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MACHINERY FAILURE ANALYSIS HANDBOOK

Sustain Your Operations and Maximize Uptime



LUIZ OTÁVIO AMARAL AFFONSO

Machinery Failure Analysis Handbook

*Sustain Your Operations
and Maximize Uptime*

Luiz Otávio Amaral Affonso



Houston, Texas

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Sustain Your Operations and Maximize Uptime

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Gulf Publishing Company
2 Greenway Plaza, Suite 1020
Houston, TX 77046

10 9 8 7 6 5 4 3 2 1

Printed in the United States of America.

Printed on acid-free paper.

Text design and composition by Ruth Maassen.

Library of Congress Cataloging-in-Publication Data

Affonso, Luiz Otávio Amaral.

Machinery failure analysis handbook : sustain your operations
and maximize uptime / Luiz Otávio Amaral Affonso.

p. cm.

Includes bibliographical references and index.

ISBN 1-933762-08-X (alk. paper)

1. Machinery—Maintenance and repair—Handbooks, manuals, etc.
2. Plant maintenance—Handbooks, manuals, etc. I. Title.

TS191.A34 2006

658.2'7—dc22

*John Mitchell once dedicated a book
to those professionals who, on a cold night,
would pack their instruments and travel
to a plant to decide if a hot piece
of machinery would last until dawn.*

*Modestly, I dedicate this one to the same
professionals, but thinking that many times
the machine did not survive the next day and the
same person had to find out what went wrong, so
that the mistake would not be made again.*

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Preface

Process industries are home to a huge number of machines, most of them critical to the industry's mission. Failures of these machines can result in consequences that range from the simple replacement of a cheap bearing to an accident that may cost millions in lost production or cause injuries or pollution.

Competition forces corporations to try to keep pace in optimization. On the machinery side of the history, this means improving efficiency and reliability and reducing maintenance cost. Design and purchase specifications, shop testing, installation, maintenance, and operation all play a role in these efforts.

The objective of this book is to help anyone involved with machinery reliability, be it in the design of new plants or the maintenance and operation of existing ones, to understand why the process machine fails, so some preventive measures can be taken to avoid another failure of the same kind.

The book is divided in four parts:

1. An introduction that discusses the fundamental causes of failures, including some definitions and examples. Here, we examine the failure analysis procedure, including some approaches suitable for different types of problems. We also look at how plantwide failure prevention efforts should be conducted, including a discussion about the importance of the role of the plant operator in the reliability of the machinery.

2. A discussion of the failure mechanisms that affect process machinery most often, such as fractures and wear.
3. An examination of how the actual machine parts fail, with several examples of bearings, seals, and other components failures.
4. Examples of real machinery failures analyzed by the author.

I offer no warranty regarding the applicability of the information contained in this book to any particular situation.

This English edition has been based on the second Portuguese edition, published in Brazil in 2006.

Part I

Introduction to Failure Analysis

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1

Fundamental Causes of Failures

This chapter discusses the basic concepts of root cause and failure. Some examples of failures related to design, fabrication, assembly, operation, and maintenance are included.

Failure occurs when the component or equipment no longer can perform its intended function safely. The function of the component is the primary reason why it was installed on the machine.

Premature failure happens when the defect occurs within the design life of the component. Design life is a design criterion understood statistically, not deterministically, which means that the expected dispersion of the component's useful life should be taken into account. Design life usually is related to certain specific types of failure modes, anything else is considered abnormal. For example, the classical end of life failure mode for an antifriction bearing is surface fatigue. Therefore, if this bearing fails due to surface fatigue after a long enough period of time, we can say that it has reached the end of its useful life. Any other failure mode indicates an abnormality, no matter how long the period before failure. This

concept does not apply to components designed for infinite life, for example, pump shafts.

The root cause of a failure is the fundamental reason that made the failure possible. Multiple root causes are more likely when dealing with machinery failures. The selection of the root causes of a certain event is done with an eye on the usefulness of the selected root cause. If we think of the relationship among all the events observed at a certain piece of machinery before failure, we will observe that some are closer in time to the failure than others and some of them play a more important role. An effective failure analysis selects the root causes that most effectively avoid repetition of the failure and in which some action is feasible. In addition to that, root cause description should be as detailed as possible, as it is not very useful to say that the root cause of a failure has been a maintenance error, for example; not very much can be done with this information. Rather, we should say that the root cause of a failure was the installation of an antifriction bearing due to the impacts produced by the use of inadequate tools and lack of training. Now, we know what has to be done to avoid repetition of the failure.

The various types of root causes can be classified in several ways. The categories that follow are arbitrary and intended to serve academic purposes only, creating a framework that will help understand the issue.

1.1 Design Failures

Design failures are born on the drawing board, when the machine designer determines a fillet radius or the specification of an antifriction bearing. Such failures can be avoided only through redesign of the failed machine or component. Great care should be exercised before deciding that the cause of a failure has been a design deficiency. Some examples follow:

1. Notches create stress concentrations that may be the origin of a fatigue crack. They can be easily avoided most times. Notches are found in shaft shoulders and threads, for example. Another example can be seen in Figure 1.1, where a fatigue

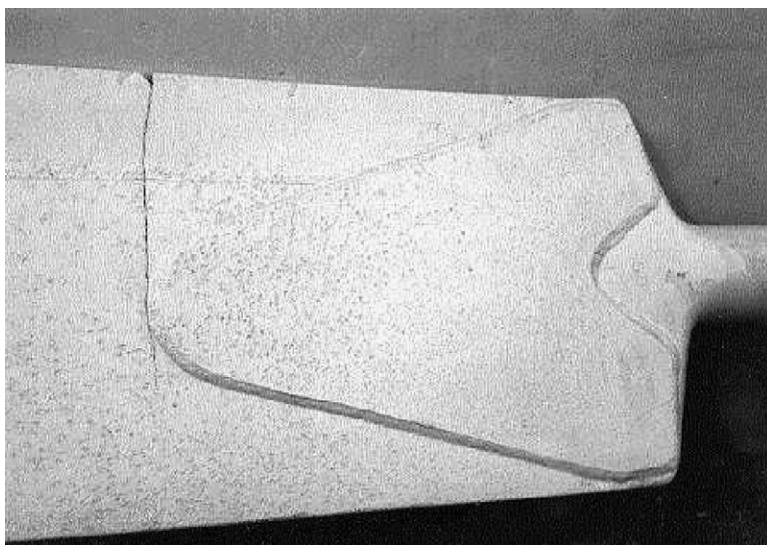


FIGURE 1.1 *Fatigue crack initiated at a stress concentration spot of a fan blade.*

crack originated at the point of stress concentration created by the reinforcement on the fan blade.

2. Inadequate design criterion can cause unforeseen demands to damage the machine or component. In such a situation, one finds that the machine simply has not been designed to handle the unexpected condition, be it a contaminant that makes a fluid corrosive or some kind of vibration. Special purpose machines are prone to this type of problem, as they are designed for a specific service and it is not very easy to test them under real working conditions. Figure 1.2 shows an example of an unexpected process condition that led to the failure of a reciprocating compressor. In this case, the unforeseen condition was the ability of the gas to polymerize and create hard deposits inside the compressor. These hard deposits damaged the valves and the stem sealing, resulting in very low reliability of the machine.
3. Design modifications can result in unexpected failures if the modification is not done carefully. For example, pumps used to



FIGURE 1.2 *Reciprocating compressor piston with hard deposits created by the polymerization of the gas.*

be designed to use packing for shaft sealing. When mechanical seals began to be used widely, many pumps were retrofitted. At that time, it was found that the vibration levels went up and reliability went down, as a result of the loss of the shaft support formerly provided by the packing.

1.2 Material Selection Deficiencies

A material selection-related failure can also be thought of as a design-related failure, as the materials of construction are selected during the design of the machine. This type of failure can be avoided through careful selection of the machine part materials. Figure 1.3 illustrates two reciprocating compressor oil scrapper rings that were damaged by high temperature. The high temperature was the result of excessive friction between the rings and the stem, as the rings were too hard. Changing the ring material to a softer rubber solved the problem.

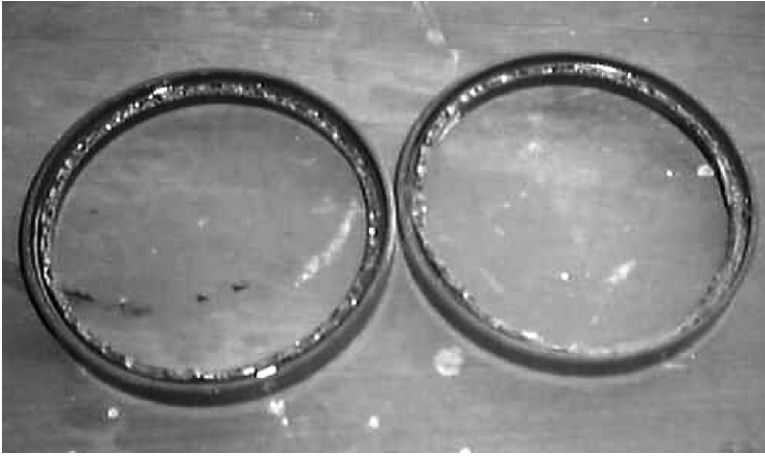


FIGURE 1.3 *Reciprocating compressor oil scraper rings damaged due to high temperature.*

Some contributing factors include

1. Structural materials normally are selected for mechanical strength. High strength may be followed by less ductility or less corrosion resistance. Failure may arise due to these or other characteristics of the material. The machine designer should consider these possibilities, and a compromise may be necessary when more than one possible failure mechanism is present.
2. Unexpected failure modes may force a change in material; a classic example is the brittle fractures of the liberty ships, which forced designers to consider other properties of the construction material.

1.3 Material Imperfections

Imperfection in the construction materials may also be the origin of machinery failures. Internal and external defects that reduce the resistance of the component are possible sources of cracks or localized corrosion, for example.

Such defects are intimately connected to the processing of the raw material during fabrication. Some classic examples include

1. Cast components: inclusions, voids, cold shots, and pores.
2. Forgings: contraction and bends.
3. Laminated parts: double lamination and lamellar decohesion.

The design of the components should take these possible defects into account, and quality control inspection should be specified accordingly.

1.4 Manufacturing Defects

Manufacturing defects occur during the processing of the raw materials used to fabricate the machine components. Although it may not be easily distinguishable from the previous type of failure, the recognition of the source of a failure is always very important for the prevention of future failures.

Some examples include

1. Cold forming introduces huge residual stresses that, if not relieved, may be the source of a fatigue fracture, if the part is cyclically loaded.
2. Machining operations can create notches that act as stress concentrators; part number inscriptions by indentation or eletro-erosion also may be a source of cracks, if done in highly stressed areas of the component.
3. Heat treatment can create various types of defects on the component: overheating, heat checking, surface decarburization, and so forth.
4. Welding can generate many types of defects. This is such a broad subject that it has become a specialized engineering subdivision.
5. Surface treatments, like electrolytic coating or acid scaling, are recognized as sources of hydrogen, which can cause

embrittlement of some types of high strength steel. This type of process should be carefully controlled.

1.5 Assembly and Installation Errors

Human imagination is the only limit to the variety of assembly and installation errors. Most of the time, these are the direct consequence of a human error and can be found anywhere, from the wrong clearances of an antifriction bearing to loose bolts and the like.

This type of error can be avoided with carefully written procedures, thorough training, correct tools, and some auditing. Figure 1.4 shows an example. In this case, an axial antifriction bearing has been assembled with a perpendicularity disalignment on the shaft. We see that one side of the bearing is damaged.

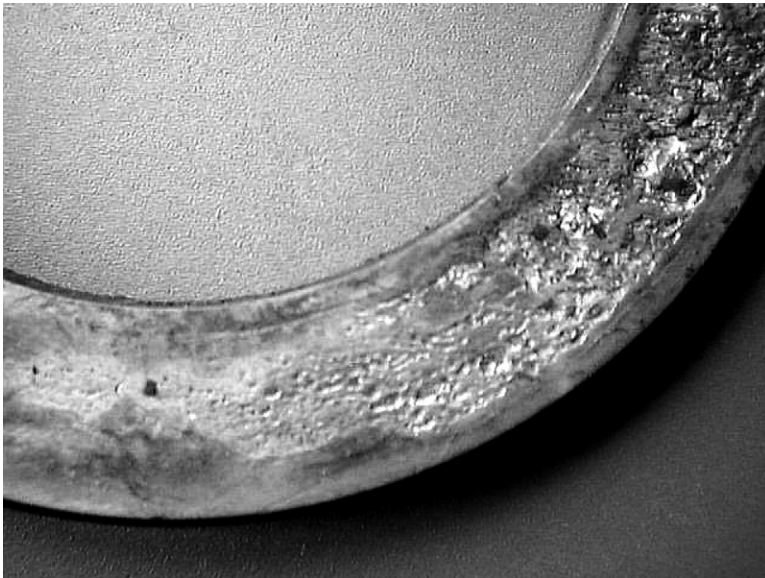


FIGURE 1.4 *Axial antifriction bearing damaged due to incorrect installation.*

1.6 Maintenance and Operation Errors

Process machinery often operate under severe conditions, which means that their maintenance and operation should be world class. Operation of process machinery outside of the design conditions not only may reduce the length of the equipment's life but also be dangerous.

Equipment startup is a critical operation, because the machine is subject to conditions not found during normal operation, including transient loads, flows, temperatures, and the like. The same transients affect the machine during shutdown, after which it is subject to the preservation conditions while inactive. The machine and system design should allow adequate preservation during these periods.

Operation and maintenance errors many times have the same origin as the assembly mistakes: human error. The same type of approach is recommended.



FIGURE 1.5 *Gear coupling with signs of severe wear due to the lack of lubrication.*



FIGURE 1.6 *Centrifugal pump impeller showing signs of wear due to operation with a low flow.*

Figure 1.5 shows a classical example of a maintenance-related failure, a gear coupling with severe wear to its teeth due to lack of lubrication. Figure 1.6 shows an example of an operation-related failure, wear on the discharge side of a water pump impeller due to operation with a very small flow. In spite of the cavitationlike appearance, the occurrence on the discharge side shows that the problem has not been caused by the suction conditions of the pump. In this case, the operation of the pump with a low flow has been caused by a system design error.

Conclusion

Analyzing a failure can be like watching a movie backward, looking for the reasons for what happened at the end of the history. However, unlike the movie, the failure analyst has no record of all the events at his or her disposal. Although the investigation can bring to light most of the history, quite often some parts of it remain unknown.

All the information and techniques contained in this book are intended to help bridge those gaps and construct a history that makes sense and, with some luck, can help avoid the repetition of the observed failure.

A word of caution: The complexity of modern machinery and the risks involved in the operations of process industries recommend that nothing be changed in the design, operation, and maintenance of machines and plants before all the consequences of the intended modification are thoroughly evaluated and a risk analysis performed by a competent professional.

2

Failure Analysis Practice

This chapter contains a description of the failure analysis procedure, including techniques, precautions, and the use of databases to organize and retrieve the results of the analysis effort.

2.1 Failure Analysis Objectives

The main objectives of machinery failure analysis are

1. Improve the reliability of the plant.
2. Reduce maintenance costs.
3. Reduce the risk of accidents and pollution.

All these objectives can be achieved when subsequent failures are avoided and the reliability of the individual machines is improved, which can be done through some preventive action to remove the root causes of the failure. A failure analysis that does not result in the removal of the root causes of the problem is of no value, because if the root causes are not found, future failures cannot be prevented.

2.2 How Far Should We Go?

How much time and effort can one spend analyzing machinery failures? A lot of other things must be done in the plant, and just like everywhere else, priorities must be set to answer this question.

The possible consequences of the failure is the guide to the amount of failure analysis effort. Most process plants place great value on safety, environmental, and production issues; therefore, the failure analysis effort should be directed to the failures that may cause the greatest impact. High-impact failures must be analyzed much more carefully than low-impact failures.

It must be stressed that the highest potential consequence of the failure has to be considered, instead of the actual impact. The reasoning is that, once a certain failure occurs, the consequences are due to chance alone. Therefore, the same failure, a mechanical seal leakage, for example, can cause no other problems than the replacement of the seal or a fire that can destroy the plant.

The recommended criteria for setting priorities is the following:

1. Nonrepetitive failures that can have no impact on safety, the environment, and production should be analyzed by the person in charge of the repair and his or her supervisor. A simple process (the five whys) can be used to help organize the analysis.
2. Repetitive failures or failures that may have an impact on safety, the environment, or production should be analyzed more carefully, normally by a multidisciplinary group.

The meaning of *safety impact* and *production impact* may be somewhat obvious, but *repetitive failures* needs definition. A repetitive failure is one whose mean time between failures (MTBF) is smaller than the plant's MTBF. If a certain bearing fails once a year and the average for the whole plant is a two-year life for the bearings, this can be considered a repetitive failure.

These criteria for setting priorities are similar to those used by many process plants and should be adjusted to a particular plant, if necessary. Both processes are discussed briefly.

2.2.1 A Simple Analysis Process: The Five Whys

A very good indication that a plant has a widespread reliability culture is the behavior of the shop floor employees. When facing a failure, people should ask themselves two questions: (1) Why did it happen? (2) What can be done to avoid repetition of the failure?

The process known as the *five whys* was created in the 1950s and is a direct consequence of human curiosity. It consists in asking for the cause of something, the cause of the cause, and so on, as shown in Table 2.1. When this question is repeated five or six times, there is a good chance that the root causes of the failure have been approached. Afterward, it is possible to devise preventive actions and an implementation plan.

This process works fine because people are naturally curious and try to learn more about what happens around them, if they have a chance. This process is nothing more than a logical arrangement of the normal thinking process and reduces the chance of someone drawing a conclusion too fast about the problem.

The biggest advantage of this process is simplicity. The mechanics and supervisor can do it themselves. The biggest drawback is that the analysis is limited by the knowledge of the people performing it, which can be avoided by providing training and technical support.

TABLE 2.1 *Example of the Five Whys*

<i>Question 1</i>	Why was the pump shut down?	Because of a seal leakage.
<i>Question 2</i>	Why did the seal leak?	Because the sealing faces were rough.
<i>Question 3</i>	Why were the faces rough?	Because the faces became too hot and the flush vaporized.
<i>Question 4</i>	Why did the flushing fluid vaporize?	Because it was too hot at the entrance to the seal gland.
<i>Question 5</i>	Why was the flushing too hot?	Because the flush cooler was plugged with debris from the cooling tower.
<i>Question 6</i>	Why was the cooler plugged?	Because it had not been washed after the cooling tower cleaning.

2.2.2 *An In-Depth Analysis Process*

When a certain failure may affect production, safety, or the environment, a more detailed and formal analysis is desirable. It is also desirable to avoid the limitations of the simplified analysis just described.

An extension of the “five whys” process can be done by a multifunctional group, where operation, maintenance, and engineering are represented. Depending on the consequences of the failure, safety, health, and law professionals also may be involved. Sometimes, even a representative of the shareholders or the workers’ union may take part in the analysis. When people from the various departments of the corporation are present, a more balanced view of the events is possible. The multifunctional group also is more knowledgeable than any individual.

A formal procedure is recommended. All the steps of the work must be written down so that the process can be thoroughly tracked. The main steps are

1. *Data collection.* It is impossible to analyze something we do not know. The first step in any failure analysis job is data collection, and this must be done in the field. All the possible information about the failure must be collected firsthand, as much as possible avoiding any conclusions without supporting evidence. The data to be gathered include the failure mode analysis of the components, maintenance and operation records, original installation and startup reports, vibration data, performance data, operator and maintenance personnel information, design data, and the like. Not all the cases will require this much information, and sometimes important data can be missing.
2. *Data organization.* The information collected must be organized to “tell the history of the failure.” Any formal process may be used to organize the data, such as a fault tree or Ishikawa diagram. It should be possible, at this point, to recreate the sequence of facts that lead to the failure.
3. *Data analysis.* Is the relationship among all the events correctly expressed in the history of the failure? Which events should be modified or removed to avoid any chance of future failures of

this sort? At this point, we look for equipment, process, people, and management issues that may be related to the failure. Note that every failure happens because someone has done something wrong or failed to do what was necessary. This means that an effective failure analysis uncovers root causes that lie well below the surface. For example, if a machine fails because of a wrong operation done by an operator, maybe this wrong operation was induced by the lack of a process instrument, which might be traced to an outdated design practice. If the operator is blamed for the failure, the real root cause is not uncovered and another operator may make the same mistake.

4. *Preventive measures and reports.* Preventive measures should be determined only after all the root causes are uncovered. It is easy to understand that, by not doing so, one runs the risk of solving the wrong problem. The failure analysis report should be as simple as possible, containing a description of the failure investigated, identification of the equipment involved, the reason for the investigation, the effects of the failure, the data collected, root causes, proposed preventive measures, and economic justification for the preventive measures. The procedure to be used for the future evaluation of the effectiveness of the preventive measure also should be listed.

2.3 Main Steps

A discussion of the main steps of a complete machine failure analysis follows. Not all failures require such a comprehensive analysis; the procedure outlined must be adapted to the situation. Sometimes, it is not possible to obtain all the information called for. In these cases, the gaps in information must be filled with reasonable assumptions. However, the accuracy of the analysis is greatly affected by the amount of information available.

The basic physical, chemical, and engineering concepts are applicable to failure analysis, which means that they should not be contradicted. It is not very likely that a new physical, chemical, or engineering law be discovered during the analysis of a machine failure.

Should this happen, it is wiser to suspect that the analysis is wrong and the available information should be verified.

The sequence of events that lead to the failure must be compatible with the way the mechanism works. The dimensions and the interactions of the components of the failed mechanism should be taken into account. The examples in Chapter 20 can clarify this point. In addition, the appearance of the damage surface can provide a lot of information about the failure mode, if we are able to understand it. The shape of a fracture or the scratches on a worn surface can give us many clues about the origin of the problem. Parts II and III provide more information on this topic.

2.3.1 Data Collection

Component failure analysis is easier if we know the history of the component and of the machine, from design to operation, including manufacturing, installation, and maintenance. It is impossible to find the root causes of a failure without knowing the related facts. Therefore, the first step to a successful failure analysis is to collect information about the following:

1. Design of the machine and the component, including dimensions, loads, stresses, natural frequencies, and the like.
2. Manufacturing reports, including chemical analysis, mechanical properties, other inspection reports, manufacturing process, mechanical running, and performance testing.
3. Operation reports, including information about the work environment, actual loads, temperatures, flow, pressure, fluid composition, and so forth. Information about abnormal conditions should be searched carefully. Keep in mind that an abnormal condition may happen for a short period and not be perceived by the operators. Careful examination of the operation reports and computer databases may uncover the abnormality.
4. Maintenance reports, including vibration analysis and previous failure analysis.
5. Pictures should be taken liberally. A good photographic report is a valuable complement to the written reports.

6. Sample selection is to be done quite carefully. The samples selected for analysis should represent the failure. It is of very little use to analyze a component that played no role on the failure. The initial approach always should be a complete visual inspection of all the components in the field. Afterward, the inspection efforts should be concentrated on the first components to fail.
7. Description of the mechanism, including dimensions, interactions between components, and the like.

As mentioned earlier, sometimes, all the information listed is not available. It is possible to analyze a machine failure without all the preceding information, but this should be done with extreme care. Every assumption made to fill an information gap, no matter how reasonable it may seem to be, is an opinion, not a fact.

2.3.2 Inspection and Testing

Damaged parts should be subject to a rigorous visual inspection even before disassembly and cleaning. The ideal situation would be to draw a “police line” that can effectively isolate the “crime scene” until it is examined by the “detective.” If anything is changed on the damaged machine, important evidence may be lost. Even the removal of seemingly unimportant debris should be avoided.

1. In visual inspection and photography, particular attention should be paid to fracture surfaces, worn or corroded surfaces, and any other damaged area. At this point, the failure mode might be identified. The preservation of the damaged surfaces is of utmost importance. Any modification that may alter the appearance of the damaged area may cause the loss of valuable evidence.
2. Nondestructive testing can be used to access the complete extension of the damage, by revealing other cracks or loss of material. The type of examination must be decided after the visual inspection of the failed machine.
3. Mechanical testing may be used to check material properties. The comparison of the originally specified mechanical properties

with those found in the failed part may uncover a material problem, such as the use of incorrect material or a deficiency of the heat treatment of the part.

4. Chemical analysis of the component material or residues found on the damaged parts may provide other clues to indicate the source of the problem.
5. Mechanism failures always should be reenacted as much as possible, to make sure that the proposed sequence of events is feasible. This proposed sequence of events must be possible with the actual interactions among the components.

2.3.3 Identification of the Failure Mode and Root Causes

This is the last step before the specification of the preventive measures. The identification of failure modes must precede the identification of root causes. Failure modes are related to machines, components, and operating conditions; root causes are related to people and work processes. More information about failure modes can be found in Parts II and III.

2.3.4 Points to Highlight for a Successful Analysis

The preceding section describes a general failure analysis procedure. Some other precautions may be helpful. Of course, we have to follow the well known general procedure: Collect information, organize the information, analyze the data, determine the failure mode, find out the basic causes of the failure, determine the best corrective actions, implement these corrective actions, and evaluate the results. This may take a lot more time and effort than it seems when we see these main steps listed so briefly.

Even if we stick to the general procedure, many things can go wrong if we do not pay attention to some other, sometimes forgotten rules. These are described next:

1. *Focus on the first component to fail.* When we examine WE shall never forget that, in most cases, only the event that has been the trigger to the failure matters to us, everything else is only consequences of this first failure. Many times, I found myself having

to persuade people not to examine all the remains of a failed machine, as the origin of the problem could be clearly identified.

2. *Do not forget to look at the weakest component on the failure sequence (or on the possibility of hidden failure modes).* This will be the exception to the first rule. Sometimes, a simple problem can destroy a machine just because there was some hidden “weak link” on the chain. This could have been the case when, for example, a small auxiliary oil pump steam turbine failure caused a huge air blower to trip, with all the deleterious effects in the process. Of course, there was a spare oil pump, and the spare pump worked properly. The problem here had been a jammed check valve that drained the oil on the emergency tank before the spare pump had time to accelerate. This hidden failure mode is one of the biggest nightmares of the instrumented safety system designer. It may happen on any component whose failure is not obvious until this component is put to work. This is easy to understand when we think of a jammed safety valve: The problem may go unnoticed until the jammed valve is required to operate and an accident happens.
3. *Look for more than one root cause.* It can be difficult to resist the temptation to finish a failure analysis when we find one valid root cause. This is exactly what we have been searching for, so it may seem that the job is done when we find one of the root causes of the event. This will not be true every time and we have to look always for more than one root cause. The main risk we run when we do not discover all of them, or at least the main ones, is that we can implement corrective action to avoid repetition of only the root causes we know. The problem is only partially solved, and the root causes that have not been eliminated may show up to repeat the failure we thought we would never see again.
4. *Examine the maintenance and operational history of the machine.* It is well known that past and present maintenance and operating conditions can greatly influence the machine life. So, it becomes obvious that they should be taken into account.
5. *Develop a machinery failure analysis database.* Our brain tends to remember things selectively. The general rule says that we

remember more easily the things that do not conflict with our overall conceptions about the world and are closer to us in time and space. Here lies the importance of using a computer database for memory expansion. We simply are unable to remember everything, so this is a great help when we have to decide the most effective ways to improve machinery reliability. It is not enough to write failure analysis reports and maintenance checklists. Reports should be used as a communication tool to inform other people in the organization on the conclusions and what must be done next. The computer database should be used, as pointed out before, as a memory expansion device.

6. *Take into consideration the operation mode of the failed mechanism.* This part may need some further explanation. In other words, we seek an explanation for the failure that does not conflict with the way the damaged machine works. The interaction among the components and between the components and working fluids, in most instances, will determine the failure mode. Another instructive way to improve our failure analysis capabilities is to analyze the way the mechanism operates and use this information to help explain the observed damage. An in-depth knowledge of the way a machine works makes failure analysis a lot easier.
7. *Learn why some machines do not fail.* Does this sound obvious? Sometimes, an easy way to improve the reliability of a certain machine is to find a similar machine in a similar service that shows good reliability. When we compare both machines, many times we discover that “equal” machines are not as “equal” as we thought, the difference being the solution to our problem.

2.4 Reports and Databases

On completing the failure analysis, its results are written down. Sometimes, they may be enough to fill in a summary of the information collected in a failure analysis database; other problems may require extensive reporting. Both reports and databases are needed for different reasons. A computer database is used as a memory ex-

pansion; a report is used as a communication tool. They complement each other.

The following information must be introduced in the database:

1. Equipment data, such as ID, manufacturer, serial number, model number, brake horsepower, fluid, operating temperature, and discharge pressure.
2. A summary of all the maintenance jobs, including the reason for the repair, dates, time to repair, what work has been done, the failure mode, root causes, cost of the service, and the like.

The database must be able to produce statistical reports based on the input data. It should be possible to find out, for example, which pieces of equipment show the least availability, which are the most common failure modes, and so forth.

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3

Failure Prevention Efforts

If the number of machinery failures in a certain plant is too large, everybody will concentrate on the “firefighting” and may lose sight of the whole situation. The safe, fast, cheap, and effective repair of failed machines is only part of the job. Even though the lessons learned through the analysis of failures should be incorporated into the day-to-day life of the plant, it should be considered a reactive action, as it is done after the fact. Effective proactive actions also should take place in a competitive plant. Some examples follow; an in-depth discussion is beyond the scope of this book:

1. Predictive maintenance, such as vibration and performance monitoring and analysis.
2. Preventive maintenance, such as cleaning and lubrication.
3. Spare parts management.
4. Operator and mechanics training.
5. Reliability-centered maintenance.
6. Benchmarking, to know the objectives.

This chapter contains a brief discussion of how to organize a plant to avoid equipment failures, using the information obtained through a failure analysis program and other methods.

3.1 Types of Failures

Spend some time in a process machinery repair shop and you will be amazed by the vast amount of different types of failures that can exist. Each piece of equipment has dozens, maybe hundreds of parts. Each part may fail in several different ways. Each process plant has hundreds of machines installed. If all the possible failure modes of all the components of all the machines are put together, too large a number of possibilities is created.

The normal way to handle big problems is to divide them into smaller ones and solve the most important one first. Therefore, similar types of failure modes can be grouped and a common solution applied to them. For example, reduced mechanical seal life is a common problem within the processing industry. Many strategies can be used to face this issue, including standardization, design reviews, and so forth.

A failure source to analyze carefully is human error. The quantity of errors made by a person is a function of the level of training, motivation, stress, and the support provided that person by the instruments and supervisors. This source of failure includes maintenance and operation errors caused by the ineffective action of the involved persons. Some general rules to avoid human errors include these:

1. Do not assign people for continuous observation of process parameters; this is best done by a computer.
2. All possible operation tasks should be automated, including emergency operations.
3. Every person involved should be trained, and this training reinforced from time to time. Periodic auditing can prove a powerful tool to avoid problems.
4. Adequate tools and equipment should be provided.

3.2 Prevention of Failures

The next step after a successful failure analysis is the implementation of preventive measures. A word of caution: No design, operation, or maintenance modifications should be done without a careful examination of all the consequences of the modification, including a risk analysis.

Preventive action can be quite varied, ranging from the modification of an operating procedure to the replacement of the failed machine. The huge number of possibilities makes it impossible to do more than just mention some examples:

1. Component standardization has enormous advantages—reduction of inventory, standardization of maintenance and operation procedures, and reduction of mean time to repair. Many components can be standardized easily, like mechanical seals, couplings, antifriction bearings, small turbine governors, and lubrication oil. The introduction of a state-of-the-art component can solve many reliability problems, especially in older plants using outdated equipment.
2. Operation and maintenance procedures can be modified to reduce failures.
3. Machines and installations may need design modifications, such as installation of new mechanical seals or more effective lubricants, improvements in the bearing housing sealing, modifications in control systems to avoid cavitation, or installation of modern antisurge systems.

The effective implementation of the preventive action depends on the importance of the problem to the plant. In the process plant industry, problems related to safety, environment, production reliability, and cost are ranked very high on the priority scale.

When the results are evaluated, the use of key performance indicators (KPI) can be helpful. The most useful KPI to evaluate rotating machinery are mean time between failures, mean time to repair (MTTR), and maintenance cost. Of course, safety and environmental indicators are included.

3.3 Machinery Monitoring and Anticipatory Action

Failure analysis is a powerful tool to improve machinery reliability. Nevertheless, keep in mind that it is not necessary to wait for any machinery failure to apply the best practices known to the industry.

In addition to analyzing failures and implementing preventive action, it is necessary to avoid foreseeable failures. This is the reasoning behind preventive and predictive maintenance strategies. Therefore, machine monitoring programs should be instituted as soon as possible. The same should be done with preventive maintenance programs, operator and mechanic training, and the like.

Machine monitoring should include

1. Performance monitoring, with periodic or on-line monitoring of head, efficiency, absorbed power, and a comparison to baseline values.
2. Condition monitoring, including vibration, bearing temperature, oil temperature, noise, seal leakage, rod drop, valve temperature, and the like.
3. Preventive maintenance, such as lubrication, cleaning, and periodic inspections.
4. Critical machinery without spare components should be monitored continuously.

3.5 Operator's Role in Machinery Reliability

Many times, process plant operators are blamed for machinery failures. Although this may be true in some places, the best plants recognize that the operators play an extremely important role in machinery reliability. This can be easily understood when we compare a careless driver with a conscientious one, normally the latter will have fewer problems with a car. The same applies to process plant operators, who should be trained and motivated on how to get the most out of the equipment.

Some examples of actions taken by a reliability focused plant, in addition to providing operator training and other working conditions, are

1. Instructing the operators about the design limits of the equipment and the consequences of off-design operation.
2. Instructing the operators on how the machines are designed and built, so they can understand the possible consequences of their acts.
3. Installing automatic monitoring of process and condition parameters, with adequate alarms and automated trips. This keeps operators doing what a computer cannot do for them, that is, inspecting the plant to make sure everything is working well.
4. Involving plant operators in all jobs related to reliability, for example, discussion of the predictive maintenance strategy and failure analysis.

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Part II

Failure Modes

This part presents a brief discussion of the main failure modes observed in process machinery components. Not all possible failure modes are included in this discussion, which focuses on those most frequently found. The Bibliography at the end of the book may be consulted for further information.

The first step to understanding machinery failure is the correct classification of its type, followed by a comparison of the observed features of the failure at hand with the existing knowledge. This way, it is possible to recognize the failure mode.

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4

Ductile and Brittle Fractures

Most components found in normal process machinery are made of metal, especially steel. Metals have a crystalline structure; therefore, the atoms are arranged according to a certain order. Inside this crystalline structure, many discontinuities are found, such as voids and inclusions, that have great influence on the material properties.

The fracture of metals under constant load and low temperature (below half the fusion temperature of the metal) may cause two mechanisms, provided no other external factor, like corrosion, has an effect:

1. Brittle fracture, in which a crack propagates unstably across the whole cross section of the component. Unstable propagation occurs under a constant or reducing load, which means that propagation continues, due to the internal stresses, even if the external load is reduced or removed. In this case, the crack tip is in a region where only localized plastic deformation can be found. The fracture surface shows no macroscopic plastic deformation, the separation occurs due to cleavage between the crystalline planes caused by large enough tension stress.

2. Ductile fracture, in which the cross section of the component is reduced due to the macroscopic plastic deformation caused by the slippage between the crystalline planes. The slippage is caused by large enough shear stress. The fracture propagation is stable, which means that propagation stops if the load is reduced. The fracture surface shows macroscopic signs of plastic deformation. A ductile fracture can be understood as the ultimate plastic deformation of a component.

A component may suffer a brittle or ductile fracture, depending on the circumstances. A reduction in temperature, for example, can be enough to modify the fracture behavior of a material.

4.1 Ductile Fracture Morphology

A ductile fracture surface normally presents three distinct regions:

1. A fibrous zone, which corresponds to the initiation site of the fracture; the fracture propagates stably in this zone. This fibrous zone is formed on the region with the highest stress triaxiality, which means it is close to the centerline of the component or near stress concentrators.
2. A radial zone, which corresponds to an unstable propagation region. This region shows a rough surface and radial marks diverging from the fibrous zone. Its appearance is similar to a brittle fracture.
3. A shear lip zone, which shows an inclination of approximately 45° to the external load direction. This is the region where stress triaxiality is reduced and shear slip of the crystalline planes is possible. The 45° inclination corresponds to the direction of maximum shear stress when the external load is axial.

These regions are shown schematically in Figure 4.1. Figure 4. 2 shows a part that suffered ductile fracture, with the macroscopic plastic deformation clearly visible.

The morphology of a specific fracture is a function of the circumstances. The conditions that may originate a brittle fracture or reduce the macroscopic plastic deformation observed in a ductile

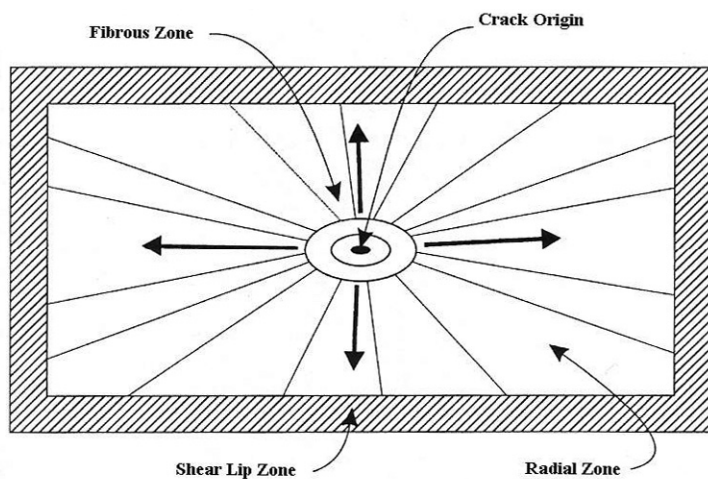


FIGURE 4.1 *A ductile fracture showing the fibrous, radial, and shear lip zones.*

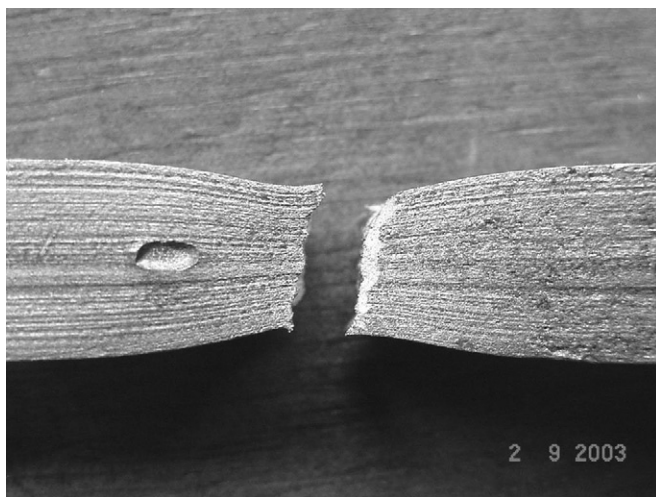


FIGURE 4.2 *A part that suffered a ductile fracture, showing the macroscopic plastic deformation.*

fracture are the same as those that make crystalline plane shear more difficult: low temperature; impact or other high-speed loading; high stress triaxiality, as found in notches, for example; or a material with fragile behavior.

If the fracture has not been caused by pure shear stress, a microscopic examination of a ductile fracture surface shows that the ductile fracture mechanism includes the formation and coalescence of microvoids, which may be somewhat spherical or parabolic. These microvoids are sometimes called *dimples*. The shape of the dimples can be related to the type of loading that caused the fracture; more traction originates spherical dimples, more shear loading causes the dimples to be more parabolic. The dimples are not observed in a pure shear ductile fracture.

4.2 Ductile Fracture Mechanism

The macroscopic plastic deformation observed on a ductile fracture is the result of shearing of the crystalline planes, which slip past one another. This slippage occurs across the direction of highest shear stress.

Figure 4.3 illustrates the behavior of a crystalline structure subject to a shear stress greater than the strength of the material. The slippage of the crystalline planes is clearly seen.

Dimple formation seems to be related to the stress concentration on the edge of a slip band that has been locked in by a defect in the crystalline grid, such as an inclusion. This stress concentration

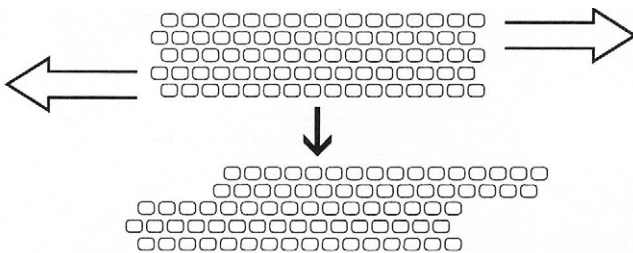


FIGURE 4.3 *The deformation mechanism caused by shear loading.*

would be relieved by the rupture of the inclusion or the rupture of the interface between metal and inclusion.

Note that dimples also might be observed in a brittle fracture of high strength steel. Ductile fracture always should be classified based on the macroscopic plastic deformation.

4.3 Brittle Fracture

A brittle fracture can be recognized because there are no macroscopic plastic deformations. Brittle fractures can be much more deleterious to machine components than ductile fractures and more difficult to avoid during the design of the machine, because of the following characteristics:

1. Brittle fractures occur when the material is subject to stresses that are smaller than the yield limit of the material. Machine design normally is based on ductile material; and the design criteria are meant to avoid plastic deformation and, in certain cases, elastic deformations. If some circumstance causes the material to behave in a more fragile manner than considered during the design of the machine, a brittle fracture may result.
2. Crack propagation in a brittle fracture is unstable, which means that, once started, the brittle fracture may propagate across the whole cross section of the component due to the internal elastic stresses only, even if the external loads are reduced. This can cause a catastrophic failure of the component and the machine.

Some methods used to avoid brittle fractures include

1. Selection of structural materials that show a ductile behavior under all anticipated operating conditions, including some abnormal situations.
2. Avoidance of triaxial stresses that occur in notches or other types of stress concentrators, like fillets, transitions, or thick areas.
3. Avoidance of impact loading or include a means to absorb the impact energy.

4. Avoidance of the tendency of hydrogen to embrittle steel, with the selection of the correct types of material and heat treatment, if exposure to hydrogen can be anticipated during operation or manufacturing processes like electroplating.

4.4 Brittle Fracture Morphology

The surface of a brittle fracture normally shows radial marks that propagate across the fracture surface, from the initiation site to close to the component periphery. These marks are formed by the interaction of the different frontlines of crack growth. They tend to disappear close to the periphery of the component because the reduced thickness reduces the triaxiality of the stress, making macroscopic plastic deformation possible. Components that have one dimension significantly smaller than the others present the characteristic chevron marks, which point to the fracture initiation spot.

This characteristic is useful to help find the origin of the fracture. The origin of the fracture is the most important location on the fracture surface, an in-depth analysis may reveal the exact damage mechanism. The following features of brittle fractures may help locate the initiation point:

1. Radial marks radiate from the initiation point. Chevron marks point toward the initiation spot. Chevron mark formation depends on the material characteristics and may not occur.
2. Fractures that initiate close to the core of the component may show a shear lip close to the component surface. Fractures that begin on the surface do not show this shear lip.
3. Structures fabricated with steel plates may show bifurcations of the cracks in the propagation direction.

Figure 4.4 shows a brittle surface, where the radial marks can be seen. This component is the spring of a relief valve, made of hardened steel.

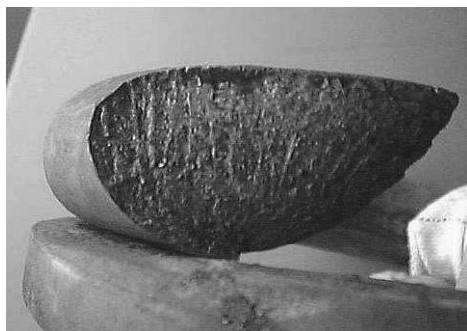


FIGURE 4.4 *Brittle fracture surface of a hardened steel component showing radial marks pointing toward the initiation point, on the lower side.*

4.5 Brittle Fracture Mechanism

A brittle fracture occurs in a direction perpendicular to the greatest tensile stress and is characterized by the separation of the atoms without slippage between the crystalline planes. This fracture mechanism is observed when such slippage is not possible, which happens when shear stress is small compared to tensile stress or material property. Figure 4.5 illustrates the concept.

A brittle fracture occurs by the separation of the atoms due to a tension stress. If the separation plane is coincident with one of the

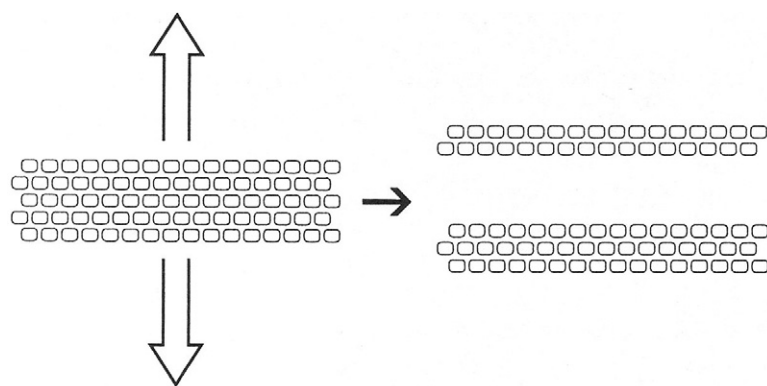


FIGURE 4.5 *The brittle fracture mechanism.*

crystalline planes of the material, the mechanism is known as *cleavage*. Under a microscope, the fracture surface shows the characteristic cleavage appearance, with the surface of each grain oriented at a slightly different angle. Some variations of this mechanism are possible, when the fracture plane does not follow the crystalline planes exactly. This is called *quasi-cleavage*, and the microscopic appearance of the fracture surface does not show the distinct cleavage planes of each grain. Dimples may be observed, sometimes.

Note that there would be tensile and shear stresses in any component subject to external load. The relative magnitude and internal distribution of these stresses, together with material susceptibility, are the main factors influencing the tendency of a fracture to be brittle instead of ductile.

Figure 4.6 illustrates two stress distributions that can result in quite different fracture mechanisms. On the left side, plane stress is illustrated. The plane stress condition can be found when the stress in one direction is much smaller than that in the other two directions, in thin components, for example. It can be seen that shear stress is large relative to tensile stress, a condition that makes plastic deformation and ductile fracture more likely. The right side of the figure shows a triaxial stress distribution. In this case, which can be found in components that are thick or have large stress concentrations, the third tensile stress greatly reduces the shear stress, reduc-

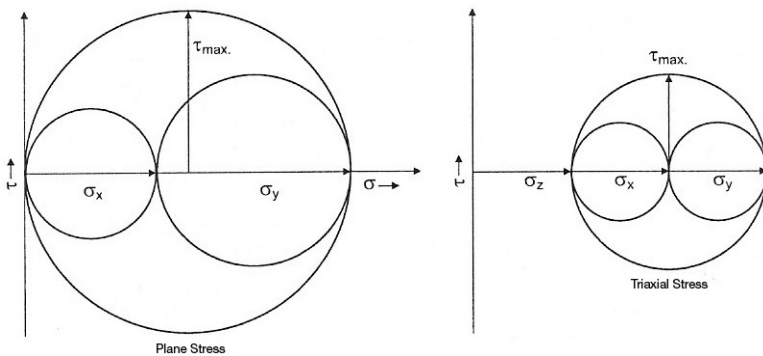


FIGURE 4.6 Mohr's circle, illustrating the reduction of the highest shear stress in a triaxial stress situation.

ing the chance of plastic deformation and increasing the chance of a brittle fracture.

The mechanism that creates stress triaxiality in stress concentration points even when the external load is uniaxial can be explained as follows: Suppose an external tensile load is applied to a notched part. The external load causes an elastic elongation in the same direction. The elastic elongation caused by the external load induces a contraction in a perpendicular direction, due to the Poisson effect. If a notch is present, the tensile stress in the direction of the external load is larger closer to the notch tip, if compared to the stress at another point closer to the component surface. Greater stress leads to greater elastic deformation, and different parts of the component are subject to different amounts of stress and elastic deformation. When one region of the component is subject to an elastic deformation different from that of an adjacent region, the need to retain continuity of the material causes stress to develop between the two regions. This is the source of stress triaxiality that makes it easier for a fracture to be brittle when a notch is present. Figure 4.7 illustrates such stresses.

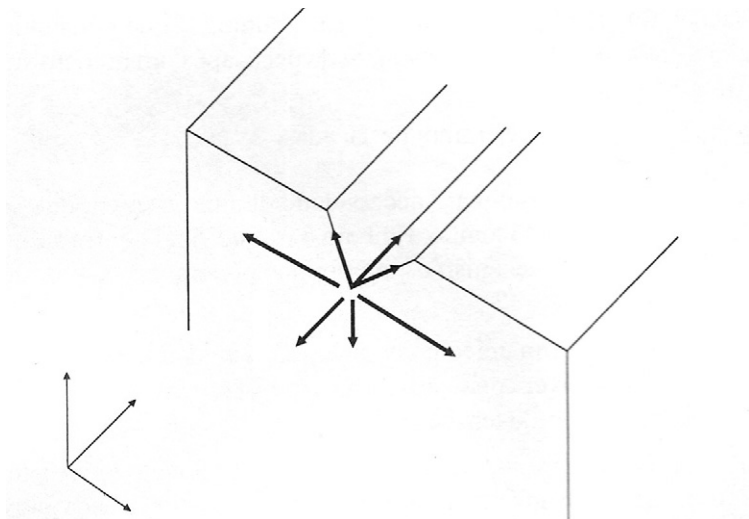


FIGURE 4.7 *The development of triaxial stresses due to a notch.*

The simplest way to evaluate a material's susceptibility to brittle fracture is impact testing. One such test, the Charpy test, uses a notched test bar made of the material of interest, which is subject to the impact of a pendulum. The amount of energy absorbed in the impact is used as a measure of the material toughness. This test was developed as the result of the analysis of the fractures of the liberty ships, which tended to break in two parts for unknown reasons, sometimes when anchored to the harbor. The investigation found that the material toughness was greatly reduced due to the low temperature, which made brittle fracture possible.

Energy absorption in the Charpy test depends on the type of material and the test temperature. Almost all types of steel absorb little energy and display a brittle behavior if the temperature is low enough. Subsequent tests at increasing temperatures show, initially, that the absorbed energy does not increase significantly, up to a certain temperature, which causes a significant increase in the absorbed energy. Further increases in temperature cause small increases in energy absorption and the fracture shows a ductile behavior. This is the transition temperature, which marks the transition from brittle to ductile behavior.

5

Fatigue Fractures

If a rotating machine component is found broken, fatigue is the most likely failure mode. This is due to the oscillating loads present when the machine is operating. The current definition of fatigue is the following: “A *progressive* and *localized* process that causes permanent structural modifications in the material. This process occurs due to conditions that cause fluctuating stresses and strains in one or more locations and can result in the formation of *cracks* and the *complete rupture* of the component after a sufficient number of cycles” (this definition is adapted from H. O. Fuchs and R. I. Stevens, *Metal Fatigue in Engineering* [New York: John Wiley & Sons, 1980]).

