual included?	Yes	Yes	No
107	Can the software accept direct input from the measurement sensors?	Yes	Yes	Yes
108	What type of sensors can it accept? (see list)	Laser–detectors	Laser–detectors	SPM lase–detector
109	Patented?	Yes	Yes	Yes
110	What is your warranty period?	12 months	12 months	1 year

FIGURE 15.38 (continued)

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Shaft alignment software questionnaire

1. Are you a manufacturer or author or programmer of shaft alignment software programs?
   ____ Yes
   ____ No

2. Are you a distributor of shaft alignment software programs? If so, please list the names or titles of the software programs and who the manufacturer or creator of shaft alignment software program is. Please indicate the company, address, phone number, contact person, and Web site (if applicable) of the manufacturer or creator. If you are a distributor only, skip the remaining questions.
   Distributor for:
   Software title _____
   Manufacturer/Creator name _____
   Address _____
   P. O. Box • Suite _____
   City _____
   State/Province _____
   Zip/Postal code _____
   Country _____
   Phone # _____
   Fax # _____
   E-mail address _____
   Web site _____
   Manufacturer or author or programmer questions only

3. Does your software run on widely used personal computers or does it require a custom computer?
   ____ Operates on standard computers
   ____ Requires a custom computer

4. If it requires a custom computer, please list the manufacturers of the custom computer and the cost of the unit.
   Manufacturer of custom computer _____
   Cost _____(U.S. dollars)

5. If the software works on widely used personal computers, what computer operating systems will the software run on?
   ___ MS-DOS
   ___ OS2
   ___ Windows 3.XX
   ___ Windows 95
   ___ Windows 98
   ___ Windows Me
   ___ Windows 2000
   ___ Windows NT
   ___ Windows XP
   ___ Macintosh versions 1.0 to 8.0
   ___ Macintosh versions 9.0 to 9.2
   ___ Macintosh OSX
   ___ Unix
   ___ Linux
   ___ Others _____

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6. If the software operates on a custom computer, can it interface with a widely used personal computer?
   ____ Yes
   ____ No (skip questions 7 and 8)
7. If it does interface with widely used personal computers, what computer operating systems will it interface with?
   ___ MS-DOS
   ___ OS2
   ___ Windows 3.XX
   ___ Windows 95
   ___ Windows 98
   ___ Windows Me
   ___ Windows 2000
   ___ Windows NT
   ___ Windows XP
   ___ Macintosh versions 1.0 to 8.0
   ___ Macintosh versions 9.0 to 9.2
   ___ Macintosh OSX
   ___ Unix
   ___ Linux
   ___ Other _____
8. If it does interface with widely used personal computers, how is the interface made?
   ____ IEEE 488
   ____ serial connection, specify _____
   ____ parallel connection, specify _____
   ____ USB
   ____ Firewire
   ____ PCMCIA card, specify _____
   ____ Memory card, specify _____
   ____ WiFi
   ____ Bluetooth
   ____ Others, specify _____
9. What is the price of the software program (in U.S. dollars)? If more than one program is available, please list separately.
   Version# _____
   Price _____ (U.S. dollars)
   Version# _____
   Price _____ (U.S. dollars)
   Version# _____
   Price _____ (U.S. dollars)
10. What was the release date of the programs?
    Release date: _____
11. Minimum RAM required _____
12. Minimum screen pixel density _____
    ____ 640×480
    ____ Others, describe _____

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13. Special graphics required?
    ____ Yes, describe _____
    ____ No
14. Mouse/Trackball required?
    ____ Yes
    ____ No
15. Number of allowable machines in drive train
    ____ 2
    ____ 3
    ____ 4
    ____ 5
    ____ 6
    ____ infinite
16. Number of allowable bolting planes per machine
    ____ 2
    ____ 3
    ____ 4
    ____ 5
    ____ 6
    ____ infinite
17. English or metric dimensional input?
    ____ Yes
    ____ No
18. Software program help incorporated in software?
    ____ Yes
    ____ No
19. Shaft alignment help incorporated in software program?
    ____ Yes
    ____ No
20. Number of detailed machine drawings in library
    _____
21. Generic machine images?
    ____ Yes
    ____ No
22. Edit fields for machine dimensions?
    ____ Yes
    ____ No
23. Selectable side direction (north, south, east, west, left, right)?
    ____ Yes
    ____ No
24. User specified side direction?
    ____ Yes
    ____ No

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25. Editable side direction?
    ____ Yes
    ____ No
26. Selectable flexible coupling design/type?
    ____ Yes
    ____ No
27. Input soft foot gap information?
    ____ Yes
    ____ No
28. Calculate and display shim shapes for soft foot correction?
    ____ Yes
    ____ No
29. Input runout information?
    ____ Yes
    ____ No
30. Input from symmetric reverse dial indicator method?
    ____ Yes
    ____ No
31. Input from asymmetric reverse dial indicator method?
    ____ Yes
    ____ No
32. Input from face–rim dial indicator method?
    ____ Yes
    ____ No
33. Front or backside face readings allowable?
    ____ Yes
    ____ No
34. Input from shaft to coupling spool dial indicator method?
    ____ Yes
    ____ No
35. Input from face–face dial indicator method?
    ____ Yes
    ____ No
36. Input from double radial indicator method?
    ____ Yes
    ____ No
37. Selectable methods at each coupling for multiple element drive trains?
    ____ Yes
    ____ No
38. Input symmetric bracket sag?
    ____ Yes
    ____ No

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39. Input asymmetric bracket sag?
    ____ Yes
    ____ No
40. Input shaft centerline readings only through 180° of rotation?
    ____ Yes
    ____ No
41. Input shaft centerline readings with less than 180° of rotation?
    ____ Yes
    ____ No
42. Manual input from stationary–movable laser systems?
    ____ Yes
    ____ No
43. Manual input from other stationary–movable systems?
    ____ Yes
    ____ No
44. Input vertical allowable movement envelope?
    ____ Yes
    ____ No
45. Input lateral movement envelope?
    ____ Yes
    ____ No
46. Input OL2R movement?
    ____ Yes
    ____ No
47. Input both vertical and lateral OL2R data?
    ____ Yes
    ____ No
48. Generic OL2R movement input?
    ____ Yes
    ____ No
49. Optical alignment OL2R data input?
    ____ Yes
    ____ No
50. Input from inside micrometer–tooling ball OL2R input (triangular setup)?
    ____ Yes
    ____ No
51. Input from inside micrometer–tooling ball OL2R input (pyramid setup)?
    ____ Yes
    ____ No
52. Calculated thermal expansion input?
    ____ Yes
    ____ No

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53. Alignment Bar (Dyn-Align/Dodd) input?
    ____ Yes
    ____ No
54. Acculign (Essinger) Bar input?
    ____ Yes
    ____ No
55. Laser–Detector OL2R input?
    ____ Yes
    ____ No
56. PIBZLT mount system input?
    ____ Yes
    ____ No
57. Ball-Rod-Tubing Connector system input?
    ____ Yes
    ____ No
58. Vernier–Strobe system input?
    ____ Yes
    ____ No
59. Indikon system input?
    ____ Yes
    ____ No
60. Calculates "shoot-for" dial indicator readings?
    ____ Yes
    ____ No
61. Calculates "target values" for laser systems?
    ____ Yes
    ____ No
62. Automatically shifts shaft positions in side and top views for OL2R data?
    ____ Yes
    ____ No
63. Solves correction moves for one machine case?
    ____ Yes
    ____ No
64. Solves correction moves for several machine elements?
    ____ Yes
    ____ No
65. Variable position overlay line (aka final desired alignment line)?
    ____ Yes
    ____ No
66. Total number of possible alignment correction moves
    ____ 1
    ____ 2
    ____ 3

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____ 4
____ 5
____ 6
____ infinite

67. Compares existing alignment-to-alignment tolerance guide?
____ Yes
____ No

68. Shows progress of each move on the alignment tolerance guide?
____ Yes
____ No

69. Shows graphical side view model of current shaft positions?
____ Yes
____ No

70. Shows graphical top view model of current shaft positions?
____ Yes
____ No

71. Show or hide grid background?
____ Yes
____ No

72. Graph or model in color?
____ Yes
____ No

73. Select grid background color?
____ Yes
____ No

74. Select shaft centerline position color?
____ Yes
____ No

75. Shows machinery side view configuration?
____ Yes
____ No

76. Shows machinery top view configuration?
____ Yes
____ No

77. Shows soft foot gap information?
____ Yes
____ No

78. Shows alignment method used?
____ Yes
____ No

79. Shows alignment data entered?
____ Yes
____ No

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80. Shows bracket sag amount?
    ____ Yes
    ____ No
81. Shows "shoot-for" readings?
    ____ Yes
    ____ No
82. Shows desired off-line side view model?
    ____ Yes
    ____ No
83. Shows desired off-line top view model?
    ____ Yes
    ____ No
84. Edit machinery names?
    ____ Yes
    ____ No
85. Edit machinery dimensions?
    ____ Yes
    ____ No
86. Edit dial indicator readings?
    ____ Yes
    ____ No
87. Edit bracket sag amount?
    ____ Yes
    ____ No
88. Edit stationary–movable data?
    ____ Yes
    ____ No
89. Edit OL2R data?
    ____ Yes
    ____ No
90. Edit scale factors in side view alignment model?
    ____ Yes
    ____ No
91. Edit scale factors in top view alignment model?
    ____ Yes
    ____ No
92. Edit number of machines in drive train?
    ____ Yes
    ____ No
93. Aligns right angle drives?
    ____ Yes
    ____ No

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94. Aligns C-flanged and vertical machinery?
    ____ Yes
    ____ No
95. Cross calculate alignment methods?
    ____ Yes
    ____ No
96. Centering a shaft in a bearing?
    ____ Yes
    ____ No
97. Start a new file?
    ____ Yes
    ____ No
98. Open an old file?
    ____ Yes
    ____ No
99. Close existing file?
    ____ Yes
    ____ No
100. Save existing file?
    ____ Yes
    ____ No
101. Save as renamed file?
    ____ Yes
    ____ No
102. Revert to last saved file?
    ____ Yes
    ____ No
103. Store each correction move made?
    ____ Yes
    ____ No
104. Print data?
    ____ Yes
    ____ No
105. Selectable items for printout?
    ____ Print machinery drive train side view configuration
    ____ Print machinery drive train top view configuration
    ____ Print side view alignment model
    ____ Print top view alignment model
    ____ Print current alignment readings
    ____ Print stationary–movable or laser data
    ____ Print bracket sag amount
    ____ Print OL2R data
    ____ Print current maximum alignment on tolerance guide
    ____ Print gaps and shim shapes for soft foot correction

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____ Print vertical allowable movement envelope
____ Print lateral allowable movement envelope
____ Print help screens
____ Print vertical and lateral movement corrections

106. Hard copy (printed) software manual included?
____ Yes
____ No

107. Can the software accept direct input from the measurement sensors?
____ Yes
____ No

108. If the software can accept input from the measurement sensors, what type of sensors can it accept?
____ electronic dial indicators
____ laser–detectors
____ optical encoders
____ LVDTs
____ CCDs
____ others, describe _____

109. If your system is patented, please send a copy of the patent (in English). If you have patents in other countries, please indicate where the patents are held and their corresponding patent numbers or identification.

_____

110. What is your warranty period? _____
111. What are the recommended calibration intervals, does the unit have to be sent back to the factory for calibration, and what is the charge for recalibration?

_____

112. Is the calibration you offer traceable back to U.S. or International Standards?
____ Yes
____ No

113. Do you offer equipment for the user to check and adjust calibration? If so, what is the cost of the test equipment?
____ Yes, cost _____
____ No

114. If you have introduced new shaft alignment systems models, do the people who own your original models have the option to trade in their older models to upgrade to a newer model? If so, what is the cost of the upgrade?
____ Yes, cost _____
____ No

115. What is the price (or price range) of the software programs?

_____

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16 Measuring and Compensating for Off-Line to Running Machinery Movement

Up to this point in the book, we discussed how to align rotating machinery when it is shut down and hopefully in a very stable position. Once the shafts are aligned within acceptable tolerances, the coupling engaged, and the machinery started up, changes in the positions of the shafts may begin to occur that could alter the accuracy of the alignment when the equipment is running.

Virtually all rotating equipments will undergo a change in position during start-up and while running that affects the alignment of the shafts. The driver machine may move a certain amount and in a certain direction as it goes from an off-line to a stable operating condition. The driven elements of the system may move different amounts and in different directions. In order for the shafts to run collinear under normal operating conditions, it is desirable to know the amount and direction of this movement to properly position the machinery during what is commonly called the "cold" (i.e., off-line or not running) alignment process to compensate for this change.

16.1 WHAT TYPE OF MACHINERY IS LIKELY TO CHANGE EQUIPMENT'S POSITION WHEN RUNNING?

The OL2R movement characteristics of the vast majority of rotating machinery drive systems in existence have never been measured. Based on observations made on a relatively small percentage of drive systems currently operating, it appears that perhaps on 60%–70% of the drive systems in existence, the amount of movement is negligible and can basically be ignored. In the remaining cases however, this movement can make all the difference between a trouble-free drive system and one that is plagued with problems. It is important to know how much movement is occurring before you deem it insignificant and ignore it. The most baffling question is which rotating machinery do you have that is moving enough from OL2R conditions where you need to measure and compensate for this movement.

Below is a broad list of machinery that is likely to change its position enough from OL2R conditions to warrant measuring this occurrence:

- Rotating machinery drive systems running at or above 200 hp and speeds of 1200 rpm or greater
- Machinery that undergoes a change in casing temperature

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For example:

(1) Electric motors and generators
(2) Steam turbines
(3) Gas turbines
(4) Internal combustion engines (diesels, etc.)
    - Speed changers (e.g., gearboxes and fluid drives)
    - Machinery that is pumping or compressing fluids or gases where the fluid or gas undergoes a change in temperature by 50° or greater from intake to discharge (this could be either a rise or drop in temperature). For example:

(1) Centrifugal or reciprocating compressors
(2) Centrifugal pumps
(3) HVAC air moving equipment
(4) Furnace fans
    - Equipment with poorly supported piping attached to the machine case where expansion or contraction of the piping induces forces into the machine case or where fluid flows can cause a reactionary moment in the piping

16.2 WHAT CAUSES MACHINERY MOVEMENT TO OCCUR?

There are a variety of factors that cause machinery to move once it is on line and running. The most common cause is due to temperature changes in the machinery itself (as it compresses gases or heats the lubricant from friction in the bearings) and is therefore generally referred to as "thermal" or "cold" to "hot" movement. The temperature change in rotating machinery is rarely uniform throughout the casing, which causes most equipment to "pitch" at some angle rather than grow (or shrink) straight up (or down). For compressors, turbines, and pumps, thermal expansion (or contraction) of the attached piping may also cause the equipment to shift.

Other sources of movement in machinery can be caused by loose foot bolts, varying weather conditions for equipment located outdoors, heating or cooling of concrete pedestals, changes in the operating condition of equipment from unloaded to loaded postures, or casing and support counterreactions to the centrifugal force of rotors as they are spinning.

Special alignment considerations must be taken into account for equipment that is started and stopped frequently or where loads may vary considerably while running. In cases like these, a compromise has to be made that weighs factors such as period of time at certain conditions, total variation of machinery movement from maximum to minimum, coupling and alignment tolerances, etc. To properly observe and record these changes, periodic checks should be made of this change in movement to fully understand how to effectively position the equipment to minimize any damaging effects from running misalignment. Continuous shaft position monitoring systems are available and are explained later in this chapter.

It has been my experience however that the majority of rotating equipment will typically maintain one specific position regardless of varying loads. What usually turns out to be a bigger problem is that some equipment may have to be offset aligned "cold" by a considerable distance making start-ups very nerve wracking. In the majority of cases, equipment will undergo the greatest rate of change of movement shortly after start-up. "Shortly" can

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mean anywhere from 5 min to 1 h for most types of equipments and may settle at some final position several hours, days, or even weeks later.

As equipment goes from a running to an off-line (R2OL) condition (i.e., coast down to shutdown), a wide variety of movement amounts and directions can occur. Some equipment may make a very rapid change immediately after shutdown. Other drive systems may flounder around and then slowly move back near their original position. To my knowledge, there has never been any published information where someone has observed several start-up and shutdown sequences to see if the machinery continues to attain exactly the same positions in the off-line and in the running conditions. Most people just assume and hope that this happens.

16.3 CONDUCTING THE OFF-LINE TO RUNNING MACHINERY MOVEMENT SURVEY

Observing the movement of rotating machinery may seem quite complicated at first glance, but these measurements are nothing more than a comparison between the position of the centerlines of rotation when the machinery is off-line to the position of the centerlines of rotation when the machinery is running, frequently referred to as off-line to running (OL2R) condition. Therefore, data must be taken when the machinery is off-line (cold) and then again when the machinery is running (hot). It is also feasible to reverse the sequence of data collection. In other words, take measurement readings when the machines are running, shut the machinery down, and take readings again (i.e., R2OL). In fact, it is recommended that both OL2R and R2OL measurements be taken just to see if the movement is consistent and repeatable.

There are many inventive ways to measure shaft alignment positions from OL2R (or vice versa) and this chapter will review several techniques. The OL2R movement data that are typically measured are often quite surprising and there is a great tendency not to believe the results. Each OL2R method has its advantages and disadvantages and it is a good idea to compare the measurements from two or more methods just to see if the results are similar.

There are currently 11 different techniques that have been used to measure this movement that will be explained in greater detail in this chapter. One very popular practice is not recommended since there are a number of potential problems with it as explained herein.

16.4 TAKING "HOT" ALIGNMENT MEASUREMENTS IMMEDIATELY AFTER SHUTDOWN

One popular, but relatively inaccurate method people have done is to take a quick "hot" alignment reading with their shaft alignment measurement system immediately after a unit has been shut down. There are some problems doing this however.

Safety tagging the machinery to prevent them from inadvertently starting up with alignment measurement devices attached to the shafts, removing the coupling guard, mounting the shaft alignment measurement system quickly enough to get a set of readings, and then getting an accurate and repeatable set of readings while the shafts are still moving proves to be a real challenge for the personnel doing the work.

This method is not recommended for the following reasons:

1. The machinery is no longer running. You cannot assume that the machinery is in exactly the same position as when it was running. Reactionary moments of the rotating elements and reactionary moments when fluids were flowing through the machine and the

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attached piping will not be present with the equipment off-line. On small drive systems, it can take but a few seconds for the shafts to stop turning. On larger drive systems with a lot of rotating momentum, it can take several minutes for the shafts to stop turning. The instant you press the stop button, the equipment begins moving. By the time the shafts stop rotating, the equipment is properly safety tagged, the coupling guard is removed, a shaft alignment system is attached to the shafts, and a complete set of alignment readings have been taken so that a period of 15 min to 1 h has already elapsed. What happened during that time frame?

2. There is a safety related issue here. The equipment you will be working around may very well be physically hot from a temperature standpoint and hot enough to physically harm you. What if somebody forgets to safety tag the machine and it starts backup while you are trying to attach the alignment tooling (or have already attached the tooling)? You are more apt to make a mistake if you are in a hurry.
3. The shafts are probably moving while you are taking the alignment readings. If you plan to attempt this, it is suggested that you take a series of readings, say every 15 to 30 min, during the cooldown period to plot the movement and extrapolate them back to the instant when the unit was first shut down to determine the actual shaft positions when running. The data are usually nonlinear and guessing the slope of the curve during the first 15 min to 1 h period after the unit has been shut down is a hit or miss proposition. For the above reasons, this technique is not recommended but if this is all you are willing to do, then it is better than guessing or doing nothing at all.

16.5 FOUR GENERAL CATEGORIES OF OL2R MEASUREMENTS

There are four broad classifications of measurement techniques employed to capture OL2R machinery movement:

- Movement of the centerlines of the machine cases with respect to their baseplate or frame
- Movement of the centerlines of the machine cases with respect to some remote reference or observation point
- Movement of one machine case with respect to another machine case
- Movement of one shaft with respect to another shaft

Just as there are advantages and disadvantages of each of the shaft alignment measurement techniques, there are advantages and disadvantages of each of the OL2R techniques shown in this chapter. After conducting these surveys for many years on a wide variety of equipment, there is always a certain level of uncertainty in the measurement data or in the setup of the equipment. Since the results of these surveys can be quite surprising, there is a tendency to disbelieve the first set of measurements. So, what do you do?

Often you repeat the test again and see if the results of the second set of measurements somewhat agree with the results of the first set. If there is a wide variation between both sets of results (e.g., the inboard end of the compressor looked like it moved 62 mils upward the first time and 38 mils downward the second time), you need to carefully review every aspect of the OL2R equipment setup, the measurement sensors, when and how you collected the data to determine how this occurred.

One of the reasons why several methods are shown in this chapter is to give you the opportunity to compare the results from two or more different techniques. It is not uncommon to repeat the OL2R survey two or three times getting very similar results each time but the information you gather may appear to be unbelievable to you and other people. So, now

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what do you do? Try a different measurement method to see if the results from that method agree with the first method performed.

There are several important issues that need to be mentioned concerning OL2R shaft movement measurements of rotating machinery:

- Fewer than 10% of the people who are responsible for alignment of rotating machinery in industry have ever actually conducted OL2R machinery movement surveys. Some people are aware of this phenomenon but only a few of them have actually tried to measure it.
- Only a small percentage (my guess is less than 5%) of the rotating machinery in existence has ever been measured for OL2R movement. Since very few drive systems have been studied, much has yet to be learned about what happens to the alignment of machinery when it is running.
- Many of the people who have conducted OL2R surveys or are aware of this phenomenon believe that machinery only moves in the up and down direction disregarding the possibility (and probability) of sideways or axial movement.
- Compared to the amount of time usually taken to align rotating machinery when off-line (which takes 3–6 h on the average), OL2R machinery movement surveys can take days, weeks, and even months to complete. These types of surveys are frequently conducted on the more critical pieces of rotating machinery in a plant and these drive systems usually cannot be turned on and turned off at the convenience of the people performing the test. So be prepared to spend more time for planning, designing, fabricating, installing, measuring, and analyzing these data than you typically would for "off-line or in-line" shaft alignment jobs.
- Without a doubt, getting accurate and repeatable information using any of these techniques is far more difficult than you can imagine. Until you actually try one or more of these techniques and attempt to get repeatability, you will not appreciate the degree of difficulty. Every time I perform one of these surveys, I am always discovering new sources of errors and trying to figure out improvements to the tooling in an attempt to better control the measurement process. About 90% of the time, you are going to have to invent some sort of fixture or holding device or establish a good reference or measurement point. Usually the measurements you take with the machinery off-line are not going to be on the same day that you capture the measurements with the machinery running in a stable operating condition. This means one of the three things:

(1) You can keep all of the OL2R equipment setup and hope nothing happens to it during the "warm-up" cycle.
(2) You can keep some of the equipment setup and store the rest until your next measurement time, then set the stored equipment backup again.
(3) You can set it all up, take it all off, and then set it all up again.

No matter what choice you pick, there are going to be any number of unforeseen events that can obliterate your efforts if you do not totally control the process. You must always be aware of any possible problem that can occur and have a backup plan if possible. You may not get everything right the first time and you might have to do it again without the same bugs biting you. Persistence, consistency, and imagination are the key ingredients to success here.

- In several cases where these data have been collected, and it was verified that the machinery was not in the right alignment position when off-line, there was a great reluctance (and in some cases pure refusal) to change the existing positions of the machinery to reflect the newly discovered information. The reasoning typically expressed

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was "the machinery has been in this position for several years, why change it now?" or "if we move the machines, we will void our warranty" or "we have to wait until the next shutdown to change the alignment (which seems to be forgotten at the designated shutdown time)."

It seems very silly to spend inordinate amounts of time, effort, and money on aligning machinery to extreme precision when the equipment is not running and ignore OL2R machinery movement. What has been accomplished if the machinery has been aligned to 1/2 mil/in. or better in an off-line condition if the machinery moves 20 or 30 mils when it is running?

16.5.1 Movement of the Centerline of the Machine Cases with Respect to Its Baseplate or Frame

The principle of operation in this category is to measure points very near the centerline of rotation at the inboard and outboard ends of each machine case in the drive train when the equipment is not running, start the machinery up, periodically observe the measurement points during operation until they have stopped moving, and compare the running positions of the points to the off-line positions of the points. The reference point is the machinery baseplate or foundation.

Tools and techniques that fall into this category are:

- Calculating machine case thermal expansion using the strain equation
- Inside micrometer–tooling ball–angle measurement devices
- Proximity probes with water-cooled stands

16.5.2 Movement of the Centerline of the Machine Cases with Respect to Some Remote Reference or Observation Point

The principle of operation in this category is to observe a point very near the centerline of rotation at the inboard and outboard ends of each machine case from a position away from the drive train when the equipment is not running, start the machinery up, periodically observe the measurement points during operation until the measurement points have stopped moving, and compare the running positions of the points to the off-line positions of the points.

Tools and techniques that fall into this category are:

- Optical alignment equipment
- Laser–detector measurement systems
- Videometry system (dual CCDs) and electronic theodolite system

16.5.3 Movement of One Machine Case with Respect to Another Machine Case

The objective of these techniques is to measure the positional change of one machine case with respect to another machine case.

Tools and techniques that fall into this category are:

- Alignment bars or custom fixtures with proximity probes
- Laser–detector systems with custom-fabricated brackets or special mounting systems
- Ball–rod–tubing connector system

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16.5.4 Movement of One Shaft with Respect to Another Shaft

Since we are trying to determine where the shaft centerlines are actually moving to from OL2R conditions, it would seem that the ultimate method would be to watch the movement of one shaft with respect to the other shaft directly. The obvious problem with this is that when the machinery is running, the shafts are rotating and placing sensors on moving shafts seems to be an impossible task yet there are two methods that have been invented to do just that.

Tools and techniques that fall into this category are:

- Vernier–strobe method
- Instrumented coupling system

16.6 CALCULATING MACHINE CASE THERMAL EXPANSION USING THE STRAIN EQUATION

At the atomic level in solid materials, the temperature and volume of the material are dictated by the vibration of the individual molecules. In other words, the hotter a solid material gets, the more the molecules vibrate and the farther apart the molecules are spaced. This phenomenon causes changes in dimensions (i.e., strain) that can be calculated by the equation shown in Figure 16.1.

The coefficients of thermal expansion for the majority of materials used in machinery casing and foundation are shown in Table 16.1. These coefficients can be used for temperatures between 32°F and 350°F. There is a slight shift in the value of the coefficient for higher or lower temperatures due to the nonlinearity of molecular vibration in materials.

There are three measurements needed: the temperature of a machine case or support structure using thermometers, noncontact infrared thermometers, or surface temperature probes such as resistance temperature detectors (RTDs) or thermocouples when the equipment is not running and in a stabilized "cold" or off-line condition; the temperature of a machine case or support structure when the equipment is running and in a stabilized "hot" or operating condition; and the distance from the point where the feet are touching at the base to the centerline of rotation of the shaft (variable "L" in Figure 16.1). You also need to determine what type of material the machine case or support leg and structure is made of to determine the coefficient of thermal expansion as listed in Table 16.1. Subtract the running temperature from the off-line temperature to get the change in temperature (variable ΔT in Figure 16.1). Then plug all this data into the strain equation to arrive at ΔL, or the change in length. Figure 16.2 shows an example of a thermal strain calculation on a multistage compressor.

Thermal strain equation

$$\Delta L = L\,(\alpha)(\Delta T)$$

where ΔL is the change in dimension/length (in.),
L is the length of the object (in.), α is the coefficient of thermal expansion/contraction (in./in.-°F), and
T is the change in temperature (°F).

FIGURE 16.1 Thermal strain equation.

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TABLE 16.1
Coefficients of Thermal Expansion for Different Materials

Material	Coefficient of Thermal Expansion (α) (mil/in.-°F)
Aluminum alloys	12.5×10^{-6}
Brass (70% Cu–30% Zn)	11.0×10^{-6}
Carbon steel (AISI 1040)	6.3×10^{-6}
Cast iron (gray)	5.9×10^{-6}
Concrete	7.2×10^{-6}
Invar	0.68×10^{-6}
Nickel steel	7.3×10^{-6}
Stainless steel	9.8×10^{-6}
Vulcanized rubber	45.0×10^{-6}
Nylon	55.0×10^{-6}

Key considerations for capturing good readings:

- Make sure the machine is in a stabilized off-line and stabilized running position by taking at least three measurements that repeat (± 2 mils) at each condition.
- Carefully observe how the machine case is attached to the baseplate to insure that you measure the support leg, machine case, sway bar, or frame support that is truly repositioning the shafts' centerline.
- Capture several temperature measurements particularly during running conditions in the event that there is an uneven temperature profile on the support mechanism.
- Measure the expansion or contraction at both ends of each machine case and realize that the amount of OL2R movement may not be the same at both ends.
- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

Advantages:
- Relatively inexpensive and easy to do

Disadvantages:
- Results may be suspected based on when and where the temperature reading was taken on the machine case
- Unable to calculate how machinery moves laterally (e.g., sideways)

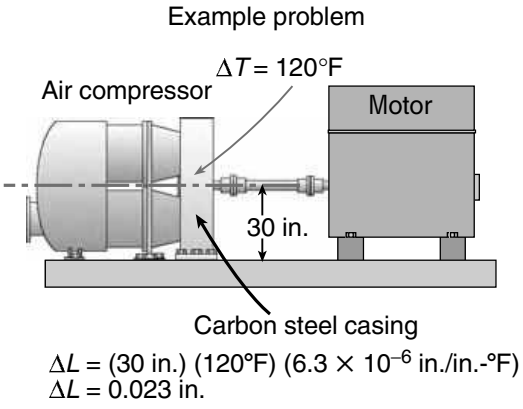


FIGURE 16.2 Example of calculating thermal expansion.

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16.6.1 Using Infrared Thermographic Equipment to Observe Thermal Profiles of Rotating Equipment

Rotating machinery casings transfer heat to the environment when running that is generated from friction, compression of process gases, steam flow, or motor stator windings. Surface temperatures at various locations on a casing vary widely and virtually no rotating equipment maintains a constant thermal gradient across the entire casing. Gas turbines, for instance, may have inlet casing temperatures below the ambient air temperature and 6 ft away have a 1200°F casing temperature at the combustor section. Since our eyesight is limited only to the visible spectrum (see Section 6.2.11), we are unable to "see" the temperature gradient profile of machine cases as they emit the longer, infrared radiation. Infrared radiation can be categorized into four general ranges:

1. Actinic range: incandescent objects such as bulb filaments having wavelengths near the visible red region
2. Hot object range: objects with temperatures around 400°F
3. Calorific range: objects with temperatures around 250°F
4. Warm range: objects having temperatures below 200°F (approximately 9000 nm)

The infrared radiation emitted from an object can be observed by thermographic equipment. The type of equipment used for thermographic studies is shown in Figure 16.3. These



FIGURE 16.3 Thermographic scanning equipment. (Courtesy AGA Corp., Secaucus, NJ. With permission.)

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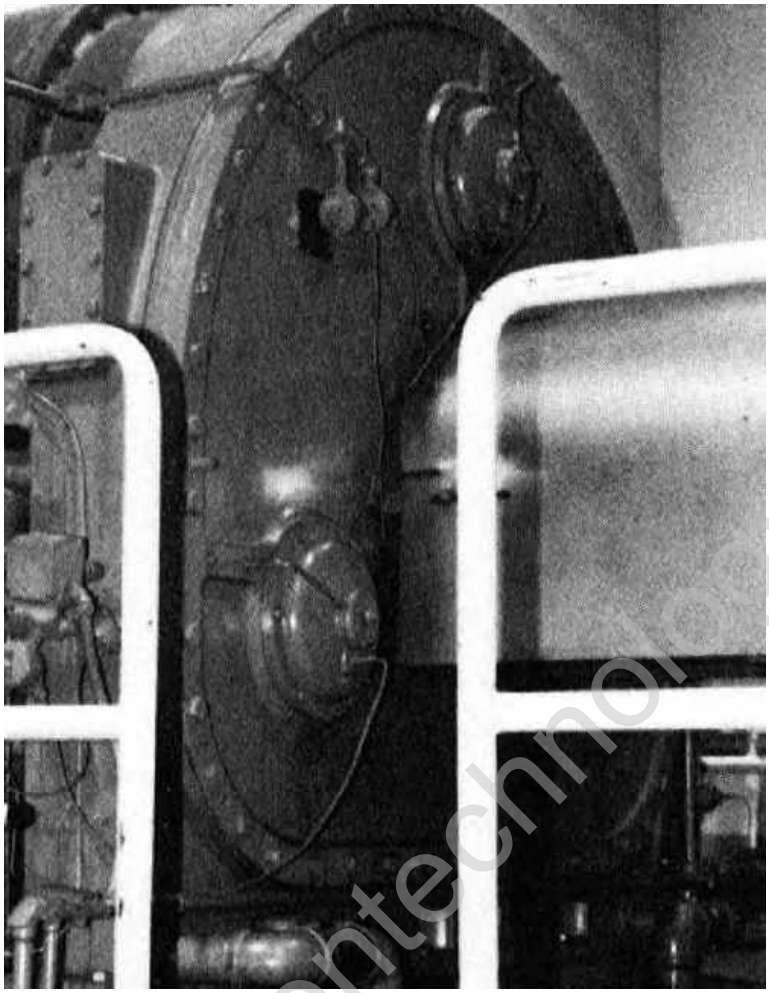


FIGURE 16.4 Compressor case.

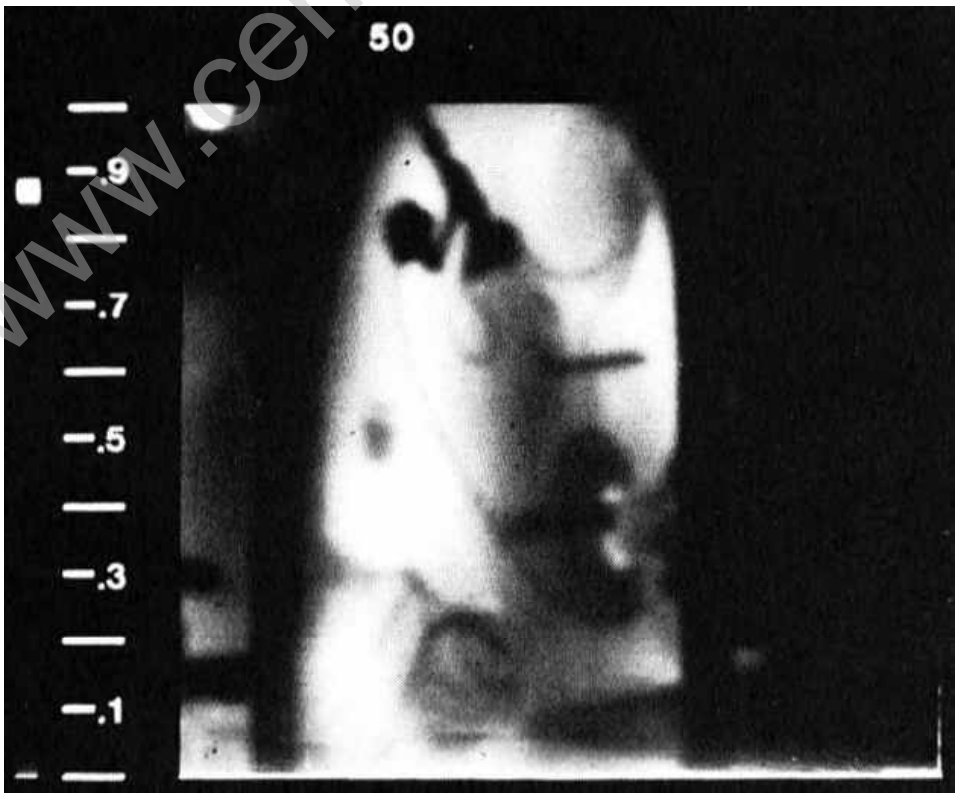


FIGURE 16.5 Thermal image of compressor case.

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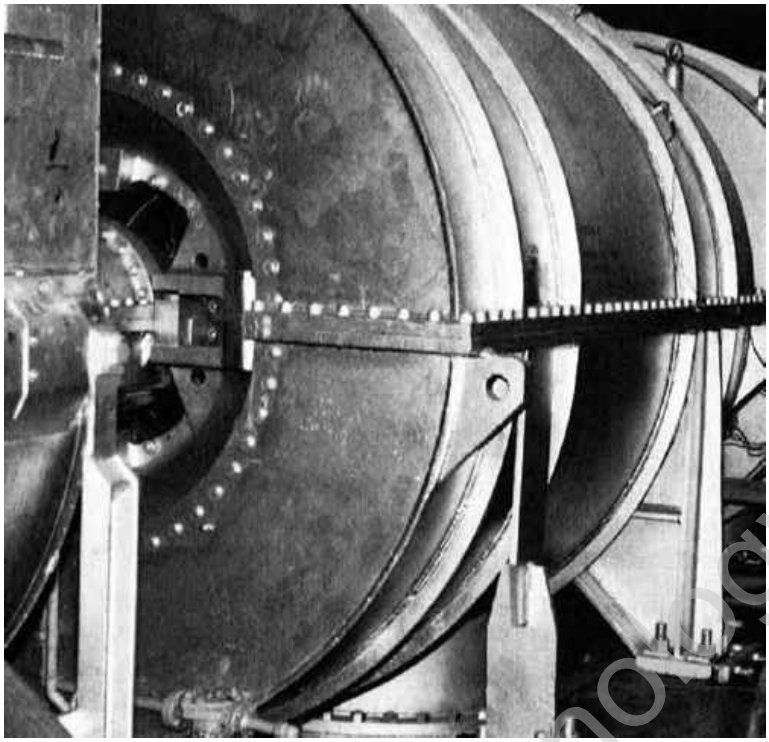


FIGURE 16.6 Axial flow compressors.

instruments scan the object for the infrared radiation and amplify the converted electrical signals from a supercooled photodetector onto a cathode ray tube (CRT), where a photographic image of the object can be recorded.

Figure 16.4 shows a three-stage centrifugal compressor case and Figure 16.5 illustrates the temperature profile when the compressor is running under full load. The white areas show where the infrared radiation (heat) is the greatest. The hottest areas in the image are approximately 135°F.

Figure 16.6 shows an axial flow compressor with rigid supports at the inlet end and flexible supports at the discharge end. Figure 16.7 illustrates the thermal profile of the discharge end with the compressor running under load (note the hot spot at the one o'clock position). Figure 16.8 shows a closer view of the flexible support leg. The lifting eye is at the left side of the photograph and the flex leg is the black portion just to the right of the lifting eye. The photograph clearly shows that the support leg stays at ambient temperatures and does not expand thermally (as originally thought when the machinery was installed).

Although movement of rotating machinery casings does not occur solely from temperature changes in the supporting structures and the casings themselves, infrared thermographic studies can assist in understanding the nature of the thermal expansion that is taking place.

16.7 INSIDE MICROMETER–TOOLING BALL–ANGLE MEASUREMENT DEVICES

Another technique that falls into the category of movement of a machine case centerline with respect to its baseplate is performed using tooling balls as reference points and measuring the distance between the tooling balls with inside micrometers or with an inside micrometer and an inclinometer (angle measuring device).

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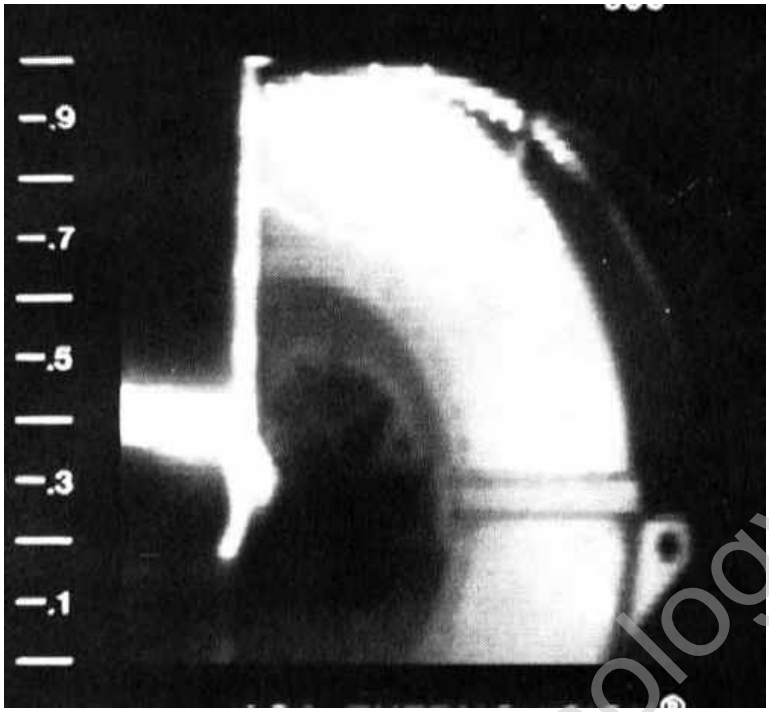


FIGURE 16.7 Thermal image of compressor end casing.

Tooling balls can be purchased from a tool and die supplier or they can be handmade. Figure 16.9 shows a fabricated tooling ball and Figure 16.10 shows how it was made. Figure 16.11 shows the basic setup of the tooling balls on the baseplate and machine cases. Figure 16.12 through Figure 16.14 show the measurements taken by employing this technique.

A traditional inside micrometer could be used for these measurements but environmental problems could occur. When capturing the running or hot measurements, any heat radiating from a machine case or even your hands could (and will) increase the temperature of the micrometer itself, changing its length. It is not uncommon to measure distances of 20 to 40 in. from tooling ball to tooling ball. If you are taking a 30 in. measurement and the carbon steel inside micrometer goes from 60°F to 120°F, the micrometer length will change by 0.013 in. (13 mils). Not consistently accurate enough when you are trying to measure ± 1 mil in positional change. Figure 16.15 and Figure 16.16 show a custom made set fabricated from invar to considerably reduce the inside micrometer thermal expansion error.

Tooling balls or similar reference point devices are rigidly attached to the foundation and to the inboard and outboard ends of each machine case as near as possible to the centerline of rotation as shown in Figure 16.17 through Figure 16.19. Distances between the tooling balls (and angles if desired) are captured for each tooling ball when the machinery is at rest and then measured again when the equipment is running and has stabilized thermally. Three tooling balls can be set up in a triangular pattern as shown in Figure 16.20 at each bearing on each machine in the drive train. A more accurate method is to set up four tooling balls in a four-sided ''pyramid'' arrangement at each bearing on each machine in the drive train as illustrated in Figure 16.21.

These measurements can then be triangulated mathematically into vertical and lateral components (using the triangular arrangement) or into vertical, lateral, and axial component distances (using the four-sided pyramid arrangement). By comparing the coordinates of the tooling ball mounted on each end of all the machine cases from OL2R (or from R2OL)

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FIGURE 16.8 Thermal image of the support leg.

positional changes can be determined. Figure 16.22 shows the mathematics for a triangular tooling ball arrangement and Figure 16.23 for a pyramid arrangement.

Key considerations for capturing good readings:

- Remember that you will probably be dealing with oblique triangular arrangements not right angle triangles (i.e., watch your math).
- Important to have stable positions for the tooling balls.

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FIGURE 16.9 Tooling ball fabricated from 0.5 in. steel ball and 1.5 in. × 1.5 in. × 0.25 in. steel plate with the ball welded to the plate.

- The tooling ball on the machine case should be located as close as possible to the centerline of rotation since we are trying to determine where the shafts are going (if the bearing moves, the shaft is sure to move with it).
- Recommend that concave tips be used at both ends of the inside micrometer to consistently seat on the round tooling balls when taking measurements.
- Keep the inside micrometer away from heat sources to prevent the mike from thermally expanding.

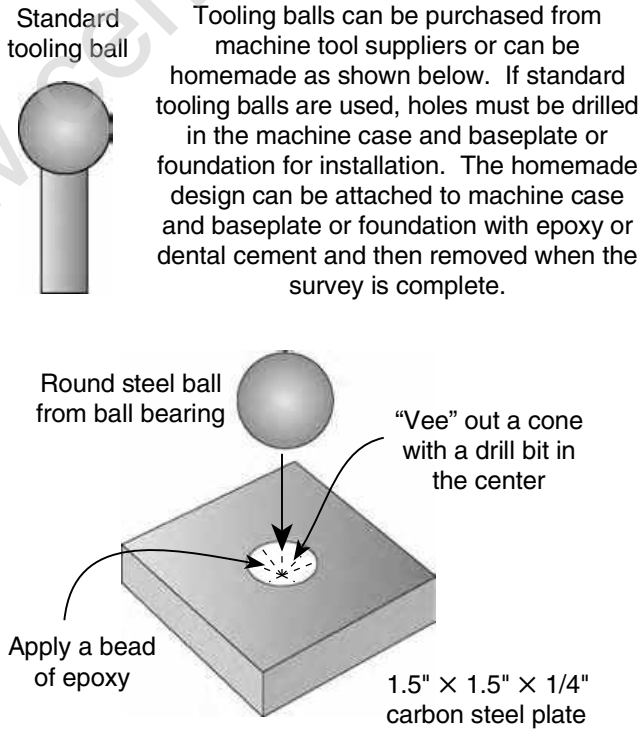


FIGURE 16.10 How to construct a tooling ball.

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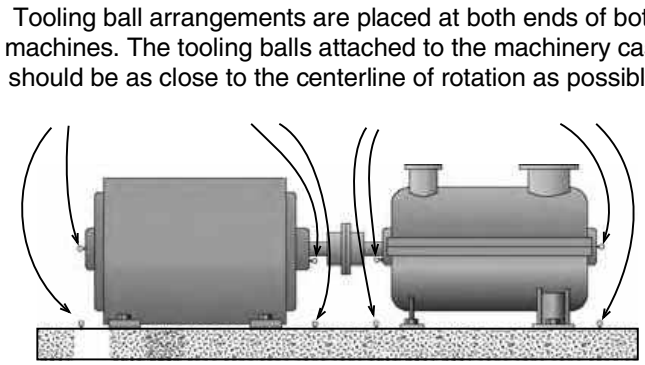


FIGURE 16.11 Basic tooling ball setup on the machinery.

- During measurements have a reference standard length comparator to insure the micrometer itself is not thermally expanding or contracting.
- Triangular tooling ball arrangements assume that there will be motion in the horizontal and vertical planes only which may not necessarily be the only directional change that is occurring (namely axial).
- For best accuracy, use the pyramid arrangement with four tooling balls.
- Have two or more people to take measurements and compare notes to insure the readings are identical (or at least close).
- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

Advantages:
- Relatively inexpensive
- Somewhat easy to set up



FIGURE 16.12 Measuring between two tooling balls with inside micrometer.

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FIGURE 16.13 Measuring a distance with the Acculign invar rods.

Disadvantages:

- Mathematics somewhat tedious particularly on four-sided pyramid arrangements.
- Caution must be taken during running measurements since one end of the inside micrometer is frequently near a rotating shaft.
- If one or more tooling balls disengage from their positions (i.e., it worked out of its hole or the epoxy gave away), you will probably have to start over.

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FIGURE 16.14 Measuring the angle with the Acculign inclinometer.

Figure 16.24 shows the results of an OL2R survey conducted on a motor-fluid drive-boiler feed water pump using the inside micrometer-tooling ball method. A pyramid tooling ball arrangement was used on this drive system. Notice the amount of movement in not only the up and down and side-to-side directions but also the axial amount of movement. Figure 16.25



FIGURE 16.15 Acculign kit. (Courtesy of Acculign, Austin, TX, www.acculign.com. With permission.)

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FIGURE 16.16 Acculign micrometer in calibration fixture. (Courtesy of Acculign, Austin, TX, www.acculign.com. With permission.)

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FIGURE 16.17 Tooling ball setup to measure outboard bearing on motor.

shows the desired off-line shaft position alignment models for the side and top views for the motor-fluid drive-boiler feed water pump shown in Figure 16.24.

16.8 VERTICAL, LATERAL, AND AXIAL OL2R MOVEMENT

Before we go any farther into these methods, it would be prudent to closely examine the OL2R data observed on the motor-fluid drive-boiler feed water pump drive system in Figure 16.24. Pay particular attention to the amount of movement that was observed in the axial direction.

As you can see, there was more movement of each machine case axially than there was in the vertical or lateral (side-to-side) directions. On the motor, the outboard end moved 19 mils



FIGURE 16.18 Tooling ball setup to measure inboard bearings on motor and hydraulic clutch.

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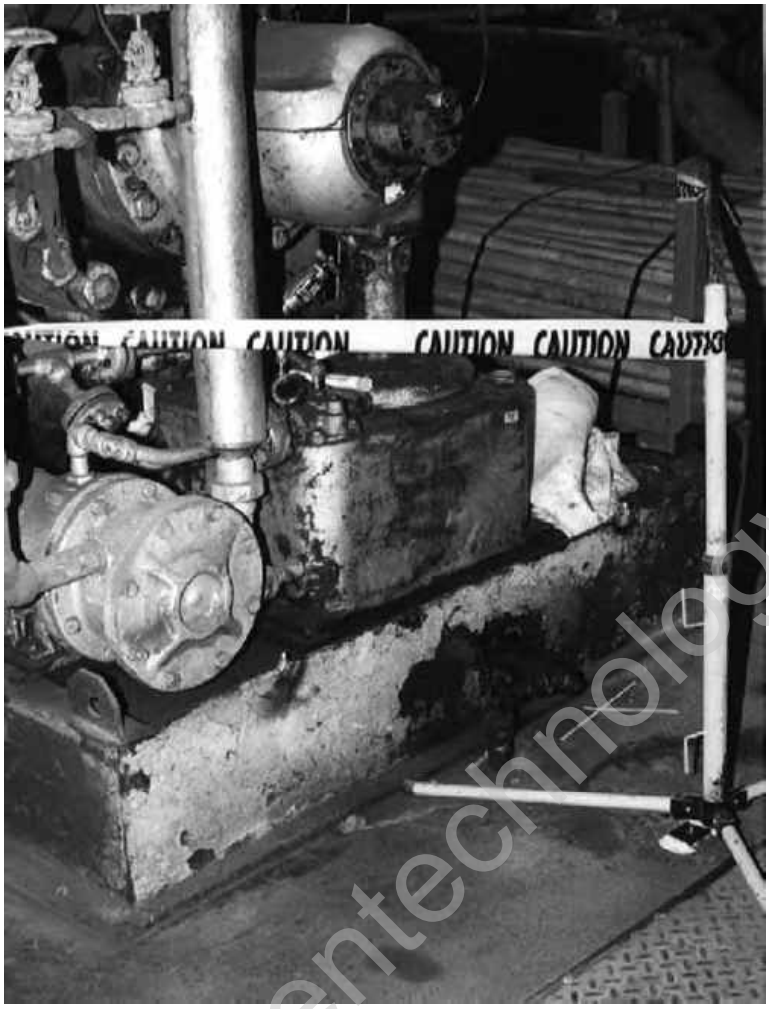


FIGURE 16.19 Tooling ball setup to measure outboard bearing on pump.

to the west and 35 mils to the east at the inboard end for a total of 54 mils of axial expansion. The fluid drive moved 22 mils to the west on the motor end and 8 mils to the east at the pump end for a total of 30 mils of axial expansion. The pump moved 51 mils to the west on the fluid drive end and 88 mils to the east at the outboard end for a total of 139 mils of axial expansion (that is over 1/8 in.).

If we bolt and dowel pin the pump to the baseplate in each corner, and the pump case expands one eighth on an inch and the baseplate does not expand at all, something has to give. Either the foot bolts and dowel pins have to bend or shear, or the pump case has to distort, or both. If the pump distorts, rotating parts inside the pump may begin contacting stationary parts inside the pump damaging the rotor and potentially resulting in a catastrophic failure.

To prevent case distortion from thermal expansion, transverse keys are sometimes used as shown in Figure 16.26. At the coupling end of the pump, a key is placed between the lower pump casing and the baseplate at a 90° angle to the centerline of rotation. The purpose of this key is to hold the pump case here and any axial expansion occurs outward from this point. This key is placed near the coupling end to minimize the amount of movement of that machine toward the other machine. Another key is placed at the outboard end of the machine but this key is placed between the lower pump casing and the baseplate at a 0° angle to the centerline of rotation. This allows the casing to expand in line with the key but prevents the machine case from moving from side to side. Additionally, the inboard bolts are tightened to

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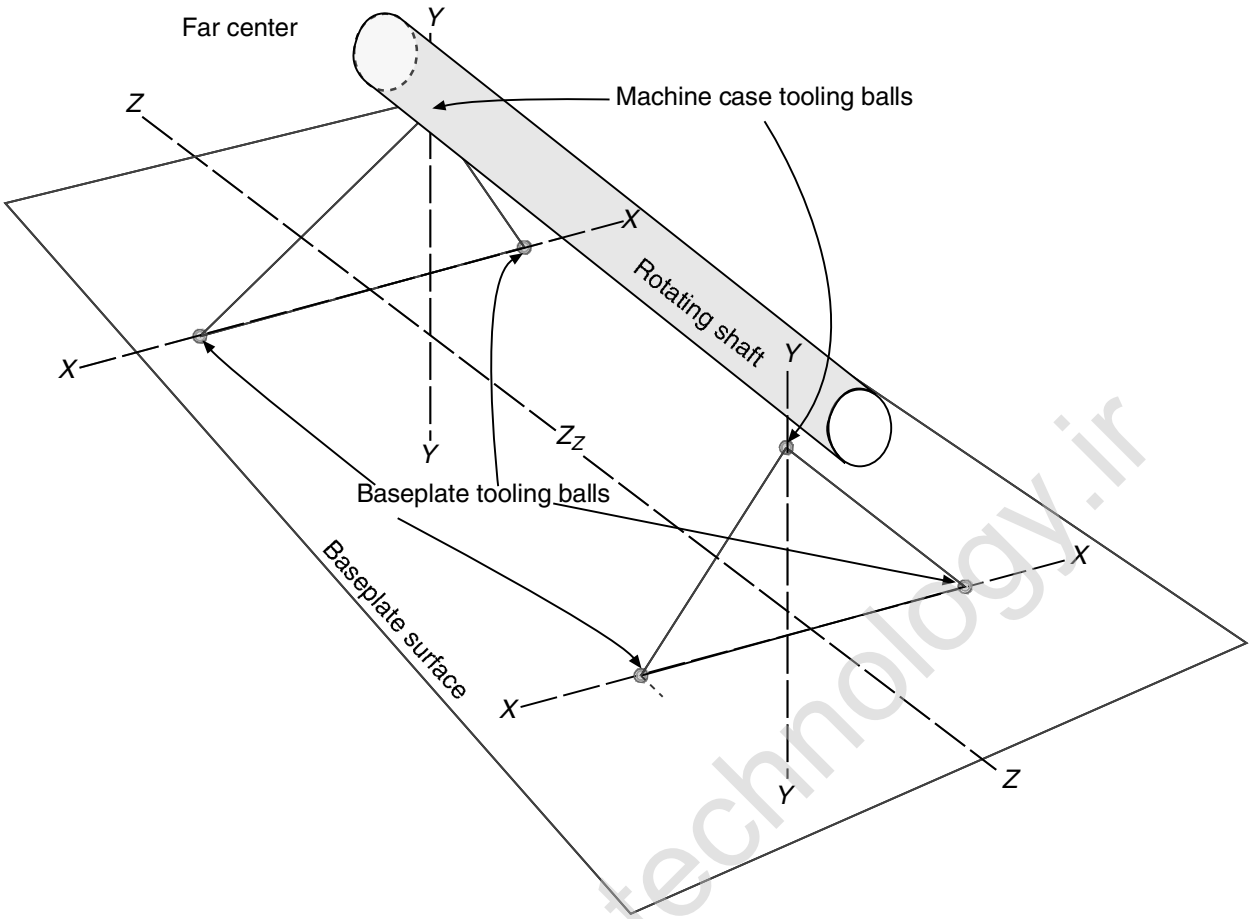


FIGURE 16.20 Triangular tooling ball setup.

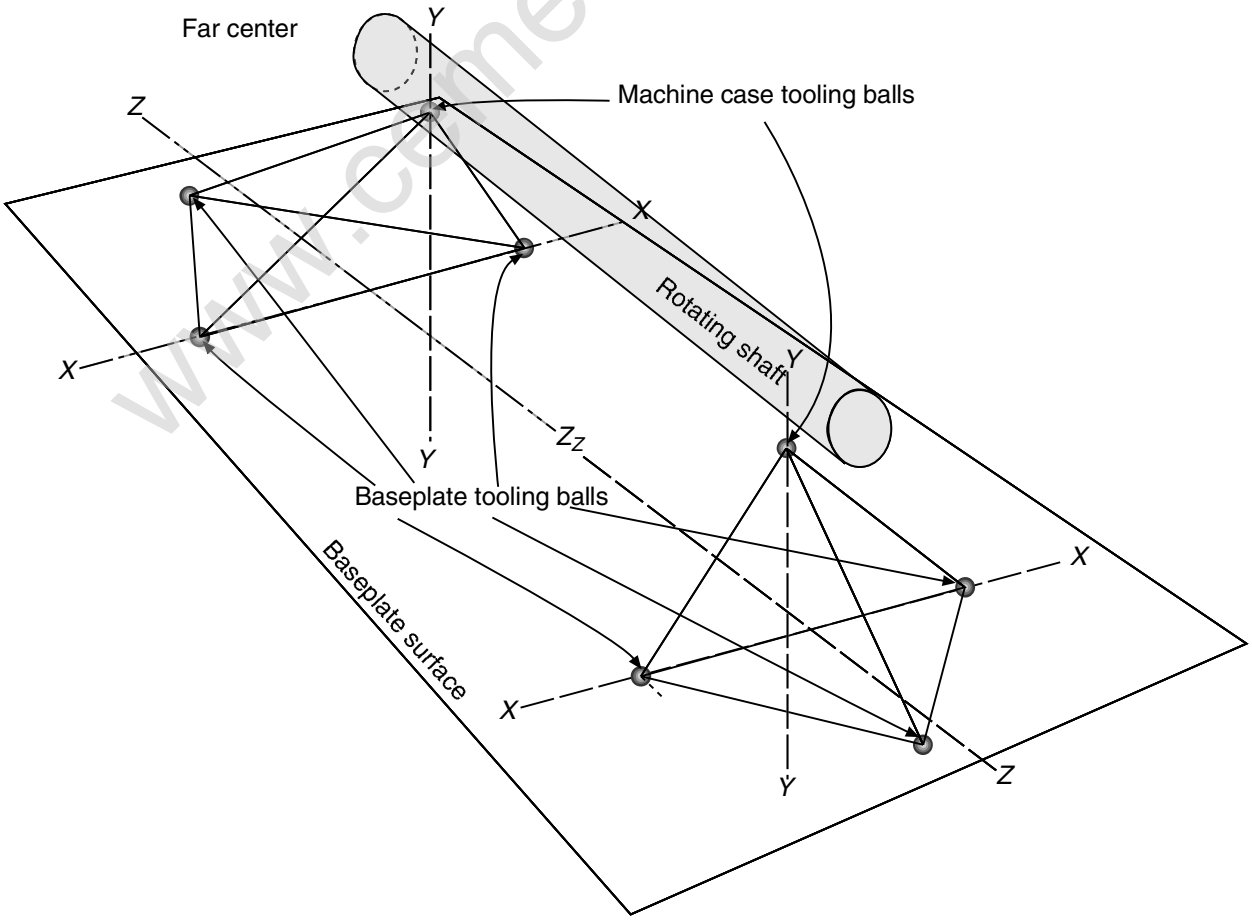
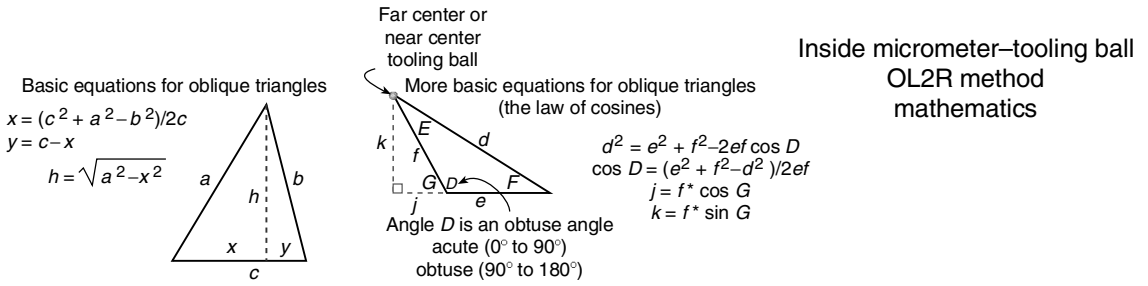
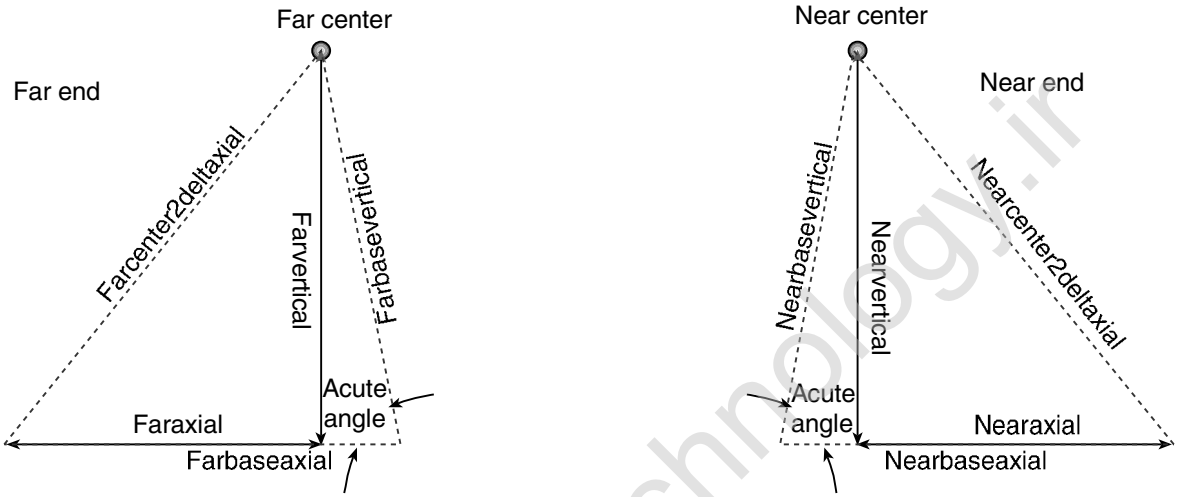


FIGURE 16.21 Pyramid tooling ball setup.

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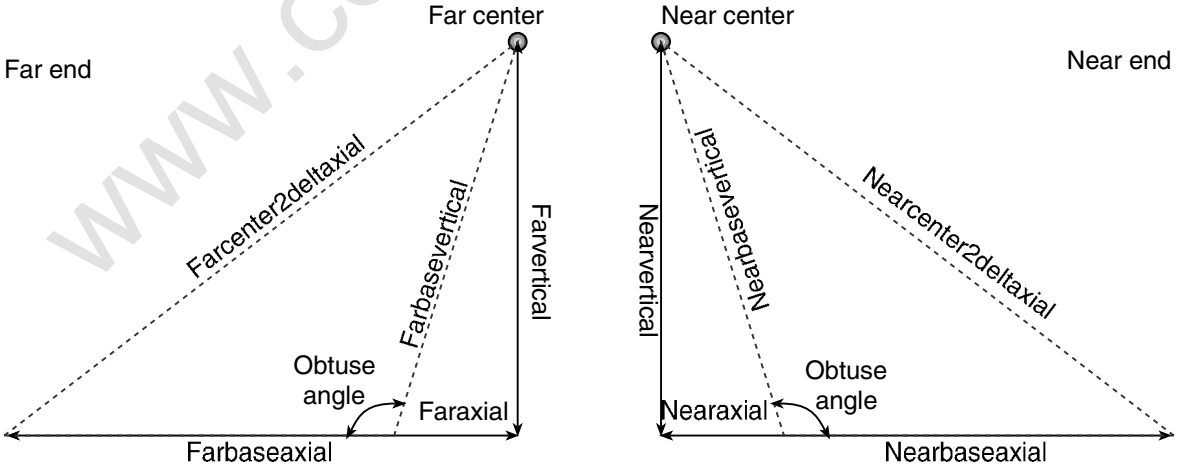


Far end math

Note: Use these equations if the angle formed at farbaseaxial and farbasevertical is an acute angle:
Faraxial=((farbaseaxial^2)+(farcenter2deltaxial^2)–(farbasevertical^2))/(2*farbaseaxial)
Farvertical=SQR((farcenter2deltaxial^2)–(faraxial^2))

Near end math

Note: Use these equations if the angle formed at nearbaseaxial and nearbasevertical is an acute angle:
Nearaxial=((nearbaseaxial^2)+(nearcenter2deltaxial^2)–(nearbasevertical^2))/(2*nearbaseaxial)
Nearvertical=SQR((nearcenter2deltaxial^2)–(nearaxial^2))



Far end math

Note: Use these equations if the angle formed at farbaseaxial and farbasevertical is an obtuse angle:
Faraxial=(farbasevertical*cosG)+farbaseaxial
Farvertical=farbasevertical*sinG

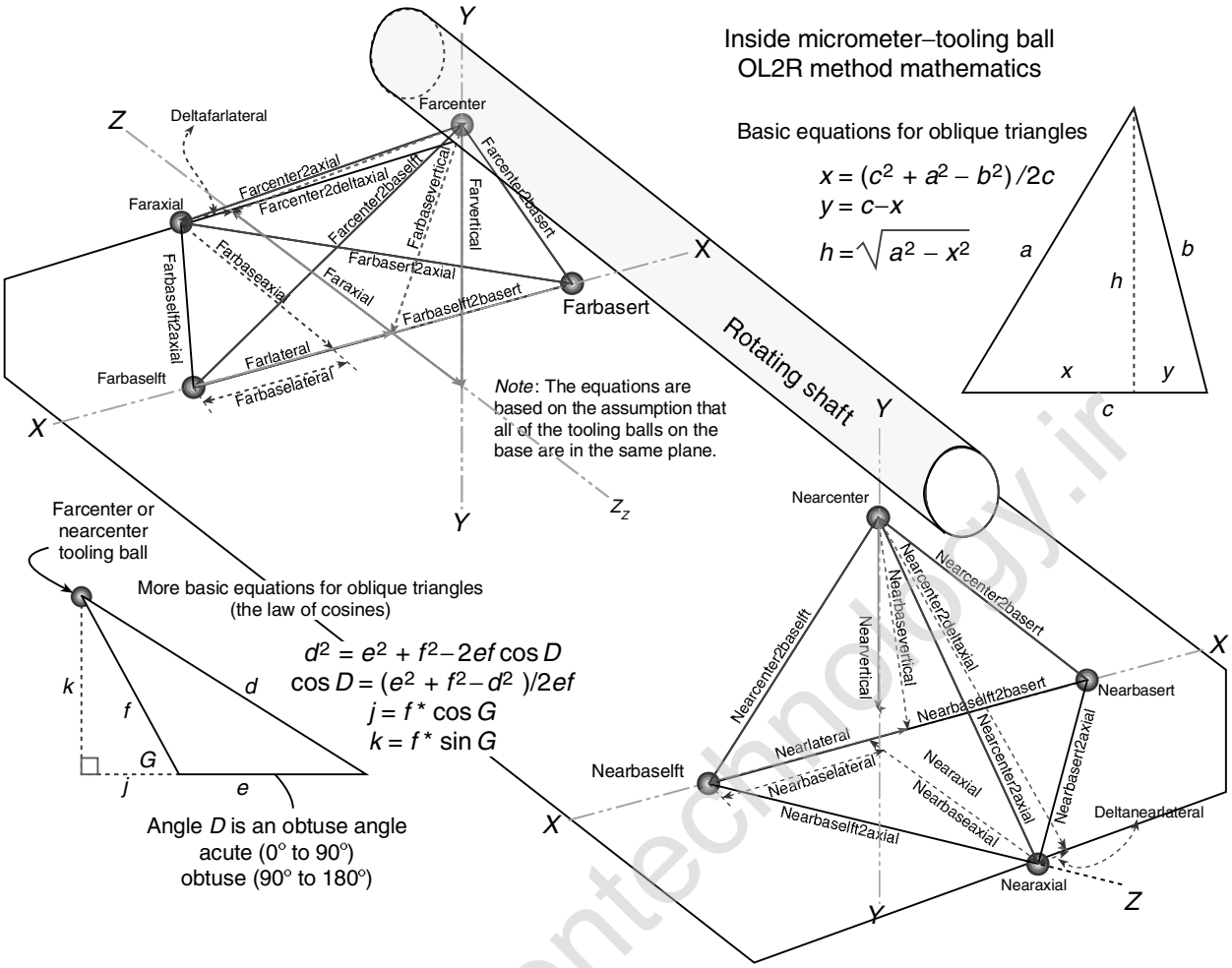
Near end math

Note: Use these equations if the angle formed at nearbaseaxial and nearbasevertical is an obtuse angle:
Nearaxial=(nearbasevertical*cosG)+nearbaseaxial
Nearvertical=nearbasevertical*sinG

FIGURE 16.22 Triangular tooling ball mathematics.

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Far end math

Farlateral=((farbaselft2basert^2)+(farcenter2baselft^2)−(farcenter2basert^2))/(2*farbaselft2basert)
Farbasevertical=SQR((farcenter2baselft^2)−(farlateral^2))
Farbaselateral=((farbaselft2basert^2)+(farbaselft2axial^2)−(farbasert2axial^2))/(2*farbaselft2basert)
Farbaseaxial=SQR((farbaselft2axial^2)−(farbaselateral^2))

Deltafarlateral=farlateral−farbaselateral or vice versa if farlateral < farbaselateral
Farcenter2deltaxial=SQR((farcenter2axial^2)−(deltafarlateral^2))
Note: Use these equations if the angle formed at farbaseaxial and farbasevertical is an acute angle:
Faraxial=((farbaseaxial^2)+(farcenter2deltaxial^2)−(farbasevertical^2))/(2*farbaseaxial)
Darvertical=SQR((farcenter2deltaxial^2)−(faraxial^2))
Note: Use these equations if the angle formed at farbaseaxial and farbasevertical is an obtuse angle:
Faraxial=(farbasevertical*cosG)+farbaseaxial
Farvertical=farbasevertical*sinG

Near end math

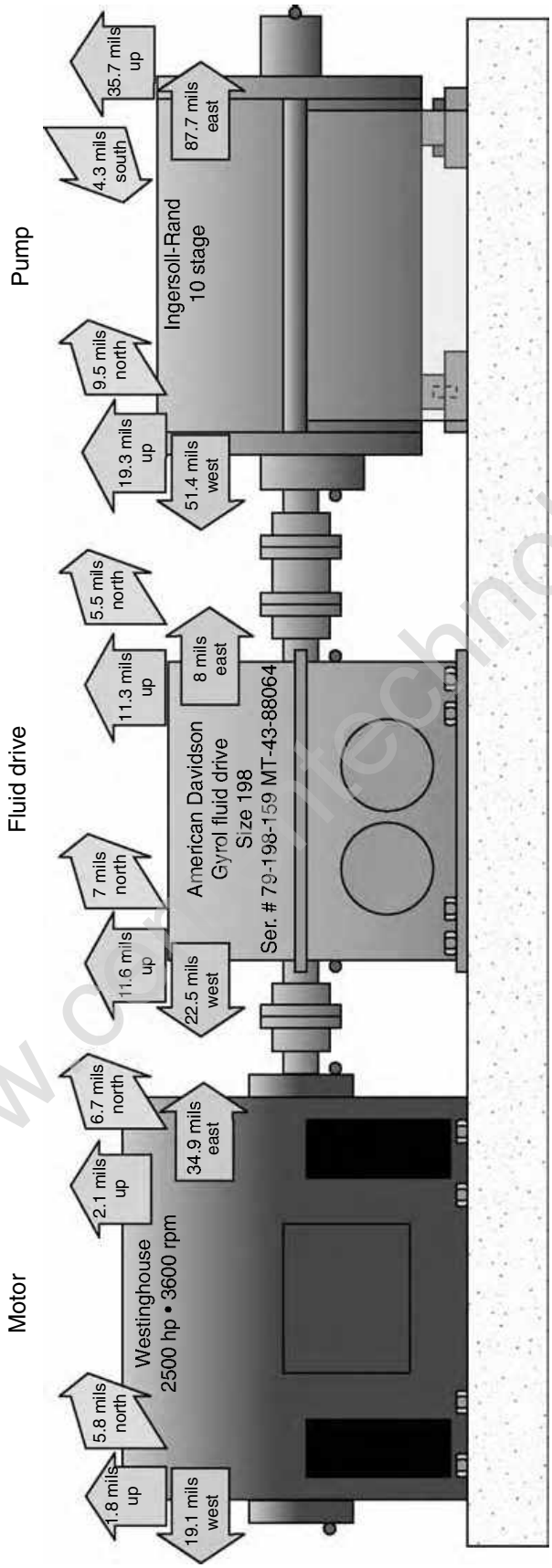
Nearlateral=((nearbaselft2basert^2)+(nearcenter2baselft^2)−(nearcenter2basert^2))/(2*nearbaselft2basert)
Nearbasevertical=SQR((nearcenter2baselft^2)−(nearlateral^2))
Nearbaselateral=((nearbaselft2basert^2)+(nearbaselft2axial^2)−(nearbasert2axial^2))/(2*nearbaselft2basert)
Nearbaseaxial=SQR((nearbaselft2axial^2)−(nearbaselateral^2))

Deltanearlateral=nearlateral−nearbaselateral or vice versa if nearlateral<nearbaselateral
Nearcenter2deltaxial=SQR((nearcenter2axial^2)−(deltanearlateral^2))
Note: Use these equations if the angle formed at nearbaseaxial and nearbasevertical is an acute angle:
Nearaxial=((nearbaseaxial^2)+(nearcenter2deltaxial^2)−(nearbasevertical^2))/(2*nearbaseaxial)
Nearvertical=SQR((nearcenter2deltaxial^2)−(nearaxial^2))
Note: Use these equations if the angle formed at nearbaseaxial and nearbasevertical is an obtuse angle:
Nearaxial=(nearbasevertical*cosG)+nearbaseaxial
Nearvertical=nearbasevertical*sinG

FIGURE 16.23 Pyramid tooling ball mathematics.

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Above movements were calculated by resolving all base mounted tooling balls into the *X–Z* plane

FIGURE 16.24 Observed movement on a motor-fluid drive-boiler feed water pump drive system from OL2R conditions.

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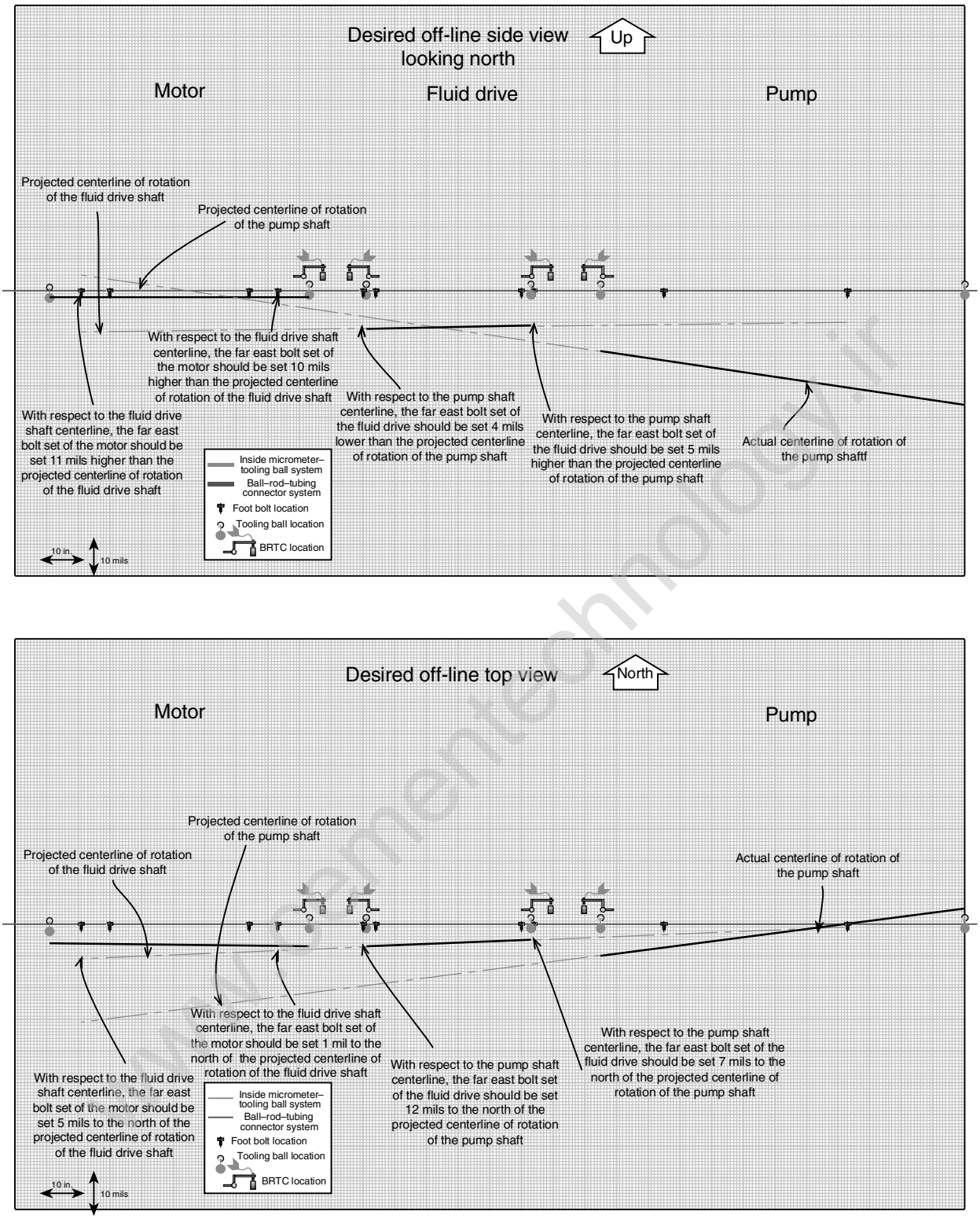


FIGURE 16.25 Desired off-line shaft position alignment models for the side and top views for the motor-fluid drive-boiler feed water pump drive system.

pinch the machine case to the baseplate. The outboard bolts are sleeved and do not pinch the case to the baseplate but allow the case to slide preventing distortion from occurring.

Not only is the pump case expanding, but so too is the shaft expanding. Notice that the machine cases (and probably the shafts) are moving toward each other from OL2R

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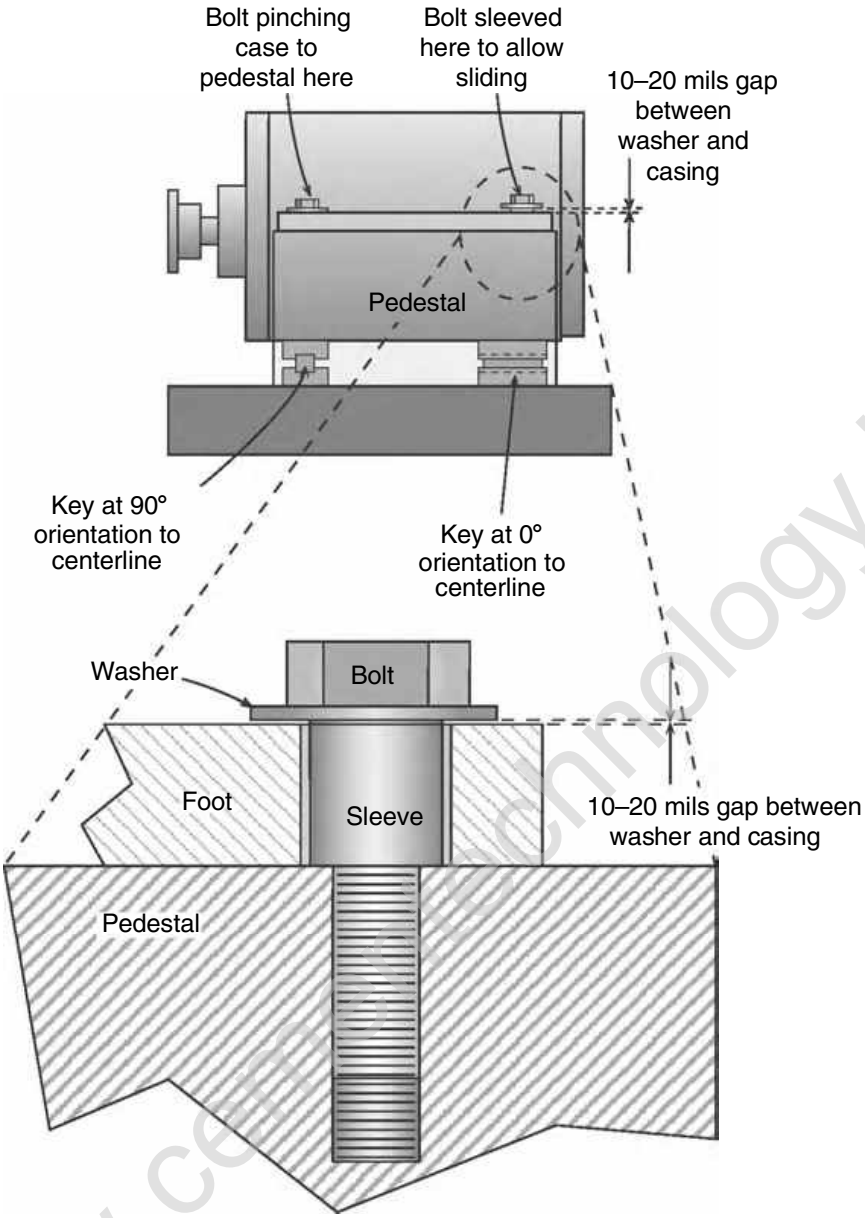


FIGURE 16.26 Transverse keys and sleeved bolt to allow for axial expansion.

conditions. Some flexible coupling designs allow axial movement of the shafts without transferring axial forces during the movement (or expansion). On the drive system shown in Figure 16.24, thankfully gear couplings were used between the motor, fluid drive, and pump. If another type of coupling design was employed that was not forgiving in axial movement, the thrust bearing loads would increase.

Based on how the measurements were taken, it is not known in this particular drive system if each of the machine cases expanded symmetrically. Since the measurements were taken on tooling balls located directly under the shafts, the machine cases could have bowed outward near the center of the machines as shown in Figure 16.27. If indeed this "bell-shaped" distortion occurs, then any OL2R technique that attaches devices near the bearings could give a false indication of what is happening to the shafts. Later on, we will examine several methods where devices are attached near the bearing so that I thought it would be prudent to mention this just as a precautionary note.

As mentioned previously in this chapter, I would again like to make it perfectly clear that we have not collected enough OL2R data on rotating machinery to conclusively state what

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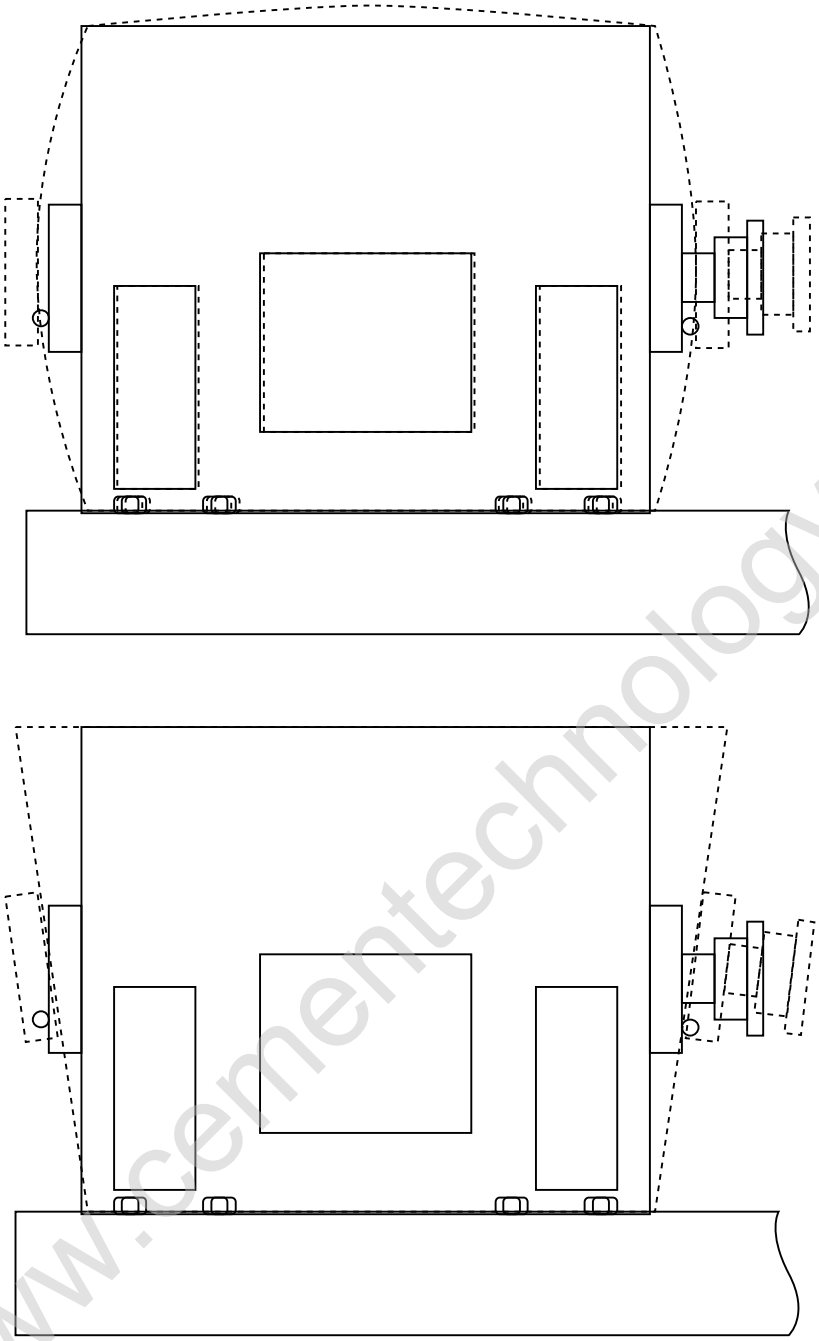


FIGURE 16.27 Possible thermal distortion shapes.

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happens to the majority of machinery in existence. These data you see are just scratching the surface of the behavior of machinery as they transit from off-line to operating conditions. We have much to learn about this phenomenon.

16.9 PROXIMITY PROBES WITH WATER-COOLED STANDS

Another technique that falls into the category of movement of a machine case centerline with respect to its baseplate is performed using water-cooled stand attached to the foundation and proximity probes held by the pipe stand to observe targets at the inboard and outboard ends of every machine in the drive system. This technique was conceived and popularized by

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Charlie Jackson and has been successfully employed on many rotating machinery drive systems.

The proximity probes are attached via a bracket to a water-cooled pipe stand, which is firmly anchored to the machinery foundation near each bearing. To maintain a stable reference point, water should be circulated through the pipe stand or the pipe should be insulated and filled with a water–glycol or antifreeze solution to prevent as little dimensional change as possible to the pipe stand itself from radiant heat emitted from the machinery. The probes are mounted on a bracket attached to the pipe stand and positioned to monitor a metal block (usually steel) affixed to each end of every machine case in the drive train. OL2R movement can be monitored in the horizontal, vertical, and axial directions. The probes could also be positioned to monitor the movement of the shaft directly since it is really the position of the shaft that is trying to be determined from OL2R conditions. Figure 16.28 shows the basic arrangement for water-cooled stands, proximity probes, and targets. Figure 16.29 through Figure 16.34 show some installations on rotating machinery.

Key considerations for capturing good readings:

- Insure that the pipe stands are rigidly attached to a stable reference point on the frame or foundation and that they maintain a constant temperature through the OL2R measurement process.
- Insure the target surfaces are at a precise 90° angle to the probes.
- The targets should be attached as close as possible to the centerline of rotation of the shaft to insure that the probes see shaft movement, not casing expansion.
- Insure that the probe tips are far enough apart to prevent any cross-field effects from one probe to another that will affect accurate gap measurements.
- Probes should always be statically calibrated to the same type of material that is observed since the gap versus voltage characteristics are different from one material to another.
- If the direction of machinery movement is not known when the probes are initially gapped, some adjustments may be necessary after the first attempt in case the target (or shaft) is moving too close or too far away from the probe tip to keep the probe within its linear range.
- Standard probes (200 mV/mil sensitivity) are usually good for gap changes near 80 mils, and some manufacturers can supply special probes able to measure up to a half inch of gap change.
- LVDT sensors could also be used instead of proximity probes.

Advantages:

- Extreme accuracy possible with a good setup
- Capable of monitoring motion in all three directions (vertical, lateral, and axial)
- Continuous monitoring possible

Disadvantages:

- Pipe stands must be mounted at both ends of each machine case
- Cannot measure any change in the machinery baseplate or foundation itself
- Somewhat expensive since pipe stands have to be fabricated; probes, cables, proximitors, readout devices, and power supplies have to be purchased
- Potential for inaccurate measurement shaft positional changes when monitoring points away from the centerline of rotation

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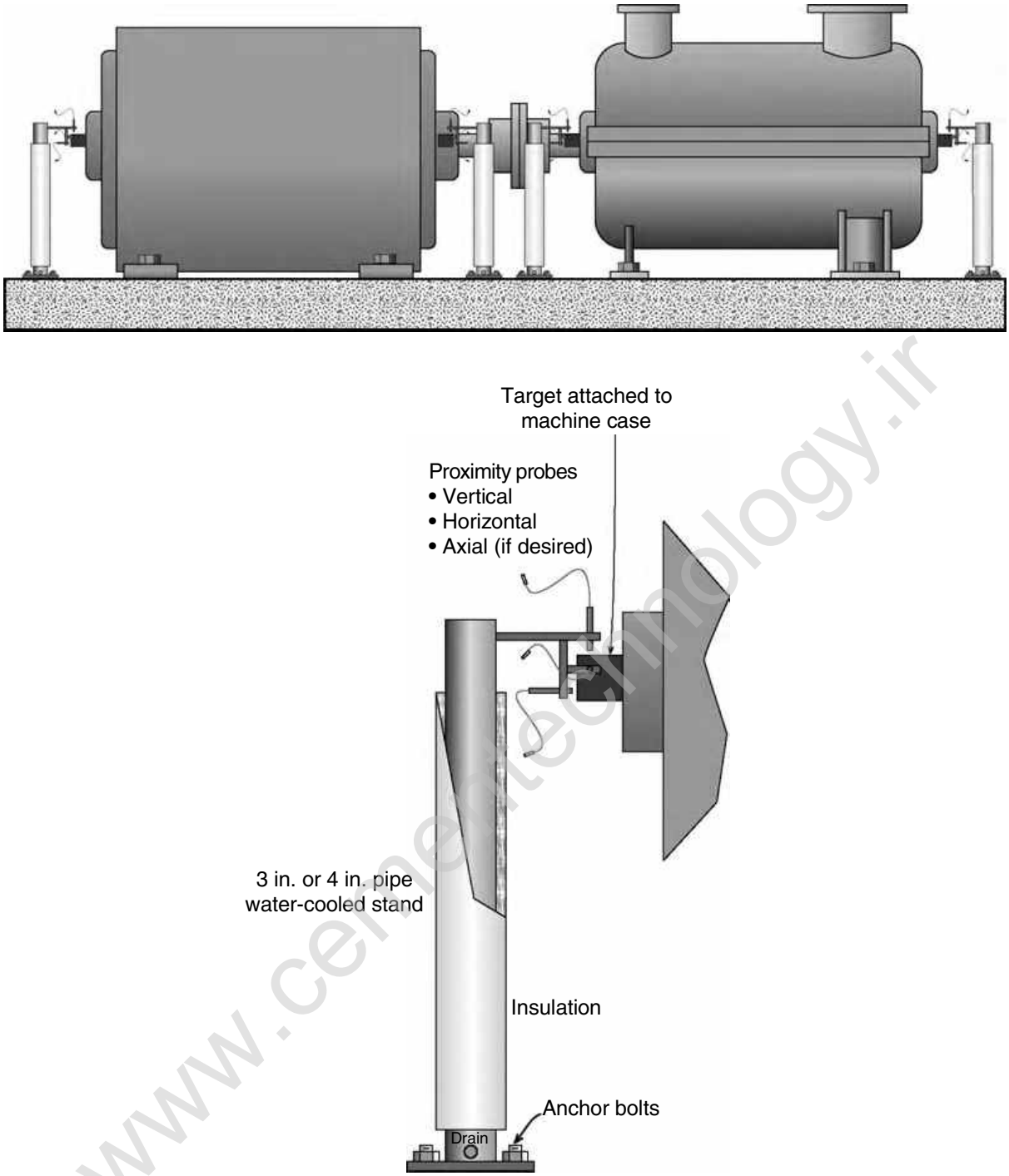


FIGURE 16.28 Basic setup for water-cooled stands, proximity probes, and targets.

- Potential for inaccurate measurement when monitoring the shafts directly particularly if a considerable amount of movement occurs (if you take a reading on a curved surface)

16.10 OPTICAL ALIGNMENT EQUIPMENT

This method falls into the category of observing movement of a machine case from a remote observation point. Optical tooling levels and jig transits are the most versatile measurement systems available to determine rotating equipment movement. Figure 16.35 and Figure 16.36 show the two most widely used optical instruments for machinery alignment. This section will deal specifically with their ability to measure OL2R movement but in no way will begin to

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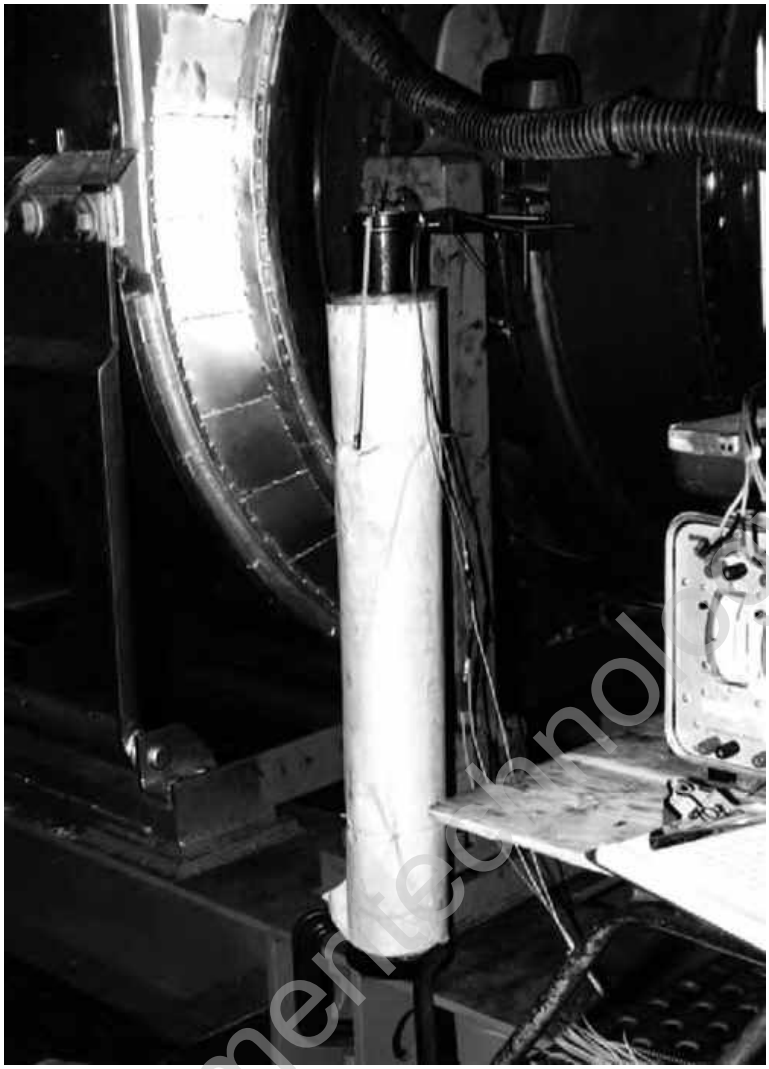


FIGURE 16.29 Water-filled pipe stand observing vertical, lateral, and axial positions at exhaust end of gas-power turbine.

explain their full potential for many other uses such as leveling foundations, squaring frames, roll parallelism, and a plethora of other tasks involved in level, squareness, flatness, vertical straightness, etc.

A detail of a 3 in. scale target is shown in Figure 16.37. Optical scale targets can be purchased in a variety of standard lengths (3, 5, 10, 20, and 40 in.) and metric scales are available. The scale pattern is painted on invar bars to minimize thermal expansion or contraction of the scale target itself. The scale targets are held in position with magnetic base holders as shown in Figure 16.38 and Figure 16.39.

There are generally four sets of paired line sighting marks on the scales for centering of the crosshairs when viewing through the scope as shown in Figure 16.37. An optical micrometer, as shown in Figure 16.40, is attached to the end of the telescope barrel and can be positioned in either horizontal or vertical direction. The micrometer adjustment wheel is used to align the crosshairs between the paired lines on the targets. When the micrometer wheel is rotated, the crosshair appears to move up or down along the scale target (or side to side depending on the position of the micrometer). Once the crosshair is lined up between a set of paired lines, a reading is taken based on where the crosshair is sighted on the scale and the position of the

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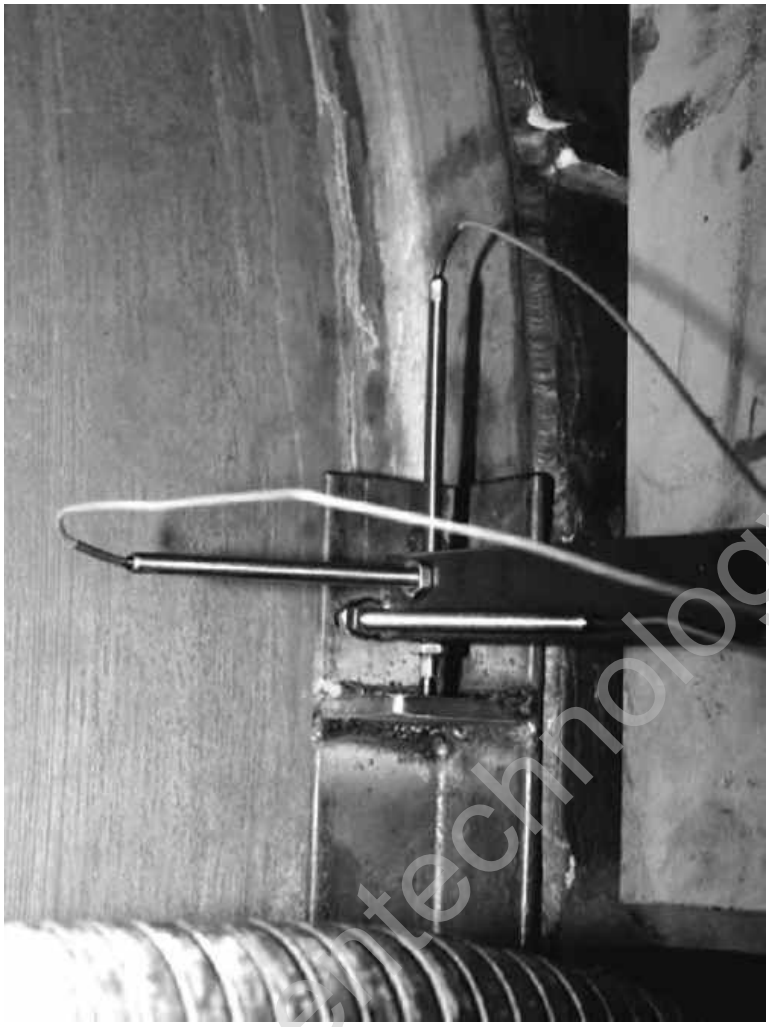


FIGURE 16.30 Close-up of probes shown in Figure 16.26.

optical micrometer. The inch and tenths of an inch reading is visually taken by observing the scale target number where the crosshair aligns between a paired line set, and the hundredths and thousandths of an inch reading is taken on the micrometer drum as shown in Figure 16.41.

The extreme accuracy (one part in 200,000 or 0.001 in. at a distance of 200 in.) of the optical instrument is obtained by accurately leveling the scope using the split coincidence level mounted on the telescope barrel as shown in Figure 16.42.

Before using any optical instrument, it is highly recommended that a Peg Test be performed. The Peg Test is a check on the accuracy of the levelness of the instrument. Figure 16.43 shows how to perform the Peg Test.

Figure 16.44 and Figure 16.45 show the basic procedure on how to properly level the instrument. If there is any change in the split coincidence level bubble gap during the final check, go back and perform this level of adjustment again. This might take 0.5 to 1 h to get this right, but it is time well spent. It is also wise to walk away from the scope for about 30 min to determine if the location of the instrument is stable and to allow sometime for your eyes to uncross. If the split coincident bubble has shifted during your absence, start looking for problems with the stand or what it is sitting on. Correct the problems and relevel the scope.

I cannot overemphasize the delicacy of this operation and this equipment. There is no way for people in a big hurry with little patience. If you take your time and are careful and attentive when obtaining your readings, the accuracy of this equipment will astonish you.

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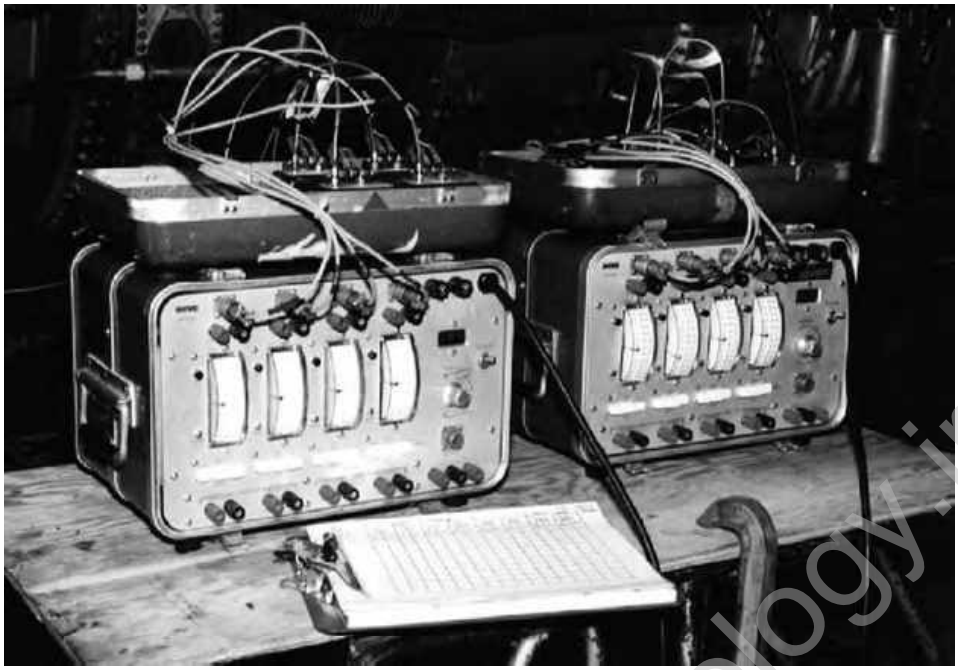


FIGURE 16.31 Power supply and signal conditioners for proximity probes.

16.11 OPTICAL PARALLAX

As opposed to binoculars, 35 mm cameras, and microscopes that have one focusing adjustment, the optical scope has two focusing knobs. There is one knob used for obtaining a clear, sharp image of an object (e.g., the scale target) and another adjustment knob that is used to focus the crosshairs (reticle pattern). Since your eye can also change focus, adjust both these knobs so that your eye is relaxed when the object image and the superimposed crosshair image are focused on a target.

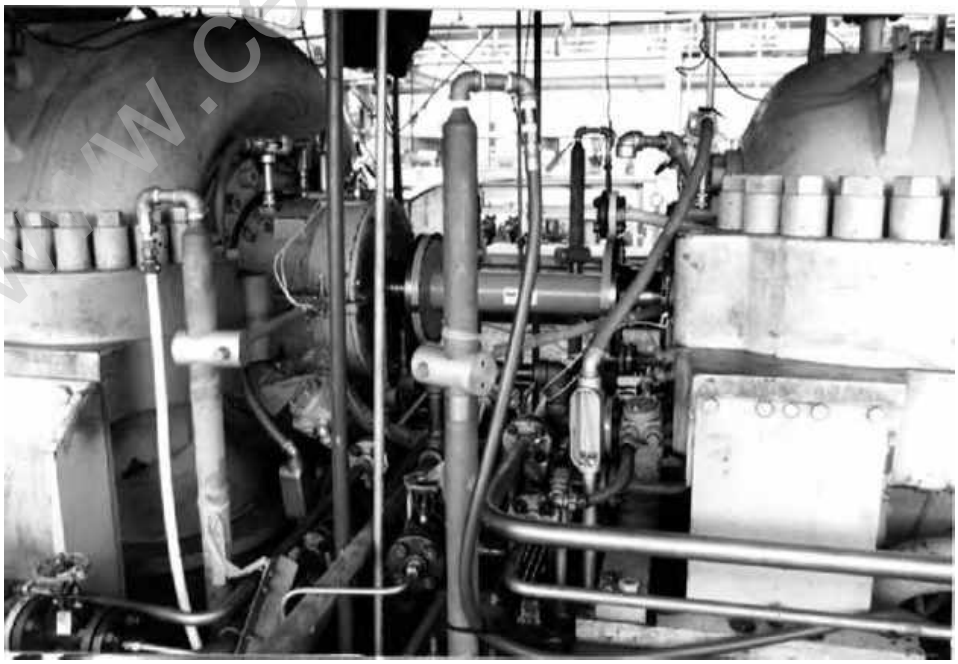


FIGURE 16.32 Water-cooled stands with proximity probes. (Courtesy of Charlie Jackson, Texas City, TX. With permission.)

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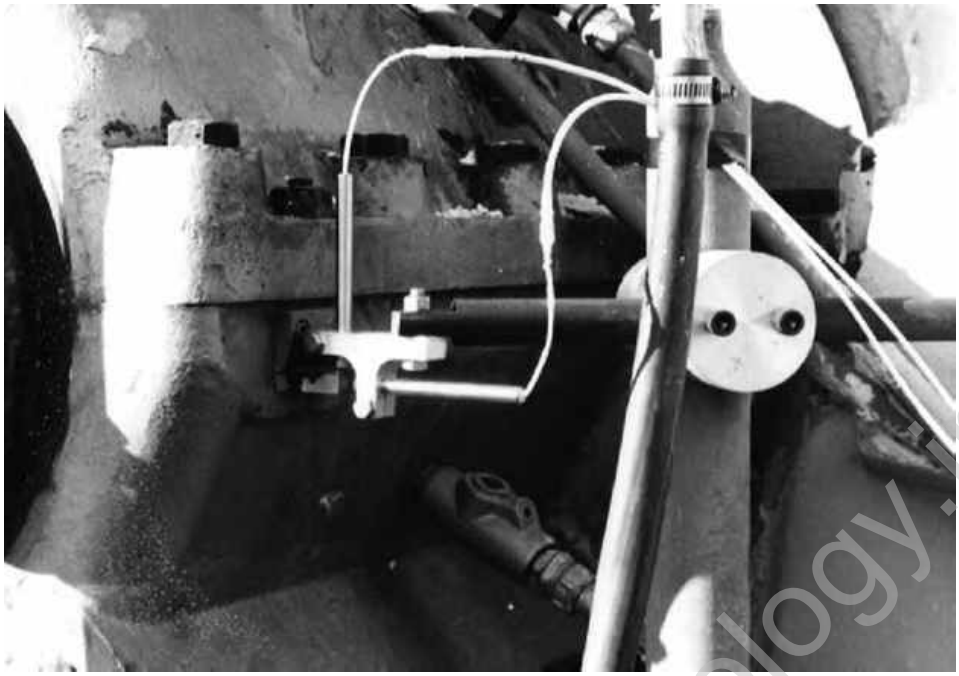


FIGURE 16.33 Close-up of water-cooled stand, proximity probes with holding bar, and target attached to machine case near its centerline of rotation. (Courtesy of Charlie Jackson, Texas City, TX. With permission.)

Adjusting the focusing knobs:

1. With your eye relaxed, aim at a plain white object at the same distance away as your scale target and adjust the eyepiece until the crosshair image is sharp.
2. Aim at a scale target and adjust the focus of the telescope.



FIGURE 16.34 Water-cooled stands with proximity probes observing position of coupling hubs. (Courtesy of Charlie Jackson, Texas City, TX. With permission.)

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FIGURE 16.35 Optical tooling level (right) and jig transit (left).



FIGURE 16.36 Jig transit. (Courtesy of Brunson Instrument Co., Kansas City, MO. With permission.)

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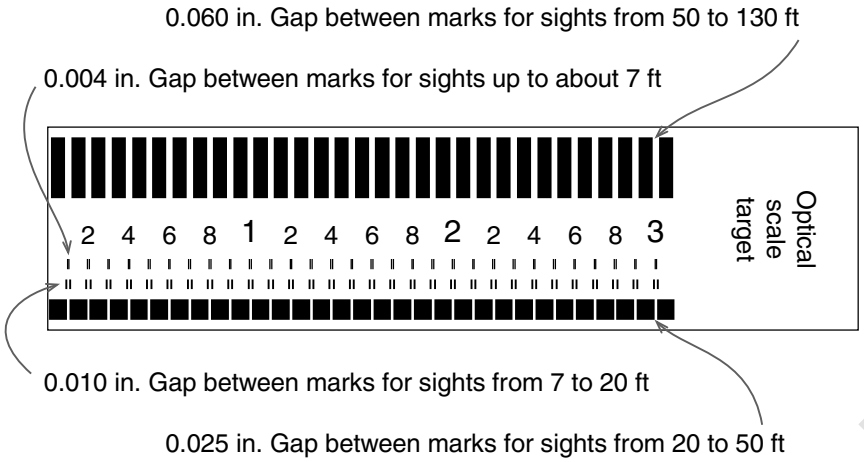


FIGURE 16.37 Three inch optical scale target.



FIGURE 16.38 Scale targets mounted on gearbox and fan bearing near their centerline of rotation.

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FIGURE 16.39 Scale targets mounted at the inboard and outboard bearings of a feed water pump.

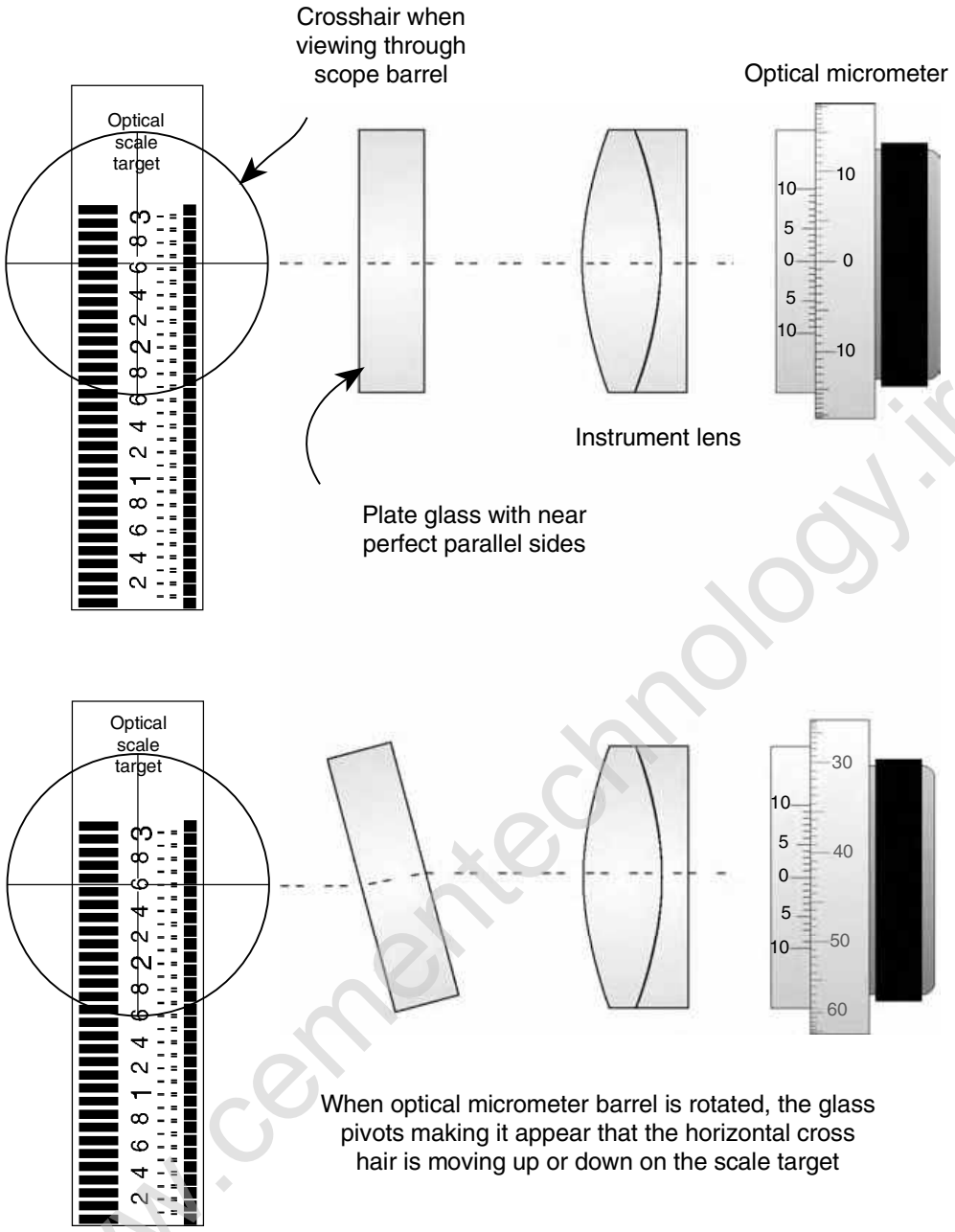
3. Move your eye slightly sideways and then up and down to see if there is an apparent motion between the crosshairs and the target you are sighting. If so, defocus the telescope and adjust the eyepiece to refocus the object. Continue alternately adjusting the telescope focus and the eyepiece to eliminate this apparent motion.



FIGURE 16.40 An optical micrometer. (Courtesy of Brunson Instrument Co., Kansas City, MO. With permission.)

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Notice in the upper drawing that when the optical micrometer is in the zero position, the horizontal crosshair is between 2.6 and 2.7 on the scale target but the crosshair is not exactly aligned with any of the marks. By rotating the micrometer drum, the horizontal crosshair is aligned at the 2.6 mark on the scale target. The inch and tenths of an inch reading is obtained off the scale target, the hundredths and thousandths of an inch reading is obtained off the micrometer drum position.

FIGURE 16.41 Principle of an optical micrometer.

16.12 USING OPTICAL TOOLING FOR MEASURING MACHINERY MOVEMENT

As described earlier, this OL2R method attempts to determine how points at the inboard and outboard ends of each machine are moving up or down and side to side. Selecting where the measurement points are on each end of each piece of equipment is very critical. We are trying

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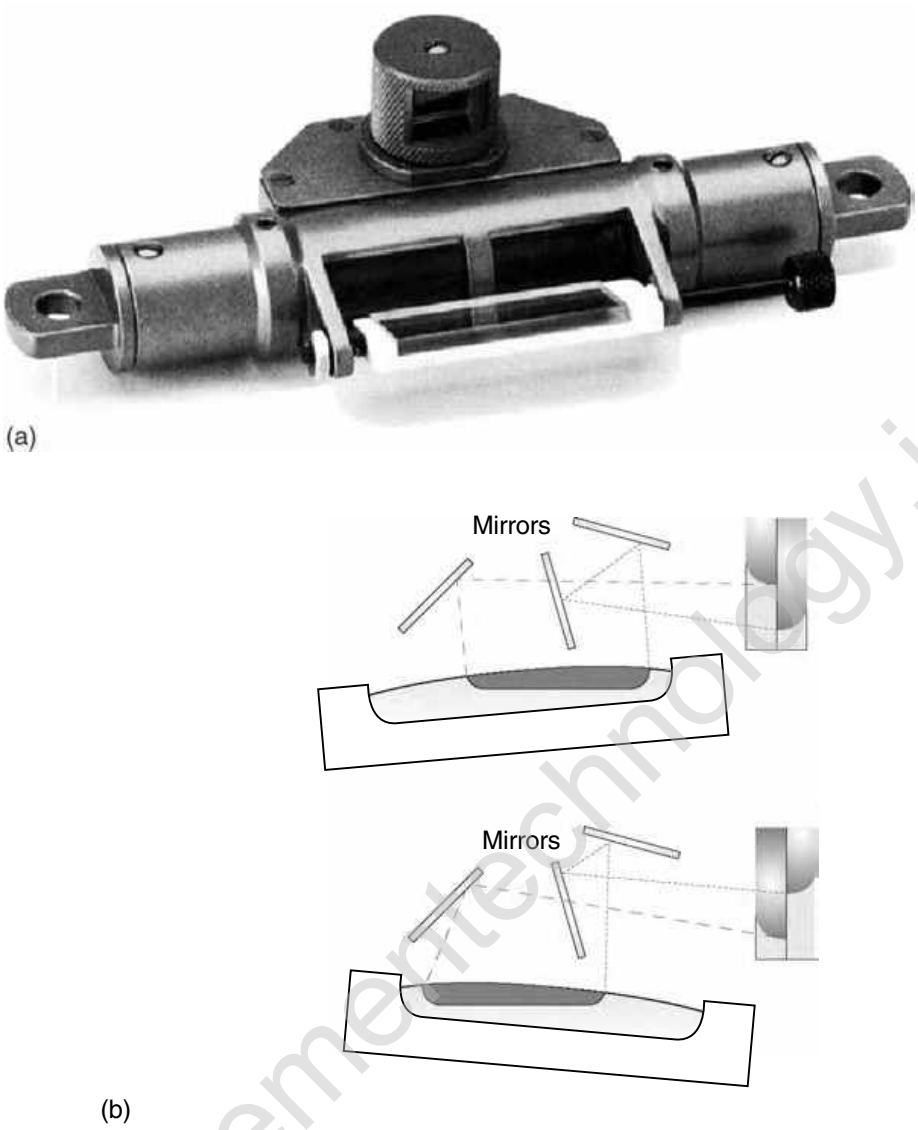


FIGURE 16.42 Principle of the coincidence level. (Courtesy of Brunson Instrument Co., Kansas City, MO. With permission.)

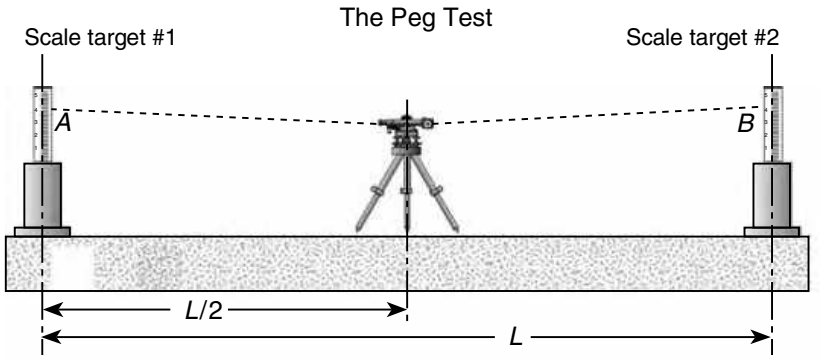
to determine where the centerlines of rotation are moving to and therefore must measure points as close as possible to the centerline of rotation of the shafts.

Either an optical tilting level or a jig transit can be used to measure the up and down movement of the scale targets as shown in Figure 16.46. An optical tilting level will typically not work as well as a jig transit to measure side-to-side movement since the scope barrel is not capable of sweeping through a large enough vertical arc to capture the readings. Two people are required to measure side-to-side movement, the scale target holding person and the observer. Measuring side-to-side movement is little more tedious since the scale targets have to be held by hand and there needs to be good communication between the scale target holding person and the observer to insure that the scale targets are at a precise right angle to the observer's line of sight. Figure 16.47 shows the procedure for taking side-to-side measurements and Figure 16.48 demonstrates how to insure that the scale target is at a precise 90° angle to the line of sight. Figure 16.49 and Figure 16.50 show a scale target, held in position against a stationary point on the machine cases directly above the centerline of rotation for lateral measurements.

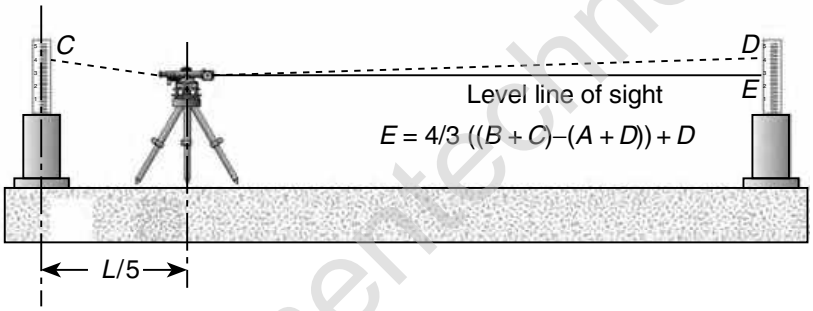
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Before using any optical instrument, it is recommended that the Peg Test be performed to insure measurement accuracy. At 40 ft, the accuracy of the scope is plus or minus 0.0024 in.



1. Set two scales apart by distance *L* (usually 40 ft) on stable platforms. Position the optical telescope or transit exactly half way between both scales. Accurately level the instrument using the split coincident level.
2. Alternately take four readings on scale target #1 (reading *A*) and scale target #2 (reading *B*). Record and average these readings.



3. Move the scope to the 1/5 *L* position, level the scope, and alternately take four readings on scale no. 1 (reading *C*) and scale no. 2 (reading *D*). Record and average these readings.
4. If the instrument is calibrated, *A* minus *B* should equal *C* minus *D* (at 32 ft , this should be no more than 0.002 in.).

If the error is greater than that, adjust the split coincidence levels as follows:

A. Set the optical micrometer drum to the hundredths and thousandths value of *E*. For example, if *E* = 4.656, set the micrometer drum to 0.056.

B. Using the tilting screw, tilt the scope barrel to align the horizontal crosshair to the inch and tenths of an inch mark on scale target #2. For example, if *E* = 4.656, align the horizontal crosshair to the paired lines at 4.6. At this point, the split coincident level will not be coincident.

C. Adjust the nuts holding the split coincident level to the scope barrel to bring the bubble halves into coincidence.

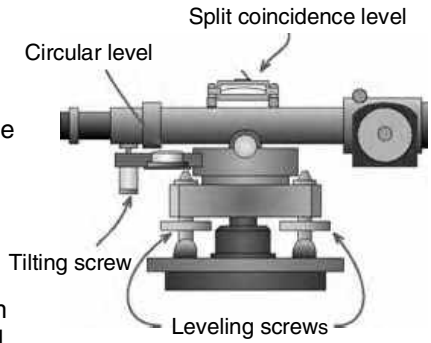
D. Perform steps 1 through 4 above to verify that the adjustment worked. Should this not be the case, the coincident level calibration adjustment nuts can be adjusted to position the leveled line of sight to be set at reading *E*.

FIGURE 16.43 Coincidence level calibration test.

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How to level optical tilting levels and jig transits



1. Set the instrument stand at the desired sighting location, attach the alignment scope to the tripod or instrument stand, and level the stand using the "rough" circular bubble level on the tripod (if there is one on the tripod). Insure that the stand is steady and away from heat sources, vibrating floors, and curious people who may want to use the scope to see sunspots.
2. Rotate the scope barrel to line up with two of the four leveling screws and adjust these two leveling screws to roughly center the split coincidence level bubble in the same tilt plane as the two screws that are adjusted as shown.
3. Rotate the scope barrel 90° to line up with the other two leveling screws to completely center the bubble in the circular level as shown.
4. If the circular level is still not centered, repeat steps 2 and 3.

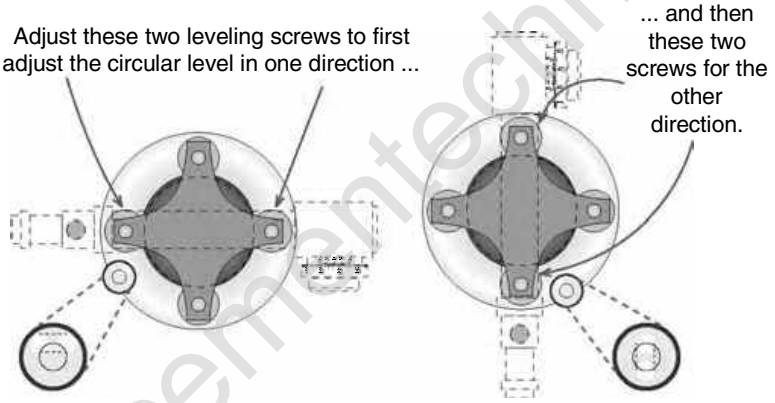


FIGURE 16.44 How to level a tilting level or jig transit, parts 1 through 4.

16.13 ESTABLISHING REFERENCE PLANES

Since there is a good possibility that the off-line measurements will be taken at a different time than the running measurements, it is suggested that vertical and lateral reference positions be established. Understand that if you take a set of elevations on scale targets attached to the machinery bearings as shown in Figure 16.46 when the equipment is off-line, then dismantle the optical instrument and tripod, and if you go to set the instrument and tripod backup at a later time, the question now becomes "what was the original elevation of the instrument when the first set of vertical measurements were taken?"

One of the primary considerations when using any method where you are observing points on a machine case with respect to a remote observation point is to establish stable, nonmoving reference "planes" in the vertical and horizontal directions that enable you to reestablish or "buck in" to that same reference plane for comparison of your off-line and your running measurements. This sounds easy but realistically, this is very difficult to accomplish. Remember you are trying to measure distances as small as 1 mil (0.001 in.).

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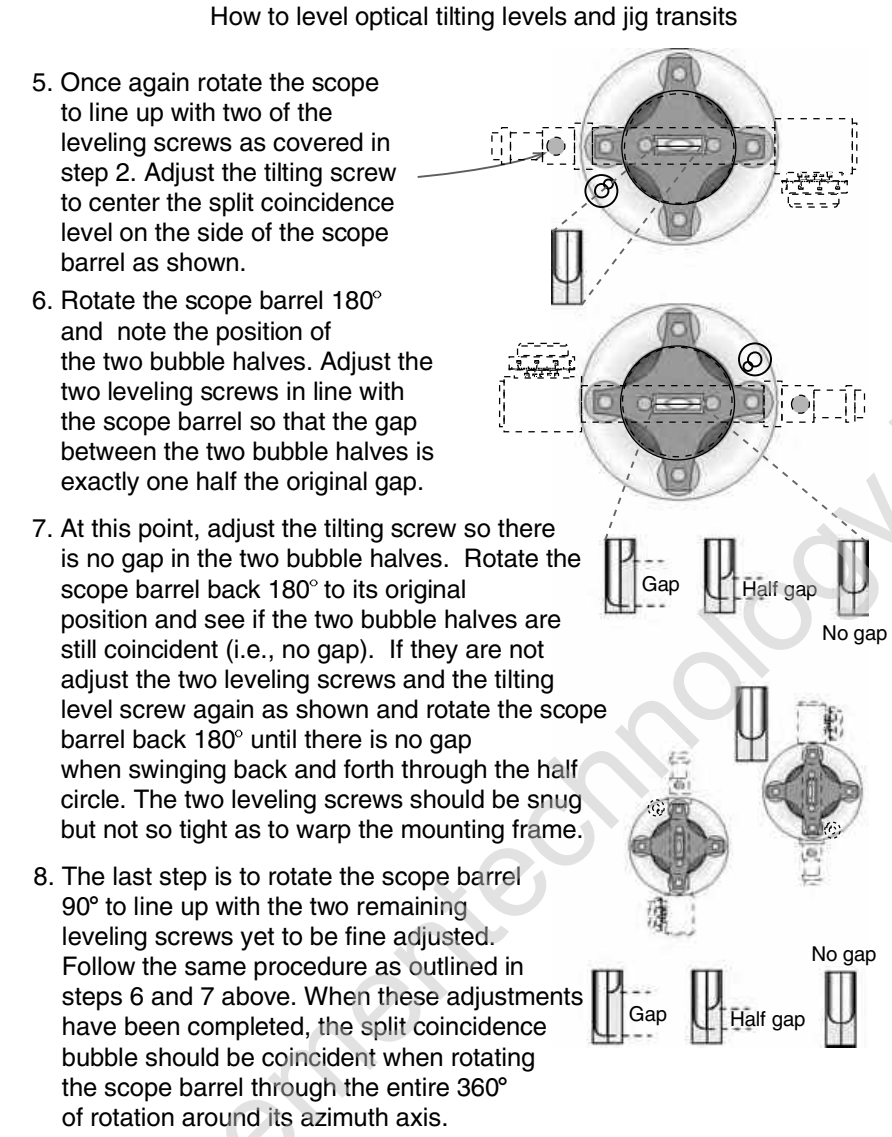


FIGURE 16.45 How to level a tilting level or jig transit, parts 5 through 8.

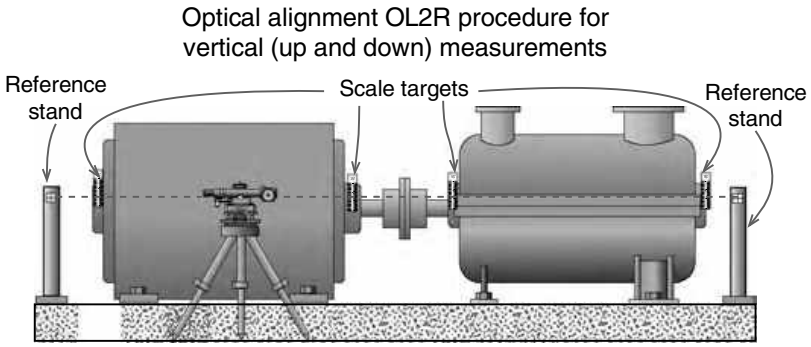
The type of reference plane that you establish is somewhat dependent on the period of time between the off-line measurements and the running measurements. If you plan on taking the off-line and running measurements within a 2-h time period, then adhesive-backed targets could be placed on walls or building columns or any fairly stable object. Examples of this are shown in Figure 16.51 through Figure 16.54. If however the time period between off-line and running measurements will be over 2 h or perhaps even days later, you should establish a more stable reference position since building walls or structural steel could very well change shape and position over a long period of time.

For vertical elevation measurements, it is suggested that one of the two methods be employed. One way is to fabricate water-cooled pipe stands and attach adhesive-backed targets to them as shown in Figure 16.55. It is recommended that at least two reference points be established, usually one at each end of the drive train. Another way is to use an invar extension rod contacting a tooling ball, which is rigidly attached to the concrete foundation or baseplate or floor as shown in Figure 16.56.

For lateral or axial measurements, adhesive-backed targets can be placed on concrete foundations or floors, or better yet, permanently anchored reference targets imbedded in the concrete as shown in Figure 16.57.

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1. Check calibration of the instrument (see Peg Test).
2. Select suitable scale positions at the inboard and outboard ends of each piece of machinery in the drive train. The "platforms" that the scale targets will be sitting on should be stable and slightly below the centerline of rotation and usually near the bearings.You can use 2 in. × 2 in. pieces of angle bearing housing.Try to insure that the surface that the scale targets will sit on is relatively level and flat. It is also advisable to install reference stands at each end of the drive train and affix a "stick-um" crosshair target to the reference stand.You can make these stands out of 3 in. or 4 in. pipe, fill them with a water–glycol or antifreeze solution, insulate the pipe, bolt or clamp them to the frame or floor, and monitor the water temperature to insure thermal stability.
3. Set the optical instrument and stand at some remote reference point away from the drive train where a stable point in space can be established but close enough to maintain the maximum possible accuracy of the readings.
4. Accurately level the instrument and take a set of readings at each target scale mounted on the machinery when it is "off-line" (i.e., not running) occasionally checking back to the reference targets at each end of the drive train to insure that you are maintaining the same vertical elevation (i.e., shooting through the same horizontal plane).
5. Run the machinery at normal conditions and allow the equipment to stabilize its position (this can take hours or even days).
6. Check the level accuracy and take a similar set of readings at each target scale occasionally checking back to the reference targets at each end of the drive train to insure that you are maintaining the same vertical elevation.
7. Compare the off-line set of readings to the running set of readings to determine the amount and direction of the movement of each scale.

FIGURE 16.46 How to take optical vertical OL2R measurements.

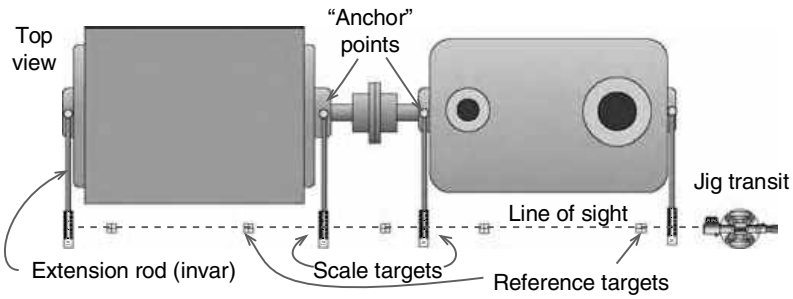
Key considerations for capturing good readings:

- Provide stable platforms for the optical scale targets
- Establish several (minimum of two, three suggested) vertical, lateral, and axial reference positions to ''buck'' back into
- Scale target should be located as close as possible to the bearings since we are trying to determine where the shafts are going (if the bearing moves, the shaft is sure to move with it)
- Magnetic base holders should be used to hold the scale targets insuring a stable target position
- Clamp on circular level bubble sets should be used on the scale target to insure scale targets are in a pure vertical position

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Optical alignment OL2R procedure for
horizontal (side-to-side) measurements



1. Check calibration of the jig transit (see Peg Test). Two people are required to do this procedure—the scale target holding person and the observer.
2. Select suitable scale "anchor" points at the inboard and outboard ends of each piece of machinery in the drive train. The points or "anchors" that the scale targets (and probably extension rods) will be touching should be stable and directly above or below the centerline of rotation and usually near or at the bearings. You can have tooling balls firmly affixed to the machine care or bearing housing as the "anchor" points to hold the scale target against for reference.
3. Set the jig transit and stand at some position along one side of the drive train where a stable point in space can be established insuring that measurements can be taken at each bearing location when the scale target or extension rod is placed at each "anchor" point at every bearing location. Orient the optical micrometer on the scope barrel to allow variation in the position of the vertical crosshair when the micrometer barrel is rotated.
4. Accurately level the instrument, then loosen the vertical sweep axis screw allowing the scope to tilt up and down. Affix several (at least two) adhesive crosshair targets to the foundation or floor along the full length of the drive train establishing a vertical reference line or plane. Take a set of readings at each bearing location by holding the scale target or extension rod when placed at each "anchor" point when the machinery is "off-line" (see "waving scales") occasionally checking back to the adhesive crosshair targets attached to the foundation or floor to insure that you are maintaining the same horizontal position (i.e., keeping in the same vertical reference plane).
5. Run the machinery at normal conditions and allow the equipment to stabilize its position (this can take hours or even days).
6. Check the level of accuracy and take a similar set of readings at each bearing location occasionally checking back to the adhesive crosshair targets attached to the foundation or floor to insure that you are maintaining the same horizontal position.
7. Compare the off-line set of readings to the running set of readings to determine the amount and direction of the movement of each scale target and bearing location.

FIGURE 16.47 How to take optical horizontal (side-to-side) OL2R measurements.

- If possible, try to keep the scale targets in place from OL2R conditions.
- Keep the scale target holding platforms clean.
- Move slowly when working around the tilting level or jig transit and stand (if your foot or arm bumped the stand, it is probably not in the level or in the same vertical or horizontal plane any more).
- Readings should be taken at night when equipment is located outdoors to prevent thermal instability of the tripod or stand when the sun heats or cools the stand.

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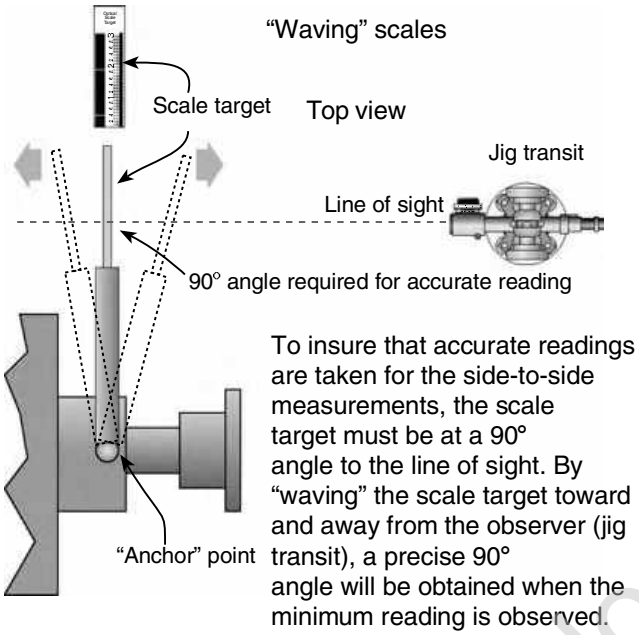


FIGURE 16.48 How to take accurate optical horizontal measurements.

- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.
- Several sets of measurements (minimum of two, three suggested) should be taken for the off-line measurements and also for the running measurements to verify that the equipment is in both a stable off-line and stable running position.

Advantages:

- Extreme accuracy possible (1–2 arc seconds) with a precisely leveled and calibrated instrument
- Excellent repeatability if instrument is properly calibrated, consistent positioning of scale targets is done, precise leveling is achieved, and stable reference points have been established



FIGURE 16.49 Jig transit and scale target on outboard pump bearing for lateral measurement.

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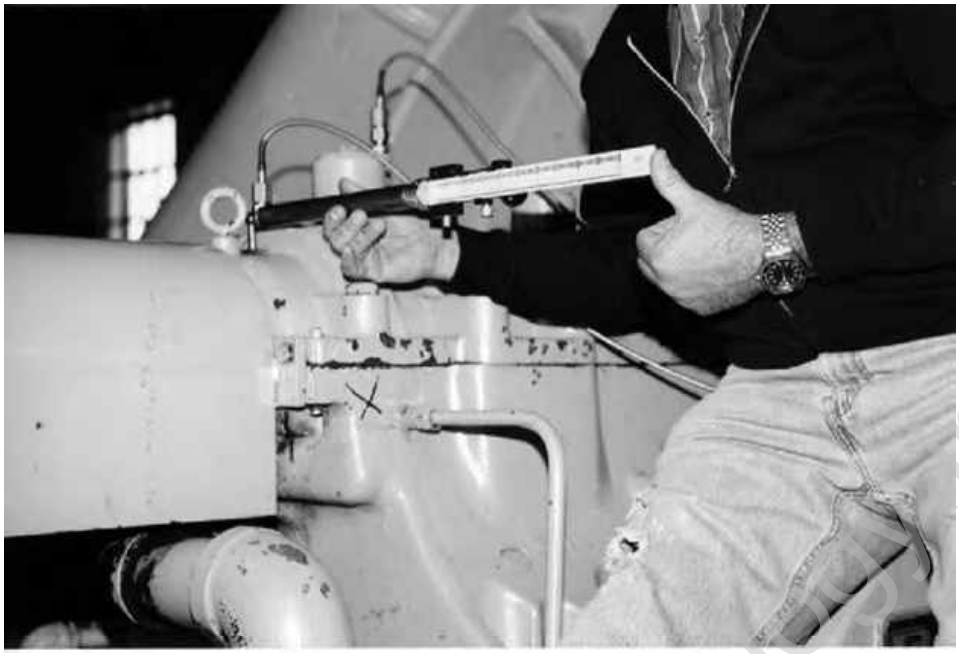


FIGURE 16.50 Holding scale target with invar rod extension for lateral measurement.

- Tremendous versatility and range of possible measurement points
- Measuring instruments can usually be placed away from heat sources

Disadvantages:

- Knowledgeable, patient, and dedicated personnel required for successful measurements and evaluation of data.
- If the machinery is vibrating excessively, it appears that the scale targets cannot be focused when taking the running measurements (a strobe light could be used to measure the high and low points of the target by tuning the strobe slightly off the vibration frequency of the object it is sitting on however).
- Scale and reference target holding devices usually need to be custom fabricated and carefully placed to insure long-term stable position.
- Instrumentation, targets, and fixturing relatively expensive.

16.14 ALIGNMENT BARS WITH PROXIMITY PROBES

This method falls into the category of observing movement of one machine case with respect to a position on the other machine case. This device was invented in the early 1970s by Ray Dodd while working at Chevron. The machinery alignment bar OL2R system is based on the principle of the reverse indicator method explained in Chapter 7.

Two ''bars'' are used, a ''probe bar'' and a ''target bar.'' The probe bar is attached near the inboard (coupling end) bearing as close as possible to the centerline of rotation on one machine case. The target bar is attached near the inboard (coupling end) bearing as close as possible to the centerline of rotation on the other machine case. The probe bar and target bar ''shadow'' each other but they do not touch. Four proximity probes are attached to the probe bar, two vertically oriented probes and two horizontally oriented probes. These probe sets are mounted at two locations along the length of the bar, which observe two positions on the

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FIGURE 16.51 Adhesive-backed target attached to I-beam.



FIGURE 16.52 Adhesive-backed target attached to I-beam.

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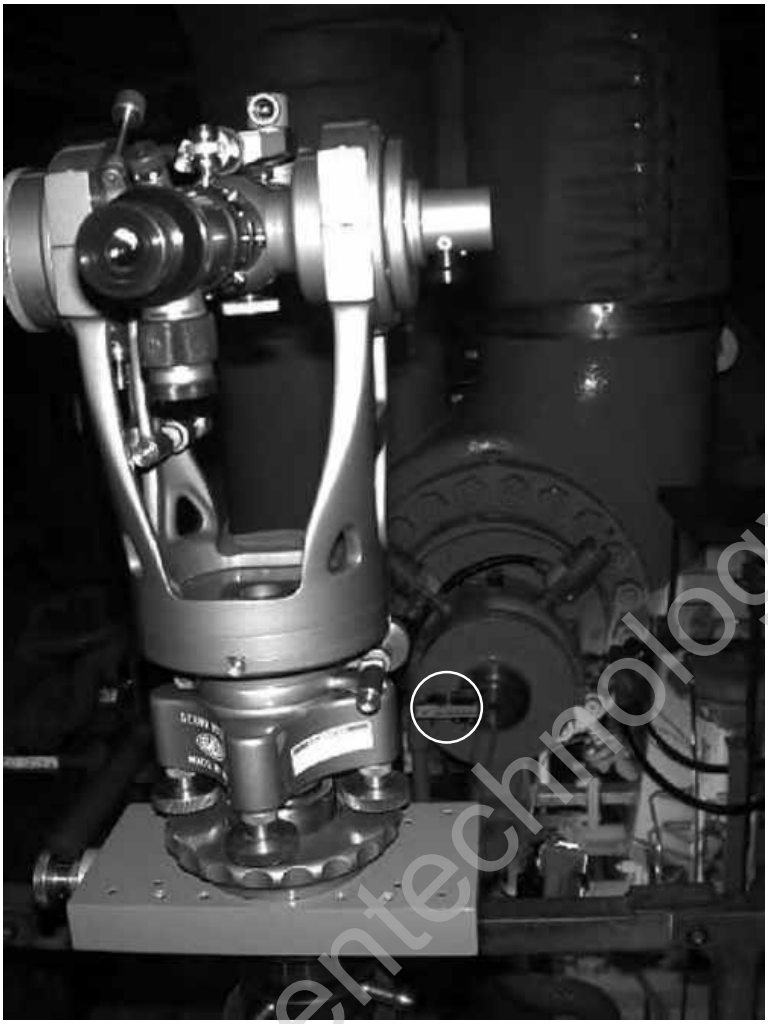


FIGURE 16.53 Transit with scale target in background.



FIGURE 16.54 Adhesive-backed target attached to wall.

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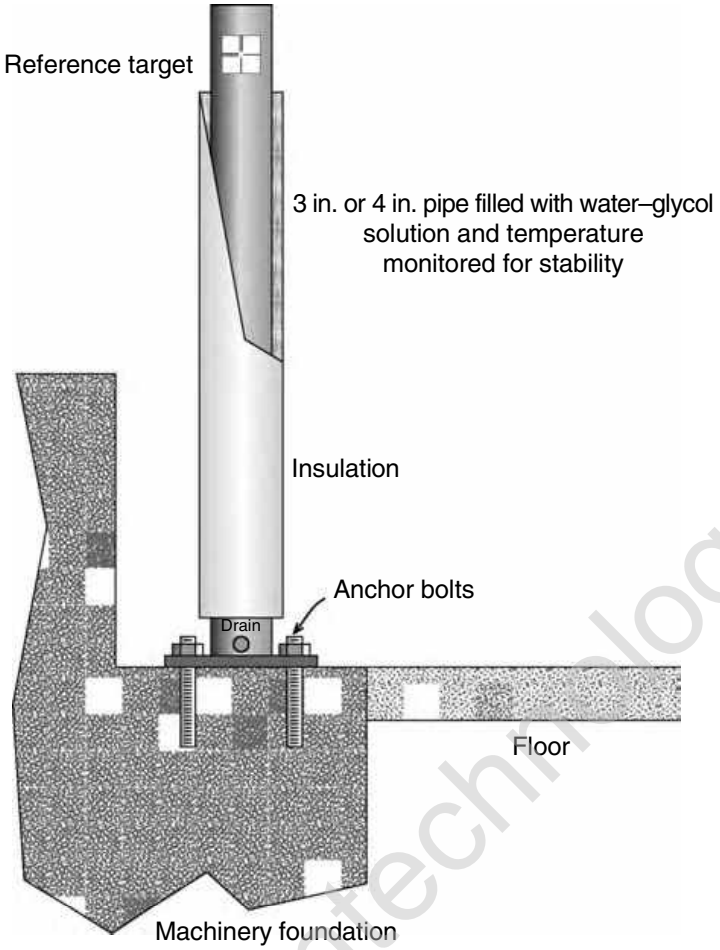


FIGURE 16.55 Water-filled pipe reference stand.

target bar. The targets could be blocks of steel attached to tubing or they could be the target bar surface itself as shown in Figure 16.58 and Figure 16.59.

Before starting the drive train when the machinery is off-line and in a stable position, a set of gap readings are taken on each of the two vertically oriented probes and each of the two horizontally oriented probes. The drive system is then started up and operated under normal running conditions until the probe gaps have stabilized. Relative machinery casing movement can be determined by comparing the gaps on each probe before and after the equipment is running. Strip chart recorders (or similar devices) can be set up to monitor the rate of change of gap during warm-up and on-line operating conditions and to see when the positions have stabilized (Figure 16.60).

Key considerations for capturing good readings:

- The probe and target bars should be attached to each machine case as close as possible to the centerlines of rotation to accurately determine shaft motion, not casing expansion or bearing housing warpage (see Figure 16.26).
- Important to have target surfaces at precise 90° angles to the proximity probes.
- The bars can be positioned inside or outside the coupling guard or shroud with precautions taken to prevent excessive vibration of the bars from coupling windage if mounted inside the guard.
- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

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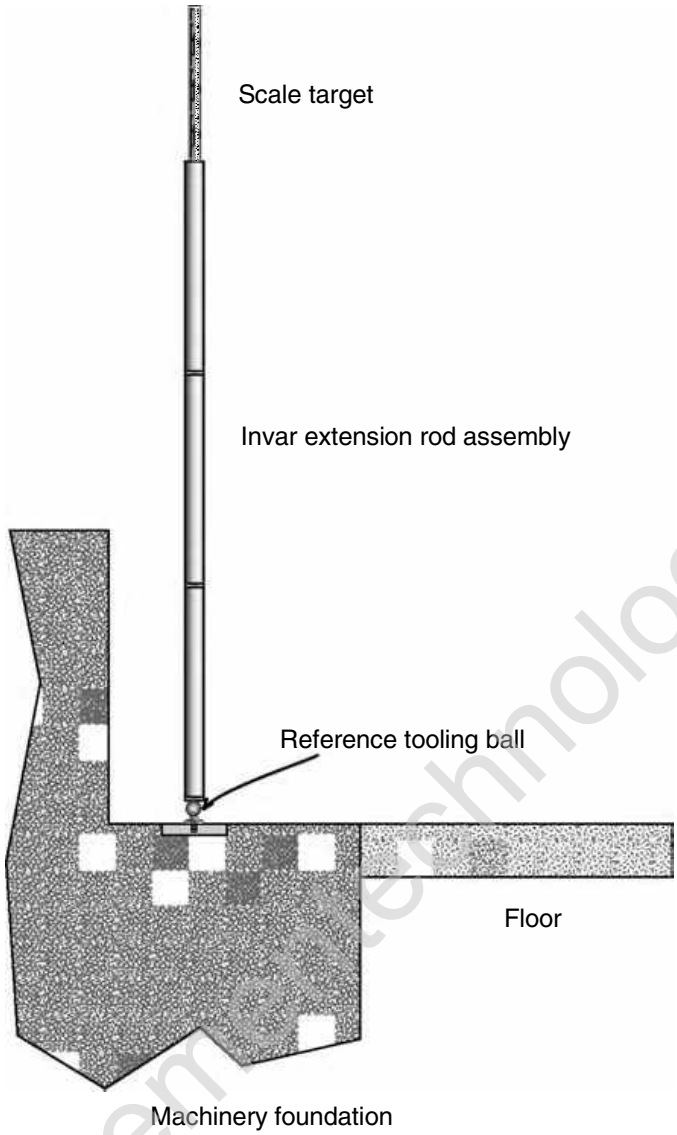


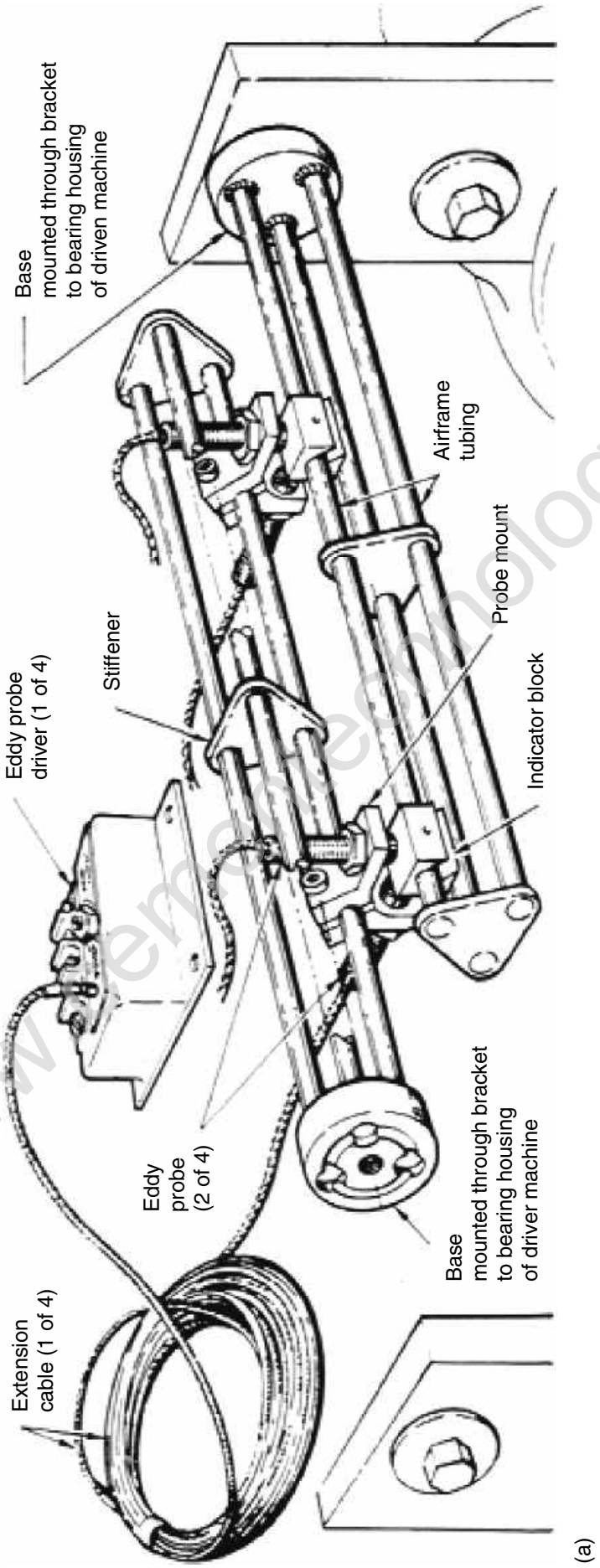
FIGURE 16.56 Invar rod and scale contacting tooling ball on foundation.



FIGURE 16.57 Permanent floor target.

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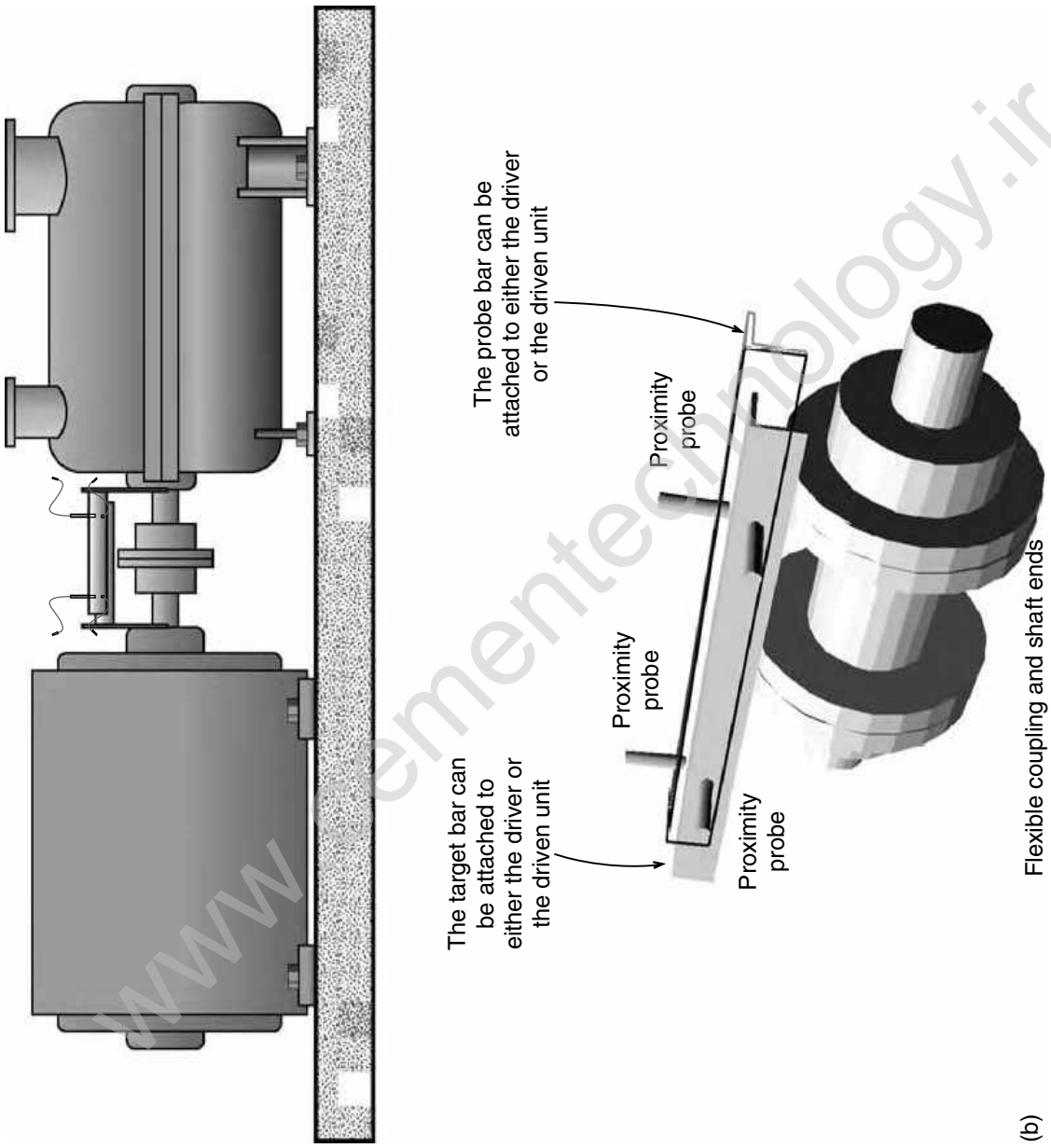


FIGURE 16.58 Alignment bar arrangements. (Upper diagram: Courtesy of Scientific Atlanta/SKF Condition Monitoring, San Diego, CA. With permission.)

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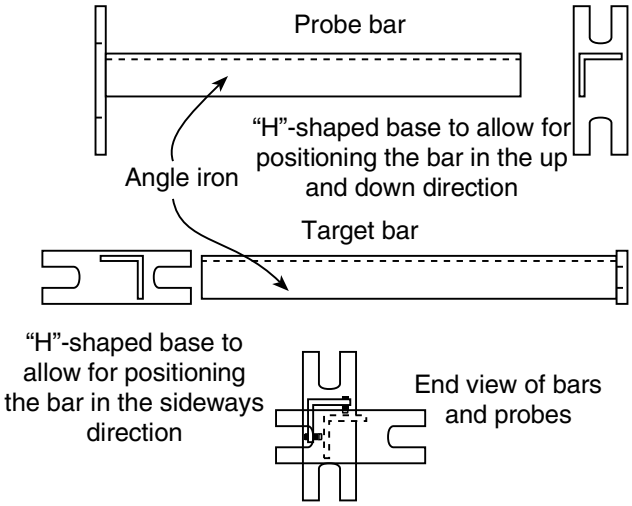


FIGURE 16.59 Fabricated alignment bar design.

Advantages:

- Fairly accurate measurements possible with proper setup.
- Capable of measuring movement in vertical and horizontal directions.
- Can continuously monitor positions of machinery without disturbing sensors or bars.
- If the machinery is vibrating excessively when taking the running measurements, the proximity probes average the oscillation effectively to insure accurate distances between the probe tips and the targets.
- Can be equipped to measure axial growth if desired.

Disadvantages:

- Knowledgeable, patient, and dedicated personnel required for successful measurements and evaluation of data.
- Custom brackets and bars have to be designed, fabricated, and carefully installed on each machine case as close as possible to the centerline of rotation.
- If the machinery is moving considerably from OL2R conditions, the proximity probes can bottom out on the target or move out of the linear range of the probes.
- Thermal casing distortion where the probe and target bars are attached could result in data that are not representatives of the movement of the centerline of rotation of the shafts.
- Somewhat expensive since custom braces and bars have to be fabricated; probes, cables, proximitors, readout devices, and power supplies have to be purchased.

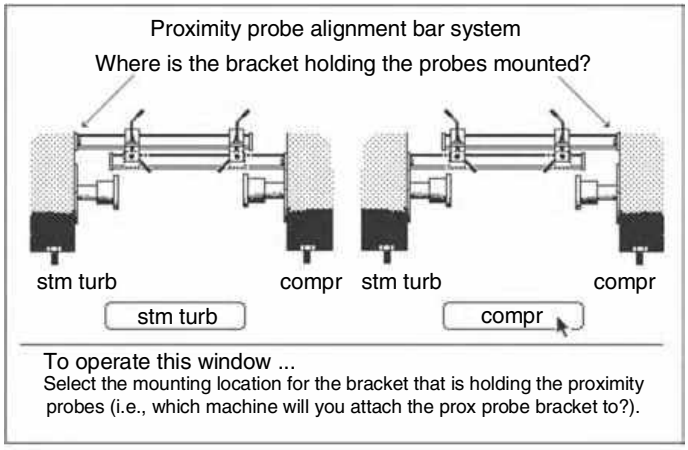
16.15 APPLYING LASER–DETECTOR SYSTEMS FOR OL2R MEASUREMENTS

Laser–detector systems can also be used to measure OL2R machinery movement in the machine case to machine case measurement category.

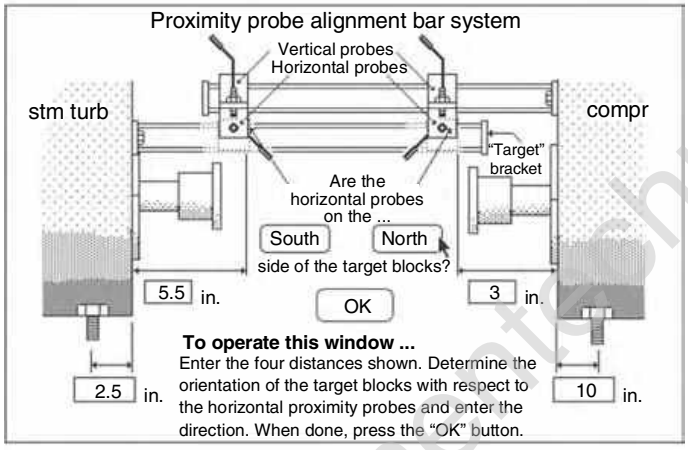
In a very simple setup, where small amounts of relative movement between machinery cases is present, a laser (or laser–detector depending on what system you use) could be mounted at or near the centerline of rotation of one machine near the inboard bearing, and the detector (or prism depending on what system you use) could be mounted at or near the centerline of rotation of the other machine near its inboard bearing as shown in Figure 16.61. With the machinery off-line, the laser–detector system can be oriented to have the laser strike in the

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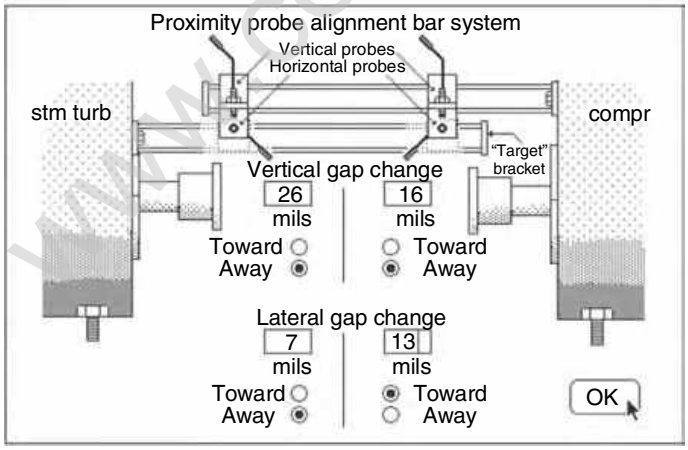
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The probe and target bars can be mounted on either machine case but you have to know what is mounted where.



The distances from the inboard feet to the points where the proximity probes are taking readings on the target bar must be known, as well as the orientation of the horizontal probes.



The gap changes from off-line to running (or vice versa) need to be recorded. Remember to capture not only the amount of the gap change but also the direction (increasing gaps means the probe moved away from the target).

FIGURE 16.60 Alignment bar setup.

center of the detector targets. The laser–detector system is kept in place on the machine cases and the unit started up and operated until movement stops. If movement occurs between the machine cases, the laser will be striking at a different position in the photodetector.

With the plethora of laser shaft alignment systems in existence, it is somewhat amazing to find out that very few people have tried to use their systems for this purpose. The underlying

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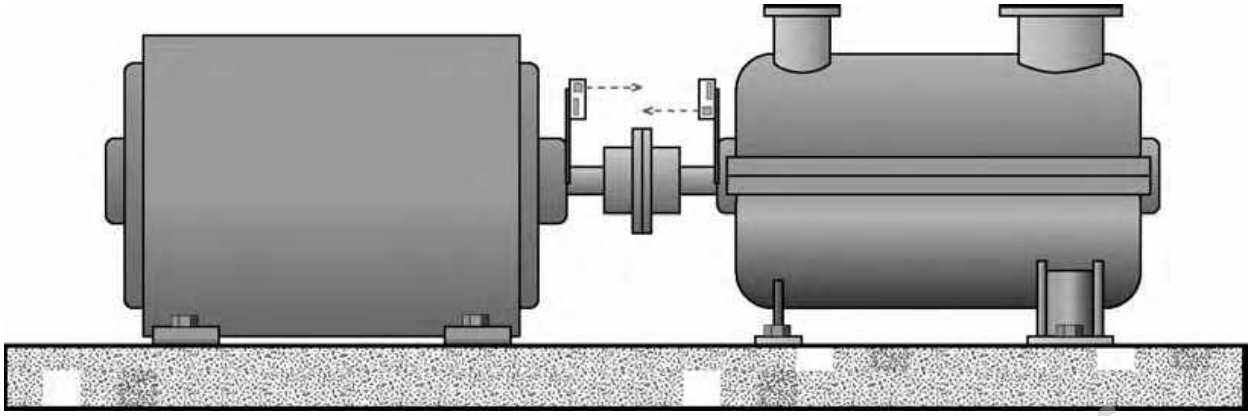


FIGURE 16.61 Basic setup of laser–detector systems used to measure OL2R movement.

reluctance seems to originate from the difficulty in mounting the lasers and detectors to the machine cases. Since there is such a wide variation in machinery design, custom brackets are usually required to hold the equipment in place similar to those shown in Figure 16.62 and Figure 16.63. These brackets must not only hold the laser and detector in a stable position, but also the laser and detector need to have positional adjustments for centering the beam. Additionally, if laser–detector–prism systems or two single-axis laser–detector units are used, two sets of laser–detector prisms are needed to capture all the required information (vertical offset, vertical angularity, horizontal offset, and horizontal angularity). With mounts having a precise 90° rotation feature, one set of such two-axis equipment is sufficient. More detail on such mounts will be given later.

Universal laser–detector mounting brackets can also be purchased as shown in Figure 16.64 through Figure 16.67. These brackets have the capacity to rotate the laser–detectors through a precise 90° arc to capture all of the measurements needed as mentioned previously, thus avoiding the need for two sets of laser–detector equipment.

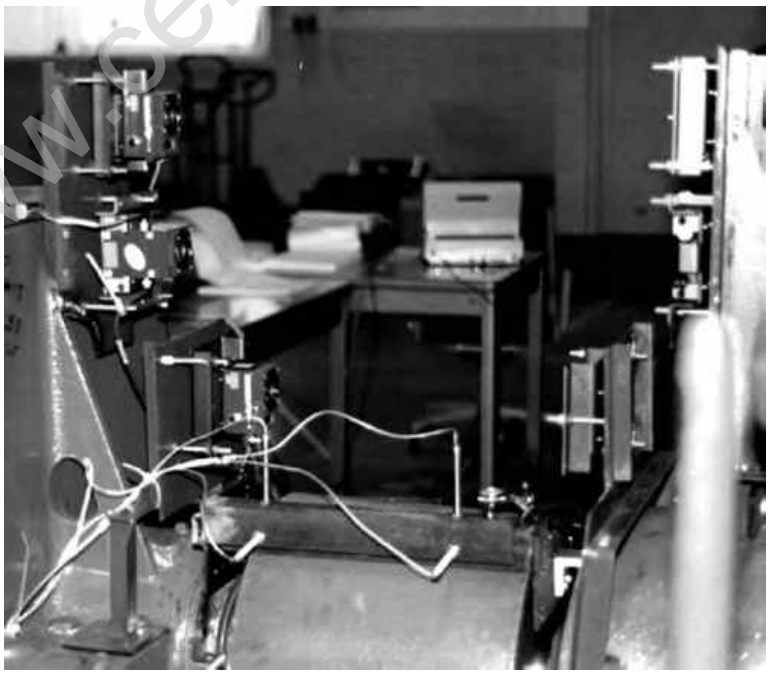


FIGURE 16.62 Laser–detector–prism systems setup on custom mounts measuring gear and compressor OL2R movement. Alignment bar setup also used for comparison.

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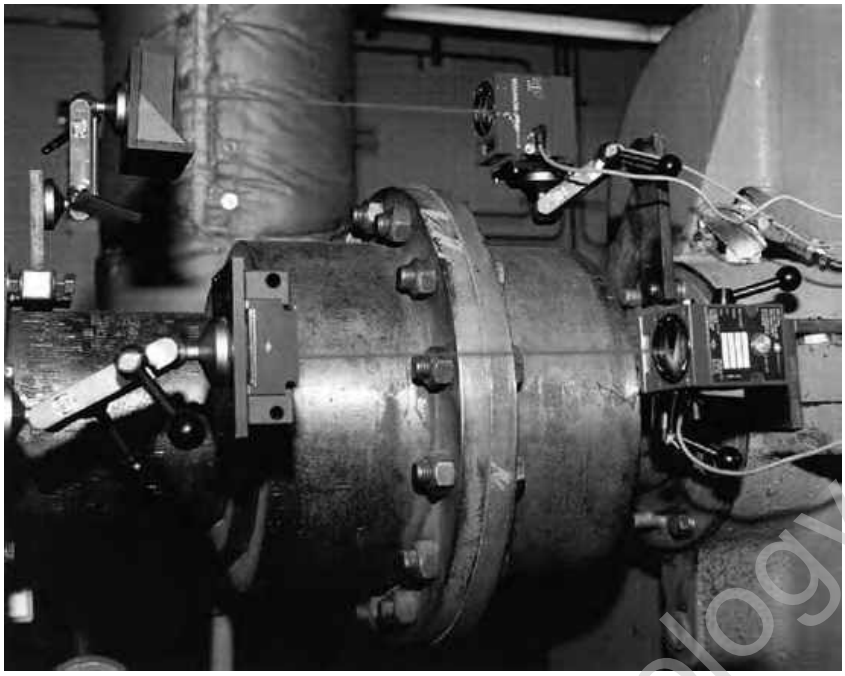


FIGURE 16.63 Permalign systems mounted across coupling. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

If a considerable amount of movement occurs between the machinery cases, the laser beam could potentially traverse outside of the detector target surface area, inhibiting complete and accurate OL2R measurements. Another type of laser–detector mounting system was created to overcome this problem and is referred to as the plug in back zeroing laser target (PIBZLT) mounts as shown in Figure 16.68 through Figure 16.75. These mounts have been designed such that it will allow attachment of different types of lasers, detectors, or prisms, with the capacity to remove and install them repeatably with good remounting precision. There is a ball pivot placed in between the laser–detector mounting plate and the machine case attachment plate with adjusting screws to tilt or pitch the laser or detector, enabling one to zero the

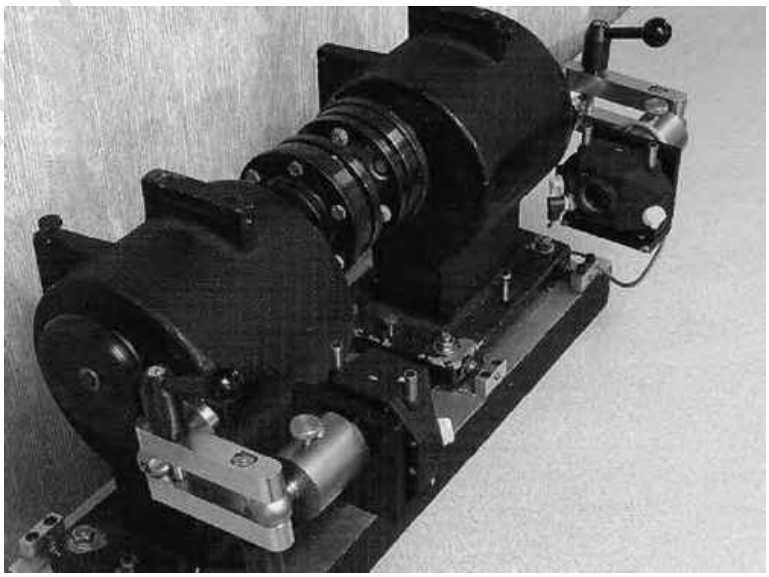


FIGURE 16.64 Permalign M3 brackets and laser–detector mounted across coupling. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

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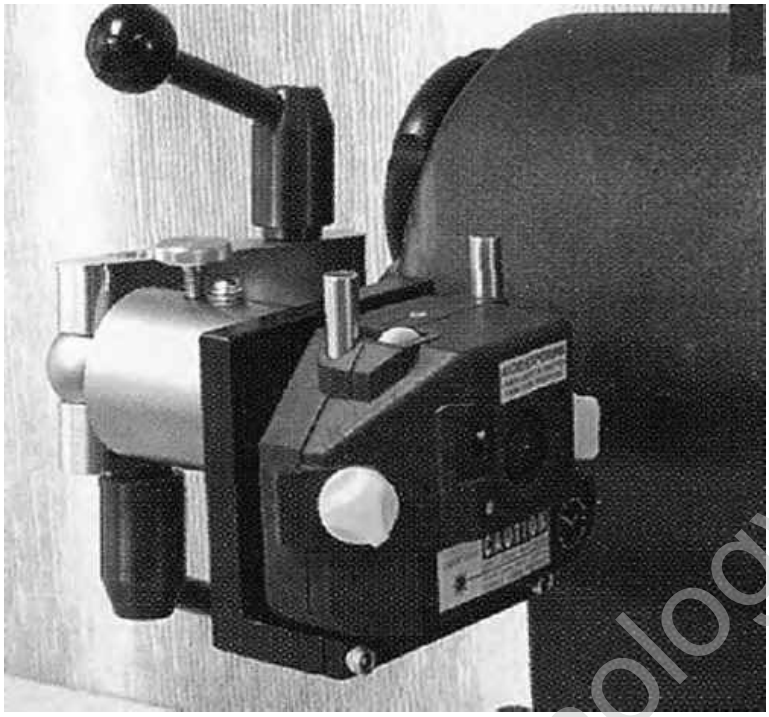


FIGURE 16.65 Permalign M3 brackets and laser–detector mounted across coupling. (Courtesy of Prüftechnik, Ismaning, Germany. With permission.)

laser–detector system. Tooling balls are placed at the twelve-, three-, six-, and nine o'clock positions on both the backing and mounting plates to measure the change in position of the mounts as shown in Figure 16.71. By measuring the position of the mounting and backing plates via the tooling balls with the machinery off-line, adjusting the laser–detector to rezero the beam, and then measuring the position of the mounting and backing plates via the tooling



FIGURE 16.66 Laser–detector OL2R mounts. (Courtesy of Vibralign Inc., Richmond, VA. With permission.)

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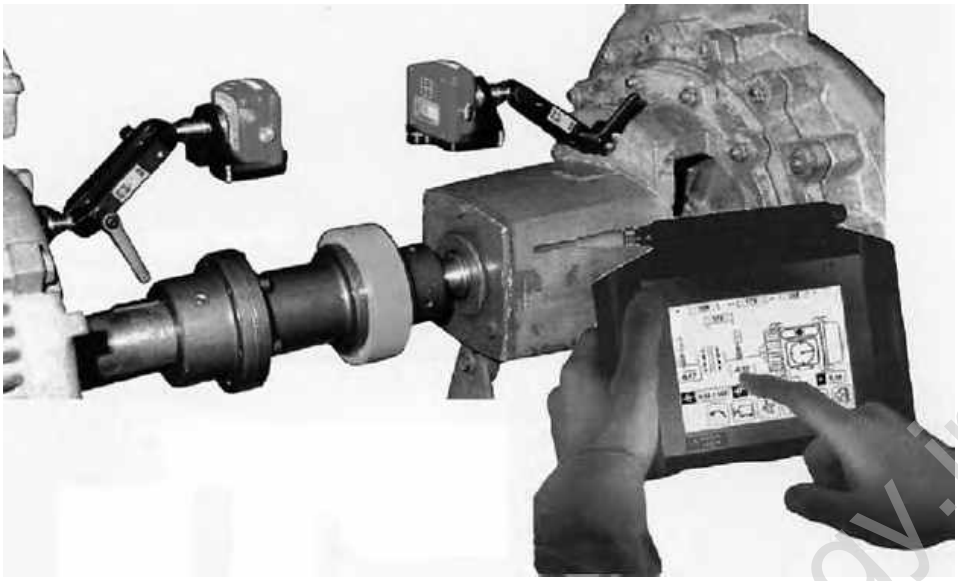


FIGURE 16.67 Laser–detector OL2R mounts. (Courtesy of Vibralign Inc., Richmond, VA. With permission.)

balls with the machinery running, the OL2R movement can be measured. The tooling ball measurements taken with the PIBZLT system are based on the face–face measurement method (see Chapter 14 for reference). The back zeroing feature greatly extends the measurement range. It also corrects for laser target poor linearity if this is present (30%–40% linearity error found on some targets at range limits).

Key considerations for capturing good readings:

- The laser–detector mounts should be attached to each machine case as close as possible to the centerlines of rotation to accurately determine shaft motion, not casing expansion or bearing housing warpage.

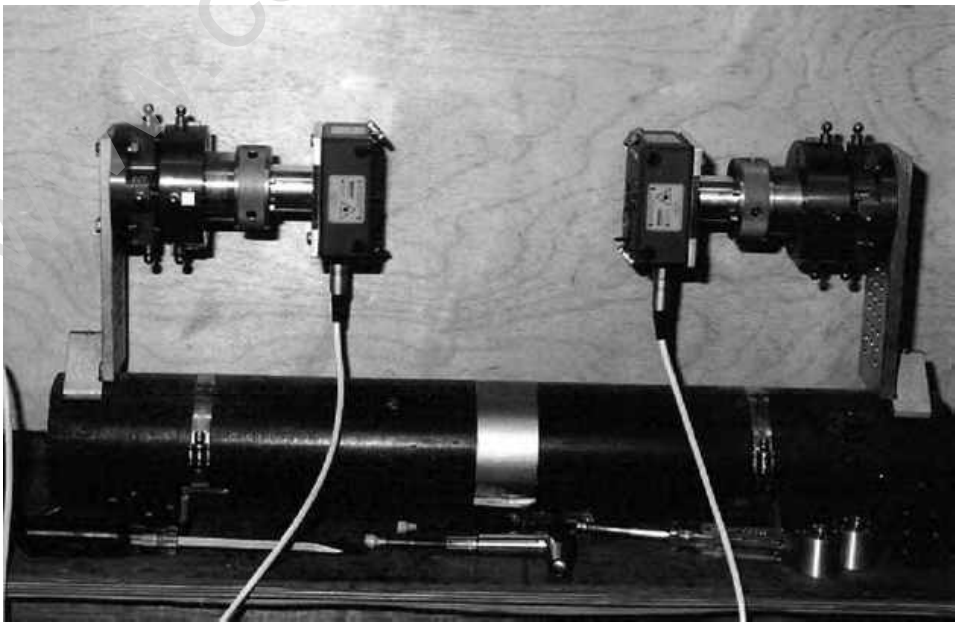


FIGURE 16.68 Plug in back zeroing laser-target mounts. (Courtesy Murray & Garig Tool Works, Baytown, TX. With permission.)

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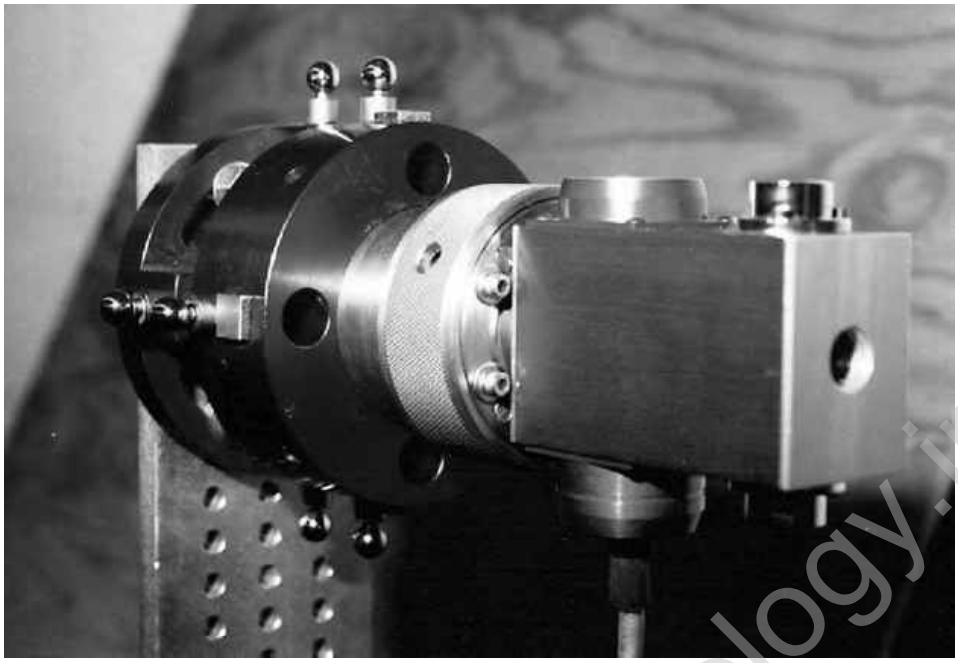


FIGURE 16.69 Close-up of four-axis laser target on PIBZLT mount. (Courtesy Murray & Garig Tool Works, Baytown, TX. With permission.)

- Mounting devices must be stable and rigid enough to maintain a precise position of the lasers and detectors even under moderately long time periods with machinery running and vibration present.
- Accurate positioning mechanisms for laser and detector required on mounting fixtures.

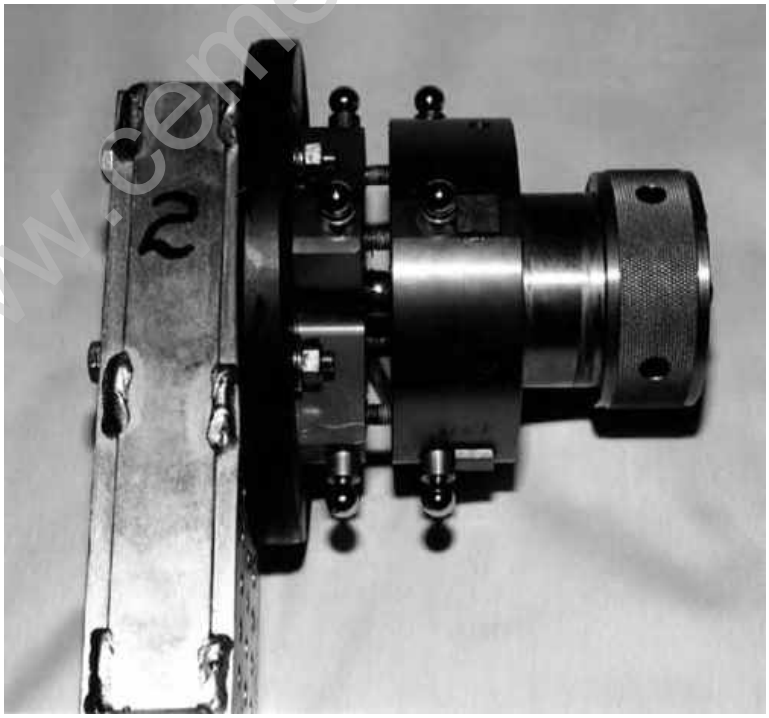


FIGURE 16.70 Side view of PIBZLT mount. (Courtesy Murray & Garig Tool Works, Baytown, TX. With permission.)

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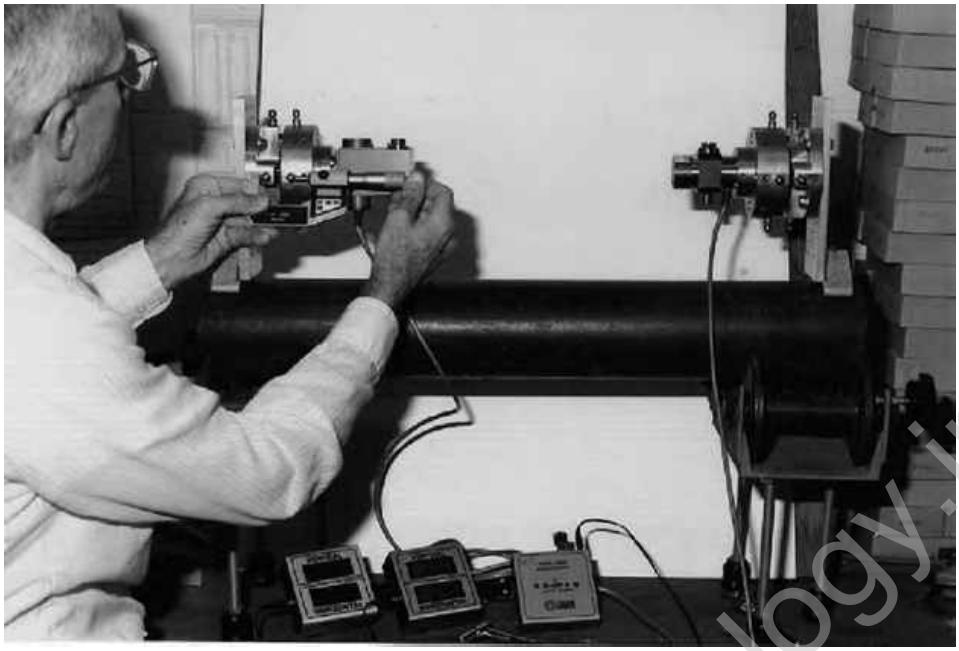


FIGURE 16.71 Taking micrometer measurements on tooling balls on PIBZLT system. The four-axis laser target is on the left mount and the laser source is on the right mount. (Courtesy Murray & Garig Tool Works, Baytown, TX. With permission.)

- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

Advantages:

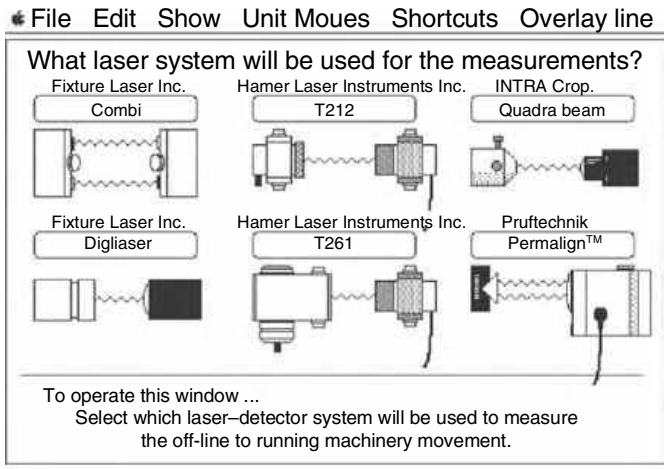
- Fairly accurate measurements possible with proper setup.
- Capable of measuring movement in vertical and horizontal directions.
- Can continuously monitor positions of machinery for moderate periods of time with stable fixtures.
- If the machinery is vibrating excessively when taking the running measurements, the photodetector can average the position of the center of the "bouncing" laser beam fairly accurately.

Disadvantages:

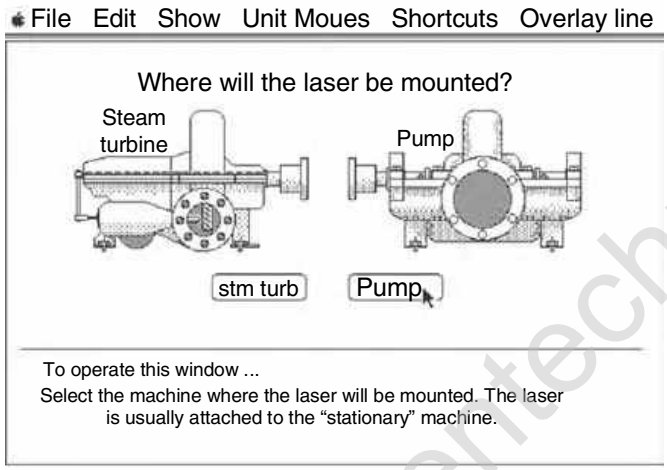
- Knowledgeable, patient, and dedicated personnel required for successful measurements and evaluation of data.
- Other light sources, steam, water vapor, etc. can affect the ability to capture accurate readings.
- If the machinery is moving considerably from OL2R conditions, the laser beam can move out of the range of the photodiode surface area unless the mounts have the capability of getting positioned knowing what the before and after positions were (e.g., PIBZLT mounts).
- Custom brackets have to be designed and fabricated or purchased and carefully installed on each machine case as close as possible to the centerline of rotation.
- Possibility of inaccurate measurements due to uneven thermal distortion of machine case where the holding fixtures are attached.

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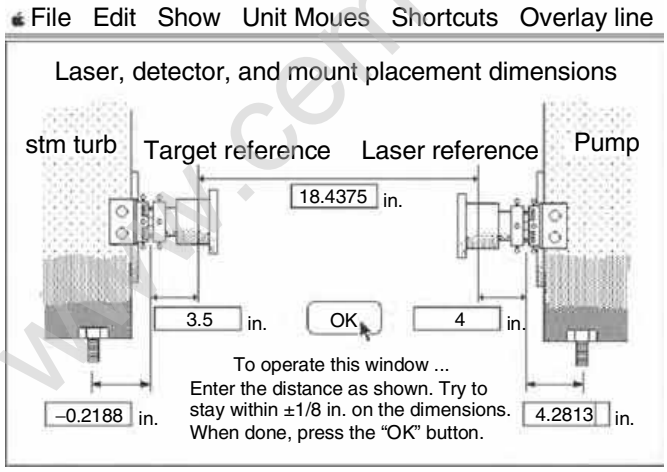
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Determine which laser system will be used with the mounts. The data capture sequence will be slightly different depending on which system is used.



Select where the lasers and where the detectors will be mounted.



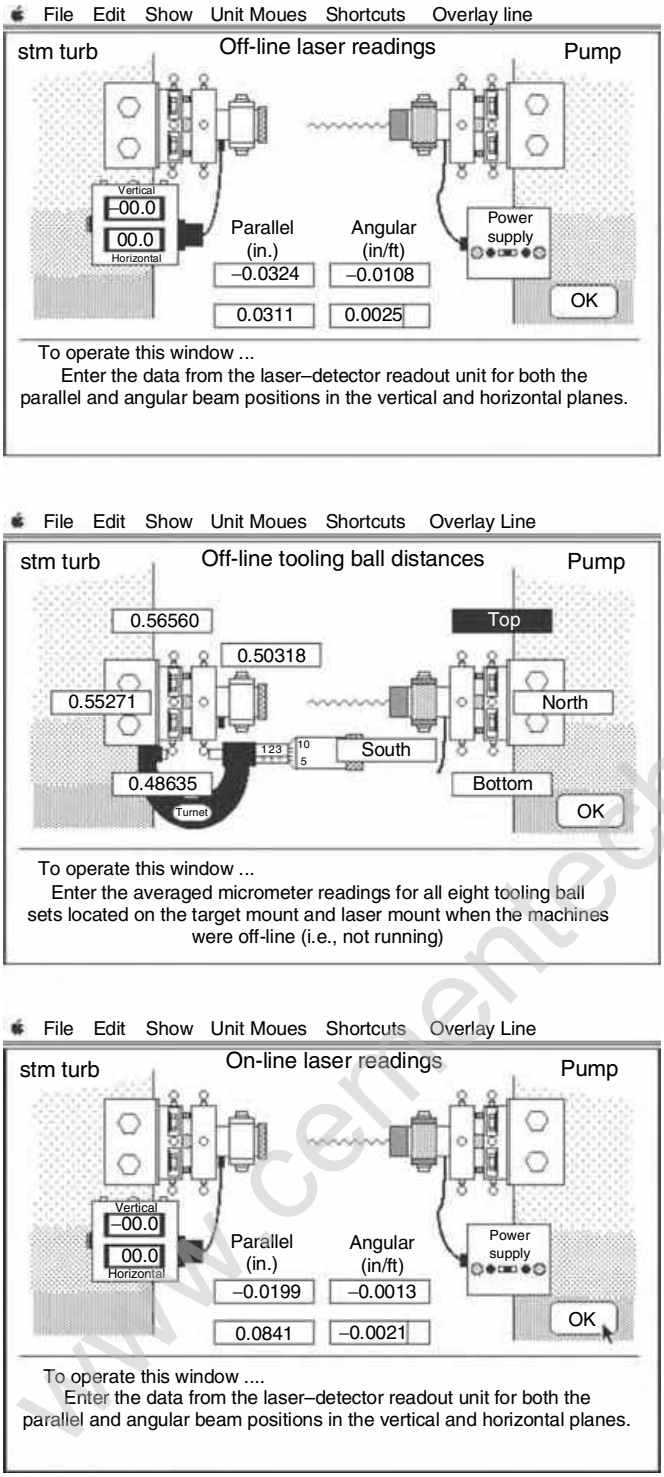
Measure the distances from the mounting locations of the lasers and detectors to the pivot point in the PIBZLT mount and then to the inboard feet of the machinery (or other reference points on the drive train).

FIGURE 16.72 PIBZLT setup information.

- Relatively expensive since lasers, detectors, readout devices, and mounts have to be purchased.
- Incapable of measuring axial shaft or casing movement.

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Zero the laser beams into the detectors by positioning the tilt or pitch adjustment screws on the mounts with the machinery off-line.

After the beam has been centered, measure the gaps at all eight tooling ball points (four on each mount) and record the dimensions.

Start the machinery up and allow sufficient time for the readings to stabilize. Rezero the beam by adjusting the tilt or pivot screws.

FIGURE 16.73 PIBZLT measurement information.

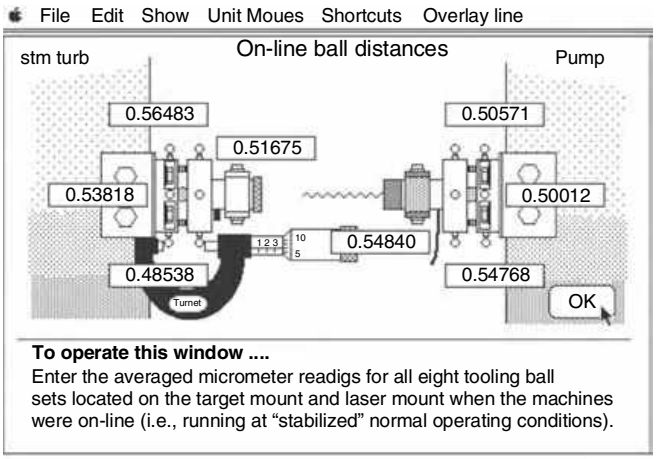
16.16 BALL–ROD–TUBING CONNECTOR SYSTEM

The ball–rod–tubing connector system, referred to as the BRTC system, falls into the category of measuring OL2R machinery movement from one machine case to the other machine case. The basic setup of the system is shown in Figure 16.76.

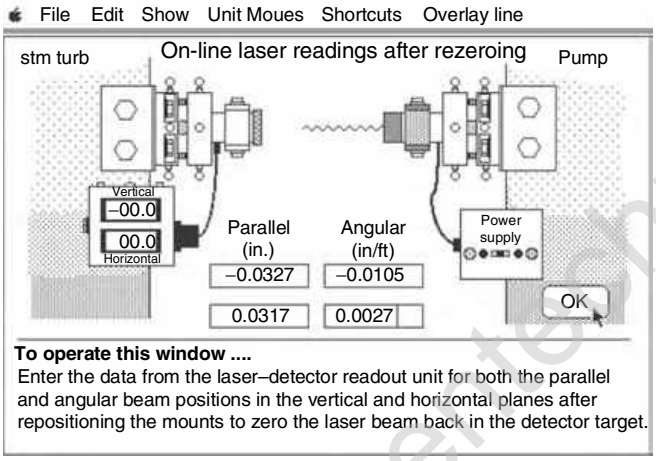
Two base blocks are attached to each machine case near their inboard bearings. Two short rods with a round ball attached to the end of each rod are indexed through a hole in

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If the beam has moved out of the range of the detector, rezero the lasers into the detectors by adjusting the tilt or pivot adjustment screws on the mount. Measure the distances between all eight tooling ball pairs and record the distances.



In the event that you are unable to exactly rezero the beam, record the *X*–*Y* coordinates of the beams in the detectors.

Note: The procedure shown may vary slightly depending on the system used.

FIGURE 16.74 PIBZLT measurement information.

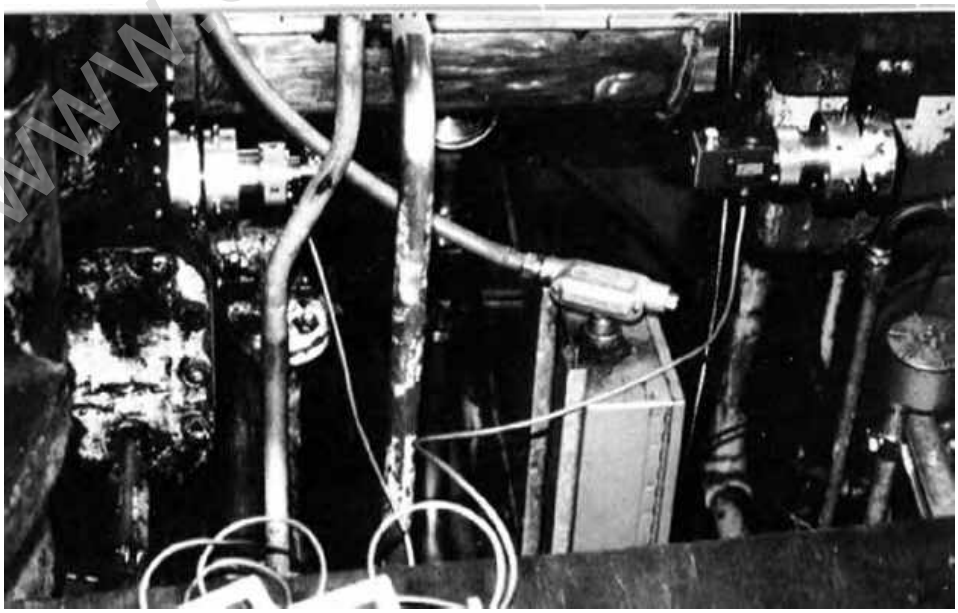


FIGURE 16.75 PIBZLT system measuring movement of steam turbine and compressor. (Courtesy Murray & Garig Tool Works, Baytown, TX. With permission.)

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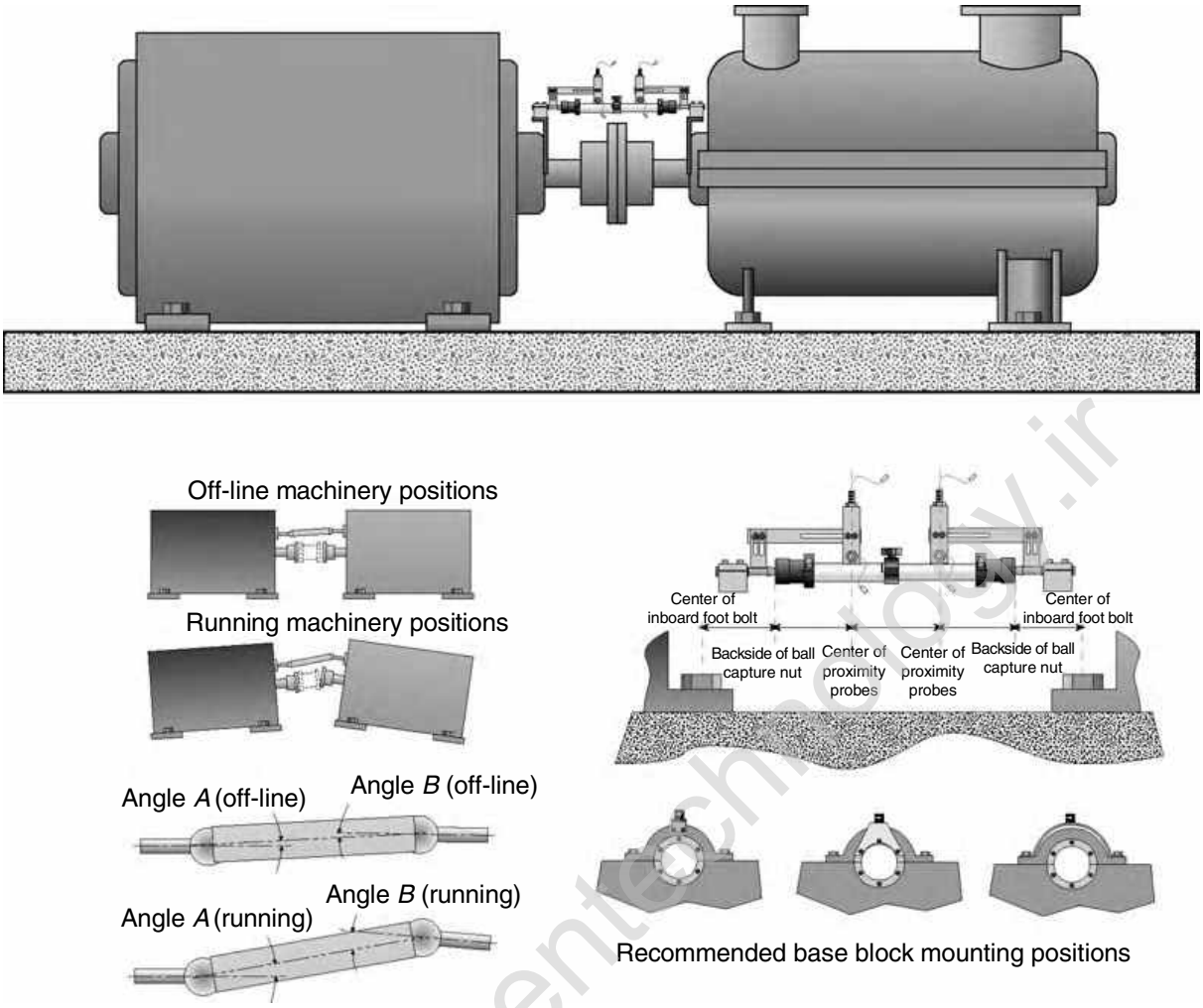


FIGURE 16.76 BRTC system basic setup and measurement principles.

each base block and clamped into place. Telescoping tubing is affixed to each ball end. A brace is attached to each rod that spans out to the tubing connector that holds two proximity probes (a vertical and a horizontal probe). Proximity probe target surfaces are attached to the tubing connector.

Gap readings are taken at all four proximity probes with the machinery off-line. The BRTC system remains attached to each machine case and the equipment is started up and allowed to stabilize in its final operating position. Gap readings are taken again during operation. This system is based on the shaft to coupling spool measurement principles adapted to measure OL2R machinery movement.

Key considerations for capturing good readings:

- The base blocks should be attached to each machine case as close as possible to the centerlines of rotation to accurately determine shaft motion, not casing expansion or bearing housing warpage.
- During operation, one rod end should be allowed to "float" to allow for any axial movement of the machine cases and then locked in place to capture running proximity probe gaps.
- The tubing connector must be able to freely pivot at each ball end.
- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

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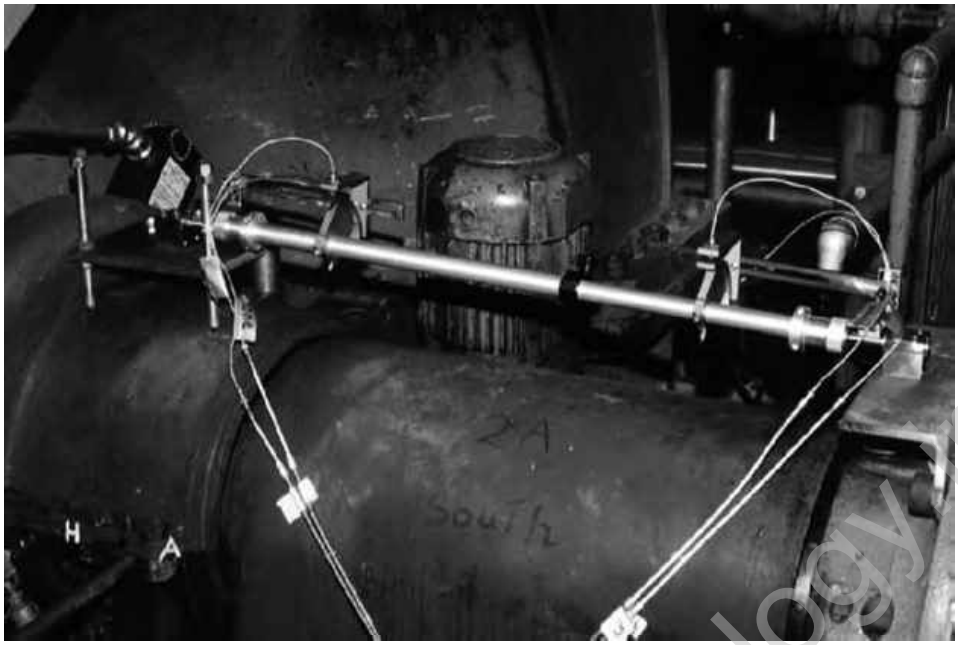


FIGURE 16.77 BRTC system measuring OL2R movement of steam turbine and pump.

Advantages:

- Fairly accurate measurements possible with proper setup.
- Capable of measuring movement in vertical and horizontal directions.
- Can continuously monitor positions of machinery without disturbing sensors or bars.
- If the machinery is vibrating excessively when taking the running measurements, the proximity probes average the oscillation effectively to insure accurate distances between the probe tips and the targets.

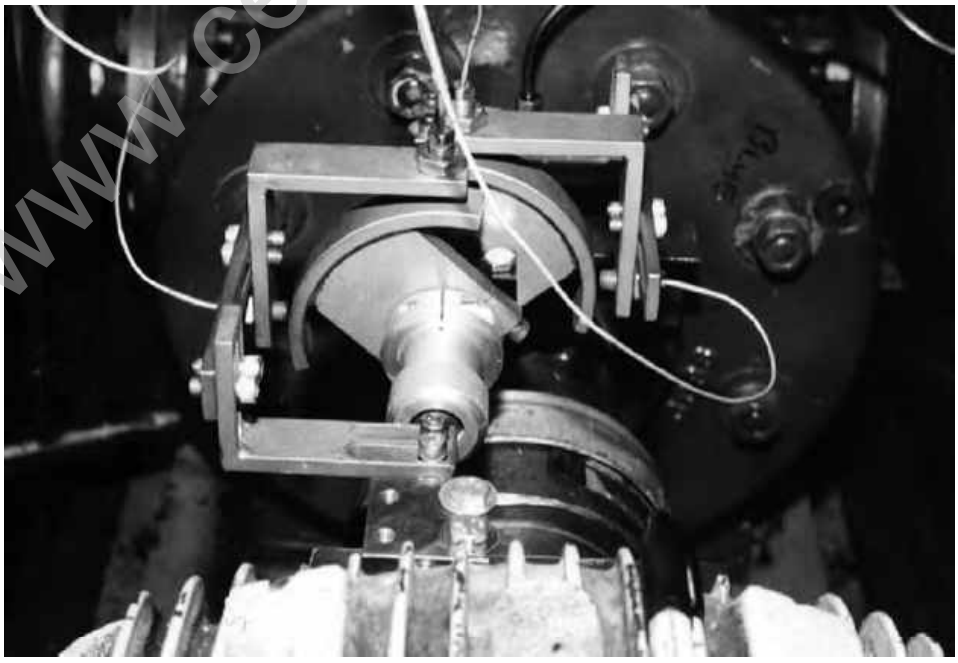


FIGURE 16.78 BRTC system measuring OL2R movement of motor and compressor.

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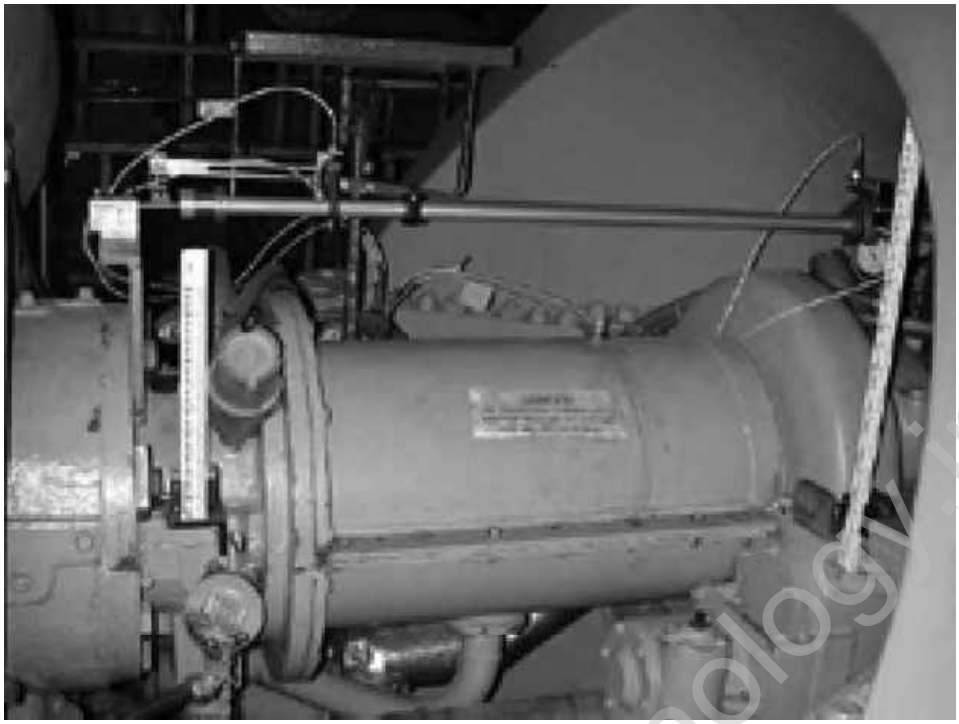


FIGURE 16.79 BRTC system measuring OL2R movement of steam turbine and pump.

- The rod ends, probes, targets, and tubing connector can be removed and reinstalled without disturbing the position of the base blocks attached to the machinery cases.
- If the machinery is moving considerably from OL2R conditions and the proximity probes bottom out on the target or move out of the linear range of the probes, the probe span can be changed (shortened or lengthened) to accommodate this excessive movement.

Disadvantages:

- Knowledgeable, patient, and dedicated personnel required for successful measurements and evaluation of data.
- Custom brackets and bars have to be designed, fabricated, and carefully installed on each machine case as close as possible to the centerline of rotation.
- Somewhat expensive since custom braces and bars have to be fabricated; probes, cables, proximitors, readout devices, and power supplies have to be purchased.

16.17 VERNIER–STROBE SYSTEM

The vernier–strobe system utilizes the face–face measurement principles adapted to measure OL2R machinery movement. This method measures movement of the shafts themselves from OL2R conditions (Figure 16.80).

Two small vernier scale sets are firmly attached across each flexing point in a coupling as shown in Figure 16.81. Two more vernier scale sets are attached 180° apart for a counterbalance. A set of readings is taken at each vernier at the twelve-, three-, six-, and nine o'clock shaft positions when off-line. The machinery is started up (with the verniers still attached to the coupling) and allowed to stabilize in its final operating position. A variable rate strobe light is then aimed at the coupling area to visually "freeze" the shaft positions. By varying the

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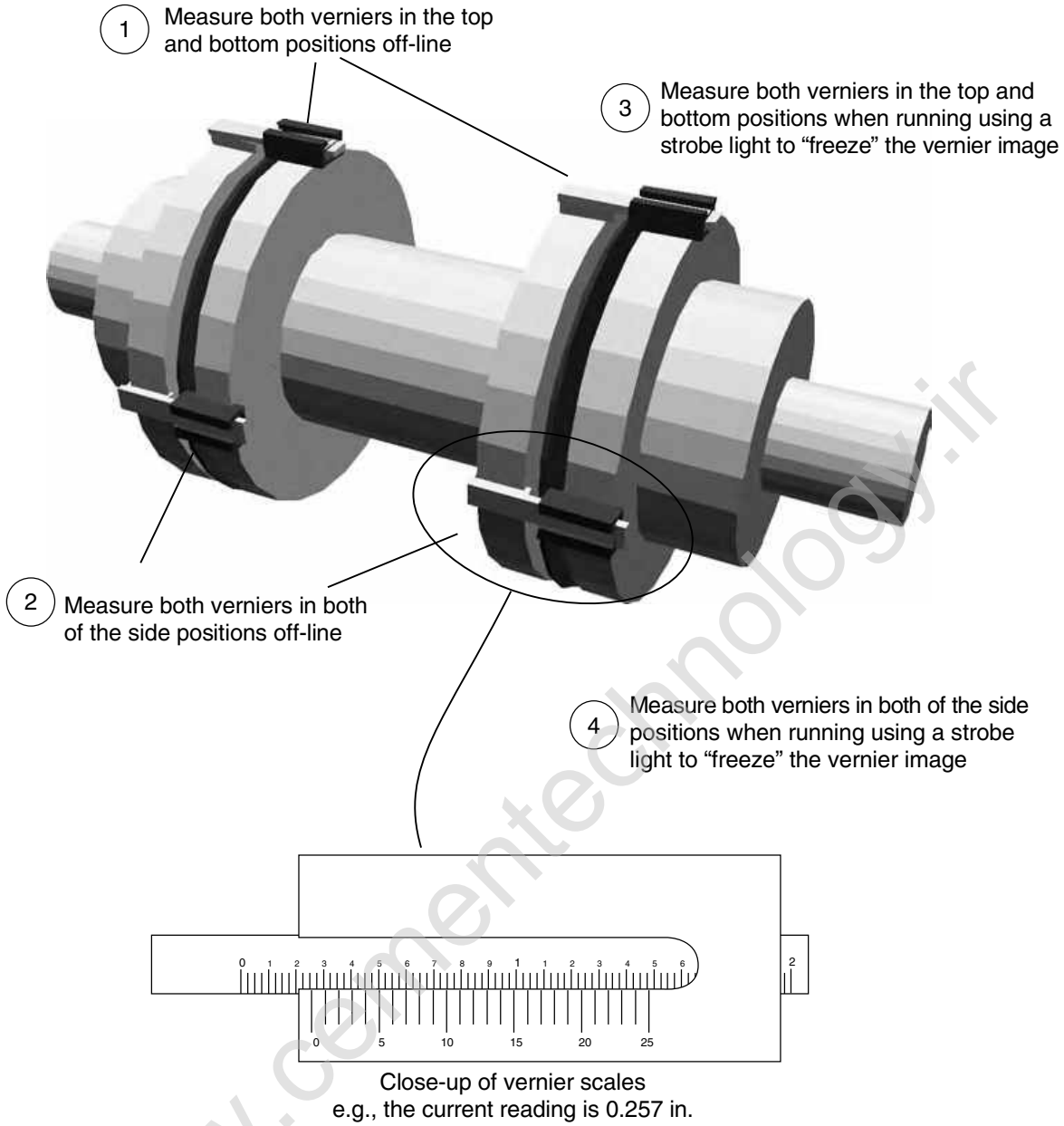


FIGURE 16.80 Basic setup of vernier–strobe system.

strobe rate slightly, the vernier scales can be rotated to the twelve-, three-, six-, and nine o'clock shaft positions when readings are taken during running conditions.

Brian Howes of Beta Machinery Analysis Ltd. in Calgary Alberta, Canada, made an interesting discovery a number of years ago. He found that many machines have synchronous axial movement in their shafts or coupling spacers while running. In effect, this means that "raw" vernier–strobe reading differences across 180° may consist of two components, the misalignment and the portion of the gap difference caused by this synchronous axial movement. Solving for the gap at the often inaccessible six o'clock position using the validity rule, is therefore unreliable. This is because the rule depends on three measurements, which are themselves unreliable as they are the combinations of alignment and axial movement data.

Brian solved the problem quite cleverly by noting that primary face measurements in machines having sleeve bearings are also subjected to error if the shaft moves axially between the initial and 180° opposite positions. He then utilized the method explained in Chapter 11

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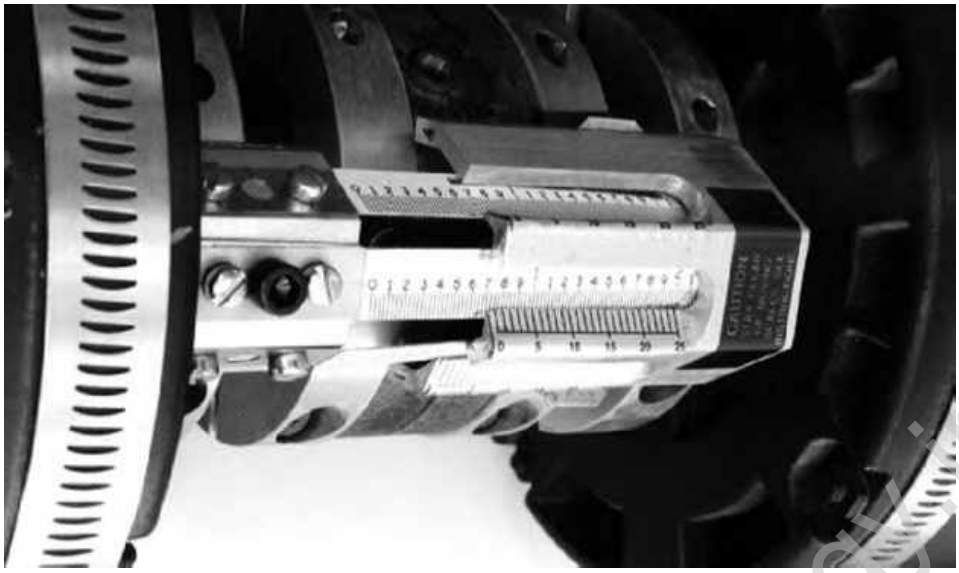


FIGURE 16.81 Vernier scale attached to flexible coupling. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

(see Figure 11.13 and Figure 11.14), which gets rid of axial float, leaving only the face misalignment.

With a single vernier caliper, getting the bottom reading can become difficult or dangerous trying to use a mirror or borescope to observe the running reading there. A modified design shown in Figure 16.82 shows a "roof-shaped" triple vernier arrangement, which aids to observe the bottom measurement without standing upside down, laying on your back, or using a mirror, borescope, or midget helper. The three scales will not, for a variety of reasons, have identical readings at any given rotational position. It is important, therefore, to always



FIGURE 16.82 Strobe light observing verniers on coupling during operation. (Courtesy of Murray & Garig Tool Works, Baytown, TX. With permission.)

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read the same scale in each primary and 180° opposite position. Remember, we are not looking for absolute measurement, but differences in readings of the same scale across 180° of rotation.

Key considerations for capturing good readings:

- The vernier scales must be firmly attached to the coupling hubs and spool piece.
- During operation, it is advisable to use a camera to capture the vernier scale readings for safety.
- A metal mesh coupling guard should be used during operation.
- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

Advantages:

- Direct shaft measurements
- Fairly accurate measurements possible with proper setup
- Capable of measuring movement in vertical and horizontal directions as well as axial shaft growth
- Relatively inexpensive
- Good portability
- Usable with photography for easy reading and for future reference

Disadvantages:

- Knowledgeable, patient, and dedicated personnel required for successful measurements and evaluation of data.
- Potential for bodily harm if the vernier scales part company from the shaft during operation.
- Can only be used on couplings with a connector spool piece (e.g., flexible disk pack or gear-type couplings).
- Machinery must be easily stopped and started.

16.18 INSTRUMENTED COUPLING SYSTEMS

The instrumented coupling system is manufactured by Indikon Corp. and this system utilizes the face–face measurement principles adapted to measure OL2R machinery movement (Figure 16.83).

Four proximity probes are housed inside the coupling spool piece. Two axially positioned probes are attached at each end of the coupling spool observing target surfaces that are attached to the ends of each shaft. Two noncontacting, rotating or stationary coil sets transmit power to the probes and capture the signal from each probe. The instrumented coupling transmits the proximity probe gap information during off-line and running conditions.

Key considerations for capturing good readings:

- Recommend that manufacturers installation instructions be followed very carefully to insure proper operation.
- Capture a set of readings from OL2R conditions and another set of readings from R2OL conditions to determine if there is a consistent pattern of movement.

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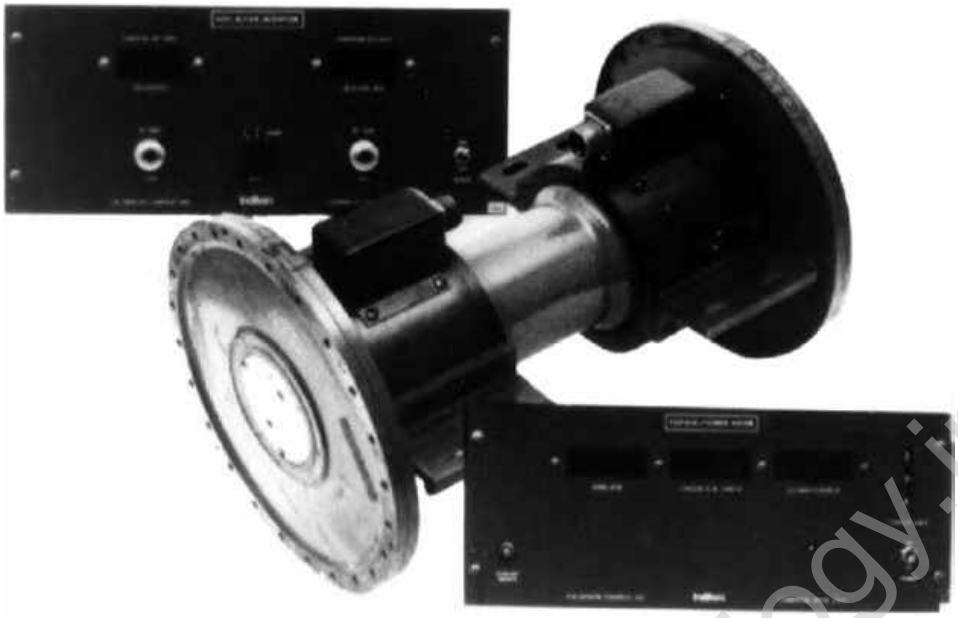


FIGURE 16.83 Instrumented coupling system. (Courtesy of Indikon Corp. With permission.)

Advantages:

- Direct shaft measurements
- Fairly accurate measurements possible with proper setup
- Capable of measuring movement in vertical and horizontal directions
- Can continuously monitor positions of machinery shafts
- Can measure axial shaft growth

Disadvantages:

- Knowledgeable, patient, and dedicated personnel required for successful measurements and evaluation of data
- Can only be used on couplings with a connector spool piece (e.g., flexible disk pack or gear-type couplings)
- Relatively expensive

16.18.1 Aligning Rotating Machinery to Compensate for OL2R Machinery Movement

Once accurately and hopefully repeatable measurements have been collected and analyzed on how the equipment moved from OL2R conditions in the field, the machine elements can then be properly positioned during the off-line shaft-to-shaft centerline alignment process to compensate for this movement to insure collinear shaft centerlines during operating conditions. Depending on how the OL2R data were collected, there are different procedures used to interpret the information to finally obtain the desired off-line shaft positions.

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16.18.2 Determining the Desired Off-Line Shaft Positions When Using the Machine Case to Baseplate or Machine Case to Remote Reference Point Methods

If you employed one of the following techniques to measure OL2R movement, the data you collected show how each end of the machinery moved from OL2R conditions (Figure 16.84 and Figure 16.85) .

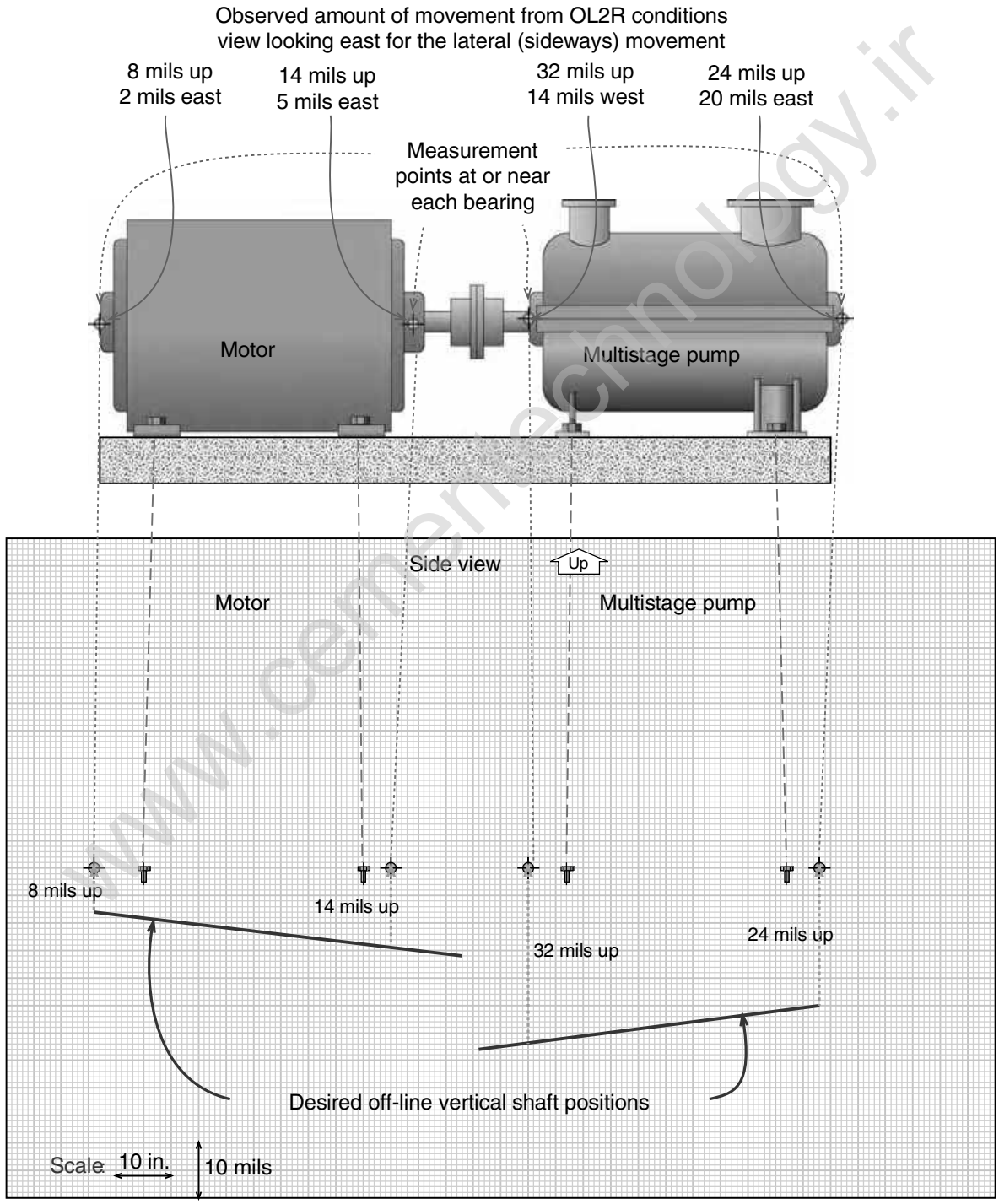


FIGURE 16.84 Example of a desired off-line side view (vertical) shaft position alignment model using the machine case to baseplate or machine case to remote reference point methods.

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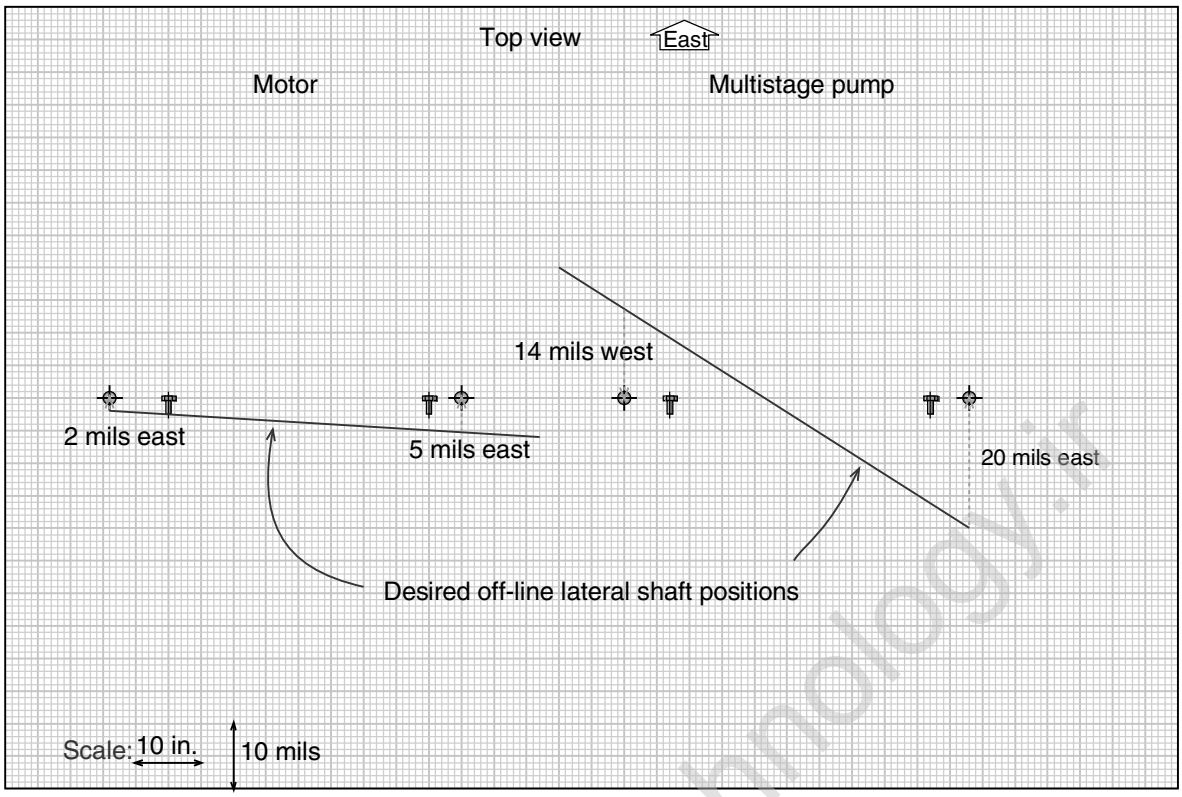


FIGURE 16.85 Example of a desired off-line top view (lateral) shaft position alignment model using the machine case to baseplate or machine case to remote reference point methods.

- Calculating machine case thermal expansion using the strain equation
- Inside micrometer–tooling ball–angle measurement devices
- Proximity probes with water-cooled stands
- Optical alignment equipment

Graph paper similar to what is used for the graphing or modeling techniques covered in Chapter 8 can be used to show the desired off-line shaft positions. The graph centerline will represent the final position of the shafts, which is often referred to as the "hot operating position" or running shaft positions. If the machinery shafts move from OL2R conditions, lines will be drawn on the graph paper to represent what position they should be in when off-line, so that when they move during operation, they will come in line with each other (i.e., end up on top of the graph centerline).

Along the graph centerline, mark where the OL2R measurements were taken at the inboard and outboard ends of each piece of machinery. Other critical points such as the dial indicator (or laser–detector) reading point locations and foot bolt points can be shown. Once the desired off-line shaft positions are drawn, "shoot for" dial indicator readings can be determined for the shaft positions when off-line.

It should become apparent by this time that if you are using dial indicators and brackets that have sag and that the shafts should not be in line with each other when off-line, you should never want to "spin zeros" for the dial indicator readings.

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16.18.3 Determining the Desired Off-Line Shaft Positions When Using the Machine Case to Machine Case Methods

If you employed one of the following techniques to measure OL2R movement, the data you collected show how one machine case moved with respect to the other machine case from OL2R conditions:

- Alignment bars or custom fixtures with proximity probes
- Laser–detector systems with custom-fabricated brackets or special mounting systems
- Ball–rod–tubing connector system

Graph paper similar to what is used for the graphing or modeling techniques covered in Chapter 8 can be used to show the desired off-line shaft positions. The graph centerline will represent the final position of the shafts, which is often referred to as the "hot operating position" or running shaft positions. In these OL2R methods, it is not known how each machine moved from OL2R conditions with respect to a fixed point in space (as opposed to the previously covered methods which do). What is known is how one machine saw the other machine move. Therefore, one of the two machine cases or shafts is used as a reference shaft and its position is placed directly on top of the graph centerline. The other machine case or shaft is then drawn on the graph paper to reflect how it moved with respect to the reference shaft.

Along the graph centerline, mark where the OL2R measurements were taken at the inboard and outboard ends of each piece of machinery. Other critical points such as the dial indicator (or laser–detector) reading point locations and foot bolt points can be shown. Once the desired off-line shaft positions are drawn, shoot for dial indicator readings can be determined for the shaft positions when off-line.

If the alignment bar system was used to determine the machinery movement, the desired off-line side view (vertical) shaft position alignment model setup might look like Figure 16.86. A little bit of thought is going to have to be put forth to recall how the probes were positioned when reading the targets and what decreasing or increasing gaps mean when setting up the chart. It is easy to make a mistake here by misinterpreting the movement data, so it is wise to make sure both the amount of movement and the direction of movement are correct and that you have gone over the graph setup at least twice before running out and positioning the machinery with shoot for readings that are wrong. Figure 16.87 shows how the desired off-line side view (vertical) shaft position alignment model might look if you used a laser–detector system with custom-fabricated brackets or generic mounting brackets or if you used the BRTC system.

16.18.4 How to Determine the "Shoot for" Off-Line Dial Indicator Readings (Also Known as "Target Values")

So far in this chapter, we have reviewed a number of methods to determine how machinery will move from OL2R conditions. In addition, we have been able to take these data and plot the information onto a graph showing where the shafts should be when the equipment is not running. As you can see, if all of the shafts in the drive system do not move in unison with each other (i.e., the same amount and in the same direction), the shaft centerlines should not be collinear when off-line. Since the shafts should not be in line with each other when off-line, what should the off-line alignment measurements be to insure the shafts are in the desired off-line positions similar to what is shown in Figure 16.86 through Figure 16.88. What would the

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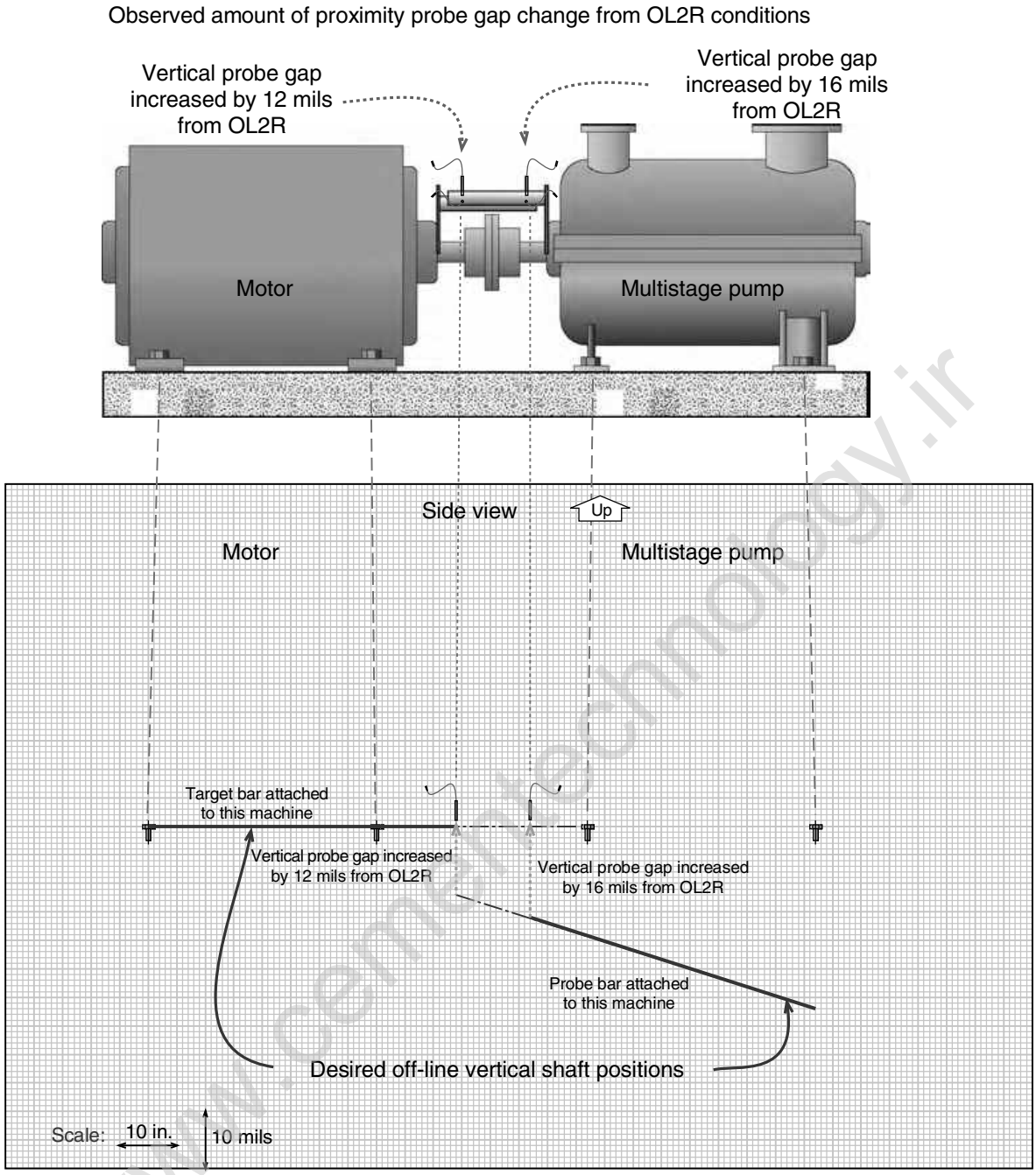


FIGURE 16.86 Example of a desired off-line side view (vertical) shaft position alignment model using the alignment bars or custom fixtures with proximity probes. The desired off-line top view (lateral) shaft position alignment model is not shown.

alignment readings be if you were using the reverse indicator method, face–rim, double radial, shaft to coupling spool, or the face–face method?

16.18.4.1 Reverse Indicator Shoot for Dial Indicator Readings

If you will be using the reverse indicator method to align your machinery, apply the following procedures to determine what the shoot for readings will be when aligning your machinery to compensate for OL2R movement:

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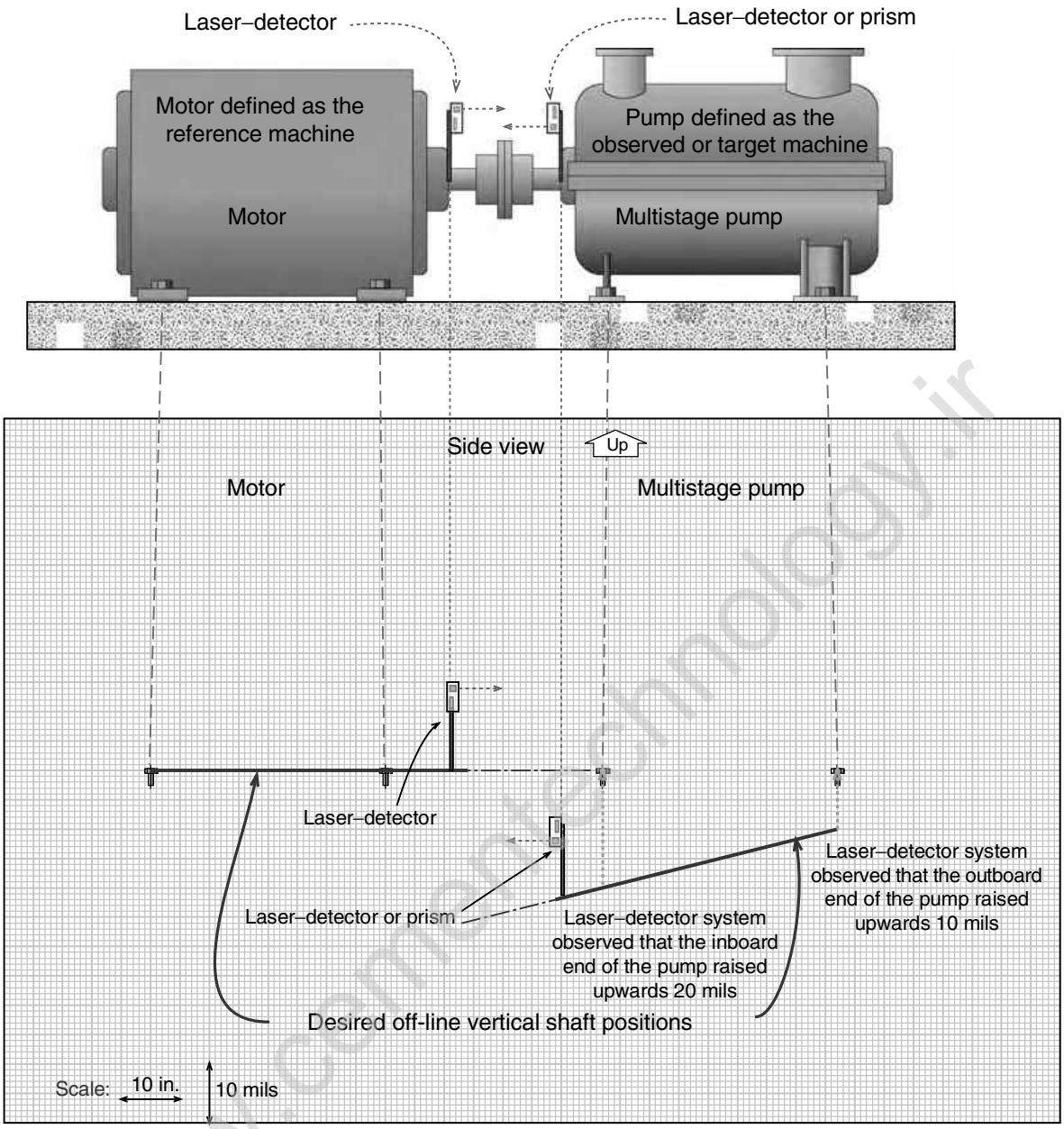


FIGURE 16.87 Example of a desired off-line side view (vertical) shaft position alignment model using a laser–detector system with custom-fabricated brackets or special mounting systems. The desired off-line top view (lateral) shaft position alignment model is not shown.

1. Plot the desired off-line shaft positions of both the driver and driven units. Figure 16.89 shows a motor and a pump plotted in both the side and top views. The amount of movement of these shafts are based on the data collected from any of the OL2R measurement techniques explained in this chapter.
2. Based on the chosen scale factor from top to bottom on the chart, measure the *A* and *B* gaps.
3. Determine whether the bottom readings taken on each shaft are positive or negative by applying the following rules. Rules to determine the sign (+) or (−) of the measurements:
   a. If the actual centerline of a unit is toward the bottom of the graph with respect to a projected centerline, the reading will be positive (+).

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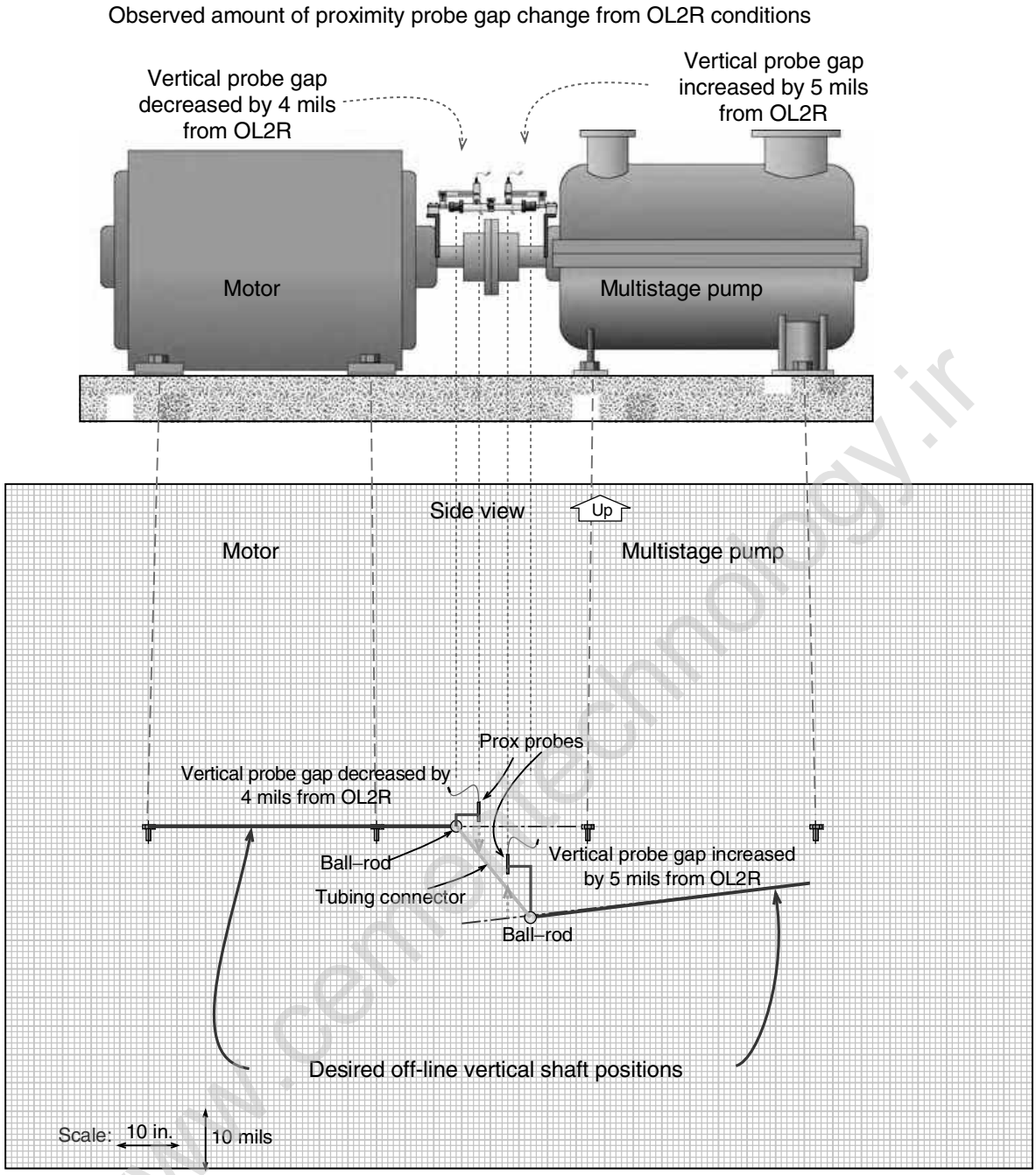


FIGURE 16.88 Example of a desired off-line side view (vertical) shaft position alignment model using a ball–rod–tubing connector system. The desired off-line top view (lateral) shaft position alignment model is not shown.

b. If the actual centerline of a unit is toward the top of the graph with respect to a projected centerline, the reading will be negative (−).

In other words, try to visualize what is going to happen to the dial indicator stem as it traverses circumferentially from top to bottom on the shaft of each machine. Is it going to move outward (negative) or inward (positive)? In Figure 16.89, the side view shows that the motor centerline appears to be higher from the vantage point of the pump, therefore the dial indicator stem will move outward as it rotates to the bottom of the pump shaft producing

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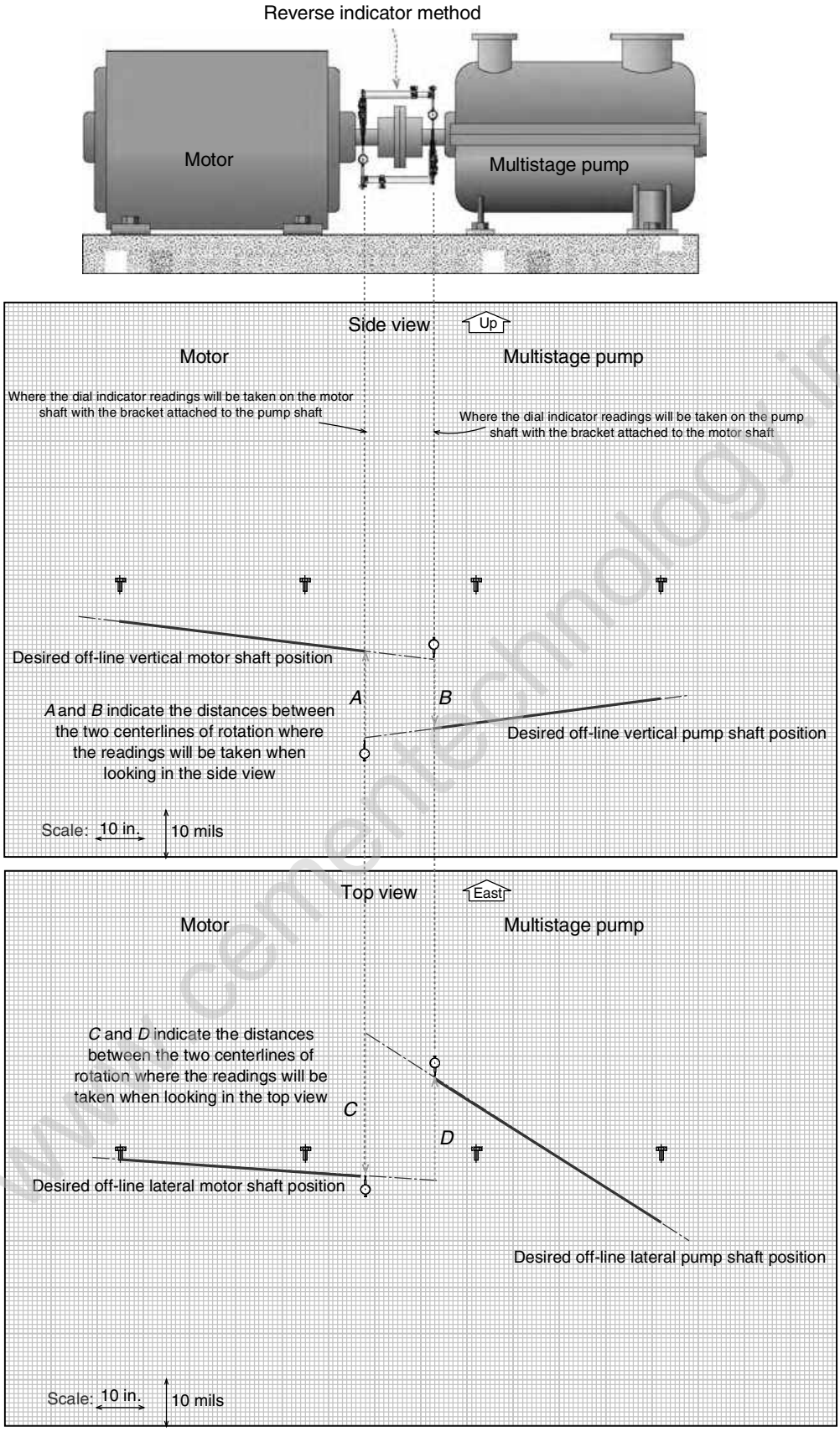


FIGURE 16.89 Example of desired off-line side and top view alignment models of a motor and a pump to calculate the shoot for reverse indicator measurements.

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a negative reading. From the vantage point of the motor, the dial indicator stem will move inward as it rotates to the bottom of the pump shaft producing a positive reading.

4. Based on the chosen scale factor from top to bottom on the chart, record the *C* and *D* gaps as shown in Figure 16.89 for the top view. Remember, you should always zero your indicator on the side that is pointing toward the top of your graph paper, in this case, it is east. Apply the same logic explained in step 3 to determine if the reading will be positive or negative.
5. Apply the appropriate gaps at *A*, *B*, *C*, and *D* into the equations shown in Figure 16.90 and solve. The shoot for reverse indicator readings solution for the desired off-line shaft positions in the side and top views for Figure 16.89 is shown in Figure 16.90 assuming that there is 10 mils of bracket sag.

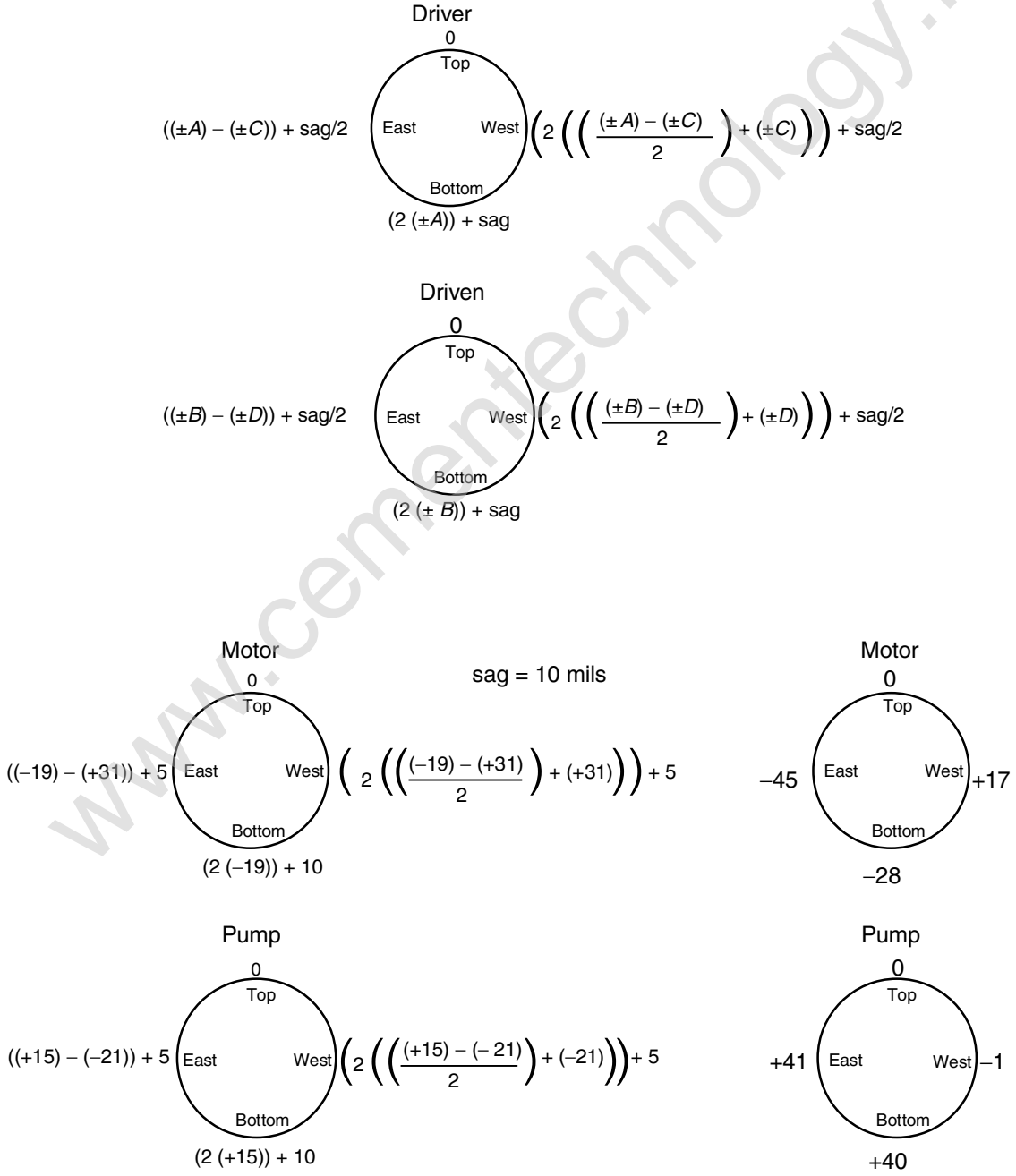


FIGURE 16.90 General equations to calculate the shoot for reverse indicator measurements and a sample calculation based on the shaft positions shown in Figure 16.89.

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16.18.4.2 Face–Rim Shoot for Dial Indicator Readings

Figure 16.91 shows the desired off-line shaft positions of a motor and a pump in both the side and top views. In this particular case, "front side" face readings were taken on the pump shaft and the T-bar overlay was used to model the desired shaft positions. Similar to the procedure for reverse indicator, measure the gaps *A*, *B*, FV, and FL. Figure 16.92 shows the general equations needed to solve for the shoot for face–rim readings as well as the specific face–rim shoot for readings you would obtain for the desired off-line shaft positions shown in Figure 16.91.

16.18.4.3 Double Radial Shoot for Dial Indicator Readings

Figure 16.93 shows the desired off-line shaft positions of a motor and a fan in both the side and top views. Similar to the above procedure for reverse indicator, measure the gaps *A*, *B*, *C*, and *D*. Figure 16.94 shows the general equations needed to solve for the shoot for double radial readings as well as the specific double radial shoot for readings you would obtain for the desired off-line shaft positions shown in Figure 16.93.

16.18.4.4 Shaft to Coupling Spool Shoot for Dial Indicator Readings

Figure 16.95 shows the desired off-line shaft positions of a gear and a motor in both the side and top views. Similar to the procedure for reverse indicator, measure the gaps *A*, *B*, *C*, and *D*. Figure 16.96 shows the general equations needed to solve for the shoot for shaft to coupling spool readings as well as the specific shaft to coupling spool shoot for readings you would obtain for the desired off-line shaft positions shown in Figure 16.95.

16.18.4.5 Face–Face Shoot for Dial Indicator Readings

Figure 16.97 shows the desired off-line shaft positions of a motor and a calender roll in both the side and top views. In this particular case, "front side" face readings were taken from both the motor to the drive shaft (also known as coupling spool) and from the calender roll shaft to the drive shaft. The T-bar overlay was again used to model the desired shaft positions. Similar to the procedure for face–rim, measure the gaps FA, FB, FC, and FD. Figure 16.98 shows the general equations needed to solve for the shoot for face–face readings as well as the specific face–face shoot for readings you would obtain for the desired off-line shaft positions shown in Figure 16.97.

16.19 ALIGNING SHAFTS FOR RUNNING CONDITIONS (ALSO KNOWN AS RUNNING ALIGNMENT OR "HOT OPERATING ALIGNMENT")

The graphing or modeling techniques shown in Chapter 8 illustrated how to align two shafts with each other to insure they were collinear when off-line. If you do not want the shafts to be collinear when they are not running but want them to be in a specific desired off-line position is similar to what is shown in Figure 16.89 through Figure 16.98.

The trick to offset aligning rotating machinery shafts is to shift the position of the shafts to where they will be when they are running and align the running shaft positions. Once the shafts have been shifted to their running positions, the vertical and lateral movement restrictions can be superimposed onto the model, and the overlay line can then be used to determine the appropriate vertical and lateral repositioning movements required to put the shafts in the desired off-line positions.

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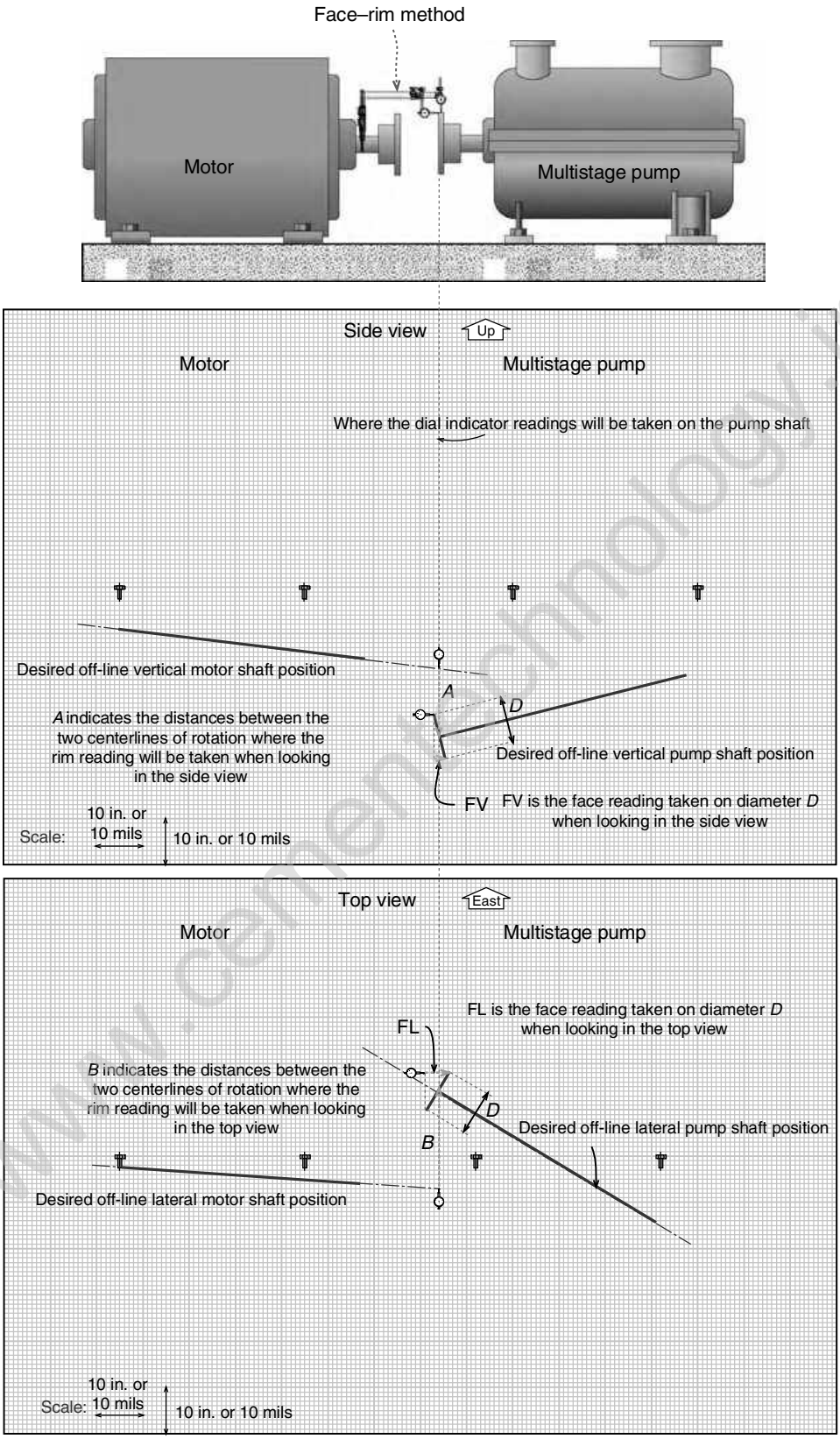


FIGURE 16.91 Example of desired off-line side and top view alignment models of a motor and a pump to calculate the shoot for face–rim measurements.

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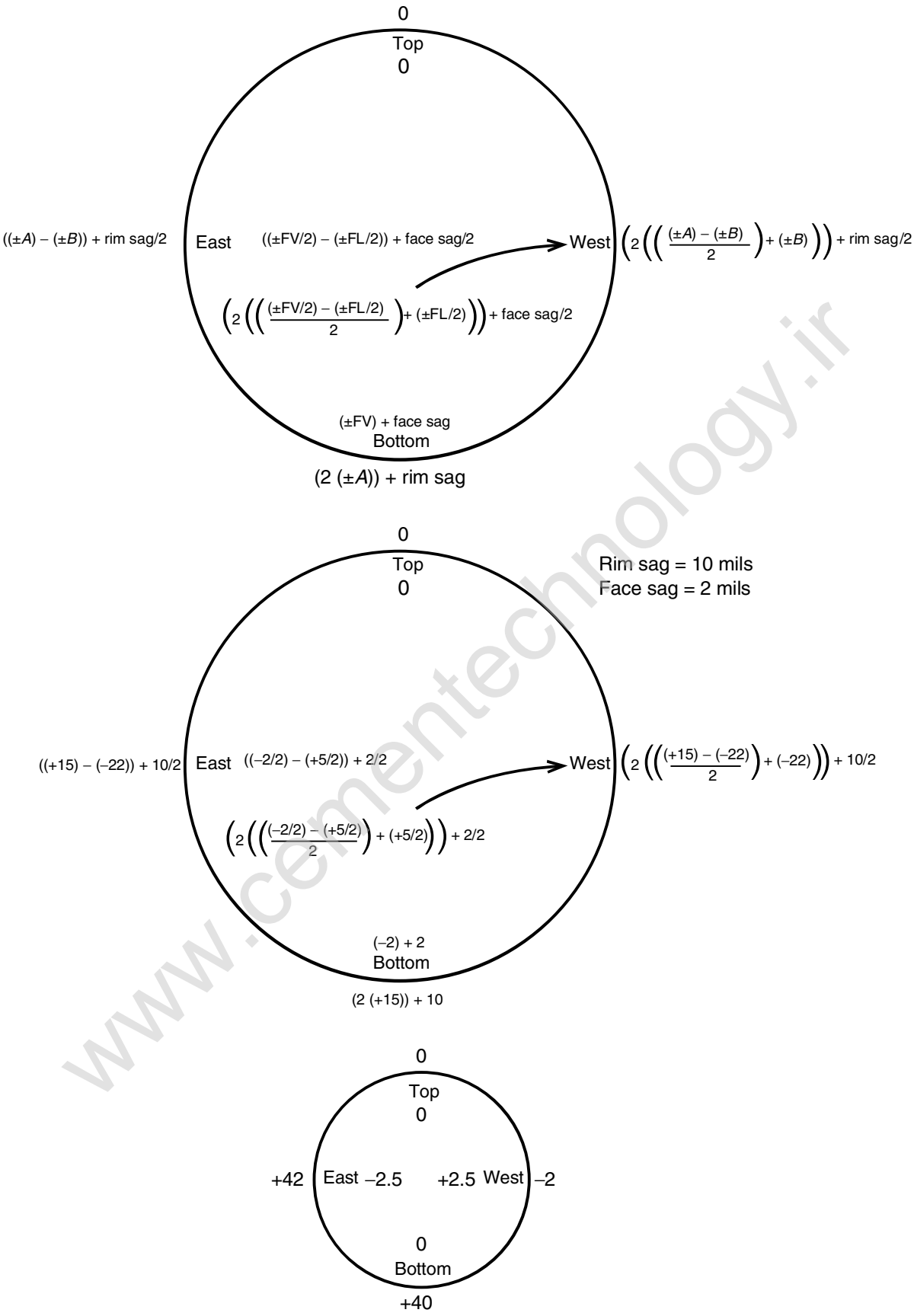


FIGURE 16.92 General equations to calculate the shoot for face–rim measurements and a sample calculation based on the shaft positions shown in Figure 16.91.

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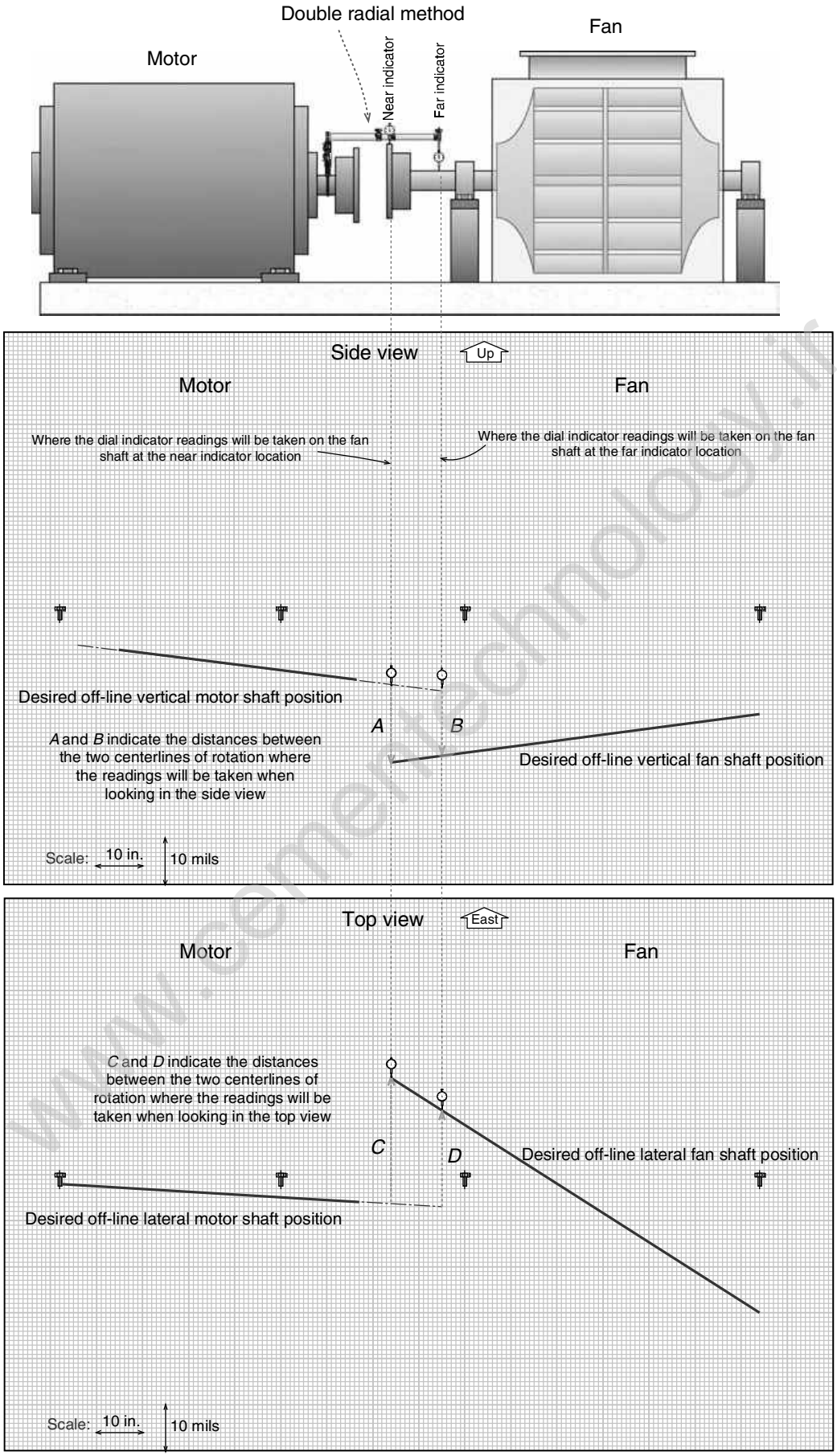


FIGURE 16.93 Example of desired off-line side and top view alignment models of a motor and a fan to calculate the shoot for double radial measurements.

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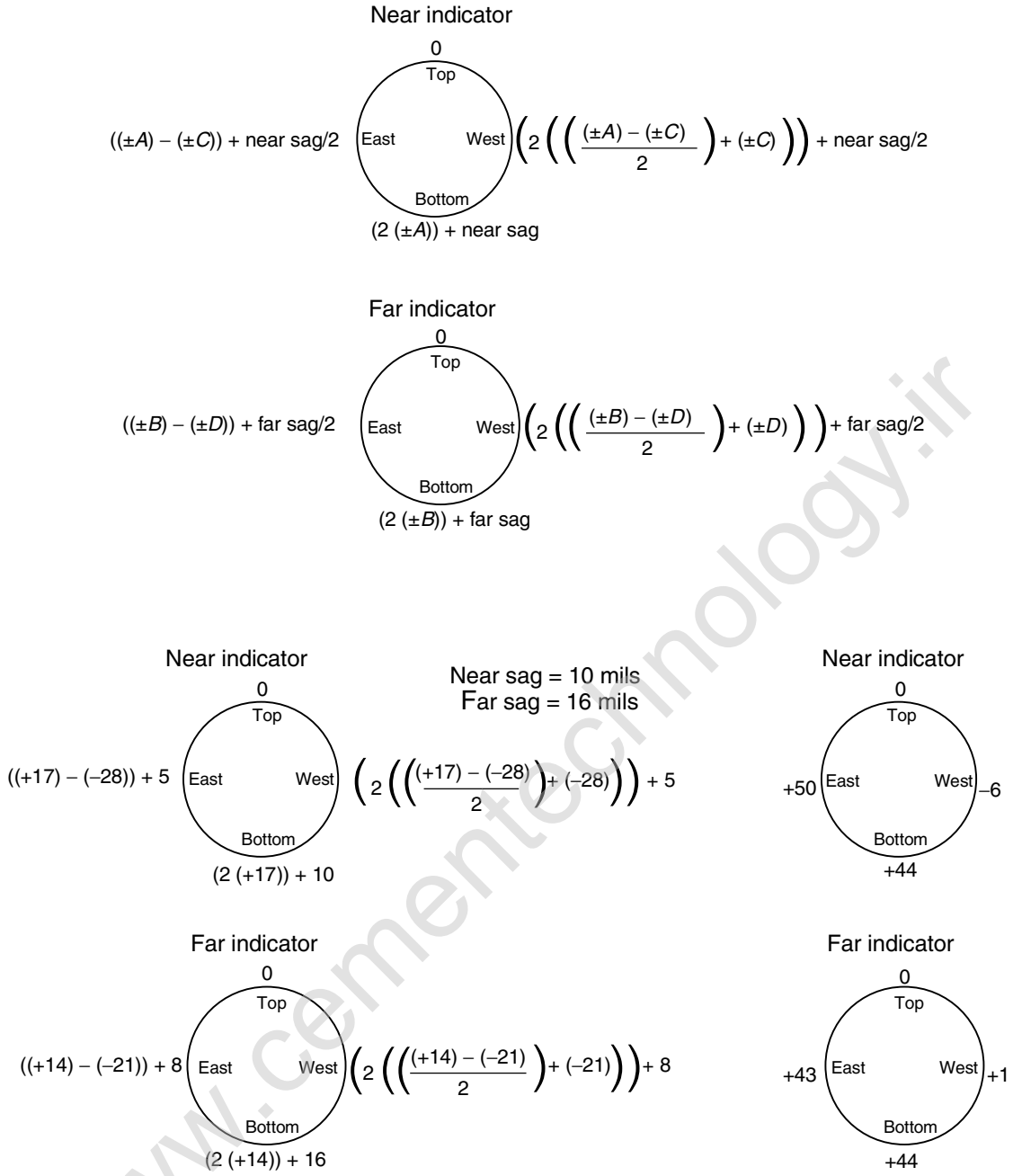


FIGURE 16.94 General equations to calculate the shoot for double radial measurements and a sample calculation based on the shaft positions shown in Figure 16.93.

Figure 16.99 shows the type of information that needs to be gathered on a two element drive train before final alignment.

