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A Simple Guide to Understanding Compressors

James M. Watterson



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DR. JAMES M. WATTERSON



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First published in 2018 by
Momentum Press[®], LLC
222 East 46th Street, New York, NY 10017
www.momentumpress.net

ISBN-13: 978-1-94708-370-7 (print)
ISBN-13: 978-1-94708-371-4 (e-book)

Momentum Press Engineering Technology Collection

Cover and interior design by S4Carlisle Publishing Service Private Ltd.
Chennai, India

First edition: 2018

10 9 8 7 6 5 4 3 2 1

Printed in the United States of America

ABSTRACT

This straightforward guide to compressors seeks to unveil a lot of myths surrounding compressors. In this book, we will be looking at most types of compressors, including the centrifugal compressors, the air compressors, and of course the most troublesome of all compressors, the reciprocating compressors.

Having a compressor with minimal operating problems does not only depend on the selection of the right type and size for your job. Detailed specifications of all auxiliary equipment and operating conditions, as well as keeping constant vigilance over the engineering and installation is imperative. The *Simple Guide* will explain in a simple yet definitive manner which compressor type is best used for which job and what it can produce.

Dr. Watterson is not a pure academic and often works at the “Coal Face” during plant shutdowns and turnarounds, and commissioning. His experience spans across many continents and varying industries. Compressors is one of the subjects he has trained with many refineries across the Middle East and Africa.

He has also published “diagnostic skills” and aid to troubleshooting with compressors, pumps, and turbines.

KEYWORDS

API piping plans, compressor efficiency, compressors, failure modes, piston rods, reciprocating compressors, rotary-lobe compressors, rotary-screw compressors, safe operations, surge control, troubleshooting

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ACKNOWLEDGMENTS

I would like to thank a number of people who helped me make this book possible, most of all my wife, who supplied endless cups of coffee at all times. The names of other people are too numerous to mention, so I simply say many many thanks to them.

CHAPTER 1

UNDERSTANDING COMPRESSORS



Having a compressor with minimal operating problems depends not only on the selection of the right type and size for the job. Detailed specifications of all auxiliary equipment and operating conditions, as well as keeping a constant vigilance over the engineering and installation phases, are also essential.

A compressor is frequently an expensive device, a critical item in a process, and a rather complicated piece of equipment to specify and purchase because of the many alternatives open to the engineer. Faced with this wide choice, on what basis can a selection be made so that the chosen compressor will do the intended job?

Since each type of compressor has specific characteristics, these must first be understood. Then, the many factors that enter into the final decision to purchase a compressor and auxiliaries are considered.

COMPRESSOR TYPES

There are two basic mechanical methods of increasing the pressure of gas: reducing its volume and increasing its velocity so that the velocity energy may be converted into pressure.

Positive-displacement machines that increase the pressure by reducing the volume are as follows:

- Reciprocating compressors, which have a piston moving within a cylinder (Figure 1.1).
- Rotary-screw compressors, in which gas is squeezed between two rotating intermeshed helices and the casing in which they are housed.
- Rotary-lobe compressors, through which gas is pushed by intermeshing lobes.

LEARN HOW A ROTARY-SCREW COMPRESSOR WORKS!

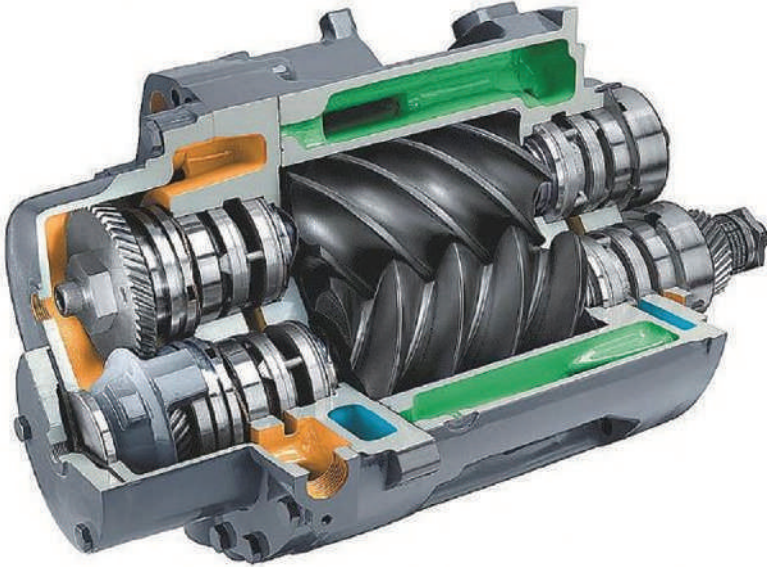
On this page I will tell you about the pros and cons of the rotary-screw compressor, what to look for when buying one, and common breakdowns that might occur during the operation.

The rotary-screw compressor uses two rotors (helical screws) to compress the air. There's a "female" rotor and a "male" rotor. The rotors are of different shape, but fit each other exactly.

When the rotors start turning, air will get sucked in on one side and get "trapped" between the rotors. Since the rotors are continuously turning, the air gets pushed to the other end of the rotors (the "pressure side") and new fresh air gets sucked in.

Because this is a continuous process, this kind of compressor doesn't make a lot of noise; it runs quietly and smoothly.

Compared to the piston-type reciprocating compressor, the rotary-screw compressor is much more expensive, but it will use a lot less energy over the years, resulting in a lower overall cost.



Compressor element (oil-free type). (Photo courtesy of Atlas Copco.)

OIL-FREE OR LUBRICATED COMPRESSORS

The rotary-screw compressor is available as oil-injected and oil-free versions. The basic principle is the same (the rotors “push” the air to one side), but they are quite different machines.

Oil-inject models are by far the most common oil-injected screw-type compressors. When you need a lot of air in your workshop or factory, this type of compressor is usually the best choice!

Oil-free models are used for specific special applications. I’ve mostly seen them on big factories like oil/gas or chemical refineries, big food factories, or other places where the compressed air must be 100 percent oil free (otherwise it could contaminate the food, product, or chemical process).

OIL-INJECTED ROTARY-SCREW COMPRESSORS

How do they work? As its name implies, oil is injected in the compressor element (where the two rotors turn), during the compression of the air. What we end up with is a mixture of oil and air under pressure (commonly about 7 bar).

In a special oil separator, the oil is separated from the air. Most of the oil is removed by the centrifugal force, the remaining less percentage of oil is separated by the separator (filter) element (it looks just like a big

air filter). The separator element should be renewed every 2,000 running hours or so (depending on the manufacturer/model).

The oil is cooled and fed back to the compressor element to do its job again. The compressed air, now without the oil, is directed to the pressure outlet of the compressor, usually through an after-cooler (the air gets very hot when it's compressed).

There is no special oil pump to do all this; the oil flows by the pressure differences inside the compressor.



Example of a small rotary-screw compressor. (Photo courtesy of Atlas Copco.)

Pros:

- Quiet operation
- High volume of air, steady flow
- Low energy cost

Cons:

- Expensive compared to piston compressors
- More suitable for continuous operation only

OIL-FREE ROTARY-SCREW COMPRESSORS

The basic workings are the same as that of the oil-injected screw compressor, only this time, there's no oil, only air! Because of this, the rotors used

are of superior quality with very little space between them. They do not touch each other though; otherwise, they would wear down too quickly.

Because there's no oil injected during compression, the compression is usually done in two stages, because if we would compress the air in one go from 1 to 7 bars, it would get really, really hot.

Stage one compresses the air to a few bars (say 3.5 bars). The air will be very hot at this time, so it flows through an inter-cooler first before entering the second stage. Stage two will compress the air further from 3.5 bar to the end level, mostly 7 bar.

Normally, the two stages will be built on one gearbox, with one electromotor driving them at the same time.

Pros:

- 100 percent oil-free air

Cons:

- More expensive than the oil-injected type
- Servicing/repairing more difficult and more expensive than the servicing/repairing of the oil-injected type
- More noisy than oil-injected compressors



An example of a big rotary-screw compressor (old-fashioned oil-free type).
(Photo courtesy of Atlas Copco.)

FAILURES

Rotary-screw compressors can fail in a number of ways. The most common problem is oil in the compressed air. Most of the times this happens because the oil separator is not doing its job properly. The chances are that the separator element is saturated with oil (read: you didn't service the compressor in time! There's a fixed time limit [running hours] to change the element!).

Another problem often encountered is water in the compressed air. Since the compressor takes in a huge amount of air (with water vapor) and compresses it to seven times a smaller volume, a lot of water will be produced. Normally this water is drained using an electronic or mechanical automatic drain. If this drain is broken, the water will stay in your compressed air and fill up your air receiver and piping.

If the problem persists, it will most probably be a defective pressure switch (which will start/stop, load/unload the compressor), or a defective inlet valve (which opens and closes the air inlet of the compressor). If it's closed, the compressor is running in an "unloaded" condition and won't supply any air.

ROTARY-SCREW AIR COMPRESSORS

Rotary-screw compressors are another variety of compressors available to industrial and DIY compressed air users.

CONCEPT

If you have ever used an electric drill in a piece of wood or metal, you may have noticed that the chips or spirals of metal follow the contour of the flutes up and out of the hole (most of them, at least).

A similar phenomenon occurs inside the rotary-screw compressor housing.

At the wide end of the screw (sometimes there is one screw operating against a housing, sometimes more than one), an inlet valve allows free air into the screw housing when there is a demand. Free air flows into the housing from the outside as there is a partial vacuum formed inside the rotary-screw housing as the screw(s) rotate.

Inside the screw housing are the screws in a bath of oil. The oil is there to provide a viscous, flowing, sealing method to help trap the air in the rotary-screw flutes.

The air–oil mixture in the screw housing moves along the flutes from the wide end of the screw toward the narrow end, pulling a vacuum behind, thus drawing more air into the screw housing.

As the air–oil blend is pulled along the flutes of the screw, the space in which the air is contained gets smaller and smaller. The diameter of the screw is larger at the inlet end and smaller at the discharge end, thus compressing the air. The amount of air trapped in the screw flutes does not change as the air is moved along the narrowing path, but the volume that air is in gets steadily smaller, thus compressing the air.

Manufacturers of rotary-screw compressors have their own ideas of what constitutes the ideal geometry of the screw within their air compressor.

Rotary-screw compressors may have just one screw (also sometimes known as augers) or maybe two or more. Single-screw compressors function the same way as multiple-screw units, with the air being compressed between the housing of the screw compartment and the screw itself, rather than between two or more screws.



The following drawing will give you an idea of how the rotary-screw concept works with two screws. The actual guts of the rotary-screw compressor will vary depending on the designs of the company that manufactured that particular compressor. The drawing shows two screws. They would be housed inside the screw compartment of the compressor, in a bath of oil.

At the narrow end there would be an outlet valve, which feeds the compressed air–oil mixture from the screw compartment and into a separator.

The separator has the job of removing as much oil from the compressed air as possible, and then to release that compressed air into the compressor receiver or into the plant main air lines.

FEATURES/COMMENTS

Although rotary-screw compressors are available in lower horsepower ratings, it would appear that it is in the 20 to 25 horsepower and higher range that many industrial compressor applications tend to move toward using a rotary-screw compressor solution instead of other types of compressors.

One major manufacturer states that the rotary-screw air compressor has become the most popular source of compressed air for industrial applications.

That may be because of the need for a compressor with a continuous duty cycle. Some rotary-screw compressor manufacturers claim a duty cycle of 24/7/365, which is pretty incredible for any electromechanical device.

Rotary-screw compressors are available with a direct motor-to-screw drive; others are belt driven. Each has its benefits and its own drawbacks, the details of which are best obtained from the actual compressor manufacturer.

Less Maintenance

The perception, one that is claimed by some manufacturers, is that rotary-screw units have the least maintenance issues of all types of air compressors and are touted as being the easiest to maintain in terms of both regular maintenance and unscheduled downtime.

Reputed for Lower Cost

When you move up into the higher horsepower units, rotary-screw units are reputed for their lower cost over a comparably sized reciprocating

compressor, and further, they boast lower operating costs than either vane or reciprocating units.

Oil Carryover

Some manufacturers suggest their oil carryover from the compressor to the compressed air supply of the plant is calculated in parts per million per day, rather than the ounces or more of oil that can enter the plant-air stream from older reciprocating models and some well-used vane models.

Lower Operating Noise

Other firms suggest that their rotary-screw units boast a very low operating decibel rating, and claim noise output levels far below other types of compressors, an important issue to be considered for the benefit of workers in the area. It is our experience that the lower operating sound levels may not eliminate the need for a soundproof housing, unless the compressor is well equipped with sound-deadening cladding.

Know Your Air Requirements

As noted earlier in this chapter, when you are moving toward selecting a rotary-screw (or any compressor type), you first need to know how much air you'll need in cubic feet per minute (CFM) at the psi you need for your plant, your tools, and all ancillary equipment for now and for the future. You'll want to build in excess volume available, as one statistic we've seen says, on average, more than 10 percent of all compressor capacity is lost through leaks, despite the best efforts of the plant to reduce wasted air.

Once you got the compressor size figured out (link; I hope the information here will help), you will want to review the up-front cost of compressors from a host of manufacturers. Check their mean time, between-failure rates, their parts and service costs, the life expectancy of the unit with the duty cycle you will require, and the particular operating costs.

Compressing air is expensive, and one compressor might provide lower up-front capital costs, yet end up being far more expensive in the longer term due to higher operating costs.

All factors having been considered, and certainly this is claimed by many of the manufacturers of the rotary-screw type of compressor, the rotary-screw compressor may surface as your best choice for industrial application.

- Sliding-vane compressors, where an eccentric cam (into which sealing vanes slide) rotates inside a housing.
- Liquid-piston type, in which a partially liquid-flooded case creates the equivalent of sliding vanes.
- Diaphragm compressors, in which a flexible diaphragm is pulsed inside a concave housing.

The two types of compressors that convert velocity into pressure are:

- Radial-flow compressors, generally called "centrifugal compressors"
- Axial-flow compressors, known as "axial compressors"

In centrifugal compressors, the gas enters the eye of the impeller, and the rotative force moves the fluid to the rim of each wheel or stage. Diffusers convert the velocity head into pressure, and return passages are then used to lead the gas to the compressor discharge or to the next impeller stage.

In axial compressors, flow occurs through a series of alternating rotating and stationary blades, and in a direction basically parallel to the compressor shaft. Each passage through the rotating blades increases the velocity of the fluid, and each passage through the stationary diffuser blades converts the velocity head into a pressure head.

SELECTION CONSIDERATIONS

Not all types of machines are made in all pressure–volume ranges. Figure 1.1 indicates, in a very general manner, the capacities of reciprocating, centrifugal, rotary-screw, and axial compressors available. The more common usage is indicated by the deeper shading. Although this figure does not indicate the theoretical or engineering limits of any design (the limits are continually being expanded), it may be used as a guide to current technology.

Since sealing systems for axial compressors are not as versatile as for other types, normally only those gases whose leakage to the atmosphere can be tolerated should be handled by this type of machine.

Rotary-lobe, sliding-vane, liquid-piston, and diaphragm compressors have relatively small capacities and, typically, atmospheric-pressure suction. Of these four types, the rotary-lobe compressor can deliver the most gas, as its maximum suction volume is about 30,000 actual cfm (acfm). A maximum discharge pressure of about 40 psia can be

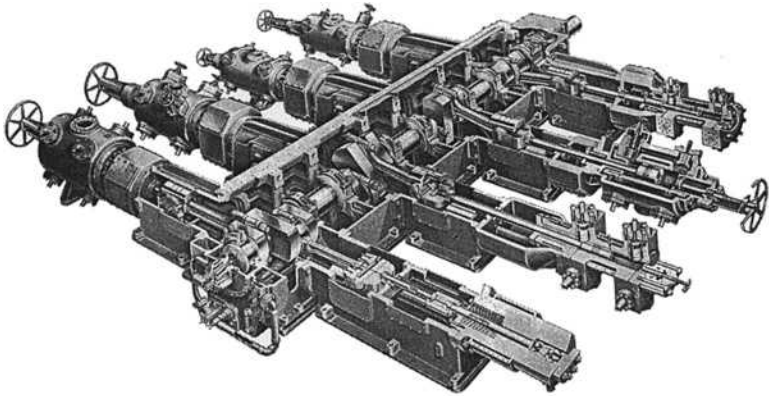


Figure 1.1. A process reciprocating compressor operates from vacuum to over 36,000 psi

attained. However, rotary-lobe compressors are most competitive at capacities of 17,500 acfm or less and discharge pressures of about 22 psia.

Maximum inlet capacities of sliding-vane units are about 3,000 acfm, or double this amount if a duplex compressor is used. The latter consists of two compressors attached to a single drive. Maximum discharge pressures of standard machines are about 65 psia in a single stage and 140 psia in two stages.

The liquid-piston compressor has a maximum capacity of about 10,000 acfm and can deliver this amount of air (or gas) at about 30 psia.

Volumes of 300 acfm or less can be compressed to about 115 psia.

The three foregoing compressor types can produce moderate to high vacuum, particularly in multiple stages.

Diaphragm compressors have much smaller volumetric capacities, with maximum flows ranging from 40 to perhaps 200 acfm. These machines, however, can develop pressures up to 40,000 psi.

Before selecting a compressor type, one must decide how many machines will be needed to handle the process load. In former years, reciprocating machines were used for almost all process applications. Since the compressor capacity was low, large plants would require trains of machines. As machine reliability and capacity increased, the tendency to install two machines started, each with 55 percent or 60 percent capacity, perhaps with a third unit as a spare.

The spare unit ensured operation at full capacity, but at an increased compressor cost of about 50 percent. If the spare compressor were omitted, but two half-size machines installed, one could still be reasonably sure

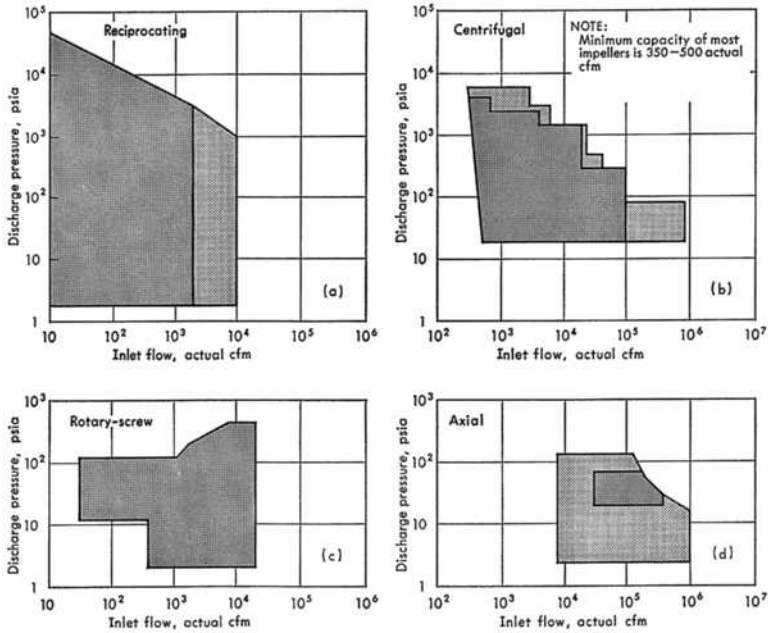


Figure 1.2. The pressure–volume operating range (discharge pressure vs. feed volume) of various compressors

of continued operation at all times. This was particularly important when the process included equipment that could not be shut down frequently, such as furnaces.

Later, to take advantage of larger machine capability, several services were placed on the same frame.

Today, the situation is somewhat different, as more centrifugal compressors are being used (Figure 1.3). For one thing, the downtime of a rotating equipment generally is appreciably less than that of a reciprocating equipment. Therefore, in many instances a single centrifugal compressor may be satisfactory. However, it must be recognized that when a compressor is down, it will usually take longer to repair or overhaul a centrifugal unit than a reciprocating one—unless a complete spare rotor is available.

Also, the pricing structure of centrifugal compressors is quite different from that of reciprocating ones. As a first very rough approximation, one may assume that halving the size of a reciprocating compressor will halve its cost. Yet, halving the size of a small centrifugal compressor may only decrease its cost 20 percent, and halving the size of a large machine may only reduce its cost 30 percent.

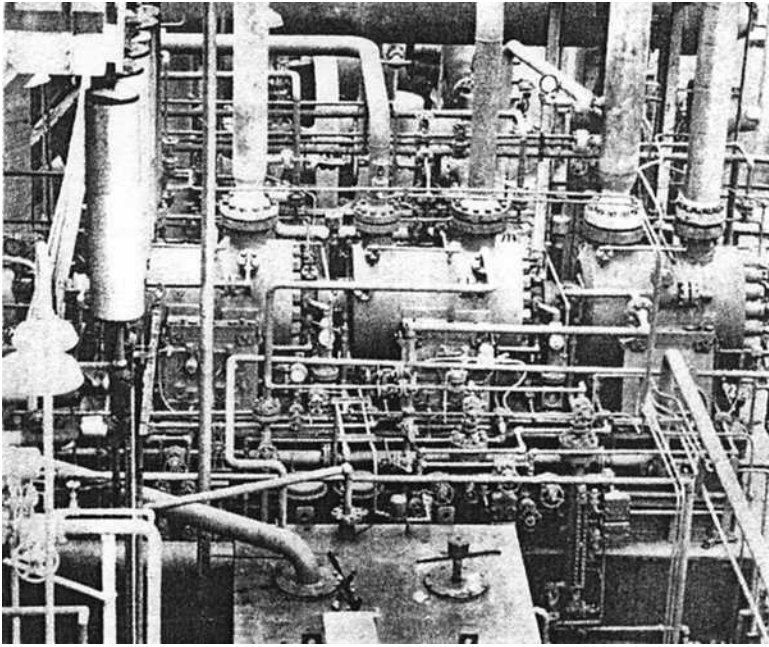


Figure 1.3. A train of three barrel-type centrifugal compressors in a 1,400-ton/day ammonia plant

Furthermore, because of their flat operating characteristics, the running of centrifugal compressors in parallel may result in surging unless very careful attention is given to avoiding unstable operation. Therefore, in many process applications for which one centrifugal compressor will have adequate capacity, an installed spare is not provided. In these instances, a complete spare rotor may be bought.

The choice between reciprocating and centrifugal compressors is not always simple, particularly for high-head, medium-capacity service such as gasfield repressuring. If several reciprocating compressors are used, each can be multiple staged to develop the desired head. The shutting down of one machine would merely cause a decrease in plant output. But if several centrifugal compressors were used in series, the failure of one would stop the entire operation.

OPERATING CHARACTERISTICS

A positive-displacement compressor is characterized by a pressure rise–volume curve that is almost vertical. (It is not completely vertical because

there is mechanical clearance, and slip and leakage from the discharge to the suction; the slip increases as the compression ratio rises.) The compressor delivers its gas against any pressure head up to the limit of its mechanical strength and drive capacity. Capacity is almost directly proportional to speed.

The characteristics of a centrifugal compressor are appreciably different. Generally, the pressure rise–volume curve is quite flat (Figure 1.4a). (It may be somewhat steeper if a heavier gas is being compressed.) A small change in the compression ratio produces a marked effect on the compressor output. As the discharge pressure increases, the flow decreases, and if the flow decreases too much, the machine will start to surge.

Surging occurs when the velocity of gas leaving an impeller wheel is too low to move the fluid through the machine. With no gas leaving the impeller, the discharge pressure may drop. Should this occur, the machine will again start to compress gas, and the cycle will be repeated. Such intermittent operation may severely damage a compressor. The characteristic curve can be modified by the installation of adjustable inlet guide vanes (Figure 1.4b). These are most effective on machines having a few stages. Adjustable diffuser vanes have been used on some machines.

In some installations, process requirements may dictate that the compressor be run at the far right of the characteristic curve, where it is very steep. Operating in this area requires careful control and is accomplished at some penalty of compressor efficiency.

The volumetric capacity of a centrifugal compressor is almost directly related to its speed; its developed head, to the square of the speed. (The horsepower requirements are thus related to the cube of the speed.) The efficiency of centrifugal compressors is lower than that of reciprocating machines by perhaps 5 to 20 percent.

These characteristics establish the sensitivity of the compressor to variations in flow conditions. For example, a change in the density of the fluid being compressed will have little effect on either the volume of gas pumped or the discharge pressure developed by a reciprocating machine, although one would have to be sure that no component parts of the compressor were being mechanically overstressed. Any variation in the density of a gas being compressed will result in a proportionate change in the weight of gas pumped.

On the other hand, because the head developed by a centrifugal compressor depends only on the velocity developed, a change in gas density will be directly reflected by a proportionate change in the developed discharge pressure. However, at a given density, if the discharge pressure can

be permitted to change slightly, one can obtain large variations in volumetric gas flow through the compressor.

The axial compressor has a very steep characteristic curve (Figure 1.4c). The unit's surge capacity is thus close to its operating capacity. However, by providing a method of adjusting the angle of stator blades and inlet guide vanes, a greater operating range can be obtained (Figure 1.4d).

Generally, the efficiency of an axial compressor exceeds that of a multistage centrifugal machine by perhaps 5 to 10 percent. The axial compressor does not contain diaphragms that expand radially as the compressed gas gets hot. This mechanical factor, combined with higher efficiency, leads to greater freedom from temperature limits and permits a higher compression ratio per case than do centrifugal units.

SPEED CONSIDERATIONS

The type of mechanical drive (including gears) that is used may influence the choice of compressor.

Compressor and drive speeds are very pertinent if one wishes to avoid gearing. The accompanying table 1.1 provides speed ranges of the most common types of compressors and drives. There are specially designed units, however, that do not fall within the ranges listed. One of these, for example, is a carbon dioxide compressor with a suction volume of approximately 50 acfm at the last wheel, which rotates at 25,000 rpm and delivers gas at 5,000 psi. The tip speed of this compressor's impeller is approximately 650 ft/sec. The compressor itself is directly driven by a specially designed 1,000-hp steam turbine.

COST COMPARISONS

In very general terms, at low pressures and large flows, the purchase cost of a reciprocating compressor may be estimated to be perhaps twice that of a centrifugal machine of the same capacity (Figures 1.5–1.7). The cost differences narrow as pressure increases or actual flow decreases. At high pressures and low flows, costs may be quite close to each other. A reciprocating compressor will need a more massive foundation, more protection from the environment, and a more careful piping design to avoid vibration and pulsation.

On the same rough basis, one may estimate the costs of rotary-screw and axial compressors to be about the same or less than that of centrifugal units. In their most suitable applications, the costs of the screw and of the axial compressors may be considerably lower.