Four key words have been highlighted:

- *Progressive* implies that a certain period of time is needed for the process to develop, although the final fracture occurs suddenly, normally without warning.
- *Localized*, as a crack can be initiated in a single point within the component, if local conditions allow.
- *Complete rupture* is the consequence of *crack* growth above a certain critical size. This critical crack size is that which allows the uncontrolled crack growth to final fracture of the part.

High cycle fatigue is the term used to designate the situation when more than 1,000 to 10,000 cycles are needed before final rupture. In this case, average stresses are usually below the yield strength of the material, and most of the time is spent to initiate a crack. Consequently, macroscopic plastic deformations can be found at the final fracture area only. This is the fracture mode most frequently found in machinery components. The term *low cycle fatigue* is used for the opposite situation, that is, when a smaller number of cycles is necessary before fracture. In this case, most of the time is spent in crack growth and stresses usually are higher than the yield strength of the material. Macroscopic plastic deformations can be found in any part of the component.

5.1 Fatigue Fracture Mechanism

The atoms within a metallic part are arranged in a certain order, which is known as the *crystalline structure*. Some crystalline planes within this structure slip across their surrounding planes more easily than others.

Each grain within the metal has its crystalline structure oriented in a different direction, which makes each more prone to deformation in a different direction. When an external load of sufficient magnitude is applied, shear stress causes some of the crystalline planes to slip. If the load is steady, deformation may happen in one direction only; this deformation may be observed on the material surface as shown in Figure 5.1. Cyclical loads may cause deformations in opposite directions. The deformations caused in one direction will not be exactly the reverse of the deformations caused in the other direction, and this process may result in a notched surface, as shown in Figure 5.2.

The deformation bands formed due to cyclical stress are excellent stress concentrators, making it easy for microcracks to develop. These microcracks may propagate following the direction of the highest shear stress until they are large enough to continue growing following a direction perpendicular to the highest tensile stress. The crack size that results depends on the material ductility, stress concentration, and so forth.

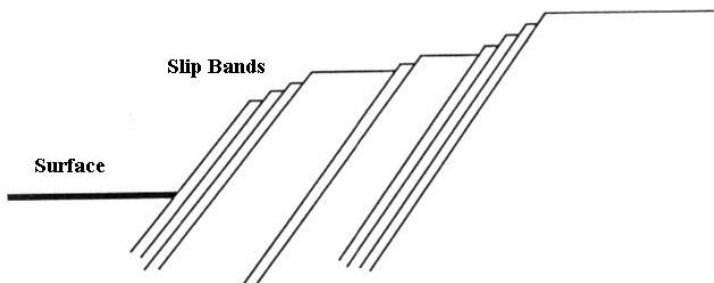


FIGURE 5.1 *The deformation bands resulting from a monotonic stress.*

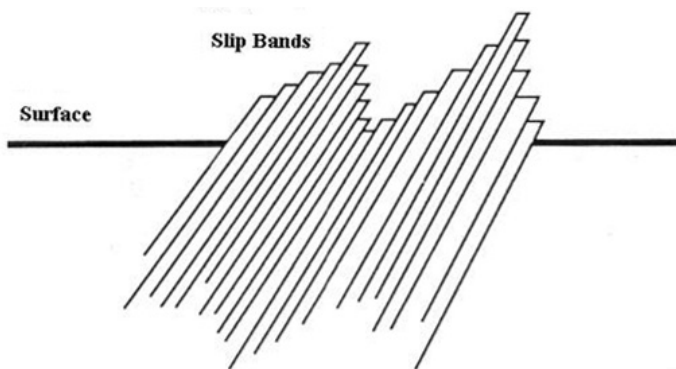


FIGURE 5.2 *The deformation bands resulting from a cyclical load.*

The relation between the deformation bands and fatigue crack nucleation was proven when the electrolytic removal of the deformation bands were found to cause an extension of the fatigue life of the test sample.

In addition to the formation of surface defects, the cyclic microscopic plastic deformation of the material tends to drive all the crystalline grid defects in the same direction. This means that voids, for example, move in the direction of the shear plastic deformation until they accumulate at the crack initiation point.

The nucleation and growth of a fatigue crack are illustrated in Figure 5.3, which shows the nucleation of the crack in the region where the deformation bands formed, the crack growth following

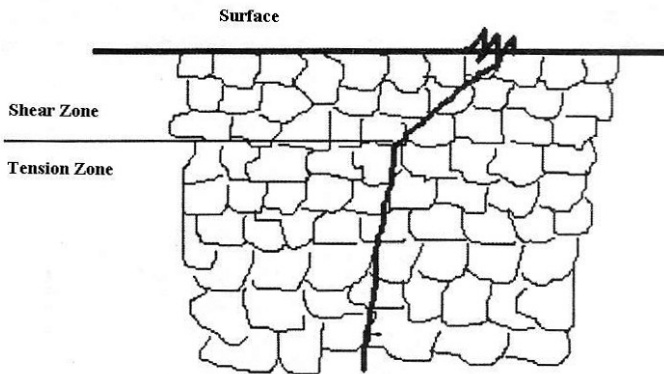


FIGURE 5.3 *The beginning of a fatigue crack.*

the shear stress direction close to the surface, and the subsequent crack growth perpendicular to the tensile stress.

The fatigue fracture process is usually divided into three parts:

1. The first stage corresponds to crack nucleation from the accumulation of discontinuities due to *localized plastic deformation*. This growth occurs across slip planes under the influence of shear stress. The damage cannot be observed with an unaided eye, as the cracks are very small. This stage may require up to 90% of the *stress cycles* to complete when high cycle fatigue is present. The presence of stress concentrators can greatly reduce the duration of this phase.
2. The second stage corresponds to the crack growth perpendicular to the main *tensile stress*. The fracture surface presents a characteristic appearance, with the progression marks (sometimes called *beach marks*) visible with an unaided eye most of the time. The second stage cracks can cover most of the fracture surface, even if not taking most of the load cycles.
3. The third stage is the final rupture of the component, which happens in the last load cycle. This final rupture happens when the crack reaches the critical crack size for unstable propagation. The final fracture area corresponds to the applied mean stress, which means that greater loads normally lead to a

greater final fracture area. Sensitivity of the material to crack propagation also plays a role; materials that are more sensitive show a larger final fracture area.

Therefore, localized plastic deformation, stress cycles, and tensile stresses are the necessary conditions for a fatigue fracture.

5.2 Fatigue Fracture Surface Morphology

The inspection of a fatigue fracture surface can be more meaningful when done with a low magnification sight glass. This examination can show several morphological details of the fracture surface that may be important evidence related to the fracture origin and the type of external loading that caused the fracture.

The most well known macroscopic characteristic of a fatigue fracture surface are the progression marks (beach marks). These marks are produced by modifications of the stress that causes the progression of the crack tip. The progression marks indicate that the fracture progression occurs in a direction that changes somewhat from time to time. They do not indicate the propagation of the fracture at each load cycle and should not be mistaken for fatigue striations, which are microscopic and do indicate the progression of the fracture at each load cycle. The progression marks often can be seen with an unaided eye. An example is shown in Figure 5.4. A change in the direction or magnitude of the external load is enough to generate the progression marks. It is possible to see fatigue fractures without progression marks if the stresses causing the propagation do not change, which cannot be achieved easily in a real-world machine.

Microscopic observation of a fatigue fracture surface reveals the propagation mechanism. Fatigue striations are almost parallel marks perpendicular to the direction of propagation. The fatigue striations are microscopic and indicate the position of the crack tip at each load cycle. Most often, they are found in ductile materials, such as the ones used for machine construction. One progression mark may contain thousands of fatigue striations.

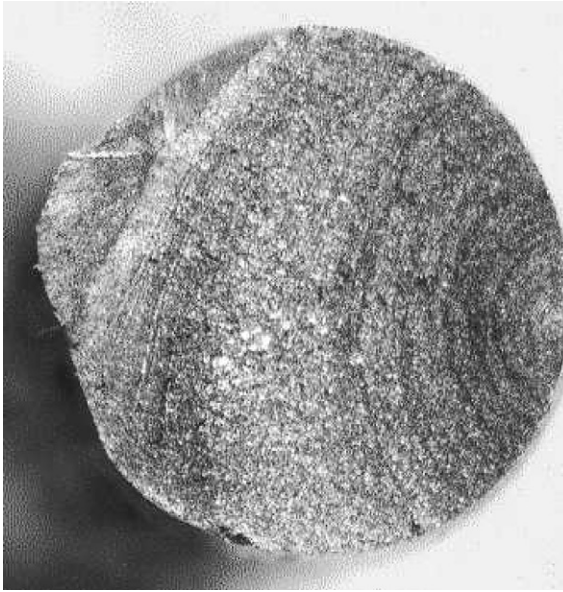


FIGURE 5.4 *Fatigue fracture surface showing the characteristic progression marks.*

A summary of the morphological aspects of a fatigue fracture surface follows:

1. One or more crack nucleation spots always exists. The nucleation spots usually are located in the place to which the progression marks converge. They may not always be visible.
2. The fatigue process involves the nucleation and propagation of one or more cracks until the final fracture. The propagation of the macroscopic cracks occurs in the direction perpendicular to the highest tensile stress.
3. The region where the progression marks can be seen may cover almost any fraction of the total fracture surface, from less than 1% to close to 100%. The final fracture area increases if the mean stress increases. It is also greater if the material has less resistance to unstable crack propagation, such as low toughness materials. Figure 5.5 illustrates these differences.

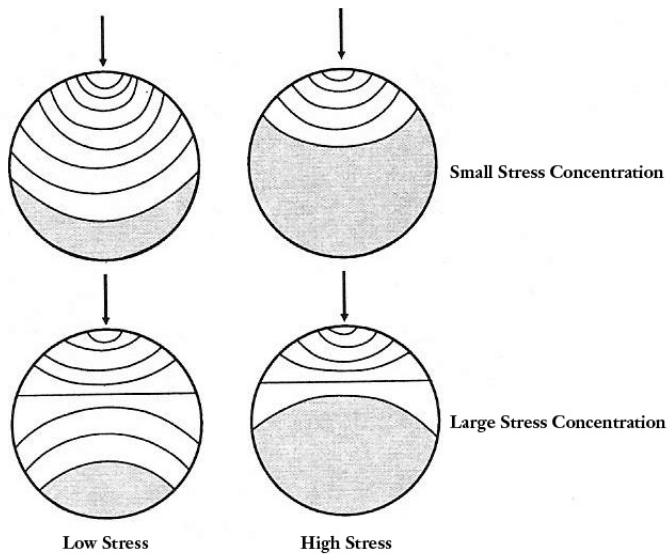


FIGURE 5.5 *The influence of stress concentration, mean stress, and material toughness on the relative size of the final fracture region.*

4. Stress concentration on the component surface results in faster propagation of the cracks close to it. The fracture surface shows deformed progression marks, which are also shown in Figure 5.5.
5. Fatigue cracks usually are nucleated on the component surface, where the stresses can be higher and other factors may contribute, like stress concentration or corrosion.
6. Multiple nucleation spots normally indicate severe stress concentration. If there are many crack nucleation points, these small cracks may merge eventually and form a large fracture that may continue to propagate. Some radial lines, known as *ratchet marks*, may be observed if the initial nucleation of the cracks occurred in different planes and steps are formed at the places where they join each other.
7. If the component suffers flexion and rotation, like a shaft, for example, the centerline of the propagation region is inclined in a direction opposite to the rotation of the shaft. This can be

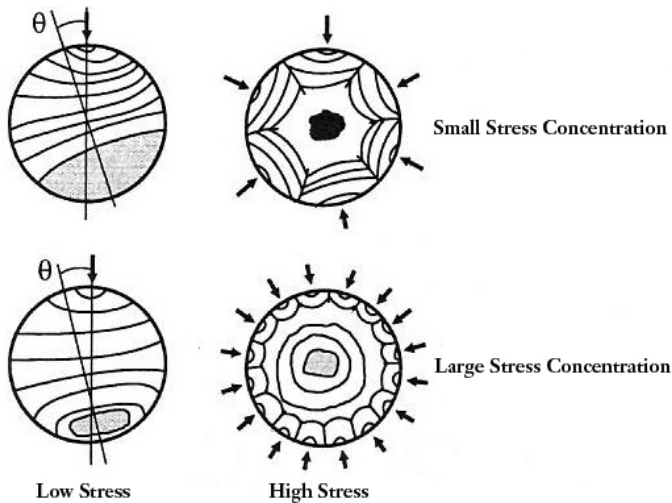


FIGURE 5.6 *The morphology of the progression marks on a rotating shaft subject to flexion.*

explained by the loss of symmetry of the stress distribution at the crack tip. The rotation of the shaft in relation to the load causes all the parts of the shaft to be exposed, alternatively, to traction and compression stresses. The crack tip tends to open and close when subject to traction and compression. When the crack region is compressed, it behaves as if not cracked and provides support for the external load, which does not happen when the crack is tractioned and opens. Figure 5.6 illustrates the concept.

8. If a component is subject to torsion, the fracture propagates at a 45° angle from the torsion centerline, as in shafts or helical springs. This direction is perpendicular to the highest traction stress for this type of loading.

5.3 Factors That Influence Fatigue Strength

Fatigue essentially is a localized process; therefore, local phenomena play important roles. Factors like stress concentrations, surface finish, residual stresses, corrosion, and the like can greatly diminish

the resistance of a component to fatigue. A brief qualitative discussion of the influence of each of these factors on the fatigue strength of a machine component follows.

The first element in the fatigue strength discussion is the number of load cycles to which the component is exposed. The number of load cycles that cause the fatigue fracture of a metal component is smaller with increasing alternating stress. In the specific case of steel components, reduction of the stress below a certain magnitude helps prevent fatigue failure of the component. This is known as fatigue strength. Other metals, such as aluminum or titanium, do not show this same behavior, and fracture occurs even with an unbounded reduction of alternating stress on the part. Even steel may not present the fatigue strength, if exposed to a corrosive environment at the same time. Figure 5.7 illustrates the relation between number of cycles and stress that cause a fatigue fracture.

Note that the fatigue strength of a certain material increases as the monotonic tensile strength increases. Even though, as the sensitivity of higher strength materials to imperfections is greater, a careful analysis is required before deciding to change a component material to avoid a fatigue fracture of a specific machine component.

Surface finish, like notches, plays an important role in the fatigue strength of a component. Fatigue is a localized process; therefore, even a small and localized imperfection may help the nucleation of a

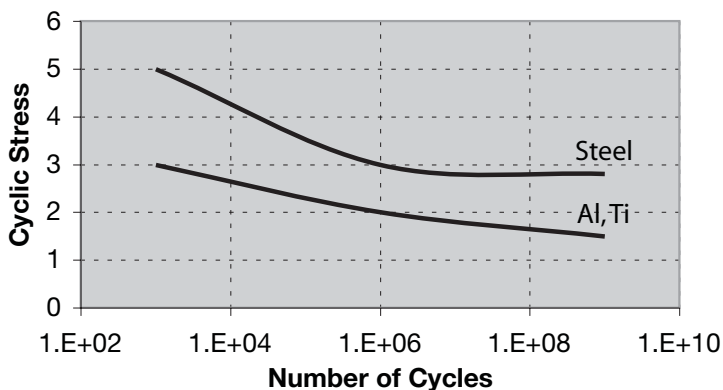


FIGURE 5.7 *Fatigue strength as a function of number of cycles.*

fatigue crack. A polished part has greater fatigue strength than a machined one. Figure 5.8 illustrates the influence of surface finish on the fatigue strength of steel components. Components with a larger surface area are more prone to some kind of surface imperfection that may be the origin of a fatigue crack.

Some design features may act the same way. Fillets, threads, notches, and others also can increase local stress beyond the fatigue limit of the material, making an otherwise safe component prone to failure. This effect is illustrated in Figure 5.9.

Residual stresses eventually present inside a component are superimposed on the stresses generated by external loads and can cause local conditions for the nucleation of a fatigue crack.

As a general rule, different failure mechanisms acting at the same time hasten the failure more than any one could. Corrosion and fretting are two common examples, which are discussed later.

In addition to normal working loads, process machinery components are subject to some situations that can bring on fatigue failure. Some examples are discussed next.

Surface fatigue is related to the contact stress between two surfaces. This phenomenon can be observed in such components as antifriction bearings and gears. The underlying theory to calculate contact stress was developed by Hertz. When two surfaces touch

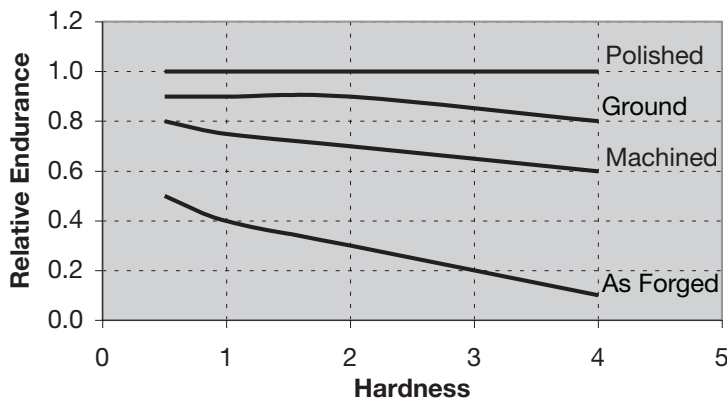


FIGURE 5.8 *The relative fatigue strength of steel components as a function of surface finish and hardness.*

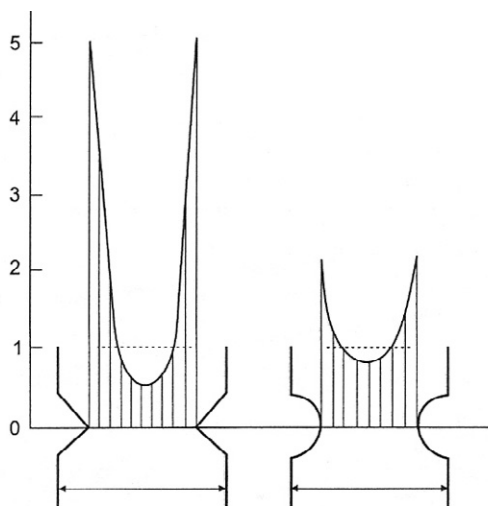


FIGURE 5.9 *The increase in local stresses, caused by a notch on the component surface, when subject to the same mean stress.*

each other, compressive stresses develop in the direction perpendicular to the surface and shear stresses originate parallel to the surface. The maximum shear stress is somewhat below the surface. The combination of these stresses, if they are cyclical, can result in the conditions for the nucleation of a subsurface fatigue crack, which usually is observed as surface peeling, if the crack is allowed to propagate. Surface fatigue is the classic failure mode of antifriction bearings and gears. Some real failures are discussed in Chapters 12, “Antifriction Bearings,” and 15, “Gears.”

Thermal fatigue is associated with thermal gradients across the machine. Thermal gradients cause differences in thermal expansion in different parts of the machine or component, and this unequal thermal expansion can cause severe internal stresses. A simple calculation can show that, if thermal growth is restrained, a temperature increase of about 80°C can stress a mild steel component to about 200 MPa, a stress level that can be higher than the fatigue strength of the material. Uneven warm-up of high temperature machines, like turbines or expanders, can result in low cycle fatigue due to the differential thermal expansion.

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6

Wear

The various forms of wear are the most commonly found failure modes of machine components. Therefore, a thorough understanding of the basic theory of wear is an indispensable prerequisite for successful machinery failure analysis.

This chapter contains a description of wear mechanisms, focusing on the mechanically-related material removal mechanisms. The discussion covers sliding wear, both lubricated and nonlubricated; hard particle wear; liquid impingement wear; and cavitation.

6.1 Sliding Wear

Sliding wear occurs when two solid surfaces slide against each other with a certain normal load. This wear mechanism is sometimes called *adhesive wear*, as microwelding of the surface roughness can play a major role in some metal combinations, although this is not always the case.

Industrial machines are built mainly with steel and other metals; therefore, this chapter discusses metal sliding wear. Both dry and lubricated wear are considered.

6.1.1 *Sliding Wear Theory and Mechanism*

The basic characteristics of sliding wear of rough metal surfaces are related to the characteristics of the contact between these surfaces:

1. Effective contact between the surfaces occurs only at the points of contact, the high spots of both surfaces.
2. The effective contact area is smaller than the total surface. The effective contact area is equal to the sum of all the high spot contact areas and is proportional to the applied load.
3. In most cases and usually in the case of metals, the high spots suffer plastic deformation, which means that the contact stress is greater than the yield stress of the softer material.

Using these basic characteristics, Archard devised a mathematical model that relates normal load and material hardness to the wear rate, the amount of material removed by unit sliding distance, as follows:

$$Q = K \times W/H$$

where

Q = wear rate, the volume of the material removed for each meter of sliding between the surfaces (mm^3/m);

W = normal load, measured in Newtons;

H = material hardness, measured in N/mm^2 ;

K = wear coefficient, adimensional and always smaller than 1.

The concept of a wear coefficient can be used to compare the wearing properties of different materials. Wear coefficients usually are measured with a standard test device, which contacts a rotating hardened steel disk with a sample of the material to be tested. The test conditions have a great influence on the actual value of the wear coefficient. Keep in mind that the test conditions do not closely reproduce the wear conditions found in a real machine, and the wear coefficients calculated during these experiments should be used as design data only with great caution. The wear coefficient can be understood as the ratio between the volume of deformed material and the volume of the material removed in a sliding wear situation.

The conditions of the interface play an important role in understanding sliding wear. A simple experiment can show the importance of the oxide layer on the wear conditions, using a brass cylinder in contact with a rotating steel disc. Brass has been used in many of these experiments, as its behavior is easily reproduced in the laboratory.

The wear rate of a brass sample as a function of the contact load is shown in Figure 6.1. We see that the wear rate grows in proportion to the load up to a certain point, when a sharp increase is followed by a return to the initial wear rate, after some point. The contact electric resistance also is shown in the figure; it reduces with small loads and is almost constant after a certain load.

This behavior can be explained by the role of the oxide film on the sample surface. The oxide film increases the electrical resistance and reduces the wear rate. When the test is run with small loads, the formation of the oxide film on the surface is faster than its removal by the sliding contact; therefore, a certain amount of oxide is always on the surface. The result is called *mild* or *moderate wear*. Above a certain threshold load, oxide removal is faster than the oxidation of the surface, and little or no oxide is there to prevent intimate contact

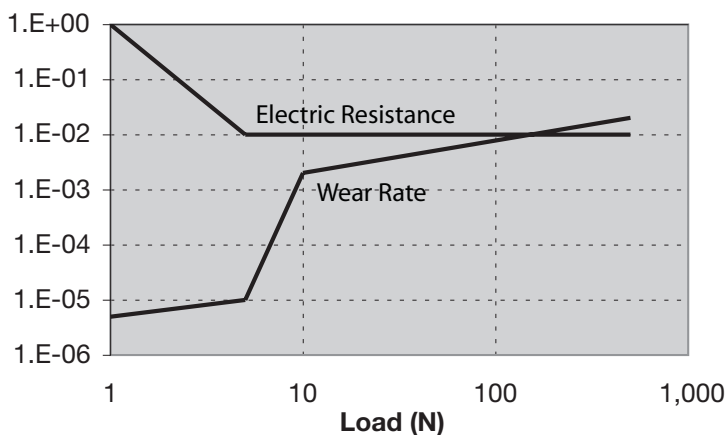


FIGURE 6.1 The wear rate and electrical resistance during a wear test with a brass sample.

between the samples. The result is severe wear and no further reduction of the contact resistance.

This transition of behavior due to the removal of oxides can be observed in many metals and alloys. It can result not only from the increase in load but an increase in relative speed or relative distance of contact. Debris removed from one or both surfaces may cause hard particle wear.

Sliding wear of metals is the result of the interaction of two mechanisms: plasticity and adhesion and oxidation.

Plasticity and Adhesion

As mentioned earlier, the contact of metal surfaces generally results in plastic deformation of the softer surface. Relative movement can cause deformation in the direction of the movement, resulting in cracks and material removal. Figure 6.2 illustrates the adhesion and plasticity occurring between two metal surfaces. The softer surface

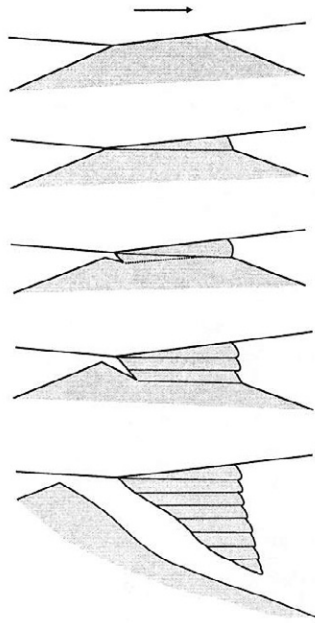


FIGURE 6.2 *The plasticity and adhesion that occurs when two metal surfaces contact each other.*

(below) is deformed by the movement of the harder one, and cracks may result in the removal of debris. Note that the process could occur even if there were no adhesion, due to the geometric interaction of the surface roughness.

The huge plastic deformation causes severe microstructural modifications on the surface. The grains are deformed in the direction of the movement. The external layer contains a mixture of oxides and metal debris as well as material transferred from the other surface.

Oxidation

The energy dissipated by the friction during the wear process can generate high heat, even at low speeds. If oxygen is present, these high temperatures result in fast oxidation of the exposed active metal. As mentioned previously, the oxide layer can have a beneficial effect on wear rate, because adhesion is made more difficult. A high ambient temperature causes an increase in the oxidation rate. Sliding wear accelerates oxidation because active metal is continually exposed. This increase in temperature can also cause cracks due to heat chocks. An example is shown in Figure 6.3, where the balancing disk of a centrifugal pump displays the characteristic radial cracks.

The oxide removal mechanism seems to be the following: The oxide layer is removed at the contact point and accumulates nearby. Oxides removed from adjacent places may accumulate above this layer, which starts to become thicker and thicker with more oxide accumulation until it becomes a high spot itself, which can be removed by the contact with a high spot on the other surface.

Figure 6.4 illustrates a situation where severe wear resulted from a rub. Some cavities can be observed at the points where material removal occurred, indicating adhesion between the surfaces. This is the balancing disc of a boiler feed pump. Although the water could have lubricated the components, the contact has been so severe that the fluid film was destroyed.

Figure 6.5 shows a machine component subject to mild wear. The surface finish difference can be noted easily, when compared with Figure 6.4. This is the stem of a turbine control valve. The amplitude of the movement and the magnitude of the normal load that caused the wear were not very high.



FIGURE 6.3 *Surface subject to sliding wear. Concentrically placed scratches caused by the sliding can be seen together with radial cracks caused by the heat chock.*



FIGURE 6.4 *Metallic surface subject to severe wear and material removal.*



FIGURE 6.5 *Metallic surface subject to mild wear.*

6.1.2 Lubricated Wear

The effective distance between the surfaces is the main parameter that influences lubricated wear behavior. This distance usually is measured in terms of the ratio between the effective distance between the surfaces and the root mean square of roughness height (λ). In the most common mechanical systems, the main factors that reduce λ are increased load or surface roughness and reduced relative speed or lubricant viscosity.

Figure 6.6 illustrates the relation between the friction coefficient (μ), the wear coefficient (k), and λ . Values of λ greater than 3 results in little or no solid contact, which causes negligible wear and small friction. This is the characteristic hydrodynamic lubrication found in regular turbomachinery bearings, for example.

When λ falls between 1 and 3, some solid contact will result. Part of the load is still carried by the lubricant film pressure, but wear and friction are greater. This is termed *elastohydrodynamic lubrication*, in which the elastic deformation of the contacting surfaces plays a role in the friction. This lubrication regime is commonly found in antifriction bearings.

There is heavy solid contact in the situation when λ is smaller than 1. In this case, sometimes called *boundary lubrication*, friction and wear are even greater than before, as most of the load is carried

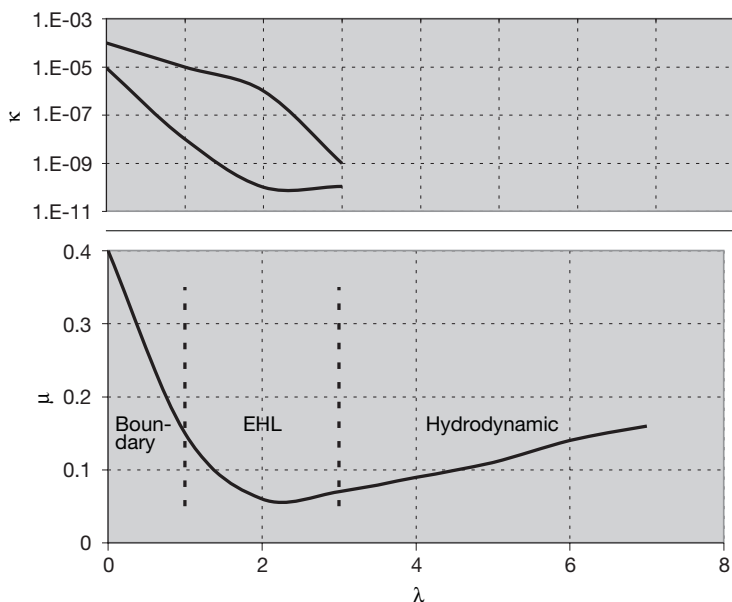


FIGURE 6.6 The behavior of friction and wear coefficients as a function of relative film thickness.

by the solid contact between the surfaces asperities. This lubrication regime can be found in heavily loaded gears and some mechanical seals, in this situation, oils with EP (extreme pressure) additives can help reduce wear. The EP additives act by forming an adsorbed layer that makes asperity adhesion more difficult, thereby reducing wear.

Figure 6.7 illustrates the film thickness and the resulting distribution of load support between the lubricant film pressure and the solid contact. Hydrodynamic lubrication is shown on the left side. We see that the entire load is supported by the fluid film pressure, with no solid contact and virtually no wear. The center figure shows the intermediate situation, while the right side figure shows that most of the load is supported by the solid contact when λ is smaller than 1. Note that the contact stress is the yield strength of the softer material, as explained previously.

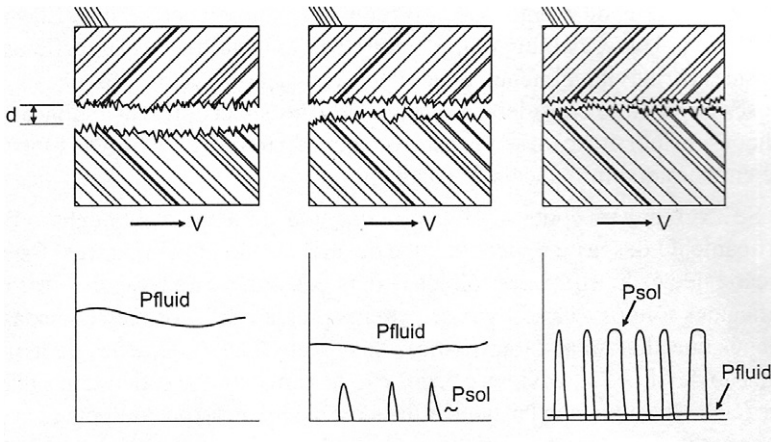


FIGURE 6.7 *The load support division between solid contact and fluid film pressure for different lubrication regimes.*

When metal sliding wear occurs for a length of time, friction and wear of the contacting surfaces are gradually reduced. This reduction can be attributed to the formation or increase of the surface oxide film, which contributes to a reduction of the microasperities' adhesion, and the removal of high spots, which increases λ for the same distance between the surfaces. Other surface modifications can reduce the wear, for example, hardening due to the formation of martensite in the case of steel surfaces.

Figure 6.8 shows a hydrodynamic bearing, a machine component that uses a thick oil film to reduce friction and wear when in normal operation. The picture shows some slight wear indications, caused during the startup and coast-down of the machine, when the shaft touches the bearing due to the insufficient film thickness caused by the reduced relative speed. This could be considered a mild wear situation.

Figure 6.9 shows an example of severe wear of a vehicle transmission component, caused by a lubrication failure. This severe wear occurs due to the unlubricated sliding wear with heavy solid contact. The wear was minimal on the other bearings, which had no loss of lubrication.



FIGURE 6.8 *Hydrodynamic bearing showing some signs of mild wear.*



FIGURE 6.9 *Crosshead pin with signs of heavy wear, caused by a loss of lubrication.*

6.1.3 Fretting Wear

Fretting wear, as the term implies, is a special form of sliding wear that can be observed when the relative movement between the sliding surfaces is of very small amplitude. The basic material removal mechanism is almost the same, but there are some important differences between fretting and sliding wear:

1. The relative movement between the sliding surfaces is small, usually 1–100 microns.
2. Fretting wear normally occurs between surfaces that were not supposed to have movement against each other. This microscopic relative movement, many times, is the result of vibration or elastic deformation of the involved body under load.
3. Debris formed by fretting wear normally are not removed from the place where they were generated. Metal oxides usually occupy a greater volume than the corresponding metal, which may result in a sticking effect, making relative movement impossible. The oxide accumulation also may result in a reduction of friction and wear.
4. Fretting wear may facilitate the nucleation of a fatigue crack. This issue is discussed later.

The fretting wear mechanism can be understood with the help of a simple experiment, pressing a sphere against a flat vibrating surface. Hertz's theory on the contact of elastic bodies shows that contact stress is greater in the center of the contact region, reducing toward the edges. When a small lateral movement is caused on the plane surface, sliding is more likely close to the edges, as the friction forces that impede sliding are proportional to the contact stress.

Figure 6.10 illustrates the experiment. A sphere is pressed against a plane surface by a force W . Contact pressure is designated P ; the dashed line shows that it is smaller at the edges of the contact region. The circular figures below indicate this contact region, the highlighted part being the part of the sphere where fretting wear has been observed after the experiment. The applied force W is reduced from top to bottom. Figure 6.10(a) shows that W was large enough to avoid any relative movement between the sphere and the

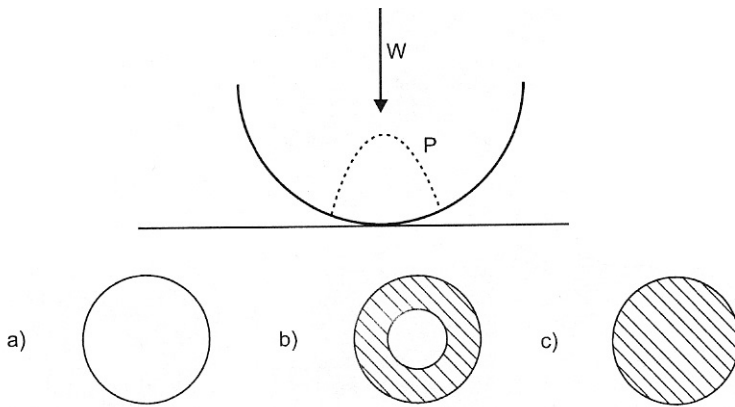


FIGURE 6.10 *The behavior of fretting wear between a sphere and a plane surface with reducing contact stress.*

plane surface; no fretting wear is visible. Figure 6.10(c) shows that W was small enough to allow fretting to occur over the whole surface. Figure 6.10(b) represents a load W that caused a contact stress large enough at the center to avoid fretting but small enough at the edges so that fretting was able to damage the sphere. The role of the elastic deformations of the contacting bodies on the possibility of fretting wear on part of the surface is evident here.

This simple experiment shows that wear rate in fretting wear can be a function of the amplitude of the relative movement, up to a certain point, where the mechanism of material removal would be considered to be sliding wear instead of fretting wear.

Figure 6.11 illustrates the variation of the wear rate as a function of the increase in the movement amplitude. Some distinct regions can be seen. Relative amplitudes smaller than 1 micron normally result in little or no material removal. Above 300 microns, it no longer is possible to find any difference relative to sliding wear. The fretting wear region is between 1 and 300 microns. For between 1 and about 20 microns, only some portions of the parts in contact show fretting, a situation made possible by the elastic deformations of the components under load. Above about 20 microns, the relative movement amplitude becomes large compared to the elastic deformation of the components, and there is relative move-

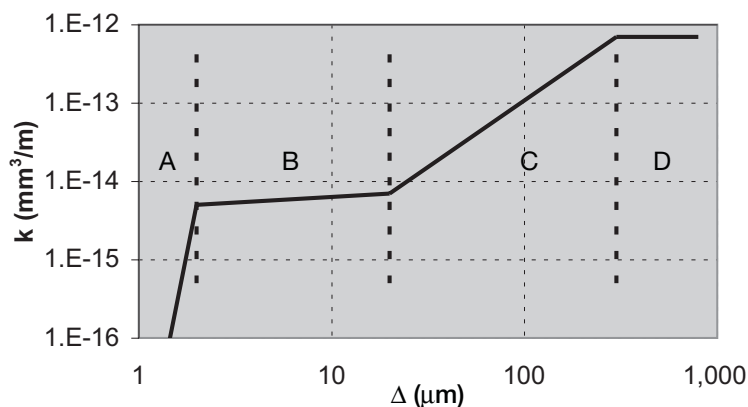


FIGURE 6.11 *The variation in wear rate (k) as a function of relative movement amplitude (Δ). In region A, there is no slippage; B indicates relative movement in some parts of the contacting surfaces; in C, there is relative movement across all the contacting surfaces; and D shows the region where fretting turns to sliding wear.*

ment of the surfaces, as if they were rigid bodies. The values of relative movement amplitude listed previously were measured during the experiment and are not to be considered generally applicable.

Figure 6.12 shows a machine component subject to fretting wear. This is the thrust-bearing pad support plate of a centrifugal compressor. The damage shown results from the vibration of the pad support when the compressor is running. Other examples can be found in Part III.

Even though fretting wear can loosen junctions and cause leakage in pressurized components and other problems, the main deleterious effect is the possibility of causing a reduction in the fatigue resistance of the component.

Figures 6.13 and 6.14 illustrate such a problem. Figure 6.13 shows a split ring designed to hold the piston rod of a reciprocating compressor to the crosshead with the help of a hydraulic nut. The black indication on the inside diameter of the ring is the result of fretting wear due to contact with the piston rod. Figure 6.14 shows the broken piston rod. The fracture has been nucleated at the contact point between the two components. A modification of the split



FIGURE 6.12 *A centrifugal compressor thrust-bearing support plate with indications of fretting wear.*



FIGURE 6.13 *Compressor split ring with indications of fretting wear.*

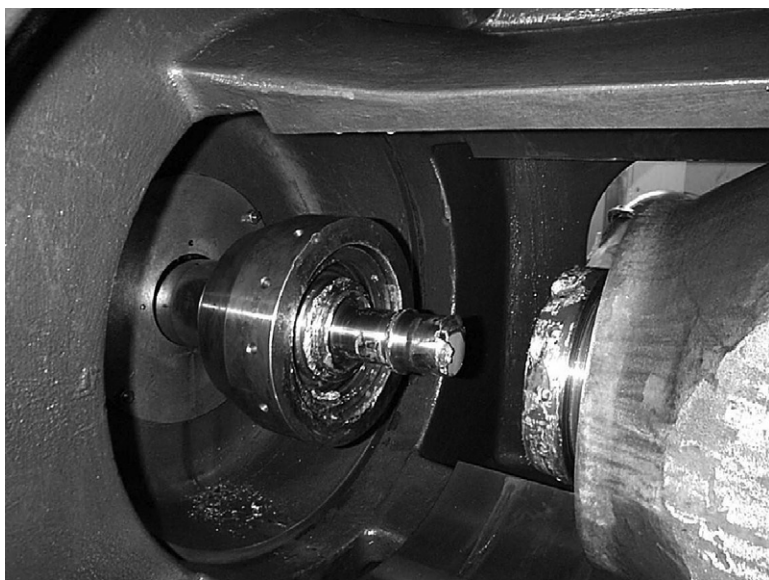


FIGURE 6.14 *Compressor piston rod that failed due to a fatigue fracture caused by fretting wear between the split ring and the piston rod.*

ring, making this contact impossible, prevented the repetition of the failure.

When the compressor is running, the piston rod is subject alternatively to tension and compression, which causes elastic deformations. If the split ring touches the piston rod, relative movement and fretting are likely to occur.

Fretting may cause the initiation of fatigue cracks due to alternating stresses on the surfaces. These stresses are large enough to cause the actual rupture of the microasperities found on the surfaces; therefore, they are large enough to induce a fatigue crack nucleation. Once nucleated, this crack becomes a stress concentrator that can locally increase the alternating stresses caused by the external loads to the point where crack growth is possible. Just like any other combination of damage mechanisms, the combination of fretting and fatigue may bring about failures faster than when one of them acts alone.

6.1.4 *Avoiding Sliding Wear*

The first step to the effective implementation of measures to avoid sliding wear is to identify correctly the wear mechanism and the allowable wear rate. When the wear mechanism and the estimated wear rate are known, it becomes possible to investigate the effect of some modifications on the severity of the problem.

Although it is not easy to avoid wear completely, very often it is possible to reduce the wear rate to acceptable levels. In general terms, all the factors that influence wear rate could be considered candidates to modification. In the practical world, it may not be easy to change some operating conditions, such as contact loads, relative speeds, or removal of contaminants. The most commonly used strategies to reduce machinery wear are discussed briefly:

1. Lubrication, perhaps, is the most commonly used technique to avoid machinery wear. As discussed previously, in the ideal lubrication condition, the lubricant film is thick enough to separate completely the sliding surfaces. Even when complete separation is not possible, lubrication can reduce wear by making adhesion more difficult, for example. This is how extreme pressure oil works. Sulfur and phosphor are common EP additives; they make steel adhesion more difficult. It is no surprise that high-quality weldable steels have only a very small amount of these two elements.
2. Material selection also can control wear. The possibilities here are almost endless. As a rule, the use of harder materials reduces wear.
3. Surface engineering is another popular strategy. Some traditional applications include surface hardening or coating and controlled surface roughness or profile. The topology of a surface can make it easier for the oil to remain in certain “sculpted” pockets and provide some lubrication, even in difficult situations.
4. Fretting wear can be reduced when the contributing features are avoided. Those contributing factors are microscopic relative movement and surface contact. This can be done by

restricting the relative movement, such as welding the parts together or increasing the contact load to the point that friction can hold the parts. Another way to avoid fretting wear is to avoid contact between the parts, increasing the clearance between them, for example. Except for lubrication, the same techniques used to reduce sliding wear can be used to reduce fretting wear. Lubrication is not useful because the relative speed between the parts normally is small, making it impossible for an oil film to develop.

6.2 Hard Particle Wear

The three most common mechanisms of hard particle wear, classified according to how the normal load is imposed to the particle against a surface, are

1. Two-body abrasion, when the hard particles actually are hard protrusions on one of the contacting surfaces.
2. Three-body abrasion, when the hard particles are pressed between two sliding surfaces.
3. Erosion, when the hard particles are projected against the surface by a stream of fluid.

Other classifications and other names for the mechanisms just described are possible. Figure 6.15 illustrates this type of wear.

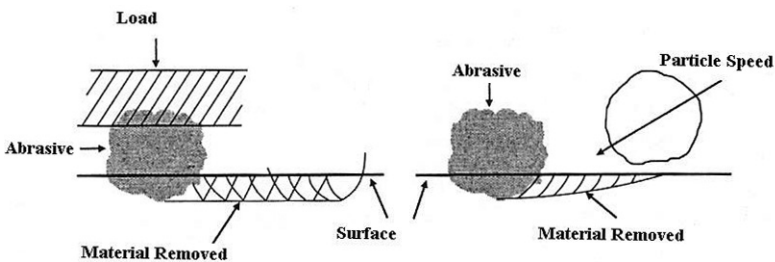


FIGURE 6.15 *The hard particle wear material removal mechanism.*

The hard particles may be present because of the normal working environment, the result of the intrusion of foreign material, due to the deterioration of the machine components. Earth moving is an example of the first situation, oil contaminants in a lube oil system may be a second type of source for the hard particles, and rust or wear of the internal parts is an example of the last one.

6.2.1 Properties of the Hard Particle

The mechanical and geometrical properties of the hard particle can influence the mechanism and wear rate. The most important ones are hardness, shape, and size.

Particle Hardness

The higher the ratio between particle and surface hardness, the higher the wear rate. In the case of a spherical particle contacting a plane surface, the application of Hertz's theory leads to the conclusion that the maximum contact stress is approximately 80% of the particle hardness, which means that the particle hardness needs to be at least 125% of the surface hardness to cause significant surface deformation. Most machines are made of steel, and silicon oxide, the most common material on earth, is many times harder than steel. This means plenty of hard particles everywhere are ready to cause machinery wear.

The observation that a hard material would scratch a softer one and that the softer one would not do the opposite inspired the creation of the Mohs hardness scale, with an integer number assigned to 10 minerals. The comparison between the Mohs scale and Vickers scale is shown in Figure 6.16. We see that the ratio between the hardness of two neighboring minerals is approximately 1.6, except for the diamond. This ratio is slightly higher than the minimum required for hard particle wear.

The mechanical resistance of the particle material also can influence the wear rate and mechanism. Remembering Hertz again, when a hard particle contacts a surface, plastic deformation is significant when the contact pressure is higher than a certain value. If the hard particle breaks or deforms with a load smaller than that required for the surface deformation, the wear rate is greatly reduced.

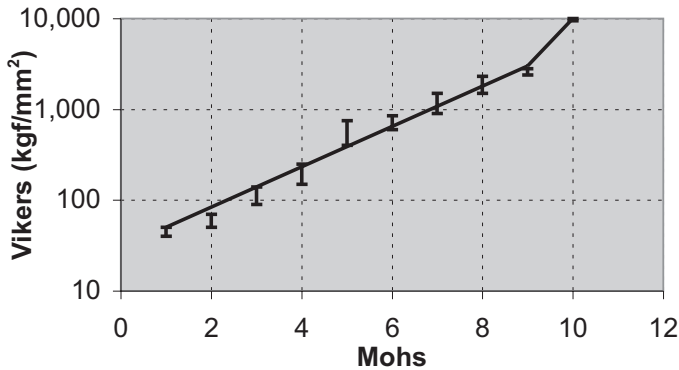


FIGURE 6.16 *Comparison between the Mohs and Vickers scales.*

Particle Shape

In the same way that a sharp knife makes cutting easier, particles with sharp edges can cut more material from the surface. It is not easy to attribute a mathematical definition to the sharpness of a particle, although this characteristic can be readily noticed by inspection.

The sharp edges act as stress concentrators that cut through the surface material. The flow of removed material around a sharp edge also can be easier. Unfortunately, plenty of sharp-edged particles are ready to penetrate and destroy process machines.

Particle Size

The size of the particles capable of causing wear covers a wide range. Most of the damage observed in real-world machinery is caused by particles greater than 5 μm and smaller than 500 μm . A simple reason for this is that the clearances usually found in rotating process machinery are in this range, with few exceptions. Particles sized below 150 μm cause less damage than larger ones, possibly because there are fewer possible nucleation spots for plastic deformation.

Another way to understand the greatest erosive effect of medium-sized particles is the relation of particle size to the clearances normally found in the process machinery. In the case of two- or three-body abrasion, if the particle is much smaller than the clearance, it will not be pressed between the surfaces. On the other

hand, if the particle is much larger than the clearance, it will not be able to penetrate the narrow space between the components. Oversized particles may not be carried efficiently by a fluid stream to promote erosion, and the small inertia of very small particles makes it difficult for these to hit the surfaces very hard.

6.2.2 Abrasive Wear Mechanism

The abrasive wear mechanism involves plastic deformations, ductile fractures, and brittle fractures. The ductility of the material and the wear conditions determine the prevailing mechanism. Most materials used for machine construction are ductile, and this section describes the plastic deformation wear mechanism only. The brittle fracture wear mechanism can be found in materials like ceramics or carbides. Although carbides wear can be important in mechanical seal failures, the mechanism is not discussed here.

Abrasive wear happens due to indentation of a surface by a hard particle, followed by the lateral movement of the particle, which extends the deformation and, ultimately, causes material removal. A simple mathematical model results in a wear equation similar to Archard's equation, introduced earlier.

The quantity of material removed in an abrasive wear situation is directly proportional to the distance covered by the hard particle and inversely proportional to the surface hardness. Relative resistance to wear is the inverse of the quantity of material removed. Figure 6.17 illustrates the relationship between relative wear resistance and the hardness of pure metals. An almost linear relationship is immediately obvious. Metal alloys show somewhat different behavior, depending on the heat treatment provided to the component.

Even though plastic deformation and strain hardening increase the hardness of the material, the effect of the strain-related hardness on the wear resistance of a metal usually is small. This behavior can be explained by the wear mechanism itself, considering that the abrasive wear on ductile metals causes local plastic deformation to the point of rupture of the material. The introduction of alloying elements that increase hardness are useful only if the element can increase hardness in an extreme plastic deformation situation. Car-

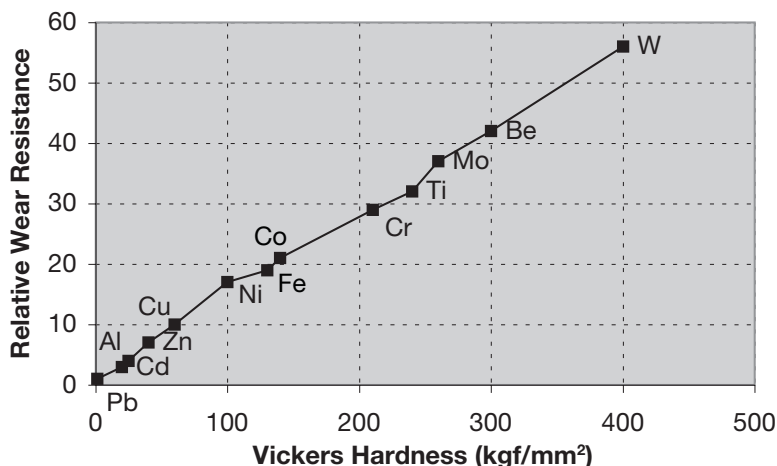


FIGURE 6.17 *Relative wear resistance and hardness of pure metals.*

bide precipitations in steel are an example of a wear resistance improving an alloying element. Heat treatment to increase hardness generally increases wear resistance.

The effect of lubricants can be somewhat surprising. If the particle size is bigger than the lubricant film thickness, lubrication increases wear, as friction between the hard particles and the surface are reduced, which makes lateral movement and material removal easier.

Figure 6.18 shows a typical example of two-body abrasion; in this case, the picture shows the surface resulting from the grinding of a welded hard-facing deposit on a 410 stainless steel plate. The hardness of the weld overlay is approximately 57–59 RC (Rockwell C). The individual scratches made by each hard particle can be seen easily.

Some wear-resistant materials are composed of a soft matrix with a dispersion of a hard phase. In these cases, wear resistance depends on the size ratio between the abrasive particles and the hard matrix particles. If the matrix is finer than the abrasive, wear resistance increases greatly, as plastic deformation inhibited by the dispersed particles. On the other hand, if the abrasive is smaller than the dispersed hard particle, the soft matrix may abrade with a greater wear rate than in the former case.