In summary:

- Gather specific information on the drive train such as how the machinery will move from OL2R conditions, piping fit up problems, the total shim thickness that exists under the machinery feet, how far can each unit be moved sideways at the feet, what positions the shafts should be in when off-line, and what are the ''shoot for'' readings.
- What positions are the shafts actually in when off-line?
- How will the movement restrictions affect the final chosen alignment line?

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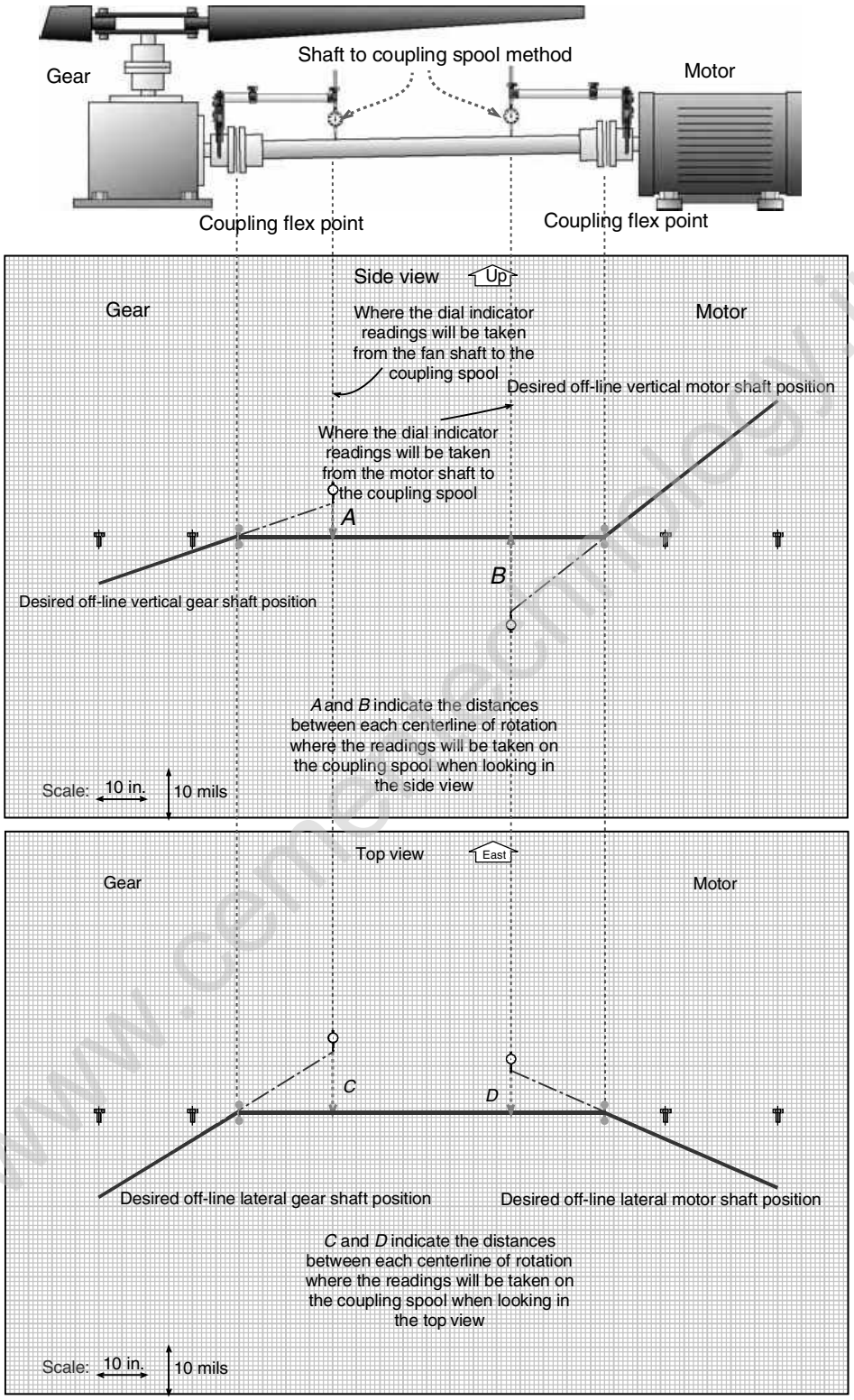


FIGURE 16.95 Example of desired off-line side and top view alignment models of a gear and a motor to calculate the shoot for shaft to coupling spool measurements.

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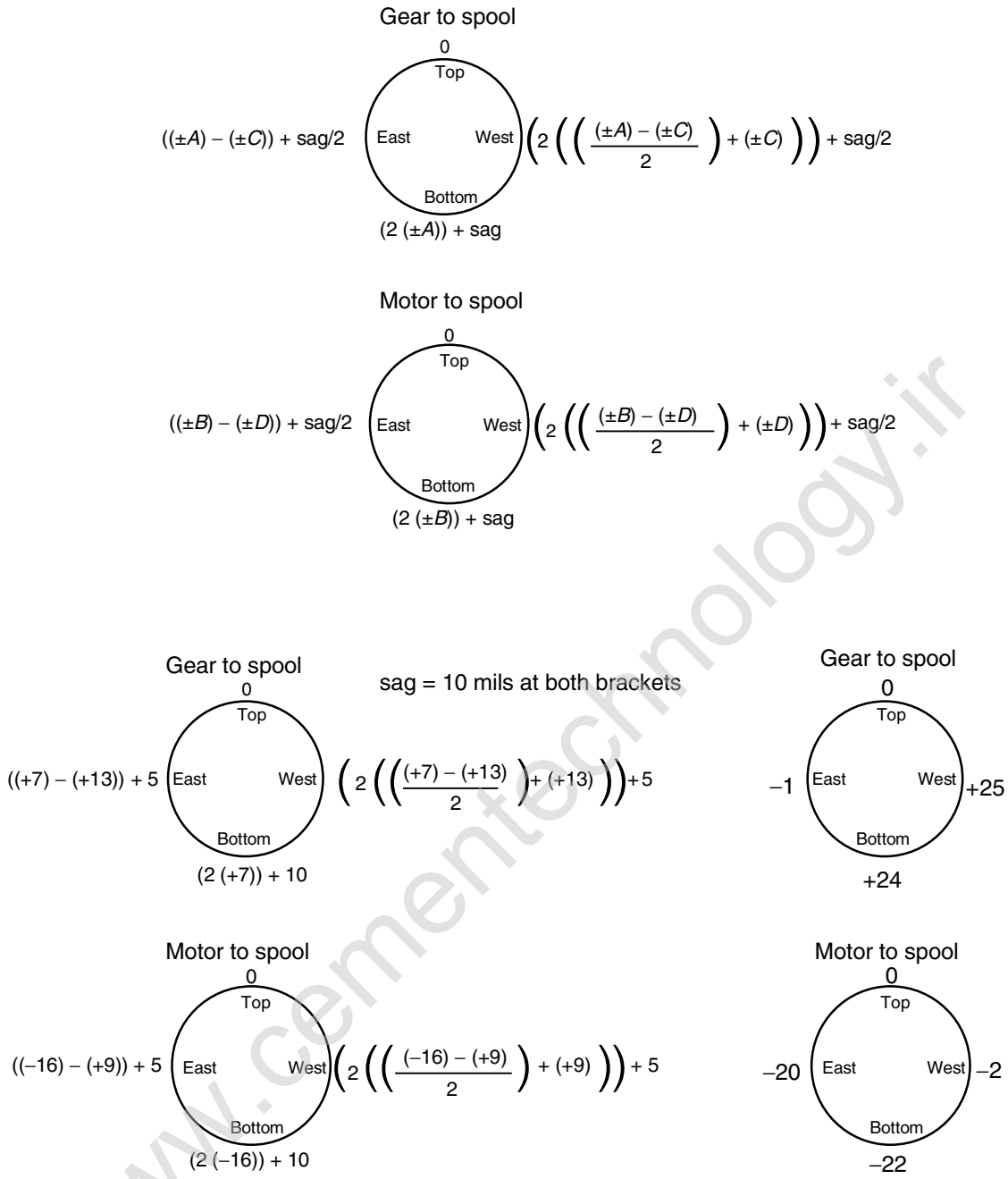


FIGURE 16.96 General equations to calculate the shoot for shaft to coupling spool measurements and a sample calculation based on the shaft positions shown in Figure 16.95.

Figure 16.100 shows the side and top view alignment models for the motor and multistage pump. The models show both the actual off-line shaft positions and the running shaft positions. To better clarify the alignment condition, Figure 16.101 shows only the running shaft positions. Superimposed on the side view, the total shim thickness that exists under each of the bolting planes have been shown and hence we know how far down each shaft could be moved. Superimposed on the top view, the lateral movement restrictions at each of the bolting planes have been shown and so we know how far to the east or west each shaft could be moved without getting bolt bound. Figure 16.101 also shows possible solutions in the side and top views by superimposing an overlay line. Bear in mind that there are other possible solutions besides the ones shown.

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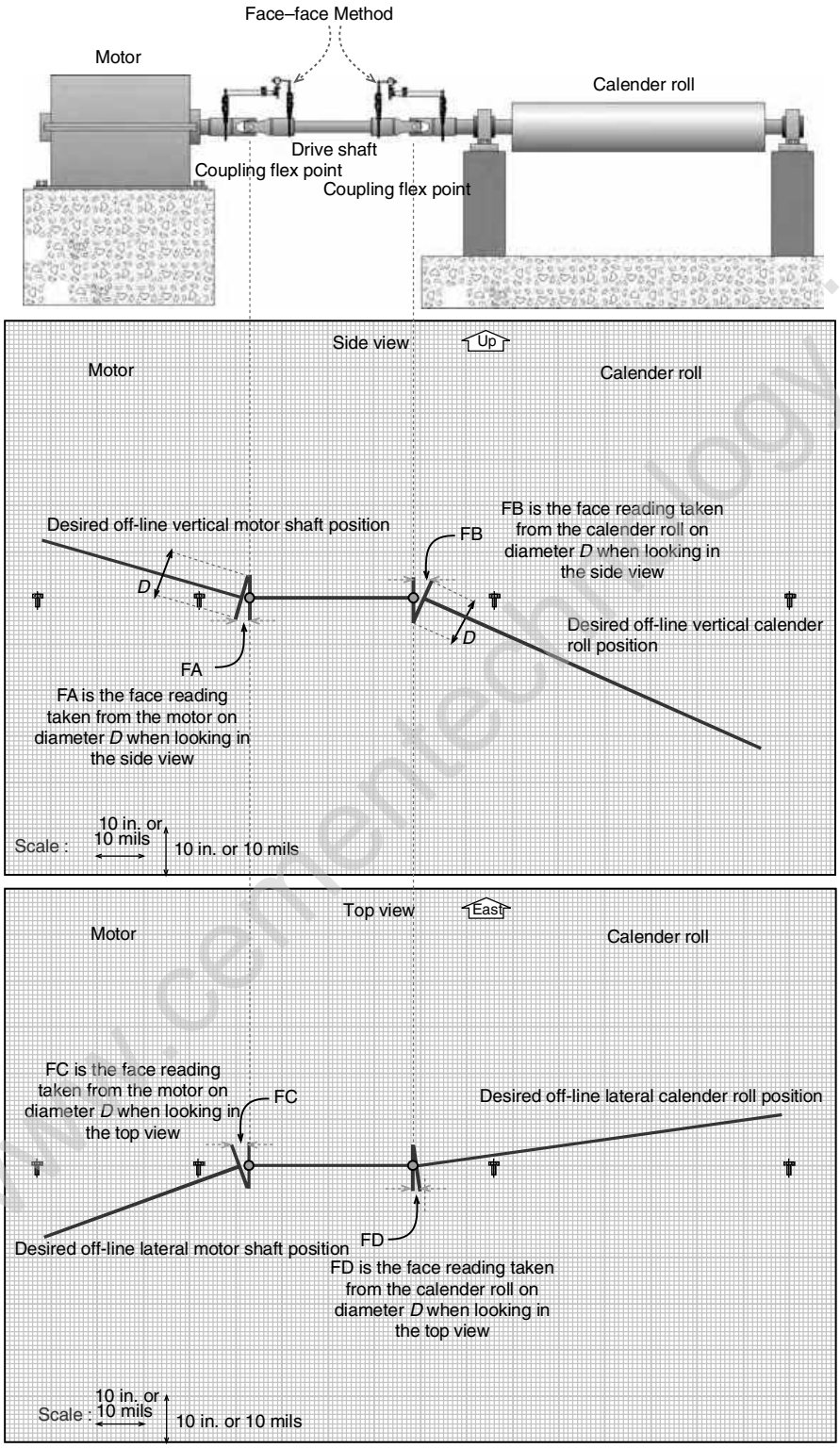


FIGURE 16.97 Example of desired off-line side and top view alignment models of a motor and a calender roll to calculate the shoot for face–face measurements.

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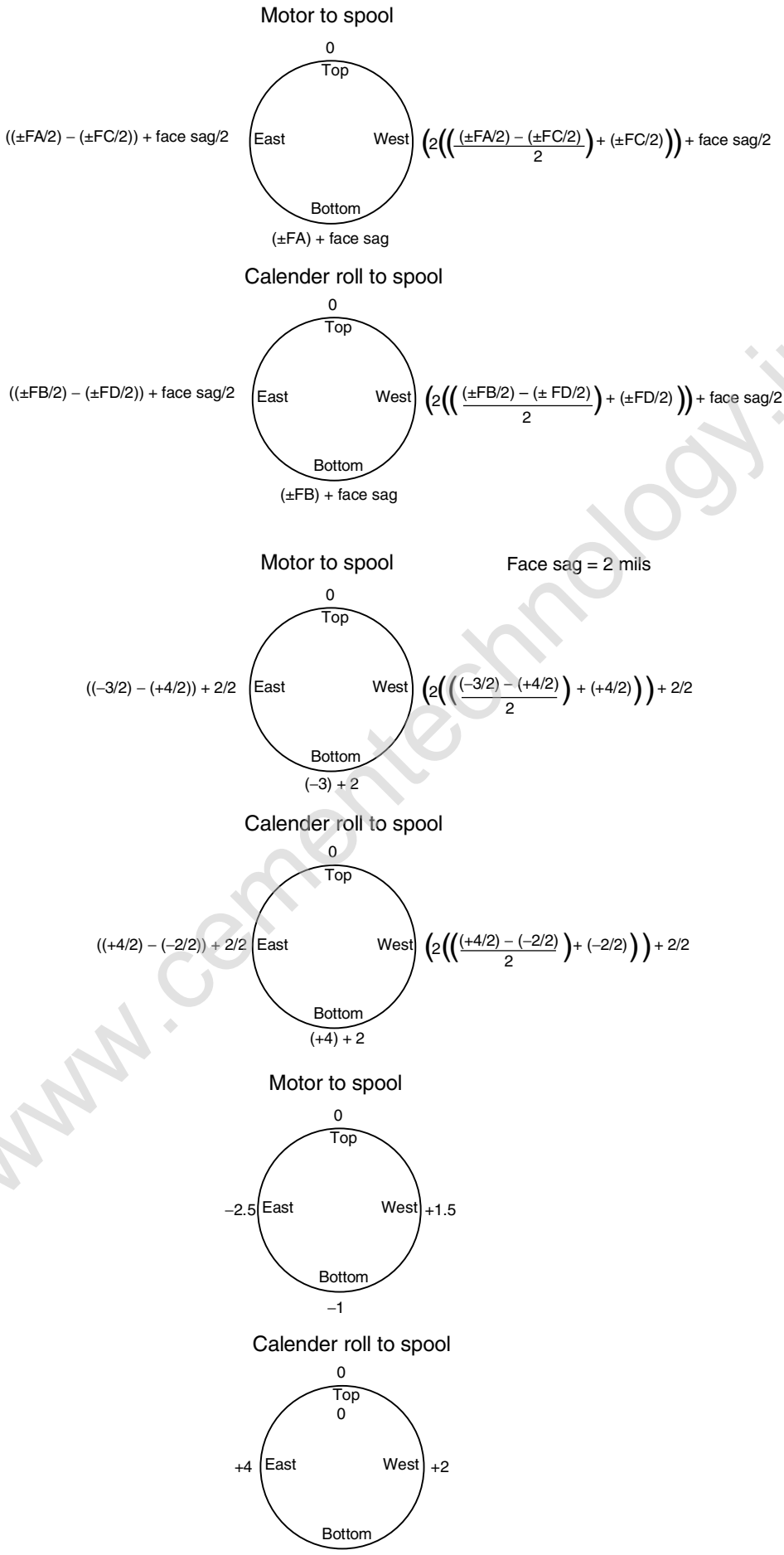


FIGURE 16.98 General equations to calculate the shoot for face–face measurements and a sample calculation based on the shaft positions shown in Figure 16.97.

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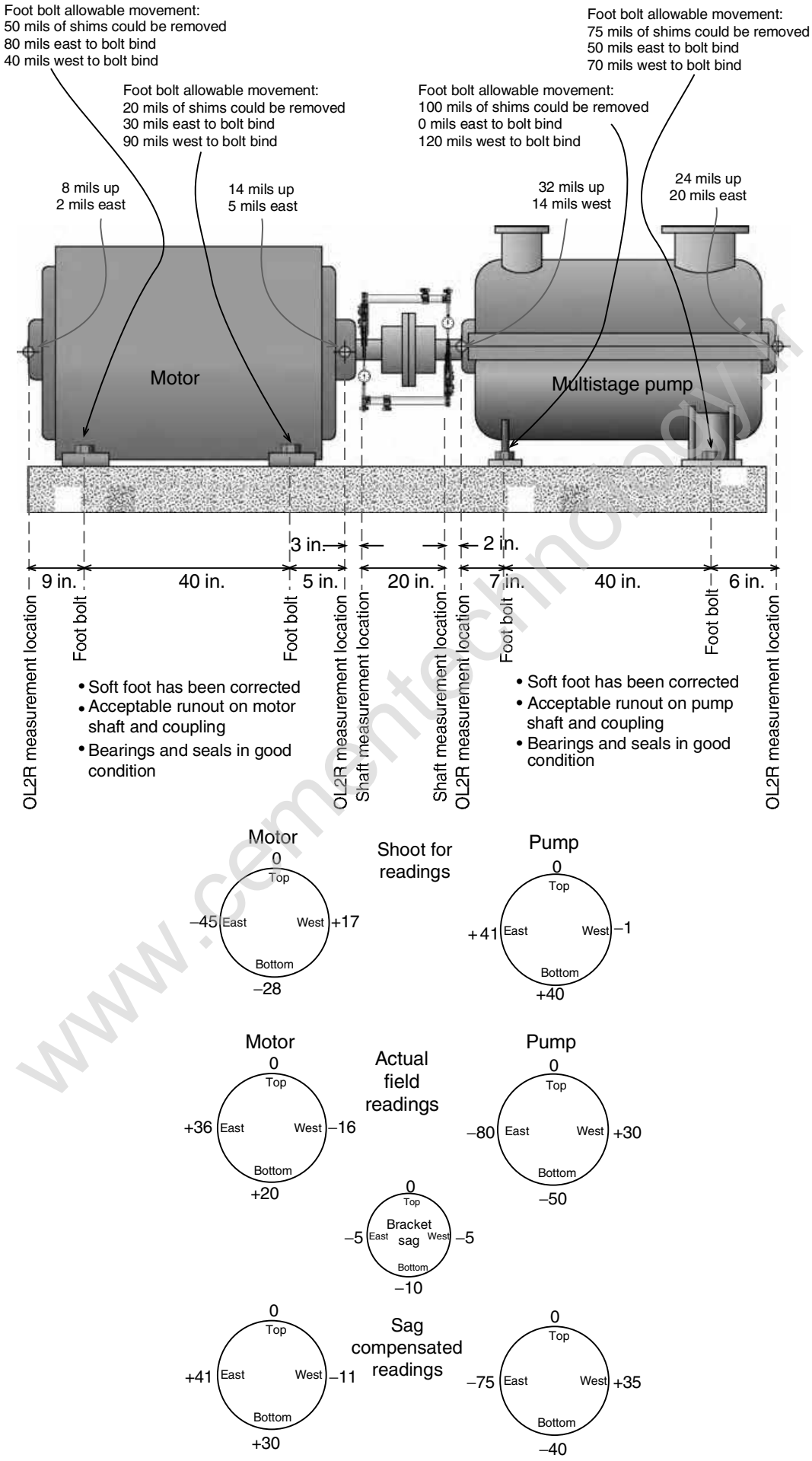


FIGURE 16.99 Typical information to be gathered on a two element drive train before final alignment.

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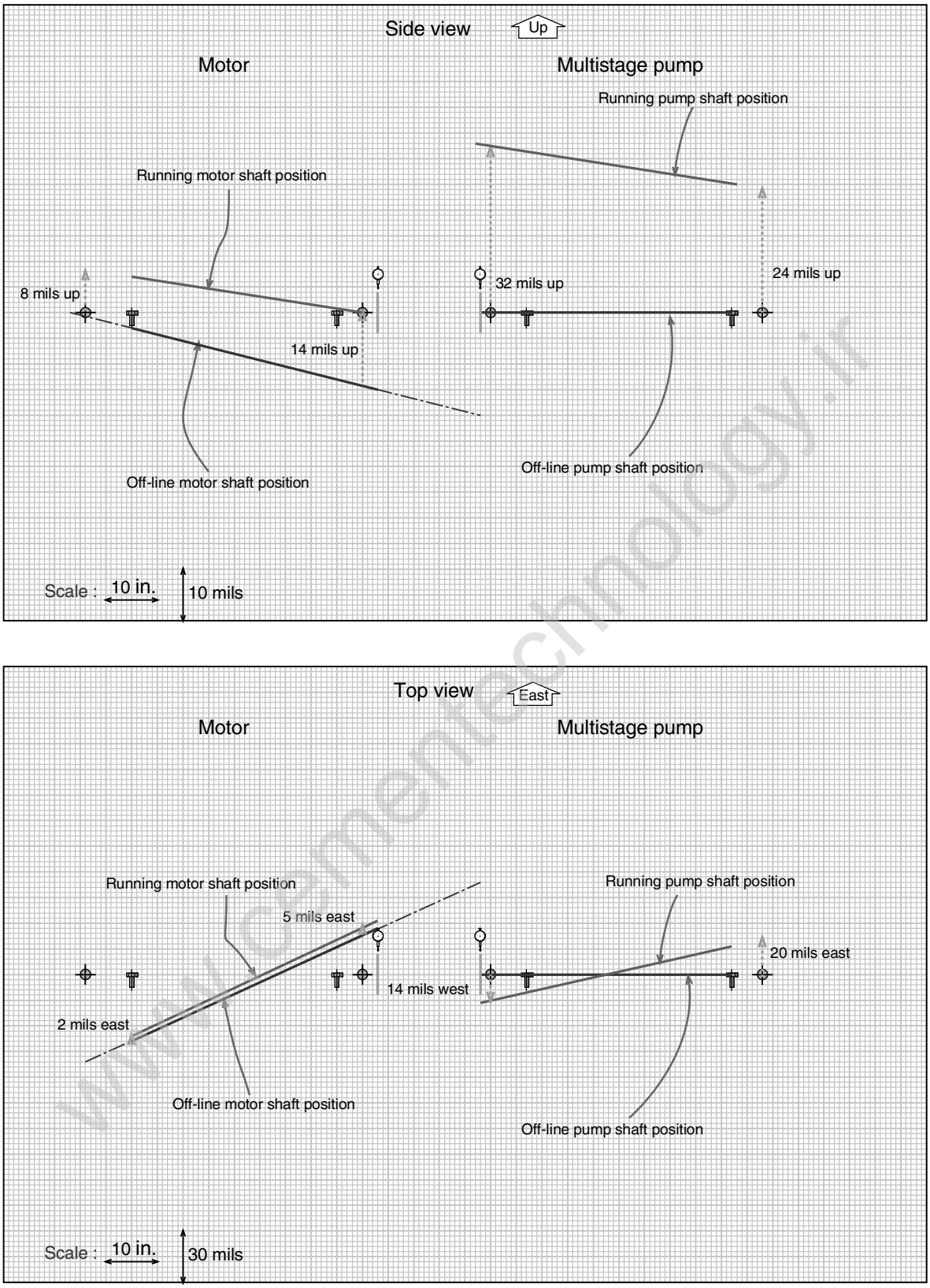


FIGURE 16.100 Side and top view alignment models for the motor and multistage pump shown in Figure 16.99.

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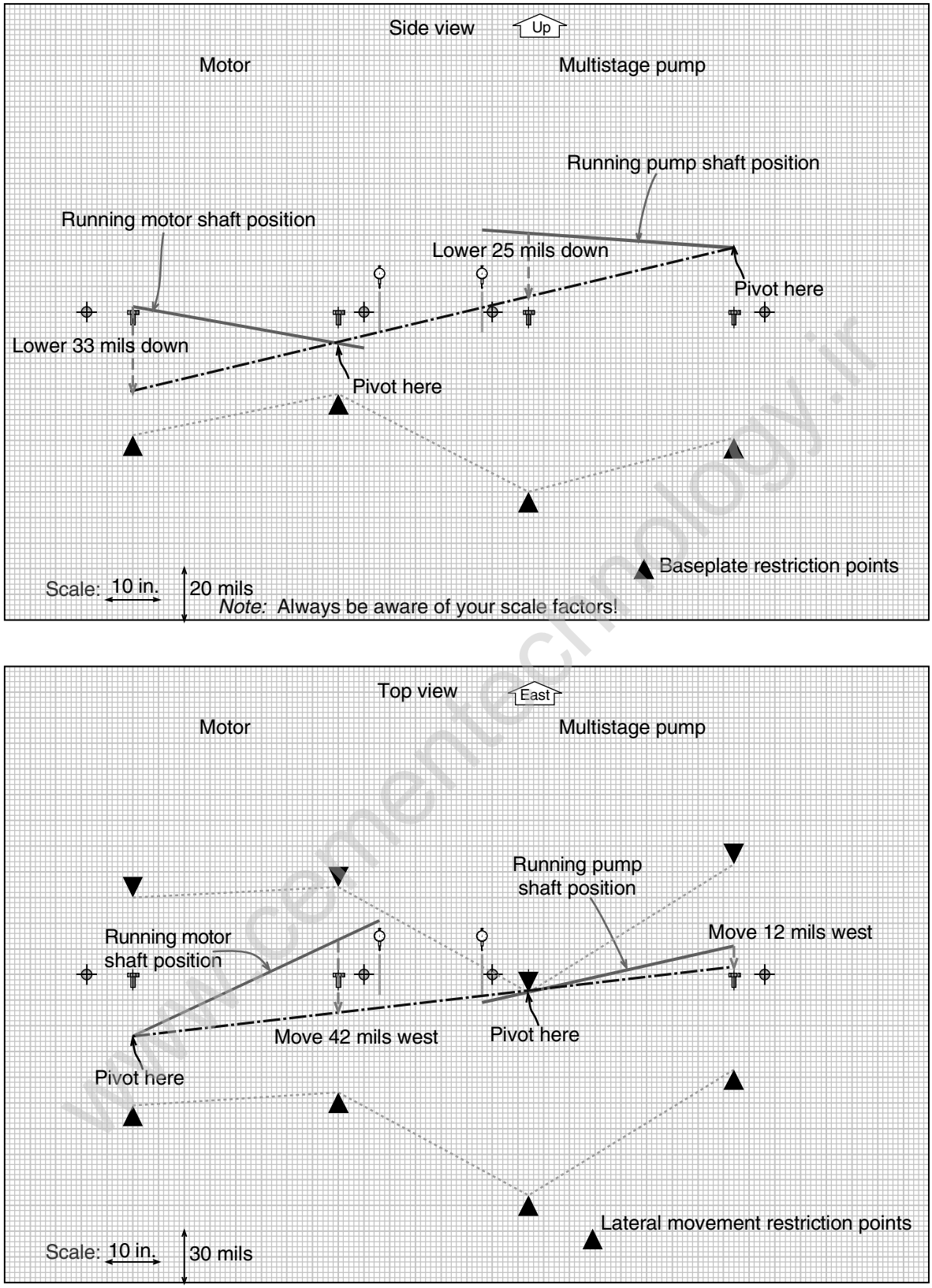


FIGURE 16.101 Side and top view alignment models for the motor and multistage pump showing the running shaft positions, boundary restrictions, and suggested possible alignment correction moves.

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As mentioned at the beginning of this chapter, very few people conduct OL2R machinery movement surveys. Assuming that your rotating machinery does not move from OL2R conditions is nothing more than burying your head in the sand and pretending that everything will automatically align itself when operating. All of this data gathering, modeling, and calculations seem quite tedious, but in many cases this extra effort may spell the difference between machinery that seems to be plagued with problems or equipment that operates successfully for long periods of time.

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17 Aligning Multiple-Element Drive Systems

The majority of rotating machinery drive systems in the world consists of two separate machines mounted on a common base. For example, there are many electric motors driving pumps and fans in virtually every industrial plant around the globe. If life were only that simple, this book might have ended here. But real life is never that easy.

There are a considerable number of rotating machinery drive systems where there are three or more machines coupled together to form what is commonly referred to as a drive "train." Some examples of typical drive trains are as follows:

- Motor–gear–pump systems
- Motor–gear–compressor systems
- High-pressure steam turbines–Intermediate-pressure steam turbines–Low-pressure steam turbines–generators–exciter (HP–IP–LP–Generator systems)
- Motor–generator–generator–generator systems (sometimes called MG sets)
- Steam turbine–Gas Turbine–Gear–Low stage compressor–Gear–High stage compressor (commonly found in refining and chemical industries)
- Drive motor–clutch–gear–pinion–brake–right-angled gear–clutch–slow-speed turning motor (commonly found in cement plants on kiln or ball mill drives)
- Rack and pinion gear set–pinion drive shaft–right-angled gear–motor–pinion drive shaft–rack and pinion gear set (found in automotive industries at vehicle assembly plants on "body drop" overhead cranes and various other industries having cranes that translate in two directions)

The variation and mixture of rotating machinery drive systems in industry is as diverse as one could possibly imagine. These machine cases can be horizontally mounted and coupled end-to-end like a railroad train or they can be arranged at a right angle and some even have been set up in a U-shaped or zigzag configuration.

In virtually every case, these drive trains are very expensive and frequently the heart and soul of the operation of the plant. Some multiple-element drive trains fall in the small horsepower range (10–500 hp) but there are a large percentage of these drive trains that are 500–50,000 hp costing millions of dollars. Consequentially, they are the most critical pieces of machinery in the operation and the ones that seem to get everyone very nervous when something goes wrong. Sometimes, small problems with these drive trains turn out to be just minor distractions but when big problems occur, these systems turn out to be absolute nightmares for the people involved in correcting the malady. Most people would just as soon have pleasant dreams than nightmares, so a lot of effort is expended toward getting the situation fixed correctly so that major problems occur infrequently.

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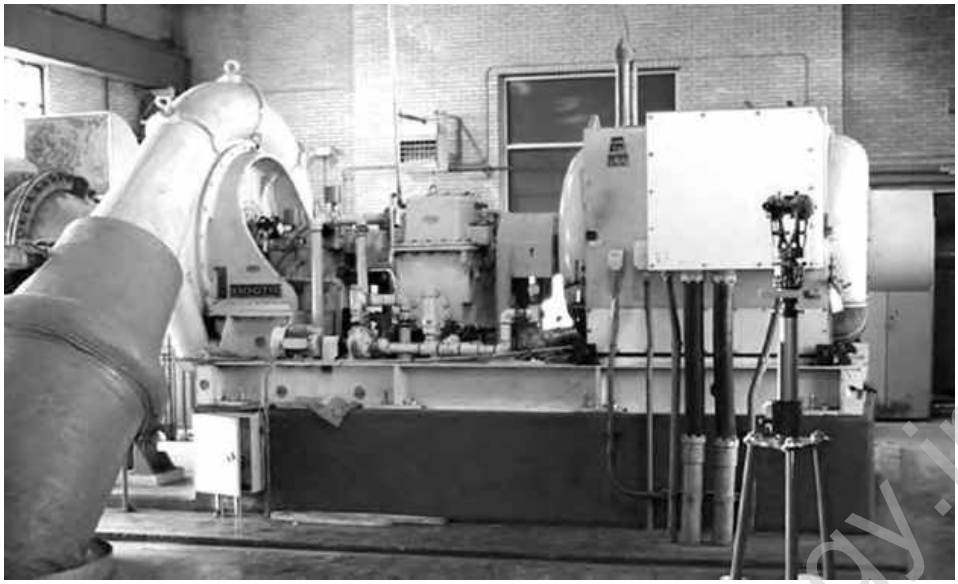


FIGURE 17.1 Motor–gear–blower drive system.

It is highly recommended that you become very adept at aligning two-element drive systems before you try to tackle a multiple-element drive train. Every shred of knowledge you have gained in your alignment experiences will be tested to your limits when one of these systems has to be installed or completely rebuilt.

17.1 MULTIPLE-ELEMENT DRIVE TRAIN ALIGNMENT LAWS

Below are some suggestions and guidelines that may help in successfully aligning a multiple-element drive system:

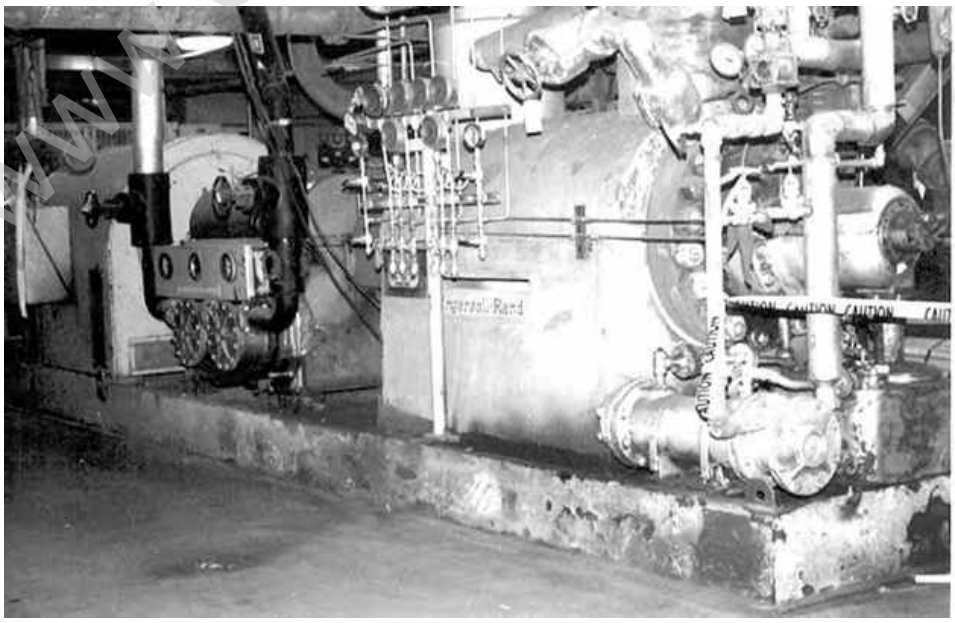


FIGURE 17.2 Motor–fluid drive–pump drive system.

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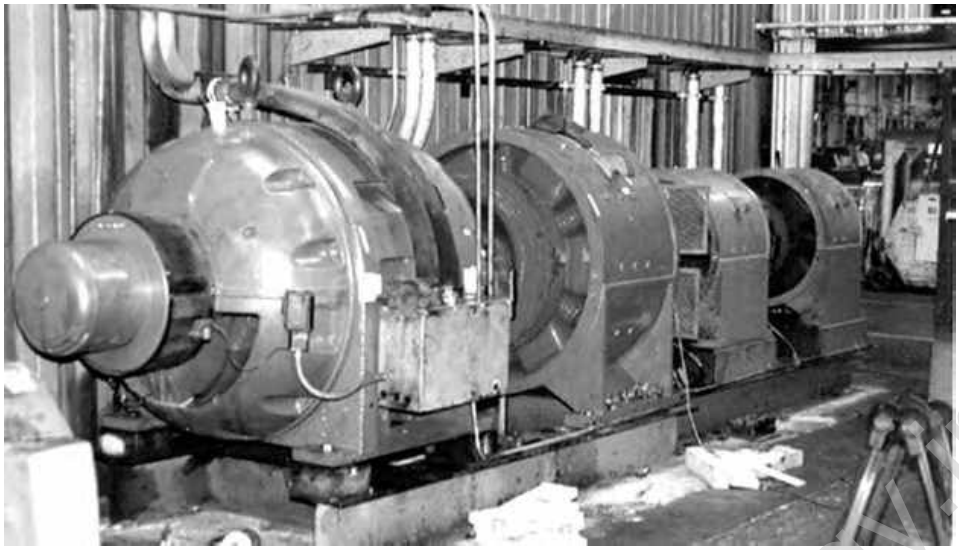


FIGURE 17.3 Motor–generator set.

- Do not move any machine case until you know where all of the centerlines of rotation are.
- Every machine case is movable. Some machines are just more difficult to move than others.
- Once you have determined where each centerline of rotation on every machine is, define the allowable movement envelope before you begin repositioning the machinery.
- Make sure you know (or at least have a good guess) where the centerlines of rotation will move to when the machines are running under normal operating conditions. This subject was covered in Chapter 16.
- Calculate how much time you think it will take to align the drive train and multiply this number by 1.5.
- Never consider a drive train fully functional until you surpass 100 h of successful operation.

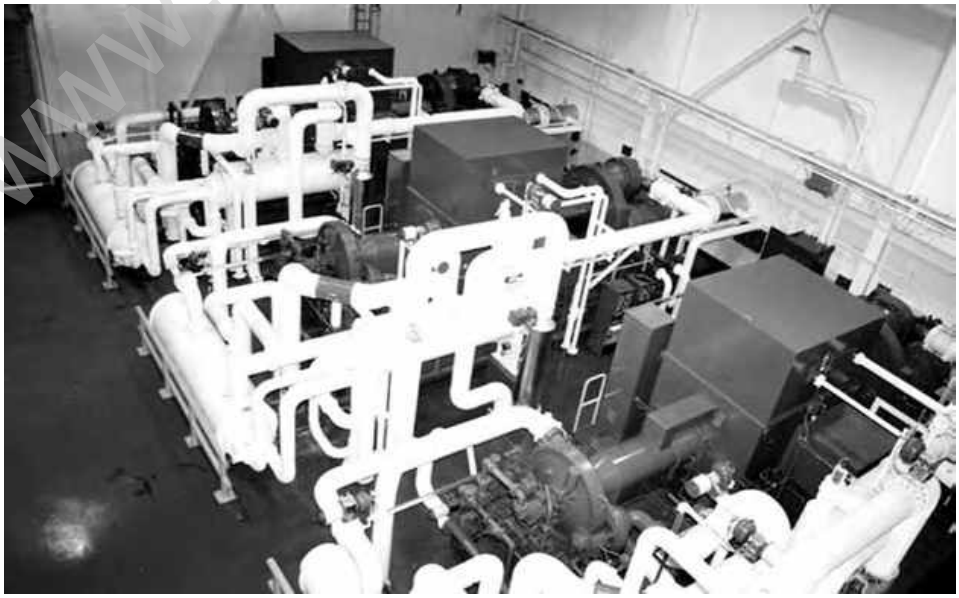


FIGURE 17.4 Multiple-stage compressor and motor drive systems.

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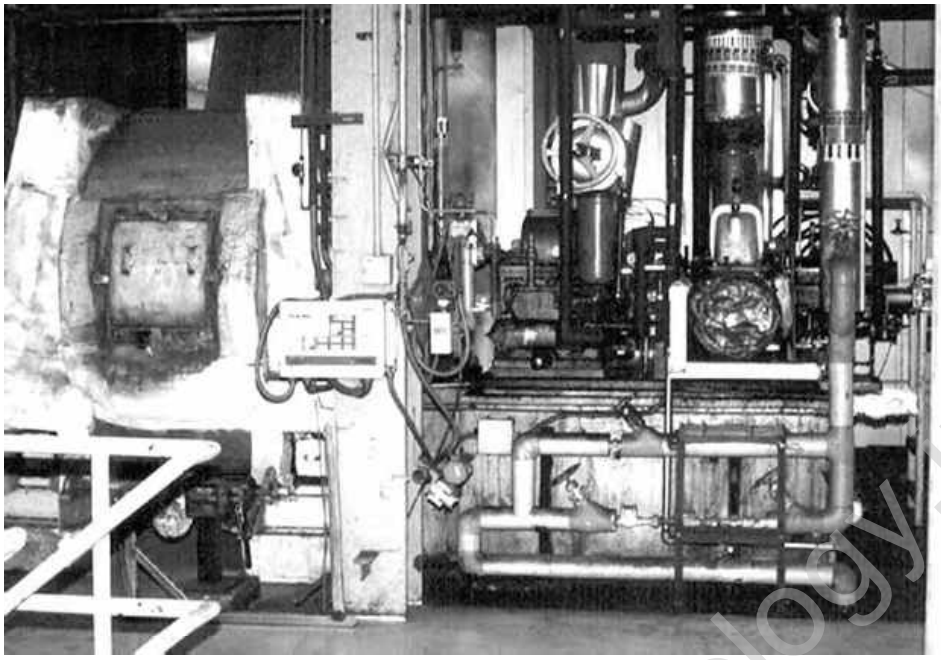


FIGURE 17.5 Fan–gear–steam turbine drive system.

17.2 MULTIPLE-ELEMENT DRIVE TRAIN: GRAPHING AND MODELING TECHNIQUES

The graphing and modeling shaft alignment techniques covered in Chapter 8 through Chapter 14 illustrated the effectiveness of using alignment models to find the most reasonable solution to reposition the rotating machinery casings to achieve collinear shaft operation. The advantages of graphing or modeling your alignment problems are as follows:

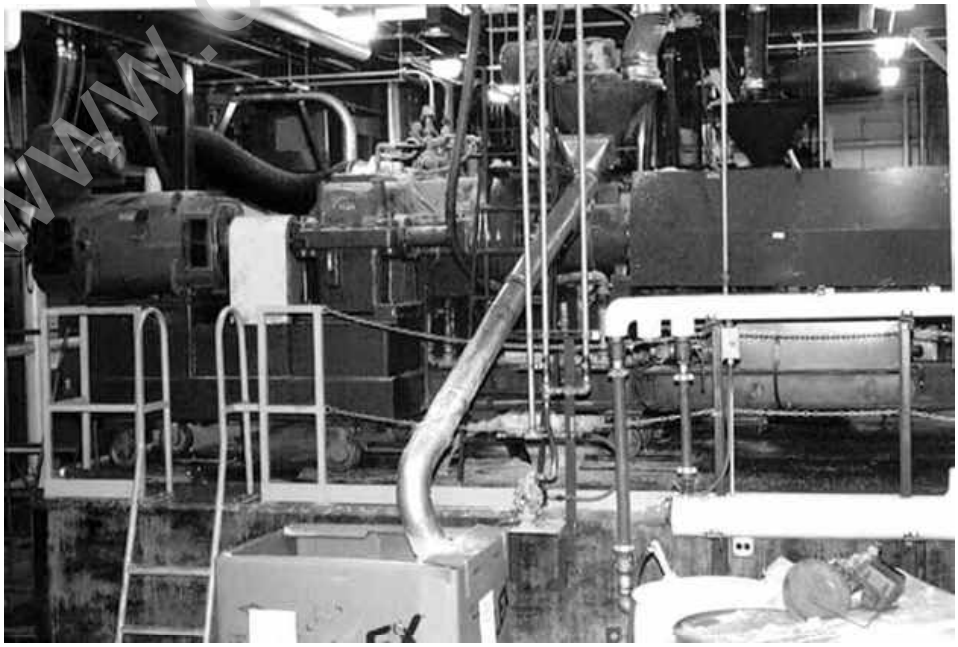


FIGURE 17.6 Motor–gear–extruder drive system.

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- It is an accurate visual representation of the centerlines of rotation of the machinery you are attempting to align.
- The graphs can be generated by people who do not have extensive backgrounds in geometry, trigonometry, or software programming.
- It is inexpensive as all you need are paper, pencil (sometimes with a large eraser), and a straightedge.
- You can use the model to show the possible movement envelope in the up-and-down and side-to-side directions to prevent wasting time on grinding baseplates or machine casings or cutting bolt shanks down or filing foot bolt holes open.
- By using an overlay line there can be an infinite number of possible alignment solutions available (again, some of the solutions make sense and others do not).
- It is an excellent mechanism to use when explaining to your coworkers or managers what your plan of action is and why you chose the solution you did even if they may not understand exactly how the graph was generated.
- The model can be used to determine if you are (or are not) within acceptable alignment tolerances.
- The model is a permanent record of the alignment that can be kept for future reference that can show as found and final alignment positions or any alignment conditions in between.

It should be somewhat obvious by this time that if you can show the relative positions of two machinery shafts on an alignment graph or model, why cannot you show 3 machinery shafts or even 10? Well you can, and here is how.

17.3 MULTIPLE-ELEMENT DRIVE TRAIN MODELING—ONE SET OF SHAFTS AT A TIME

Figure 17.7 shows a multiple-element drive system consisting of a motor, gear, and pump assembly. A set of reverse indicator readings were captured between the motor and the gear and also between the gear and the compressor. Two sheets of graph paper were then used to

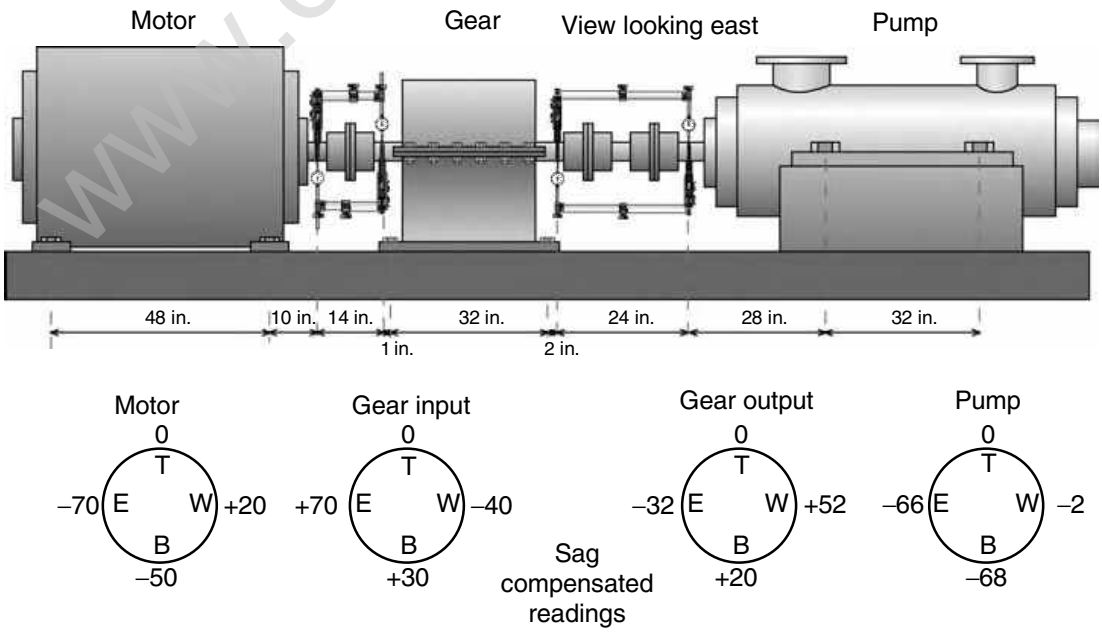


FIGURE 17.7 Motor–gear–pump drive system layout.

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construct the relative positions of the motor and the gear shafts in the side and top views using the point-to-point reverse indicator modeling technique as shown in Figure 17.8. Two more sheets of graph paper were then used to construct the relative positions of the gear and the pump shafts in the side and top views using the point-to-point reverse indicator modeling technique as shown in Figure 17.9. Bear in mind that gearboxes obviously consist of more than one shaft, but because these shafts are housed in a single case, the input and output shafts can be shown on the alignment model as a single centerline of rotation. Notice that the gear shaft position appears on the side and top view models in both Figure 17.8 and Figure 17.9. The goal here is to generate an alignment model with all three shafts on the same graph paper.

To accomplish this, let us determine what moves need to be made to align the motor to the gear and the pump to the gear. Figure 17.10 shows the same shaft positions as in Figure 17.8 except here the centerline of rotation of the gear input shaft has been extended toward the motor shaft. The required moves to bring the motor into alignment with the gear in the up or down direction is to remove 63 mils of shims under the outboard feet of the motor and remove 31 mils of shims under the inboard feet of the motor. The required moves to bring the motor into alignment with the gear in the lateral direction is to move the outboard end of the motor 10 mils to the east and the inboard end 37 mils to the east.

Figure 17.11 shows the same shaft positions as in Figure 17.9 except here the centerline of rotation of the gear input shaft has been extended toward the pump shaft. The required moves to bring the pump into alignment with the gear in the up or down direction is to remove 93 mils of shims under the outboard feet of the pump and remove 63 mils of shims under the inboard feet of the pump. The required moves to bring the pump into alignment with the gear in the lateral direction is to move the outboard end of the motor 220 mils to the east and the inboard end 120 mils to the east.

The dimensions of all three machines have been scaled on the side and top view alignment models in Figure 17.12. The gear shaft was then placed directly on top of the graph paper centerline. The required vertical moves to bring the motor into alignment with the gear shaft (63 mils down at the outboard and 31 mils down at the inboard end) were then transferred from Figure 17.10. The required lateral moves to bring the motor into alignment with the gear shaft (10 mils east at the outboard and 37 mils east at the inboard end) were also transferred from Figure 17.10.

In Figure 17.13, the required vertical moves to bring the pump into alignment with the gear shaft (93 mils down at the outboard and 63 mils down at the inboard end) were then transferred from Figure 17.11. The required lateral moves to bring the pump into alignment with the gear shaft (220 mils east at the outboard and 120 mils east at the inboard end) were also transferred from Figure 17.11. Figure 17.13 now shows the alignment condition of all three shafts in both the side and top views.

17.4 MULTIPLE-ELEMENT DRIVE SYSTEM GRAPHING—MODELING ALL THE SHAFTS AT ONE TIME

Graphing two shafts at a time can be somewhat tedious and a slight waste of paper. Six alignment models were needed to generate the multiple-element alignment models shown in Figure 17.13. If we attempt to graph the motor, gear, and pump shafts on one alignment model using the point-to-point reverse indicator modeling technique, there would be two gear shafts appearing on the model, which cannot be true. Here is the advantage of the line to points reverse indicator modeling technique. By drawing one shaft directly onto the graph paper centerline, we can then use it as a reference to draw the other shafts.

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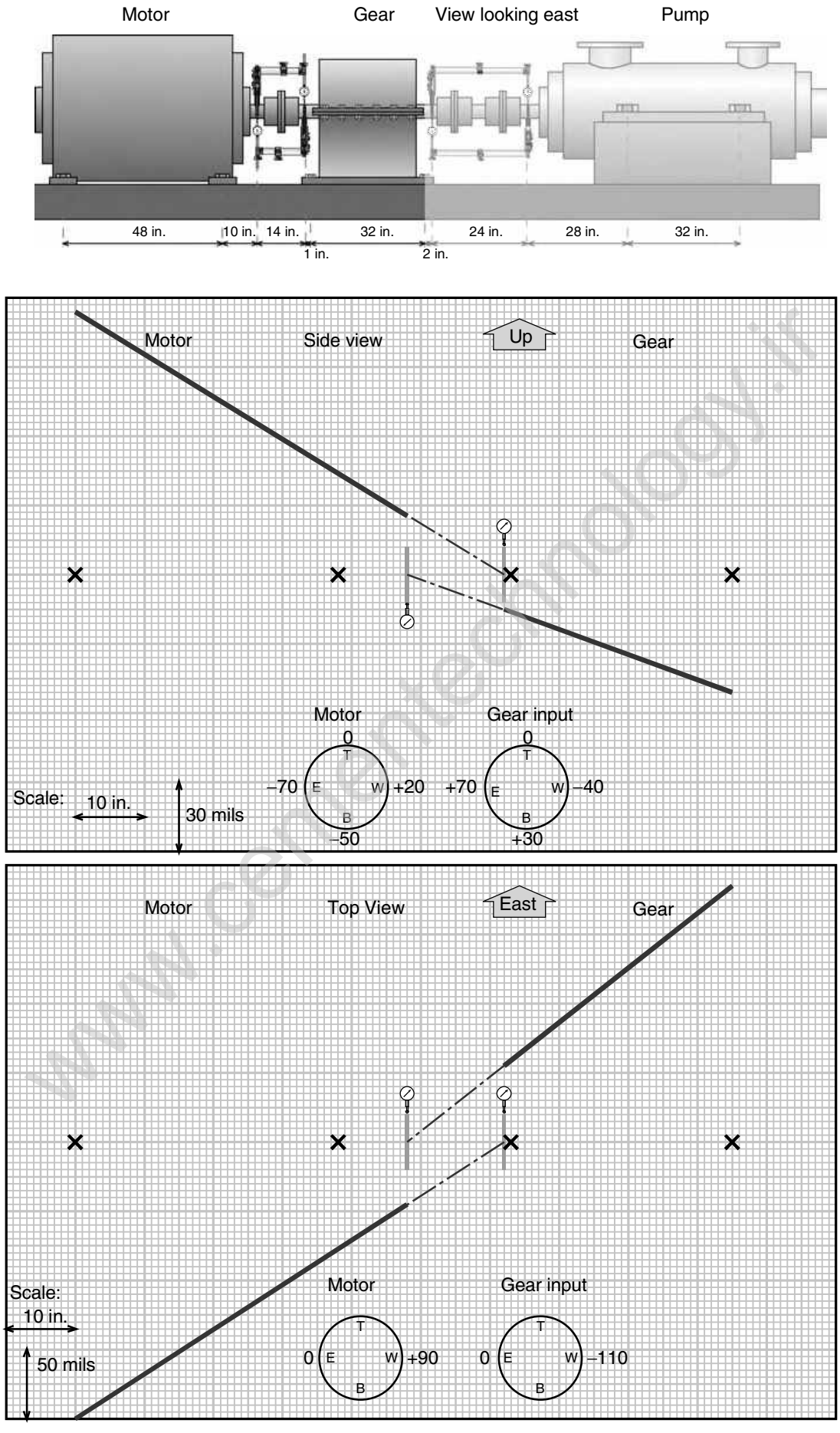
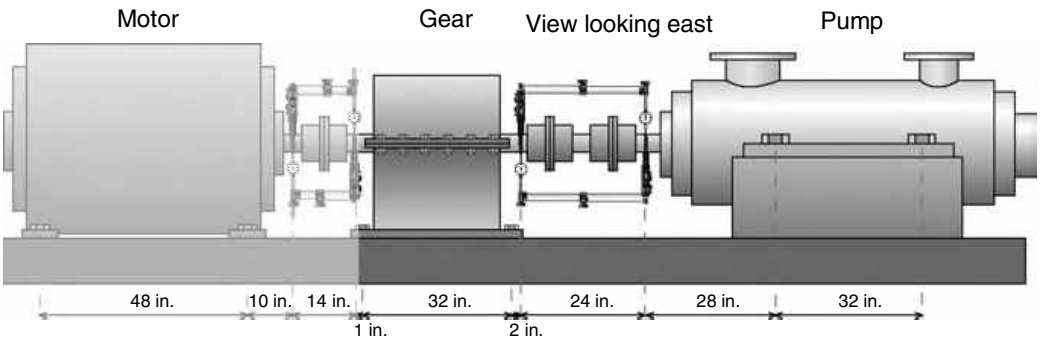
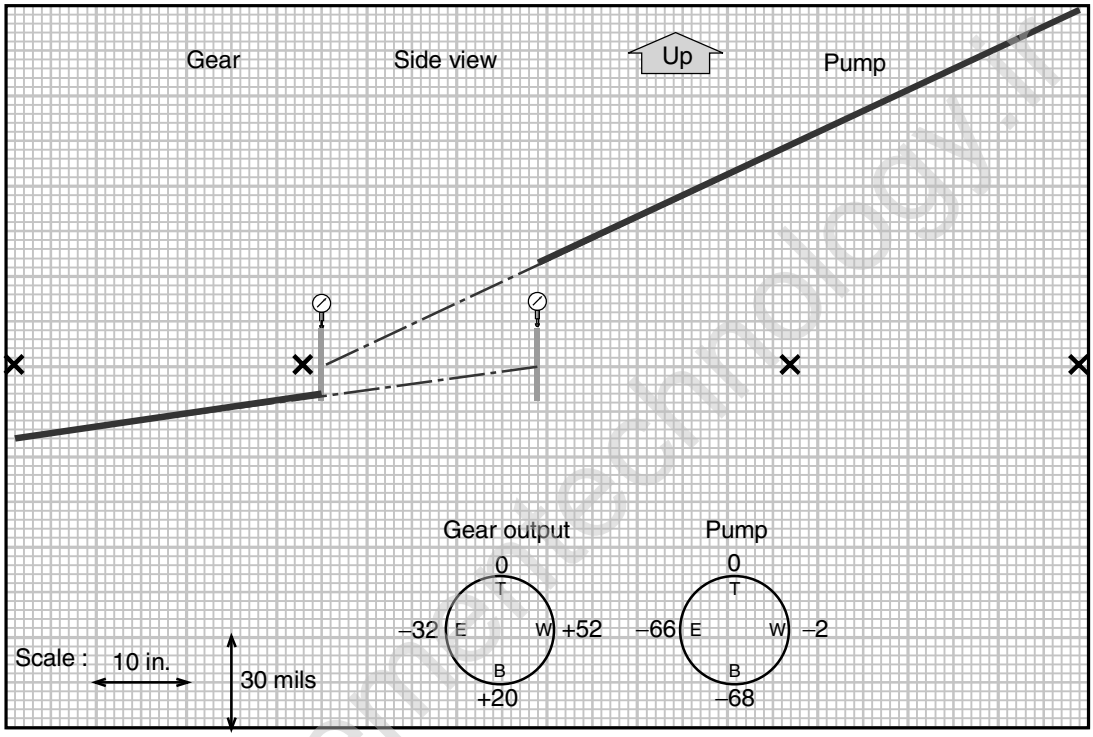


FIGURE 17.8 Side and top view alignment models of just the motor and the gear.

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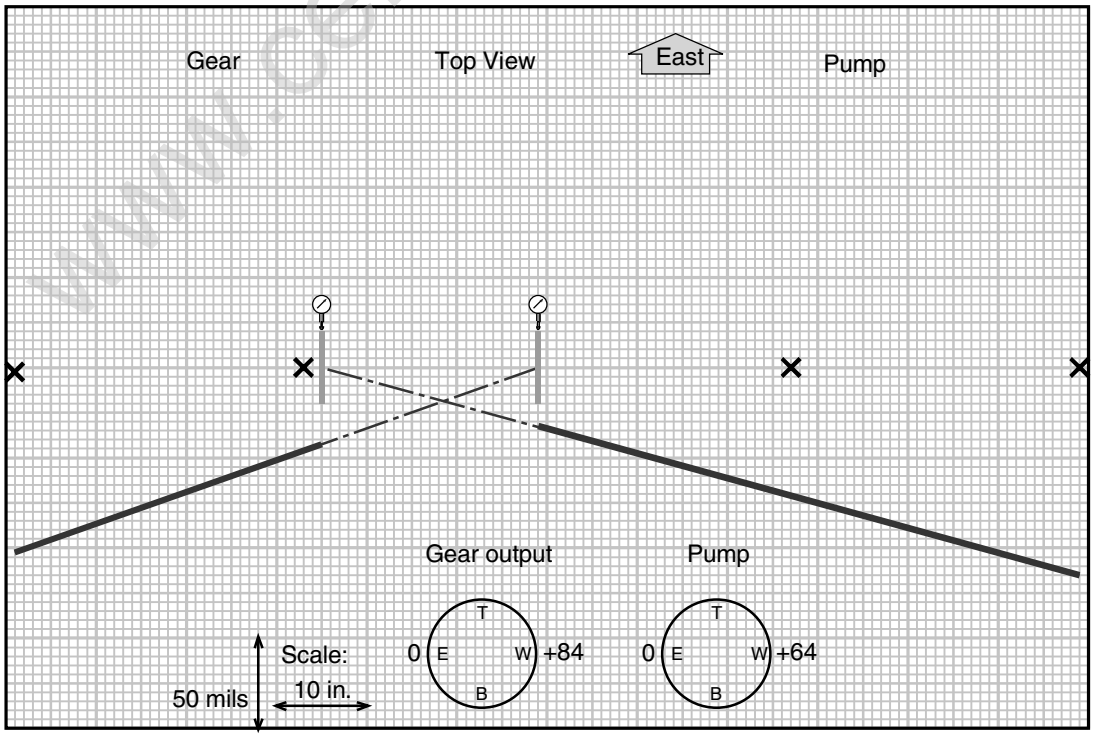


FIGURE 17.9 Side and top view alignment models of just the gear and the pump.

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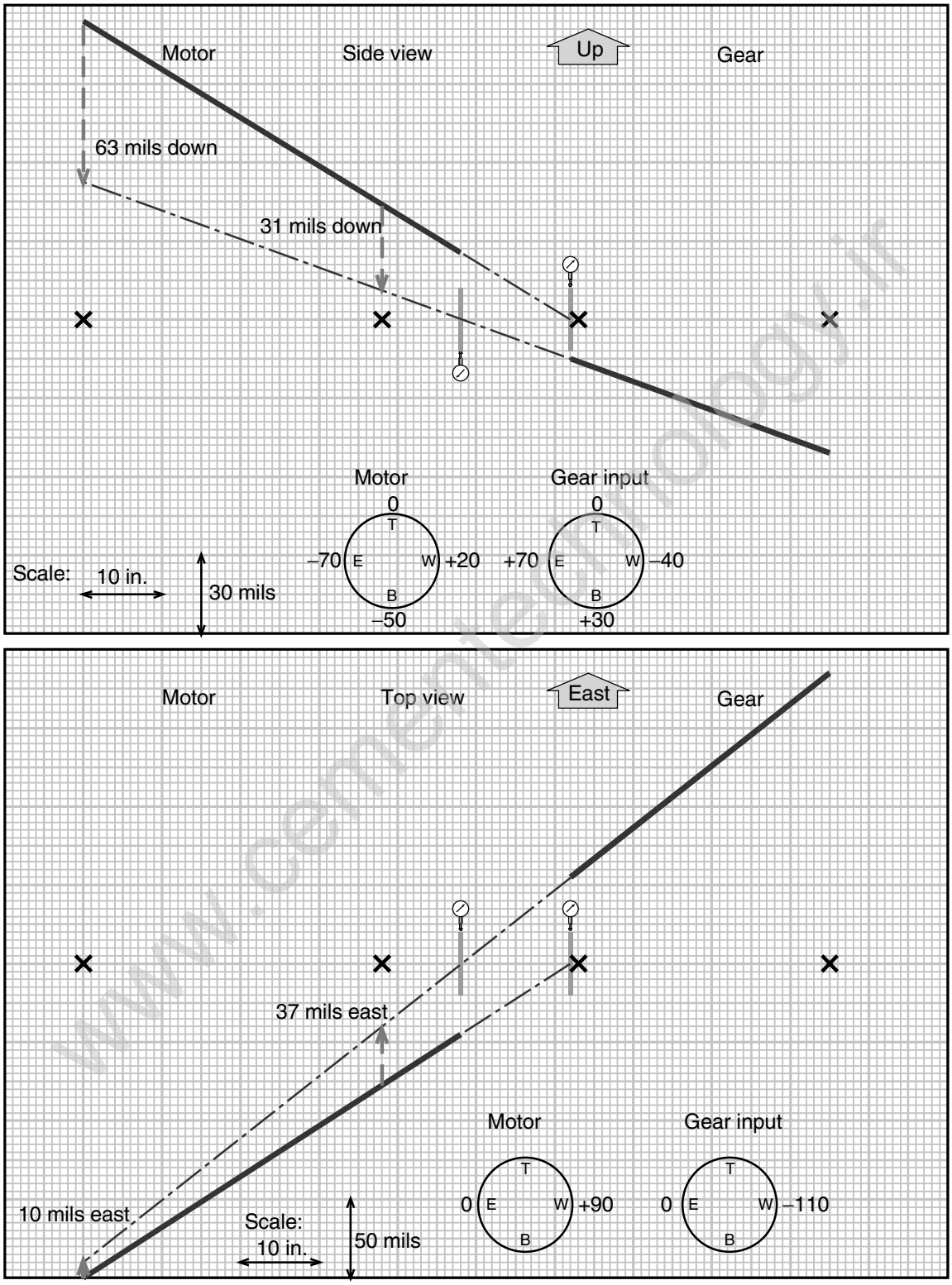


FIGURE 17.10 Required moves to align the motor to the gear.

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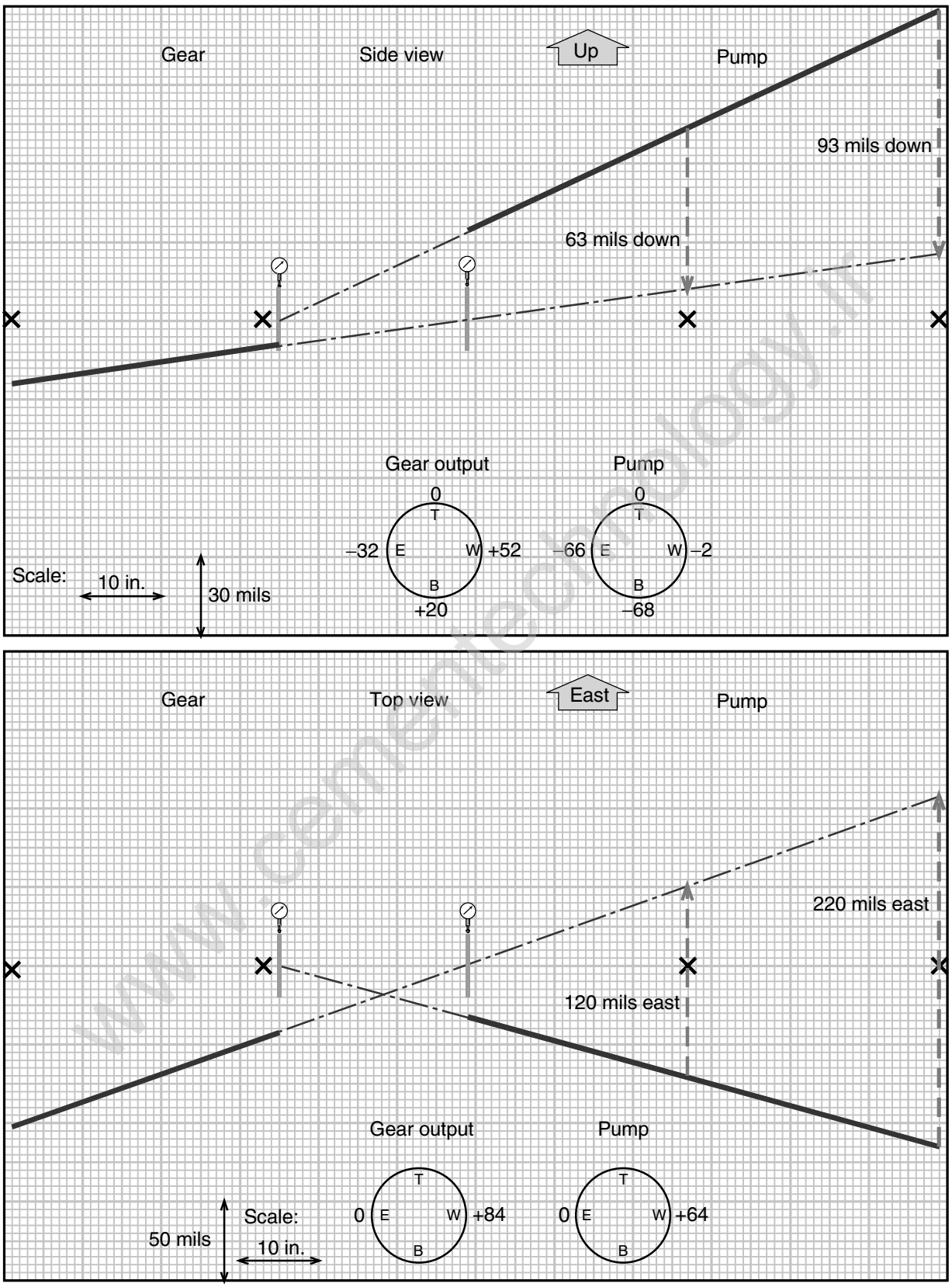


FIGURE 17.11 Required moves to align the pump to the gear.

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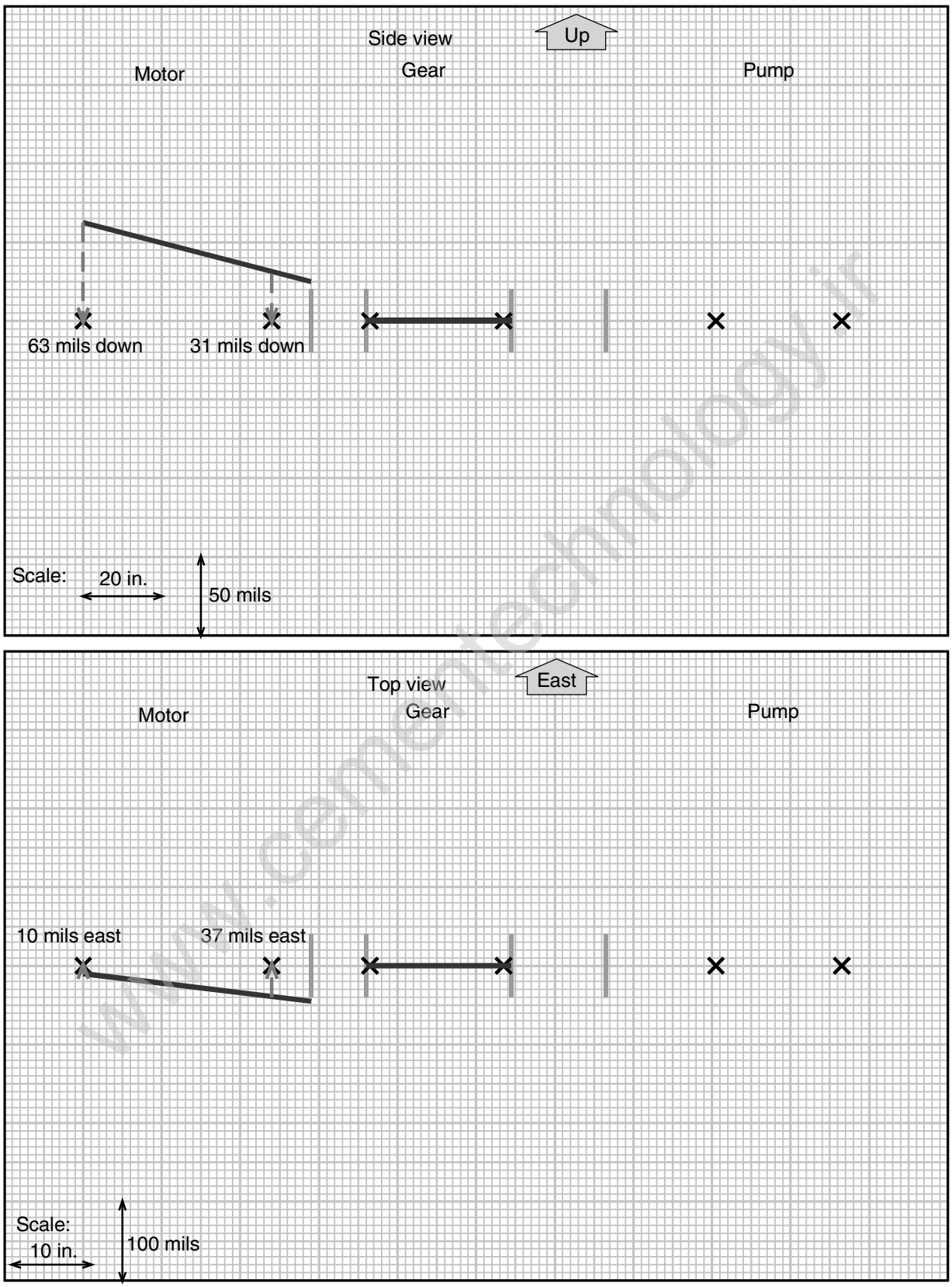


FIGURE 17.12 Three-element alignment model showing the gear and motor shafts only.

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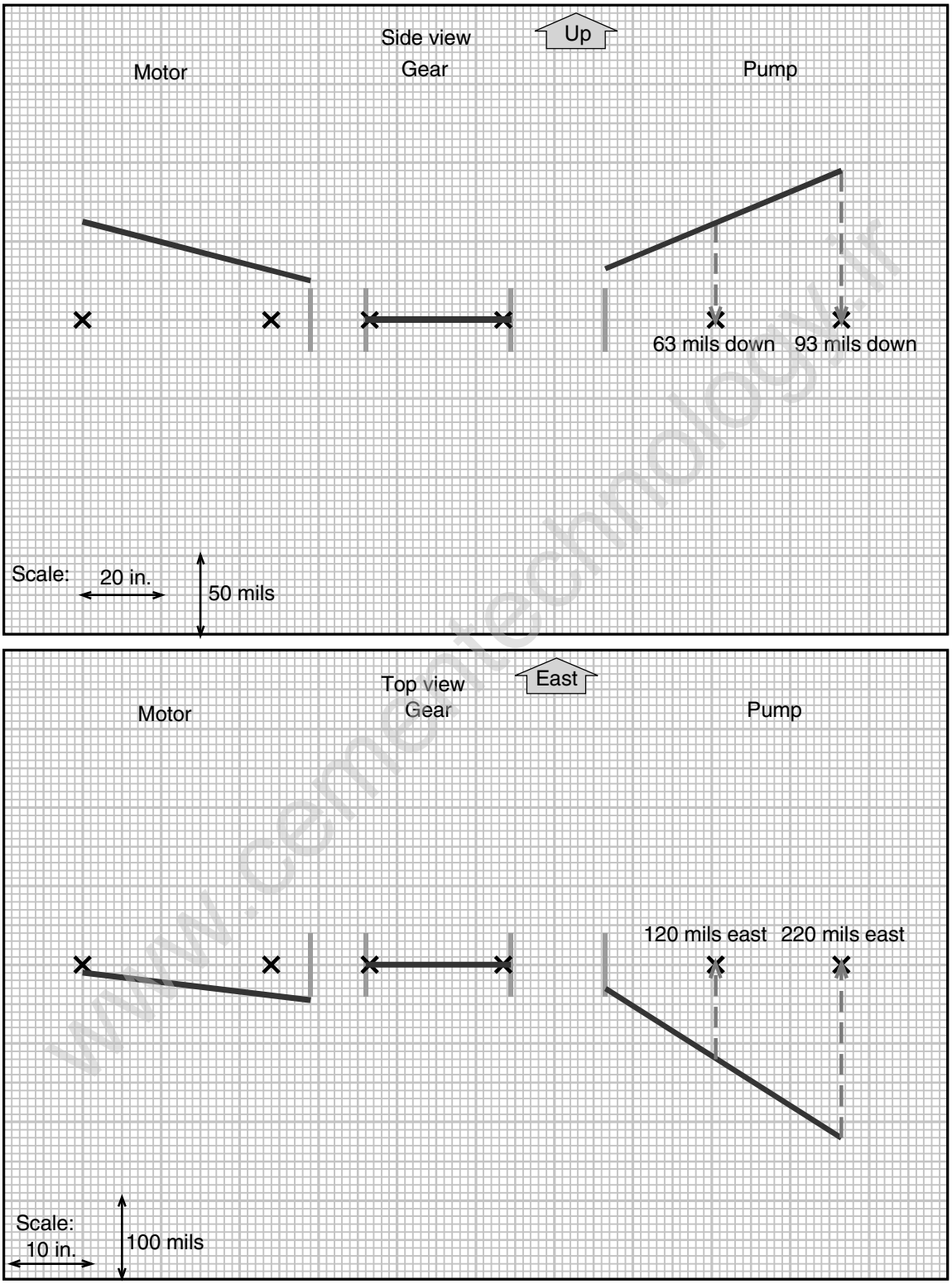


FIGURE 17.13 Three-element alignment model showing all three shafts.

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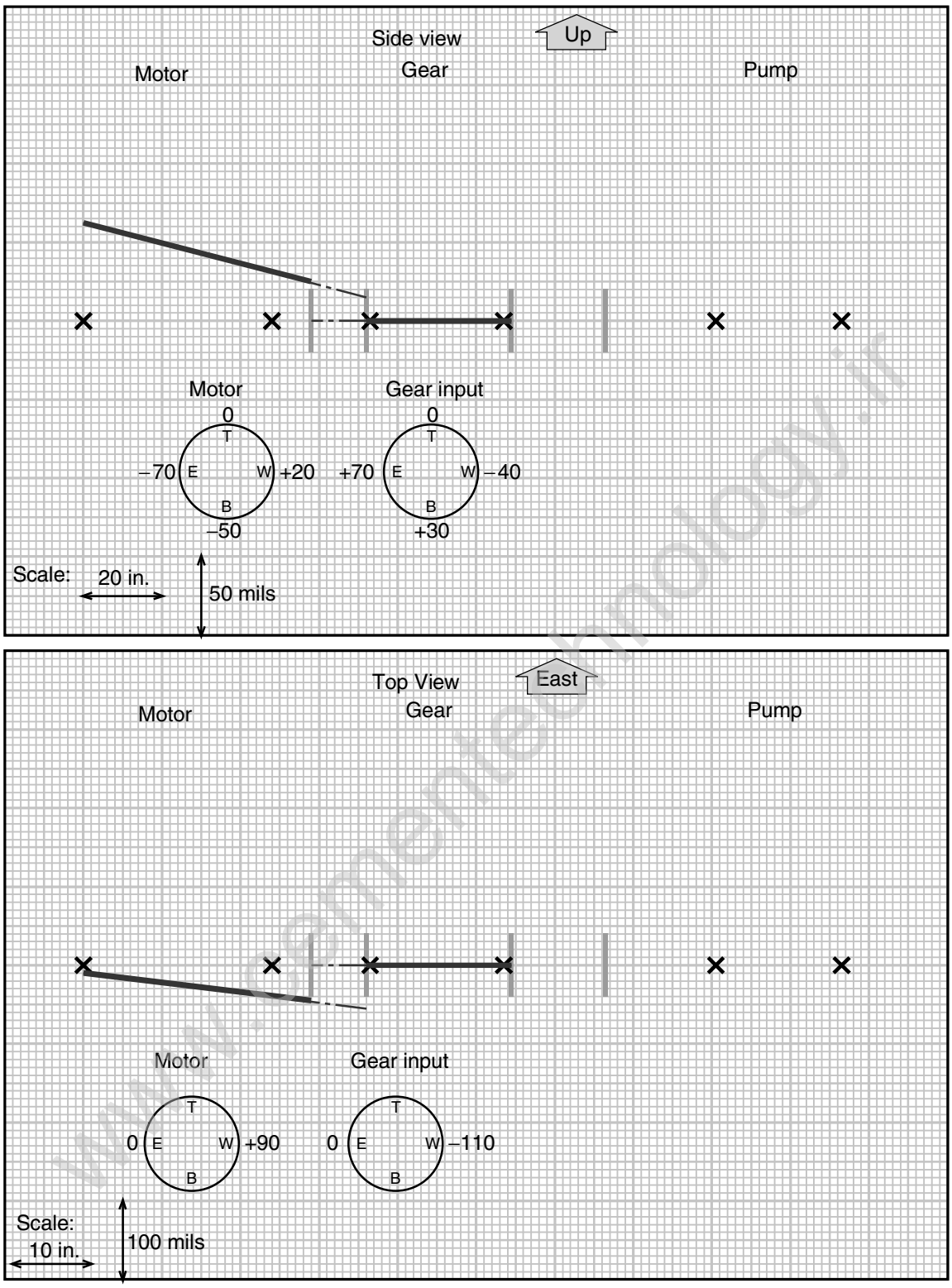


FIGURE 17.14 Three-element alignment model plotting the motor and gear shafts using the line to points reverse indicator modeling technique.

Figure 17.14 shows the misalignment between the motor and gear shafts in the side and top views using the line to points reverse indicator modeling technique. Figure 17.15 shows the misalignment between the gear and pump shafts in the side and top views. It does not matter which set of readings is plotted first, the motor to gear or the gear to pump. Notice that the

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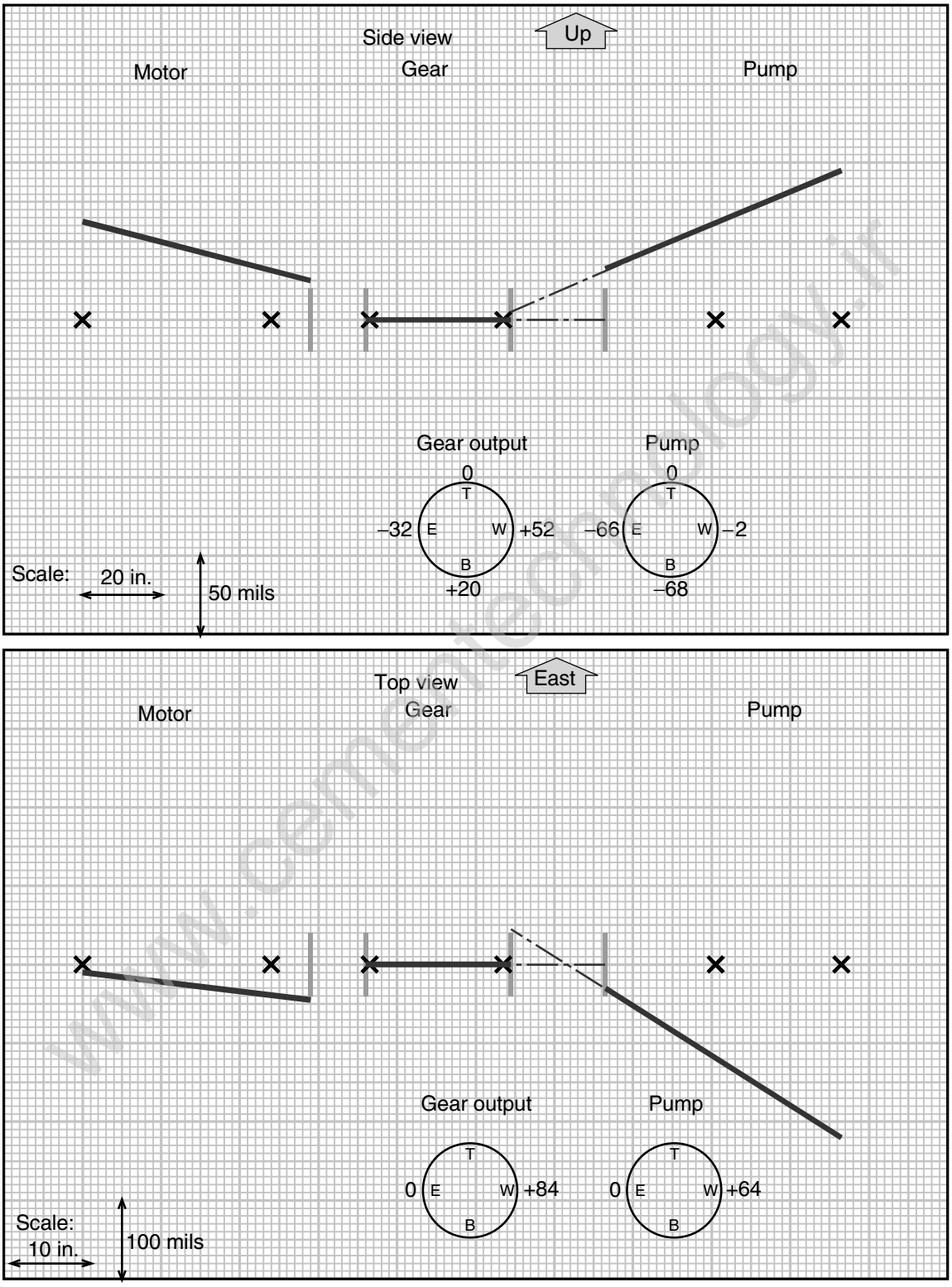


FIGURE 17.15 Three-element alignment model plotting the gear and pump shafts using the line to points reverse indicator modeling technique.

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positions of the shafts in Figure 17.13 are identical to the positions of the shafts in Figure 17.15 even though two different methods were used to generate the two graphs.

Again, just as in modeling two-shaft drive systems, all that has been accomplished with the graph is that the positions of the shafts have been determined. At this point, it is imperative that the restrictions in the up or down and east or west directions be determined at every bolting plane on all three machines and transferred onto the graph. Once this has been done, the final desired overlay line can be drawn onto the graph and the movement solutions at all of the bolting or translation planes can be determined and executed. An example of this is illustrated in Figure 17.16 showing the amount of existing shims and the lateral movement restrictions at each bolting plane. Notice that in the side view if you were to name any one of the machines as a ''stationary'' machine, the vertical restriction would have prevented you from aligning the drive system; and in the top view if you were to name either the gear or the pump as a stationary machine, the lateral restriction would have prevented you from aligning the drive system.

17.5 MIXING DIFFERENT ALIGNMENT MEASUREMENT METHODS

In some situations, more than one alignment measurement method could be (or may have to be) employed to measure each set of shafts. Figure 17.17 through Figure 17.19 show how you can use different alignment measurement methods and still model the machinery positions.

Two electric motors are coupled together to drive a gearbox and a compressor. A laser–detector shaft alignment system was used to measure the alignment between the two motors (Motor A and Motor B). Reverse Indicator readings were taken between Motor B and the gearbox. The shaft to coupling spool method was employed between the gear output shaft and the compressor. Figure 17.17 shows the dimensions of the four-element drive system, the information gathered from the laser–detector shaft alignment system, the reverse indicator measurements (before and after sag compensation), the shaft to coupling spool measurements (before and after sag compensation), the amount of existing shims under each bolting plane, and the lateral movement restrictions at each bolting plane. You need every piece of information shown in Figure 17.17 to determine how to correct the misalignment condition that exists on this drive system.

Figure 17.18 shows the side view alignment model of all four shafts and the vertical restriction boundary. The gear shafts were placed on the graph paper centerline and each shaft was referenced from the gear outward. Figure 17.19 shows the top view alignment model of all four shafts and the lateral restriction envelope. Again, the gear was placed on the graph paper centerline and each shaft was referenced from that point outward. Carefully study both alignment models to determine how the shafts were constructed.

Despite the fact that three different alignment measurement methods and tools were used to determine the relative positions of each set of shafts, the entire drive train can still be modeled on graph paper. As you can see from the shaft positions in the side view, attempting to call Motor A, Motor B, or the gear as the ''fixed'' or stationary machine will result in a lot of headaches and unnecessary work. In the top view, attempting to call any of the four machines as the fixed or stationary machine will result in unforeseen situations and unnecessary work. It should become obvious why it is recommended that the multiple-element drive train alignment laws mentioned above should be adhered to when aligning drive systems of this complexity.

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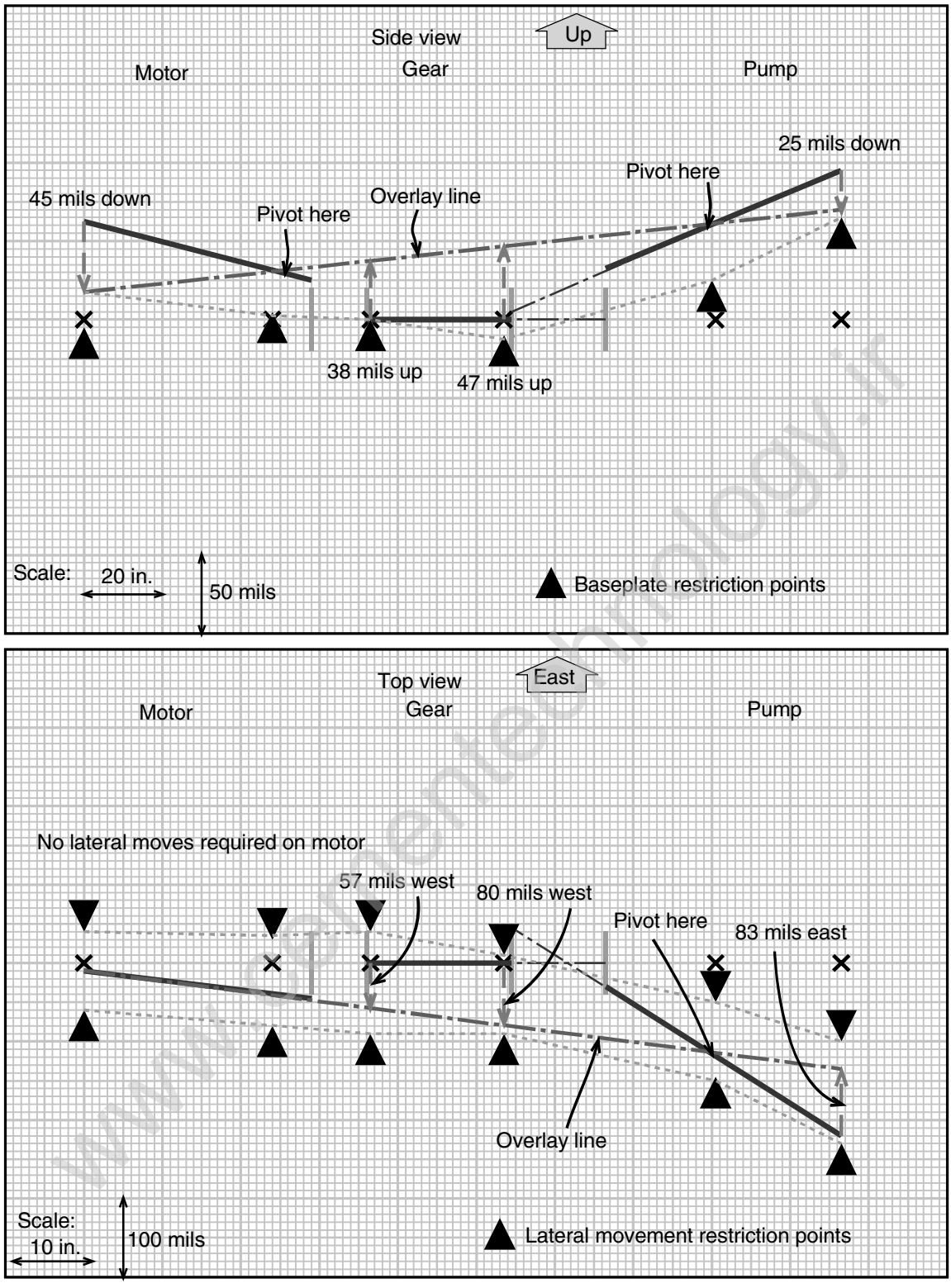


FIGURE 17.16 Possible alignment corrective moves after overlaying the boundary conditions.

17.6 MODELING RIGHT-ANGLE DRIVE SYSTEMS

So far, we have examined rotating machinery drive systems that are horizontally mounted, direct in-line machinery. But not all drive systems are configured that way. Some drive trains are arranged in an "L" shape, commonly referred to as right-angle drives. A right-angled

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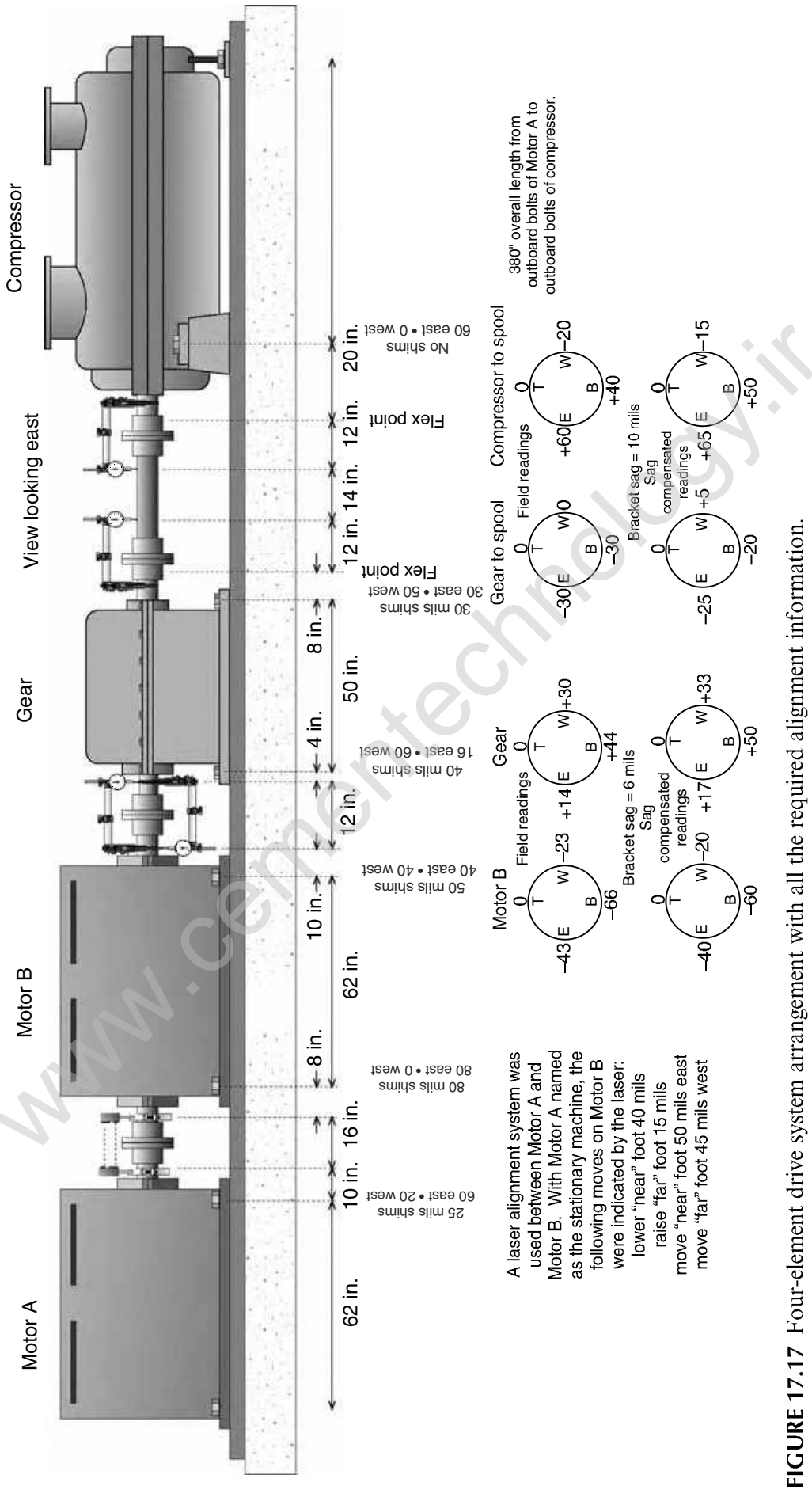


FIGURE 17.17 Four-element drive system arrangement with all the required alignment information.

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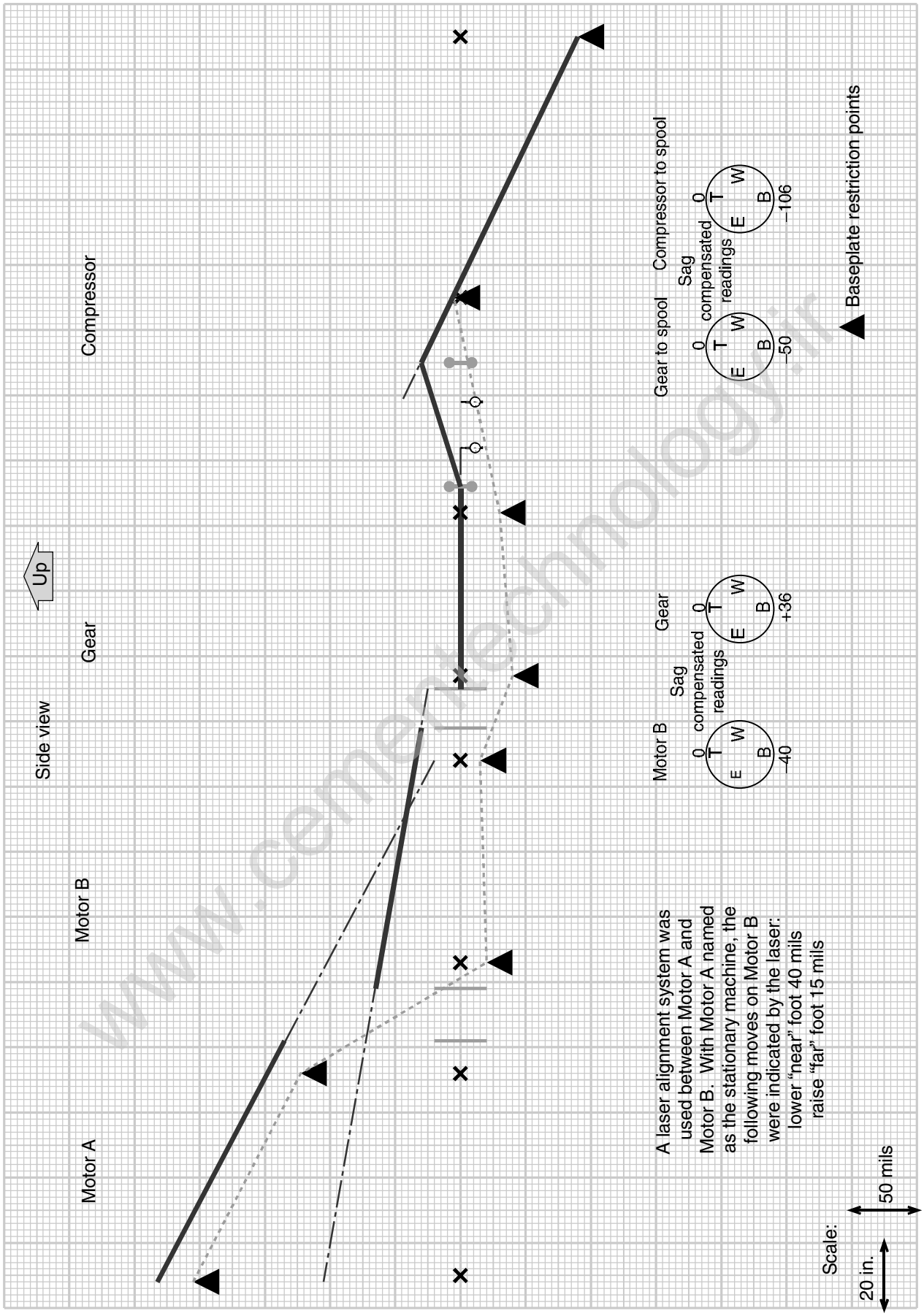


FIGURE 17.18 (**See color insert following page 322.**) Four-element drive system side view alignment model of all four shafts and the vertical restriction boundary.

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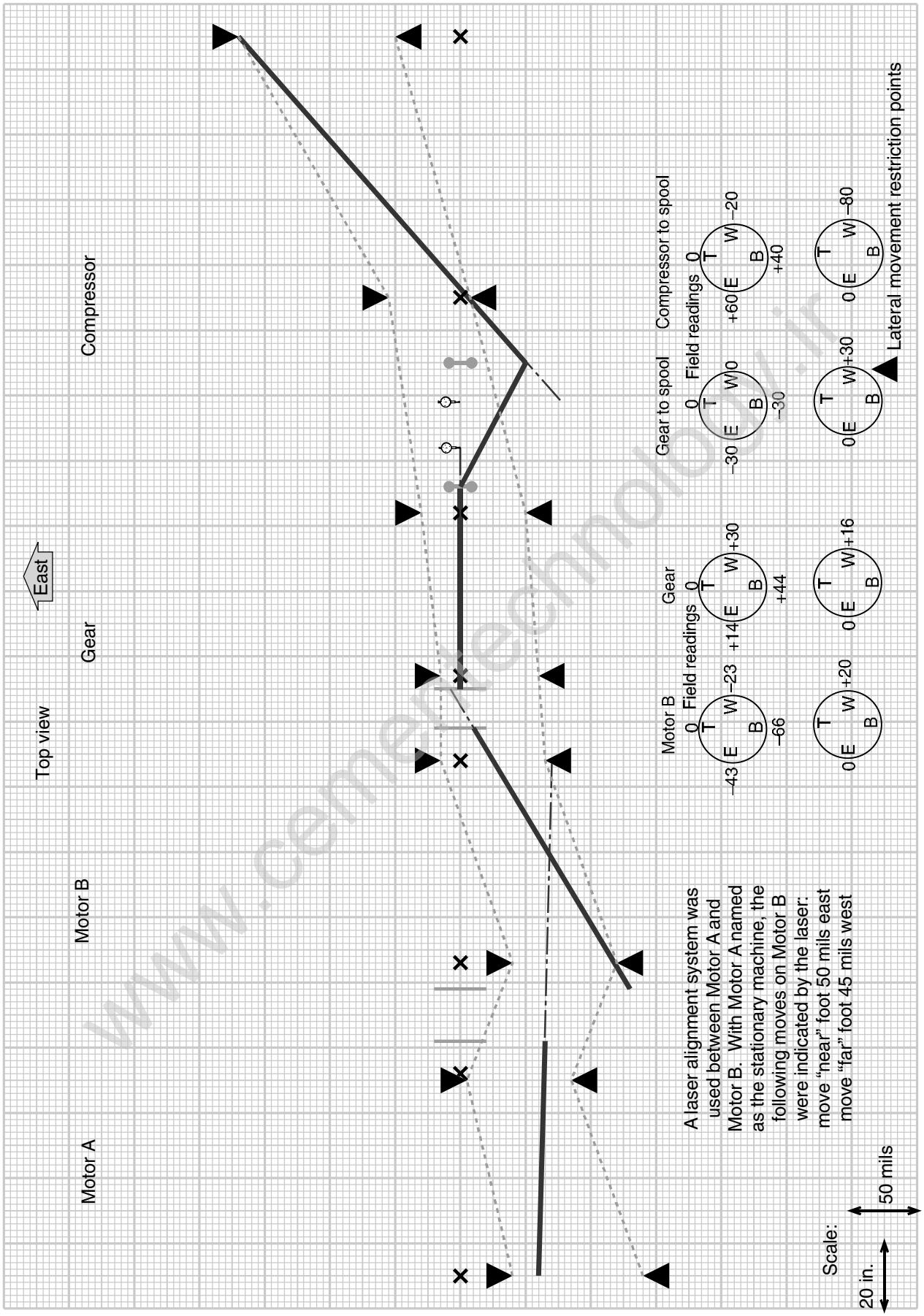


FIGURE 17.19 (**See color insert following page 322.**) Four-element drive system side view alignment model of all four shafts and the lateral restriction envelope.

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gearbox is flexibly coupled to a driver on the input side and to some driven machine on its output side. The goal is to align the centerline of rotation of the driver and the input shaft of the gear and the driven unit to the output shaft of the gear.

There are two tricks to this modeling method. One is to plot the graph in the "dual scale" mode similar to the method used for face–rim plotting covered in Chapter 11. The other trick is to "fold" one of the views of the alignment model where the right angle occurs in the drive system.

The sample problem shown in Figure 17.20 comprises an electric motor coupled to a speed reducing right-angled gear that drives a roll. The reverse indicator was employed between the

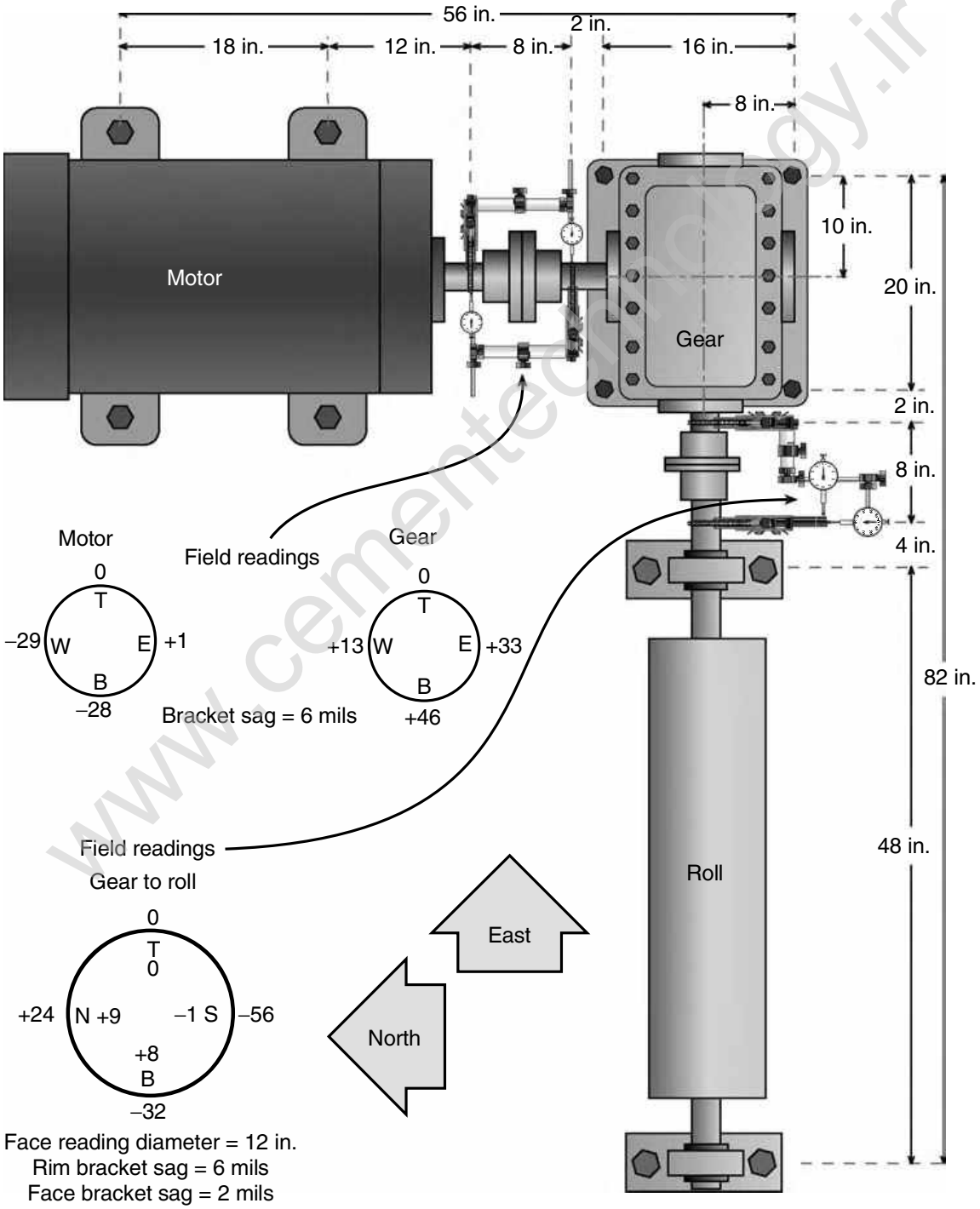


FIGURE 17.20 Right-angle drive system.

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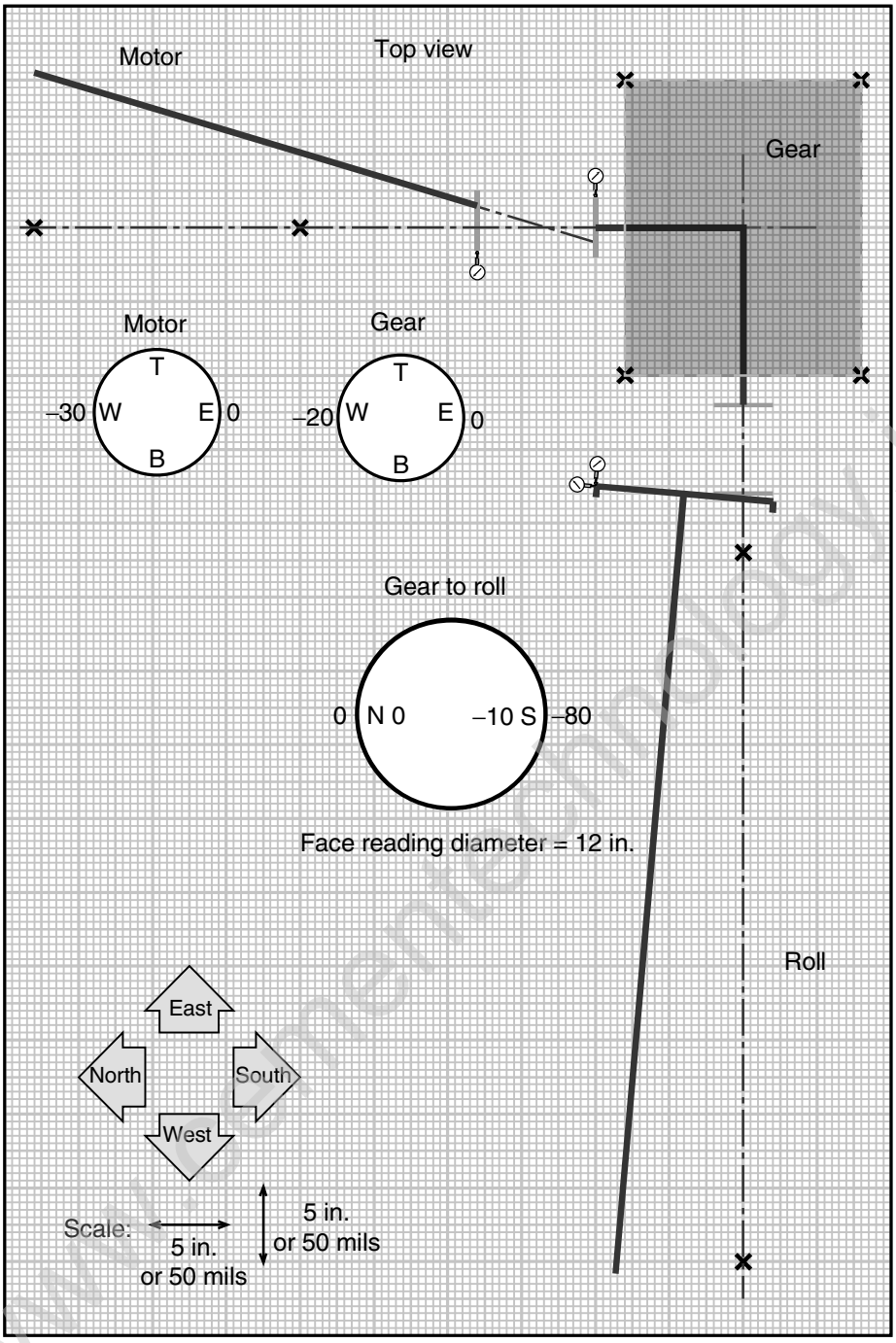


FIGURE 17.21 Right-angle drive system top view alignment model.

motor and the input shaft of the gear. The face–rim method was used to capture the alignment data between the motor and the output shaft of the gear.

Figure 17.21 shows the top view of the motor–gear–roll drive system. The alignment plotting techniques in this view are straightforward and follow the modeling principles explained in Chapter 10 and Chapter 11.

Figure 17.22 shows the side view of the drive system. There is a slight "trick" in this alignment model view however. The side view alignment model is split or "folded" as if you were viewing it from two different directions. Figure 17.23 shows how the alignment model is folded through the center of the right-angled gear (i.e., it is as if you folded the graph paper

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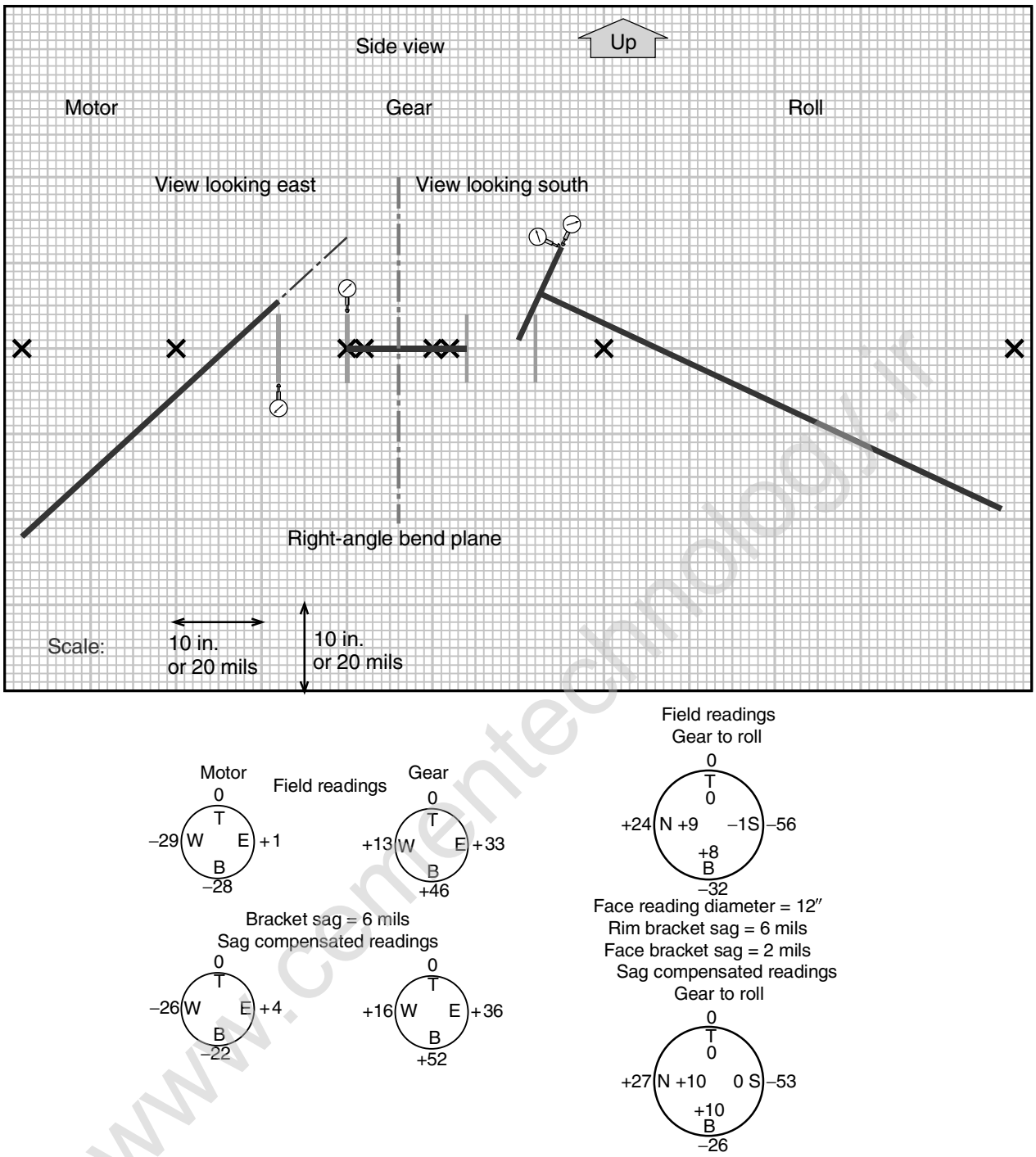


FIGURE 17.22 Right-angle drive system side view alignment model.

through the center of the right-angle gear through its vertical axis). The left side of the side view alignment model shows what the position of the motor shaft and input shaft of the gear would look like if you were looking to the east. The right side of the side view alignment model shows what the position of the output shaft of the gear and the roll would look like if you were looking to the south. Figure 17.24 illustrates how each bolting plane on the right-angled gearbox is identified depending on which direction you are viewing in the side view alignment model. Figure 17.25 shows a possible alignment corrective move in the vertical direction assuming there were no shims under any of the machinery feet.

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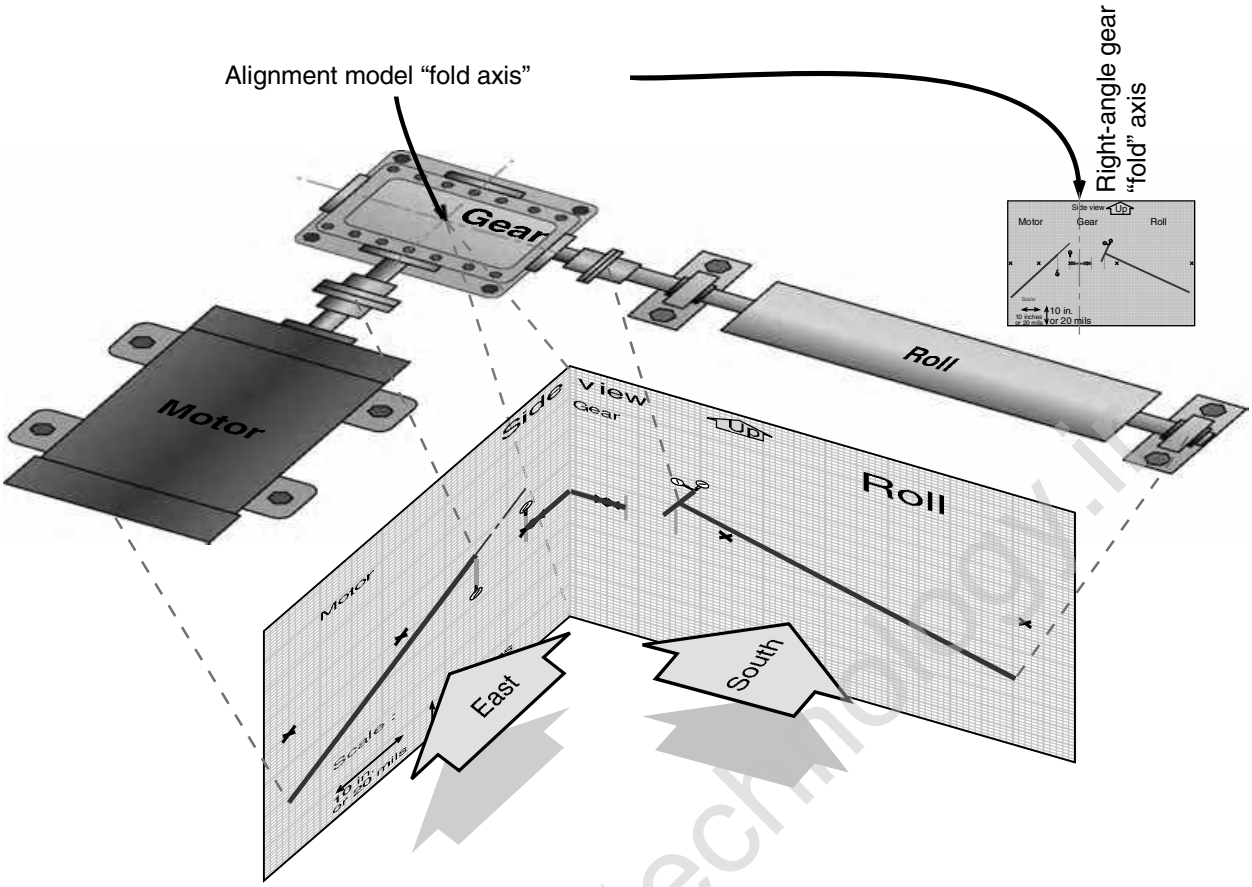


FIGURE 17.23 (**See color insert following page 322.**) Right-angle drive system "folded" side view alignment model.

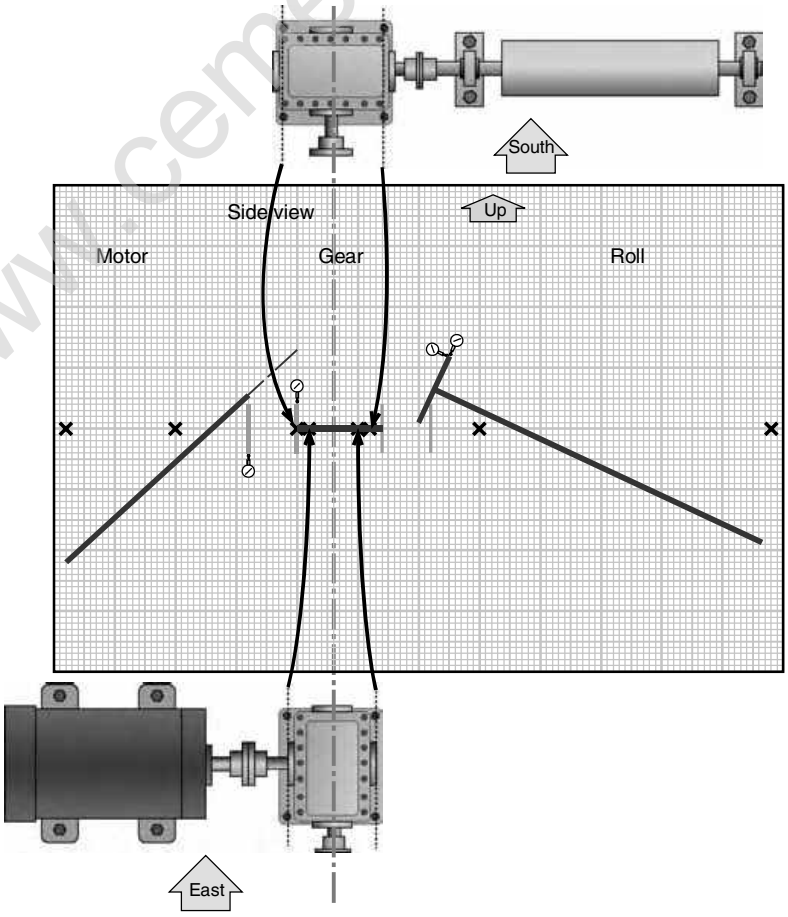


FIGURE 17.24 Bolt plane identification on the right-angled gear.

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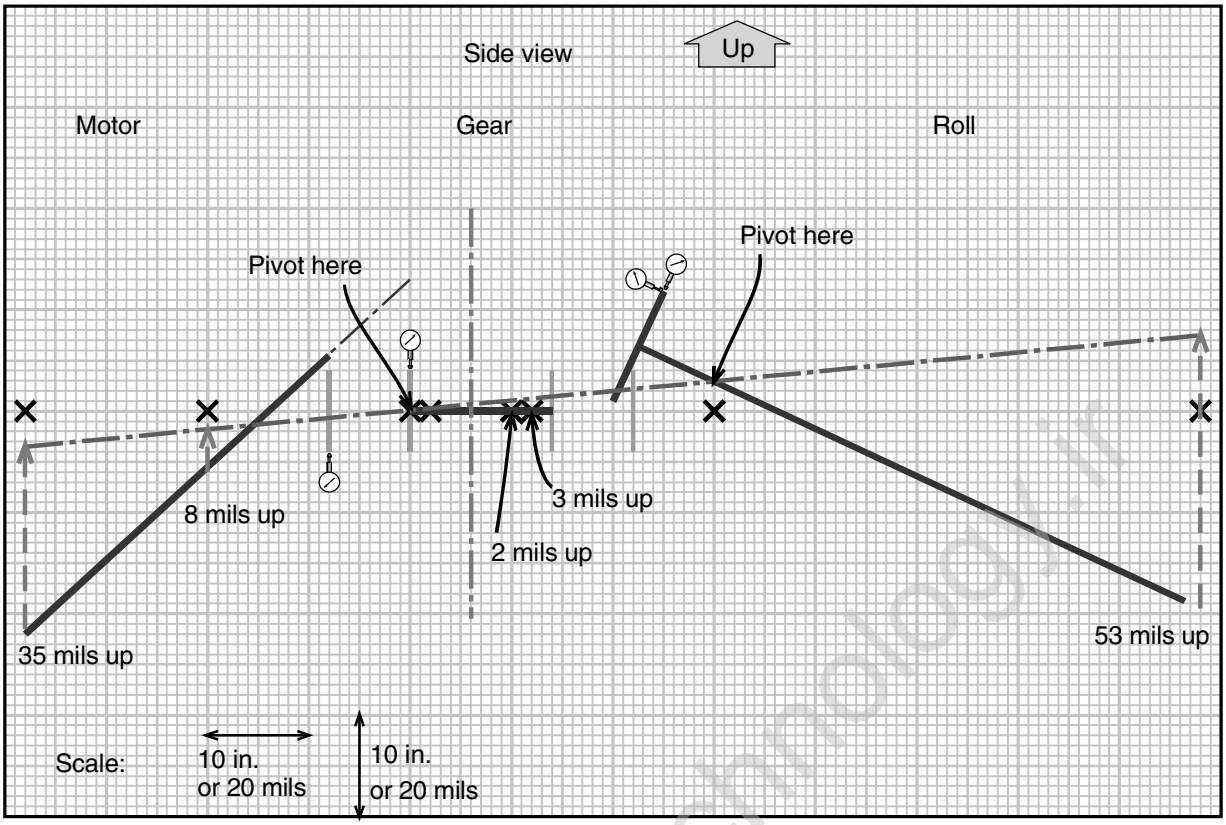


FIGURE 17.25 Possible vertical alignment corrective moves with no shims under any machinery feet.

To determine the corrective moves for this drive system in the top view, a clear transparency with the right-angled gear shafts transferred to the transparency will be of assistance in determining what lateral moves need to be made to correct the misalignment condition as shown in Figure 17.26. In addition, we need to determine the lateral movement restrictions (i.e., the bolt bound conditions) on all three machines in the drive system. The gaps between the foot bolts and the holes in the motor and the pillow block bearings in the roll are fairly straightforward. Because the motor will only need to be positioned in the east to west direction and the roll in the north to south direction, the gaps between the foot bolt and holes on the east and west sides and the gaps between the pillow block bearing and bolt holes on the north and south sides, respectively, need to be detrmined. The gaps in the holes of the ight-angled gear however need to be determined in all four directions as the gear may have to be rotated through its vertical axis. Figure 17.27 shows what gaps are needed to be measured on the gear feet. These restrictions are then transferred to the top view alignment model. The restrictions on the motor and roll can be illustrated as single-plane restriction points, the restrictions on the gear need to be illustrated as ''restriction rings'' as shown in Figure 17.28. The clear transparency showing the right-angled gear shafts and the four corner bolts is pitched to insure that the projected centerlines of the input and output shafts of the gear fall within the lateral movement boundaries of both the motor and the roll and also within the restriction rings on the gear itself as shown in Figure 17.29. Once the transparency is positioned to stay within the lateral movement boundaries of all three machines, the required moves can then be determined to correct the misalignment condition.

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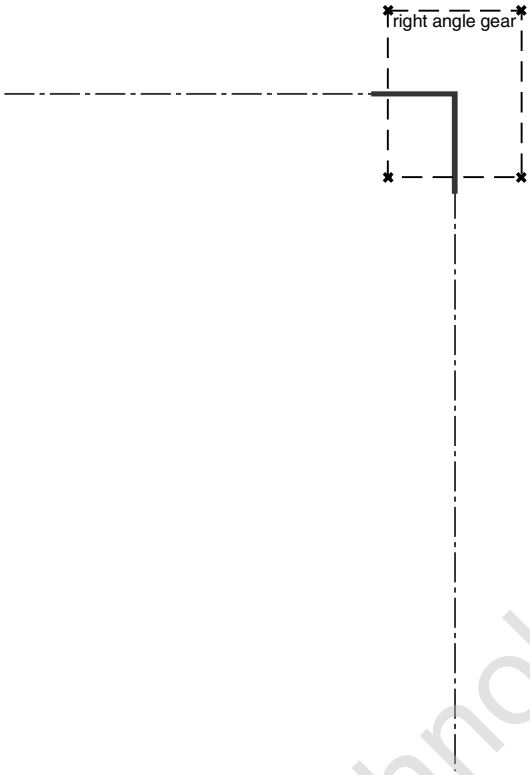


FIGURE 17.26 Right-angle gear overlay transparency for assistance in the top view alignment model.

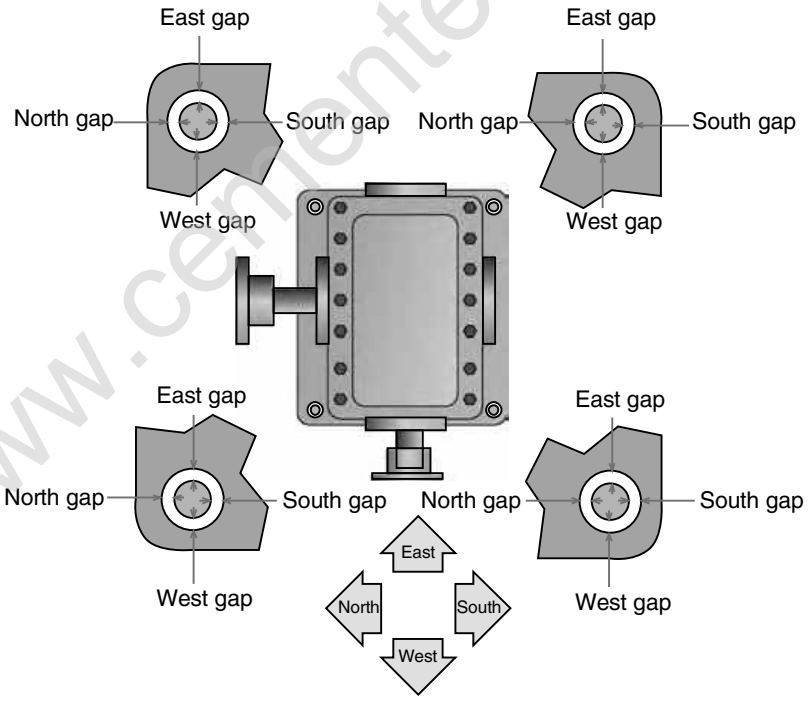


FIGURE 17.27 Lateral movement restrictions on the gear in all four directions are needed.

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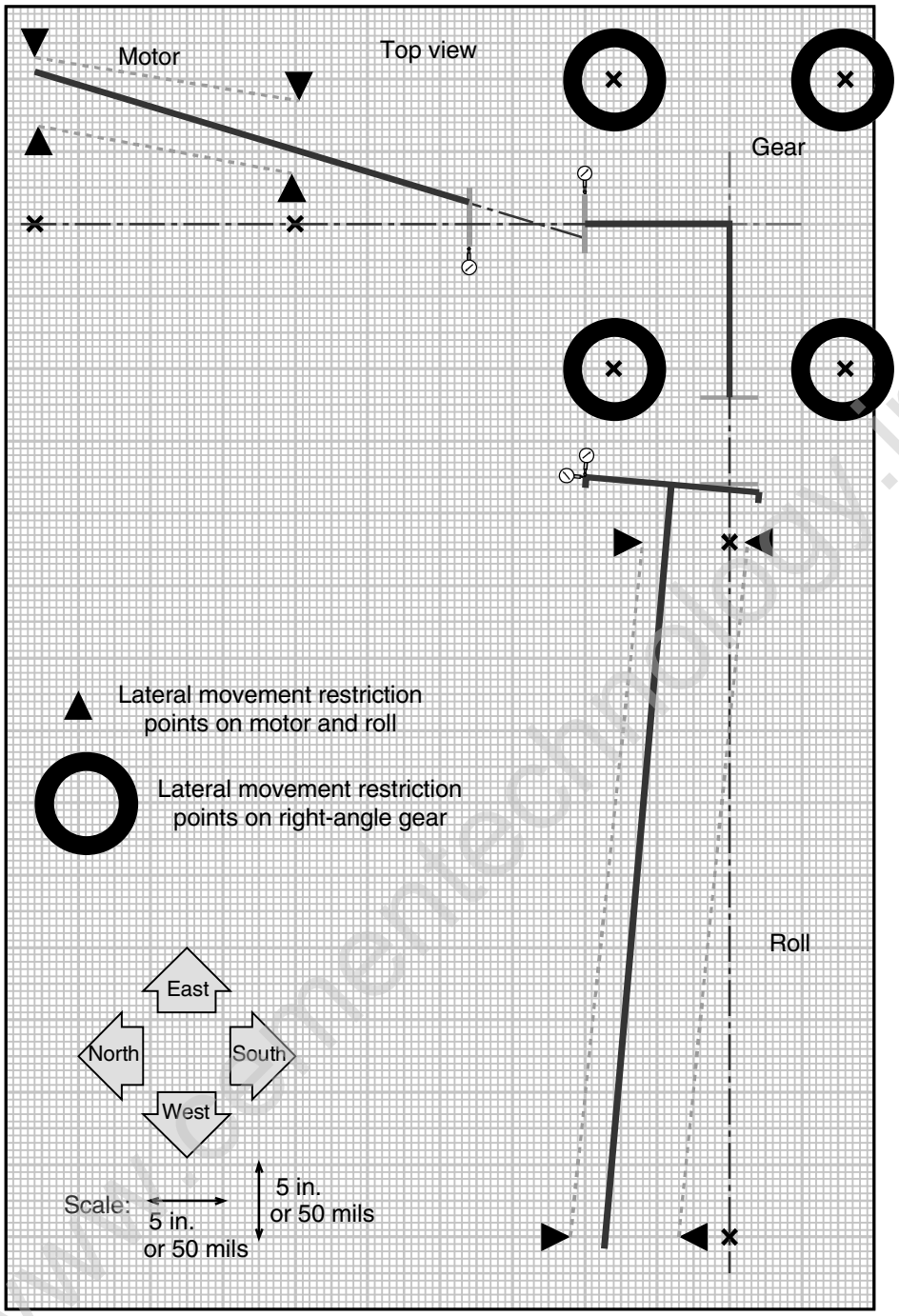


FIGURE 17.28 Top view alignment model showing the boundary conditions.

17.7 FINAL COMMENTS ON ALIGNING MULTIPLE-ELEMENT DRIVE TRAINS

Multiple-element drive trains can get extremely complex if you do not keep your wits about you when aligning the machinery. Some drive systems can consist of up to 20 or more elements in "L" shapes, "U" shapes, "S" shapes, or other ridiculously complex arrangements. Regardless of how many pieces of machinery are in the drive train or how the shaft-to-shaft measurements were taken, the positions of all the machinery can be accurately illustrated on

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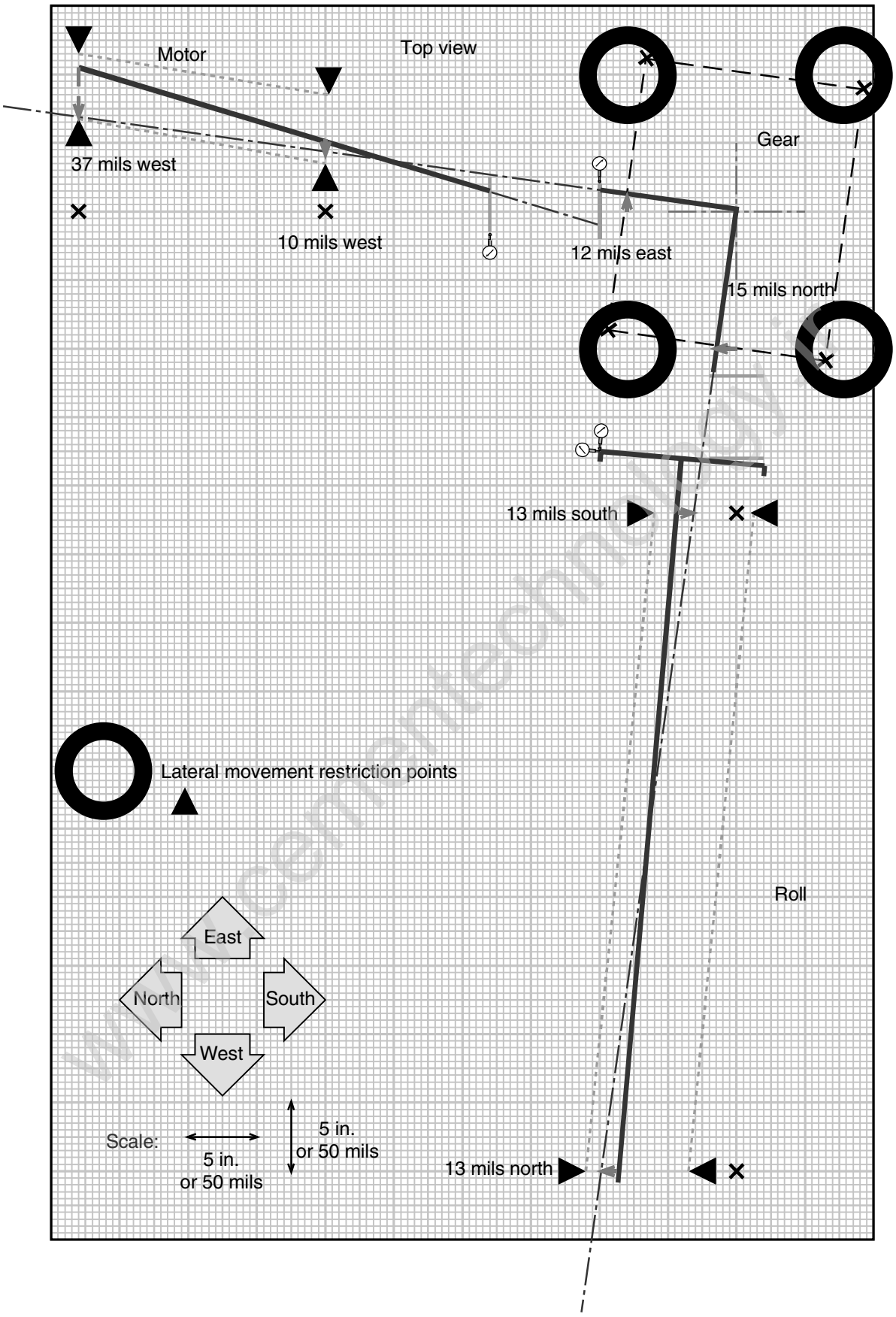


FIGURE 17.29 (**See color insert following page 322.**) Top view alignment model showing possible alignment corrective moves in the lateral direction staying within the boundary conditions.