Table 1.1. Speed Range for Compressors and Drives

Compressor Types	Usual Speed Range, Rpm.	Remarks
Large reciprocating compressors	300-600	Some even 1,000 to 1,500 rpm.
Small reciprocating air and refrigeration machines	1,000-1,500	
Rotary-screw	3,000-10,000	
Process centrifugal units	3,000-12,000	Some large-horsepower machines up to 17,000 rpm.
Special, small-volume high-head air centrifugals	30,000-50,000	
Axial compressors	3,000-6,000	Some up to 16,000 rpm.
Large internal-combustion engines and reciprocating gas expanders	300-600	
Small rotary and radial engines	3,000-8,000	
Mechanical-drive gas turbines and centrifugal expanders (over 1,000 hp.)	10,000 or less	Small gas-turbine compressor drives have operated at up to 50,000 rpm.
Mechanical-drive, back-pressure steam turbines (3,000 to 40,000 hp.)	16,000 or less	Condensing turbines have lower maximum speeds.
Electric motors	3,600 or less	

COMBINATIONS OF COMPRESSORS

At times, combining compressors may be worth considering. For example, in compressing to a very high pressure, it may be possible to use a centrifugal machine or a rotary-screw machine for a lower pressure and then pipe the gas to a reciprocating unit. In some instances, axial and centrifugal

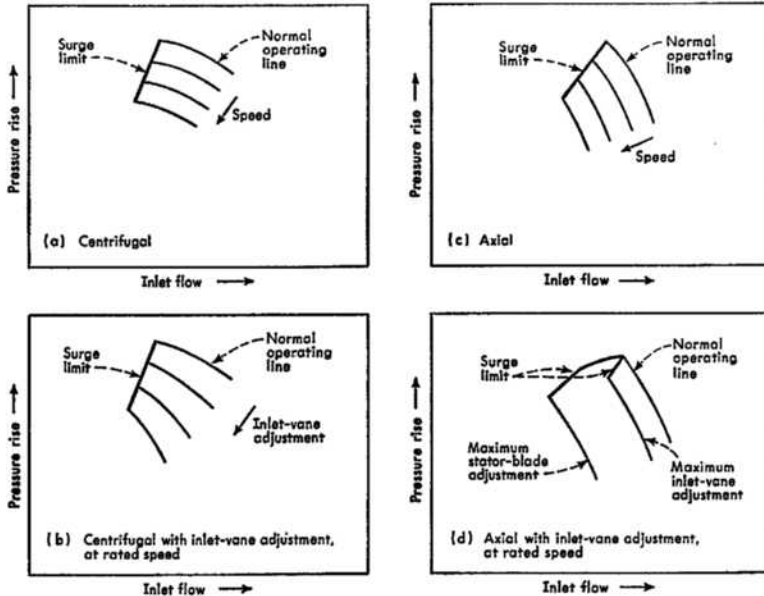


Figure 1.4. Operating characteristic curves for centrifugal and axial compressors

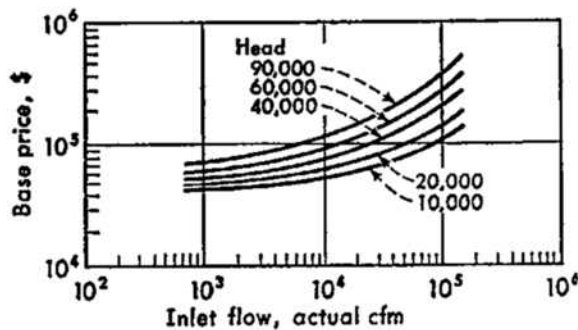


Figure 1.5. The cost of centrifugal compressors according to the inlet volume and head output (head is feet of gas)

impellers may be placed on the same shaft. In addition, one might also resort to placing axial- and centrifugal-compressor cases in a common drive train.

As an alternative to an axial compressor, three or four single-stage centrifugal compressors may be connected by a gear train to a single drive. With the gas cooled after each stage of compression—and gears designed to permit each stage to be run at its optimum speed—the efficiency of

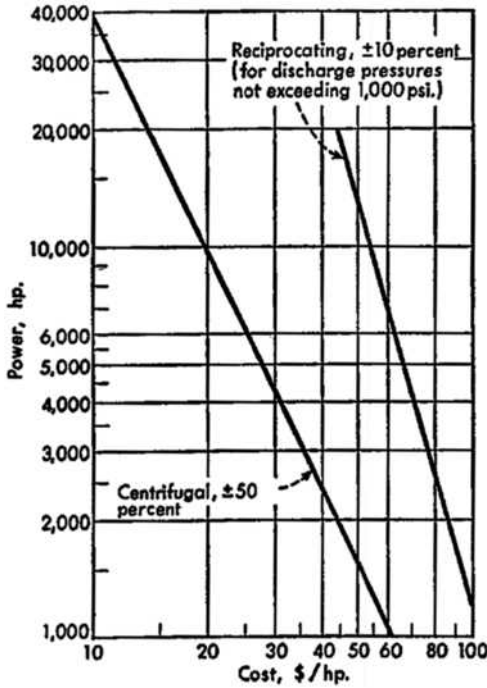


Figure 1.6. The base price of reciprocating and centrifugal compressors according to horsepower unit

these centrifugals is comparable to that of an axial compressor, while their operating characteristics are those of a centrifugal machine. Extensive gearing, however, is a distinct disadvantage.

Additional significant characteristics of each type of compressor will be discussed in a forthcoming article.

DRIVING MECHANISMS

Smaller units are usually electric-motor driven (direct or belt); for medium to large units, a wide choice of drives are available. These include motors (synchronous, induction, low or high speed); steam turbines (back-pressure, condensing or controlled extraction); internal-combustion engines (integral or direct connected); gas turbines (single or double shaft); and expanders.

The selection of a drive depends to some extent on the compressor service, but more important are the overall energy balance, energy

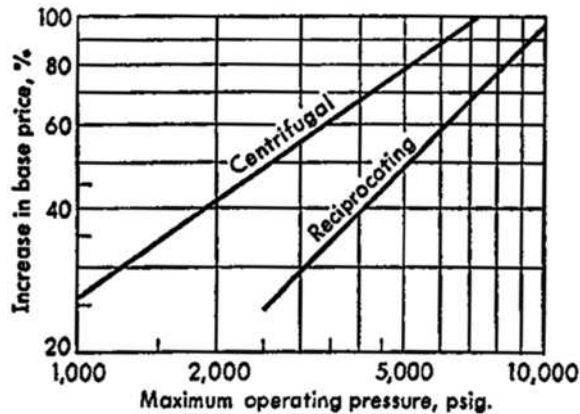


Figure 1.7. The effect of operating pressure on the cost of centrifugal and reciprocating compressors

utilization and availability, and heat-rejection methods. Within the limits imposed by these criteria, the selection should stress a drive system that is simple, dependable, and straightforward. The compressor is the reason for the drive, not the other way around.

The drives of internal-combustion engines and steam turbines can ordinarily be operated over a fairly large range of speeds. This may not be the case, however, if gas-turbine or electric-motor drives are used.

Let us first consider gas turbines, almost all of which have axial-type air compressors for pressure–air supply to the compressors. For single-shaft units (air compressor, gas turbine, and driven unit on one shaft), the speed range is most often determined by the steep performance curve of the axial compressor rather than by the much flatter curve of the centrifugal process compressor. Double-shaft machines permit constant speed for the axial air compressor and variable speed for the process compressor. The selection of sizes, speeds, and horsepower outputs of commercially available gas turbines is limited. Very often, a wide freedom of choice is not available as to single- or double-shaft units.

Motor drives usually have a constant speed: In a limited number of cases, variable-speed couplings, wound-rotor or multipole (PAM) motors may be used. Large motors may be of the synchronous or induction kind. For a unit driven at above-synchronous speeds (3,600 rpm for 60 Hz), the choice should be based on the total cost of the motor and the speed increaser. Thus, a 1,800-rpm induction motor and its speed increaser may cost less (including operating costs) than a 1,200-rpm synchronous motor with its speed increaser.

Constant-speed centrifugals in process plants tend to operate at a high enough average load so that the economic rewards of power-factor correction obtained by the use of a synchronous motor are minor.

Fossil-fuel drives are used when initial and operating costs are more attractive than those of steam or motor drives, sufficient electric power is not available, and electric or steam sources are not reliable. In this last case, the entire system must be carefully specified to ensure that minor items such as cooling-water pumps, pressure switches, control air, etc. are independent of any source of power, not as dependable as the compressor drive.

Internal-combustion engines are usually turbo-supercharged and may be two or four cycle, integral with or separate from the compressor. The type of engine can normally be selected on the basis of drive features, including accessories and costs (purchase order, installation, fuel consumption, spare parts, maintenance) independent of the compressor. It is best, however, to include the drive as part of the compressor system.

The gear mechanical rating, including the American Gear Manufacturers Association (AGMA) service factor, should be selected so that the gear rating does not become the limiting factor in the compressor and drive train. Steam-turbine drives combined with a gear (with the turbine at a lower speed than the compressor) are sometimes lower in cost than higher-speed turbines. The policy of the user and his insurance carrier on warehouse spares for gears affects the choice, since the gears and additional couplings increase the probability of outage.

MARKETING OF COMPRESSORS

Compressor systems and their drives range from small through medium to large. Marketing methods range from those suitable for catalog items handled by distributors to engineered systems oriented to specific market areas such as chemical processing, gas distribution, petroleum refining, and electric-power generation. Engineers holding discussions with equipment suppliers, manufacturer representatives, or suppliers of package or skid-mounted units should recognize these elements of supplier organizations; such knowledge will aid in establishing the scope and the detail most useful to include in purchase requisitions and specifications.

Catalog items require little more than hardware description, as performance is specified in published information. Engineered systems, on the other hand, require the definition of performance requirements for the overall compressor system. Hardware definitions also are needed to establish the quality level of the system and its components, and to define

the number and type of auxiliary elements such as oil coolers, governor, pumps, etc. Other hardware items include controls, heat-rejection systems, drives, gears, piping, ducts, and electrical wiring (cable, conduit, trays, etc.).

The environment (indoor or outdoor) should also be made clear so that due allowances can be made for access for construction and maintenance, sound control and isolation, and area electrical classification.

SYSTEM SPECIFICATIONS

The form of a purchase specification should be the one most familiar to, and most commonly used within, the issuing organization; typical forms are given in API standards 617 and 618. Here we shall consider the content of the specification.

Performance criteria must be carefully defined for the end use that the compressor must have within the overall system. Following should be included in the specification:

- The range of mass and volumetric flow as influenced by variations in the inlet temperature, pressure, molecular weight, gas composition (vapor loading, compressibility factor, etc.), discharge pressure, temperature, and flow of cooling fluids (water, air, etc.)
- Startup, standby, and shutdown conditions of the compressor and of the entire system
- Mention of even traces of vapors, liquid droplets, dusts, or gases that may be minor items for the chemistry of the process but may cause fouling, gunking, seal problems, etc., either by themselves or when mixed with lubricants or sealing fluids (items such as these may appreciably influence the choice of the compressor type)
- Range of ambient temperature
- Altitude
- Area electrical classification
- Applicable codes and standards from such organizations as the Tubular Exchanger Manufacturers Association (TEMA) and the American Society of Mechanical Engineers (ASME)

Purchase specifications must define quality requirements for auxiliary equipment such as seals, piping systems (material and arrangement), type and quality of control elements and systems, level of redundancy, and shop testing (if any). Checklists for such items may be prepared based on the available knowledge within an organization, as well as on

accepted references on the inspection of completed installations, such as Chapter X of the API “Guide for Inspection of Refinery Equipment.” Purchase specifications—or those prepared by the customer for use by an engineer-constructor—should not limit the bidders from using their knowledge and experience.

Required controls cover a very wide range of supply. The compressor specification should include all elements that directly measure and control any part of the compressor system. This includes local panels, receivers from external inputs, and any items to provide outputs to external devices. Devices for volume control as such—or those used to control mass flow and provide anti-surge control—must be carefully defined as to which elements are supplied as part of the compressor system and which elements are external. Thus, such items as inlet guide vanes are best included in the compressor system, while anti-surge and recycle devices are usually best considered as external to the system.

An increasing area of interest for controls consists of the types of diagnostic devices used to measure, indicate, alarm, and record vibration (velocity and displacement), axial movement, bearing temperatures, and drive-motor-copper temperatures. Axial-movement and motor-copper temperature indicators are best used for both alarm and shutdown. Other instruments are most suitable for alarm only and as trend indicators. The compressor supplier is in the best position to select the points of pickup and recommend types of pickup and readout devices. Controls for units to be attended only by remote or occasional local surveillance require very careful attention.

Job cost and completion time is improved with proper use of shop-assembled units. Typical packages, including skid-mounted units, comprise refrigeration (chilled water and low-temperature brines) and complete instrument and plant-air units. Purchase specifications should therefore call for or permit packaged units to be offered where feasible. The units should be such that they need merely to be set on simple foundations, and have the power, cooling-water, and supply and discharge piping connected.

FACTORS THAT MAY BE OVERLOOKED

Perhaps the most important single factor to determine is whether a proposed piece of machinery has been used in a similar service and what its history has been. This by no means implies that one should never install a newly designed compressor or use an older design for a new application. The first to use a new design may enjoy an advantageous position

before competitors follow suit. Furthermore, if the decision to begin the construction of a plant cannot be delayed, it may be a matter of either installing something new or installing a proven design that one realizes may soon be outmoded. Features of a design that have not been proven should be reviewed in detail, and perhaps consideration should be given to courses of action to be taken if unexpected difficulties occur. Attempting to foresee possible failures and developing corrective courses of action is very time consuming, complex, and usually not very rewarding. But when trouble does occur, such planning may more than compensate for the effort spent.

If a compressor and its drive are investigated separately, one should not overlook the direction of rotation (which may not always be changeable) and its effect on gear requirements. One supplier should be given the responsibility of completing a combined torsional and vibration analysis of the entire system. An agreement as to who will do this work should be reached as soon as possible, so that any required design changes can be made with minimum difficulty.

Similarly, it is advisable to have one supplier assume the responsibility for collecting and correlating design data pertaining to noise emission and for making the final recommendation for noise suppression. The best procedure is to select one supplier and have him assume responsibility for the overall unit.

When reviewing vendors' proposals, there are several items, in addition to price and energy requirements, that should not be overlooked.

One may question what spares the supplier will generally have in his shop. If the purchaser does not buy a spare rotor, how long will it take to obtain a new one in the case of an emergency? Is there another company that might share the cost of a standby spare? Such sharing has not been generally accepted because even though costs are reduced, the risk of extended outage is increased. Nevertheless, this course of action may be worth considering.

When a centrifugal compressor is bought, one should check the closeness of the operating point to the surge point. On low-capacity, high-head wheels, these points may be quite near each other. It may not be possible to reduce speed very much without resorting to bypass control or to the installation of suction or discharge valves, etc. Critical speeds should also be reviewed to be sure they are far enough away from any desired operating speed, particularly if operation at reduced capacities may be considered.

Allowable noise levels are a function of frequency as well as intensity, and levels must be lower at higher pitches. The amount of noise generated depends on the type of compressor, its horsepower rating, compression ratio, speed, etc. Silencers or acoustic coverings may be used to reduce the

emitted noise to levels acceptable by the user and by regulatory authorities. Methods of estimating compressor noise levels, and the effect of various kinds of silencers, are detailed in the literature.

Drive and compressor characteristics must be evaluated. Operating at the maximum continuous speed of the centrifugal compressors, which is generally 5 percent greater than the speed at the compressor rated point, may call for a power expenditure 15 percent greater than the rated horsepower. If the compressor is driven by a turbine, increasing the latter's capacity by 15 percent may impose a significant increase in the cost of the drive and its auxiliary facilities, as well as a penalty of poorer efficiency when running the drive at the rated compressor speed. Normally, the permissible maximum mechanical speeds of the drive and the driven unit should be the same.

It may also be good to review guarantees. In general, the following applies (API Std. 617, 618) unless other representations are made: For centrifugal compressors operating at a constant speed, the capacity is guaranteed; the head may vary within +5 percent and -0 percent of that specified; the horsepower (when corrected to the specified head-capacity conditions) may vary by not more than 4 percent of the stated horsepower. For centrifugal compressors operating at a variable speed (i.e., gas- or steam-turbine drive in most instances), the capacity and the head are guaranteed but the speed is not. Horsepower may vary ± 4 percent.

For reciprocating compressors, one may specify a guaranteed capacity with no-negative tolerance, as well as a guaranteed maximum horsepower and a specified speed. However, process industries frequently accept a manufacturer's capacity guarantee of ± 2 to 3 percent rather than pay more for the no-negative tolerance. When motor drives are to be used, obviously one must review compressor-speed guarantees, much more carefully than when turbine drives are involved.

INSTALLATION ENGINEERING

After the compressor unit is selected and a purchase order issued and accepted, the next steps require continued vigilance. This is not simple because many more people in the engineering and supplier organizations now become involved. Follow these guidelines to prevent certain items from being neglected:

1. For all but simple catalog units, prepare a process and instrumentation diagram or an engineering flow diagram for the complete compressor system.

2. Establish the layout requirements including those determined by operator assignment, that is, the number of operators assigned to the compressor during normal operations, during startup only, etc. Provide terminals for remote control if an operator will not be in attendance at all times.
3. If a large, complex compressor is involved, hold meetings with the supplier's engineering group to establish schedules for the submission, review, and approval of the supplier's engineering data and drawings, and tentative plans for use of his servicemen during installation and startup.
4. Review the compressor manufacturer's drawings and those of his suppliers to ensure that quality and performance criteria are being met.
5. Review the torsional-vibration analysis and lateral critical studies completed by the compressor and drive supplier to make certain that no contemplated operating condition will cause the machine to operate at a hazardous speed.
6. Review unpriced supplier orders. (Priced orders would not be made available and are unnecessary.)
7. Review control plans, including startup, normal operation, scheduled and forced shutdowns, protective and safety devices for alarm and shutdown, and the duties to be assigned to the operators.
8. Submit to the supplier, for his comments and review, design bases and installation drawings for foundations, piping, and pipe supports. Such information should include the calculated forces and moments (hot and cold) exerted by the piping on the equipment flanges. Guidelines for allowable values are established by the compressor and turbine suppliers on their outline drawings.
9. Review requirements for shop and field pressure and performance testing. In most applications, test procedures established by the supplier are sufficient. Establish procedures for shop erection and match marking of the prefabricated pipe to be furnished by the compressor supplier. Establish what shop tests are to be witnessed.
10. Establish requirements for such items as operating and maintenance access (cranes, monorails, etc.); noise control within buildings and other enclosures; protection from fumes and dust; winterizing; and piping systems, including drains, vents, and access for field flushing and cleaning.
11. Provide methods so that the installed dimensional accuracy of piping right at the compressor is high, and thus compatible with the level of dimensional exactness required by the machinery. Neglect of this may require field changes in the piping arrangement to secure and

maintain the acceptable compressor alignment. Arrange the major piping so that supports can be taken from concrete substructures rather than from elevated steel structures. This is most important in reciprocating compressors because it helps in attenuating vibration.

12. Make provisions for shop inspection during fabrication and assembly, as well as during shop testing.
13. Obtain copies of expediting and inspection reports. Monitor delivery schedule.

VIGILANCE DURING INSTALLATION

Job pressures during installation, run-in, and startup create many hazards to the achievement of quality results from the compressor system. To minimize the hazards, follow these guidelines:

Manufacturer's Representatives—Err on the side of too much participation in the field of representatives of the supplier's service organization and of his major sub-suppliers (turbines, gears, motors). Most suppliers include a specific number of days for such representatives in the original proposal price (and thus in the purchase order), with a per diem rate beyond this limit. Do not save these "free" days for when troubles may be encountered. Use the days to avoid trouble. Do not hesitate to obtain such service beyond the limit of days included in the purchase order for such items as machine setting and grouting, alignment, initial run-in, and actual startup and demonstration.

Field Checking—During field checking of all kinds, not only construction drawings but also flowsheets, operating manuals, etc. must be consulted.

Foundations and Superstructures—Check these vital elements of the system, using supplier drawings and installation recommendations, as well as the engineer's construction or working drawings. Dimensional accuracy and quality of construction are both vital.

Materials of Construction—Check to see whether mill certificates have been received and whether they are acceptable.

Piping, Ductwork, and Supports—Ascertain accuracy and structural adequacy, so that excessive loads are not imposed on equipment flanges. Also make sure that provisions are made for controlled movement due to thermal expansion; proper line slopes are maintained; noise and vibration are attenuated; and resonant conditions

are avoided. Conduct hydrostatic and leak tests, remove all temporary blanks, and install rugged line strainers.

Flush, degrease, mechanically clean, and, when pertinent, clean with chemicals. Chemical cleaning (pickling) of carbon steel piping for such services as lubricating oil should also be done. Chemical cleaning of field-assembled systems should be considered very carefully because the results can be very hazardous if any of the chemical solution enters the compressor during startup operations. Proper drainage and venting provisions are essential for effective and safe removal of cleaning fluids. Chemical cleaning, moreover, does not prevent trouble from mud, stones, welding rods, and slag.

Small piping systems such as sealing, venting, drain, and control connections must be checked for continuity and completeness. Frequently, these systems are not given enough attention. This kind of piping (usually field run) must be arranged so as to permit access to the compressor, as well as for lubrication and maintenance. The supports must be sturdy to avoid leakage or rupture from vibration during normal and upset operating conditions.

Instrumentation and Controls—For the proper functioning of instruments and controls: (1) conduct completeness and continuity checks—hydraulic, pneumatic, electrical; (2) commission, field-calibrate, and establish set points; (3) check for accessibility for operator use and vision, and for adjustment and maintenance; (4) check mounting locations, and methods to minimize vibration pickup—avoid mounting directly on the compressor, light platforms, or hand rails; (5) check the supply of instrument air and control power for adequacy and reliability.

Electrical Power and Lighting—For power, test for continuity, electrical-insulation soundness, proper grounding, and settings of relays (including the correctness of thermal overload trip devices), tightness and quality of all connections, sealing of fittings, and use of flexible connections for equipment and instruments.

As for lighting, check to see that it is adequate for those areas frequently missed by general lighting such as control stations, instruments, and lubrication points.

Thermal Insulation and Painting—Check to see that proper insulation will: (1) avoid hazard to personnel; (2) prevent thermal shock to the piping, compressor, and turbine from rain; (3) prevent fire from occurring as a result of oil spillage on a hot metal surface.

Avoid “overcompleteness” such as (1) paint on valve stems, instruments, etc.; (2) unnecessary insulation on flanges and flange bolting;

(3) external insulation on internally insulated brick or refractory piping and ducts; (4) insulation restraint on expansion joints; and (5) insulation that will limit the freedom of the required pipe movement.

DYNAMIC COMPRESSORS

INTRODUCTION

There are two forms of energy in any system. One is called potential energy and the other is called kinetic energy. For example, compressed gas in a static state exerts its pressure in all directions, as shown in Figure 1.8. When the outlet valve is opened, the gas flows out at very high velocity. Depending on the flow rate, the pressure in the cylinder drops down. In this case, the pressure energy is converted into kinetic energy. This kinetic energy is capable of doing work such as driving a pneumatic wrench, hammer, etc.

The higher the pressure, the higher will be the velocity and hence the kinetic energy of the gas leaving the system.

A dynamic compressor adds energy to gas in the same manner that an electric fan does. Consider a fan in operation and note the following points:

1. It is the rotating blades of the fan that force the air to move.
2. Air that is at rest tends to remain at rest.
3. As the fan blades start turning, they push on the air. The stationary air resists the push of the blades.
4. As the air resists the blades, the molecules of the air are brought closer together.
5. When the air molecules are compressed, the volume of the air decreases.
6. As the volume of the air decreases, its pressure increases.
7. The blades of the fan overcome the resistance of the air and thrust the air forward.
8. The faster the blades turn, the faster the air is pushed.
9. The fan, by doing work on the air, actually increases the pressure and velocity of the air.

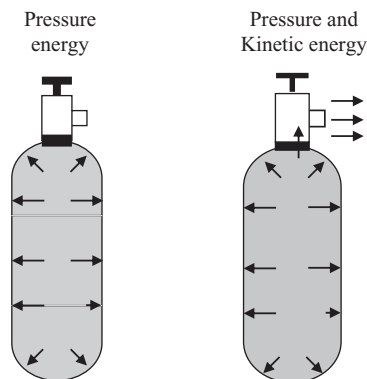


Figure 1.8. Two forms of energy

10. When velocity and pressure are added to a gas, its total energy increases.
11. A dynamic compressor increases the total gas energy by adding pressure and velocity to the gas.
12. The total energy of a gas leaving a compressor is greater than the total energy of the gas entering the compressor.
13. The energy that a gas gains in a compressor is due to the work done on it.

CENTRIFUGAL COMPRESSORS

If a body is set in motion, it tends to continue in motion unless some force acts in the opposite direction to stop it. If there is no gravity pull, nor any obstacle to deflect it, a body in motion travels in a straight line.

Suppose a ball attached to a string is set in motion, as shown in Figure 1.9. Assume that there is no gravity and that the string has no effect on the ball. The ball moves in a straight line.

Suppose the string is fastened to a fixed pivot point and then the ball is set in motion, as shown in Figure 1.10. At first, the ball moves in the direction of motion. When the string becomes taut, it deflects the ball. Because of the deflection, the ball actually travels in an arc, or a circle. Assuming it has enough energy, the ball continues to move in a series of arcs. At each instant of its travel, the physical tendency of the ball is to travel in a straight line. But instead, the ball travels in a circle because it is held or deflected by the string.

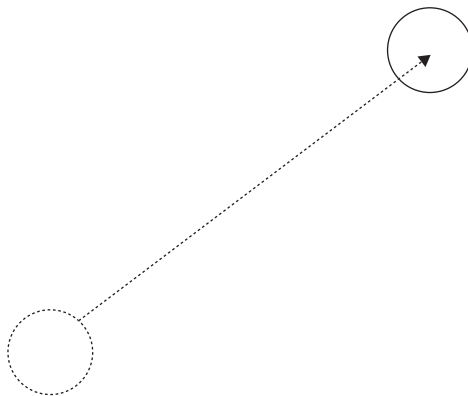


Figure 1.9. Motion of a ball in straight line

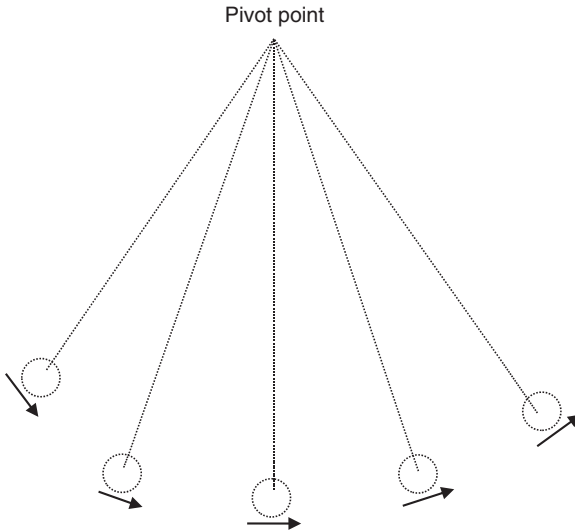


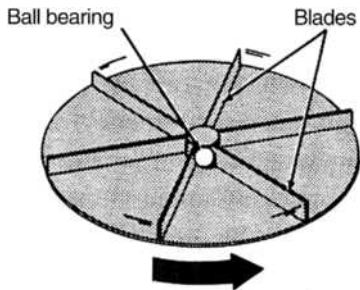
Figure 1.10. Centrifugal motion of a ball

The string actually applies centripetal (pulling-in-toward-the center) force, causing the path of the ball to change or curve. If the string breaks, the ball flies out in a straight line. Any object traveling in a circle is kept in that path of travel by a force called centripetal force. The force holding the ball in the circle of motion, that is, from the ball to the pivot point is the centripetal force.