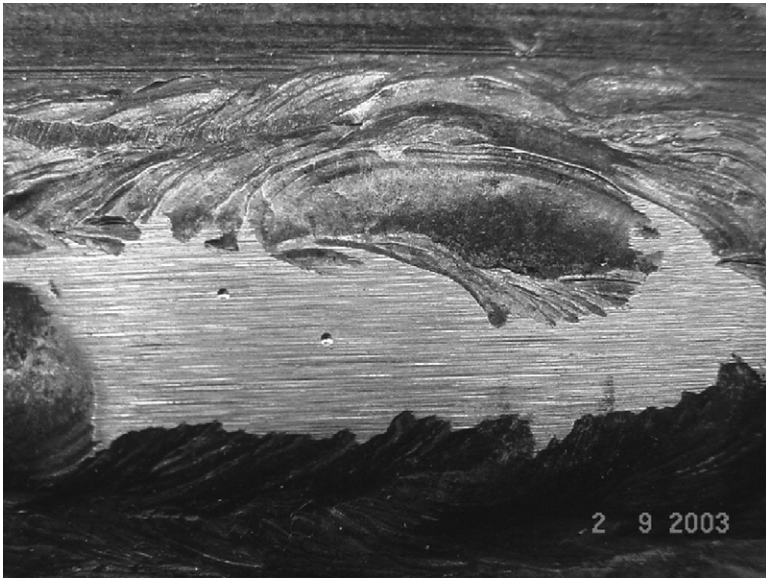


FIGURE 6.18 *Weld overlay showing signs of two-body wear.*

6.2.3 Hard Particle Impingement Erosive Wear

Erosive wear occurs when hard particles are thrown on a solid surface. The main difference from the preceding section can be found in the source of the forces acting on the hard particle. In this case, the prevailing force that presses the particle against the surface is its velocity or inertia. In hard particle abrasion, the quantity of removed material is proportional to the distance traveled by the hard particle and the normal load. In erosion, material removal depends on the particle's velocity or inertia and the total mass of particles that hits the surface. Everything else is the same; that is, the mechanism of material removal is related to the plastic deformation of the surface material. Figure 6.19 shows a centrifugal pump throttle bushing that suffered erosion from particulate matter present in the pumped fluid. Unlike the part shown in Figure 6.18, in this case, the particle size was small enough to produce a smooth surface, as the individual scratches caused by each particle are too small to be seen with an unaided eye.



FIGURE 6.19 *Severe erosive wear on a pump throttle bushing caused by iron fluoride particles suspended in the pumped fluid.*

6.2.4 Reduction of Hard Particle Wear

The classic methods to reduce hard particle wear are the following:

1. Removal of the hard particle abrasive, which is possible only when the hard particles are avoidable contaminants, such as sand inside a lube oil system.
2. Surface hardening, if the hardening is not produced by plastic deformation. Any other surface-hardening process reduces wear, be it flame hardening, nitriding, hard facing, chrome plating, or something else. Stellite is a common weld overlay applied to reduce wear.
3. Surface softening can be quite effective to reduce wear, when other conditions do not impede its use. An example of an impeding condition is high temperature, as the soft surface overlay usually is made of rubber. The reason why a soft surface can have a high wear resistance is that the hard particle causes elastic deformations only, especially when the hard particle is projected against the surface at a low speed. Automobile tires are a good example of this technique, also used in dredging pumps.

6.3 Liquid Impingement Wear

The wear on solid surfaces may happen because of the high-velocity impact of liquid droplets, even if no solid particles are present. The wear mechanism seems to be the following:

1. The high-velocity impact generates a high pressure on the impact point because of the deceleration of the impacting fluid. This high pressure may cause surface cracks or deformation.
2. After the collision with the surface, the liquid droplet flows outward in a radial direction, traveling at high speed. As it travels, the liquid may collide with some surface irregularities and remove some material.

The contact pressure resulting from the impact is enough to damage the surface. If the droplet hits a region where some damage already exists, further damage is hastened. The eroded surface loses material and soon shows a characteristic rough appearance. The surface material plays an important role in the extent of the possible damage. As in the preceding situations, higher strength and harder materials are more resistant to erosion.

Erosion can be found on the last stages of condensing steam turbines, cooling tower fans, and the like. Figure 6.20 illustrates the wear mechanism. Figure 6.21 shows a steam turbine casing that suffered severe erosion from the impact of condensate droplets. This type of damage normally occurs when the turbine operates in a con-

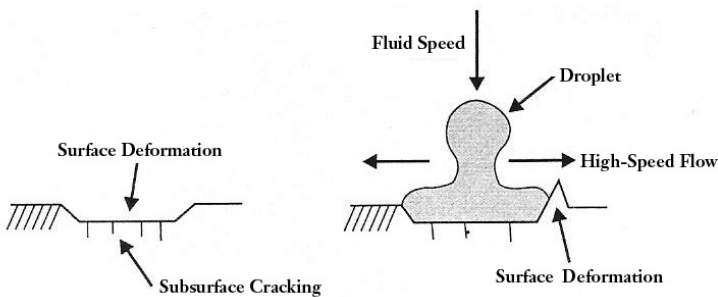


FIGURE 6.20 *The erosion wear mechanism.*



FIGURE 6.21 *Erosion on a steam turbine casing.*

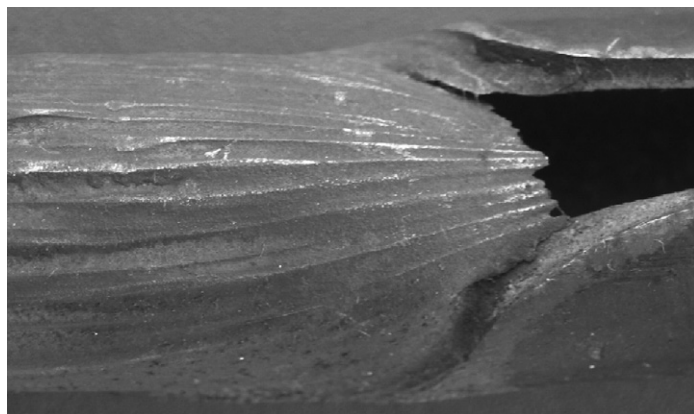


FIGURE 6.22 *Erosion on the surface of a high-pressure water preheater.*

dition that allows too much condensation of the steam in the last few stages before the condenser. Figure 6.22 shows a high-pressure water preheater tube that suffered severe erosion from the impact of high-speed water droplets. These droplets originated from a water leak on the tube sheet.

6.4 Cavitation

Cavitation is the formation and subsequent collapse of vapor bubbles in a liquid stream. This can happen due to local variations in pressure, as in a centrifugal pump or control valve, or temperature, as in the cooling system of a diesel engine. The mechanism is the following:

1. Local reduction of pressure or increase in temperature up to the point that causes vaporization of the liquid.
2. Collapse of a vapor bubble previously formed due to an inversion of conditions. The collapse of the bubble generates a high-speed jet of liquid that travels within the bulk of the liquid.

If the collapse of the bubble occurs close to a solid surface, cavitation damage may result, although a large number of microjets is necessary before any significant damage results. Figure 6.23 illustrates the cavitation mechanism.

Some contributing factors may aggravate or attenuate the cavitation damage:

1. Gases dissolved in the liquid dampen the impact. When the vapor bubble is formed, a certain amount of the dissolved gas occupies part of the bubble volume. The vapor inside the bubble condenses almost instantaneously. The gas inside the bubble is unable to go back in solution as fast as this, so when the bubble collapses, the gas inside it acts as a cushion.

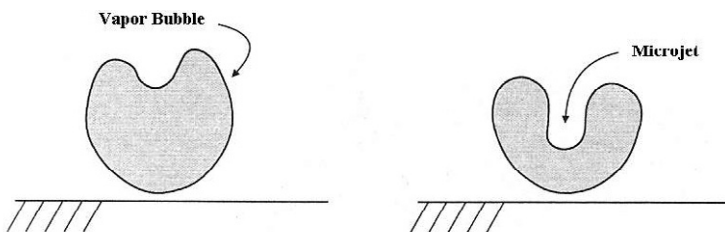


FIGURE 6.23 *The collapse of a vapor bubble and the impact of a microjet on a nearby surface.*

2. A high ratio between vapor- and liquid-specific volume increases the volume of the bubble. A bigger vapor bubble can generate a faster liquid jet on condensation of the vapor. This is why water pump cavitation produces more damage than hydrocarbon pump cavitation.

The surface modifications that reduce cavitation damage are the same as the ones that reduce liquid impingement damage. However, it usually is more effective to avoid cavitation damage by avoiding the formation of vapor bubbles, reducing temperature, increasing pressure, or reducing the vapor pressure of the liquid.

Centrifugal pump cavitation occurs due to the pressure reduction caused by a change in the direction of the fluid flow when entering the impeller, if suction pressure is too close to the vapor pressure of the fluid. The pressure reduction causes vaporization, and the subsequent increase in pressure as the fluid travels through the impeller causes vapor bubble collapse. Figure 6.24 shows a centrifugal pump impeller that suffered severe cavitation damage.



FIGURE 6.24 *Cavitation damage in a centrifugal pump impeller.*

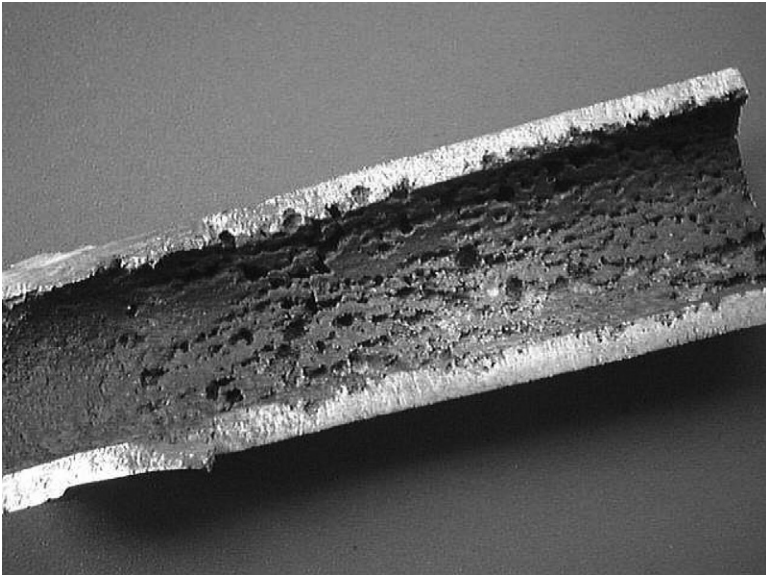


FIGURE 6.25 *Cavitation damage on the inside surface of a boiler drain.*

Figure 6.25 shows a tube that has been used as the drain of a boiler with signs of cavitation damage. The portion of the tube shown had a globe valve to control the flow, installed on the left side of the figure. Flow direction was from left to right. When flowing past the valve, pressure reduction was enough to vaporize water. The cavitation damage is concentrated at a certain distance from the valve position, in the place where the vapor bubbles collapsed. Vapor bubble collapse happened here because of the pressure recover after the vena contracta.

7

Corrosion

Corrosion is the deterioration of the useful properties of a material caused by chemical or electrochemical interaction with the environment. The thinning of a pressure vessel wall is an example, where the useful property, in this case, the capacity to resist internal pressure, is lost if thickness reduction is significant. The staining of a mirror is another example, where the oxidation of a very thin layer of the mirror material can cause the loss of its useful property, reflectivity. These simple examples show that the allowable loss of material can be quite different for different devices. Even a small amount of corrosion can make a mechanical component useless. Other types of corrosion may cause hydrogen embrittlement despite very little loss of material.

Corrosion may cause the failure of machine components directly or interact with other failure modes to make component failure more likely. Due to the complexity of the subject, only the types of corrosion that affect rotating process machinery most often are discussed here.

The factors that influence the corrosion mechanism and rate are summarized here. It is not easy to enumerate completely all the

influencing factors involved in a specific corrosion problem, as very small percentages of contaminants in the electrolyte can drastically alter the corrosion behavior of the material.

1. Chemical composition, microstructure, and homogeneity of the material, including the existence of dissimilar materials, especially dissimilar metals. The processing of the material, including heat treatment and other fabrication processes, also influence the corrosion conditions.
2. Chemical composition and contaminants, including the oxygen content, of the environment. Concentration gradients can play a role as well.
3. Temperature and temperature gradients, the interface between the material and the environment, and relative velocity.

7.1 Electrochemical Corrosion Mechanism

The majority of the corrosion processes observed in rotating equipment is metal corrosion in an aqueous environment. These are electrochemical corrosion processes, characterized by the need of electricity to flow between anodic and cathodic areas. All chemical reactions require the exchange of electrons, but in electrochemical corrosion, this electron exchange occurs over a distance greater than interatomic spacing. The order of magnitude of this electron exchange distance can be as small as the grain size, in the case of uniform corrosion of a metal in an acid environment, or as large as a few kilometers, when a buried pipeline is corroded by electric current transmitted by a nearby railway.

The first condition necessary to make electrochemical corrosion possible is that both the metal and the aqueous media show little electrical resistivity. This condition is easily fulfilled in the case of metal corrosion in aqueous media. The electric resistivity of metals is normally small, and aqueous media can display this characteristic because of the dissolved ions. Atmospheric corrosion is a special case, where the electrolyte is provided by the humidity in the air. The oxide layer formed on corroded surfaces also can act as a solid electrolyte.

The electrochemical nature of metal corrosion has been depicted in a simple experiment by West, the saline droplet experiment. In this experiment, a droplet of a 3% NaCl aqueous solution is placed over a polished iron surface. The solution contains a small amount of potassium ferricyanide, which turns blue in the presence of iron ions (Fe^{2+}) and phenolphthalein, which turns pink when in contact with hydroxyl ions (OH^-).

Soon after contact with the iron surface, some scattered blue and pink indications are seen, as illustrated in Figure 7.1(a). After some more time, the distribution of these colored areas change, with a concentration of blue indications in the center and pink on the edge. This distribution can be seen in Figure 7.1(b). A brown area can be observed in between.

The main reactions taking place inside the saline droplet are the formation of iron ions in the center and hydroxyl ions on the outer edge. Oxidation of iron (the anodic reaction) is represented as



Some electrons are produced as a result of the oxidation. The iron ion makes the potassium ferricyanide turn blue. The reduction of water in the presence of oxygen (cathodic reaction) is represented as

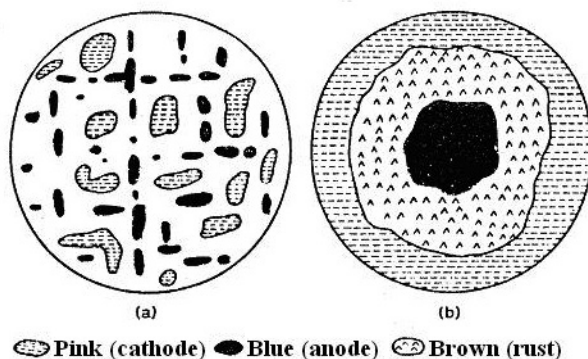
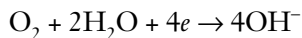


FIGURE 7.1 West's experiment with electrochemical corrosion.

The electrons produced previously are consumed in this reaction, because they were able to travel through the highly conductive iron plate. The reason why the cathodic reaction takes place close to the edge of the water droplet after some time is the need for oxygen for the cathodic reaction. Oxygen reaches the metal surface, where the reactions take place, after diffusion through the water. More oxygen is found in the regions closer to the atmosphere, which is the source of the oxygen.

Many other secondary reactions take place at the same time. One of them is the formation of the brown deposit observed in between the blue and pink areas. The chemical composition of this brown deposit is quite complex, but it is composed mainly of FeOOH and Fe_3O_4 , formed by the reaction between the iron and hydroxyl ions. This is why it precipitates mainly in between the cathodic and anodic areas. The brown deposit is also known as *rust*.

The last secondary reaction to be mentioned is the result of the increase in the pH of the solution at the outer edge of the droplet. In this region, the prevailing conditions allow the formation of a thin oxide layer on the metal surface. This is called *passivation*, and the resulting oxide layer acts as a protecting barrier for the bare metal, reducing the corrosion rate without impeding the flow of electrons necessary for the whole process.

7.2 Uniform Corrosion

Uniform corrosion of metals is the simplest and most frequently observed form of corrosion, being found when items are exposed to the atmosphere, the soil, and an enormous variety of aqueous solutions. Uniform corrosion normally affects metallic surfaces that have a uniform chemical composition and a microstructure with homogeneous access to the corrosive solution, which itself needs to be uniform.

The corroded surface may become covered with the corrosion products or completely clean, depending on many factors, such as the relative velocity of the corrosive solution and the solubility of the corrosion products in the electrolyte.

The uniform corrosion rate normally increases with temperature. The interface temperature should be considered the important parameter, because the corrosion reactions take place here. The interface temperature can be significantly different from the fluid or metal temperature, especially when there is significant heat exchange. The chemical reactions themselves follow Arrhenius's law, which states that the reactions double in speed for a 10°C increase in temperature. When dealing with aerated solutions, also remember that oxygen content, an important parameter for the corrosion conditions, is strongly influenced by temperature. High temperature may reduce the oxygen content of aqueous solutions, influencing the corrosion rate.

Concentration of the corrosive chemical in the solution has a strong effect on corrosion rate. For most corrosives, an increase in concentration from a much diluted solution causes an increase in corrosion rate up to a certain point. Very concentrated solutions may show a reduction in corrosion rates, if the concentration is high enough to make the reaction starve for water.

Some common strategies used to reduce this type of corrosion are

1. Selection of materials that can be corrosion resistant, which means the corrosion rate is small enough for the application. The resistance of these materials to corrosion usually is a result of passivation.
2. Introduction of a barrier between the metal and the corrosive environment. Painting is an example of an organic barrier; chrome plating, an inorganic barrier. When using this type of strategy, always consider the effects of small defects on the coating. A defect on a chrome coating creates an anodic spot that suffers a high corrosion rate, because the ratio between anodic to cathodic areas results in a high density of anodic current. The same defect on a galvanized plate does not create an anodic spot, because zinc is anodic relative to steel, and the region below the damaged coating is protected from corrosion.
3. Modifications of the corrosive medium. While it may not be easy to change a corrosive solution for something totally different,

the introduction of corrosion inhibitors may reduce the corrosion rate to an acceptable level. This strategy is commonly used in the processing industry.

4. Cathodic protection, forcing the metal to assume an electric potential relative to the corrosive medium that will not allow corrosion to take place. This is often done in buried pipelines with the introduction of an external power source to change the aforementioned potential.
5. Anodic protection, with the introduction of anodes in contact with the main structure and the corrosive medium. This often is done to protect oceangoing ships and other structures. A zinc or aluminum anode is attached to the ship hull and consumed instead of the steel.

Rusting of steel and darkening of silver in contact with the atmosphere are two examples of uniform corrosion. Figure 7.2 shows a pump impeller made of cast iron exposed to a hydrofluoric



FIGURE 7.2 *Pump impeller that suffered uniform corrosion.*

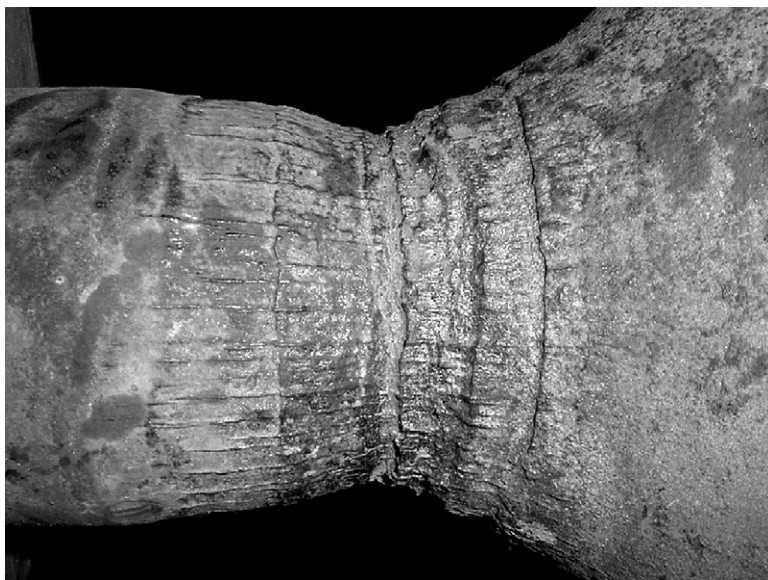


FIGURE 7.3 *Acid corrosion on the shaft of an induced draft fan.*

acid solution on the outlet of an acid neutralizer. Figure 7.3 shows an induced draft fan shaft subject to corrosion due to a leakage of atmospheric air to the inside. The lower temperature of the air caused condensation of sulfur compounds on the shaft surface.

7.3 Corrosion Fatigue

Corrosion fatigue is a failure mode that combines the effects of both fatigue and corrosion. Generally, whenever two or more failure modes are present at the same time and interact with each other, the failure progression is much faster than when they act separately. As an example, high strength steel may have its fatigue strength reduce by as much as 90% when exposed to seawater, when compared to the dry air strength.

The corrosive environment influences the nucleation and the progression of the fatigue crack. The exact interaction mechanism is not known. Corrosion fatigue has been found in metals exposed to aqueous solutions and components subject to periodic condensation

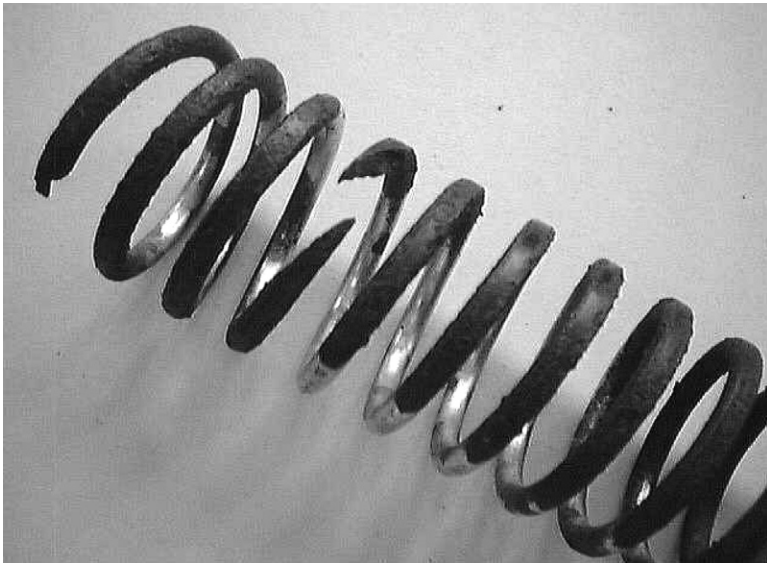


FIGURE 7.4 *Spring showing corrosion fatigue.*

of vapor. The reduction of the fatigue resistance in corrosive environments is time related, much as normal fatigue is related to the number of load cycles. A longer exposure to a corrosive medium and cycling loads reduces the fatigue strength of the material.

Although not exactly known, the crack nucleation mechanism depends on the type of material. Carbon steel components, for example, usually nucleate cracks in the bottom of the corrosion pits, containing significant quantities of corrosion products, although this is not a necessary condition. Unlike in the case of stress corrosion cracking, specific combinations of corrosive medium and metal are not necessary.

Crack propagation also is affected in an unknown way. The corrosive damage to the fracture surfaces makes a detailed analysis very difficult. In a certain way, this is the same problem found when a corrosion fatigue problem is analyzed in the field. It always is very difficult to find concluding evidence, and the diagnosis of the failure mode often is based on indirect indications.

It is important to recognize that corrosion fatigue is not necessarily the failure mode when corrosion and fatigue do not act at the same time. To make things even worse, corrosion does not always accelerate a fatigue failure.

Figure 7.4 shows a spring that failed due to corrosion fatigue. As usually is the case, concluding evidence has not been found and the diagnosis was made through the exclusion of other possible failure modes.

7.4 Pitting Corrosion

Pitting corrosion is a localized corrosion phenomenon that produces distinct, well-defined cavities on the component surface. Corrosion on the inside of the pits is uniform. Any metal that can be passivated can suffer this type of corrosive attack, which happens when the corrosion resistance of the material is borderline because of localized rupture of the oxide film. The protective oxide film can be ruptured in a number of ways, including local dissolution or mechanical damage. Concentration differences found under deposits can be enough to promote pitting corrosion.

When pitting corrosion is found on a clean, uniform metal surface, a slight increase in the corrosiveness of the environment results in general corrosion. An important characteristic of pitting corrosion is the ratio between cathodic and anodic areas, the ratio between the total passivated area and the total pit area. The larger this ratio, the larger the current density inside the pits and the higher the penetration velocity.

The small area and small material removal caused by the pit make detection difficult. It is possible that the affected equipment will show leaks with a very small loss of mass. Pitting corrosion also may begin under localized surface deposits, making detection even more difficult.

The mechanism that allows the fast penetration of pitting corrosion is the formation of an excess of positive loads inside the pit, due to the fast dissolution of the metal. This excess provides for the migration of electrolyte ions to the inside of the pits, making the process self-sustainable.



FIGURE 7.5 *Pitting corrosion on the inside surface of a steam condensate pipe.*

Some common causes of pitting corrosion are

1. Surface heterogeneities on the material.
2. Local loss of passivity due to the chemical or mechanical rupture of the oxide film.
3. Formation of differential concentration cells under deposits.

This process is usually related to specific combinations of metal and medium. A classic example is the occurrence of pitting corrosion in austenitic stainless steel exposed to environments containing chlorides. Carbon steel may suffer pitting corrosion when exposed to moderately corrosive environments. Figure 7.5 shows a carbon steel tube with signs of pitting corrosion due to the exposure to steam condensate.

7.5 Galvanic Corrosion

When two dissimilar metals contact each other while immersed in an electrolyte, the difference in the electrochemical potentials result in faster corrosion of the metal with the higher potential and

protection of the other metal. This, called *galvanic corrosion*, is readily recognized at the dissimilar metals interface.

The appearance of the corroded surface is similar to uniform corrosion. The higher the potential difference, the faster the corrosion of the less noble metal. Examining the position of the metals in the galvanic series, it is possible to find out beforehand if there is a chance of galvanic corrosion and which metal will be corroded. The less noble metal in the galvanic series is magnesium, followed by zinc, galvanized steel, aluminum, cast iron, 410 stainless steel (active), 304 stainless steel (active), lead, copper, nickel (active), nickel (passive), Monel 400, 410, and 304 stainless steel (passive), Hastelloy C, silver, titanium, graphite, gold, and platinum.

If the exposed area of the less noble metal is smaller than the other one, faster corrosion also results. Some commonly found combinations of metals that result in galvanic corrosion are copper and stainless steel, brass or bronze and steel, stainless steel and carbon steel. Figure 7.6 shows an example of galvanic corrosion. In this case, a carbon steel nut has been used in a stainless steel cutting tool. Figure 7.6 shows an example of galvanic corrosion. In this case, a carbon steel nut has been used in a stainless steel cutting tool, and the nut has been heavily corroded.

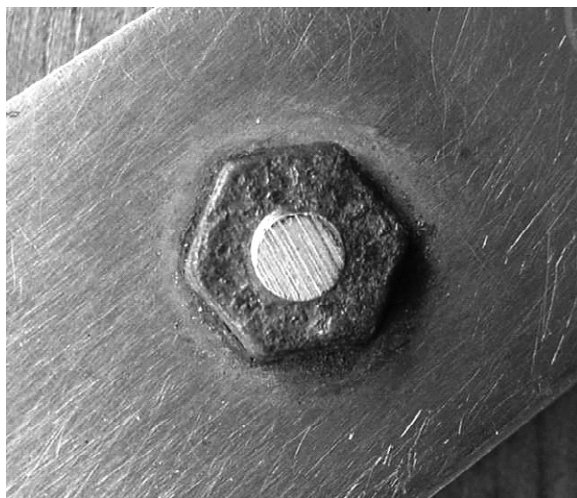


FIGURE 7.6 *Galvanic corrosion in a carbon steel nut installed in a stainless steel cutting tool. The small area of the nut, relative to the tool, speeds corrosion.*

7.6 Corrosion Erosion

Corrosion erosion is another failure mode combination that produces a new and faster damage mechanism. In this case, electrochemical corrosion combines with mechanical hard particle wear.

Starting with a metal surface relatively protected from corrosion due to a thin oxide film, the flow of fluid carrying solid abrasive particles can remove the oxide layer and expose the unprotected active metal. Electrochemical corrosion may act again, as oxidation of bare metal is very fast. Now, we go back to the beginning, and the hard particles, once again, remove the oxide layer and so on. The most commonly used structural material, steel, produces a protective iron oxide layer. Iron oxide is softer than steel, being easily removed by the abrasive action of hard particles.

Corrosion erosion is commonly found in components subject to a high-velocity flow of moderately corrosive fluids containing small quantities of solids in suspension. The inside of centrifugal pump casings is a good example, the damage being found in the high-velocity flow areas of the pump, such as the volute or impeller.

The appearance of the corrosion erosion damaged surface often is smooth, sometimes showing grooves oriented in the direction of fluid flow. Figure 7.7 shows a centrifugal pump casing with

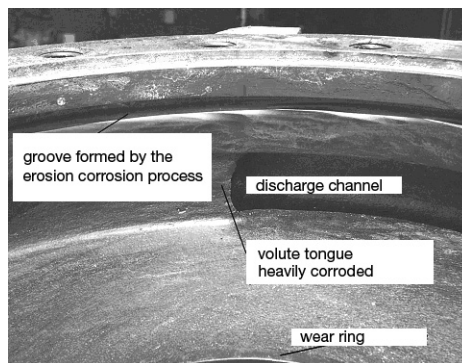


FIGURE 7.7 *Corrosion erosion inside a centrifugal pump casing.*

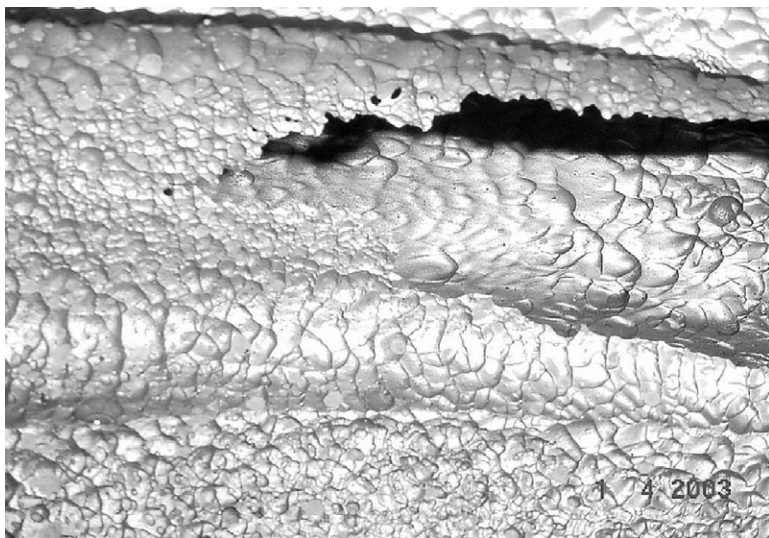


FIGURE 7.8 *Pump casing that suffered corrosion erosion.*

signs of corrosion erosion. The smooth surface indicates that material removal due to abrasion has been large. Figure 7.8 shows another pump casing subject to corrosion erosion. In this case, the rough surface indicates the prevalence of material removal due to acid corrosion.

7.7 Stress Corrosion Cracking

Stress corrosion cracking (SCC) is the failure mode that causes cracks in certain materials subject to monotonic stresses smaller than the strength limit and exposure to certain specific corrosive environments. As usual, the combination of individual failure modes results in a much faster fracture of the component.

The exact mechanism of crack induction due to the corrosive effects is not known. Corrosion is known to facilitate the initiation and propagation of the crack. Any kind of tension stress, due to externally applied loads or internal residual stresses, can cause stress corrosion cracking to develop. The initial cracks are microscopic

and may take a long time to develop, which does not mean that further propagation cannot be very fast.

The cracks resulting from SCC can be branched or not, inter- or transgranular. This phenomenon requires specific combinations of metal and corrosive environment. Some metals and environments widely recognized as prone to SCC are listed in Table 7.1.

The crack initiation can be impossible to detect. Crack propagation is impossible to stop without a change in conditions. These two characteristics make this failure mode dangerous, due to the possibility of unexpected and undetectable failures.

The safest and easiest, although sometimes expensive, way to avoid SCC is to recognize this possibility in a certain environment and use resistant materials. Special care should be taken with abnormal operating conditions, when unexpected fluid may contact the equipment and an off-design condition arise.

Figure 7.9 shows a safety valve bellows cracked due to SCC. The bellows material is 316 austenitic stainless steel. Although no chlorides are in the process fluid, the connection between the valve and the flare line made it possible for some chlorides to travel backward to the valve. Figure 7.10 shows a micrograph of the crack tip, where a branched intergranular crack can be seen.

TABLE 7.1 *Classic Examples of Metal and Environment Combinations Prone to SCC*

<i>Metal</i>	<i>Corrosive Medium</i>
Carbon and low-alloy steel	Caustic solutions, under certain conditions Humid H ₂ S Ammonia Nitrates
Austenitic stainless steel	Chlorides and hypochlorides Some caustic solutions
Brass	Ammonia, amine Mercury salts

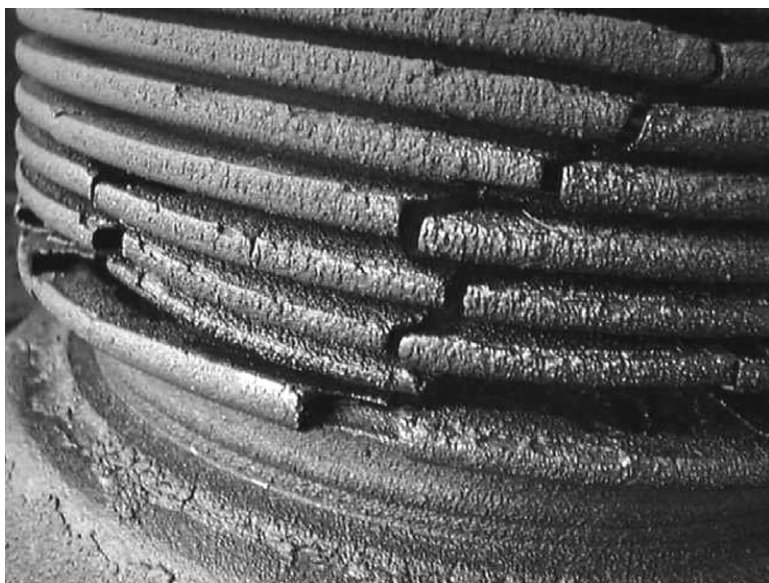


FIGURE 7.9 *SCC on a safety valve bellows.*

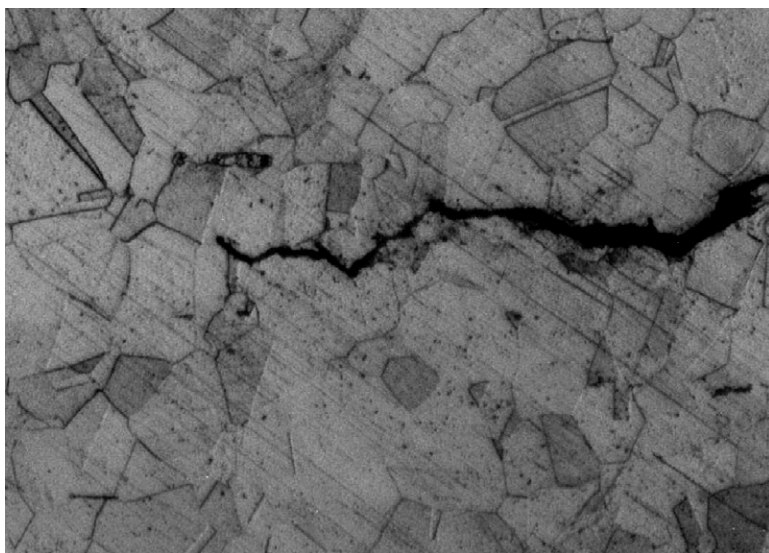


FIGURE 7.10 *Micrograph of the crack shown in Figure 7.9.*

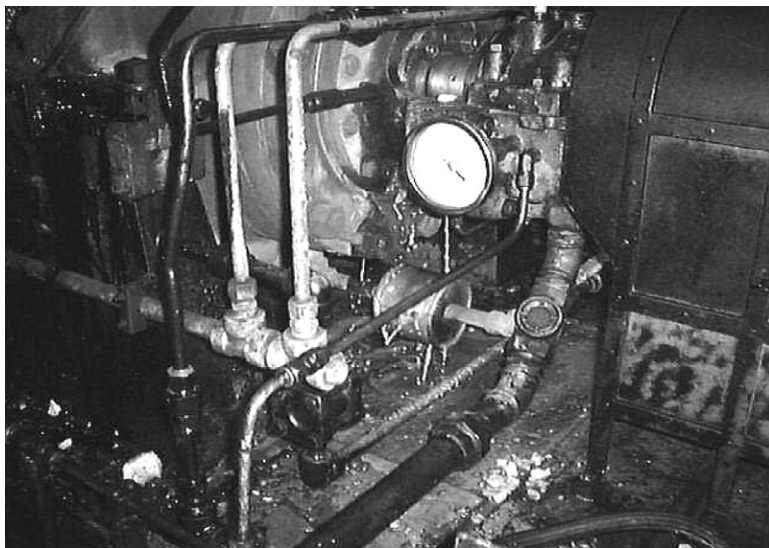


FIGURE 7.11 *Chloride bearing seal quench water drips over an austenitic stainless steel drain, causing SCC.*



FIGURE 7.12 *Drain piping with stress corrosion cracks.*

Figures 7.11 and 7.12 show another situation where off-design conditions caused SCC in a drain piping. In this case, quench water from the pump seals dripped over the drain lines. This is a high-temperature pump, the casing and piping made of 316 austenitic stainless steel. The quench water contains chlorides in a concentration small enough to cause SCC. Evaporation of the water in contact with the hot drain pipe caused the chloride concentration to increase, and SCC resulted.

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8

Incrustation

Incrustation is a failure mode in which deposition of material from the environment or process on the surface of the components of a machine causes deterioration in its performance. This deterioration can take several forms:

1. Parts supposed to have relative movement can be stuck together by the adhesive action of the deposits. A good example is the leakage of a mechanical seal due to the loss of flexibility caused by incrustation.
2. Components designed to interact may be impaired by the deposits. For example, the plug of a globe valve may not seat correctly if the contacting surfaces are covered with a foreign material, resulting in leakage.
3. Flow obstruction can be observed in a variety of situations, such as in steam turbine blades or the inside of tubes.

Formation of the deposits may have different causes, depending strongly on the type of process in which the equipment is installed. The type of fluid and contaminants present, the operating pressure, temperature, the relative speed, and the type of surface (material,

roughness, shape) influence the deposition mechanism. These factors also determine the rate of material deposition on the surface and the characteristics of the deposit, like hardness and solubility.

Deposits may be divided in two different types: deposits that are a direct consequence of the process environment and those that result from an abnormality. When the incrustation is a consequence of the process, the design of the machine and the system should take this in account.

This means that, during the design, measures should be taken to avoid the deleterious effects of the unavoidable incrustation. Some examples are the use of flushing systems in valves seats and pump wear rings, the installation of washing systems in centrifugal compressors and steam turbines, and the multitude of coatings designed to avoid incrustation inside rotating equipment.

A different scenario can be found when incrustation happens because of a process abnormality. Design countermeasures may not be necessary here.

The chemical nature of the deposits found in process industry rotating machinery is one of the following:

1. Coke deposition.
2. Deposition of salts or other inorganic material.
3. Incrustation of living organisms.

8.1 Coke Deposition

Coke incrustation is the typical petrochemical type of deposit. When hydrocarbons are exposed to high temperatures and the correct combination of low relative speed and residence time, the molecules may crack. The cracked hydrocarbons have a strong tendency to polymerize, due to the number of radicals made available. Unsaturated hydrocarbons are more prone to polymerization.

The result of this process is a hard, dark layer of coke, composed mainly of carbon. This coke layer can be very hard or waxy, depending on the conditions of formation. It can be insoluble by almost every known chemical and very difficult to remove mechanically.

Figure 8.1 shows the plug of a globe valve covered with coke incrustation. This incrustation was formed by the decomposition of heavy hydrocarbons processed at 500°C.

Figure 8.2 shows the head of a reciprocating cylinder with coke incrustation, created by the polymerization of the process gas. The temperature increase during compression helped polymerization of the unsaturated parts of the gas. The rounded shape of the deposits indicates that they were liquid at some time in the past. This deposition is a direct consequence of the process; the compressor should have been designed to withstand this effect.

Figure 8.3 shows a centrifugal compressor with heavy coke incrustation. This compressor operates in the same type of plant as the former one, but in this case, the coke formation was a result of the incorrect operation of the impeller washing system, installed to avoid this type of problem. Figure 8.4 shows the nozzles used for on-line washing of a compressor.



FIGURE 8.1 *Coke deposit on the plug of a globe valve.*

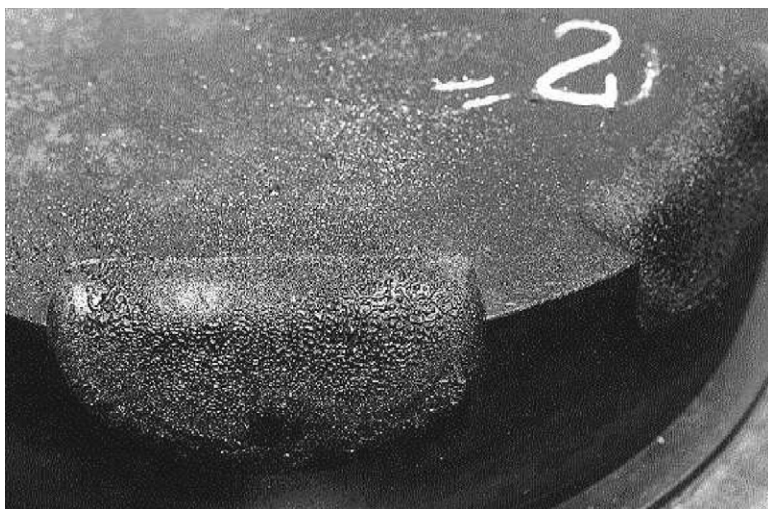


FIGURE 8.2 *Coke deposit on the head of a reciprocating compressor.*



FIGURE 8.3 *Coke incrustation in a centrifugal compressor.*



FIGURE 8.4 *Nozzles installed for on-line washing of compressor rotor.*

8.2 Salt Deposition

This is the precipitation of salts dissolved in the process fluid. Some examples follow:

1. Silica and other salts can be found on steam turbine blades, for example. This deposition can be a consequence of water treatment deficiencies or the inability of the boiler to avoid water droplets being carried by the steam leaving the boiler drum. Figure 8.5 shows a steam turbine with salt deposition.
2. Ammonium chloride sometimes is found in equipment that uses ammonium compounds as corrosion inhibitor. This type of salt is also found as a by-product of some catalyst regeneration operations, in certain plants. Figure 8.6 shows a centrifugal compressor rotor covered with ammonium chloride due to an abnormal operation during catalyst regeneration.

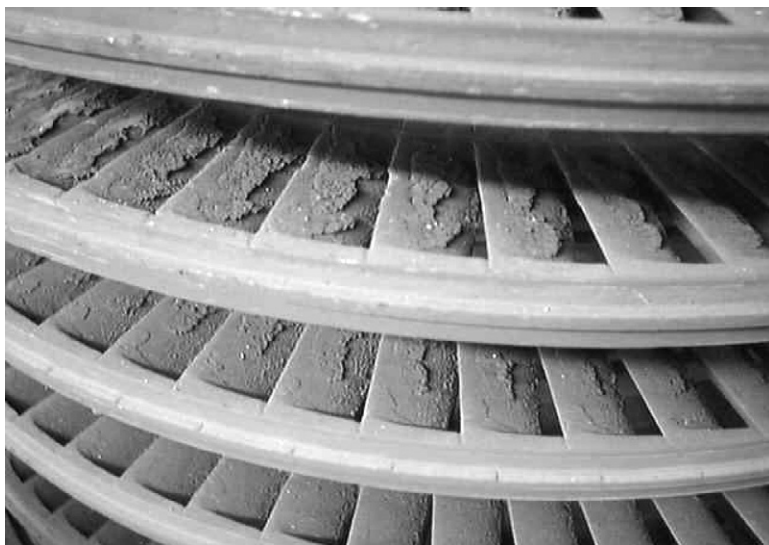


FIGURE 8.5 *Silica and calcium deposits on steam turbine blades.*



FIGURE 8.6 *Ammonium chloride incrustation in a compressor rotor.*



FIGURE 8.7 *Iron fluoride incrustation on a mechanical seal carbon ring.*

3. Various fluorides can be found inside the machines installed in HF (hydrofluoric acid) alkylation plants. Figure 8.7 shows a mechanical seal carbon ring covered with iron fluoride incrustation. These deposits jeopardize seal operation, leading to leakage.

Even air compressors may be affected by incrustation of airborne impurities. The most common strategies to avoid this problem are the use of filters and cleaning systems. Antiadherent coatings can also be used. This Teflon-based coating makes deposition more difficult.

8.3 Biological Incrustation

Biological incrustation is the development of vegetables or animals on the equipment surfaces. This is a problem well known to the offshore industries, where a great variety of organisms adhere to ship hulls and offshore platform structures. The inshore process



FIGURE 8.8 *Seawater pump suction strainer with heavy biological incrustation.*

industry sometimes suffers from this type of alien invasion, especially in the raw water systems. Seawater pumps can be heavily affected.

Several techniques have been developed to avoid this type of incrustation, with the use of toxic pigments in the painting, the introduction of chemicals to the water, or the recirculation of hot water from time to time. Attention should be paid to the possible environmental impact of the designed countermeasures.

Figure 8.8 shows a seawater pump suction strainer covered with a variety of sea creatures. Pump performance is affected by the clogged inlet strainer.

9

Electric Discharge Damage

A turbomachine may be transformed into an electricity generator under certain conditions. If any part of the machine is magnetized, the relative movement between this part and others may generate electricity, as inside a conventional electricity generator. This phenomenon, called *electromagnetism*, happens harmlessly in a large number of generators around the world but may be dangerous in a machine not designed to handle the electric current. Another possibility is the generation of electricity due to the flow of certain types of fluid, like damp air or wet steam. This static electricity generation mechanism is similar to the one that creates thunder and lightning in a storm cloud.

The circulation of electricity inside a conventional turbomachine can be harmful in two ways:

1. Generalized heating of the component because of the Joule effect.
2. Localized melting in close clearance areas due to sparking.

Heating due to Joule effect may diminish the strength of the material, which may lead to failure. High temperature oxidation

also may be found; the component surface will show a characteristic overheated appearance. Thermal growth may cause rubbing between close clearance components that have relative movement.

Localized melting due to the sparks may happen in close clearance areas if there is current flow and sparks. These close clearance areas can be the bearing or seals of the machine. A surface affected by electrical discharge sparking shows localized indications, which may spread throughout part of the surface. The macroscopic appearance is similar to a sand blast surface.

Figure 9.1 shows a steam turbine governor primary element that was affected by both these mechanisms. The sand blasted-like areas are the result of sparking, the center area oxidized due to the high temperature resulting from the Joule effect. Some components of the turbine were magnetized for unknown reasons, and the electricity generated has been enough to damage the component.

Hydrodynamic bearings are especially sensitive to this failure mode, as the Babbitt usually has a low melting temperature. The



FIGURE 9.1 *Primary element of a steam turbine governor with signs of electric discharge damage.*

bearing surface may show a sand blasted appearance. The damaged region often is at the close clearance spots, where sparks are more likely to develop.

As the material removal continues, the bearing profile changes and the bearing loses its load-carrying capacity. If the oil film is lost, the contact between the shaft and bearing cause massive destruction, removing any indication of the electrical discharge damage. Failure analysis is very difficult without evidence of the electric discharge damage, and it is likely that this failure might be diagnosed as a lubrication problem.

The macroscopic appearance of the damaged surface is best observed with a low magnification lens. The sand blasted appearance is immediately obvious. Figure 9.2 shows a turbine bearing with signs of electrical discharge damage. An examination with a low magnification microscope (20–30 $\times$) shows a series of rounded indications with a shiny bottom, the result of the localized melting of the surface. Figures 9.3 and 9.4 show examples of bearing surfaces with this type of damage.

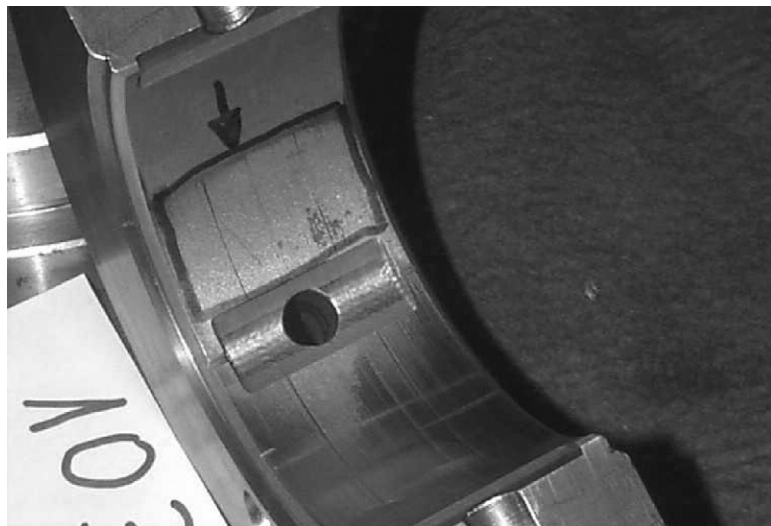


FIGURE 9.2 *Electrical discharge damage on a steam turbine bearing.*

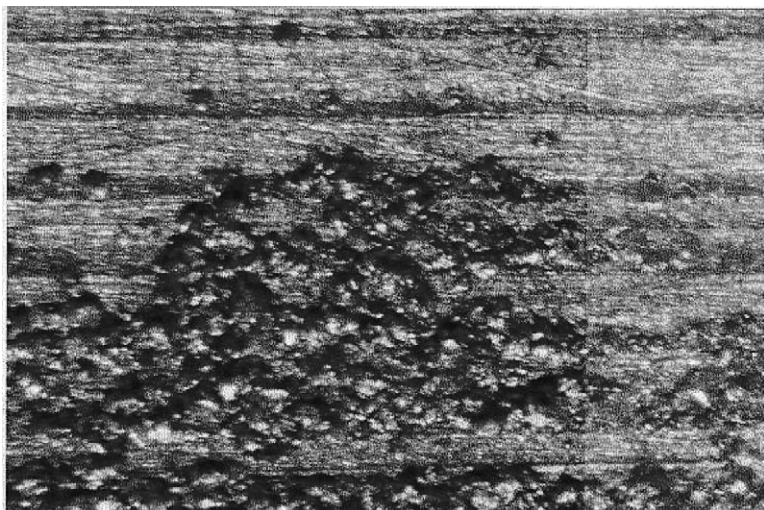


FIGURE 9.3 *Micrograph of electrical spark damage on a bearing surface.*

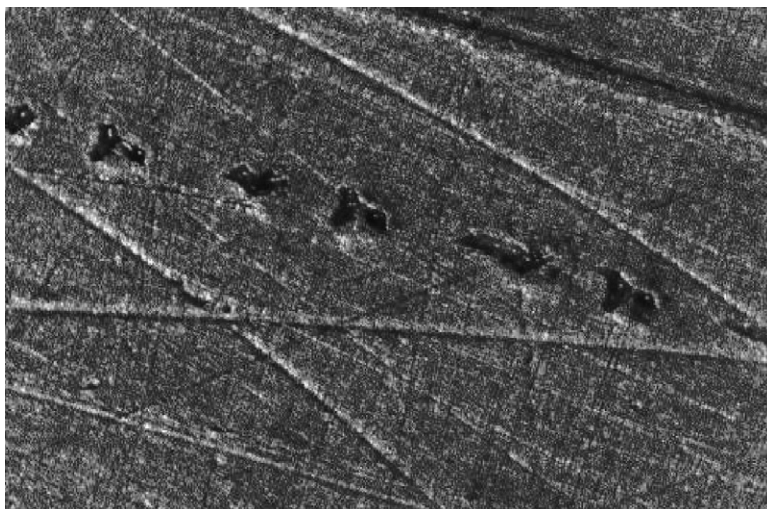


FIGURE 9.4 *Micrograph of a thrust bearing surface with indications of electrical discharge damage.*

Shafts also may be affected, but the appearance of the damage is somewhat different, due to the great difference in material properties. Figure 9.5 shows an example of a steam turbine shaft with the characteristic sand blasted finishing. Antifriction bearings may be affected as well.