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an alignment model as shown in this chapter. Once the movement restrictions are superimposed onto the alignment model to visualize your boundary conditions, usually by moving more than one element, an effective alignment solution can be attained.

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Piotrowski, J., Aligning multiple-element drive trains and right-angle drives,'' *P/PM Technology*, 5(2), 1992.

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18 Aligning V-Belt Drives

Up to this point, the machinery discussed in this book has been directly driven by flexible or rigid couplings, but there are alternative power transmission schemes that are frequently used in industry known as belt drives. Examples of belt drive machinery can be seen in Figure 18.1 through Figure 18.4.

18.1 BELT DRIVE SYSTEMS—ADVANTAGES AND DISADVANTAGES

There are several advantages and disadvantages of belt drive equipment:

Advantages
- Inexpensive way to reduce or increase speed.
- Provides some cushioning against shock loads during start up and operation.
- Does not require lubrication like a gearbox.
- Because of the flexibility in belts, alignment is not as critical as direct drive systems.
- In certain instances, placing two shafts parallel to each other rather than in line can save space.
- When the drive system is off-line, visual inspections of the belts are fairly easy to do.
- If the speed of the driven machine is too fast or too slow for the operational requirements, changing the diameter of the sheaves to adjust the speed is fairly inexpensive.

Disadvantages
- Incapable of delivering a precise nonvarying speed of the driven machine as belts will wear and eventually begin to slip over time.
- Correct belt tension is difficult to achieve and may change dramatically with new belts in a short period of time.
- Belt tension places radial loads on bearings and shafts even if the alignment is correct.
- Sheaves are frequently found to be eccentric or skew bored causing excessive vibration and reduced belt life.
- Some sheave designs have a taper lock bushing that will, if the installer is not careful, draw the sheave onto the bushing incorrectly inducing an excessive radial or face runout condition.

18.2 V-BELT STANDARDS INFORMATION

Standard cross-sectional and length dimensions have been established for V-belts as shown in Figure 18.5 and Figure 18.6. Figure 18.7 shows how to calculate the required length of a V-belt.

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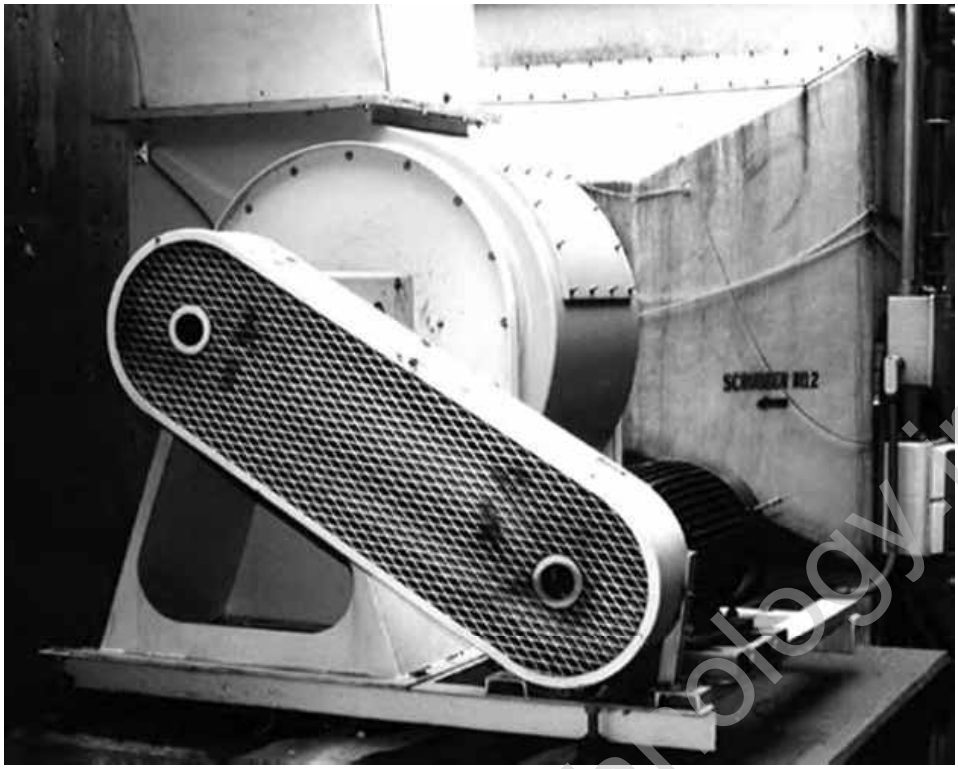


FIGURE 18.1 Belt driven fan.



FIGURE 18.2 Belt driven fan.

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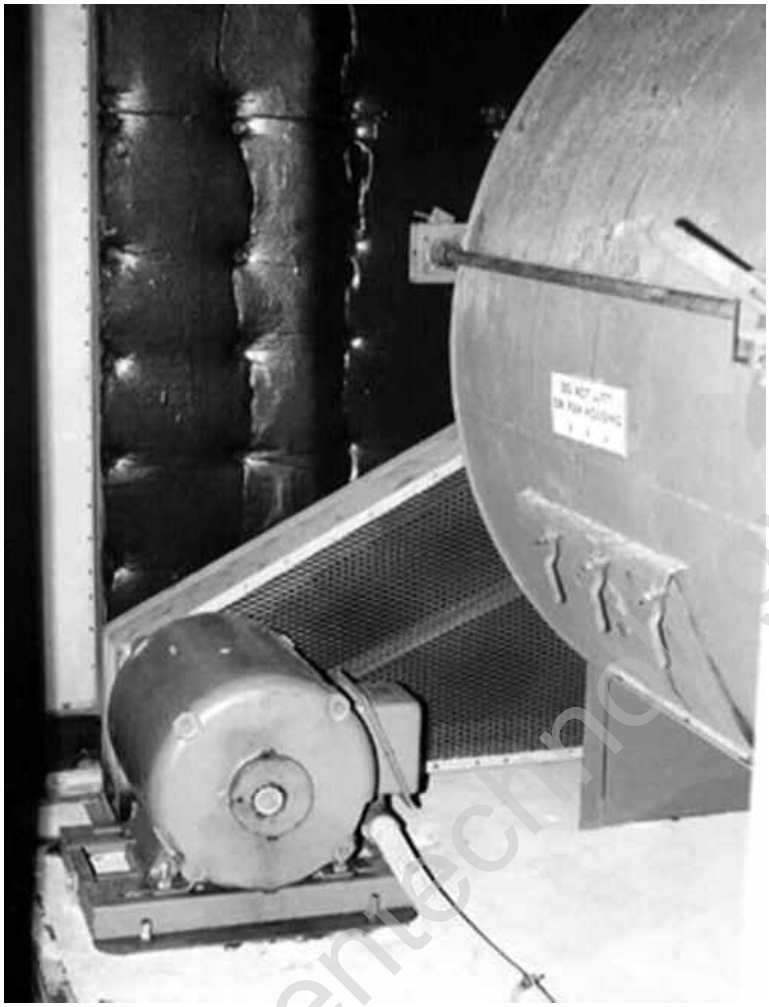


FIGURE 18.3 Belt driven fan.

18.3 SHEAVE INFORMATION

Standard information on sheaves is shown in Figure 18.8.

18.4 V-BELT RECOMMENDATIONS AND RULES OF THUMB

Here are some guidelines for successful, long-term belt operation:

- 4000 ft/min is a good speed for V-belts to run at. Trouble can begin to occur below 1000 ft/min and above 5000 ft/min.
- Typically, the center-to-center distance between shafts should neither exceed three times the sum of the sheave diameters nor should it be less than the largest sheave diameter.
- Belts should be tensioned to 1/64 in. deflection per inch distance between shaft centers (refer to belt tensioning guidelines and procedures).
- Typically, sheave misalignment should be less than 1.5° (i.e., 1/8 in. per foot or approximately 10 mils/in.).
- Do not mix new and old belts or belts from different manufacturers.
- Keep debris from jamming between belts and sheaves during operation or replacement.
- Salads get dressing, not belts.

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FIGURE 18.4 Belt driven dredge.

- Worn sheaves may shorten belt life by 50%.
- For a specific shaft speed, shorter belts will not last as long as longer belts.

18.5 SHEAVE AND BELT WEAR

Sheave and belt wear indicator should be used to determine if these components are worn excessively. These Go/No go gauges are typically made out of plastic and can be obtained from your power transmission supplier. Figure 18.9 shows what to look for when using the gauges.

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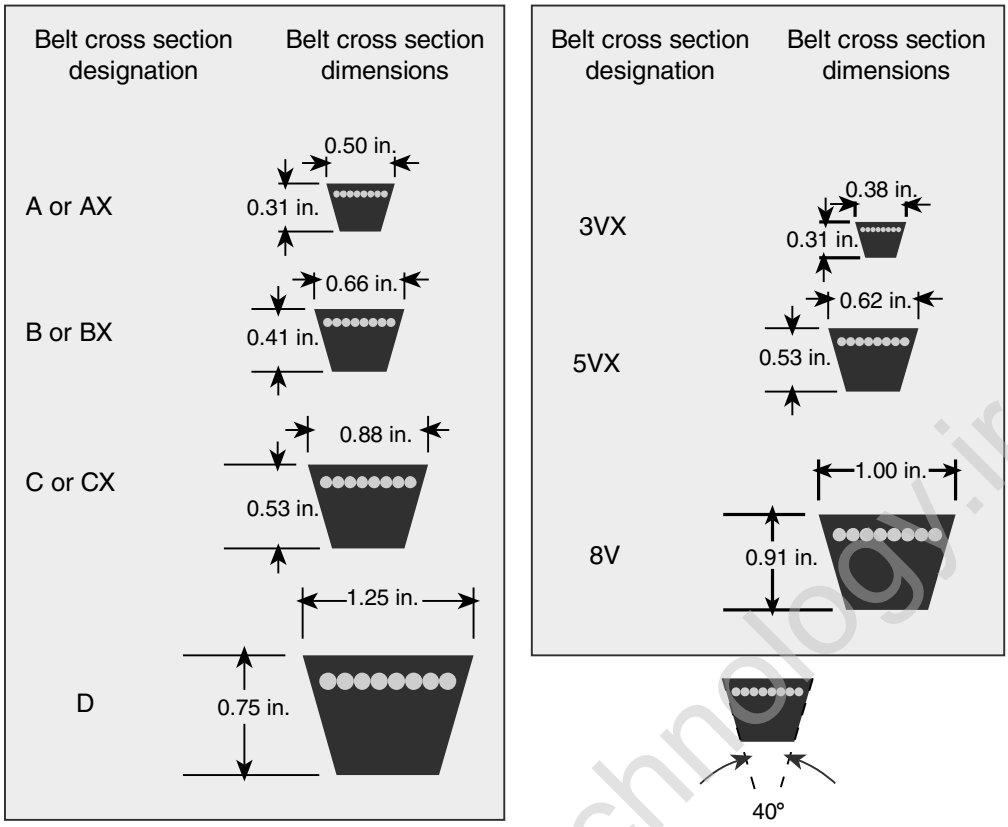


FIGURE 18.5 V-belt cross section standard dimensions.

18.6 ADJUSTING BELT TENSION

Possibly the most critical requirement for successful belt installation and belt life is to establish the correct belt tension. It is often the most difficult to do and subject to a great deal of speculation and guesswork. There are two general guidelines that are followed. One

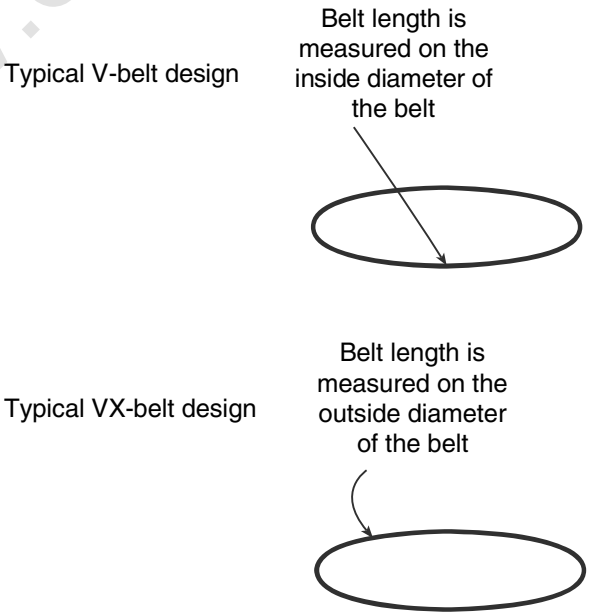


FIGURE 18.6 V-belt length measurements for V- and VX-type belts.

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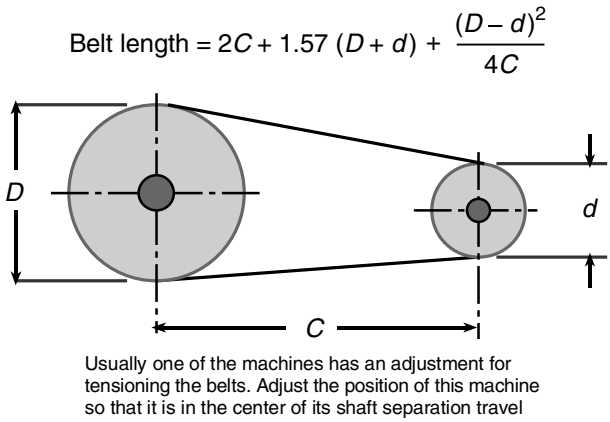


FIGURE 18.7 How to determine the correct length of a belt.

using a belt tensioning gauge as shown in Figure 18.10 and the other without one as shown in Figure 18.11. As mentioned previously, if you have installed new belts, it is recommended that you check and if necessary, adjust the belt tension after 24–48 h of operation since the belts will stretch slightly after their initial use.

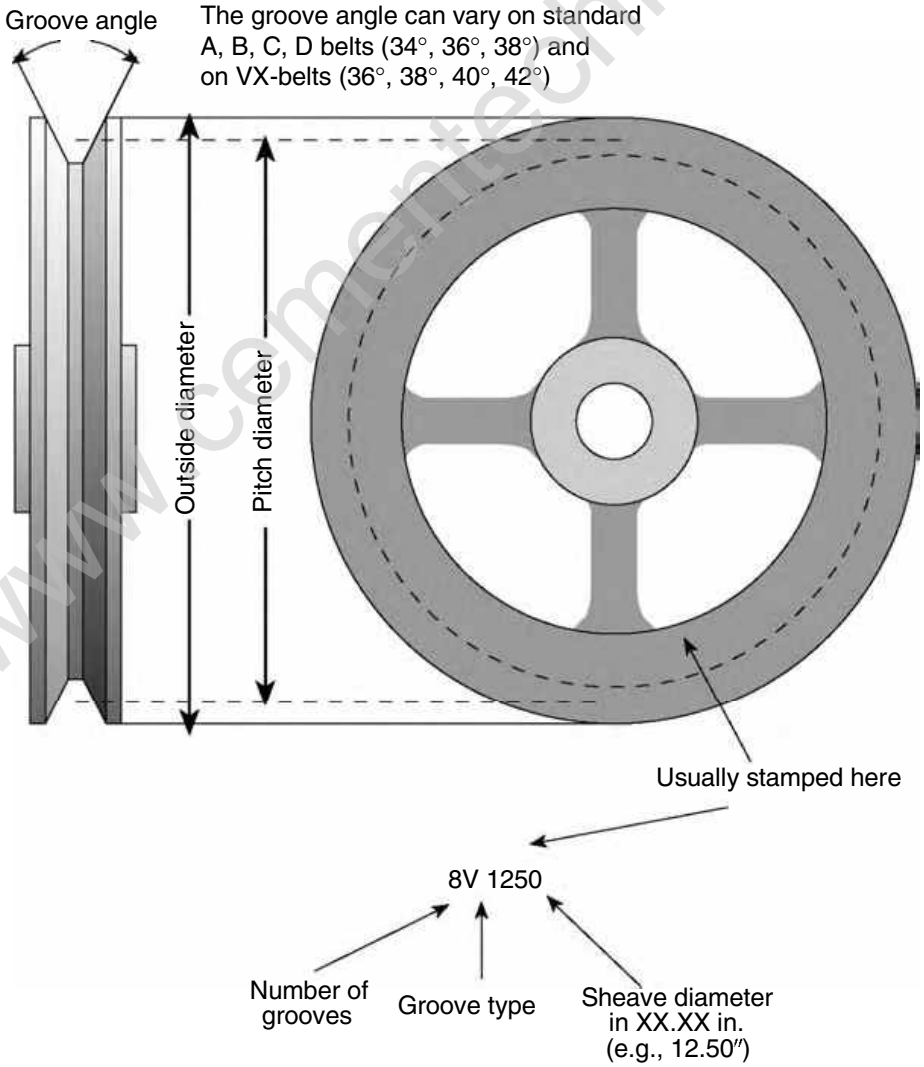


FIGURE 18.8 Sheave nomenclature and dimensions.

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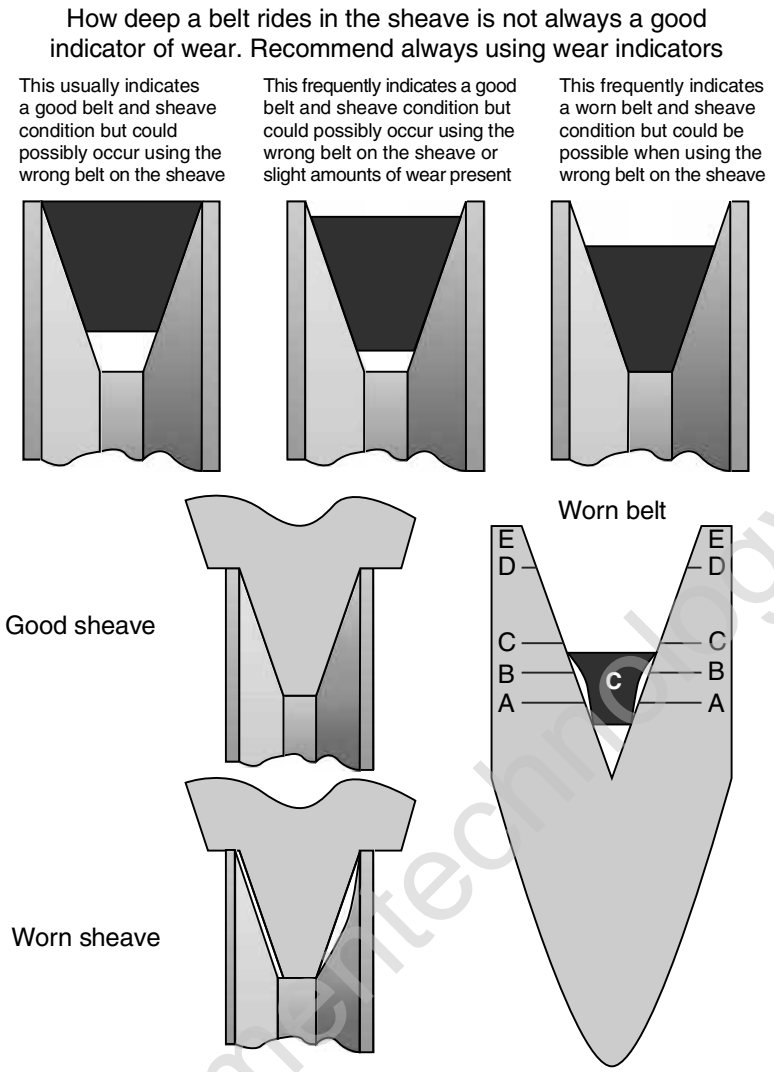


FIGURE 18.9 V-belt and sheave wear.

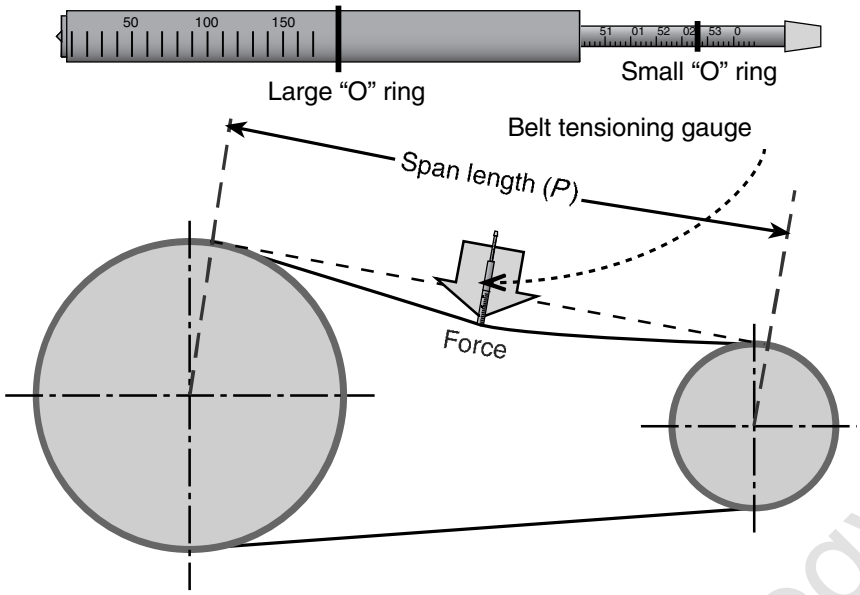
18.7 PRELIMINARY ALIGNMENT CHECKS FOR V-BELTS AND SHEAVES

Machinery driven by V-belts is subject to the same types of soft foot problems as found on direct drive machinery. Any soft foot conditions should be corrected before starting the alignment procedure and definitely before tensioning of the belts. Refer to Section 5.6 on finding and correcting soft foot problems. Both radial and face runout checks need to be made on the sheaves as shown in Figure 18.12 through Figure 18.14.

There seems to be a higher incidence of excessive runout problems on sheaves that on flexible or rigid coupling hubs. Many sheaves are attached to shafts using taper lock bushings. The advantage of taper lock bushings is that you can quickly and easily position the sheave at different axial positions along the end of the shaft to assist in the belt/sheave alignment process. The disadvantage of taper lock bushings is that it is very easy to cock/skew the sheave onto the tapered bushing if you tighten the first bolt in the bushing to its final torque value and then lock it in a skewed position when the rest of the bolts are tightened. To eliminate this problem, first finger tighten each taper lock bolt to just begin seating the sheave onto the tapered bushing. Setup a magnetic base onto the machine frame and position a dial indicator stem on the face of the bushing as shown in Figure 18.15.

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1. When installing new V-belts, shorten the center-to-center distance to allow for the new belts to be placed in the sheave groove without forcing or prying the belts into place. Arrange the belts so the top and bottom spans have the same amount of sag. Begin applying tension to the belts by increasing the center-to-center distance until the belts are snug and have a "springy" action when struck with your hand. If possible, it is recommended that the drive system be hand rotated to assist in seating the belts during tensioning.
2. Measure the span length (*P*) (see diagram). Use a white piece of chalk or pencil to mark the center of the span.
3. Place a straightedge on the outer surface of the belt insuring that the straightedge is contacting the belt at the points on each sheave where the span length (*P*) was measured.
4. Slide the large "O" ring along the barrel of the Span length scale of the tension tester to align with the measured span length. Slide the smaller "O" ring along the barrel of the pounds of force scale of the tension tester to align with zero pounds of force.
5. Position the Span length scale of the tension tester at the center of the belt span, grasp the other end with your hand and firmly apply force to the belt until the bottom edge of the large "O" ring is even with the bottom of the straightedge. The smaller "O" ring will side to a specific force mark on its scale.
6. Determine what belt cross section is on the drive system (e.g., A, CX, 5VX, etc.). Refer to the chart that came with the tension tester to determine if you are 1 to 1.5 times the required belt deflection force. If the force is too high, decrease the center-to-center span. If the force is too low, increase the center to center span.

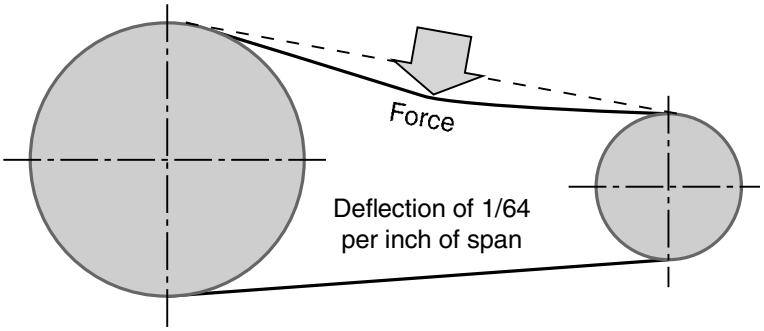
   Note: You may want to run the test two or three times to insure repeatability of the measurements.

FIGURE 18.10 Adjusting belt tension using a gauge.

Slowly hand rotate the shaft observing where the high and low spots are on the sheave. Based on where the high (or low) spot is on the sheave, tighten the bolt nearest to the high spot to begin squaring up the bushing to the centerline of rotation of the shaft. Slowly continue hand rotating the shaft observing where the high and low spots are on the sheave. Tighten each bolt incrementally until the face runout has been eliminated or reduced to acceptable levels. If the face runout cannot be reduced to acceptable levels, remove the

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1. When installing new V-belts, shorten the center-to-center distance to allow for the new belts to be placed in the sheave groove without forcing or prying the belts into place. Arrange the belts so the top and bottom spans have the same amount of sag. Begin applying tension to the belts by increasing the center-to-center distance until the belts are snug and have a "springy" action when struck with your hand. If possible, it is recommended that the drive system be hand rotated to assist in seating the belts during tensioning.
2. Determine the direction of rotation of the drive sheave to indicate which side of the belt will be in tension and which side will be "slack." At the moment of start-up, carefully watch the slack side of the belt while listen for an excessive amount of "squealing" or "chirping" indicating that slippage is occurring. A slight bowing of the slack side during start up and during normal operation usually indicates adequate tension. Operate the drive system for about 5 min to seat the belts in the sheave grooves. Again, observe the belts for a slight bowing of the slack side. If the slack side remains taut during normal operation, the drive is too tight.
3. Check the tension on newly installed belts and sheaves several times during the first 24 h of operation by observing the slack side span.

FIGURE 18.11 Adjusting belt tension based on deflection at center span.

sheave from the taper lock bushing and try it again. If, after two to three attempts, you may have to try to rotate the sheave to a different angular position and try again. If after several attempts, regardless of what angular position the sheave was positioned, if the face or rim runout is unacceptable, get a new sheave and bushing. The general guideline for acceptable face runout is 1 mil/6 in. of sheave diameter.

Face and rim runout should gradually and linearly increase or decrease as the shaft is rotated. Notice in Figure 18.16 that through 180° of rotation there is no face runout. Through the other 180°, the runout increases and then decreases. This is an indication that the sheave is permanently bent and should be discarded.

A word of warning. I have witnessed, on several occasions, people attempting to align belt/sheave drive systems without checking the face and rim runout. All of the alignment measurement methods and tools rely on the fact that the outer surface of the sheave (or the "V" of the sheave) is perfectly perpendicular to the centerline of rotation of the shaft that it is attached to. If you would prefer not to check this, be ready to perhaps spend several days attempting to align the two machines and get absolutely nowhere.

18.8 TYPES OF SHEAVE MISALIGNMENT CONDITIONS

The apparent simplicity of these drives obscures the complexity of the misalignment condition that may exist. Figure 18.17 shows the three basic types of misalignment that can occur: offset, pitch, and skew conditions. Usually there is a combination of all three of these

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FIGURE 18.12 Rim runout check on sheave.

conditions as shown in the lower illustration in Figure 18.17. Adjustments to the machines may therefore require precise, controlled moves in the vertical, lateral, and axial directions.

18.9 USING A STRAIGHTEDGE TO MEASURE MISALIGNMENT

Belt and sheave driven equipment poses a slightly different type of alignment problem than equipment directly coupled together. The basic objective is to insure that the shaft centerlines are parallel to each other.

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FIGURE 18.13 Face runout check on sheave.

For decades, the most widely used alignment tool is either a string or a straightedge. Today there are far more elaborate ways to perform belt–sheave alignment as shown later in this chapter, but most often, acceptable belt alignment can be accomplished using a simple straightedge. Bear in mind that most manufacturers of belt and sheave drives suggest that the sheaves should be aligned to within 1/8 in. per foot distance between shaft centerlines. That is about 11 mils/in., much more forgiving that direct drive systems, which are typically aligned to around 1 mil/in.

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FIGURE 18.14 Rim runout check on sheave.

Straightedges work fine for distances under 4 ft but when the distance between the driver shaft and the driven shaft begin to exceed that, a string should probably be used. One tool developed by Max Roeder called the A-String works extremely well and produces very accurate results shown in Figure 18.18 and Figure 18.19. The A-String has an adjustable base that enables one to compensate for centerline offset of sheaves as shown in Figure 18.20. To properly align sheaves, you must compensate for any difference in the actual center of the V in each sheave. Measure the width of the groove; then measure the flange outer thickness on each sheave to determine what the offset may have to be if the outer flange widths are not the same.

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FIGURE 18.15 Face runout check on sheave.

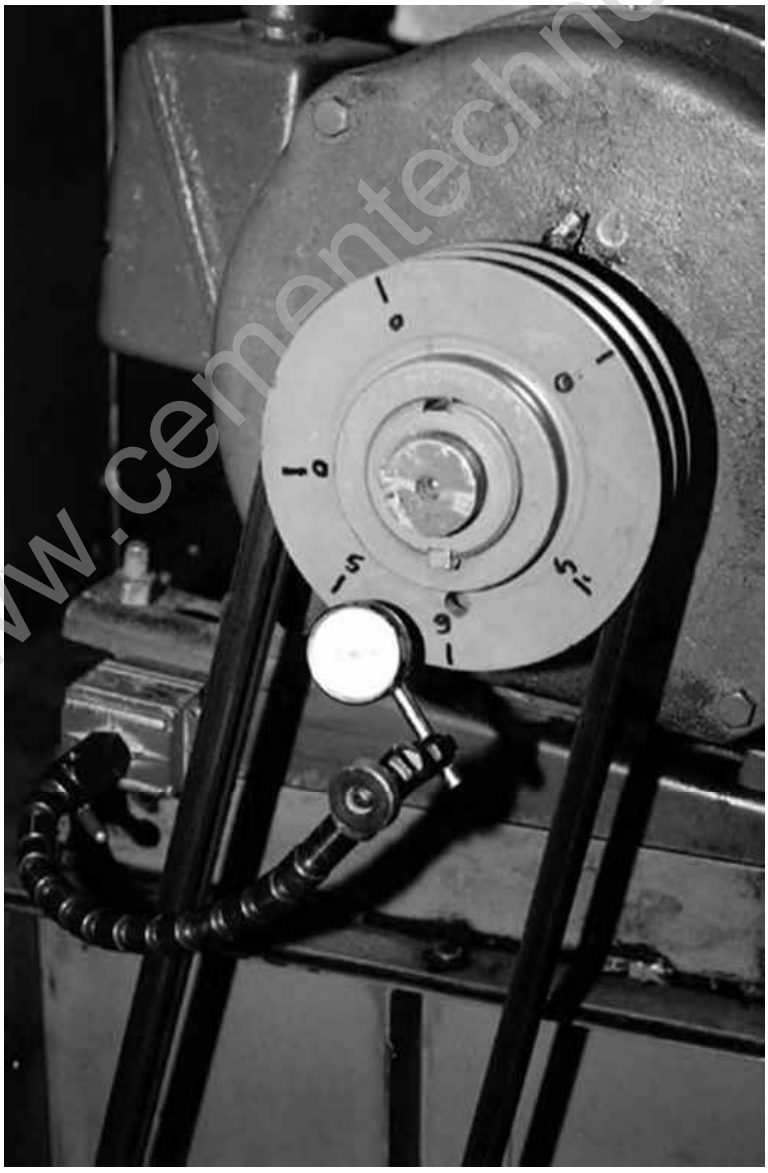


FIGURE 18.16 Bent sheave.

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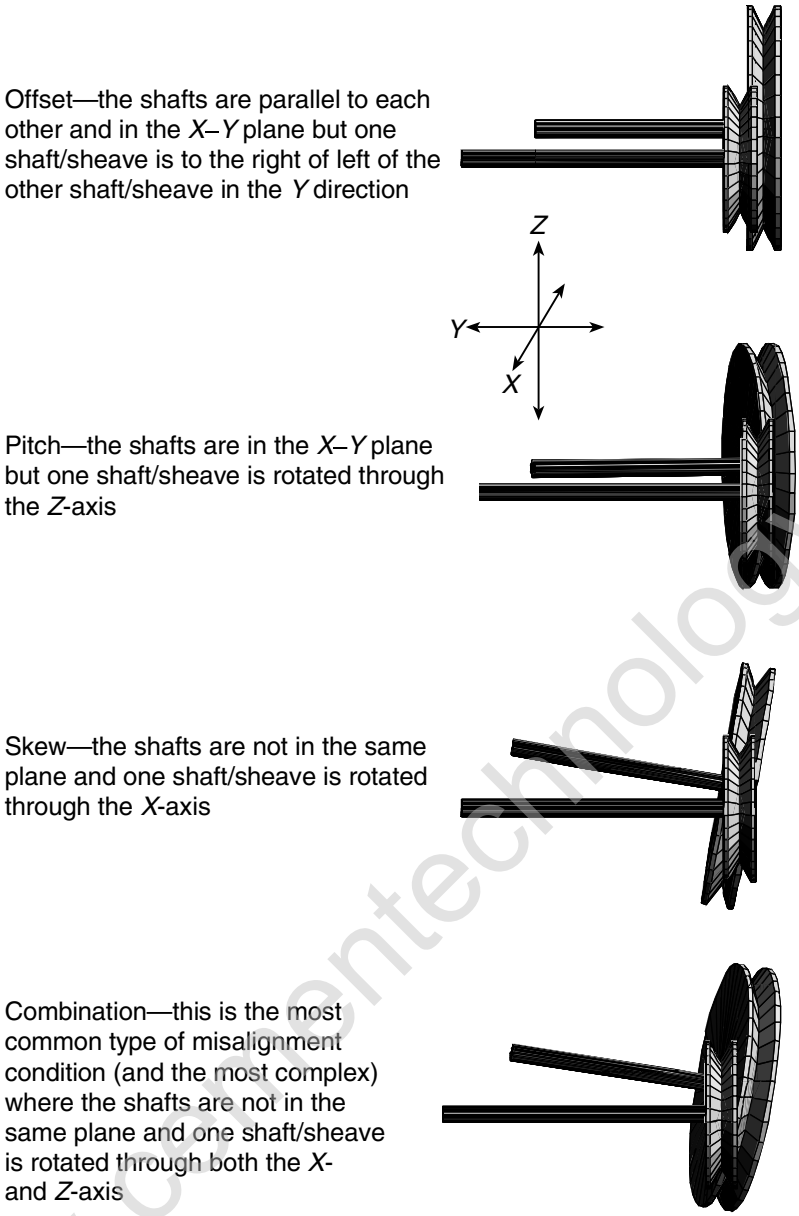


FIGURE 18.17 (**See color insert following page 322.**) Types of belt and sheave misalignment conditions.

18.10 MEASURING THE MISALIGNMENT AT THE SHEAVES

To measure the amount of offset, pitch, and skew that exists between the shafts and their sheaves, measurements with a straightedge need to be taken at two different places on the outer surface of the sheaves as shown in Figure 18.21. Measure the distances across each sheave at the upper and lower gap measurement locations. Determine what type of gap condition you have based on the four different configurations shown in Figure 18.22. Using feeler gauges, measure and record the amount of the gaps (in mils) between the straightedge and the surface of the sheaves as shown in Figure 18.23 and Figure 18.24.

18.11 V-BELT MACHINE MEASUREMENTS

In addition to the gap measurements taken on the sheaves as shown in Figure 18.21 through Figure 18.24, dimensional measurements of the two machines need to be taken as shown in

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FIGURE 18.18 A-String sheave alignment tool.

Figure 18.25. A recording sheet shown in Figure 18.26 can be used to record all the required information to generate an alignment model of the misalignment condition.

18.12 MODELING V-BELT ALIGNMENT PROBLEMS

Alignment models can also be used to visualize the misalignment condition on belt and sheave drive equipment. You will have to generate two different views of your drive system. One view will be generated from above (i.e., the top view), which will show any offset and pitch conditions between the two sheaves. The end view will show any offset and skew conditions that exist between the two sheaves. You can use two T-bar overlays (see Face–Rim graphing method on Chapter 11) to represent each shaft/sheave.

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FIGURE 18.19 A-String sheave alignment tool.

18.13 V-BELT ALIGNMENT MODELING SAMPLE PROBLEM

Figure 18.27 shows an electric motor driving a fan. The critical dimensions needed to generate an alignment model of this drive system are shown in Figure 18.27. Use one of the T-bar overlays to scale off the distance from the inboard-to-outboard bolts of the motor (15 in.) and the distance from the inboard bolt of the motor to the edge of the sheave where the straightedge measurements are taken (5 in.). On the top of the T-bar overlay, scale off the 6 in. distance to represent where the straightedge gaps or contact point were measured. Similarly for the fan, use the other T-bar overlay to scale off the distance from the inboard-to-outboard bolts of the fan (12 in.) and the distance from the inboard bolt of the fan to the edge of the sheave where the straightedge measurements are taken (4 in.). On the top of the T-bar overlay, scale off the 8 in. distance to represent where the straightedge gaps or contact points were measured.

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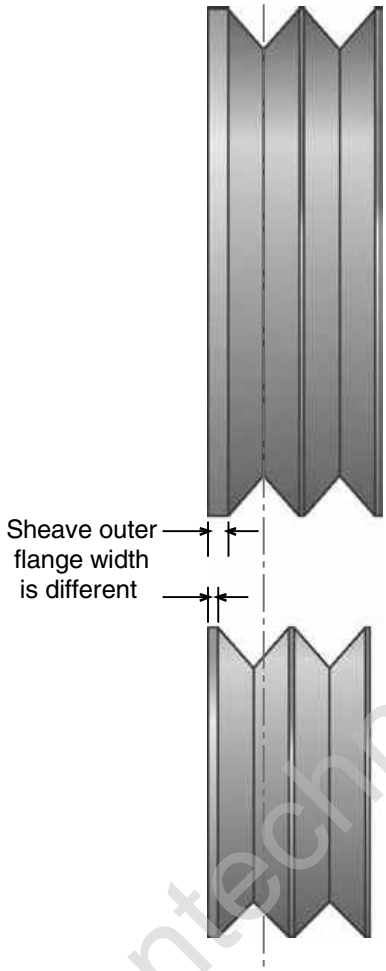


FIGURE 18.20 Measuring centerline offset of sheaves.

Figure 18.28 shows the top view of the motor and fan shafts. Notice that you want to pitch each T-bar overlay at the midpoint between the gap you measured at the top of each sheave and the gap you measured at the bottom of each sheave. In the case of the motor sheave gaps, the upper gap was 10 mils and the lower gap was 16 mils. (10 + 16 mils = 26 mils; 26 mils/2 = 13 mils, i.e., the midpoint). In the case of the fan sheave gaps, the upper gap was 26 mils and the lower gap was 8 mils. (26 + 8 mils = 34 mils; 34 mils/2 = 17 mils, i.e., the midpoint). Notice that the motor shaft and fan shaft are not parallel to each other.

Figure 18.29 shows the end view of the motor and fan shafts. In this view, the motor is toward us and the fan is away from us. Also notice that in this particular case, the distance from the upper to lower straightedge measurements on the motor was 6 in., the same distance the straightedge measurements were taken from the inboard-to-outboard edge of the motor as shown in the top view. If the distance from the upper to lower straightedge measurements is not the same as it was between the inboard and outboard edges, you must scale off whatever the upper to lower straightedge measurements actually were when viewing the shafts and sheaves in the end view. Again, notice that you want to pitch each T-bar overlay at the midpoint between the gap you measured at the top (upper edge) of each sheave and the gap you measured at the bottom (lower edge) of each sheave. In the case of the motor sheave gaps, the upper gap was 10 mils and the lower gap was 16 mils. The midpoint at the upper edge of the motor sheave is 5 mils, and the midpoint at the lower edge of the motor sheave is 5 mils. The T-bar for the motor should be pitched to intersect the midpoint at its upper and lower

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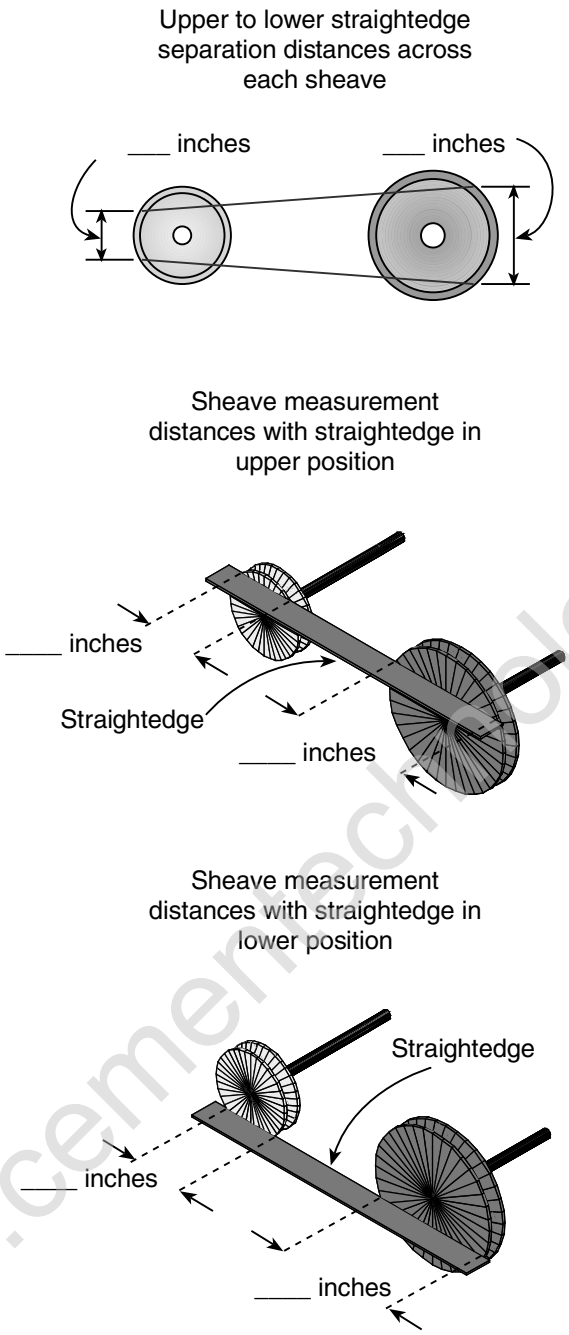


FIGURE 18.21 Measure the gap conditions on the sheaves at two different locations.

points. In the case of the fan sheave gaps, the upper gap was 26 mils and the lower gap was 8 mils. The midpoint at the upper edge of the fan sheave is 13 mils, the midpoint at the lower edge of the fan sheave is 4 mils. The T-bar for the fan should be pitched to intersect the midpoint at its upper and lower points.

Why do we position the T-bar overlays at the midpoints of the gaps? Because the actual centerline of rotation is midway between the 6 in. (on the motor) and 8 in. (on the fan) measurement points where the straightedge was positioned on each sheave. Graphically, the top of the T-bar overlay is represented as a straight line. When viewing the sheaves from the top or end views, the sheaves would actually appear as ellipses.

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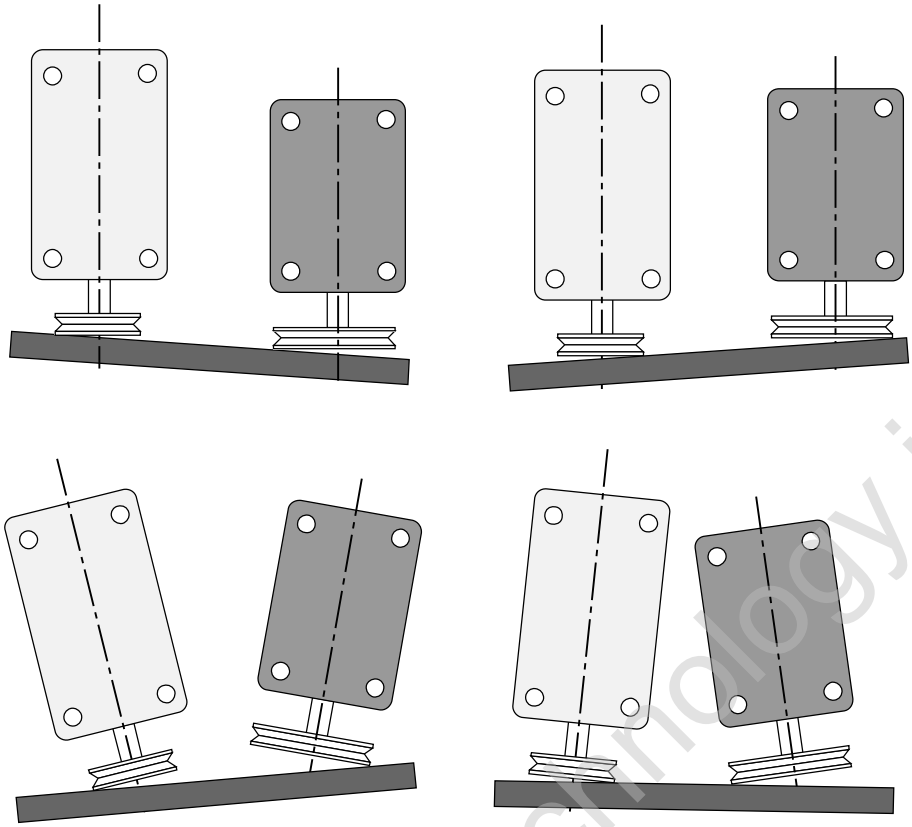


FIGURE 18.22 Four possible gap conditions.

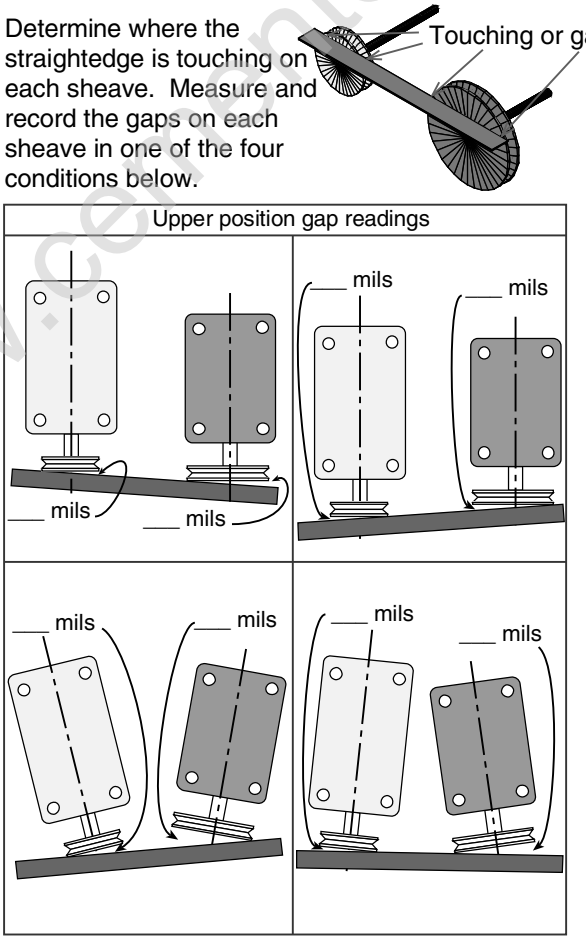


FIGURE 18.23 Measure the gap conditions at the top of the sheaves.

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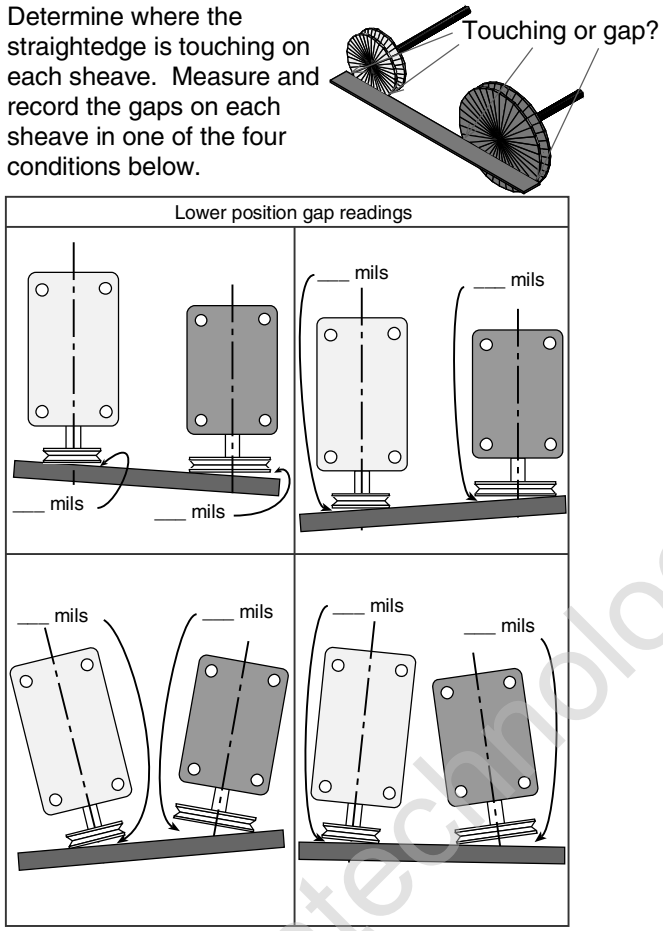


FIGURE 18.24 Measure the gap conditions at the bottom of the sheaves.

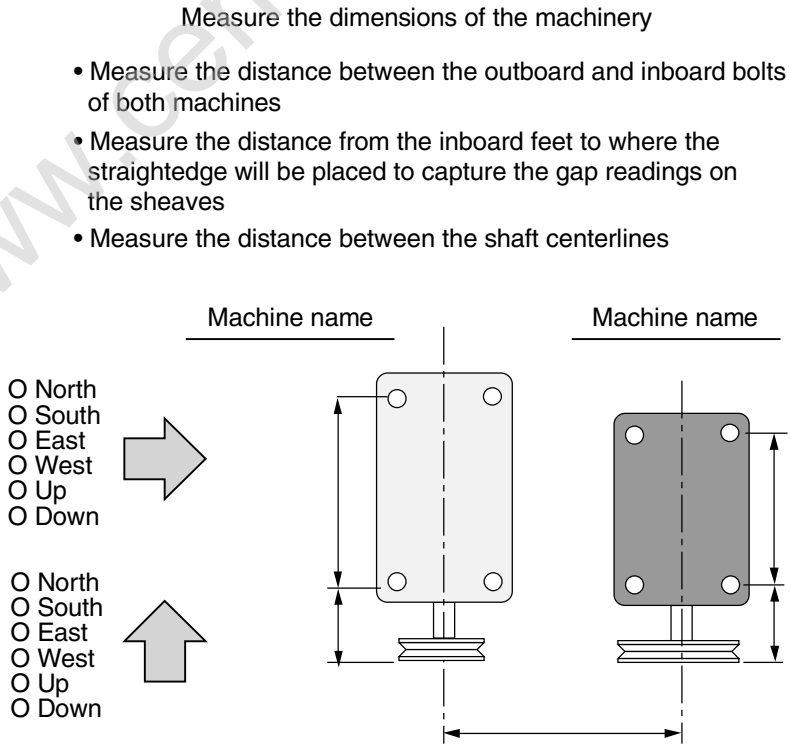


FIGURE 18.25 Dimensional measurements of the machines.

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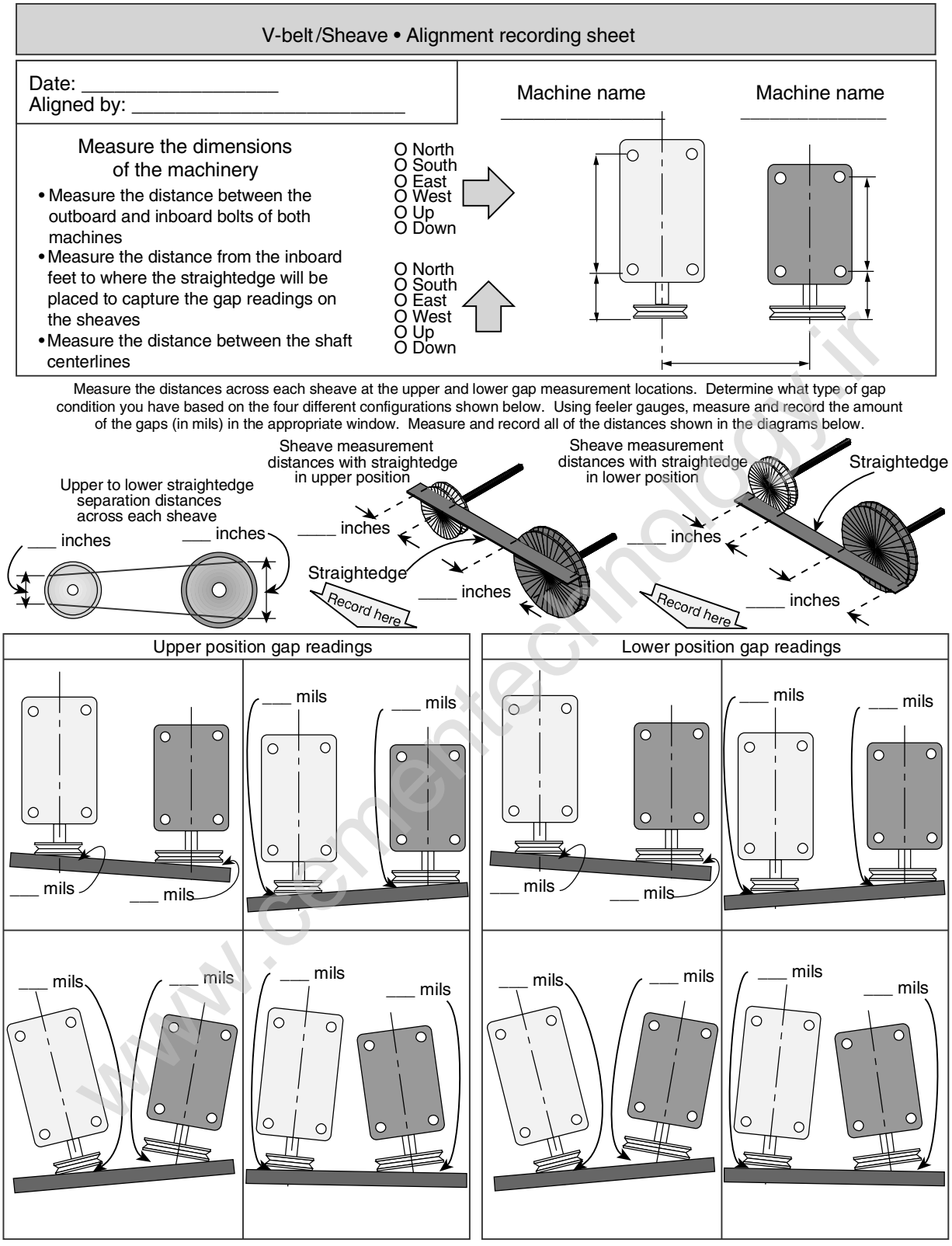


FIGURE 18.26 (**See color insert following page 322.**) V-belt and sheave alignment recording sheet.

To correct the misalignment (i.e., nonparallelism) between the motor and fan shafts in the top view, the next step would be to determine the allowable lateral movement envelope on both machines. That is, how much room is there between the foot bolts and the holes on the motor and the fan in the north to south direction? You will have to remove the foot bolts at the inboard and outboard end of each machine, look down the hole and see how much room

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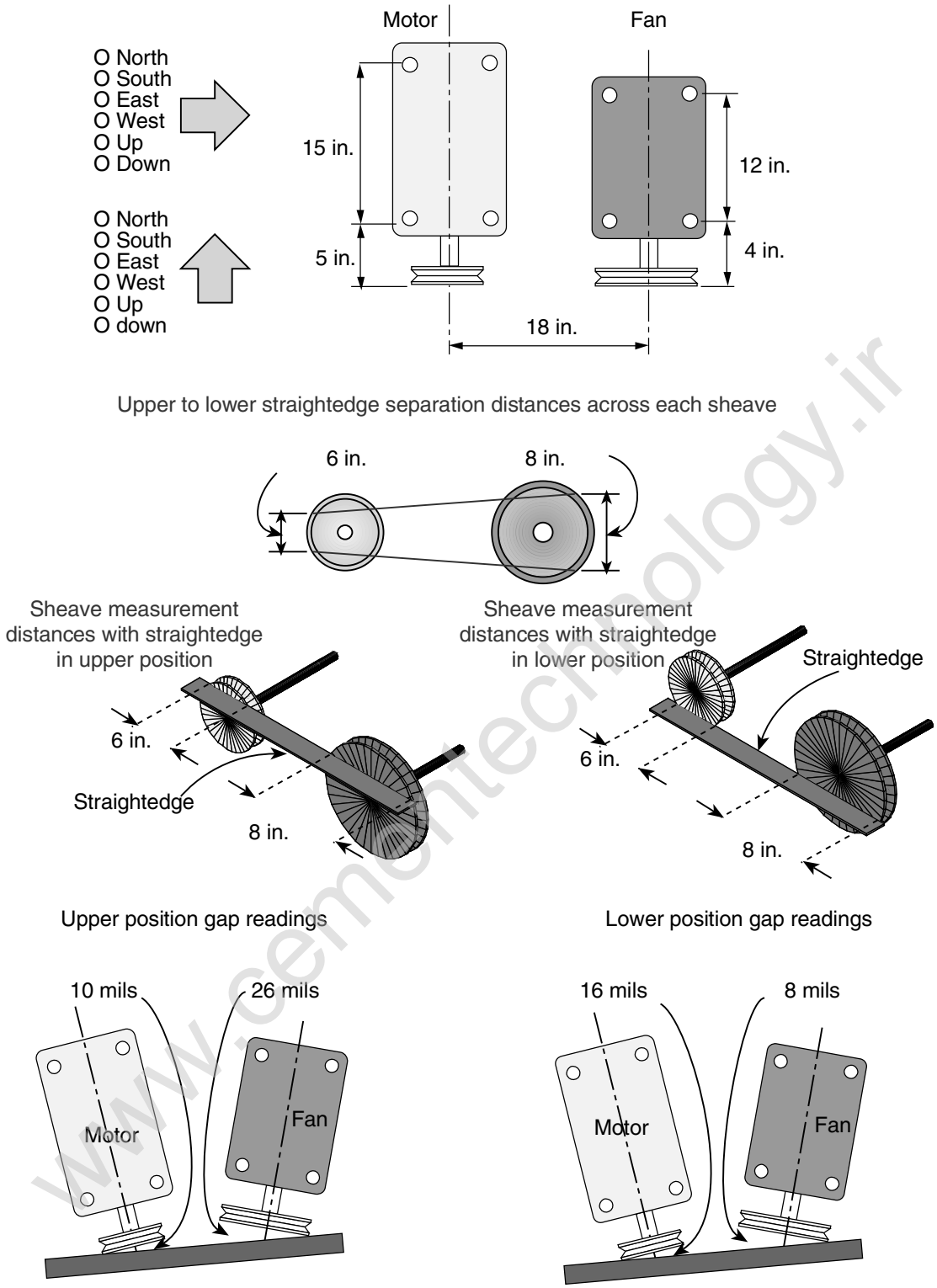


FIGURE 18.27 Fan and motor V-belt drive dimensions and sheave gaps.

is there to move each machine north to south before you get bolt bound. As shown in many of the alignment modeling examples in Chapter 8 and Chapter 10 through Chapter 14, superimpose these boundary conditions on the alignment model at the machinery feet on the motor and fan, then position the motor T-bar and the fan T-bar to bring the top of each T-bar into a straight line (i.e., position the centerlines so they are parallel to each other). Figure 18.30 shows one possible alignment solution in the top view. Figure 18.31 shows one possible alignment solution in the end view.

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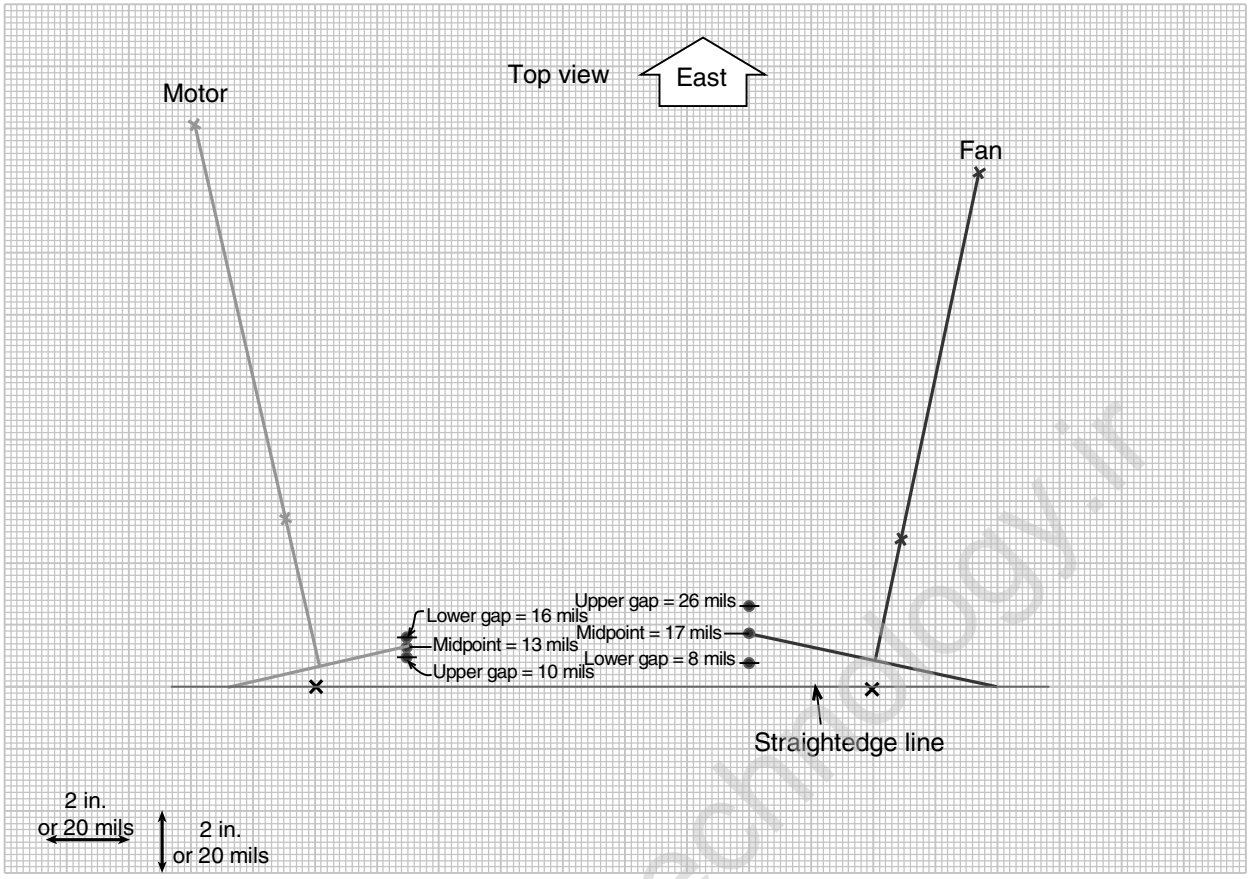


FIGURE 18.28 Top view of motor and fan shafts.

18.14 LASER ALIGNMENT SYSTEMS FOR V-BELTS AND SHEAVES

Around 1998, several companies began to develop laser alignment systems for V-belt drives. There are two different approaches that these manufacturers have taken. One method is to attach the laser to the outer surface of one of the sheaves and project the laser beam toward

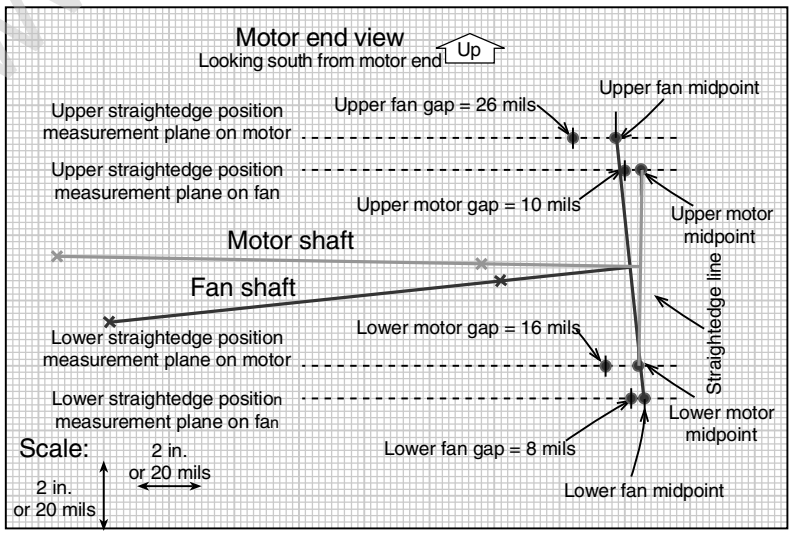


FIGURE 18.29 End view of motor and fan shafts.

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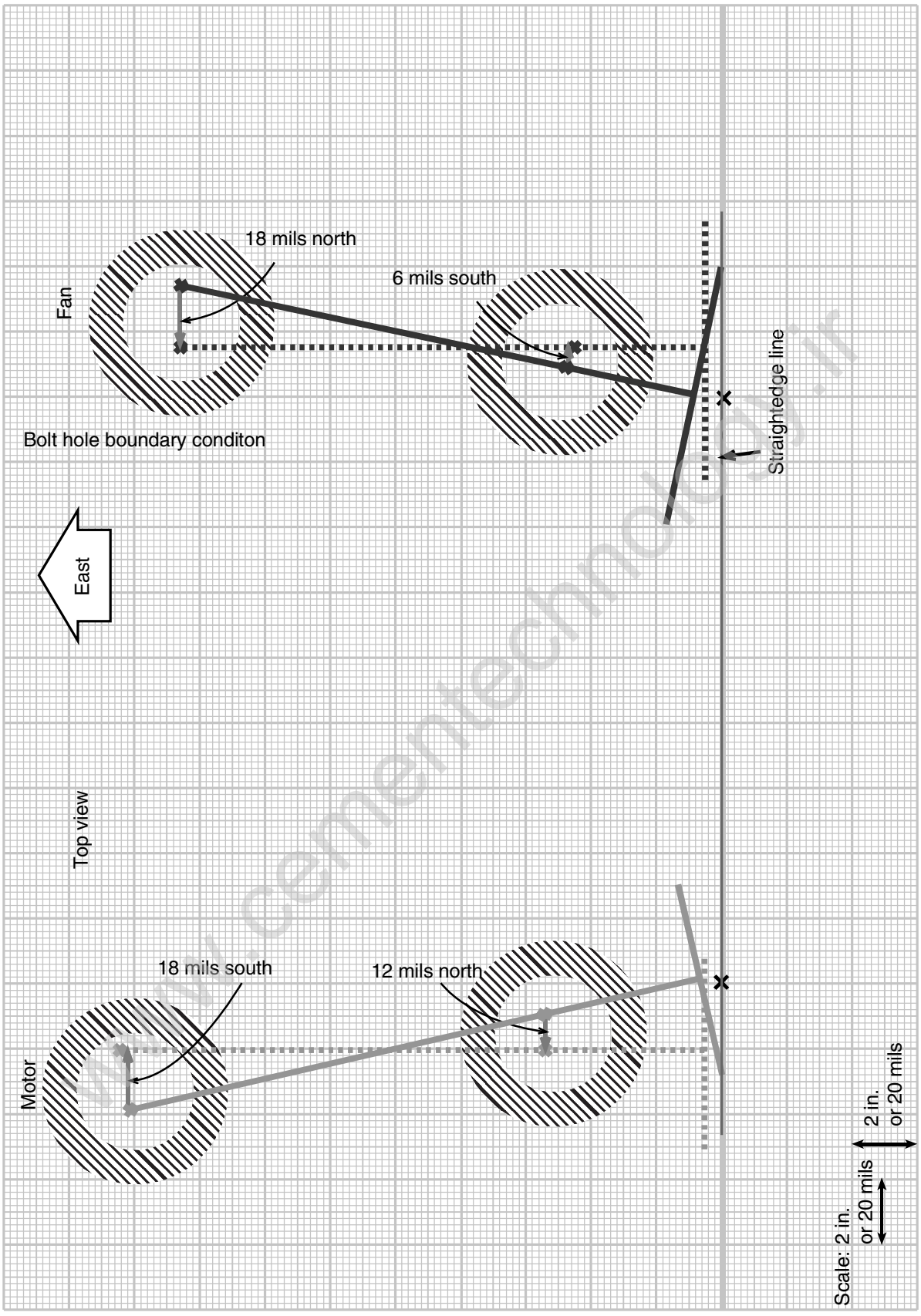


FIGURE 18.30 (**See color insert following page 322.**) Possible alignment corrections for the motor and fan in the top view.

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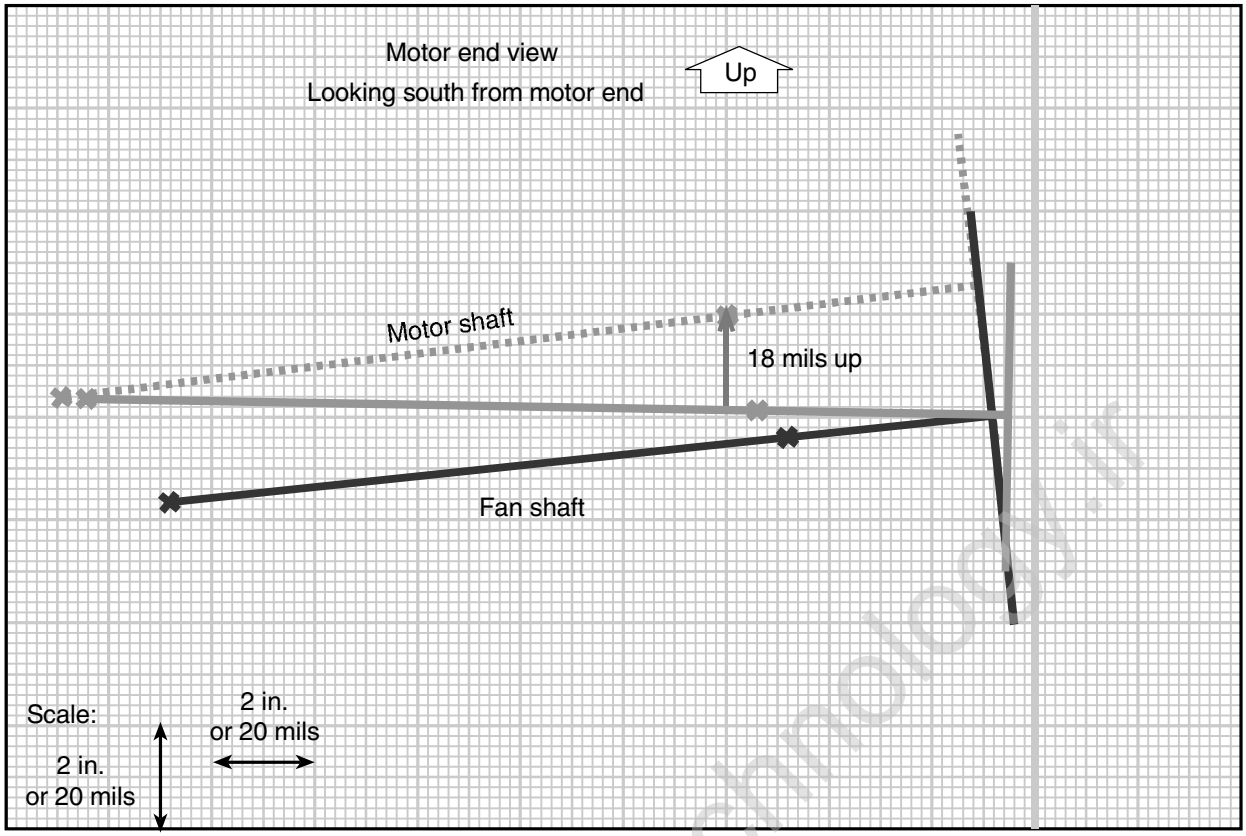


FIGURE 18.31 Possible alignment corrections for the motor and fan in the end view.

visual sighting targets attached at different points on the other sheave. The laser and the target are held in place with magnets. Figure 18.32 through Figure 18.35 show systems that use this approach.

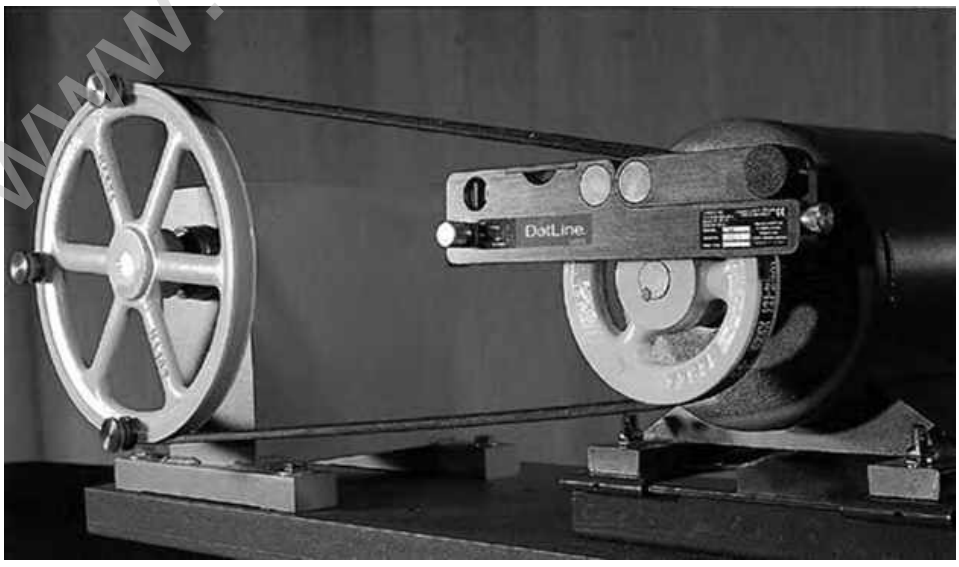


FIGURE 18.32 Dotline laser system. (Courtesy of Ludeca Inc., www.ludeca.com, Doral, FL. With permission.)

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FIGURE 18.33 SheaveMaster system. (Courtesy of Ludeca Inc., www.ludeca.com, Doral, FL. With permission.)



FIGURE 18.34 D200 BTA Digital system (Courtesy of Damalini AB, Molndal, Sweden. With permission.)

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FIGURE 18.35 D80 BTA Compact system (Courtesy of Damalini AB, Molndal, Sweden. With permission.)

The other approach is to position the laser into the grooves of one of the sheaves and a photodiode target into the grooves of the other sheave. Figure 18.36 and Figure 18.37 show systems that use this approach.

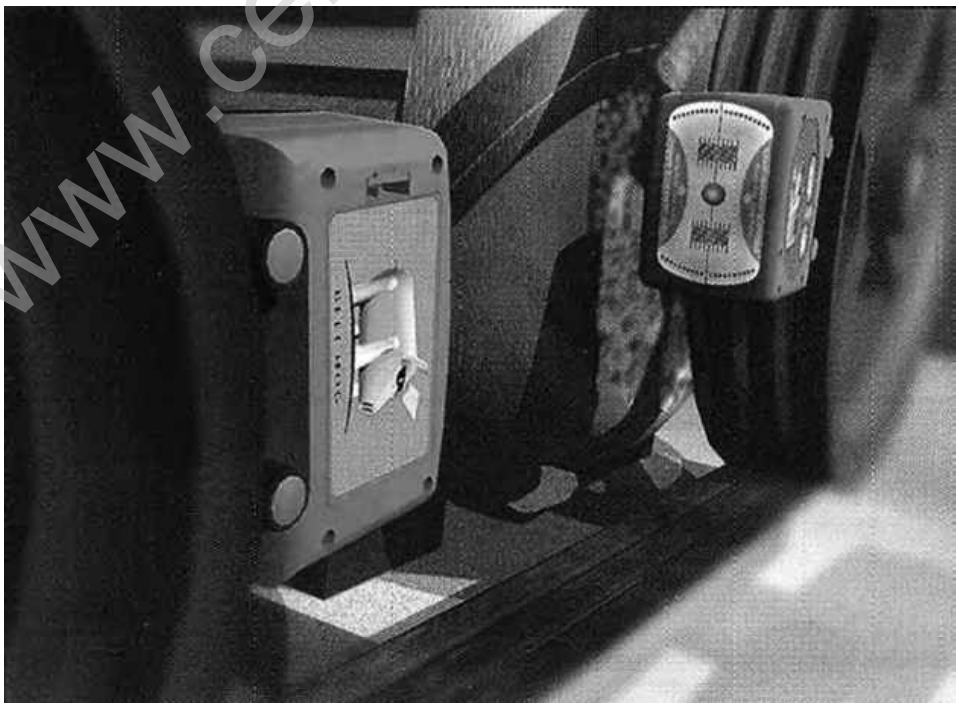


FIGURE 18.36 Belt Hog. (Courtesy of VibrAlign Inc., Richmond, VA. With permission.)

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FIGURE 18.37 S600 system. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)

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19 Bore Alignment

The alignment of rotating machinery shafts, as discussed in the previous chapters, concentrates on measuring the centerline of rotation of one shaft with respect to another shaft. These shafts are usually solid cylinders of various lengths supported by a bearing at each end. The position of the two bearings that support each shaft dictates the location of that shaft's centerline of rotation. If we are aligning two shafts, each of which is supported in two bearings, then the goal is to align the centers of all four bearings in the two shafts. Therefore, shaft alignment and bearing alignment both really mean the same thing.

If the shafts were made out of a perfectly clear, transparent material (e.g., glass), we could then visually look down the centers of the transparent shafts from each end and observe if the centers of the supporting bearings were collinear. Figure 19.1 shows what you might see assuming that your line of sight was coincident with both centerlines of rotation.

But on the other hand the shafts of machinery are not made out of a transparent material and we cannot look straight down the centers of the shafts. Consequentially, to find the centerline of rotation of these shafts, we have to observe a point at a fixed distance from the centerline—typically the outer surface of the shaft. All of the tools and methods used to perform shaft alignment as shown in Chapter 10 through Chapter 15 are based on this premise.

What if we want to align two hollow cylinders with each other where we could look down the center of the cylinders? What if these hollow cylinders could not rotate? What if one cylinder could rotate but the other could not?

19.1 ALIGNING A ROTATING SHAFT WITH A STATIONARY HOLLOW CYLINDER

Figure 19.2 shows an electric motor and a hollow cylinder. The electric motor will eventually drive a screw that is placed inside the barrel (i.e., hollow cylinder). The screw is used to move a viscous material down through the barrel and is expelled at the discharge end of the barrel under pressure. Drive systems of this type are called extruders and are used extensively in the food and plastics industry. In some cases, the motor directly drives the screw, in other cases, the motor is flexibly coupled to the input shaft of a gearbox and the output shaft of the gearbox drives the screw. It is not uncommon to have only 5–20 mils of clearance between the blades of the screw and the bore of the barrel so any misalignment between the drive shaft and the screw will cause the screw to drag against the inside of the barrel wearing away both the screw and the bore of the barrel. Frequently, the screw is not supported in bearings and the viscous material, under pressure, will act to force the screw to the center of the barrel.

To align the drive shaft with the barrel, the screw is removed so that one can visually sight down the barrel to the end of the drive shaft. The goal is to align the centerline of rotation of the motor shaft with the centerline of the bore of the barrel. Understand that it is possible to align the centerline of the bore of the barrel to the center of the end of the shaft and still have a misalignment problem as shown in Figure 19.3.

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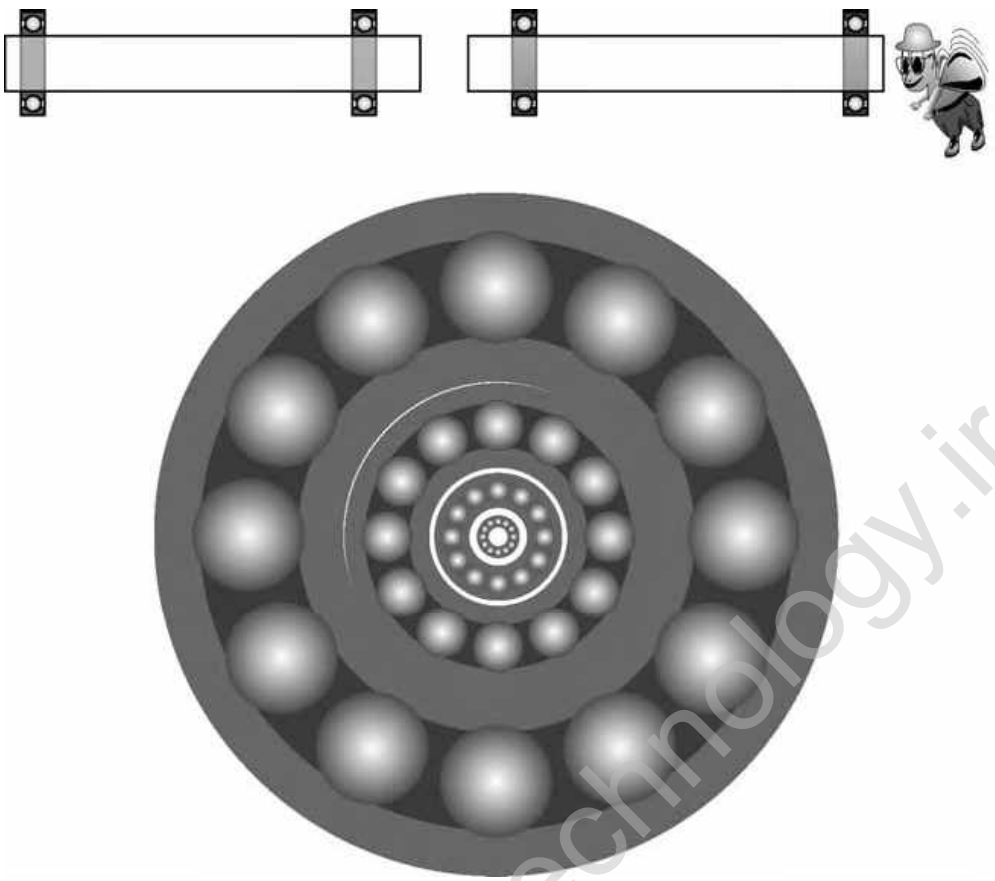


FIGURE 19.1 View through the axis of rotation.

One way to accomplish this measurement is to perform the double radial method (refer to Chapter 12). As shown in Figure 19.4, the dial indicator measurements can be taken on the inside bore of a cylinder rather than capturing the measurements on the outside of a cylinder (e.g., a shaft). Remember that you will have to compensate for the bracket sag that occurs at both the near and far indicators. You also have to be aware of the fact that you are reading an inside diameter and the sign (+/−) of the measurement from the top to the bottom (or side to side) will be opposite of what it would be if you were measuring the outside diameter.

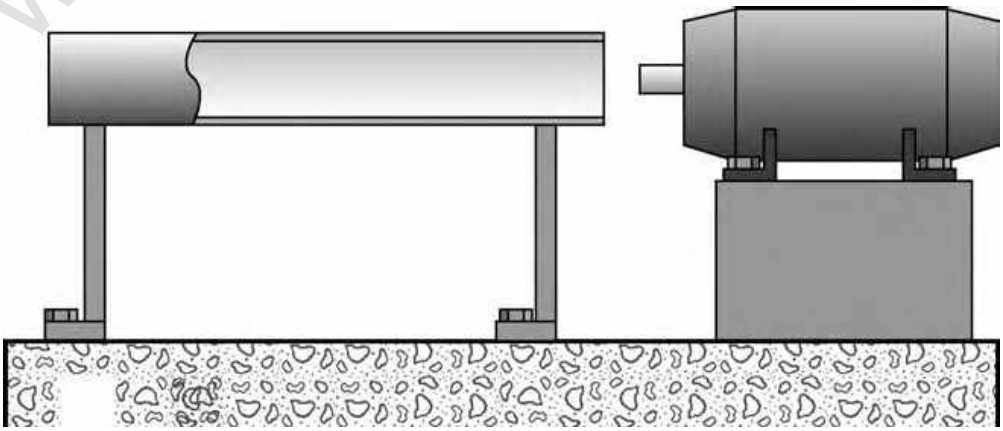


FIGURE 19.2 Motor and barrel.

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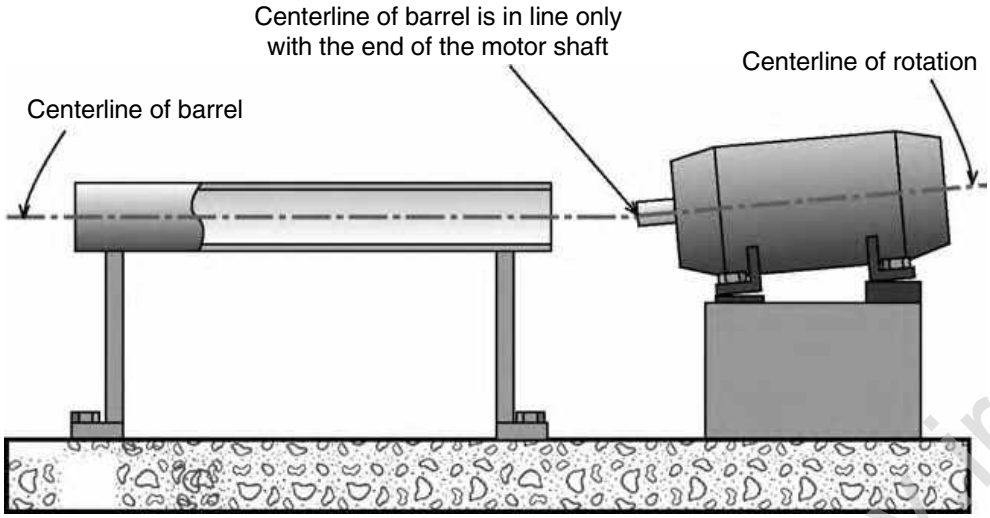
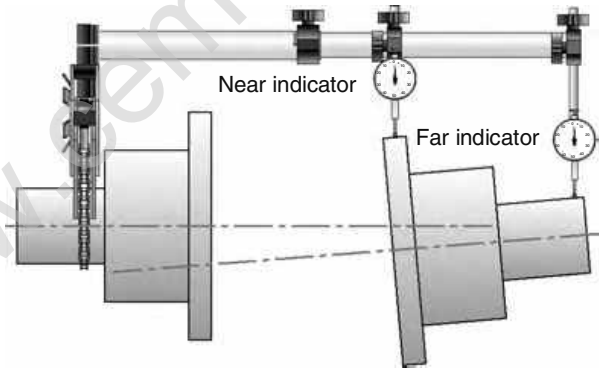


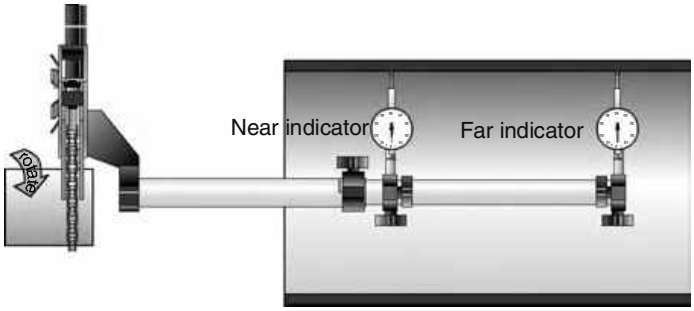
FIGURE 19.3 Pure angular misalignment of motor shaft and barrel centerline.

For example, with the indicator set up to take a reading on the inside of the barrel, if the indicator is zeroed at the top of the inside of the barrel then rotated to the bottom and the dial indicator measured a +20, the barrel appears to be ''high'' at that point. If instead, the indicator was set up to take a reading on the outside of the barrel, if the indicator is zeroed at the top of the outside of the barrel then rotated to the bottom, the dial indicator would measure a −20.

Figure 19.5 shows the dimensions and double radial measurements that were taken on the motor and barrel. Figure 19.6 and Figure 19.7 show the side and top view alignment models.



Taking measurements on the outside of a cylinder



Taking measurements on the inside of a cylinder

FIGURE 19.4 Double radial method measuring outside and inside of cylinders.

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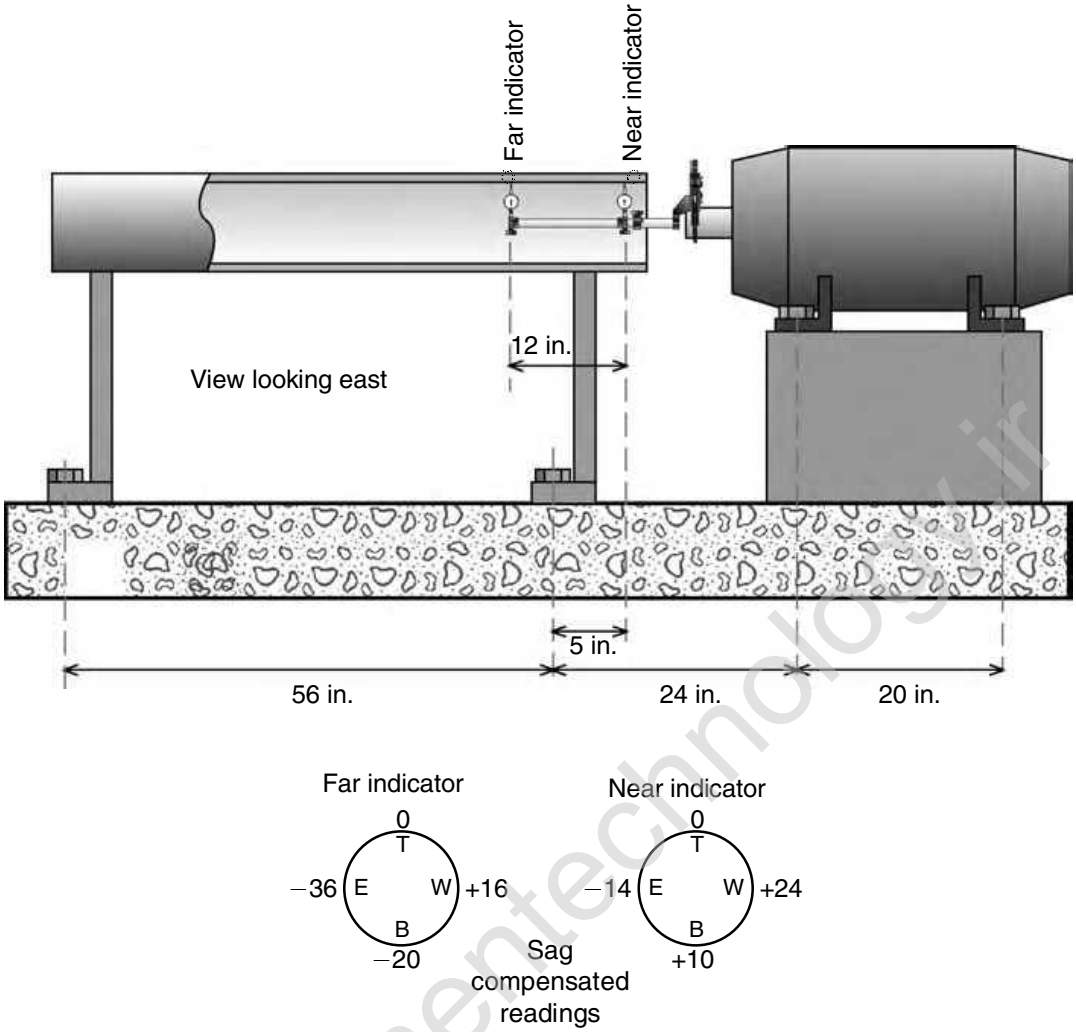


FIGURE 19.5 Motor and barrel dimensions and measurements taken on bore using the double radial method.

19.2 ALIGNING TWO HOLLOW CYLINDERS

Next, let us examine how you would align two hollow cylinders with each other. The assumption is that the cylinders are perfectly straight (i.e., not bowed) and that the inside diameters of the cylinders are consistent along the full length of both cylinders. Either cylinder may, or may not, have the capability to rotate on an axis that is coincident with the centerline of its bore. The measurement device that we will use for this basic procedure is an optical jig transit (refer to Figure 6.11) held in position with an appropriate tripod or stand. The stand must have a translation slide and a precision vertical lift. The jig transit must also have an optical micrometer attached to the end of the telescope barrel (see Figure 6.15). The optical micrometer can be positioned to translate either the horizontal or vertical crosshair by rotating the micrometer through a 90° arc on the end of the telescope. The problem with doing this is that you run the risk of inadvertently moving the scope to a different line of sight, if you jar the scope when repositioning the optical micrometer. To reduce the need to rotate the micrometer, a coordinate optical micrometer enables the user to measure target offsets in both the vertical and horizontal planes without having to rotate a single axis micrometer 90° on the barrel of the telescope to capture both measurements as shown in Figure 19.8.

An additional tooling that is required are bore sighting targets and fixtures to position and hold the sighting targets in the cylinders. Optical bore sighting targets are shown in

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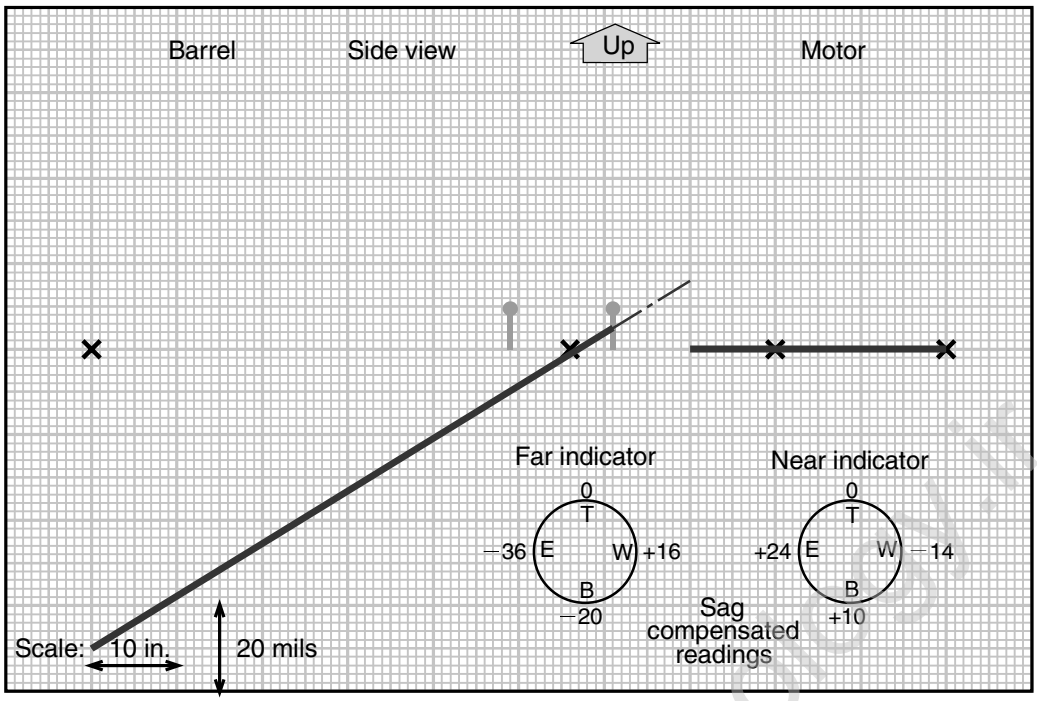


FIGURE 19.6 Side view alignment model of motor centerline and barrel centerline.

Figure 19.8. These sighting targets are fabricated from nylon with an accurately painted pattern of paired lines set 90° apart precisely positioned from the center of the target. A small battery operated light source (e.g., a flashlight) can be used to illuminate the translucent target from behind, as this target will usually be placed inside a dark cylinder. Other sighting targets shown in Figure 19.9 are made out of thin wires or a pattern cut out of a thin piece of metal, which will allow you to view objects behind the target acting as if they were transparent. The see-through target is typically mounted as the nearest target to the jig transit

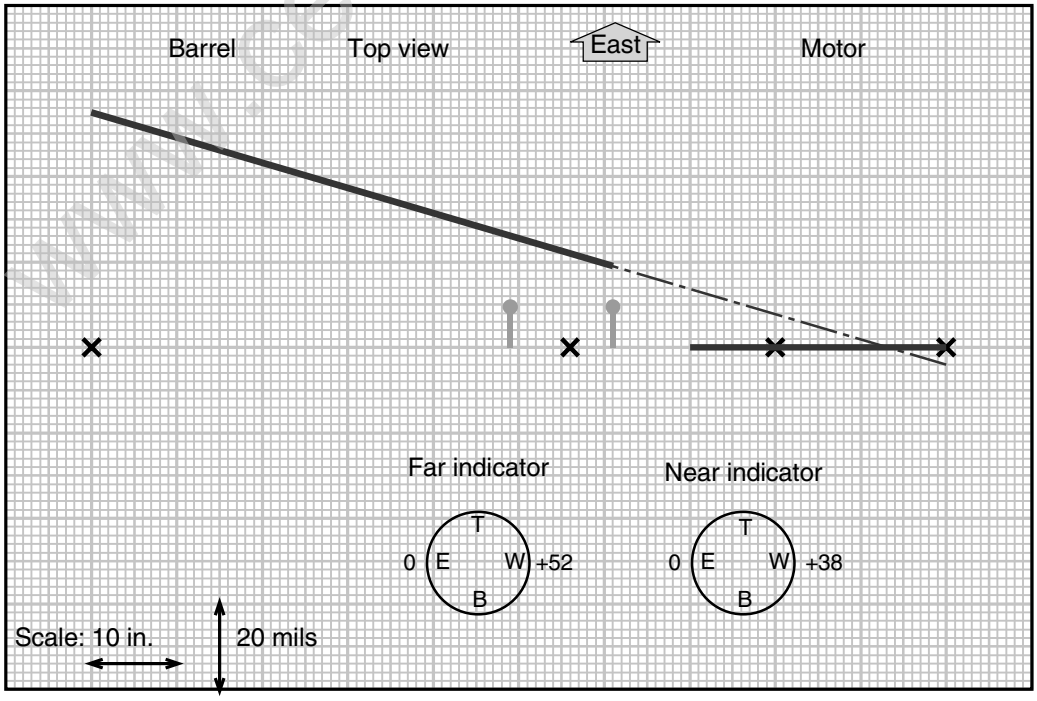


FIGURE 19.7 Top view alignment model of motor centerline and barrel centerline.

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FIGURE 19.8 Coordinate optical micrometer. (Courtesy of Brunson Instruments, Kansas City, MO. With permission.)

enabling visual sighting of targets down range without having to move the see-through target from its position.

The sighting targets will be placed at different points in the center of the cylinder and they do not have the capacity to automatically center themselves. Therefore a sighting target

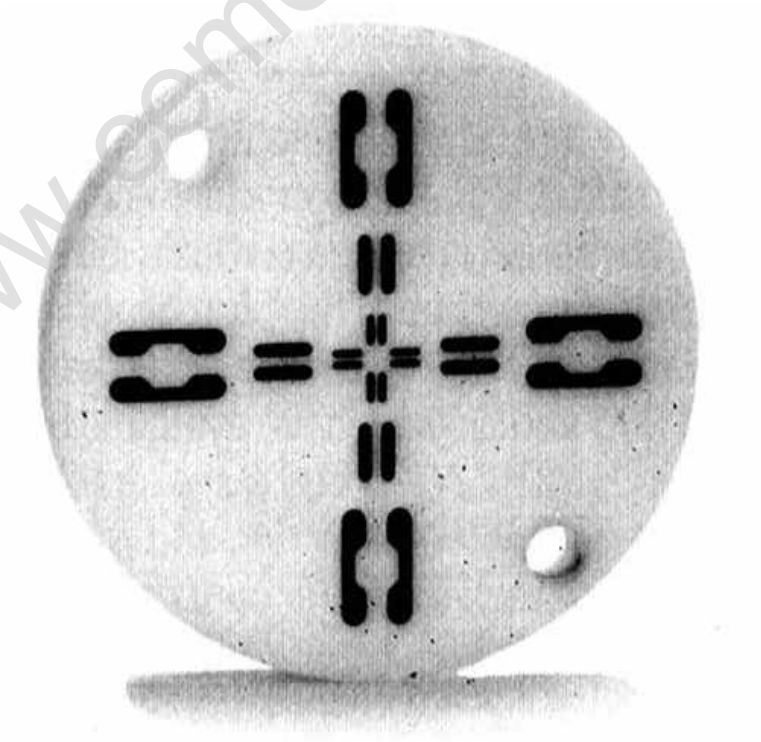


FIGURE 19.9 Translucent bore sighting target. (Courtesy of Brunson Instruments, Kansas City, MO. With permission.)

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FIGURE 19.10 See-through bore sighting target. (Courtesy of Brunson Instruments, Kansas City, MO.)

fixture is needed not only to hold the target in position, but also to center the target in the cylinder. This is true whether the sighting targets are visual targets like the ones shown in Figure 19.9 and Figure 19.10 or they are photodiode targets used with laser bore alignment systems. Fixtures similar to the ones shown in Figure 19.11 and Figure 19.12 are examples of adjustable target holding and centering devices needed for this process. The fixture on the left in Figure 19.11 uses eight setscrews, two at each 90° angle on the fixture to center and square the fixture in the bore of the cylinder. This fixture is held rigidly in a cylinder supported in



FIGURE 19.11 Bore sighting target fixtures.

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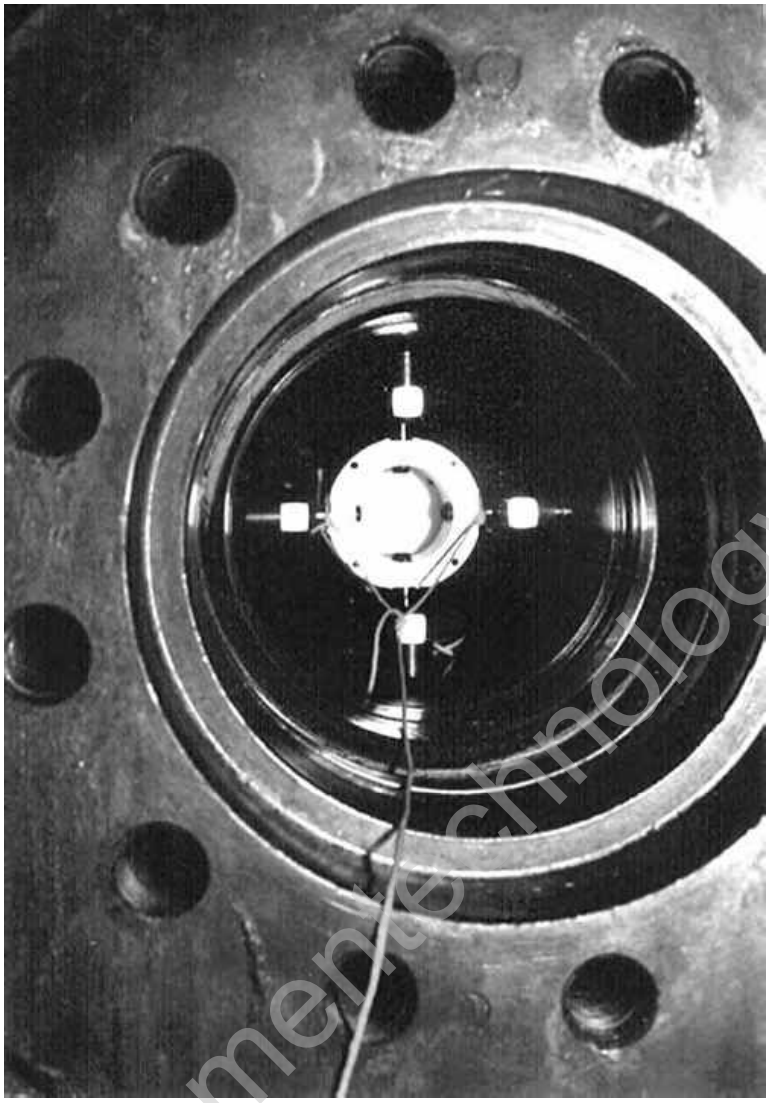


FIGURE 19.12 Bore target fixture inside a barrel.

bearings and the entire cylinder with the target installed can be rotated. The fixture on the right in Figure 19.11 is similar in design and function but with four bars that can move radially outward on guide bolts with ball bearings at each end of the bar that allow the entire target and fixture to rotate inside a nonrotatable cylinder. Figure 19.12 shows this fixture and a sighting target inside a barrel. An additional feature of this device is that the target can also be translated axially enabling several measurements to be taken along the entire length of the cylinder via a string that is used to pull the target through the barrel.

19.3 BASIC MEASUREMENT PRINCIPLES AND NOMENCLATURE

Figure 19.13 shows the typically used coordinate system and jargon. Figure 9.14 shows the basic layout of the measurement process, the hollow cylinders with their bore centerlines shown as dashed lines, and some of the terminology and nomenclature that will be used in this procedure. As illustrated, the centerline of the bore of the near and far cylinders can be out of alignment with each other in both the vertical and lateral (sideways) directions. Although the jig transit telescope has the capability of being leveled within one arcsecond (0.001 in. 17 ft), it is important to understand that the centerlines of the bores of the two cylinders can be in line with each other (coincident) but not necessarily level. Leveled and aligned does not mean

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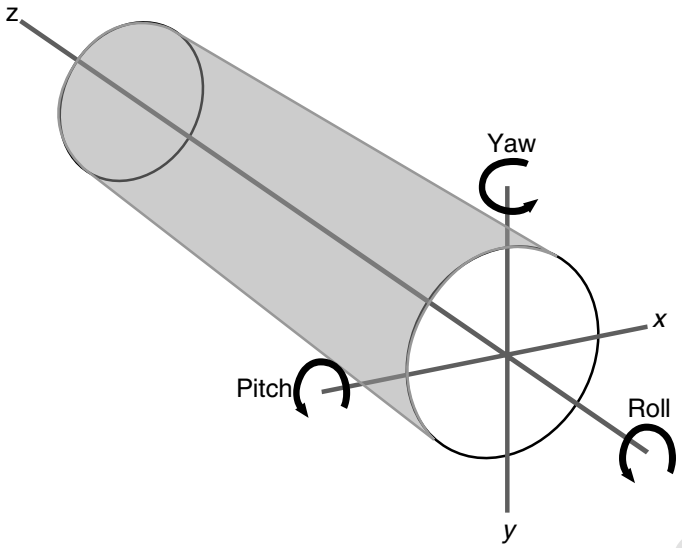


FIGURE 19.13 Coordinate system and rotation nomenclature.

the same thing (did I mention this before?). Therefore, the process of leveling the scope will not be discussed in this procedure.

Figure 19.15 shows an angled view of the near and far cylinders with targets situated at each end of both cylinders. These targets must be accurately positioned to insure that they are truly in the center of the cylinder. To accomplish this, the target must be able to rotate through at least a 180° arc and ideally through 360° of rotation. This can be done in one of two ways: rotating the entire cylinder itself or rotating the target inside the bore of the cylinder. Figure 19.16 shows what the observer would see from the jig transit position looking at a target at the near end of the near cylinder.

19.4 CYLINDER ALIGNMENT PROCEDURE

Aligning two cylinders with each other requires an eight-step process:

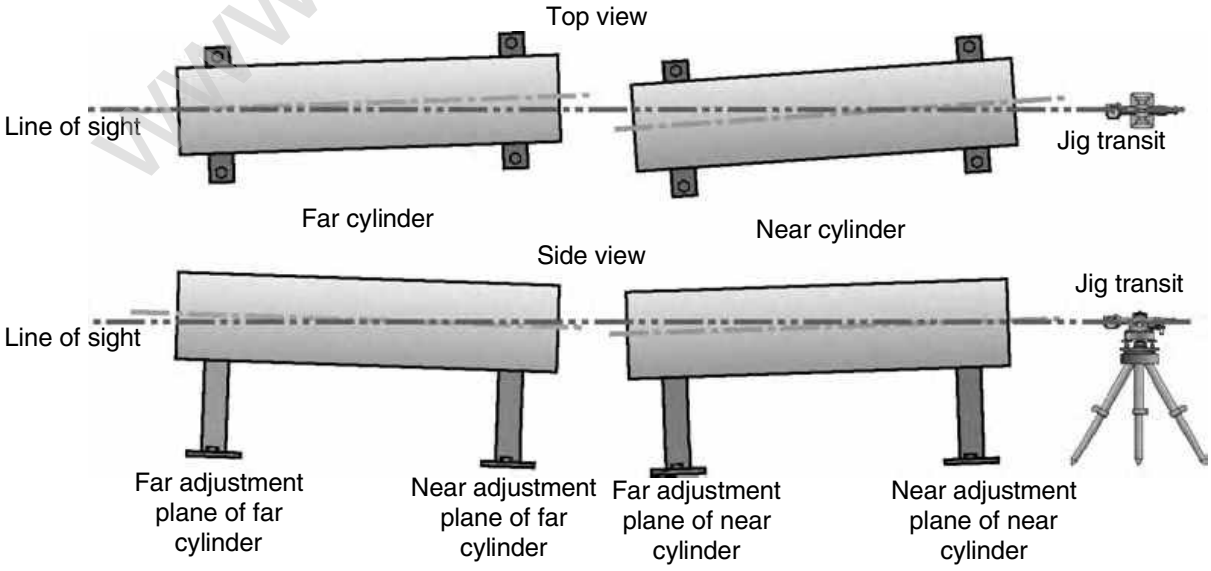


FIGURE 19.14 Line of sight observing two misaligned cylinders in the side and top views.

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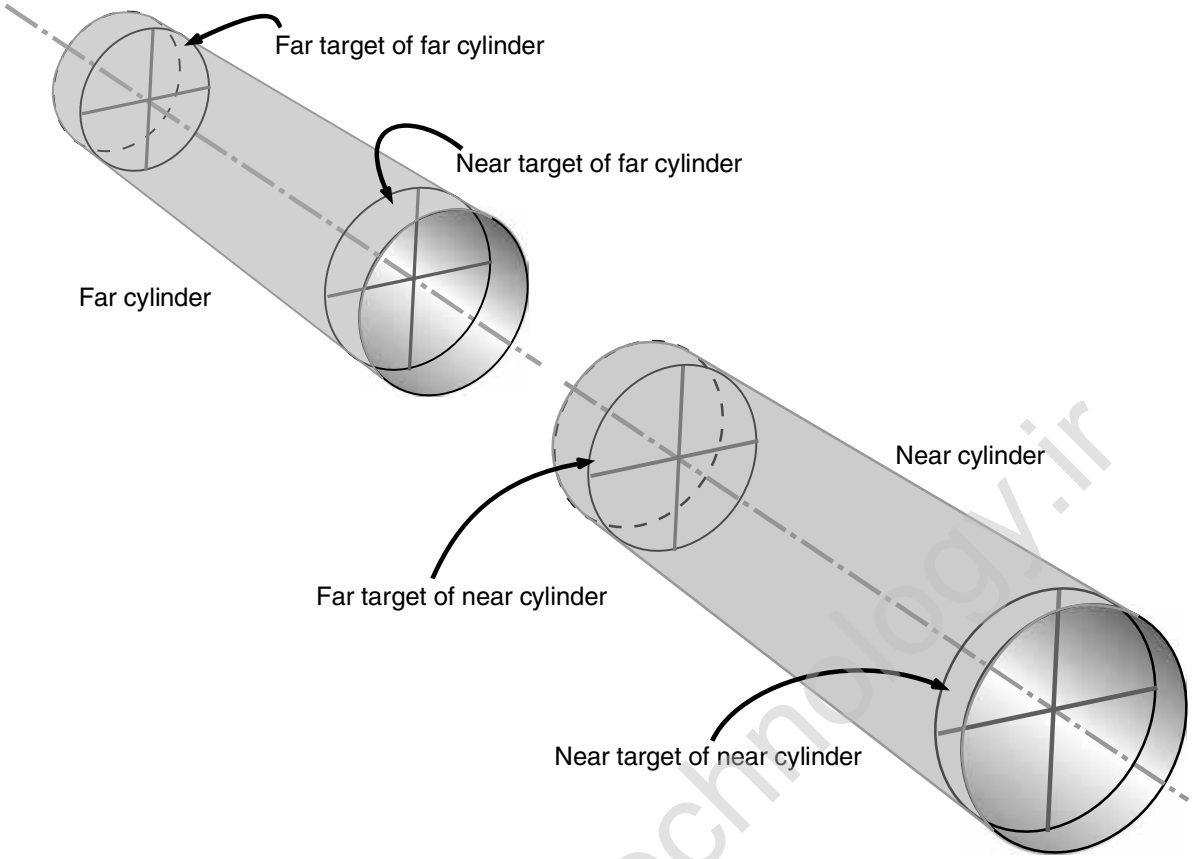


FIGURE 19.15 Near and far targets placed in near and far cylinders.

1. Install and center a see-through target at the near end of the near cylinder. To center the target in the bore of the cylinder, study and perform the tasks shown in Figure 19.17 through Figure 19.20.
2. Install and center a target at the far end of the near cylinder. Again, perform the tasks shown in Figure 19.17 through Figure 19.20 to center the target in the bore of the cylinder.
3. Buck in the jig transit line of sight to the targets installed in the near cylinder at the near and far ends. Refer to the "bucking in" procedure and perform the tasks shown in Figure 19.21.

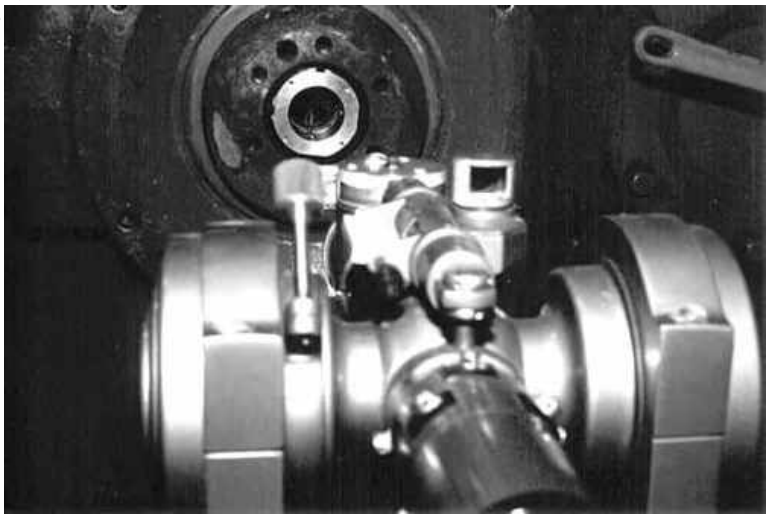


FIGURE 19.16 Near target of near cylinder as observed from the jig transit.

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Step 1. Prior to installing the target and fixture, roughly adjust the positioning screws to loosely fit (20–50 mils of clearance) the target and fixture in the bore. Slide the target into the cylinder and adjust the centering screws to snugly fit the target to prevent movement from occurring during rotation.

Step 2. At the jig transit, focus on the target position and adjust the vertical and lateral tangent screws to aim the transit crosshairs at the target center as shown in the illustration below.

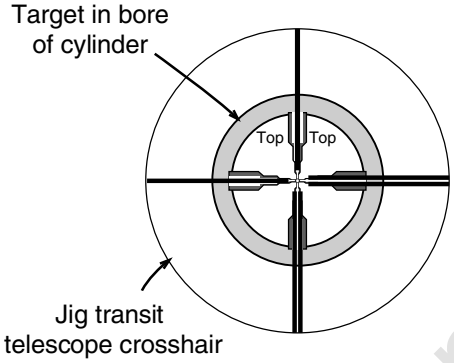


FIGURE 19.17 Step 1 and step 2 for centering a target in a cylinder.

4. Install and center a target at either the far or near end of the far cylinder. Again, perform the tasks shown in Figure 19.17 through Figure 19.20 to center the target in the bore of the cylinder.
5. Install and center a target at the other end of the far cylinder. Again, perform the tasks shown in Figure 19.17 through Figure 19.20 to center the target in the bore of the cylinder.
6. Buck in the jig transit line of sight to the targets installed in the near cylinder at the near and far ends. Refer to the bucking in procedure and perform the tasks in Figure 19.21.

Step 3. Rotate the target 180° and observe the target position through the jig transit telescope. Using the optical micrometer, measure the amount of vertical and lateral offset that exists between the target center and the center of the telescope crosshairs as shown in the figure below.

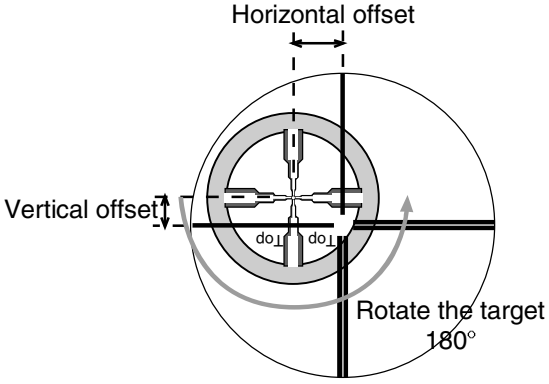


FIGURE 19.18 Step 3 for centering a target in a cylinder.

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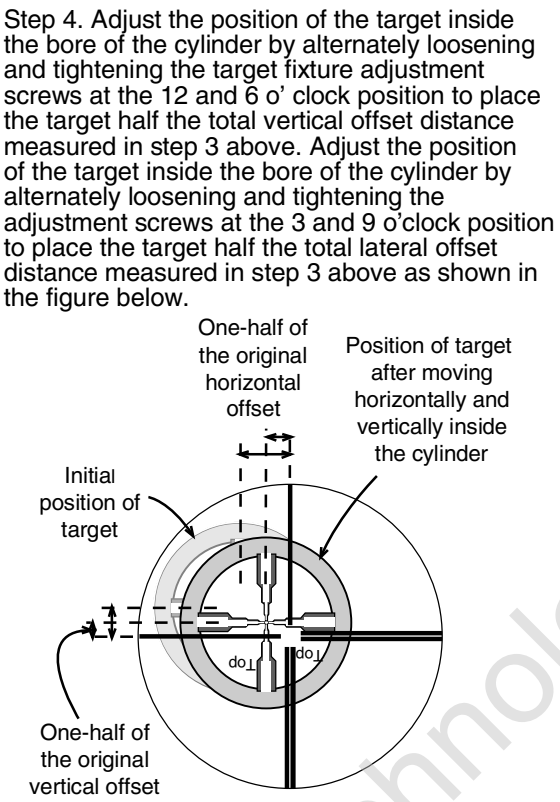


FIGURE 19.19 Step 4 for centering a target in a cylinder.

7. Using the optical micrometer, measure the positions of the targets at the near and far ends of the far cylinder in the vertical and sideways directions. Study and perform the information in Figure 19.22. Record the information.

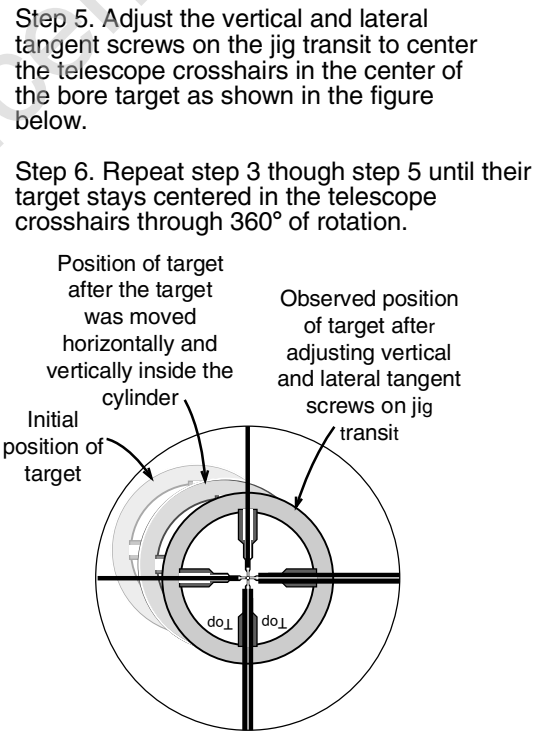


FIGURE 19.20 Step 5 and step 6 for centering a target in a cylinder.

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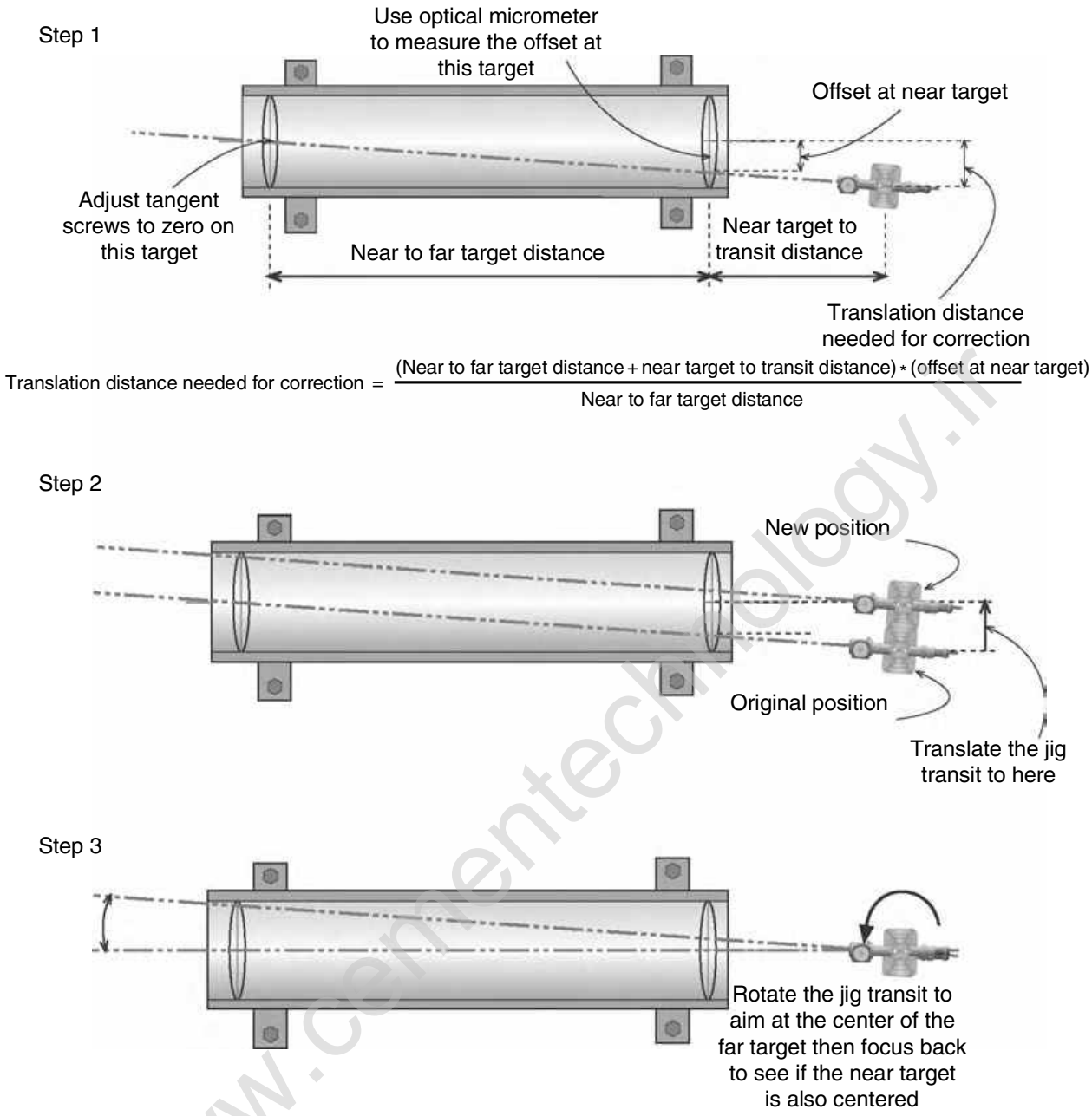


FIGURE 19.21 Bucking in your line of sight to the centerline of the bore of the cylinder.

8. If the targets at the near and far ends of the far cylinder are not coincident (in line) with the targets at the near and far ends of the near cylinder, position either the near or far cylinder to bring the bore centers into alignment. Refer to the "correcting the misalignment" procedure below and study Figure 19.23 and Figure 19.24.

19.5 BUCKING IN PROCESS

1. Center the bore targets at both ends of the cylinder as shown in Figure 19.21. Measure the distance between the bore targets and the distance from the near target to the center of the jig transit (i.e., where it rotates through its azimuth or Z-axis). Focus on the far target and using the tangent screws, center the telescope crosshairs on the target. Focus on the near target and observe its position with respect to the telescope crosshairs. If you are lucky the near target is centered in the telescope crosshairs.

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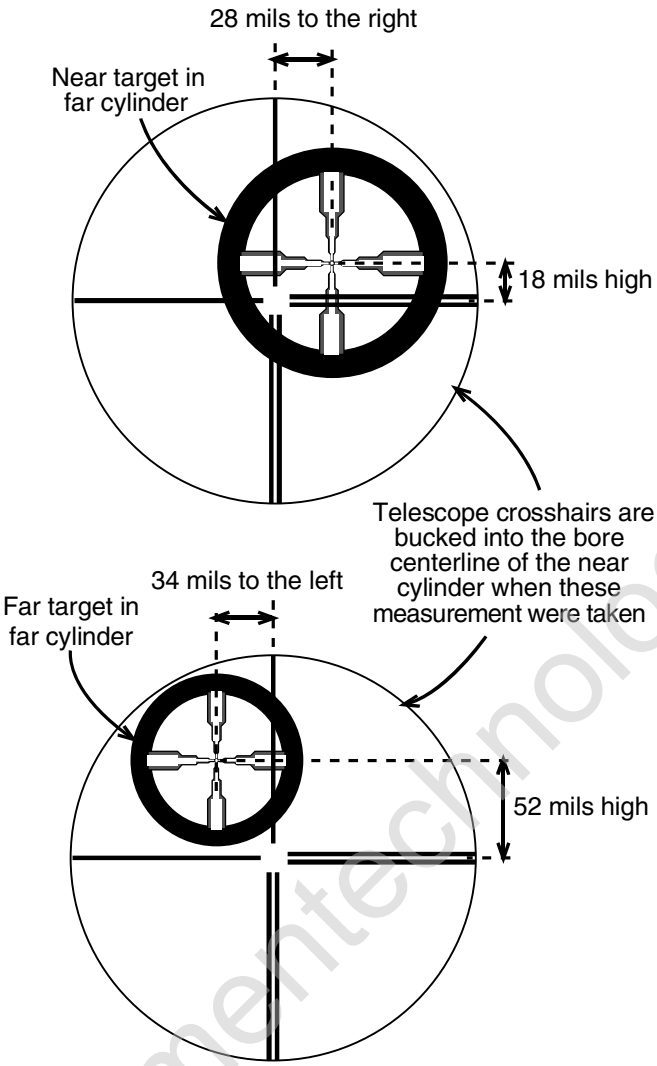


FIGURE 19.22 Measuring the amount of misalignment of the near and far targets in the far cylinder.

OK, so much for dumb luck. The near and far targets are not directly in line with each other. What you have to do now is translate the entire jig transit in the sideways direction and rotate the jig transit through its azimuth (Z) axis to align the vertical crosshairs of the telescope with the vertical paired lines on the near target. Similarly, you have to raise or lower the jig transit in the vertical direction using the precision lift mechanism and plunge (i.e., pitch) the jig transit through its pivoting (X) axis to align the vertical crosshairs of the telescope with the horizontal paired lines on the near target. Now there are two ways to do this: trial and error and mathematics. Both work, mathematics just happens to be slightly faster but requires a little bit of number crunching.

Plug the scale target reading at the near target and the distances into the formula to obtain the necessary translation distances.

2. Translate (i.e., move) the entire jig transit in the sideways direction to the amount you calculated in the equation. You can either use the rotary indicator wheel on the translation table where the scope is mounted or you can focus the scope on the near target, set the optical micrometer to the desired lateral translation distance (assuming it is under 100 mils), and begin translating the scope until the crosshairs line up on the near target's horizontal and vertical paired lines.

   Raise (or lower) the entire jig transit in the vertical direction the amount you calculated in the equation by adjusting the vertical lift mechanism. You can focus the

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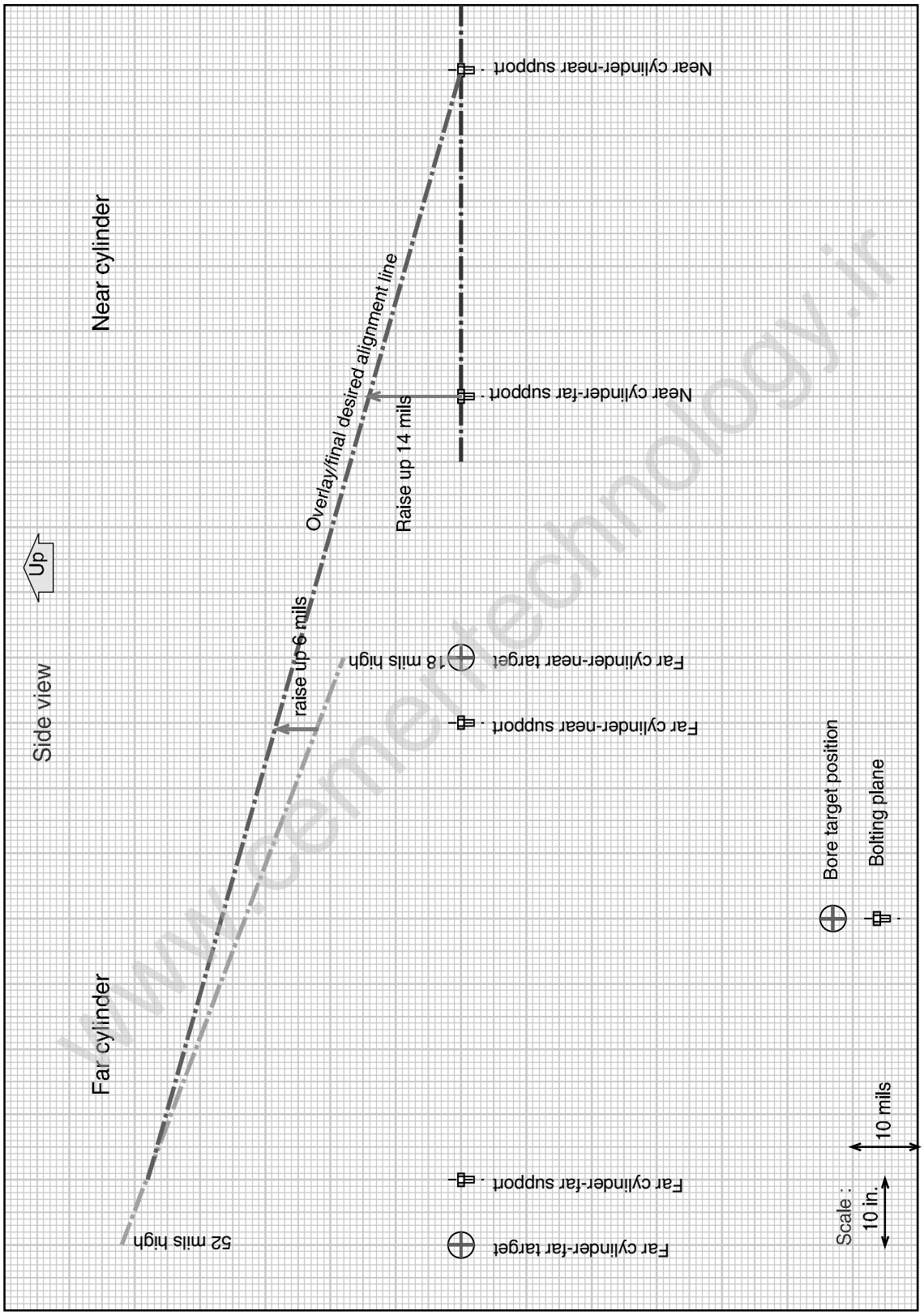


FIGURE 19.23 Side view alignment model with one possible alignment solution shown.

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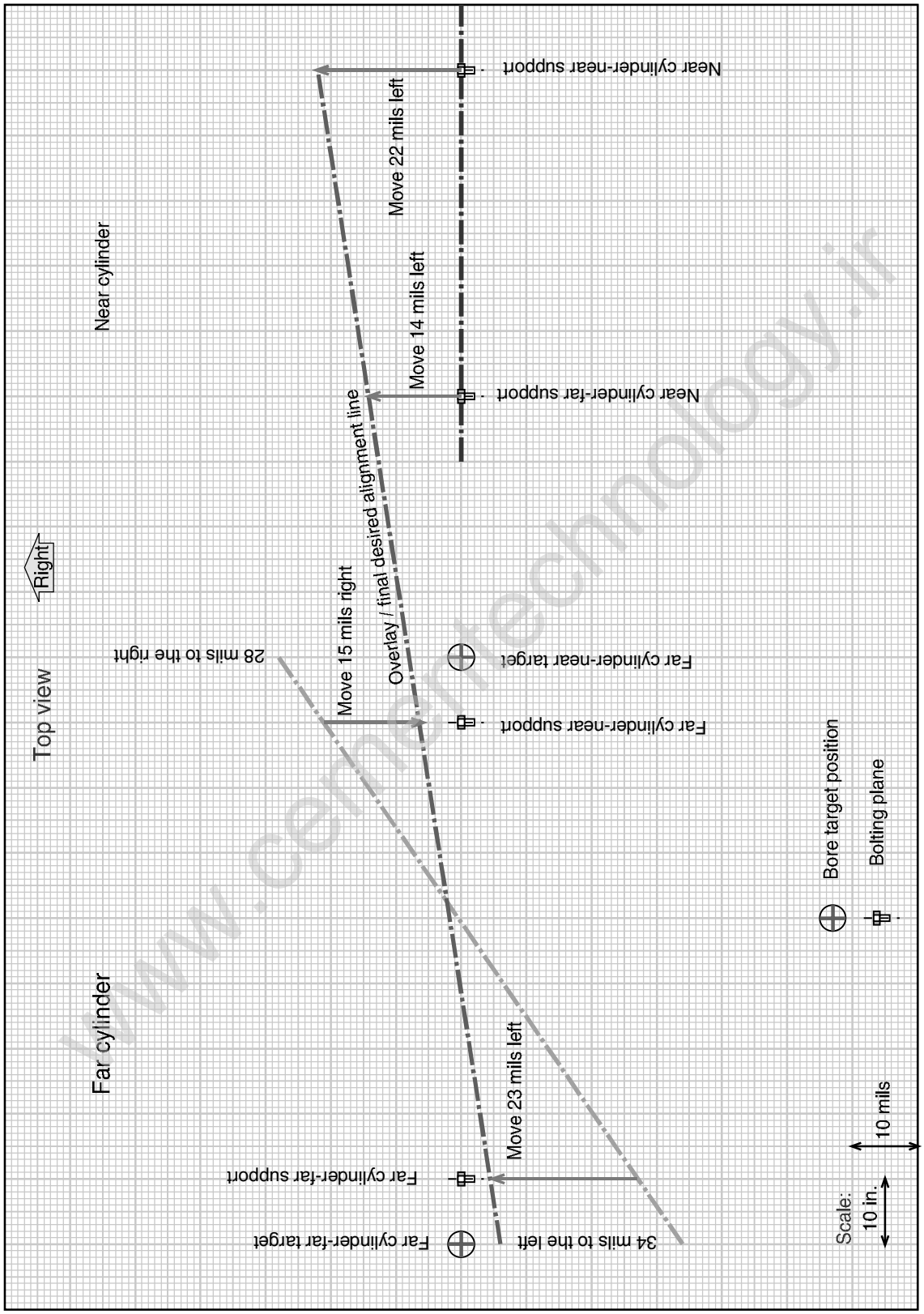


FIGURE 19.24 Top view alignment model with one possible alignment solution shown.