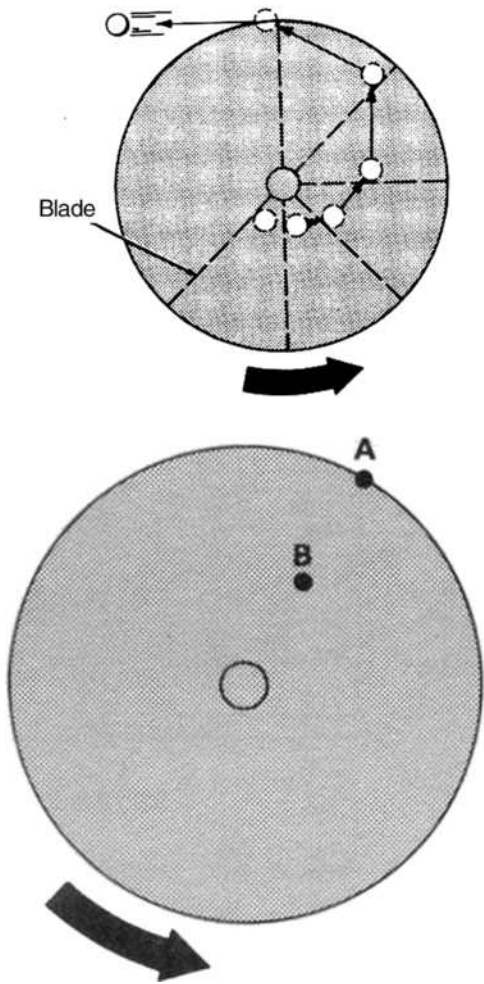
To hold the ball in its position and path, an opposite force is required. That force is called the centrifugal force. If the centripetal force is eliminated, the object then moves in a straight line.

The force pulling an object in a circular path toward the center is centripetal force. The centrifugal tendency of the object is its tendency to pull away from the center of rotation, or to pull against the centripetal force. The centrifugal tendency acts in the direction opposite to the centripetal force.

The centrifugal tendency is actually not a force but is the result of the tendency of the object to move in a straight line while being pulled toward a center of rotation by the centripetal force. Assume a ball bearing is placed close to the center of a disk that has blades, as shown in the figure below. As the disk begins to move, one of the blades forces the ball bearing to move. The ball bearing tends to travel in a straight path. The drawing shows the actual path of the ball bearing as the disk rotates.



When the disk rotates, the bearing is forced away from the center of the disk, as shown in the figure below.



Let us consider two points A and B located on the disk. Point A is at the tip of the disk, while point B is closer to the center of the disk.

When this disk starts rotating, as shown in the figure above, point A covers a larger distance than point B. When the disk is rotating, point A moves faster than point B. Anything that is being carried along by the rotation of the disk has a greater velocity when it is near the outer rim of the disk. If anything being carried along by the rotation of the disk also travels outward from the center to the outer rim, it gains velocity.

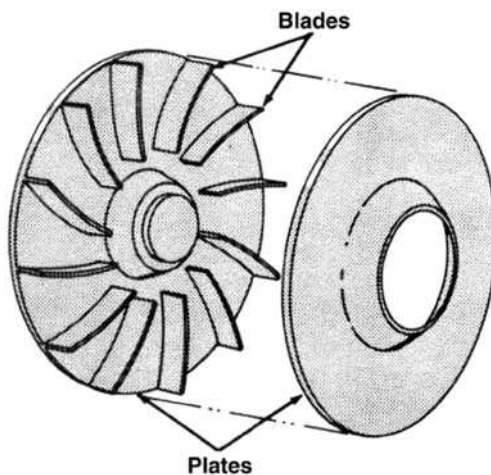
The velocity of point A is proportional to the RPM (revolution per minute), or the rotating speed of the disk. The energy picked up by the material at point A is given by

$$KE = m \times v^2/2g_c$$

If D is the diameter of the disk in meters and N is the RPM of the disk, the longitudinal velocity is equal to $\pi \times D \times N/60$ m/sec (i.e., $3.28 \times \pi \times D \times N/60 = 0.17181 \times D \times N$ m/sec. The kinetic energy gained will be $m \times v^2/2g_c$.)

To achieve this kinetic energy, work has to be done on the disk or impeller.

The figure given below is a compressor impeller. An impeller is made of two plates separated by blades.

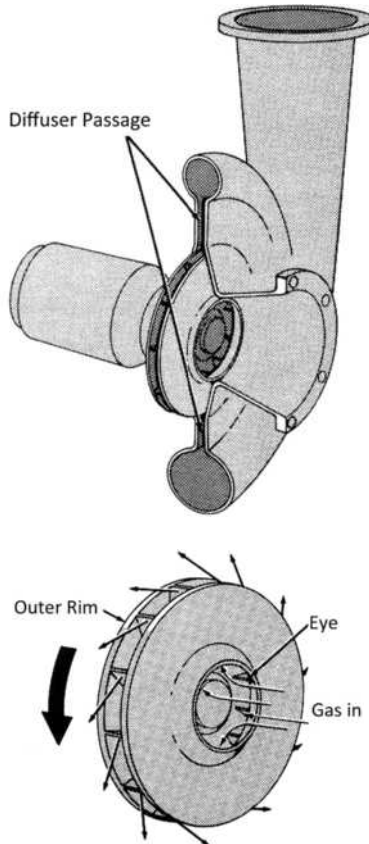


When the impeller begins to rotate, the blades force the air in the impeller to move. Air molecules tend to travel in a straight line. Because there is no centripetal force, the rotation forces the air molecules outward from the center, or eye, of the impeller. As the air molecules move outward, they gain velocity, or speed. The air also tends to oppose the push of the blades, so the pressure of the air is increased. The impeller adds both pressure and velocity to the air.

The tendency of air or gas to move outward from the center of a rotating impeller is the centrifugal tendency. A compressor that uses centrifugal tendency to impart pressure and velocity to a gas is called a centrifugal compressor.

The part of the centrifugal compressor that moves the gas is the impeller. As the impeller rotates, it moves the gas toward its outer rim. As the gas moves toward the outer rim of the impeller, its velocity increases.

This increase in velocity away from the eye creates a low-pressure area at the eye. This low-pressure area causes a suction, which allows more gas to enter. The impeller does work on the gas. The work is converted into the energy that the gas gains, which is in the form of both pressure and velocity. When the gas is at the tips of the impeller blades, it is at maximum velocity. As the gas leaves the impeller, it is thrust into a passageway called the diffuser (refer figures given below). When the gas enters the diffuser, the impeller is not acting directly on the gas.



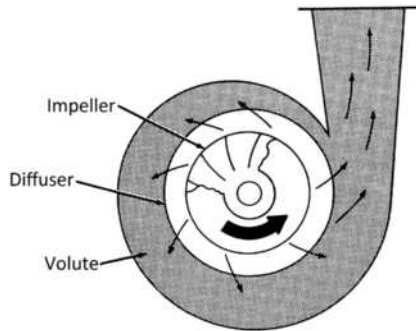
The radius of the diffuser is larger than the radius of the impeller. Due to the larger radius, the flow path of the gas through the diffuser is in a larger spiral. Since the flow path is longer and there is no direct action by

the impeller blades, the velocity of the gas decreases. As the velocity of the gas decreases, its pressure increases.

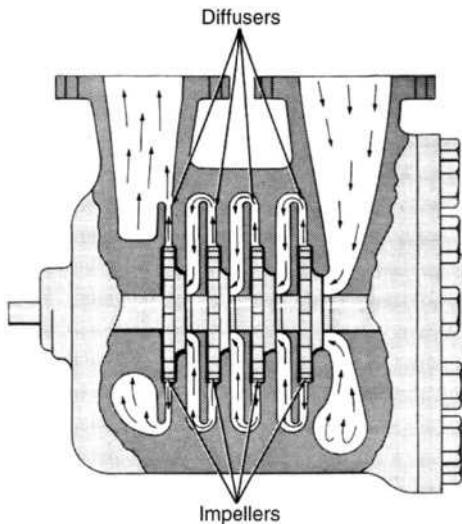
The diffuser converts the velocity of the gas to increased gas pressure. Gas passes from the diffuser into the volute. In the volute, the conversion from velocity to pressure continues.

Gas passes from the diffuser into the volute as shown below (single stage/last stage of a multistage compressor).

In the volute, the conversion from velocity to pressure continues. In a centrifugal compressor, work is done on a gas to impart both pressure and velocity.



A centrifugal compressor, by doing work on gas, imparts both pressure and velocity to the gas. Then, the velocity of the gas is converted into pressure within the compressor. Look at the compressor below.



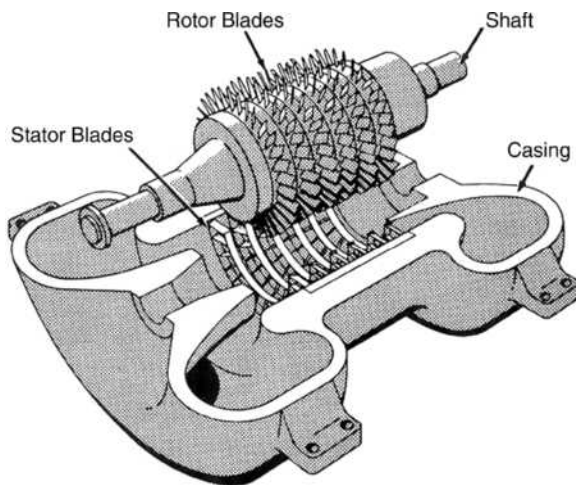
- It has four separate impellers.
- Each impeller and diffuser makes a stage.
- This is a four-stage centrifugal compressor.

As the gas leaves the first impeller, it gains some velocity and pressure. The increased velocity is partially converted into pressure in the diffuser.

As the gas leaves the diffuser, it enters the return passage, which guides it into the eye of the next impeller. When the gas enters the eye of the second impeller, it has greater pressure than when it entered the eye of the first impeller. Each impeller adds to the total energy of the gas. It may be noted that the velocity added by the impeller is converted into pressure energy within the diffuser. When the gas leaves the compressor, its pressure is higher than the inlet pressure. The work done by a compressor is the total energy added to the gas through impellers. A gas leaving the compressor has added energy in the form of pressure and temperature.

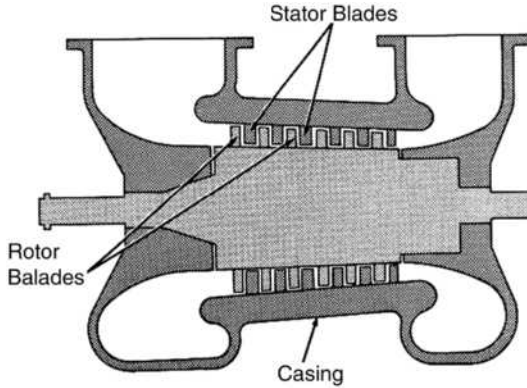
AXIAL COMPRESSORS

Motion along the axis of a shaft is called axial motion. This takes place in a straight line. A compressor in which the gas moves parallel to the axis of its shaft is called an axial compressor. An axial compressor has stator and rotor blades, as shown below.



The rotor blades are attached to the shaft and rotate with the rotary motion of the shaft. The stator blades are attached to the casing, as shown in the above figure.

The arrangement of the blades is such that there is a set of stator blades between each two sets of rotor blades, as shown in the figure below.

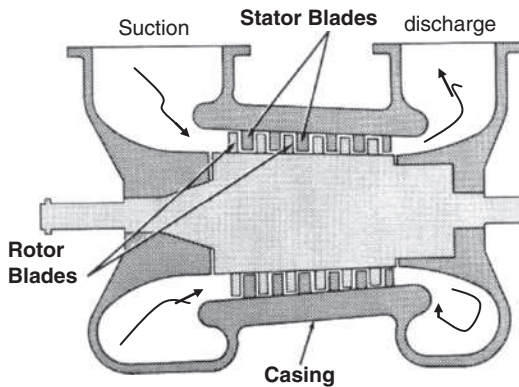


The rotor blades behave in the same manner as the blades of a fan. As they rotate, they force the gas to move. The rotor blades impart both pressure and velocity to the gas.

The rotor blades force the gas into the stator blades.

As the gas is thrust into the stator blades, the openings between the blades act as diffusers and reduce the velocity of the gas. With the decrease in the velocity, the pressure of the gas increases. The stator blades guide the gas into the next set of rotor blades. The gas entering the second set of rotor blades has a slightly higher pressure. Thus, each set of stator and rotor blades increases the gas pressure.

In axial compressor, the pressure increase of a gas is achieved by using many sets of stator and rotor blades. The blades in an axial compressor are not of the same size. The blades get gradually smaller toward the discharge end of the compressor, as shown below.



As the gas flows through an axial compressor, it occupies less volume successively in its flow path. Thus, the gas pressure increases. The

flow of gas in an axial compressor is somewhat linear and in the direction of its axis.

REVIEW

Two forms of gas energy are pressure and velocity. Energy cannot be created or destroyed, but it is convertible from one form to another. By doing work on a gas, the compressor adds energy to the gas. The total energy of a flowing gas is a function of its pressure, velocity, and temperature. Where the gas velocity reduces, pressure increases (Bernoulli's Theorem).

RATIO OF COMPRESSION

While the difference between suction and discharge pressures denotes the work done on a gas system, the ratio of absolute discharge pressure to absolute suction pressure is known as compression ratio. When a gas is compressed, part of the energy input or work done is converted into heat and friction losses.

The ratio of compression, R , is the relation between the absolute discharge pressure and the absolute suction pressure. If P_2 is the discharge pressure and P_1 is the suction pressure, then the compression ratio $R = P_2/P_1$.

This means the compression ratio denotes how many times the discharge pressure is greater than the suction pressure. In determining compression ratio, only absolute pressures must be used. To get absolute pressure, add the atmospheric pressure to gage pressure.

For example, compressor discharge pressure = 300 psig: Absolute discharge pressure = $300 + 14.7 = 314.7$ psia.

Example:

Air is compressed to 100 psig using an air compressor. What is the compression ratio of the compressor?

Air enters at atmospheric pressure. Therefore, $P_1 = 14.7$ psia

Discharge pressure = 100 psig = $100 + 14.7 = 114.7$ psia

Therefore, compression ratio $R = 114.7/14.7 = 7.81$

Example (metric units):

Air is compressed to 6 kg/cm²g using an air compressor. What is the compression ratio of the compressor?

Air enters at atmospheric pressure. Therefore, $P_1 = 1.033$ kg/cm²a

Discharge pressure = 6 kg/cm²g. Absolute pressure = $6 + 1.033 = 7.033$

Compression ratio $R = 7.033/1.033 = 6.81$

Note how the pressures have been converted into absolute pressures.

COMPRESSOR CAPACITY

The capacity of a compressor is the volume of gas that it handles in a given period of time. For example, CFM indicates the volume of the gas handled by the compressor in 1 minute. The flow rate of a gas in CFM in a pipe line depends on the velocity of the gas and the diameter of the flow path. For the same velocity, the rate in CFM is higher for a larger diameter passage.

Example:

Air passes through two pipe lines, one with 6 in. diameter and another with 8 in. diameter. The velocity of air in both the pipes is 500 ft/min. What is the flow rate of air through these two pipe lines?

Denote the diameter of the 6 in. pipe as d_1 and that of the 8 in. pipe by d_2 .

- a. Area of cross-section of the 6 in. pipe $= \pi \times d_1^2/4 = (22/7) \times (6/12)^2/4$
 $= 0.19643 \text{ ft}^2$
- b. Volume of air flow = velocity $\times$ cross-section area $= 500 \times 0.19643$
 $= 98.2 \text{ cfm}$
- c. Area of cross-section of the 8 in. pipe $= \pi \times d_2^2/4 = (22/7) \times (8/12)^2/4$
 $= 0.34921 \text{ ft}^2$
- d. Volume of air flow = velocity $\times$ cross-section area $= 500 \times 0.34921$
 $= 174.6 \text{ cfm}$

Result: Air flow rate in 6 in. and 8 in. diameter pipes for the velocity of 500 ft/min are 98.2 and 174.6 cfm, respectively.

If the gas velocity is greater, then the pressure at the discharge section is lower.

During compression, the volume of gas entering the compressor is greater than the gas leaving the discharge. The ACFM is measured at the suction of the compressor.

HEAD OF COMPRESSION

For compressing a gas to higher pressures, certain amount of work has to be done on the gas. The work done on a gas may be expressed as foot pounds or kilogram meters.

When 1 lb of gas is lifted or moved to a distance of 1 ft, then the work done by the compressor is 1 ft·lb. Similarly, when 10 lb of gas is moved to a distance of 1 ft, then the work done by the compressor is 10 ft·lb. When 1 lb of gas is lifted/moved to a distance of 10 feet, then also the work done by the compressor is 10 ft·lb.

The head developed by a compressor is the distance or height to which a column of gas can be moved at the average density of gas. This is an important factor in calculating the head developed by a compressor. For each pound of gas the compressor raises the head, a corresponding amount of work in foot pounds has to be done on the gas. As shown above, if the head increases, the number of foot pounds of work to be done on the gas per pound also increases. Pressure may also be converted to head, as shown in the example below.

Example:

1,000 cfm of air is compressed by a compressor to a pressure of 6 kg/cm²g pressure. What is the work required to be done by the compressor on the air and what is the head developed?

- I. Air inlet pressure = atmospheric pressure = 14.7 psia
- II. Air outlet pressure = $6 \times 14.22 + 14.7$ psia = 100 psia
- III. Air density at atmospheric pressure = $28.84/359$ lb/ft³ = .0803 lb/ft³
- IV. Weight flow rate of air = $1,000 \times 0.0803$ lb/min = 80.3
- V. Air density at 100 psia = $28.84/50.19$ = 0.5746 lb/ft³
- VI. Average density of air = $(0.0803 + 0.5746)/2$ = 0.3274 lb/ft³

$$\begin{aligned}\text{Differential pressure developed by the compressor} &= 6 \times 14.22 \text{ psi} \\ &= 85.32 \text{ psia} \\ &= 12,286.08 \text{ lb/ft}^2\end{aligned}$$

$$\begin{aligned}\text{Dividing the above value by the average air density, we get} \\ &= 12,286.08/0.3277 \text{ ft} \\ &= 37,491.8 \text{ ft}\end{aligned}$$

$$\text{Work done on 1,000 cfm air} = 80.3 \times 37,491.8 \text{ ft·lb/min}$$

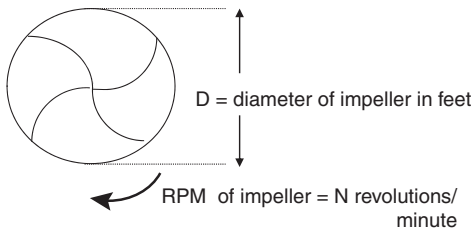
This is the work to be done by the compressor on 1,000 cfm of air.

RPM AND HORSEPOWER

For all rotating machines, RPM is an important parameter. The impeller of a centrifugal compressor has to rotate (revolve) to move the gas.

As the RPM of the impeller increases, the velocity of the gas also increases.

Gas velocity is proportional to the RPM of the impeller, as shown below.



$$\text{RPM of impeller} = N \text{ revolutions/minute}$$

$$\text{Gas velocity at the tip of the impeller} = \pi \times D \times N \text{ ft/minute}$$

Diameter	RPM	Gas velocity (feet/min)
5	8,000	1,25,714
5	10,000	1,57,142
5	12,000	1,88,571

For any given RPM, a set amount of work in foot pounds or kilogram meter is done on the gas per unit weight. Since this velocity is converted into head or pressure, for a constant RPM, the head developed is fairly constant.

Whether the gas is heavier or lighter, the work done per pound of gas is the same for the same RPM.

At a given RPM, the ACFM of gas the compressor compresses will be constant. But the weight of the gas compressed will be more, in the case of a heavier gas. When more weight of gas is compressed, the work done on a heavier gas will be more.

When the rate of work done on a gas increases, the horsepower required to compress a heavier gas also increases.

Example:

Gas	Vol flow (cfm)	RPM	Density (lb/ft ³)	Flow (lb/min)	Work (ft·lb/min)
Hydrogen	1,000	8000	0.0056	5.60	5.60
Ammonia	1,000	8000	0.0473	4.73	4.73
Air	1,000	8000	0.0803	8.03	8.03

The density of a gas does not affect the head developed, but affects the discharge pressure of the compressor.

Two identical compressors handle 200 cfm of air and hydrogen at 12,000 rpm and as per the compressor characteristics; the differential head developed is 20,000 feet at discharge conditions. What will be the discharge pressure under these conditions? (Densities at discharge conditions for air and hydrogen are 0.65 and 0.15 lb/ft³, respectively.)

Case 2.

Discharge head for hydrogen = 20,000 ft: Density = 0.15 lb/ft³

Discharge pressure = head x density = 20,000 × 0.15 lb/ft²

= 3,000 lb/ft²

= 20.83 lb/in.² = 1.46 kg/cm²g

BHP refers to the break horsepower that is required by the compressor shaft to achieve the desired compression ratio. Because the gas density fluctuates very often in industrial systems, a centrifugal/dynamic compressor tends to change power consumption while in operation.

The ratio of compression, R , is the absolute discharge pressure divided by the absolute suction pressure. R is an indicator of the amount of pressure that the compressor adds to the gas.

At a given RPM, a dynamic compressor adds a certain head to the gas. The total head added depends on:

- The design of the compressor
- The amount of gas flow and
- The operating RPM (speed)

As RPM increases, the total head developed by the compressor increases.

At a fixed RPM and CFM, the compressor attains approximately the same feet of head, regardless of the weight of the gas handled.

The head developed by the compressor does not depend on the density of the gas being handled.

The feet of head or the meter of head can be converted into PSI or kg/cm^2 equivalent.

Similarly, PSI or kg/cm^2 can be converted into feet or meters, respectively.

When a compressor at a given RPM is handling a heavier gas, the work it does over a pound of gas is the same as the work done on a pound of lighter gas.

Head represents the amount of foot pounds or kilogram meter of work done per unit weight.

A compressor at a given RPM handles two different gases. The gas that requires a larger volume per unit weight is the lighter gas.

SURGING

When a compressor is connected to a large system, which needs a large amount of gas for the process, the gas intake will vary with the process requirement. During startup, the gas demand will be very high. Hence the compressor will run at its full capacity.

As the process proceeds, the demand for gas will start reducing. If the system does not use the gas as quickly as the compressor delivers, the system pressure will increase.

With the increase in the system pressure, the resistance to discharge of the compressor increases.

Due to this, the compressor capacity reduces.

When the head required maintaining the flow increases above the maximum head of the compressor, the gas flow stops.

Under this condition, the pressure within the compressor becomes less than the system pressure. This results in the gas flow from the system to the compressor, called “flow reversal.”

When some quantity of gas has gone to the compressor section, the system pressure will drop.

When the system pressure becomes less than the maximum head of the compressor, the compressor starts delivering the gas to the system.

The compressor operates at a lower capacity and higher head/pressure.

If the system pressure system builds up again to a high pressure, the flow reversal will repeat. The cycle continues.

The rapid flow of gas back and forth the compressor is called surging.

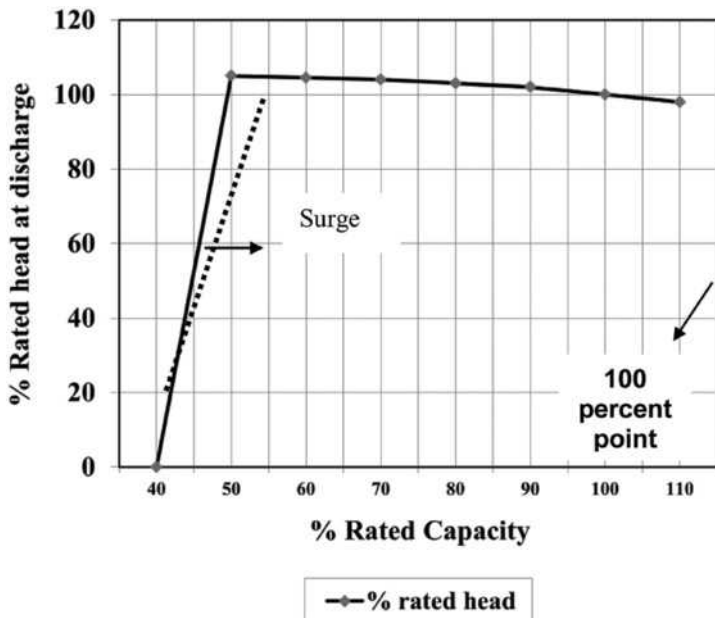
Surging occurs when the compressor operates below the minimum capacity.

The rapid reversals of surging set up severe vibrations in the compressor and piping, which may cause damage to the compressor.

A compressor goes into surging because the flow of gas drops down below the minimum stable limit.

Most compressors are protected against surging by anti-surge control methods as explained below.

Refer to the figure shown below.



Y axis shows the percentage of rated head and x axis shows the percentage of rated capacity of the compressor. A point higher on the graph shows higher head. The compressor capacity varies between 40 to 110 percent. The curve shows, as the flow increases, head decreases.

The compressor is discharging into a system that requires 100 percent of its rated head. According to the graph, the compressor is operating at 100 percent of its capacity.

Suppose the discharge system does not use as much gas as the compressor delivers the pressure at the discharge end of the compressor increases as may be seen from the graph.

When the gas flow reduces to 90 percent, the head developed increases to 102 percent of the rated head. As the gas discharged from the compressor reduces, the pressure developed by the compressor increases. When the flow reduces to 50 percent of the rated flow/capacity, the compressor no longer delivers the gas to the system and virtually the gas flow stops.

When the compressor pressure becomes lower than the system pressure, the gas will flow from the system to the compressor (flow reversal).

When this condition is reached, the system pressure will start dropping down, and the compressor will discharge gas to the system again. This repeated process of forward and reverse flow is called “surging.”

INTERPRETATION OF CURVES

Performance curves show the limits of the compressor, which are the surge point on the curve and the normal capacity limit on which the compressor must operate.

Compression is controlled by making permissible changes in pressure, flow, and temperature to keep the compressor from surging.

SURGE CONTROL

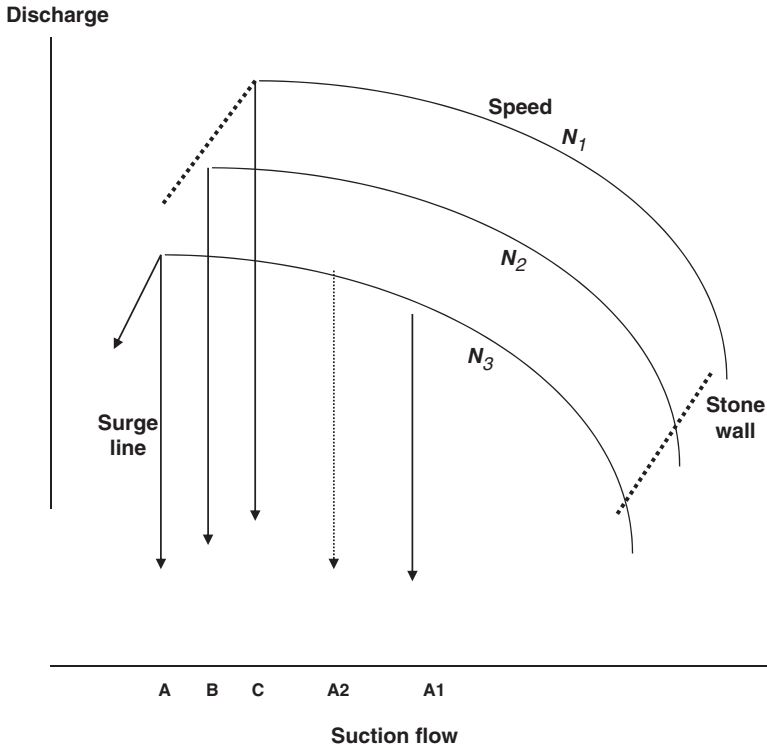
A centrifugal compressor has continuously falling pressure flow characteristics. A family of parallel curves represents a compressor operating at different speeds.