The main sources of electricity in a rotating machine are

1. Static electricity, generated by the flow of fluids inside the machine. This is a common situation for condensing steam turbines and air compressors. The quantity of energy generated this way usually is small, and the problem often can be solved with the introduction of effective shaft grounding.
2. Electromagnetism, generated by relative movement of magnetized components, may result in appreciable energy generation. Current densities usually are larger than before and it is necessary to demagnetize the machine completely to avoid damage.

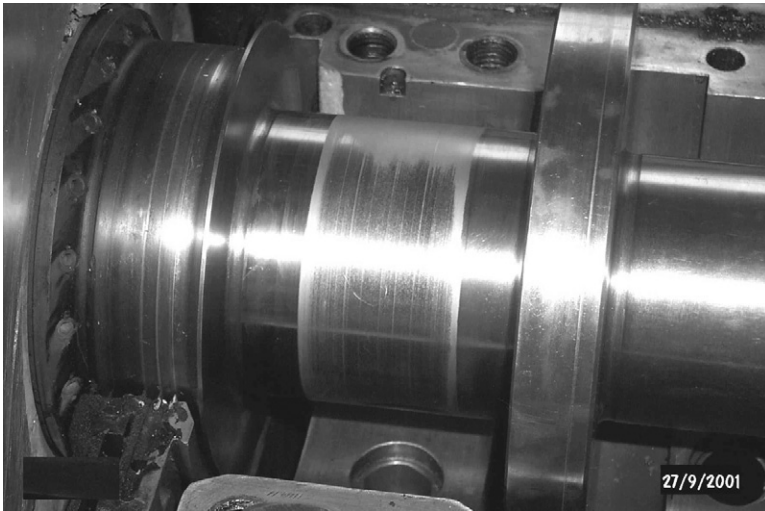


FIGURE 9.5 *Condensing steam turbine shaft with indications of electric discharge damage.*

Machine components may become magnetized for several reasons:

1. The use of magnetic plates during machine operation.
2. Magnetic particle inspection without proper demagnetization.
3. Flow of electricity through the machine, possibly because of welding discharge or suction piping without proper grounding.

Some effective ways to avoid this type of failure mode are

1. Make sure all the components of the machine are demagnetized. A portable Gauss meter may help. Residual magnetism below 5 gauss is recognized by the API (American Petroleum Institute) standards as a safe level. Some authors suggest that rotating components should be demagnetized to less than 2 gauss residual magnetism.
2. Correctly insulate the generator and motor bearings from the casing, especially for the large electric machines.
3. Install grounding brushes and keep them in good condition, especially for machines at risk of developing static electricity. Note that most large motors and generators use carbon grounding brushes, which may be somewhat ineffective, as they need a high potential for polarization and are easily insulated from the shaft by a thin layer of oil. Metal filament brushes may be more effective.
4. Avoid electric arc welding close to the machine. If this is not possible, make sure that the grounding does not allow any current to flow through the machine.

Part III

Machinery Component Failures

The failure modes described in the preceding part manifest themselves in some typical forms, depending on the type of machine part involved. An analysis of the way those failure modes show up in different machinery components makes it easier for the failure analyst to find the root causes of a problem. This part examines the typical failure modes found in the most common rotating equipment components. A summarized review of the function and operating mechanism of the components is included.

The appearance of the surface of a failed machine component can say a lot about what happened. The great number of pictures found in this part can be used as a “component failure atlas,” to make failure mode identification much easier. The importance of examining the failure surface recommends that the greatest possible care be taken to preserve it. Any modification of the surface may hide important evidence and make the failure analysis more difficult.

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10

Shafts

The function of the shaft is to support the rotating components or transmit torque from one part of the train to the other. These functions usually are associated with a requirement for a specific lateral and torsional stiffness. The shaft stiffness is controlled in an attempt to control the deflexions and resonance frequencies of the rotor, which can have a great influence on the dynamic behavior of the machine.

Machine shafts work in quite varied conditions and environments. The load acting on the shaft can be any combination of torsion, flexion, and traction. The useful life of shafts generally is indefinite, which means that any shaft failure should be considered an abnormality.

Most shaft failures are fatigue fractures. The origin of these fractures usually is found in stress concentrators such as fillets, keyways, or threads. Ductile and brittle fractures are rarely found and normally are associated with poor material selection, fabrication problems, or gross overload.

The steps described in Chapter 2 should be followed as a preparation for a shaft failure analysis. The first and most important step is data collection. The most important information to be collected is

1. Design and fabrication of the shaft: all the available information about the stresses, deflection, natural frequencies, chemical analysis, heat treatment, mechanical properties, a detailed drawing, and the like.
2. Operational history of the machine, including vibration and maintenance records.
3. Loads acting on the shaft and the environmental conditions.
4. Abnormal operating conditions.

Special attention should be paid to stress concentration regions. Also very important is the alignment of shafts and bearings. The failure surface should be inspected very carefully. Abnormal indications are quite revealing, especially fretting, because it reduces fatigue strength.

Figure 10.1 shows an example of a shaft fracture. This high-speed steam turbine shaft failed due to fatigue.

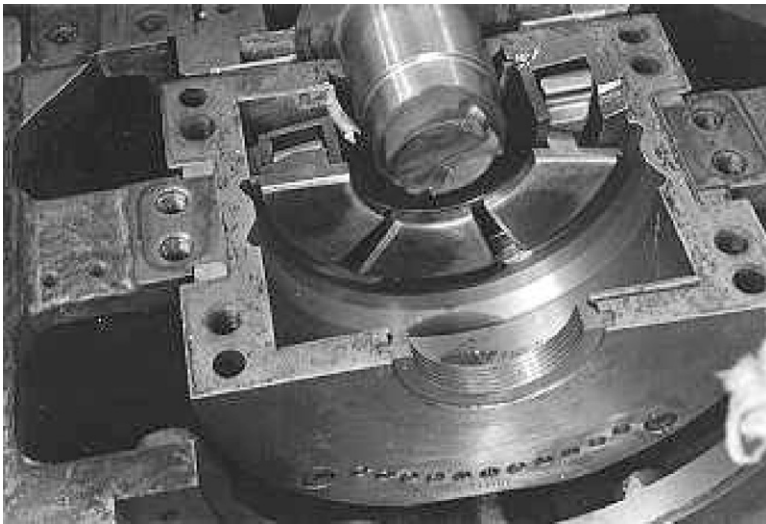


FIGURE 10.1 *Fatigue fracture of the shaft of a 9 MW, 7,000 rpm steam turbine.*

10.1 Stresses Acting on a Shaft

External loads acting on a body generate internal stresses. The stress distribution is a function of the distribution of the external loads and the shape and mechanical properties of the body. If these internal stresses cause a fracture, the fracture face shape will correspond to the stresses inside the body. Fatigue and brittle fractures grow in a direction perpendicular to the main tension stress direction. Ductile fractures develop in the direction of the highest shear stress.

During failure analysis, the correlation between fracture shape and internal stresses and the connection of internal stresses to external loads can help identify the loads that caused the fracture. Figure 10.2 illustrates the main stresses that develop from four typical external loadings. Keep in mind that there often are combinations of these load types, instead of only one.

Note also that modern turbomachine shafts are designed based on a deformation criterion. Therefore, the stresses acting on the shaft usually are small, at least under normal operating conditions.

Figure 10.3 shows a pump shaft that suffered a brittle fracture from a torsion load. The helical shape of the fracture face is readily seen. The fracture propagates at a 45° angle to the shaft centerline, because this direction is perpendicular to the main tension stress. This is typical for a brittle fracture.

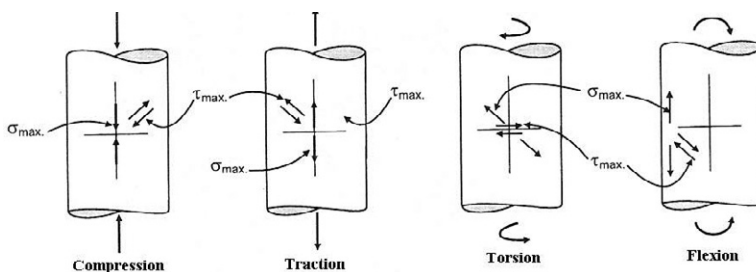


FIGURE 10.2 Internal stresses acting on a shaft, as a function of the external load.

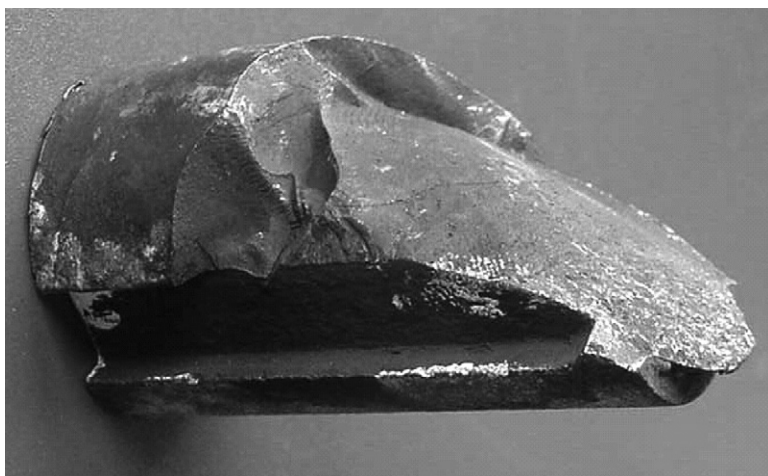


FIGURE 10.3 *Brittle fracture of a shaft due to torsion.*

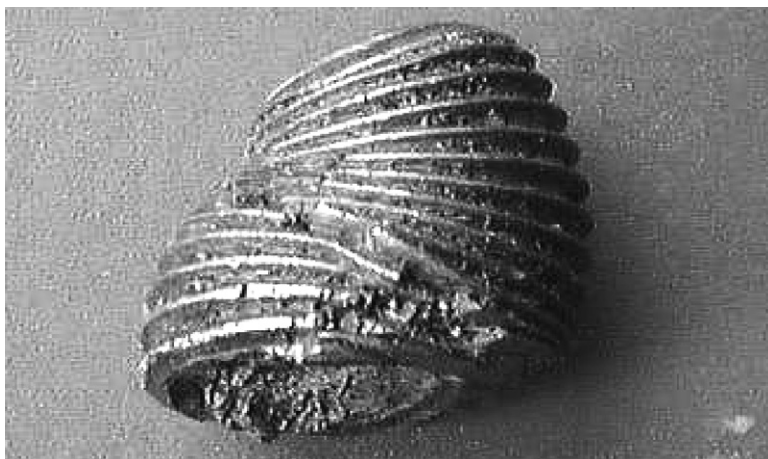


FIGURE 10.4 *Bolt damaged by axial overload.*

Figure 10.4 shows a screw that failed due to a compressive axial load. As the material is ductile, the fracture direction is at a 45° angle from the bolt centerline, in the direction of the main shear stress, in this case.

10.2 Fatigue Failures

The cyclical loads that may cause a fatigue failure of a shaft may or may not be associated with the rotation of the shaft. A machine driven by a two-pole induction motor is very common in the process industry. The 3,550 rpm of the shaft turns out to be about 5 million turns a day. The fatigue endurance of steel usually does not change above 1–10 million cycles. Therefore, if a shaft of such a machine suffers a fatigue fracture after more than a few days of work, we should look for the loads that caused the fracture somewhere other than in the rotation of the shaft. Of course, this assumes that conditions do not change too much until the shaft fracture. Some typical factors that may contribute are fretting and cavitation.

Figure 10.5 shows a fatigue fracture of a steam turbine shaft. The average stress acting on the shaft was very small, as indicated by the small final fracture area. The initiation site can be seen clearly on the right side of the picture. Surprisingly, the fracture did not initiate at the point with a higher stress concentration, the keyway. Another driving force caused the initiation at the point indicated, but this was not found at the time of that failure.

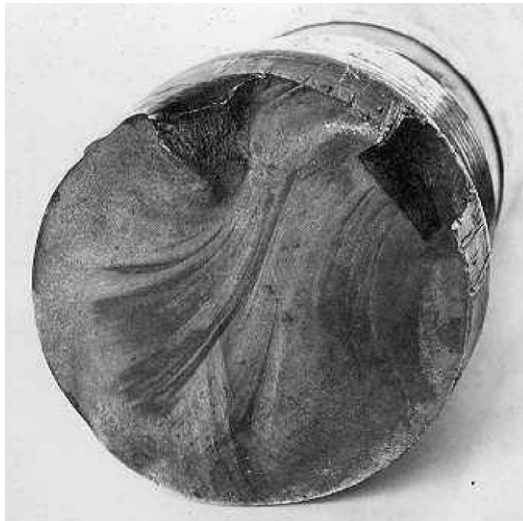


FIGURE 10.5 *Fatigue fracture of a steam turbine shaft.*

This shaft fracture clearly shows the three characteristic elements of a fatigue fracture. One should not assume that this is always the case, as there is a certain probability of damage to the fracture surface. The most common problems that cause the destruction of the evidence provided by the fracture face are rubbing between both faces shaft rotation and rust, or poor preservation after the failure.

It is not very easy to calculate accurately the loads that act on the shaft. Nevertheless, this can be very instructive when analyzing a failure. Machine design handbooks provide the methods to perform these calculations.

Torsional fatigue fractures display the same characteristic indications. In this case, the stress concentration provided by keyways can be highly detrimental to shaft endurance. Fatigue failures at the keyway may happen if the coupling hub is loose on the shaft and the machine is subject to oscillating torque, like reciprocating pumps and compressors. The shaft in these fractures may seem to have been “peeled” all around or cracks may develop close to the edges. Figure 10.6 shows a keyway fracture caused by a loose key. The

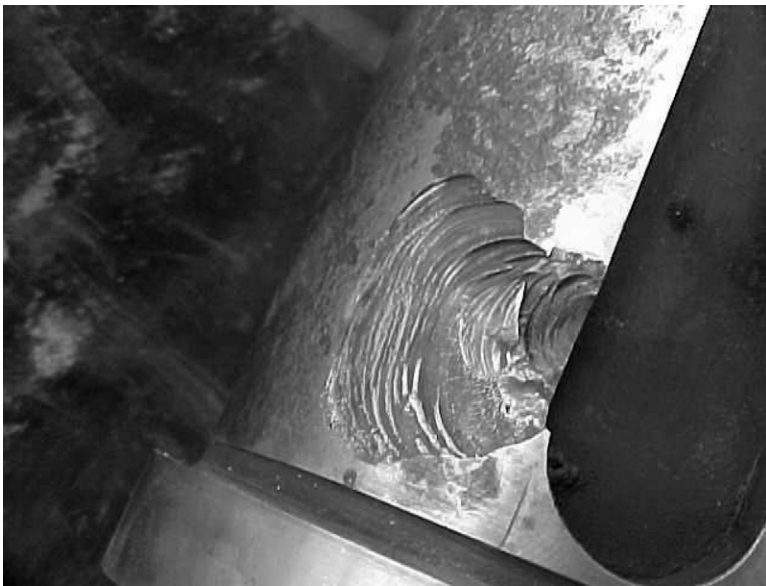


FIGURE 10.6 *Keyway cracks due to a loose key.*

loose key was not perfectly aligned with the keyway, causing a concentration of the load at the contact point.

10.3 Shaft Wear

Shaft wear usually is observed in the bearings and seals area. The reason for this is very simple: These are places where clearance among components with relative movement is close, a situation that makes rubbing likely.

More often than not, shaft wear is a result of another failure, probably of the bearing or seals. In these cases, it is more effective to analyze the first component to fail, as everything else may be only consequences of the first failure.

Figure 10.7 shows an example of shaft wear. This shaft was installed in a gear pump that pumped fuel oil to a boiler. The pump design counted on the pumped fluid to lubricate the bearings, a common arrangement for lube oil pumps. In this case, the fuel oil contained many contaminants, both solid and liquid, and, obviously, was not a good lubrication oil. The result is damaged bearings and shaft, caused by the poor choice of equipment.

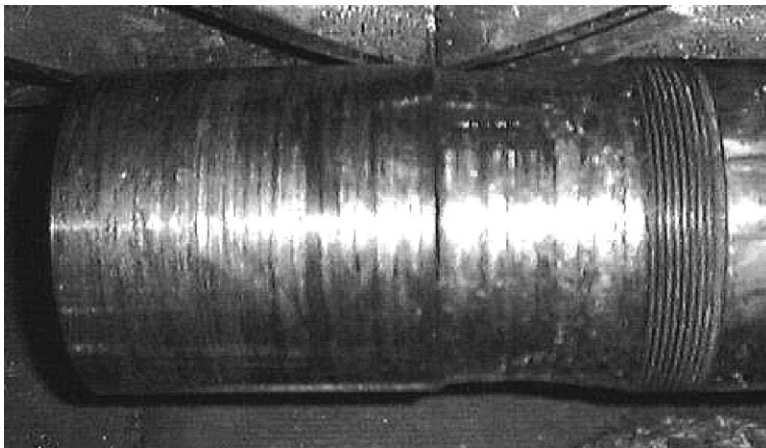


FIGURE 10.7 *Gear pump shaft with heavy wear at the bearings area.*

Another common example of wear failure in shafts is the case of vertical pumps that have radial bearings lubricated by the pumped fluid. Many times the pumped fluid is not a good lubricant, and high bearing and shaft wear results.

10.4 Shaft Distortion

Shaft distortion usually is the result of some gross accidental overload. It is not very likely for a design deficiency to cause shaft distortion. Given the small clearances inside a typical rotating machine, even very small distortions may lead to internal rubbing. Even if the distortion does not result in internal rubbing, high-speed machinery may experience high vibration levels with shaft deformation as small as a few hundredths of a millimeter.

Some classic situations in which shaft distortion may happen are

1. Induced draft fans, which may operate at high temperatures. If the fan remains inactive and exposed to the high temperature flue gas, shaft distortion may happen.
2. Vertical pumps with long slender shafts. This type of arrangement is particularly sensitive to shaft misalignment, and it is particularly difficult to obtain a proper setup during the initial installation or any maintenance work.

11

Hydrodynamic Bearings

Hydrodynamic bearings are intended to provide support to a rotating shaft. This support can be either radial or axial. In addition to supporting a certain load, these bearings usually are designed to influence the rotodynamic behavior of the machine, providing specific quantities of stiffness and damping. This is especially important for high-speed machinery (machines that operate above the first critical speed), as bearing stiffness influences the resonance frequency and bearing damping influences the shaft vibration when the operating speed is close to the critical speed. There are several different types of hydrodynamic bearings but the operation principles are the same.

The useful life of this type of bearing is considered indefinite. The expected end-of-life failure mode is sliding wear due to startup and shutdown rubbing. If a hydrodynamic bearing operates continuously, the potential life span is infinite, as there is no metal to metal contact.

The dynamic behavior (dynamic stiffness and damping) of a hydrodynamic bearing depends on the shape of the bearing (cylinder, tilting pad, pressure dam, etc.), the clearance between shaft and bearing, the shaft speed and load, and the oil viscosity. A modification in any of these parameters can change the bearing behavior,

with the resulting high vibration, rubbing, or high temperature. A rotodynamic analysis should be performed before the modification of any of these factors. These factors can be modified by changes like a modification in the coupling or a change in the seal type. Low-speed machines are less sensitive to a change in these features than high-speed machines.

11.1 Operation of a Hydrodynamic Bearing

The basic operation of hydrodynamic bearings was explained long ago by Reynolds. The lubrication conditions of a hydrodynamic bearing are the same as described in Chapter 6, “Wear.” The three possible lubrication modes are

1. *Hydrodynamic lubrication or full film lubrication.* The minimum film thickness is between 0.0025 and 0.025 mm, much greater than surface asperity height, and the friction coefficient is smaller than 0.01. There is virtually no wear, as there is no metal to metal contact. This is the normal operation mode of most turbomachine bearings.
2. *Elastohydrodynamic lubrication or thin film lubrication.* Film thickness is approximately the same as surface asperity height, there is occasional metal to metal contact, and the friction coefficient is much higher than before. Wear is moderate, as is metal to metal contact. Operating temperatures may be high, bearing life is reduced.
3. *Boundary lubrication.* There is continuous metal to metal contact, film thickness is negligible, and the friction coefficient is high. Wear and temperature are very high. A bearing operating in this regimen does not last very long.

Figure 11.1 illustrates the behavior of the shaft inside the bearing when it begins to rotate. No hydrodynamic forces develop at 0 rpm, and the shaft rests on the bearing. When the shaft begins to turn, the metal to metal contact forces it to roll over the surface of the bearing in the direction of rotation. The shaft tries to “climb” the slope provided by the curvature of the bearing, but it soon begins to slide and rotate. The oil in contact with the shaft is drawn with the shaft as it

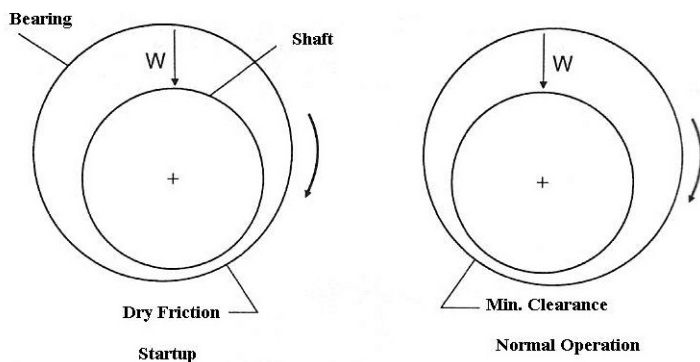


FIGURE 11.1 *The startup of a hydrodynamic bearing.*

rotates, because of the oil's viscosity. This causes the flow of oil into the wedge formed by the converging surfaces of the shaft and the bearing. The oil is “smashed” inside this wedge, and this generates the hydrodynamic force that keeps the shaft suspended.

Figure 11.2 illustrates the oil pressure distribution resulting from the effects just described. The hydrodynamic pressure generation depends only on the rotation of the shaft and the bearing characteristics, no external pressurization is necessary.

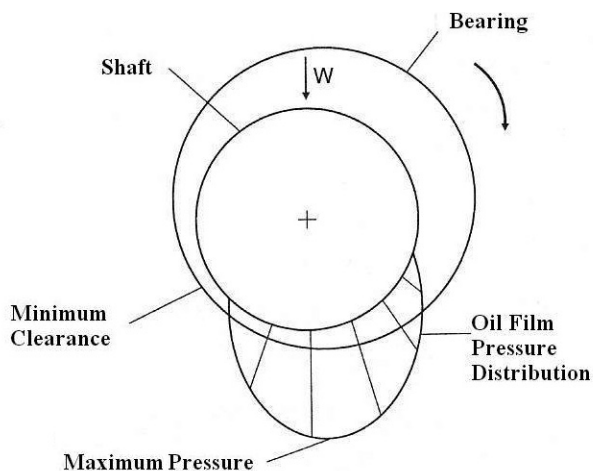


FIGURE 11.2 *The pressure distribution around a hydrodynamic bearing.*

11.2 Hydrodynamic Bearing Construction

The materials used in bearing construction should have the following characteristics:

1. High compression and fatigue strength.
2. Ability to not adhere to the shaft material in case of an accidental rub and high resistance to startup wear.
3. Ability to embed foreign hard particles and avoid shaft wear.
4. High thermal conductivity and easy conformability.
5. Resistance to oil corrosion.
6. Low cost, easy fabrication, and ready availability.

No single material has all these characteristics, and bearings are constructed with a combination of metals. Process industry rotating machinery normally uses a steel base covered with Babbitt metal to reach most of these objectives.

11.3 Hydrodynamic Bearing Failure Analysis

Chapter 2 pointed out that the first and most important step in failure analysis is data collection. In addition to the information listed in Chapter 2, the following should be investigated:

1. Work load and velocity.
2. Operating temperatures of oil and metal.
3. Failure, operation, vibration, and maintenance histories.
4. Shaft and bearing alignment.
5. Characteristics and actual condition of the lubricant, especially color, impurities, and viscosity.
6. Possible sources of contaminants to the oil.
7. Residual magnetism of the machine components.

Repetitive failures of a certain bearing may indicate a design problem. Design verification may be helpful, especially if persistent

vibration problems are present. This design verification can be especially interesting for high-speed machinery.

Inspection of the failed bearing is one of the first data collection steps. Totally disassemble and inspect the bearing before cleaning it, as debris or incrustation may provide important evidence about the failure mode.

Parts that have movement related to each other, such as bearing pivots and equalizer rings, should have only minor, uniform wear. Special attention should be paid to thrust collars, any loss of perpendicularity may accelerate the wear of the pad pivots.

The bearing surface should be smooth and uniform, with only minor scratches. Some regions may look shiny, but normally this is no problem, as long as the surface is regular and no cracks or heavy wear is present.

11.4 Fatigue Failures

Fatigue failure of Babbitt metal is similar to any other metal fatigue. The crack initiation point can be stress concentration caused by an embedded particle, a high-stress region due to shaft misalignment, high temperature, and so forth. The fatigue crack may seem to open in the direction of rotation. Pieces of metal may be carried in the direction of the shaft rotation.

Many fatigue cracks initiate on the interface between Babbitt metal and steel. The huge difference in elasticity modulus and mechanical strength creates a so-called metallurgical notch, which facilitates crack initiation in the same way as a mechanical notch.

Figure 11.3 shows the surface of a bearing that suffered fatigue due to local overload caused by shaft misalignment. The metal seems to have been peeled from the steel backing.

11.5 Bearing Metal Wear

Wear on the bearing metal of a correctly designed and operated machine happens only during startup and shutdown. Normal operation wear usually is negligible, being caused by very small particles

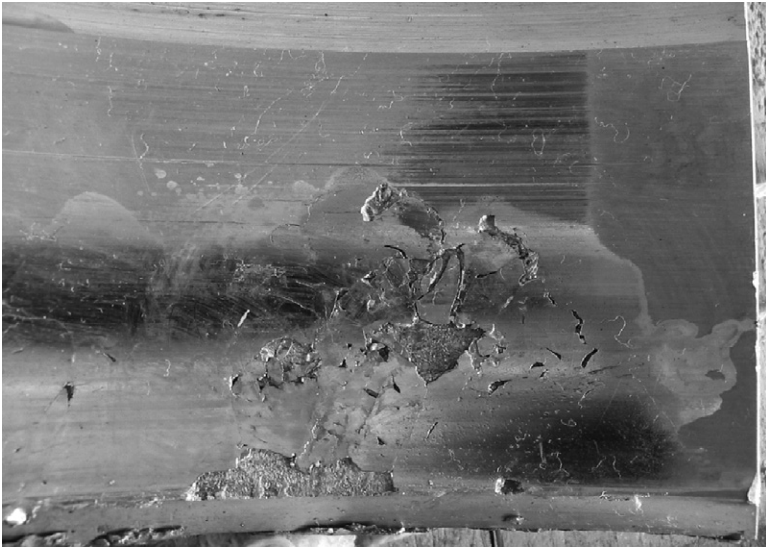


FIGURE 11.3 *Surface fatigue on a bearing.*

suspended in the oil. All the wear mechanisms mentioned in the preceding part can be found in bearings.

A bearing surface with a polished appearance and minor scratches after a long operation time can be considered normal. Original clearance is not greatly affected by normal wear. Figure 11.4 shows a bearing with only moderate wear. We see that the larger part of the surface is uniform and displays almost the original finishing. Only minor scratches can be seen and the shiny areas, which indicate shaft contact, are uniformly distributed.

A bearing that suffered minor scratches can be remachined and reused, if the resulting clearance still is acceptable. Deeper scratches are reasons for scrapping a bearing. Like a new automobile tire, the circumferential grooves are fluid escape paths, but unlike the tire, this can be deleterious to bearing performance. The reduced oil film may result in high operating temperature and wear. Figure 11.5 shows a portion of a bearing surface that has been deeply grooved by hard foreign particles.

Fretting of the pad pivots or the bearing seat is another form of wear. In this case, it is not likely that the bearing will be damaged

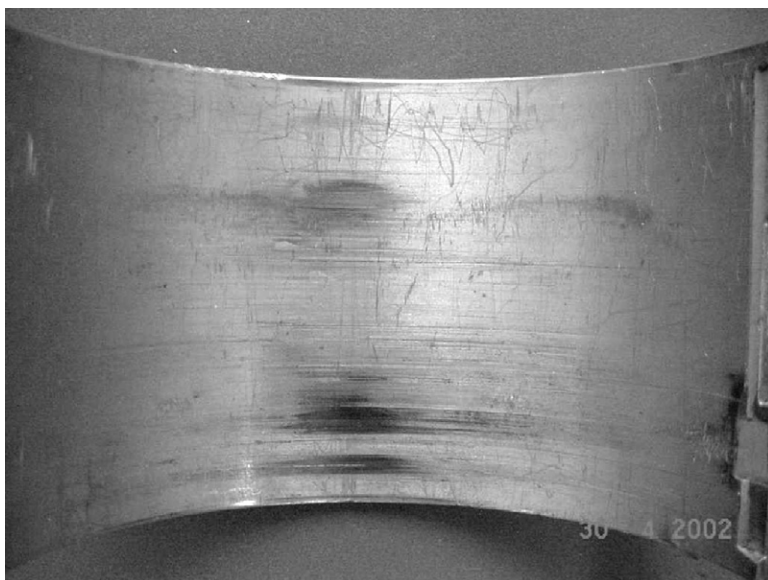


FIGURE 11.4 *High-speed steam turbine bearing with normal wear after many operating hours.*



FIGURE 11.5 *Radial bearing of a reciprocating compressor grooved by hard particles.*

directly, but high vibration of the shaft may result, due to the increased clearance between the components and consequent loss of stiffness. Some bearing designs that use an elastically flexible pad pivot were introduced some years ago to avoid this problem.

11.6 Corrosion

Bearing corrosion usually is caused by oil contamination or decomposition. The decomposition of mineral oils creates acid compounds that can affect the bearing materials. Oil can be contaminated by water or process fluids. The resulting effect is influenced by the type of contaminant. Water corrosion may create a dark oxide layer on the bearing surface. Process fluid contamination usually is associated with some kind of seal failure. This can happen more easily in machines that have combined seal and lubrication oil systems.

Corrosion can be distinguished easily from surface fatigue by noting that corrosion initiates on the surface and fatigue on the metal interface.

11.7 Effect of Hard Particles on Bearings

The most common cause of abnormal bearing wear is hard particles in the oil system. These hard particles may be steel fragments, sand, or any other type of residue. They can enter the oil system through vents, breathers, wear, or corrosion of other machine parts in contact with the oil. Critical machine oil systems should always have adequate filters and be constructed in stainless steel to avoid most of these problems. The particle size and hardness are the controlling factors in wear:

1. Particles much smaller than the bearing clearance slowly remove bearing material through abrasion. Very small particles may not be able to hit the bearing surface strong enough to remove any material.
2. Particles much larger than bearing clearance cannot reach the bearing surface but may become fixed in a position to remove only a thin metal stripe from the shaft, a phenomenon called *wire wooling*, because of the wire wool formed in the process. Even when wire wooling does not happen, the shaft may be damaged by these hard particles, as shown in Figure 11.6.
3. Intermediate-sized particles may become embedded on the bearing metal and normally do not damage the shaft. These particles may create a stress concentration spot that could initi-

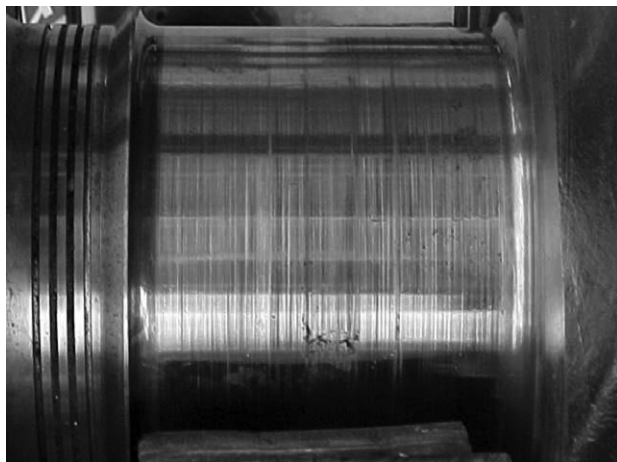


FIGURE 11.6 *Shaft damaged by the action of hard particles.*

ate a fatigue crack. If a slice of metal is removed because of the fatigue fracture, the particle that originated the stress concentration may be removed as well. This can be somewhat misleading when considering the root causes of the failure, because the particle will not be found on inspection of the bearing. The stress concentration caused by an embedded particle is illustrated in Figure 11.7. Another effect of the embedded particle is surface deformation, which may reduce a clearance point, leading to higher operating temperatures.

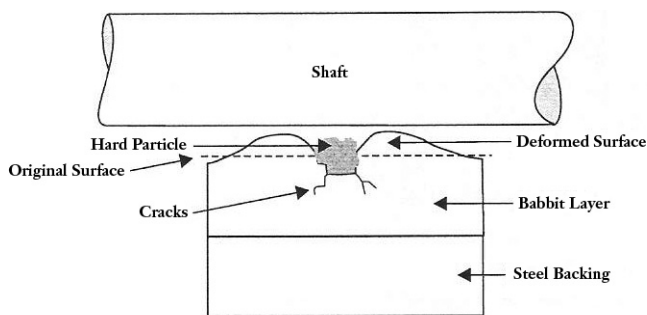


FIGURE 11.7 *The surface deformation and stress concentration caused by an embedded particle.*

11.8 Effect of Lubrication

Good bearing lubrication can reduce the effects of adverse conditions such as a bad surface finish or a certain amount of debris, but even the best oil on the market cannot make a starving bearing survive. The design and position of oil circulation grooves are critical. These channels should neither have sharp edges, which act as oil wipers, nor be located in a high-pressure area of the bearing. Some bearing designs use oil channels in the high-pressure area as a resource to control the pressure distribution of the oil film or correct the dynamic behavior of the bearing.

Some causes of inadequate lubrication are machine startup without prelubrication, insufficient clearance, low oil flow to the bearing, and defects in the lube oil pump or system.

11.9 Effect of Temperature

A hydrodynamic bearing can be overheated in a variety of ways, and the effects include discoloration, cracks, deformation, and melting of the bearing metal. Oil additives may plate the hot surfaces, creating a varnished appearance. Heat checking can result in cracks due to thermal fatigue. Figure 11.8 shows the surface of a bearing that suffered surface cracks due to thermal fatigue.

The maximum operating temperature of common bearing metals is not much higher than 100°C. High temperature operation reduces bearing life because of the great reduction in the mechanical strength of the material. Some common factors that may cause a bearing to operate in high temperatures are lubrication failures (as mentioned previously), overload, excess speed, rubbing, and cooling problems (such as a plugged oil cooler or obstructions to air flow around an air-cooled bearing).

Operation of the bearings in high temperatures often causes deformation of the Babbitt metal in the direction of shaft rotation. Figure 11.9 shows a high-speed gear bearing damaged due to high temperature caused by a failure of the lubrication system. The metal deformation of the bearing metal in the direction of shaft rotation is readily noticed.



FIGURE 11.8 *Bearing surface with thermal fatigue cracks.*

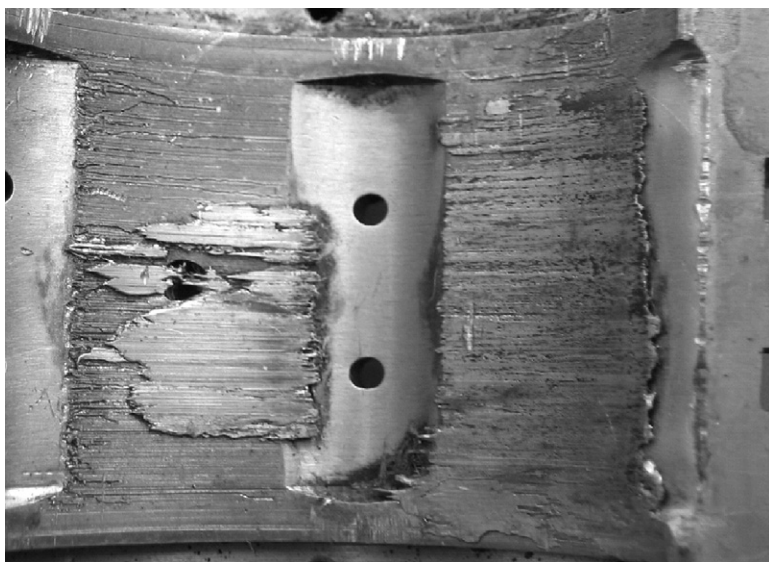


FIGURE 11.9 *High-speed gear bearing damaged by high temperature.*

11.10 Effect of Overloads

An overload pushes the shaft in the direction of the bearing. The greater the load, the closer the shaft is from the bearing surface. This reduced thickness of the oil film increases the bearing temperature and the probability of rubbing.

The appearance of an overloaded bearing is similar to an overheated bearing. Close investigation of the operating conditions may be necessary before a diagnosis can be made.

11.11 Assembly Deficiencies

An inadequate assembly may force the bearing to operate under off-design conditions, with local overload on some part of its surface. This local overload may cause local failures, which may later cause the complete failure of the bearing.

A common situation is a too tight or loose fit on the bearing seat, which may cause overheating or high vibration, respectively. Other examples are shaft or bearing misalignment, which causes local overload, and misalignment of oil passages, which may cause the bearing to starve.

An incorrect assembly usually causes irregular wear on the bearing due to the irregular contact. A simple way to avoid this is to check the bearing contact with Prussian blue during assembly. Figure 11.10 shows a gearbox bearing with surface fatigue on one side. This irregular damage pattern results from misalignment of the bearing relative to the shaft. The poor contact between the shaft and bearing caused local overload beyond the possibilities of the bearing material on the damaged region.

11.12 Electrical Discharge Damage

Although it is not difficult to recognize the indications of electrical discharge damage, it can be very difficult to find evidence of this type of problem if the damage to the bearing is extensive. Figure 11.11 shows an unusual example of the result of this type of damage. The spark damage indications are easily recognized with microscopic inspection. Large cracks can be seen on the bearing surface, caused by

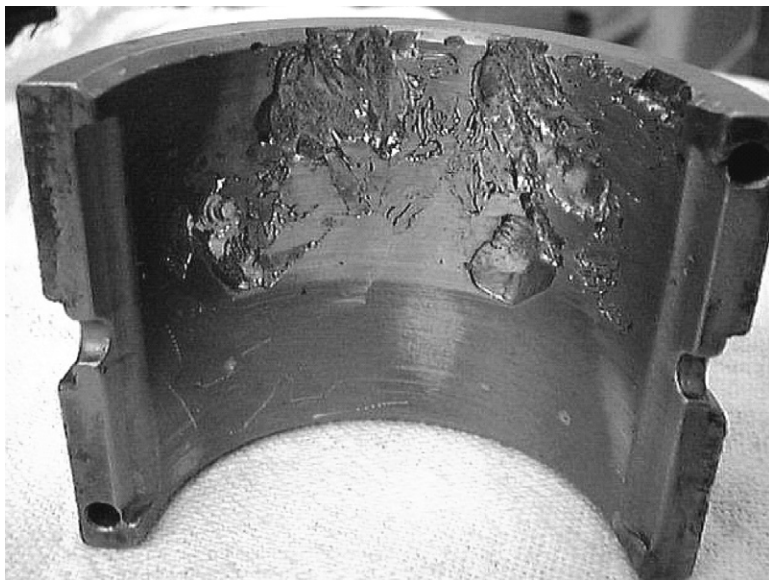


FIGURE 11.10 *Irregular wear on a gearbox bearing due to misalignment.*

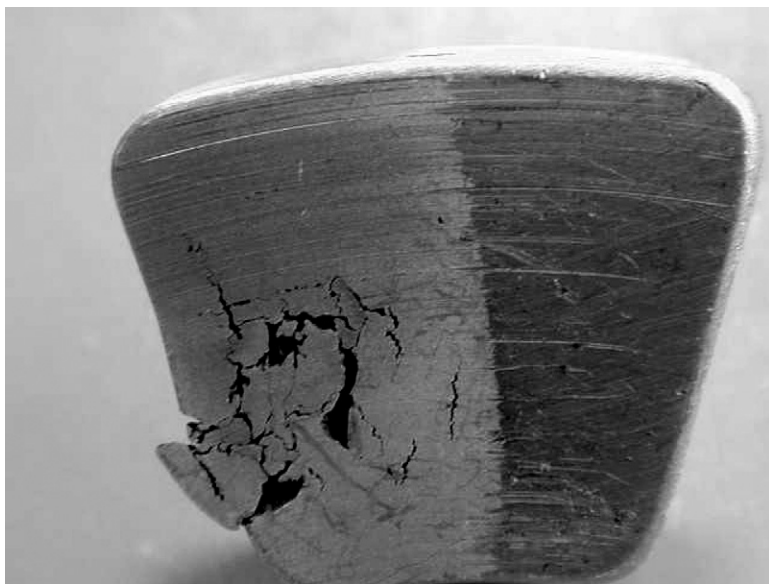


FIGURE 11.11 *Thrust bearing pad showing signs of spark damage.*

the overload that resulted from the loss of the oil film, generated by the modification of the surface caused by the electrical sparks.

The unusual situation shown here is that this bearing pad was found in a transition situation between displaying only the electric discharge damage and being totally destroyed. Should this bearing be disassembled and inspected some time later, it probably would not be possible to find the characteristic indications of electrical discharge, because the bearing metal would have been removed. This would have made failure analysis very difficult, if not impossible.

11.13 Fabrication-related Failures

Although not very common, manufacturing defects can be found from time to time. A manufacturing defect should be suspected only if some evidence is found. In addition to the more obvious cases, when dimensions or surface finishing is not correct, some other situations can be found:

1. Insufficient adherence between the bearing metal and the steel backing can cause detachment of the Babbitt metal. An ultrasonic inspection can find the adherence loss, but the results of the inspection are not easily interpreted. The large difference in mechanical and physical properties between steel and Babbitt metal cause strong reflections of the sound wave, even if adherence is perfect. A comparison with a standard bearing of known quality is the only way to perform an effective inspection. Figure 11.12 shows an example of lack of adherence, in this case quite easily noticed even with an unaided eye.
2. Blistering and porosities result from gas evolution during casting. This problem is easily avoided with careful quality control during fabrication. Figure 11.13 shows an example of a bearing with a surface blister. In this case, the blister was located close to the edge of the bearing and had no effect on its load carrying. Should the blister be found on the load-bearing area, it would probably create a weak point prone to failure.
3. Material quality problems may cause the bearing metal to be softer or harder than desired. This may cause premature failure of the bearing, if it is too soft, or shaft damage due to hard

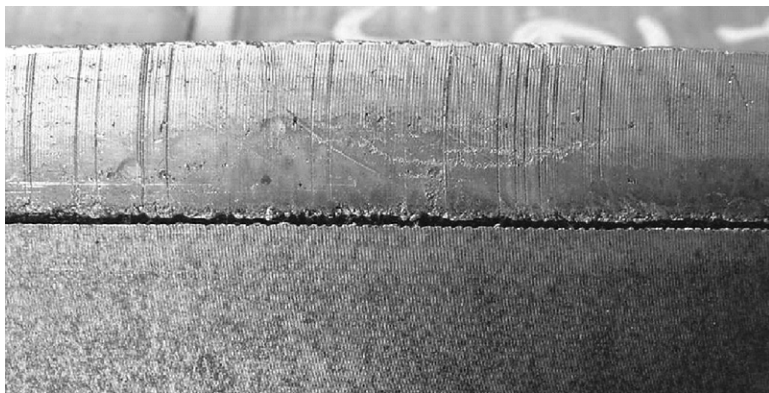


FIGURE 11.12 *Insufficient adherence between the Babbitt metal and steel backing.*



FIGURE 11.13 *A blister on the surface of a bearing.*

particle or contact wear, if it is too hard. Another problem is the application of a new Babbitt layer over an old one to restore a used bearing. Although not an approved procedure, sometimes it may be found, because of poor maintenance practices. Figure 11.14 shows an example.

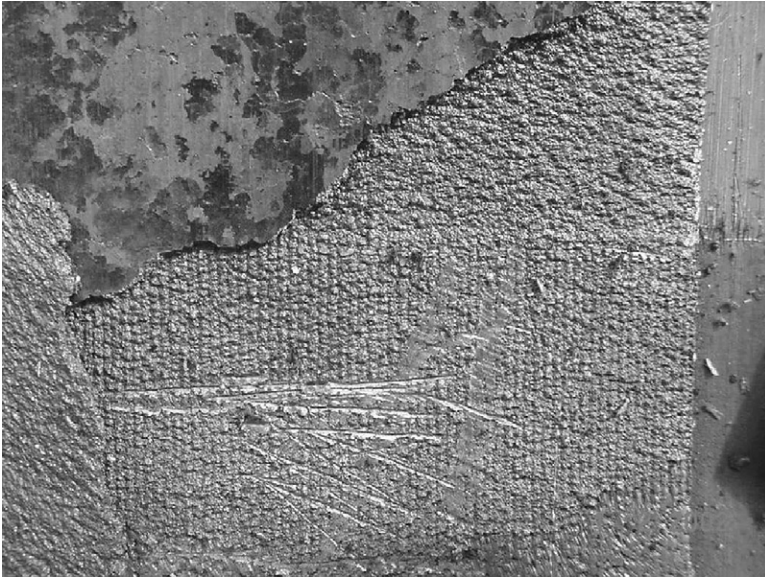


FIGURE 11.14 *Detachment of the Babbitt layer due to an improper restoration procedure.*

11.14 Design-related Failures

The issue of design deficiencies can be found in connection with a variety of machinery failures. Hydrodynamic bearings design failures can be related to a series of situations:

1. Improper load-carrying capacity, which should be suspected when the bearing is overloaded, overheated, or both.
2. Improper dynamic behavior, when the component presents dynamic stiffness and dampness that are inadequate for the desired machine rotodynamics. This type of problem should be suspected if the machine displays persistent high vibration, not reduced with normal maintenance.
3. Oil system design failures, when the oil system cannot supply the adequate quantity of clean, cool oil for proper bearing operation.

Figure 11.15 shows an example of the last situation. This is an electric motor fitted with hydrodynamic bearings, like many others. The size of this motor, 1.5 MW, and its speed, 3,550 rpm, had an influence on the choice of proper bearing cooling system, which was borderline. A slightly smaller motor would be adequately served by an air-cooled bearing, the next larger size motor would require pressurized oil lubrication for proper cooling. The manufacturer chose to install a hybrid system, with an oil ring and oil reservoir under the bearing and a small oil flow to increase the heat removal.

The oil circulation system designed for the motor required a very small injection pressure, 0.1 kgf/cm^2 . Most forced lubrication systems use injection pressure of about 1 kgf/cm^2 . Using such a low pressure made it easy for the motor to loosen its oil injection, following a variety of small problems, ranging from dirty oil screens to foam on the reservoir. An increase in the injection pressure caused

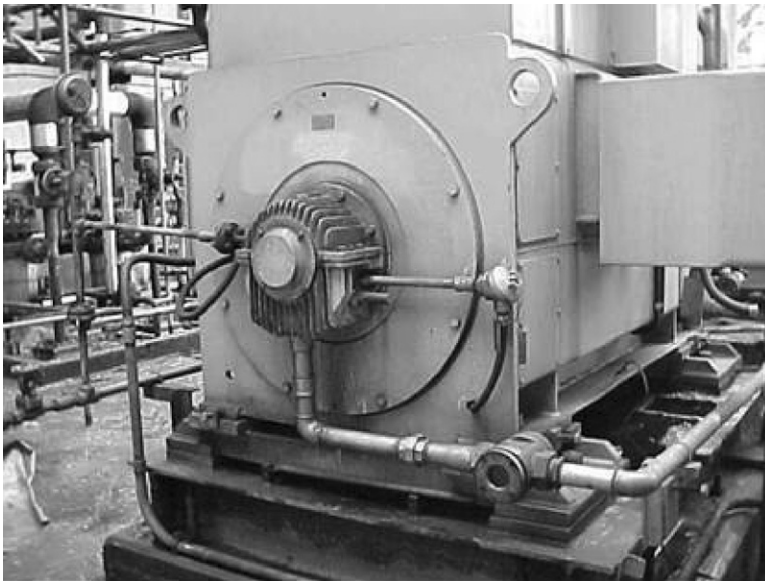


FIGURE 11.15 *Electric motor that had its bearings and the oil return line modified to allow for proper oil flow.*

the oil to leak past the motor labyrinths, because the return line had a small diameter, causing a large pressure drop.

The bearings were replaced by normal pressurized bearings and the return line (shown in Figure 11.15) enlarged to 1½ inches nominal diameter. The reduction in the pressure drop in the return line made it possible to feed the bearings with the proper amount of oil. These simple modifications solved the problem.

12

Antifriction Bearings

Antifriction bearings are intended to provide support to rotating shafts. Unlike hydrodynamic bearings, antifriction bearings are seldom used to control the dynamic behavior of the shaft because of their negligible dampening and very high stiffness, when compared with the other bearings.

The other major difference between antifriction and hydrodynamic bearings is that the former usually is selected to have a finite life. This finite life is calculated based on the ratio between actual and allowable loads. The calculated bearing life can be reached only if

1. The operating conditions, like loads, speed, and temperature, are within the design.
2. The bearing is lubricated by uncontaminated oil of the right quality and quantity.
3. Bearing installation is done correctly, with no damage, distortion, or misalignment.
4. The dimensions of the parts related to the bearing are correct, within close tolerances.
5. The bearings have no manufacturing defects.

When all these factors are correct, a specified percentage (usually 90%) of the bearings reach the end of the design life. In this case, the end of life failure mode is surface fatigue. Two conditions must be fulfilled before a bearing is considered to have reached the end of its useful life: The actual life of more than 90% of the bearings should be greater than the design life and the failure mode should be surface fatigue. A more detailed discussion about bearing design life is presented further in this chapter.