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scope on the near target, set the optical micrometer to the desired vertical translation distance (assuming it is under 100 mils), and begin translating the scope until the crosshairs line up on the near target's horizontal and vertical paired lines.

3. Once you have translated the scope you must now rotate the scope through its azimuth axis so your vertical crosshair lines back up with the vertical paired lines on the near target. Similarly, you now must plunge or pitch the scope through its *X*-axis so your horizontal crosshair lines up with the horizontal paired lines on the near target.

If everything works well, the telescope's line of sight should be centered at the near and far targets. If not, repeat step 1 through step 3 until the telescope crosshairs and the centers of both targets are in line with each other. At this point, the line of sight of the pivoting scope on the jig transit is parallel to the bore centerline of the cylinder.

19.6 CORRECTING THE MISALIGNMENT

Once the bore targets were centered in the far cylinder and the telescope's line of sight was bucked back into the centerline of the bore of the near cylinder, measurements were taken at the near and far targets of the far cylinder. To help visualize the misalignment between the two cylinders and assist in correcting the misalignment condition, construct side view and top view alignment models.

As shown in Figure 19.22, assume that the following measurements were taken at the far cylinder targets:

Near target: 18 mils high and 28 mils to the right
Far target: 52 mils high and 34 mils to the left

Figure 19.23 shows an exaggerated misalignment condition between the near and far cylinders in the side view (up or down direction). Figure 19.24 shows an exaggerated misalignment condition between the near and far cylinders in the top view (left or right direction). Notice that the target positions and bolting plane positions have been accurately scaled on the graph paper from left to right and the bore centerline of the far cylinder has been accurately scaled from top to bottom on the graph (see scale factors in lower left hand corner). Now a straight line can be drawn on top of the graph and the cylinders can be moved to that overlay or final desired alignment line.

19.7 LASER BORE ALIGNMENT SYSTEMS

Laser–detector systems as shown in Figure 19.25 through Figure 19.30 have been developed to accomplish the task of bore alignment. If you have not already done so, you might want to review the information on lasers and photodiode detectors in Chapter 6 to get a basic understanding of how these components work.

The principles of bore alignment with laser–detector systems are virtually identical to the process using optical alignment equipment explained in this chapter. The laser beam is substituted for the visual line of sight established with a jig transit. Rather than visually observing a sighting target placed in the center of a hollow cylinder, a photodiode is centered in the cylinder and a cable transmits the position of the laser beam on the surface of the detector.

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FIGURE 19.25 D630 Extruder system. (Courtesy of Damalini, Molndal, Sweden. With permission.)



FIGURE 19.26 D630 Linebore system. (Courtesy of Damalini, Molndal, Sweden. With permission.)

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FIGURE 19.27 Fixturlaser Extruder system. (Courtesy of Fixturlaser, Molndal, Sweden. With permission.)

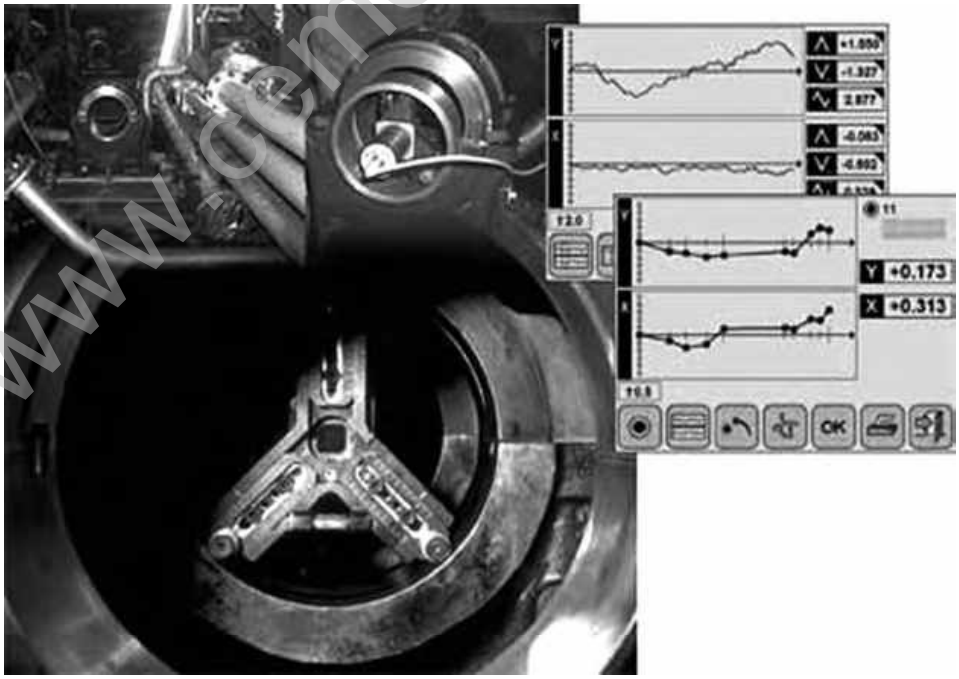


FIGURE 19.28 Fixturlaser Centering system. (Courtesy of Fixturlaser, Molndal, Sweden. With permission.)

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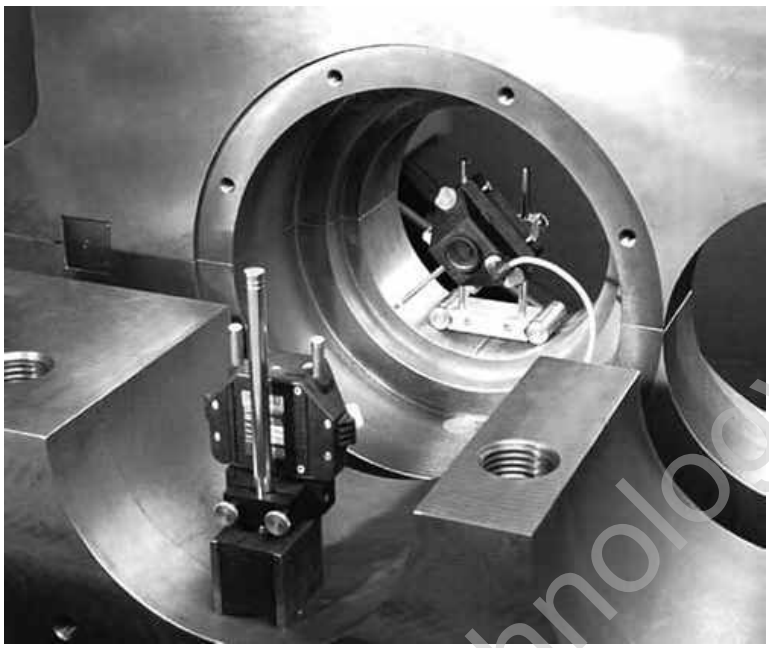


FIGURE 19.29 Prüftechnik Boralign system. (Courtesy of Pruftechnik, Ismaning, Germany. With permission.)



FIGURE 19.30 Prüftechnik Centralign system. (Courtesy of Pruftechnik, Ismaning, Germany. With permission.)

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20 Parallel Alignment

Chapter 18 covered alignment of V-belt-driven equipment. As you observed, the goal of aligning belts and sheaves is to get the driver shaft parallel to the driven shaft and the belts to track straight in the sheaves. To accomplish this task, the outer surfaces of the sheaves or the grooves of the sheave itself were used as the reference positions. The assumption is that the outer surfaces of the sheaves or the grooves of the sheave itself are perfectly perpendicular to its centerline of rotation. To verify the perpendicularity of the sheave to its shaft, face runout measurements are taken as shown in Figure 18.13 and Figure 18.15. Once we are sure that the sheaves are indeed perpendicular to their centerlines of rotation, we can then use straight edges, strings, wires, or laser beams to align the sheaves, bringing the two shafts into a parallel position. It seems cumbersome to use the sheaves as the reference positions but the shafts are typically buried inside the machine casings, making it virtually impossible to use the shafts themselves to measure from. If only the whole length of both shafts were exposed! Well, in some cases, they are.

There are drive systems in industry that encompass a series of cylinders, shafts, or rolls where they must be positioned so they are parallel to each other. Examples of this are frequently found in the paper, plastic, printing, and steel industry.

20.1 ROUGH ALIGNMENT OF PARALLEL ROLLS

As a quick review, Figure 20.1 shows the coordinate system and terminology used in aligning cylinders or rolls. The goal in aligning rolls is to get all rolls so that their individual y–z planes are parallel or coplanar and their x–z planes are parallel or coplanar.

Perhaps the simplest method of measuring roll parallelism is to use a standard tape measure. Effectively you wrap the tape measure around each roll, once at one end, record the distance around and between the two rolls there, then again at the other end as shown in Figure 20.2 and Figure 20.3. Compare the distance at the near and far ends. If the measurements are the same, assuming both rolls are level with respect to gravity, the centerlines of rotation of the rolls are parallel to each other.

If however, the rolls are not level with respect to gravity, their y–z planes may be parallel, but the centerlines of rotation may not be parallel. Figure 20.4 shows how the two tape measurement distances at the near and far end can be the same but the centerlines of rotation of the rolls might not be parallel.

Bear in mind that the x–z plane does not have to be referenced to gravity, that is, the x–z plane does not necessarily have to be level. Remember, level and aligned do not mean the same thing. It is nice and convenient that the rolls are level but they do not have to be. If their slopes are the same (i.e., the x–z plane is not level, but at a fixed angle), placing the rolls parallel to each other is still achievable.

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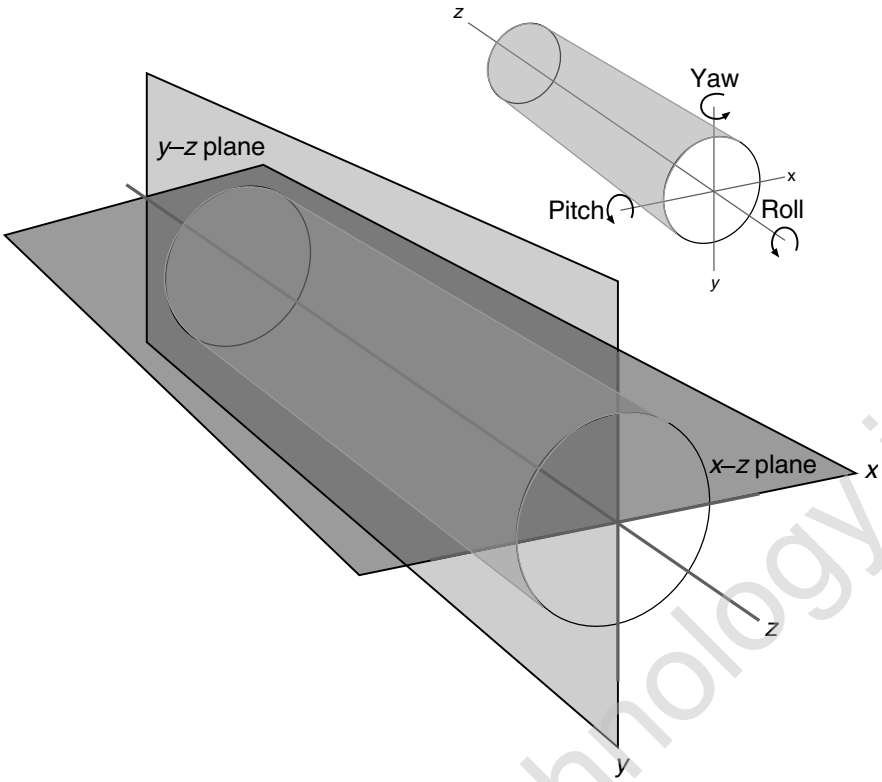


FIGURE 20.1 Coordinate system for cylinders.

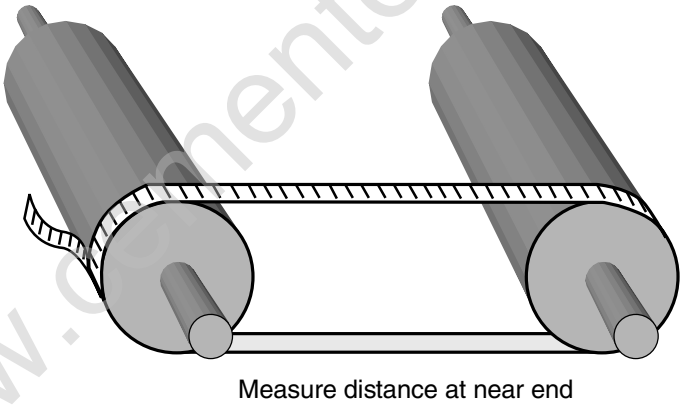


FIGURE 20.2 Using a standard tape to measure the distance between the rolls at the near end.

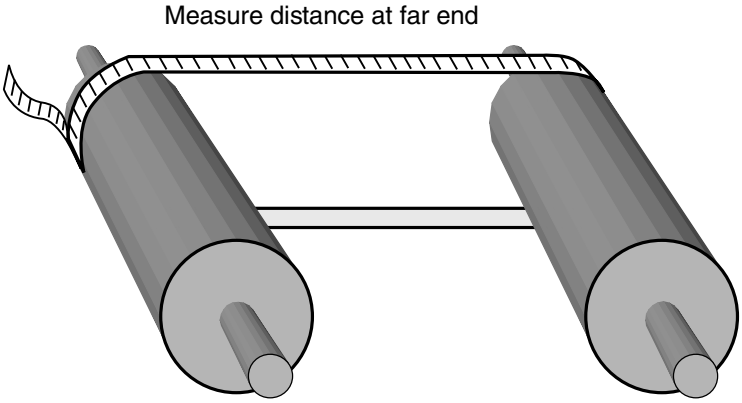


FIGURE 20.3 Using a standard tape to measure the distance between the rolls at the far end.

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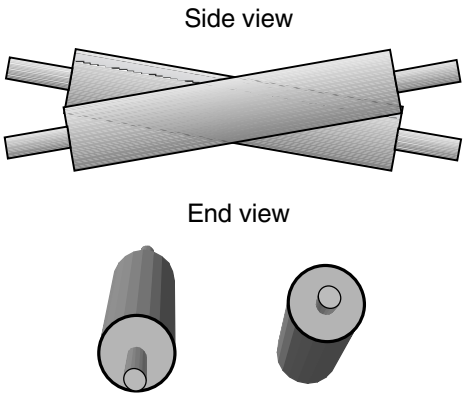


FIGURE 20.4 Centerlines of rotation are parallel in the *y–z* plane but skewed in the *x–z* plane.

20.2 USING OPTICAL ALIGNMENT EQUIPMENT FOR ROLL PARALLELISM

The extreme accuracy and versatility of optical alignment equipment makes it ideal for use in accurately positioning rolls for parallelism and perpendicularity. To align the rolls in the vertical, horizontal, and if necessary the axial direction, two instruments are needed, two telescopic transit squares or one telescopic transit square and a jig transit as shown in Figure 20.5 and Figure 20.6. The transit square has a second scope affixed to the yokes of the transit that is at a precise 90° angle to the main sighting scope.

20.3 ALIGNING THE ROLLS IN THE VERTICAL (UP/DOWN) DIRECTION

For this procedure you either need the telescopic transit square or a jig transit, a stand, and a lateral translation slide.

1. Setup the optical transit on its tripod or base so that your line of sight is slightly above the one of the shafts (or rolls). In this case, we set the equipment up to first observe the position of the upper shaft (or roll) as shown in Figure 20.7.
2. Level the scope. Take a reading at scale target A and record the measurement. Take a reading at scale target B and record the measurement. Measure the distance between the scale targets and their respective positions to the bearings that support both ends of the upper shaft (or roll). If the diameter of the roll is the same at both measurement positions, and the readings at A and B are exactly the same, then the roll is level with respect to gravity. If the diameter of the roll is the same at both measurement positions, and the readings at A and B are not the same, then the roll has a slope with respect to gravity. Determine what the slope is by obtaining the difference between the measurements at A and B and divide it by the distance between target A and target B. This will tell you the slope in mils/inch (or mils/foot if you prefer).
3. Move the optical equipment so you can sight along the top of the lower shaft (or roll) as shown in Figure 20.8. Level the scope. Take a reading at scale target C and record the measurement. Take a reading at scale target D and record the measurement. Again, measure the distance between these scale targets and their respective positions to the bearings that support both ends of the lower shaft (or roll). It is convenient, but not necessary that the distance between scale targets A and B is the same as the distance between scale targets C and D. If the diameter of the roll is the same at both measurement positions, and the readings at C and D are exactly the same, then the roll is level with respect to gravity. If they are not the same, determine the slope as described in step 2.

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FIGURE 20.5 Telescopic transit square. (Courtesy of Brunson Instruments Co., Kansas City, MO. With permission.)

4. By comparing the difference between the readings you observed at scale targets A and B and the difference between the readings you observed at scale targets C and D, you will be able to determine if the rolls are parallel to each other in the up–down direction.

20.4 ALIGNING THE ROLLS IN THE LATERAL (SIDE TO SIDE) DIRECTION

For this procedure you need either a telescopic transit square and a jig transit or two telescopic transit squares, a stand with lateral translation slides for each instrument, and a light source for one transit.

1. The best way to perform this procedure is to use two telescopic transit squares or one telescopic transit square and a jig transit. Setup the telescopic transit square on its stand so that your line of sight is slightly to one side of one of the shafts (or roll) as shown in Figure 20.9. This transit will be referred to as the measurement transit.

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FIGURE 20.6 Jig transit. (Courtesy of Brunson Instruments Co., Kansas City, MO. With permission.)

2. Level the telescopic transit square. Position scale targets on the side of the shaft (or roll) at both ends as shown in Figure 20.9. Measure the distance between the scale targets and their respective positions to the bearings that support both ends of the lower roll. You should also measure the distance from the nearest scale target to the center of the universal transit square (i.e., where it rotates through its azimuth axis). Loosen the locking screw that will allow the pivoting telescope to rotate through the X axis. What you are going to do now is position the universal transit square so the pivoting telescope's line of sight is parallel to the shaft (or roll) as shown in Figure 20.10 and Figure 20.11.

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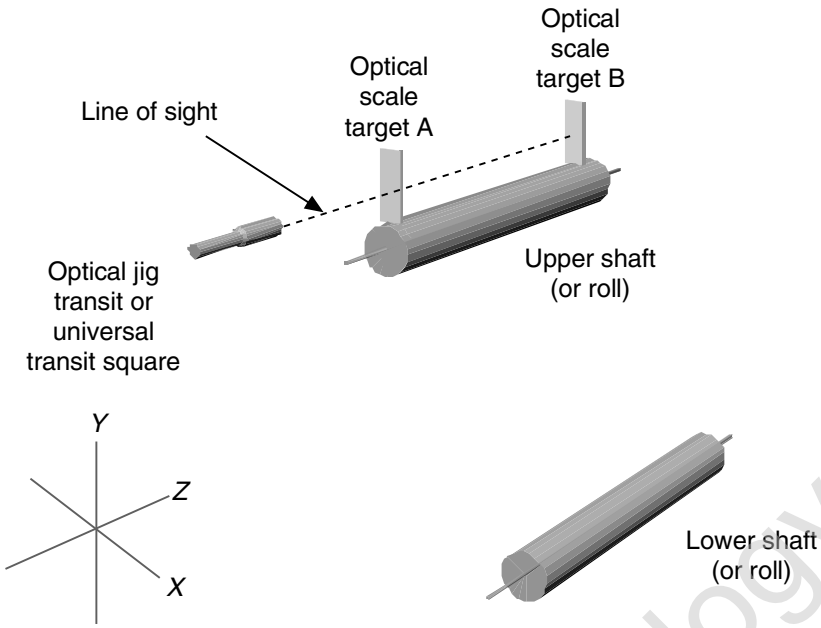


FIGURE 20.7 Observing the vertical position of the upper roll.

3. Take a reading at scale target E and record the measurement. Take a reading at scale target F and record the measurement. If you are really lucky, the readings at scale targets E and F are exactly the same (i.e., the telescope's line of sight is parallel to the roll).
4. OK, so much for dumb luck. The readings at scales E and F are not the same. What you have to do now is translate the entire universal transit square in the *X* direction and rotate the universal transit square through its azimuth (*Y*) axis so the pivoting telescopes line of sight is parallel to the lower shaft (or roll). Now there are two ways to do this, trial and error or mathematics. Both will work, but mathematics just happens to be slightly faster and requires a little bit of number crunching.

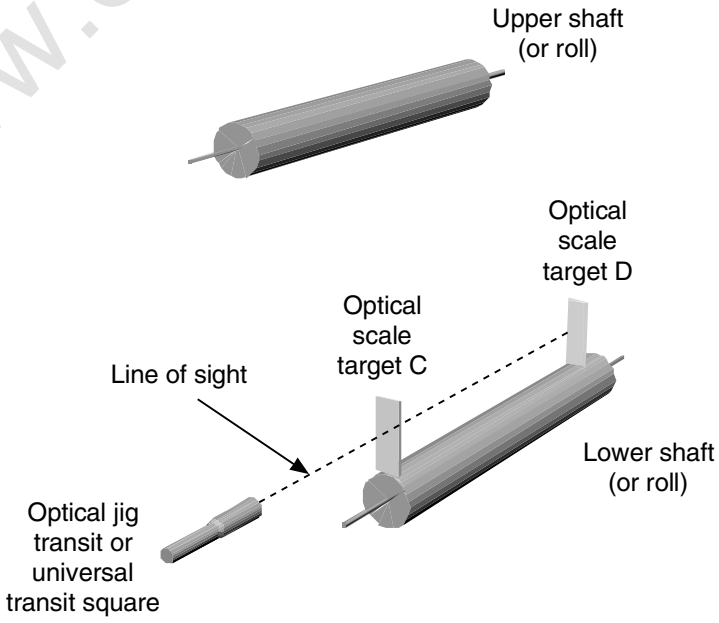


FIGURE 20.8 Observing the vertical position of the lower roll.

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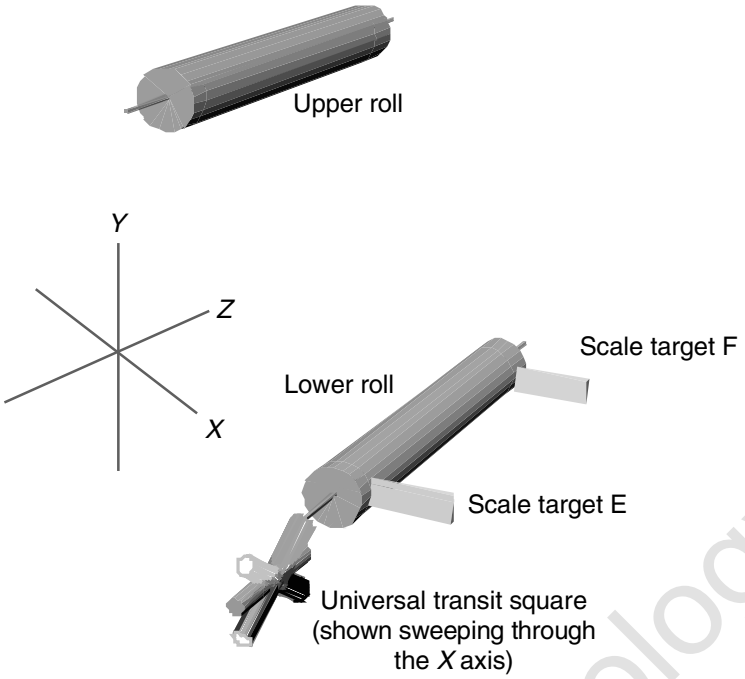
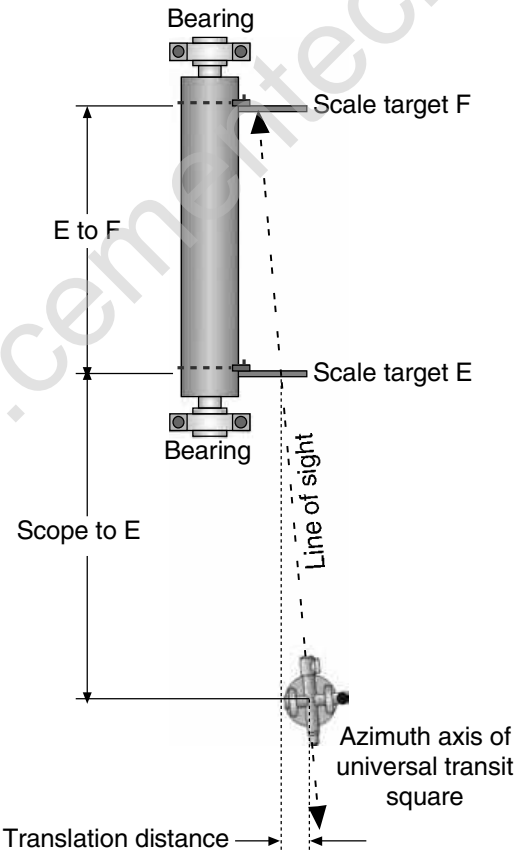


FIGURE 20.9 Observing the lateral position of one roll.



$$\text{Translation distance} = \frac{(\text{reading at scale E} - \text{reading at scale F}) \times \text{scope to E}}{\text{E to F}}$$

FIGURE 20.10 Top view of sighting the roll in the lateral direction.

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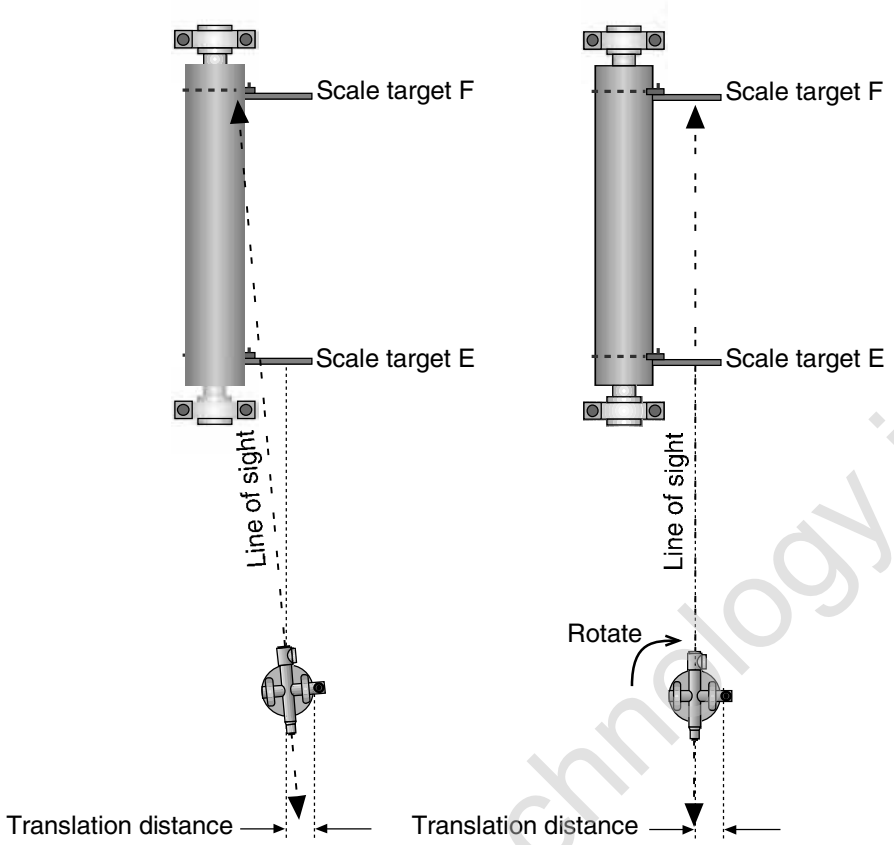


FIGURE 20.11 "Bucking in" so your line of sight is parallel to the roll.

5. Measure the distance between scale target E and scale target F as well as the distance between scale target E and the location of the telescopic transit square (specifically the azimuth axis point).
6. Take a reading at scale target E and scale target F and record the readings. Plug the scale target readings at E and F and the distances into the formula to obtain the necessary translation distance as shown in Figure 20.10.
7. Using the lateral slide, translate (i.e., move the entire universal transit square in the *X* direction) the amount you calculated in the equation. You can either use the rotary indicator wheel on the translation table the scope is mounted to or you can focus the scope on target E and begin translating the scope observing the readings on the scale target.
8. Once you have translated the scope you now must rotate the scope through its azimuth axis so your vertical crosshair lines back up with the reading you first observed at scale target E as shown in Figure 20.11.
9. If everything worked well you should now have the same reading at scale target E and scale target F. If you do not, repeat step 3 through step 8 until the readings match. At this point, the line of sight of the pivoting scope on the universal transit square is parallel to the shaft (or roll). You should probably lock the scope in this position to prevent rotation through the azimuth axis in case you inadvertently bump the unit.
10. The next step is to establish a perpendicular line of sight to this shaft (or roll). Set the other telescopic transit square (or a jig transit) on its stand at a position that is perpendicular to the telescopic transit square used to buck in the two scale targets on the shaft (or roll) as shown in Figure 20.12. This transit will be used to establish a line of sight that is at a precise 90° angle to the shaft (or roll) that you bucked into. This

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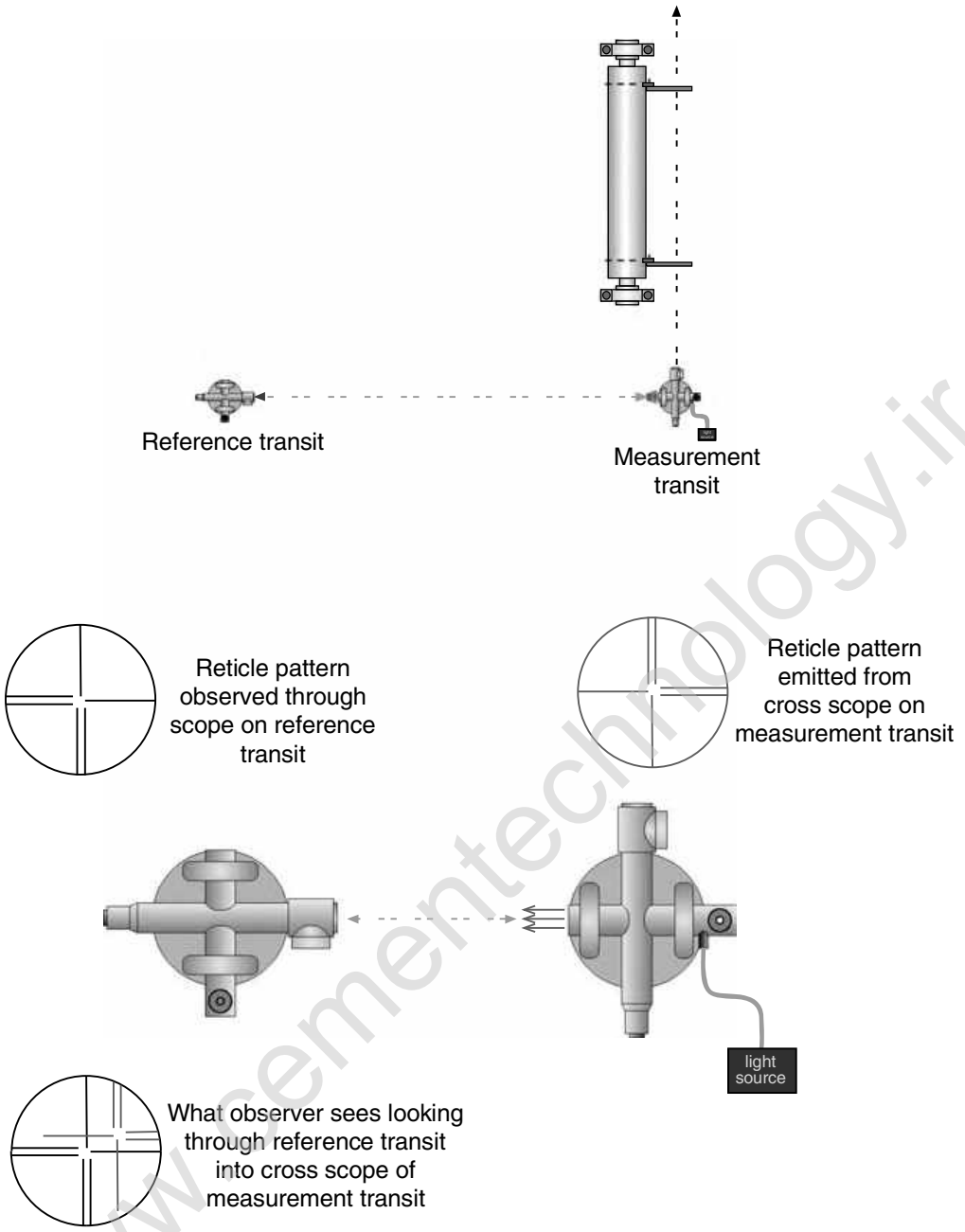


FIGURE 20.12 Setting a reference transit at a right angle to the measurement transit.

transit will be referred to as the "reference transit." To accomplish this, you need to collimate the reference transit to the scope that is locked in position by the support yolk at a precise 90° angle to the pivoting telescope as shown in Figure 20.12. Collimating the two transits sets their lines of sight parallel to each other. Insert a light source into the cross scope of the measurement transit to illuminate the reticle pattern inside the cross scope. As the cross scope is permanently focused at infinity, the light rays emitted from the scope are parallel to each other. Set the height of the reference transit at the same height as the measurement transit, and precision level the reference transit. Look into the barrel of the measurement transit's cross scope to sight the illuminated reticle pattern inside the scope barrel. Buck in the reference transit to the cross scope of the measurement transit so the illuminated reticle pattern inside the cross scope is superimposed on the reticle pattern in the reference scope as shown in

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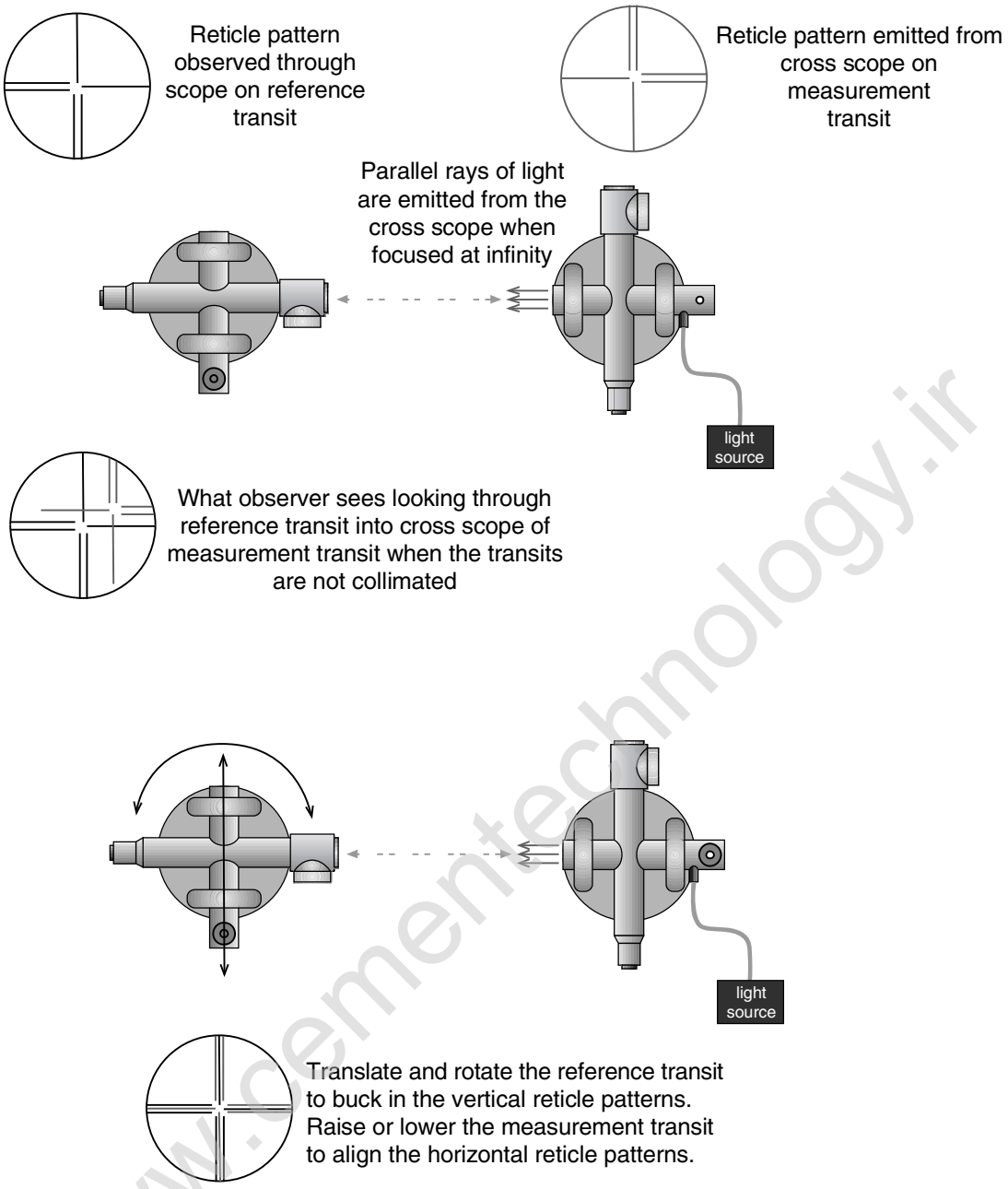


FIGURE 20.13 Collimating the transits.

Figure 20.13. Double check the readings on scale targets E and F to insure the measurement transit is still parallel to the roll. If you are sure you will finish the entire job in a few hours, skip the next step.

11. If you are going to check the shafts (or rolls) periodically to verify parallelism, it is a good idea to establish either a temporary or a permanent vertical reference plane that is perpendicular to the rolls. Once this vertical reference plane is established, you can use it to set the measurement transit more quickly for future alignment work. Adhesive backed reference targets attached to the floor can be used as a temporary reference plane as shown in Figure 20.14. Permanent floor targets (also known as monuments) can be installed in the floor as shown in Figure 20.15. Using the reference transit, plunge the scope downward to sight points along the floor. One by one, align the adhesive backed (or monument) targets to the vertical crosshairs in the reticle pattern. Positioning temporary or permanent targets within 1 mil can

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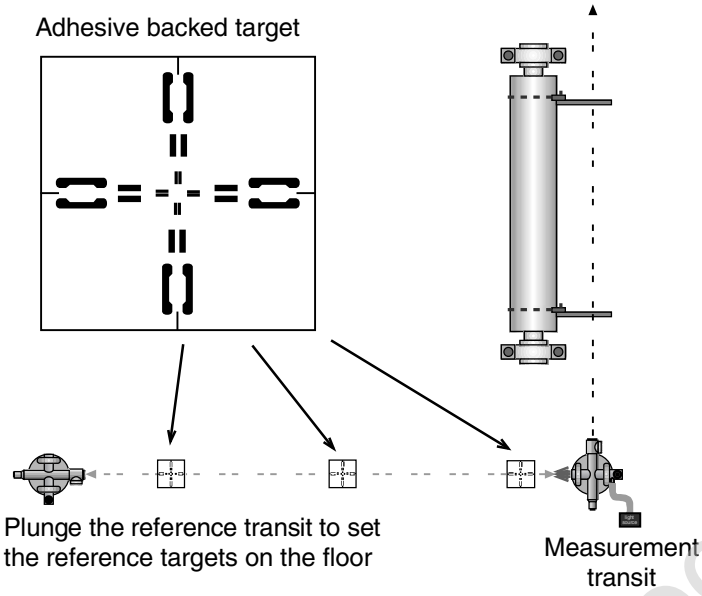


FIGURE 20.14 Placing adhesive backed target to establish a temporary reference line.

be very time consuming and tedious. Be patient, eventually you will get them where you want.

12. Move the measurement transit assembly so you can now sight along the side of the upper shaft (or roll). Try to position the transit so the azimuth axis passes through the reference line you just established as shown in Figure 20.16. Make sure that the translation slide is perpendicular to the vertical reference line on the floor. Level the scope, then align the main scope to the vertical reference line.



FIGURE 20.15 A permanent floor monument.

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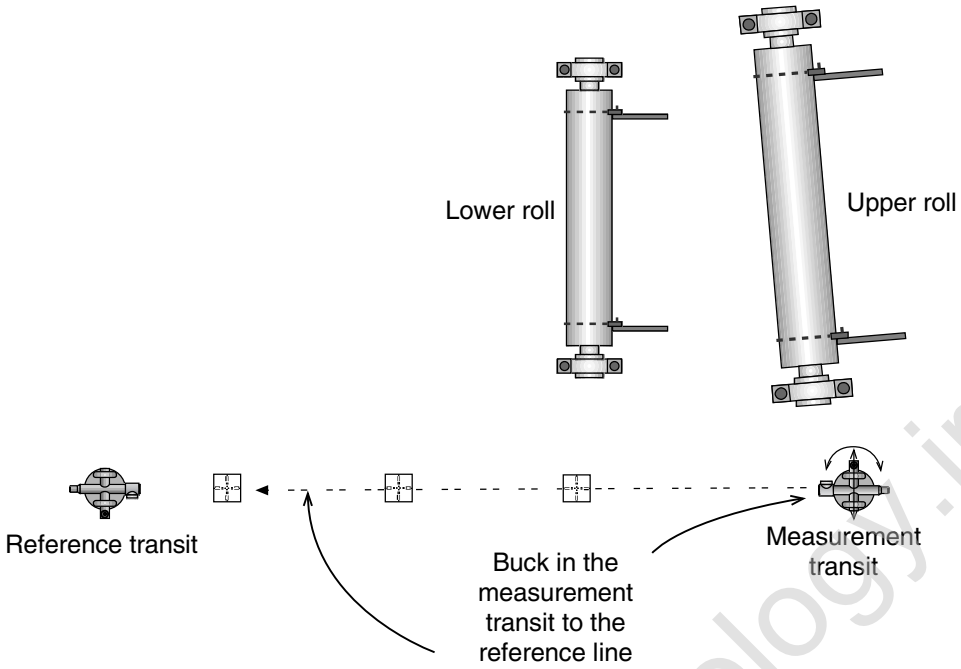


FIGURE 20.16 Move transit in position to check the upper roll.

13. Plunge the pivoting telescope on the measurement transit so the vertical crosshair is aligned to one of the reference targets you placed on the floor. Plunge the transit to observe another reference target placed on the floor. If you got really lucky, all of the targets will align with the vertical crosshair. If not, buck the measurement transit into the vertical reference plane.
14. Once you have bucked the measurement transit back in to the reference plane, rotate the measurement transit 90° through its azimuth axis so it is now pointing at the targets placed on the side of the upper roll. The cross scope of the measurement transit should now be aiming at the reference transit. Insert the light source into the cross scope (assuming you removed it) to illuminate the reticle pattern as shown in Figure 20.17.

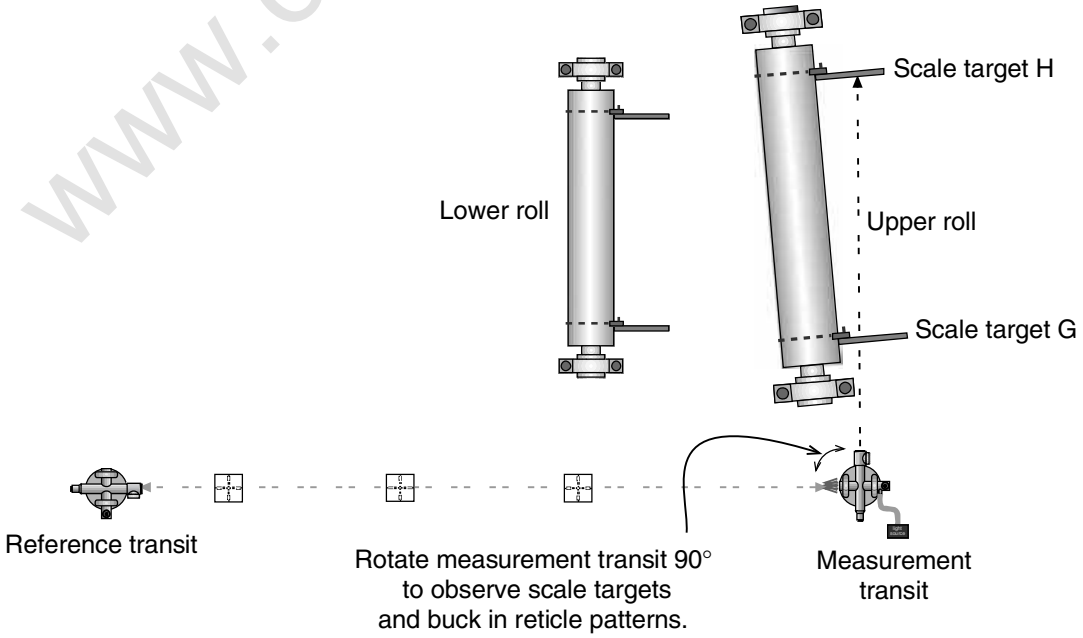


FIGURE 20.17 Measure the near and far scale targets on the upper roll.

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Go down to the reference transit and see if the reticle patterns are aligned as described in step 10. If they are not aligned, rotate the measurement transit through its azimuth axis to align the reticle patterns. If the measurement scope is at a different height, you could raise or lower it using the precision lift. However, if the vertical crosshairs are aligned on both reticle patterns that are necessary to insure that the transits are in the vertical reference plane and at a 90° angle to each other.

15. Take a reading at scale target G and record the measurement. Take a reading at scale target H and record the measurement. If the readings at G and H are the same, the two rolls are parallel to each other. If they are not, measure the distance between these scale targets and their respective positions to the bearings that support both ends of the lower shaft (or roll). Adjustments to the bearing positions on either or both of the shafts (or rolls) can be made to bring them into a parallel condition.

20.5 USING LASER–DETECTOR SYSTEMS TO MEASURE PARALLELISM

A laser–detector system can also be used to measure parallelism of shafts or rolls. There are three primary components: a laser, a detector, and a pentaprism. A pentaprism is a five-sided prism. When a collimated beam of light enters the pentaprism, it exits at a precise 90° angle regardless of the position of the pentaprism as shown in Figure 20.18.

The principles are similar to the procedure for using optical alignment instruments. The laser is positioned on the vertical reference plane and the pentaprism is positioned so the beam is diverted at a 90° angle along the side of a roll. Photodiode targets are placed on the outer surface of a roll at each end and measurements are taken at each photodiode. The pentaprism is then moved to the next roll and measurements are taken on the photodiode targets on that roll as shown in Figure 20.19. Figure 20.20 through Figure 20.31 show some of the laser alignment systems available for these measurements.

20.6 USING ROLL, PITCH, AND YAW POSITIONS OF ROLLS TO MEASURE PARALLELISM

As shown in Figure 20.1, the position of a shaft or roll can be described by its pitch, roll, and yaw positions. For shafts or rolls to be parallel, they would share the same roll, pitch, and yaw

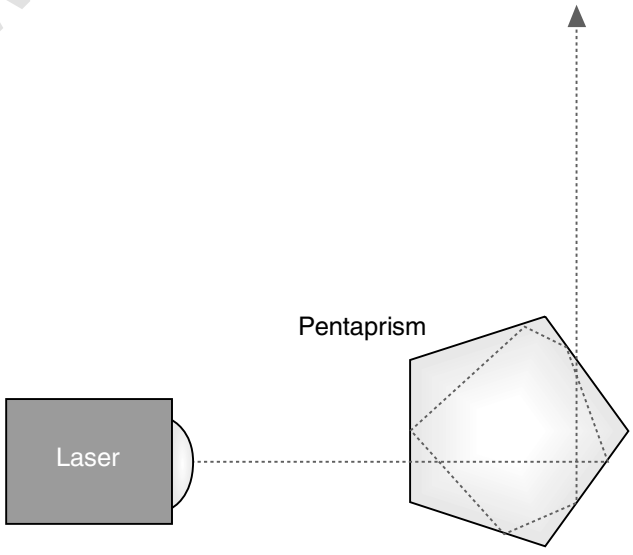


FIGURE 20.18 Laser beam exits at a precise 90° angle with a pentaprism.

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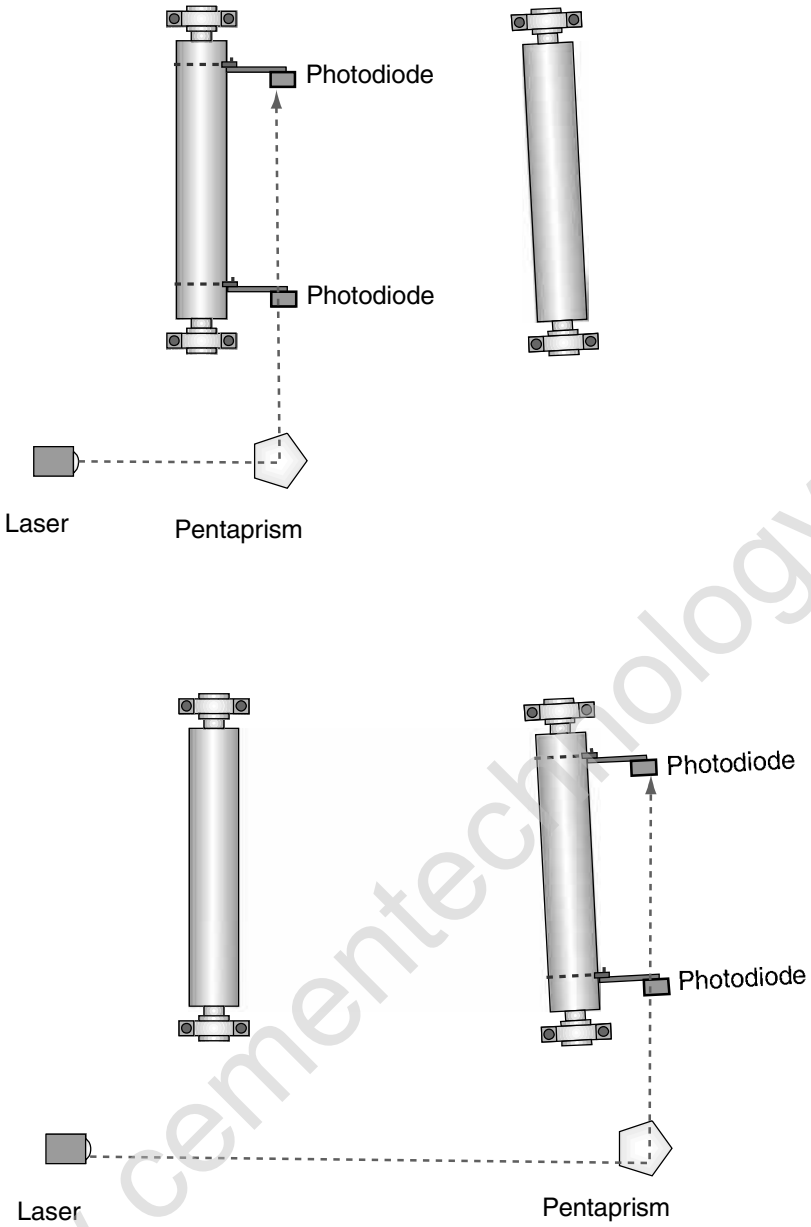


FIGURE 20.19 Using a laser and pentaprism to take measurements on parallel rolls.

attributes. Again, keep in mind that shafts or rolls can be parallel to each other but not necessarily level with respect to gravity. If a comparison of the roll, pitch, and yaw characteristics is made between two or more rolls, one could determine if they are, or are not parallel.

The Prüftechnik Paralign system incorporates three devices called ring laser gyroscopes set orthogonal to each other as shown in Figure 20.31. One ring laser gyroscope measures the roll, another the pitch, and another the yaw of the assembly. By placing the Paralign system on the outer surface of a roll and sweeping the assembly along the outer surface of the roll through an angle, the roll, pitch, and yaw information of that roll is obtained and stored in memory. The Paralign system is then positioned on the outer surface of another roll and swept again to measure its roll, pitch, and yaw information. Now, a comparison can be made between the first and second roll. Additional rolls can be measured and stored into memory.

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FIGURE 20.20 Damalini Model D670 roll parallelism measurement system. (Courtesy of Damalini, Molndal, Sweden. With permission.)

20.7 ALIGNING ROLLS AND THEIR DRIVES—SAMPLE PROBLEM

Figure 20.32 shows a side view of two motors flexibly connected to drive rolls and a third idler roll. Figure 20.33 shows the arrangement in the top view. Drive motor A is intentionally offset to its roll. As the couplings are universal joints, we want to insure that the centerline of rotation of the drive shaft is parallel to its roll to eliminate any Cardan error (refer to Chapter 4, Figure 4.13). We also want the rolls to be parallel to each other in both the up and down and east to west position. To fully understand this sample problem, it is recommended that you familiarize yourself with the information in Chapter 4, Chapter 7, Chapter 8, Chapter 13, Chapter 14, and Chapter 17. For this particular example, we will assume that there is no off-line to running (OL2R) machinery movement in any of the components.



FIGURE 20.21 FixturLaser roll system. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)

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FIGURE 20.22 FixturLaser Model FL100 laser. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)

Because there is an intentionally designed offset between motor A and its drive roll, a good alignment measurement method to use here would be the face–face method (Chapter 14). We have a choice of methods to measure motor B and its drive. We could use the reverse indicator or face–rim methods but there would be a considerable amount of bracket sag due to the distance from the motor shaft to the roll or we could use a laser alignment system (Chapter 15). Assuming you do not have a laser alignment system, a good method to use here would be the shaft to coupling spool method (Chapter 13). To capture the positions of roll A, roll B, and the idler roll, the optical or laser alignment equipment described in this chapter could be used. Figure 20.34 shows the complete set of measurements taken between the motor shafts and the rolls and the measurements on the rolls themselves. The target on the south side of each roll was used as a reference to the target on the north side.

Figure 20.35 shows the alignment model in the side view. Notice that the three rolls are not parallel to each other. Do not forget that the horizontal lines on the graph paper do not represent lines perpendicular to gravity. The centerlines of rotation of motors A and B are projected out toward their respective rolls to help visualize the nonparallelism. The triangles represent the movement restrictions in the down direction (i.e., the total thickness of shims under the feet of the motors or bearings that support the rolls).

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FIGURE 20.23 FixturLaser Datum Line Fixture. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)

Figure 20.36 shows the top view. Again notice that the three rolls are not parallel to each other. Do not forget that the horizontal lines on the graph paper do not necessarily represent the desired lateral positions of the rolls or motor shafts in the east to west direction. The triangles represent the lateral movement restrictions in the east to west direction (i.e., bolt-bound conditions). The goal is to position the rolls so they are parallel to each other, position motor A shaft so it is parallel to its roll, and either align motor B shaft so it is collinear or at least parallel to its roll.

Figure 20.37 shows one possible alignment solution in the side view and Figure 20.38 shows one possible alignment solution in the top view. As you are hopefully aware, these are not the only possible solutions.

For those of you who work in the paper, plastic, printing, or steel industries, the above sample problem is simplistic compared to the dozens of rolls you have to align on your machines. The power of the alignment modeling techniques comes from the fact that you can put as many pieces of machinery on the model as you want. In some cases, you may need some large sheets of graph paper.

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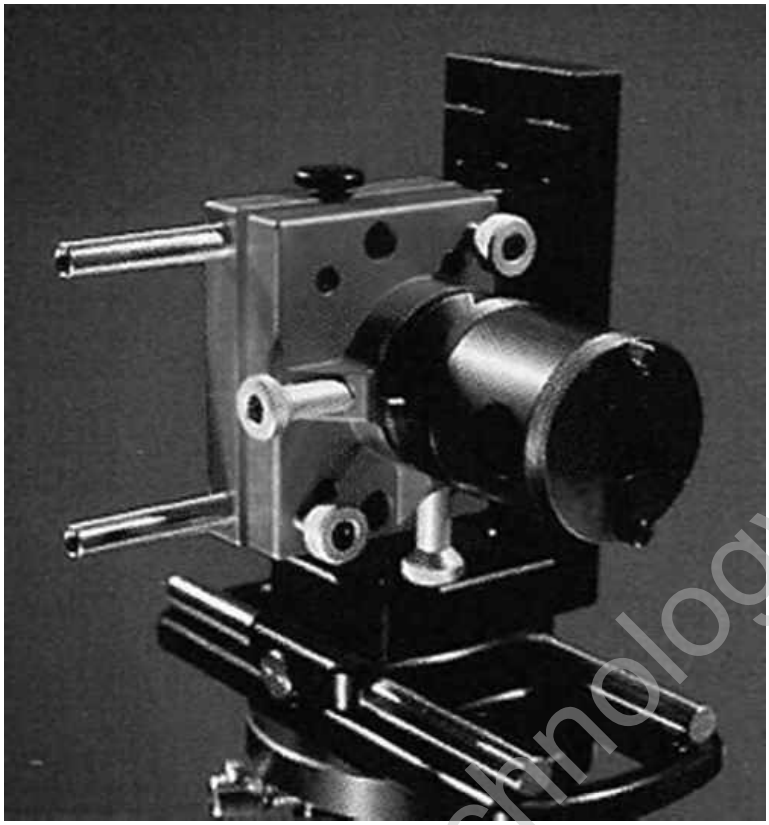


FIGURE 20.24 FixturLaser angle prism. (Courtesy of FixturLaser, Molndal, Sweden. With permission.)



FIGURE 20.25 Hamar Model L-720 laser. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)

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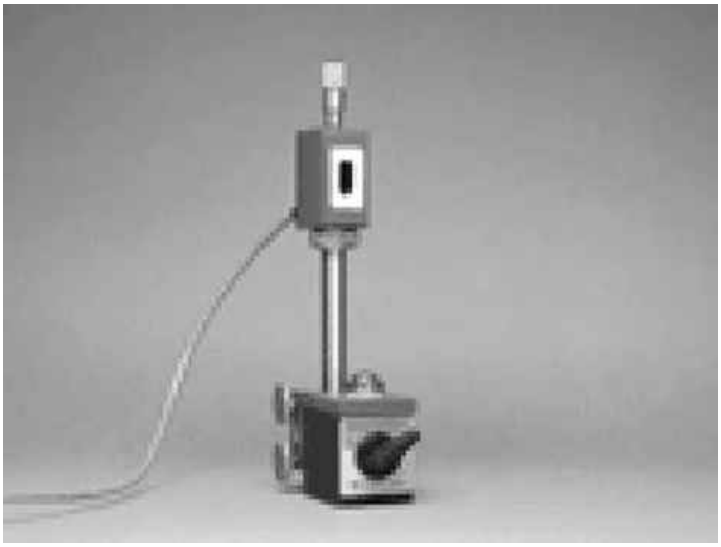


FIGURE 20.26 Hamar Model A-517 single axis target. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)



FIGURE 20.27 Hamar Model T-261A 4 axis target. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)



FIGURE 20.28 Hamar Model T-212 4 axis target. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)

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FIGURE 20.29 Hamar Model P-405 optical square. (Courtesy of Hamar Laser Instrument Co., Danbury, CT. With permission.)

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FIGURE 20.30 Prüftechnik Paralign system. (Courtesy of Pruftechnik, Ismaning, Germany. With permission.)



FIGURE 20.31 Paralign internals showing the three ring laser gyroscope in orthogonal positions. (Courtesy of Pruftechnik, Ismaning, Germany. With permission.)

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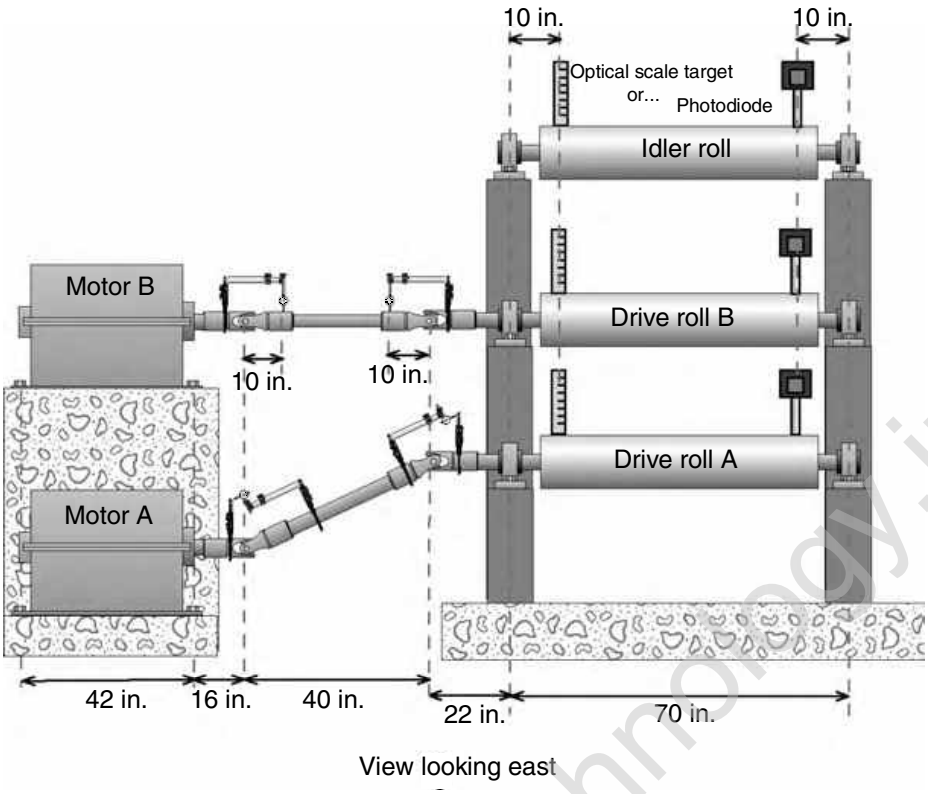


FIGURE 20.32 Side view of rolls and the drive motors.

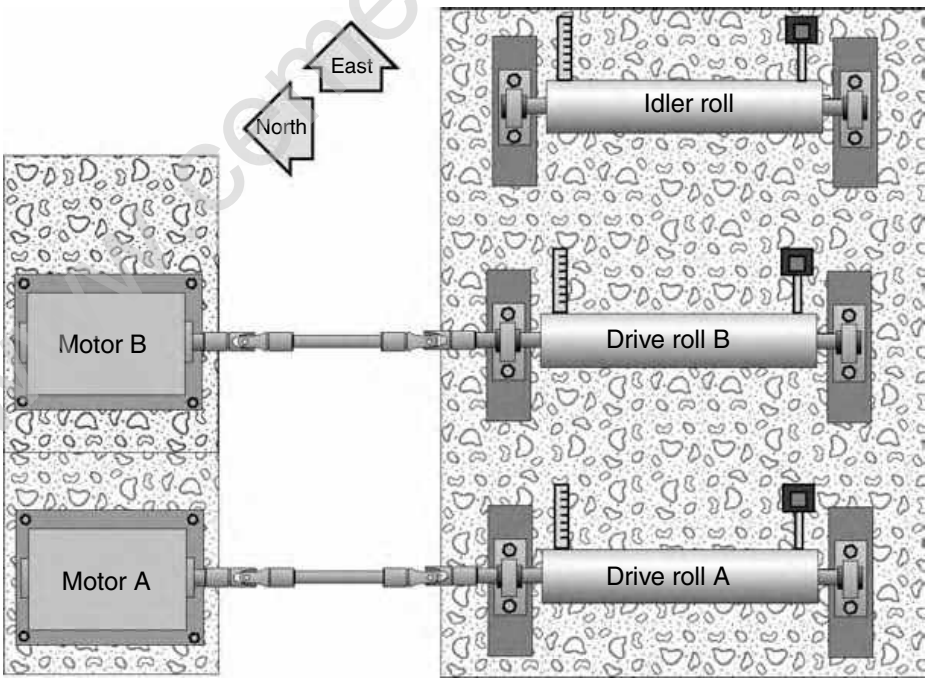


FIGURE 20.33 Top view of rolls and the drive motors.

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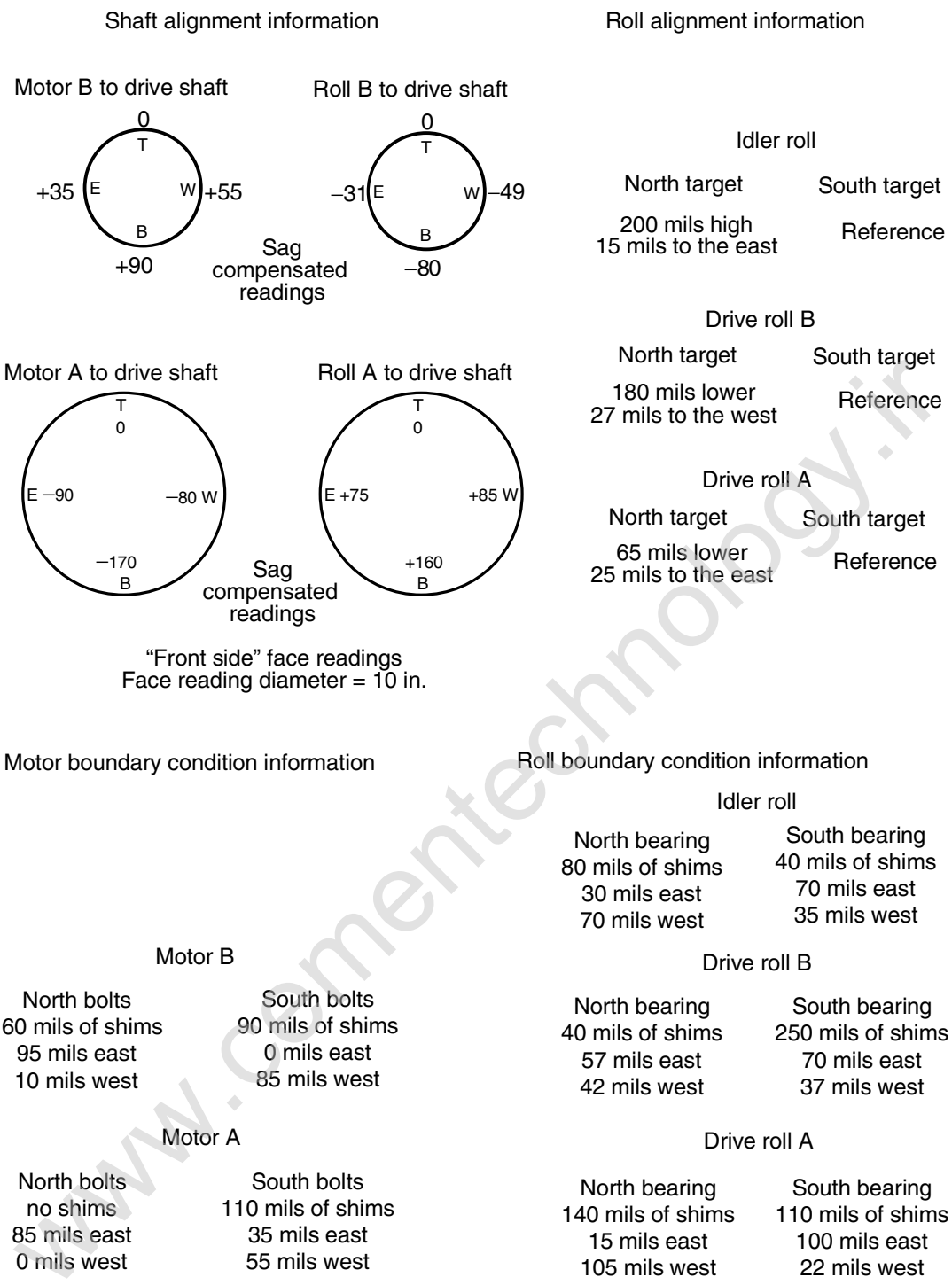


FIGURE 20.34 Shaft alignment, roll alignment, and boundary condition information.

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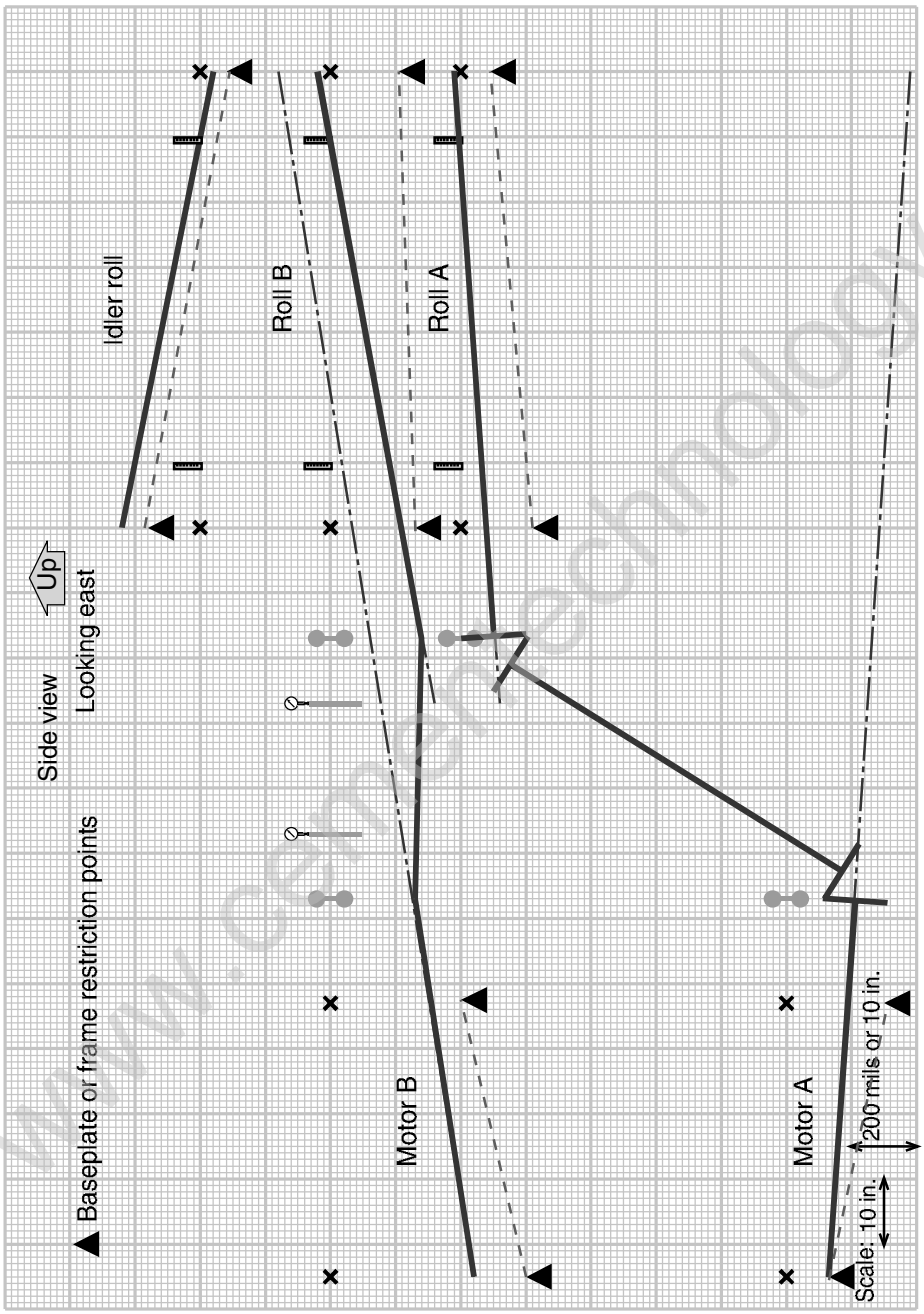


FIGURE 20.35 Side view of roll drive system.

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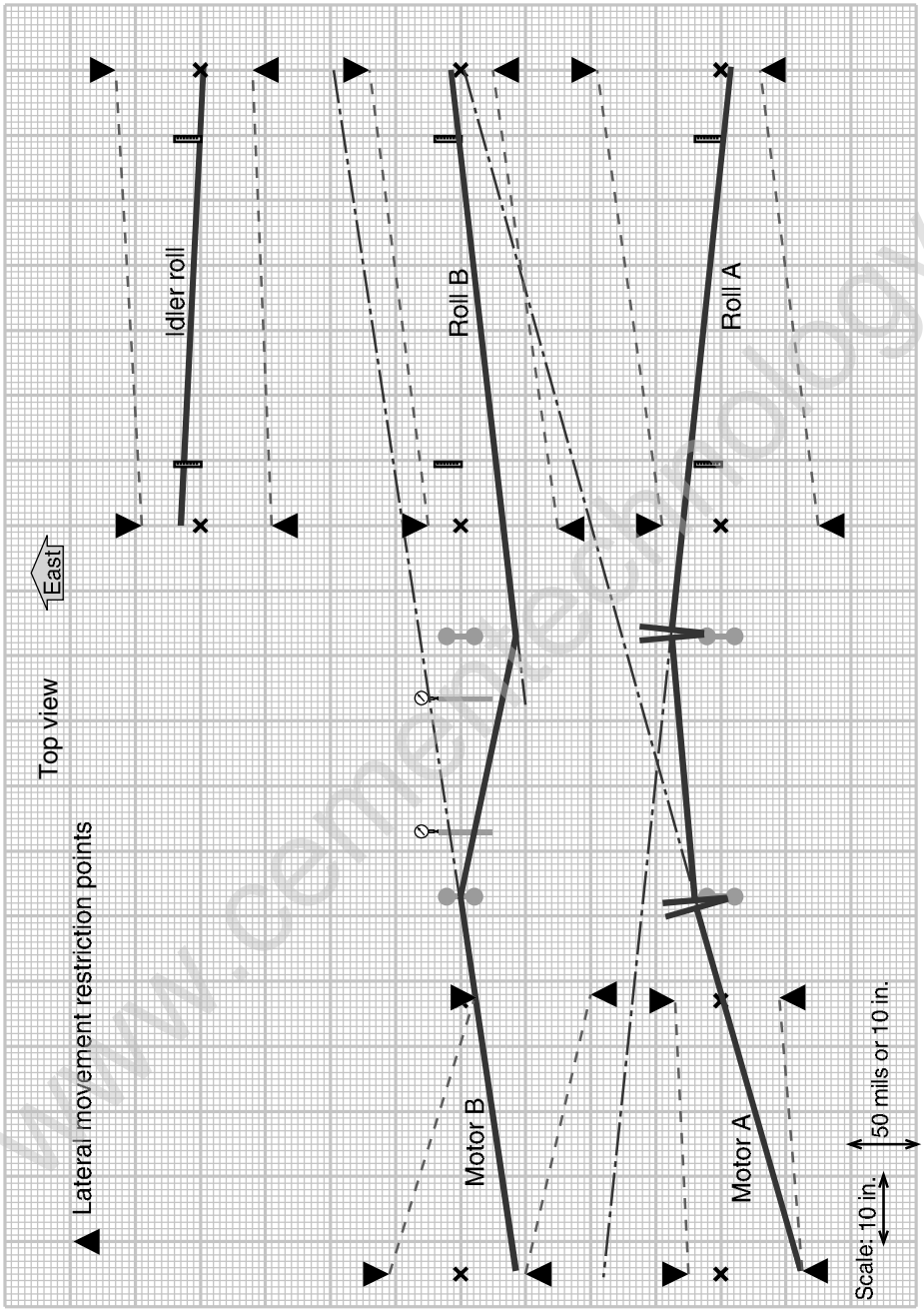


FIGURE 20.36 Top view of roll drive system.

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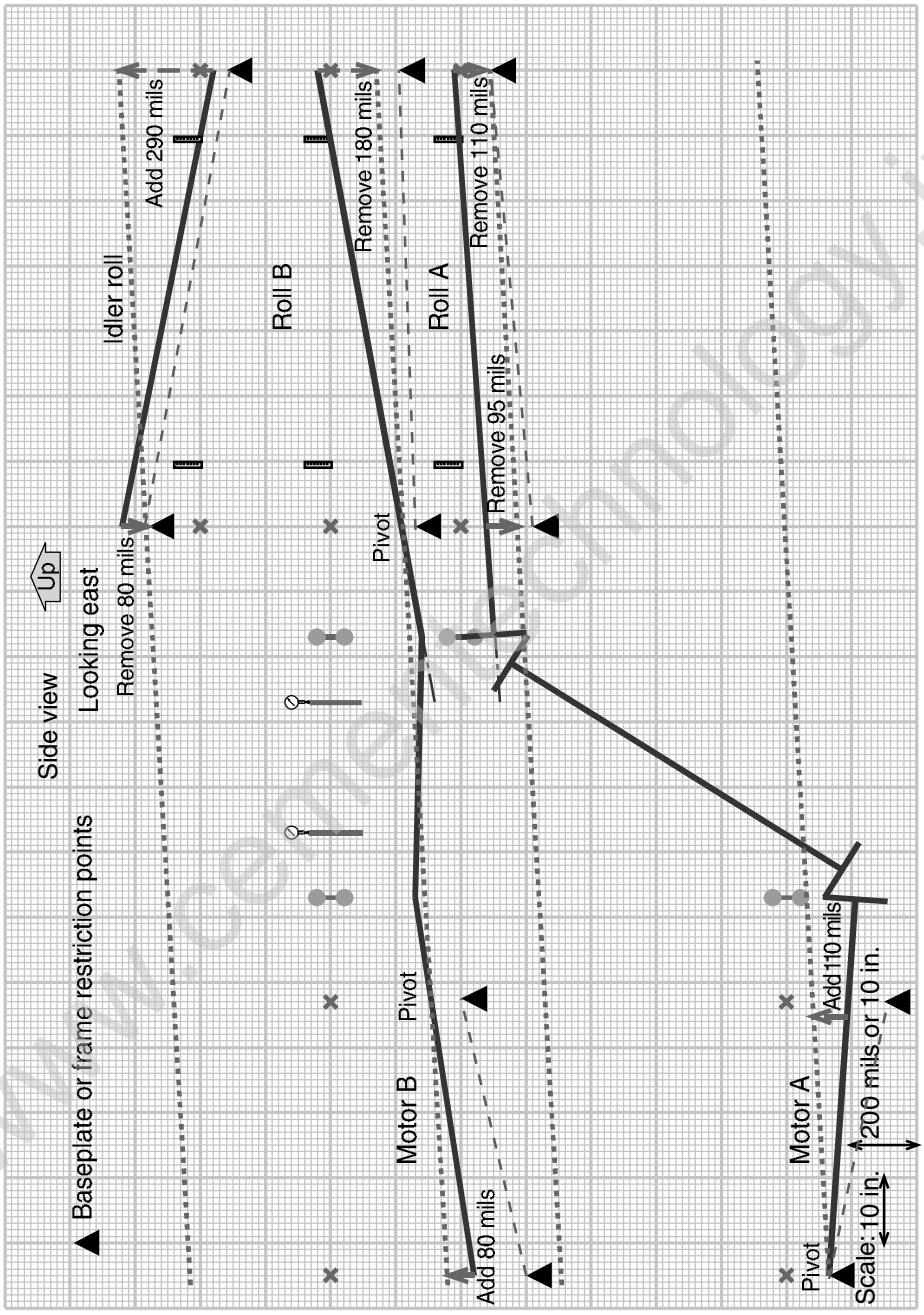


FIGURE 20.37 (**See color insert following page 322.**) Side view alignment model showing one possible solution.

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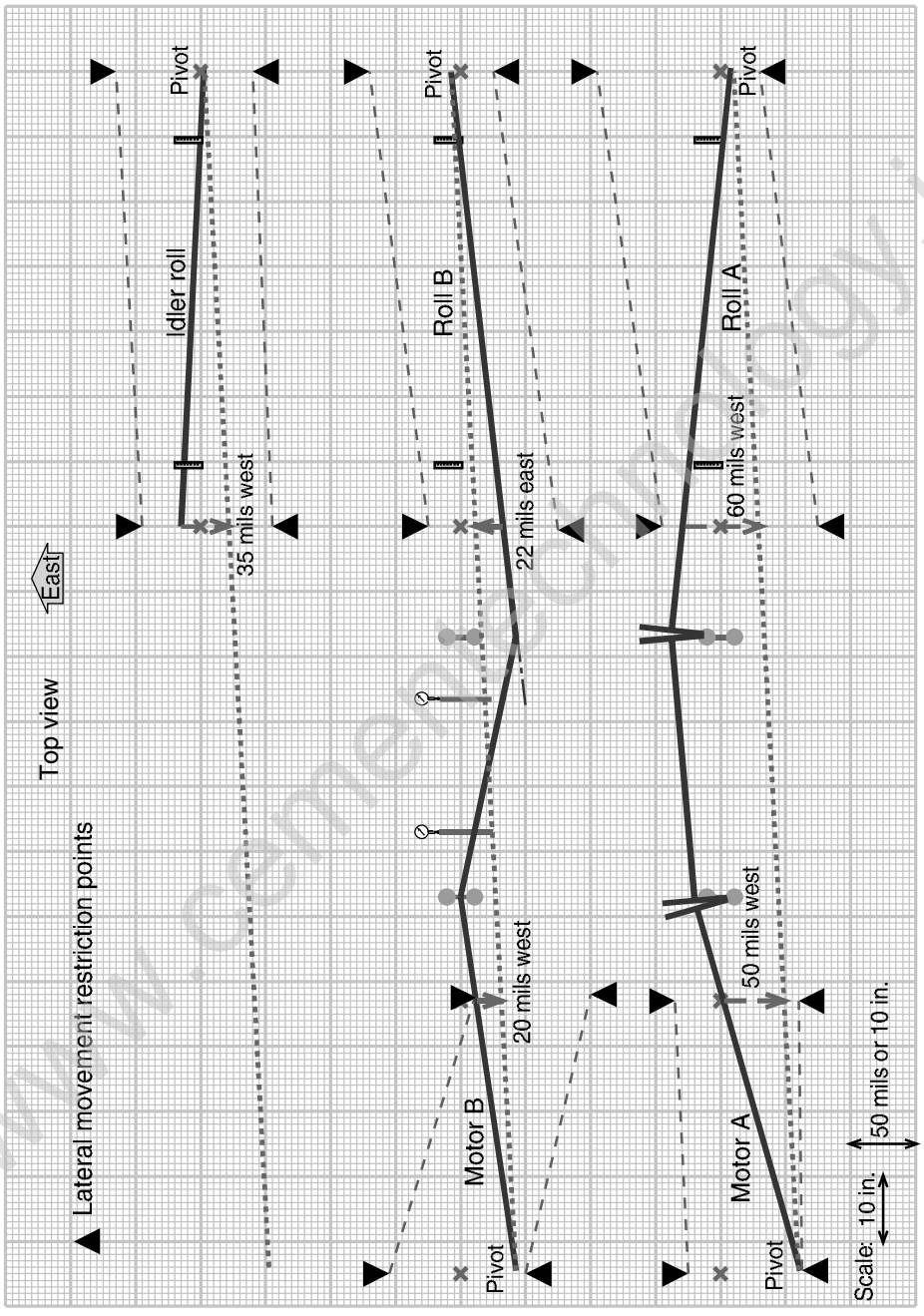


FIGURE 20.38 (**See color insert following page 322.**) Top view alignment model showing one possible solution.

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21 Alignment Considerations for Specific Types of Machinery

Up to this point, fairly broad generalizations have been made about rotating machinery. The shaft alignment measurement methods and tools discussed in Chapter 10 through Chapter 15 did not mention any specific type of machinery that the methods should be used on. Intentionally so, since most of these methods can be used on any type of machinery regardless of their specific function. The rotating equipment could have been electric motors, steam turbines, pumps, fans, compressors, or whatever. It really does not matter. The examples of graphing and modeling techniques covered in Chapter 8 through Chapter 20 had specific machine names on the diagrams, but for all practical purposes, the names of the machinery were not relevant, only the graphing concepts. The OL2R methods discussed in Chapter 16 should work on virtually any type of machinery but certain OL2R methods are better suited for certain situations than some of the others mentioned.

This is not to suggest however that a wide variety of rotating machinery behaves the same way or should be aligned the same way. It is important to know a considerable amount of information about each piece of machinery before, during, and after the alignment process. Knowledge about how it works internally, what operational function does it perform, how does the process affect its operational performance, and how it interacts with its frame and foundation, and influence from external connections such as piping are important in thorough understanding of the behavior of the machine. The example in Appendix A illustrates the type of information that should be kept on each piece of rotating machinery.

This chapter will explore some of the specific information relating to alignment on common rotating machinery equipment. Much of the information contained herein is based on actual field measurements but should not be construed as hard and fast rules. Always consult the manufacturer of the specific type of rotating machinery for information pertaining to your particular machine. But above all, do your own investigative analysis and learn the behavior of your machines.

Most of the OL2R movement ranges indicated in each machine category are based on actual field measurements. These data indicate how the centerline of rotation of the shaft might move from OL2R (or how the inboard and outboard bearing positions change from OL2R and vice versa). The OL2R movement amounts reflect average ranges of motion; in other words, the high end values could, and in many cases, have been in excess of the numbers indicated, sometimes by a factor of 300% and up. Again, most manufacturers of rotating machinery do not conduct OL2R measurements at a customer's plant site. If you consult the original equipment manufacturer for OL2R information, ask how the measurements were taken, what the environmental and operational conditions were during the test, and why they feel these data would be indicative of the machinery you have in operation.

The following information is an attempt to give you an overview of what to consider when installing and aligning these different types of machinery. It is therefore recommended that

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this information be used strictly as a starting point for you to initially align new rotating machinery systems to allow you the opportunity to safely conduct field studies on your rotating equipment.

21.1 DRIVERS

21.1.1 Electric Motors

Electric motors (AC induction or synchronous or DC motors) are perhaps the best behaved types of rotating machinery from an alignment standpoint. Electric motors up to 500 hp frequently are outfitted today with antifriction-type bearings so they do not pose any major installation problems assuming they are mechanically and electrically sound (Figure 21.1). Medium to large electric motors are frequently outfitted with sliding-type bearings (Figure 21.3). When power is applied to motors where the armatures are supported in sliding-type bearings, the electromagnetic field wants to center the rotor with respect to the stator field. This phenomenon is often referred to as "magnetic center" and needs to be taken into consideration for proper shaft-to-shaft spacing. Many electric motors have no thrust bearing *per se* and rely on electromagnetic forces to center the rotor. To find magnetic center, uncouple the motor and run it "solo." Once the field is applied, you may notice the shaft "hunting" back and forth axially for a short period of time and then it will typically settle out at one specific axial position. Very carefully scribe a line on the rotating shaft with a felt tip pen or soapstone near a stationary reference fixture such as the bearing seal. Keep your fingers away from keys or keyways and do not let any loose clothing, tools, rags, or other stationary objects attached to your body or near you hit the shaft. Drop the field (i.e., shut the motor off) and let the rotor stop completely. It is unlikely that the shaft's axial position is directly on magnetic center so after safety tagging the breaker, hand rotate the shaft pushing or pulling it axially until the scribe mark you made lines back up with the stationary reference fixture you picked. Now measure and set the shaft-to-shaft distance with the machine it is driving.



FIGURE 21.1 Typical small electric motor.

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FIGURE 21.2 Medium size electric motor.

If misalignment conditions are severe enough on electric motors and the shaft or armature elastically bends a sufficient amount, the rotor to stator air gap can get out of tolerance (the accepted tolerance for air gap eccentricity differential is ±10% of the total air gap). From a vibration standpoint, eccentric air gap problems will frequently exhibit a spectral peak at twice line frequency (120 Hz in North America refer to Section 2.2.3).

Typical OL2R movement range of electric motors (horizontally mounted):
Vertical movement: 1 to 5 mils upward (5 to 200 hp); 3 to 30+ mils upward (200+ hp), typically symmetrical (i.e., inboard and outboard ends move up the same amount)
Lateral (*sideways*) *movement*: 0 to 4 mils (usually much less than vertical movement)
Axial movement: 5 to 10 mils (5 to 200 hp); 8 to 50+ mils (200+ hp)



FIGURE 21.3 Large electric motor supported in sliding bearings.

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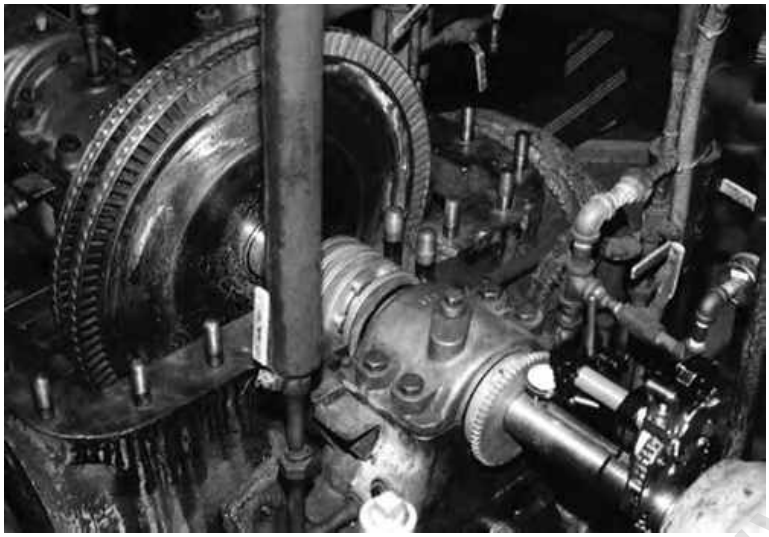


FIGURE 21.4 Small steam turbine with upper casing removed.

21.1.1.1 Additional Information on Electric Motors

Moderate to excessive soft foot conditions have been experienced on virtually every size motor regardless of frame construction design. Uneven air gap problems found occasionally due to improper positioning of end bells or housing distortion due to uncorrected soft foot. Inboard (coupling end) bearings may run hotter due to misalignment conditions. Excessive vibration may be due to improperly bored coupling hubs. Infrared thermography surveys and motor current signature analysis are very helpful in diagnosing problems.

21.1.2 Steam Turbines

Steam turbines can range in output from 20 to 100,000+ hp with speeds up to 25,000+ rpm and therefore become some of the more interesting equipment for OL2R surveys and consequentially some (Figure 21.4 through Figure 21.6) of the more difficult equipment to maintain and operate properly. Steam pressures can range from 200 to 4000+ psig and temperatures from 400°F to 1100°F. Due to the fact that a high-temperature gas is used to propel blades for shaft rotation, extensive frame and casing design considerations concerning

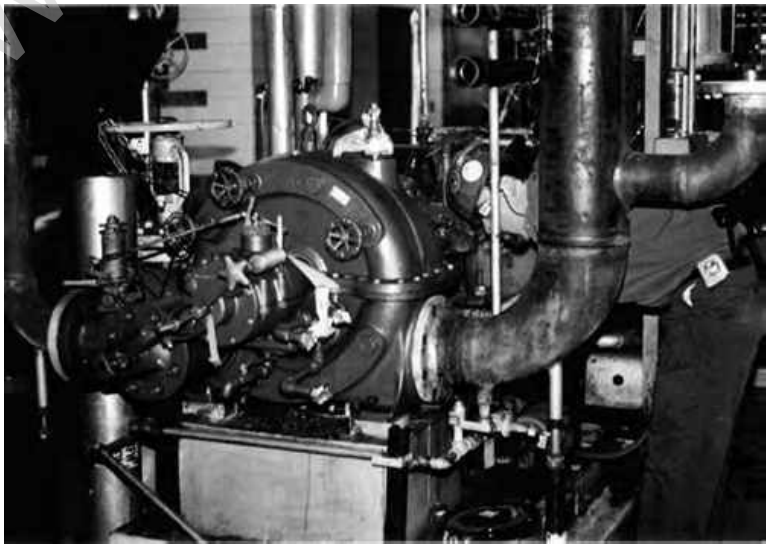


FIGURE 21.5 Small steam turbine.

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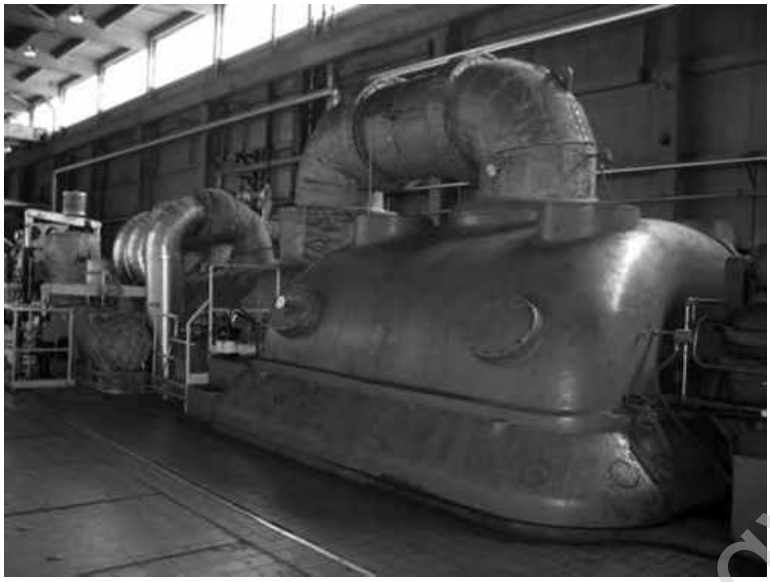


FIGURE 21.6 Large steam turbine.

casing and rotor expansion and contraction are taken into account to minimize excessive positional change of the rotor during operation. However, movement of the shaft invariably occurs from OL2R conditions that can range considerably from unit to unit. In addition, rotor expansion must be taken into consideration when selecting a flexible coupling to prevent thrust transfer from one rotor to another, causing premature bearing or coupling failure. On several occasions, the condensing end of the steam turbine has been observed to move downward during operation. The cooler temperatures and the "vacuum draw down" effect of the condenser may actually move the condenser end opposite of what one might expect. Again, since there is such a wide variety of equipment in existence, it is always best to consult with your equipment manufacturer for initial installation, design modification, overhaul, or operational problems with these units. Thank them for their input, but always do your own research.

Typical OL2R movement range of steam turbines (horizontally mounted):
Vertical movement: –10+ to 25 mils upward (5 to 500 hp); 5 to 40+ mils upward (500+ hp), typically asymmetrical (i.e., inboard and outboard ends do not move up the same amount)
Lateral (*sideways*) *movement*: 0 to 40+ mils (can be as much or considerably more than the vertical movement)
Axial movement: 10 to 100+ mils (5 to 200 hp); 20 to 250+ mils (200+ hp)

21.1.2.1 Additional Information on Steam Turbines

Moderate to excessive off-line soft foot conditions have been experienced on virtually every size steam turbine regardless of frame construction design. Frequently, on small- to medium-sized steam turbines, one end of the casing is rigidly bolted to the frame and a "sway bar" or flexible support is mounted at the other end to allow for axial expansion to occur to prevent casing warpage during operation. Sometimes on larger steam turbines, the casing is keyed at the casing centerline and the hold-down bolts are not tightened to lock the casing against the frame support but are kept loose to allow for symmetric lateral and axial casing expansion to occur.

The lateral movement that occurs is often directly related to the expansion and contraction of the steam piping connected to the steam turbine casing and proper design and installation of the piping system is imperative to minimize static (off-line) and dynamic (running) nozzle loads.

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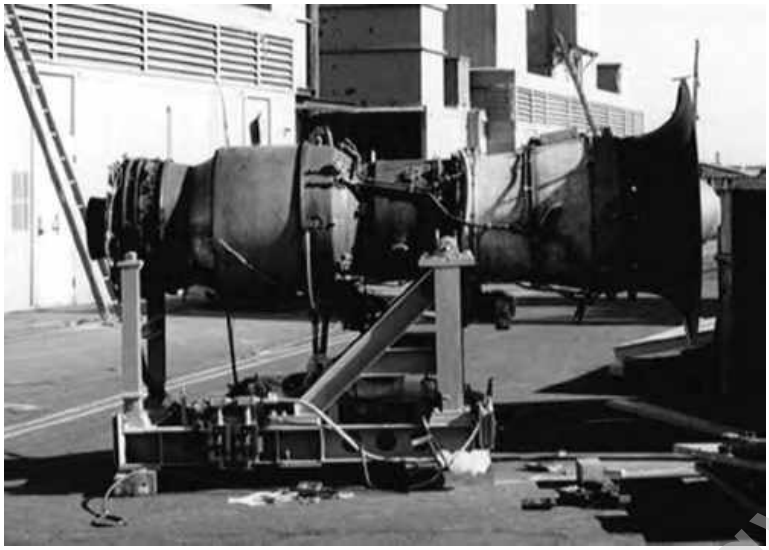


FIGURE 21.7 Gas turbine.

Most steam turbines are supported in sliding-type bearings and therefore exhibit a certain amount of axial clearance between the thrust runner and the active–inactive thrust bearings (often referred to as thrust float). When setting the machinery axial positions off-line, seat the thrust runner against the active thrust bearing before measuring and adjusting the shaft-to-shaft distance. Bear in mind that the axial movement amounts mentioned above are for the casing and housing. The shaft may expand more than that and may influence how you should set the off-line shaft end to shaft end distances.

21.1.3 Gas Turbines

Industrial gas and power turbine drivers are used in a wide variety of applications ranging from compression of gases and electrical generation to propulsion systems for ships (Figure 21.7 and Figure 21.8). The Brayton cycle (i.e., a gas turbine) compresses air via a

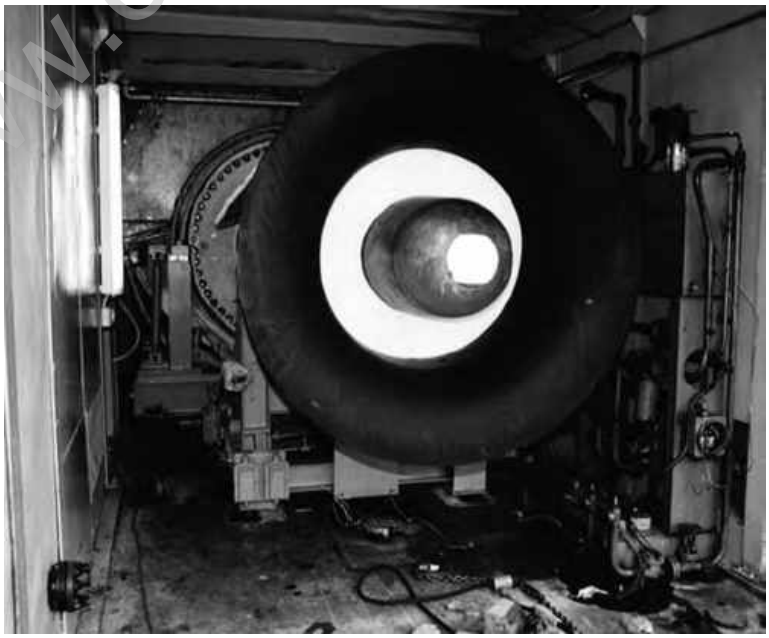


FIGURE 21.8 Gas turbine driving an electric generator.

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centrifugal or axial flow compressor where the compressed air is mixed with fuel (liquid jet fuel or natural gas) and burned. The hot, high-velocity gas then impinges on a series of several stages of curved blade sets (power turbine) that is used to rotate the driven machinery. Frequently, the gas and power turbines, although separate rotors supported in their own bearings, share a common casing and frame. The residual high-velocity gas is then vented through ductwork that sometimes houses a heat exchanger for a closed loop system or for use in heating liquids for other purposes.

The gas turbine produces a tremendous amount of forward thrust in reaction to the high-velocity gas escaping out of the tail end of the machine. A considerable amount of heat is generated in the cycle and a twisting or torsional counter reaction occurs in the frame during operation. These factors all contribute to some of the most radical OL2R machinery movement in any type of driver used today.

Typical OL2R movement range of gas or power turbines:
Vertical movement: Intake end—10+ mils downward to 10+ mils upward; exhaust end—5 to 80+ mils upward
Lateral (*sideways*) *movement*: Intake end—2 to 20+ mils; exhaust end—2 to 60+ mils
Axial movement: See additional information

21.1.3.1 Additional Information on Gas Turbines

Moderate to excessive off-line soft foot conditions have been experienced on virtually every size gas and power turbine regardless of frame construction design. Movement in the axial direction from OL2R conditions can also be excessive. Forward movement of gas turbines (i.e., toward the intake end) has been observed to translate 180+ mils. Gear- or diaphragm-type couplings have been employed at the output shaft to drive the equipment. If the coupling is a diaphragm-type (or any flexible disk-type) and there is movement toward the intake end, damage could occur to the coupling and the thrust forces can be transmitted to the driven machine. The shaft-to-shaft distance between the power turbine and the driven equipment shaft is usually 40+ in. in an attempt to minimize the effect from large amounts of OL2R movement and to minimize any heat transfer from the exhaust duct work to the driven machine. Bear in mind that the axial movement amounts mentioned above are for the casing and housing. The shaft may expand more than that and may influence how you should set the off-line shaft end to shaft end distances.

21.1.4 Internal Combustion Engines

Very few field studies have been conducted (or at least published) on how internal combustion engines move from OL2R conditions (Figure 21.9). Diesel engines, for example, are frequently used to drive backup electrical generators, fire pumps, and portable air compressors. In the wastewater treatment industry, biogas engines can be used to drive the air compressors. The crankshaft is typically set very low in the casing and engine mounts can be found below, at, or slightly above the centerline of rotation of the crankshaft. The relatively few studies that have been done have still shown OL2R machinery movement regardless of the casing support mounting location. Flexible coupling design is somewhat critical since variations in torque occur as each piston delivers rotational force at varying intervals.

Typical OL2R movement range of internal combustion engines:
Vertical movement: 1 to 5 mils upward (5 to 200 hp); 2 to 20+ mils upward (200+ hp), typically symmetrical (i.e., inboard and outboard ends move up the same amount)

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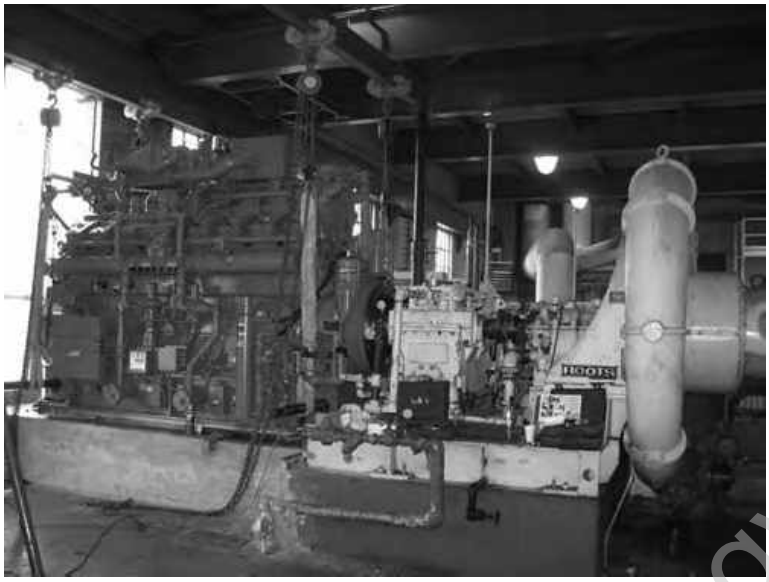


FIGURE 21.9 Sixteen cylinder biogas engine coupled to a gearbox and compressor.

Lateral (*sideways*) *movement*: 0 to 4 mils (usually much less than any vertical movement)
Axial movement: Unknown

21.1.4.1 Additional Information on Internal Combustion Engines

Moderate to excessive off-line soft foot conditions have been experienced on virtually every size internal combustion engine regardless of frame construction design. On medium and large engines, distortion of the engine frame during installation is a concern. To insure that the crankshaft bearings are not distorted, web deflection tests are conducted as shown in Figure 21.11 and Figure 21.12. A web deflection test determines if the distance between the crank webs is changing when the crankshaft is rotated. If the bearings are misaligned due to casing distortion, or there is an excessive amount of shaft misalignment with the coupling engaged, the distance between the crank webs will vary when the crankshaft is rotated. If the gap variation between the web is excessive, shims must be added between the engine and the soleplates to relieve the distortion of the casing.

21.1.5 Horizontally Mounted Centrifugal Pumps

Without a doubt, one of the most common drive systems in virtually every industry is a motor-driven, horizontally mounted, centrifugal pump (Figure 21.13 through Figure 21.15). There are several hundred designs of centrifugal pumps and it would be difficult to cover every characteristic of each design used in industry. Their purpose is basically to move an incompressible fluid from point A to point B. The temperature of the fluid conveyed has a great effect on the OL2R conditions of the pump. As discussed in Chapter 5, the piping attached to the pump can have a tremendous influence on obtaining and maintaining accurate alignment, so that many people are unwilling to even try to reposition pumps, henceforth declaring them the "stationary" machine when aligning them.

Typical OL2R movement range of centrifugal pumps:
Vertical movement: 0 to 80+ mils upward typically asymmetrical (i.e., inboard and outboard ends do not move up the same amount)

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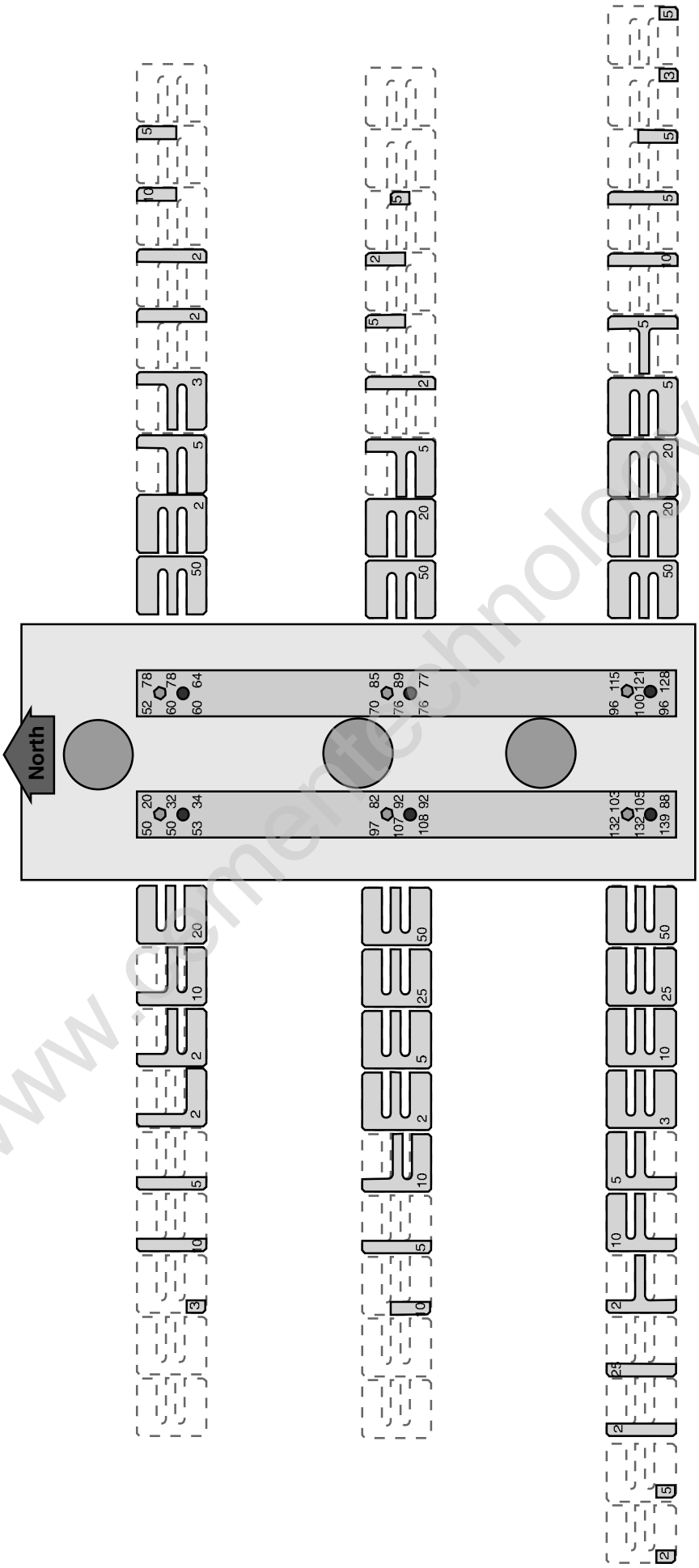


FIGURE 21.10 Soft foot map between engine frame and soleplates on biogas engine shown in Figure 21.9.

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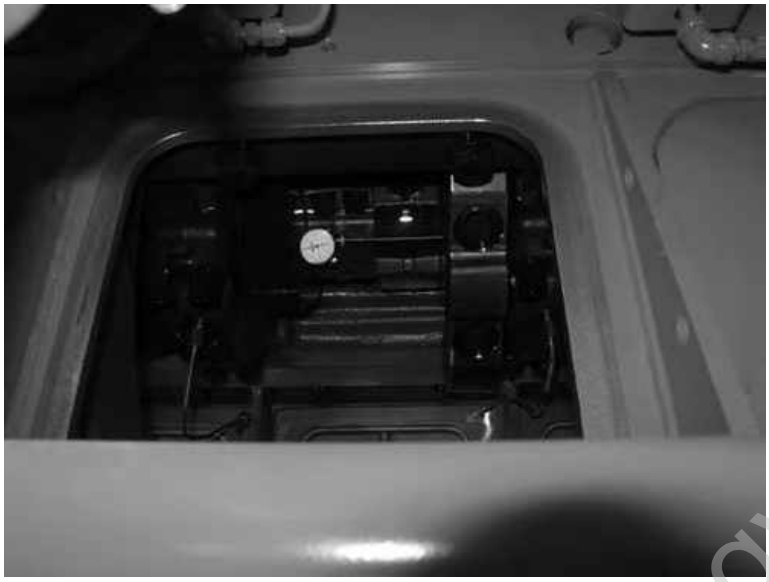


FIGURE 21.11 Inside dial gauge used to measure web deflection.

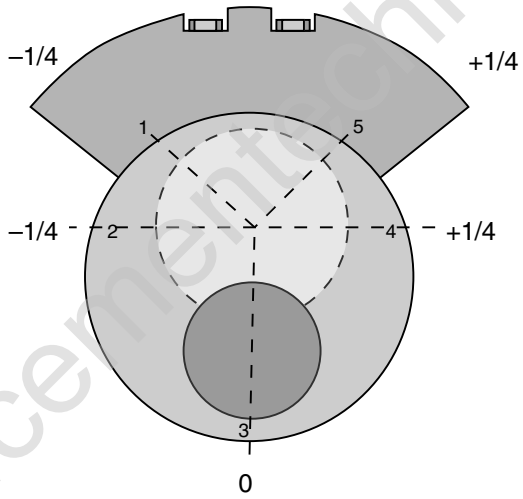


FIGURE 21.12 Web deflection measurements typically taken at five positions.

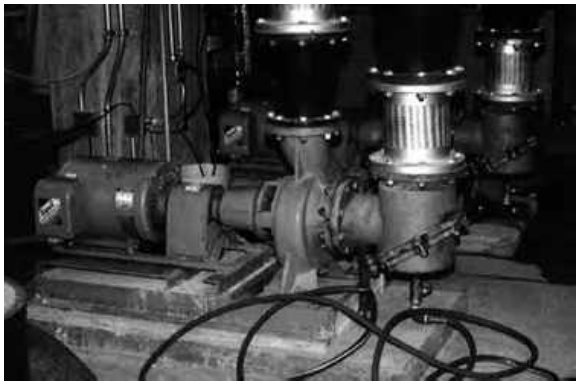


FIGURE 21.13 Single-stage centrifugal pumps with overhung impeller.

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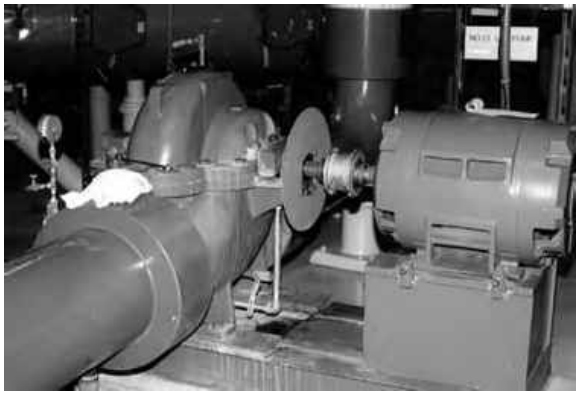


FIGURE 21.14 Single-stage centrifugal pump with centered impeller.

Lateral (*sideways*) *movement*: 0 to 90+ mils (can be much greater than vertical movement and is usually asymmetrical)
Axial movement: 0 to 150+ mils, frequently dependent on temperature of process fluid

21.1.5.1 Additional Information on Horizontally Mounted Centrifugal Pumps

Moderate to excessive off-line soft foot conditions have been experienced on virtually every centrifugal pump regardless of frame construction design. Maintaining long-term alignment of ANSI- and API-type pumps can be difficult due to the loosely supported inboard (coupling) end of the pump case. Failure of mechanical seals can often be attributed to misalignment conditions. Excessive leakage on mechanically packed pumps can also be attributed to misalignment conditions. Pumps can experience internal rubs due to rotor distortion caused

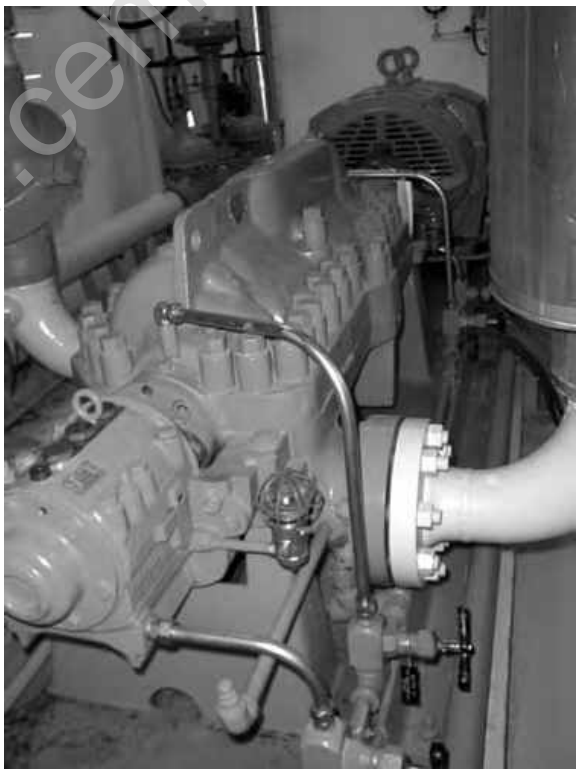


FIGURE 21.15 Multistage centrifugal pump.

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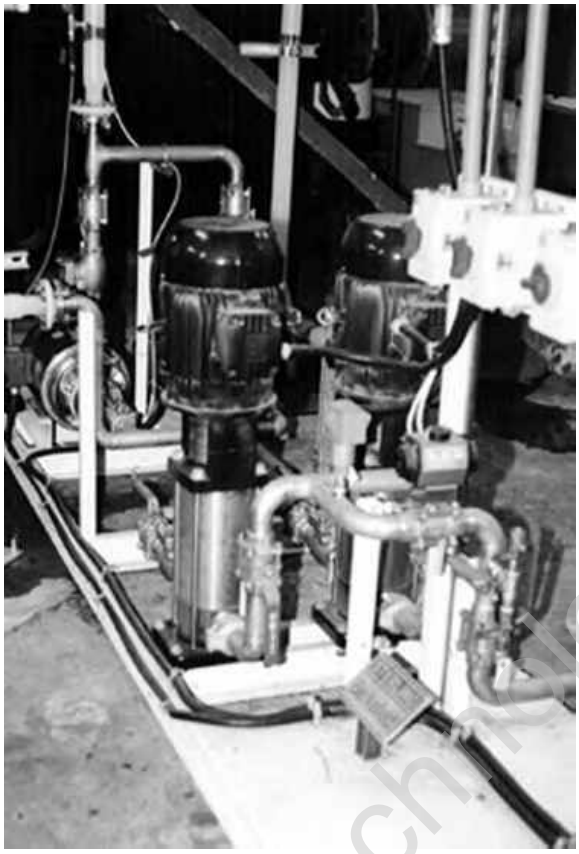


FIGURE 21.16 Small vertical pumps.

by moderate to excessive misalignment conditions. Bear in mind that the axial movement amounts mentioned above are for the casing and housing. The shaft may expand more than that and may influence how you should set the off-line shaft end to shaft end distances.

21.1.6 Vertically Mounted Centrifugal Pumps

There are several different types of vertical pumps such as well water pumps, in-line pumps, and reactor coolant pumps. In most cases, vertical pumps are driven by C-flanged motors. These motors are bolted to a cylindrical casting that is attached to the pump casing. In some situations, the pump is supported in its own bearings and the motor is flexibly coupled to the pump. In other situations, the pump is rigidly coupled to the motor shaft and the thrust load is supported by a thrust or radial bearing at the top of the motor. The assumption that many people have is that no alignment is required for these types of machines since the motor, connector casting, and pump casing are perfectly machined, rabbeted fits that precisely align the motor shaft to the pump shaft. In most cases, this is not true. Misalignment can and does occur on these types of drives as often as a horizontally mounted drive system (Figure 21.16 through Figure 21.18).

Figure 21.19 shows a large vertical pump driven by a 2500-hp motor, which is bolted to the pump casing with 12 bolts. The pump was new and was experiencing excessive vibration where misalignment was suspected as the cause. The coupling connecting the motor shaft to the pump shaft is a rigid coupling. The upper bearing of the motor has a thrust bearing that supports the weight of the armature and the weight of the pump shaft. Upper and lower bronze bushings act as the radial bearings for the pump shaft. These bushings are lubricated by the water that is pumped upward from the impeller at the lower end of the pump shaft.

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FIGURE 21.17 Medium-sized vertical pumps.

As mentioned previously, any attempt to align shafts that are connected together with a rigid coupling are futile. The misalignment can be severe and the shafts will elastically bend to accommodate the misalignment condition making it appear that the alignment is acceptable when capturing readings across the engaged rigid coupling. To properly align a unit like this, the coupling must be disengaged. In doing that however, the pump shaft drops down from its own weight and the impeller touches the housing at the bottom. Any attempt to rotate the pump shaft after the coupling has been disengaged can potentially damage the impeller. Since the motor shaft can still be rotated, either the face–rim or double radial alignment methods could be used. In this particular case, the double radial method was used to check the alignment between the two shafts. Before disconnecting the coupling however, runout measurements can be taken to determine if the coupling hubs are bored properly (i.e., concentric) and if the motor or pump shafts are permanently bent. Figure 21.20 shows the runout measured on the shafts and the coupling hubs.

After the runout measurements were taken, the mechanical seal was removed and the coupling was disengaged. The specified distance between the end of the motor shaft and the end of the pump shaft was 0.250 in. There is an adjustment nut on the top of the pump

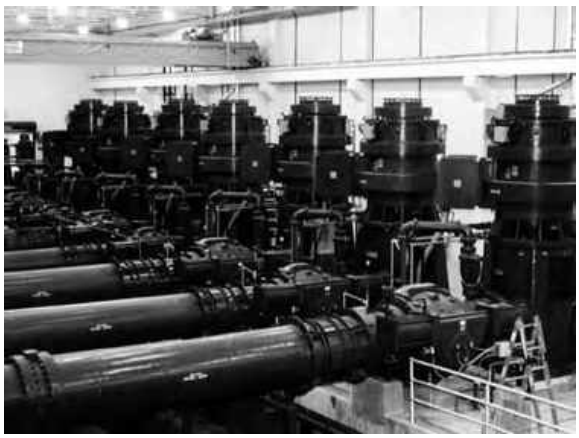


FIGURE 21.18 Large vertical pumps.

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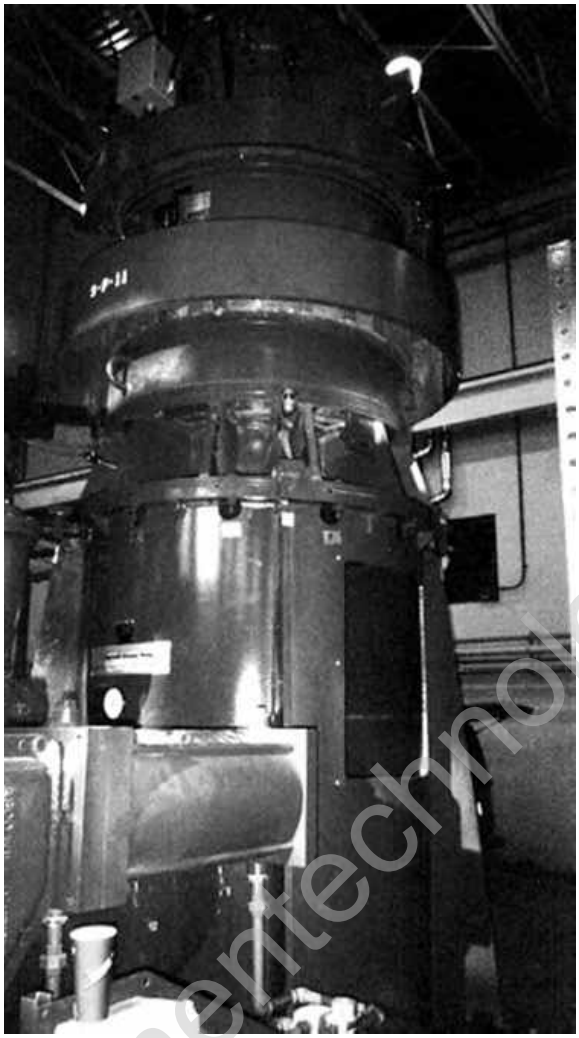
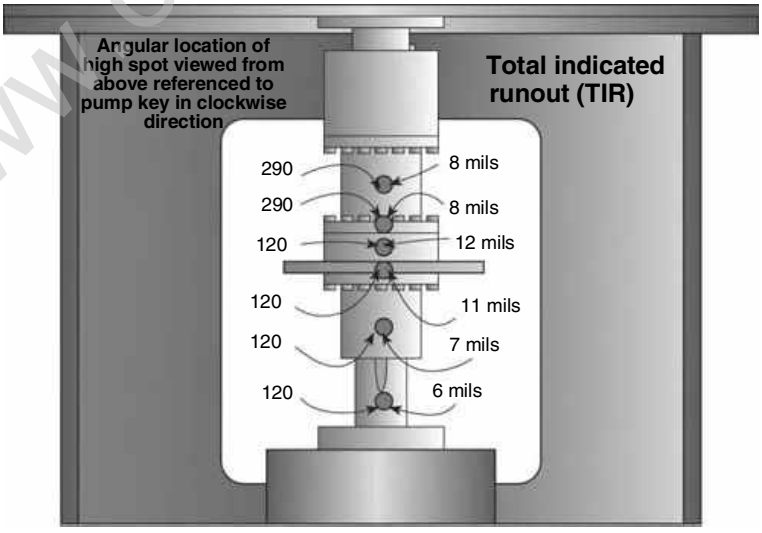


FIGURE 21.19 Vertical pump.



As found gap between rotating mechanical seal cartridge and top of stuffing box = 0.293 in.

As found adjusting nut to coupling spool clearance with coupling bolts removed = 0.263 in.

FIGURE 21.20 Runout measurements taken on pump shown in Figure 21.19.

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FIGURE 21.21 Centering the pump shaft in its upper bushing.

shaft that can be rotated to obtain the desired shaft-to-shaft distance with the coupling disengaged. The pump shaft was then centered in its upper bushing as shown in Figure 21.21 and Figure 21.22. Feeler gauges were used to measure four points between the shaft and the bushing as shown in Figure 21.23. The specified total radial clearance in the bushing was to range from 6 to 12 mils. Notice that the measured gaps exceed the specified amount and that they are not the same in the east to west direction compared to the north to south direction.

Now the pump shaft is centered in the upper bushing and the coupling disengaged, alignment readings can be taken between the shafts. Figure 21.24 shows capturing the alignment reading on the adjustment nut and Figure 21.25 shows capturing the alignment reading on the pump shaft just above the stuffing box area. Another reading was taken on the balance ring just below the adjustment nut. The as-found alignment readings are shown in Figure 21.26.

To plot the misalignment condition, two views will be generated. One view will show the misalignment in the north to south direction as shown in Figure 21.27 and another view will show the misalignment in the east to west direction as shown in Figure 21.28. A T-bar overlay will be used in this modeling method. The top part of the T-bar overlay will represent the mating flange surface where the motor bolts to the pump housing. Each bolting plane has been scaled off on the top part of the T. Carefully study the bolt plane designations in

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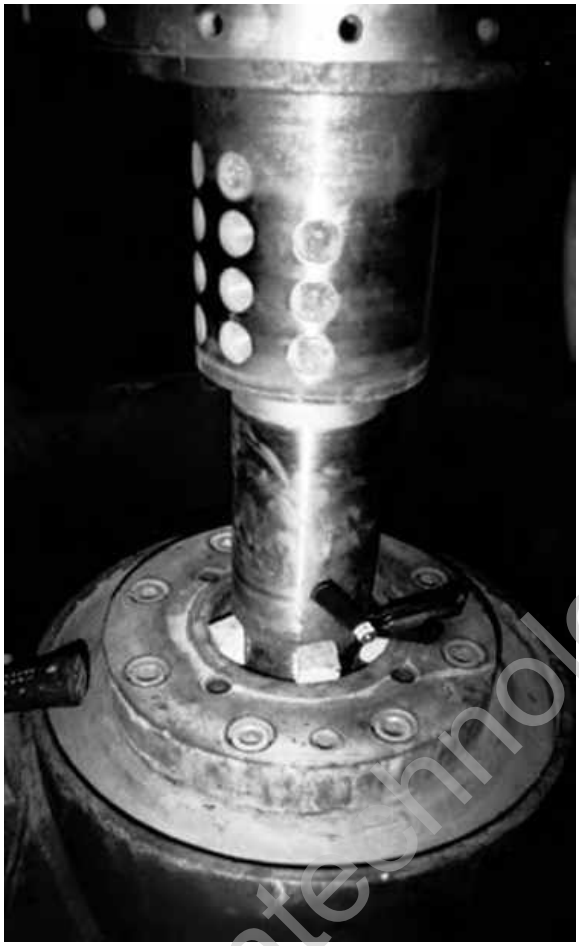


FIGURE 21.22 Wooden wedges were used to keep the pump shaft centered.

Figure 21.27 and Figure 21.28. The base to the T-bar overlay will represent the centerline of rotation of the motor shaft.

Since there was runout observed on the motor and pump coupling hubs, before disengaging the coupling, the shafts were rotated so that the high spots were placed on the north side of the shafts. With the high spots physically positioned on the north side, we can compensate for

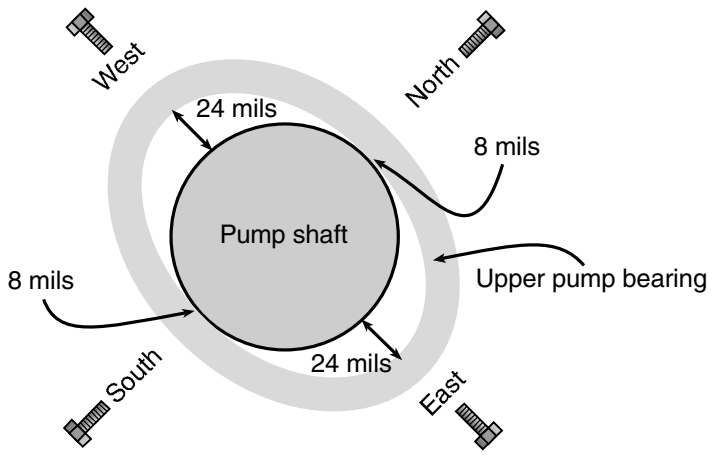


FIGURE 21.23 Gap measurements at the upper bushing of the pump.

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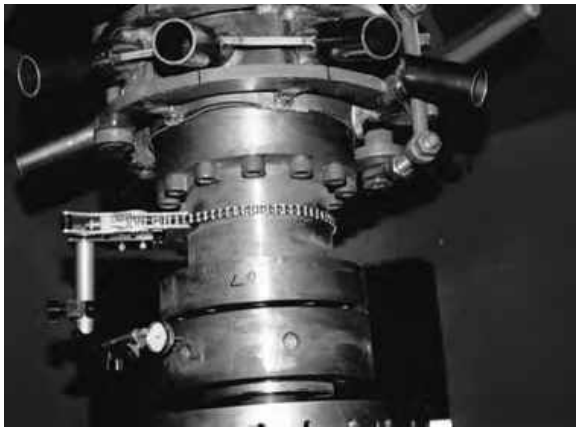


FIGURE 21.24 Alignment readings taken at the top of the pump shaft at the adjustment nut.

the eccentricity and determine where the actual centerline of rotation of the pump shaft is as shown in Figure 21.26.

The top part of the T-bar overlay shows what thickness of shims need to be installed between each of the 12 bolts that mate the motor housing to the pump housing to correct for the angular misalignment. Bear in mind that each of the 12 flange bolts appear in both views and that there is an angular misalignment condition in both views. Bolt by bolt, add the number of shims required to correct the angular misalignment from the north–south and east–west views. Once the shim totals for each bolt have been added together, determine



FIGURE 21.25 Alignment readings taken at the bottom of the pump shaft just above the stuffing box.

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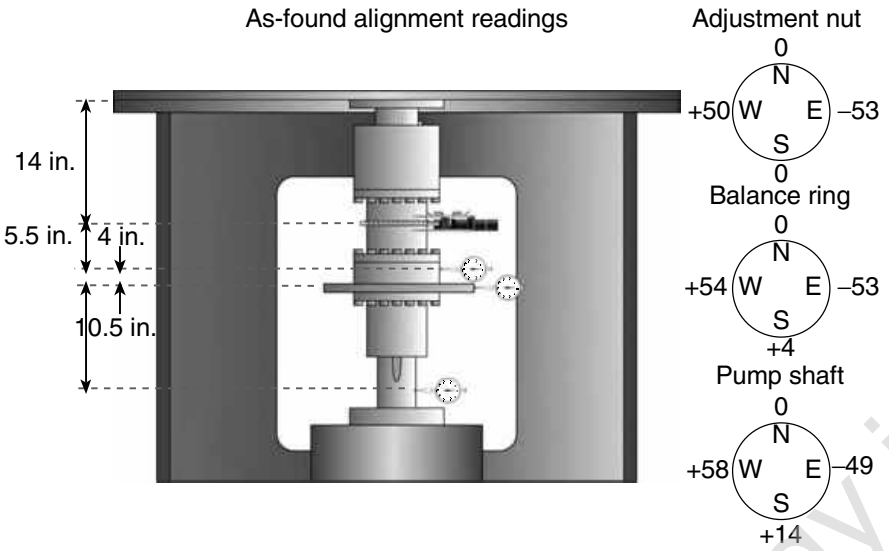


FIGURE 21.26 As-found double radial alignment measurements.

which bolt requires the least amount of shims and subtract that amount from all the bolts as shown in Figure 21.29. Ironically, the least amount of shims happens to occur at bolt A.

At this point, all of the bolts were loosened and soft foot gaps were measured at each bolt as shown in Figure 21.30. Improper contact can occur on these types of machines also.

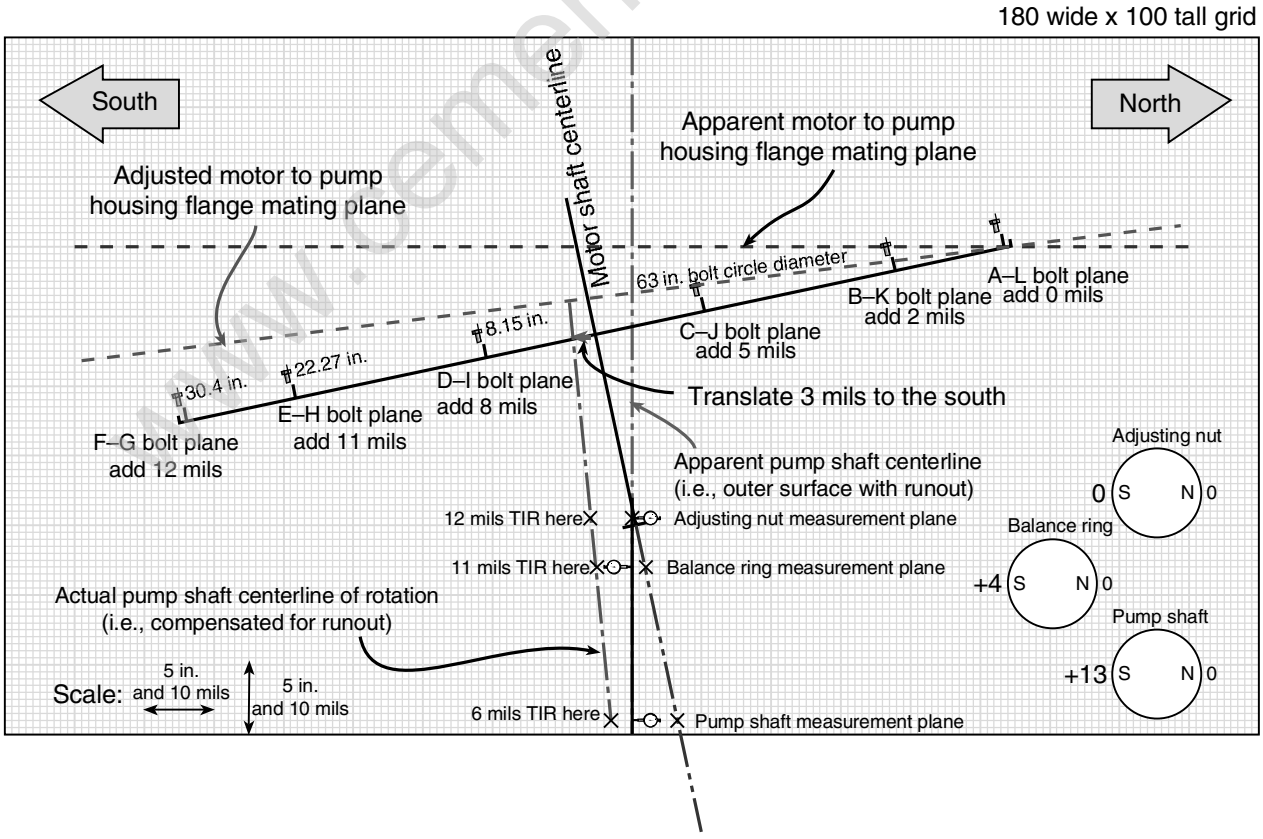


FIGURE 21.27 (**See color insert following page 322.**) As-found alignment model of motor and pump shaft as viewed in the north to south direction.

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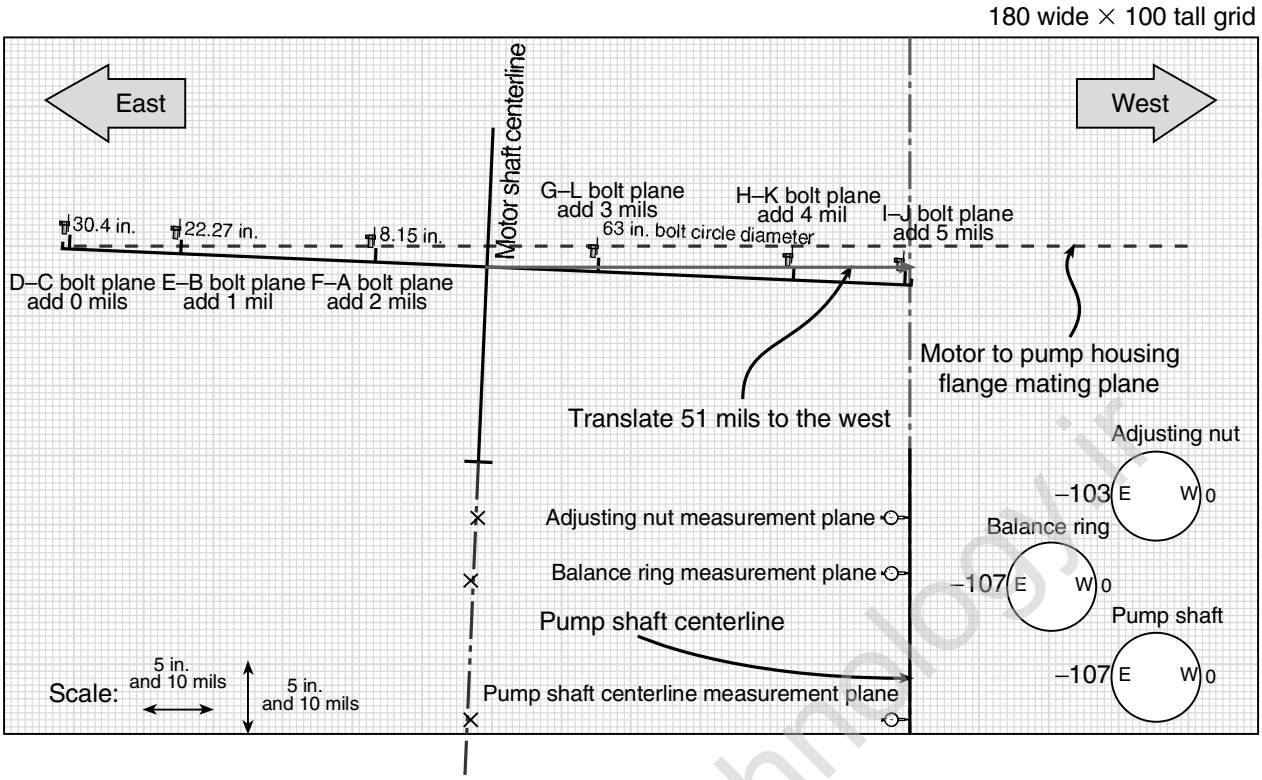


FIGURE 21.28 (**See color insert following page 322.**) As-found alignment model of motor and pump shaft as viewed in the east to west direction.