For a compressor operating at constant speed, the pressure increases as the flow decreases and vice versa.

If the reduction in flow continues, at one point, the compressor operation becomes unstable and a momentary flow reversal takes place. This leads to a very rapid pulsating flow backward and forward through the compressor internals, resulting in severe vibration and consequent damage.

This phenomenon, known as “surge,” occurs at a surge point at a particular flow and a particular speed.

The surge point shifts to the right as the speed increases, as shown in the figure below.



Suppose the compressor is running at RPM N_3 and the inlet gas flow rate is A1. When the flow rate is reduced due to lower system intake, the flow rate drops to A2. When the flow drops down further to the level A, on RPM N_3 , the surge limit is reached. This results in a drastic drop in the gas flow and the pressure leading to flow reversal from the system to the compressor.

The line joining the surge points of curves at different speeds is the compressor surge line. To avoid surge, compressors are always operated at flows greater than the surge point, that is, at a reasonable distance to the right of the surge line.

As the flow increases, a point is reached at the right-hand extreme of the compressor curve, after which the flow cannot increase, no matter what changes are made on the outlet side. The point is known as stonewall for the compressor. No adverse effect occurs at stonewall conditions. The operating regime of a centrifugal compressor lies between two limits: surge limit and stonewall.

SURGE CONTROL TYPES

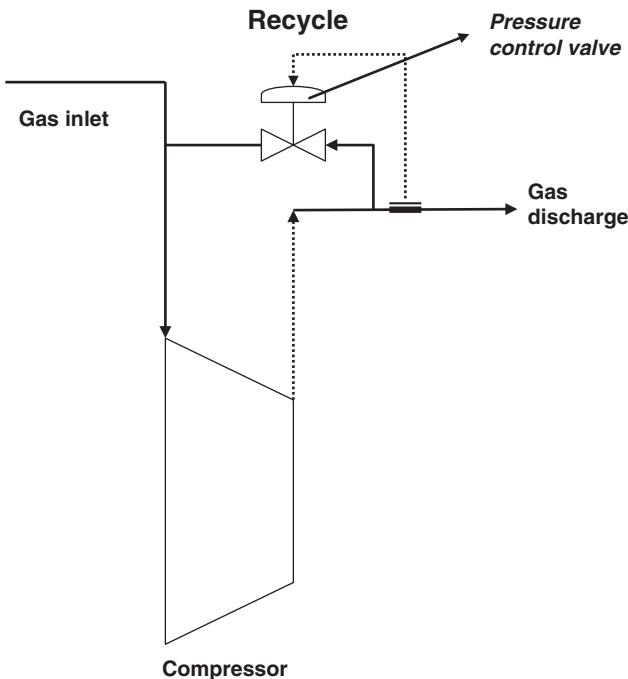
RPM Control

In this method, when the gas flow rate starts coming down, the RPM (speed) of the compressor is increased by sending the signal to the speed control governor. This will increase the steam rate and the RPM for the same flow rate of gas.

Recycling/Pressure Control

In this method, the discharge pressure is controlled by a spill-back mechanism, which recycles the excess gas back to the suction, as shown in the figure below. In this control system, the gas passing through the compressor is compressed to the required pressure and passes to the system. The pressure control valve located at the discharge section will maintain the header pressure.

When the pressure starts increasing, the spill-back control valve will open and recycle the excess gas to the suction, without affecting the system pressure.



TROUBLESHOOTING COMPRESSOR PROBLEMS

Normal problems associated with compressors may be broadly classified into two categories: mechanical problems and operational problems.

In industrial practice, centrifugal compressors are usually run by steam turbines. Motor-driven centrifugal compressors are also used in small-capacity compressors. The problem associated with the compressor may not necessarily be due to the compressor alone, but may be due to the accessories also. Hence, for troubleshooting compressor problems, related accessories must also be checked. A checklist for the compressor and accessories is given below.

Mechanical Problems

Mechanical problems associated with the compressors are:

- Excessive vibrations of bearings
- Axial displacement of compressor rotor
- Bearing running too hot
- Bearing running rough
- Pulsating pressure
- Too low discharge pressure
- Casing temperature running high
- Excessive noise in running

Excessive Vibration of Bearings

All high-capacity compressors are provided with a trip mechanism when the vibration goes above a stipulated level. Vibration increase is a gradual process. If the vibration shows an increasing tendency, check all the bearing temperatures, gas flow rate, pressure developed, etc.

Normally, vibration will start only if some misalignment/roughness of the bearing has developed. Vibration may also occur when the compressor surges.

If the compressor surging is frequent, the bearing alignment and/or the condition of the bearing may have deteriorated, resulting in excessive vibration. When the vibration starts increasing, the compressor must be taken out of service and inspected for bearing conditions and alignment.

Axial Displacement of the Rotor

All centrifugal compressors are provided with an axial displacement trip system. Up to 0.5 to 1.0 mL of displacement is tolerated in certain machines. When the displacement exceeds this limit, the compressor will trip automatically. Axial displacement takes place from the high-pressure end to the suction side (low pressure). This displacement is prevented by a thrust bearing, provided in the suction end. If the thrust bearing is damaged due to surging, critical speed, etc., the displacement of the rotor will tend to move toward the suction end. When the displacement exceeds a certain limit, the rotor blades will tend to contact the diffuser. Due to the high speed of the machine and the friction heat generated, there will be total damage to the compressor if it fails to trip on the stipulated displacement limit. Frequent compressor surging of the compressor increases the axial displacement.

Bearing Running Too Hot

Most compressors are provided with bearing temperature indicators or local bearing thermometers. When the temperature of the bearing shows an increasing tendency, the immediate step is to check for proper bearing lubrication. Inadequate and/or improper lubrication may lead to high bearing temperatures.

An experienced person will be able to even foresee the consequences of high bearing temperature. From a historic log data of the compressor, it is possible to determine whether or not the compressor is running or not.

The following table is a historic reading of a compressor, which gives the most important readings of a compressor. The bold lines indicate the design/stipulated values of the compressor operating parameters.

Month	Bearing 1 Temp °C	Bearing 2 Temp °C	Displace- ment mL	Vibration Bearing 1 mL	Vibration Bearing 2 mL
Maximum limit	100	100	0.05	1.5	1.5
1	90	82	0.01	0.028	0.01
3	105	85	0.01	0.250	0.05
5	112	87	0.02	0.580	0.07
7	121	89	0.034	1.05	0.09

Compressor RPM 15,000/Full Load

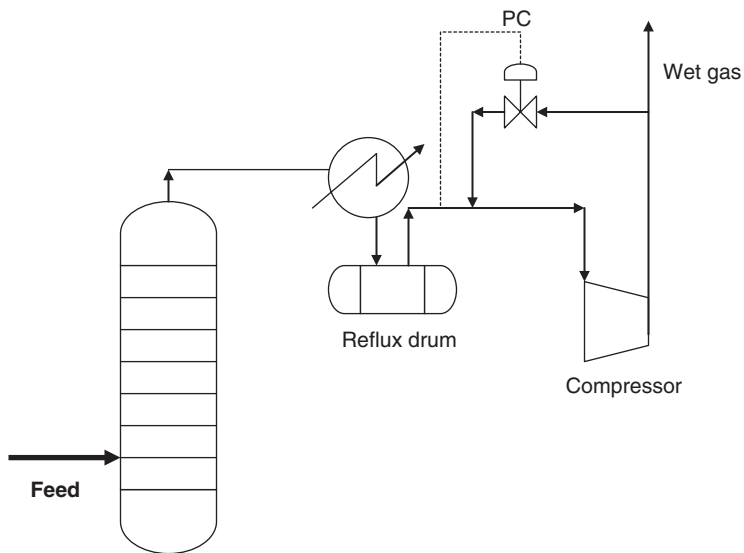
It may be noted from the above data that bearing 1 has a problem, which is reflected in the vibration levels. Another indirect method of bearing metal loss is to collect the lube oil sample from the abnormal section and analyze for the presence of metals. If the lube oil shows the presence of any metal, matching with that of the bearing, it may be assumed that the bearing thickness is deteriorating and may fail at certain point of time.

Surging in Compressors

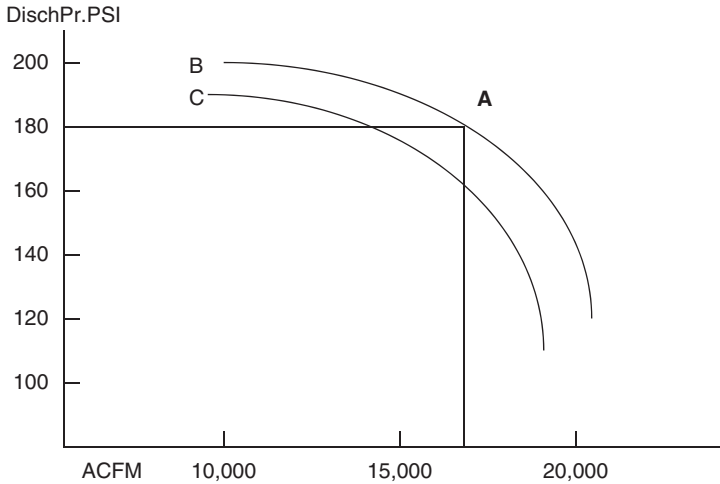
Besides flow variations, changes in the molecular weight of the gas being compressed may also lead to surging, as shown below. The process conditions linked to the compressor play an important role in compressor surges.

Example:

The process refers to a gas compressor, which compresses a mixture of methane, ethane, propane, and propylene coming from a cracking process. The gas mixture is compressed to recover heavy hydrocarbons from the mixture by condensation in a condenser and to recover the dry gas for further processing. The compressor is located as shown.



The following figure is the characteristic curve of the gas compressor shown above.



Observations: Point “A” is the operating point
Points “B” and “C” are surge points

OPERATING PARAMETERS

Gas flow: 17,000 acfm
Molecular weight of gas: 34
Suction pressure: 20 psig
Discharge pressure: 200 psig

The compressor develops 180 psig differential pressure under normal operating conditions as given above. The discharge pressure is fixed by the absorber pressure. Suction pressure is controlled by the PC, as shown above.

The compressor handles 17,000 acfm of gas, of which 10,000 acfm goes to the absorber, while 7,000 acfm is recycled.

When the drum pressure starts rising, the PC will start closing. This will result in low gas inflow and the operating point will move to the left, toward the surge point.

When the molecular weight starts decreasing, the differential pressure will also start decreasing, and at certain point of time, the compressor will start surging.

Hence, it is necessary to check the molecular weight of the gas regularly to avert compressor surging and damage to the compressor.

Similarly, lube oil analysis data for:

Flash point

Water content

Impurities and wear particles

Viscosity and viscosity index

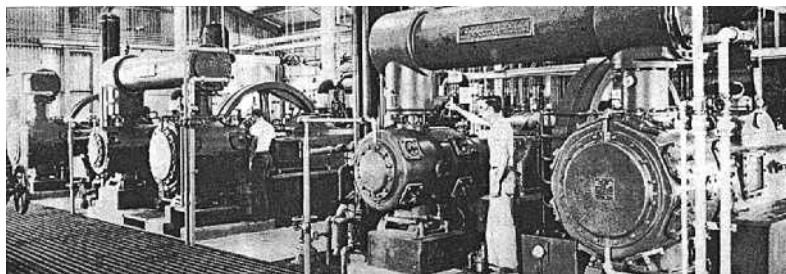
Total acidity may be used to diagnose the compressor-related problems for remedial action.

CHAPTER 2

COMPRESSOR EFFICIENCY

DEFINITION MAKES A DIFFERENCE





Compressor efficiency ratings can be as misleading as interest rate figures. To be certain of getting the most for your money when buying compressors, you must be sure that efficiency comparisons are made on the same basis.

The compression of air and other gases consumes a significant amount of power in the chemical process industry. In the production of industrial gases, such as oxygen, nitrogen, and helium, or in the rapidly growing field of natural-gas liquefaction, compression power accounts for more than 80 percent of total energy requirements. Clearly, compressor efficiency has a direct impact on product costs.

This article compares isothermal and adiabatic efficiencies, shows the effect of intercooling between stages of compression, and provides convenient ways for obtaining preliminary figures on compression power.

ISOTHERMAL COMPRESSION

The basic equation used to calculate the work of gas compression in a steady-flow process (and flow through a reciprocating compressor may be considered steady, despite the pulsations of the individual compression strokes) is:

$$w = \int_1^2 v \, dp \quad (1)$$

where w is the work required per lb-mole, v the molal volume, and p the pressure.

For preliminary calculations, the ideal-gas equation of state may be used to relate pressure, volume, and temperature:

$$Pv = RT \quad (2)$$

In fact, this equation will yield quite accurate results at pressures below 10 atm and at temperatures well above the critical temperature.

If the gas could be compressed without friction and isothermally, the work required to raise the pressure from p_1 to p_2 is:

$$w_{iso} = RT_1 \ln (p_2/p_1) \quad (3)$$

In this equation, R is the gas constant, 1.986 Btu/(lb-mole) (°R); T_1 the initial temperature, °R; and work has the units Btu/lb-mole.

The isothermal efficiency of a compressor is defined as the ratio of work calculated by Eq. (3) to the actual work required, or:

$$\eta_{iso} = w_{iso}/w_a \quad (4)$$

In many cases, the numerical value of isothermal efficiency may appear to be low, for example, 65 percent, and yet the actual work required may be less than that of a compressor with an adiabatic efficiency of 80 percent. We shall see how this apparent discrepancy comes about.

ADIABATIC COMPRESSION

An ideal gas compressed adiabatically and without friction requires work according to the expression:

$$w_{ad} = RT_1 \left(\frac{k}{k-1} \right) \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] \quad (5)$$

where k is the ratio of specific heats, C_p/C_v , and all the other terms are as defined before.

Adiabatic efficiency is defined similarly to isothermal efficiency:

$$\eta_{ad} = w_{ad}/w_a \quad (6)$$

It is clear that the two definitions of compression efficiency vary only in the standard of comparison, that is, ideal isothermal work versus ideal adiabatic work. Figure 2.1 shows the variation in the ideal adiabatic work as a function of the pressure ratio, p_2/p_1 , and the specific heat ratio, k . In all cases, this work is greater than the isothermal work shown by the lowest curve. The influence of the specific heat ratio is pronounced. Monatomic gases, such as helium and argon, have $k = 1.66$, and therefore, the adiabatic

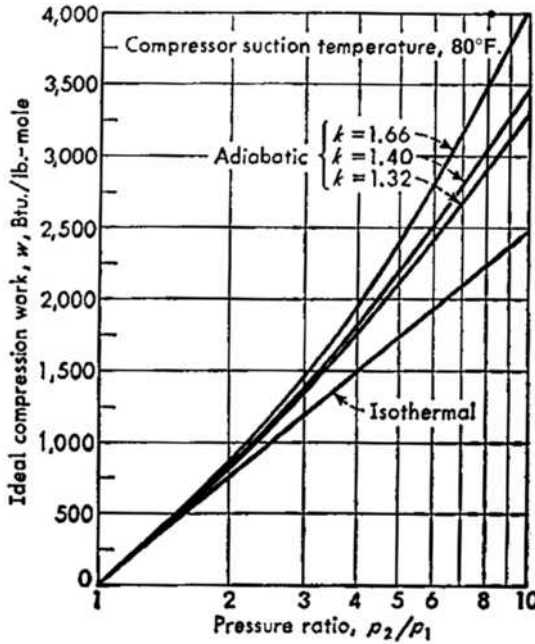


Figure 2.1. Adiabatic efficiency is greater than isothermal work

work for the compression of such gases is significantly greater than the isothermal work. Diatomic gases, such as nitrogen, oxygen, hydrogen, and air, have $k = 1.4$. More complex gaseous molecules have a lower value of k , such as 1.32 for methane.

As an example, consider the compression of air at 14.0 psia and 80°F to 56 psia. From the appropriate equation, or from Figure 2.1 with $k = 1.40$ and $p_2/p_1 = 4.0$, we find $w_{iso} = 1,487$ Btu/lb.-mole and $w_{ad} = 1,824$ Btu/lb.-mole.

If the actual work required by a compressor is 2,280 Btu/lb.-mole, the compressor efficiency may be expressed as $\eta_{iso} = (1,487/2,280)(100) = 65$ percent and $\eta_{iso} = (1,824/2,280)(100) = 80$ percent. Both of these efficiency definitions are equally valid, although the adiabatic efficiency is usually used when no effort is made to cool the gas during or between stages.

EQUAL-WORK EFFICIENCY RATIO

We have seen in the preceding example that for the same actual work of compression, the adiabatic and isothermal efficiencies vary by a factor

of 1.23. From Eq. (3–6), the equal actual work of compression, w_a , results when:

$$\frac{\eta_{ad}}{\eta_{iso}} = \frac{\left(\frac{k}{k-1}\right) \left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1 \right]}{\ln(p_2/p_1)} \quad (7)$$

The equal-work efficiency ratio given by Eq. (7) is plotted as a function of pressure ratio in Figure 2.2. With the same conditions as in the earlier example, $k = 1.4$ and $p_2/p_1 = 4.0$, the equal-work efficiency ratio is 1.23.

To illustrate how the equal-work efficiency ratio may be used, consider the problem of evaluating compressor bids. Again, we shall compress air from 14.0 psia and 80°F to 56 psia. If one vendor guarantees 80 percent adiabatic efficiency and another 70 percent isothermal efficiency, the quotation of the latter must indicate a lower power requirement, despite the lower efficiency. We found that the equal-work efficiency ratio for these conditions is 1.23. The ratio of quoted efficiencies is 1.14. Whenever the actual ratio is less than the equal-work ratio, the compressor whose efficiency is based on the isothermal efficiency will require less power. Above the equal-work curve, the adiabatic efficiency machine requires less power.

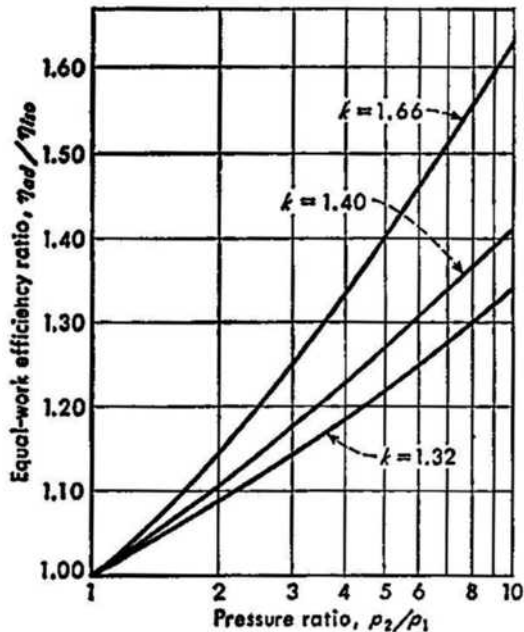


Figure 2.2. Ratios for the actual work of compression

MULTISTAGE COMPRESSION WITH INTERCOOLING

Whenever the overall pressure ratio exceeds 4, consideration must be given to multistage compression with intercooling. This is necessary to keep the temperature of the compressed gas at a safe value, for example, 365°F for air compressors with hydrocarbon lubricants as well as to reduce the overall power requirement.

Ideal intercooling would reduce the temperature between stages to the suction temperature. The effect of staging with intercooling is shown in Figure 2.3. The ideal work of compression in steady flow, given by Eq. (1), is represented by the areas to the left of the curves. The isothermal curve represents the minimum work requirement, whereas the adiabatic curve encloses a much larger area. The intermediate dashed curve represents two-stage compression with intercooling to the suction temperature, 80°F.

Now let us consider a two-stage compressor, compressing air from 14.0 psia and 80°F to 140 psia, with intercooling to 80°F. Should the efficiency of a compressor be based on the ideal isothermal or the ideal adiabatic compression, as shown by the solid curves in Figure 2.3? Actually, we should refer to stage efficiency but clearly keep the overall efficiency in mind.

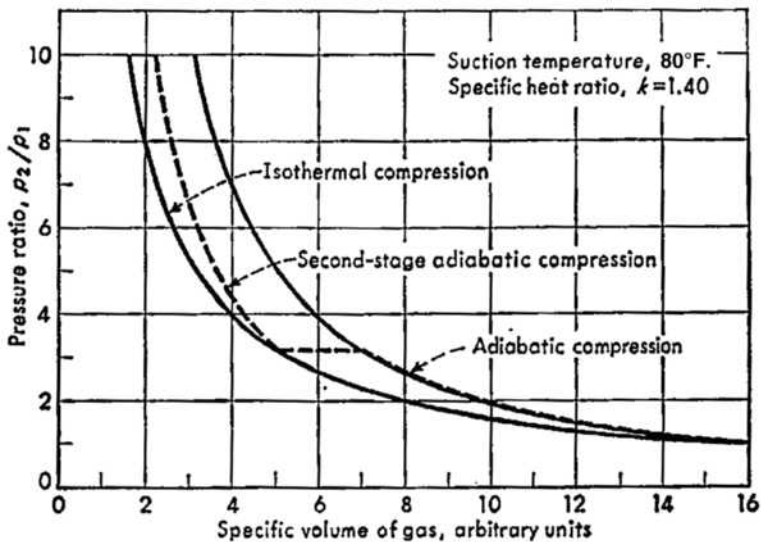


Figure 2.3. Dashed curve represents two-stage adiabatic compression with intercooling to the suction temperature; solid curves depict adiabatic and isothermal compression of gases

Because, in this example, the overall pressure ratio is 10.0, we may compute the ideal work of compression as: $w_{iso} = 2,470$ Btu/lb-mole and $w_{ad} = 3,480$ Btu/lb-mole.

These numbers may also be found from Figure 2.1. For each stage of compression, with a pressure ratio of 3.16, the ideal adiabatic work of compression is 1,460 Btu/lb-mole, or a total, for two stages, of 2,920 Btu/lb-mole. Using a realistic adiabatic stage efficiency of 80 percent, the actual work of compression is $2,920/0.80 = 3,650$ Btu/lb-mole. We now have three equally accurate, but quite confusing, compressor efficiencies:

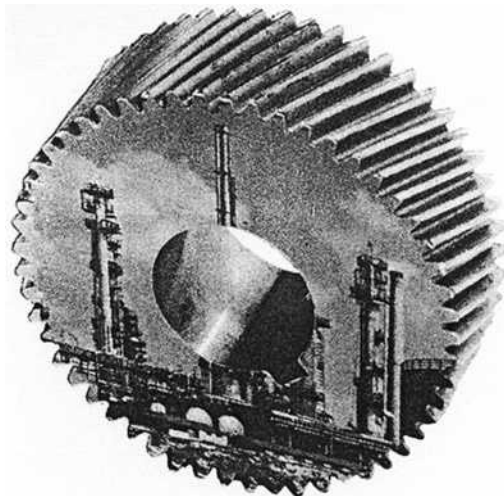
Adiabatic compressor efficiency: $(3,480/3,650) (100) = 95\%$

Adiabatic stage efficiency: $(2,920/3,650) (100) = 80\%$

Isothermal compressor efficiency: $(2,470/3,650) (100) = 68\%$

Efficiency is a word with many meanings. We have seen that a higher numerical value of efficiency based on one definition may actually represent poorer performance than a lower numerical value based on another definition. The equations and graphs presented in this discussion are designed to refresh practicing engineers on these points. There is no doubt, however, that for a given compression job, the compressor that requires the least horsepower is the most efficient.

HOW TO ACHIEVE ONLINE AVAILABILITY OF CENTRIFUGAL COMPRESSORS?



Here is practical information that will ensure the best selection and proper maintenance of centrifugal compressors and their drivers,

bearings, seals, couplings, speed gears, and lubrication and control systems.

Centrifugal compressors are an integral part of the chemical process industries (CPI). They are used extensively because of their smooth operation, large tolerance to process fluctuations, and higher reliability than other types of compressors.

Centrifugal compressors range in size from pressure ratios of 1.3:1 per stage to as high as 12:1 on experimental models. We will limit our discussion to compressors having pressure ratios below 3.5:1 because these are the ones used extensively in the CPI.

A proper selection of these compressors involves making complex decisions, since the successful operation of many CPI plants depends on the smooth and efficient functioning of such units. To ensure the best selection and proper maintenance of centrifugal compressors, the engineer must have a wide knowledge of many engineering disciplines.

SPECIFICATION CRITERIA

Detailed compressor specifications can vary from customer to customer. Some provide only the basic information, such as pressure, flow rate, type of gas, driver, and site conditions. Others present a lengthy document detailing types of bearings, rotor response, lubrication system, acceptable tolerance on performance, etc. The latter requires that the engineer be very conversant with compressors and their total support systems. A starting point for the specifications is to be found in the publications of the American Petroleum Institute (API) for turbomachinery. Some of the applicable ones are listed in Table 2.1.

Specifications for the API standards are written by user engineers, with the input of manufacturers and engineering contractors. Thus, the standards represent a wealth of experience and are a very good base from which to start turbomachinery specifications.

Many decisions, regardless of details contained in specifications, have to be made in advance by the engineer. Some of these decisions may involve company philosophy about various units; others could be strictly job oriented. In writing a job specification, we must have an understanding of the major problem areas. These will now be covered.

COMPRESSOR LAYOUT

The general topography of the plant must be known so that the proper site can be established. Whether the unit will be grade mounted or mezzanine

Table 2.1. Sources for compressor specifications

Standard	Title
API 611	General Purpose Steam Turbine for Refinery Services
API 612	Special Purpose Steam Turbine for Refinery Services
API 613	High-Speed Special Purpose Gear Units for Refinery Services
API 614	Lube and Seal Oil Services
API 616	Combustion Gas Turbines for General Refinery Services
API 617	Centrifugal Compressors for General Refinery Services
API 670	Noncontacting Vibrating and Axial Position Monitoring Systems

Available from: American Petroleum Institute, **2101** L St., N.W., Washington, DC 20037.

mounted is important in determining the foundation characteristics. Enough space should exist for the ducting, so that the inlet conditions to each stage allow the flow to enter without large distortions of velocity and pressure.

Accessibility requirements should be kept in mind so that repair and maintenance work on the unit can be performed with relative ease. The location of the oil system for the unit is also an important aspect. It is advisable to locate the oil reservoir away from the base plate, with the bottom sloped toward the low drain point. Enough space should be provided so that the return-oil lines can enter the reservoir away from the oil-pump suction. This would greatly reduce the disturbance of the pump suction, and also help in keeping the reservoir-retention time to around 10 minute.

COMPRESSOR ENVIRONMENT

The environment in which a machine operates is as important as any other factor. In many cases, this detail is often overlooked, or described in a phrase such as “extreme cold climate.” The vendor needs to know much more. He must know what extreme cold means (usually below -25°F) and what the transition weather is.

Weather creates many problems—though few precautions are taken because operating problems are either not recognized or are glossed over.

All cold-weather applications need to have some icing protection, especially in the air-inlet supplies. Fuel-supply ventilation, pneumatic controls, and actuators need to have some degree of deicing. Many types of deicing systems exist. The two most common use exhaust gases or compressor bleed-air.