12.1 Antifriction Bearing Lubrication

As discussed previously, the ideal lubrication is found when the oil film is able to separate completely the moving surfaces. Antifriction bearing lubrication is no different. Figure 12.1 illustrates the oil film formation of an antifriction bearing. The wedge is provided by the curvature of the elements. The relative speed is provided by the bearing rotation.

The conditions for the formation of the oil film are somewhat similar to those of a hydrodynamic bearing. The oil film is thicker with an increase in oil viscosity and speed and a reduction in load. The usual selection of bearings is based on the load-carrying capacity information published by the manufacturers. For these normal selections, there is a direct relation between bearing speed and minimum oil viscosity to create an adequate oil film. Note that the vis-

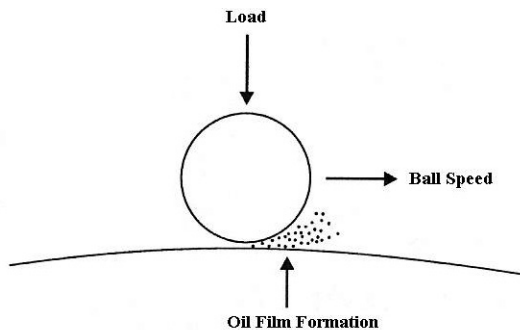


FIGURE 12.1 *The oil film formation in an antifriction bearing.*

cosity is required at the operating temperature of the bearing, not at room temperature. Oil viscosity is usually specified at 40°C, corrections should be made for any different temperature. This minimum viscosity information is also published by the bearing manufacturers. Oil viscosities greater than the minimum required result in larger film thickness and longer bearing life. Smaller viscosities, on the other hand, cause reduced bearing life.

Although similar to the oil film formation of hydrodynamic bearings, the mechanism for antifriction bearings has one important peculiarity: Contact pressures are much higher, which results in much less film thickness, which increases the importance of the elastic deformation of the bearing components and the oil viscosity.

The contact pressure can be as high as 10,000 kgf/cm². At this level of contact pressure, oil film formation is not possible with the usual viscosities of common oils, because the oil would be expelled from the contact area. Just as viscosity increases with a reduction in temperature due to a reduced intermolecular distance, oil viscosity increases greatly when the pressure goes up. At the normal contact pressure found between antifriction bearing elements, oil viscosity can be 10,000 higher than ambient pressure viscosity. The oil is somewhat like the asphalt used to pave a street.

The oil film formation mechanism depends on this viscosity increase with pressure. Water, on the other hand, exhibits no such pronounced increase in viscosity with pressure. This is why water contamination can be so deleterious to antifriction bearings.

The other important difference, a thinner layer of oil film, makes antifriction bearings much more sensitive to contaminants than hydrodynamic bearings. Even small particles can damage the bearing surface. This high sensitivity to contaminants means that bearing box sealing and oil cleanliness should be treated seriously. High vibration also can interfere with the oil film and cause metal to metal contact.

12.2 Antifriction Bearing Design Life

The design life of a bearing is usually designated L_{10} , which means that at least 90% of the bearings reach or surpass this useful life.

Bearing life is subject to great dispersion and the average life of antifriction bearings, also termed L_{50} , is four or five times larger than L_{10} . Some examples of typical design and average life expectancy for commonly found process machines are listed in Table 12.1. Do not conclude that the bearing life listed in the table represents the actual life as measured in the field. The design life listed is a common specification for this type of machine, and the average life is the life that should result due to the aforementioned dispersion.

End of useful life is the cause of bearing failure only if the failure mode is surface fatigue and average bearing life is close to the values mentioned in the table. A surface fatigue bearing failure may not mean that the bearing has reached the end of its useful life, if the average life is smaller than mentioned.

Not many plants can reach an average bearing life as large as that mentioned in the table, because it is not easy to control all the affecting factors mentioned at the beginning of the chapter. The impact of some of the factors that reduce bearing life can be appreciated with examples.

Bearing life can be reduced with an increase in load. The effect here is the same as the reduction in the number of cycles before final fracture by fatigue of a cyclical loaded component. The life of the bearing can be reduced by 8 to 10 times with twice the design load.

Water has a tremendous effect in the lubrication conditions and the bearing's life. A small contamination with water reduces the oil's capacity to have its viscosity increased by contact pressure. Only 100 ppm of water can reduce bearing life to about half of that designed. Considering that water begins to be visible in the oil at

TABLE 12.1 *Some Examples of Design and Average Life of Antifriction Bearings*

<i>Type of Machine</i>	<i>Design Life, L_{10}</i>	<i>Average Life, L_{50}</i>
Small electric motor	18,000 h	90,000 h (11 yrs)
Large electric motor	40,000 h	200,000 h (25 yrs)
API centrifugal pump	25,000 h	125,000 h (15 yrs)
ANSI centrifugal pump	18,000 h	90,000 h (11 yrs)

about 300–400 ppm, it is possible to understand how sensitive the antifriction bearings are to water.

The small thickness of the oil film also makes it sensitive to small particles. Even a small quantity of hard particles in the oil may reduce bearing life up to 20 times. This large reduction is possible because the particles penetrate the oil film and cause a large stress concentration on the bearing race or rolling element.

12.3 Contact Patterns on Bearing Races

Before examining damaged bearings, it is interesting to learn the contact patterns of normal bearings. Such contact patterns are surface modifications caused by contact with the rolling elements. These are normal indications that do not reduce bearing life and indicate how precise the bearing rotation is.

If the bearing components are considered infinitely rigid, the contact area is reduced to a point in the case of ball bearings and to a line for roller bearings. Considering that elastic deformations happen, these contact areas become a circle and a ribbon, respectively. The rotation and the loads acting on the bearing cause the distribution of these contact areas over all or part of the bearing circumference. Figures 12.2, 12.3, and 12.4 illustrate the contact pattern for bearings that worked properly with radial, axial, and combined axial and radial loads, respectively.

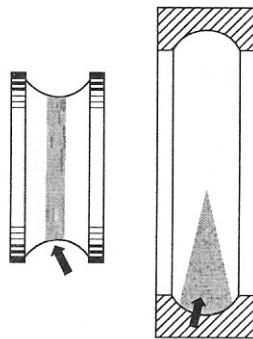


FIGURE 12.2 *Normal contact pattern on the races of a bearing subject to radial loads.*

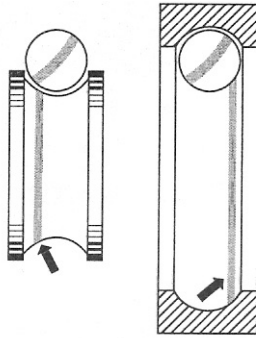


FIGURE 12.3 *Normal contact pattern on the races of a bearing subject to axial loads.*

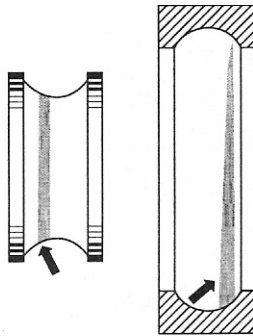


FIGURE 12.4 *Normal contact pattern on the races of a bearing subject to axial and radial loads.*

12.4 Antifriction Bearing Failure Analysis

As usual, antifriction bearing failure analysis begins with data collection. Many times a bearing is subject to off-design conditions, which have great influence on its useful life. The following information should be obtained:

1. Working speed and load.
2. Operation temperature of the oil and metal.
3. Operation, maintenance, vibration, and failures history.
4. Shaft and bearing alignment.

5. Oil characteristics and contaminants.
6. Potential sources of debris and contaminants.

Carefully inspect the failed bearing, starting with the contact pattern. Abnormal indications may indicate specific defects, as shown in the next sections. Carefully examine any debris or contamination in the bearing or bearing box.

Assuming that an antifriction bearing failed due to surface fatigue, the following stages can be observed:

1. The contact between the rolling element and race creates a certain subsurface stress distribution, described by the Hertz theory. The maximum stress can be found a certain distance below the surface, as illustrated in Figure 12.5. If the contact stress is high enough to cause a fatigue fracture, the initiation site will be below the surface, at the point where the stress is maximum. The fatigue fracture begins under the surface and can go unnoticed before it reaches the surface.
2. Continuation of bearing operation may cause the initial fracture to grow. The growing crack eventually may reach the surface of the bearing race. This is the first opportunity for an external observation of this crack. At this time, machine vibration measured from the outside may start to display the bearing's natural

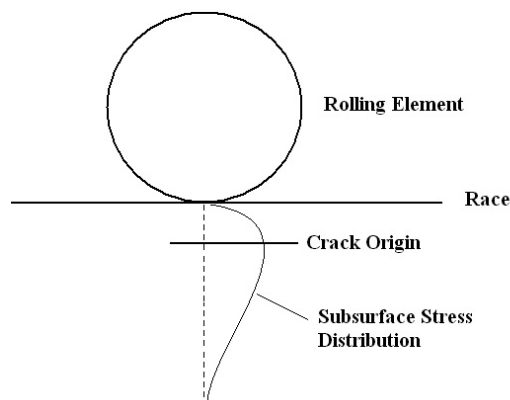


FIGURE 12.5 *The stress distribution caused by the contact between the rolling element and the race of an antifriction bearing.*

- frequencies, as if the contact between the rolling element and the surface crack caused the bearing to “ring.”
3. If the bearing continues to run, the initial crack may develop and a slice of the surface material may be removed. Figure 12.6 shows a bearing that had some surface material removed due to surface fatigue caused by the rolling contact. At this time, external vibration measurements show the characteristic bearing defect frequencies. This is a good opportunity to replace the bearings, as further operation greatly increases the risk of a catastrophic failure.
 4. Figure 12.7 shows a bearing whose surface is heavily damaged by fatigue. At this point, almost all the race periphery has been affected by the surface fatigue. Vibration measurements no longer show the characteristic bearing failure frequencies, only random noise. At this time, the machine should be immediately shut down, as the probability of a catastrophic failure is high.

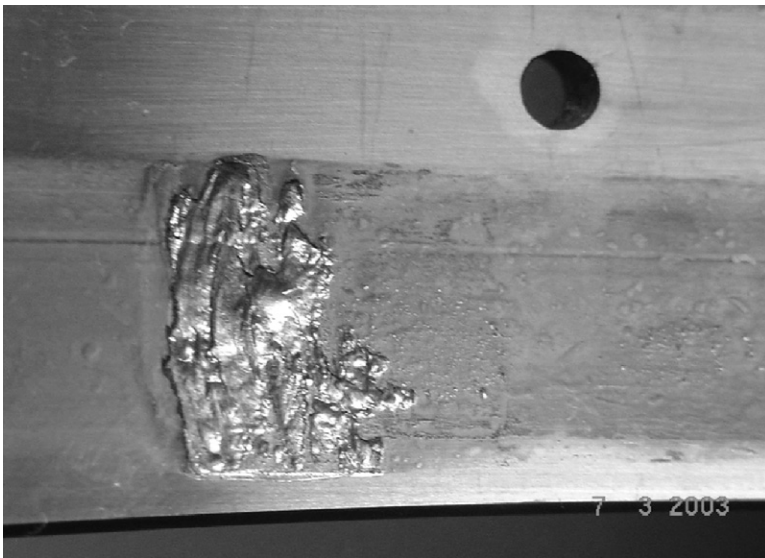


FIGURE 12.6 *Bearing surface scaling caused by fatigue.*

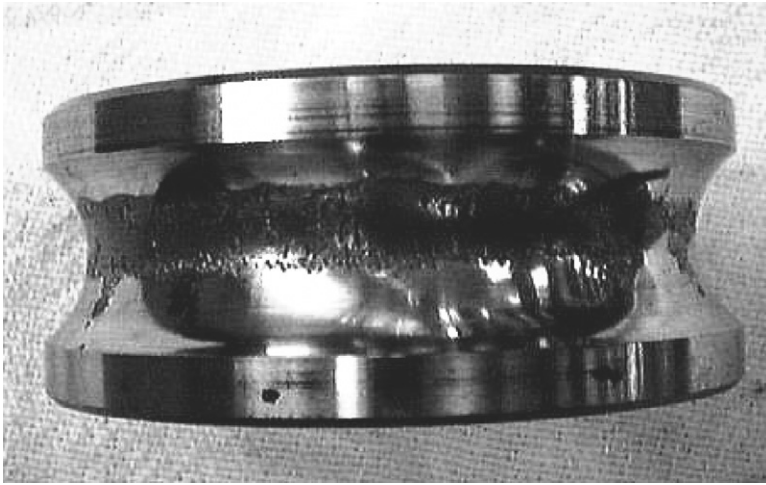


FIGURE 12.7 *Bearing inner race with heavy signs of surface fatigue.*

12.5 Types of Failure

Most bearing failures are not associated with the end of its useful life but to the following causes, each discussed in the following sections:

1. Defective seating of the shaft or bearing box.
2. Misalignment.
3. Incorrect assembly or handling.
4. Incorrect fit.
5. Inadequate lubrication.
6. Ineffective sealing.
7. Vibration when the machine is not running.
8. Design or manufacturing deficiencies.

12.5.1 Defective Seating of the Shaft or Bearing Box

Despite being made of high strength steel, bearing rings are highly flexible. Even small deflections can cause the rolling elements not to run smoothly and load distribution not to be uniform. The lack of uniform load distribution causes accelerated failure of the bearing.

Figure 12.8 illustrates the contact pattern of a bearing installed in an oval bearing box. This is a common example of an antifriction bearing assembly defect. When seated in a bearing box that was not perfectly round, race deformation caused the rolling elements to touch the external race on two sides, causing great local overload. Some common causes of this type of defect are machining errors of the bearing box and wear of axially split bearing boxes.

If a bearing is installed in a shaft whose diameter is too big, excessive interference results. This excessive interference causes the inner ring to expand and may eliminate the internal clearance of the bearing. The result is that the bearing operates overloaded. The contact pattern may show contact between the rolling elements and all the periphery of the outer race. This effect is shown schematically in Figure 12.9.

The installation of a bearing in a conical shaft results in irregular load distribution within the bearing. Figure 12.10 shows a bearing that has not been perfectly seated on the shaft. In this case, a two-row bearing was installed in such a way that the entire load is carried by one row of the rolling elements. The resulting overload caused the observed damage to the bearing race.

Even if the bearing seats are perfect, foreign bodies may unseat the bearing. As mentioned before, the bearing races are quite flexible; any irregularity on the seat causes deformation of the race and local overload.

12.5.2 Misalignment

Antifriction bearings are quite sensitive to misalignment. The combination of small internal clearances and high stiffness of the bearing components make them especially prone to local overload in case of misalignment. Table 12.2 lists the permissible angular misalignment for some types of bearings. Note that a pair of angular contact and two-row ball bearings are the most sensitive.

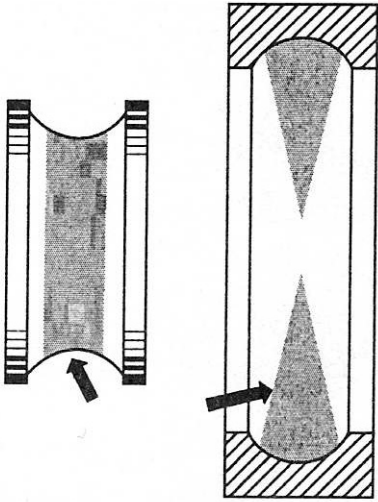


FIGURE 12.8 *Contact pattern on the races of a bearing that operated inside an oval box.*

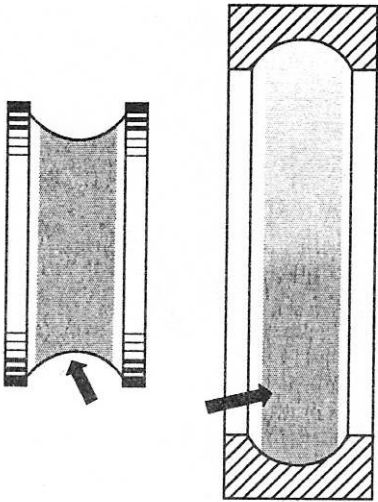


FIGURE 12.9 *Contact pattern of a bearing installed in a shaft with too large a diameter.*

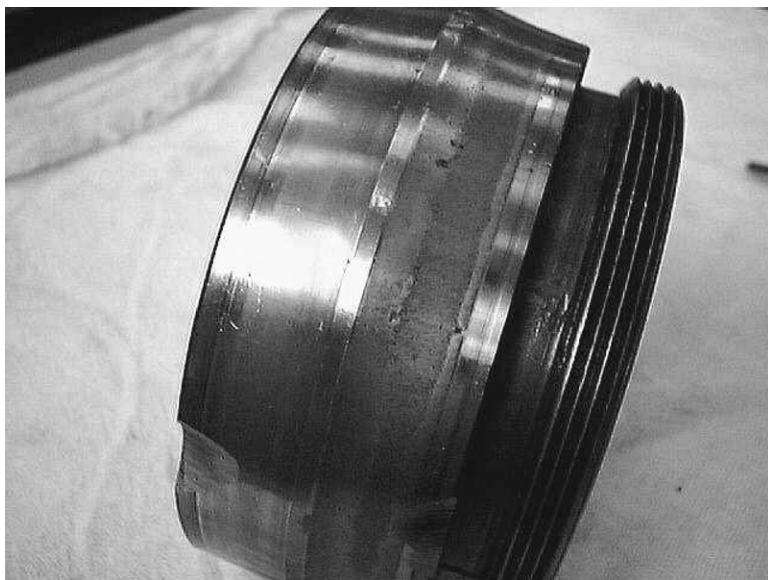


FIGURE 12.10 Bearing seated irregularly on the shaft, showing the damage that resulted from the irregular load distribution.

TABLE 12.2 Misalignment Tolerances of Some Antifriction Bearings

Type	Tolerance
Single row	2–10'
Double row	<2'
Angular contact (pair)	~0
Roller bearings	3–4'

Shaft and bearing misalignment are common causes of bearing failures. Bearings may be misaligned when the shaft, bearing boxes, and bearings are not on the same centerline. The contact patterns observed in bearings that worked while misaligned are characteristic. Figure 12.11 illustrates some possible ways to misalign bearings, all of them able to overload the component. An eccentric bearing box forces the shaft to deflect; a bearing box seat that is not perpendicular to the shaft forces the rolling elements against the races, just

like a bent shaft. Figures 12.12 and 12.13 illustrate typical contact patterns generated by misalignment.

Figure 12.14 shows a double row bearing with sign of distress caused by misalignment. We see that one of the races has been heavily damaged by the local overload caused by the misalignment and the other one is perfect.

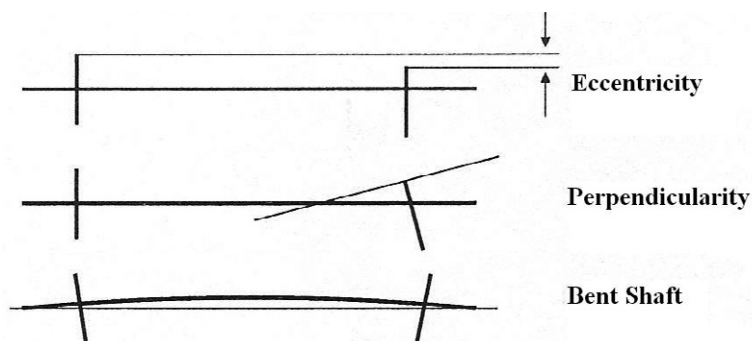


FIGURE 12.11 Some possible ways to have bearings misaligned.

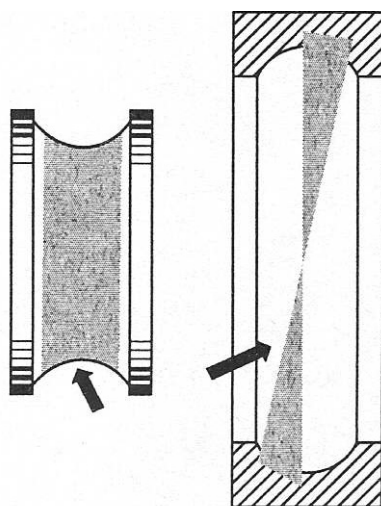


FIGURE 12.12 Contact pattern produced in the bearing races when the outer race is not perpendicular to the shaft.

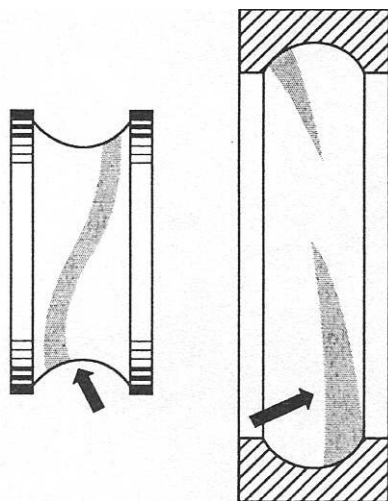


FIGURE 12.13 *Contact pattern produced in the bearing races when the race is not perpendicular to the shaft.*



FIGURE 12.14 *Double row bearing with heavy damage due to misalignment.*

Incorrect Assembly and Handling Deficiencies

Bearing assembly manuals contain a lot of information about how to install a bearing correctly. The main problems found in the real world are lack of cleanliness in the components, tools, and working environment and the use of incorrect tools to install the bearing.

It has already been mentioned that even a small amount of impurity may greatly reduce bearing life. Foreign particles may be pressed against the races and cause indentations that are the origin of further damage.

The use of undue force during the installation of a bearing may damage the bearing races. Although the races are made of high strength steel hardened to more than 60 RC, it is possible to have the rolling elements plastically deform the races, much in the same way as in a Brinell hardness test. That is the origin of the term *brinelling*. Figure 12.15 shows the inner race of a double row bearing that has been damaged by the application of undue force during assembly. The original damage is the indentations made by the balls on the inner race surface. The picture shows the secondary damage caused by the rotation of the bearing under load after the original

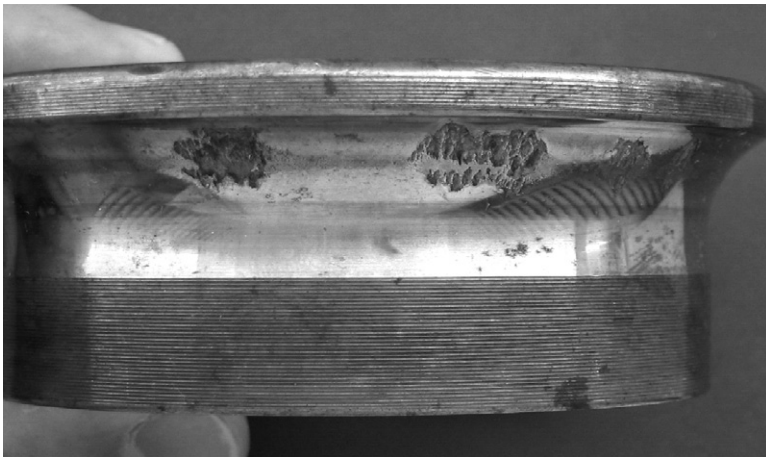


FIGURE 12.15 Angular contact ball bearing with signs of damage caused by the application of excessive force during assembly.

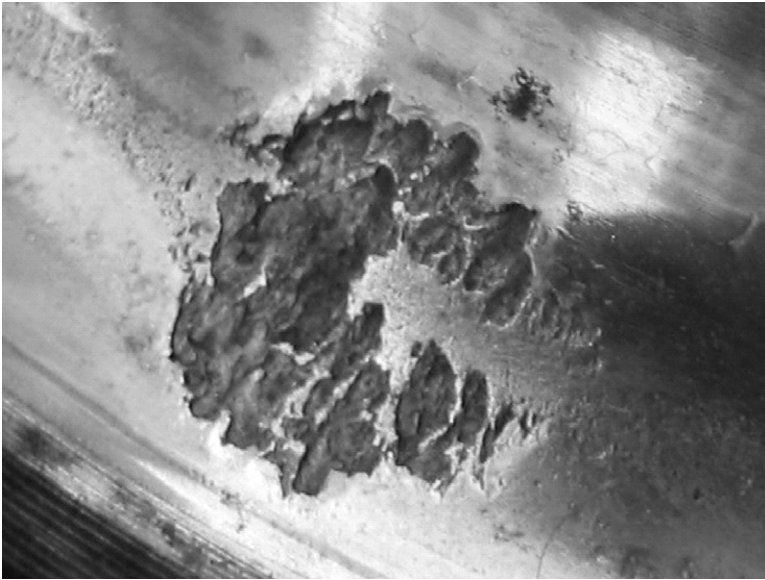


FIGURE 12.16 *Enlarged view of the damage shown on the bearing in Figure 12.15.*

indentations were made. Note that the distance between each damaged area is the same as the distance between the balls, a characteristic feature of this type of defect. Figure 12.16 shows an enlarged image of one of the craters found in Figure 12.15.

Another example of an installation mistake is an error in the relative position of the bearings in the machine, which causes overload and rapid failure of the bearings. This type of situation can be found in machines whose bearings have been designed to absorb thermal growth, for example. If the bearings are assembled in such a way as to lose their freedom to move, damage may result because of the thermal growth loads. The internal clearances of the bearings are very small, being enough to absorb only very small outside movement.

Figure 12.17 shows damage caused by the incorrect setting of the relative position of the bearings on the shaft. The bearings shown were installed in an air-cooled heat exchanger fan. The

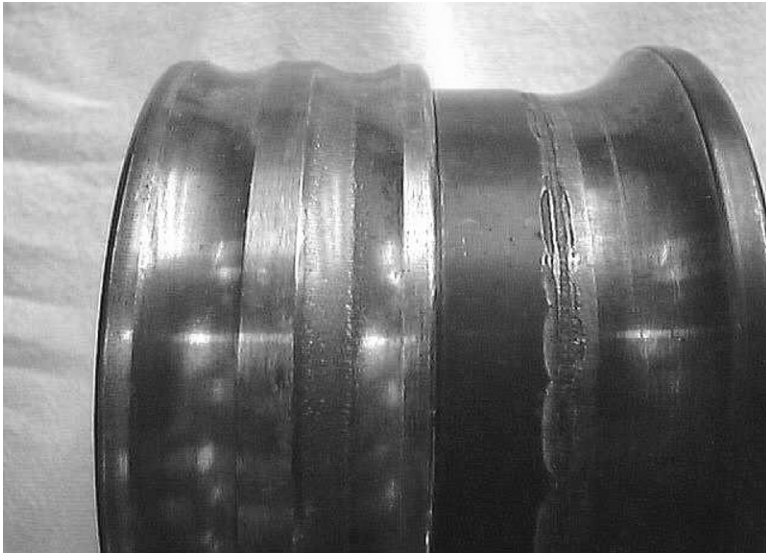


FIGURE 12.17 *Bearings installed incorrectly in a fan. The contact patterns show clearly that they have shared the thrust load in the wrong way.*

angular contact bearing was supposed to withstand the vertical axial load, but defective assembly caused the double row bearing to absorb the axial thrust. Both bearings have been damaged. The contact patterns of both bearings indicate that they worked improperly: The radial bearing shows indications of the large axial load, and the angular contact bearing shows indications that the ball ran out of its race.

A bearing can be handled incorrectly in many ways. Figure 12.18 shows an example of improper handling of a centrifugal pump that resulted in damage to the bearings. When suspended by the shaft, the whole weight of the pump is supported by the bearings. When the pump operates, the indentations formed on the bearing races are initiation spots for surface failure, the damage caused is similar to that shown in Figure 12.15. Other examples of improper handling of bearings are inadequate storage, transportation without care, and removing the bearings from the original package long before installation.

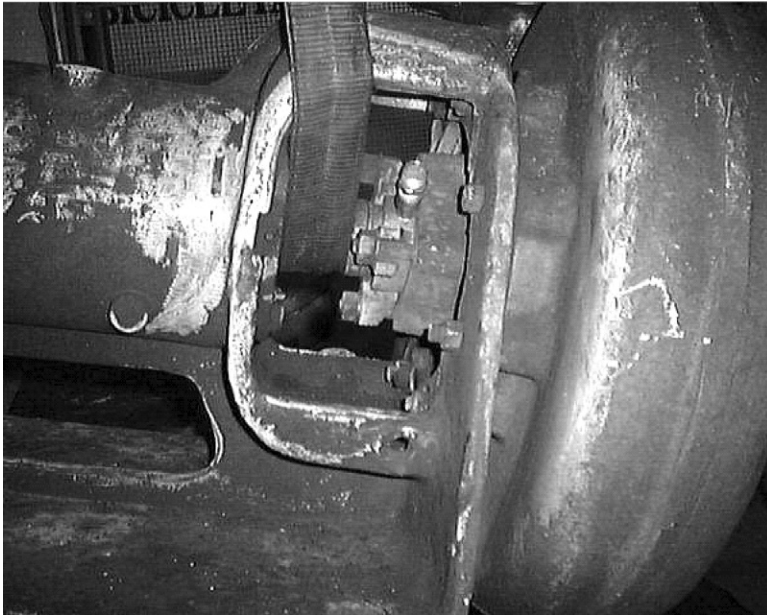


FIGURE 12.18 *Incorrect handling of a pump can damage the bearings.*

12.5.3 Incorrect Fit

As a rule, the bearing race that rotates relative to the load should have an interference fit; the other one may have a loose fit. The higher the load, the higher the interference. The clearances or interference fits adequate for each use can be found in the material published by the bearing manufacturers. The typical situation, in the case of rotating machines, is to have the inner race mounted with an interference fit on the shaft and the outer race installed with a loose fit.

Inner race fit problems can be due to a loose fit, which may allow the inner race to rotate on the shaft and be damaged, or a too tight fit, which may eliminate the internal clearances and overload the bearing. Outer race fit problems can be similar. A too tight fit can have the same effect as indicated previously, a too loose fit may cause high vibration and fretting on the bearing or bearing box, as shown in Figure 12.19. It is not very likely that outer racing bearing



FIGURE 12.19 *Bearing outer race with signs of fretting due to a loose fit.*

fretting can result in a fatigue crack, as the predominant loads result in compressive stresses.

12.5.4 Lubrication Failures

An antifriction bearing has several rubbing surfaces: balls against races, balls against cages, rollers against race flanges. The lubricant provides an oil film between running surfaces and removes friction-generated heat. Sometimes, the lubricant also must remove debris from the bearing and avoid the entrance of contaminants to the races.

A lubrication failure may mean a failure in any one of these functions. The appearance of a bearing that failed due to a lubrication problem is characteristic and easy to recognize. The reason why the lubrication failure occurred may not be so obvious.

If a bearing is operating with a lack of oil, there is more metal to metal contact than desired. The first indication of this improper lubrication may be a slight polishing of the surface or increased temperature. Another early sign of improper lubrication is cage wear. More severe damage shows a rough surface and severe overheating. Figure 12.20 shows a bearing cage with signs of wear. Cage wear can happen because of inadequate lubrication, even when no other damage is obvious, as the cage is not as hard as the other parts.

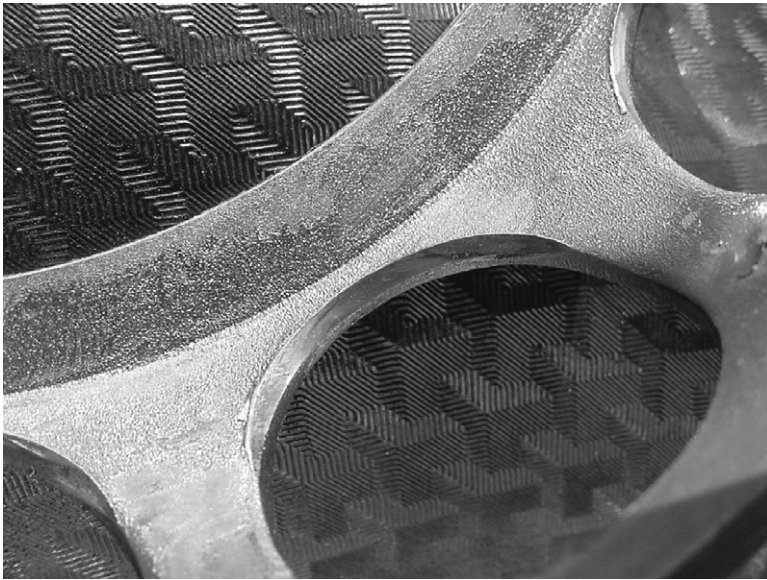


FIGURE 12.20 *Cage of a ball bearing with signs of wear.*

A slightly more severe lack of lubrication causes surface polishing due to metal to metal contact. High bearing temperature may also result. Elevated temperatures may cause plating of the oil additives, metal discoloration, or even melting of some components. Figure 12.21 shows an example of additive plating due to high temperature.

Figure 12.22 shows bearing rollers subject to inadequate lubrication, which resulted in high temperature and surface polishing. The rollers turned blue.

The next lubrication problem can be regarded as somewhat extreme. In this case, the thrust bearing of a centrifugal pump suffered a catastrophic failure due to lack of lubrication. The bearing temperature was high enough to melt the brass cage. Figure 12.23 shows the bearing as found on pump disassembly. Centrifugal process pumps often are found with a pair of angular contact bearings to serve as thrust bearings. In this case, this common arrangement was



FIGURE 12.21 *Axial bearing race with signs of oil additive plating due to high temperature.*



FIGURE 12.22 *Rollers subject to poor lubrication, with the consequent polishing of the surface and oxidation.*



FIGURE 12.23 *Thrust bearing of a centrifugal pump that suffered a catastrophic failure due to lack of lubrication.*

lubricated with the help of an oil flinger and an oil channel that made it possible for the oil to reach both bearings. The oil channel has passages that go through the bearing box cover, and there is a gasket between the bearing box and the cover. Two holes in this gasket are needed to allow the flow of oil. Figure 12.24 shows the position of the gasket as found during disassembly. The gasket was installed in the wrong position during the assembly of the pump, covering the oil passages. A better design of the gasket would not require a precise positioning, allowing gasket installation in any position.

There are many other possibilities for lubrication failure. Another example is a problem well known to the maintenance professional: Mixing different types of greases can cause unwanted reactions among the grease components. The same is true for lube oils. Figure 12.25 shows an electric motor bearing in which a high-temperature grease was applied erroneously. When the high-temperature grease mixed with the regular-grade grease remaining in the bearing, they turned into a hard compound that made it impossible for the motor to run.

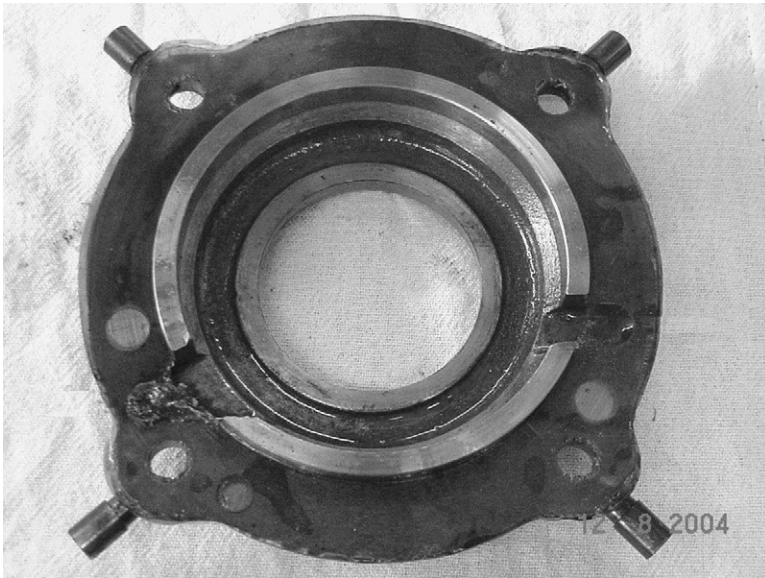


FIGURE 12.24 *Gasket of the bearing box cover of the bearing shown in Figure 12.23. Note that the wrong assembly position made it impossible for the oil to flow to the bearing.*

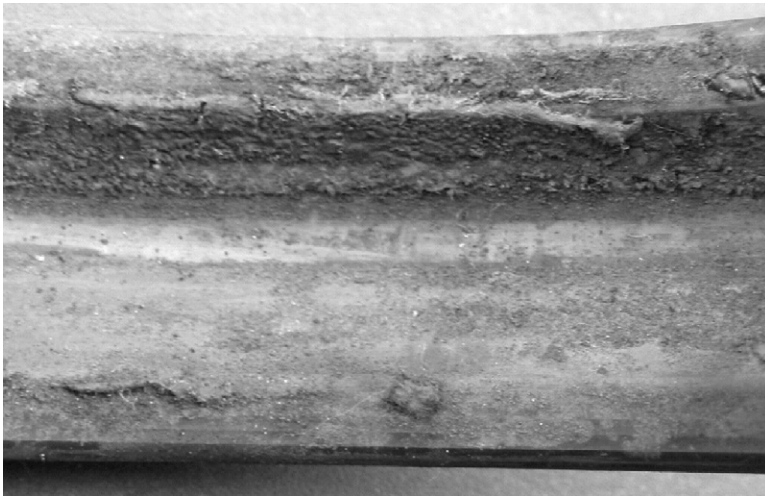


FIGURE 12.25 *Outer race of an electric motor bearing with residue of the product resulting from mixing two incompatible types of grease.*

12.5.5 Ineffective Sealing

Although foreign particles may be introduced to a bearing or bearing housing during installation, the most direct and continuous source of external contaminants is an ineffective seal. This is also the most common design error in machines containing antifriction bearings.

As mentioned previously, the oil film formed between the rolling elements and the races is very thin. Being so thin, even tiny particles can be large enough to be greater than the oil film. When such a particle is pressed against the rolling elements and race, a huge stress concentration forms at the contact point, which may cause an indentation on the bearing. This indentation is an initiation spot for further damage.

Different indentations are caused by different types of particles. Hard, relatively large particles, like coarse sand, cause individual indentations of an appreciable size. Very small particles cause uniform wear of the races and rolling elements. Figure 12.26 shows the

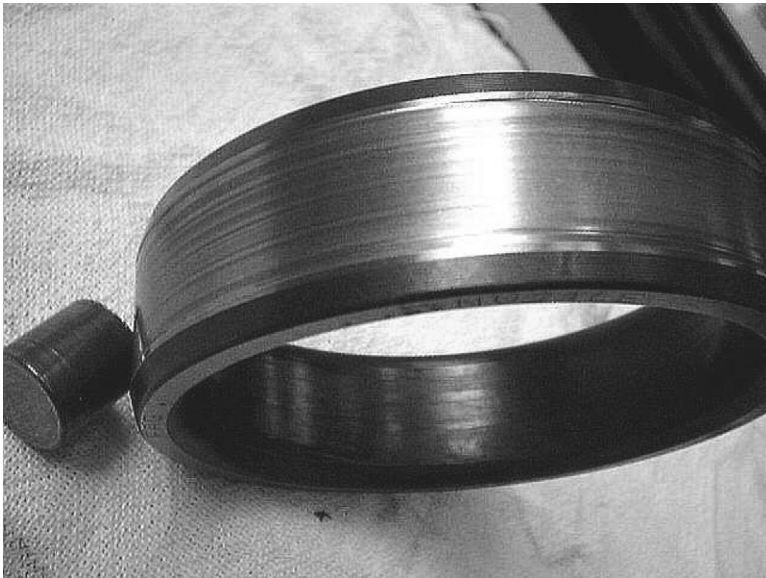


FIGURE 12.26 *A bearing that suffered abrasion due to oil contamination with a very fine alumina-based catalyst.*

inner race of a bearing that suffered abrasion due to the contamination of the oil with an alumina-based catalyst of submicron granulometry. The surface appears polished. The bearing was installed in a centrifugal pump and, despite a greatly increased radial clearance, was operating with reasonable levels of vibration.

If a small particle can cause a uniform surface abrasion, larger particles may act as road bumps, causing the rolling elements to bump and hit the bearing surface. In addition to the stress concentration provided by the particle itself, the impact of the rolling element on the race is a source of concentrated damage.

An ineffective seal may allow water to enter the bearing housing. Small quantities of water may severely reduce the bearing life, as mentioned before. Larger quantities of water in the oil can cause bearing corrosion, also extremely deleterious to bearing life. Figure 12.27 shows a high-speed bearing with signs of corrosion caused by the entrance of water into the bearing box.

Many different types of shaft seals can be used to avoid contamination of the bearing environment. Some use labyrinths, others are based on mechanical seals. This effective and cheap upgrade can be done to many types of process machines.



FIGURE 12.27 *Corrosion due to contamination of the oil with water, caused by ineffective sealing of the machine.*

12.5.6 *Vibration When the Machine Is Not Running*

In a standing bearing, the rolling elements touch the races with no oil film in between them, as the oil film formation depends on the bearing speed. This situation causes the surface irregularities to touch each other with a contact stress approximately equal to the yield stress of the material. If external vibration is superimposed, local stress may be greatly increased and fretting may occur due to the microscopic relative movement of the parts. This fretting wear caused by the external vibration when the bearing is not running is also called *false brinelling*. When the machine starts to run, the race defects act as starting points for subsequent failure.

The term *false brinelling* was coined because of the similarity of the damage that results from the indentation on the races caused by the balls in case of overload when the bearing is not running. Figure 12.28 shows an example of a bearing damaged due to false brinelling. In this case, the bearing of a high-speed vertical pump was being “shaken” by the high vibration of a nearby pump. As usual, the distance between cavities is the same as the distance between the balls.

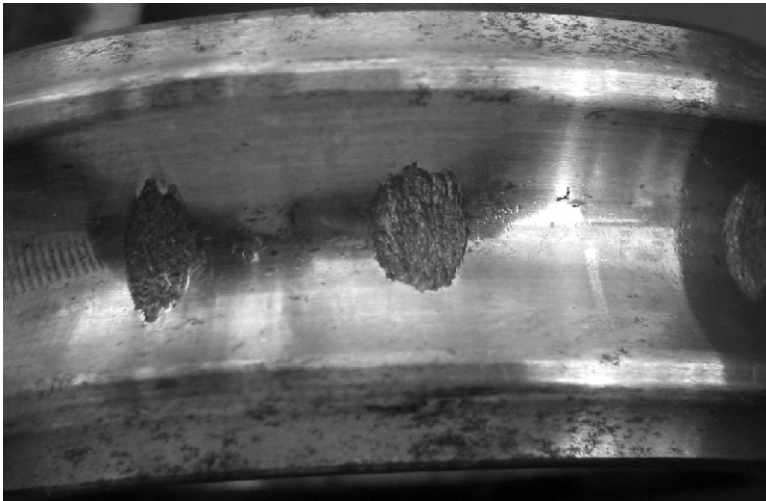


FIGURE 12.28 *Bearing damage due to false brinelling.*

When analyzing a bearing that failed due to fretting, the most difficult part is to differentiate between fretting and brinelling. Keep in mind the damage mechanism: Brinelling is caused by overload indentations, no material removal is observed. Fretting is a wear process, and material removal does occur. If these two defects can be observed before the bearing has a chance to operate and increase the damage, we could notice that the original grinding marks can be seen on the bottom of the brinelling indentation, as no material has been removed. On the other hand, the original grinding marks are not seen inside the fretting cavity, due to the material removal.

If the bearings have a chance to run, further damage makes it impossible to differentiate between them by inspecting the grinding marks. The operating history of the machine provides the missing evidence to help solve the problem. Bearing failures due to brinelling usually occur soon after a maintenance intervention, and fretting needs an external source of high vibration acting for a period of time when the machine is not running.

12.5.7 Design and Manufacturing Deficiencies

Modern process machinery is normally the result of a long development process. This process has made design errors something found infrequently. Nevertheless, errors happen from time to time, some examples are discussed here:

1. Undersizing is the situation in which the loads that act on the bearing during normal operation result in a useful life shorter than required. This type of error has been found, for example, in the case of an electric motor that drives and supports a vertical pump. If the pump axial load is not considered when sizing the motor thrust bearing, its life can be much shorter than desired. Another example is the situation when motor bearings have not been designed for a belt drive, which exerts additional radial loads on the bearings.
2. Oversizing usually is thought beneficial to bearing life. Although this can be true in some instances, too much oversizing may result in a bearing that is not loaded enough to provide friction between rolling elements and races to avoid ball or roller

skidding. Ball skidding damages the bearing surface. Figure 12.29 shows skidding indications on the race of a pump bearing that was greatly oversized.

3. Inadequate arrangement of the bearings may result in inadequate load distribution. Thermal growth of shafts, for example, may overload the bearings of a machine, if none of them has some freedom to move. Figure 12.30 shows an example of poor design that caused thermal growth to damage the bearings. In this case, the bearings of the hot gas damper were assembled when all the equipment was at room temperature. When the hot gas began to flow, the damper vanes were hotter than the walls of the duct, due to the internal refractory lining. The difference in thermal growth caused severe damage to the bearings.
4. The selection of the wrong type of bearing may also lead to a useful life shorter than required. A common example of this design error is the use of polyamide cage bearings in heavy-duty machinery. This type of bearing should be restricted to light-duty equipment. Figure 12.31 shows an example of a polyamide cage bearing installed in a heavy-duty gearbox driving a cooling tower fan.

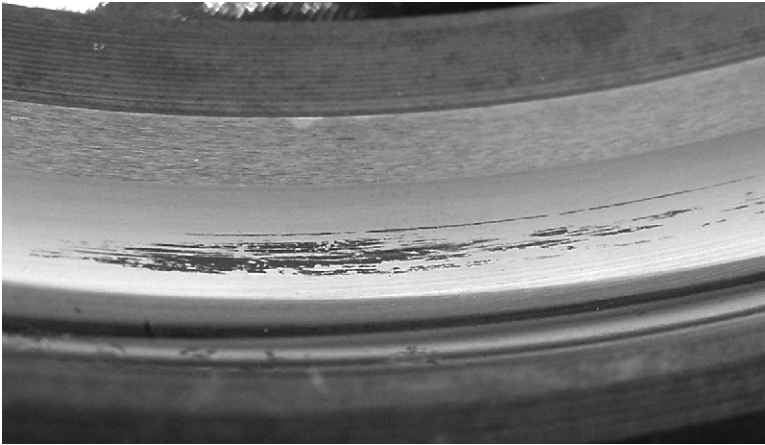


FIGURE 12.29 *Outer race of an angular contact bearing that suffered ball skidding due to oversizing.*

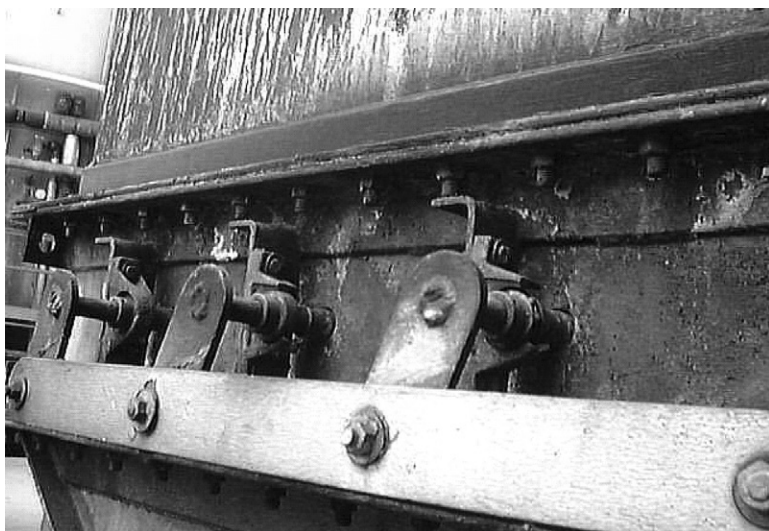


FIGURE 12.30 *Damper vanes bearings damaged by differential thermal growth.*

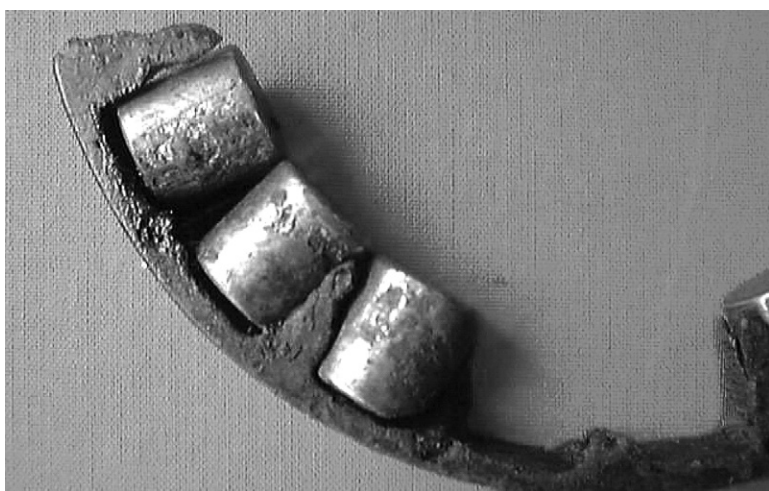


FIGURE 12.31 *Polyamide cage damage. This type of cage should not be used in heavy-duty machinery.*

5. An ineffective seal, the most commonly found design error, has already been mentioned.

Bearing manufacturing is a precision industry, at least for the most reputable vendors. Production and quality control usually are very strict, which results in a small number of manufacturing defects reaching the final user of the bearing. In many cases, a manufacturing deficiency is blamed just because no other root cause has been found. A manufacturing error should be suspected only if the evidence suggests it. Some examples of evidence of fabrication problems found in antifriction bearings are

1. Oval races, which can be confirmed by careful measurement.
2. Material problems, which can be confirmed by a metallurgical examination. Figure 12.32 shows an example of a quite uncommon bearing manufacturing problem. This problem probably was caused by the use of the extremes of the ingot to forge the ball. Slag and dirt may accumulate at the extremities of the ingot during solidification of the casting. If the ingot is not properly cut before forging, these impurities may find their way into the finished product. A material problem has been suspected in this case because the bearing suffered a catastrophic failure and some of the balls seemed to be hollow. Cutting some of the remaining balls proved that they actually were hollow.

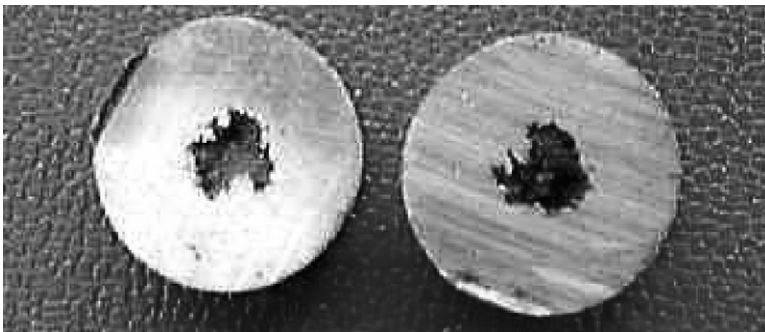


FIGURE 12.32 *An uncommon example of manufacturing defect, a hollow bearing ball.*

13

Mechanical Seals

Mechanical seals are intended to avoid uncontrolled leakage of fluid through the region where the shaft enters the machine casing. Many process industries handle dangerous fluids; a highly reliable shaft sealing mechanism is of utmost importance.

Mechanical seal failures represent a large fraction of the total machinery failures in the process industry. Depending on the situation of the plant, from close to 20% up to more than 60% of the machine failures can be attributed to mechanical seals. This high percentage of the total failures makes mechanical seal failure analysis very important.