Bolt ID	Shims for north–south view	Shims for east–west view	Total shims for both views
A	0	2	2
B	2	1	3
C	5	0	5
D	8	0	8
E	11	1	1 2
F	1 2	2	14
G	1 2	3	15
H	1 1	4	15
I	8	5	1 3
J	5	5	1 0
K	2	4	6
L	0	3	3

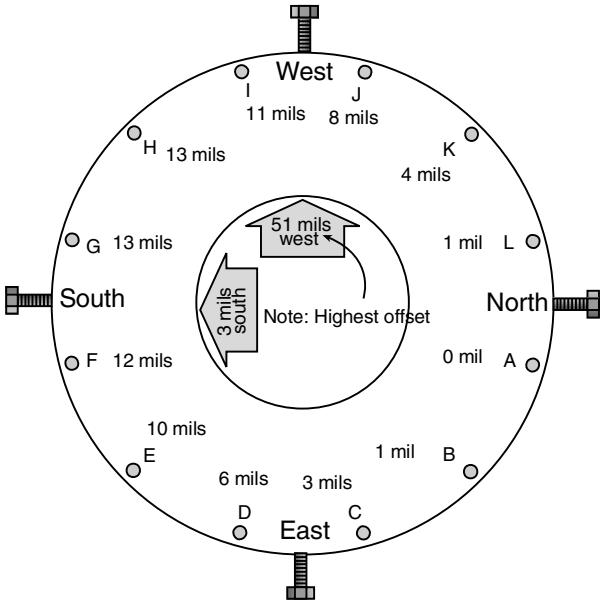


FIGURE 21.29 Shims to correct the angular misalignment and translations to correct the offsets.

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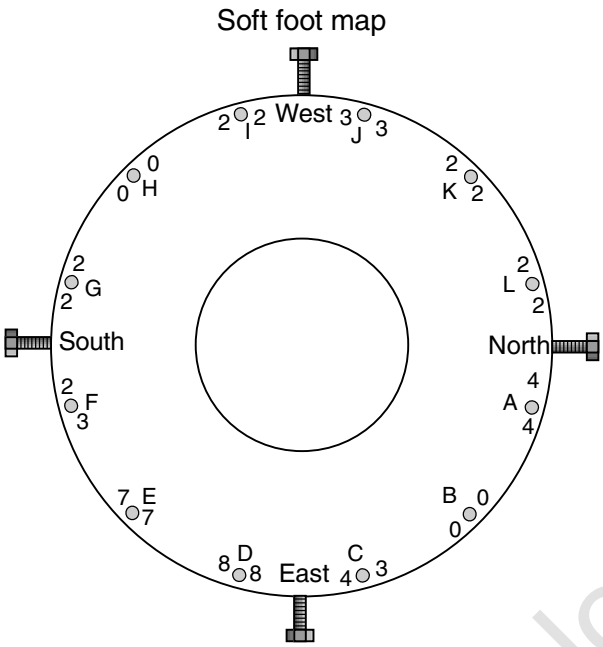


FIGURE 21.30 Soft foot gap measurements at all the flange bolts.

Notice that there is a greater amount of angular misalignment in the north–south direction than the east–west direction. This is not uncommon with C-flanged equipment. The worst misalignment condition was not the angular problem but the offset particularly in the east–west direction. Notice in Figure 21.28 that the motor shaft had to be translated 51 mils to the west to align it to the pump shaft. To accomplish this, the lateral positioning jackscrews were adjusted as shown in Figure 21.31.

Despite the fact that this unit had only 20 h of operation on it, the excessive offset misalignment in the east–west direction was so severe that it damaged the upper pump bushing as shown in Figure 21.32. This explained why there was an excessive amount of clearance in the east to west direction of the upper pump bushing as shown in Figure 21.21.

Another design variation in vertical pumps incorporates a shaft extension that is attached to the end of the pump shaft via a threaded coupling. This shaft extension then goes through a hollow motor shaft and is then attached to the upper bearing of the motor via a (+)-shaped hub with an adjustment nut to raise the pump shaft a specified amount. There are no



FIGURE 21.31 Adjusting the jackscrews to correct the offset misalignment.

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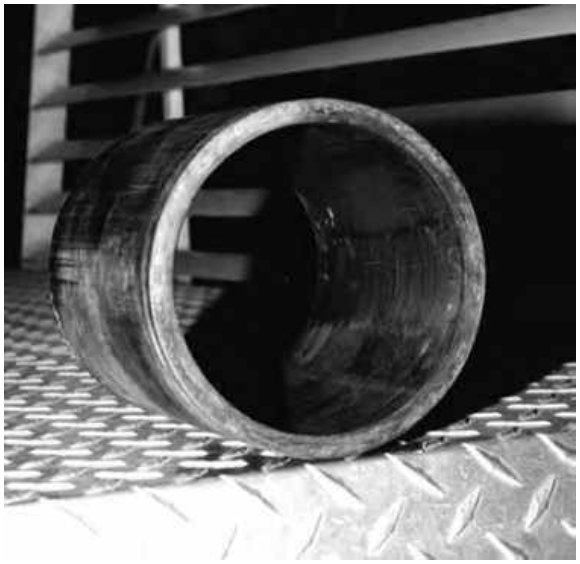


FIGURE 21.32 Excessively worn upper pump bushing due to excessive lateral misalignment.

provisions in this design to align the centerline of rotation of the motor with the centerline of rotation of the pump shaft similar to the pump previously discussed. The assumption with this design is that the pump shaft, the pump shaft extension, the motor shaft, the mating flange surfaces of the motor and pump, the threaded coupling, etc. are all machined to tolerances that "automatically align" the drive system when it is assembled. Problems do occur on these machines and there are ways to verify that the machining has been done properly so the alignment is correct. The verification process consists of the following measurements:

1. With the unit assembled, measure the clearance between the pump shaft extension and the hollow motor shaft as shown in Figure 21.33. Ideally the gaps should be even all the way around the shaft.

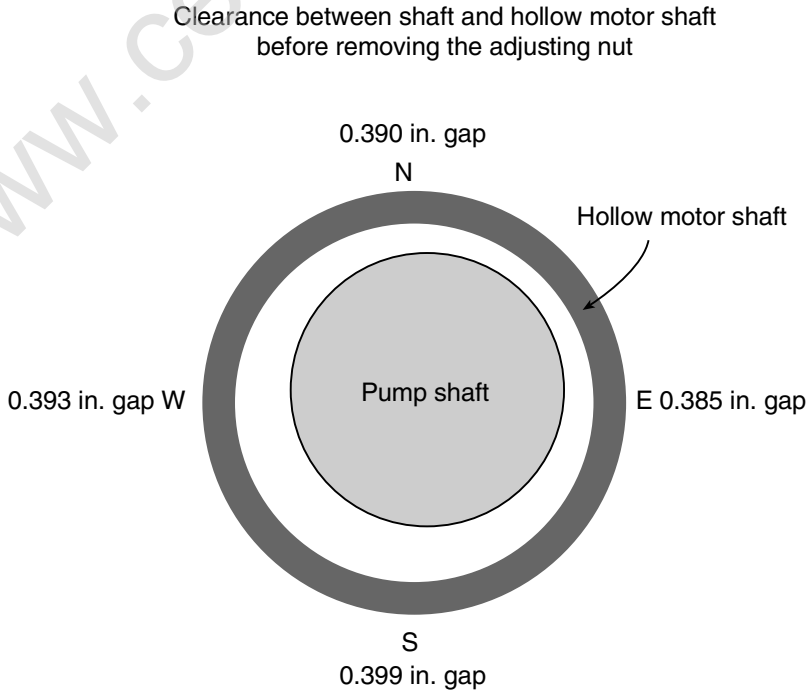


FIGURE 21.33 Measured clearances between the pump shaft extension and the hollow motor shaft.

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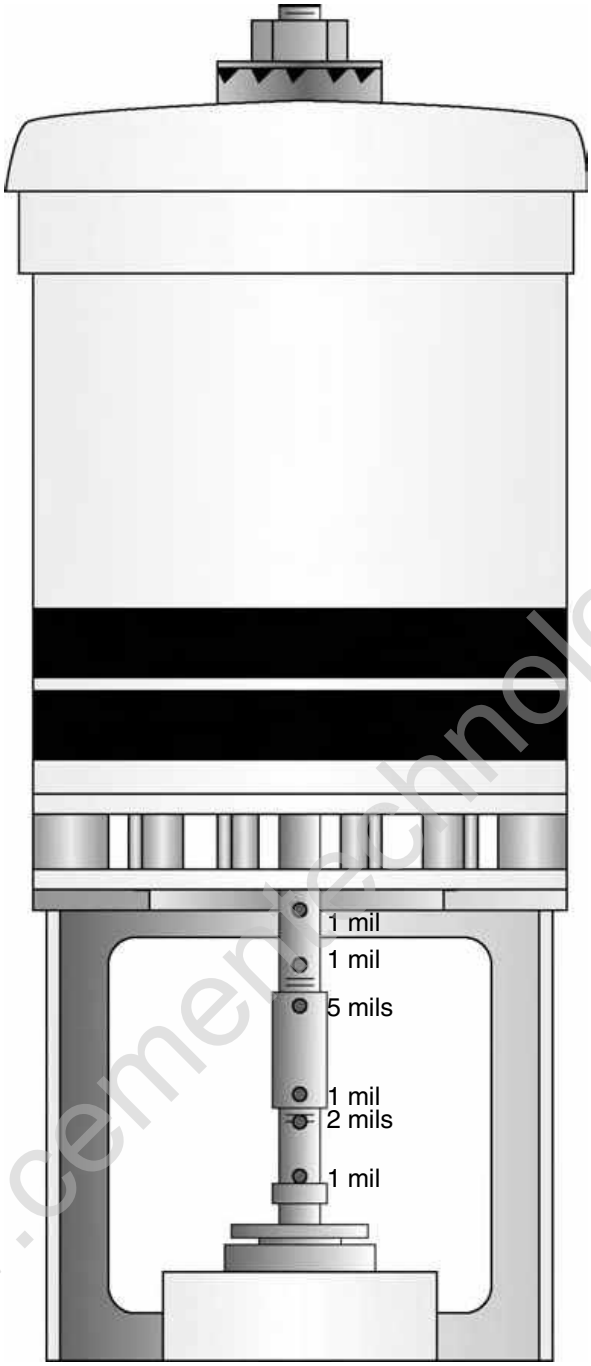


FIGURE 21.34 Measured runout at several points along the exposed pump shaft, threaded coupling, and pump shaft extension.

2. With the unit assembled, measure the runout at several points along the exposed pump shaft, threaded coupling, and pump shaft extension as shown in Figure 21.34. Also measure the runout at the upper hub that attaches the pump shaft extension to the upper motor bearing as shown in Figure 21.35 and Figure 21.36. Ideally the runout should adhere to the guidelines discussed in Chapter 5.
3. With the motor removed as shown in Figure 21.37, center the pump shaft in its upper bushing and measure the eccentricity on the pump shaft and the pump shaft extension as shown in Figure 21.38 through Figure 21.40. This requires fabricating a split collar to position a rolling element bearing that will be used to hold a bracket and indicator for

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FIGURE 21.35 Measuring runout on the upper hub.

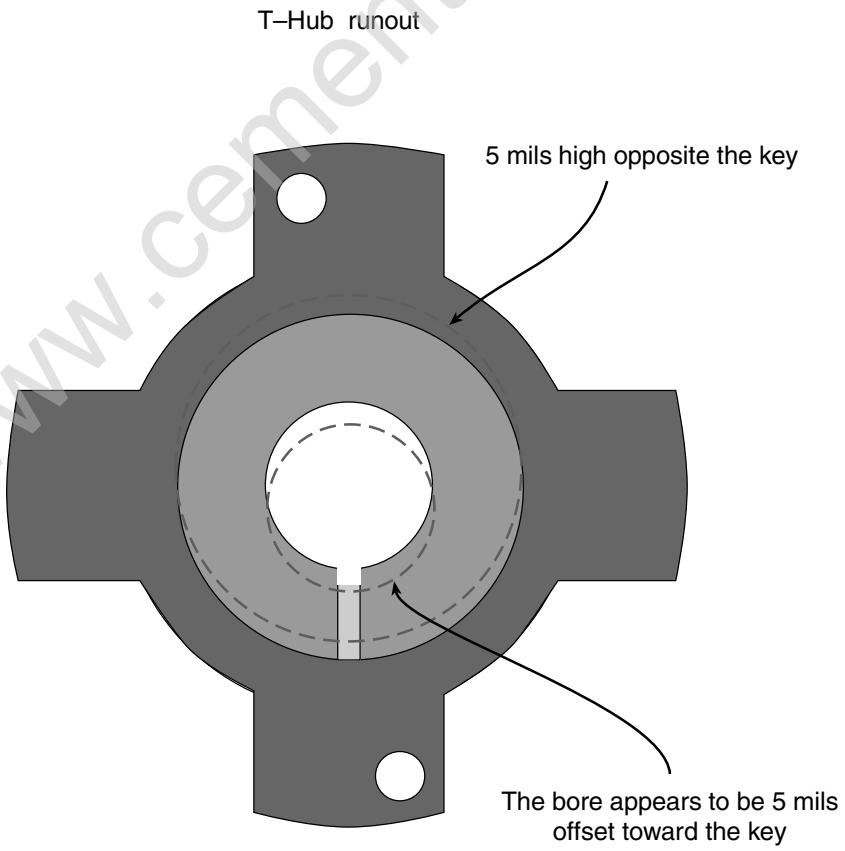


FIGURE 21.36 Runout at the upper hub that attaches the pump shaft extension to the upper motor bearing.

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FIGURE 21.37 Remove the motor.

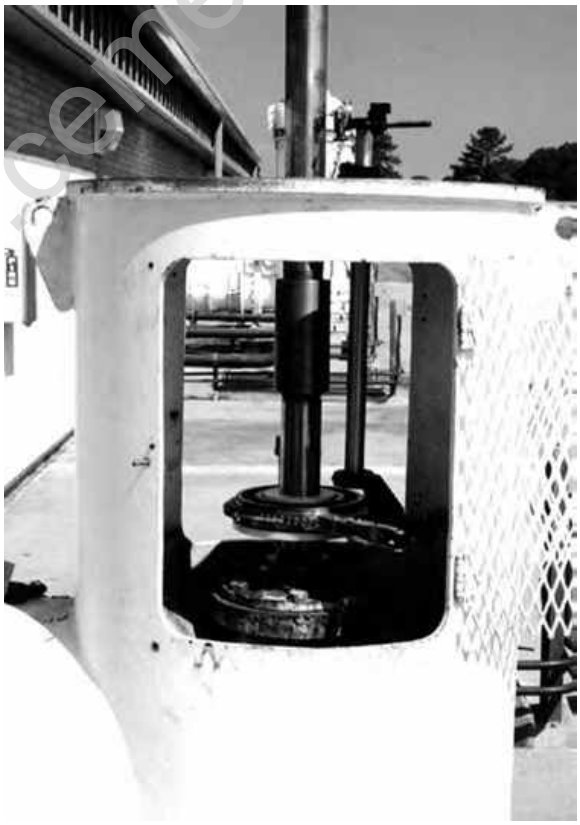


FIGURE 21.38 Measure the eccentricity at the lower end of the pump shaft.

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FIGURE 21.39 Measure the eccentricity at the upper end of the pump shaft.

the measurements as shown in Figure 21.41 and Figure 21.42. Ideally the runout should adhere to the guidelines discussed in Chapter 5.

4. Determine if the pump flange face is centered and perpendicular to the pump shaft as shown in Figure 21.43 through Figure 21.45. Since the mating flange faces are often rabbeted fits, the rabbet surface is used to check for concentricity and the flange face surface is used for perpendicularity checks.

21.1.6.1 Additional Information on Vertically Mounted Centrifugal Pumps

When the motor and pump are bolted together, they effectively become one contiguous frame and OL2R machinery movement has very little effect on the alignment of the shafts. Since the entire drive system is typically attached to a floor or the structure of the building, checks should be made to insure that the motor–pump assembly is firmly attached to the floor via the anchor bolts. Leakage at the packing gland and frequent replacements of the mechanical seal are indications that a misalignment condition or excessive runout may be the culprit. Excessive vibration can often be attributed to unbalance conditions in the motor armature or pump shafts but can also be traced to excessive runout conditions in the rotating assembly.

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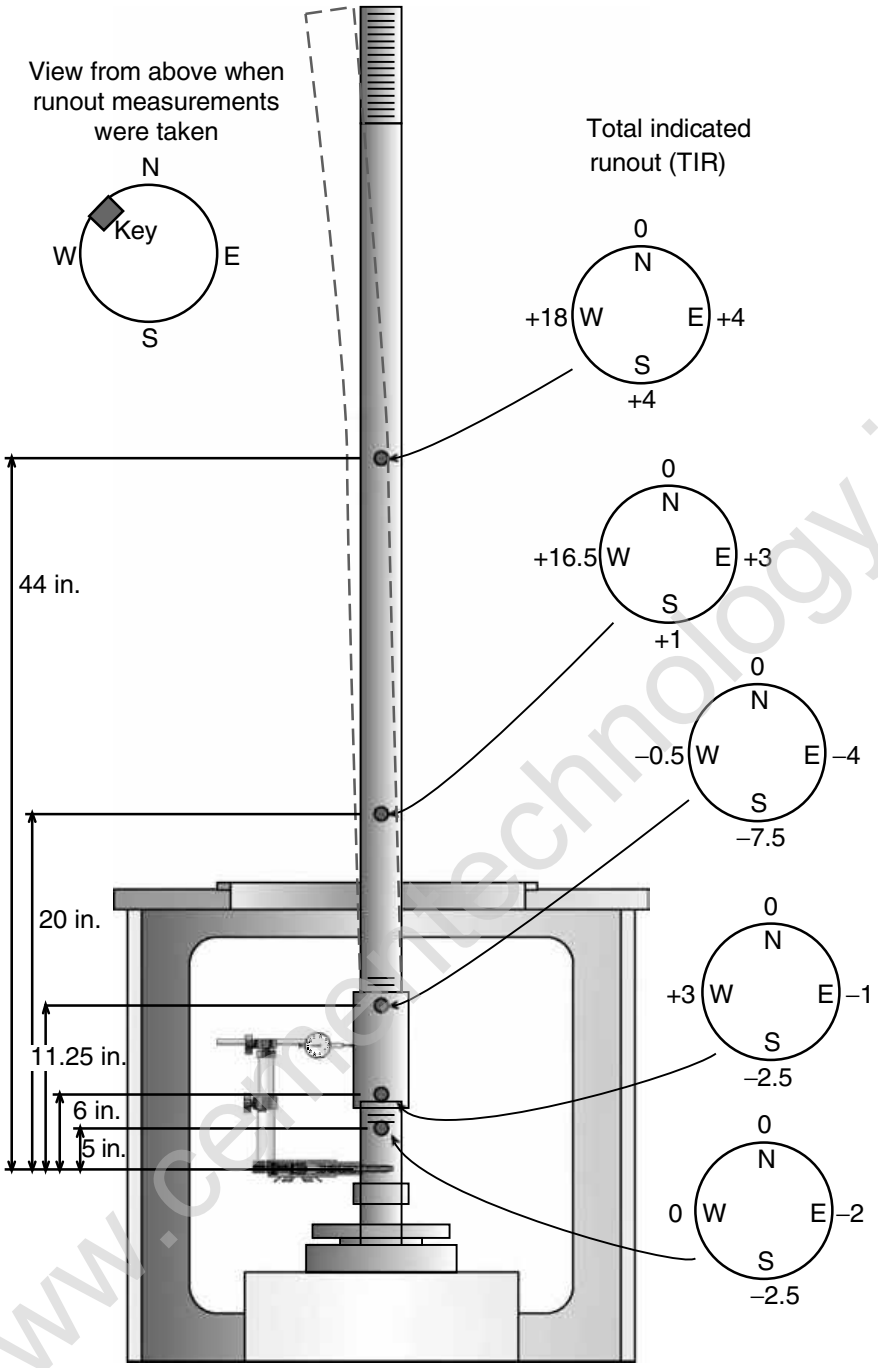


FIGURE 21.40 Eccentricity measurements along the pump shaft and its extension.

21.1.7 Blowers and Fans

There are several different designs of fans and blowers and again, it would be difficult to cover every aspect of these types of machines. Similar to pumps, their purpose is basically to move large volumes of a compressible fluid at low pressures from point A to point B. A large majority of smaller horsepower (5 to 200 hp) units are belt-driven as shown in Figure 21.46. Larger units are more frequently direct-driven as shown in Figure 21.47 and Figure 21.48. Again, the temperature of the gas that is conveyed has a great effect on the OL2R conditions of the fan.

As discussed in Chapter 5, the ductwork attached to the fan can have a tremendous influence on obtaining and maintaining accurate alignment, so that many people are

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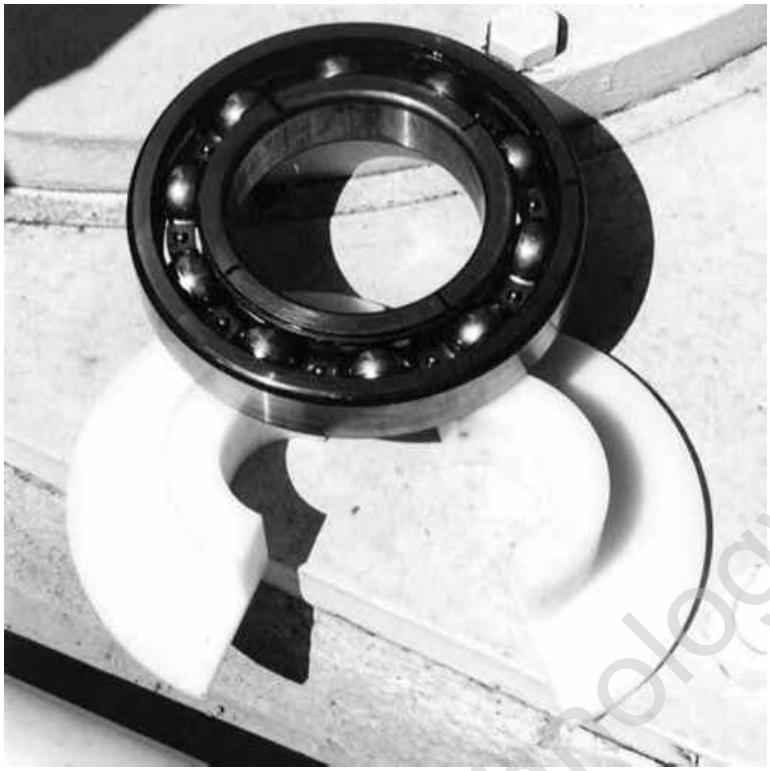


FIGURE 21.41 Split collar and bearing.

unwilling to even try to reposition fans and blowers and henceforth declare them the "stationary" machine when aligning them. In some situations, where the fan blades are center mounted on the shaft and the shaft is supported by bearings at each end, the position of the shaft is dictated by the positions of the bearing pedestals that are not directly attached to the fan housing. The fear in altering the position of the fan bearings is that internal fan blade to shroud clearances could be upset and rubs could occur. Here again, the graphing or modeling technique can be used not only to align the shafts, but also to position the fan housing to properly set fan blade to shroud clearances.

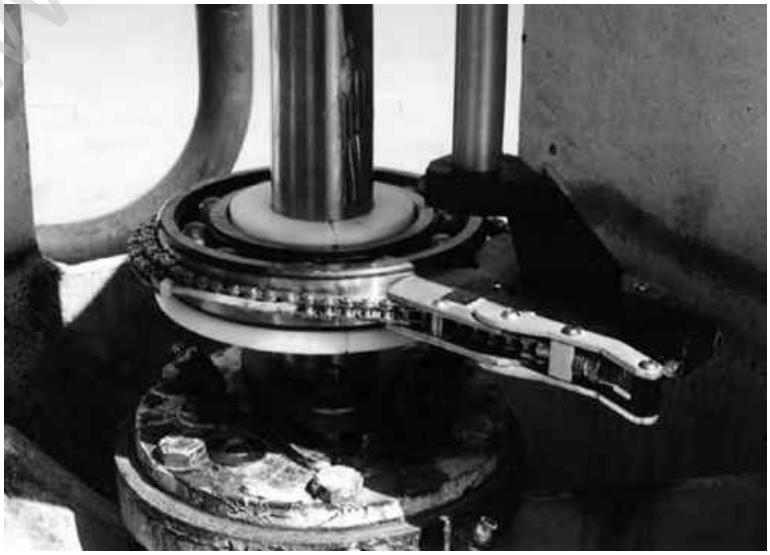


FIGURE 21.42 Bracket attached to outer race of rolling element bearing held in place with the split collar.

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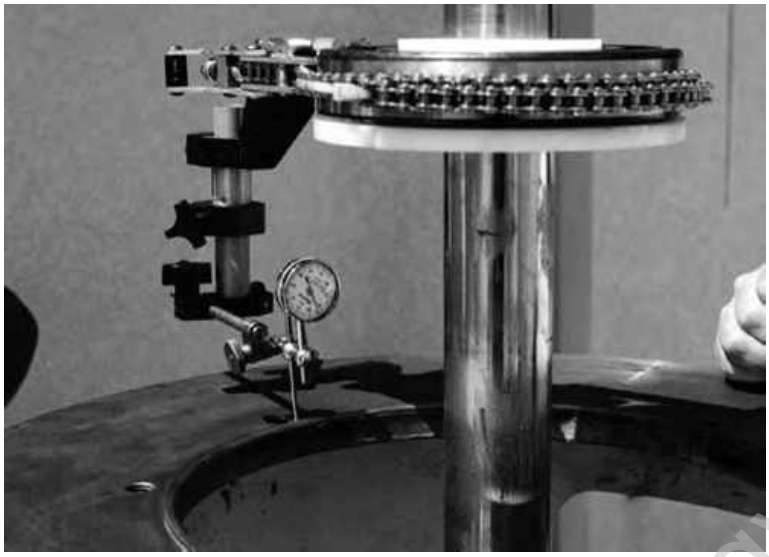


FIGURE 21.43 Checking for concentricity on the rabbeted surface.

Figure 21.49 shows a motor-driven fan, where the fan shaft and wheel are supported on pedestals that are separated from the fan housing itself. It is possible for the centerline of the fan housing not to be collinear with the centerline of rotation of the fan shaft. To determine where the center of the fan housing is, take gap measurements between the fan wheel and the shroud at the top, bottom, and both sides at the inboard and outboard ends. For example, top and bottom gap measurements were taken between the fan wheel and the stationary



FIGURE 21.44 Checking for perpendicularity on the flange surface.

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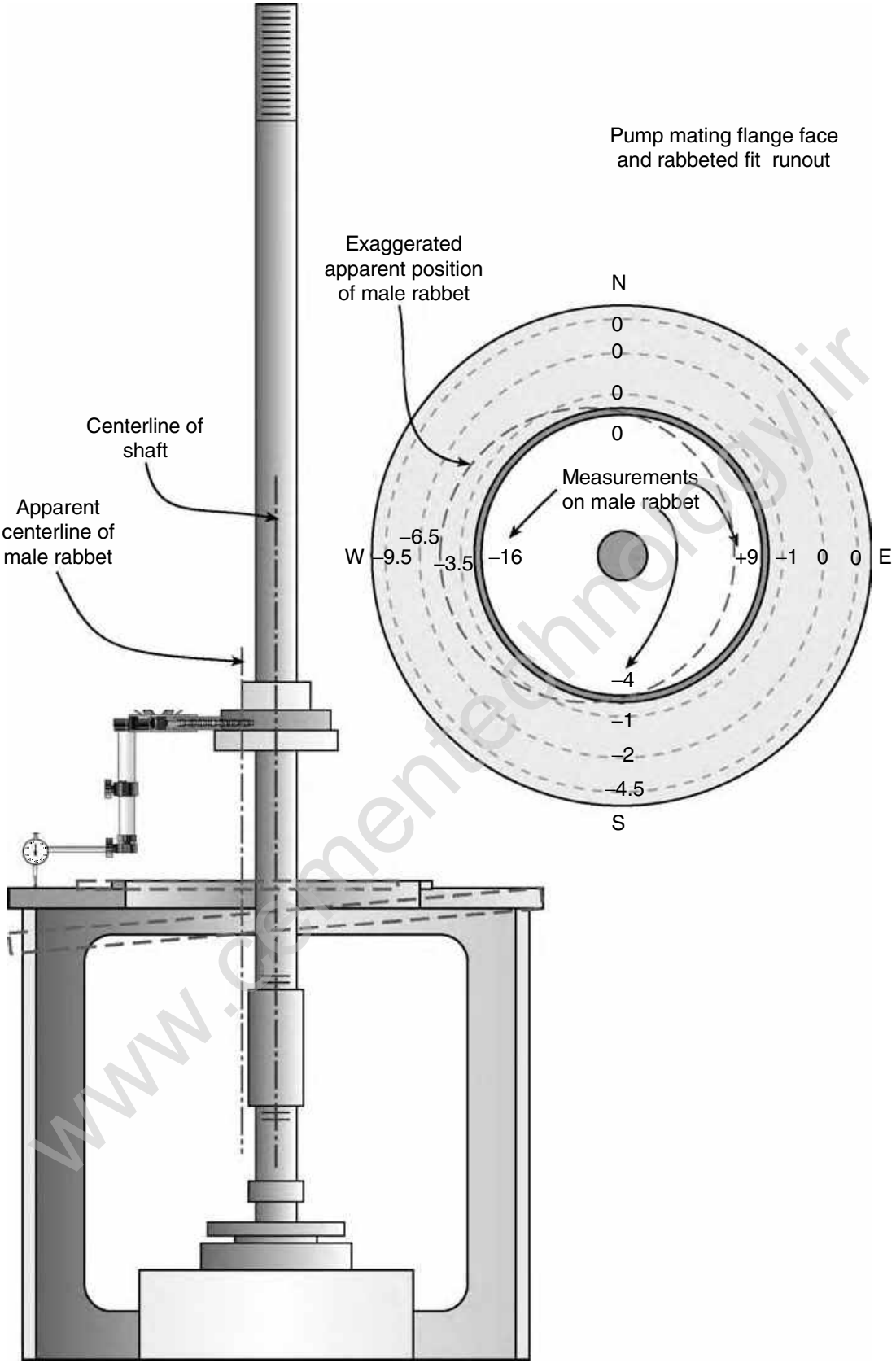


FIGURE 21.45 Concentricity and perpendicularity measurements on pump.

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FIGURE 21.46 Belt-driven fan.

shroud at both the inboard and outboard ends as shown in Figure 21.49. Notice that the gap at the twelve o'clock position at the inboard end is 30 mils and at the six o'clock position the gap is 270 mils. By adding these two gaps together, the total gap is 300 mils. For the fan wheel to be centered in the up and down direction, there should be a gap of 150 mils at both the twelve- and six o'clock positions. Since the gap is greater at the six o'clock position (270 mils), the housing is 120 mils low at that point with respect to the fan shaft and wheel. Likewise the outboard end is 40 mils too low.



FIGURE 21.47 Direct-driven fan.

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FIGURE 21.48 Aligning a direct-driven fan.

Figure 21.50 shows the side view of the motor shaft, the fan shaft, and the centerline of the bore of the fan housing. Assuming that there are no shims under any of the motor, fan bearings, or fan housing bolts, the overlay line could be projected to pivot at the outboard end of the motor and the outboard bearing of the fan and shims can be added under the inboard end of the motor, the inboard bearing of the fan, and the inboard and outboard foot bolts of the fan housing as shown in Figure 21.50. A similar alignment model can be generated in top view showing the lateral positions of the motor shaft, the fan shaft, and the centerline of the bore of the fan housing to achieve correct lateral alignment. Once again, the alignment modeling method gives us the opportunity to add as many pieces of information about our drive system as necessary to enable us to align all the components regardless of its complexity.

Typical OL2R movement range of blowers and fans:
Vertical movement: 0 to 80+ mils upward typically asymmetrical (i.e., inboard and outboard ends do not move up the same amount)
Lateral (*sideways*) *movement*: 0 to 20+ mils
Axial movement: 0 to 50+ mils

21.1.7.1 Additional Information on Horizontally Mounted Blowers and Fans

Moderate to excessive off-line soft foot conditions have been experienced on virtually every type of fan and blower regardless of frame construction design. Frequently, the fan bearings are bolted to the fan frame or to a pedestal, which may also have a soft foot condition. Bear in mind that the axial movement amounts mentioned above are for the casing or housing. The shaft may expand more than that and may influence how you should set the off-line shaft end to shaft end distances. For belt-driven fans, excessive vibration is often traceable to excessive face or rim runout in sheaves. Slow increases of vibration on fans often occur due to a dirt buildup on the fan wheel or uneven erosion or corrosion of the fan wheel. Since access to the fan wheel is usually accessible in the field, balancing of fans is frequently done *in situ*. Make sure the alignment and runout are acceptable and that any dirt buildup on the fan wheel has been removed before balancing. Balancing is the last thing I do. There is nothing more frustrating than attempting to "balance out" an alignment or runout problem.

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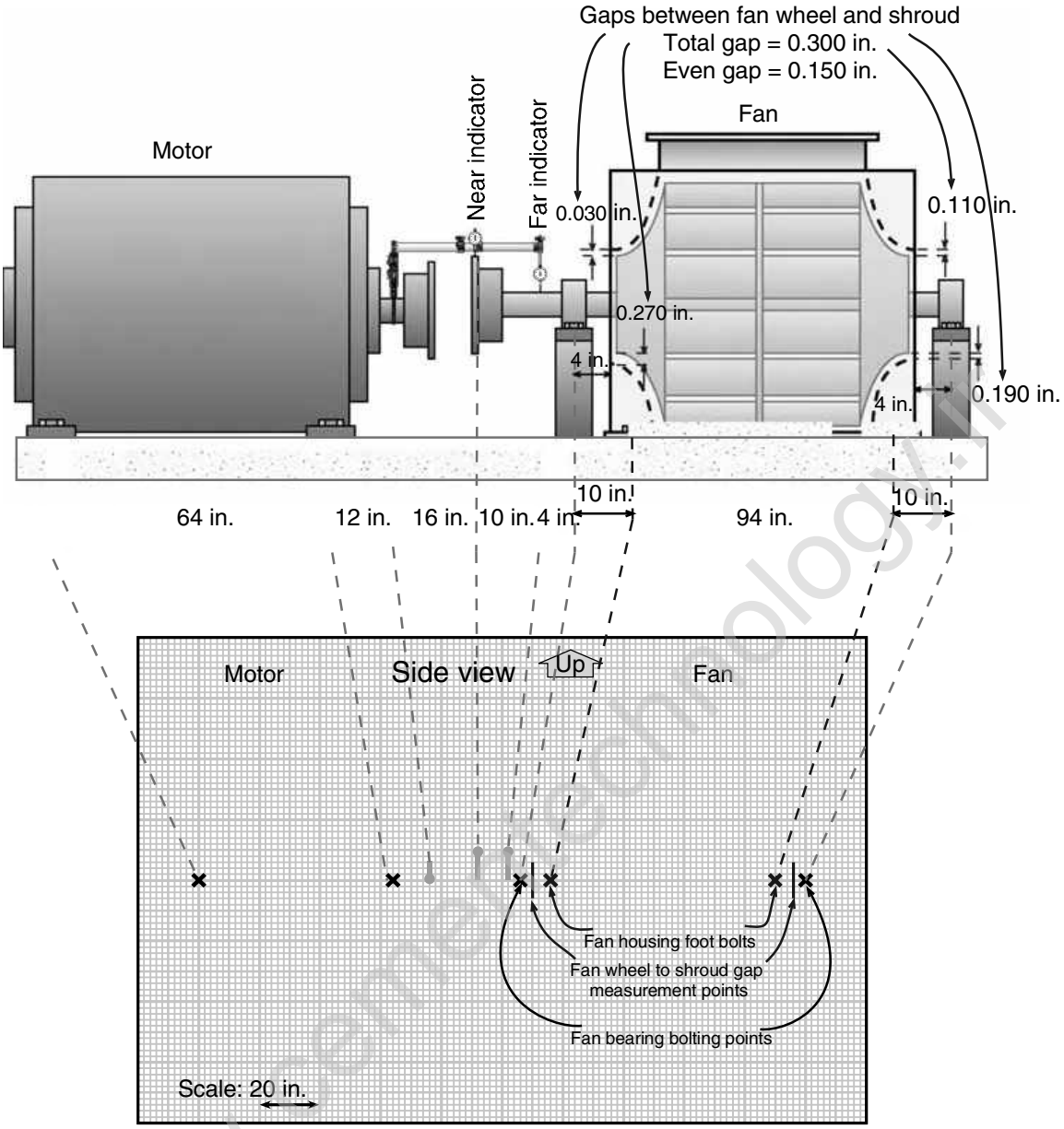


FIGURE 21.49 Motor-driven fan where fan shaft is supported on separate pedestals.

21.1.8 Compressors

The variation in compressor design is as diverse as pumps and fans. Figure 21.51 shows a small biogas compressor. Figure 21.52 shows a multistage horizontally split compressor with the upper casing removed. Figure 21.53 shows a multistage bullgear-driven compressor. Figure 21.54 shows axial flow compressors. Figure 21.55 shows a V-shaped two-stage reciprocating compressor. Figure 21.56 shows a chiller compressor. In the refining and chemical industries, it is not uncommon for several compressors to be driven by a single steam or gas turbine.

When gases are compressed, heat is generated and thereby casing expansion typically occurs. On multistage compressors such as the ones shown in Figures 21.51 through

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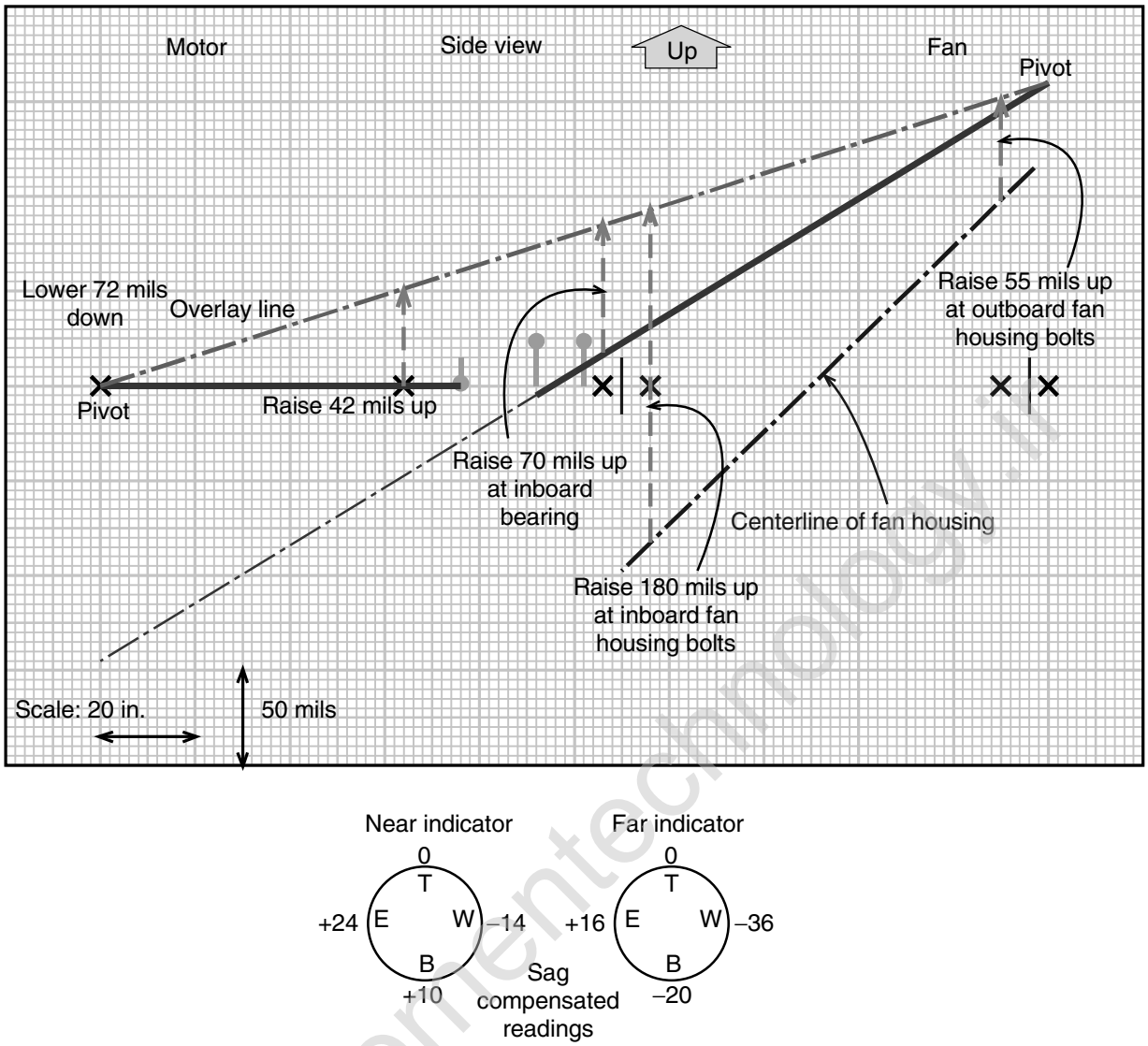


FIGURE 21.50 Side view of motor shaft, fan shaft, and fan housing bore.

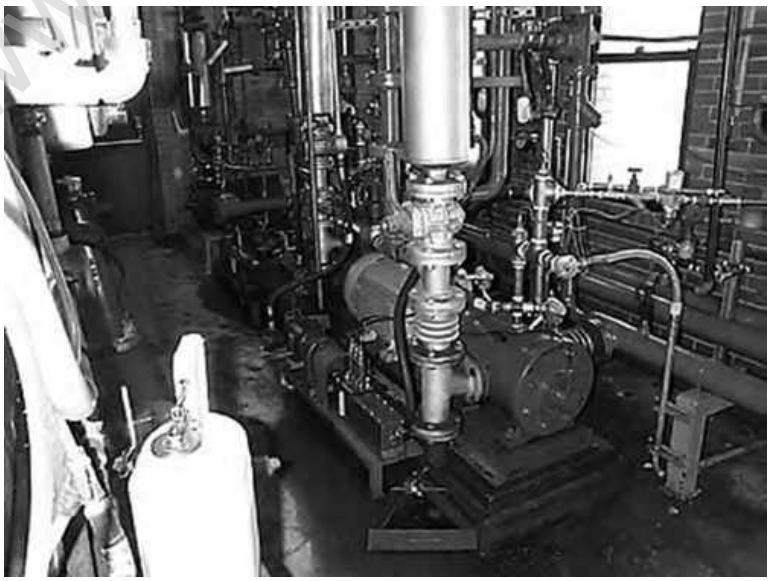


FIGURE 21.51 Small biogas compressor.

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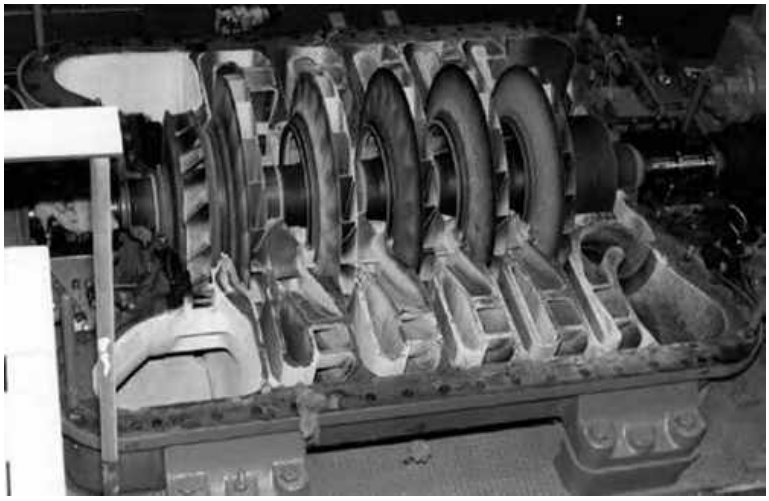


FIGURE 21.52 Multistage horizontally split compressor.

Figure 21.54, since the compressible fluid is entering the compressor at a much lower temperature than the discharged gas temperature, uneven OL2R movement frequently occurs. Depending on how the suction and discharge piping is attached to the compressor case, lateral (sideways) OL2R movement can occur as the attached piping expands or contracts. Refrigeration compressors can move downward during operation.

Typical OL2R movement range of compressors:

Vertical movement: 10+ mils downward to 80+ mils upward typically asymmetrical (i.e., inboard and outboard ends do not move up the same amount)

Lateral (*sideways*) *movement*: 0 to 30+ mils (usually much less than the vertical movement but can be greater than vertical movement in certain applications)

Axial movement: −10+ to 100+ mils

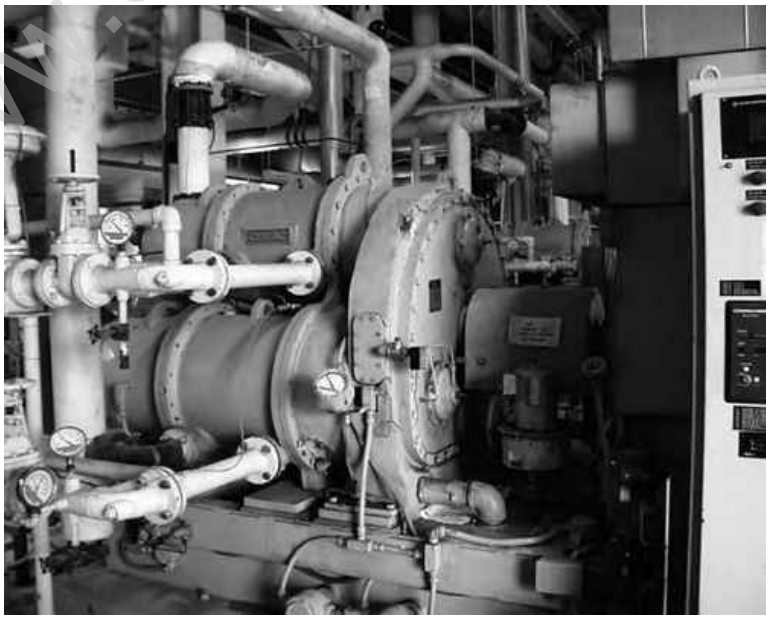


FIGURE 21.53 Multistage bullgear-driven compressor.

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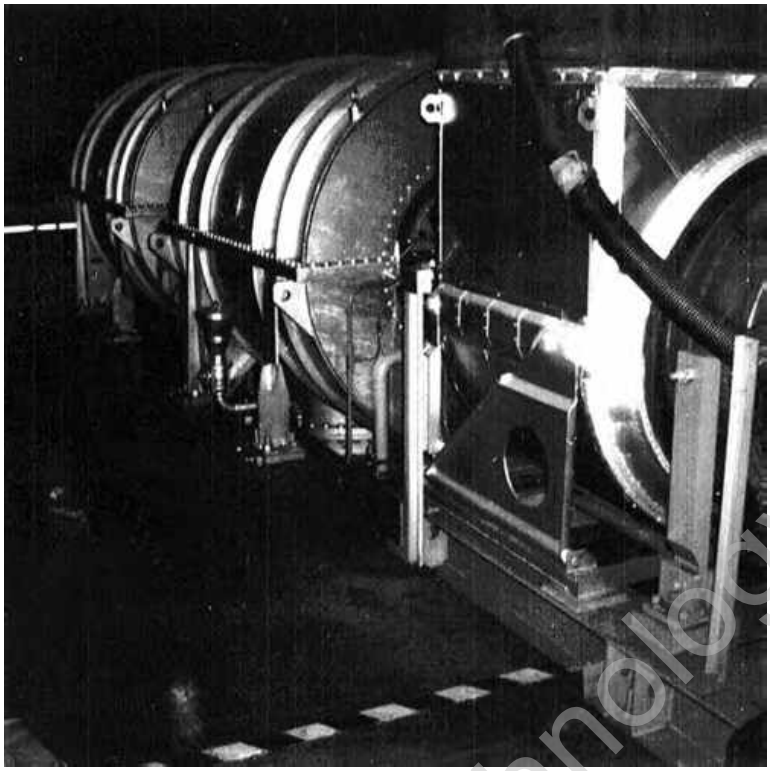


FIGURE 21.54 Axial flow compressors.

21.1.8.1 Additional Information on Compressors

Moderate to excessive off-line soft foot conditions have been experienced on virtually every type of compressor regardless of frame construction design. On medium to large compressors, the main lube oil pump is often driven off the compressor or driver shaft where misalignment has been observed between the pump and the shaft it is connected to despite the fact that the compressor and driver shaft may be aligned properly. During operation, if there is a backflow (i.e., stall or surge), the compressor shaft may make a rapid movement in the axial direction. If disk or diaphragm couplings are installed, damage could occur to the coupling due to this violent axial movement. Bear in mind that the axial movement amounts for OL2R conditions

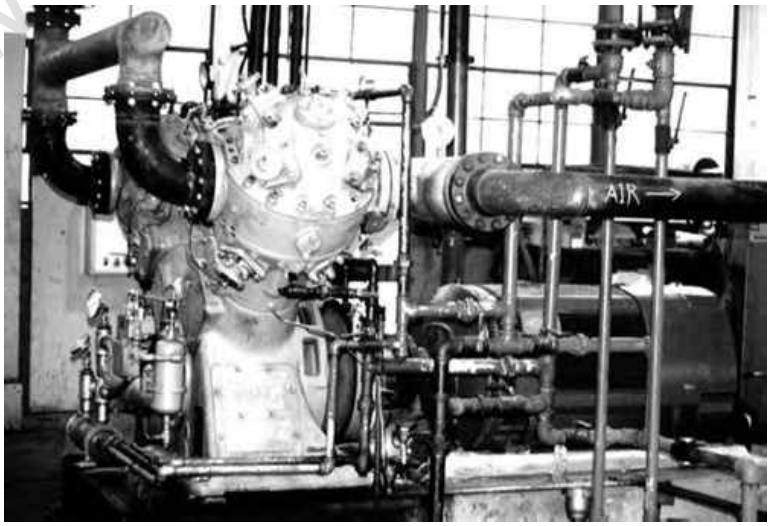


FIGURE 21.55 Two-stage reciprocating compressor.

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FIGURE 21.56 Chiller compressor.

mentioned above are for the casing or housing. The shaft may expand more than that and may influence how you should set the off-line shaft end to shaft end distances.

21.1.9 Horizontally Mounted Electric Generators

It is difficult to imagine what our world would be like without electricity. Minor power outages due to severe weather conditions can make life seem unbearable and power companies scramble to get us back on line as quickly as possible so we can get back to "normal." The vast majority of people have no idea what it takes to generate and deliver uninterrupted power to the grid. It is projected that California alone will increase their need for electricity at a staggering 1000 MW/year. Where will it come from? Mostly it will come from rotating electric generators driven by steam or gas turbines as shown in Figure 21.57.

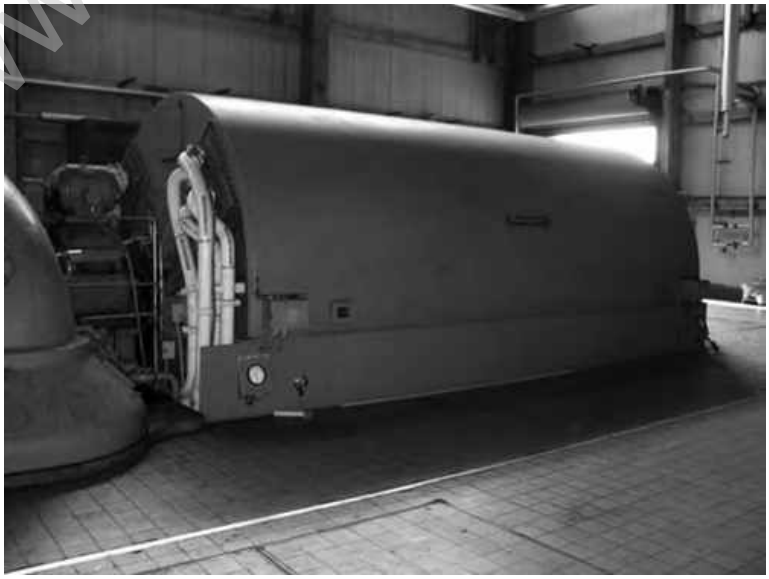


FIGURE 21.57 Typical alternating current electric generator.

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FIGURE 21.58 Typical motor–DC generator set.

Both alternating (AC) and direct (DC) current generators are used in a variety of industries. For base load AC electric generating stations, the generators are frequently driven by high-, intermediate-, or low-pressure steam turbines and are usually set on a catenary curve (refer to Figure 9.12). Smaller generators may be driven by a single steam turbine. Standby, AC generators are frequently driven by diesel engines at hospitals or industrial plant sites in the event of a power interruption. DC generators are typically driven by an electric motor. Several DC generators can be coupled together, referred to as motor–generator (MG) sets, as shown in Figure 21.58. Electric generators (similar to their cousins—the electric motors) are perhaps the best behaved types of rotating machinery as far as OL2R machinery movement is concerned. As with motors, the maximum recommended rotor to stator air gap eccentricity differential is $\pm 10\%$ of the total air gap.

Where generator armatures are supported in one bearing as found in MG sets, correcting any air gap clearance problems can also be taken into account when aligning the machinery casings. For a quick review, examine Figure 11.9 that covers the 16-point alignment method. As mentioned in Chapter 11, some drive systems are supported in three bearings. On MG sets, the motor armature is supported in two bearings, the generator armature is supported with one bearing, and the armatures are connected together with a spigot fit rigid coupling. A similar arrangement is often found between an AC generator and its exciter in the electric power industry. Remember what I have said about attempting to align an engaged rigid coupling? Don't try it.

Figure 21.59 shows a three bearing drive system. Rather than use the 16-point method which requires the rigid coupling to be disengaged just enough to provide a gap at the coupling flange faces while still engaged in the spigot or rabbet fit, another method is to provide a temporary support at the inboard end of the generator armature as shown in Figure 21.59. With the temporary support in place, the coupling can be completely disengaged and both armatures can be rotated. Now, the system you choose, the reverse indicator, face–rim, double radial, or a laser–detector system, could be used to capture the shaft-to-shaft measurements. Remember, the positions of the shafts are dictated by the positions of the bearing pedestals and the air gap clearances between the armatures and stator windings are dictated by the positions of the machine casings which in this case are independent of the bearing pedestals. In addition to the shaft-to-shaft alignment readings, air gap clearances need to be taken at both ends of both machines.

The alignment modeling technique for this drive system is similar to what was discussed in this chapter on fans as shown in Figure 21.49 and Figure 21.50. In this case, rather than finding the location of the bore of the fan housing, air gap measurements are taken between the armature and stator at both ends of both machines to determine the centerline of the bore

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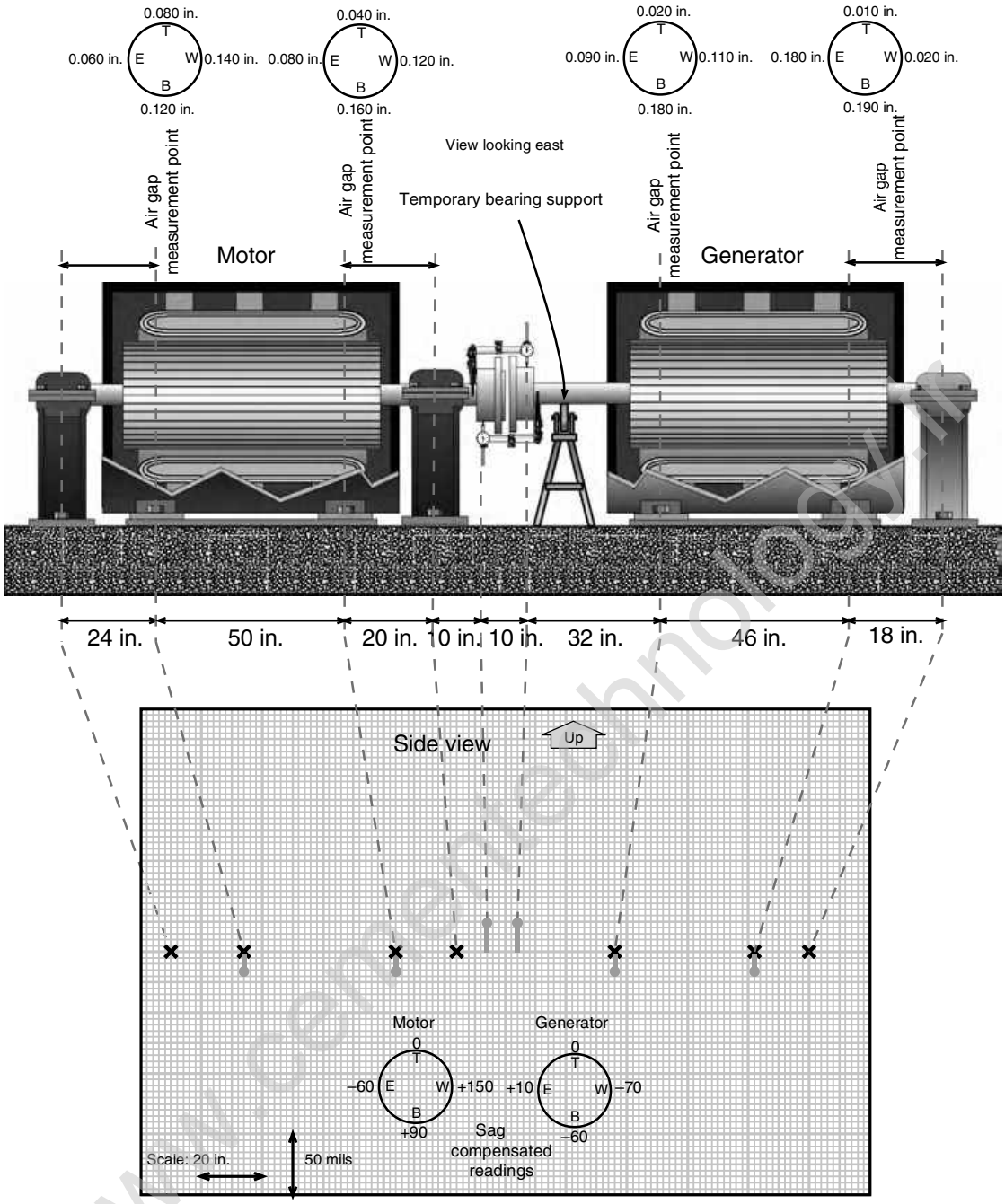


FIGURE 21.59 Three bearing drive system.

of the stator windings as shown in Figure 21.60. Assuming there are no shims under either stator and all three bearing pedestals, the overlay line in Figure 21.60 illustrates one possible alignment solution. Again, the alignment model can be used not only to align the shafts, but also to center the armature in the stator bore. Could you generate the top view alignment model from this information?

Typical OL2R movement range of horizontally mounted electric generators:

Vertical movement: 1 to 5 mils upward (5 to 200 hp); 3 to 30+ mils upward (200+ hp), typically symmetrical (i.e., inboard and outboard ends move up about the same amount)
Lateral (*sideways*) *movement*: 0 to 4 mils (usually much less than any vertical movement)
Axial movement: 5 to 40+ mils

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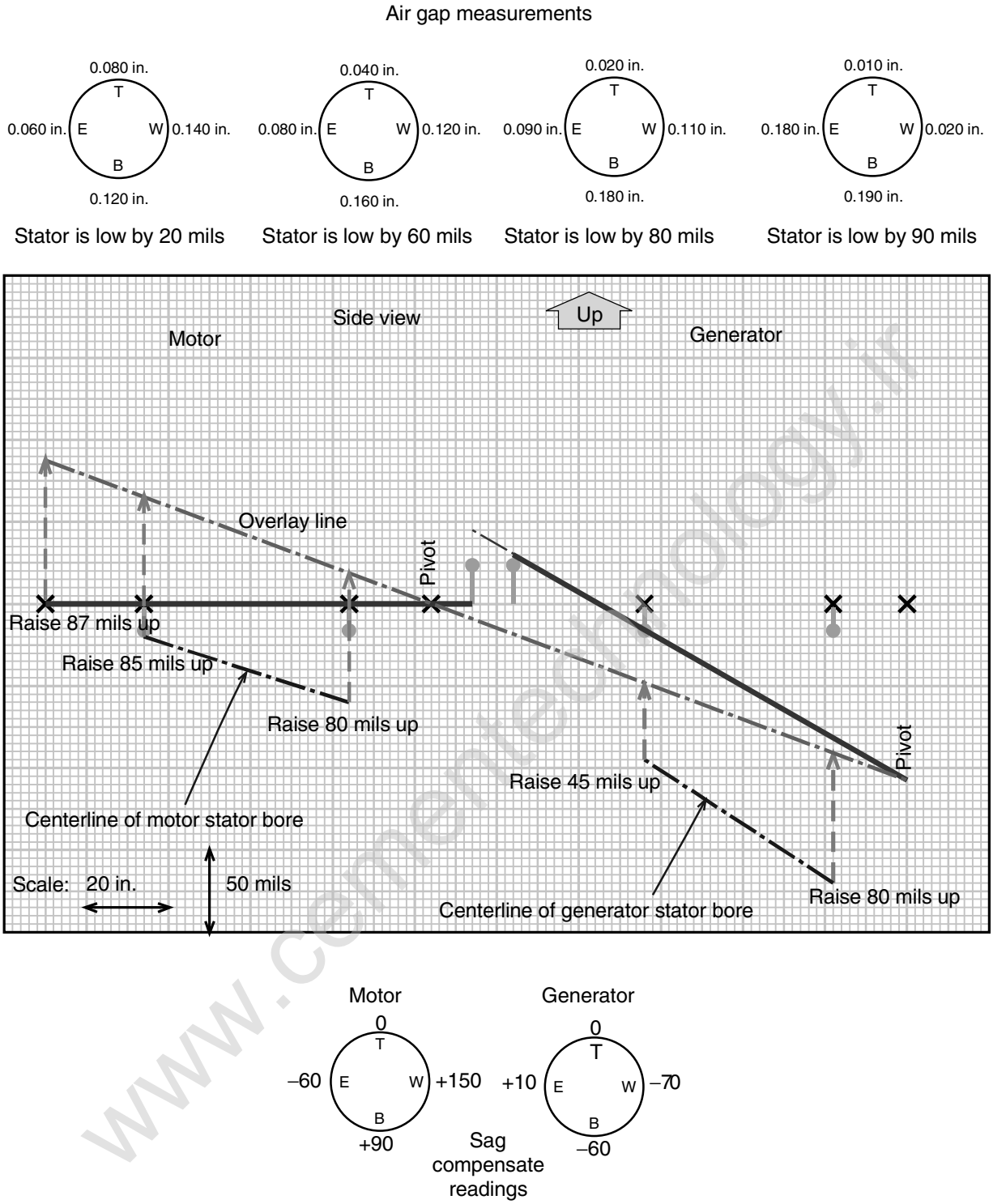


FIGURE 21.60 Side view alignment model of the three bearing drive system shown in Figure 21.59.

21.1.9.1 Additional Information on Electric Generators

Observed soft foot conditions on generators are the same as for motors. Large generators are often held down with several bolts, necessitating soft foot corrections to be made at every bolting point.

When aligning medium to large generators, because the rotors weigh a significant amount, it is often extremely difficult to rotate them by hand. If you decide to use an overhead crane or

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other similar device to aid in rotation, insure that you have made the desired rotation of the shaft to measure the side or bottom shaft positions, so that you remove the tension on the crane or chain fall before taking your measurement. Any excessive tension on the lifting and rotational device may elastically bend the shafts causing erroneous readings.

As mentioned in Chapter 9 and illustrated in Figure 21.61, metal is elastic, shafts are made out of metal, long metal cylinders supported at both ends will have a tendency to sag in between the supports. When connecting two shafts together with a rigid coupling, effectively you are assembling one continuous shaft supported at four points with the capability of disconnecting them between the middle two bearings. The ideal situation is to insure that each of the four bearings carries 25% of the entire load (assuming that the rotors are axially symmetric). Since the shafts are not infinitely stiff as shown in the top drawing in Figure 21.61 then to achieve even loading, it would be most desirable to match the curved centerline of rotation to the natural curve that occurs on the elastically bending shafts as shown in the second from the bottom drawing. However, in many instances, I see people aligning rigid couplings with a larger gap at the bottom than at the top as shown in the bottom drawing in Figure 21.61. Would this not have a tendency to strain the shafts, slightly unload the inboard

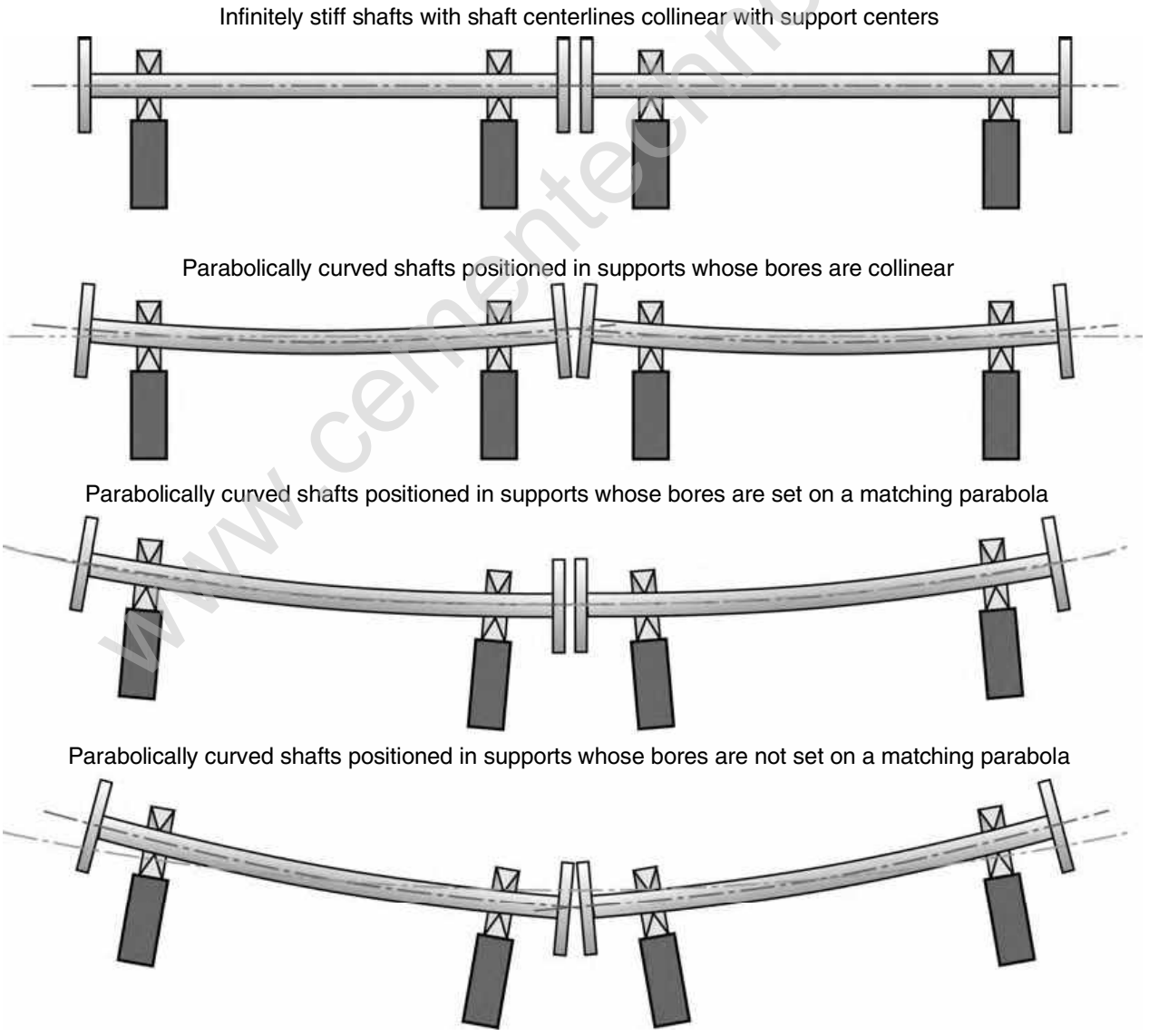


FIGURE 21.61 Which alignment condition is correct?

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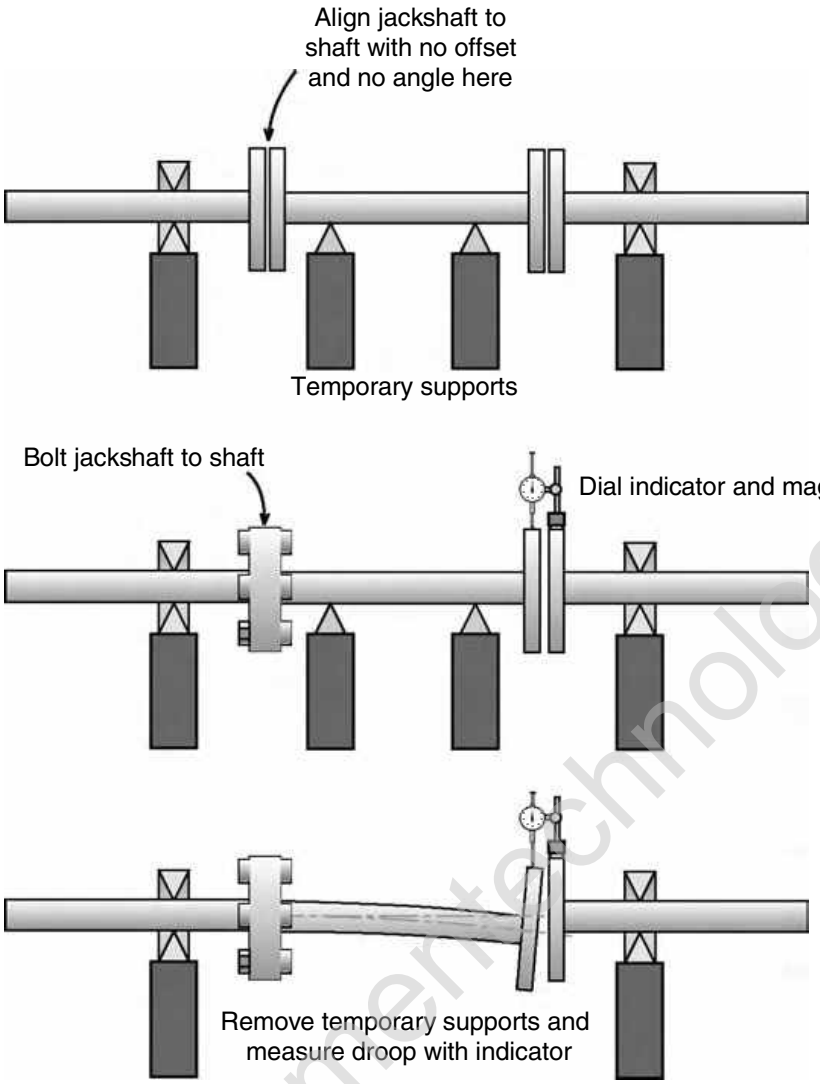


FIGURE 21.62 Measuring droop in a jackshaft.

bearings, and slightly overload the outboard bearings? Of course, the best way to determine the load on a bearing is to install strain gauges in each bearing right? Where pray tell has this been done?

On some larger turbine generator drive systems, there may be a long spool piece (often referred to as a jackshaft) connecting two of the turbines together. One practice is to remove a set of coupling bolts at one turbine only and perform a 20-point alignment check from the jackshaft to the turbine. Bear in mind that the overhung weight of the jackshaft will cause the disengaged end to "droop" a certain amount. The amount of droop is often calculated and rarely measured. Figure 21.62 shows a method to actually measure cantilevered droop on a jackshaft to verify the calculated droop.

21.1.10 Vertically Mounted Electric Generators

Vertically oriented electric generators are principally found at hydroelectric generating stations. The generator is mounted to a floor or building structure and a rigid coupling connects a water turbine shaft to the lower end of the generator armature as shown in Figure 21.63. The upper bearing of the generator is both a radial and thrust bearing that supports the weight of the armature and the water turbine. A lower radial bearing is located above the rigid

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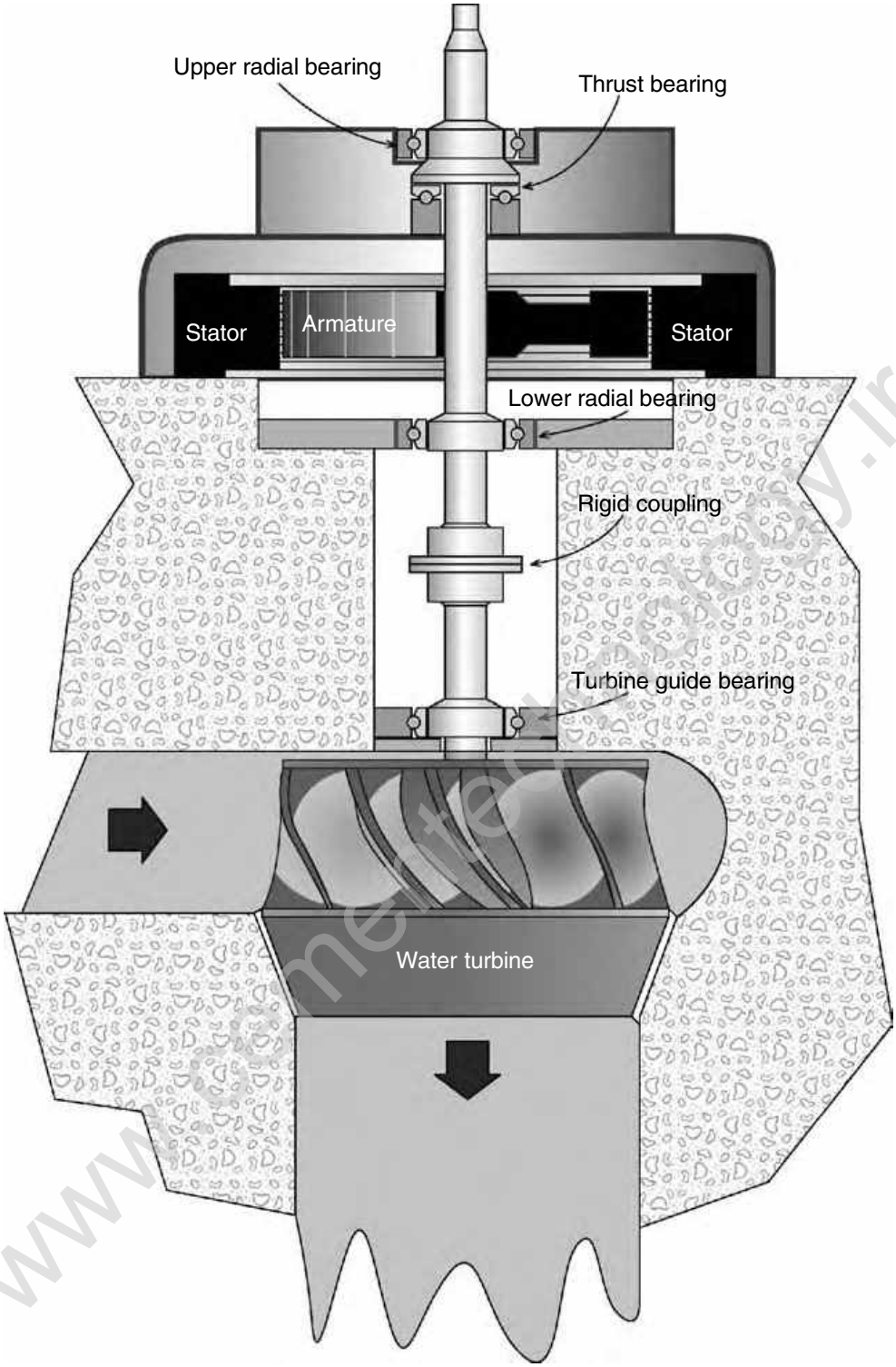


FIGURE 21.63 Hydroelectric generator drive system.

coupling. Another design has a single radial–thrust bearing assembly located below the generator armature. All designs incorporate one, if not two turbine guide bearings.

In an ideal world, the generator and turbine shaft would be perfectly plumb the thrust bearing evenly loaded on all of its pads and the centerline of rotation of the generator shaft would be concentric with the bore of all the radial bearings. In the real world, the thrust runner may not be perpendicular to the centerline of rotation of the generator shaft; the

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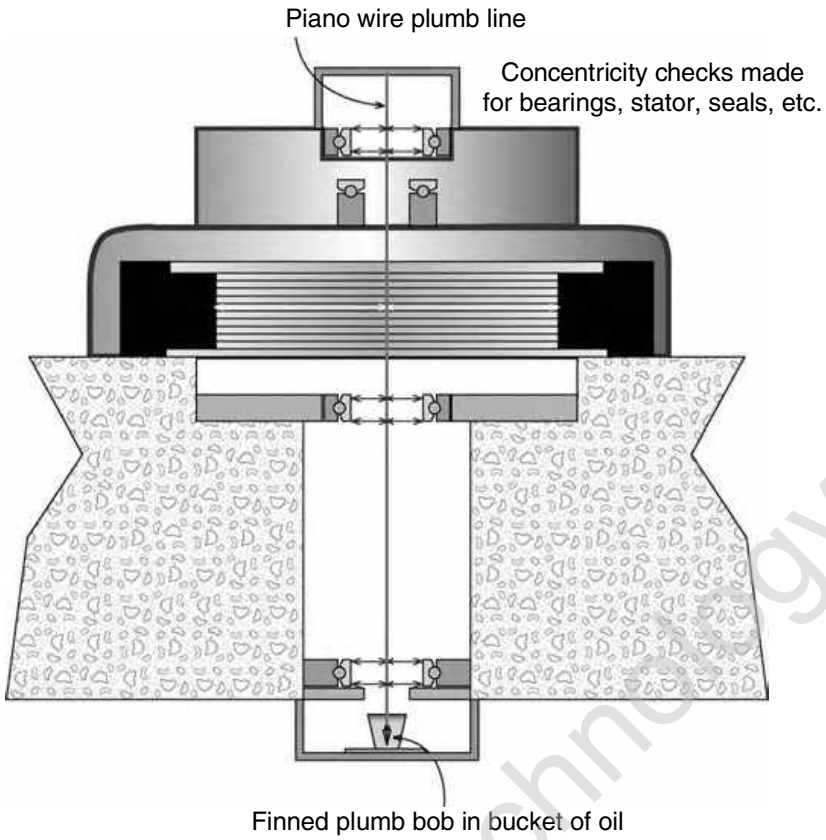


FIGURE 21.64 Plumb line positioned for concentricity checks.

generator and turbine shafts could have excessive radial runout; the coupling hub flange faces may not be perpendicular to the centerline of rotation of their shafts; the coupling bolt hole circle could be eccentric; and the radial bearings could be misaligned. There are four major issues to be dealt with: concentricity, perpendicularity, plumb, and straightness.

With the shafts removed, concentricity of the stationary components with respect to a vertical plumb line can be measured. Figure 21.64 shows a plumb line made out of stainless steel, nonmagnetic piano wire suspended from a metal frame above the upper bearing. At the other end of the piano wire is a finned plumb bob that is suspended in a bucket of viscous oil (e.g., SAE 90 weight) below the turbine guide bearing to dampen any movement of the wire if it is touched. The bucket should be sitting on a nonconductive frame (e.g., wood).

At the upper bearing, the plumb wire is translated in the north, south, east, and west direction so that it is concentric with the top of the upper bearing. To center the wire, an electric inside micrometer is used to measure the distance from the bore of the bearing to the suspended wire. The electric micrometer consists of an inside micrometer, headphones, and a battery as shown in Figure 21.65. One end of the micrometer is anchored against the bore of the bearing. The micrometer drum is then turned slowly to increase the length of the micrometer assembly. When the tip of the micrometer touches the wire, current begins to flow that can be heard in the headphones as a "click." If the electric micrometer is continually touching the wire, a "hissing" sound will be heard. If you know that the inside micrometer is touching the bearing surface and the wire and no noise is heard, you may not have electrical continuity from the frame to which the piano wire is attached to the bearing bore. To verify continuity, use a volt–ohm-meter set to measure resistance. Touch

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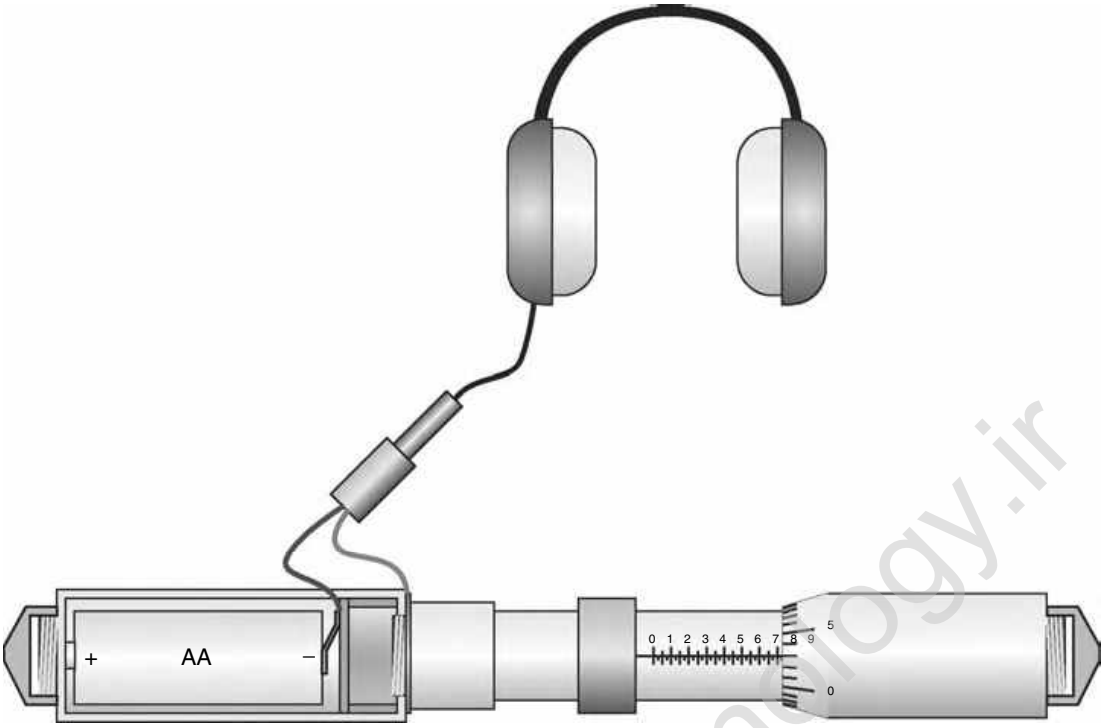


FIGURE 21.65 An electric inside micrometer.

one lead to the piano wire and the other lead to the bore of the bearing. If you have continuity (i.e., the capacity for electric current to flow), the resistance will be low (e.g., less than 10 ohms). If the resistance is high or at infinity, you may have to run an insulated wire from the piano wire frame to the bearing (and all other measurement points) to complete the circuit.

To insure that the micrometer is at a perpendicular angle to the piano wire, it is adjusted until the inside micrometer is at its shortest length to barely contact the wire. The distance is then recorded. Three additional measurements are taken at 90° from the initial inside micrometer reading and recorded. If all four distances are the same, the wire is centered. If not, the position of the wire is adjusted from the upper support based on the mathematical average of the four inside micrometer readings. Once the wire is centered at the top of the upper bearing, measurements at any point along the length of the wire to a stationary object can then be measured.

Measurements are then taken at the lower part of the upper bearing, the stator windings, the upper and lower points on the lower bearing, the upper and lower points on the turbine guide bearing, and any other critical clearances along the length of the rotating assembly. Many radial bearings incorporate wedge-shaped segments, so care must be taken when measuring distances at the bearings to insure that the inside micrometer tip is at the same point on each wedge. If the bearings have been removed, then the measurements are taken to the bearing-housing bore. The general rules of thumb for concentricity are as follows:

- *Bearings*: 20% of diametral bearing clearance
- *Stator*: 5% of design air gap (where stator air gap is $\pm 5\%$ of design air gap)
- *Seal ring*: 10% of diametral seal ring clearance

If for some reason one or more of the components are not concentric with a plumb centerline, then you will have to position the device to make it concentric. This could be the upper

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generator bearing, the lower generator bearing, the stator, the turbine guide bearing, the seals, the upper and lower seal or wear rings of the turbine, etc. An alignment model can be constructed to show the position of each component (bearings, stator, and seals). You must then determine which of the components are not concentric, how far and which way they must be moved, and what you have to do to place them where they need to be. In some cases, that might be an easy task, in other cases, someplace between a headache and a pure nightmare. Hopefully the three bearings and all the seals are concentric and only the stator needs to be repositioned. Once we know that the stationary components are concentric to a plumb line that represents the desired centerline of rotation, we can now begin our investigation of the rotating assembly itself. There is another way to determine if the stationary components are concentric with the shafts installed.

Four issues concerning the condition of the rotating assembly need to be measured and corrected if they are out of tolerance.

1. Is the thrust runner perpendicular to the centerline of the generator shaft?
2. Are both coupling hub flange faces perpendicular to their respective shafts?
3. Are the shafts straight?
4. Is the centerline of rotation out of plumb?

The first three items effectively deal with runout conditions discussed at length in Chapter 5 with ''preliminary'' as the key word here. Figure 21.66 shows what would happen if the

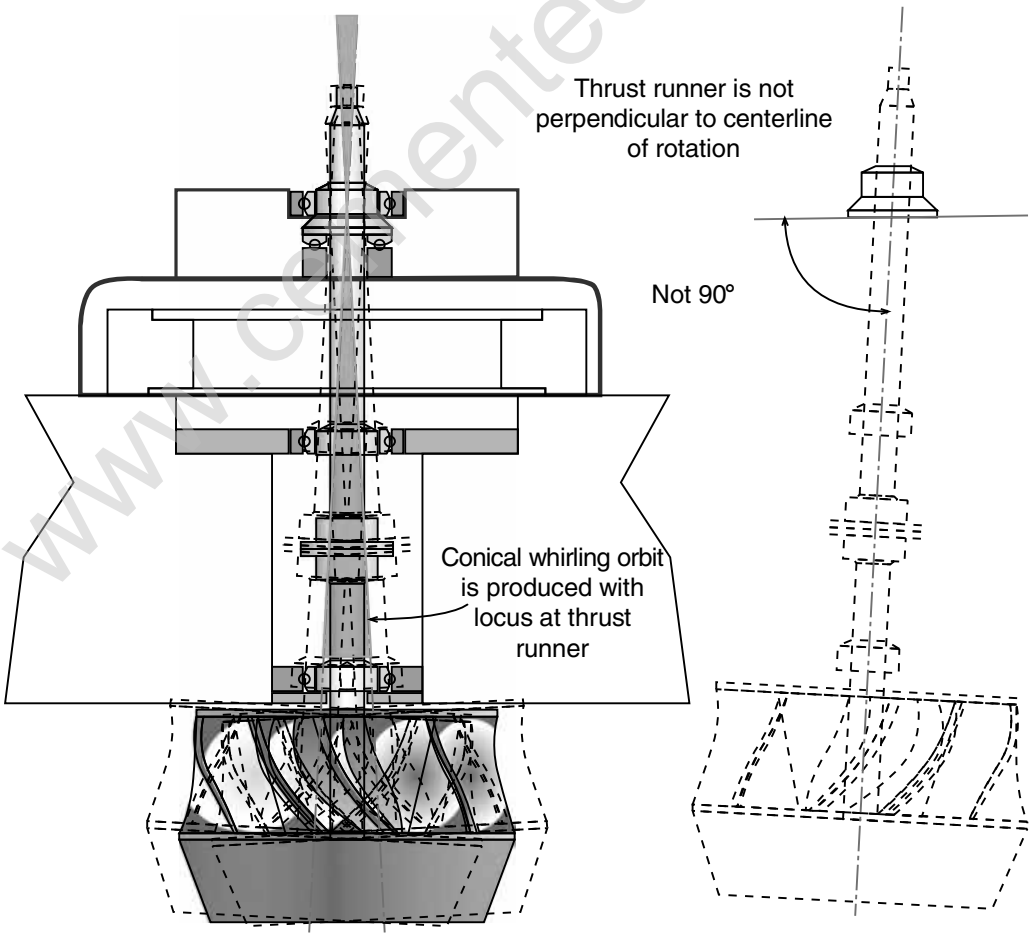


FIGURE 21.66 **(See color insert following page 322.)** A nonperpendicular thrust runner face will force the rotating assembly to orbit.

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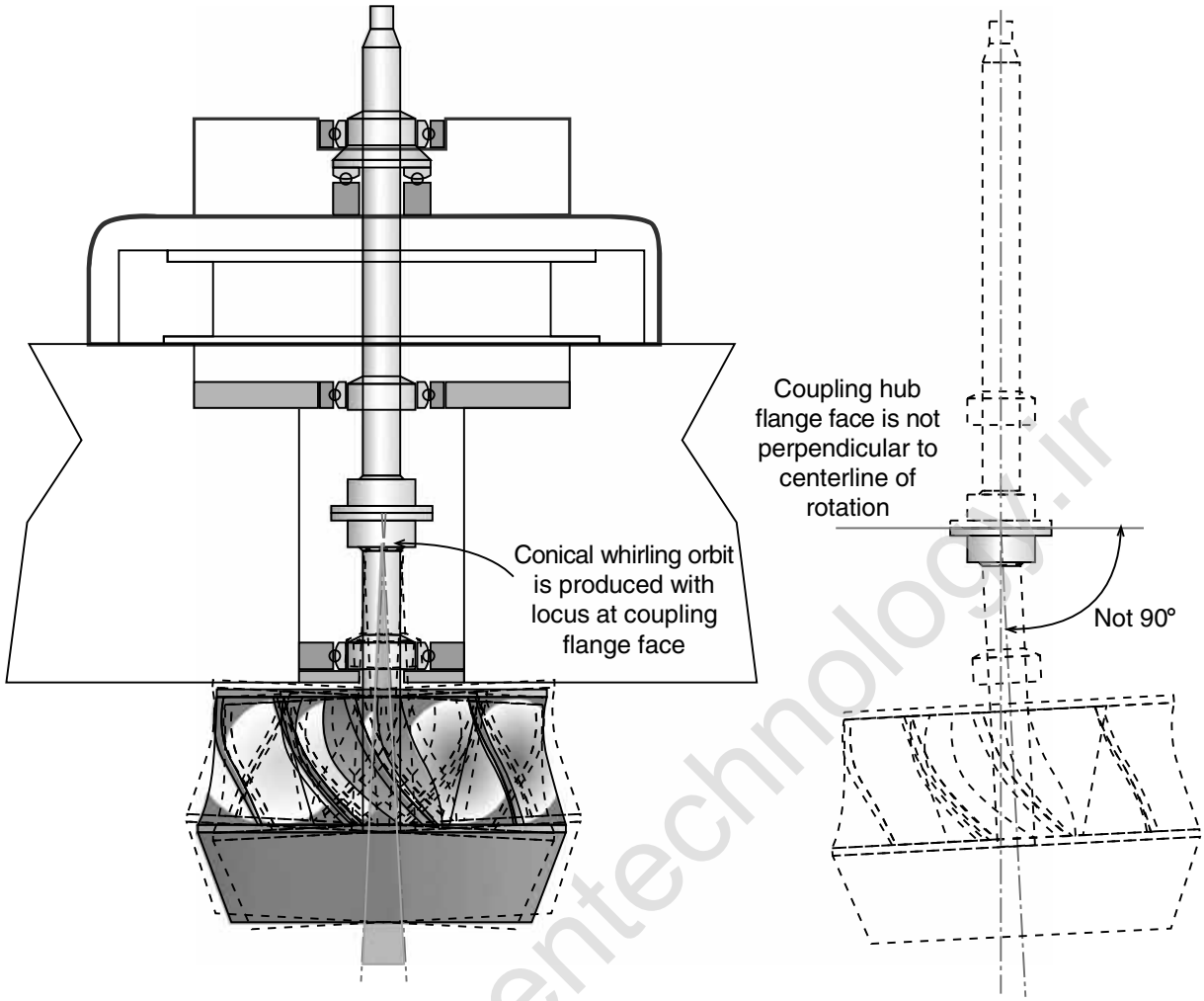


FIGURE 21.67 A nonperpendicular coupling hub flange face will force the turbine shaft to orbit.

thrust runner is not perpendicular to the shaft with the centerline of rotation plumb. Figure 21.67 shows what would happen if either or both of the coupling hub flange faces are not perpendicular to their shafts with the generator shaft plumb. One of our objectives is to determine if the centerline of rotation of the entire rotating assembly (i.e., the coupled generator and turbine shafts) is plumb. In other words, if the thrust bearing is not level and the thrust runner is not perpendicular to the shaft, the rotating assembly would produce a spinning cone and its centerline of rotation would be pitched at an angle as shown in Figure 21.68.

If we happen to deal with the condition shown in Figure 21.68, the shafts have stopped rotating the high spot of the runout, which will stop at some random angular position (e.g., southeast). If we then attempt to measure the plumb of the shafts with north, south, east, and west as our reference points, how do we know how much of what we are measuring at those 90° positions is due to an out of plumb condition and how much is due to a runout condition? We do not know. We first have to establish how much runout is present before we attempt to measure any type of alignment condition.

It makes no sense to attempt to align machinery with bent shafts, problems with the coupling, and problems with one of the components on the rotors (e.g., a thrust runner). Why do you waste your time in aligning shafts that have problems with them? If you align the centerlines of rotation and then discover that there is an issue with one or more of the shafts, all of the effort you spent in aligning is waste of time. Why? Because you may have to replace

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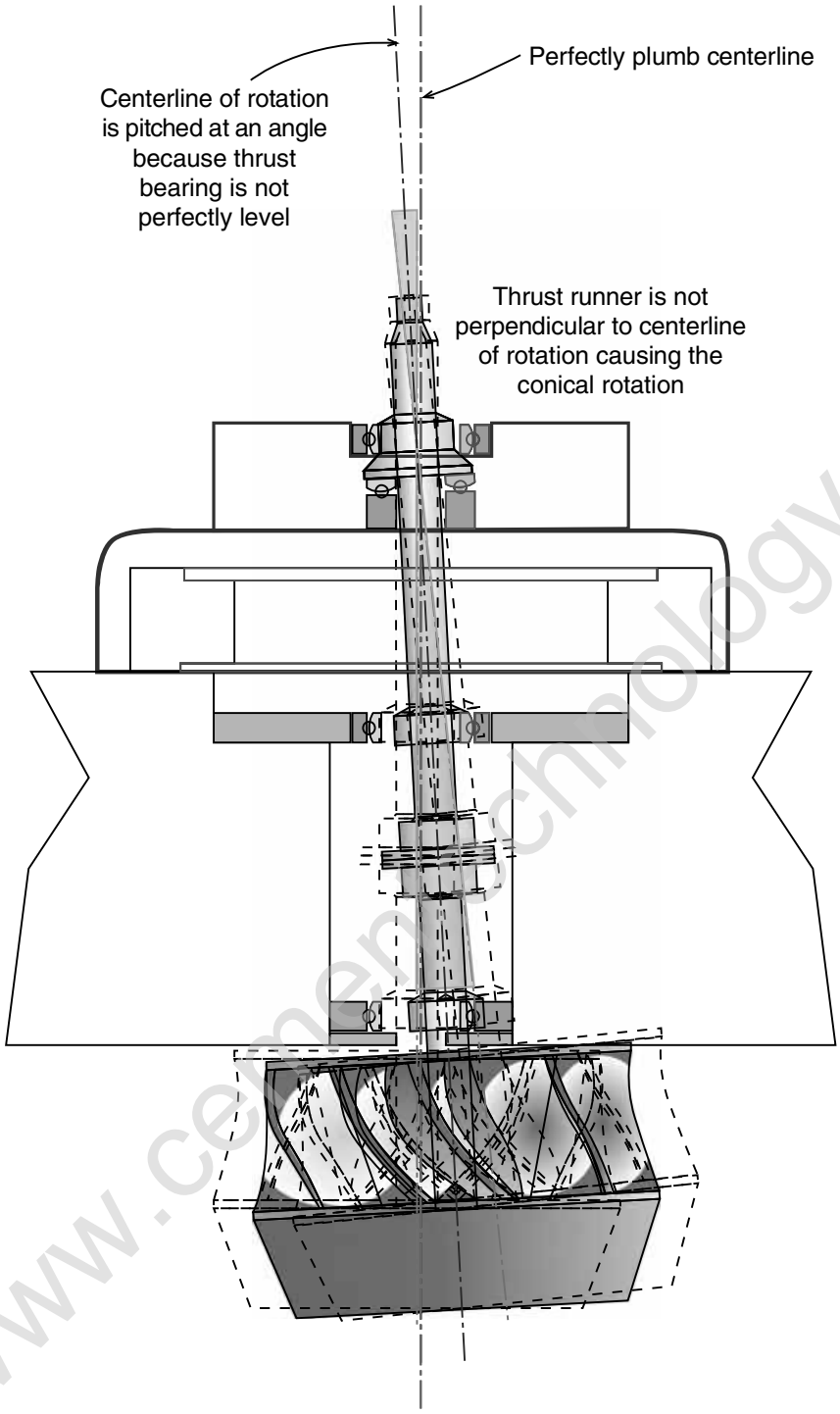


FIGURE 21.68 Combination of out of plumb centerline of rotation combined with a thrust runner that is not perpendicular to its centerline of rotation.

a shaft, a coupling hub, a thrust runner, or at least machine with the component to correct those problems. To do that, you may very well have to disassemble the machines. Once the problems have been corrected within acceptable criteria, you are back aligning again. Save yourself some grief, discover and correct these problems before you ever start aligning. If we check the runout first, then we can not only ascertain if the runout is acceptable, we also know where the high spots are and can place them at an angular position of our choosing (e.g., north). Then we know how much of what we are measuring is due to runout and how

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much is due to an out of plumb condition. If you are not clear on this, you may want to go back and study Figure 21.20 and Figure 21.27.

The rotor in Figure 21.68 will have a propensity to want to run on a pure vertical plumb line and the guide bearings would force the shafts to their respective positions but the radial forces on those bearings would be undesirable and the thrust runner would only be contacting a few of the thrust shoes on the "high" side of the thrust bearing greatly diminishing its life span. The whirling cone would produce excessive vibration particularly at the lower generator bearing and the turbine guide bearing. Although the runout and perpendicularity problems are exaggerated for clarity as shown in Figure 21.66 through Figure 21.68, you can see that if the guide bearings were in place, the capacity to accurately measure the position or runout of the rotor would not be possible. To accurately measure the runout and plumb condition of the rotor, the upper and lower generator bearings and the turbine guide bearing should be removed. The entire rotor assembly should only be sitting on the thrust bearing and somewhat centered. The rotor should be free to swing back and forth freely with just a slight amount of force.

The perpendicularity condition is typically measured by the amount of runout near the turbine guide bearing rather than as a face runout condition on the thrust runner itself or on the coupling hub flange faces. The goal is to level the thrust bearing so the center of the cone of runout is within acceptable limits and still stay within an acceptable cone orbit diameter. The general rules of thumb are as follows:

- Radial runout not to exceed 0.003 in. (3 mils) along the entire length of both shafts.
- Center of the cone of runout (plumb line of runout) is 0.000025 in. times the length of the shaft measured from the highest to lowest plumb reading. For example, if the distance from the highest to lowest plumb reading is 100 in., then the maximum recommended out of plumb deviation of the centerline of rotation would be 0.0025 in.
- Cone orbit diameter (also known as static runout) is 0.002 in. times the length of the shaft from the thrust bearing to the runout measured above the turbine guide bearing divided by the diameter of the thrust runner. For example, if the length of the shaft from the thrust bearing to the runout measured above the turbine guide bearing is 300 in. and the diameter of the thrust runner is 30 in., then the maximum static runout should not exceed 20 mils (0.020 in.).

Figure 21.69 shows the setup for measuring the runout using dial indicators. Two indicators, 90° apart should be set up at the upper generator bearing to observe the position of the rotor as the assembly is rotated to determine if the shaft is sliding laterally on the thrust bearing surface during rotation. Another indicator is placed just above the turbine guide bearing to measure the runout at that point. Additional indicators could (and probably should) be placed at various points along the length of the shafts to aid in discovering possible problems such as a bent shaft, cocked coupling hub flange faces, eccentric couplings, and a cocked thrust runner. At least two rotations should be made to verify the consistency of the amount and angular position of the high spots. You may want to initially set up just the indicator above the turbine guide bearing. If the runout condition there is excessive, then the additional indicators should be set up to determine the source of the runout condition.

One technique used to measure the plumbness of the rotating assembly is to install four plumb lines similar to the arrangement shown in Figure 21.71. The plumb lines are suspended from a metal frame below the lower generator bearing and spaced 90° apart. It is not necessary to position each plumb line at exactly the same distance from the shaft. What you will be measuring is the difference in distance from each plumb line to the shaft at several points along the length of that plumb wire. Again, an electric inside micrometer will be used to

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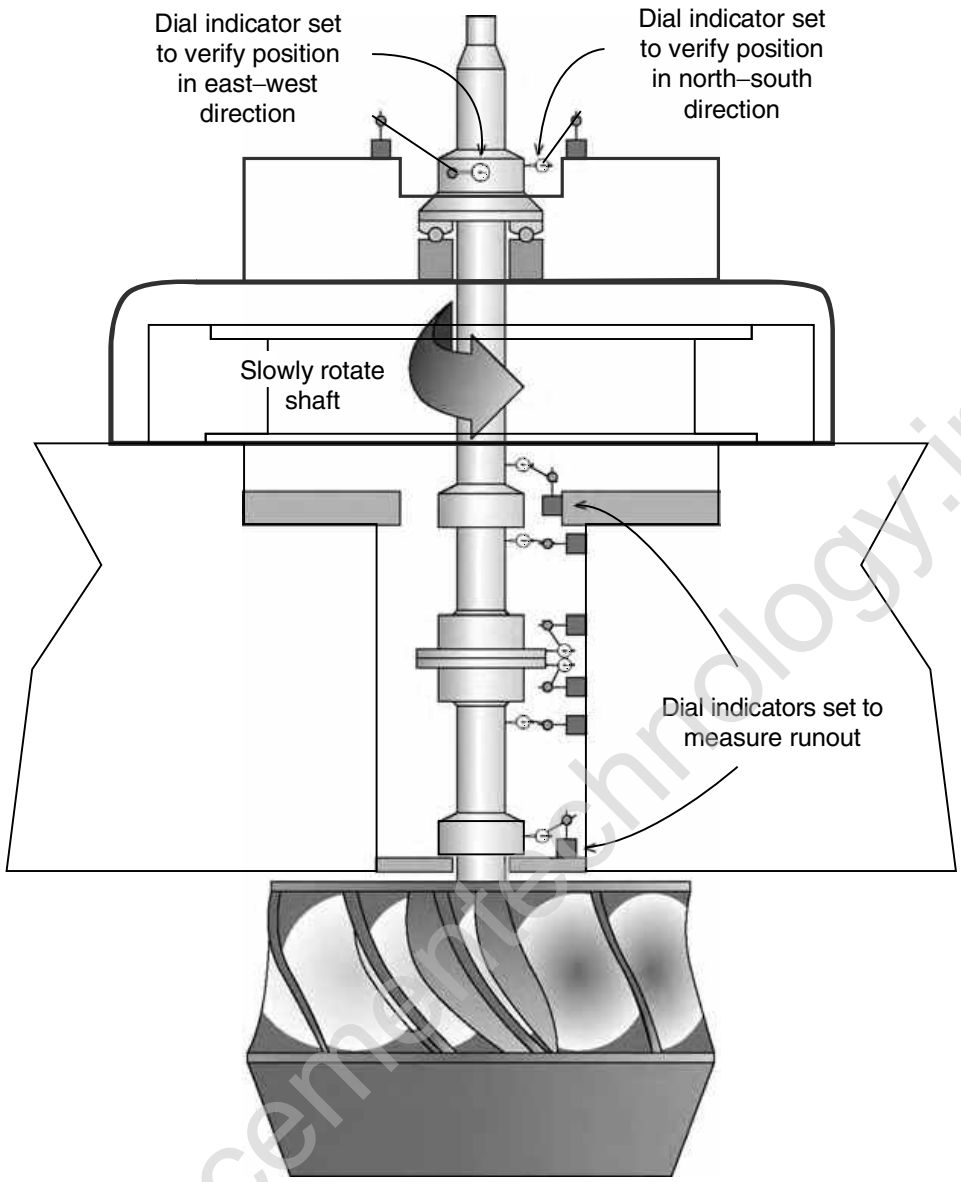


FIGURE 21.69 Runout checks.

measure those distances. To aid in positioning the micrometer against the shaft, a Y-shaped seat is attached to one end of the electric micrometer. Temporary bands are placed around the shaft at four or more points. The banding will provide a mechanical rest for the Y-shaped seat to rest against for stability and consistency. In addition to the measurements taken at the four plumb lines, gap measurements need to be taken at four points between the shaft and the bore of each bearing housing as well as the gaps at the upper and lower seal rings of the turbine.

Figure 21.72 shows an example of the dimensions needed to plot the plumb and clearance measurements. As we have done with all our alignment models, the dimensions in Figure 21.72 are scaled onto two pieces of graph paper. One model will show the position of the rotors in the north–south direction and the other in the east–west direction.

Figure 21.73 shows the information that needs to be gathered to determine the position of the rotor with respect to the bearings and wear rings. Figure 21.74 shows the information gathered for the distances from each plumb line to the outer surface of the shaft at each measurement location. As previously mentioned, any runout that exists in the rotor will affect the bearing and wear ring clearance measurements as well as the plumb line to rotor measurements with some caveats.

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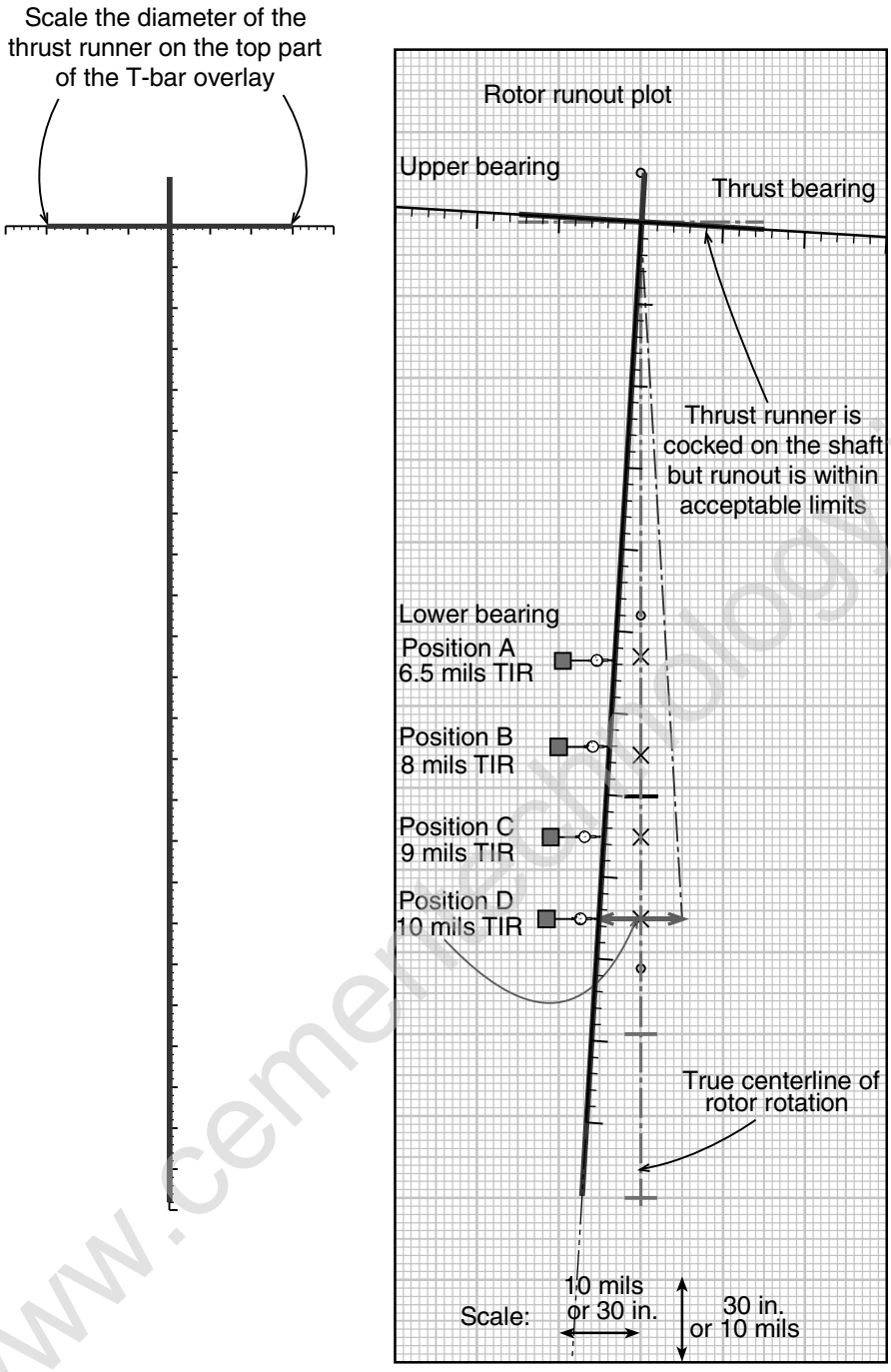


FIGURE 21.70 Sample runout plot.

If the rotor is in some random angular position, how much of what was measured from the plumb lines to the rotor is due to an out of plumb rotor position and how much is due to the runout in the rotor?

The most effective way to determine the actual out of plumb rotor position is to place the high spots on the rotor at a known angular position (e.g., in line with the north plumb line) before measuring the distances between the plumb lines and the shaft. Since we know exactly where the high spot is, we can then accurately determine the position of the centerline of rotation of the rotor.

For example, if we measured 10 mils of radial runout just above the turbine guide bearing as shown in Figure 21.70, which is perhaps within our acceptable maximum runout limit, and we have identified the angular position of the high spot at that point, then we know that the

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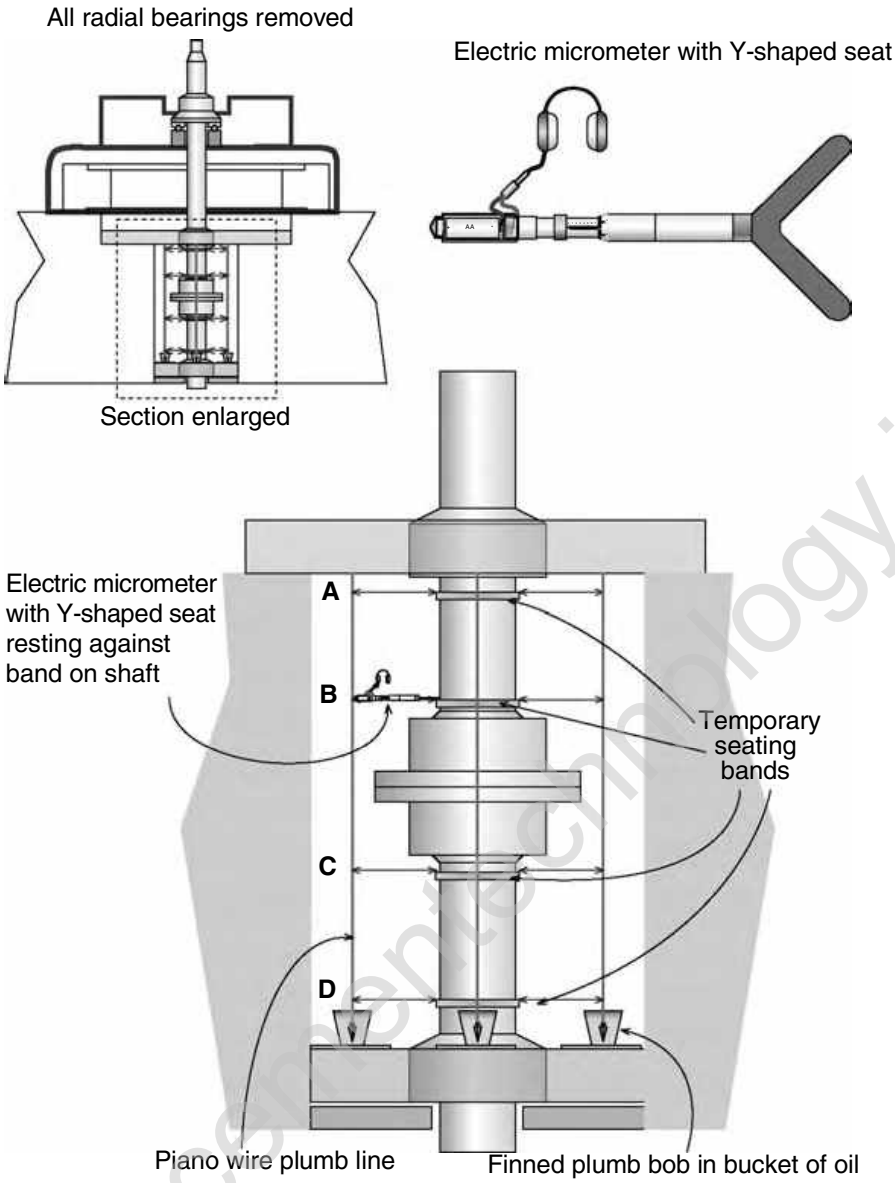


FIGURE 21.71 Checking the rotor position for plumbness using four plumb lines.

actual centerline of rotation of the shaft is 5 mils (half the total radial runout) toward the opposite side of the rotor (i.e., 180° from the high spot). If we placed the high spot at the same angular position as the north plumb line, then 5 mils of what we measured with the electric micrometer is due to the runout at that point on the rotor. If the runout that was taken at the turbine guide bearing is at the same elevation where we measured the shaft to plumb line at position D, we can then compensate for the runout in our alignment model which shows the north to south position of the rotor.

The angular pitch of the rotors shown in Figure 21.75 is due to the out of level condition of the thrust bearing. As illustrated, the south side of the thrust bearing must be raised 6.5 mils and the west side must be raised 5.5 mils. Bear in mind that once the adjustments are made to level the thrust bearing, another set of plumb line measurements must be taken to verify that the centerline of rotation is within acceptable limits.

Laser alignment systems have been developed to perform out of level measurements. Bear in mind that any system that only measures levelness will not be able to differentiate between an out of level condition and the rotor that is pitched due to a runout condition.

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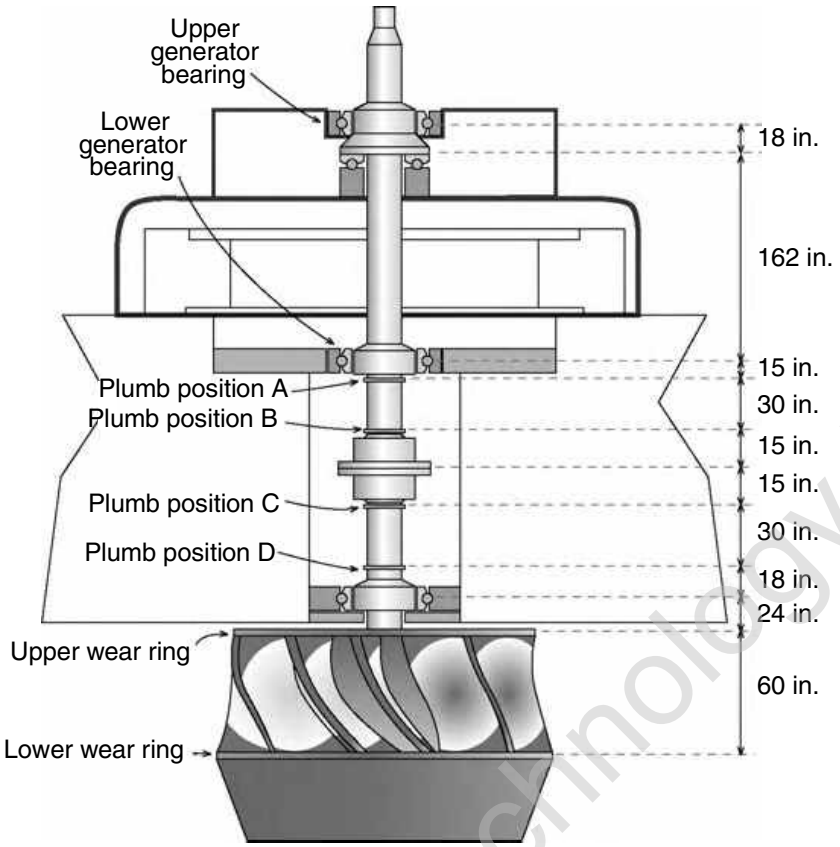


FIGURE 21.72 Dimensional information for modeling vertical turbine generator rotors.

Measurement location	Compass position	Measurement device	Field measurement	Zero on north side (north–south) + to the south – to the north	Zero on west side (west–east) + to the east – to the west	Ideal gaps
Upper generator bearing	North	Snap gauge and OD mike	1.25500			1.2500
	South		1.24500	0.0100		1.2500
	West		1.24625			1.2500
	East		1.25375		–0.0075	1.2500
Lower generator bearing	North	Snap gauge and OD mike	1.24850			1.2500
	South		1.25400	–0.0055		1.2500
	West		1.25125			1.2500
	East		1.24875		0.0025	1.2500
Turbine guide bearing	North	Snap gauge and OD mike	1.49100			1.5000
	South		1.50900	–0.0180		1.5000
	West		1.50525			1.5000
	East		1.49475		0.0105	1.5000
Upper wear ring	North	Feeler gauges and OD mike	0.01975			0.0300
	South		0.04025	–0.0205		0.0300
	West		0.03600			0.0300
	East		0.02400		0.0120	0.0300
Lower wear ring	North	Feeler gauges and OD mike	0.01675			0.0300
	South		0.04325	–0.0265		0.0300
	West		0.03825			0.0300
	East		0.02175		0.0165	0.0300

FIGURE 21.73 Bearing and wear ring clearance measurements.

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Plumb line ID	Measurement position	Measurement device	Field measurement	Zero at position A	Orientation of shaft
North plumb wire	Position A	Electric mike	0.5120	0.0000	
	Position B		0.5085	– 0.0035	+to the south
	Position C		0.5055	– 0.0065	–to the north
	Position D		0.5025	– 0.0095	
South plumb wire	Position A	Electric mike	0.6280	0.0000	
	Position B		0.6315	0.0035	+to the north
	Position C		0.6345	0.0065	–to the south
	Position D		0.6375	0.0095	
West plumb wire	Position A	Electric mike	0.7230	0.0000	
	Position B		0.7210	– 0.0020	+to the west
	Position C		0.7190	– 0.0040	–to the east
	Position D		0.7170	– 0.0060	
East plumb wire	Position A	Electric mike	0.3480	0.0000	
	Position B		0.3500	0.0020	+to the east
	Position C		0.3520	0.0040	–to the west
	Position D		0.3540	0.0060	

FIGURE 21.74 Plumb line to shaft distance measurements.

Typical OL2R movement range of vertically mounted electric generators:

Vertical movement: The generator may thermally expand due to heat generated in the windings in the vertical direction from 5 to 30+ mils but this will probably have very little effect on the performance of the machine.

Lateral (*sideways*) *movement*: Again, the generator may thermally expand due to heat generated in the windings in the lateral direction from 10 to 50+ mils but this will probably have very little effect on the performance of the machine since the expansion is usually symmetric.

Axial movement: There may be 5 to 20+ mils of thermal expansion of the generator shaft and there may be 5 to 20+ mils of thermal contraction in the turbine shaft due to the probability of lower ambient temperature of the water, which would have very little effect on the performance of the machine.

21.1.11 Speed Changing Devices—Gearboxes and Fluid Drives

Gearboxes come in a wide variety of designs and sizes and are used to precisely increase or decrease the shaft speed from one machine to another. Some examples of gearboxes are shown in Figure 21.76 through Figure 21.78.

Fluid drives (also known as hydraulic clutches) are designed to disengage a driven component or provide a variable rate of output speeds. An example of a fluid drive is shown in Figure 21.79.

A large percentage of gearboxes have a continuous lube oil system where the lower casing half acts as the oil reservoir and an oil pump sprays lubricant into the meshing gears and floods the bearings where it returns back to the reservoir. Heat is generated in the bearings and meshing gear teeth during operation and the oil temperature is frequently controlled by a heat exchanger. Operating oil temperatures are typically kept around 120°F to 150°F.

Typical OL2R movement range of gearboxes:

Vertical movement: 5 to 20+ mils upward typically slightly asymmetrical (i.e., inboard and outboard ends do not move up the same amount, but very nearly so)

Lateral (*sideways*) *movement*: 0 to 10+ mils (see notes)

Axial movement: 0 to 30+ mils

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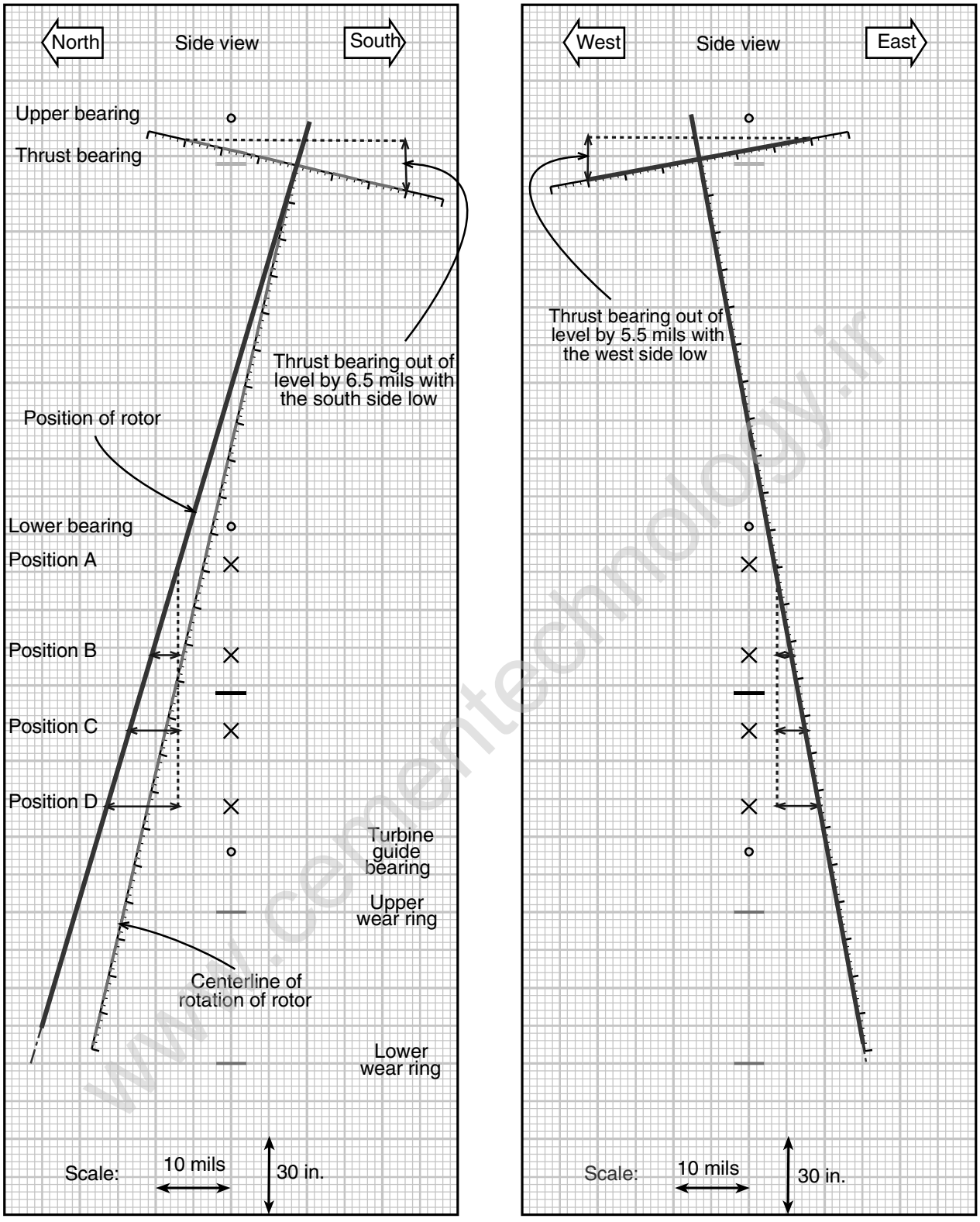


FIGURE 21.75 Alignment models of the rotor in the north to south and west to east positions.

21.1.11.1 Additional Information on Gearboxes and Fluid Drives

Moderate to excessive off-line soft foot conditions have been experienced on virtually every gearbox regardless of frame construction design. Gearboxes are frequently bolted to the frame in more than four points and soft foot correction can be more difficult to correct the

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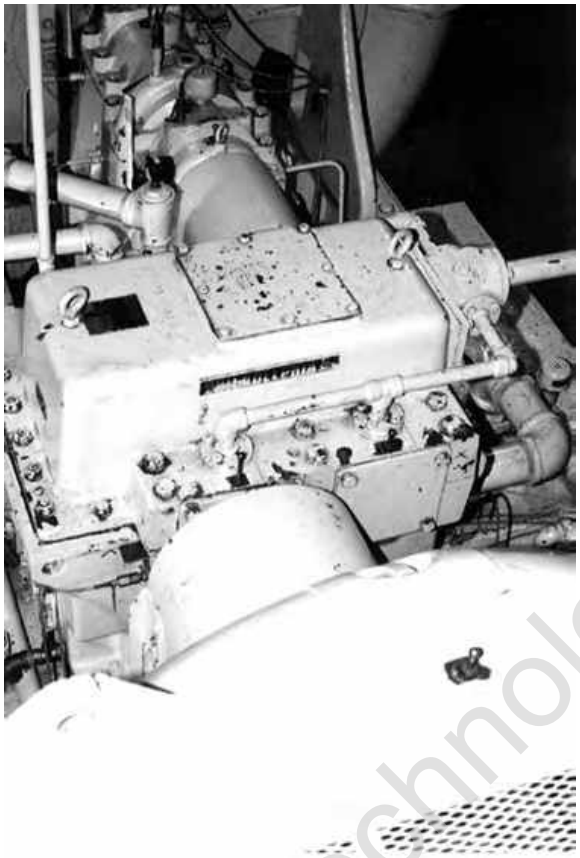


FIGURE 21.76 Gearbox.

machines bolted in three of four points. Uncorrected soft foot can distort the housing causing meshing problems as shown in Figure 21.80 through Figure 21.82.

Since there is typically a rise in casing temperature from OL2R conditions, not only will the shafts move upward, but they will also spread apart. Several OL2R studies have shown that gearboxes and fluid drives can twist or warp when operating. If dowel pins are used, casing distortion can occur if all four corners are pinned to the frame.

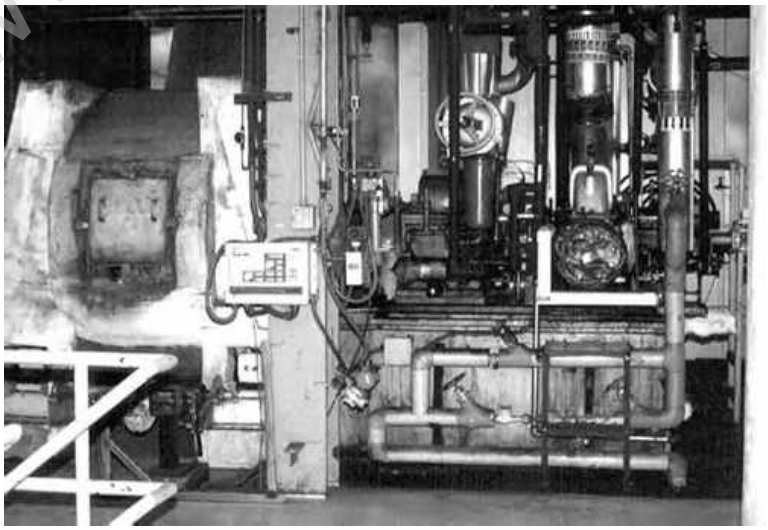


FIGURE 21.77 Steam turbine, gearbox, and fan drive system.

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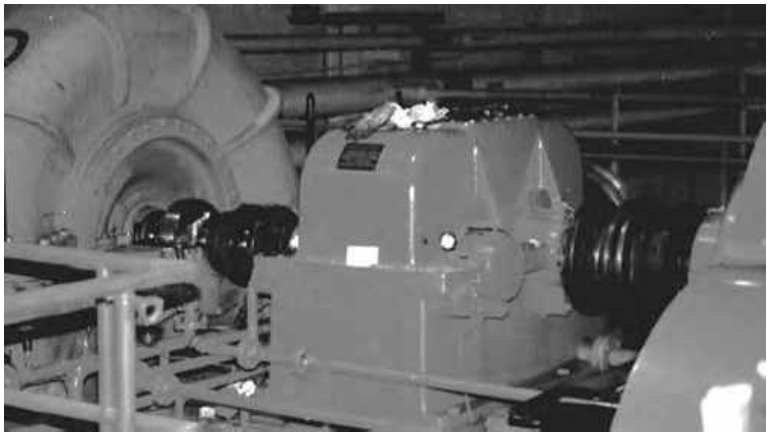


FIGURE 21.78 Motor, gearbox, compressor drive arrangement.

One method to compensate for the lateral expansion of a gearbox is to pin the high-speed shaft and allow the low-speed shaft to move into alignment during normal operation as shown in Figure 21.84. The one dowel pin at the high-speed shaft acts as the control point for the gearbox. The dowel pin at the opposite end of the high-speed shaft allows expansion in the axial direction but prevents the gear case from translating laterally. The bolts nearest to the dowel pins should be 90%–100% of final torque value. The torque on the foot bolts should be less as the distance from pins increases. This lower torque setting will hopefully allow the case to slide between the underside of the bolt head and the gear case foot, yet still provide a hold-down force to the baseplate or soleplates.

21.1.12 Cooling Tower Fan Drives

Although cooling tower fan drives are not usually thought of as glamorous rotating machinery systems, they are very critical to the operation of the plant and can experience alignment problems as acute as any other type of rotating equipment. In fan drive systems where a right-angled gearbox drives a six- or eight-bladed fan assembly where the drive motor is located outside the plenum and the motor is connected to the input shaft of the gear by a long spool piece or "jackshaft," OL2R movement is usually not measured and in many cases ignored. The saving factor in these designs is that the flexing points in the coupling are separated by a



FIGURE 21.79 Motor, fluid drive, pump drive arrangement.

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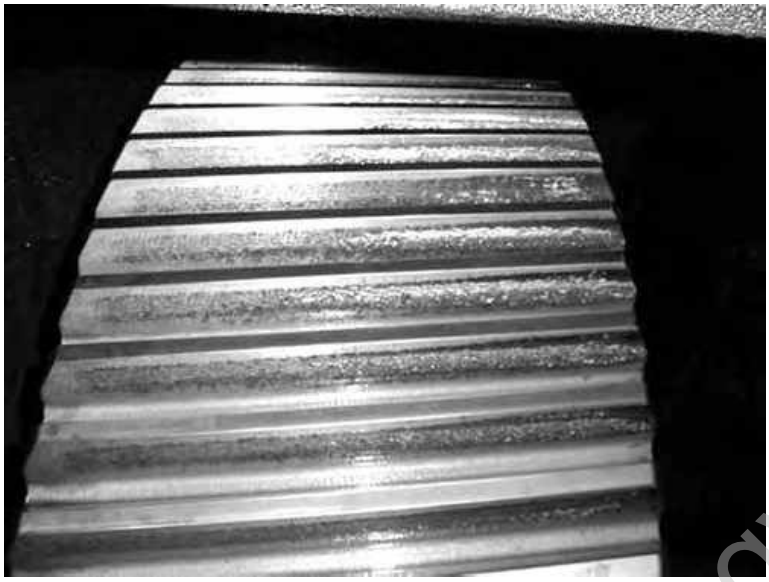


FIGURE 21.80 Irregular gear tooth wear pattern due to a soft foot condition distorting the gear housing.

considerable distance, thereby allowing for considerable amounts of centerline-to-centerline offsets at the flexing points. For example, if there is a 100-in. separation between the flexing points, you could have up to 100 mils of centerline-to-centerline deviation and still be at 1 mil/in. misalignment (100 mils/100 in. = 1 mil/in.).

The shaft to coupling spool method shown in Chapter 13 or the face–face technique shown in Chapter 14 is recommended for aligning these types of drives. Since most cooling towers are located outside, an interesting phenomenon can occur when aligning these drive systems during daylight hours with the sun shining. If the drive is kept stationary, the long coupling spool can get unevenly heated from the sun and thermally bow the spool piece. As you begin rotating the shafts to capture a set of readings, the hot or sunny side of the spool piece now begins to rotate into the shade and the sun starts to heat a different side of the spool piece. As the hot side cools and the shaded side warms up, the spool piece begins to change its shape causing erroneous readings.



FIGURE 21.81 Corner of gearbox in Figure 21.80 showing that the foot bolt is not supported on the outer edge (in addition to the soft foot problem shown in Figure 21.83).

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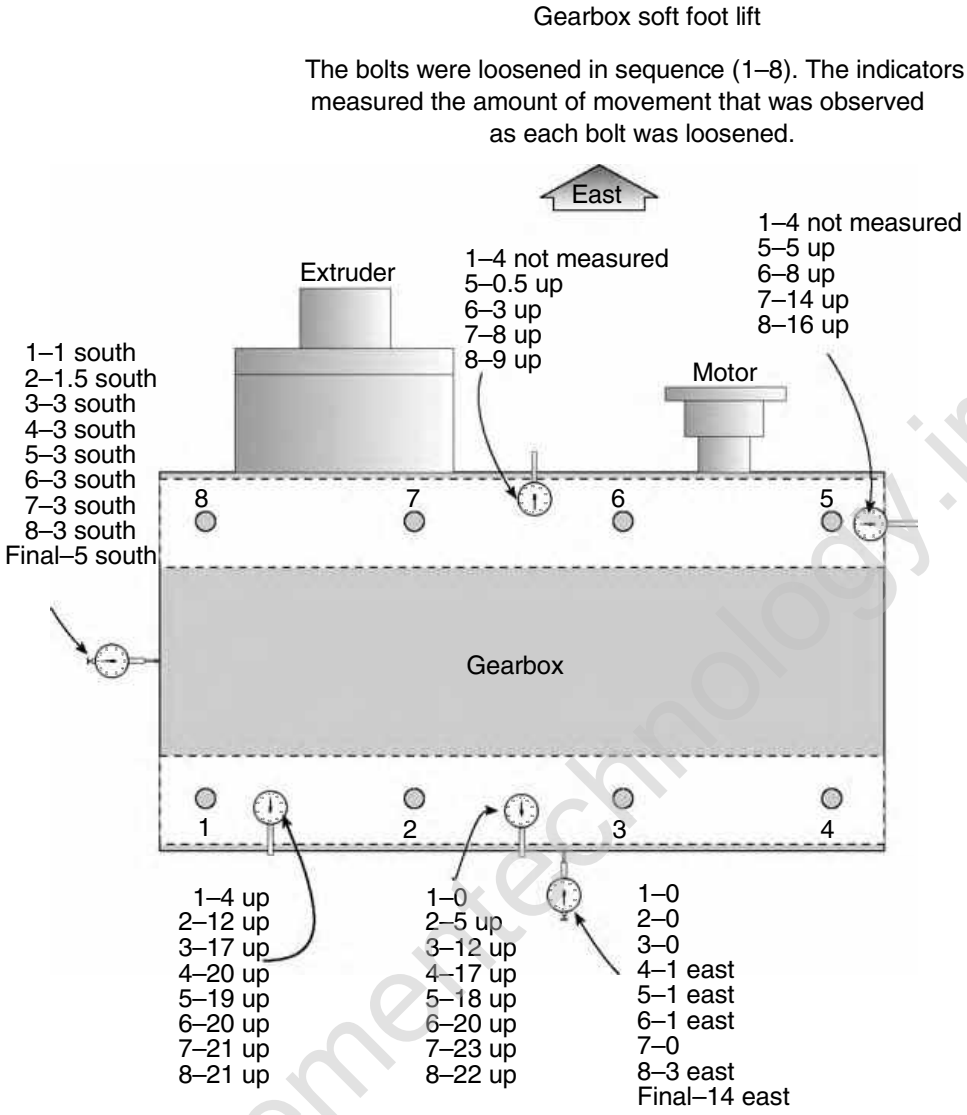


FIGURE 21.82 Soft foot lift check on above gearbox.