Tropical climates present their own problems, such as excessive corrosion, high moisture content, and high ambient temperatures that increase both the horsepower required and the required cooling capacity of the lube-oil systems. Desert locations require special filtration systems to prevent the erosion of the blades and special sealing on joints to prevent micron-sized sand particles from entering the lube system, etc.

Machine usage is an environmental consideration because intermittent operation can be the most severe kind of service. The aggressiveness of the gas to be handled and its temperature determine the materials of construction to be used.

COMPRESSOR TYPES AND ARRANGEMENTS

It is not always obvious what type of compressor is needed for an application. Of the many kinds, some of the more significant are the centrifugal, axial, rotary, and reciprocating. Figure 2.4 will aid in the selection of a compressor. For very high flows and low pressure ratios, an axial-flow compressor would be best. Axial-flow compressors usually have a higher efficiency but a smaller operating region than a centrifugal machine. Centrifugal

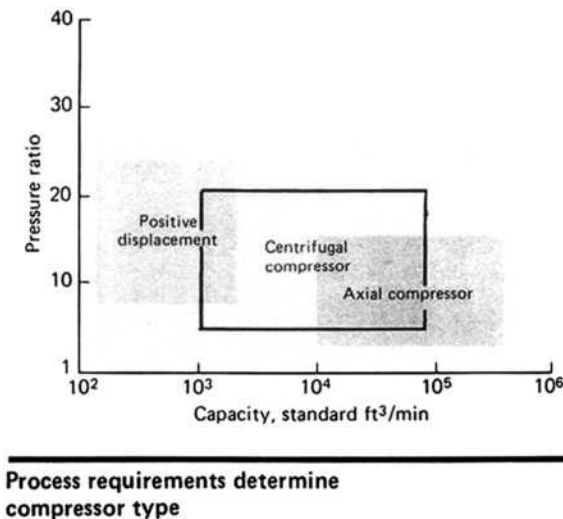


Figure 2.4. Compressor capacities

compressors operate most efficiently at medium flow rates and high-pressure ratios. Rotary and reciprocating compressors (positive-displacement machines) are best used for low flow rates and high-pressure ratios.

The general configuration for compressors and their drive trains must fit the location, environment, and type of compressor. A decision must be made as to whether the units are to operate in a series or in parallel, and it requires a knowledge of the necessary discharge pressure and flow. In most cases, a number of casings are connected together to form a “compressor train.” This is nothing more than connecting various compressors in a series. The limit to the number of casings so connected is due to the rotor dynamics of the coupled train.

A decision must also be made as to what type of mounting is desirable for a given arrangement. Most turbomachinery is mounted on structural-steel platforms, referred to as base plates or skids. These platforms are then positioned on a mass of concrete at the job site by installing them on sole plates or through direct grouting. Platforms should be considered part of the foundations, with great care exercised in their design. Insufficient rigidity can allow the rotating machinery to excite these platforms.

DRIVER SELECTION

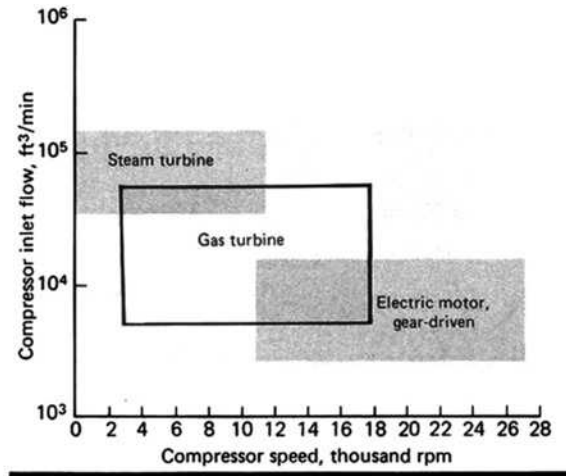
The three main types of drives for centrifugal compressors are (1) steam turbines, (2) gas turbines, and (3) electric motors. The decision of which drive is best is not always easy. Selection depends on many factors, such as location, process, and unit size.

For remote locations, gas turbines are mostly used, due to their low maintenance and the ability to prepackage the units. Their light weight makes them a must for offshore platforms. For chemical plants, steam turbines are widely used, due to the needs for process steam. In this manner, CPI plants can use energy more efficiently. For smaller flows, electric motors drive the compressor, usually through speed-increasing gears.

Typical ranges for the various drives are shown in Figure 2.5, from which we note that the higher the flow, the lower the speed. At high flows, the compressor diameter must be large; therefore, the speed must be reduced to maintain the same stress levels in the machinery.

COMPRESSOR DESIGN AND CONFIGURATION

To properly design a centrifugal compressor, we must know the operating conditions—the type of gas, and its pressure, temperature, and molecular



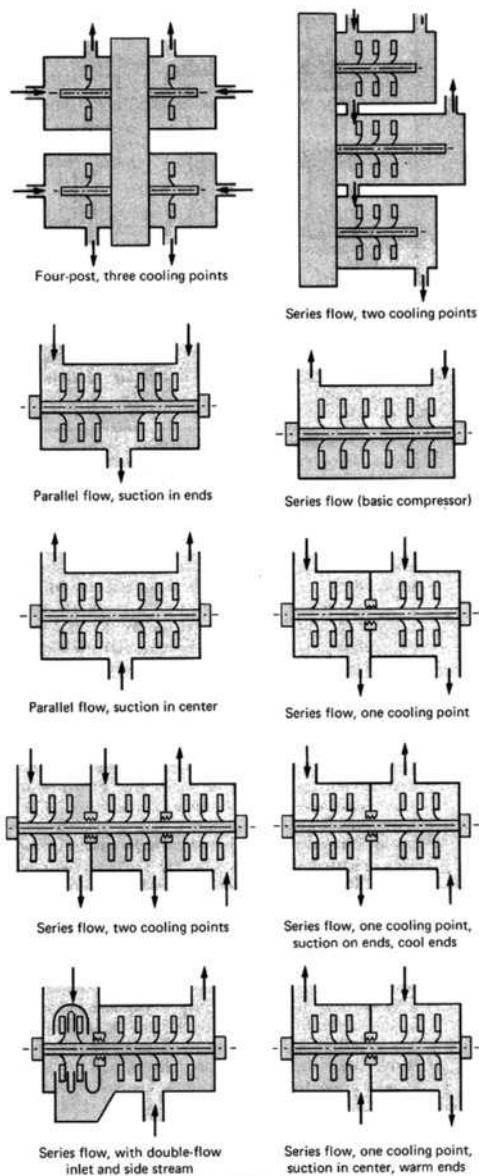
Driver selection depends on many factors

Figure 2.5. Different compressor speeds

weight. We must also know its corrosive properties so that proper metallurgical selection can be made. Gas fluctuations due to process instabilities must be pinpointed so that the compressor can operate without surging.

Centrifugal compressors for industrial applications have relatively low pressure ratios per stage. This is necessary so that the compressors can have a wide operating range, and stress levels can be kept at a minimum. Due to the low pressure ratios for each stage, a single machine may have a number of stages in one “barrel” to achieve the desired overall pressure ratio. Figure 2.6 shows some of the many configurations. Some of the factors to be considered when selecting a configuration to meet plant needs are:

1. Intercooling between stages can considerably reduce the power consumed.
2. Back-to-back impellers allow for a balanced rotor thrust and minimize overloading the thrust bearings.
3. Cold inlet or hot discharge at the middle of the case reduces oil-seal and lubrication problems.
4. Single inlet or single discharge reduces external piping problems.
5. Balance planes that are easily accessible in the field can appreciably reduce the field-balancing time.
6. Balance piston with no external leakage will greatly reduce the wear on the thrust bearings.



Centrifugal compressors have many configurations in order to meet the needs of the process plant

Figure 2.6. Centrifugal Compressor Configurations

7. Hot and cold sections of the case that are adjacent to each other will reduce thermal gradients and thus reduce case distortion.
8. Horizontally split casings are easier to open for inspection than vertically split ones, reducing maintenance time.

9. Overhung rotors present an easier alignment problem because shaft-end alignment is necessary only at the coupling between the compressor and the driver.
10. Smaller, higher-pressure compressors that do the same job will reduce foundation problems but will have greatly reduced operational range.

IMPELLERS AND THEIR FABRICATION

Centrifugal-compressor impellers are either shrouded or unshrouded. The blading for the impellers can have one of three configurations—most commonly radial, followed by backward-curved, and, in some rare cases, forward-curved. Backward-curved blades have the largest operating range and are usually higher in efficiency. The radial-bladed impeller produces a higher head but has a smaller operating range and a slightly lower efficiency than the one with backward-curved blades.

Open, shrouded impellers that are mainly used in single-stage applications are made by investment-casting techniques or by three-dimensional milling. Such impellers are used, in most cases, for the high-pressure-ratio stages. In the CPI, the shrouded impeller is the most common.

Figure 2.7 shows several fabrication techniques. The most common type of construction is seen in Figure 2.7a and 2.7b, where the blades are fillet-welded to the hub and shroud. In Figure 2.7b, the welds are full penetration. The disadvantage of this type of construction is the obstruction to the aerodynamic passage. In Figure 2.7c, the blades are partially machined with the covers and then butt-welded down the middle. For backward-lean-angle blades, this technique has not been very successful, and there has been difficulty in achieving a smooth contour around the leading edge.

Figure 2.7d illustrates a slot-welding technique and is used where the blade-passage height is too small, or the backward-lean angle too high, to permit conventional fillet welding. In Figure 2.7e, an electron-beam technique is used to weld on the shroud or the hub. This technique is still in its infancy and work needs to be done to perfect it. Its major disadvantage is that electron-beam welds should preferably be stressed in tension, but for the configuration of Figure 2.7e, they are in shear. The configurations of Figure 2.7g through 2.7j use rivets. Where the rivet heads protrude into the passage, aerodynamic performance is reduced.

Materials for fabricating these impellers are usually low-alloy steels, such as AISI 4140 or AISI 4340. AISI 4140 is satisfactory for most applications; AISI 4340 is used for larger impellers requiring higher strengths. For corrosive gases, AISI 410 stainless steel (about 12 percent chromium)

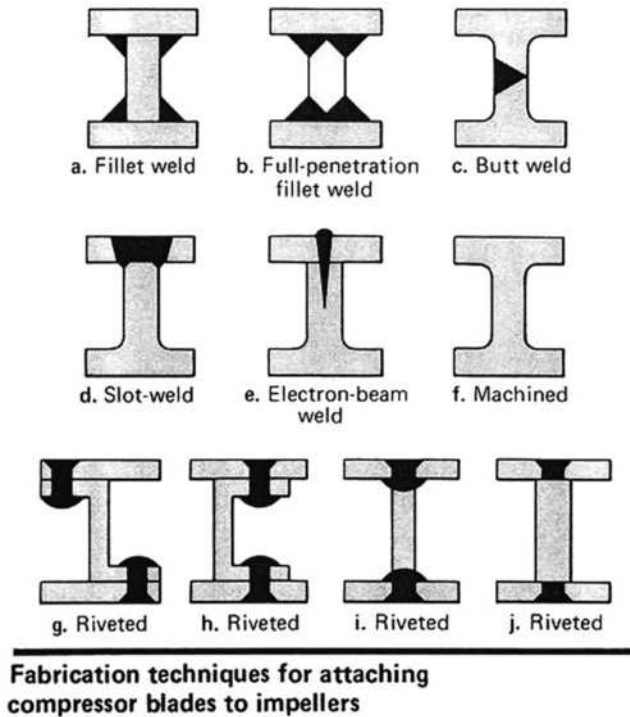


Figure 2.7. Fabrication types for compressor blades

is used. Monel K-500 is employed in halogen-gas atmospheres and in oxygen compressors because of its resistance to sparking. Titanium impellers have been applied to chlorine service. Aluminum-alloy impellers have been used in great numbers—especially at lower temperatures (below 300°F). With new developments in aluminum alloys, this range is increasing. Aluminum and titanium are sometimes selected because of their low density. This can cause a shift in the critical speed of the rotor, which may be advantageous.

ROTOR DYNAMICS

Rotor movement and its effect on the performance of the entire unit is the most important aspect of a centrifugal-compressor design. Most compressors for the CPI are built in accordance with API 617 specifications. The natural frequency of the rotor should not occur in the variable-speed range of the compressor, so as not to excite any of these frequencies. In newer high-speed compressors, many operate above

their first critical speed. Shafts that operate above the first critical are said to be “flexible shafts.”

It is desirable that the first critical not be around half the design speed. Otherwise, a problem known as “oil whirl” may be induced. Oil whirl is a major cause of instability in turbomachines. It may occur in journal bearings or in the seals in which the shaft and the stationary seal are separated by a film of fluid.

Many types of whirling motions occur in turbomachinery and are generated by various factors, such as:

- Hydrodynamic or “oil whip” or half-frequency whirl is caused by the instability of the fluid film in the bearing.
- Aerodynamic whirl is usually induced by tip clearances on compressors and turbines. It cannot be eliminated on balancing.
- Dry-friction whirl is usually initiated by rubbing caused by a large unbalanced force.
- Torque-induced whirl occurs in large flexible rotors under large power levels.
- Gyroscopic-induced whirl may be caused by bowed shafts and skewed disks.
- Asymmetric whirl is caused by the difference in shaft stiffness and is most violent near the first critical.

Balancing is a major problem for the newer flexible rotors that operate, in many cases, above the first critical and, often, above the second or third critical. Figure 2.8 shows the various modes that the rotor shaft undergoes as it passes through these criticals. High-speed balancing of these rotors is sometimes a must for a smooth operation. This inevitably means field balancing, because there are only a few test-stands that can balance these rotors at design speed.

Figure 2.9 is a typical rotor-response curve for a four-stage rotor. Here, the rotor is operating above the first critical, but the steepness of the curve near the design point is a cause for concern. Modification of the rotor and a change in bearing stiffness will move the slope from the design point.

BEARINGS FOR HIGH-SPEED MACHINES

Journal and thrust bearings are among the most important components to assure the maintenance-free running of high-speed turbomachines. Bearings in these machines range from simple journal bearings and flat thrust

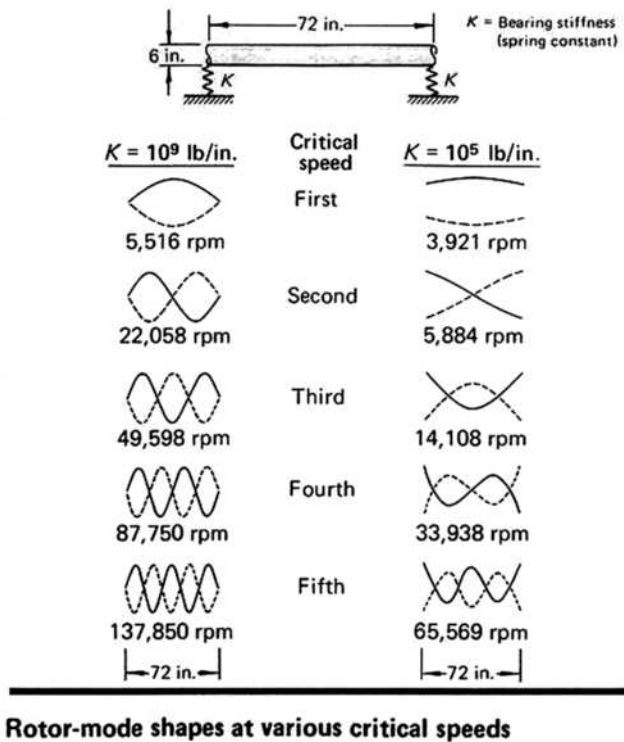


Figure 2.8. Rotor shapes & speeds

bearings to multiwedge designs for both thrust and journal bearings. Some of the many factors that enter into the selection of such bearings are:

- Speed range of shaft
- Maximum misalignment that can be tolerated by the shaft
- Critical-speed analysis, and influence of bearing stiffness on this analysis
- Loading of the compressor impellers
- Oil temperature and viscosity
- Foundation stiffness
- Axial movement that can be tolerated
- Type of lubrication system and its contamination
- Maximum vibration levels that can be tolerated

All rotating machines vibrate when operating, but the failure of the bearings is mainly due to their inability to resist cyclic stresses. The level of vibration that a unit can tolerate is shown in the severity charts (Figure 2.10).

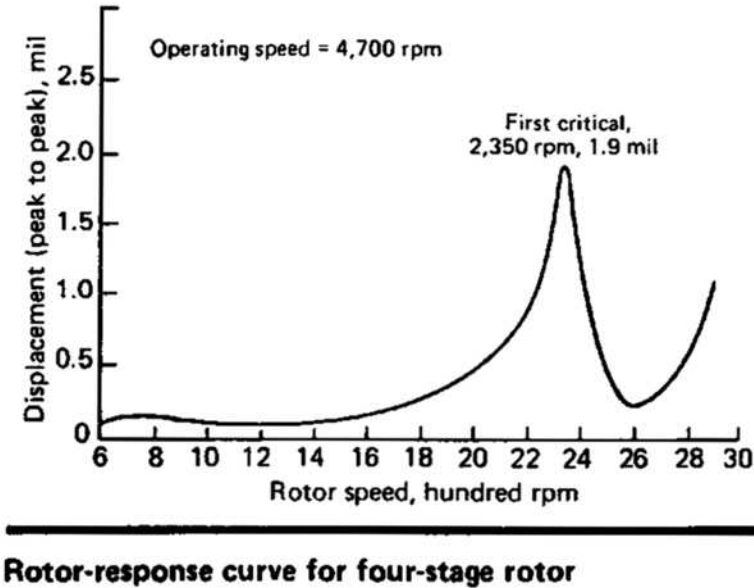


Figure 2.9. Rotor speeds & response curves

These charts are modified by many users to reflect the critical values for their machines.

JOURNAL BEARINGS

The journal bearing for turbomachinery has a fluid film that carries the load. Film thickness in most applications ranges from 0.0003 in. for gases to 0.008 in. for hydrostatic oil-lubricated bearings.

In this chapter, we will discuss only journal bearings in which a positive supply of lubricant is fed to the bearing at all times. Figure 2.11 shows a number of such bearings. The circumferential-grooved bearing normally has the oil groove at half the bearing length. This provides better cooling, but reduces load capacity by dividing the bearing into two parts. The cylindrical bearing, used in turbines, has a split construction with two axial oil-feed grooves at the split. The pressure-dam bearing is used where bearing stability is required.

The most common bearing is the tilting-pad type, whose most important feature is self-alignment when the bearing is used with spherical pivots. This bearing offers the greatest increase in fatigue life because of these advantages:

1. Self-aligning for optimum alignment and minimum limit
2. Thermal conductivity backing material to dissipate heat developed in the oil film

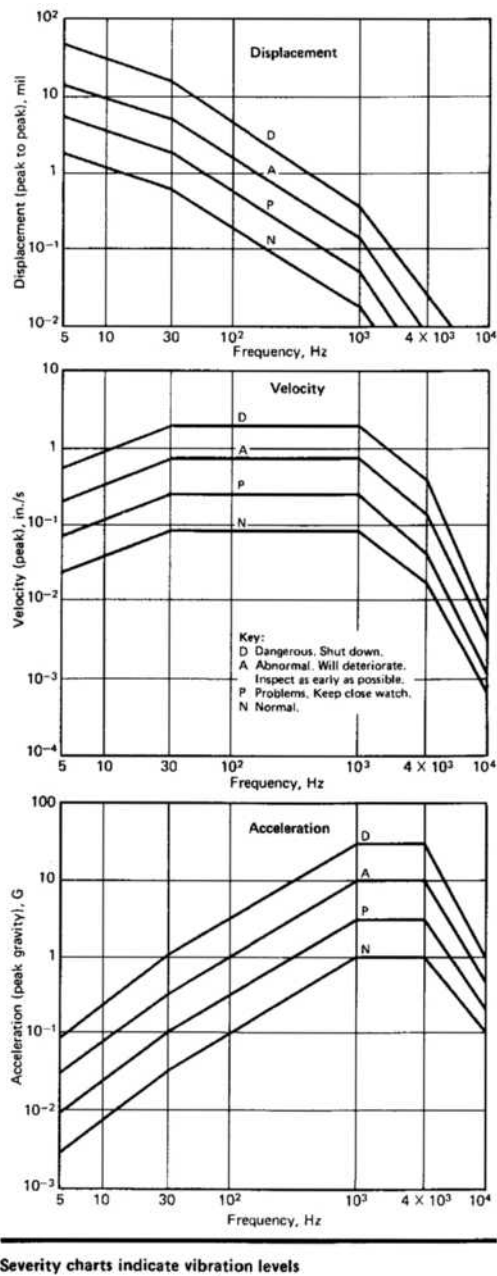
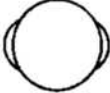
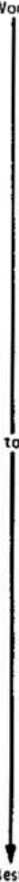
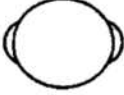


Figure 2.10. Vibration levels for compressors

- 3. Thin babbitt layer, centrifugally cast to a uniform thickness of about 0.005 in. (thick babbitts greatly reduce bearing life; babbitt thickness of about 0.01 in. reduces the bearing life by more than half)

Bearing type	Load capacity	Suitable direction of rotation	Resistance to half-speed whirl	Stiffness and damping
Cylindrical bore 	Good	 	Worst  to Best	Moderate
Cylindrical bore with dammed groove 	Good			Moderate
Lemon bore 	Good	 		Moderate
Three-lobe 	Moderate	 		Good
Offset halves 	Good			Excellent
Tilting-pad 	Moderate	 		Good

Journal bearings for turbomachinery

Figure 2.11. Bearing types

- 4. Oil-film thickness can be varied by changing the number of pads, directing the load onto or in between the pads, or changing the axial length of the pad. Oil-film thickness is critical when making bearing stiffness calculations.

THRUST BEARINGS

The most important function of a thrust bearing is to resist the unbalanced force developed in the working fluid of the machine and to maintain the rotor in its position within the prescribed limits. A complete analysis of the thrust load must be conducted. Compressors with back-to-back rotors greatly reduce the load on thrust bearings.

Figure 2.12 shows a number of different types of such bearings. When properly designed, the tapered-land thrust bearing (Figure 2.12a) can take and support a load equal to that of a tilting-pad thrust bearing. With perfect alignment, it can match the load of even a self-equalizing tilting-pad thrust bearing. Figure 2.12b is a nonequalizing tilting-pad

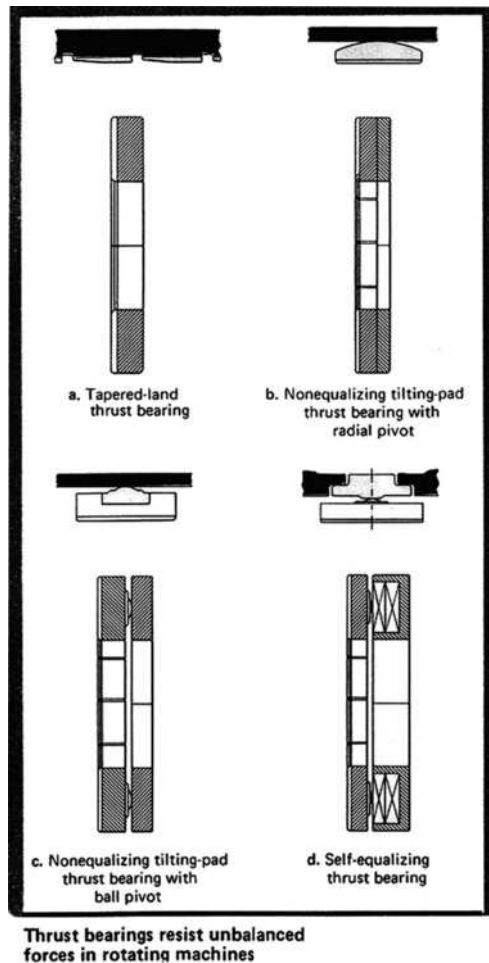


Figure 2.12. Thrust bearings Vs unbalanced forces

thrust bearing that pivots on the back of the pad along a radial line. Figure 2.12c is a nonequalizing tilting-pad bearing whose pads are supported on spherical-pivot points. Since this allows the pads to pivot in any direction, alignment is not as serious a problem as in the other two types. Figure 2.12d is the Kingsbury-type self-equalizing thrust bearing. This bearing virtually eliminates the problem of misalignment. The major drawback is that standard designs require more axial space than do a nonequalizing type.

MISALIGNMENT

The amount of misalignment that can be tolerated in high-speed turbomachines depends on the types of journals and thrust bearings used. Figure 2.13 shows the misalignment for such bearings. In a journal bearing, misalignment will cause the shaft to contact the end of the bearing. Thus, journal length is a criterion in the amount of misalignment that a bearing can tolerate. Obviously, a shorter length bearing can tolerate more misalignment. In a thrust bearing, misalignment will cause the loading of one segment of the thrust-bearing arc and the unloading of the opposite segment. This effect is more pronounced at higher loads and with less-flexible bearings.

To adjust for misalignment other than by using tilting-pad bearings, various techniques must be employed. The simplest and most common is the so-called the cold alignment (also known as base alignment) method. Once this is done, hot alignment is carried out. Hot-alignment techniques measure the changes when the unit is operational and temperature growth is stabilized, so that accurate alignment data are obtained. There are many ways to perform these tests. The recommended procedure is to do cold alignment by using the “reverse-indicator graphical plotting” and the hot alignment by using an optical alignment technique.

Reverse-indicator graphical plotting is normally obtained when the unit is cold. This is done by first laying out the desired hot operating line on a graph paper (Figure 2.14). This line shows the desired equilibrium operating conditions. Then, the desired cold position of the shaft is plotted, based on the thermal-growth information that must be supplied by the manufacturer. Actual positions of the shaft in the field are taken. This information is plotted, the difference computed, and shims added to the supports. (The major assumptions here are that the computation measures actual vertical growth, and that growth in the horizontal direction is zero.) This procedure is then repeated, after hot checking the alignment.