According to the definition presented earlier, the failure of a component is characterized by the inability to perform its function safely. A mechanical seal is considered to fail when the leakage is greater than a specified value. The great variety of operating conditions, seal designs, and local regulations makes it impossible to form a general rule regarding permissible seal leakage.

This chapter focuses on the failure analysis of the types of mechanical seals commonly found in the process industry, with

springs or metal bellows, carbon, silicon carbide or tungsten carbide faces, and elastomer O-rings.

13.1 How a Mechanical Seal Works

Mechanical seal operation is the result of a complex integration between solid friction due to asperity contact and hydrodynamic forces generated by the fluid in the interface due to the relative movement of the faces. The combination of these two effects generates the forces that keep both faces slightly apart when in operation.

Liquid film formation is similar to a hydrodynamic thrust bearing, except that the axial load usually is much smaller, the fluid often is a poor lubricant, temperatures can be very high, and the fluid film cannot always separate the faces completely and often are partially vaporized, as in the case of liquified petroleum gas (LPG) or hot water seals.

There always is some leakage past the seal when it is operating. This leakage is very small, invisible to the naked eye. Figure 13.1 illustrates the interaction between the surface asperities and normal leakage. This does not mean that it is impossible to avoid all leakage of the pumped fluid. Some of the fluid that contacts the seal will leak, but it is possible to avoid the leakage of even minute quantities of the pumped fluid if double seals are used.

When a mechanical seal begins to rotate under pressure, the sealing faces are subject to the forces of the springs, the static pressure forces of the sealed fluid, and the hydrodynamic forces generated by the fluid at the interface. The factor most influential on seal life is the fraction of total face load supported by hydrodynamic fluid pressure. This is the well known lubricated friction problem. As usual, if the film thickness is large enough to completely avoid solid contact, wear is very low. The hydrodynamic forces are generated by the interaction of the fluid with the moving irregularities of the rotating face. Unlike a hydrodynamic bearing, the mechanical seal normally is not designed to provide a thick fluid film, as this would lead to high leakage. This is the mechanical seal dilemma:

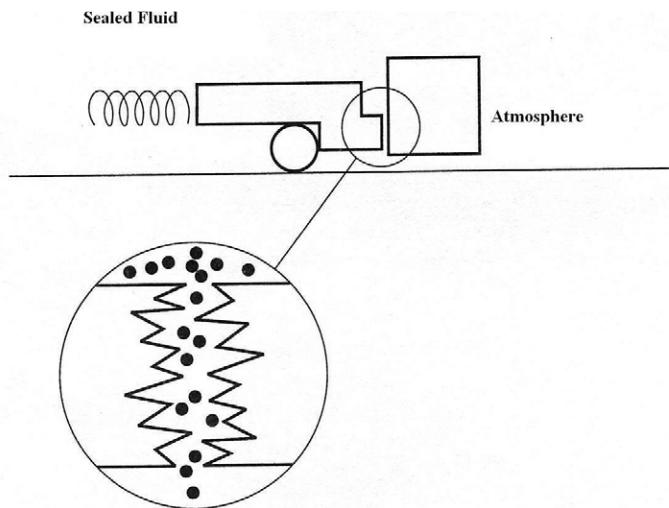


FIGURE 13.1 *The working mechanism of a mechanical seal.*

The lubrication conditions that result in longer life also result in more leakage. These, clearly, are conflicting requirements.

The necessity of little leakage means that the film thickness should be very thin. To obtain good lubrication conditions with a thin film, surface asperity height should be very small, too. Seal surfaces usually are lapped to a mirrorlike finish to reduce asperity contact.

Even when correctly lapped, the seal faces have some asperities. The normal operating condition of mechanical seals causes some solid contact. This solid contact causes the removal of the highest spots of the softer surface.

Special face configurations sometimes are used to improve lubrication, when poor lubricity fluids have to be sealed. Figure 13.2 shows a mechanical face seal with grooves designed to improve the hydrodynamic effects. This can be useful in poor lubricity fluids, like hot water, LPG, and even gases. Note that a portion of the surface, close to the inside diameter, is flat. This is done to limit the seal leakage.



FIGURE 13.2 *Mechanical seal face with grooves to improve the hydrodynamic effect.*

13.2 Seal Mechanical Design

Although a complete analysis of the mechanical design of a seal is outside the scope of this text, a simplified discussion of some important design features can be quite revealing. A summary of the main advantages and disadvantages of some typical construction details is presented.

13.2.1 Rotating versus Stationary Flexible Element

A rotating flexible element can be less sensitive to incrustation, as the centrifugal force provided by the rotation forces heavier particles to the outside. In addition, they are usually shorter, simpler to assemble, and cheaper. On the other hand, if the shaft rotates at a high speed, above 3,600 rpm, it can be easier to keep a stationary flexible element seal balanced and centered. Centrifugal pump mechanical seals normally have a rotating flexible element. Cen-

trifugal compressor mechanical seals usually are stationary, as they tend to operate at high speed.

13.2.2 Internal versus External Pressurization

Almost all the seals used in the process industry are externally pressurized. The simple reason for this characteristic is that thermal deformation of externally pressurized seals cause the faces to converge to the inside, providing an extra entrance to the interface for fluid. This makes cooling more effective. In addition, the stresses developed in the face rings of externally pressurized seals are compressive. The materials used for seal faces are brittle and much stronger when compressed. Remember that brittle materials develop cracks in a direction perpendicular to the main tension stress. On the other hand, large diameter seals may be subject to elastic instability when pressurized from the outside.

13.2.3 Width of Hard and Soft Faces

Most mechanical seals have a larger width on the hard face. Soft face wear is greater than hard face wear; if the soft face is wider than the hard face, a deep groove may be worn. This groove causes the hard face to lock radially, and any relative radial movement causes great leakage.

13.2.4 Design of the Secondary Sealing

The most commonly used secondary sealing elements are O-rings and metal bellows. O-rings are easily available in many sizes and materials, usually are cheap, provide torsional damping, allow for greater axial travel, and can be subject to a higher pressure, when compared to metal bellows. The main disadvantage of O-rings is the limited allowable temperature range. High temperature, typically, is the reason why metal bellows seals are specified. The introduction of perfluorelastomer compounds a few years ago greatly extended the high temperature capabilities of O-rings. Elastomer O-rings may cause shaft sleeve fretting, swell from exposure to chemicals, and be a source of high friction between the seal and the sleeve, making face adjustment more difficult.

Most mechanical seals used in the process industry use O-rings as secondary sealing elements. Metal bellows seals normally are used only for temperatures above 200 or 250°C.

13.2.5 Single versus Multiple Springs

Multiple spring seals usually are more expensive and more difficult to assemble, because of the greater number of components. The larger spring of a single spring seal can be more resistant to corrosion, due to the larger dimensions of the spring, although the Hastelloy springs used in modern mechanical seals are quite resistant to almost any fluid found in the process industry. Multiple springs also result in a more uniform load distribution on the faces. Most seals used in the process industries use multiple springs.

Some seal designs have springs protected from the contact with the fluid. This can greatly increase seal life because of the reduced probability of a seal hangup due to incrustation of the springs, especially when the sealed fluid contains solids in suspension or can cause incrustation.

13.2.6 Mechanical Seal Standardization

Standardization of machinery components has some well known advantages, like reduced inventory, greater familiarity of the crew with the standard parts, and availability. Old plants also benefit from a standardization program in other ways, as the number of different installed seal designs is usually large, not to mention that most of them may be outdated.

Some useful characteristics should be introduced in a standard seal:

1. Seal design should be current and adequate for most plant services. A standard seal should use high-quality carbon against silicon carbide faces, Hastelloy springs, and the like. Refineries and petrochemical plants should consider using API Standard 682, which has a complete description of some standardized seals and sealing systems.
2. Dimensions should be standard regardless of the seal manufacturer, to make a replacement easier, if so desired.

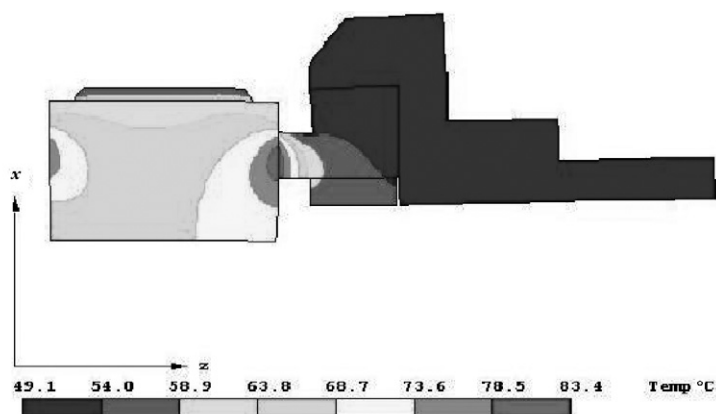


FIGURE 13.3 *The calculation of the temperature distribution of a mechanical seal.*

13.3 Seal Hydrodynamic Design

As with hydrodynamic bearings, the hydrodynamic design of mechanical seals is an involved subject, usually done by the seal manufacturers with the help of specialized computer programs. These computer programs calculate the hydrodynamic support provided by the fluid film, considering the actual shape of the faces, as resulting from pressure and thermal distortion. An in-depth discussion of this subject is beyond the scope of this text. Figure 13.3 shows the output of one of these programs, in this case, illustrating the temperature distribution resulting from a specific operation condition.

13.4 $P \times V$

A simplified evaluation of the severity of a sealing service can be done with the help of a simple index number. The product of the fluid (or face) pressure and peripheral speed is directly related to the heat generated at the interface. The idea behind this simplification is that pressure is related to face load, which is related to the friction forces. So, this product is related to the product of friction force and speed, the dissipation of energy by a moving body subject

to external friction forces. Note that this is a simplification; a more detailed analysis may be required for difficult sealing services.

There are two PV products: total PV , termed PV_t , and net PV , termed PV_n :

$$PV_t = P \times V$$

$$PV_n = [P \times (B - K) + P_{\text{spring}}] \times V$$

where

P = sealing box pressure (kgf/cm²);

V = speed of the centerline of the seal face (m/s);

B = seal balancing;

K = pressure distribution factor, the ratio between the average pressure on the interface and the fluid pressure (usually equal to 0.5 for liquids);

P_{spring} = the face pressure caused by the action of the springs (kgf/cm²).

Total PV is used to evaluate the severity of a sealing service, regardless of seal design. Net PV can be used to evaluate the severity of the operation of the seal to the seal faces and depends on the seal design. Seal manufacturers publish the PV limits of their seals in their catalogs. Commercial seals that use carbon against tungsten or silicon carbide faces have PV_t and PV_n limits of about 1,000 kgf/cm² × m/s and 300 kgf/cm² × m/s, respectively. Ceramic faces are limited to about 60% of these values.

Seal balancing is the ratio between the face pressure and the fluid pressure. Figure 13.4 illustrates the concept.

The PV limit of commercial seals that use a carbon face versus a tungsten carbide or silicon carbide face is close to 1,000 kgf/cm² × m/s for PV_t and 300 kgf/cm² × m/s for PV_n . Carbon faces running against ceramic have PV limits around 650 and 200 kgf/cm² × m/s, respectively.

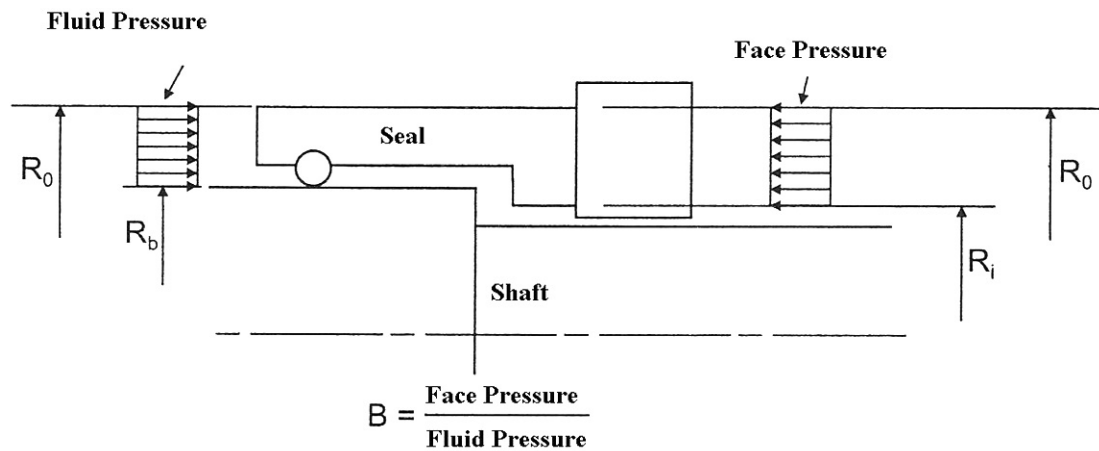


FIGURE 13.4 *Balancing a mechanical seal.*

13.5 Sealing System

A mechanical seal lasts longer when working in an adequate environment, which means that the temperature, viscosity, and purity of the fluids in contact with the seal are within certain limits. The pumped fluid does not always have the correct characteristics, so it is necessary to create a seal support system. The control of the seal working environment through the adequate selection of the sealing system has an enormous influence on the seal's useful life.

Sealing systems can be selected with the help of the literature and API Standard 682, which has a great deal of information about seal and sealing system selection. Pay special attention to pumps that operate with dangerous fluids, whether toxic, explosive, or flammable. If a barrier fluid different than the pumped fluid is used, the following characteristics are needed:

1. Compatibility between the barrier and pumped fluids, which means that mixing the two fluids should not result in undesirable reactions, like formation of deleterious compounds or vaporization.
2. Compatibility with the seal materials, avoiding corrosion of the metallic parts and degradation of the elastomers.
3. Lowest possible volatility and toxicity, so eventual leakages are not dangerous.
4. Viscosity below 500 cSt at the lowest operating temperature and greater than 1 cSt at the highest.
5. Cleanliness of the fluid.

The inexistence of a normative criterion for what constitutes a clean fluid has raised some debate on this topic. Also, no objective information is available about how much crystallization or polymerization of the sealed fluid should be allowed. A reasonable criterion is to consider that fluids containing more than 100 ppm solids are "dirty," and special precautions are needed. This should not be considered a mandatory requirement, as it can be difficult to find

fluids cleaner than this in the process industry, and a great number of seals successfully seal fluids containing more than 100 ppm solids.

An example stresses the importance of the correct choice of sealing system. A boiler feed pump was originally self-sealed. The high temperature of the water, above 80°C, caused vaporization between the faces. A shortened seal life resulted from this. Figure 13.5 shows the usual appearance of the pump; high seal leakage was a normal situation. The steam cloud is the result of poor seal interface conditions. The installation of an improved sealing system, in this case, one with a heat exchanger to cool down the seal area, made it possible for the seal to work without vaporization between the faces. The result is shown in Figure 13.6, and a much longer seal life could be obtained.

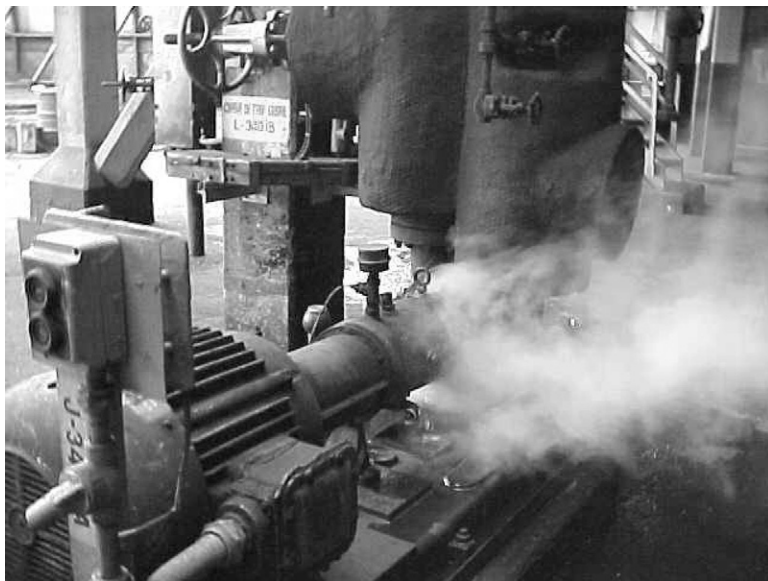


FIGURE 13.5 *Boiler feed pump with seal leakage caused by an inefficient sealing system.*

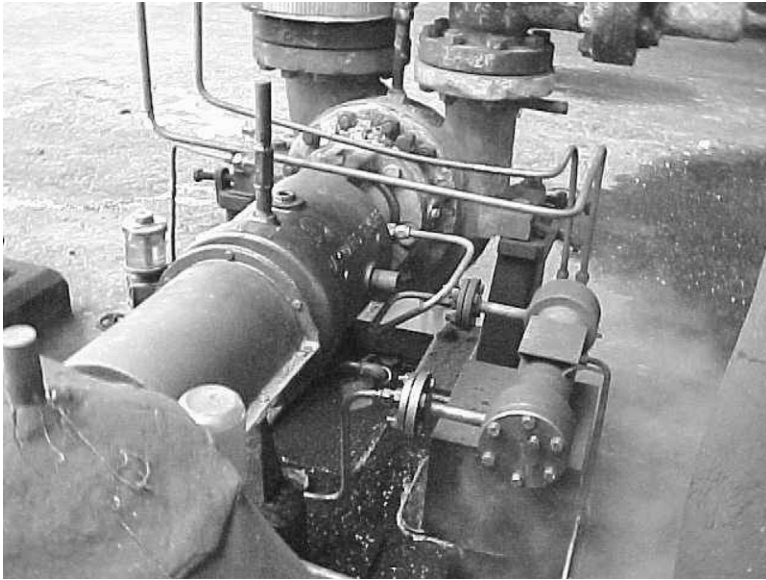


FIGURE 13.6 *The same pump, after the introduction of an improved sealing system.*

13.6 Mechanical Seal Failure Analysis

A mechanical seal in good condition always shows a negligible leakage. Failure occurs when this leakage increases to unacceptable levels. The permissible seal leakage may depend on many circumstances, including toxicity and flammability of the fluid and local regulations. The issue is not discussed here.

The characteristic end-of-life failure mode of mechanical seals is sliding wear of the soft face caused by the contact with the asperities of the hard face. The soft face is designed to withstand a certain amount of wear without leakage, if this wear is uniform enough not to affect face finishing or cause leakage. It is not easy to find out beforehand the expected seal life, due to the great number of variables involved. Any seal failure that happens due to this mechanism should be considered abnormal.

Seal failure analysis should begin with data collection, as suggested for the other component failures discussed. The following information should be added to the list described in Chapter 2:

1. A seal cross section drawing with a list of materials.
2. Operating data, like speed, pressures, and temperatures.
3. A flowchart of the pump and the sealing system.
4. Characteristics of the pumped and sealed fluid, like viscosity, contaminants, and vapor pressure.

13.7 Face Contact Patterns

Face contact patterns can be quite revealing about the mechanical interaction of the seal faces. Seal faces that operated in good conditions display a regular, centered, smooth, and light contact pattern. Figure 13.7 shows such a seal face, in this case, a silicon carbide ring that operated for a long time. Little or no wear can be observed, and many times the contact area becomes smoother than the original face finish. Seals with such wear indications normally have a long life.



FIGURE 13.7 *Silicon carbide seal ring showing a normal contact pattern after a long time in service.*

13.8 Failure Mechanisms and Causes

The most common causes of mechanical seal failures are

1. Inadequate handling of the seal components. The seal faces are precision components made of brittle materials. Scratches and edge chipping should be carefully avoided through proper handling. Cleanliness is also of utmost importance.
2. Incorrect assembly, involving a wrong position of the components or incorrect preload.
3. Inadequate design. The shorter life may be due to fast face wear, heat checking, corrosion, and the like.
4. Operation problems of the equipment where the seal is installed, like sealing system upsets, incorrect startup, or contamination of the sealing fluid.
5. Mechanical problems in the equipment, like high shaft run-out or high vibration.

Seal failure modes form three main groups:

1. Chemical attack, like metal corrosion and elastomer swelling.
2. Mechanical damage, such as face wear, O-ring cuts, and ring fractures.
3. Thermal damage, like heat chocking and elastomer burning.

Each of these failure modes is discussed in the following sections.

13.9 Corrosion of Seal Components

The corrosion mechanism of the seal components depends on the specific component being analyzed. The metal parts may be subject to the corrosion mechanisms described in the Chapter 7. However, unlike metal corrosion, which is nearly always an electrochemical process, carbon rings and elastomers may be subject to specific corrosion processes. Some examples of carbon and elastomer corrosion are discussed here.

13.9.1 Seal Face Corrosion

Mechanical seal faces should be completely inert to the sealed fluid, due to the need for a highly polished finish. The slightest corrosion on seal faces impairs the sealing capacity, as even small material removal can significantly modify the seal face. The usual seal materials, impregnated carbon and silicon or tungsten carbide, can be made quite resistant to a variety of conditions. Nickel-bonded tungsten carbide shows a higher corrosion resistance than the cobalt bonded variety. Most times, face corrosion results from an incorrect application or installation of a component different from that originally specified.

Figure 13.8 shows a carbon ring subject to severe corrosion. This is a regular phenolic impregnated carbon ring installed in a HF alkylation pump by mistake. A Teflon impregnated ring should have been used.



FIGURE 13.8 *Carbon ring showing severe corrosion due to misapplication.*

13.9.2 Secondary Sealing Corrosion

Several damage mechanisms may affect some elastomer secondary sealing elements. Fluorelastomer, perfluorelastomer, and graphite secondary sealing, on the other hand, are inert to almost all the fluids found in the process industry. Again, elastomer damage often is a result of misapplication.

Elastomers may swell if fluid absorption is possible. This swelling results in an increased diameter, which may impair seal movement, distort faces, or even fracture a seal face. Allowable swelling for a static O-ring is about 50% in volume. A dynamic O-ring may begin not to work properly if swollen by more than 15 or 20% in volume. An easy way to prove the swelling of the ring is to compare actual versus catalog dimensions.

As mentioned previously, fluorelastomers and perfluorelastomers are virtually inert to almost any fluid found in the process industry. A correct specification can solve most problems. O-ring swelling, normally, is due to misapplication. Figure 13.9 shows a swollen O-ring. The volume increase has been so large that the seal face is broken, as shown in Figure 13.10.



FIGURE 13.9 *A swollen O-ring, shown on the outside. The ring on the inside is a new one, introduced into the picture to make comparison possible.*

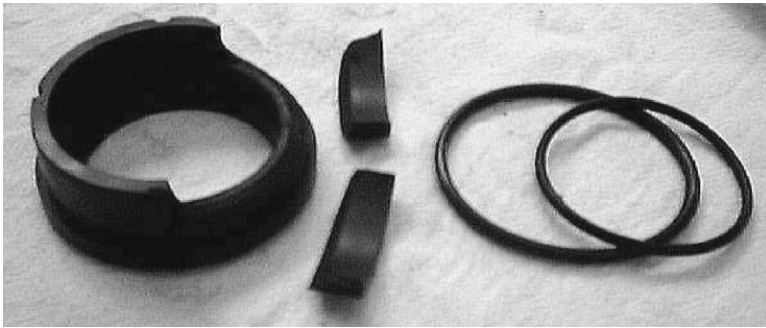


FIGURE 13.10 *Carbon ring broken by the swollen O-ring shown in Figure 13.9.*

Another example of misapplied O-ring is shown in Figure 13.11. In this case, the material has been almost completely dissolved by the sealed fluid, a hydrocarbon. The fluorelastomer originally specified for this seal would not have been affected. It is not easy to correctly identify the O-ring material; great care should be exercised in the identification of purchased and stored elastomer parts.



FIGURE 13.11 *Elastomer O-ring that suffered chemical attack due to exposure to hydrocarbons.*

13.10 Mechanical Damage

Mechanical damage of seal faces can be found as sliding wear, abrasion, fractures, distortion, or chipping. Secondary sealing mechanical damage can be in the form of deformation or cuts. The seal mechanism may mechanically hang up due to incrustation. Some typical problems and examples are discussed in the following sections.

13.10.1 Face Wear

The normal wear mechanism of mechanical seal faces is the removal of the high spots on the softer surface by contact with the harder surface. If no external hard particles are present, there will be no abrasion and the resulting face finish can be even smoother than the original. Figure 13.12 shows two seal rings that worked with propane for about one and a half years. Propane is not a good lubricant, and under these conditions, a seal operates with a large part of the total face load supported by the solid contact. No other adverse condition was present, and the seal face was able to operate without a leak even after losing a few millimeters of surface material.



FIGURE 13.12 *Faces of a mechanical seal that operated in a propane pump. Note the smooth surfaces, despite an approximately 3 mm material loss on the soft surface.*

Some general rules make the identification of seal wear easier. Most seals operate with a certain amount of solid contact. The fraction of the total load supported by the solid contact is a function of the seal design and operating conditions, but as a general rule for conventional design seals, the smaller the viscosity of the fluid, the greater the solid contact fraction of the total face load. Sliding wear is the result of this contact. If no other adverse condition is present, face wear is slow and smooth, a large increase in leakage is not expected before a significant portion of the faces is removed.

Hard particle wear displays a characteristic feature, some concentric grooves made by the individual particles. Note that this feature depends highly on particle size. Very small particles may result in a polished surface not easily distinguished from the previous case. Figure 13.13 shows typical abrasive wear on a seal face. In this case, the particle size was large enough to produce clearly visible grooves on the silicon carbide ring surface.



FIGURE 13.13 *Seal ring with grooves produced by hard particles.*

Erosion also may be found on mechanical seal ring surfaces. Unlike the former cases, it normally does not affect the seal faces. For example, poorly designed flushing ports can produce a high velocity jet of fluid that may be directed against the seal rings. If the flushing fluid contains solids in suspension, these solids may erode the seal components. Figure 13.14 shows a seal carbon ring with heavy signs of erosion due to an improperly located restriction orifice. The restriction orifice should never have been installed on the seal gland.

Many plants use line filters at the flushing inlet. These line filters are intended to avoid hard particle wear and seal hang-up. They should not be used, as the screen opening is usually around 0.5 mm, large enough to allow the most harmful particles to enter the seal, and the total screen area is too small, being easily clogged. The lack of flushing can do more harm to the seal than dirt in the fluid.



FIGURE 13.14 *Carbon ring with erosion due to high-velocity fluid impact.*

13.10.2 Seal Face Fracture

A seal face can be broken in many ways, and the brittleness of the materials commonly used in its construction make it even easier for a fracture to develop. The most obvious source of face fracture is improper handling, which needs no further comment.

A less obvious, but commonly found, fracture mechanism is thermal chock. The internal stresses that develop from thermal gradients easily break a seal ring. The causes can be quite varied, too. Irregular flushing injection, too high PV , and operation errors are a few examples. Figure 13.15 shows an outdated metallic seal ring with signs of heat chocking fractures on the surface. This particular example has been caused by a too large heat generation (PV larger than the material limit). The radial crack pattern is typical of this type of failure mode.

Figure 13.16 shows a silicon carbide ring fractured due to the thermal chock produced by an operation error. The pump was started with the flushing injection shut off. When the operator got

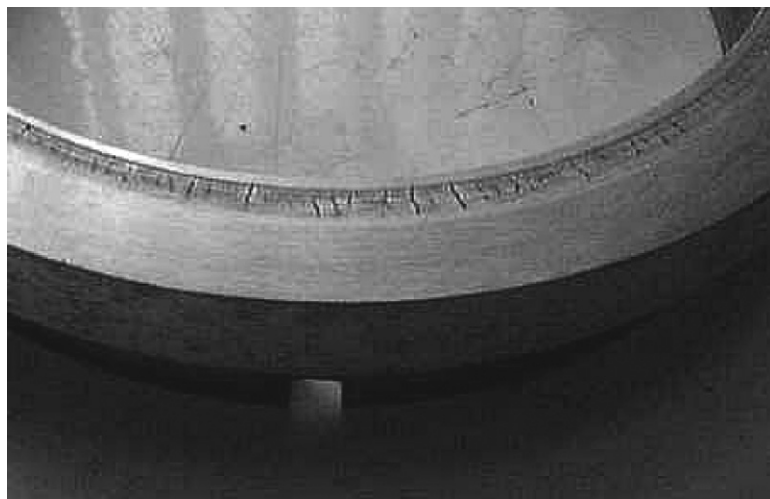


FIGURE 13.15 Seal ring with the characteristic radial cracks caused by heat chocking.



FIGURE 13.16 *Silicon carbide seal ring broken due to thermal chock.*

back to pump and opened the flushing injection, the seal rings were hot enough to be damaged by the inflow of cold fluid. Silicon carbide is particularly sensitive to heat chocks, although its high thermal conductivity and low friction coefficient, when contacting carbon, makes this an unusual problem.

The mechanical conditions of the pump can have an effect on seal behavior. High vibration, high shaft run-out, and the like may damage the seals. The extent of the damage can range from a mere increase in the contact pattern width to seal ring fracture due to the impact with pump parts. Figure 13.17 shows a carbon ring broken by impact with the seal cavity. The reason for the impact was a combination of high vibration due to pump cavitation with an outdated pump and seal design, with a very little clearance around the seal. Figure 13.18 shows a tungsten carbide ring damaged by the impact of the seal sleeve. The shaft sleeve was brought into contact with the seal ring because of a bearing problem.

Excessive or oscillating driving torque can fracture the carbon ring. The reasons for this problem can be a too viscous liquid or inadequate lubrication conditions. Hot water, for example, is a difficult



FIGURE 13.17 *Carbon ring broken by impact with the inside of the sealing box, caused by pump cavitation.*



FIGURE 13.18 *Tungsten carbide ring damaged by contact with the shaft.*



FIGURE 13.19 Carbon ring damaged by drive mechanism impact from oscillating lubrication conditions.

fluid to seal, because of the low viscosity and high vapor pressure. Hot water seals are prone to a condition called *slip-stick*. This condition occurs when friction between the seal faces changes rapidly, and the same happens with the driving torque. This situation may result in impacts on the drive mechanism of the seal, which may cause fractures. Figure 13.19 shows a boiler feed pump seal, subject to slip-stick from the rapid variations in drive torque caused by the sporadic vaporization of the hot water between the faces. The seal manufacturer designed a stronger drive mechanism to avoid damage from torque variations, but this was not enough.

This problem was solved with the introduction of a heat exchanger to cool the seal area and a pumping ring to move the water to the heat exchanger. The lower temperature increased the viscosity of the water and reduced its vapor pressure, avoiding lubrication problems.

13.10.3 Seal Face Distortion

A small distortion of the seal faces may not be enough to fracture the seal rings, but even the smallest deformation of the seal faces

increases the leakage. A distorted face is evident due to the irregular contact pattern. The most frequent causes of seal face distortion are

1. Uneven cooling or insufficient heat removal from the interface. The distortion mechanism in this case is differential thermal expansion of the seal ring. If the face is hotter than the rest of the seal, the faces are convergent and the seal may operate with contact at the inner diameter only. Edge chipping is likely in this situation, due to stress concentration on the contact circle. Figure 13.20 shows a seal face distorted due to poor heat removal from the interface. It is possible to note that there has been contact on the internal diameter of the ring. Internal edge chipping also easily is noticed. The reason why this seal was not properly cooled was an inadequate sealing system, which did not allow for proper circulation of cooling fluid around the seal.



FIGURE 13.20 *A carbon ring subject to thermal distortion due to poor heat removal from the interface. Note that there has been contact on the inside diameter only, meaning that the faces have been made convergent by the deformation.*

2. High sealing pressure can cause the opposite effect, if the seal is externally pressurized. The seal deformation causes the faces to become divergent. Divergent seal faces restrict the access of the cooling fluid to the faces and cause wear at the outside diameter of the rings.
3. Irregular loads may cause face distortion. The high precision required for face flatness and the flexibility of the faces enable even small irregular loads to cause harmful distortion. These irregular loads may be caused by mechanical problems, irregularities of the drive mechanism, and the like. Figure 13.21 shows a seal face distorted by a problem with the drive mechanism. Note that the distorted region is polished by the heavier contact. The carbon ring was fractured by this distortion. The irregular load provided by the drive mechanism caused the carbon ring not to be perfectly flat. The harder contact at the high spot resulted in faster wear and high friction, which means higher temperature. The higher temperature increased the distortion, making the overall effect worse.



FIGURE 13.21 Carbon ring mechanically distorted by a drive mechanism problem. A crack developed at the distorted point.

13.10.4 Seal Hang-Up

Incrustation of the seal mechanism may result in a seal hang-up. Entrainment of dirt on the springs or secondary seals areas may jeopardize the capacity of the seal to move axially. If the seal cannot move axially, its faces cannot adjust themselves to each other and leakage may increase.

An efficient way to avoid problems with seal incrustation is to use a seal design that keeps the whole mechanism protected from the sealed fluid. In this case, the probability of problems is much smaller.

Metal bellows seals should be treated with special care. Any particulate of higher density than the sealed fluid is forced to the outside by the centrifugal force generated by seal rotation. If a metal bellows is installed with the possibility of having a dirty fluid in contact with the inside diameter, rotation of the seal forces the particles into the bellows convolutions. Once this happens, the bellows are locked. It is impossible to clean a bellows in this situation, unless the solid particles are soluble in some known cleaning agent.

Figure 13.22 shows the inside diameter of a metal bellows seal covered with hard coke particles. This seal lost its ability to move axially. One way to avoid this accumulation would be to have a



FIGURE 13.22 *Metal bellows with coke accumulation on the inside.*

rotating bellows with the outside diameter exposed to the dirty fluid, so that centrifugal force would help keep particles away from the seal.

13.10.5 O-Ring Shearing

O-rings are very sensitive to shearing, as they are made of soft materials. This shearing sometimes happens during seal assembly, when the O-ring is forced past a sharp edge or fillet on the shaft or sleeve surface. Figure 13.23 shows an O-ring cut due to the sharp edge on a shaft. A simple way to avoid this problem is to round all fillets and transitions.

13.10.6 Seal Misalignment

Like any other mechanical component, mechanical seals require a certain precision in their alignment with interrelated components. If the rotating part of a mechanical seal is not perpendicular to the shaft, there will be axial movement of the flexible component every turn of the seal. This reciprocating movement can cause excessive wear of the torque transmission devices, as shown in Figure 13.24. The figure shows a carbon ring that had the torque transmission



FIGURE 13.23 *O-ring cut by the sharp edge on a shaft.*



FIGURE 13.24 *Carbon ring displaying heavy wear of the torque transmission device.*

device worn from excessive axial movement caused by the lack of perpendicularity.

It may seem that a large error in perpendicularity is necessary to cause such accelerated wear, but this is not exactly true. A 0.2 mm error in a seal that rotates at 3,550 rpm causes a total axial sliding of around 2 km a day. This is quite a long distance.

13.11 Thermal Damage

In addition to the possibility of thermal chocks, already mentioned, operation at high temperature may damage a mechanical seal in two other ways:

1. Carbon rings are more resistant to thermal chocks than hard rings. Nevertheless, they may be subject to high temperature

damage in a different way. If a carbon ring is exposed to conditions that result in excessive fat heating to high temperature, the impregnated resin may evaporate and cause surface blistering. This surface blistering starts with the formation of a high spot on the surface at the place where the resin boiled off. The high spot is subject to harder contact with the other face, increasing the temperature even further. The process ultimately may finish with the loss of surface material and the formation of a crack. This phenomenon is rarely seen with the carbon materials used nowadays.

2. Secondary sealing usually is made of elastomers, whose temperature limit generally is not very high. Most elastomers used in the industry are damaged if exposed to temperatures higher than 150 or 200°C, only a few stand more than that. An overheated elastomer normally becomes brittle, due to the increase in the number of cross-links between molecules. This makes the elastomer hard and no longer able to deform elastically, as originally intended; the inability to adapt to the movement of other parts results in leakage. Figure 13.25 shows a fluorelastomer made brittle by exposure to high temperature in a hydrocarbon environment. Although this is an obvious example, some situations require careful analysis, as this failure mode can be confused with chemical attack.

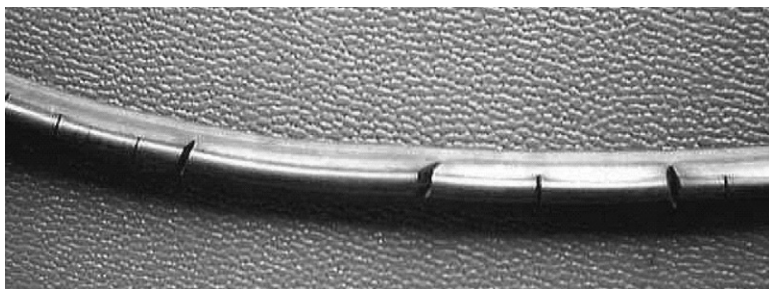


FIGURE 13.25 *Fluorelastomer O-ring damaged by exposure to high temperatures.*

13.12 Design and Manufacturing Defects

Design deficiencies are as varied as seal designs. In addition, a feature considered an inadequate design today may have been the state of the art some time ago. A design deficiency is considered if a specific feature of the seal played a prominent role in the failure. Some examples of design problems have been discussed in connection with the failure modes described previously. Other examples follow.

Figure 13.26 shows a seal ring with a tungsten carbide insert. This design has been used in the past but is no longer acceptable in the petrochemical industry because of the tendency of the insert to move out of its correct position and cause leakage.

Figure 13.27 shows another seal design popular in the past but no longer used. The Teflon wedge is not flexible enough to deform and absorb small axial movements without sliding on the sleeve, causing fretting.



FIGURE 13.26 Seal ring with inserted tungsten carbide face.



FIGURE 13.27 *Seal sleeve with signs of fretting caused by the Teflon wedge.*

Manufacturing defects seldom are found in a mechanical seal. These problems usually are related to face-finishing problems, like roughness or flatness errors. Construction of the seals may be done with materials not appropriate for the project, or dimensions may be wrong. A manufacturing defect should not be suspected without direct evidence.

14

Bolts

The function of bolts is to transfer load from one machine component to another. This load transmission may be to allow for transmission of power, as an alternative compressor rod bolted to the piston; to keep components together and avoid leakage, such as in machine flanges; and so forth. Bolt life is considered indefinite, an objective easily achieved with a correct design and installation.

When selecting and installing bolts consider these factors:

1. The magnitude and direction of the load to be transmitted.
2. The reason behind the union.
3. The type, thickness, and stiffness of the components to be joined.
4. The work environment.

The variety of factors involved in bolt selection means that great care should be exercised when replacing a bolt. Except for the simplest of jobs, an identical bolt always should be used for replacement.

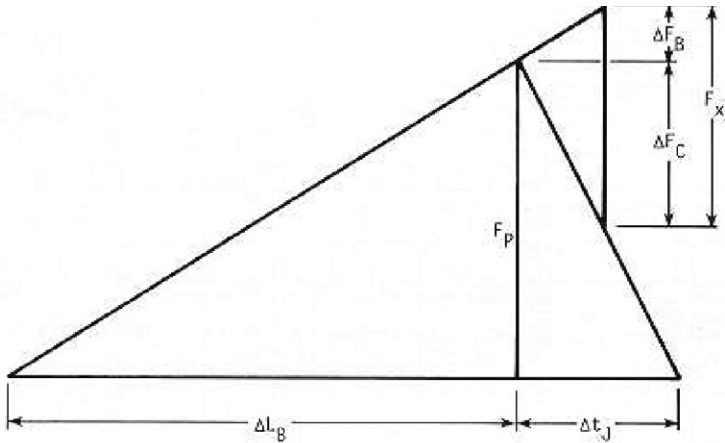


FIGURE 14.1 *The deformations caused by the preload (F_P) and by modification of the component loads when an external force (F_x) is applied.*

14.1 How a Bolt Works

Load distribution over the bolt threads is not uniform. Material elasticity causes the concentration of loads on the threads closer to the load-bearing surface. When the bolt is preloaded during installation, there will be some elastic deformation of the bolt and junction. The amount of elastic deformation of each component depends on its stiffness. The bolt is the more flexible part of the junction and suffers larger elastic deformation. Figure 14.1 shows schematically the elastic deformation of the bolt and junction. This figure shows that the elastic deformation of the bolt is larger.

When an external load is applied, part of the junction load is relieved, and the bolt load increase is smaller than the applied load. This is the reason why bolts subject to cyclic loads may suffer fatigue fracture when not correctly preloaded.

14.2 Application of the Preload

Although the preload is quite an important part of bolt design, the application of the correct preload in a field installation may not be easy. Fortunately, only critical service bolts require accurate applica-

tion of the preload. An indication of the accuracy of the applied preload as a function of method of application is shown in Table 14.1.

The preload reduces spontaneously after some time. The reasons for the reduction are the local plastic deformation that occurs on the load-bearing interface. As explained in Chapter 6, “Wear,” the contact between the load-bearing surfaces causes local plastic deformation of the contact spots. This mechanism may reduce the preload error by 2–10%.

When a junction uses several bolts, the elastic interactions among the various bolts cause the preload error of one bolt to be relieved when the adjacent bolt is torqued. This effect can relieve the preload error by 10–80%, introducing the need for multiple tightening of the bolts.

The relation between applied torque and bolt preload is calculated by

$$T = K \times d \times F_{\text{pt}}$$

where

T = applied torque ($N \cdot m$);

F_{pt} = preload (N);

K = experimental friction factor, shown in Table 14.2.

Several methods are used to prevent bolt loosening, such as self-locking nuts and adhesives. Welding should be avoided, as the heating can reduce the bolt strength and cause plastic deformation.

TABLE 14.1 *Estimated Preload Error as a Function of the Application Method*

<i>Application Method</i>	<i>% Error</i>
Manual tools	±80
Pneumatic impact tools	±50
Hydraulic elongation tools	±20
Bolt heating	±15
Measurement of bolt elongation	±5
Strain gauges	±1

TABLE 14.2 *Experimental Friction Factors*

<i>Type of Lubrication or Coating</i>	<i>K</i>
Cadmium plated	0.15–0.33
Galvanized	0.26–0.40
Oxidized	0.11–0.28
Molibdenum bissulfide	0.14–0.17
As received steel bolts	0.16–0.27
Copper-based grease	0.08–0.23
Oil	0.20–0.23

14.3 Reusing Bolts

Bolt reutilization should be decided with care in critical applications. Examples of critical applications are reciprocating compressor heads, high-speed coupling, and other services subject to cycling or very large loads. The bolt to be reused should be carefully inspected to make sure that there is no thread deformation, cracks, bolt elongation, or other damage.

Although utilization of a used bolt can save money in the case of special bolts, it is probably cheaper to replace a general-purpose bolt, like the ones found in a valve bonnet or piping flange, instead of reusing them. Figure 14.2 shows the cylinder head bolts of a reciprocating compressor inspected with a dye penetrant before re-installation. Self-locking bolts should not be reused.

14.4 Bolt Failure

The most commonly found bolt failure mechanism is fracture. These fractures often initiate at the transition between the cylindrical and threaded regions of the bolt. Common causes for bolt fracture are the use of components that lack the required resistance, normally because of a design error; adjustment problems between the junction components; generating loads that are not aligned with the bolt centerline; and inadequate preload.

Figure 14.3 shows a fatigue fracture of a bolt. This bolt originally was installed on the bonnet of a plug valve that operated with

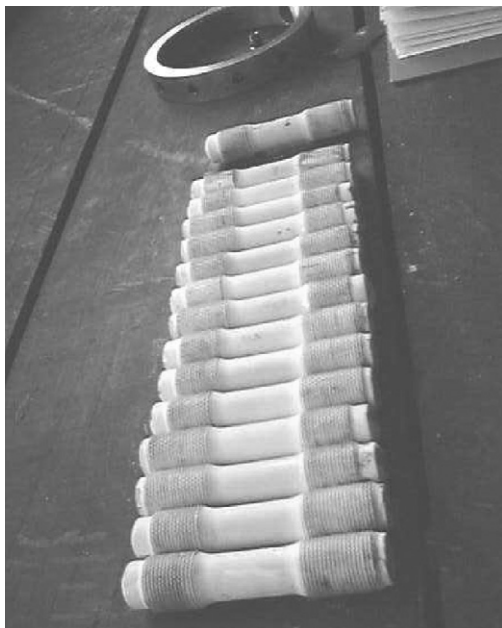


FIGURE 14.2 *Reciprocating compressor cylinder head bolts being inspected before reutilization.*



FIGURE 14.3 *A fatigue failure of a valve bolt. Note that a certain portion of the fracture surface is oxidized.*

high-temperature hydrocarbon. The process required this valve to open and close twice a day; the mechanism of the valve caused a momentary increase in the bolt load during opening and closing the valve. Figures 14.4 and 14.5 show a view of the valve and the bolt that precipitated the failure, with the other bonnet bolts broken by

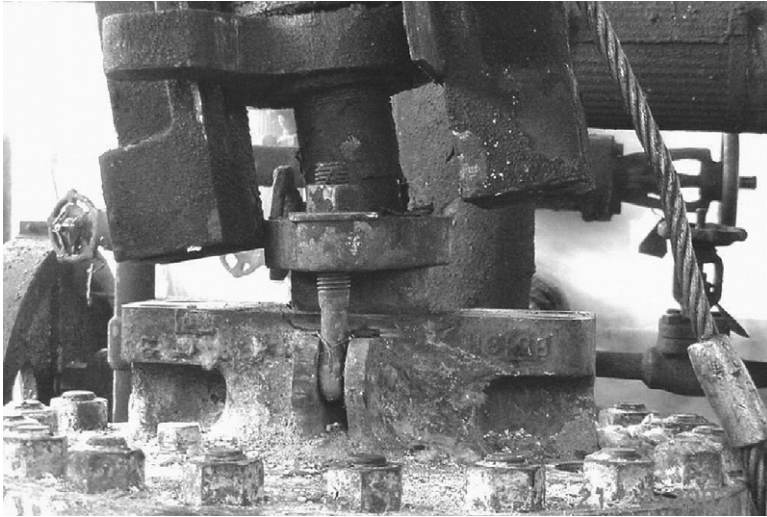


FIGURE 14.4 *Valve bonnet with four bolts fractures.*



FIGURE 14.5 *Bolt of the valve shown in Figure 14.4 that failed due to the overload from the fracture of the first bolt.*

the overload resulting from the first failure. The root cause of this failure is a design error. The valves originally were designed for manual operation. The manufacturer introduced a motor actuator without redesigning the junction. The problem was solved by the installation of larger diameter bolts.

Figure 14.6 shows a carbon steel bolt erroneously installed in a corrosive environment, which contained hot hydrocarbons. The corrosion made it possible for the bolt to fall loose.

Figure 14.7 shows a reciprocating compressor that suffered extensive damage from a bolt failure. The compressor casing was broken by the impact of the piston against the bolt, which fell inside the compressor cylinder after the fracture, as shown in Figure 14.8. Figure 14.9 shows the fracture face of the bolt, where ratchet marks are clearly seen, indicating a fatigue fracture. Figure 14.10 shows the point of initiation of the fatigue fracture, a manufacturing defect at the root of the thread. This manufacturing defect was caused by the rolling of the thread, indicating a quality control problem. Note that the position of the defect makes detection through inspection very difficult.



FIGURE 14.6 *A bolt corroded by contact with hot hydrocarbons.*



FIGURE 14.7 *Reciprocating compressor casing damaged by a broken bolt.*

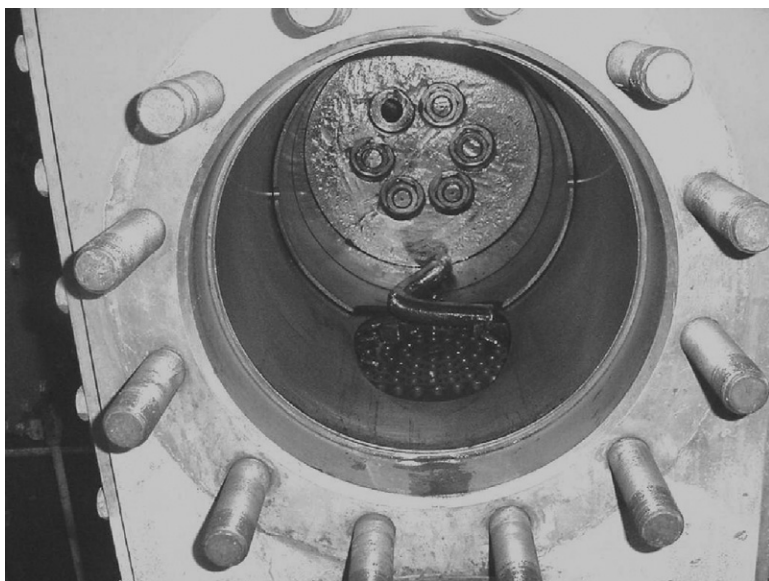


FIGURE 14.8 *Inside of the cylinder, showing the broken bolt at the bottom.*



FIGURE 14.9 *Fracture surface, displaying ratchet marks.*

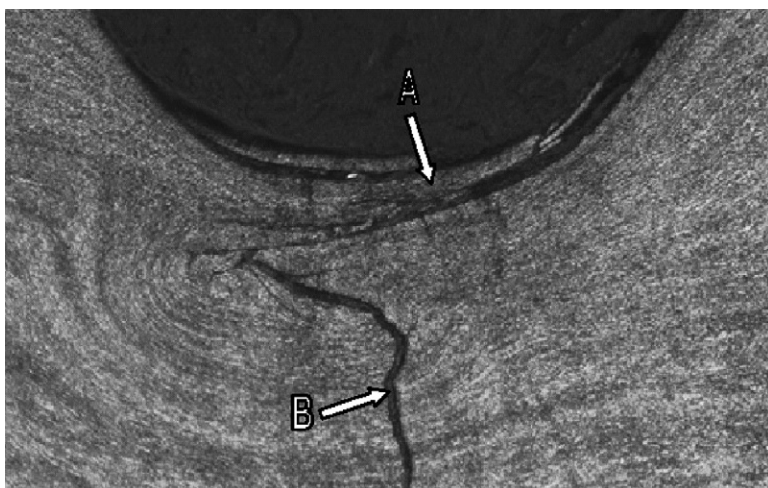


FIGURE 14.10 *Manufacturing defect found at the thread root (A), and the initiation spot of the fracture (B).*

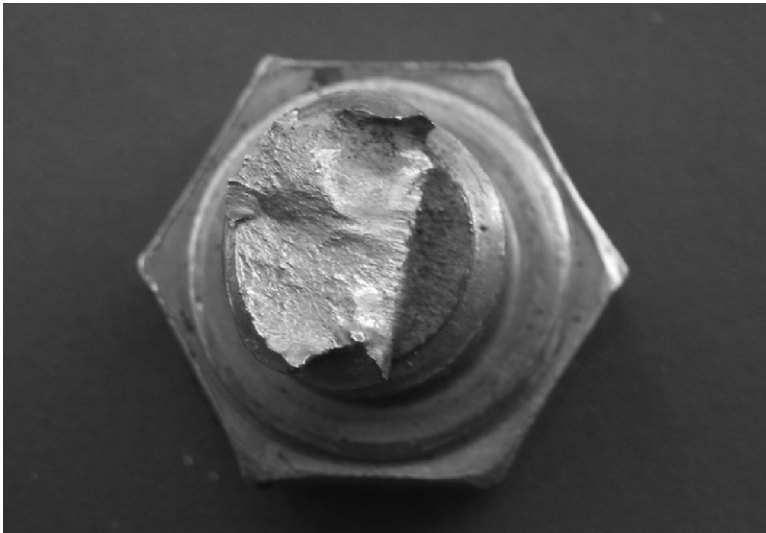


FIGURE 14.11 *A bolt sheared by the overload caused by a process upset.*

The last example of bolt failure is shown in Figure 14.11. This is a 316 stainless steel bolt used to hold the water distribution manifold inside a water treatment vessel. A process upset caused the resin inside the vessel to overload the water distribution manifold, shearing the bolt.