21.1.13 Aligning Ship Rudders

This section may seem like a radical departure from typical rotating machinery alignment measurement methods but alignment problems occur in virtually every industry and the marine industry is not an exception. In the life of a seagoing vessel, over time, the main steering device known as a rudder will require repairs to its stationary and rotating components. Erosion or corrosion of these devices and the occasional mishap of the rudder hitting an object or the sea floor will require maintenance and replacement of defective components. Figure 21.85 shows a view of the major components of a ship rudder.

The key components are the rudder horn (part of the ship hull), the rudder, the rudder stock (which is the shaft that rotates the rudder), the pintle pin, and the bearings. The pintle pin is the hinge pin for the rudder. It carries about 95% of the horizontal force of the rudder when the rudder is turned. The pintle pin is secured in a pintle pin bore in the rudder and held in place with a nut. The pin is forced on with significant force to form an interference fit. The rudder has a cast iron pintle pin boss as part of its structure and the rest of the "skin" of the rudder is welded on using plate steel on internal frames. The rudder is hollow and is usually filled to about 1 psi with air. It actually floats. The rudder is held vertically by

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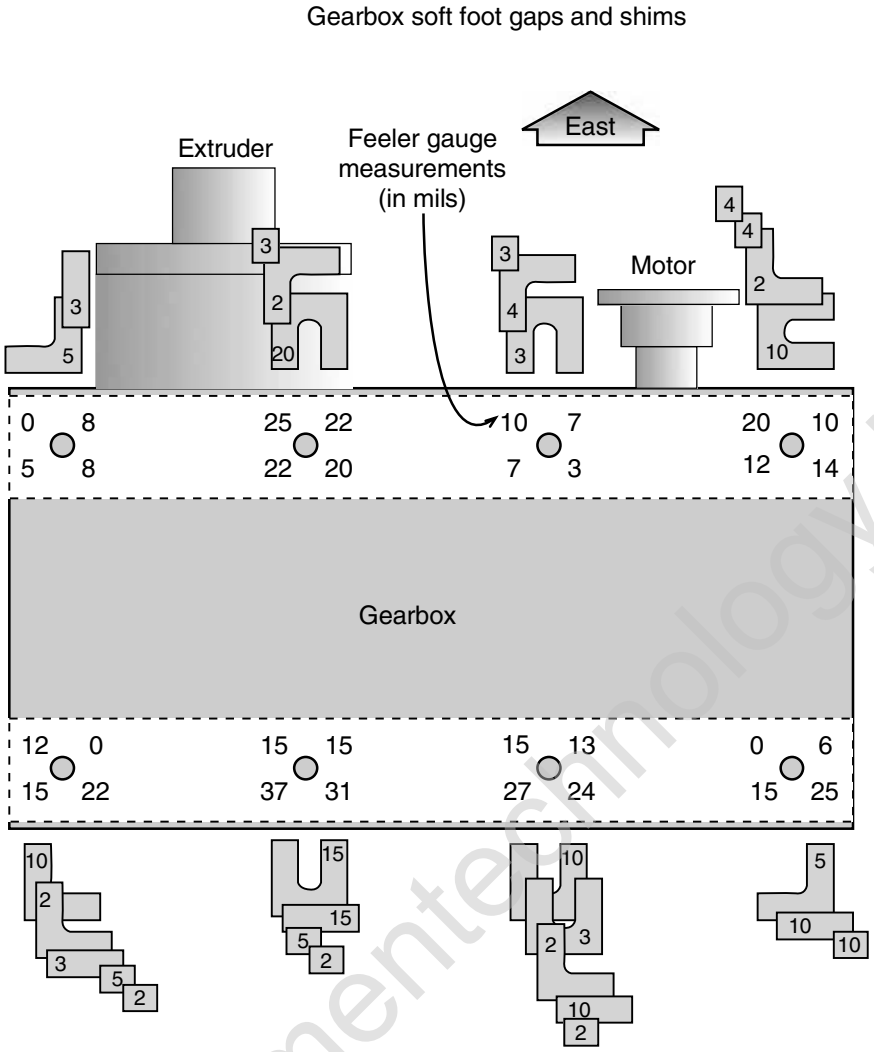


FIGURE 21.83 Soft foot map of above gearbox.

the pintle pin riding in a bearing surface on the rudder horn, by the rudder stock which is secured by multiple bearing strakes internal to the hull, and by the mechanical arrangement inside the hull which moves the rudder stock.

The rudder stock bore and the pintle pin bore are located in the rudder. They are about 20 in. ID at the tip and about 18 in. at the bottom with a taper of 1 in 12 in. The bores are usually of cast iron welded to rudders that can weigh over 35 t. The pintle pins are usually a metal-to-metal tapered fit (some with keys, others without) and 85% fit is usually required. The rudder stock bore has at least one keyway, which mates up with a key on the rudder stock. Sometimes (especially older German vessels) the keys are in the bore and the keyway is on the rudder stock. The keys are very large and are always bolted securely in place. The rudder is usually removed from the ship (usually weighs about 35 t) for this sort of work.

The typical disassembly sequence is shown in Figure 21.86 and Figure 21.87. The rudder is held in place with chain falls and the rudder stock is then removed along with the mechanical connections inside the ship. The rudder stock is then lifted out through holes in the ship decks. The rudder is then lowered and tilted until the pintle pin is free of its bearing surface in the rudder horn. The access panels in the rudder are cut out after the ship is in dry dock. The rudder is then removed and set up vertically in a work bay. The pintle pin nut access panel is removed and the pintle pin nut is removed. The pintle pin is then removed.

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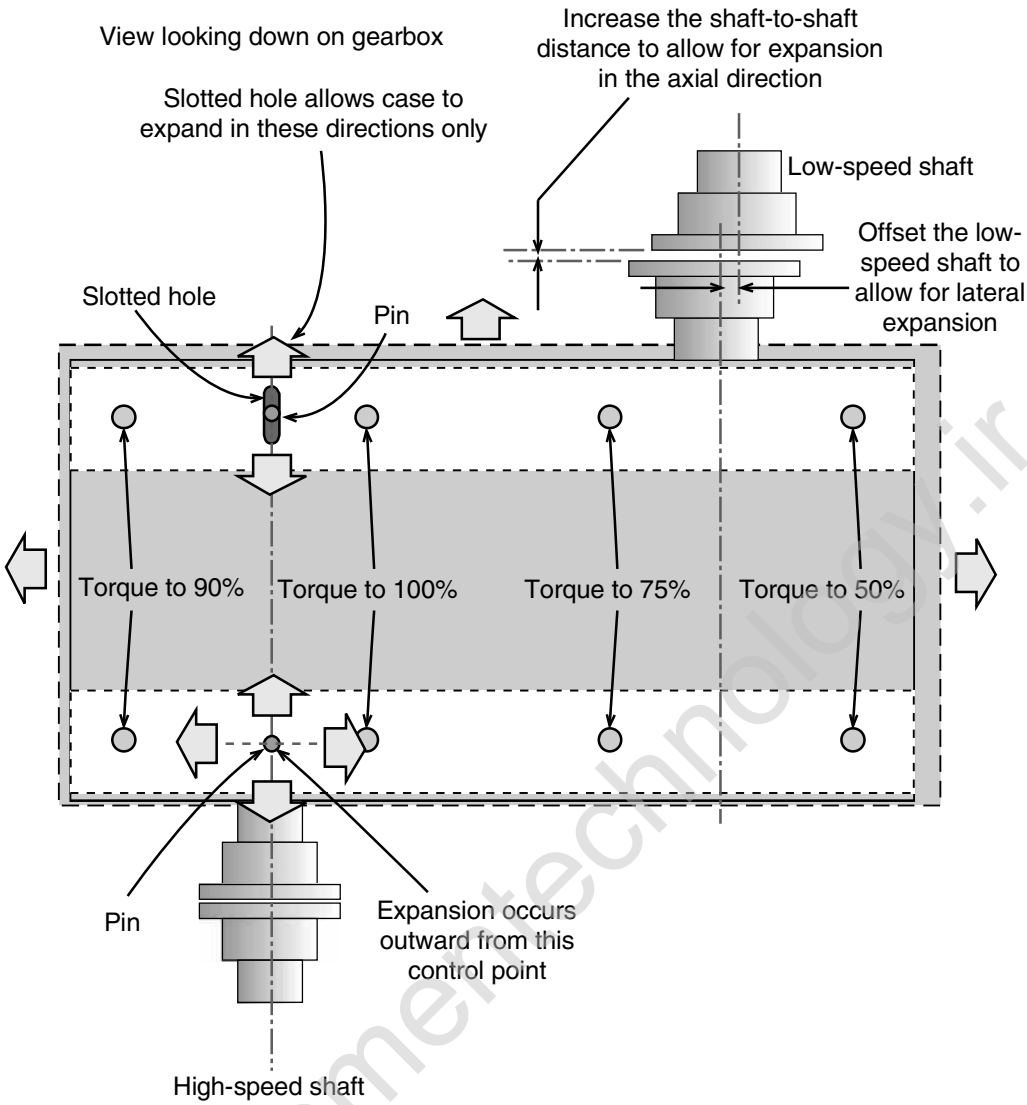


FIGURE 21.84 Pinning a gearbox at the high-speed shaft compensating and allowing the gear case to expand without warpage.

Now that the rudder has been removed from the ship, another issue needs to be addressed. Before getting into the tapered pintle pin alignment, one should investigate whether the centerline of rotation of the rudder stock is concentric to the bore of the pintle pin bearing in the rudder horn. If it is possible for the ship to run aground or hit something under water, it is possible for the rudder horn to get bent causing a misalignment between the rudder stock bearing and the pintle pin bearing. Assuming the rudder has been removed and the rudder stock is in place, Figure 21.88 shows how the double radial method could be used to measure if the bore of the pintle pin bearing is not collinear with the centerline of rotation of the rudder stock.

If you jumped to this part of the book because your boat is out of the water with a damaged rudder and are confused about the above diagram, you should probably review the basics of the double radial method as explained in Chapter 12.

The amount of angular rotation of the rudder stock shaft may be limited to no more than about 20° to 30° from side to side (total of about 60°). If you are limited in the amount of possible angular rotation, you could mathematically determine the full angular sweep based on the information given in Section 6.11. If you are not much of a math whiz and find that the section is boring or too intense, here is a trick I use in the event that you cannot rotate a shaft through 360°.

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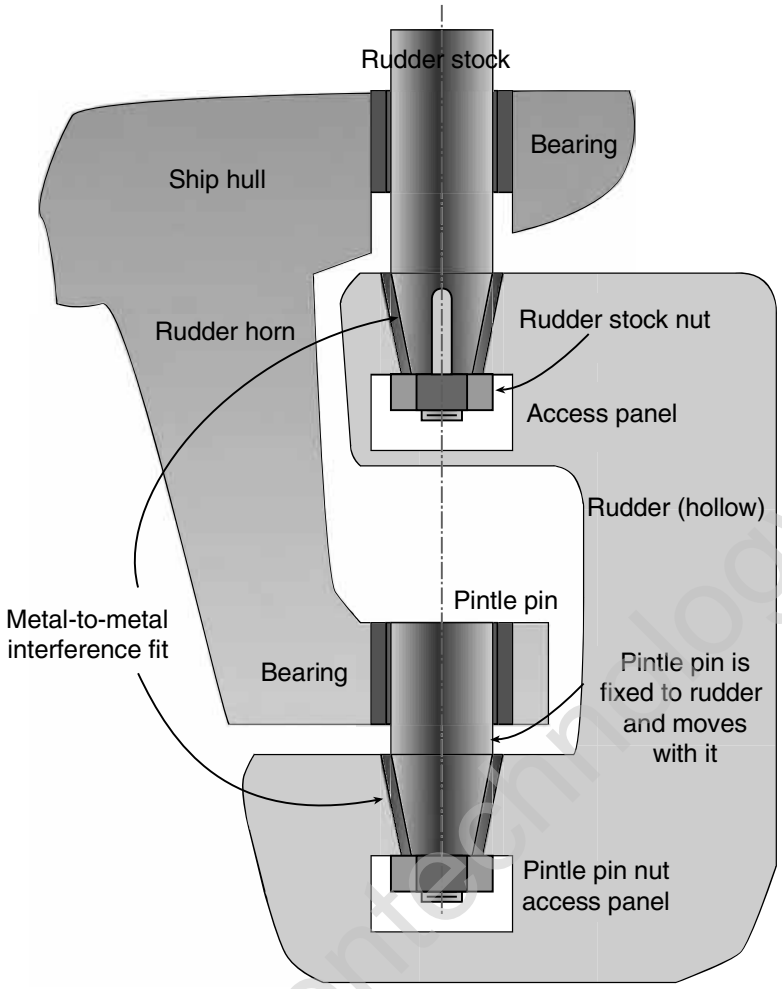


FIGURE 21.85 A typical rudder arrangement on a large ship.

1. Mark off the inside or outside of the cylinder (i.e., shaft or bearing bore) into 90° arcs. Since the rudder stock is vertically oriented, I usually try to use compass directions (N, S, E, W) or ship coordinates (fore, aft, port, starboard) to designate the position at each quadrant.
2. Rotate the shaft (rudder stock) all the way in one direction until it stops. Clamp the bracket to the shaft, set the indicator at one of the quadrant marks, and zero the indicator.
3. Rotate the shaft as far as it can go in the other direction (in this case 60°) taking care to observe what the indicator is reading as you do the rotation. When the shaft stops its rotation, record the dial indicator measurement and also scribe a mark with a pencil or soapstone exactly where the tip of the indicator stopped on the surface of the shaft or bearing bore. In this case, that is the bearing in the rudder horn.
4. Rotate the shaft back to its starting position, loosen the bracket on the shaft, rotate the entire bracket or dial indicator arrangement so that the tip of the indicator is positioned where it stopped at the pencil or soapstone mark, tighten the bracket, dial in the measurement you observed at this point, and start rotation again. Keep in mind that you only have 30° to go before so that you get to your first quadrant mark.
5. Repeat step 2 through step 4 until you get all the way around the shaft (see Section 6.10, i.e., you do not have to rotate all the way around).

Once the measurements have been taken at the top and bottom of the bearing bore, you could plot or model these measurements as described in Chapter 12. Remember, you are

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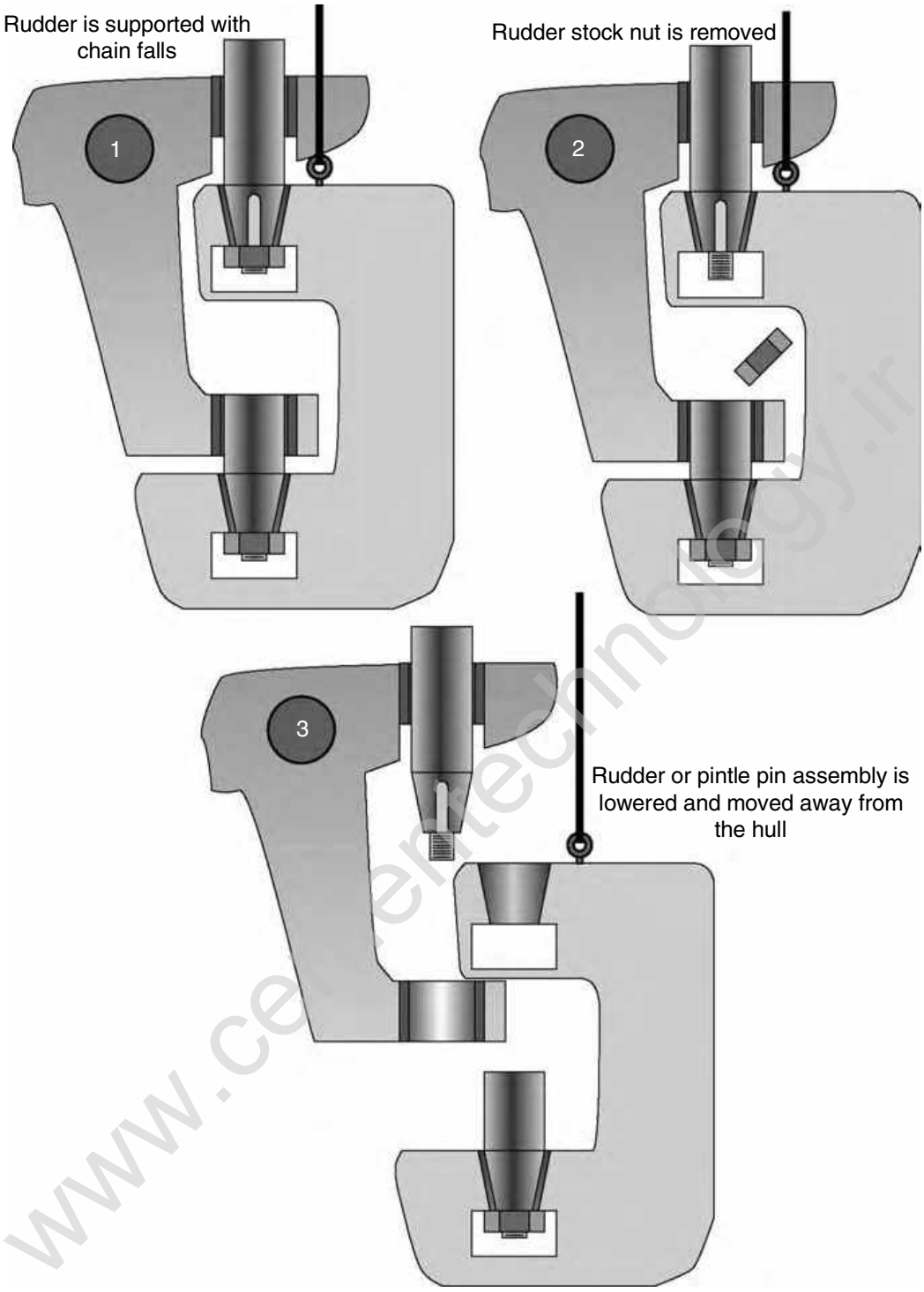


FIGURE 21.86 Removing the rudder from the ship.

taking bore measurements, not OD measurements; so be careful how you plot the points. If the centerline of rotation of the rudder stock shaft is concentric with the bore of the bearing in the rudder horn, you will sweep zeros all the way around the bore at the top and bottom of the bearing in the rudder horn. Notice that in Figure 21.88, they are not zeros all the way around. This is telling you that the rudder horn bearing is not aligned with the rudder stock shaft. You have got a major problem. Somehow, someway, you are going to have to position the rudder horn bearing to the centerline of rotation of the rudder stock shaft. If you do not figure out

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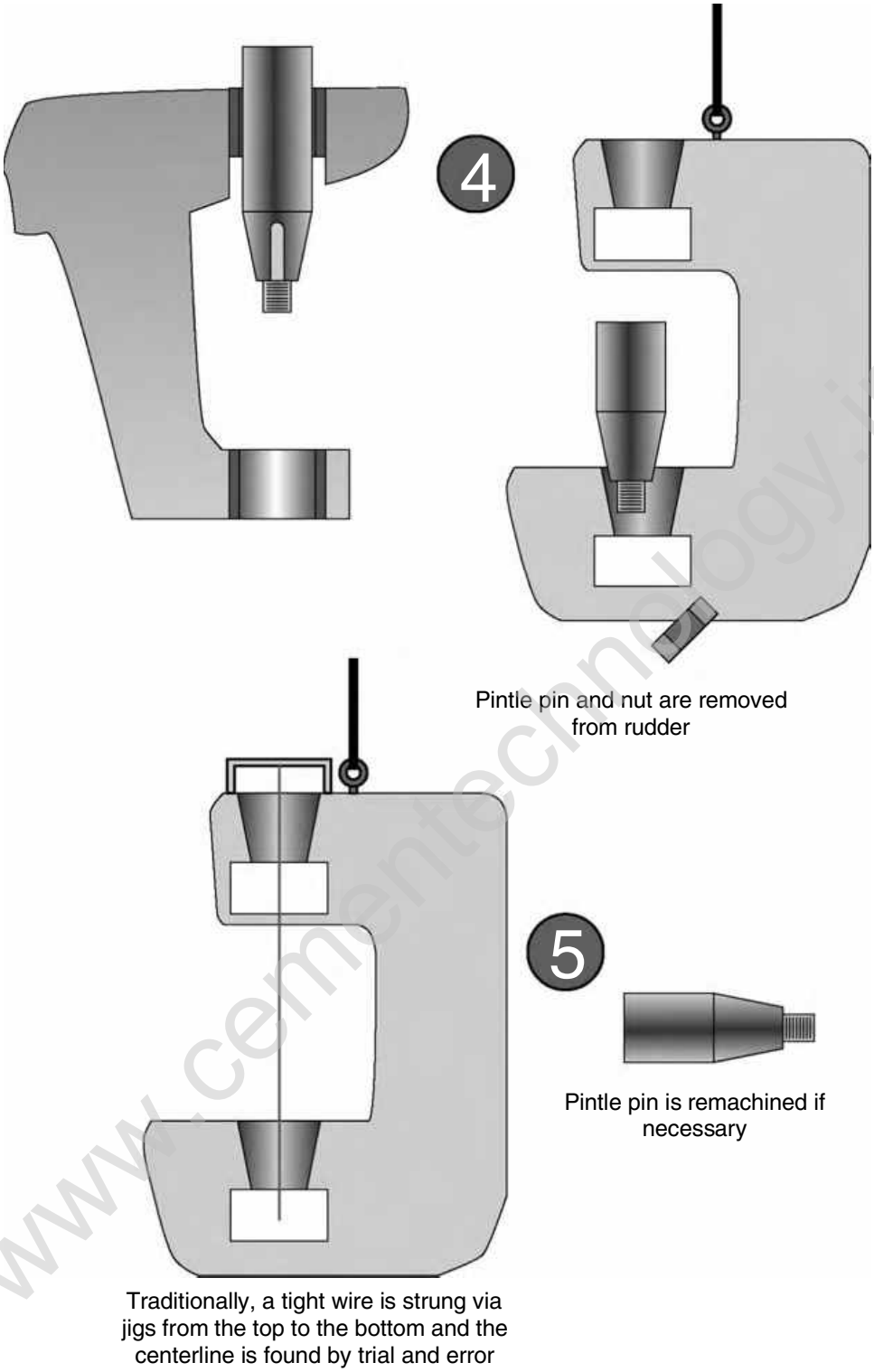


FIGURE 21.87 Final steps of removal.

how to do this at this point, the rest of this procedure is not going to help you. If we get the tapered bores of the rudder to be collinear, the alignment of the rudder stock bearing and the pintle pin bearing are not in line with each other, and none of this is going to work right. As far as I am concerned, the pintle pin is nothing more than an extension of the rudder stock shaft. For those of you who luckily swept zeros or painstakingly positioned the rudder horn bearing so it does sweep zeros, we can now go back to getting the rudder and pintle pin right.

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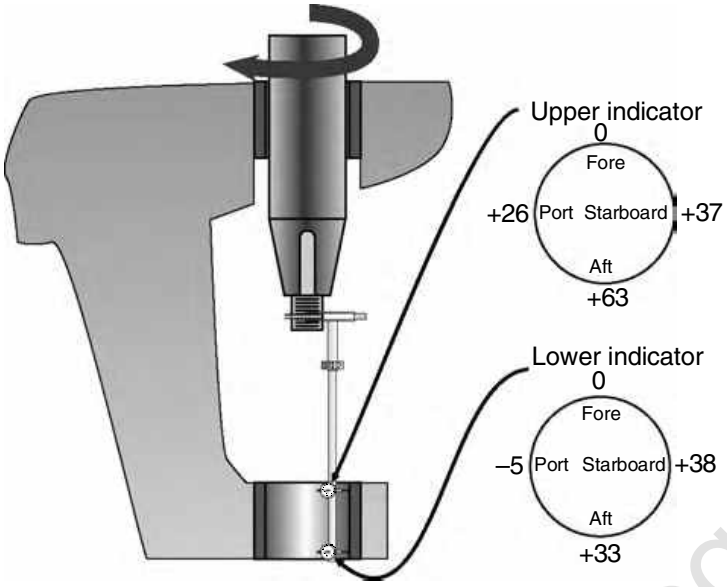


FIGURE 21.88 Using the double radial method to determine if the centerline of rotation of the rudder stock is in line with the bore of the bearing on the rudder horn.

For ships that have been in service for a while, usually the pintle pin is very loose in the pintle pin bore which is the reason the rudder is removed. Clearances of up to 1/4 in. have been found where metal-to-metal interference fit should exist. At this point, the rudder and pintle pin are either completely replaced or the tapered bores on the rudder or the pintle pin itself are repaired. It is not uncommon for the inside of the bores to have erosion or corrosion pits 1/4 in. deep. Repairs to the damaged bores and pintle pin are usually less expensive than replacing the entire rudder and pin. The tapered section of the pintle pin can be repaired by welding metal to the outside surface or applying a metal-based polymeric product ("liquid metal") and then machining the taper in a lathe. The real challenge is to repair the tapered bores in the rudder and insure that the centerline of the tapered bores at the top and bottom of the rudder are collinear.

One method is to weld up the damaged cast iron surfaces then bore them back to specs. However, there are inherent problems when welding cast iron. Additionally, the possibility of thermal distortion of the rudder could occur during the welding process. After the welding is completed, machining the tapered bores requires an extremely large lathe.

Another technique to accomplish this is to use the machined pintle pin as a male mold to reform the damaged tapered bore in the rudder using a metal-based polymeric epoxy product. But how do you correctly position the new (or repaired) pintle pin to get it concentric with the centerline of the upper and lower tapered bores of the rudder? Alignment of this very large and heavy pin can be tricky. It is imperative that the centerlines of the pintle pin bore (in the rudder) and the rudder stock bore (in the rudder) be the same. This is very difficult due to the mass of the rudder, the age of the ship (groundings, collisions, etc.), and the lack of a smooth-machined surface to take readings from.

Traditionally, a tight wire (piano wire) is usually strung via jigs from the top of the rudder stock to the bottom of the pintle pin bore on the rudder as shown in step 5 in Figure 21.87. Then by trial and error the centerline of the two bores is found. Rarely are the two bores perfectly in line. When the tight wire is centered as best it can, punch marks are made on the top and bottom of each bore equidistant from the wire at the cardinal points. The wire is then removed and jacking bolts are welded on to the top and bottom of the bores to move the pin after it is installed. Calipers are then used to measure from the punch marks to the outside of the pintle pin and the jacking bolts adjusted to center the pin in the bore. Measurements are

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noted, the pin is removed, and the metal-based polymeric epoxy product is applied to the tapered bore. The pin is recentered using the jacking bolts mentioned above, the epoxy is allowed to cure and then the pin is jacked out. The result is a molded metal-to-metal surface. The nut is applied, torqued down, and the rudder is replaced on the ship. If the rudder stock has problems, it is solved in a similar fashion.

An alternative method to find the centerlines of the tapered bores is shown in Figure 21.89, which shows a side and a top view of the arrangement. Rather than using a tight wire, a piece

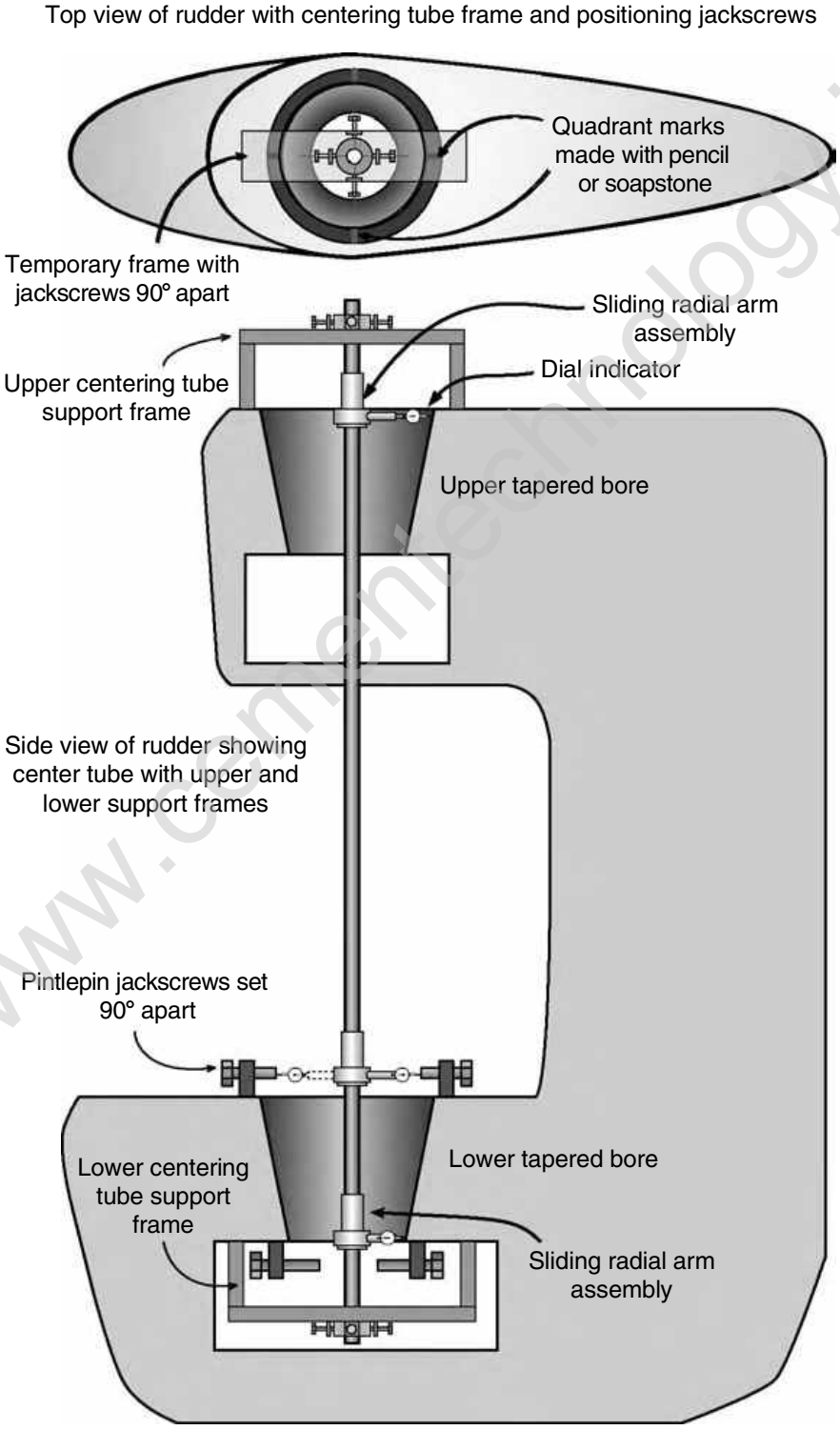


FIGURE 21.89 Centering tube and radial arm arrangement.

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of tubing is used to support a "radial arm" holding a measuring device, in this case, a dial indicator. Figure 21.90 shows a close-up of the radial arm assembly.

The radial arm has the capacity to slide up and down the center tube enabling one to measure any point along the length of the tube. Due to the taper of the bores, the radial arm must have the capacity to reach out different distances so that the radial arm is fabricated from two tubes that can telescope in each other. The center tube is held in place with fixtures at the top of the rudder where the rudder stock indexes into its tapered bore and at the bottom of the lower tapered bore in the rudder, the pintle pin indexes into its tapered bore. The upper and lower fixture has jackscrews that allow one to move top and bottom of the center tube to establish a precise centerline. The general procedure would be as follows:

1. Position the upper and lower center tube support frames as shown in Figure 21.89. Slide the center tube through the upper support partway and then slide the radial arm onto the center tube. Slide the center tube until it indexes into the lower support bearing. Roughly center the fixtures using a tape measure from the outside diameter of the center tube to points on the inside of the tapered bores (i.e., roughly within ± 0.125 in.). Rigidly attach the upper and lower center tube support frames to the rudder by drilling and tapping holes or by tack welding the fixture in place (you might be able to use strong magnets to hold them but you run the risk of bumping it later on).
2. Attach a dial indicator to the end of the telescoping tubes of the radial arm assembly. Slide the radial arm so that it is in line with the top of the upper tapered bore and lock it

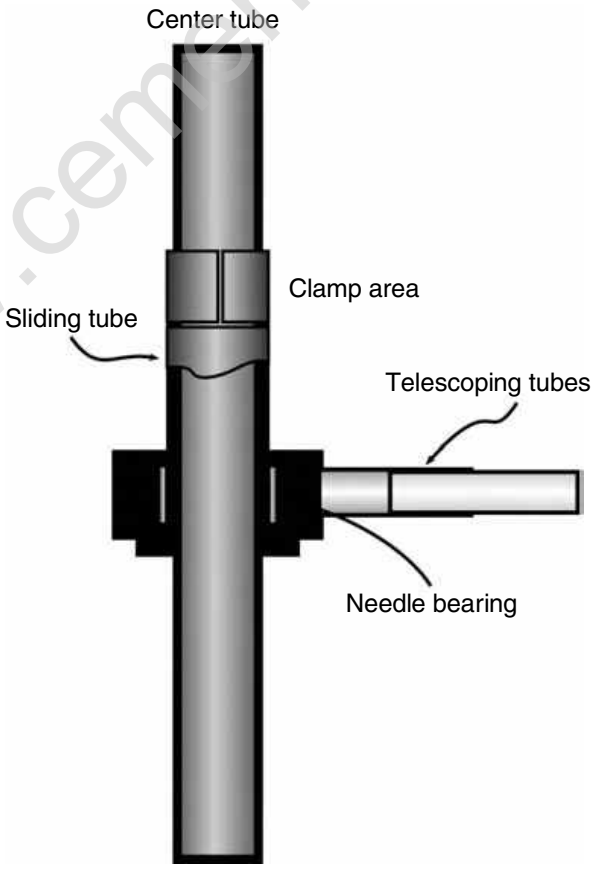


FIGURE 21.90 Radial arm assembly.

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in this position. Rotate the radial arm so that it is in line with one of the quadrant marks, zero the indicator, and rotate the radial arm through a 180° arc observing the dial indicator as you do so. Adjust the jackscrews so that the dial indicator reads half of the total stem travel. Rezero the indicator and rotate 180° back to its original point. If the indicator is not zero, adjust the jacking screws until you sweep zero from side to side. Rotate the radial arm through a 90° arc to the other two quadrant marks and repeat the centering procedure described in this step. When the indicator stays at zero through the entire 360° of rotation, the centering tube is coincident with the bore of the tapered hole at that point.

3. Slide the radial arm so that it is in line with the bottom of the lower tapered bore and lock it in this position. Rotate the radial arm so that it is in line with one of the quadrant marks, zero the indicator, and rotate the radial arm through a 180° arc observing the dial indicator as you do so. Adjust the jackscrews so that the dial indicator reads half of the total stem travel. Rezero the indicator and rotate 180° back to its original point. If the indicator is not zero, adjust the jacking screws until you sweep zero from side to side. Rotate the radial arm through a 90° arc to the other two quadrant marks and repeat the centering procedure described in this step. At this point, the center tube should be positioned at the centerline of the bore of the two tapered bores. You could take additional measurements at the bottom of the upper tapered bore and at the top of the lower tapered bore to see how much of a variation exists at these two points.
4. Slide the radial arm to a position where the pintle pin locating jackscrews will be placed. Affix four jackscrews at 90° arcs on the top and bottom of each tapered bore (i.e., a total of eight jackscrews).
5. Remove the radial arm, center tube, and fixturing mechanisms.
6. Proceed with the temporary installation of the pintle pin and bore repair as shown in Figure 21.91.
7. Once the metal-based polymeric epoxy product has been poured and hardened, remove the pintle pin and jackscrews.

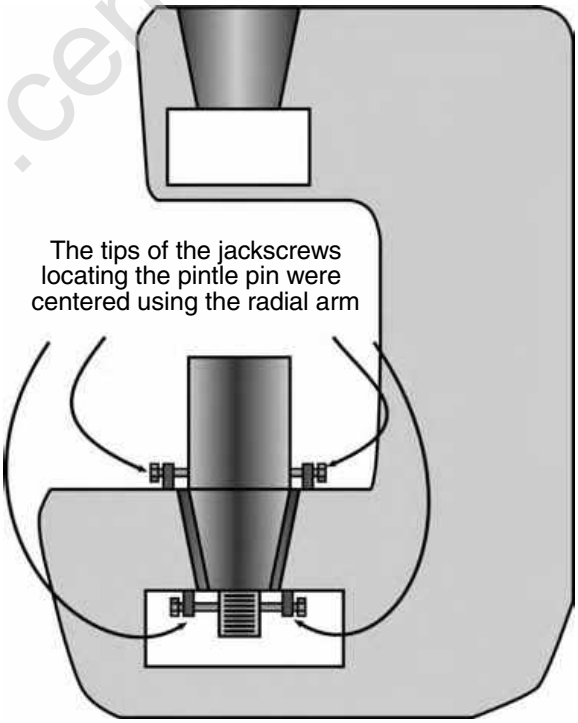


FIGURE 21.91 Pintle pin held in position with jackscrews ready for the metal-based polymeric epoxy to be poured.

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22 The History of Machinery Alignment

The historical path of shaft alignment encompasses many interwoven engineering disciplines. To understand how shaft alignment progressed to its current state of technology will require an exploration of a wide variety of topics. As necessity is the mother of invention, the necessity of aligning rotating machinery shafts is directly linked to the development of rotating machinery and therefore important for us to look at the progression of this equipment throughout history. The subject of shaft alignment is also concerned with mechanical measurement, mathematics, metallurgy, vibration analysis, statics and dynamics, optics, and electronics—each contributing to the use of the current tools and techniques described in this book.

Our story will start 450,000 years after man first began to gather in groups and use fire in a controlled fashion during the mid-Pleistocene era. The crux of engineering triumphs during the Neolithic era (11,000 to 6000 BC) was the emergence of permanent dwellings, the beginnings of agriculture, and to a very basic extent—metallurgy. Even through 4000 years of inhabitation from 8000 to 4000 BC in the oldest cities of Jericho, Ain Ghazal, and Catal Hüyük (located in Israel, Lebanon, and Turkey), very little developed in terms of tools, measurement, architecture, and science.

From this point forward begins the emergence of "modern" men's eventual development of shaft alignment displayed to you in snapshots of time events.

–4000 to –3500—Smelting of gold and silver known. Copper alloys are used by Egyptians and Sumerians.

–3000 to –2500—Cheops pyramid in Egypt built to extremely precise dimensions showing knowledge of geometry and measurement. A first attempt to establish a standard of measurement by Egyptians was the cubit (changed from ≈18 to 20.63 in. around 3000 BC). Iron objects first appear.

–2000 to –1500—Babylonia uses highly developed geometry for astronomical measurements. Assyrians and Babylonians establish the units of measurement as: the cubit (now 20.5–20.6 in.), the span (10.5 in.), and the digit (≈0.653 in.). Egyptians use knotted rope to construct right angles illustrating Pythagorean theorem, which is also known in China during this period. Water level believed to be used in Mayan culture for construction of irrigation systems (see –600 to –500).

–1000 to –900—Chinese textbook of mathematics shows principles of plainemetry, proportions, arithmetic, root multiplication, geometry, equations with unknown quantities, theory of motion.

–900 to –800—Iron and steel production in Indo-Caucasian culture.

–600 to –500—Theodorus of Samos, a sculptor, credited with inventing ore smelting and casting, water level, carpenters square, and lathe.

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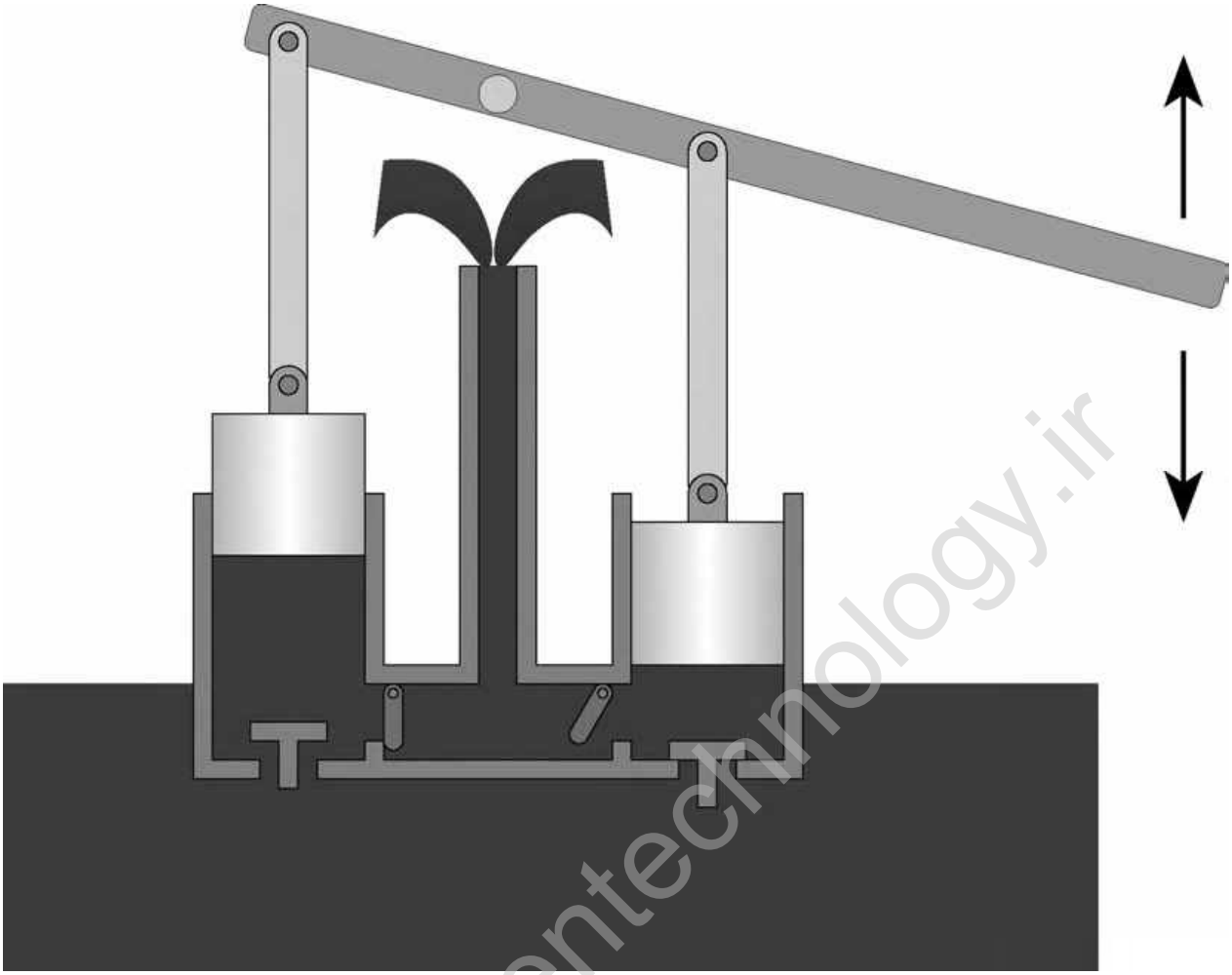


FIGURE 22.1 Ctesibius pump.

–384—Aristotle was born. Credited with much of the initial discoveries in physics, biology, and psychology contained in his book *Historia Animalum*. Aristotle or his student Straton publishes *Mechanika* discussing the lever and gearing.

–323—Euclid writes his first book on geometry called *Elements*.

–300 to –200—Ctesibius of Alexandria invents the force pump (Figure 22.1) and Archimedes of Syracuse invents the screw pump. The appearance of gears leads to the development of the ox-driven water wheel for irrigation. Universal joint used by Greeks (see 1550).

100—Hero of Alexandria describes the principles of an aeolipile (Figure 22.2), a simple reaction steam turbine, in *Pneumatica*, describes levers, gears, motion on an inclined plane, velocity, and the effects of friction in his book *Mechanics*. Hero's book, *On the Dioptra*, describes a type of theodolite, and explanations of plain and solid geometrical figures, conic sections, formula for calculating the area of a triangle from the lengths of its sides, and a method for determining the square root of a nonsquare number appear in his book *Metrica*. Theodosius of Nithynia (also known as Theodosius of Tripoli) authors "Sphaerica" dealing with spherical geometry.

250—Diophantus of Alexandria writes first book on algebra.

285—Pappus of Alexandria describes operation of current machines in use: cogwheel, lever, pulley, screw, and wedge.

700—Water wheels for mills used all over Europe.

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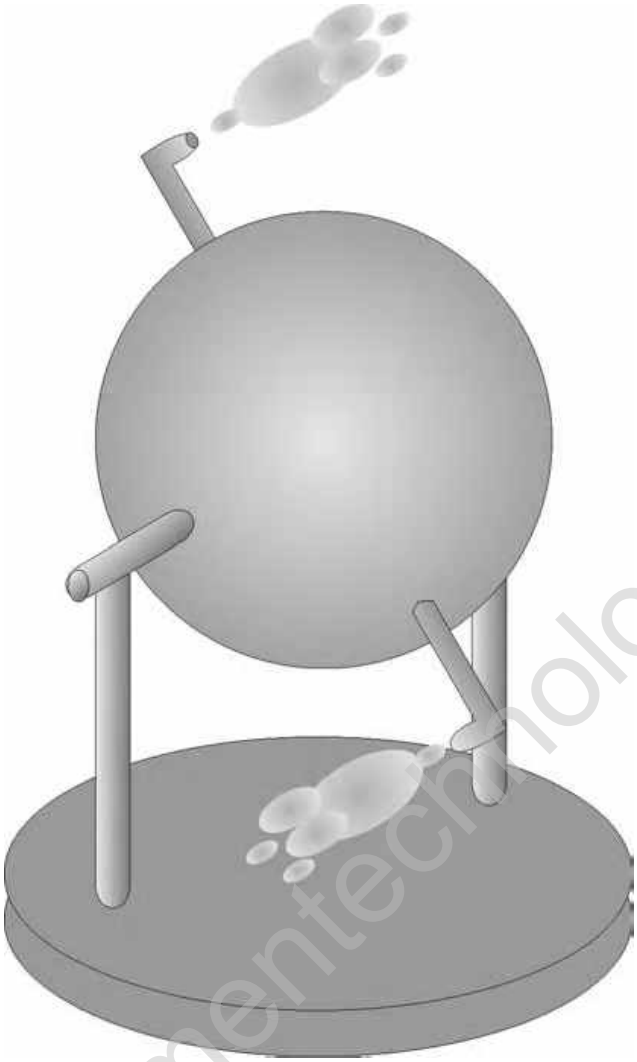


FIGURE 22.2 Hero's aeolipile.

1280—Roger Bacon discusses the basic operation of the telescope by use of "... glasses or diaphanous bodies that may be formed that the most remote objects may appear just at hand..." in his book *Epistola ad Parisiensem* (see 1600).

1305—Edward I of England issued a decree that read... "It is ordained that three grains of barley dry and round make an inch, twelve inches make a foot, three feet make an ulna, five and a half ulna make a rod, and forty rods in length and four in breath make an acre."

1510—Leonardo da Vinci designs horizontal water wheel and proposes a "smokejack," which uses the hot gases rising from a fire to propel a vertically oriented shaft driven by blades attached to the shaft as shown in Figure 22.3. This device was later patented by John Dumbell of England.

1540—Filde Nanez Salaciense of Portugal, while working on a system to more accurately read angles for his map making work, devises a rotary scale where a series of divisions were marked equally around the circumference of an outer scale and an inner rotary scale with a series of equally marked divisions one division less than the outer scale. Although the device did not work very well due to the difficulty of accurately scribing equally spaced lines by hand, the idea became the basis of the vernier scale (see 1631).

1550—Jerome Cardan (Geronimo Cardano) Italian physician, mathematician, and friend of DaVinci is credited with the "invention" of the universal joint (also known as Hooke

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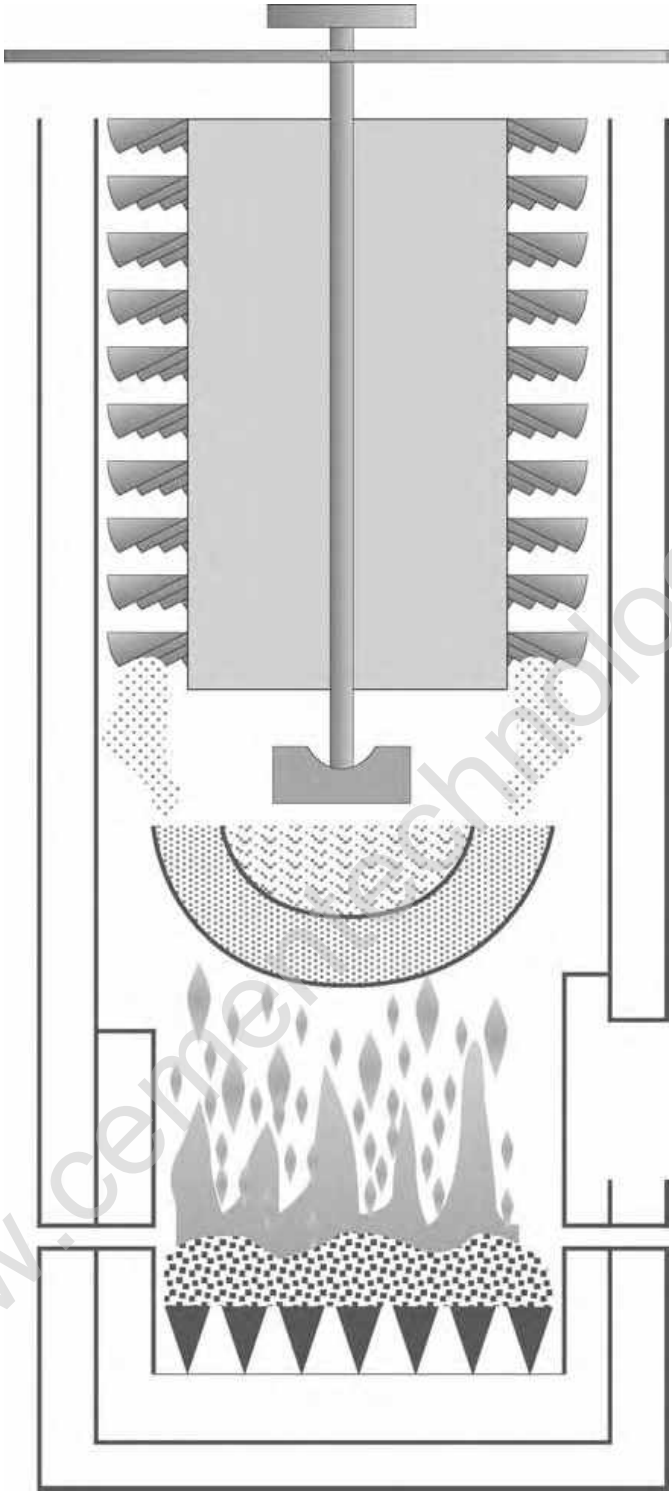


FIGURE 22.3 Da Vinci's smokejack.

or Cardan joint). Georgius Agricola (Georg Bauer) writes *De Re Metallica*, a 10-volume series of books in chemical and mining engineering covering ores, theory of formation of mineral veins, surveying, tools, machines, pumps, hoists, water power, ore preparation, smelting, and manufacture of salt, soda, alum, vitriol, sulfur, bitumen, and glass. These works were later translated to the English language by American President Herbert Hoover.

1576—François Vieté introduces use of decimal fractions.

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1600—Dutch optician Johann Lippershey invents telescope, which is adapted for astronomical observations a year later by Galileo Galilei who manufactured hundreds of telescopes that were in great demand by "amateur" astronomers of his time. Galileo is also credited with the invention of the thermometer, was the first person to coin the term "moment" meaning the effect of force, founded the science of strength of materials, and his fascination with the periodic swing of pendulums began his inquisition into falling objects laying the ground work for gravity and acceleration.
1601—Giovanni Battista della Porta develops principles of condensing steam turbine.
1611—Marco de Dominus publishes scientific explanation of a rainbow (electromagnetic spectrum).
1614—Scottish mathematician, John Napier publishes *Canonis Descripto*, describing his discovery of logarithms. Napier was the first to use the decimal point to express fractions.
1621—William Oughtred devises the first slide rule using Napiers logarithms.
1629—Giovanni Branca describes using a jet of steam impinging on blades projecting from a wheel to produce a rotating shaft.
1631—Pierre Vernier invents slide caliper.
1639—Prior to his death at the age of 24 in the Civil War of 1642, astronomer William Gascoigne invents micrometer from his work of attempting to determine the diameter of celestial objects. By devising a caliper whereby two fingers were moved toward or away from each other simultaneously by left hand and right hand threads, the image in Gascoigne eyepiece could be determined by the finger distance and triangulation principles. It is not clearly known where William got the precisely cut threads for his micrometer but a French engineer Besson built a lathe capable of cutting threads in 1569 (see also 1791).
1650—Robert Hooke uses universal joint in clock. In 1655 he was employed by Robert Boyle who used his technical expertise to assist in constructing the "air pump." Boyle later publishes a book in 1660 entitled *New Experiments Physico-Mechanical Touching the Spring of Air and Its Effects*.
1652—German scientist Otto von Guericke invents an air pump and 11 years later constructs a frictional electric generating machine.
1665—Francis Grimaldi explains diffraction of light and Isaac Newton invents differential calculus.
1673—Gottfried Leibnitz invents the "Leibnitz Wheel," the first mechanical device to perform addition, subtraction, multiplication, and division.
1690—Denis Papin devises noncondensing, single acting steam pump with piston and the steam safety valve.
1698—Thomas Savery obtains patent for steam driven, water raising engine and first coins the term "horsepower." The success of the coal-fired steam engines are directly linked to the sixteenth century energy crisis caused by the lack of wood fuel from the deforestation of England (see Figure 22.4).
1702—Alain Manesson Mallet invents a telescopic sight incorporating a level bubble called the dumpy level.
1704—Isaac Newton publishes *Optics* defending the emission theory of light. From 1662 to his death in 1727 he coinvents calculus with Leibnitz, develops the generalization of the concepts of force, mass, and the principle of effect and counter-effect.
1705—Thomas Newcomen (Figure 22.5) and John Cawley invent condensing steam turbine.
1714—D.G. Fahrenheit invents mercury thermometer followed by Réaumurs alcohol thermometer 2 years later followed by Anders Celsius centigrade thermometer 28 years later.

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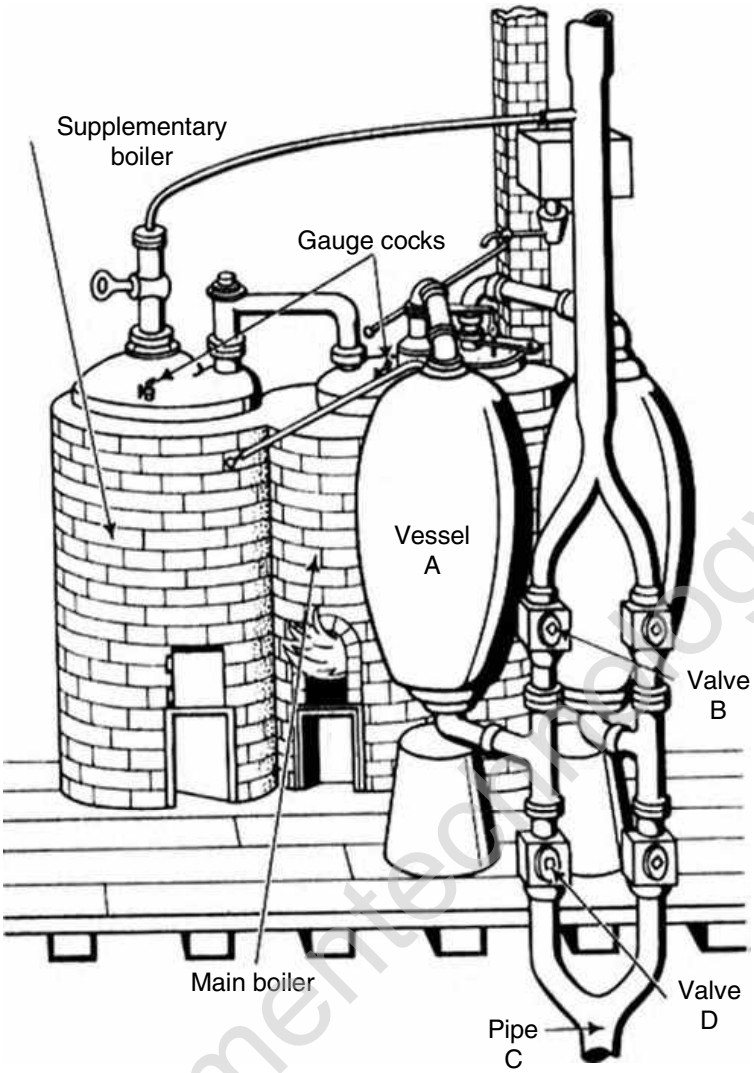


FIGURE 22.4 Savery steam engine.

1720—Theodolite made by Sisson Benjamin Cole in London was the first to employ two spirit levels (see Figure 22.6).

1729—English scientist Stephen Gray discovers that some bodies are conductors and nonconductors of electricity.

1745—Ewald Jurgen von Kleist invents capacitor (Leyden jar).

1750—Swiss mathematician Leonard Euler and his son Albert experiment with impulse driven water turbines. Leonard also developed equations describing buckling of struts, the catenary curve, and formulated the laws governing the flow of fluids and the relationship of pressure to flow (see 1770).

1754—P. Van Musschenbroek at the University of Leyden in Holland first demonstrated that when two insulated metal plates are brought in close proximity to each other without making contact, considerably more electrical charge could be stored than in a single plate (the Leyden jar) commonly known today as a capacitor. When a wire is connected to one plate of an electrically charged Leyden jar and then made to touch the other plate, an electrical discharge occurs.

1762—Cast iron first converted to malleable iron at Carron ironworks in Scotland.

1764—James Watt invents steam condenser for improvement to steam engines patenting his improvements in 1769. Files a second patent in 1781 describing sun and plant wheels

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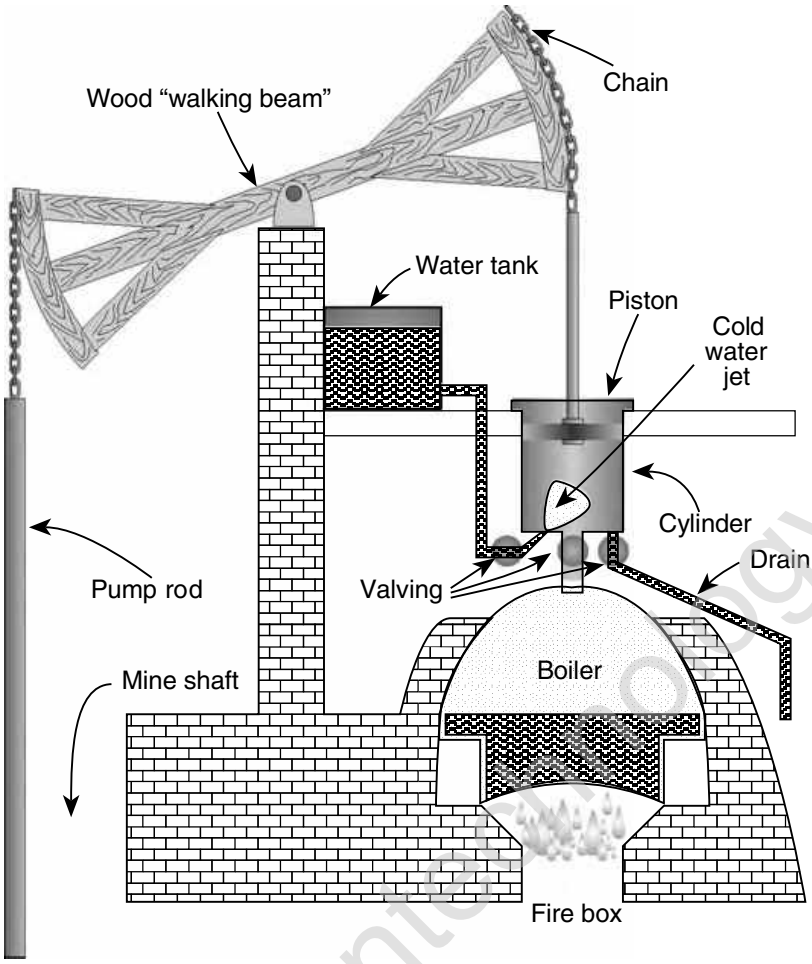


FIGURE 22.5 Newcomen's mine pump.

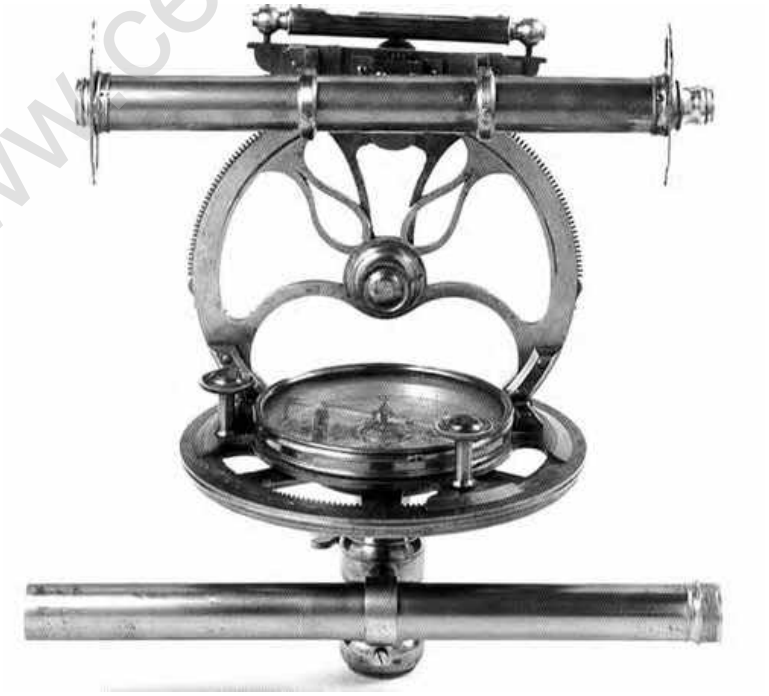


FIGURE 22.6 Eighteenth century theodolite.

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and a flywheel. Files another patent in 1782 describing the use of double action whereby steam is injected on one side of a piston and vacuum on the other.

1768—French mathematician and physicist Jean Baptiste Fourier born.

1770—Leonard Euler publishes *Introduction to Algebra* followed by another book on mechanics, optics, acoustics, and astronomy 2 years later.

1774—John Wilkinson constructs boring mill to manufacture cylinders for steam engines.

1786—Galvani discovered electric current occurs when two dissimilar metals come into contact with each other. By suspending zinc and copper plates in an acid solution Galvani showed that a steady flow of current would flow (chemical battery).

1787—Ernst Chlandi experiments with sound patterns on vibrating plates.

1790—The French National Assembly committee members decided that the meter would be one ten-millionths of a quadrant of the Earth's meridian. In 1799, a platinum–iridium end bar was produced and became known as the "Metre des Archives," the master standard of length in the world. The bar's length was based on a slightly inaccurate geodetic survey made to establish the distance of the Earth's meridian. As of 1983, the meter is currently defined as the distance light travels in a vacuum after 1/299,792,458 of a second.

1791—John Barber patents first gas turbine. After working with Joseph Bramah, 22-year-old British engineer Henry Maudslay starts his own business and develops a metal lathe (most former lathes were mostly made from wood) capable of accurately cutting threads. Using his new machine, Henry cuts 50 threads per inch in a long rod that is eventually used as a micrometer to check his work. Henry is also credited with the leather "U" seal when working with Bramah in the development of the hydraulic press.

1800—William Herschel discovers existence of infrared solar rays, Richard Trevithick constructs low-pressure steam engine, and Alessandro Volta was the first to produce electricity from zinc–copper battery.

1802—John Dalton introduces atomic theory in chemistry.

1806—Oersted discovers that a magnetic field is produced around a wire where electric current is flowing proving for the first time that electricity and magnetism are indeed related.

1815—Augustin Fresnel begins research on diffraction of light.

1816—Ernst Werner von Siemens born in Hanover. Credited with the invention of the armature initially used in telegraphy and later used in the larger generators (dynamo) demonstrating the dynamo-electric principle. Along with his brothers Karl Wilhelm (developed a type of governor for steam engines), Friedrich, and nephew Alexander founded the Siemens Company.

1824—Joseph Aspdin of England patents portland cement process naming it after its resemblance to portland stone, a limestone quarried at Portland, England.

1829—J.B. Nelson introduces the hot blast furnace.

1831—Michael Faraday, while taking a hiatus on his work in chemistry after discovering benzene and butylene, and manufacturing the first stainless steel, began work on getting electricity from magnetism. His now infamous experiment whereby he thrust a permanent magnet into and out of a coil of wire producing a flow of current laid the ground work for much of the work about to be done in electric generation and basis for operation of the vibration sensor now known as the velocity pickup (seismometer).

1843—Jonval introduces axial flow turbine.

1848—The first screw caliper patent (i.e., micrometer) was issued to French mechanic Jean Laurent Palmer.

1849—James B. Francis builds a radial inflow water turbine wheel (172 kW/230 hp) based on a patent by Samuel Howd achieving an efficiency of 80%. Twenty-year-old Lester

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Pelton ventures to California in his failed attempt to find gold but experiments with water wheels used in gold mining to design what is now known as the Pelton wheel.

1853—Lord Kelvin discovered that current must pass back and forth between two plates in a condenser to be able to emit electromagnetic waves introducing the concept of an oscillating electrical circuit, the key to all radio transmission.

1856—Henry Bessemer decarburized molten iron by blowing cold air through the iron forming mild-carbon steels upon cooling.

1864—Scottish physicist James Maxwell theorizes that not all of the energy in Kelvin's oscillator is dissipated as heat but a certain percentage must radiate into space as electromagnetic radiation. Maxwell, based on the findings by Wilhelm Weber and Friedrich Kohlrausch showing that the velocity of electricity through a wire and the velocity of light in a vacuum are the same, concludes that there is a relationship between the two giving birth to the concept of the electromagnetic theory of light.

1867—Parisian gardener Joseph Monier obtains patent for reinforced concrete. J.R. Brown and Lucian Sharpe while visiting the Paris Exposition saw a Palmer micrometer (see 1848). Taking the best features of the Palmer micrometer and another micrometer designed by S.R. Wilmot (superintendent of Bridgeport Brass) Brown and Sharpe released the first U.S. made micrometer in 1867.

1868—R.R. Musket introduces tungsten into steel manufacturing a self-hardening metal used for cutting tools. The U.S. Navy's Bureau of Steam Engineering adopts William Sellers Unified Screw Thread design using 60° as the thread angle. Not until after the end of World War II did American, British, and Canadian representatives finally agree on this standard thread design.

1869—W.J. Rankine publishes paper "On the Centrifugal Force of Rotating Shafts" in *Engineer* (Vol. 27, p. 249).

1872—F. Stolze of Germany develops gas turbine consisting of a separately fired combustion chamber, a heat exchanger, and a multistage axial flow compressor coupled to a multistage reaction turbine.

1877—Sir Charles Parsons begins work at Armstrong Works in Elswick England. In 1884, serving for a year on the experimental staff of Messrs. Kitson of Leeds where he patents the modern day steam turbine. Leroy S. Starrett invents combination square.

1878—Centralized generating station first proposed by St. George Lane Fox (England) and Thomas Edison (United States). Carl De Laval builds a small 42,000 rpm reaction steam

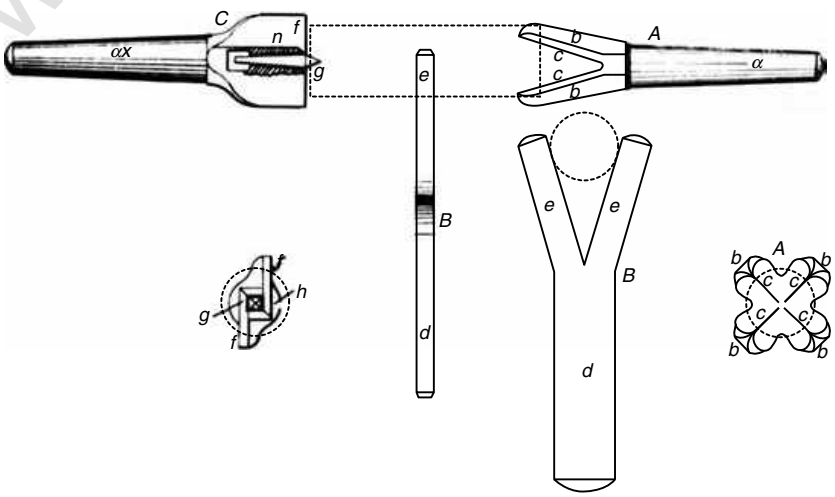


FIGURE 22.7 William's shaft centerer patent, 1863.

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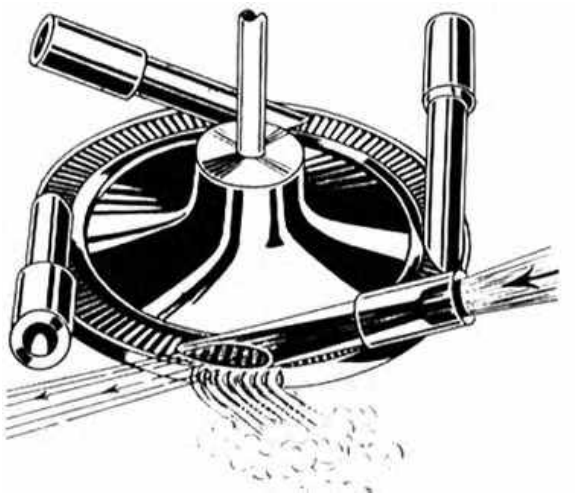


FIGURE 22.8 De Laval turbine.

turbine for cream separators (see Figure 22.8). De Laval continues improvement to the steam turbine by utilizing hyperbolic blade designs.

1880—American Society of Mechanical Engineers formed.

1881—Lucien Gulard and John Gibbs obtain English patents for "series alternating current systems of distribution." These patents were purchased by George Westinghouse in 1885.

1882—First electric direct current generating stations installed in London, England, on January 12th and in New York city on September 4th. British electrical engineer, William Ayrton invents ammeter, electrical power meter, improved voltmeters, and meters to measure self and mutual inductions.

1883—John Logan of Waltham, Massachusetts files a U.S. patent for the dial indicator as shown in Figure 22.9.

1884—Nikola Tesla begins work with Thomas Edison's company patenting the induction, synchronous, and split phase electric motors and new forms of generators and transformers. In 1892, Edison General Electric and Thomson-Houston Electric companies merged to form General Electric. American Society of Electrical Engineers formed.

1886—George Westinghouse and William Stanley (credited with perfected the transformer) first demonstrated the practicality of generating and transmitting alternating current over long distances in Great Barrington, MA. F. Hooks theorizes idea behind flexible disk coupling. British electrical engineer, Sebastian Ferranti, working at Grosvenor Gallery Co. in London also proposes using high-voltage alternating current for power transmission that would be utilized at discrete sites through step-down transformers.

1887—Heinrich Hertz of Leipzig experimentally confirms Maxwell's prediction by observing radiation emanating from an oscillating electric circuit. While a professor at Case School of Applied Science in Cleveland, OH, Albert Abraham Michelson devises the interferometer capable of measurements to one-millionths of an inch.

1889—De Laval builds a large number of steam turbines ranging in size from five to several hundred horsepower and in 1892 builds a 15 hp turbine for marine applications. British engineer, Charles Parsons forms his own company after developing the multistage steam turbine while working at Clarke, Chapman and Co. in Gateshead, England.

1893—Rudolph Diesel invents engine named after him. Sulzer of Switzerland acquires patent rights to the diesel 4 years later.

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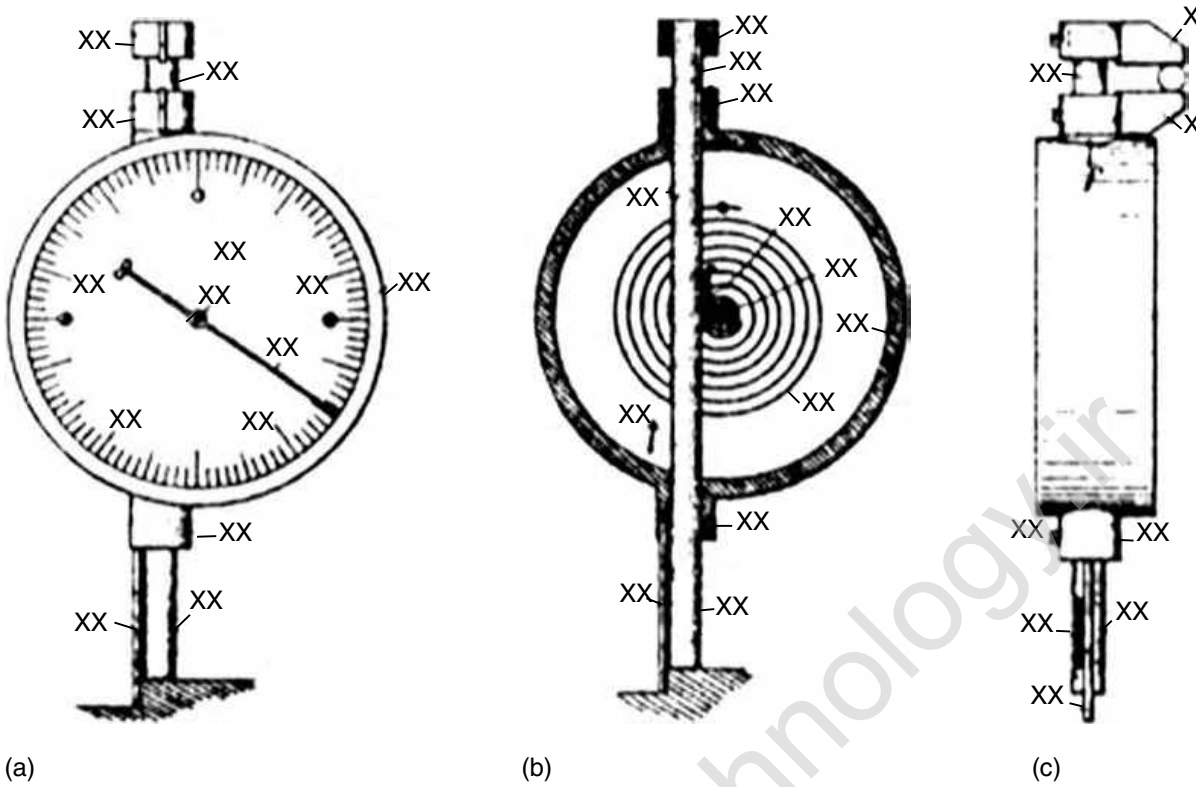


FIGURE 22.9 Logan's dial indicator patent, 1883.

1894—S. Dunkerley publishes paper "On the whirling and vibration of shafts" in *Phil. Trans*. (Vol. 185).
1897—English physicist J.J. Thompson discovers electron.
1900—Max Planck shows that if you vibrate atoms severely enough, separate bursts of radiation are emitted. These bursts of energy he termed as "quanta."
1903—R. Armengaud and C. Lemale built first successful gas turbine demonstrated for Societe des Turbomoteurs in Paris achieving efficiencies of 3%.
1904—C. Chree publishes paper "The whirling and transverse vibration of rotating shafts" in *Philadelphia Magazine* (Vol. 7). British electrical engineer, John Fleming conclusively shows that the rectifying property of a thermionic valve was still operative at radio frequencies eventually leading to the "diode valve."

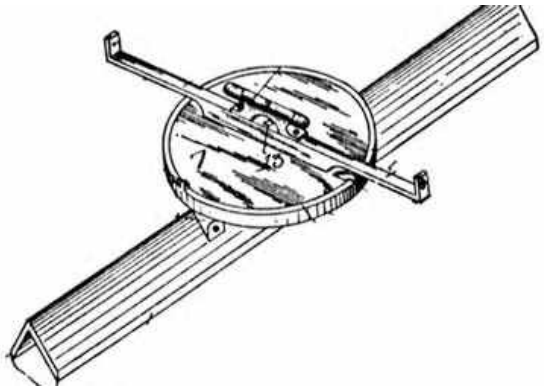


FIGURE 22.10 Campbell's shaft leveler patent, 1893.

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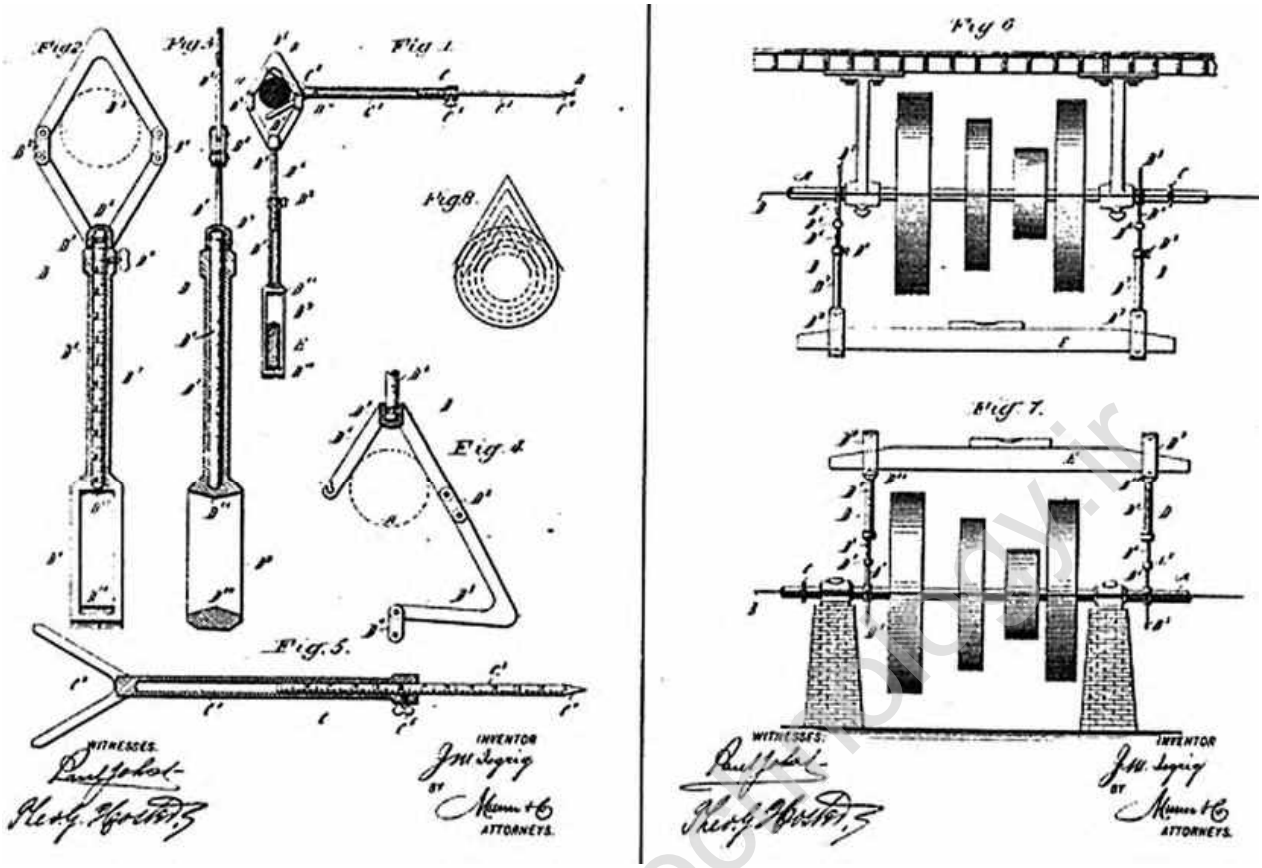


FIGURE 22.11 Isgrig's shaft leveler patent, 1894.

1907—U.S. inventor Lee DeForest patents the thermionic grid-triode vacuum tube, the forerunner of the transistor.

1908—Albert Einstein is awarded the Nobel Prize for physics where he showed that photon absorption releases electrons (the photoelectric effect).

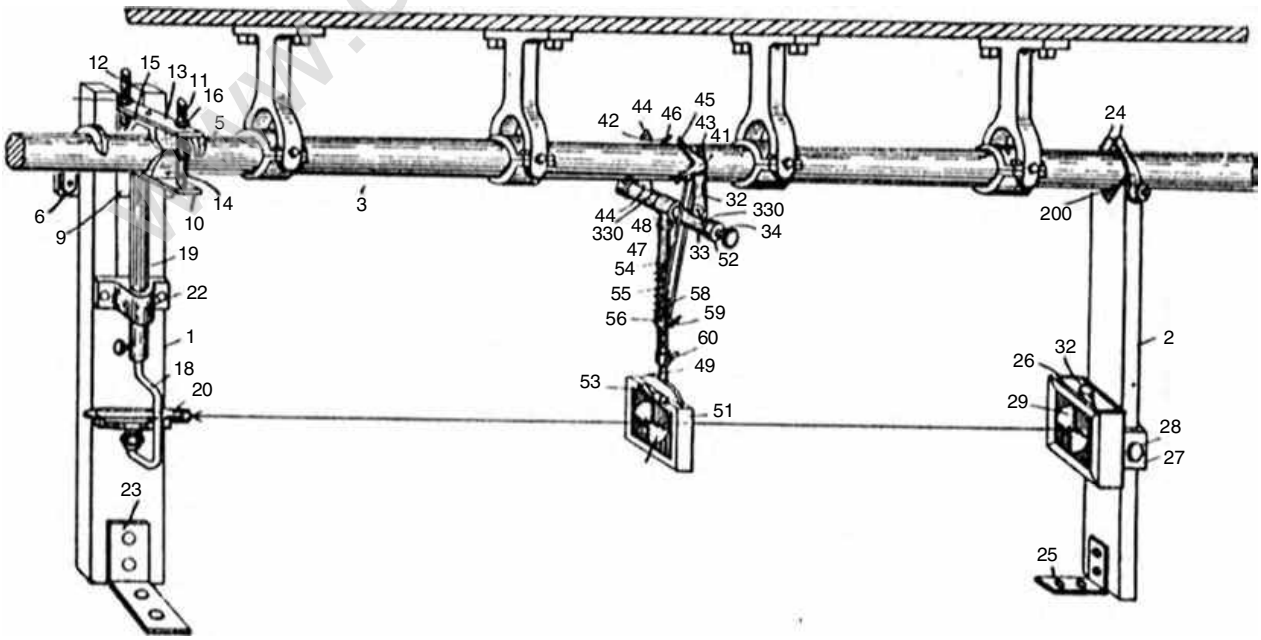


FIGURE 22.12 Kinkead's shaft leveler patent, 1901.

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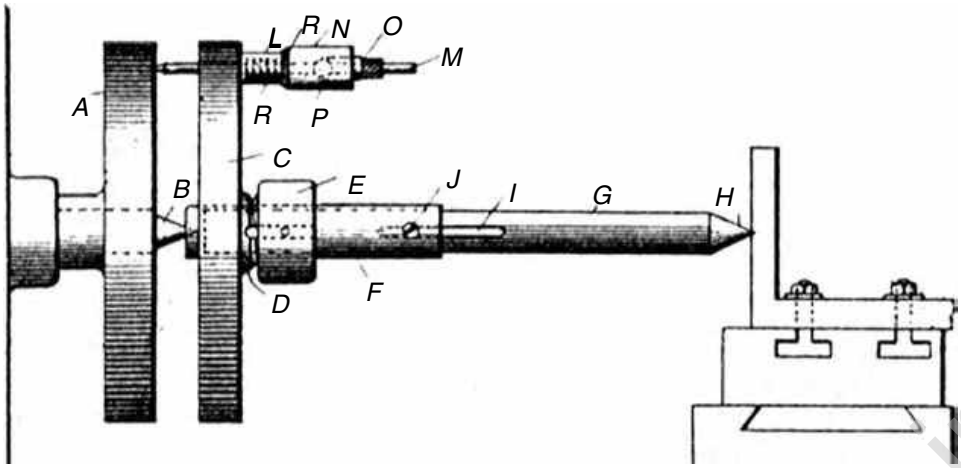


FIGURE 22.13 Miller's shaft centerer patent, 1901.

1914—*Scientific American* describes operation of chain coupling in May issue.
1916—Albert Einstein proposes the idea of stimulated emissions of radiation.
1919—H.H. Jeffcott publishes "The lateral vibration of loaded shafts in the neighborhood of a whirling speed—The effect of want of balance" in *Philadelphia Magazine* (Vol. 37, p. 304); Ernest Rutherford identifies the particles in the nucleus of an atom as positively charged particles.
1920—Austrian engineer Viktor Kaplan patents design for adjustable rotor blades on a water-driven turbine.
1924—B.L. Newkirk publishes article on "shaft whipping" describing the critical speeds of rotors.
1946—Joseph Christian files U.S. patent for shaft alignment device utilizing the face–face measurement method as shown in Figure 22.15.
1949—Robert Voss files U.S. patent for shaft alignment device utilizing the face–peripheral (rim) measurement method as shown in Figure 22.16.
1953—John Callahan files U.S. patent for shaft alignment device utilizing the double radial measurement method as shown in Figure 22.17.
1954—Charles Townes at Columbia University experiments with generated radio/micro waves by stimulating the emission of energy that was stored in ammonia molecules in a container thereby naming their device as the maser (acronym for microwave amplification by stimulated emission of radiation). This was the first demonstration of generating electromagnetic waves without using traditional devices such as flames, electric lamps, neon and fluorescent lights or natural devices such as the sun, aurora borealis, or lightning. Simultaneously, Aleksander Prohorov and Nikolai Basov in Moscow completed detailed calculations explaining the requirement for maser action to occur. All three men were awarded the Nobel Prize for their work in 1964.
1957—Gordon Gould writes proposals for visible wavelength laser (referred to as an optical maser by Townes and Schawlow) in notebooks eventually filing patents in 1977, 1979, 1987, and 1988. Gould is credited for coining the term "laser."
1958—Charles Townes and Arthur Schawlow (Bell Lab researcher) file U.S. patent for maser.
1962—The first semiconductor junction lasers were almost simultaneously demonstrated by teams at General Electric Research Laboratories, IBM Watson Research Center, and MIT Lincoln Laboratories using gallium arsenide diodes cooled to 77 K and pulsed with high current.

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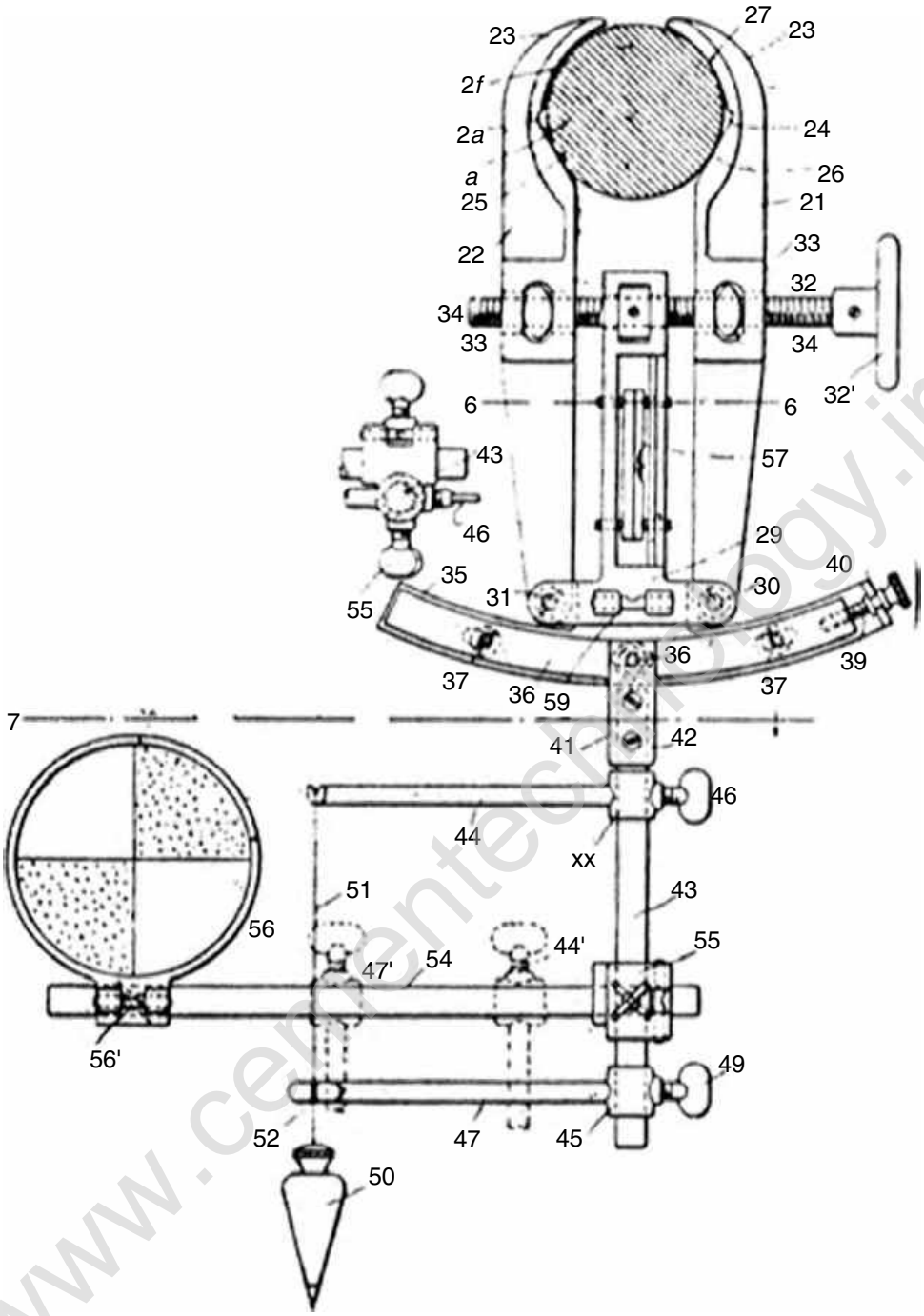


FIGURE 22.14 Ferris' shaft aligner patent, 1908.

1970—The Charge Couple Device (CCD) was proposed by W. Bell and G. Smith from Bell Laboratories as an alternative digital information storage device that would have the electronic equivalent of magnetic bubble devices.

1972—V. Ray Dodd files patent for "alignment bars" utilizing proximity probes and targets used to measure off-line to running machinery movement data. Malcolm Murray files U.S. patent for machine alignment calculator utilizing a "T" inked onto a clear piece of plastic overlaid onto a scaled grid.

1973—Malcolm Murray files U.S. patent for machinery positioner.

1976—Donald Bently files U.S. patent for shaft alignment device utilizing proximity probes for measurement (see Figure 22.18).

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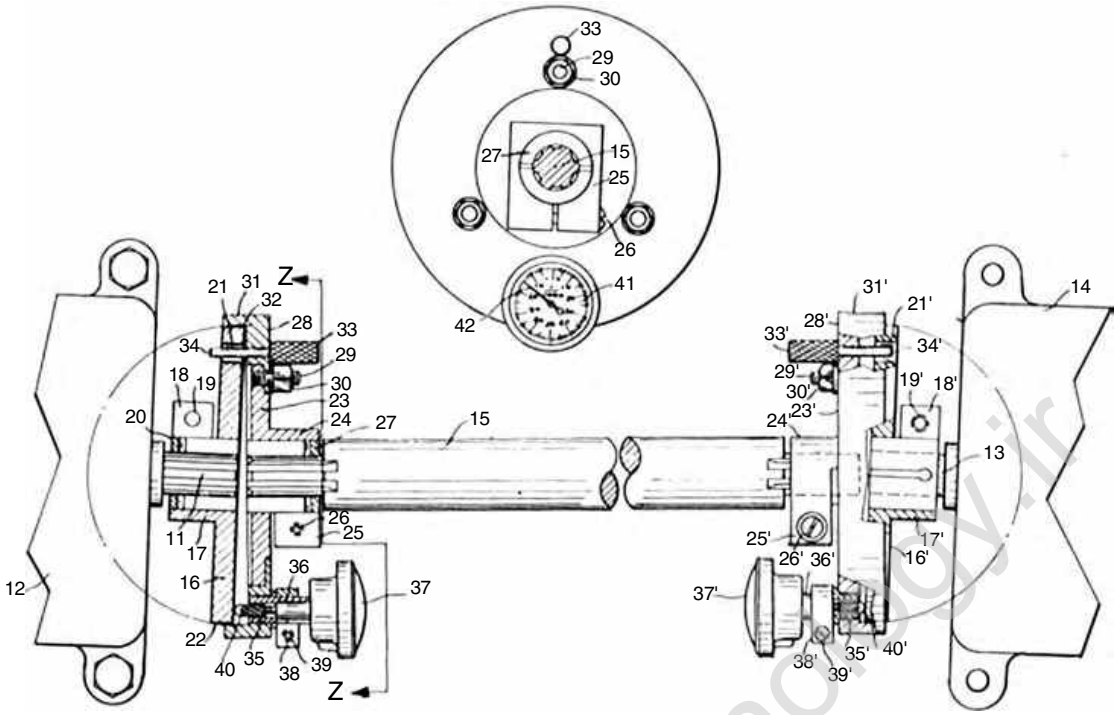


FIGURE 22.15 Christian's shaft alignment device, 1946.

1983—Malcolm Murray files U.S. patent for alignment brackets illustrating use of reverse indicator method (see Figure 22.19).
1984—John Zatazelo files U.S. patent for electronic shaft alignment calculator.
1986—John Zatazelo files U.S. patent for a variety of methods used to determine shaft alignment.
1987—Heinrich Lysen files U.S. patent for laser–detector and prism shaft alignment system. Brian Morrisey files U.S. patent for horizontal machinery positioning device.

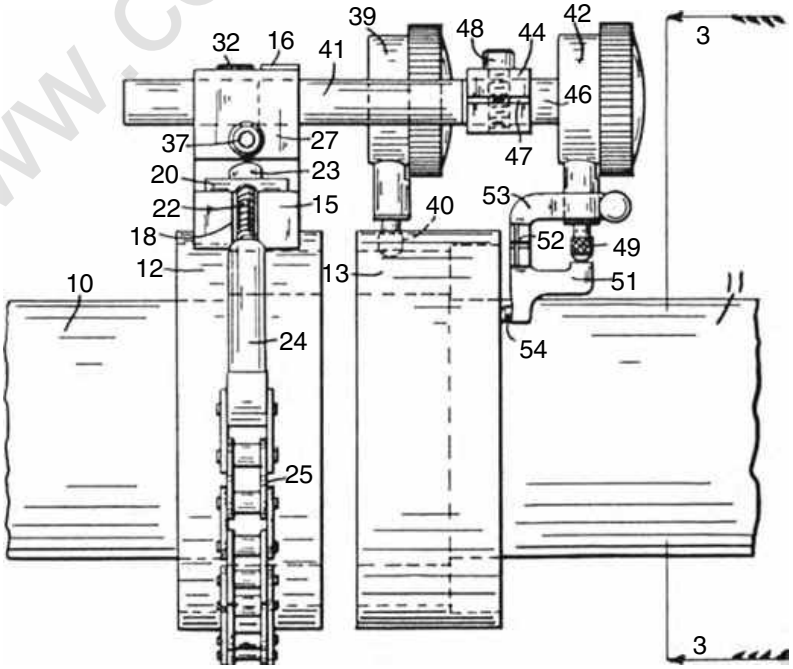


FIGURE 22.16 Voss' shaft alignment device, 1949.

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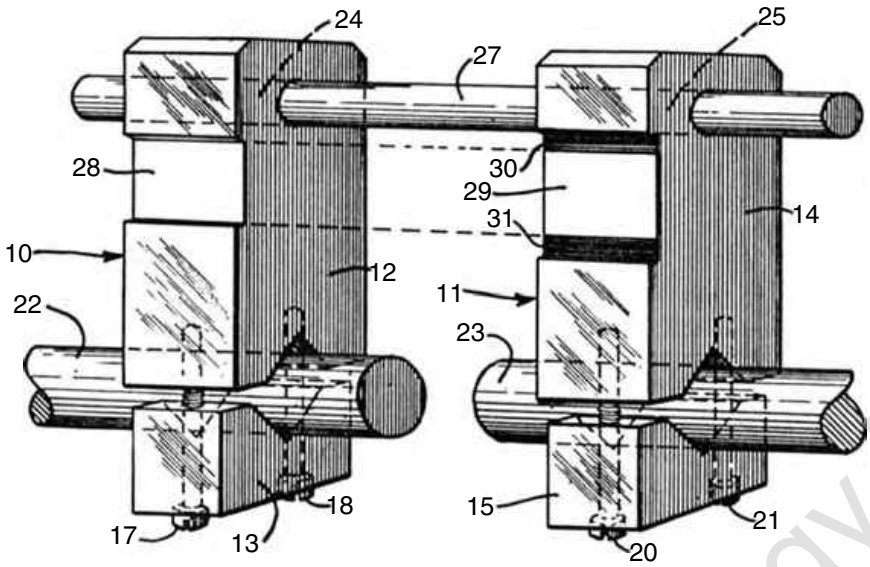


FIGURE 22.17 Callahan's shaft alignment device, 1953.

1990—Malcolm Murray files U.S. patent for vernier-strobe method used to measure off-line to running machinery movement data.

1991—Dieter Busch files U.S. patent for laser–detector–prism system used to measure off-line to running machinery movement data. Paul Saunders files U.S. patent for electronic shaft alignment system utilizing optical encoder.

There you have it. A brief run through the last 6000 years of human existence, illustrating the engineering triumphs during periods of unparalleled growth in science and technology interspersed with periods of dismal ignorance. In many cases, it is not clear who actually "invented" certain methods and devices.

The vast majority of U.S. patents for aligning and leveling machinery filed from 1860 to 1946 fell into the following categories:

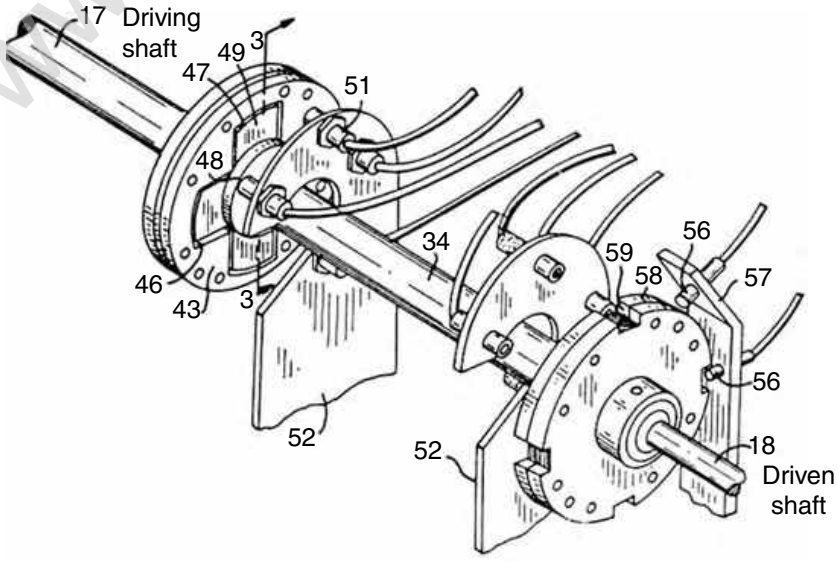


FIGURE 22.18 Bently's shaft alignment device, 1974.

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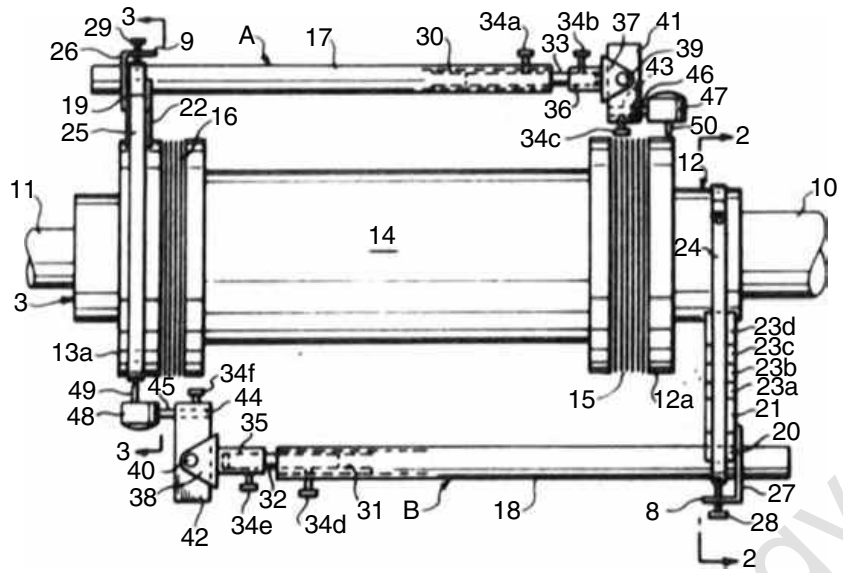


FIGURE 22.19 Murray's shaft alignment device, 1983.

- Use of spirit/dumpy levels to set the positions of shafts in line with each other
- Centering devices for work pieces on lathes
- Use of a tight wire for alignment of engine cylinder bores

The technology of shaft alignment as we recognize it today, really did not emerge until the 1940s and 1950s followed by a flurry of patent activity that persists to the present day. Between 1860 and 1950, only 11 U.S. patents were filed relating to measuring off-line shaft centerline positions, the majority of which exclusively dealt with aligning and leveling line shafts typically used in the paper industry. Over 60 years had passed since John Logan patented the dial indicator before Joseph Christian actually filed a patent using this device for alignment of rotating machinery shafts.

However, this does not necessarily mean that shaft alignment was not performed on rotating machinery during the early part of the Industrial Revolution. As we have seen throughout history, several methods and devices had been in existence for long periods of time before legal patent documents were filed claiming their originality. Regrettably, the true inventors of these mechanisms and procedures will probably never be known.

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Appendix A
Machinery Data Card

Coupling information

○ Elastomeric ○ Gear ○ Flex disc ○ Diaphragm ○ Metal ribbon ○ Chain ○ Other _____

Mfg.	Model No.	Serial No.

Shaft-to-shaft spacing in. ± in.	Bolt torque ft-lb	Wrench size in.	
Driver shaft diameter in. ± in.	Interference fit mils	Shaft taper in. ft	
Driven shaft diameter in. ± in.	Interference fit mils	Shaft taper in. ft	

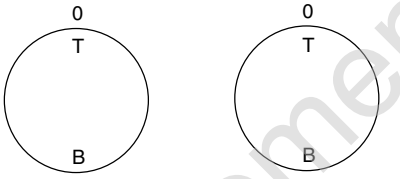
Lubrication info ○ Grease ○ Oil ○ Continuous feed oil

Viscosity or type	Amount of lube oz/pints
Recommended Mfg.	Secondary Mfg.

Shaft alignment information

○ Reverse indicator ○ Face–peripheral ○ Double radial
○ Face–face ○ Shaft to coupling spool
○ Laser ○ LVDT ○ Prox probe ○ Other _____

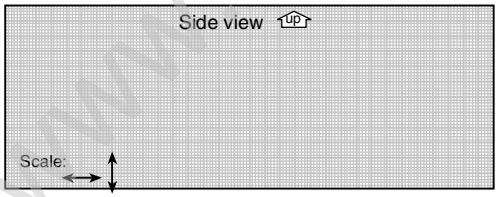
"Desired" dial indicator readings
(No bracket sag)



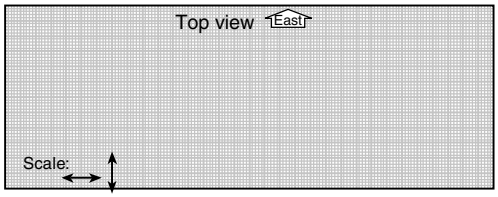
Indicate where any soft foot corrections were made, the shape and thickness of the shims or custom wedges, and indicate how the corrections are oriented under each of the machinery feet.

Top view of machinery

Desired off—Line vertical shaft positions



Desired off—Line lateral shaft positions



Maintenance history Date of installation_____

Date	Work performed

(*continued*)

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Appendix A (continued)

Machinery data card

Equipment name _____ Location _____

Equipment photo/sketch

Driver unit Information

◯ Induction motor ◯ Synchronous motor ◯ Steam turbine ◯ Gas turbine ◯ Diesel ◯ Other _____

Mfg.	Model No.	Serial No.
HP	Speed RPM	Total unit weight/rotor weight / lb
Trip speed RPM	"Critical" speed(s) RPM	Service factor

Bearing information Type of bearings: ◯ Antifriction ◯ Sliding ◯ Magnetic

Inboard (coupling) Brg.	Mfg.	Mdl./Brg. No.	Clearance
Outboard Brg.	Mfg.	Mdl./Brg. No.	Clearance
Thrust Brg.	Mfg.	Mdl./Brg. No.	Axial float

Off-line to running machinery movement

	Vertical movement	Lateral movement	
Outboard end	_____ mils ◯ Up ◯ Down	_____ mils N S E W	Foot bolt size _____ in.
Inboard end	_____ mils ◯ Up ◯ Down	_____ mils N S E W	Wrench size _____ in.

Driven unit Information

◯ Pump ◯ Gear ◯ Compressor ◯ Generator ◯ Fan ◯ Other _____

Mfg.	Model No.	Serial No.
HP	Speed RPM	Total unit weight/rotor weight / lb
Trip speed RPM	"Critical" speed(s) RPM	Service factor

Bearing information Type of bearings: ◯ Antifriction ◯ Sliding ◯ Magnetic

Inboard (coupling) Brg.	Mfg.	Mdl./Brg. No.	Clearance
Outboard Brg.	Mfg.	Mdl./Brg. No.	Clearance
Thrust Brg.	Mfg.	Mdl./Brg. No.	Axial float

Off-line to running machinery movement

	Vertical movement	Lateral movement	
Outboard end	_____ mils ◯ Up ◯ Down	_____ mils N S E W	Foot bolt size _____ in.
Inboard end	_____ mils ◯ Up ◯ Down	_____ mils N S E W	Wrench size _____ in.

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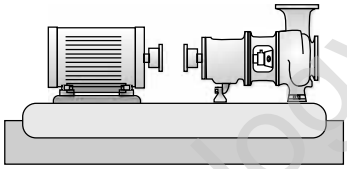
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Appendix B
Sample Preliminary Alignment Record Sheet

Preliminary alignment checks

Company: _____
Equipment name: _____
Location: _____
Equipment identification #: _____



Static piping stress checks

Date: _____

Dial indicators were located on the . . .
O Motor O Pump

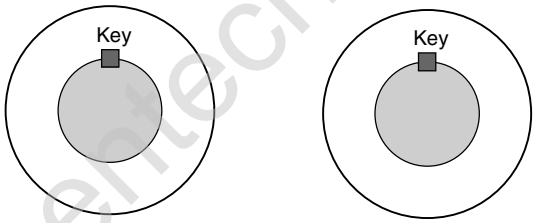
With piping attached and after loosening the pump foot bolts, the dial indicators read the following values . . .

_____ mils vertical
_____ mils lateral/horizontal

Shaft and coupling hub runout checks

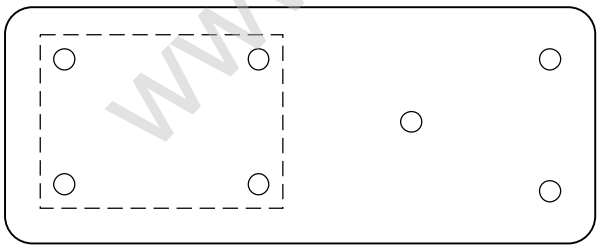
Motor shaft Pump shaft

Date: _____

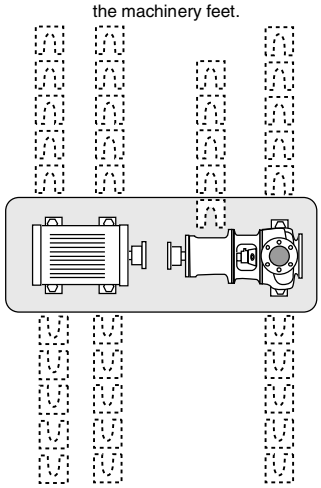


"Soft Foot" checks and corrections

Loosen or remove all foot bolts. Clean underside of machinery feet and points of contact on baseplate. Rough align both units. With the foot bolts in place but not tightened down, measure four points around each bolt hole with a set of feeler gauges and record the readings.



Indicate where any "soft foot" corrections were made, the shape and thickness of the shims or custom "wedges," and indicate how the corrections are oriented under each of the machinery feet.



Piping stress test by: _____
Runout checks by: _____
"Soft Foot" checks by: _____

Notes:
_____
_____
_____
_____

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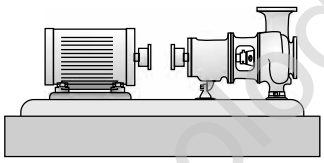
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Appendix C
Sample Installation and Shaft Alignment

Report record sheet

Installation and shaft alignment report

Company: _____
Equipment name: _____
Location: _____
Equipment identification #: _____



Static piping stress checks

Date: _____

Dial indicators were located on the . . .
O Motor O Pump

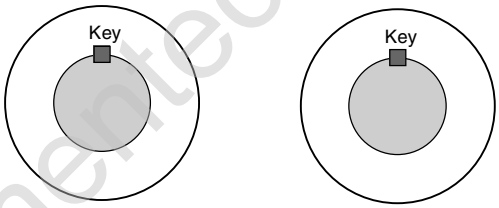
With piping attached and after loosening the pump foot bolts, the dial indicators read the following values . . .

_____ mils vertical
_____ mils lateral/horizontal

Shaft and coupling hub runout checks

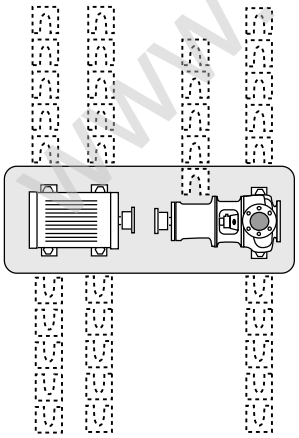
Motor shaft Pump shaft

Date: _____



"Soft Foot" checks and corrections

Indicate where any "soft foot" corrections were made, the shape and thickness of the shims or custom "wedges", and indicate how the corrections are oriented under each of the machinery feet.



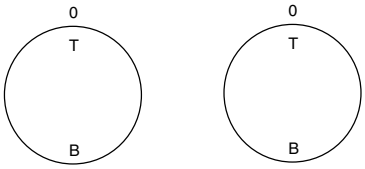
Piping stress test by: _____
Runout checks by: _____
"Soft Foot" checks by: _____
Final alignment readings by: _____

Shaft alignment information

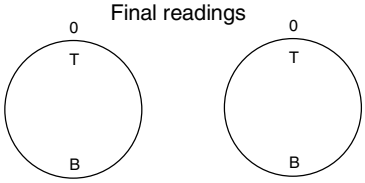
Indicate how the shaft positions were measured with the machinery off-line.

O Reverse indicator O Face–peripheral
O Shaft to Coupling spool O Face–face
O Double radial O Laser O Other _____
Amount of bracket sag _____ mils

"Desired" / "Shoot-for" Dial indicator readings



Final readings



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Appendix D
Torque Values
(SAE Grade 2 Bolts)

Recommended Torque Values for SAE Grade 2 Nuts and Bolts Unified National Coarse (UNC)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.164	32	19 in.-lb	14 in.-lb	546
0.19	24	28 in.-lb	21 in.-lb	734
0.25	20	67 in.-lb	50 in.-lb	1,313
0.3125	18	120 in.-lb	90 in.-lb	2,175
0.375	16	21 ft-lb	16 ft-lb	3,188
0.4375	14	32 ft-lb	24 ft-lb	4,388
0.5	13	52 ft-lb	39 ft-lb	5,850
0.5625	12	69 ft-lb	52 ft-lb	7,500
0.625	11	100 ft-lb	75 ft-lb	9,300
0.75	10	190 ft-lb	142 ft-lb	13,800
0.875	9	293 ft-lb	220 ft-lb	11,400
1	8	427 ft-lb	320 ft-lb	15,000
1.125	7	8,678 ft-lb	650 ft-lb	18,900
1.25	7	1,155 ft-lb	865 ft-lb	24,000
1.375	6	1,467 ft-lb	1,100 ft-lb	28,575
1.5	6	1,667 ft-lb	1,250 ft-lb	34,800

(55 kpsi proof strength–69 kpsi tensile strength for sizes 0.250 ft to 0.500 in.)
(52 kpsi proof strength–64 kpsi tensile strength for sizes 0.500 ft to 0.750 in.)
(28 kpsi proof strength–55 kpsi tensile strength for sizes 0.750 ft to 1.500 in.)

Recommended Torque Values for SAE Grade 2 Nuts and Bolts Unified National Fine (UNF)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.164	36	23 in.-lb	17 in.-lb	622
0.19	32	35 in.-lb	26 in.-lb	852
0.25	28	75 in.-lb	56 in.-lb	1,500
0.3125	24	156 in.-lb	108 in.-lb	2,400
0.375	24	23 ft-lb	18 ft-lb	3,600
0.4375	20	37 ft-lb	28 ft-lb	4,913
0.5	20	55 ft-lb	41 ft-lb	6,600
0.5625	18	80 ft-lb	60 ft-lb	8,400
0.625	18	110 ft-lb	83 ft-lb	10,575

(*continued*)

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Appendix D (continued)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.75	16	192 ft-lb	144 ft-lb	1,5375
0.875	14	184 ft-lb	138 ft-lb	1,2600
1	12	274 ft-lb	205 ft-lb	1,6425
1	14	280 ft-lb	210 ft-lb	1,6800
1.125	12	397 ft-lb	297 ft-lb	21,150
1.25	12	553 ft-lb	415 ft-lb	26,550
1.375	12	746 ft-lb	559 ft-lb	32,550
1.5	12	979 ft-lb	734 ft-lb	39,150

(55 kpsi proof strength–69 kpsi tensile strength for sizes 0.250 ft to 0.500 in.)
(52 kpsi proof strength–64 kpsi tensile strength for sizes 0.500 ft to 0.750 in.)
(28 kpsi proof strength–55 kpsi tensile strength for sizes 0.750 ft to 1.500 in.)

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Appendix E
Torque Values
(SAE Grade 5 Bolts)

Recommended Torque Values for SAE Grade 5 Nuts and Bolts Unified National Coarse (UNC)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.164	32	52 in.-lb	40 in.-lb	890
0.19	24	73 in.-lb	55 in.-lb	1,114
0.25	20	96 in.-lb	76 in.-lb	2,025
0.3125	18	204 in.-lb	156 in.-lb	3,338
0.375	16	31 ft-lb	23 ft-lb	4,950
0.4375	14	50 ft-lb	37 ft-lb	6,788
0.5	13	76 ft-lb	57 ft-lb	9,075
0.5625	12	109 ft-lb	82 ft-lb	11,625
0.625	11	150 ft-lb	112 ft-lb	14,400
0.75	10	266 ft-lb	200 ft-lb	21,300
0.875	9	430 ft-lb	322 ft-lb	29,475
1	8	644 ft-lb	483 ft-lb	38,625
1.125	7	794 ft-lb	596 ft-lb	42,375
1.25	7	1,120 ft-lb	840 ft-lb	53,775
1.375	6	1,470 ft-lb	1,100 ft-lb	64,125
1.5	6	1,950 ft-lb	1,462 ft-lb	78,000

(85 kpsi proof strength–120 kpsi tensile strength for sizes 0.250 ft to 0.750 in.)
(78 kpsi proof strength–115 kpsi tensile strength for sizes 0.750 ft to 1.000 in.)
(74 kpsi proof strength–105.5 kpsi tensile strength for sizes 1.000 ft to 1.500 in.)

Recommended Torque Values for SAE Grade 5 Nuts and Bolts Unified National Fine (UNF)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.164	36	53 in.-lb	40 in.-lb	1,023
0.19	32	66 in.-lb	49 in.-lb	1,279
0.25	28	120 in.-lb	87 in.-lb	2,325
0.3125	24	228 in.-lb	168 in.-lb	3,675
0.375	24	35 ft-lb	26 ft-lb	5,588
0.4375	20	55 ft-lb	41 ft-lb	7,575
0.5	20	85 ft-lb	64 ft-lb	10,200
0.5625	18	122 ft-lb	91 ft-lb	12,975
0.625	18	170 ft-lb	128 ft-lb	16,350

(*continued*)

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Appendix E (continued)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.75	16	297 ft-lb	223 ft-lb	23,775
0.875	14	474 ft-lb	355 ft-lb	32,475
1	12	705 ft-lb	529 ft-lb	42,300
1	14	721 ft-lb	541 ft-lb	32,275
1.125	12	890 ft-lb	668 ft-lb	47,475
1.25	12	1,241 ft-lb	930 ft-lb	59,550
1.375	12	1,672 ft-lb	1,254 ft-lb	72,975
1.5	12	2,194 ft-lb	1,645 ft-lb	87,750

(85 kpsi proof strength–120 kpsi tensile strength for sizes 0.250 ft to 0.750 in.)
(78 kpsi proof strength–115 kpsi tensile strength for sizes 0.750 ft to 1.000 in.)
(74 kpsi proof strength–105.5 kpsi tensile strength for sizes 1.000 ft to 1.500 in.)

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Appendix F
Torque Values
(SAE Grade 8 Bolts)

Recommended Torque Values for SAE Grade 8 Nuts and Bolts Unified National Coarse (UNC)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.138	32	22 in.-lb	16 in.-lb	815
0.164	32	41 in.-lb	31 in.-lb	1,254
0.19	24	59 in.-lb	44 in.-lb	1,568
0.25	20	143 in.-lb	107 in.-lb	2,850
0.3125	18	24 ft-lb	18 ft-lb	4,725
0.375	16	43 ft-lb	32 ft-lb	6,975
0.4375	14	69 ft-lb	52 ft-lb	9,600
0.5	13	106 ft-lb	79 ft-lb	12,750
0.5625	12	153 ft-lb	115 ft-lb	16,350
0.625	11	211 ft-lb	158 ft-lb	20,325
0.75	10	376 ft-lb	282 ft-lb	30,075
0.875	9	606 ft-lb	454 ft-lb	41,550
1	8	908 ft-lb	681 ft-lb	54,525
1.125	7	1.288 ft-lb	966 ft-lb	68,700
1.125	7	1.288 ft-lb	966 ft-lb	68,700
1.25	7	1.817 ft-lb	1362 ft-lb	87,225
1.375	6	2.381 ft-lb	1786 ft-lb	1,03950
1.5	6	3.161 ft-lb	2371 ft-lb	1,26450

(120 kpsi proof strength–150 kpsi tensile strength for sizes 0.250 ft to 1.500 in.)

Recommended Torque Values for SAE Grade 8 Nuts and Bolts Unified National Fine (UNF)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.138	40	25 in.-lb	18 in.-lb	910
0.164	36	43 in.-lb	32 in.-lb	1,321
0.19	32	68 in.-lb	51 in.-lb	1,792
0.25	28	163 in.-lb	122 in.-lb	3,263
0.3125	24	27 ft-lb	20 ft-lb	5,113
0.375	24	49 ft-lb	37 ft-lb	7,875
0.4375	20	77 ft-lb	58 ft-lb	10,650
0.5	20	119 ft-lb	89 ft-lb	14,400
0.5625	18	171 ft-lb	128 ft-lb	18,300

(*continued*)

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Appendix F (continued)

Bolt Size (in.)	Threads per Inch	Torque (Dry)	Torque (Lubricated) (75% of Dry Torque)	Clamp Load (lbf +/−25%)
0.625	18	239 ft-lb	179 ft-lb	2,3025
0.75	16	419 ft-lb	314 ft-lb	3,3600
0.875	14	668 ft-lb	501 ft-lb	4,5825
1	12	994 ft-lb	745 ft-lb	5,9700
1	14	1,019 ft-lb	764 ft-lb	6,1125
1.125	12	1,444 ft-lb	1,083 ft-lb	7,7025
1.25	12	2,011 ft-lb	1,508 ft-lb	9,6600
1.375	12	2,711 ft-lb	2,033 ft-lb	11,8350
1.5	12	3,557 ft-lb	2,667 ft-lb	14,2275

(120 kpsi proof strength–150 kpsi tensile strength for sizes. 0.250 ft to 1.500 in.)

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Appendix G
Shaft Alignment and Related U.S. Patents

US Patent #	Held by	Month	Day	Year	Description
39608	Williams	August	18	1863	Shaft centerer
283627	John Logan	August	21	1883	Improvement to gauges (dial indicator)
431054	Henderson	July	1	1890	Pipe clamping device
458055	George Hunt	December	30	1890	Gage for aligning engines
487427	Poole	December	6	1892	Level hanger for shafting
521306	Humphrey Campbell	June	26	1893	Shaft setting device
541754	Isgrig	June	25	1895	Line shaft alignment
575857	Sly	January	26	1897	Lantern bracket
651024	Thomas	June	5	1900	Aligning and leveling instrument for shafting
679591	James Barns	March	27	1901	Centering device for lining up engines
685288	Miller	October	29	1901	Indicator for lathes
685290	John C. Miller	March	9	1901	Aligning work piece to CL of lathe spindle
685455	Kinkead	October	29	1901	Instrument for hanging and lining up shafting
807085	Newton	December	12	1905	Shaft aligner
868074	Ernest Clark	April	13	1906	Shaft liner and leveler
958736	Edgar Ferris	August	26	1908	Shaft aligning device
1221507	Buesse	April	3	1917	Pipe clamp base
1231479	Blumer	June	26	1917	Roll locating device
1295936	Spellman	March	4	1919	Measuring instrument
1339384	Douglas	May	11	1920	Gage
1351663	Koch	August	31	1920	Measurement gage comparator
1477257	Lewis Fritz	July	8	1922	Shaft alignment gauge
1505313	Alvah J. Colwell	June	16	1923	Crank shaft throw parallel gauge
1516288	Frank Godfrey	April	5	1924	Bevel protractor
1559230	William Eccles	January	22	1921	Aligning meter for gears and shafts
1591485	Albert Guillet	August	25	1925	Lining block and measure for leveling machinery
1616084	Albert Guillet	April	1	1926	Leveling and lining spinning frames
1799739	John Elering, et al.	November	13	1926	Precision measuring device
1907959	Albert Guillet	April	7	1928	Lining and leveling means for machinery
2395393	Arche Brilliantine	June	3	1944	Electric alignment micrometer
2402567	Milner	October			airfoils
2451720	Davis	October	19	1948	Centering device
2461143	Clifford	February	8	1949	Gaging device/shaft bracket
2499753	Hubbard	May	2	1946	Chain tightening and securing device
2516854	Joseph Christian	January	16	1946	Gauging apparatus for aligning shafts
2624944	Pujda, A.G.	January	13	1953	Layout measuring tool
2634939	Robert Voss	July	2	1949	Shaft aligning mechanism
2638676	Callahan	May			alignment device
2656607	Harding	October			device
2692527	Wetzel, T.A.	October	26	1954	Optical aligning apparatus for machine tools
2703505	Senn, J.A.	March	8	1955	Apparatus for aligning machine elements

(*continued*)

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Appendix G (continued)

US Patent #	Held by	Month	Day	Year	Description
2726058	Luther Foltz	March	22	1954	Shaft align bracket
2833051	Cunningham	May			bracket
2923202	Trimble, R.H.	August	21	1948	Dual field optical system
2929922	Townes, Schawlow	March	22	1960	Masers and maser communications system
3163037	Kawabata, J.S.	December	29	1964	Dynamic torque meter
3176403	Meyer	April			(no sag)
3187439	Leach, R.I.	April	28	1961	Alignment gage
3192631	Goguen, D.J.	June	29	1962	Shaft alignment apparatus
3244392	Sheets	April			bracket
3279086	Schlitt, et al.	October	18	1966	Compensated gyroscopic directional reference
3525158	Torlay	August			bracket
3551057	Hamilton, et al.	December	29	1970	Laser beam alignment apparatus
3578281	Nielsen, Turner	May	11	1971	Precision alignment fixture
3604121	Harold Hull	June	7	1968	Roll alignment
3631604	Schenavar, Stanley	January	4	1972	Shaft align bracket
3664029	Glucoft, Westerfield	May			Shaft align bracket
3723013	Stirland, et al.	March	27	1973	Alignment system
3733706	Arthur Blohm	July	15	1970	Machine aligning device
3783522	V. Ray Dodd	April	4	1972	Method & apparatus for shaft alignment
3789507	Malcolm Murray	February	17	1972	Machine element alignment system
3816000	Fiedler, George C.	June	11	1974	Three axis alignment means
3849857	Malcolm Murray	July	5	1973	Machine element alignment positioner
4033042	Donald E. Bently	October	10	1974	Shaft alignment apparatus and method
3901604	Louis Butler	August	26	1975	Wheel alignment
4053845	Gordon Gould	October	11	1977	laser
4060719	Daltnowski	July	23	1976	Geometric calculator
4102052	Heinz Bloch	December	14	1976	Deflection indicator for couplings
4115925	Malak, Stephen	September	26	1978	Shaft aligner
4161068	McMaster, T.M.	July	17	1979	Apparatus and method for aligning shafts
4161436	Gordon Gould	July	17	1979	Laser
4215482	Richard Szewczyk	April	1	1978	Workpiece centering device for lathe
4216587	Willice Stone	January	29	1979	Shaft align bracket
4231161	Flavio Belfiore	April	18	1979	Pulley alignment tool
4234924	LaVance, et al.	September	11	1978	Baseline measure for electronic positioning
4244111	James Heard	April	24	1979	Shaft align bracket
4249294	Belfiore, Flavio	February	10	1981	Pulley alignment device
4283688	Lloyd, et al.	August	11	1981	Laser autoalignment system
4367594	Murray	January			brackets
4413514	Stovall, David T.	November	8	1983	Shaft alignment tool
4428126	Burke Banks	December	21	1981	Continuous monitoring shaft positions
4439925	Brian Lock	March	3	1982	Concentricity measuring
4447962	Joseph Grosberg	May	24	1982	Adjustable bore target/gauge
4451992	Malak, Stephen	September	30	1982	Shaft positioning device/method
4463438	Zatezalo, John	July	31	1984	Shaft alignment calculator
4502233	Gary Boitz, et al.	July	7	1983	Shaft align apparatus and method
4516328	Massey, Charles	May	14	1985	Shaft align brackets
4518855	Malak, Stephen	May	21	1985	Shaft alignment bracket
4586264	John Zatezalo, et al.	November			Alignment methods
4623979	John Zatezalo, et al.	January	31	1984	Shaft alignment calculator

(*continued*)

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Appendix G (continued)

US Patent #	Held by	Month	Day	Year	Description
4698491	Lysen, Heinrich	May	29	1984	Laser alignment system
4704583	Gordon Gould	November	4	1987	Laser
4712953	Witzel, Weisgerber	December	15	1987	Method and apparatus for aligning cutting tools
4746021	Gordon Gould	May	24	1988	Laser
4709485	Bowman, Donald E.	December	1	1987	Device and method for OL2R movement
4764010	Bachmann, Doker	August	16	1988	Method for aligning brackets
4790507	Brian Morrissey	May	18	1987	Tool for precise movement of machinery
4928401	Malcolm Murray	May	29	1990	Vernier strobe system
4964224	Lawrence Jackson	July	18	1989	Shaft align bracket
4984173	Imam, Carlson	January	8	1991	System for aligning a rotating line shaft
4991965	Dieter Busch	May	11	1988	Laser monitoring machinery movement
5026998	Roland Holzl	April	4	1990	Alignment methods
5056237	Paul R. Saunders	July	2	1990	Electronic shaft alignment device
5077905	Malcolm Murray	January	7	1992	PIBZLT system
5185937	K. Piety, D. Nower	February	16	1993	Alignment bracket
5263261	K. Piety, D. Nower	November	23	1993	Shaft alignment device
5430539	Lysen, Heinrich	July	4	1995	Checking alignment of body axes for parallelism
5435073	Sullivan, David L.	July	25	1995	Alignment tool for rotating equipment
5450245	Grotzinger, Yager	September	12	1995	Laser alignment apparatus
5508609	Parkinson, James R.	April	16	1996	Detecting axial position and alignment of a shaft
5684578	Nower, Gaddis	November	4	1997	Laser alignment head for use in shaft alignment
5691523	Calvo, Frank A.	November	25	1997	Machinery shaft alignment calculator
5715609	Nower, Daniel L.	February	10	1998	Stationary shaft alignment apparatus
5734108	Walker	March	31	1998	Sensing shaft displacement and strain
5896672	Harris, G. Danny	April	27	1999	Precision shaft alignment system
5980094	Nower, Daniel L.	November	9	1999	Analysis of alignment data
5987762	Toth, Deneszczuk	November	23	1999	Pulley alignment device
6034763	Slater, Kramer	March	7	2000	Dual beam laser device for alignment
6040903	Lysen, Heinrich	March	21	2000	Determining the relative position of two bodies
6046799	Lysen, Heinrich	April	4	2000	Device for misalignments of two shafts
6049378	Busch, et al.	April	11	2000	Device and method for mutually aligning bodies
6098297	Belfiore, F.C.	August	8	2000	Pulley alignment
6374507	Lehto, Rick A.	April	23	2002	Pulley alignment
6411375	Hinkle, et al.	June	25	2002	Shaft alignment methodologies
6434841	Stahl, Israel, et al.	August	20	2002	Pulley alignment
6434849	Hermann, Michael	August	20	2002	Lateral and/or angular offset
6615904	Lindgen, Hangkan	May	22	2002	Detecting a misaligned roller portion of a roller
6739923	Murakami, Masatoshi	January	9	2003	Engine alignment jig assembly
6763597	Lysen, Heinrich	July	20	2004	Parallel alignment method
6782766	Parkinson, James R.	September	13	2002	Torque, axial position, and alignment of shaft
6784986	Lysen, Heinrich	August	31	2004	Device for alignment of machine shafts
6915582	Engels, Geoffrey P.	July	12	2005	Alignment structure

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Appendix H
Shaft Alignment Training Questionnaire

If you decide that you would like to have an outside contractor conduct shaft alignment training courses for you and the personnel in your company, this questionnaire should help in assessing the type of training courses that are offered, the content of the courses, the qualifications of the instructors, and the costs associated for the training.

1. What are the name(s) or title(s) of your training course(s), and the duration of the course(s)?

Course title _____
Hours of training _____ h.

Course title _____
Hours of training _____ h.

Course title _____
Hours of training _____ h.

2. Do you offer in-house shaft alignment training courses at your facility? If so, what is the course fee(s) per student?

____ No
____ Yes
Course title _____
Fee per student _____ (U.S. dollars)

____ Yes
Course title _____
Fee per student _____ (U.S. dollars)

____ Yes
Course title _____
Fee per student _____ (U.S. dollars)

3. Do you conduct shaft alignment training courses at a customer's facility or plant site? If so, what is the course fee(s), do you charge per student or per course, and what is the minimum and maximum number of attendees allowed in the course?

____ No

____ Yes
Course title _____
Fee per student charge _____ (U.S. dollars)
Fee per course charge _____ (U.S. dollars)
____ minimum number of attendees
____ maximum number of attendees

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____ Yes
Course title _____
Fee per student charge _____ (U.S. dollars)
Fee per course charge _____ (U.S. dollars)
____ minimum number of attendees
____ maximum number of attendees

____ Yes
Course title _____
Fee per student charge _____ (U.S. dollars)
Fee per course charge _____ (U.S. dollars)
____ minimum number of attendees
____ maximum number of attendees

4. Do you provide classroom material for each student? If so, describe the training material (e.g., books, course notes, CD, etc.) that each student receives in the course.
____ No

____ Yes
Course title _____
Course material _____

____ Yes
Course title _____
Course material _____

____ Yes
Course title _____
Course material _____

5. Can the classroom material be purchased without attending the course? If so, what is the cost of the classroom material?
_____ (U.S. dollars)

6. Do the attendees perform hands-on work on alignment training simulators in the course? If so, indicate the number of students per training simulator.
____ no training simulators used
____ 1 student per training simulator
____ 2 students per training simulator
____ 3 students per training simulator
____ 4–6 students per training simulator
____ 1 training simulator for entire class
____ 1 training simulator for instructor only
____ Others, describe _____

7. Can a training simulator be purchased? If so, what is the cost of the training simulator?
_____ (U.S. dollars)

8. Please indicate what topics are taught in your course(s) and whether it is covered in a lecture (LCTR), and/or a real life example or case history (XMPL), and/or if it is practiced with the hands-on training simulators (LAB):

LCTR	XMPL	LAB	Content
___	___	___	Symptoms of misaligned rotating machinery
___	___	___	Definition of shaft misalignment
___	___	___	Alignment tolerance guidelines

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___ ___ ___ Difference between alignment and flexible coupling tolerances
___ ___ ___ Design principles of flexible and rigid couplings
___ ___ ___ Finding and correcting coupling problems
___ ___ ___ Design principles of bearings
___ ___ ___ Static and dynamic forces in machinery
___ ___ ___ Using vibration analysis to detect misalignment
___ ___ ___ Using infrared thermography to detect misalignment
___ ___ ___ Step-by-step procedure for the alignment process
___ ___ ___ How much time each step of the alignment process requires
___ ___ ___ How to measure runout on mechanical couplings and machinery shafts
___ ___ ___ What causes excessive runout conditions
___ ___ ___ Finding and correcting soft foot conditions
___ ___ ___ Finding and correcting excessive piping strain
___ ___ ___ Face–rim alignment method
___ ___ ___ Reverse indicator alignment method
___ ___ ___ Double radial alignment method
___ ___ ___ Shaft to coupling spool alignment method
___ ___ ___ Face–face alignment method
___ ___ ___ Measure and compensate for bracket/bar sag
___ ___ ___ Validity rule for alignment measurements
___ ___ ___ Mathematical corrections for face–rim
___ ___ ___ Mathematical corrections for reverse indicator
___ ___ ___ Mathematical corrections for double radial
___ ___ ___ Mathematical corrections for shaft to coupling spool
___ ___ ___ Mathematical corrections for face–face
___ ___ ___ Alignment graphing/modeling for face–rim
___ ___ ___ Alignment graphing/modeling for reverse indicator
___ ___ ___ Alignment graphing/modeling for double radial
___ ___ ___ Alignment graphing/modeling for shaft to coupling spool
___ ___ ___ Alignment graphing/modeling for face–face
___ ___ ___ Basic operating principle of all laser alignment systems
___ ___ ___ Basic operating principle for a specific laser alignment system
___ ___ ___ Aligning multiple element drive trains
___ ___ ___ Aligning right angled drives
___ ___ ___ Aligning vertically oriented shafts
___ ___ ___ Off-line to running (OL2R) machinery movement basics (aka "hot" and "cold" alignment)
___ ___ ___ Calculated machine case thermal expansion method
___ ___ ___ Inside micrometer-tooling ball-angle measurement methods
___ ___ ___ Proximity probes with water-cooled stands method
___ ___ ___ Optical alignment tooling method
___ ___ ___ Alignment bars with proximity probes method
___ ___ ___ Laser–detector systems method
___ ___ ___ Rod-tubing connector system method
___ ___ ___ Vernier-strobe method
___ ___ ___ Other OL2R machinery movement methods
___ ___ ___ Compensating for OL2R machinery movement
___ ___ ___ Other topics _____
___ ___ ___ Other topics _____
___ ___ ___ Other topics _____
___ ___ ___ Other topics _____
___ ___ ___ Other topics _____
___ ___ ___ Other topics _____
___ ___ ___ Other topics _____

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9. Do you offer or provide testing or exams for your courses? If so, what is the cost of the exams?

Exam description _____
Exam cost _____ (U.S. dollars)

Exam description _____
Exam cost _____ (U.S. dollars)

Exam description _____
Exam cost _____ (U.S. dollars)

Exam description _____
Exam cost _____ (U.S. dollars)

10. Are your tests or exams recognized by a professional affiliation or organizations? If so, what are the professional affiliation or organizations?

Professional affiliation or organizations:

_____
_____
_____
_____
_____

11. Please indicate the number of years of experience of the training instructor(s), their previous job functions and time in that job, and educational experience.