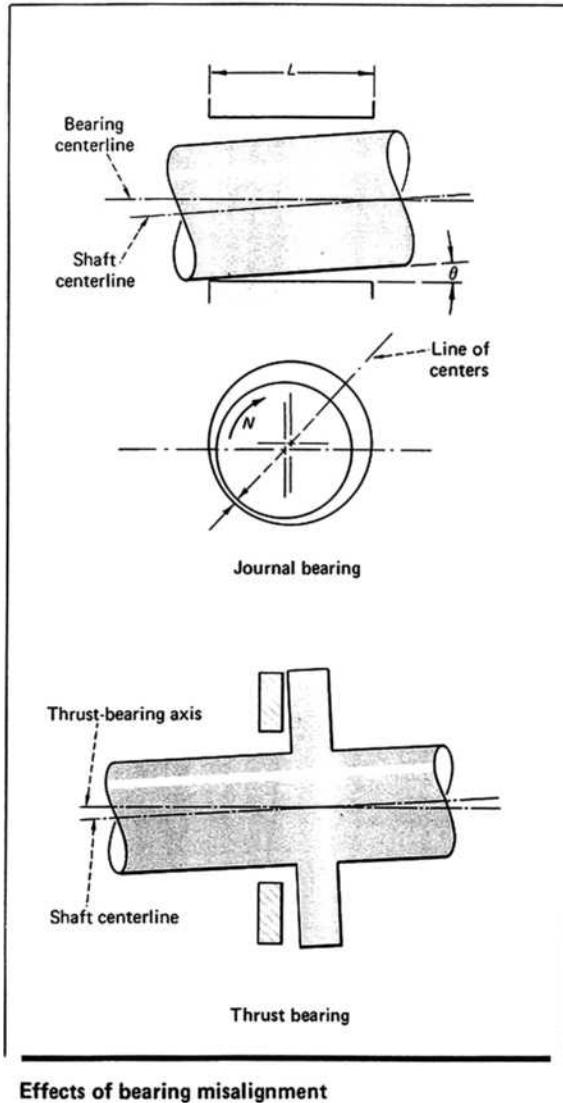
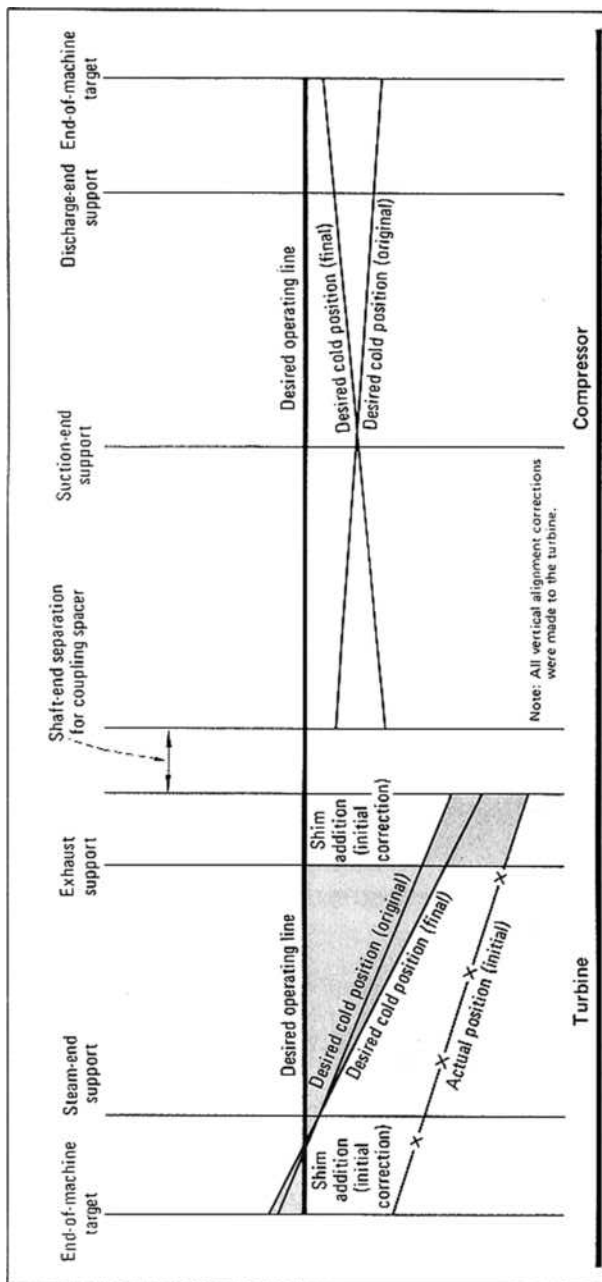


Figure 2.13. Bearing misalignment

For hot checking, an optical alignment procedure is recommended. The heart of this technique is the optical instrument with built-in optical micrometers to measure the displacement from a precise and referenced line of sight. After the initial cold alignment, the train is ready for startup. Optical reference points are established at each end and each side of each unit of the train, and as close to the couplings as possible. On most units, this is at the bearing housing.



Graphical plot shows steps in alignment procedures for compressor train

Figure 2.14. Alignment techniques

A jig transit is readied, a line of sight established, and readings taken at each reference point in the compressor train in the vertical plane. For the horizontal plane, a somewhat similar procedure is followed, and data are recorded for reference points. The train is then started up and operated at design or near-design conditions, and temperatures are allowed to stabilize. Then, another set of optical readings are taken. In this manner, when a comparison is made with the cold readings, actual thermal growth can be calculated. Thus, the actual thermal growth can be plotted on the graph, and new corrections can be computed.

The technique outlined here is used on new machines. On old machines, a reverse technique can be applied. First, a hot-alignment check is made, then a cold optical check, followed by the mechanical reverse-indicator readings. The information is plotted, and realignment measurements are taken.

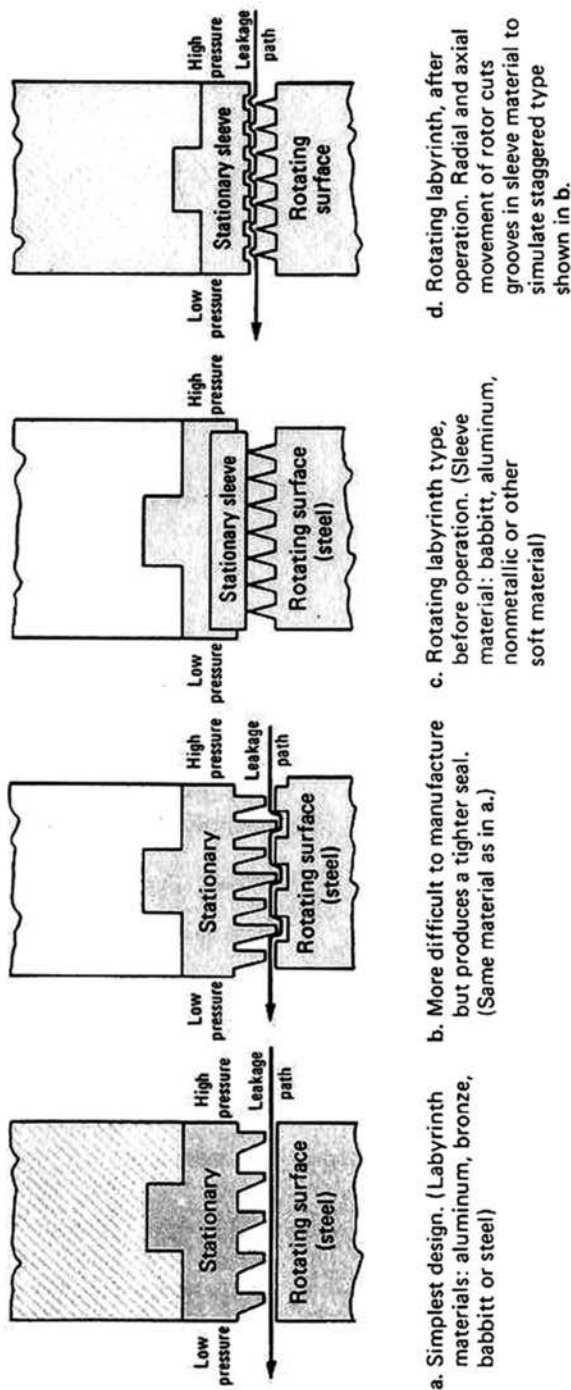
The outline of the technique given here is simple, but in practice one must develop this skill. Some of the major problems encountered in alignment are due to pipe strain. This is caused by the piping being offset from the intake or exhaust nozzles from a few thousandths of an inch to several inches. Many engineers cannot understand how a “small” pipe is able to move a large, heavy piece of machinery. Tension on pipe hangers can change the vibration level considerably.

Another contributor to the alignment problem is the gear casing. Thermal growth in the new fabricated cases has been unpredictable. In some instances, gear cases rise with a twist to create another problem that is very difficult to correct. Short couplings also present a problem and magnify misalignment. Some users are now specifying that the coupling spacer will be at least 18 in. long.

The preceding discussion has presented a general outline of alignment techniques. The alignment of high-speed machinery must be accurate; otherwise, major problems will arise. Sometimes, to prevent major shutdowns to correct the misalignment, heaters are added to one or the other leg of the unit to align them while running. This technique is not a cure but rather a temporary relief when shutdown is impossible.

COMPRESSOR SEALS

The internal seals that prevent leakage around the impellers are usually labyrinth type, as shown in Figure 2.15. They have a series of circumferential knife edges that are positioned close to the rotating impeller. If damaged by rubbing, erosion, or corrosion, these knife edges will lose



Labyrinth seals prevent leakage around the impellers of centrifugal compressors

Figure 2.15. Function of labyrinth seals- explanation of abc shows the leakage paths

their effectiveness. In some cases, the knife edges are machined onto the rotating part, while a sleeve of soft material is positioned on the stationary part. The rotating part then cuts a groove into the stationary sleeve, reducing leakage considerably and thus increasing the compressor efficiency. The stationary sleeve can be manufactured in babbitt-lined steel, ceramic, compressed steel-fiber material, etc., while the knives are made of high-quality steel.

Shaft seals are usually mechanical-contact or liquid-film types, as seen in Figure 2.16. In some cases where a small amount of leakage can be tolerated, labyrinth seals are used. The oil- or liquid-film seal consists of two stationary bushings that surround the shaft with a clearance of a few thousandths of an inch. Oil at a nominal rate of 10 gpm is introduced between the bushings at a positive pressure higher than that of the process gas and leaks in both directions along the shaft. The oil is retained in the seal housing by “O” rings. To limit the inward oil leakage, the differential pressure across the inner bushing is only a few pounds per square inch. The inner-leakage rate varies from 1 to 4 gph, depending on the size of the seal, but is independent of the gas pressure being contained.

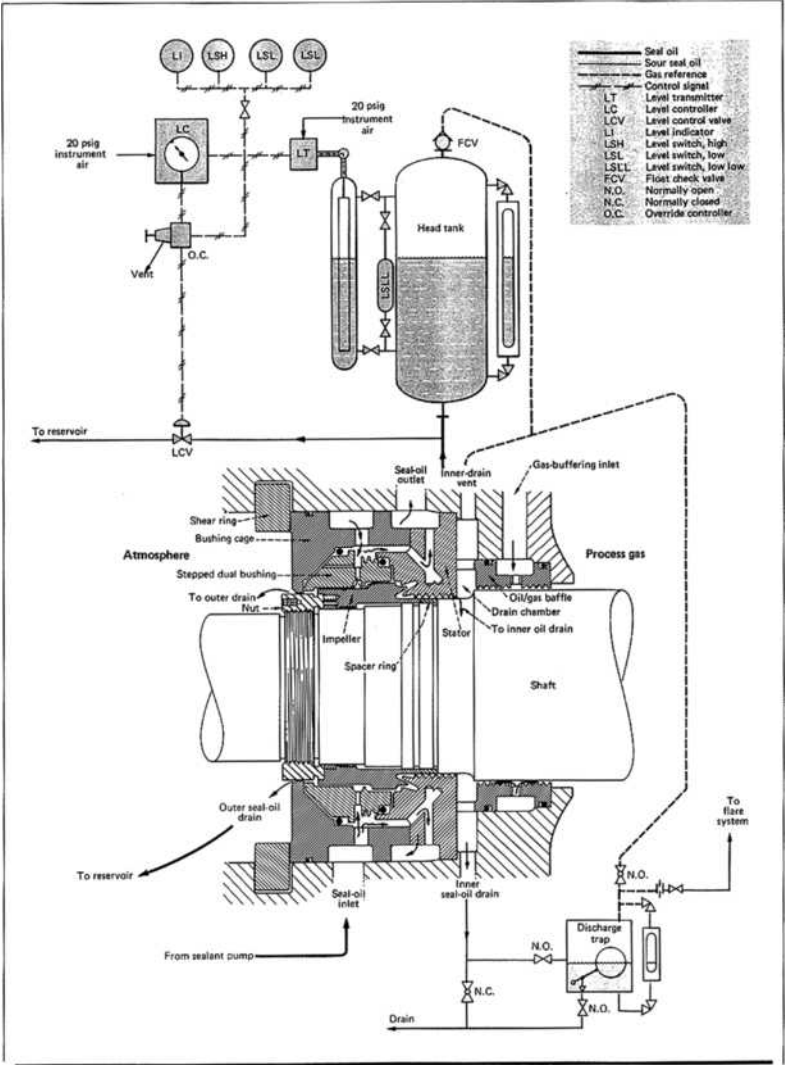
This leakage is collected in a chamber that is usually separated from the gas stream by a labyrinth seal. Overflow of oil from the leakage chamber and its subsequent entry into the compressor is the biggest problem.

Mechanical-contact seals have two major elements (Figure 2.17). These are the oil-to-process gas seal, or carbon ring, and the oil-to-uncontaminated-seal-oil-drain seal, or breakdown bushing. This seal can maintain a lower inner-leakage ratio with higher oil-to-gas differential pressure.

In operation, the seal-oil pressure is maintained at about 25 to 50 psia over the process-gas pressure against which the seal is sealing. High-pressure oil enters the seal cavity, completely filling it. Some of the oil (ranging from 2 to 8 gph) is forced across the carbon-seal face and flows out the contaminated-oil drain. The mechanical seal's great advantage over the oil-film seal is that it has a minimum effect on rotor dynamics. On the other hand, when the oil-film bushings lose their free-floating feature, they can upset the stability of the rotor when operating at high speeds.

BALANCING

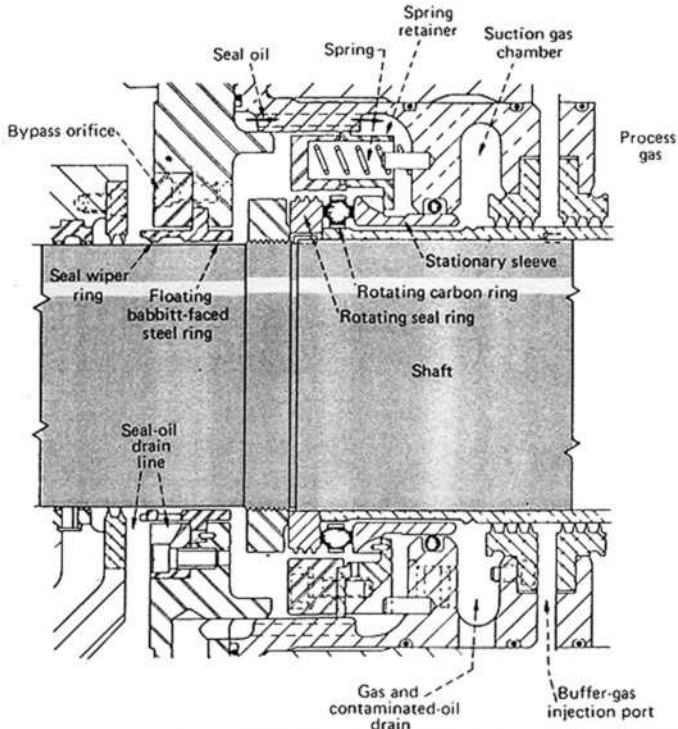
Balancing of high-speed rotating machinery is very crucial in the smooth running of the machine. Machines designed to operate higher than their



Liquid-film seal for compressor shaft provides continuous flow of seal oil

Figure 2.16. Liquid film seal function

first critical speed need even more care. Presently, in most vendors' shops, each wheel is balanced separately and then shrink-fitted to the shaft. During this process, shaft runout must be closely watched, since large changes could indicate that the impeller is not installed square to the shaft. When all rotors are fitted on the shaft and the final balancing is done, correction is usually made to the last impeller installed if an unbalance exists.



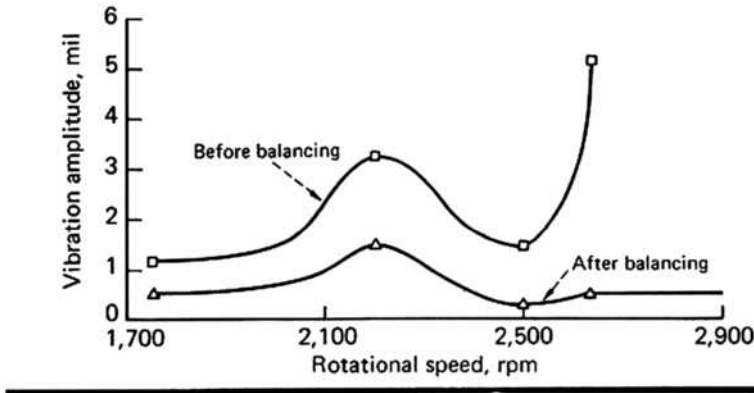
Mechanical-contact seal for compressor shaft

Figure 2.17. Function of contact seals

The best technique for high-speed rotors is to balance them at their rated speed, and not in low-speed machines. This is not always possible in the shop, and so is often done in the field. To achieve best results, balancing should be performed at all rotating planes. This is done with the influence-coefficient technique. Here, the effect of unbalance on each plane, and its influence on the other planes, is noted. Once the influence coefficients of each plane are known, balancing can be performed at each plane and a smooth rotor obtained. Figure 2.18 shows rotor-response curves before and after balancing.

COUPLINGS

Flexible gear couplings have been the most widely used. In many new machines, they are being replaced by disk-type couplings. Gear-type



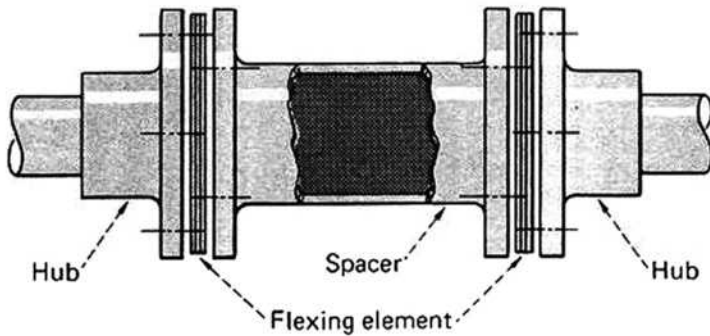
Rotor-response curves show effects of balancing

Figure 2.18. Balancing techniques before & after

couplings consist of a hub gear and a sleeve gear. In most cases, male teeth are integral with the hub, but some couplings have male teeth integral with the sleeve. Most of these couplings are mounted on a tapered-shaft end. In many instances, a key and keyway are provided to transmit torque; in others, the coupling is shrink-fitted on the tapered shaft.

Lubrication of the gear coupling is accomplished by a continuous or batch-packed technique. With continuous lubrication, nozzles spray oil into the coupling's teeth. The oil is recirculated through the machine's lubricant system, which dissipates the heat generated in the coupling, and thus maintains a relatively constant temperature. With batch-packed or seal lubrication, a recommended grease or oil is sealed in the coupling, which is changed as necessary or during scheduled shutdown. The major advantages of the batch-packed technique are that the lubricant is the best available for the application and that the lubricant does not get contaminated. For high-temperature applications, continuous lubrication is recommended. Such lubrication requires a good filtration system; otherwise, a centrifuging of contaminants in the gear teeth may occur.

The disk-type or metal-flexible coupling, shown in Figure 2.19, consists of two hubs rigidly mounted by interference fit or by flange bolting to the driven and driving shafts. Flexible elements, one attached to each hub, compensate for misalignment. They are connected by a spacer (usually tubular) to span the gap between shafts. The major problem of this coupling is its potential to vibrate when excited at its resonance frequency.



Metal-flexing couple runs without lubrication

Figure 2.19. Flexible couplings

The major advantage is that it needs no lubrication system and can tolerate a higher degree of misalignment.

LUBE-OIL SYSTEMS

API Standard 614 covers in detail the minimum requirements for lubrication systems, oil-type shaft-sealing systems, and control-oil supply systems for special-purpose applications.

The base system consists of a reservoir that should be separately placed from the base plate. The working capacity of the oil tank should be at least 5 minute, based on normal flow. The reservoir-retention time should be 10 minute, based on normal flow and the total volume below the minimum operating level.

Arrangements for heating the oil should also be made. If an immersion heater is used, maximum watt density should be 15 W/in.^2 . If steam heating is used, the heating element should be external to the reservoir.

The oil system should be equipped with a main oil pump, a standby pump, and, for critical machinery, an emergency pump. Power sources for the main and standby pumps should be different. For example, if steam is used to power the main pump, the standby pump should be electrical. The emergency pump is also usually electrical, but will be driven by either a d.c. supply or a completely separate a.c. supply.

Twin oil coolers should be provided and piped in parallel, using a single multiport transfer valve to direct the flow to the coolers. The

coolers should carry water on the tubeside and oil on the shellside. Oil-side pressure should be greater than the waterside pressure. Twin full-flow filters should be located downstream of the oil coolers. Do not pipe the filters with separate inlet and outlet block valves. This could cause loss of oil flow from the possible human error of blocking the flow during filter switching.

Oil for turbomachinery should be of correct viscosity and formulated with the required chemical additives to prevent rusting, resist oxidation and sludging, be noncorrosive to machine parts, resist foaming, and separate rapidly from water. The oil should be checked periodically to detect changes in viscosity, **pH** and neutralization number, and precipitation.

HIGH-SPEED GEARS

In many compressor applications, high-speed gears are used between the drive (especially, electric motors) and the compressor. The application of gear drives to large trains involving high speeds and power levels has never been an easy task. Currently, high-hardness gearing is used, with tooth loads in the range of 1,500 to 2,000 lb/in. of face, at pitchline velocities of 20,000 to 30,000 ft/min. Normally, the gear drives used in such an application are of the single-helical or double-helical type, with rotors carried in sleeve-type bearings. The double-helical gear is the first choice for high-reliability applications.

Proper alignment of gears to the shafts is very important, and hot checks on alignment are usually a must for high-speed gearing. When possible, gears should be run-in on initial startups. Speed and load should be increased percentage wise. Lube-oil pressure and temperature should be closely observed, and adjustments made to the lube system, as required. Oil pressure must be maintained at all costs. High-speed gearing must be treated with care.

CONTROL SYSTEMS

The controls for most compressor trains consist of two major systems: one for the lubrication system and one for the compressor. For the lubrication system, minimum alarms are: low oil pressure, low oil-pressure trip (at some point lower than the alarm point), low oil level in the reservoir, high oil-filter differential pressure, high thrust-bearing metal temperature, and

high oil temperature. Each pressure and temperature-sensing switch should be in a separate housing. The switch type should be single pole, double throw, and furnished as open (deenergized) to alarm and close (energize) to trip. Pressure switches for alarms should be installed, with a “T” connection for a pressure gage and bleeder valve to test the alarms. Temperatures should be monitored in the oil piping to and from the coolers, and at the outlet of each radial and thrust bearing. Bearing-metal temperature should also be measured, since problems will show up much faster in the metal temperature than in the oil temperature. Figure 2.20 shows that the change in metal temperature is much higher than the change in oil temperature as load is increased.

Pressure gages should be provided at the discharge of the pumps, bearing header, control-oil line, and seal-oil line. Each atmospheric oil-drain line should be equipped with steel nondestructive bull’s-eye flow indicators, positioned for viewing through the sides.

For the compressor, a complete instrumentation package should be provided, as shown in Figure 2.21. In this manner, a full analysis can be made for both vibration and performance characteristics. For vibration analysis, both proximity probes and accelerometers have been suggested. Figure 2.22 shows the ranges for which the various probes are applicable. The reason for two different transducers is that proximity probes show only the movement of the shaft at the location of the probes. The probes do not pick up high-frequency problems such as blade-passing or gear-mesh frequencies.

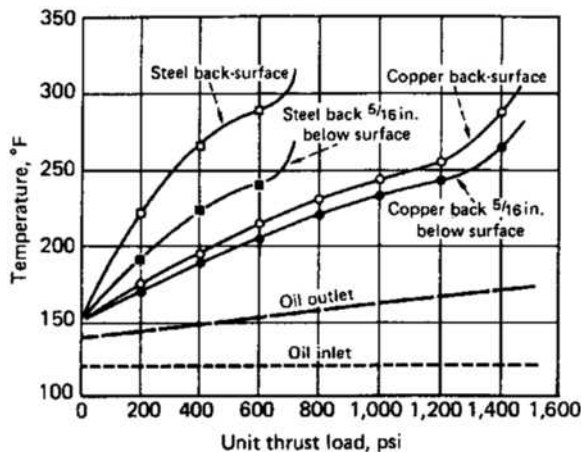


Figure 2.20. Temperature changes

Turbine	
❶	❷
2 Radial probes	2 Radial probes
1 Key phasor	1 Accelerometer
2 Thrust probes	1 Journal-bearing temperature
1 Accelerometer	1 Steam pressure
1 Journal-bearing temperature	1 Steam temperature
2 Thrust-bearing temperature	
1 Steam pressure	
1 Steam temperature	
Low case	
❸	❹
2 Radial probes	2 Radial probes
1 Accelerometer	1 Accelerometer
1 Journal-bearing temperature	2 Thrust probes
2 Process pressure	1 Journal-bearing temperature
2 Process temperature	2 Thrust-bearing temperature
	1 Process pressure
	1 Process temperature
Gear box	
❺	❻
1 Accelerometer	1 Journal-bearing temperature
1 Journal-bearing temperature	
❼	❽
1 Journal-bearing temperature	1 Journal-bearing temperature
	1 Accelerometer
	1 Key phasor
High case	
❾	❿
2 Radial probes	2 Radial probes
1 Accelerometer	1 Accelerometer
1 Journal-bearing temperature	2 Thrust probes
2 Process pressure	1 Journal-bearing temperature
2 Process temperature	2 Thrust-bearing temperature
	1 Process pressure
	1 Process temperature

Monitoring probes required for compressor train

Figure 2.21. Monitoring probes

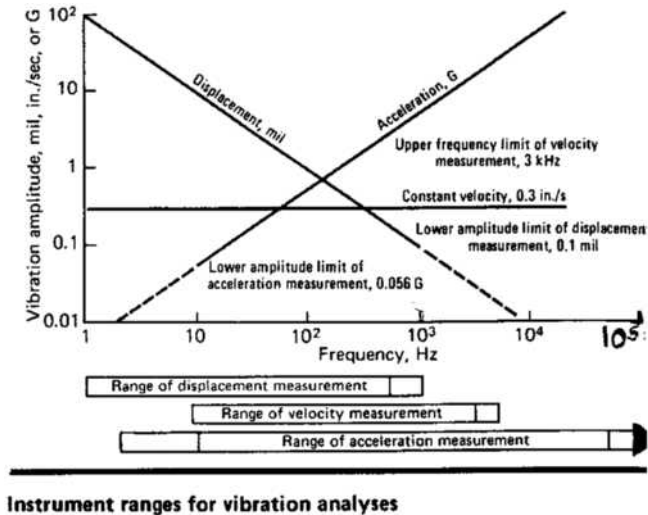
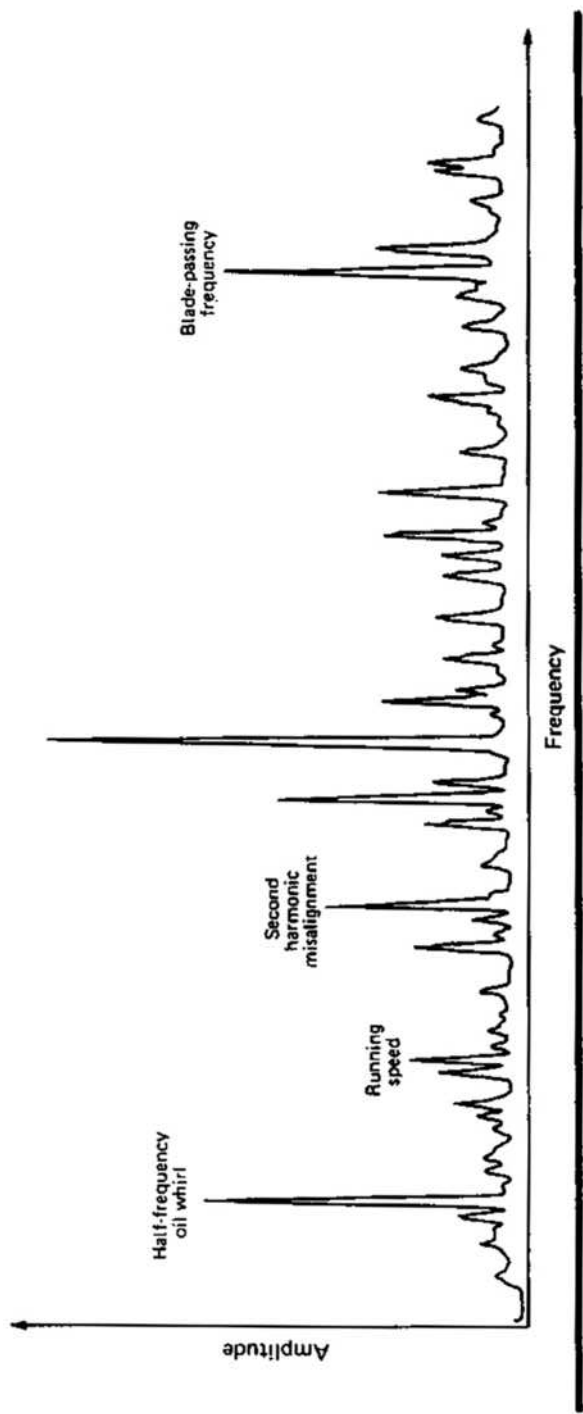


Figure 2.22. Vibration analysis

Figure 2.23 is a real-time plot that shows the blade-passing frequencies, dominating as the compressor approaches surge. High subharmonic frequencies indicate oil whirl, and second harmonics indicate some misalignment. In some cases, catastrophic failure has occurred even though no indication was given by the proximity probes. When properly used, proximity probes are excellent in locating subharmonic and misalignment problems. Thus, by using a combination of proximity probes and accelerometers, a complete picture of rotor dynamics is obtained.