15

Gears

The function of gears is to adjust the speed of rotating shafts. The great variety of conditions that require this adjustment originated the large number of gear designs in use nowadays. Under normal conditions, gear service life is determined by the teeth wear and smoothness of operation. The higher the speed, the smaller the permissible loss of tooth through wear without an unacceptable increase in gear vibration.

15.1 Gear Tooth Contact and Lubrication

The reason for the high load capacity of gears is the mechanism of tooth contact. Gear teeth are loaded only when in contact with each other, which happens during a small fraction of the total operating time. The time between contacts is sufficient for the removal of the heat generated during the contact; and heavy loads, close to the physical limits of the material, may be transmitted.

The initial contact region of the gear teeth is close to the tip of the driven tooth and close to the root of the driver tooth. This contact region moves as the gears rotate. The surfaces slide and rotate

all across the gear height, except at the pitch diameter, where only rotation is observed.

Each of the three basic gear types (spur, helical, and worm) is characterized by a specific condition of tooth contact, which describes its basic operation. The tooth contact is a direct consequence of the gear geometry. All other gear types are variations of these three basic ones. Their tooth contact is similar to the description presented here.

Spur gear teeth are straight and contact each other in a straight line parallel to the shaft centerline that covers the whole length of the tooth. The contacting pair carries the entire load transmitted by the gears, which makes load transmission from one tooth to the next less than perfectly smooth.

Helical gears have helical teeth, which means that more than a pair of teeth are in contact at any given time. The total load is distributed between these contacting pairs, and only part of the total load is transmitted to the next pair. This characteristic makes helical gear especially suited for high-speed and high-load service, as the load transmission from one pair of teeth to the next is smoother than in the previous case.

Worm gear teeth contact each other in a very different way. In this case, tooth contact covers the whole face of the involved teeth. There is no relative rotation of the teeth, only sliding is observed. The consequence of this behavior is that heat generation is large, compared to the other two gears, limiting the use of this gear to small speeds and loads. Worm gears normally are used for shafts that cross each other and can provide high-speed ratios in a small set.

Tooth contact is heavily influenced by the accuracy of the tooth profile. This accuracy results from the selection of the gear quality, as defined by the American Gear Manufacturers Association (AGMA). The AGMA quality rating associates numbers to gear quality. Low-speed, simple services may use lower quality, cheaper gears; high-speed, critical service gears, like centrifugal compressor driver gears, need AGMA 11 or 12 grade quality. The high-quality gears obviously offer smoother power transmission, with less vibration and noise and longer life, but at a price.

The precision of gear manufacture usually is checked by a tooth contact test. This test is done by the transfer of a very thin layer of Prussian blue from one gear to another, on rotation. Correct tooth contact displays a regular, evenly distributed contact region. Figure 15.1 shows the tooth contact test results for a new gear set, displaying uniform and correctly distributed contact. One word of caution: The test of gears that have a tooth profile modified to compensate for the centrifugal force and thermal growth deformation effects should take this into consideration.

Lubrication of the gear teeth has two main functions: to provide an oil film (or an extreme pressure protection film) and to remove heat. Removal of particles is a secondary function. Most high-speed gears used in the process industry are lubricated with mineral turbine oil. Normally, the driven machine and gear are lubricated with the same oil.

There are many variations in the design of the gear lubrication systems, ranging from a simple layer of grease over the teeth in

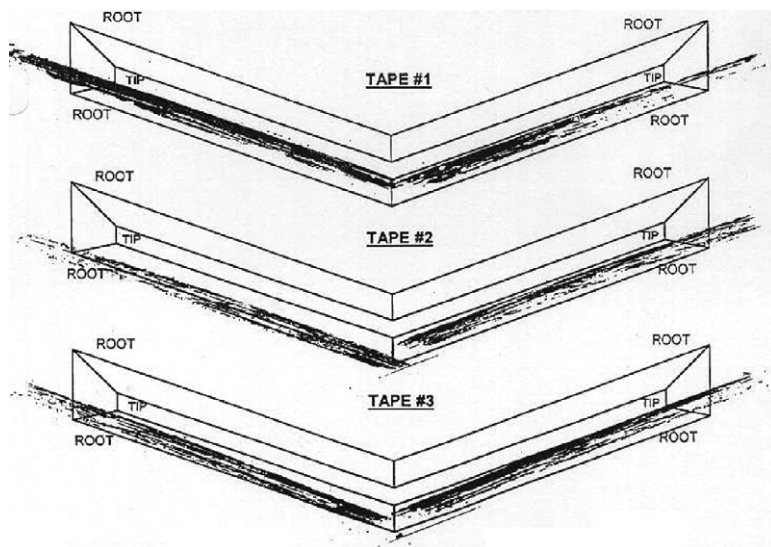


FIGURE 15.1 *Gear tooth contact tape lift, showing an evenly distributed contact.*

low-speed gears to a pressurized spray of oil directed to the contact region of high-speed machines. The heat removal function becomes more important in high-speed gears, where most of the oil is directed to the teeth that are leaving the contact area. These teeth are hotter than the rest of the gear, and heat transfer is more effective.

High-speed gears that work with little tooth contact stress may be able to form an oil film between the teeth, much in the same way as antifriction bearings. Lower speed, higher load gears normally are unable to produce a thick oil film, and a significant part of the load is supported by metal to metal contact. This type of gear benefits greatly from the use of extreme pressure oils, which form a thin protective layer over the tooth surface that makes adhesion more difficult.

Should the lubrication system be unable to remove the heat generated by tooth friction, overheating may result. Figure 15.2 shows an extreme example of gear overheating, which was enough to melt the gear teeth. This gearbox was the connection between an electric motor and a cooling tower fan. Lack of lubrication also can result in gear overheating.

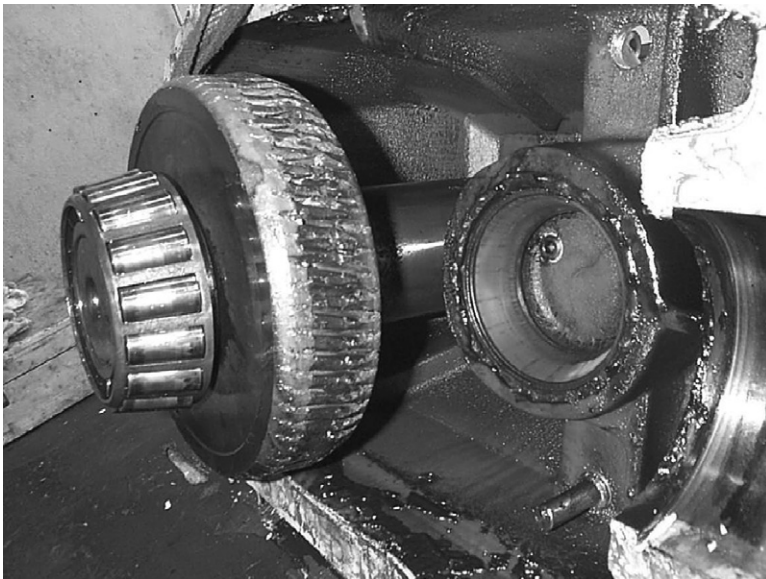


FIGURE 15.2 *Gear teeth melting due to a lubrication problem.*

15.2 Loads Acting on the Gear Teeth

The size of the load acting on the gear teeth and the amount of power transmitted are not good indications of the severity of the service. The severity of the service is related more directly to the factors that reduce gear life, like the conditions of lubrication, unexpected transients, and environmental factors, such as temperature, corrosion, and abrasives.

The main load acting on a gear tooth is the contact load. This contact load generates a certain stress distribution within the tooth. The largest stresses acting on an operating gear tooth are the contact stress at the contact point between two teeth and the flexion stress at the tooth root. Most gear failures begin in one of these two regions. The design methods used to size gear teeth are aimed at the determination of the local contact stress and the flexion stress and a comparison of these calculated values within some predefined limit.

Gear tooth materials are required to stand high contact and flexion stresses and usually are made of hardenable carbon or low alloy steel. The teeth may be through or surface hardened. Through-hardened teeth usually present hardness higher than 390 BHN; case-hardened teeth may have surface hardness of more than 58 RC, similar to an antifriction bearing hardness.

15.3 Gear Failure

The end-of-life failure mode for gears is tooth wear. Obviously, the characterization of the end of the useful life depends on the design life of the gear. Some high-speed critical service gears are designed for a 20-year life span. Therefore, the end of life of a gear may be called for only if the failure mode is tooth wear and the service life has been greater than designed.

A systematic gear failure analysis begins with data collection and inspection of the failed gear. A classification of the failure mode within the concepts described in the ANSI/AGMA 1010-E95 standard, “Appearance of Gear Teeth—Terminology of Wear and Failure,” may be helpful. This standard identifies and describes the classes of common gear failures and illustrates degrees of deterioration. Since the

decision whether or not a gear has failed depends on how much damage is tolerated by the specific service, the standard does not define failure but provides a common language to describe gear wear and failure. The standard does not discuss root causes or preventive measures for gear failures.

Most gear failures are related to operation or maintenance factors, like assembly, installation, lubrication, and overload. Gear teeth wear is discussed first.

15.3.1 Gear Tooth Wear

The wear mechanism may be classified, according to severity, as run-in, mild wear, and destructive wear. Keep in mind that the acceptable wear depends on the type and function of the gear, one gear's run-in or mild wear may be considered another gear's destructive wear.

Run-in is the wear process that removes the high spots of surface asperities during the initial operation of the gear, creating a polished surface. This is a normal process and normally is self-limited. Figure 15.3 shows a polished gear surface. A well run-in gear shows an appearance similar to the one shown and should be considered that it probably will have a long, trouble-free life. This statement is valid for the types of gears normally found in the process industry.

Mild wear manifests itself as slight material removal from the addendum and dedendum of the tooth face. Nearly no wear can be observed on the pitch line, as there is only rotation of the tooth faces over one another. Figure 15.4 shows the tooth face material removal mechanism, illustrating the reduced wear rate at the pitch line. Figure 15.5 shows a gear tooth that has been subject to mild wear. Note that surface finish is good and there is a small step on the left side of the tooth surface.

The tooth profile is not greatly affected by mild wear. The profile modification can be a problem to high-speed gears only. Mild wear of a gear tooth may be considered a problem if it is observed after a short operating time. If the operating time is long and compatible with the intended design life, it should be considered a normal process. Further inspection and gear replacement should be planned.



FIGURE 15.3 Gear displaying a polished surface, the result of a correct run-in.

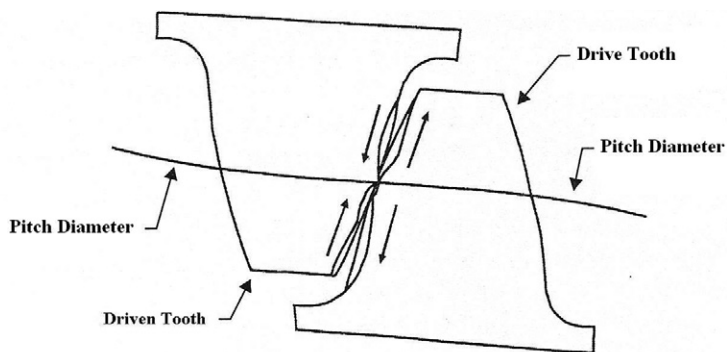


FIGURE 15.4 The wear mechanism at the addendum and dedendum of the tooth face.



FIGURE 15.5 Gear displaying mild wear.

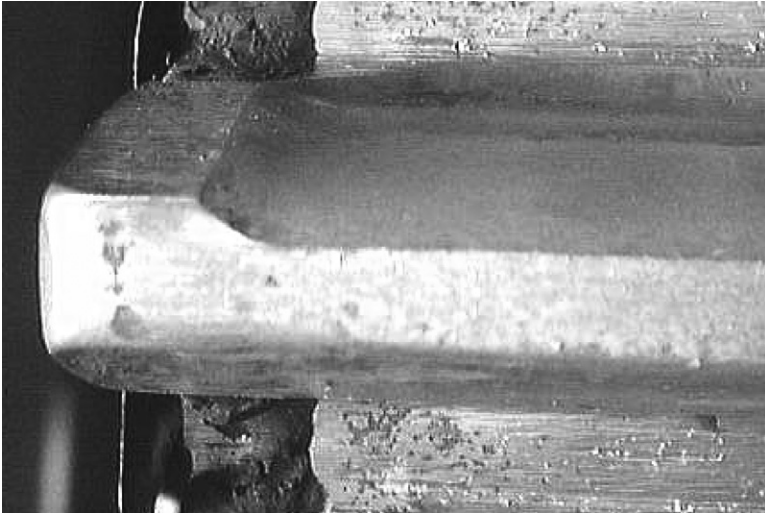


FIGURE 15.6 *Heavy destructive wear on a gear tooth surface.*

On the other hand, severe wear can be destructive, even for low-speed gears. The profile modification is large enough to make gear operation rough and noisy, even at low speeds. The gear should be replaced immediately. Figure 15.6 shows an example of a heavily worn gear. Note the huge modification on the tooth profile.

15.3.2 Surface Fatigue

Gear surface fatigue is similar to surface fatigue on other components and may happen even if operation conditions are perfect. Basically, the failure occurs in three situations.

Light pitting at the pitch diameter may be only a sign of the natural accommodation to surface irregularities. The sliding action of the teeth in the dedendum and addendum creates an opportunity for sliding wear on the asperity high spots. At the pitch line, there is no sliding action; and there is some stress concentration at the high spots. This stress concentration can remove the high spots through

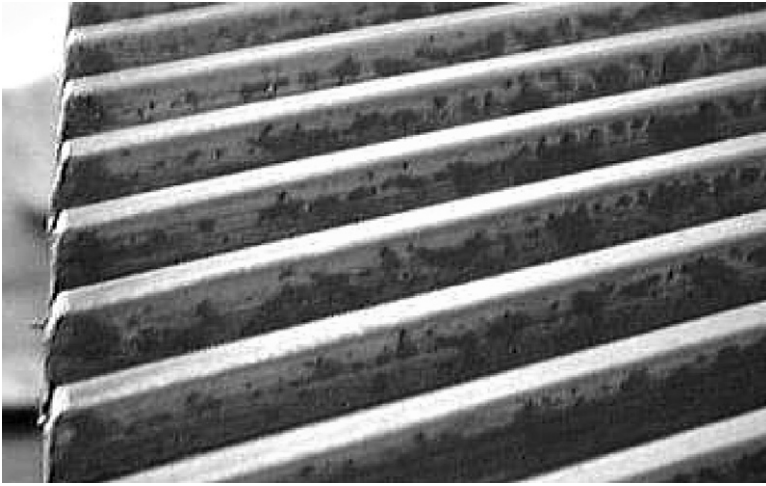


FIGURE 15.7 *High-speed gear with light surface fatigue at the pitch line.*

surface fatigue. This should not be thought a reason to replace the gear and usually is self-limited. Figure 15.7 shows an example of high-speed gear with signs of surface fatigue after a long operating period.

The second possibility is the appearance of surface fatigue on the dedendum of the driver gear. This is the region of the tooth that first makes contact with the driven gear tooth and where the sliding action is opposite to the rolling action, making stresses higher. This situation often indicates that the gears are overloaded. Figure 15.8 shows a gear displaying surface fatigue at the dedendum. Note that the pitting is evenly distributed over the tooth length, indicating that alignment was good.

The third possible situation is surface scaling due to the load concentration, caused by gear misalignment. This local overload is readily identified by the uneven distribution of the damage on the tooth surface. Figure 15.9 shows a gear damaged by this mechanism, with the concentration of the surface damage clearly seen.



FIGURE 15.8 *Overloaded gear, with surface fatigue indications at the dedendum.*

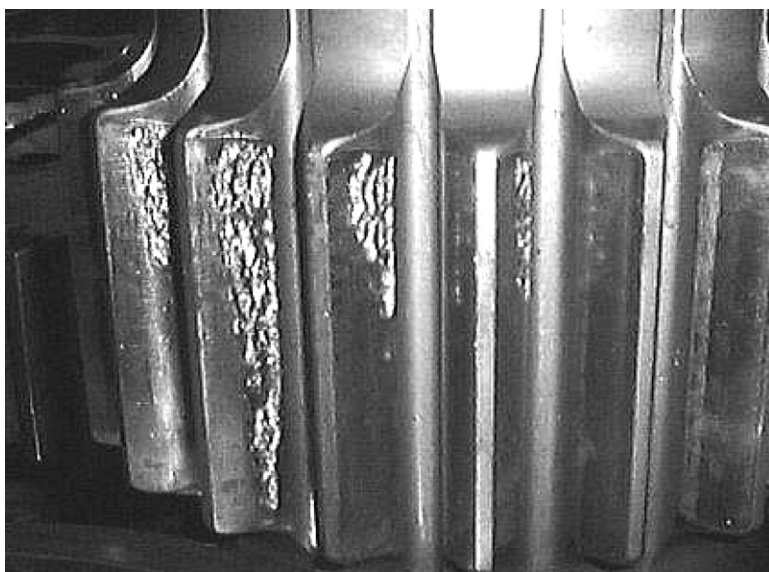


FIGURE 15.9 *Concentration of surface damage on one side of the gear tooth due to misalignment of the gear.*

15.3.3 Plastic Deformation of Gear Teeth

Plastic deformation of gear teeth occurs when the contact stresses combine with the sliding stresses and overcome the yield strength of the material. This failure mode often is associated with soft gear materials but may be found in high-strength materials.

The deformations can be classified by three different mechanisms:

1. Cold flow is the commonly observed situation where there is macroscopic plastic deformation because of high stresses. This normally is associated with soft materials; the material may be deformed beyond the tooth tip, forming a kind of flange. Continued operation under this condition results in increased vibration and noise. This type of problem can be avoided by using a harder material for gear construction.
2. Rippling is the formation of a regular wavy surface perpendicular to the movement direction. The surface seems scaled. This phenomenon is often found in hardened gear and is considered a problem only if the progression does not halt spontaneously. Low speeds and inadequate lubrication are contributing factors.
3. Ridging is the surface deformation that results in the formation of a series of high and low spots that spread in the direction of tooth movement. This occurs when a combination of high contact stress and low speed causes surface deformation in the specified direction. This phenomenon is usually found in heavily loaded worm or hypoid gears, which have high surface sliding.

15.3.4 Gear Tooth Fracture

Gear tooth fracture is caused by the same mechanism discussed previously. A tooth fracture should be considered an abnormal event. Figure 15.10 shows a fatigue failure of a gear tooth, where the usual progression marks can be seen. The operating time before fracture makes it easy to say whether a synchronous force has been responsible for the fracture, as discussed in Chapter 5.

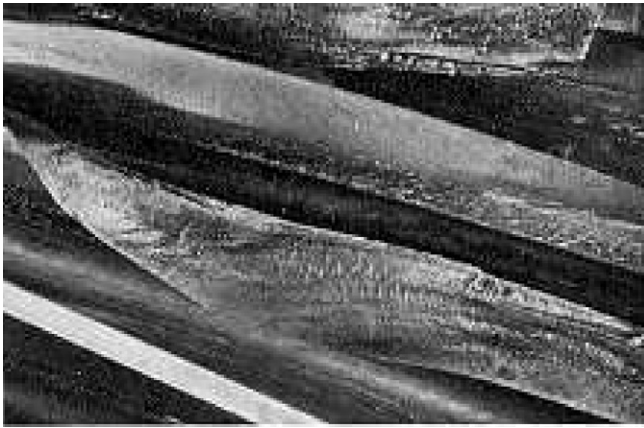


FIGURE 15.10 *Fatigue fracture of a gear tooth.*

Brittle fractures sometimes result from a local overload provided by a foreign body. A foreign body between the contacting teeth causes huge stress concentration and may originate a fracture. Figure 15.11 shows that it is a simple task to find the initiation spot and evidence that a foreign particle was involved. On the other hand, finding the root cause of the fracture and the origin of the particle can be difficult.

Local overload can cause tooth fracture. This local overload may originate, for example, from shaft misalignment. Figure 15.12

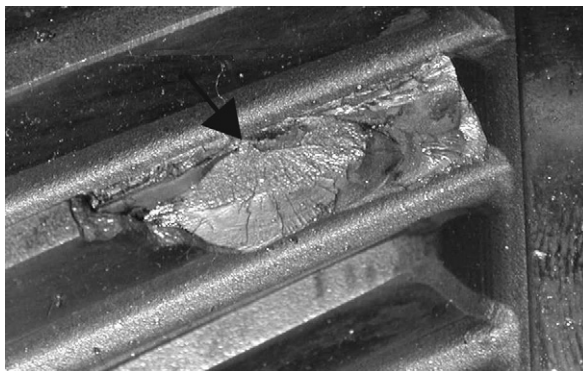


FIGURE 15.11 *Brittle fracture of a gear tooth, initiated at the point indicated by the arrow.*

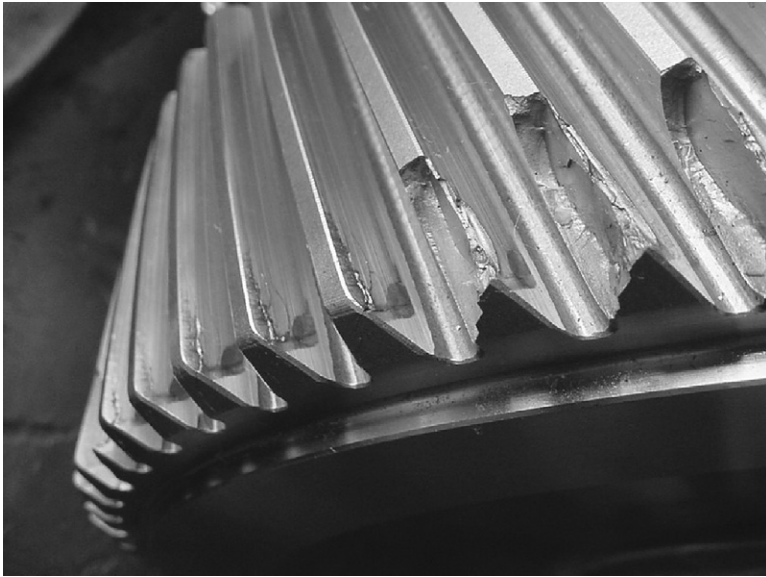


FIGURE 15.12 *Gear tooth fracture caused by local overload resulting from shaft misalignment.*

shows a gear fracture due to overload, caused by misalignment. This example is discussed further in Chapter 20.

15.3.5 Other Damage Mechanisms

Other damage mechanisms associated with gears are related to improper processing during manufacturing, environmental conditions, or accidents, for example.

Among the failures related to manufacturing are heat treatment cracks, caused by improper cooling, heat chocking, surface cracks caused by a too heavy grinding, and so on. Electrical damage also can be found, although it is rare.

Environmental conditions may result in gear corrosion, although this problem normally is avoided in the process industry through enclosure of the gear inside steel casings. One special occasion when gear corrosion may happen is when machines use a combined lubrication and sealing oil system and the compressed gas contains corrosive compounds. Oil contamination may happen in this kind of system, which may ultimately result in gear and bearing corrosion.

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16

Reciprocating Compressor Valves

The suction and discharge valves of a reciprocating compressor are intended to allow flow of gas in only one direction, with the smallest possible pressure drop. Most compressor failures are related to the valves. Part of the reason for this behavior is the existence of moving parts in contact with each other and with the compressed gas, with possible contamination from the process.

It is not possible to specify a design life for the valves, only a typical valve life for each type of service. The variety of valve designs and compressor services means that further discussion of this topic is beyond the scope of this book. A frequent design objective is to achieve valve life of 25,000 hours. This is a challenge, especially considering that, in a typical process compressor, operating at 400–500 rpm, the valves open and close more than 500 million times during this intended design life.

The types of valves most commonly found in process compressors are ring valves and poppet valves. Channel and plate valves are still found in some older machines. The sealing element material is special steel or a plasticlike glass nylon or PEEK.

16.1 How a Compressor Valve Works

Reciprocating compressor valves are automatically actuated by the gas movement. With the piston at the top dead center, shortly with speed equal to zero, the discharge and suction valves are closed. When the piston begins to move, the gas remaining inside the cylinder begins to expand and pressure is reduced, establishing a pressure differential between both sides of the valves. The discharge valve is unable to open, as the pressure differential is in the direction of the seat. When the pressure difference across the suction valve reaches a value larger than the pressure exerted by the springs, the suction valve opens and allows gas to enter the cylinder. The piston speed increases and so does the pressure drop across the suction valves.

When the piston crosses the halfway point of its stroke, velocity begins to reduce. By the time it is close to the bottom dead center, its speed is small and so is the flow of gas entering the cylinder. The reduction of gas velocity across the suction valves reduces the pressure drop across the valve, which becomes smaller than the spring pressure. At this time, the suction valve closes.

After the piston passes the bottom dead center, both valves remain closed until the cylinder pressure is higher than the discharge pressure by an amount sufficient to open the discharge valves. The operating conditions of the suction and discharge valves are different, because the piston speed is not the same when both valves open, the amount of time they are open is unequal, and the pressure and temperature of the gas flowing through the valves are not the same.

In addition to the spring pressure, the difference in the area exposed to the gas pressure at each side of the valve also generates the need for a difference in pressure to open the valve. This area is unique because the valve needs a seat. Some liquid may be present at the valves, causing a sticking that further increases a pressure difference for the valve opening. This liquid can be cylinder lubricating oil or the condensate of the gas.

No comments have been made about how much time the valve takes to open or close. A complete discussion of valve dynamics is outside the scope of this text, but a summary is presented here.

Valves close when the pressure drop due to gas flow is small enough. The springs accelerate the sealing element until it hits the seat. If the springs are too soft, the valves take too long to close and are slammed by the reversal of the gas pressure. This can increase wear or even break the sealing elements.

On the other hand, too stiff springs close the valves when the piston is too far from the dead center. The piston continues to move in the same direction, against a closed valve. The increase in pressure opens the valves again, the pressure in the cylinder drops, and the valve closes once more. This, called *valve fluttering*, can damage the valves in a very short time.

Some important parameters regarding valve efficiency and useful life are the velocity of the gas going past the valves and valve lift. The velocity of the gas on the valves is directly related to valve pressure drop and compressor efficiency. To keep the pressure drop low, a high lift is necessary; but a high valve lift leads to high impact velocity of the sealing element on the seat, which reduces valve life. These are examples of conflicting requirements in machine design.

16.2 Valve Failure

The external sign of valve failure is increased valve temperature. Further damage can lead to increased discharge temperature, reduction of compressor flow, and noise. Valve failure is characterized by the loss of the ability to allow gas flow in only one direction.

Valve leakage normally occurs through the seats, but may be observed past the gasket between valve and compressor, the valve center bolt, and between the valve body and the seat plate.

Valve failure may be related to mechanical wear and fatigue, foreign materials in the gas flow, and abnormal action of the valve or compressor components.

16.2.1 Mechanical Wear and Fatigue

Compressor valve wear is observed at the contact spots. Most compressor valve sealing elements are made of plastic material, the seats are normally made of stainless steel. Wear normally is concentrated

at the softer component. Ring or poppet wear impairs the sealing capability of the valve. Fracture of the elements may result, in the most severe situations.

Some factors increase valve wear, such as high-speed movement by the sealing element, which results in a stronger impact at the seats, or foreign matter, which may act as abrasives. Cylinder lubrication may reduce valve wear, because of the formation of a protective oil film, but too much oil may increase wear. The excess oil has a sticking effect, interfering with the valve dynamics. A valve ring stuck to the seat moves only when the pressure difference across the valve is large, being seated much faster than in a normal situation.

Figure 16.1 shows a glass-nylon valve ring subject to severe wear due to the deposition of coke on the valve seats. The original rounded profile has been destroyed. Figure 16.2 shows the inside of the cylinder where the valve was installed, with the coke incrustation readily seen.

16.2.2 Corrosion

Corrosion is rarely observed in compressor valves. As mentioned previously, most sealing elements are made of plastic materials, which are virtually inert when exposed to most gases used in the process industry. Metal rings may be subject to corrosion if not properly selected or the compressor is not properly operated.

Figure 16.3 shows a steel valve ring subject to corrosion. This ring was installed in an air compressor that operated occasionally. Water condensation due to cooling of the compressed air caused the observed damage.

16.2.3 Foreign Matter in the Gas

Even small quantities of solids and liquids entering the compressor may damage the valves. Solids may be rust, sand, or other process residues. Liquids may originate from condensation, cylinder lubrication, and the like. The main effect of solid contaminants is to create a stress concentration spot, which makes valve failure likely. Liquid entrainment, on the other hand, may interfere with valve dynamics in two ways:



FIGURE 16.1 *Reciprocating compressor valve ring subject to heavy wear due to the deposition of abrasive coke particles at the valve seats.*



FIGURE 16.2 *Compressor valve covered with coke.*



FIGURE 16.3 *Steel valve ring subject to corrosion due to water condensation.*

1. Sticking the valve rings to the seats, resulting in an increased impact against the valve plate when opening and against the seat when closing. This effect is most likely to happen when small quantities of liquid enter the cylinder.
2. Imposing an overload to the valve rings when forced by the piston movement, which may happen when a larger volume of liquid goes in the cylinder.



FIGURE 16.4 *Valve rings broken due to irregular action of the valve unloaders. Note the indentation caused by the unloader fingers.*

16.2.4 Abnormal Action of the Valves

Abnormal action of the valves occurs when the components of the valve do not act as designed. Some examples are impacts of the valve rings against the seats, fluttering, and piping resonance. The action of elements external to the valves, like valve unloaders, can be included. Valve unloaders are intended to keep the suction valve open when the compressor operates, reducing the flow of gas through the cylinder to zero. An abnormal action of the valve unloaders can affect the valve rings, as the unloader fingers touch the valve rings. Figure 16.4 shows an example of valve rings damaged by improper action of the valve unloaders. The indentation caused on the rings surface by the unloader fingers can be readily seen.

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17

Belt Transmissions

Belt transmissions are simple devices used to transmit torque and movement between rotating shafts. Power transmission occurs through the friction or engagement between a belt and a sheave.

This type of transmission displays the following characteristics:

1. Simple installation and maintenance.
2. High reliability and long life.
3. Ability to operate with high speed.
4. No transmission of vibrations and chocks.
5. Limited power transmission capabilities.

Power transmission occurs due to the difference in belt tension in the active and inactive sides of the belt. Figure 17.1 illustrates the difference in belt tension that allows the transmission of power from one sheave to another.

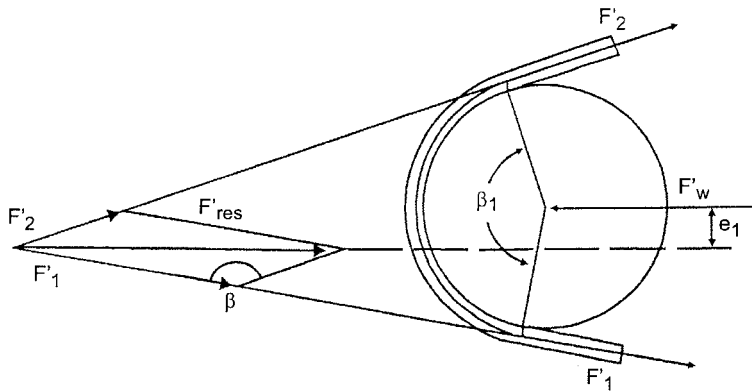


FIGURE 17.1 Forces acting on both sides of the belt. The difference in belt tension between the sides makes power transmission possible.

17.1 “V” Belts

“V” belts are nonsynchronized transmitters, as there is always a certain slip between the belt and the sheave. The wedge-shaped sheave increases the friction with the belt and reduces the bearing load for certain power transmissions. Figure 17.2 illustrates the increase in contact load due to the wedge effect of the sheave.

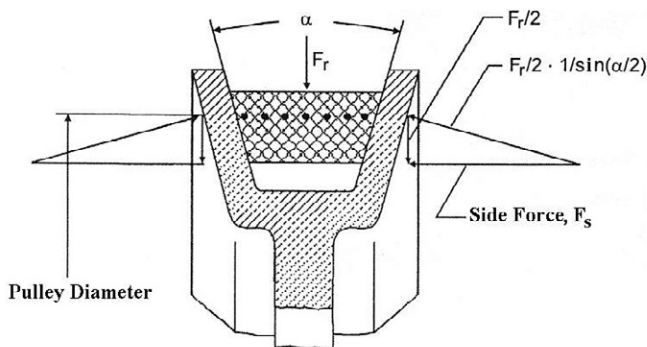


FIGURE 17.2 The wedge-shaped sheave increases friction and reduces the bearing load.

Sheave diameters should be large enough to keep bending stresses low. Rotating speeds should be kept within certain limits, reducing centrifugal force to a value that can be supported by the belt.

Belt manufacturers publish information about the service factors and power transmission capabilities of standard belts. A correctly designed and installed transmission lasts at least 24,000 hours.

“V” belts are actually endless belts with a trapezoidal cross section, assembled in sheaves that have a characteristic profile. “V” belts work smoothly and silently, absorb chocks, and do not exert great loads on the bearings.

A group of strings is responsible for the resistance to tension. This group of strings is surrounded by an elastomer body, which defines the belt shape. Some belts have an external wear resistant layer of another type of rubber. The construction of the belt is directly related to the way it works. Sheave friction pulls the elastomer body, and the elastomer body transfers the loads to the strings.

17.1.1 Factors That Influence Belt Life

Assuming that the belt has been correctly selected, the factors that influence belt life are sheave diameter and operating temperature. Sheave diameter is directly related to the flexion stress on the belt. A too small diameter results in higher bending stresses, which reduces belt life. Each manufacturer publishes notes on the influence of belt diameter on the life of the belt. As a general rule, for each 10% reduction in sheave diameter, belt life is reduced approximately 40%.

The operating temperature considered the basis for the 24,000 hours design life is 70°C. Higher temperatures reduce belt life from two effects: oxidation and softening of the rubber.

Other factors may influence belt life, such as the absence of one or more belts in a multibelt transmission, the use of pulleys that are not in perfect condition, and installation problems, such as incorrect preload or any damage.

17.1.2 “V” Belt Failures

The most common belt failure mode is deterioration due to high temperature. A 10°C increase in belt temperature reduces belt life by approximately 50%. The increase in temperature causes the rubber

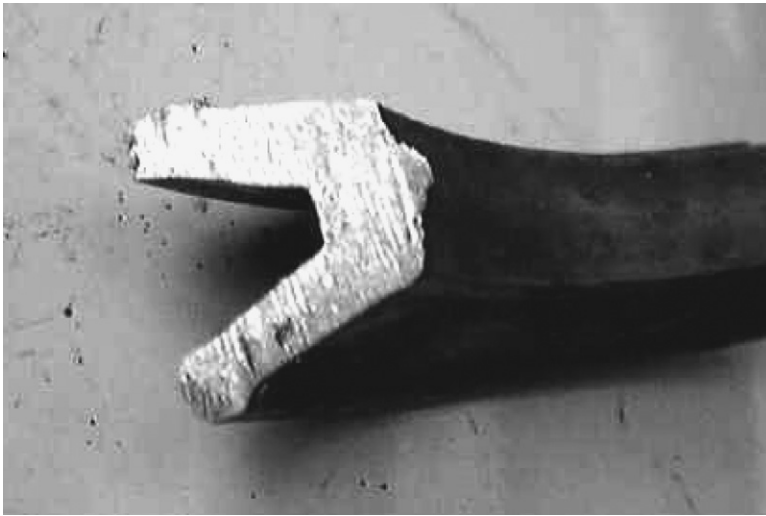


FIGURE 17.3 *Worn sheave can reduce belt life.*

to oxidize faster and makes it softer. A softer rubber concentrates the power transmission efforts on the strings closer to the surface, overloading them.

The main reasons for high-temperature operation are excessive slip of the belt on the pulley due to inadequate belt tension; high ambient temperature or belt enclosures that do not allow for cooling; misalignment of the belt on the sheave, causing friction on the pulley flanges; and excessive belt bending, due to the use of small sheaves.

A worn pulley may reduce the belt life due to the load concentration on the strings closer to the surface. Belt life reduction can be as large as 50%. Figure 17.3 shows a worn sheave.

17.2 Synchronized Belts

Synchronized belts are simply toothed flat belts. They are used mainly when a slipless transmission is necessary. The teeth that provide synchronization also allow a great power transmission in a compact device. One of the pulleys needs a flange to hold the belt in place. When not loaded, the distance between belt teeth is slightly

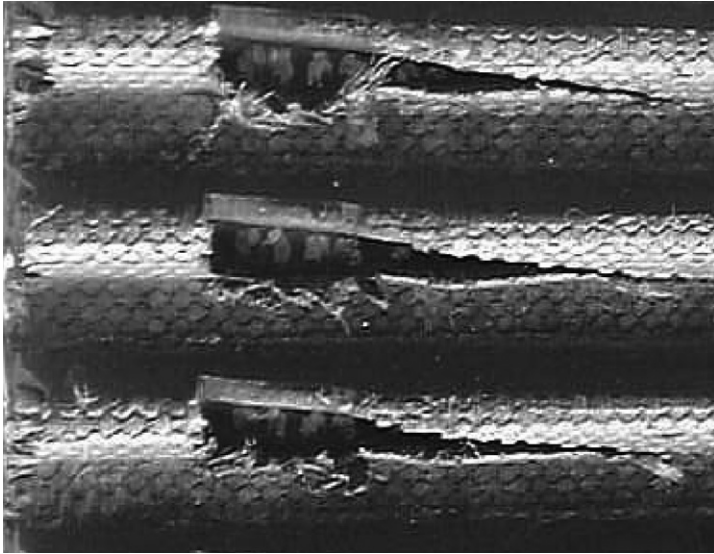


FIGURE 17.4 *Synchronizing belt displaying heavy wear from local overload caused by misalignment.*

bigger than between pulley teeth. A correctly preloaded belt fits the pulley perfectly. If belt tension is too high, tooth spacing is larger than required and the teeth on the nontensioned side of the belt are overloaded. Insufficient belt tension causes the opposite effect.

Geometry, construction, and failure modes are similar to the “V” belt, except that pulley misalignment causes more than just overheating. A misaligned toothed belt may be forced out of the flangeless pulley and locally overloaded. This can reduce belt life or even tear the belt. Figure 17.4 shows a synchronized belt torn due to misalignment. Part of the belt was not touching the pulley, causing local overload.

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18

Couplings

Couplings are used to connect rotating shafts. The functions of the coupling are

1. Torque and rotation transmission within the prescribed design limits.
2. Vibration dampening, especially rubber element couplings installed in reciprocating machinery.
3. Allowing for some shaft misalignment. This is the main reason why couplings are used instead of bolted flanges to connect rotating machinery shafts. An important issue here is how much misalignment a certain coupling can admit before a problem arises. The manufacturer's published information many times quotes allowable misalignment of 1 or 2 mm. This should be understood as the maximum allowable misalignment for the coupling only, without regard to the machines connected by the coupling. Allowable misalignment of process machinery shafts is approximately 0.05 to 0.1 mm. More misalignment puts excessive force on the bearings and causes high vibration. Figure 18.1 illustrates how a spacer coupling can

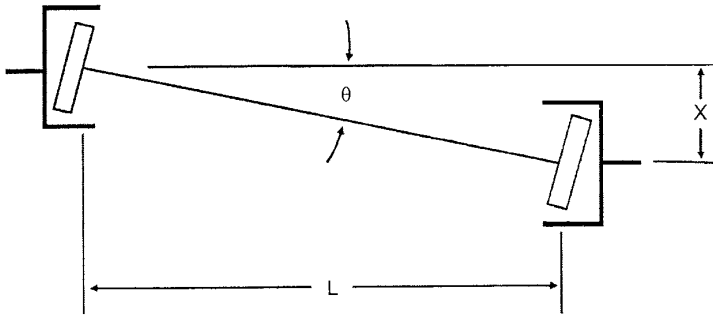


FIGURE 18.1 *How a spacer coupling tolerates lateral misalignment.*

tolerate lateral misalignment, even though each flexible element can stand only angular misalignment.

4. Influence the resonance frequencies of the machines connected. The coupling torsional stiffness can be fine-tuned to a certain value to modify the torsional natural frequencies of the train. This is a common technique used to tune natural frequencies away from existing excitation frequencies. The coupling mass and moment also can be used to influence the lateral natural frequencies but to a lesser degree.

Flexible couplings can be mechanically flexible or elastically flexible. Grid and gear couplings are mechanically flexible; there is some relative movement between coupling components to allow for misalignment tolerance. Despite having high torsional stiffness and large torque resistance in a small size, they need to be lubricated. This is a huge disadvantage compared to elastically flexible couplings, which allow misalignment because of the elastic deformation of one or more of the coupling components. Diaphragm and disk couplings are elastically flexible.

18.1 Disc Coupling Couplings

Disc coupling couplings have a flexible metallic element that deforms elastically to absorb the misalignment. This type of coupling does not require lubrication, which constitutes a great advantage.

tage. It can be designed to operate at high speeds. This type of coupling is probably the most popular in the processing industry. Most types of disc coupling couplings need a spacer to accommodate lateral misalignment.

A common failure mode of this type of coupling is fatigue failure of the flexible elements. There is a certain amount of elastic deformation to absorb the misalignment, which results in stresses on the membrane. The stress is directly related to the degree of misalignment. When the coupling rotates, the membrane is deformed cyclically to the opposite sides. Should these cyclic stresses be high enough, fatigue failure may result. Figure 18.2 shows an example of this type of failure, a cooling tower fan coupling, which has two membranes on the extremes of a long shaft. Alignment of cooling tower fan coupling can be tricky, because of the distance between coupled machines. This coupling was not perfectly aligned, and



FIGURE 18.2 *Cooling tower fan coupling elastic element displaying fatigue failure due to misalignment.*



FIGURE 18.3 *Centrifugal coupling blade that failed due to misalignment.*

fatigue failure resulted. Figure 18.3 shows another example, a centrifugal pump to electric motor coupling. The motor bearings are hydrodynamic. An increase in bearing wear caused misalignment of the shafts, overloading the couplings.

A large misalignment is not the only reason why a disc coupling may fail due to fatigue. Large oscillations in the transmitted torque may have the same result. Disc coupling couplings are not the best option in the presence of torque oscillation and chocks. Figure 18.4 shows a coupling installed in a reciprocating compressor. Some blades were broken and others have deformed due to the oscillating torque. Figure 18.5 shows the deformed blades.

18.2 Grid Couplings

Grid couplings use a shaped metal stripe installed inside the hub grooves to allow for shaft misalignment. The stripe is loosely fitted into the hubs, which are separated by a short distance. When misaligned, the hub grooves slide back and forth on the metal stripe to provide coupling elasticity. This type of coupling requires lubrication between the stripes and the hubs and a protective cover to keep the lubrication in contact with the grid. The protective cover is displaced easily, resulting in a heavily unbalanced coupling that causes

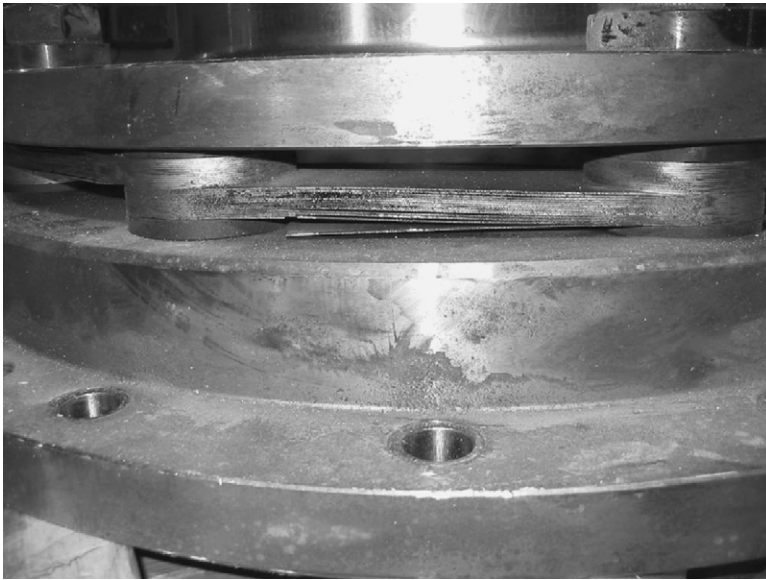


FIGURE 18.4 *Blades fractured by oscillating torque.*

high machine vibration. This problem may be particularly severe for two-pole speed machines.

This was a quite popular coupling in the past. Nowadays, the need for lubrication made disc couplings a better choice. Lack of lubrication can result in accelerated wear of the grid and hub. Excessive misalignment can have the same effect, due to the excessive sliding between the components. The failure appearance is similar. Figure 18.6 shows a coupling grid displaying excessive wear.

18.3 Gear Couplings

Gear couplings are mechanically-flexible couplings that use internal and external gears meshed to allow for misalignment. The gear teeth are barreled to accommodate small misalignments without stress concentration at the edge of the teeth. They can transmit a larger torque better than the preceding types of couplings for the same size.



FIGURE 18.5 *Blades plastically deformed by the oscillating torque.*

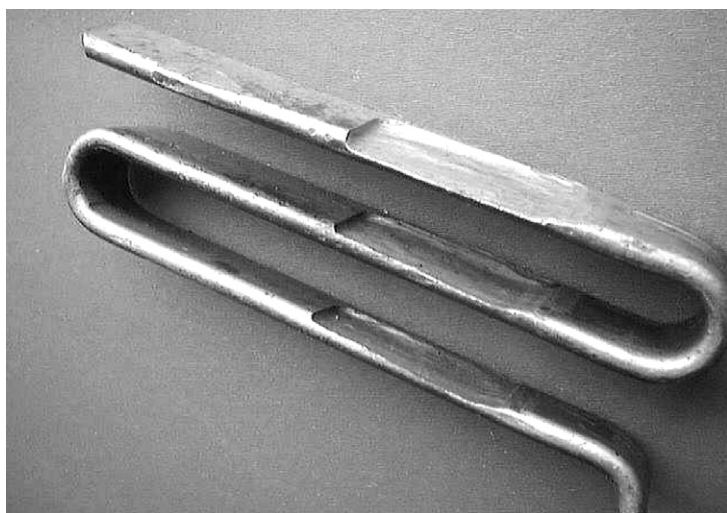


FIGURE 18.6 *Coupling grid displaying excessive wear.*

This has been the first choice for high-speed, high-power machinery in the process industry for a long time, being displaced in recent years by disc couplings and diaphragm couplings, which need no lubrication and cannot lock into position due to gear teeth friction.

The misalignment tolerance of the gear couplings is large, limited only by the internal mechanical clearances. In spite of that, high-speed machinery, with few exceptions, develops very high vibration with coupling misalignment far below the coupling limit. Figure 18.7 shows one such example. This coupling was installed between a gearbox and a 15 MW, 2,600 rpm centrifugal air blower. After disassembly, severe wear of the gear teeth indicated some kind of problem, which proved to be a shaft misalignment with a simple and careful verification. This has been one of those rare occasions when a large misalignment did not result in high shaft vibration. This behavior can be explained by the huge mass of the blower rotor, 11,800 kg, and the low speed. This combination made the

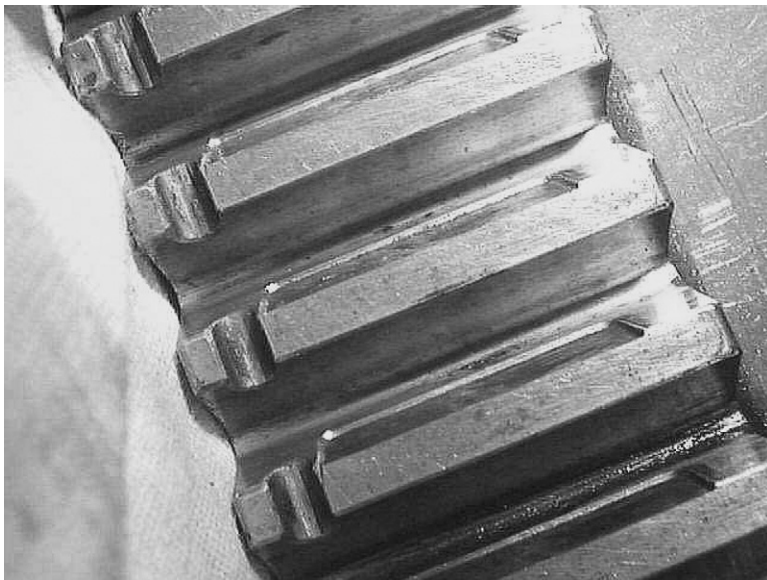


FIGURE 18.7 *Gear coupling of a large centrifugal blower with signs of heavy wear caused by shaft misalignment.*

inertia of the rotor too large for the misalignment forces, completely absorbed by the coupling.

No other problem, like a lubrication deficiency, was found. A lubrication defect can greatly accelerate the gear coupling wear, because contact pressure between the teeth may be large. In addition to accelerating wear, lack of lubrication may cause the teeth to lock in position. This reduces coupling tolerance for misalignment and may increase machine vibration.

Low-speed gear couplings also are affected by improper lubrication. Figure 18.8 shows a gear coupling installed in the transmission of an overhead crane. The coupling teeth display signs of heavy wear, to the point where the teeth edges became sharp. Corrosion covers the coupling hub. This problem was caused by improper lubrication, due to the difficult access to the overhead crane. This type of machine always should be equipped with centralized grease lubrication systems, to be used by the crane operator.

Like any other machine component, couplings are sensitive to installation errors. Figure 18.9 shows a gear coupling with teeth damaged by incorrect installation. The distance between the shaft ends was not set correctly, and the coupling teeth did not engage correctly. Note that the teeth edges were sheared by the local overload from the incorrect engagement.



FIGURE 18.8 *Gear coupling displaying heavy wear due to lack of lubrication.*



FIGURE 18.9 *Gear coupling with sheared teeth due to improper assembly.*

18.4 Coupling Standardization

As mentioned previously, mechanical seal standardization may offer large advantages for couplings, especially for older plants. When a standard coupling is selected for centrifugal pumps, the most common rotating machine in the process industry, the following characteristics should be considered:

1. Nonlubricated coupling.
2. Design according to international standards, like DIN 740 and API 610, if intended for pumps.
3. Minimum spacer length of 200 mm.
4. Flange interlocking device, to avoid loosening the spacer if the blades fail.
5. Suitability for balancing.

Several models of couplings that fit this description are on the market.