Instructors name _____
Training experience _____ years
Alignment field experience _____ years

Number of Years	Job Description
_____	Plant manager
_____	Engineering manager
_____	Maintenance manager
_____	Mechanical engineer
_____	Electrical engineer
_____	Civil engineer
_____	Industrial engineer
_____	Other engineers, _____
_____	Front line supervisor/foreman/planner
_____	Technician
_____	Mechanic
_____	Millwright
_____	Pipefitter
_____	Electrician
_____	Electronic repair/instrumentation
_____	Others (please list) _____

Formal Training

_____ K–12 (elementary education)
_____ Trade school
_____ Junior college
_____ Bachelor's degree
_____ Master's degree

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_____ Doctoral degree
Instructors name _____
Training experience _____ years
Alignment field experience _____ years

Number of Years	Job Description
_____	Plant manager
_____	Engineering manager
_____	Maintenance manager
_____	Mechanical engineer
_____	Electrical engineer
_____	Civil engineer
_____	Industrial engineer
_____	Other engineers, _____
_____	Front line supervisor/foreman/planner
_____	Technician
_____	Mechanic
_____	Millwright
_____	Pipefitter
_____	Electrician
_____	Electronic repair/instrumentation
_____	Other (please list) _____

Formal Training
_____ K–12 (elementary education)
_____ trade school
_____ junior college
_____ Bachelor's degree
_____ Master's degree
_____ Doctoral degree

12. Additional information
_____
_____
_____
_____
_____

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Appendix I
Shaft Alignment Services Questionnaire

If you decide that you would like to have an outside contractor perform shaft alignment services for you, this questionnaire should help in assessing the qualifications of the company or personnel who will be performing the services.

1. Indicate what geographical area your alignment services cover (cities, states, countries, etc.).

_____
_____
_____
_____
_____
_____
_____

2. Indicate what types of industrial rotating machinery you have worked on and for what period of time.

Number of Years	Machinery Type
_____	Synchronous motors
_____	Induction motors
_____	Steam turbines
_____	Gas turbines
_____	Water turbines
_____	Internal combustion engines
_____	Drivers up to 50 hp
_____	Drivers from 50 to 200 hp
_____	Drivers from 200 to 1000 hp
_____	Drivers from 1000 to 5000 hp
_____	Drivers from 5000+ hp
_____	Single stage centrifugal pumps
_____	Multiple stage centrifugal pumps
_____	Centrifugal air compressors
_____	Reciprocating air compressors
_____	Screw or sliding vane air compressors
_____	Gearboxes
_____	Fans
_____	Blowers
_____	AC generators
_____	DC generators
_____	Cooling tower fan drives

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_____ Multiple element drive trains
_____ Other machinery (please list)
_____
_____
_____

3. Please fill out the following form for each employee performing shaft alignment in your organization. If there are several employees who perform these functions, please attach their information to this questionnaire.

Part A. Job description

Indicate the number of years of experience of the service personnel, their current job function(s), time in that job, and educational experience. If they perform more than one function, please indicate each one.

Service personnel name _____
Alignment field experience _____ years

Number of Years	Job Description
_____	Company president or director
_____	Department manager
_____	Engineering manager
_____	Service manager
_____	Mechanical engineer
_____	Electrical engineer
_____	Civil engineer
_____	Industrial engineer
_____	Other engineers, _____
_____	Front line supervisor/foreman/planner
_____	Technician
_____	Mechanic
_____	Millwright
_____	Pipefitter
_____	Electrician
_____	Electronic repair/instrumentation
_____	Others (please list) _____

Part B. Education

_____ K–12 (elementary education)
_____ Trade school
_____ Junior college
_____ Bachelor's degree
_____ Master's degree
_____ Doctoral degree

Part C. Alignment training

Indicate what training you have received in the following shaft alignment methods, procedures, or tasks. Formal training means they attended a training course, on the job training (OTJT) means you where shown this information by a coworker or you were self taught.

Formal	OTJT	Shaft Alignment Method(s), Procedures, Tasks
_____	_____	Foundation and baseplate/soleplate installation and repair
_____	_____	Soft foot detection and correction
_____	_____	Excessive runout detection and correction

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_____	_____	Excessive piping strain detection and correction
_____	_____	Straightedge, eyeball, feeler gauge
_____	_____	Face–rim indicator
_____	_____	Reverse indicator
_____	_____	Shaft to coupling spool
_____	_____	Double radial
_____	_____	Face–face
_____	_____	Laser/detector system(s), model(s)_____
_____	_____	Calculating "growth" using the thermal strain equation
_____	_____	Inside micrometer/tooling ball/angle measurement devices
_____	_____	Proximity probes with water-cooled stands
_____	_____	Optical alignment tooling
_____	_____	Alignment bars and proximity probes (aka "Dodd"/Dynalign bars)
_____	_____	Laser/detector dystem(s), model(s)_____
_____	_____	Ball-rod-tubing connector system
_____	_____	Vernier-strobe method
_____	_____	Instrumented coupling system
_____	_____	Shaft alignment software programs
_____	_____	Others (please list) _____

Part D. Shaft alignment field experience

Indicate what off-line shaft alignment methods you have performed and for what period of time.

Number of Years	Shaft Alignment Method(s) You Have Used
_____	None
_____	Eyeball
_____	Straightedge, eyeball, feeler gauge
_____	Face–rim indicator
_____	Reverse indicator
_____	Shaft to coupling spool
_____	Double radial
_____	Face–face
_____	Optical alignment tooling
_____	Laser/detector system(s), model(s) _____
_____	Others (please list)_____
_____	
_____	
_____	

Indicate what off-line to running (OL2R) machinery movement measurement and alignment compensation methods (aka "hot" and "cold" alignment) you have performed and for what period of time.

Number of Years	OL2R Machinery Movement Measurement Method(s) You Have Used
_____	Calculating machine case thermal expansion (thermal strain equation)
_____	Inside micrometer/tooling ball/angle measurement devices
_____	Proximity probes with water-cooled stands
_____	Optical alignment tooling
_____	Alignment bars and proximity probes (aka "Dodd" bars/Dynalign system)
_____	Laser/detector system(s), model(s) _____

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_____ Ball-rod-tubing connector system
_____ Vernier-strobe method
_____ Instrumented coupling system
_____ Others (please list)_____
_____
_____

Part E. Additional talents and capabilities
Indicate any additional tasks you have performed and for what period of time.

Number of Years	Skills or Tasks
_____	Vibration analysis or monitoring
_____	Shop balancing
_____	Field balancing
_____	Infrared thermography surveys
_____	Oil analysis
_____	Motor current signature analysis
_____	Motor testing
_____	Root cause failure analysis (RCFA)
_____	Machinery overhaul and repair
_____	Sliding bearing inspection, replacement, or repair
_____	Rolling element bearing inspection or replacement
_____	Rigging
_____	Foundation installation (i.e., concrete, re-bar, forms, isolation, etc.)
_____	Basic machine tool operation (e.g., lathe, mill, grinder)
_____	Gas welding and cutting
_____	MIG welding
_____	TIG welding
_____	Pipefitting
_____	Hand tools (e.g., wrenches, ratchets, etc.)
_____	Basic measuring tools (micrometers, dial indicators, slide calipers, etc.)
_____	Power tools (e.g., drills, grinders, saws, air tools, etc.)
_____	Fork truck operation
_____	Crane operation
_____	Lubrication
_____	Flexible or rigid coupling inspection or replacement
_____	Oil seal inspection or replacement
_____	Mechanical packing inspection or replacement
_____	Mechanical seal inspection or replacement
_____	Pneumatic instrumentation
_____	Electronic instrumentation
_____	Hydraulics
_____	Basic computer skills
_____	Word processing software
_____	Database or spreadsheet software
_____	Presentation software
_____	CAD
_____	Illustration/graphics software
_____	Desktop publishing software
_____	Digital photography

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_____ Digital video
_____ Local area network installation and maintenance
_____ Internet and web browsers
_____ Software programming languages (e.g., C, Pascal, Basic, Fortran, etc.)
_____ Other skills (please list)
_____
_____
_____
_____

Part F. Certification or accreditation

Indicate any certification or achievements related to your work. Please show what level of achievement (e.g., Levels 1, 2, 3, etc.) was received, and the organization or institution the achievement was received from (e.g., ISO, ANSI, ASME, Vibration Institute, JATC, college, trade school, company, etc.).

Level	Organization	Subject or Skill
_____	_____	Shaft alignment
_____	_____	Optical alignment
_____	_____	Vibration analysis
_____	_____	Balancing
_____	_____	Infrared thermography
_____	_____	Oil analysis
_____	_____	Motor current signature analysis
_____	_____	Motor testing
_____	_____	Root cause failure analysis (RCFA)
_____	_____	Machinery overhaul and repair
_____	_____	Rigging
_____	_____	Machine tool operation
_____	_____	Gas welding and cutting
_____	_____	MIG welding
_____	_____	TIG welding
_____	_____	Pipefitting
_____	_____	Fork truck operation
_____	_____	Crane operation
_____	_____	Lubrication
_____	_____	Pneumatic instrumentation
_____	_____	Electronic instrumentation
_____	_____	Hydraulics
_____	_____	Basic computer skills
_____	_____	Word processing software
_____	_____	Database or spreadsheet software
_____	_____	CAD
_____	_____	Software programming languages
_____	_____	Others _____
_____	_____	Others _____
_____	_____	Others _____
_____	_____	Others _____

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Appendix J
Alignment Internet Web Sites

Company or Organization	Web Site
Accushim Inc.	http://www.accushim.com/v180.html
Alignment Services Inc.	http://www.alignmentservicesinc.com/
Alignment Supplies Inc.	http://www.alignmentsupplies.com/
Automated Precision	http://www.apisensor.com/
Brunson Instrument Co.	http://www.brunson.us/
Aline Mfg. Inc.	http://alinemfg.com/
Damalini	http://www.damalini.com/
Donaldson Garrett & Associates Inc.	http://www.dg-a.com/
Dreyco Mechanical Services Inc.	http://www.dreyco-mechanical.com/index.asp
Fixturlaser	http://www.fixturlaser.com/
James L. Griffin Co. Inc.	http://www.jlgriffin.com/
Hamar Laser	http://www.hamarlaser.com/
Indikon Co. Inc.	http://www.indikon.com/
Kara Co.	http://www.karaco.com/
Laser Tools Co. Inc.	http://www.lasertoolsco.com/
Ludeca	http://www.ludeca.com/
Newman Tools Inc.	http://www.newmantools.com/shaft.htm
New Standard Institute	http://www.newstandardinstitute.com/index.cfm
Oasis Alignment Services	http://www.oasisalignment.com/
Oxford Engineering Co.	http://www.oxfordengineeringco.com/
Peterson Alignment Tools co.	http://www.petersontools.com/
Pinpoint Laser Systems	http://www.pinlaser.com/
Pruftechnik	http://www.pruftechnik.com/index__.htm
R&T Factors Inc.	http://rtfactors.com/
Schaeffer Precision Alignment Inc.	http://www.schaefferprecision.com/
SPM Instrument	http://www.spminstrument.com/
Turbine Tools	http://www.turbinetools.com/
Turvac Inc.	http://www.turvac.com/
Unisorb	http://www.unisorb.com/
Universal Technologies Inc.	http://www.unitechinc.com/
Update International	http://www.update-intl.com/
Vibralign	http://www.vibralign.com/

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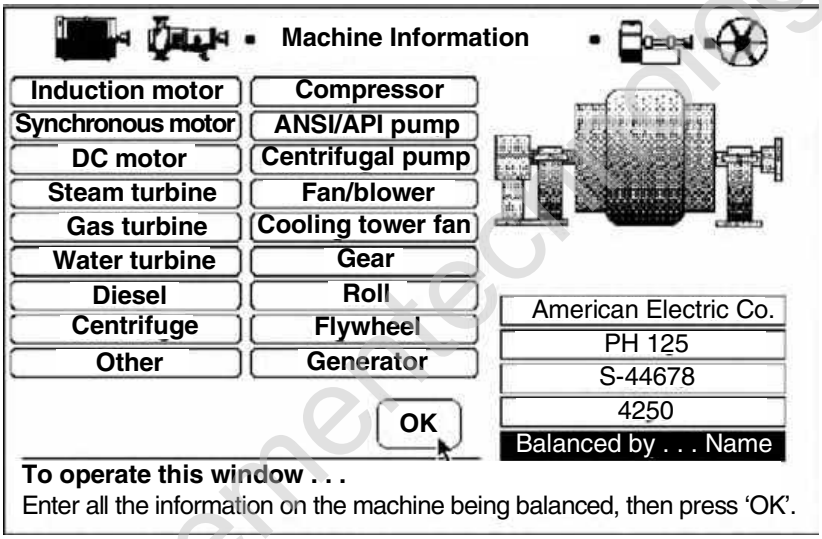
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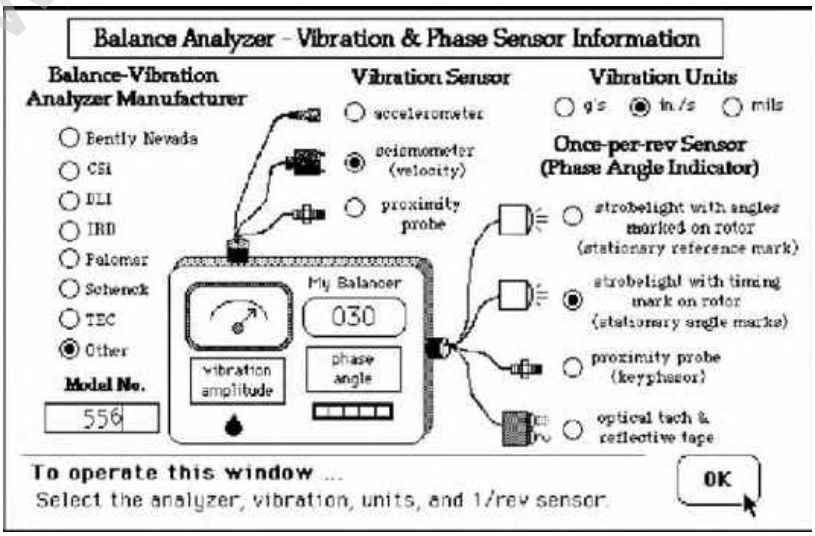
Appendix K
Single Plane Balancing

Single plane balancing procedure

1. Gather some information on the type of machine your are going to be balancing as shown below. If you ever need to balance this machine again, you can go back and review all of this information to reduce the amount of time required for re-balancing.



2. Record the information on the balance analyzer, type of vibration sensor used, vibration engineering units, and phase angle measuring device used. This is critical information for future balance runs.

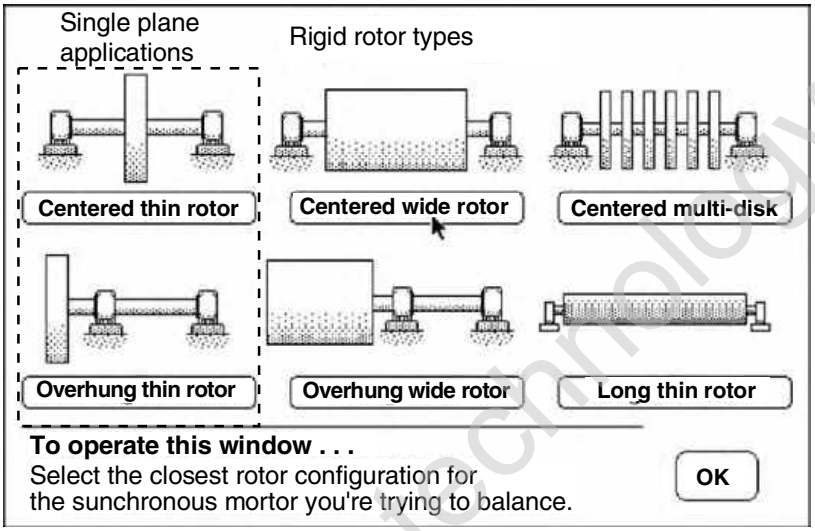


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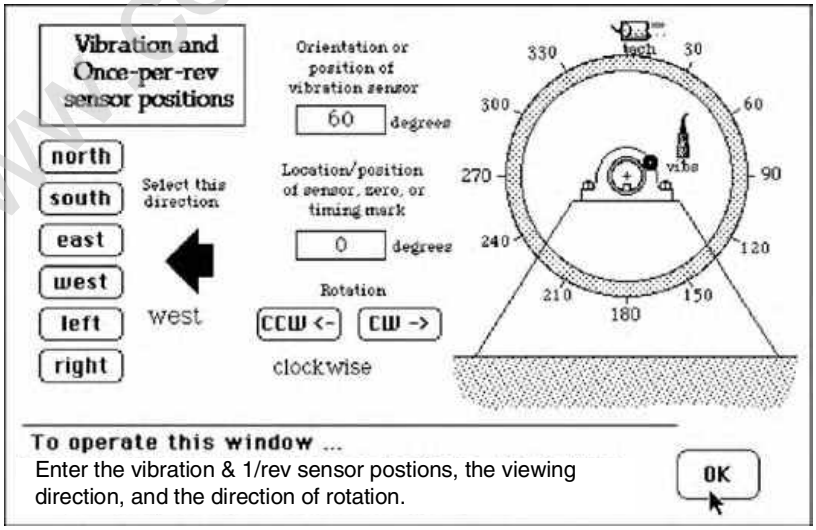
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Single plane balancing procedure

3. Usually, single plane balancing can be performed on the types of rotors shown inside the dashed box below. The other rotor types shown below could be single plane balanced but usually require two plane balancing.



4. Record the information on the placement/location of the vibration sensor and the phase angle measuring device. This is critical information for future balance runs.

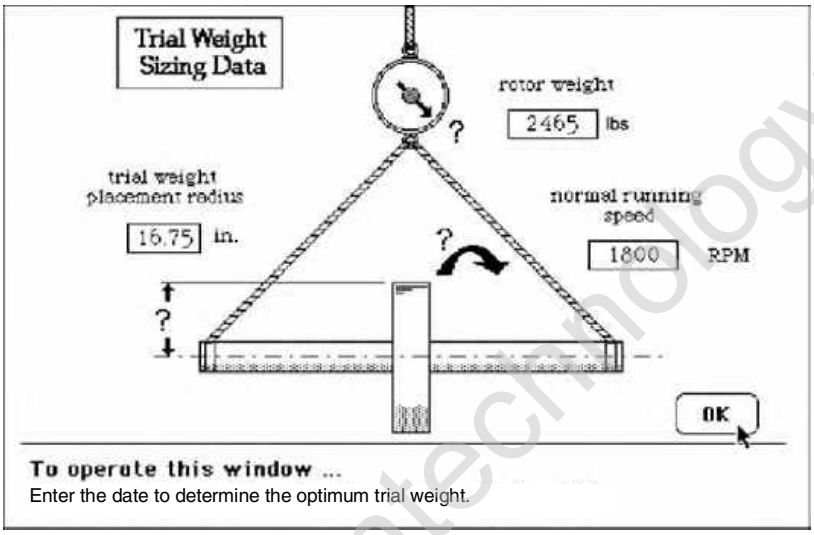


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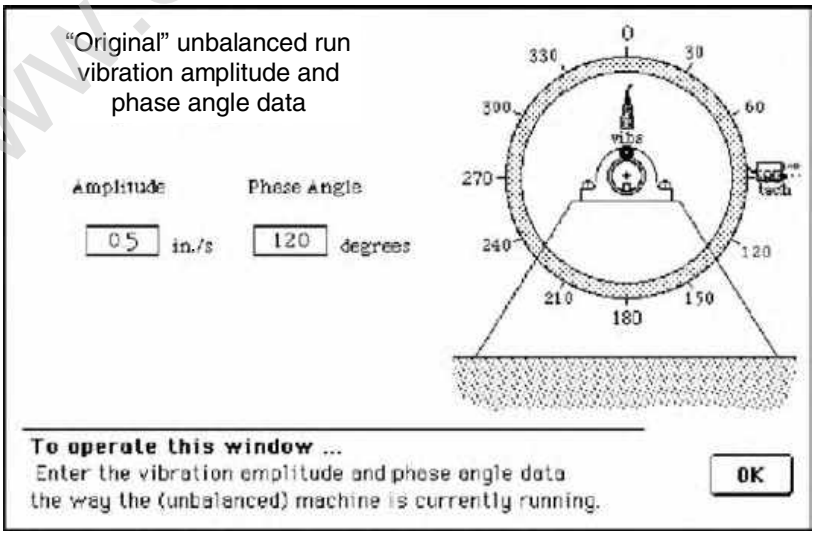
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Single plane balancing procedure

5. Gather some information on the rotor weight, the normal operating speed of the rotor, and where you are going to be installing the trial and/or correction weights. This is critical information for calculating the right amount of trial weight so you get a good response without damaging the machine trying to balance it.



6. Operate the rotor at the balancing speed, and with your analyzer filter tuned to the rotating speed of the rotor (i.e., 1 x RPM). Proceed to measure and record the original unbalance amplitude and phase data. This will be called the “Original” or “O” vector.



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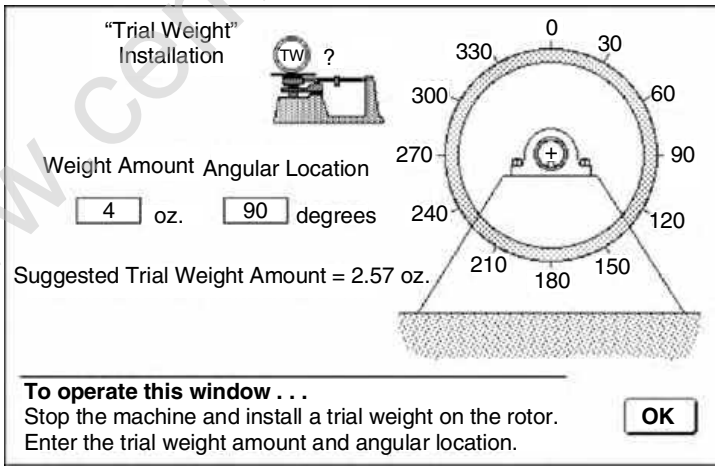
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Single plane balancing procedure

7. Stop the rotor and add a trial weight to the part. The trial weight should produce a force equal to 10% of the static weight of the rotor on one bearing. Record the amount of the trial weight (in ounces or grams) and the distance from the centerline of rotation (in inches or centimeters). Insure that the trial weight is firmly attached to the rotor.

$$\text{Trial weight} = \frac{F}{K \times R \times N^2}$$

where:
Trial weight = ounces in English system or grams in metric system
F = 10% of the static rotor weight (pounds in English system or kilograms in Metric system)
R = radius of the trial weight from the centerline of rotation (inches in English system or centimeters in Metric system)
N = rotor speed/1000 (RPM/1000)
K = 1.77 (English system) or 0.011 (Metric system)



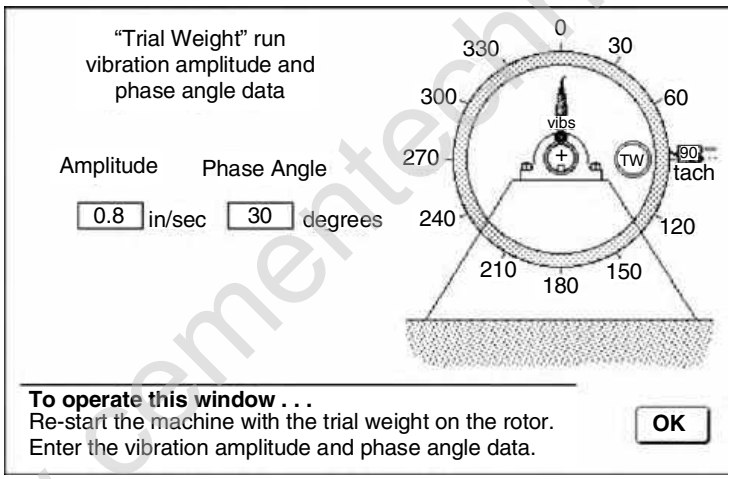
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Single plane balancing procedure

8. Restart the machine and operate the rotor at the balancing speed. Observe and record the new unbalance amplitude and phase data. This will be called the “original plus trial weight” vector (O+T).



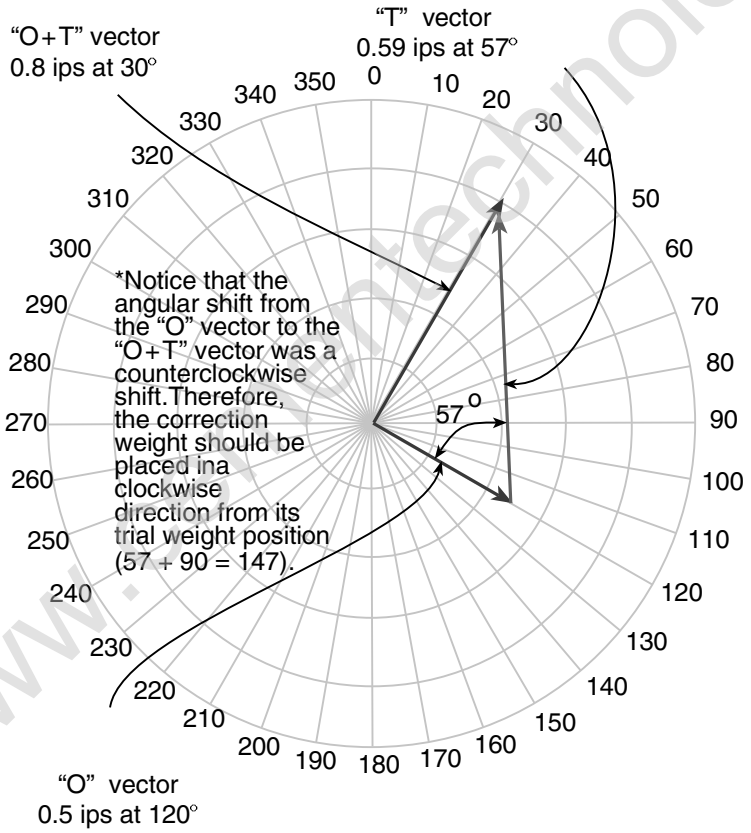
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Single plane balancing procedure

9. On a sheet of polar graph paper, plot the "original run vector" (called the "O" vector, the "original plus trial weight vector" (called the "O+T" vector). Construct the "trial weight effect" vector (aka the "T" vector) by connecting the ends of the "original" and "trial weight" vectors. The "T" vector should point from the "O" vector to the "O+T" vector.



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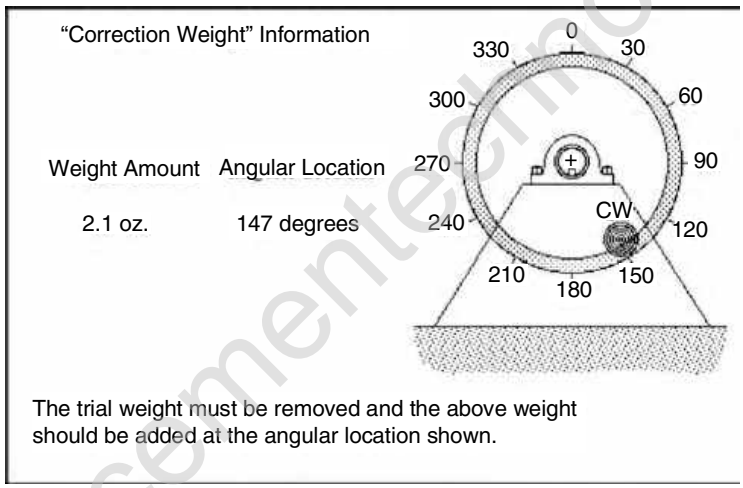
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Single plane balancing procedure

10. Measure the length of the "trial weight effect" vector and use the formula to determine the correct balance weight needed

$$\text{Correction weight} = \text{trial weight} \times \frac{\text{"original" vector amplitude}}{\text{"trial weight effect" vector amplitude}}$$

11. Using a protractor, measure the included angle between the "O" and "T" vectors. This will be called the "correction" angle.
12. Mark the spot where the trial weight is located and remove the trial weight. Install the correction weight at an angular amount equal to the "correction" angle from the point where the trial weight was located but in a direction opposite of the phase shift from the "O" vector to the "O+T" vector. Make sure the correction weight is installed at the same radius from the centerline of rotation as the trial weight.



13. Run the rotor again and record the vibration and phase angle data. If everything went OK, the rotor should now be balanced. If additional "trim balancing" is required,use this latest amplitude and phase data as a new "O+T" vector and plot it on a new polar graph paper along with the original "O" vector. Draw a new "T" vector and re-calculate the new correction weight. Repeat as often as necessary.

14. In the future, if you place the vibration and phase angle sensors in the same place, all you need do is measure and record the amplitude and phase angle data, plot it on a new piece of graph paper as the "O+T" vector along with the "O" vecor, draw a new "T" vector, plug it into the correction weight formula above and you have the new correction weight you need. Good luck and great balancing!

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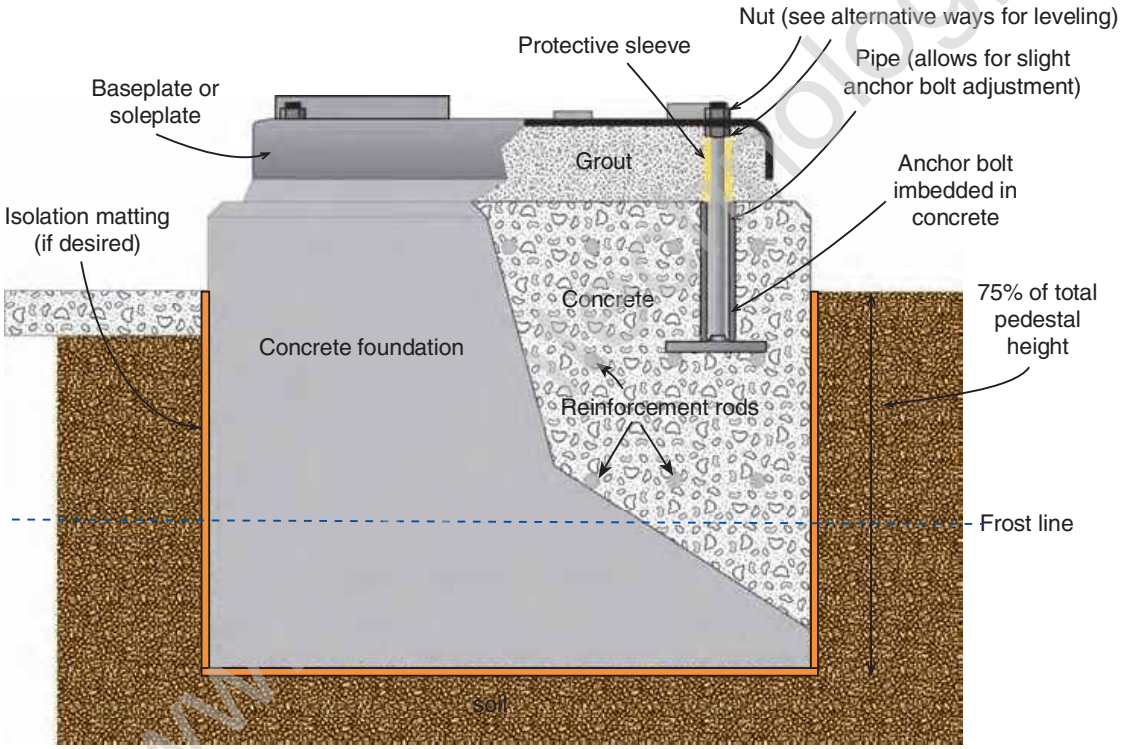


FIGURE 3.12 Section view of a typical rigid foundation.

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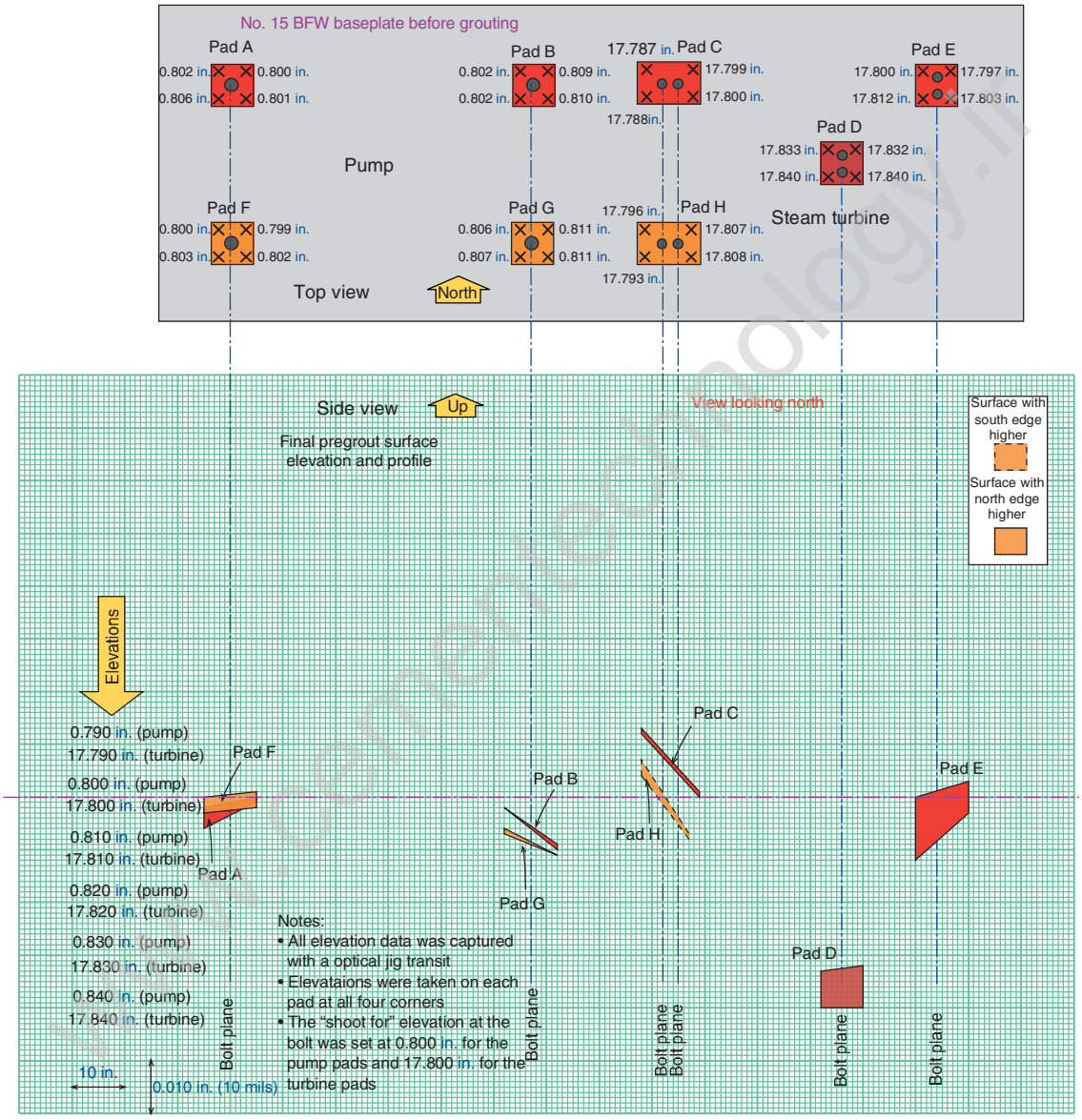


FIGURE 3.45 Baseplate elevation profile prior to grouting.

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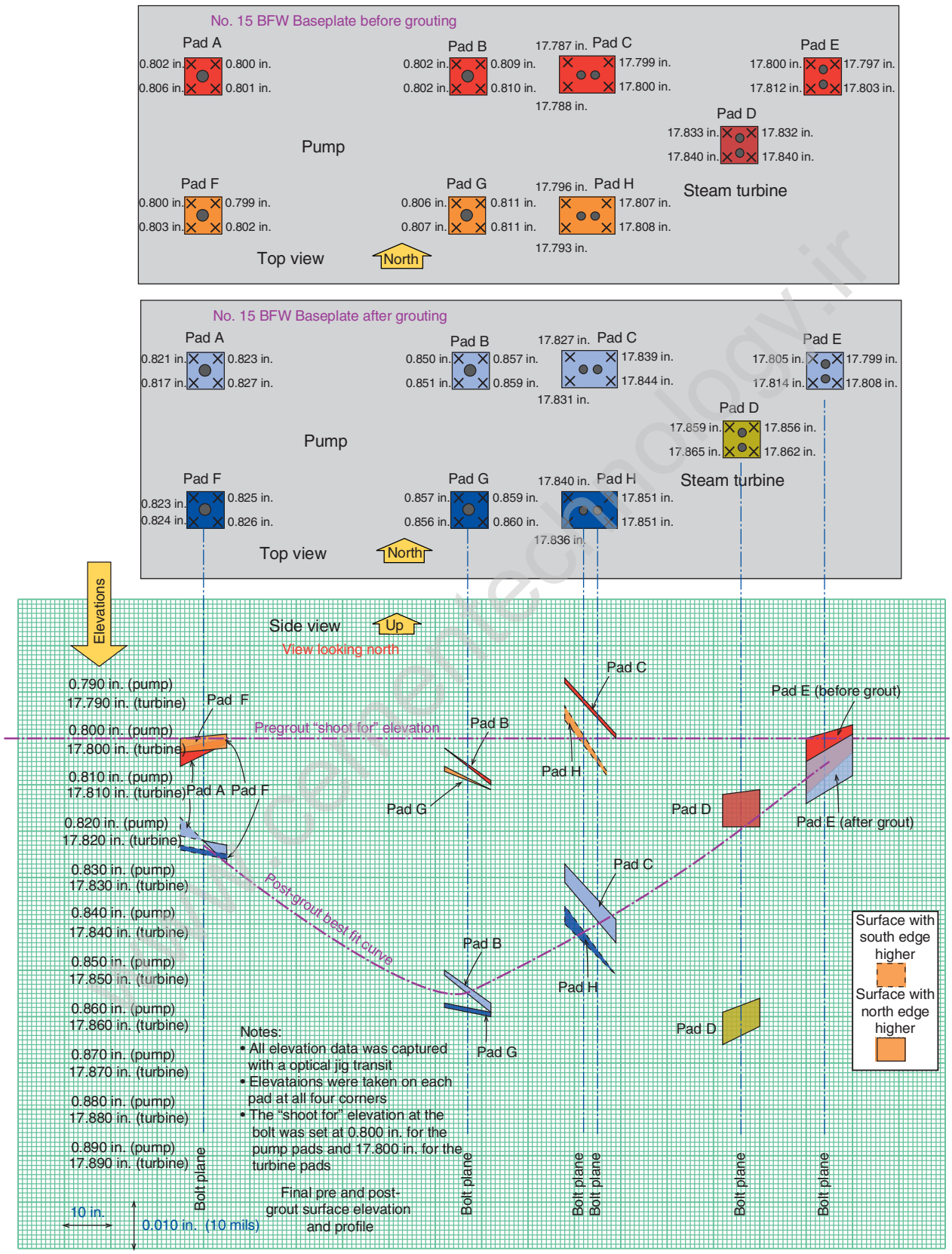


FIGURE 3.62 Elevation data and profiles before and after grout pour.

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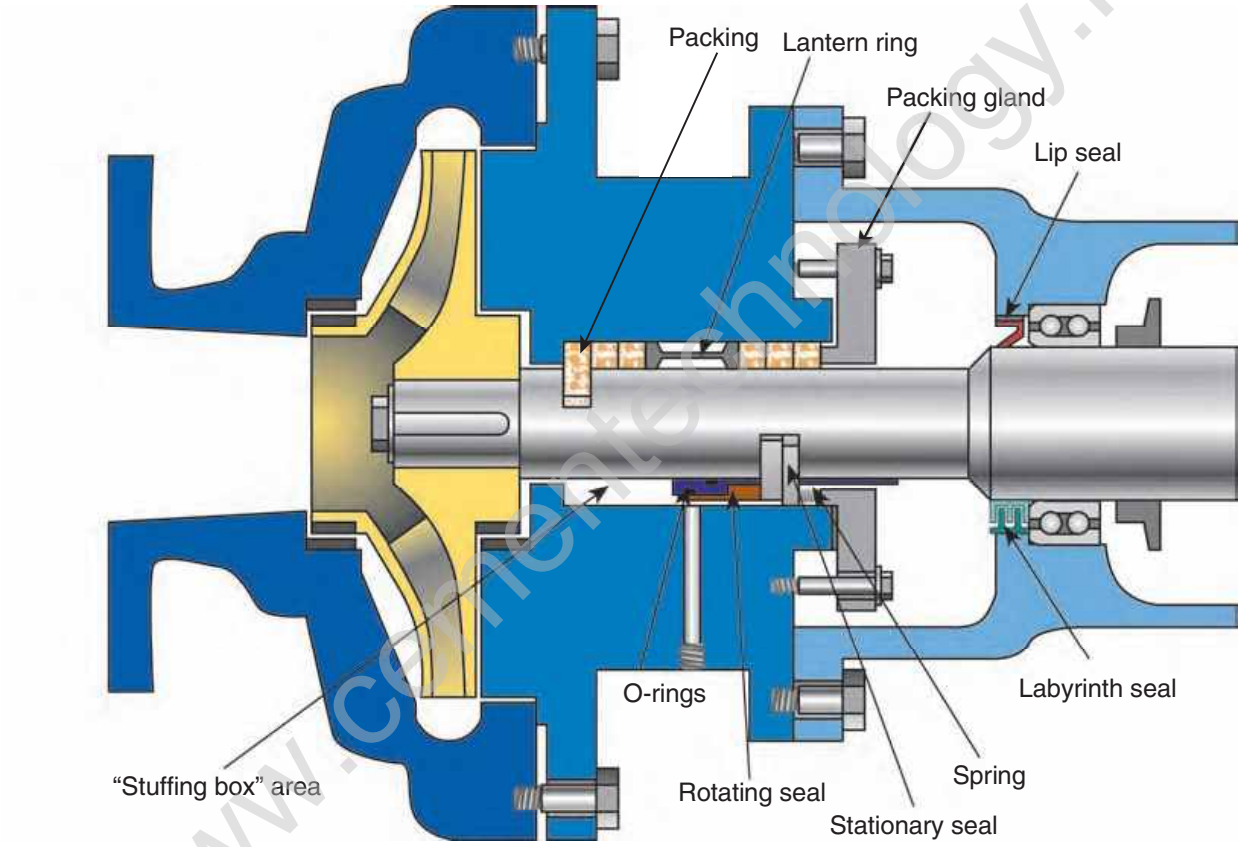
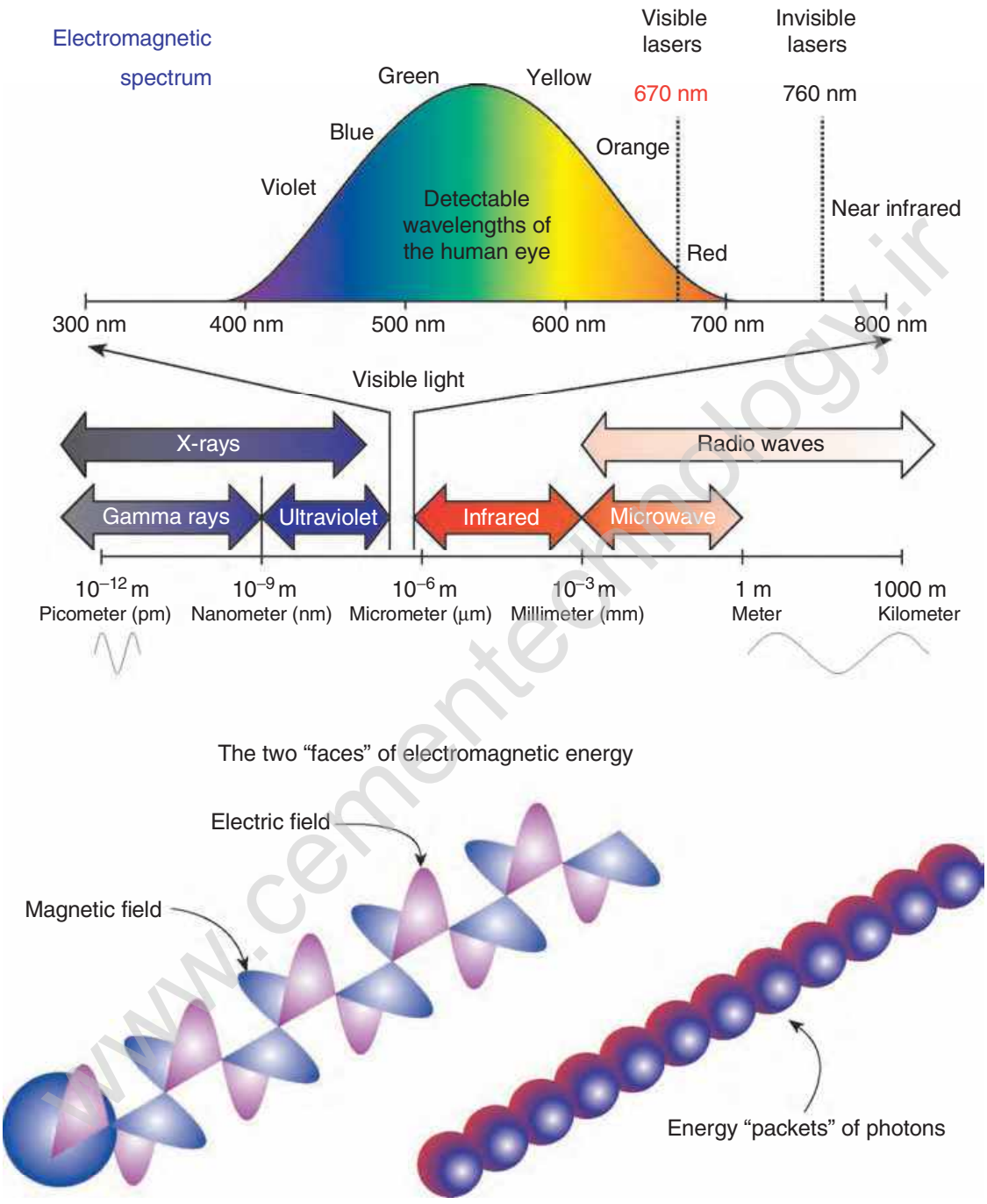


FIGURE 5.15 Four commonly used seals in rotating machinery.

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The photon is the key behind controlling an atom’s orbital energy.
Absorption occurs when electrons go from a lower to a higher orbital level (shell).
Emission occurs when electrons go from a higher to a lower orbital level.

FIGURE 6.26 The electromagnetic spectrum.

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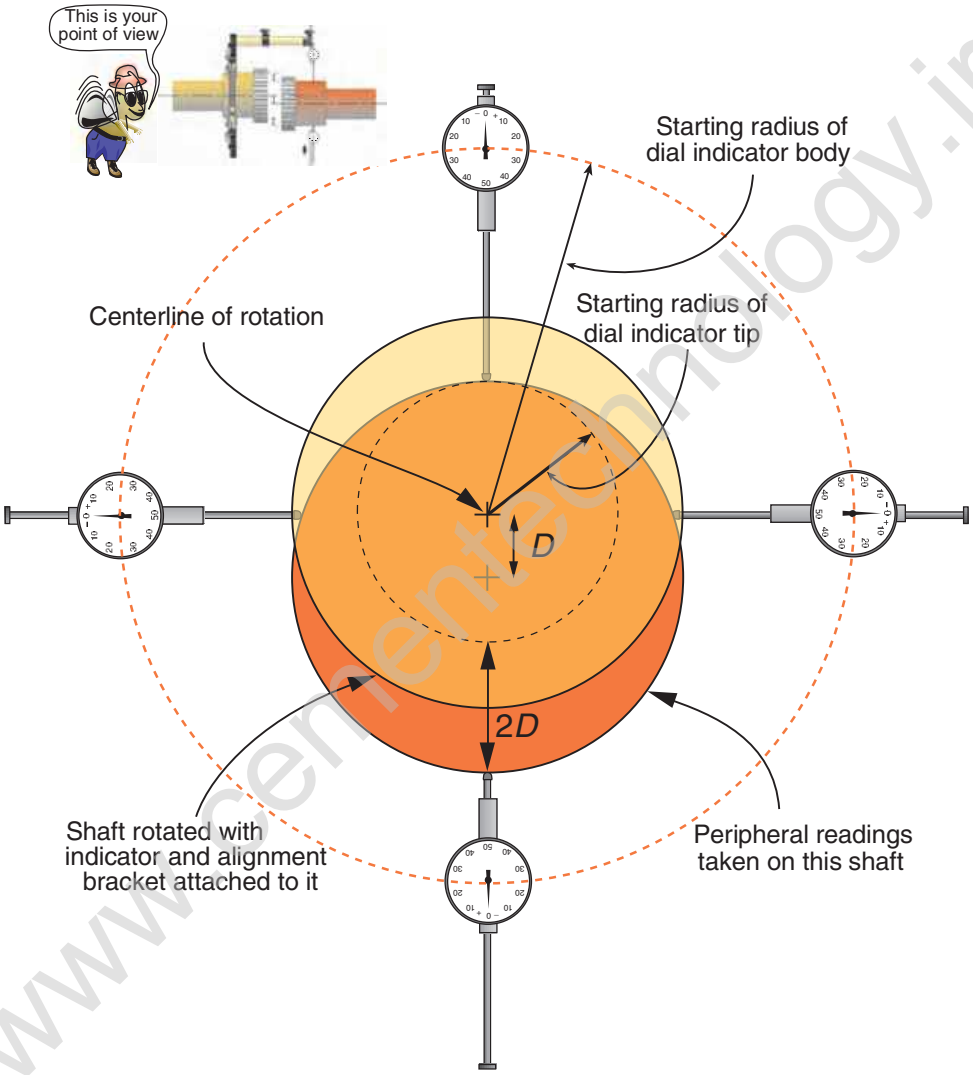


FIGURE 6.45 End view showing indicator measuring twice the actual offset of the shaft centerlines.

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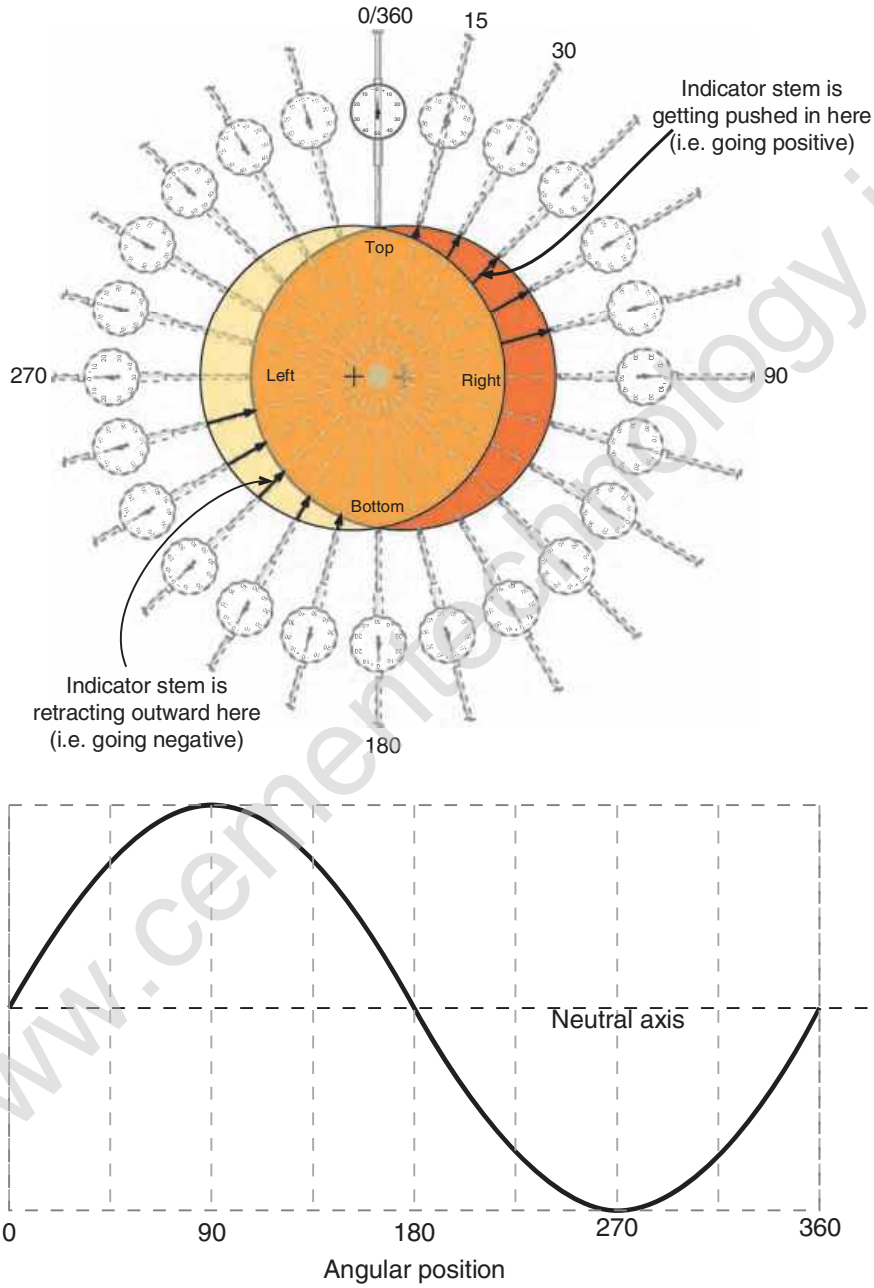


FIGURE 6.49 Plotting the measurements for a pure horizontal misalignment will produce a sine wave.

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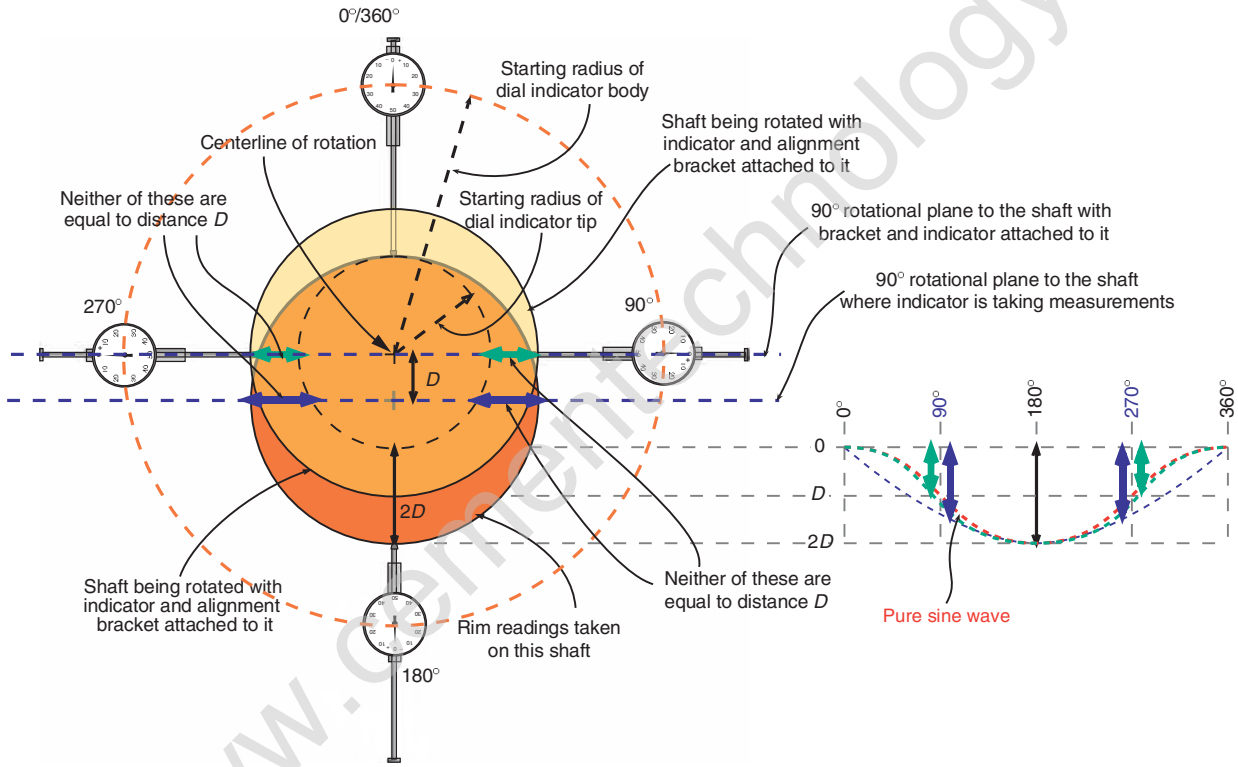
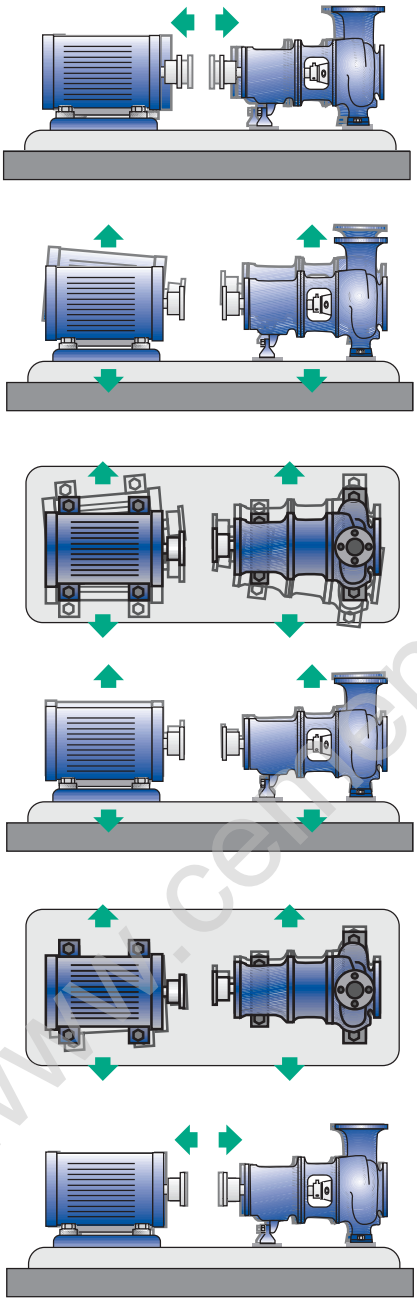


FIGURE 6.55 Error increases with increasing misalignment.

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Step 1—Roughly adjust (±100 mils) the axial position of the shafts.

Step 2—Make a fairly large adjustment to either or both of the machines in the up and down direction by adding or removing shims.

Step 3—Make a fairly large adjustment to either or both of the machines in the side-to-side direction by translating the inboard and outboard ends.

Step 4—Make a “trim” adjustment (if necessary) to either or both of the machines in the up and down direction by adding or removing shims.

Step 5—Make a “trim” adjustment (if necessary) to either or both of the machines in the side-to-side direction by translating the inboard and outboard ends.

Step 6—Make any final adjustments to the axial position of the shafts.

FIGURE 7.8 Typical sequence of corrective moves.

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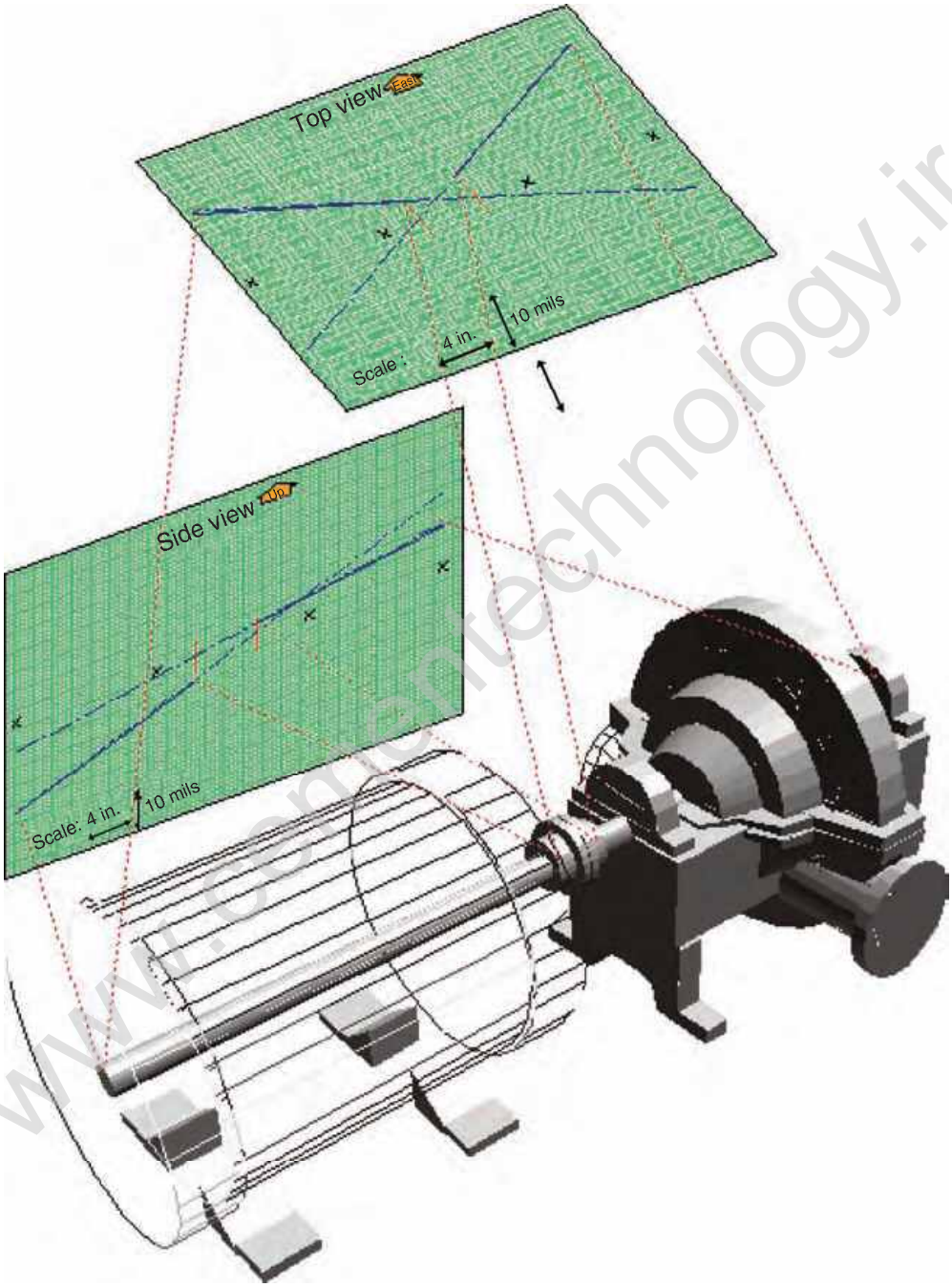


FIGURE 8.3 Three-dimensional view of the side and top views.

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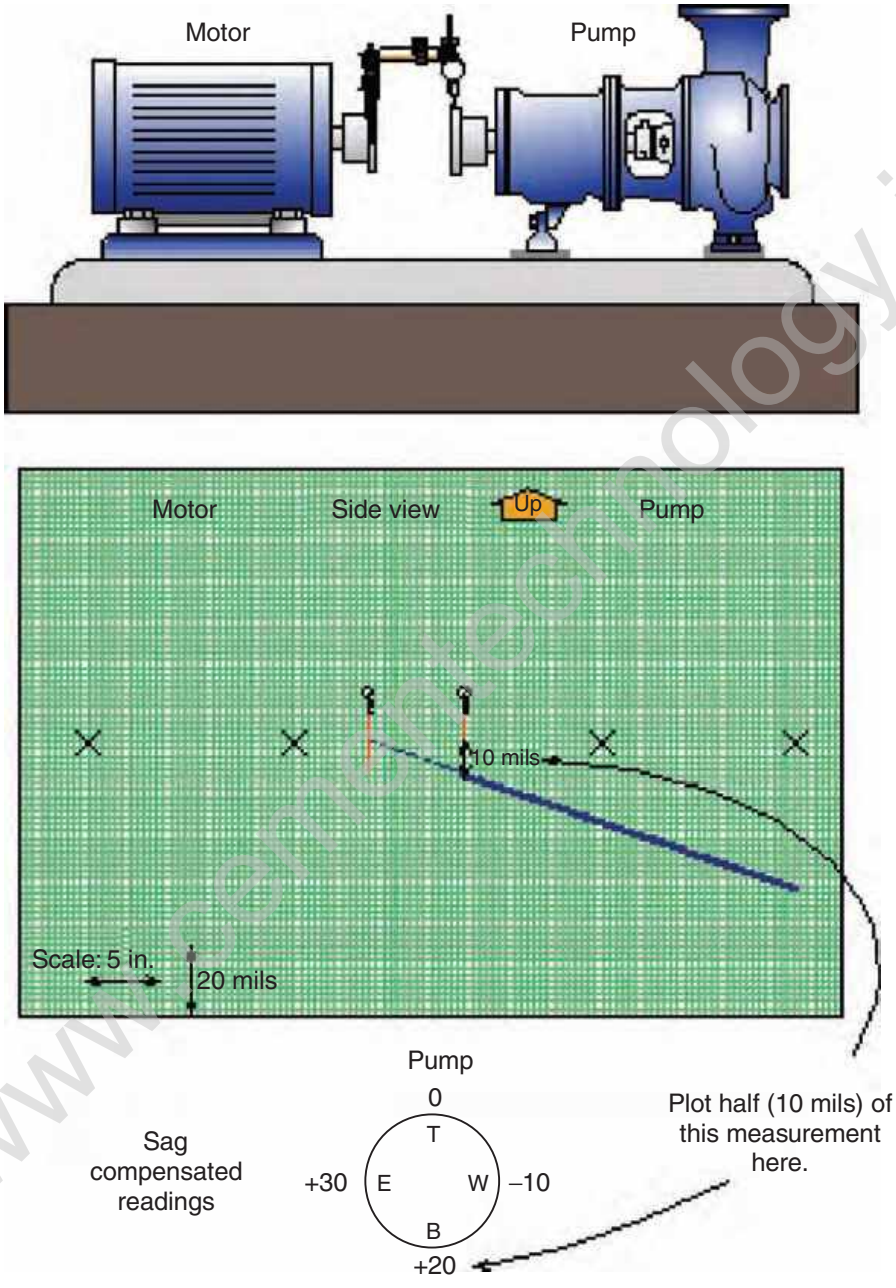


FIGURE 8.8 Plotting the pump shaft in the side view.

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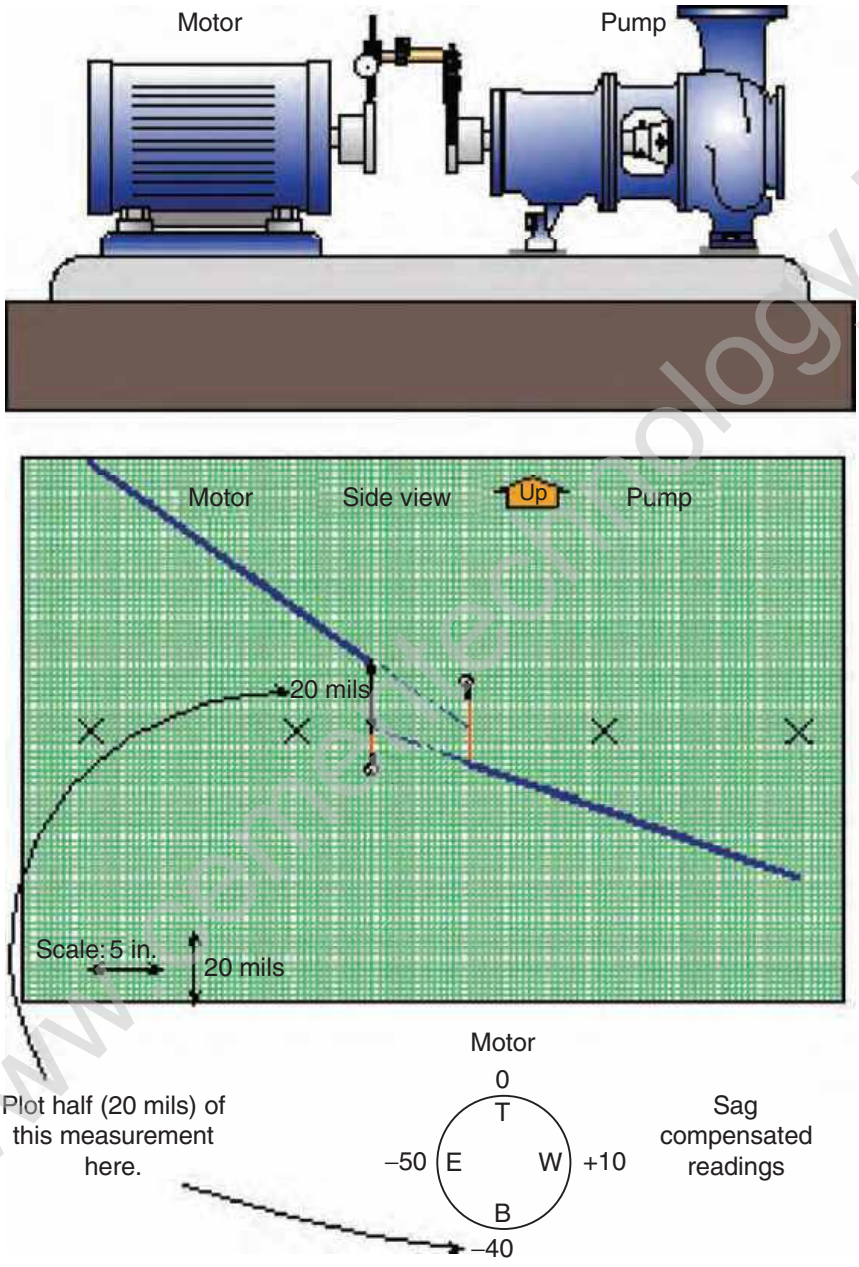


FIGURE 8.9 Plotting the motor shaft in the side view.

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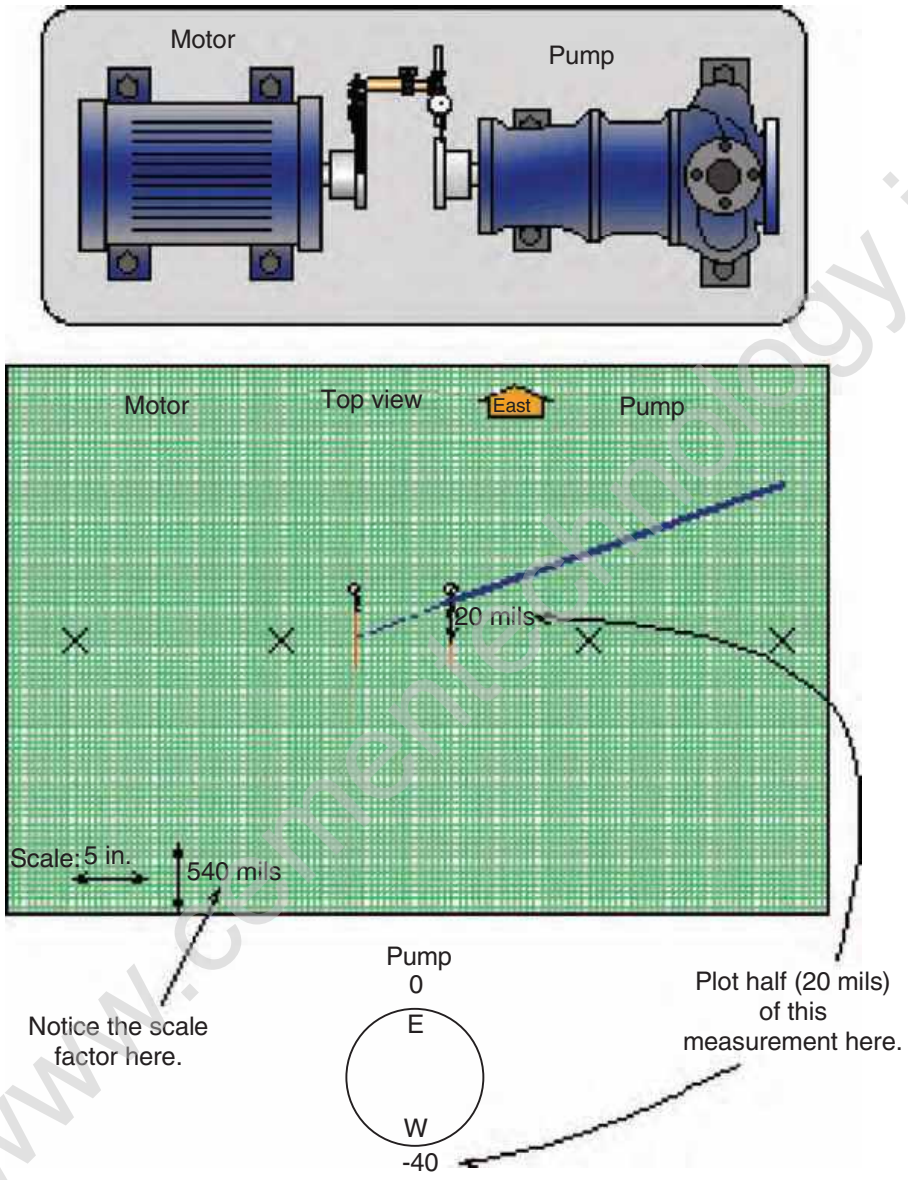


FIGURE 8.11 Plotting the pump shaft in the top view.

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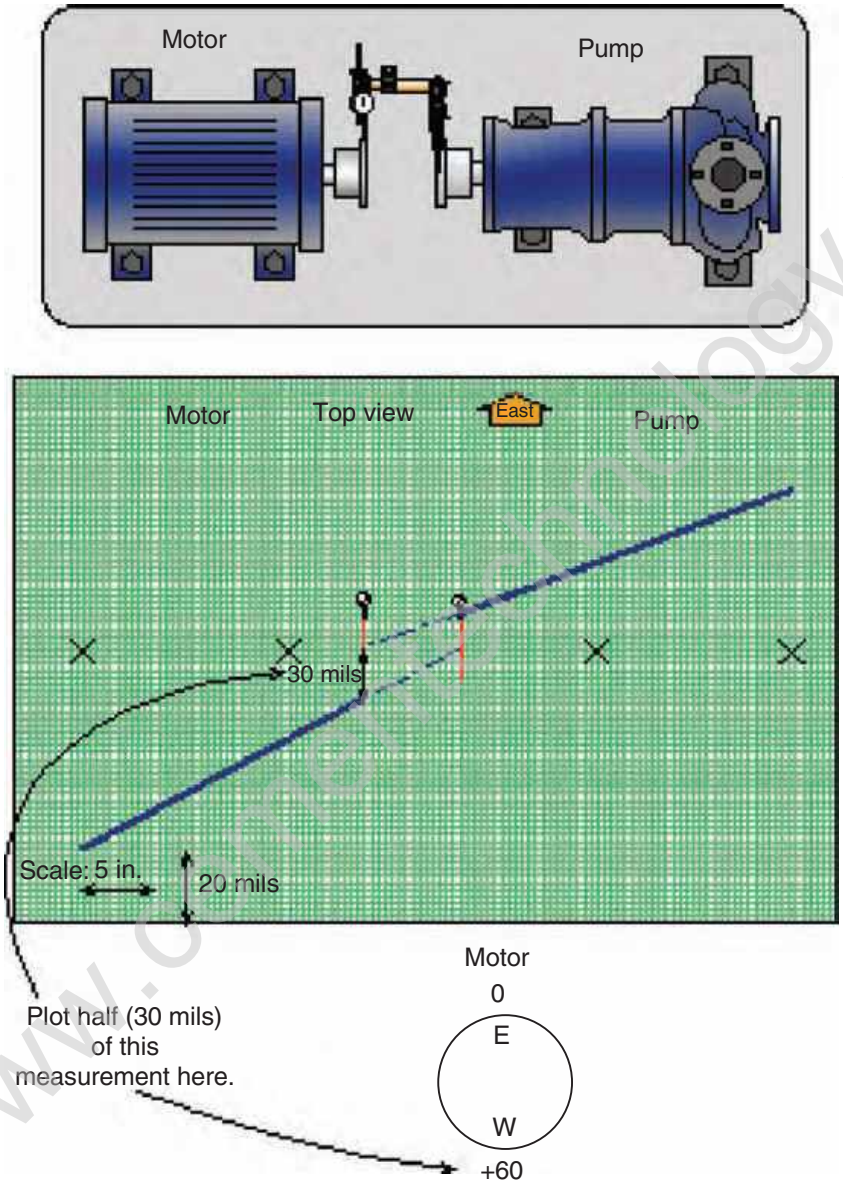


FIGURE 8.12 Plotting the motor shaft in the top view.

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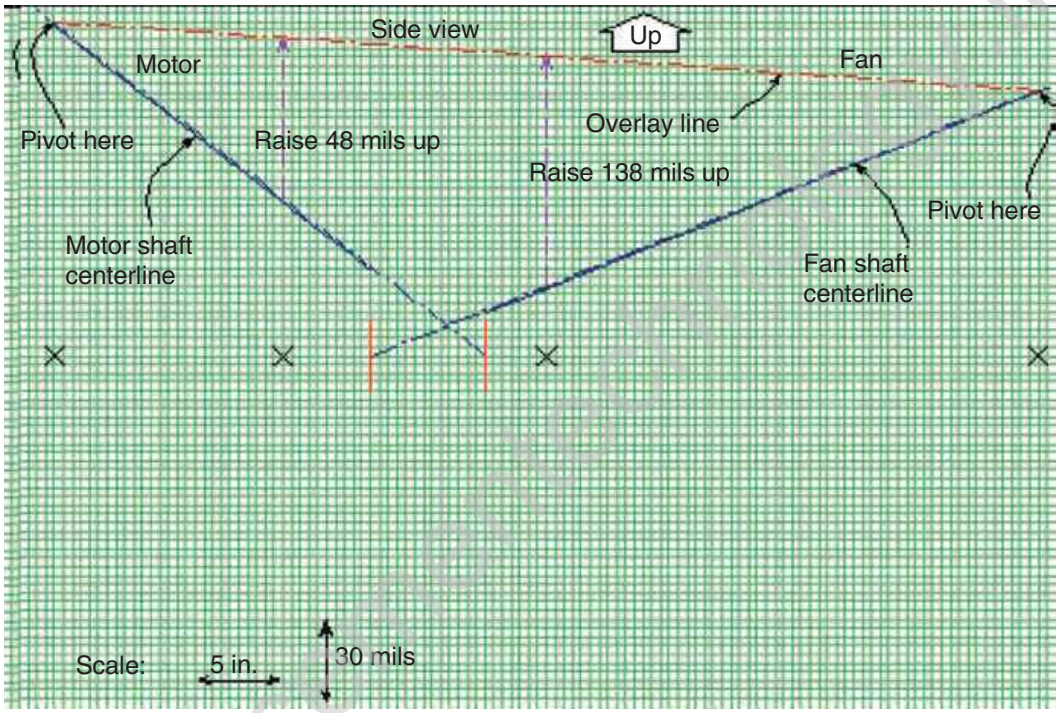


FIGURE 8.14 Movement solutions for the inboard feet of both the motor and the pump by pivoting at the outboard feet of both machines.

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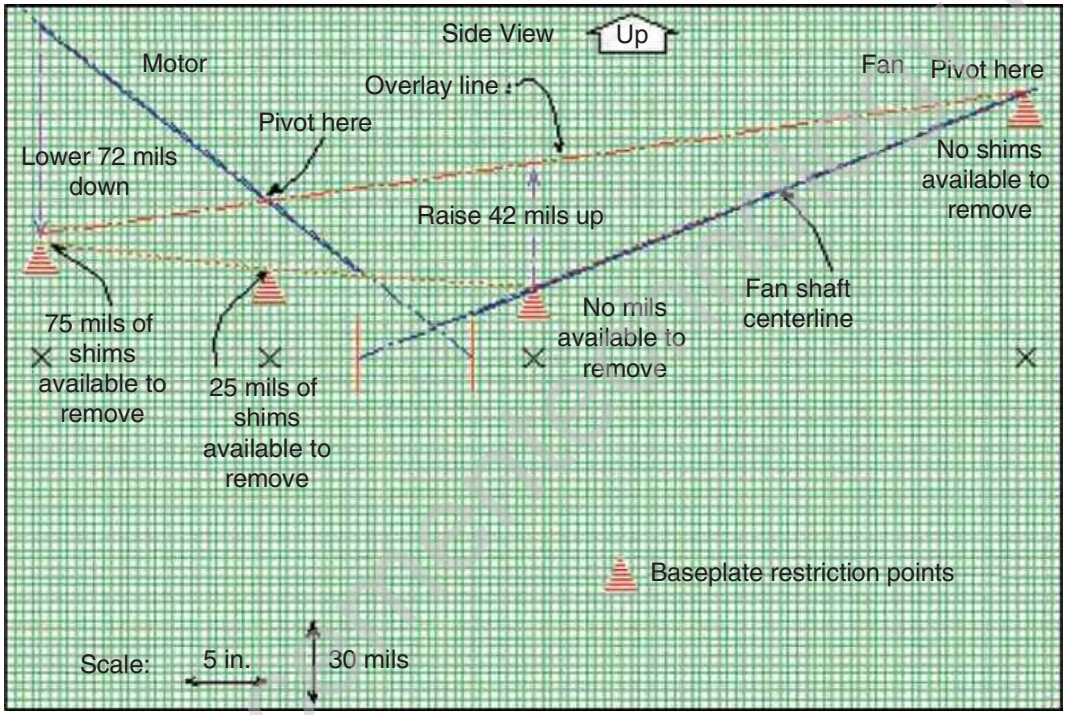


FIGURE 8.15 Movement solutions using the outboard feet of the fan and the inboard feet of the motor as pivot points.

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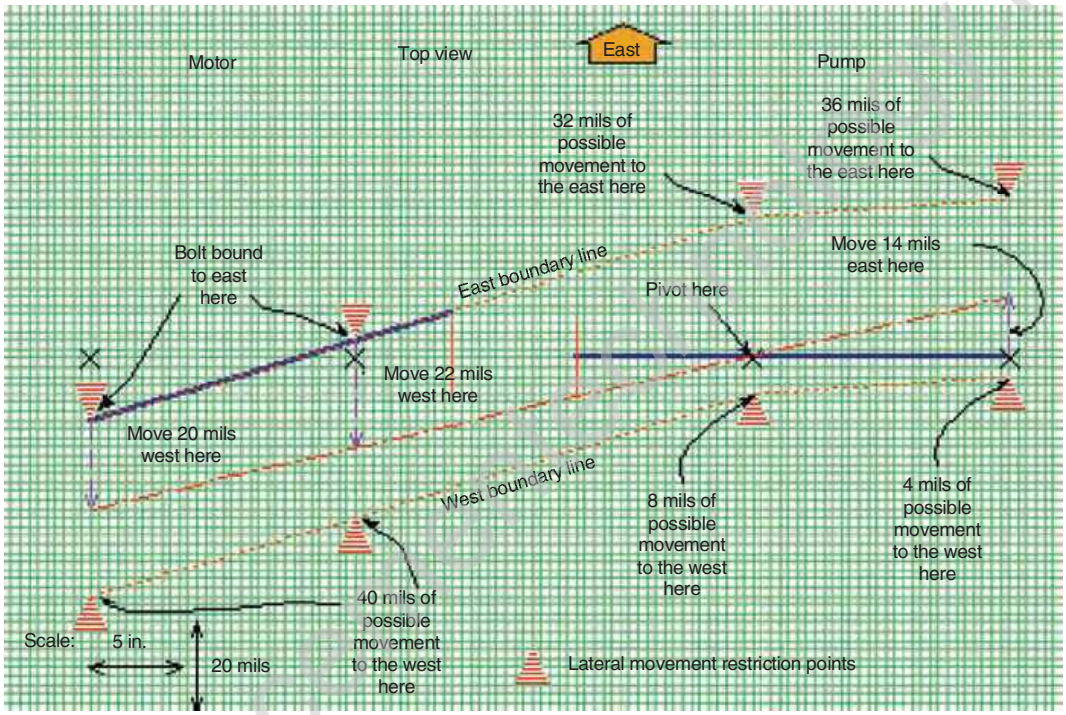


FIGURE 8.16 Applying lateral movement restrictions to arrive at an easy sideways move within the east and west corridors.

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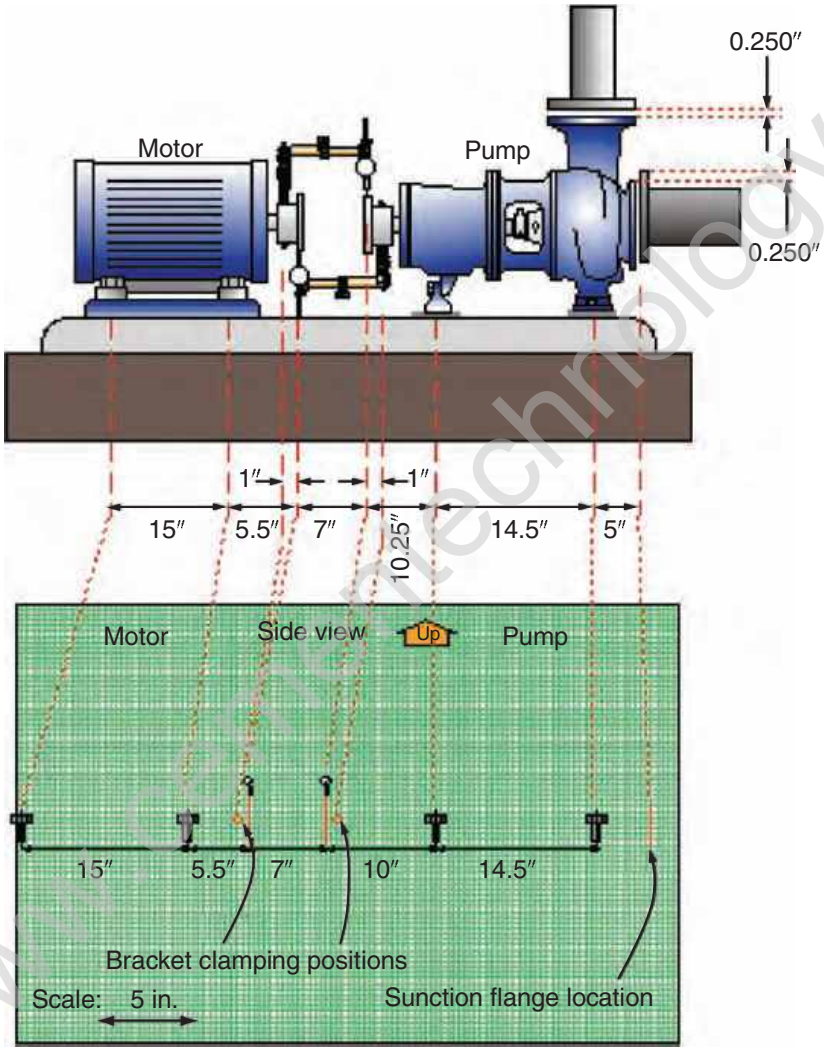


FIGURE 8.17 Scaling off the dimensions for a motor and pump including the location of the suction flange on the pump.

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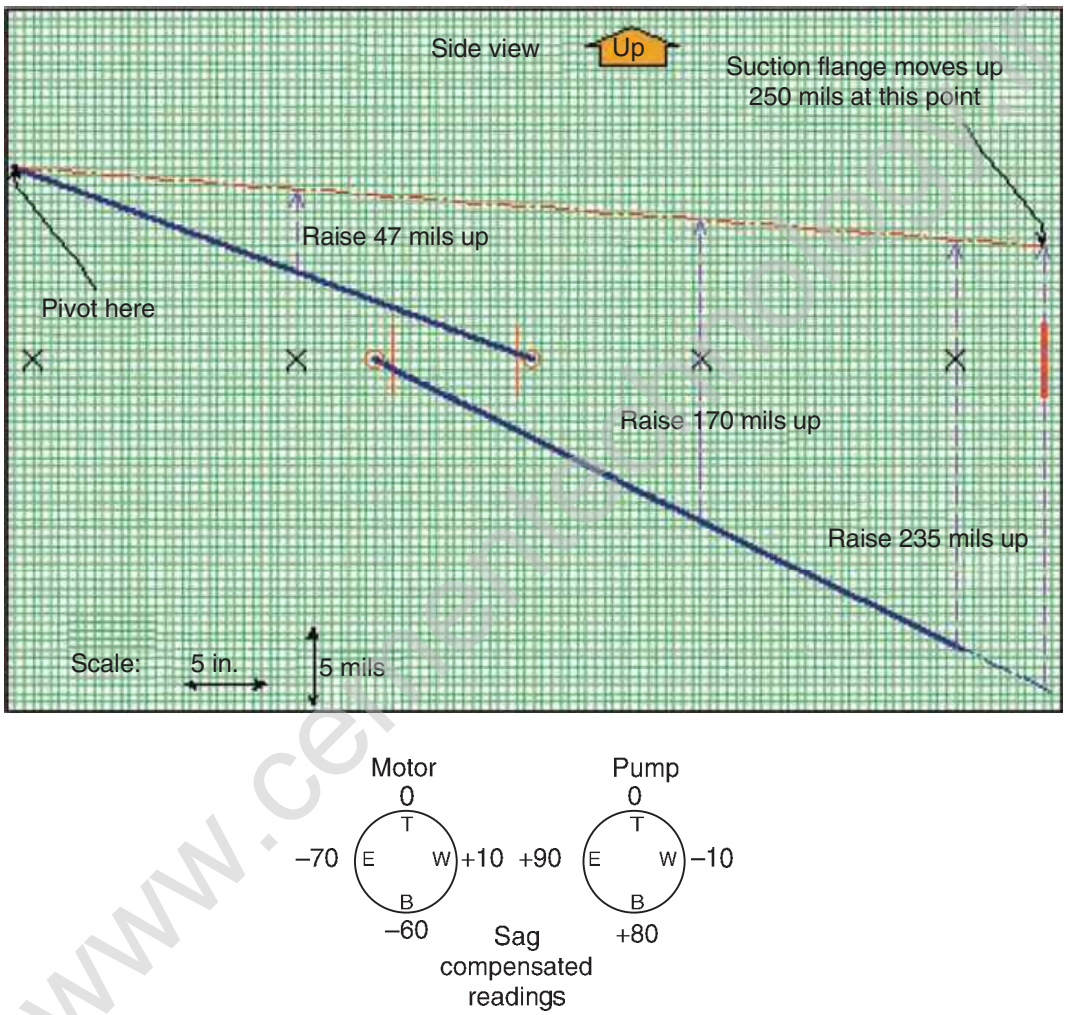


FIGURE 8.18 Overlay line positioned to correct the piping fit-up problem and align the shafts with one move.

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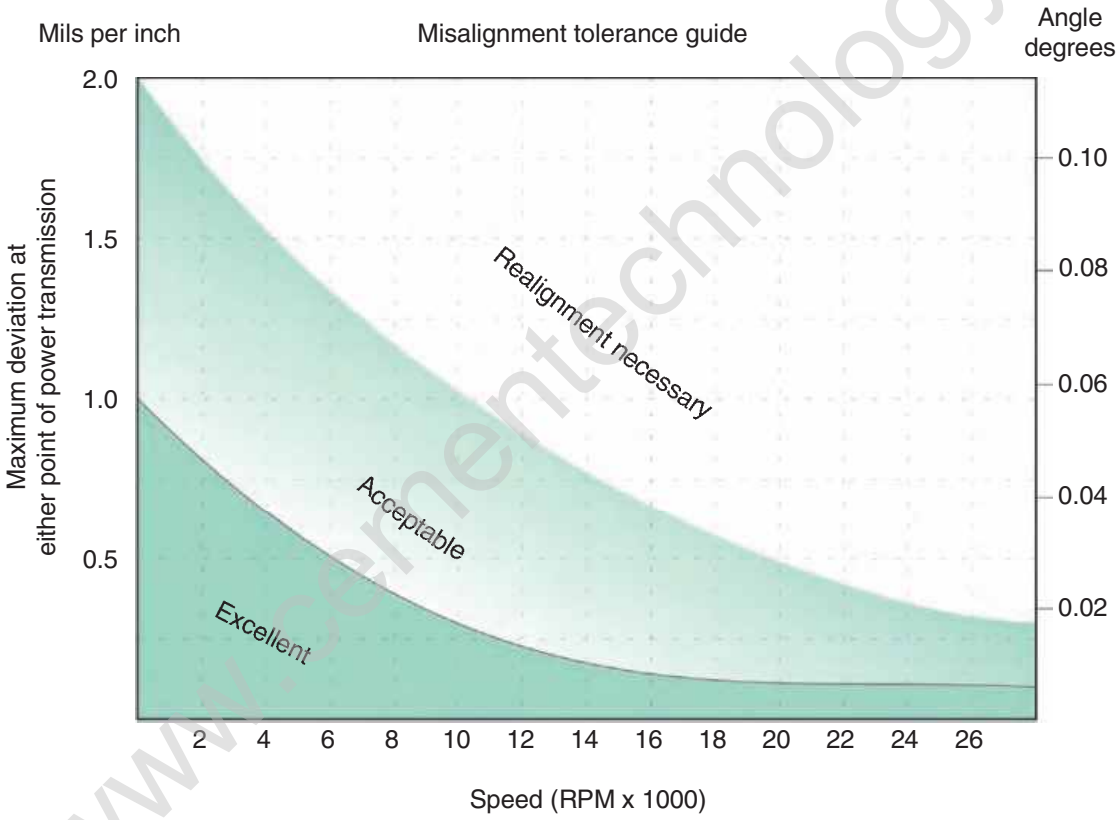


FIGURE 9.6 Recommended maximum misalignment of flexibly connected rotating machinery.

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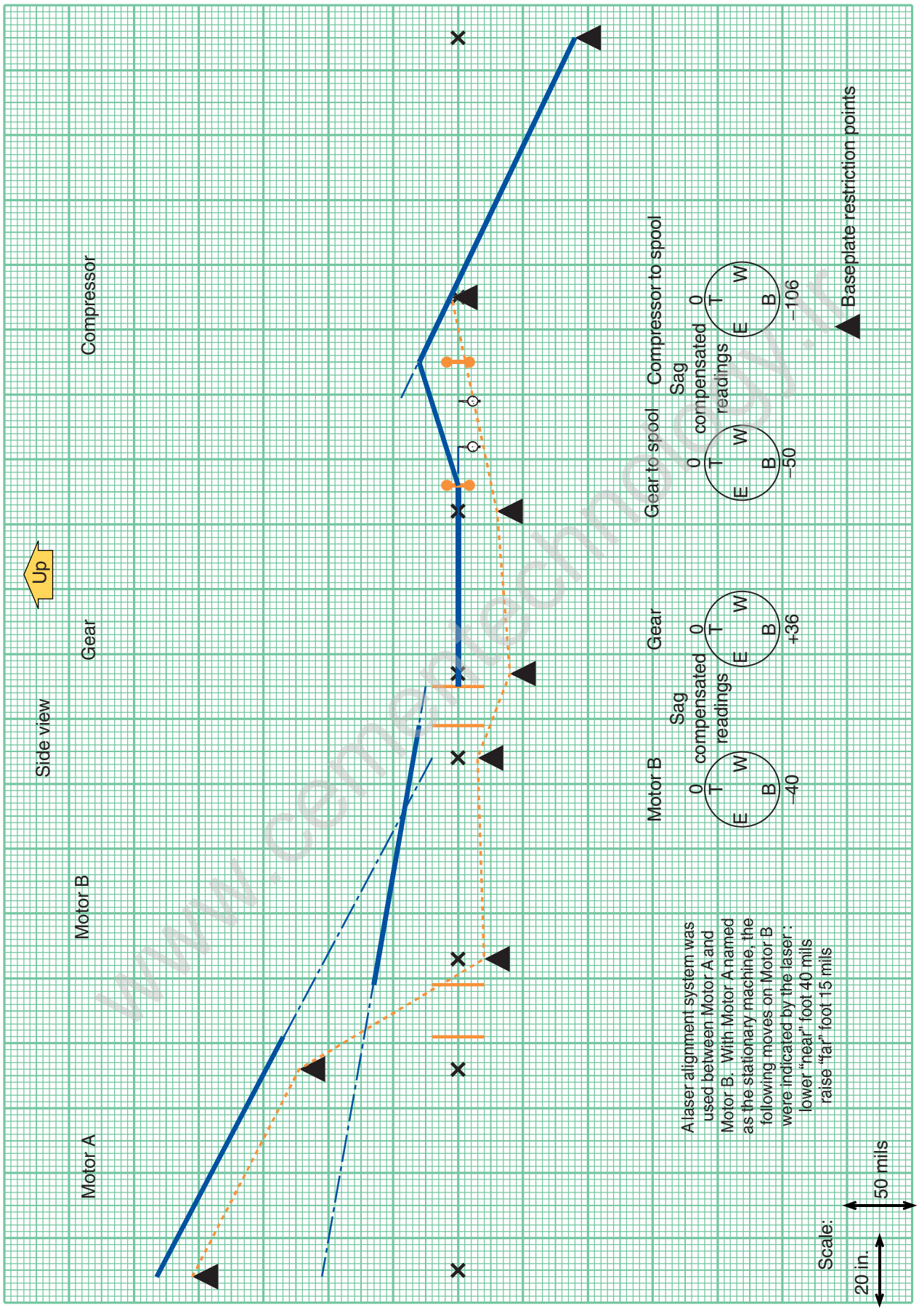


FIGURE 17.18 Four-element drive system side view alignment model of all four shafts and the vertical restriction boundary.

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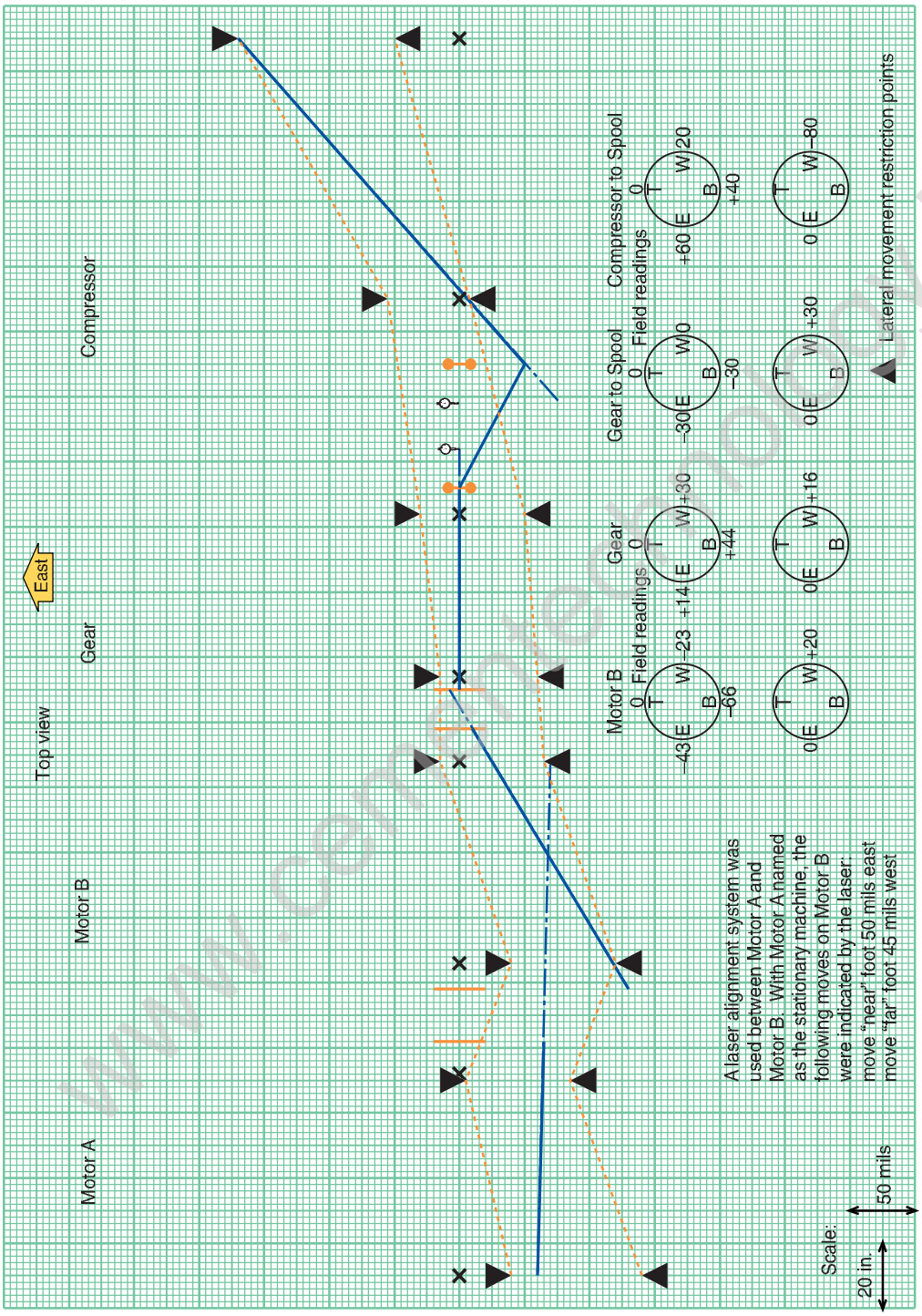


FIGURE 17.19 Four-element drive system side view alignment model of all four shafts and the lateral restriction envelope.

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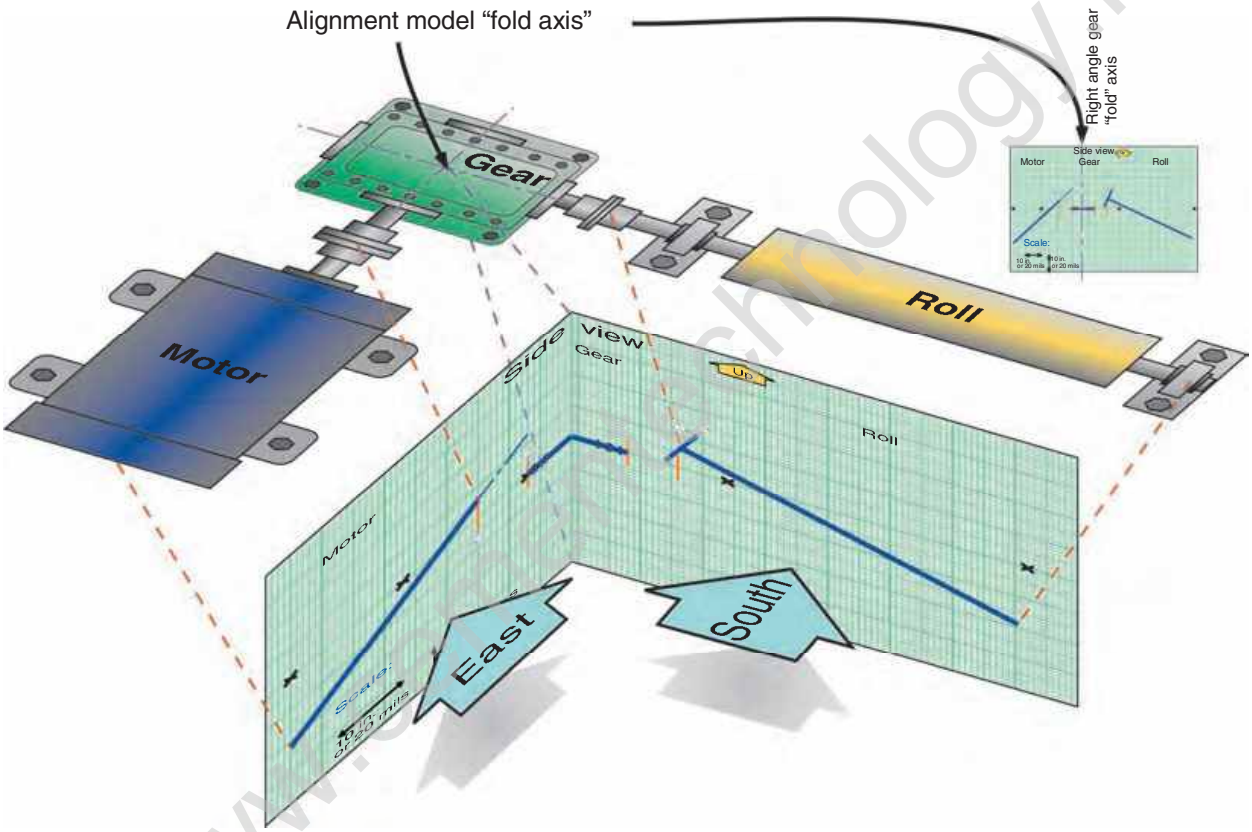


FIGURE 17.23 Right-angle drive system ‘‘folded’’ side view alignment model.

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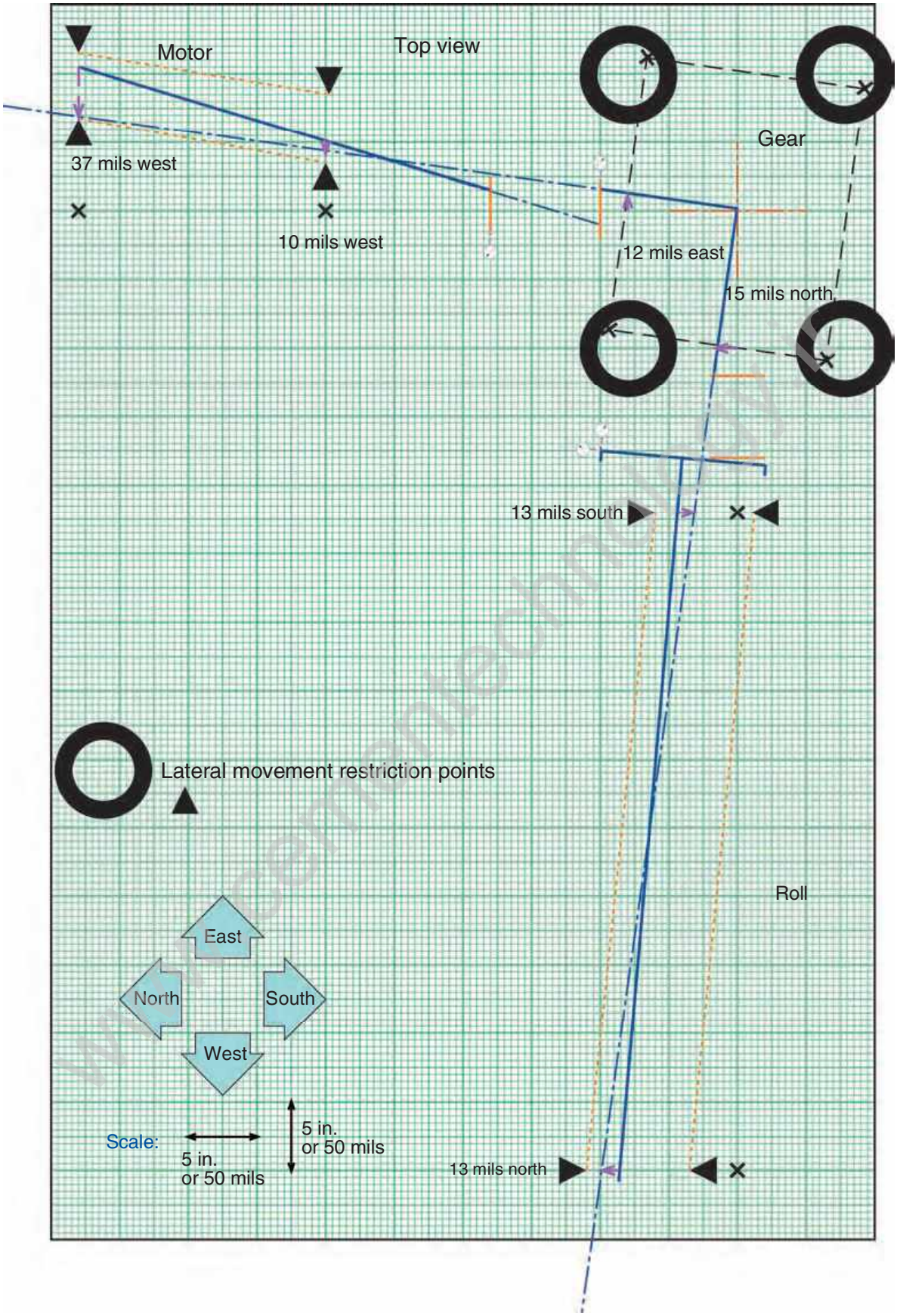


FIGURE 17.29 Top view alignment model showing possible alignment corrective moves in the lateral direction staying within the boundary conditions.

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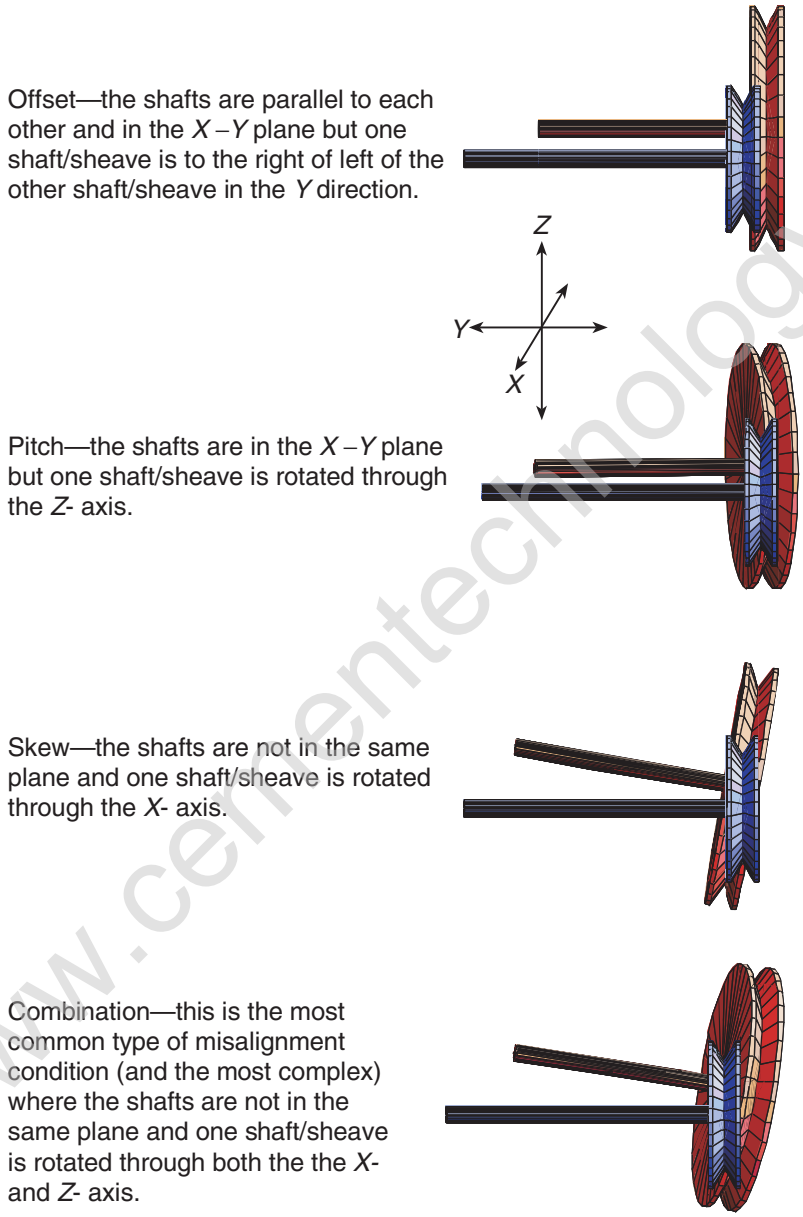


FIGURE 18.17 Types of belt and sheave misalignment conditions.

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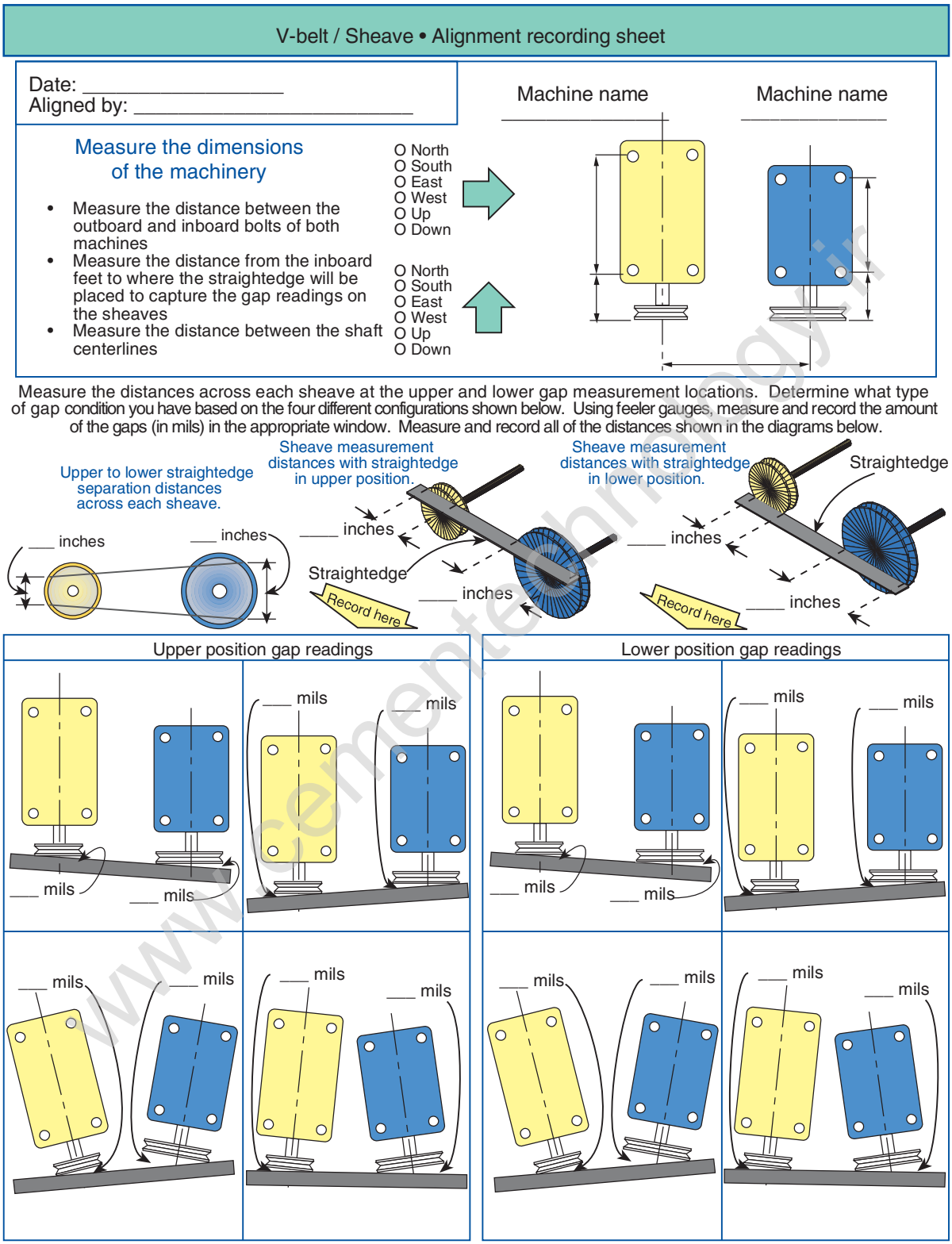


FIGURE 18.26 V-belt and sheave alignment recording sheet.

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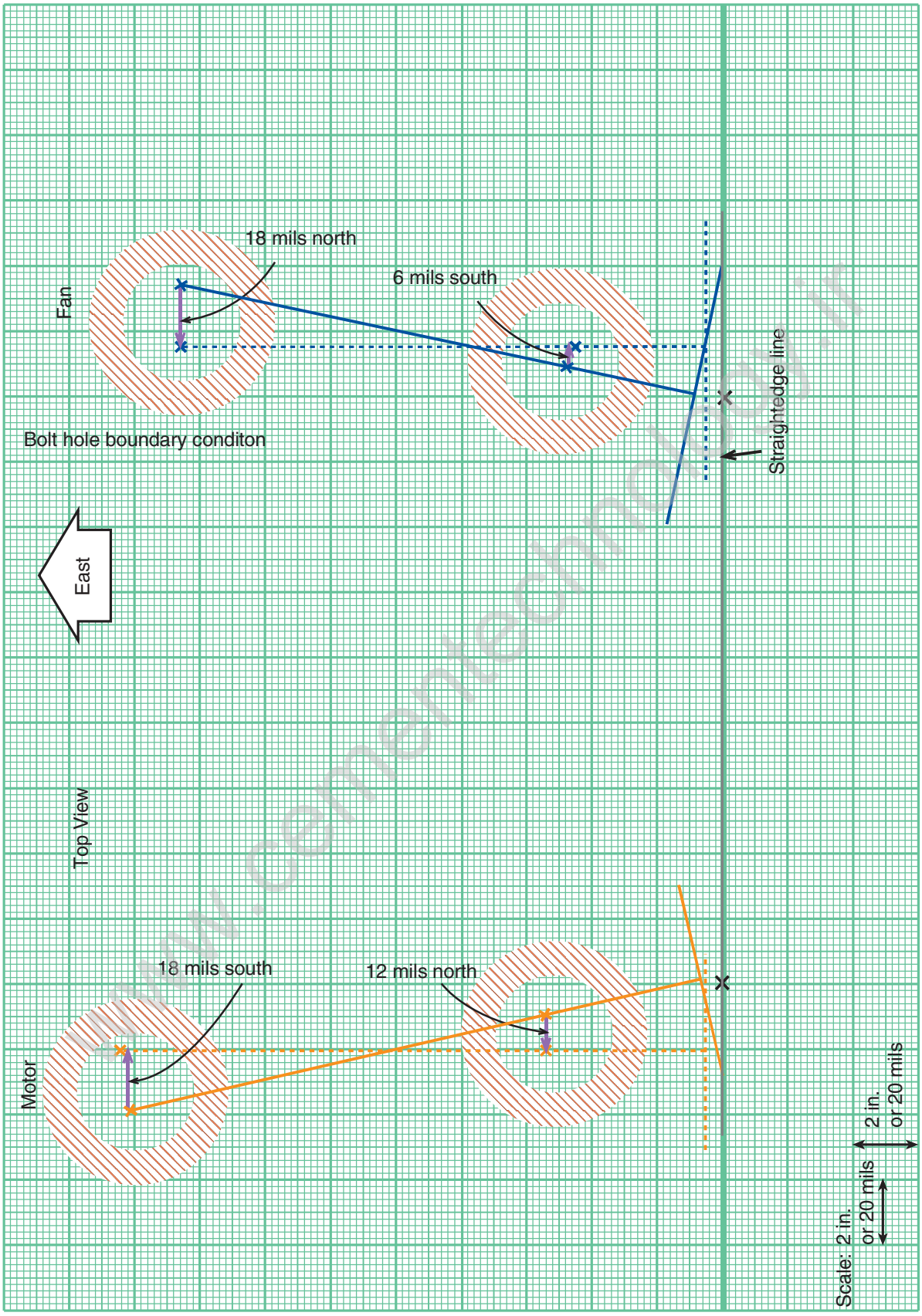


FIGURE 18.30 Possible alignment corrections for the motor and fan in the top view.

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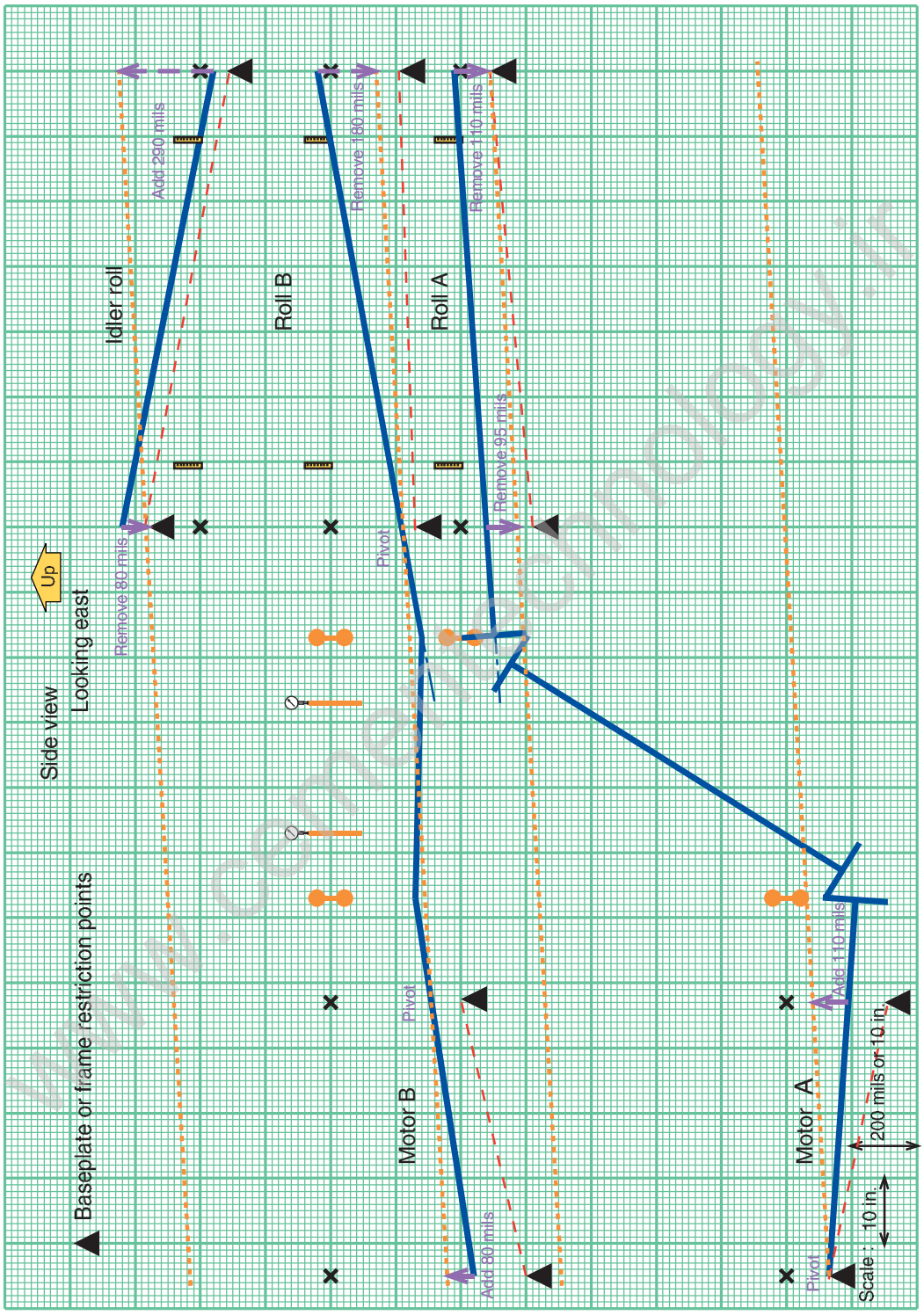


FIGURE 20.37 Side view alignment model showing one possible solution.

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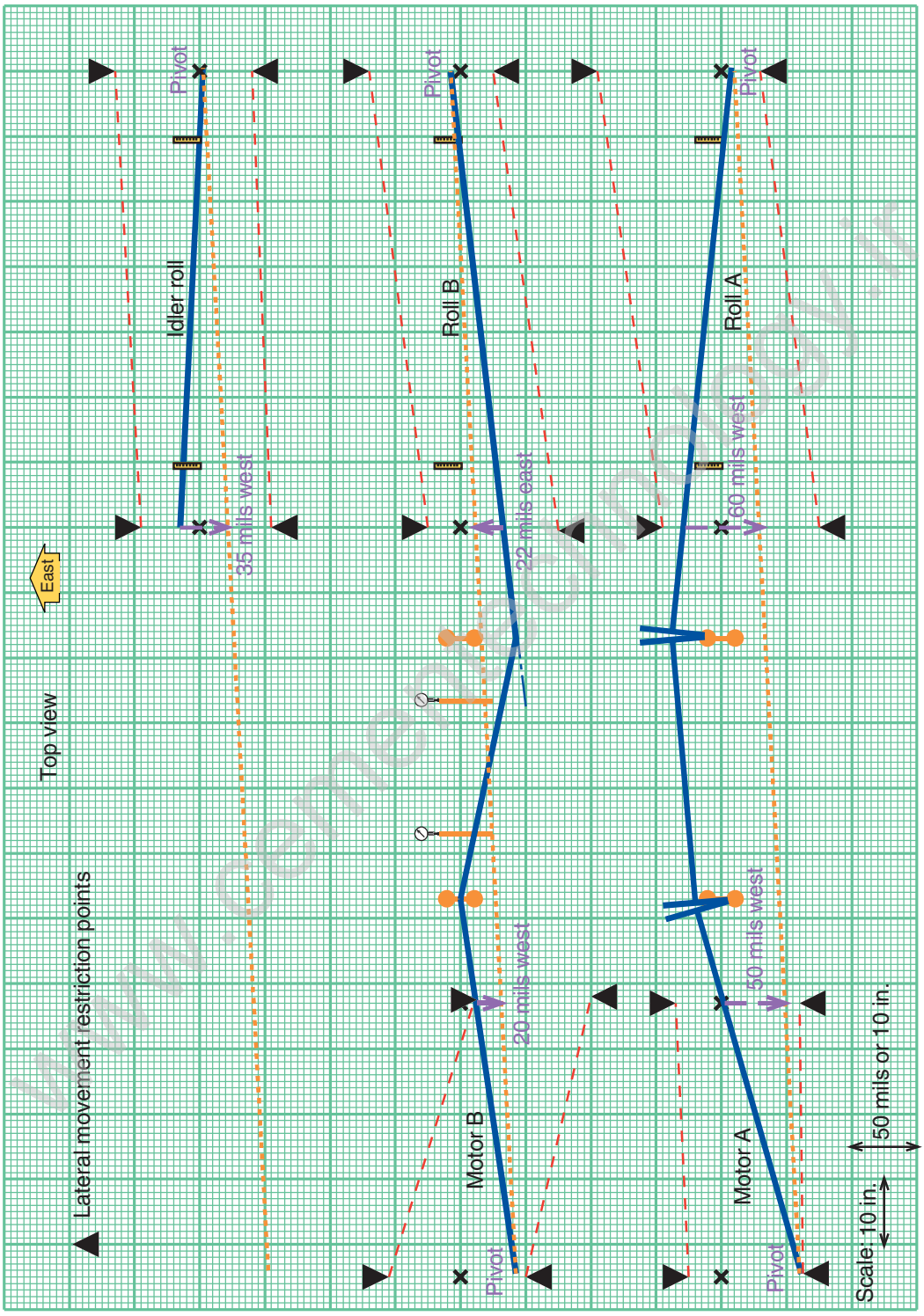


FIGURE 20.38 Top view alignment model showing one possible solution.

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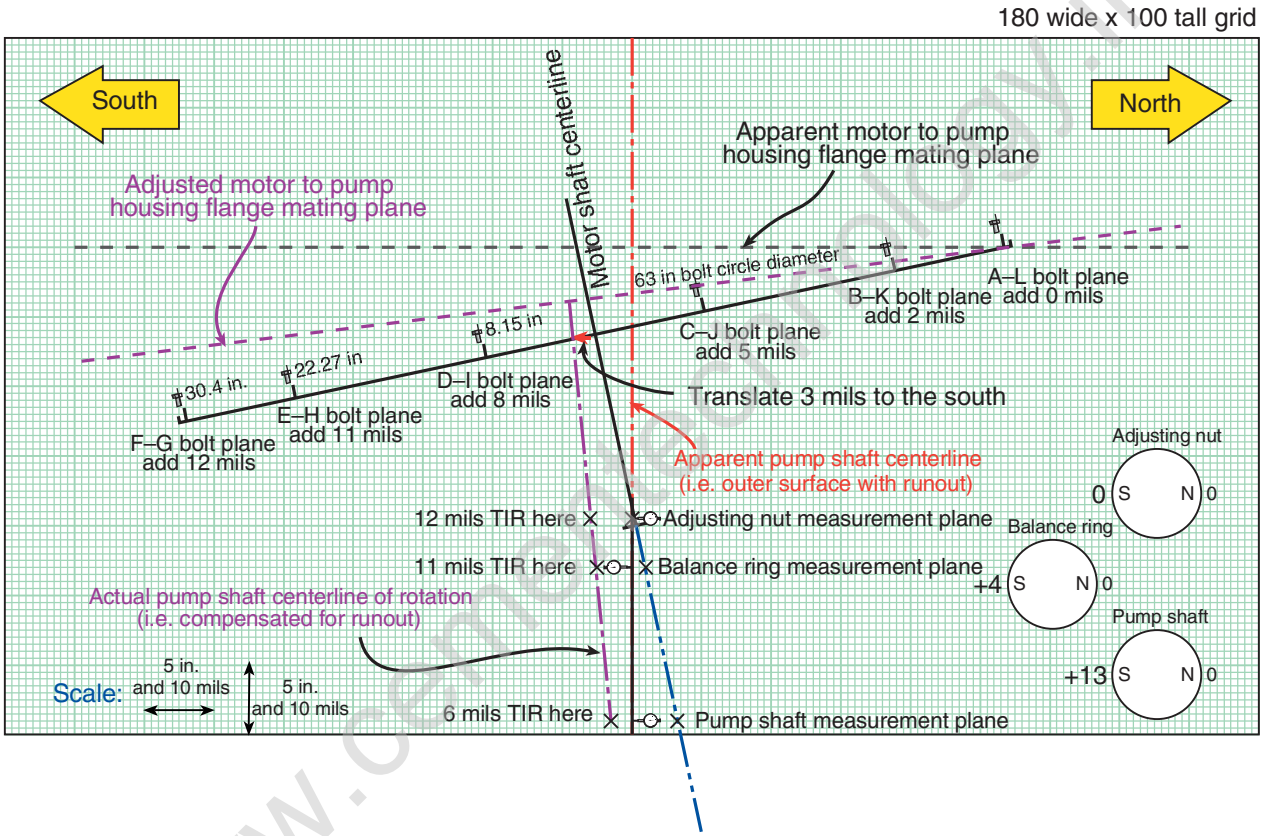


FIGURE 21.27 As-found alignment model of motor and pump shaft as viewed in the north to south direction.

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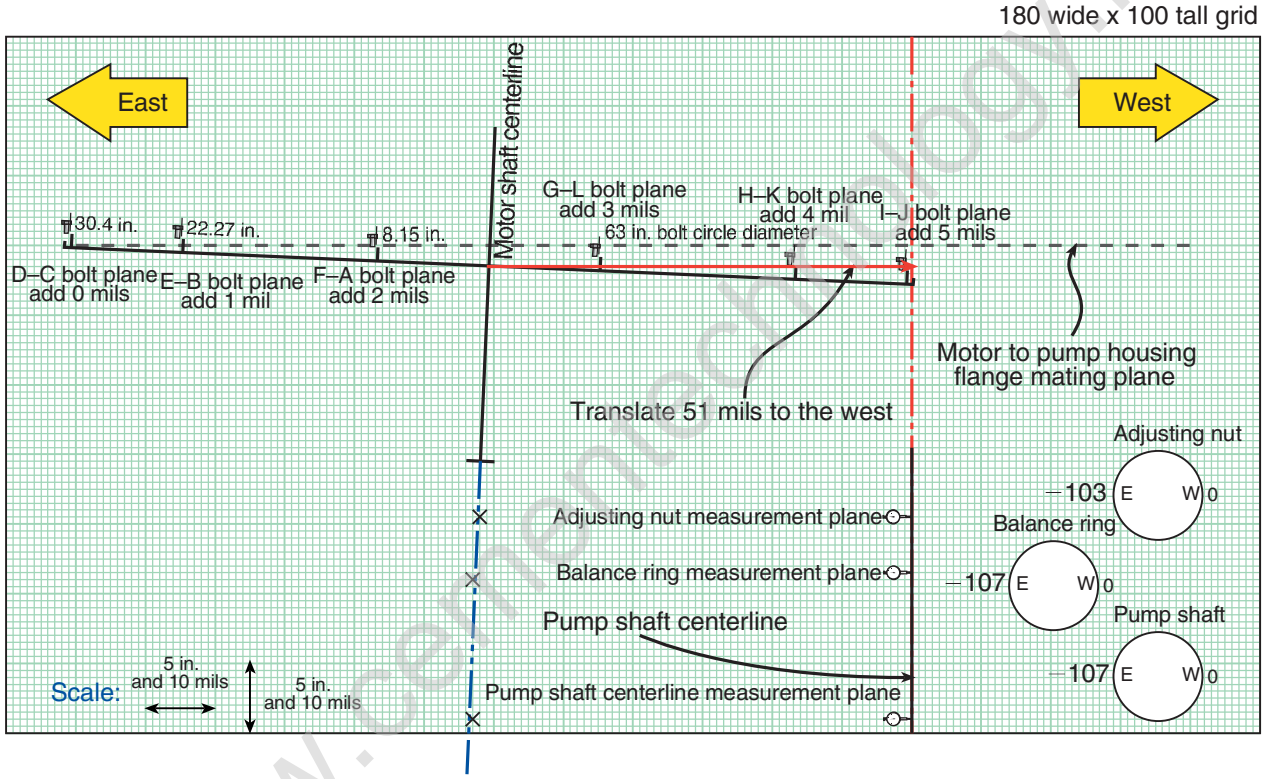


FIGURE 21.28 As-found alignment model of motor and pump shaft as viewed in the east to west direction.

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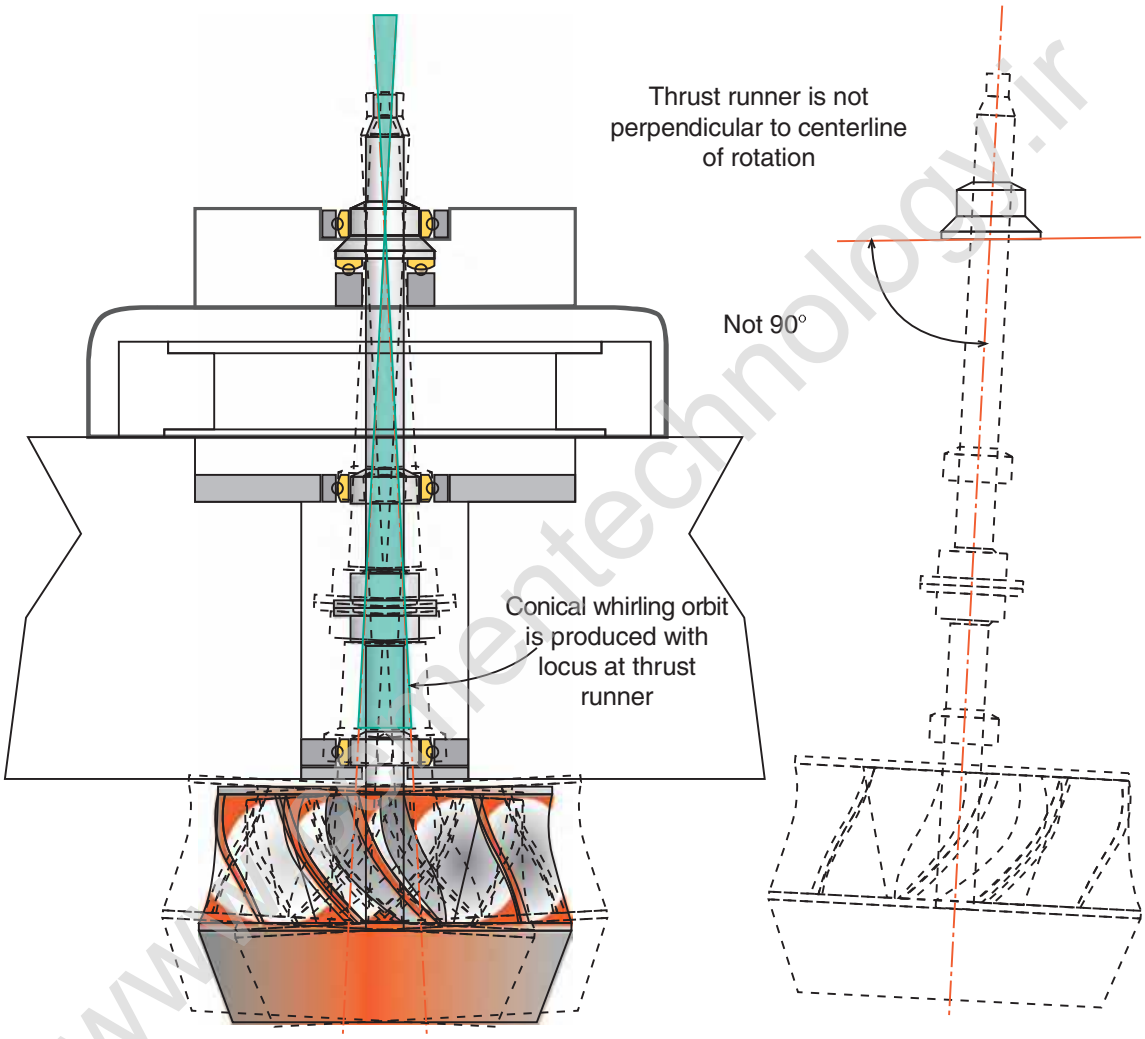


FIGURE 21.66 A nonperpendicular thrust runner face will force the rotating assembly to orbit.

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