Surge-control instruments on the market are basically static devices. These measure the pressure rise across the compressor, and sense the flow rate from pressure drop across an orifice. The pressure rise and flow rate are then compared with values programmed into the device, and when they exceed these values, the signal is given for a bypass valve to open.

The biggest drawback of this system is that it does not provide for any changes in the molecular weight of the gas or the degradation of the compressor itself. Presently, programs are under way that measure the dynamic change in the pressure head in the boundary layer. When flow reversal is sensed, the bypass valve is opened. This program is still being evaluated.



Real-time plot for a compressor shows details of critical speeds

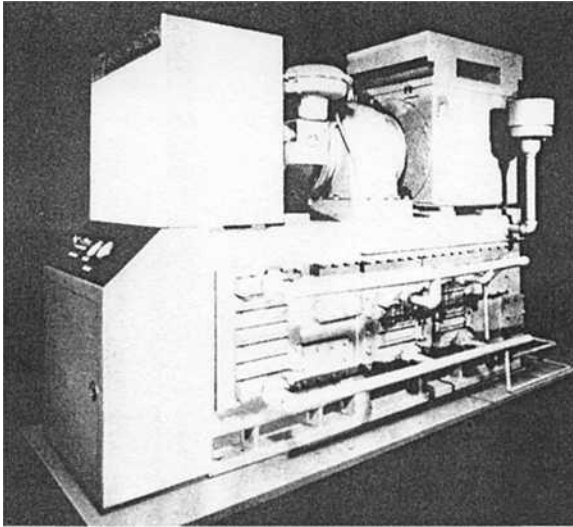
Figure 2.23. Vibration Spike Types

Until now, monitoring systems have consisted of no more than vibration monitors. With the advent of real-time analyzers and minicomputers, a whole new era is opening in monitoring and diagnostic systems. These new systems indicate problem areas, give expected life of various components, and schedule maintenance. They correlate both process and mechanical parameters to fully diagnose the compressor train and run it at its most efficient point. These systems use a performance matrix in combination with trending data and real-time analysis to properly project and diagnose problems. A good monitoring and diagnostic system can save thousands of dollars in downtime and energy.

CHAPTER 3

BASICS OF SURGE CONTROL FOR CENTRIFUGAL COMPRESSORS





Surging is an unstable operating condition that causes erratic compressor performance. Here is a basic anti-surge control scheme for centrifugal compressors, along with suggestions on how to cope with variations in operating conditions and how to handle multi-compressor arrangements.

Compressors, like other major processing equipment, should be controlled as effectively as possible to ensure efficient plant operation. In devising a control scheme, the designer must consider how to avoid an unstable operating condition known as surging.

Simply stated, surging occurs when the compressor throughput is reduced to a point sufficiently below design conditions so that erratic performance results. The job of the anti-surge system, then, is to detect the potential upset and automatically compensate for it by maintaining a flow through the compressor in excess of the surge condition.

Before getting into the details of how this system works, we must first understand the phenomenon of surging from the standpoint of compressor-system variables.

SURGING CHARACTERISTICS

A typical family of characteristic curves for a variable-speed centrifugal compressor is shown in Figure 3.1. These curves depict the adiabatic head as a function of actual inlet-volume flow, with a characteristic curve for each rotational speed.

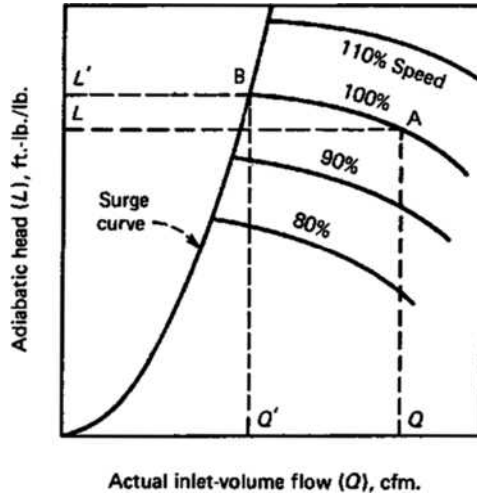


Figure 3.1. Characteristic curves for a typical centrifugal compressor are based on Eq. (3)

If we assume that the compressor is operating at Point A on the 100 percent-speed curve, the inlet flow is Q and the head is L . Now if the external load resistance gradually increases while the speed remains constant, the flow decreases and the operating point will move to the left along the 100 percent-speed characteristic curve. When it reaches Point B, the flow decreases to Q' , and the head increases to L' , the maximum head the compressor can produce at this speed.

At this point, the characteristic curve is practically flat and the operation of the compressor becomes unstable. This condition is called surging, and it appears as rapid pulsation in the flow and discharge pressure, producing high-frequency reversals in the axial thrust on the compressor shaft. In some machines, surging can become severe enough to cause mechanical damage. To avoid this danger, a control system must be devised to prevent the compressor from operating in this unstable area.

BASIC CONTROL THEORY

In all centrifugal machines (both pumps and compressors), the head produced is proportional to the square of the speed, while the flow rate is linearly proportional to the speed. These relationships are described in the following equations:

$$Q = CN \quad (1)$$

$$L = CN^2 \quad (2)$$

Substituting for N yields:

$$L = CQ^2 \quad (3)$$

This equation is used to calculate the parabolic surge curve usually provided by the compressor manufacturer.

Since there is no way to measure the adiabatic head directly, we must find another quantity to use in its place. The one most easily related to head is compression ratio, R_c .

$$R_c = P_2 / P_1 \quad (4)$$

Equation (5) is used to relate the compression ratio to the adiabatic head:

$$R_c = (1 + Lm\phi/1545T_1Z_1)^{1/\phi} \quad (5)$$

where

$$\phi = (K - 1)/K$$

When the compressor handles gas of only one composition, and when the suction temperature is nearly constant, the quantities m , ϕ , T_1 , and Z_1 are constant, and Eq. (5) reduces to:

$$R_c = (1 + CL)^{-1/\phi} \quad (6)$$

Although Eq. (6) shows that the relation between R_c and L is not linear, when this equation is plotted for most of the gases commonly encountered (e.g., air, natural gas, etc.), the curves approach straight lines over the normal operating ranges. The quantity $(R_c - 1)$ can then be substituted for L without producing significant distortion of the surge curves.

Equation (3) shows that the relation between the adiabatic head and Q^2 is linear. Figure 3.2 is a plot of L and the corresponding values of $(R_c - 1)$ versus Q^2 for the same range of heads for natural gas to show the magnitude of the error introduced by substituting $(R_c - 1)$ for L in Eq. (3). At greater heads, this error increases and may require an adjustment in the system to correct it, but in most cases, the error is small enough to be neglected.

This permits us to write the equation:

$$R_c - 1 = CQ^2 \quad (7)$$

The differential pressure across the compressor is:

$$\Delta P = P_2 - P_1 \quad (8)$$

From Eq. (4),

$$P_2 = P_1 R_c$$

Then:

$$\Delta P = P_1 R_c - P_1$$

$$\Delta P = P_1 (R_c - 1) \quad (9)$$

$$R_c - 1 = \Delta P / P_1 \quad (10)$$

Substituting Eq. (10) into Eq. (7) provides:

$$\Delta P / P_1 = CQ^2 \quad (11)$$

Nomenclature

<i>c</i>	Constant (not necessarily the same in all equations)
<i>h</i>	Differential pressure across primary in. of water flow device,
<i>K</i>	Specific heat ratio
<i>L</i>	Adiabatic head, ft.-lb./lb.
<i>m</i>	Molecular weight
<i>N</i>	Compressor speed, rpm.
<i>P</i>	Pressure, psia.
<i>Q</i>	Actual inlet-volume flow, cfm.
<i>R_c</i>	Compression ratio
<i>T</i>	Absolute temperature, °R.
<i>V</i>	Specific volume, cu. ft./lb.
<i>W</i>	Mass flowrate, lb./min.
<i>z</i>	Supercompressibility factor

Greek letters

Δ	Differential
ϕ	Ratio of (<i>K</i> - 1) to <i>K</i>

Subscripts

<i>D</i>	Design
<i>1</i>	Suction
<i>2</i>	Discharge

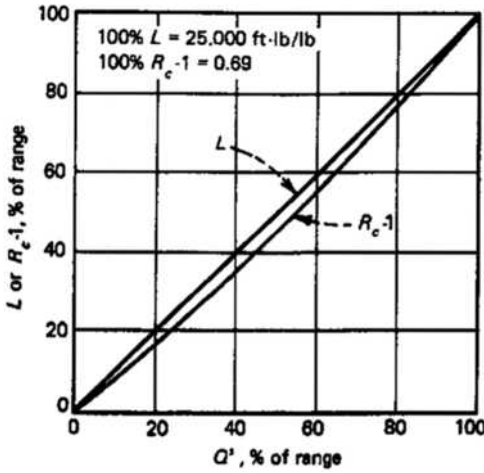


Figure 3.2. Error resulting from replacing the adiabatic head with the compression-ratio factor is insignificant

To measure Q , we must have a primary device, such as an orifice or a venturi tube, in the compressor suction line. If we let h represent the differential pressure across this primary device in inches of water, we can calculate the mass flow, W , to the compressor from the equation:

$$W = C\sqrt{hP_1 / T_1} \quad (12)$$

$$V = C(T_1 / P_1) \quad (13)$$

$$Q = WV \quad (14)$$

Substituting Eqs. (12) and (13) in Eq. (14) produces:

$$Q = C\sqrt{hP_1 / T_1} \times T_1 / P_1 \quad (15)$$

$$Q = C\sqrt{hT_1 / P_1}$$

If, as before, we assume the temperature to be constant, then:

$$Q = C\sqrt{h / P_1}$$

$$Q^2 = Ch / P \quad (16)$$

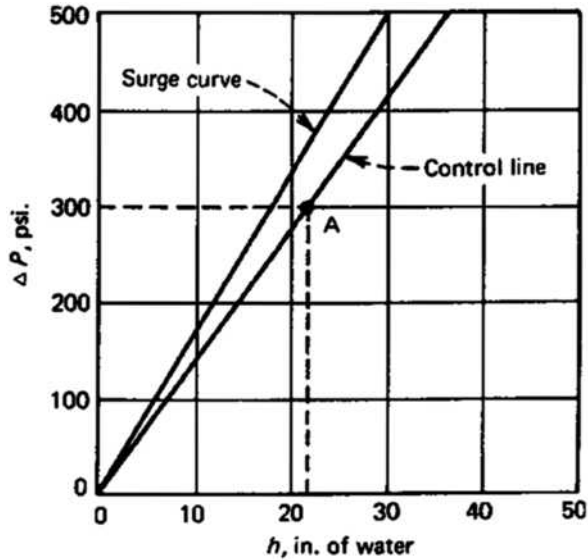


Figure 3.3. Control line is generally displaced about 10 percent of Q , or 20 percent of ΔP , from the surge curve

Substituting Eq. (16) in Eq. (11) yields:

$$\begin{aligned} \Delta P / P_1 &= C(h / P_1) \\ \Delta P &= Ch \end{aligned} \quad (17)$$

Equation (17) is used to calculate the surge curve, which will be our basic design equation. It verifies two important points:

1. There is a linear relationship between ΔP and h , as seen in Figure 3.3 and
2. The surge curve is not affected by variations in suction pressure, P_1 .

Figure 3.3 also shows the control line, displaced to the right of the surge curve. Moving the control line to the right reduces the possibility of reaching surge conditions in case a rapid decrease in flow causes an overshoot to the left of the control line. However, if the control line is placed too far to the right, gas will be bypassed unnecessarily and power wasted.

Consequently, the location of the control line represents a compromise based on the actual operating conditions of each individual system. As a general rule, the control line is displaced about 10 percent of flow,

or 20 percent of differential pressure. However, there are cases where the compressor must operate close to the surge curve, and it is necessary to reduce this margin.

EFFECT OF TEMPERATURE CHANGES

In deriving Eq. (17), we assumed that the compressor inlet temperature was constant. Since this is not always true in actual practice, it is necessary to investigate the effect of changes in this variable on the performance of the control system.

In order to determine the change in ΔP resulting from a change in T_1 , we start with Eq. (5), which expresses the relationship between compression ratio (R_c) and temperature. If it is assumed that the head and gas composition are constant, the factors L , m , and ϕ in the equation will be constant. The factor Z_1 may be somewhat affected by the temperature, but for this part of the analysis, it will be considered constant.

Equation (5) reduces to:

$$R_c = (1 + C/T_1)^{1/\phi} \quad (18)$$

Figure 3.4 shows a plot of this equation for two gases, air and natural gas, that have widely different molecular weights and specific heat ratios. The curves show the magnitude of the change in the compression ratio for a 120°F. change in the inlet temperature in a compressor producing a head of 40,000 ft·lb/lb.

From Eq. (9), $\Delta P = P_1 (R_c - 1)$, it is evident that for any particular value of suction pressure (P_1), ΔP will vary directly with $R_c - 1$, and the shape of the curves for ΔP versus T_1 is the same as that for R_c versus T_1 .

Now, we will consider the effect of changes in the differential pressure across the orifice, h , resulting from variations in suction temperature. Starting with Eq. (15),

$$\begin{aligned} Q &= C\sqrt{hT_1 / P_1} \\ Q^2 &= C(hT_1 / P_1) \\ h &= C(Q^2 P_1 / T_1) \end{aligned} \quad (19)$$

Then, for specific values of Q and P_1

$$h = C/T_1 \quad (20)$$

The curves in Figure 3.4 also show that both ΔP_1 and h vary inversely with the absolute inlet temperature, T_1 . If these temperature effects were identical, both sides of Eq. (17) would be changed equally and the system would be exactly self-compensating. However, this is not the case—although both variables change in the same direction, the magnitudes are different, with the result that there is a shift in the slope of the surge curve.

In most cases, this problem can be solved simply by placing the control line to the right of the surge curve for the highest temperature and accepting the fact that, when minimum temperature conditions exist, some gas may be bypassed unnecessarily.

If temperature variations are too great, or if the compressor must operate close to the surge curve, it will be necessary to provide temperature compensation in the control system. The instrumentation required to perform this function will be described later.

MOLECULAR-WEIGHT CHANGES

In most processes, each compressor handles the same gas all the time. Typical examples are air, refrigerant, and natural-gas pipeline compressors. In some applications, however, the compressor may be called on to handle gases of varying composition.

Since a variation in composition almost always results in a change in molecular weight, it is important to determine the effect of this change on the surge curve. Since the surge curve is plotted with ΔP versus h , we must determine how each of these variables is affected.

Referring back to Eq. (5):

$$R_c = (1 + Lm\phi / 1,545T_1Z_1)^{1/\phi}$$

for fixed values of L , T_1 , and Z_1 , this equation reduces to:

$$R_c = (1 + Cm\phi)^{1/\phi} \quad (21)$$

This equation indicates that, as with temperature, there is a nonlinear relation between molecular weight, m , and differential pressure across the compressor, ΔP . This is further complicated by the fact that a change in molecular weight is accompanied by a change in the specific heat ratio, K , and consequently the factor ϕ .

In some cases, such as the lighter saturated hydrocarbons (i.e., methane, ethane, and propane, and mixtures of these gases), there is a definite

relationship between m and K , as shown in Figure 3.5. This permits an accurate calculation of the change in R_c and ΔP for a given change in m .

However, in some processes, the compressor may be handling two or more gases where there is no correlation between m and K . An example of this would be a methane compressor that is also periodically used to purge the system with nitrogen. Under such a condition, each value of R_c and ΔP must be calculated individually to determine the magnitude and direction of the change.

To compute the effect of changes in m on the value of h , we must include m in Eq. (15), which then becomes:

$$Q = C\sqrt{hT_1 / P_1 m} \quad (22)$$

$$h = CQ^2 P_1 m / T_1 \quad (23)$$

Then, for specific values of Q , P_1 , and T_1 ,

$$h = Cm \quad (24)$$

By comparing Eqs. (21) and (24), it is apparent that the system will not be exactly self-compensating for changes in molecular weight. For lighter hydrocarbons, changes in m produce changes in the same direction in both ΔP and h , but in different magnitudes. Figure 3.6 shows how the surge line shifts in slope for changes in m from 16 to 20.

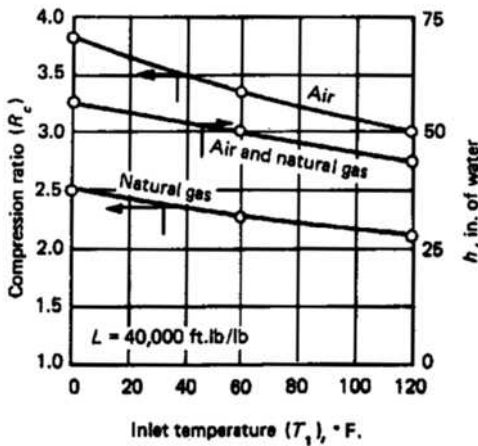


Figure 3.4. Curves for both compression ratio, R_c , and h vary inversely with gas inlet temperature

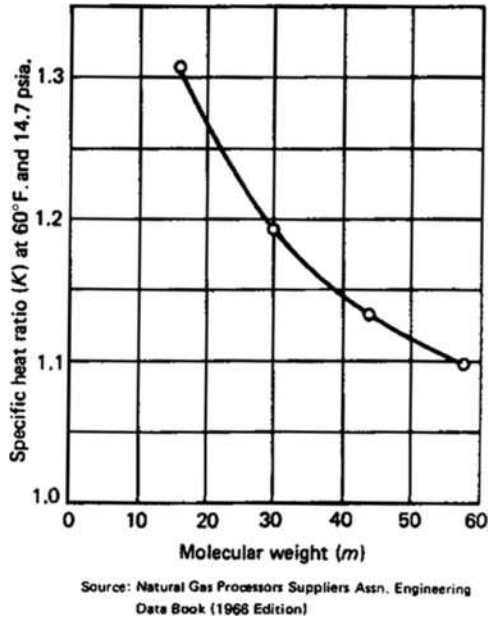


Figure 3.5. Definite relationship exists between molecular weight and specific heat ratio for lighter hydrocarbons

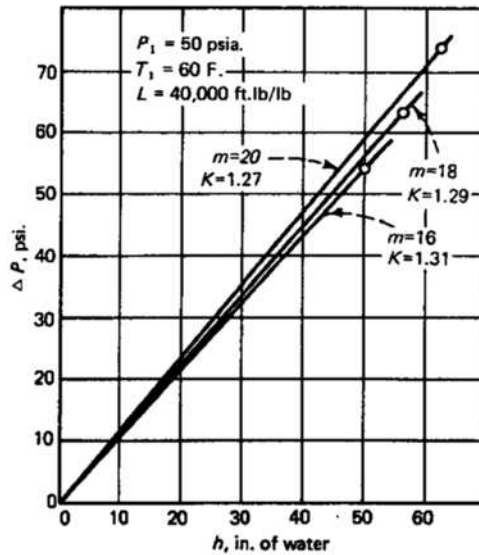


Figure 3.6. Increasing m shifts the surge curve to the left

It is not possible to generalize, however, concerning the overall effect of molecular-weight changes because of the influence of the specific-heat-ratio factor. As mentioned earlier, when two or more unrelated gases are handled by the same compressor, the slope of the surge line must be calculated for each condition. And if there is too great a divergence to be covered by a single control line, provision must be made to adjust the slope as required. This is usually done manually.

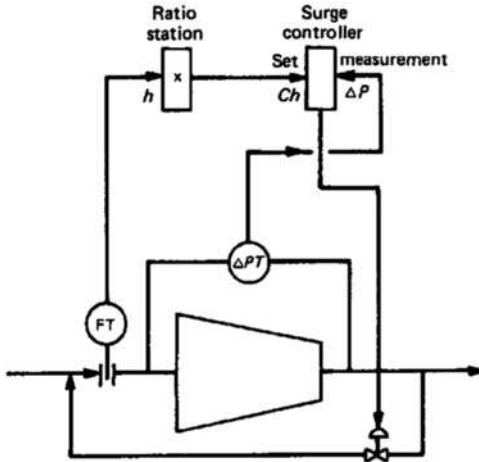


Figure 3.7. A basic scheme to provide anti-surge control

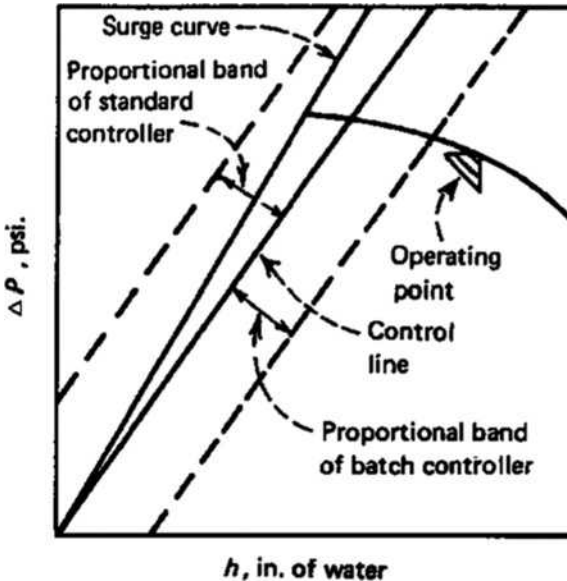


Figure 3.8. "Batch" controller provides anti-windup protection

INSTRUMENTATION

The instrumentation required to use Eq. (17) is shown in Figure 3.7. This equipment, of course, may be either pneumatic or electronic. A high-range differential-pressure transmitter measures ΔP and sends an output signal that becomes the measurement signal to the surge controller. The flow transmitter is connected across a primary device such as an orifice or a venturi in the compressor suction line, and its output signal, h , is fed to a ratio station. There it is multiplied by the constant C and becomes the set point of the surge controller.

The surge controller should have, in addition to proportional and reset functions, an anti-windup function, sometimes referred to as the “batch” feature.

To understand the need for a “batch” feature, consider Figure 3.8. Under normal conditions, the compressor operates in an area some distance from the control line. This results in an offset between the measurement and the set point of the controller. In a standard proportional-plus-reset controller, this causes the output signal to wind up to either its high or low limit.

In this condition, the proportional band and the operating point will be on opposite sides of the set point or control line, and no control action will be obtained until the measured operating point reaches the control line. If the measurement approaches the control line rapidly, it will overshoot before the controller can unwind, and the compressor may surge.

The anti-windup or “batch” function is arranged so that when the controller output reaches its limit, the reset loading is adjusted to shift the proportional band to the same side of the control line as the measurement, as indicated in Figure 3.8. Then, if it approaches the control line rapidly, the measurement enters the proportional band, and the control action starts before it reaches the control line; overshoot is thus reduced or eliminated. If the measurement change is slow—that is, no faster than the reset rate—the controller will perform as a standard instrument.

INSTRUMENT RANGES

The basic anti-surge control system shown in Figure 3.7 includes two differential-pressure transmitters—one measuring the differential pressure across the primary flow device, h , and the other measuring the differential pressure across the compressor, ΔP .

If flow measurement is used only for anti-surge control, the primary device can be designed for optimum conditions. The preferred design for maximum flow is about 25 to 30 percent above the maximum volume flow

indicated by the surge curve. To keep pressure loss as low as possible, the maximum differential pressure, h , should be as small as practical.

If flow measurement is used in the normal operation of the compressor, the primary device will probably be designed for a considerably higher maximum flow. As a result, the value of h at the surge line will be a much smaller percent of full scale and may require an undesirably high setting of the ratio station. Under these circumstances, it is better to install a second transmitter with a lower range for use only with the anti-surge system.

Calculations required to establish the range of the ΔP transmitter depend on the variables used by the compressor manufacturer to plot the surge curve. When the curve is plotted with discharge pressure versus flow for a given suction pressure, a simple subtraction produces the maximum ΔP at surge.

If the surge curve is plotted using compression ratio versus flow, the maximum value of operating point (OP) can be obtained from Eq. (9). However, if the manufacturer has used adiabatic head versus flow to draw the surge curve, calculations become a little more involved, and it is necessary to use Eq. (5) to calculate the compression ratio and then compute ΔP from Eq. (9).

Having determined the maximum ΔP , the transmitter range should be set at some reasonable amount above this value (e.g., 20–25 percent).

In order to plot the surge curve in the form shown in Figure 3.3, which is expressed by Eq. (17), about four points should be selected from the manufacturer's surge curve, and the corresponding values of ΔP and h computed.

The ΔP values can be calculated in the same manner as the range of the ΔP transmitter is determined. To compute the corresponding values of h , we use Eq. (15):

$$\begin{aligned} Q &= C\sqrt{hT_1 / P_1} \\ h &= CQ^2 P_1 / T_1 \end{aligned} \quad (25)$$

Using the subscript D to indicate the design conditions for the primary flow device, the equation for full-scale flow is:

$$H_D = CQ_D^2 P_D / T_D \quad (26)$$

and dividing Eq. (25) by Eq. (26) yields:

$$h = (h_D)Q^2 P_1 T_D / Q_D^2 P_D T_1 \quad (27)$$

If the Rowing pressure and temperature are the same as the design conditions, Eq. (27) reduces to:

$$h = h_D(Q / Q_D)^2 \quad (28)$$

Points plotted using ΔP and h will usually fall very close to a straight line. The control line in Figure 3.3 is located by adding 5 to 10 percent to the flow values and calculating the corresponding values of h . This provides the necessary safety margin to keep the compressor out of the surge area.

Some compressors with very high compression ratios have parabolic surge curves in the lower range that straighten out, or even bend to the right, at the upper end. These curves, when replotted as ΔP versus h , obviously do not produce a straight line. However, in almost every case, a control line can be drawn that will provide adequate protection over the normal operating range of the machine. If this control line does not pass through the origin, it is necessary to add bias between the ratio station and the controller.