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19

Turbomachinery Blades

The blades found inside turbines and compressors interact directly with the fluid to promote energy transfer, the main function of the machine itself. The function of turbomachines is to provide energy transfer between the shaft and the fluid, in either direction.

The design life of the blades is indefinite for almost every type of machine. Two exceptions are the hot gas turboexpander and gas turbines, where blade life is not expected to be indefinite due to the severe operating conditions. The high operating temperatures are the main reason why their life is limited. Turboexpanders also may be subject to abrasion in some services.

19.1 How a Turbomachinery Blade Works

Energy transfer in turbomachines was explained many years ago by Euler, who found out that the torque acting on the rotor is equal to the change in angular moment of the fluid. In other words, energy transfer is possible if forces are acting on the blade and blade movement. The loads that act on a turbomachine blade are

1. Direct fluid loads, resulting from the energy transfer mechanism.

2. Indirect fluid loads, such as turbulence and pressure oscillation close to nozzles and diffusers. These loads may generate high blade vibration.
3. Centrifugal force due to shaft rotation.
4. Other cyclic loads caused by machine vibration, either lateral or torsional.

In addition to these requirements, the blades should be able to stand the following possibilities of damage:

1. Erosion or abrasion due to two-phase flow, cavitation, or hard particles suspended in the fluid.
2. Corrosion due to the fluid or a contaminant action.
3. Incrustation of fluid components or contaminants.
4. High temperature creep.

19.2 Blade Failure Analysis

As usual, blade failure analysis should begin with data collection. The occurrence of damage to the turbomachinery blade is a side effect of the failure of the bearings, and high shaft vibration is not unusual, due to the small clearances and large difference in mechanical strength between blade and casing and the high relative speed.

19.2.1 *Fatigue Fractures*

Blade fracture is seldom related to the energy transfer loads, in most cases, being related to some form of blade resonance. The analysis and simulation of turbomachinery blade vibration is outside the scope of the book, but a simplified description can be helpful.

The main sources of blade vibration are the vibration of the shaft and vibration-inducing mechanisms of the flow, such as surge or stall. However, these phenomena are not powerful enough to damage a blade if resonance does not occur. Therefore, blade resonance may be thought of as the common factor underlying most blade fractures. Of course, there are other fracture mechanisms; some examples are discussed later.

The possibility of blade resonance is investigated with the help of a Campbell, or interference, diagram. In this diagram, the blade's natural frequencies are plotted together with the possible exciting frequencies, so that coincidences can be readily detected. For example, the excitation sources can be machine rotation, gear mesh frequency, nozzle pass frequency, and multiple sources.

Should a coincidence of resonance and excitation frequency be found in the operating speed range, there is risk of high blade vibration. The usual procedure is to tune the blades out of this frequency or remove the excitation source. If this is impossible, the stress acting on the blade may be calculated by considering the effect of resonance and evaluating the situation. Figure 19.1 illustrates such a Campbell diagram.

There are several ways to influence the blades natural frequency and avoid resonance. Large steam turbine blades usually are tuned out of the dangerous range, after the actual natural frequency is measured. Smaller blades may be connected with wires, which provide some damping. Damping greatly reduces vibration in case of resonance. Another possibility is to add a closing strip to the top of a blade row, so that the blades vibrate as a group. Vibration

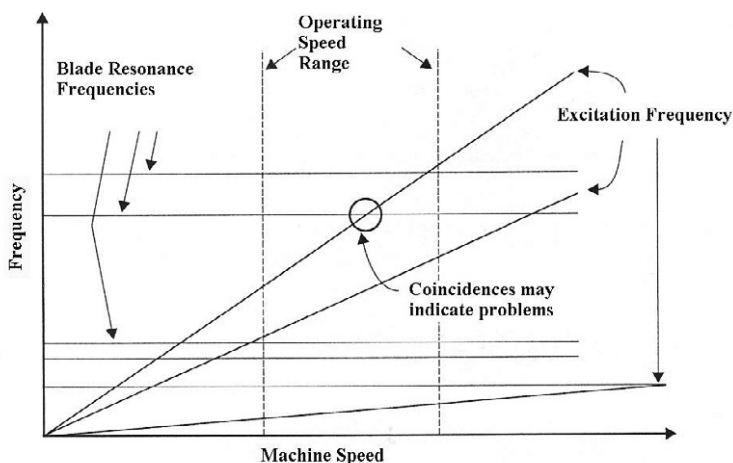


FIGURE 19.1 *Campbell diagram of blade vibration.*

modes and frequencies are greatly influenced by this technique. The vibration modes of a row of blades are illustrated in Figure 19.2.

Figure 19.3 shows the rotor of a condensing steam turbine that operated for approximately 20 years without problem, until one blade of one of the last stages suffered a fracture. This happened suddenly, without warning. Of course, no one would expect that this could happen. Figure 19.4 shows the fracture face. The fracture face features are not clearly seen, but an examination with a microscope showed the failure mode to be fatigue.

A fatigue fracture that happens after so long, obviously, is not related to the normal operating conditions of the machine. Something changed during the operating life of the turbine.

Figure 19.5 shows the erosion damage found on the shroud rivet of the failed blade. This damage cannot be explained easily, as this region should have been protected from erosion by the shroud. The origin of this erosion was investigated in detail, but no firm conclu-

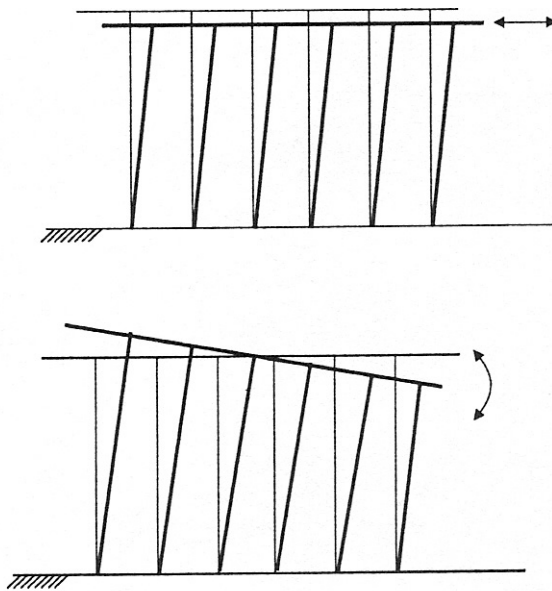


FIGURE 19.2 Two vibration modes of a group of blades connected by a shroud.

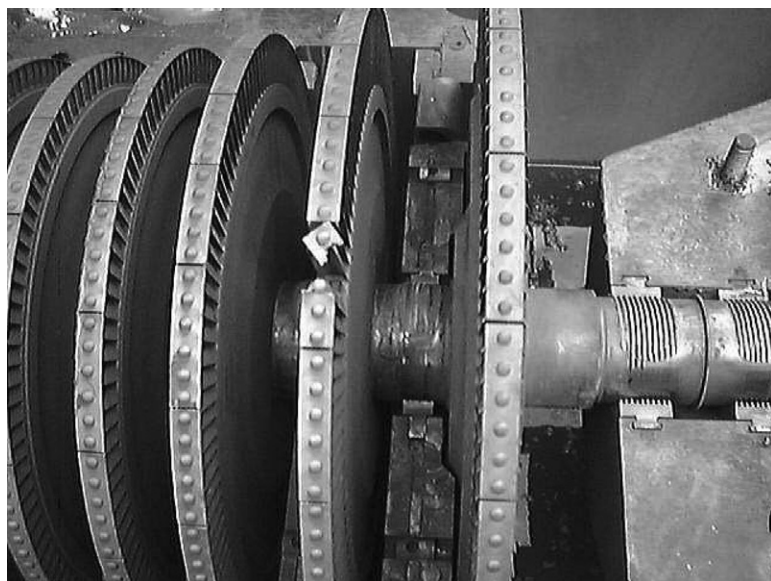


FIGURE 19.3 *Steam turbine rotor that lost a blade in operation.*

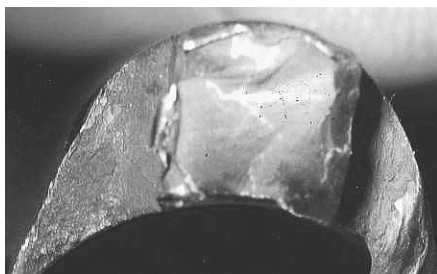


FIGURE 19.4 *Fracture surface at the blade root.*

sion was possible. The most probable cause of this erosion is the operation of the turbine with lower than specified inlet steam temperature, which resulted in heavier condensation in the last stages. Although this could have happened, the cause could not explain why the erosion damage was found only on the broken blade.

The proposed explanation for the event is the following. Erosion of the shroud rivet by steam condensate loosened the connection of



FIGURE 19.5 *Signs of erosion found at the shroud rivet on the failed blade.*

the shroud to the blade. When the blade lost its connection with the shroud, its resonance frequency changed and an unknown excitation force, possibly turbulence due to the steam flow, caused the blade to resonate and fail due to fatigue.

A quite different situation is shown in Figure 19.6. This figure shows the rotor of an axial blower damaged by ingestion of an inlet guide vane (IGV), which failed due to a fatigue fracture. The failure occurred shortly after the installation of the blower. The machine had been successfully tested at the factory before the field installation.

It is believed that the flow of air excited the IGV resonance frequencies due to localized stall. This is an aerodynamic phenomenon, where loss of continuity in the flow lines results in the formation of



FIGURE 19.6 *Axial blower rotor damaged from the fracture of the IGV blades.*

vortices, flow reversal, and turbulence. Figure 19.7 shows the fracture surface of some of the blades, which display discrete indications of the failure mode, a fatigue fracture.

The blades shown in Figure 19.7 vibrated in the first mode. Figure 19.8 shows other IGV blades, not yet broken but already cracked. The position of the cracks indicates that they vibrated in a different mode, which corresponds to a different frequency. The excitation in several frequencies is characteristic of random excitations, like turbulence.

19.2.2 Erosion, Abrasion, and Foreign Object Damage

These are different failure modes with one thing in common: They usually are caused by material that should not be in the fluid path. Some examples are excessive moisture in the exhaust section of condensing steam turbines, a catalyst entering a hot gas expander, and air impurities inhaled by a gas turbine.



FIGURE 19.7 *Fracture surface of the IGV blades, displaying discrete evidence of the fatigue failure.*

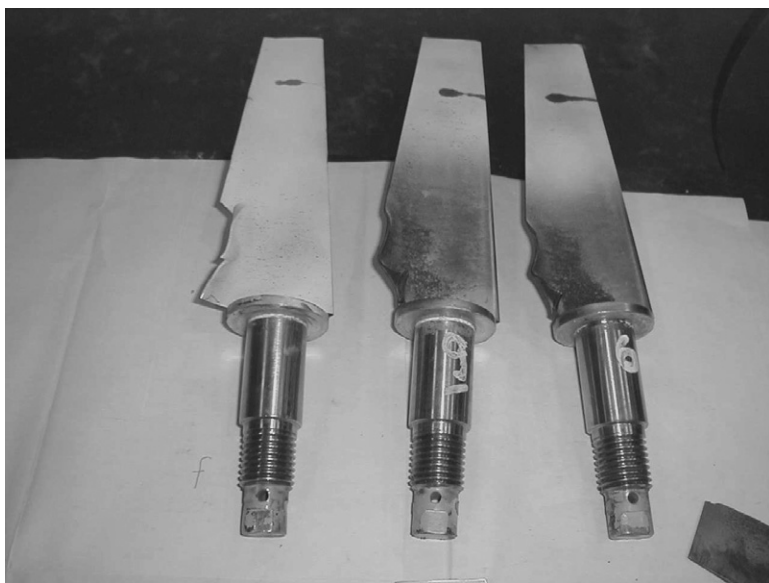


FIGURE 19.8 *Other IGV blades of the same compressor showing cracks at different positions than the ones shown in Figure 19.7. This indicates that these blades vibrated at another frequency.*

Erosion and abrasion normally are time related. Material is removed progressively from the surface, modifying the original shape of the blade. The energy transfer depends on the change of fluid momentum along the rotor; if the blade shape is altered, the direction of fluid flow and the change in momentum are modified. The exact modification on the blade shape depends on the angles of incidence and relative speed of the material causing wear.

A hot gas expander blade that suffered abrasion due to contact with some catalyst is shown in Figure 19.9. The catalyst that caused the material removal is a very fine alumina-based powder. It is possible to note that most of the wear is concentrated on the leading edge, due to the incidence angle of the particles, and at about two-thirds of the way to the trailing edge, where high relative speeds can



FIGURE 19.9 Hot gas turboexpander blade with signs of wear due to the impact of hard catalyst particles.

be found. This type of problem may be caused by problems on the catalyst separation system of the unit.

The last stage blades of a condensing steam turbine are well known candidates to suffer erosion. Being a well known problem, some field proven solutions are on the market. Figure 19.10 shows last-stage blades of a condensing steam turbine displaying signs of erosion. It is possible to note that the blade tips are hard faced and not affected. Steam quality contributes to this failure mode. Reduction in steam quality can be caused by a reduction in the live steam superheat.

If a small reduction in steam superheat may cause an increase in the moisture content in the last stages of the turbine, large reduc-

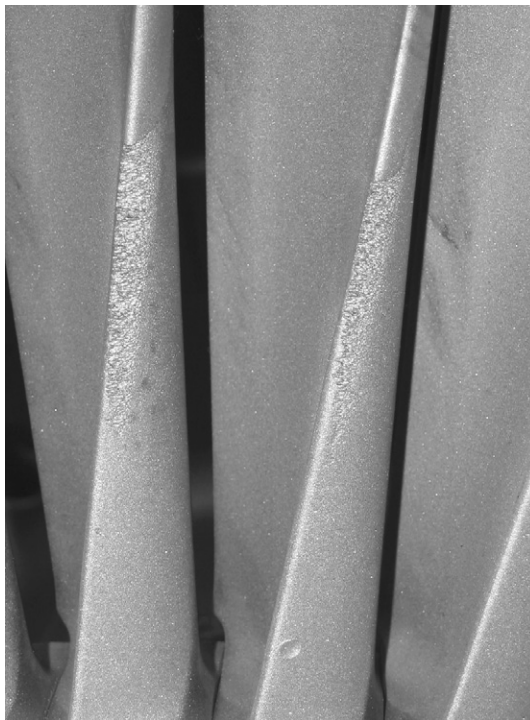


FIGURE 19.10 *Erosion of last-stage condensation steam turbine blades due to high moisture content in the steam.*

tions may cause water ingestion. Obviously, this is much more destructive to the turbine than an increase in the moisture quantity.

In addition to thermal chocks, the entrance of water in the turbine introduces loads much higher than with steam operation, which normally results in turbine damage.

Figure 19.11 shows a steam turbine that operates intermittently. The steam piping needs draining before each startup, despite steam traps on the piping. This draining operation was not done correctly, and water ingestion resulted. The nozzle ring and the first blade row are damaged. Several bolts installed in steam path components were fractured on this occasion.

A simple modification that reduces the chance of an event like this is the installation of steam traps that can keep the live steam piping hot at all the times. This can be a problem, especially for units where long steam piping feeds the turbine.



FIGURE 19.11 *Steam turbine damaged by water ingestion.*

Large foreign particles cause damage from their impact against the blades. This problem may be related to inlet filter problems, improper piping cleaning, or other maintenance problems. Figure 19.12 shows a gas turbine blade that displays indentations caused by the impact of a foreign particle. In this case, the damage was not large enough to jeopardize the operation of the machine.

An important characteristic of the particle movement inside the machine is that the particle speed is smaller than the fluid flow. The larger the particle, the smaller its speed. This means that the angle of incidence of the particle on the blade is not the same as that of the fluid. This should be taken into account when a failure is analyzed.

The “foreign” particle may not be something that enters the machine but the result of the failure of another component. Figure 19.13 shows an example of this situation, where the first rows of blades of the low-pressure casing of a steam turbine were damaged by a fractured bolt. This bolt was loose on the steam inlet, impacting the blades at will.

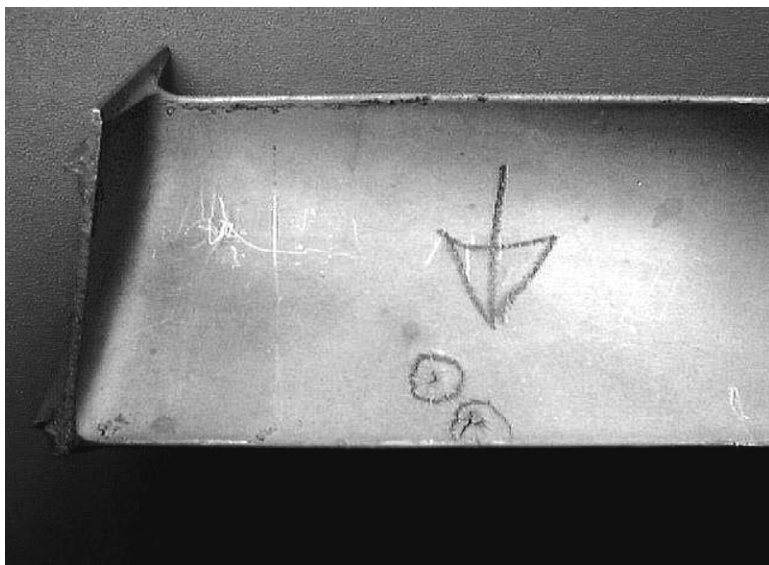


FIGURE 19.12 *Gas turbine blade displaying damage from foreign particle impact.*

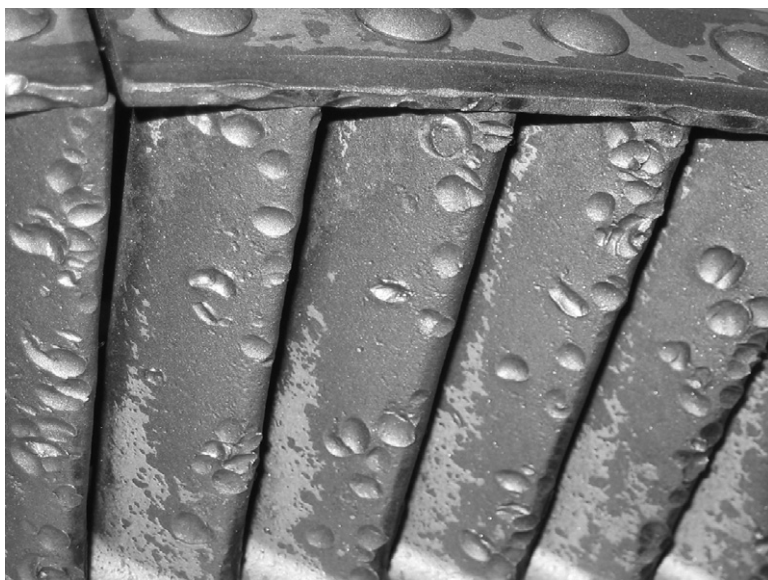


FIGURE 19.13 *Steam turbine blades damaged by a broken bolt.*

Analysis of blade corrosion problems can be quite complicated. Air compressors may be subject to electrochemical corrosion from contaminants in the air, like chlorides. Machines that work with atmospheric air usually have filters to deal with the specific site air quality. Industrial and marine environments can be the most corrosive, and filters should be maintained carefully.

Process compressors may be subject to corrosion from the action of the fluid or contaminants. A complete analysis of blade corrosion is clearly outside the scope of this book.

Figure 19.14 shows a hot gas expander rotor with blade failure. These blades were coated with an abrasion-resistant coating, as abrasives were expected on the fluid stream. The coating was not impervious and did not protect the blades from the contaminants in the gas flow.

The working fluid was the effluent gas from a catalytic cracker. This gas contains a certain amount of sulfur; the blades are made of a high nickel alloy. Sulfur and nickel form a well-known, low-



FIGURE 19.14 *Hot gas expander rotor that suffered a blade failure due to the formation of nickel sulfide.*

temperature eutectic, nickel sulfide. The formation of nickel sulfide caused the blade failure.

A failure like this usually is analyzed with the help of the machine manufacturer, who also played an important role in determining the modifications to be introduced to the system. This problem was solved by the introduction of a modified blade coating, which was resistant to not only abrasion but also corrosion.

Machines that operate at a high temperature may be subject to some kinds of thermal damage. Creep and thermal fatigue are failure modes that affect the components if the adequate precautions are not implemented.

Figure 19.15 shows the surface of a hot expander blade damaged by thermal fatigue. The superficial cracks are the typical indications of this problem. Improper control of the cooling steam flow to the machine is believed to have caused the thermal chocks observed. Hot gas expanders use steam to keep the shaft seal area at a lower temperature than the blades. When the machine is shut down, the flow of steam has to be reduced or blade cooling may be too fast.



FIGURE 19.15 *Thermal fatigue on a hot gas expander blade surface caused by improper control of the cooling steam.*

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Part IV

Case Studies

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20

Failure Analysis Examples

This chapter contains several case histories of machinery failures. The analyses are summarized and intended to give the reader some information about how machinery failure analysis is done in the real world. The most important point here is that, in addition to knowledge of the failure modes, the failure analyst must have in-depth knowledge of how the machines and components work. The main objective of these examples is to illustrate the application of the concepts discussed throughout the book.

20.1 A Pump Fire Caused by Turbine Driver Overspeed

Turbine overspeed can easily destroy the driven equipment. This case history describes the analysis of one such event, when a pump destruction and subsequent fire were caused by a failure of the turbine speed control and overspeed protection systems.

Many small turbines use a hydraulic governor directly connected to the shaft and a mechanical overspeed trip device that consists in a small mass-spring system positioned close, but not at, the shaft centerline. Figures 20.1 and 20.2 illustrate the overspeed device and the governor connection to the shaft.

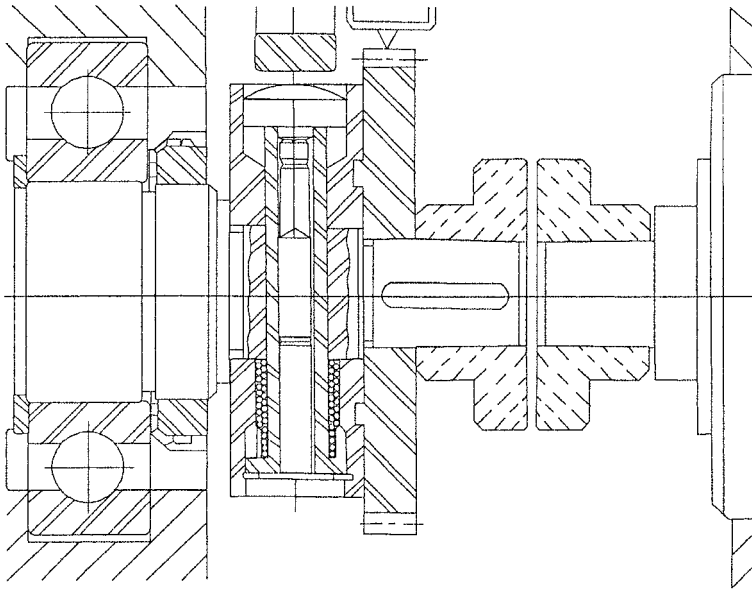


FIGURE 20.1 *The overspeed device and connection of the governor to the shaft.*

This history begins in the middle of an afternoon, when a power plant DCS (distributed control system) warned the control room operator that the discharge pressure of the centrifugal pump feeding fuel oil to a boiler had increased suddenly. The field operator was asked to investigate, and he found that the turbine speed was 4,100 rpm, much higher than normal, 3,600 rpm. The sudden increase in the turbine speed was the cause of the unexpected increase in the discharge pressure of the fuel oil pump. An attempt to reset the governor did not solve the problem, and it was concluded that the governor should be sent to the workshop for repair. By this time, the end of the shift was approaching quickly, and it was agreed by the operator and the mechanic that the pump would continue running and the governor would be removed the next morning. The motor driven spare pump was ready to operate.

By the middle of the night, under heavy rain, the alarms began to sound again, as the oil pressure suddenly dropped to almost zero.

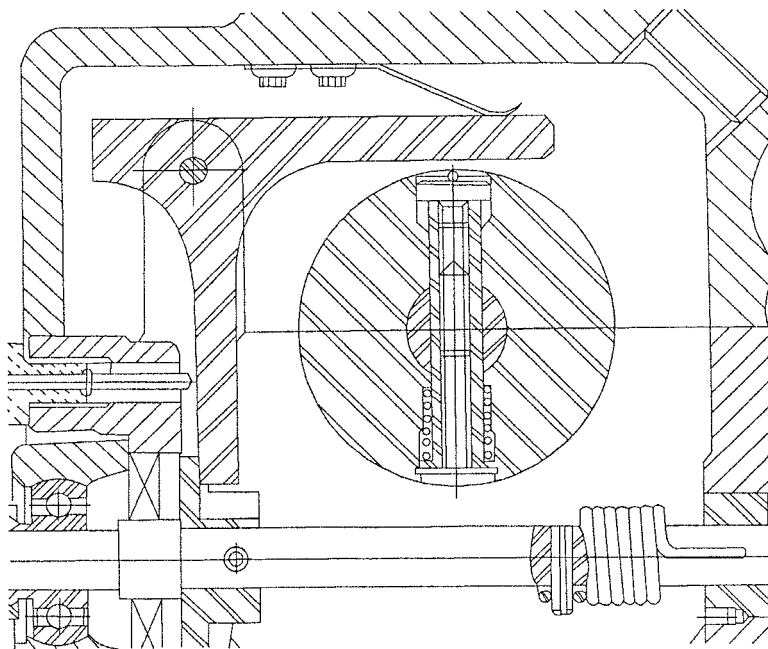


FIGURE 20.2 *Front view of the overspeed device.*

The spare pump was automatically started up, but this did not restore fuel oil pressure. Soon after, a fire alarm indicated that the problem was growing. A fire had begun at the pump station.

The whole system was shut down, and firefighters extinguished the blaze promptly. After a preliminary cleaning, the pump and piping were found to be severely damaged, causing a huge hot fuel oil leak that started the fire.

Pump inspection showed that turbine overspeed caused the accident. Little damage was found on the turbine, aside from shearing of the shaft tip close to the governor, making the governor and overspeed trip device useless. Turbine bearings were not greatly affected. Figure 20.3 shows the sheared shaft stub, the governor coupling, and the overspeed mass as found inside the bearing housing. Figure 20.4 shows the overspeed mechanism disc, also found



FIGURE 20.3 *Overspeed mass and governor coupling damaged after the accident.*



FIGURE 20.4 *Overspeed system disc displaying wear on the inside diameter.*

inside the bearing housing. Note that the heavily worn inside diameter of the disc was originally assembled on the sheared portion of the shaft, as shown in Figure 20.1.

Figure 20.5 shows the plot of the pump discharge pressure and of the deaerator pressure. This deaerator is part of the steam generation system, its pressure closely approximates the turbine exhaust pressure. It is possible to note that pump pressure was constant, despite some variations in turbine exhaust pressure until the middle of the period indicated on the plot, a clear indication that the turbine governor was controlling the pump speed. After the middle of the plot, pump discharge pressure displays variations that are the opposite of the turbine exhaust pressure, indicating that the governor no longer was controlling turbine speed. The time when the pump discharge pressure changed was the time when the DCS sounded an alarm indicating the increase in pump pressure, in the middle of the afternoon. This is when the turbine shaft was sheared and the governor lost control of the turbine, as it was no longer connected. Turbine speed became controlled by the available enthalpy drop of the steam, which is a direct function of the deaerator pressure. Pump discharge pressure is directly related to turbine speed.

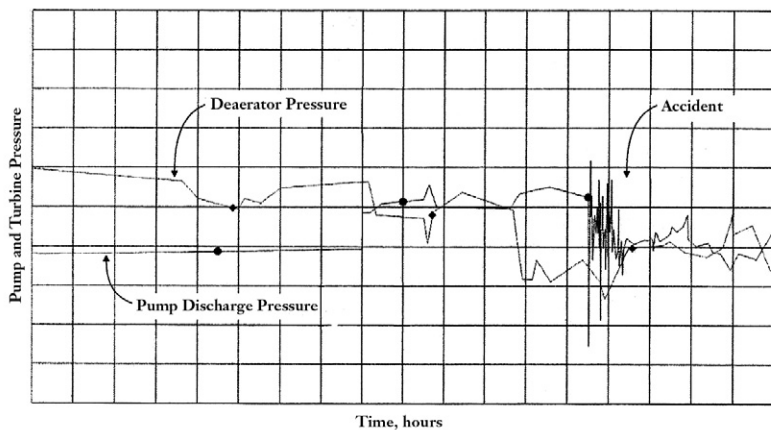


FIGURE 20.5 *Pump discharge pressure and turbine exhaust pressure a few hours before and after the accident.*

The turbine shaft was sheared by the sliding wear caused by the relative movement between the shaft and the overspeed disc. The evidence supporting this is the heavy wear found on both interfaces of the two parts, as shown in the pictures. A very different problem is to explain how this could happen, as Figures 20.1 and 20.2 show that these parts should rotate together and the overspeed mass should not allow relative movement between them.

Observing the overspeed mass in Figure 20.3, we note that the mass had been bent. The quick closing valve spring, shown on the left in Figure 20.2, displays a tiny indication that it suffered impact. When the parts were assembled on the turbine, it was found that the deformed shape of the overspeed mass fit perfectly on the space between the disc and the spring.

This means that the force that held the overspeed disc and made rubbing of the shaft possible was caused by the position of the overspeed mass, stuck between the disc and the spring. Unfortunately, the size of the bolt that holds the mass inside the disc is small enough to allow the mass to go out of position in the disc without touching the trip trigger.

We believe that the following sequence of events led to the accident:

1. The overspeed mass bolt unscrewed due to vibration, because it had not been locked in place, as required in the turbine maintenance manual.
2. The overspeed mass bolt left the disc. When leaving the disc, unfortunately in a direction that did not touch the trip trigger, it hit the steam valve stem spring, causing the disc to stop rotating.
3. The turbine shaft was sheared by the sliding wear with the disc, disconnecting the governor and the overspeed trip device. At this time, the governor lost control of the turbine and pump discharge pressure increased. Turbine speed became controlled by the available enthalpy drop.
4. In the middle of the night, the heavy rain caused a temperature and pressure reduction of the low-pressure steam system, and

the turbine accelerated to a speed high enough to destroy the pump.

The causes of the accident are

1. A maintenance problem, as the overspeed mass bolt was not locked with Loc-Tite, as instructed in the turbine maintenance manual.
2. A design problem in the overspeed trip. The construction should not allow this type of event without tripping the turbine. Figure 20.6 shows a trip system mass that has a longer bolt, long enough to trip the machine before falling out of position.



FIGURE 20.6 Comparison between the damaged (right) and a new (left) overspeed mass. Note the difference in bolt size of the new component.

3. An operation error, as the pump was allowed to continue running even after the governor problem was evident. Even if the spare pump was not available, when the governor problem was found, the decision to keep a defective machine in operation should not be taken before doing some sort of risk analysis.

The actions to avoid repetition of this problem are

1. Instruct maintenance personnel to follow the manufacturer's instructions strictly.
2. Instruct maintenance and operation personnel that defective machines should not be kept in operation without a proper risk analysis.
3. Change the design of similar trip systems, so that the trip mass could not leave its place without touching the trip trigger.

In this example, an in-depth knowledge of the machine construction and operation was more important than complicated metallurgical analysis. The relationship between the damage suffered by each component and the way the machine works revealed more about the root causes of the failure than a detailed analysis of the wear damage, for example. Also note that the damage to the pump and piping were not analyzed, as they were merely consequences of the initial turbine failure.

One feature of this example is the importance of the small details on the overall history. The bolt size, the impact damage on the spring, and other small damage were important in explaining the accident.

The concepts illustrated in this example are

1. Multiple root causes require action by multiple layers of the organization and on the machine itself.
2. The operation history of the machine and the operating conditions at the moment of the accident are important.
3. Consider the operating mode of each component during failure analysis.

4. Maintain a focus on the first component to fail. Note that the failures of the other components were not analyzed in detail.

20.2 Hydrogen Compressor Seal Failure

The example tells the history of a centrifugal compressor seal failure. This is a hydrogen recycle compressor equipped with conventional oil film seals operating at about 80 kgf/cm². The observed symptoms are

1. High leakage of oil past the internal seal.
2. The differential pressure between oil and gas was approximately half the design value.
3. Fast oscillations in the oil–gas differential pressure.

An elevated oil tank was responsible for the control of the differential pressure. The oil level in this tank was normal, and no oscillation had been observed. The piping connecting the pressure differential transmitter to the oil and gas lines were drained and some oil was present.

Figure 20.7 shows the seal rings. Oil is injected in between the rings with a pressure slightly higher than sealed gas pressure. The elevated oil tank gas space is connected to the inside of the compressor, so that this tank is kept at the same pressure as the compressor sealing chamber. The tank is fed through an oil pump and a level control is installed to keep tank level and, consequently, the differential pressure constant.

Vibration of the compressor was high at some times. A vibration analysis showed the most probable cause of this high vibration was an internal rubbing. The origin of the rubbing was not identified at that time.

Once the oil leakage was high, the compressor was shut down and the seals replaced. The seals displayed indications of heavy rubbing, as shown in Figure 20.8. This was the origin of the high vibration observed from time to time. Some signs of fretting were observed on the seal cover, at the place where the seal ring was supported.

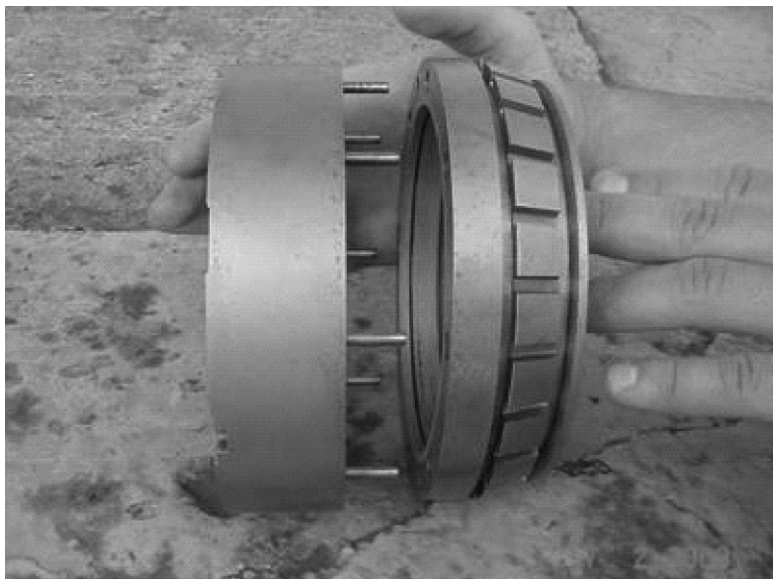


FIGURE 20.7 *Oil seal rings.*

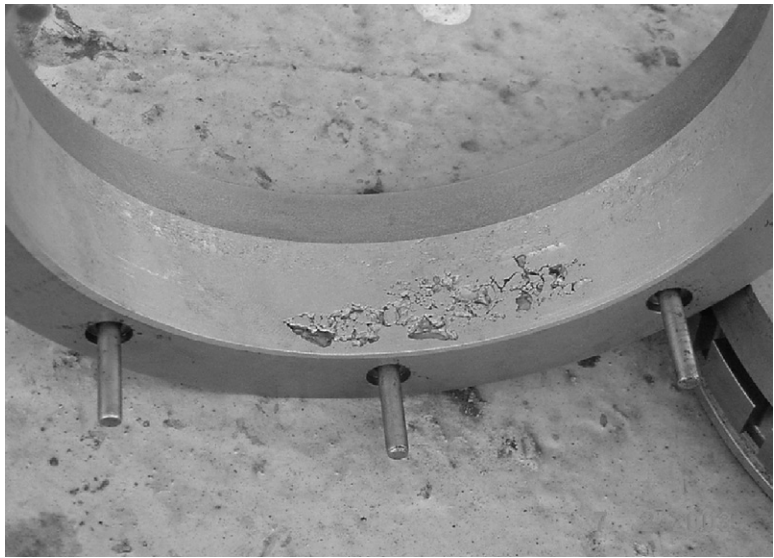


FIGURE 20.8 *Rubbing at the internal surface of the seal ring.*

An inspection of the compressor revealed that the gas pressure reference line, which should have been installed without pockets, had a low point close to the compressor, shown in Figure 20.9. This low spot allowed oil to accumulate, making the differential pressure less than desired. It was estimated that a 2 or 3 m oil column inside the pipe would be enough to reduce the differential pressure to the observed value. The oscillations in the differential pressure with no observable variation of the tank level can be explained by the fact that the pipe diameter is much smaller than the tank diameter: a large difference in the oil level on the line corresponds to a small difference on the tank level.

The reduction in the oil–gas differential pressure, caused by the accumulation of oil inside the reference gas line, made the lubrication conditions of the seal ring inadequate, and rubbing resulted. Rubbing increased the clearance and the oil leakage, the first observed symptom.

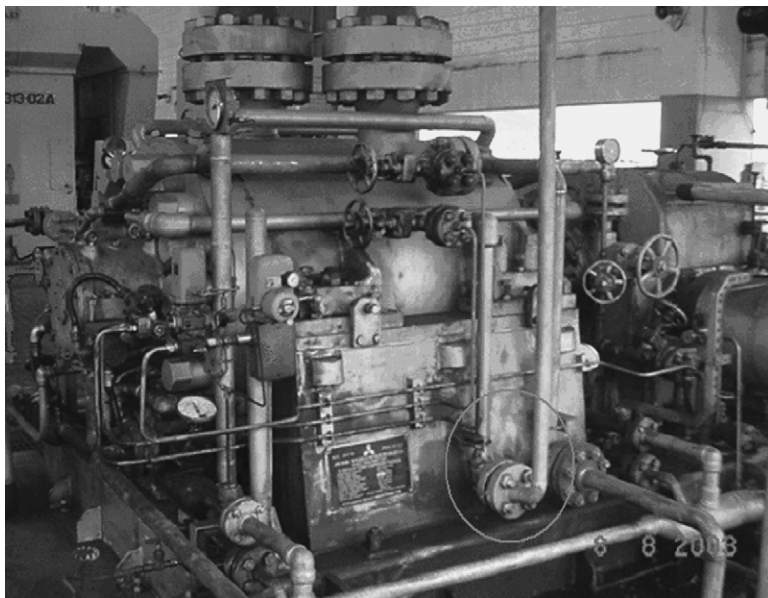


FIGURE 20.9 *A low spot on the seal gas reference line.*

Note that this compressor had operated for about five years before the problem was observed. Although the installation mistake was there from the beginning, the effect on the pressure differential was possible only after the gas pressure reference line filled with oil. The source of the oil found inside the reference gas line has not been identified, the most probable origins for this oil are loss of control of the overhead tank level or oil accumulation inside the compressor during a shutdown period.

20.3 Vibration-induced Fatigue Failures of Identical Reciprocating Compressors

Critical service reciprocating compressors usually are spared, as the reliability of one compressor is usually smaller than the reliability required for a process plant. The underlying idea is that if one compressor fails, the other is available. However, if the compressors suffer from a common problem, the plant may be shut down.

This third example describes the history of two newly installed reciprocating compressors that suffered identical failures due to a design deficiency, causing a long plant shutdown a few weeks after initial startup, before the compressors could be put back to service.

The compressors were two newly installed reciprocating compressors, operating at 514 rpm, with discharge pressure of 87.9 bar. These are throw machines feeding hydrogen to a hydrotreater, a common refinery service. Both compressors suffered crankshaft fractures after about a week of operation. Figures 20.10, 20.11, and 20.12 show the situation found when the machine frames were opened after the fracture. Although it is easy to note that the failure mode was a fatigue fracture perpendicular to the shaft, the root causes of the failure were not obvious. The fracture shape indicates that bending loads caused the fracture.

Some possible causes for a bending overload of the shaft are a design error, if the shaft is not strong enough to stand all the operating loads; axial or lateral resonance; an installation problem that resulted in high web deflection; an overload due to incorrect operating conditions; and a grossly unbalanced flywheel.

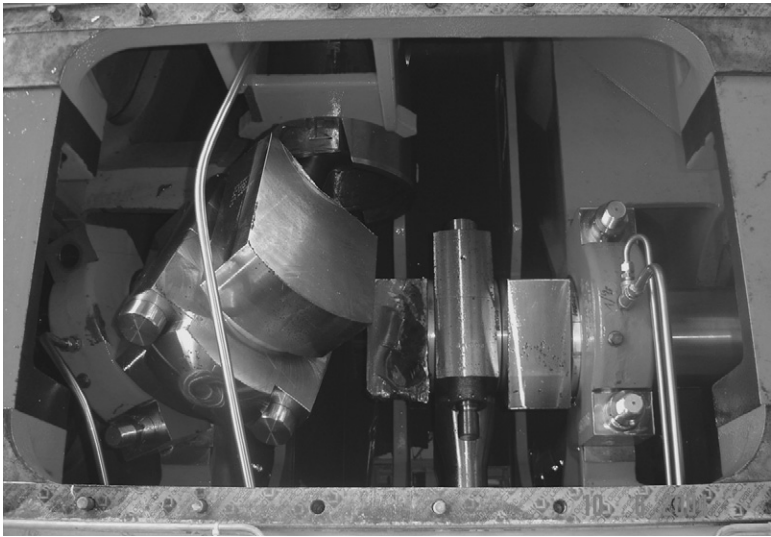


FIGURE 20.10 *View of the compressor frame with the broken shaft.*



FIGURE 20.11 *View of the shaft after removal from the machine.*



FIGURE 20.12 *Crankshaft fracture face. Note the ratchet marks on the right side, indicating a high stress concentration and several fracture initiation spots.*

The manufacturing, installation, and operation records were carefully checked and no abnormality found. All the manufacturing inspections, including mechanical test run reports, were reviewed, with no deviation from the design specifications. Installation inspection revealed nothing wrong, web deflection was within design limits, the machine base was in good conditions. All the operation parameters were within the design.

The shaft was sent to a metallurgical laboratory for analysis, and no material defects or deficiencies were found. The material mechanical properties were within the design limits, no material defect was found.

The compressor manufacturer verified the design of the crankshaft, using a finite element (FEA) model, shown in Figure 20.13.

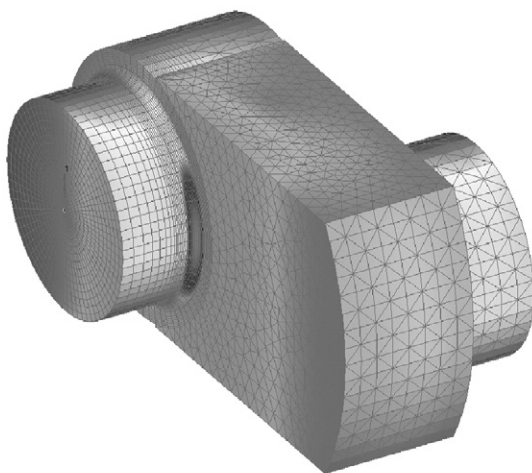


FIGURE 20.13 *Finite element model used to calculate the stresses acting on the shaft.*

The results of the FEA model were checked with the help of a stress concentration measurement with strain gauges, shown in Figure 20.14. The shaft was found to be correctly designed to stand the normal operating loads, with cyclic stresses well within the allowable limits of the material.

The resonance frequencies of the shaft were calculated and no axial or torsional resonances found. Although it is fairly easy to calculate the axial and torsional resonance frequencies, once the finite element model is built, the lateral resonance calculations are another type of problem.

The bearing stiffness and damping does not influence the axial and torsional modes but have a huge influence on the lateral vibration. Unlike centrifugal compressors or steam turbines, where a great deal of research has been done on the bearings calculations, reciprocating compressor bearing modeling is more an art than a science, nowadays. The main difficulty when modeling reciprocating compressor bearings is that the operating loads vary greatly in



FIGURE 20.14 *Measurements of the stress concentration with strain gauges.*

direction and magnitude. Bearing characteristics are load dependent, so the problem seems difficult to solve.

The lateral resonances were calculated using bearing characteristics derived from the manufacturer experience. The calculated first lateral critical speed was 17 Hz, quite close to two times the running speed. A reciprocating compressor always has a high excitation at this frequency, because of the piston movement. A double effect machine adds the gas load to the piston inertia, and the result is a high excitation at twice the running speed.

Figure 20.15 shows the mode shape for the first critical speed. The high amplitude at the coupling side of the compressor shaft is

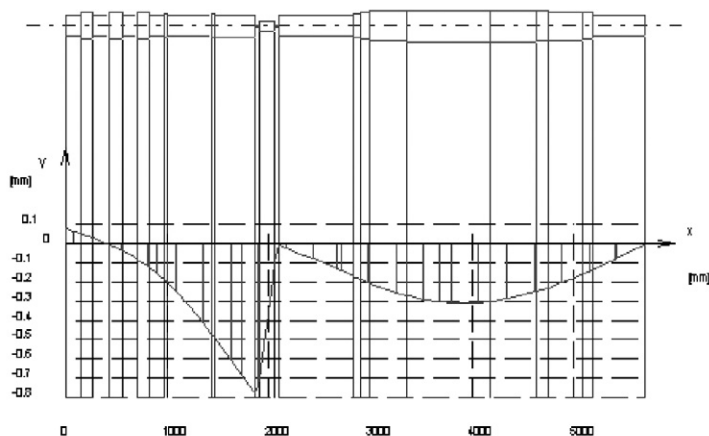


FIGURE 20.15 Mode shape for the first critical speed.

due to the heavy flywheel. Flywheel weight is significant, as a large inertia is necessary to reduce the torque oscillations caused by the normal operation of the compressor.

After this analysis, we found that the most probable cause for shaft fracture is lateral resonance. This is quite unusual, especially with this particular frame, which was a manufacturer standard with dozens of units sold and operating under similar conditions.

An unusual feature of these compressors is that they are fitted with flexible disc couplings. Such couplings are widely used in process plant turbomachinery, but in this case, some caution is necessary. Usually, the driver inertia is accounted for when sizing the flywheel, if a rigid coupling is used. As these particular machines used flexible couplings, the flywheel was larger than usual. The heavier flywheel reduces the lateral resonance frequencies of the shaft, the same effect of the flexible coupling, because of the reduced shaft stiffness. This is believed to be the reason underlying the whole problem, as the manufacturer had no experience with the flexible disc couplings for this type machine. In other words, the flexible coupling introduced the need for a larger flywheel and reduced the shaft stiffness, reducing the lateral critical speeds of the crankshaft.

Once it was found that the fracture had been caused by lateral resonance, a simple solution for the problem became evident: reduction of flywheel mass retaining the original rotating inertia. This is done easily by increasing the flywheel radius, as rotating inertia is the combination of mass and radius. The reduction of flywheel weight was used to tune the first lateral critical speed out of the dangerous range, in this case, twice the running speed.

After this analysis, the machines were fitted with modified flywheels and instrumented for a new startup. Shaft and rod stresses were measured with strain gauges and flywheel behavior was monitored with accelerometers. The machines started up again, and it was found that shaft stresses were close to the calculated values, well within the limits of the shaft material, and the actual resonance frequencies were outside the dangerous range, although the calculation could not predict it as precisely as the shaft stresses.

As I write this, the machines have been running for 16 months with no further problems. It is important to note that the assistance from the compressor manufacturer was of enormous value in solving this problem. Although clearly a design problem, the machine owner also did everything within his reach to help the manufacturer solve the problem.

Many machine manufactures, when facing a problem like this, look for a way to be relieved from the responsibility, but this is the worst possible way to behave. The owner, sometimes, tries to prove that the vendor is guilty and adopts a less than optimum posture on the accident. Once a problem arises, the only reasonable way to face it is for the owner and vendor to cooperate in the search for an adequate solution.

20.4 A Gearbox Failure from the Electric Discharge Damage of a Bearing

A step-up gearbox driving a centrifugal compressor was found one morning, sounding like stones were being thrown by the gears inside the casing. Quite wisely, the maintenance supervisor told the operators to shut down the machine and decided to open the gear-

box. This decision required a huge fertilizer plant to interrupt production, as the compressor is a critical machine with no spares.

When the gearbox cover was removed, some of the gear teeth were found broken. Figure 20.16 shows the broken gear teeth found on this occasion. This machine is a two-casing turbine driven centrifugal compressor with a step-up gear in between the two compressor casings.

A closer view of the fracture is shown in Figure 20.17, where a brittle fracture is clearly identified. The operating conditions of the compressor were modified not long before, and it was suspected that the gears were overloaded. Although the teeth fracture showed clearly that there had been a concentration of load on the fracture side, indicating shaft misalignment, the power absorbed by the compressor and the teeth stresses were calculated. As expected, the calculations show that the gears were not overloaded.

Inspection of the other gear components revealed that the pinion bearing had suffered heavy rubbing and the coupling had signs



FIGURE 20.16 *Broken gear teeth on the step-up gear.*



FIGURE 20.17 *Closer view of the fracture on the edges of the gear teeth.*

of electrical discharge damage between the teeth. Figure 20.18 shows the pinion bearing rubbing indications, and Figure 20.19 shows the coupling teeth with signs of electrical damage. When the indications of electric damage were found on the coupling, a portable Gauss meter was used to measure the residual magnetism of the gear and compressor components.

It was found that the pinion shaft was magnetized to 16 gauss, a much higher residual magnetism than allowed by the American Petroleum Institute, which specifies that no part of the machine should be allowed to have more than 5 gauss residual magnetism. Some authors require that rotating components be demagnetized to less than 2 gauss residual magnetism.

The oil system was inspected and no problems found. No indications of any problem were found on the other bearings. Shaft vibration and bearing temperature were normal before the start of the loss of oil film, a few hours before the failure. The proximity probe installed in the gearbox detected that the shaft began to move toward the bearing surface approximately eight hours before the



FIGURE 20.18 *Pinion bearing displaying heavy rubbing.*



FIGURE 20.19 *Gear coupling displaying signs of electrical discharge damage.*

accident. This became evident through the analysis of the dc voltage indication, that changed from -9 V , approximately the value measured in the past, to -6 V .

The maintenance records of the machine recorded that a rotor grounding brush had been removed about one year before the event. The grounding brush was installed at the far side of the high-pressure compressor shaft, and any current generated at the gearbox would flow through the coupling to be grounded by the brush. This explains the electric discharge damage found on the coupling.

When the brush was removed from the high-pressure compressor, the current generated by the magnetized shaft was discharged by the pinion bearing, the least resistant path. The pinion bearing was damaged to the point that the oil film was lost and the bearing failed, causing the misalignment of the gear shafts. Unfortunately, bearing damage was extensive enough to remove any evidence of the electrical discharge damage. The assumption that electric discharge caused by the pinion shaft magnetism caused the initial bearing damage is based on indirect evidence.

The magnetized pinion was installed approximately three years before the failure. It was not possible to discover how the shaft became magnetized. Although the literature indicates that shaft magnetization is possible with heavy rubbing and no sign of electric discharge damage was found on the bearing, it is believed that this is the most probable cause for the failure. The residual magnetism of all the accessible components of the train were measured and, as no indication of magnetized parts were found, the machine was started up after replacement of the damaged components. A few years later, the machine was completely disassembled in a scheduled shutdown and no residual magnetism or similar failures were found. This means that the assumptions made at the time of the failure was right.

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