RATIO SETTING

With the control line established, as in Figure 3.3, the slope of the line (the factor C in Eq. 17) represents the setting of the ratio station. Rearranging Eq. (17) produces:

$$C = \Delta P / h \quad (29)$$

In calculating C , values of ΔP and h must be expressed in terms of the percentage of full scale of their respective transmitters. This calculation can best be shown by an example. If we assume, as in Figure 3.3, that the transmitters have the following ranges:

$$\Delta P = 0\text{--}500 \text{ psi}$$

$$h = 0\text{--}50 \text{ in. of water}$$

and if we select an arbitrary point, A , on the control line where:

$$\Delta P = 300 \text{ psi}$$

$$h = 22 \text{ in. of water}$$

$$\text{then: } \Delta P = (300 / 500)100 = 60\%$$

$$\text{and : } h = (22 / 50)100 = 44\%$$

Using these values in Eq. (29) gives:

$$C = 60 / 44 = 1.36$$

and the equation becomes:

$$\Delta P = 1.36h$$

AUTOMATIC TEMPERATURE COMPENSATION

As we mentioned earlier, the slope of the surge curve changes with variations in the compressor inlet temperature. When these changes in the inlet temperature are large and occur frequently, and when the compressor must operate near the surge curve, it may be wise to provide automatic temperature compensation in the anti-surge control system.

This can be done by replacing the manually adjusted ratio station in Figure 3.7 with an analog dividing computer, as shown in Figure 3.9. By properly scaling the computer, the equation:

$$\Delta P = Ch/T_1 \quad (30)$$

can be solved. This equation indicates that the temperature correction is inversely proportional to the first power of the absolute temperature. Although, in theory, this is not precisely correct, the error introduced by this assumption is negligible.

When the largest temperature changes occur over long periods of time (e.g., seasonally), changes in the slope of the control line can be made manually to keep the system as simple as possible (as in Figure 3.7).

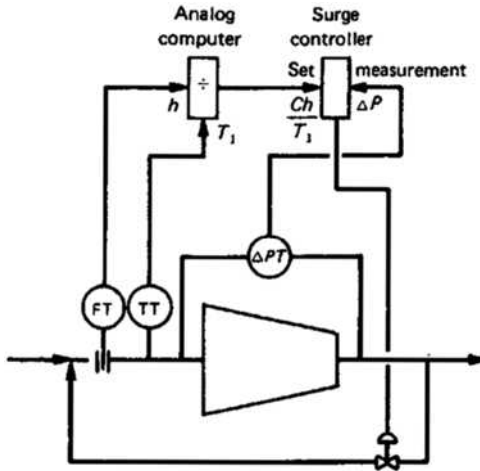


Figure 3.9. Control arrangement to provide automatic temperature compensation uses an analog dividing computer

INLET-GUIDE VANES

Constant-speed centrifugal and axial compressors are frequently equipped with adjustable inlet-guide vanes to control the output flow. In addition, the axial compressor may have adjustable stator blades. Besides influencing the compressor output, moving these vanes also changes the slope of the surge curve. In other words, there is a surge line for each vane position, just as there is for each inlet temperature.

Since the magnitude of this change in the slope of the surge line is different for each compressor design, there is no convenient way to express it mathematically. However, it is generally true that moving the vanes in the counter-rotation direction will reduce the slope of the surge line (i.e., it moves the surge line to the right). It is also generally true that the change in slope is not a linear function of the vane position.

There are several solutions to this problem. If the change in slope is reasonably small, it may be possible to use the surge line for the maximum counter-rotation vane position and set the control line accordingly, as in Figure 3.3. This places the control line on the safe side of the surge line for all vane positions. But it also provides an unnecessarily wide safety margin when the vanes are moved in the pre-rotation direction. This could result in bypassing gas at times when it is not required. This drawback must be weighed against the advantage of using the simple control system shown in Figure 3.7.

When vanes are positioned from a manual station, and the position is not frequently changed, it may be satisfactory to use the basic control system and adjust the setting of the ratio station manually. This can be done by providing the operator with a table showing the correct ratio setting for each vane position.

When frequent changes in the vane position produce large variations in the slope of the surge line (as when they are operated by a controller), it is usually desirable to make the changes in the ratio setting automatically. A system like the one for automatic temperature compensation (Figure 3.9) can be used. The signal from the controller, t , the vane operator is a measure of the vane position and, instead of the temperature measurement, can be fed into the analog computer to adjust the ratio setting. This system produces a linear relation between the ratio setting and the vane position that is not precisely correct. However, the errors introduced are usually small enough to be neglected.

DISCHARGE-LINE FLOW MEASUREMENT

The anti-surge control system based on Eq. (17) uses a primary flow-measuring device in the compressor suction line. This system offers

the very important advantage of being self-compensating for changes in suction pressure and, at least partially, for variations in suction temperature. These benefits are not realized when the flow measurement device is located in the discharge line.

In some installations, because of the piping configuration or the size of the line, it is not feasible to put the primary flow device in the suction line. This may also be true when the compressor operates with a very low suction pressure and any additional pressure drop in the line cannot be tolerated. It may be possible to resolve the problem by using the inlet eye of the compressor as an orifice. However, it must first be established that this measurement is a usable one (i.e., noise free, repeatable, and representative). It also requires individual calibration.

If this cannot be done, it is necessary to install the primary flow device in the discharge line and modify the anti-surge control system accordingly. This modification consists of adding analog computers to calculate the value of h for use in Eq. (17). The instrumentation to perform this computation is shown schematically in Figure 3.10.

Under steady-state conditions, the weight and flow (in lb/min or in scfm) are the same at both suction and discharge, that is,

$$W_1 = W_2 \quad (31)$$

Using Eq. (12) to substitute for W provides:

$$C_1 \sqrt{h_1 P_1 / T_1} = C_2 \sqrt{h_2 P_2 / T_2}$$

If the primary flow devices are designed for the same maximum values of weight flow, and if we make

$$\sqrt{h_1 P_1 / T_1} = \sqrt{h_2 P_2 / T_2}$$

then

$$h_1 = h_2 (P_2 T_1 / P_1 T_2) \quad (32)$$

With these relative maximum values of h_1 and h_2 , the same relation will hold true at any point on the scale and we can use Eq. (32) to calculate h_1 under all conditions. The surge curve and control line can be plotted in the conventional manner, and the setting of the ratio station can be computed by the method described earlier.

Although the inlet temperature may vary considerably, in some compressor applications, the ratio T_2/T_1 will be nearly constant (remember these are absolute temperatures).

If the temperature ratio is assumed constant, Eq. (32) reduces to:

$$h_2 = Ch_2 P_2/P_1 \quad (33)$$

This greatly simplifies the control system by eliminating the temperature transmitters as well as two analog computers (i.e., the one to divide T_2 by T_1 and the one to multiply the temperature ratio by the pressure ratio).

When the compressor operates at a constant speed, the instrumentation can be further simplified. Under this condition, the compression ratio is constant at the surge point, so Eq. (33) becomes:

$$h_1 = Ch_2 \quad (34)$$

This makes it possible to eliminate two more analog computers from Figure 3.10 and use the standard basic control system, by replacing h_1 with the corrected value of h_2 .

COMPRESSORS IN A SERIES

When two or more compressors are connected in a series, using the control system shown in Figure 3.7 on each individual machine will provide the best protection against surge. This, of course, requires a recirculation control valve for each compressor.

In some designs, two compressors in a series are driven by the same prime mover, and consequently, they both always run at the same speed. The machines are sometimes considered as a single unit: The manufacturer may provide one surge curve for the combination and the installation usually includes only one recirculation control valve. For surge control, the system shown in Figure 3.7 is used.

When compressors are independently driven at variable speed, but have a single recirculation control valve, the surge is controlled by the pneumatic system shown in Figure 3.11. Each compressor is equipped with its own ratio station and controller, but the two are combined into an auto-selector system to enable either controller to operate the common recirculation valve as required.

If, at times, either compressor operates singly, the three-way valves in the controller output lines should be installed to ensure a positive air supply

to the selector relay of the controller in operation. If both compressors will always operate together, the three-way valves may be eliminated altogether.

COMPRESSORS IN PARALLEL

When centrifugal compressors are operated in parallel, the problem of properly dividing the load is always present. The solution is particularly difficult when the compressors have dissimilar characteristics. But even when the characteristics are identical, some means must be provided to ensure that the desired load distribution is obtained. Controls to accomplish this will not be discussed here, except as they affect the anti-surge control system.

Figure 3.12 shows the effect of inadequate control. With a differential pressure of 80 psi and both compressors running at 100 percent speed, the load is equally divided and the flow through each machine is Q_1 .

If, however, for some reason such as dissimilar governor characteristics, one compressor is operating at only 98 percent speed and the other is at 102 percent, the flows will be Q_2 and Q_3 , respectively. If the difference in speeds is large enough, the flow through one machine will decrease sufficiently to cause surging.

From Figure 3.12, it can be seen that the flatter the characteristic curves, the more difficult the problem becomes. For this reason, centrifugal compressors require more precise methods of load distribution than axial-flow compressors, which have much steeper curves. This susceptibility to surge makes it even more essential to provide positive protection for centrifugal machines during parallel operation.

The control arrangement shown in Figure 3.13(a) allows maximum operating flexibility with minimum supervision. It consists of a standard surge control system for each compressor, which provides complete protection at all times, whether the machines are operating singly or together. It is even possible to start a compressor with the surge controller in automatic while the other compressor is running. As the newly started machine approaches its operating speed, the surge controller holds the recirculation valve open just far enough to bring the compressor up along the control line and keep it from surging. And this system can be used for any number of machines.

Figure 3.13(b) shows an alternate surge control system that requires less equipment. It uses only one ratio station and controller, one OP transmitter, and a single recirculation control valve. When both compressors are operating, the low selector relay chooses the lower flow signal and sends it to the ratio station. If only one machine is running, its flow signal is connected to the ratio station by a manual selector switch. This selection can also be performed automatically with relay logic, if desired.

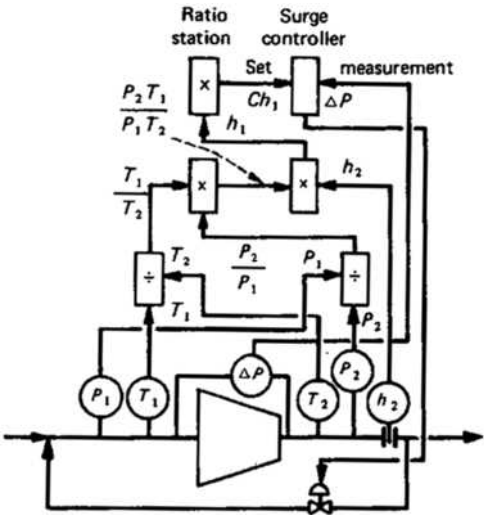


Figure 3.10. A schematic diagram shows the instrumentation required when the primary flow-measuring device is located in the centrifugal-compressor discharge line

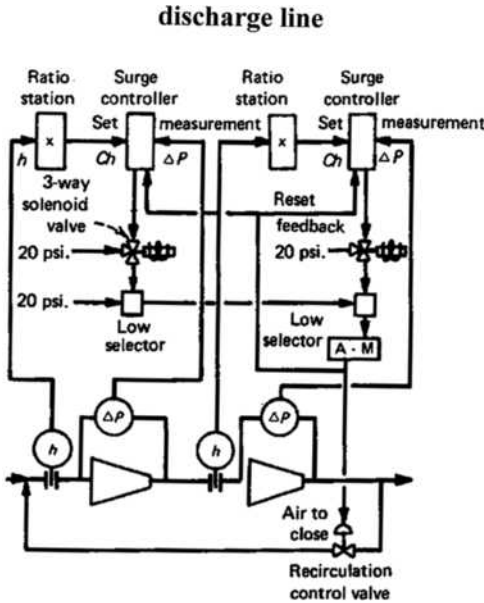


Figure 3.11. The system for controlling compressors in a series

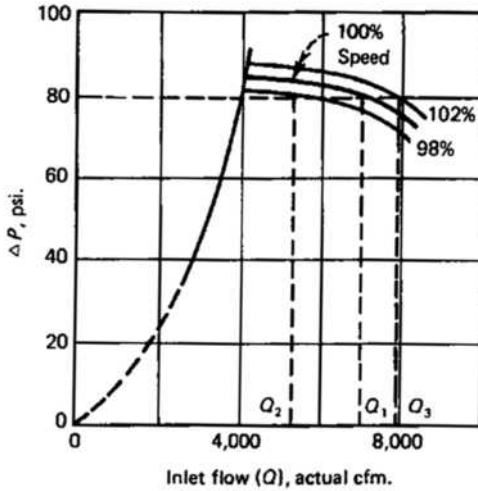


Figure 3.12. Load distribution for parallel compressors can be seriously affected if they run at different speeds

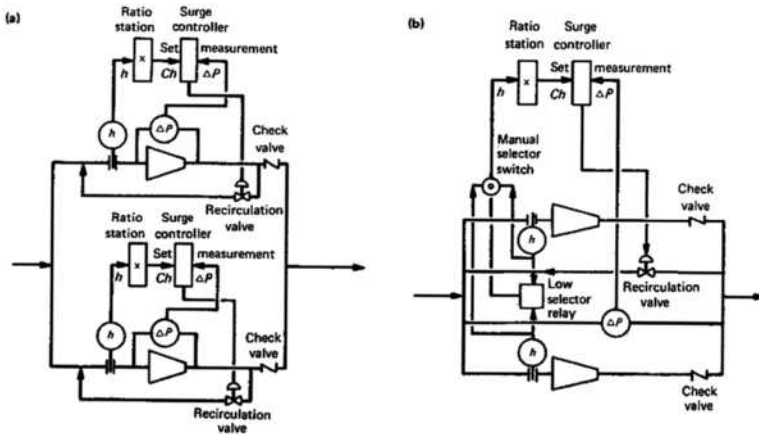


Figure 3.13. Alternate control arrangements for centrifugal compressors operating in parallel

The cost savings resulting from the use of this system, however, are not obtained without some sacrifice. For example, each compressor must be equipped with a manual recirculation valve for starting. With one machine operating, the recirculation control valve cannot be used for starting the second compressor, so the operation must be performed manually and without automatic surge protection.

IMPROVED SURGE CONTROL FOR CENTRIFUGAL COMPRESSORS

Surge may occur without being detected by conventional control systems, cutting compressor and process efficiency and causing compressor damage. Excess flow recirculation or blowoff can help avoid some surges, but energy costs are high. Here is a control system that copes with all surges without incurring such penalties.

The operation of a centrifugal compressor can become unstable due to changes in many conditions such as flow rate, pressure, and the molecular weight of gas. This causes rapid pulsations in flow, called surge. No system is immune to sudden upsets at one time or another.

We have surge-tested air compressors, gas compressors, centrifugal compressors, and axial compressors, at high and low pressures. The amplitude of the flow drop and the frequency of surge cycles vary. However, there are certain common results: The pressure change is less than the flow change in every case that we have recorded; flow usually drops extremely fast just before surge, and then always drops precipitously during surge, at least in the authors' experience, a time of 0.05 second from set point flow to reverse flow being common.

The speed of some surges is such that conventional control systems cannot detect them, let alone react to them. So, often even when records indicate that no surge has taken place, stripping down a compressor will reveal surge damage, ranging from changes in clearances, which exact a penalty in compressor efficiency, to destruction of parts.

Unstable operation—whether it is detected or not—affects the operation of the process that relies on the unit.

Typically, surge is prevented by recirculating some of the flow or by blowing off the excess flow. Constant recirculation of flow—often at 30 to 40 percent of the amount needed by the process—is common. For instance, this is found frequently in units used to compress chlorine and wet gases, among others. Obviously, this excess flow costs energy. And it does not necessarily avert all surges.

IMPROVED SURGE CONTROL

Here, we will present an improved surge-control strategy and instrumentation system. This system copes with even very rapid upsets, minimizing disturbances to process operations. The amount of recirculation is reduced sharply, saving significant amounts of energy. Furthermore, the system lends itself to actual field calculation of the surge limit.

With conventional closed-loop control schemes, once surge begins, the compressor control system oscillates, and the only way to stop this is by manual override. Here, if surge occurs, the compressor will automatically be brought out of it during the first surge cycle, and the control system will reset itself to prevent such oscillations from recurring.

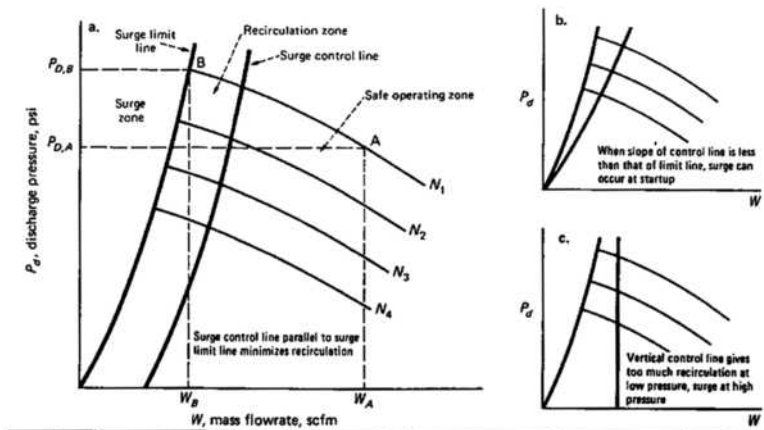
Conventional systems can be set into oscillation by operator errors; such errors are minimized by this system. Also, fluctuations in the pressure and flow rate of the compressed gas are kept to a minimum, resulting in relatively minor disturbances to the process. (The reasons for these advantages will be explained later.)

In order to develop this control system, we will first relate surge to compressor operation, then determine where surge begins and where to set the control system to prevent it. Next, we will discuss instrumentation, control strategies, and calibration of the equipment. Compressors in a series and in parallel will not be discussed here, due to limitations of space.

SURGE CHARACTERISTICS

Figure 3.14 shows characteristic curves for a typical single-stage centrifugal compressor. Each rotational speed (N_1, N_2, N_3, N_4) has its own characteristic curve.

If the compressor is operating at speed N_1 at Point A (mass flow rate W_A) and the flow is decreased, surge will occur at W_B . The pressure has increased from $P_{D,A}$ to $P_{D,B}$. Point B lies on the surge limit line, a naturally occurring



Characteristic curves and surge control lines define regions of operation for compressor

Figure 3.14. Surge characteristics

line that is peculiar to each compressor geometry. The area to the left of this line is the surge zone, where there are oscillations of flow and pressure.

The line to the right of the surge limit line is the surge control line, which is an artificial line set by an engineer. These lines are the boundaries of the recirculation zone, where recirculation or blowoff is used to prevent surge. To the right of the control line is the safe-operating zone, where recirculation is considered unnecessary. In order to develop the control strategy, we must first develop improved equations for the control and limit lines.

SURGE LIMIT LINE

Here, we will formulate an equation for the surge limit line that does **not** assume constant inlet temperature and molecular weight. Typical equations assume that these parameters are constant. Corrections must be applied when conditions change.

The head of a centrifugal compressor is a function of angular velocity, speed of rotation, inlet volumetric flow rate, and impeller diameter. Using a form of the kinetic energy of the compressor, given by Davis and Corripio, and performing a dimensional analysis:

$$\frac{g_c H_p}{N^2 D^2} = f\left(\frac{Q_3}{ND^3}, Ma, Re\right) \quad (35)$$

At high velocities at which centrifugal compressors operate, the variation in Reynolds number, **Re**, with the velocity of the gas is negligible. The variation in Mach number, **Ma**, with velocity is also negligible. Mach number can be assumed constant as long as the gas velocity does not approach the speed of sound. We will assume that operation is well below this speed in the region close to surge.

For a given compressor geometry, under these conditions, if H_p/N^2 is plotted against Q_3/N , then the performance curves for different speeds will reduce to one curve, called the universal performance curve (Figure 3.15).

The surge limit line is reduced to a single point, the surge limit. Since the surge limit is fixed for a particular geometry, the values of this point are constant:

$$\frac{H_p}{N^2} = C_1 \quad (36)$$

$$\frac{Q_3}{N} = C_2 \quad (37)$$

Thus, the curves of the surge limit line follow the fan law, which states that the inlet volume is proportional to speed, and the head is proportional to the square of speed. The surge limit may be defined by either Eqs. (36) or (37). These equations apply not only to single-stage centrifugals but also to multistage units without intercoolers.

While Eqs. (36) and (37) can be used to set the surge limit (and the surge limit line), this is impractical since H_p and Q_s depend on a measurement of molecular weight.

Variations in molecular weight cannot be continuously measured at the high speeds present in compressors and, therefore, must be eliminated from the equation of the surge line. Also, H_p and Q_s depend on variations in the inlet temperature and pressure. We will eliminate molecular weight and temperature from the equation of the surge limit line. Eqs. (36) and (37) are combined to eliminate N :

$$C_3 H_p = Q_s^2 \quad (38)$$

Polytropic head is given by:

$$H_p = \frac{C_4 Z_{av} T_3}{M} \frac{(R_c^\sigma - 1)}{\sigma} \quad (39)$$

Substituting H_p in Eq. (38):

$$\frac{C_3 Z_{av} T_3}{M} \frac{(R_c^\sigma - 1)}{\sigma} = Q_s^2 \quad (40)$$

The volumetric flow rate equals the mass flow rate divided by density:

$$Q_s = \frac{W}{\text{density}} = \frac{C_6 W}{(P_s M / Z_s T_s)} \quad (41)$$

Substituting Q_s in Eq. (40):

$$C_7 Z_{av} \frac{(R_c^\sigma - 1)}{\sigma} = \frac{W^2 Z_8^2 T_8}{P_s^2 M} \quad (42)$$

Using an equation for gas flow across an orifice in the suction of the compressor:

$$W = C_8 \sqrt{\frac{\Delta P_{or,s} P_8 M}{T_8 Z_8}} \quad (43)$$

Equation (42) becomes:

$$C_9 Z_{av} \frac{(R_c^\sigma - 1)}{\sigma} = \frac{\Delta P_{or,s} P_8 M Z_8^2 T_8}{Z_8 T_8 P_8^2 M} \quad (44)$$

Or

$$C_9 \frac{Z_{av}}{Z_8} P_8 \frac{(R_c^\sigma - 1)}{\sigma} = \Delta P_{or,s} \quad (45)$$

Equation (45) may be written for the discharge side of the compressor. Since the mass flow rate at any instant in the compressor is the same at suction and discharge, Eq. (43) can be written as:

$$W = C_8 \sqrt{\frac{\Delta P_{or,s} P_8 M}{T_8 Z_8}} = C_8 \sqrt{\frac{\Delta P_{or,d} P_d M}{T_d Z_d}} \quad (46)$$

Or

$$\Delta P_{or,s} = \Delta P_{or,d} \frac{P_d}{P_8} \frac{T_8 Z_8}{T_d Z_d} \quad (47)$$

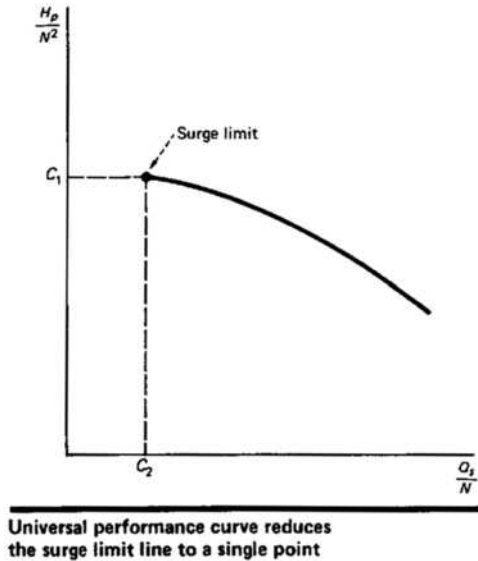


Figure 3.15. Performance curves

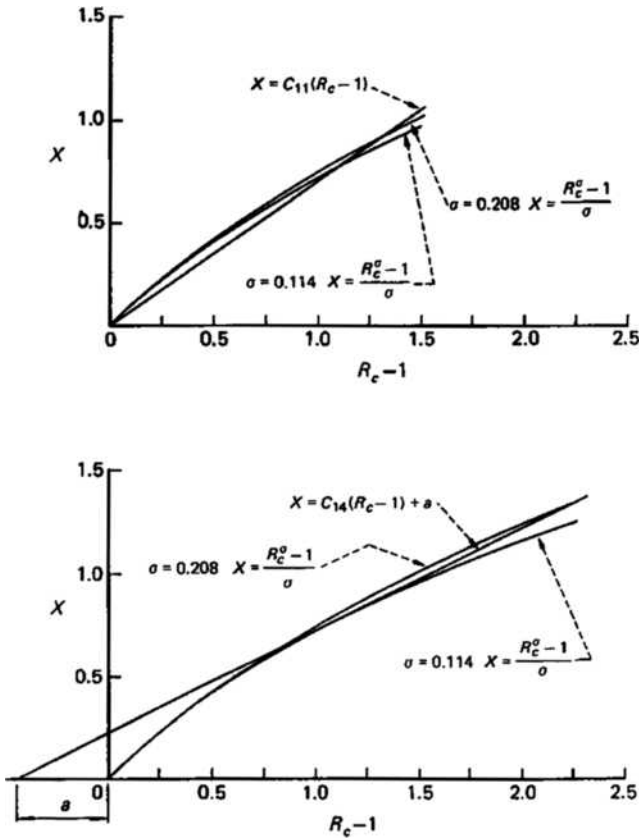


Figure 3.16. Surge control

Nomenclature

b_a, b_b, b_n	Distance between surge control line and surge limit line ($n = 1, 2, 3$). in. H_2O
C_n	Constant ($n = 1, 2, 3, \dots$). Units vary from equation to equation
D	Impeller diameter, ft
d_1	Bias added by summing device in No. 5, Fig. 7, psi
d_2	Bias added by summing device in No. 7, Fig. 7, psi
g_c	Acceleration of gravity, ft/s^2
H_p	Polytropic head, $(ft-lb)/lb$
k	Ratio of specific heat at constant pressure to specific heat at constant volume
k_l	Slope of surge control line, dimensionless

M	Molecular weight
Ma	Mach. number
N	Speed of rotation, rpm
ΔP_c	Differential pressure across compressor, psi
$\Delta P_{or,s} \Delta P_{or,d}$	Orifice pressure at suction; discharge psi
P_s, P_d	Suction pressure; discharge pressure, psi suction; discharge psi pressure, psi
Q_s	Volumetric flowrate in suction, actual cfm
R_c	Pressure ratio across compressor
Re	Reynolds number
T_s, T_d	Temperature at suction; discharge, R
W	Mass flowrate, scfm
Z_{av}	Average compressibility = $(Z_s + Z_d)/2$
<i>Greek letters</i>	
σ	$\frac{k-1}{k\eta_p}$
η_p	Polytropic efficiency

Equation (45) becomes:

$$C_{10} \frac{Z_{av} Z_d T_d P_8^2 (R_c^\sigma - 1)}{Z_8^2 T_8 P_d} \frac{1}{\sigma} = \Delta P_{or,d} \quad (48)$$

If we assume that charges in compressibility are small, then $Z_{av} = Z_d = Z_s$. The term $(R_c^\sigma - 1)/\sigma$ can be approximated by a linear function (Figure 3.16). If R_c is less than 2.5, the approximation $C_{11}(R_c - 1)$ can be used. Equation (45) and (48) become:

$$C_{12}(P_d - P_8) = C_{12} \Delta P_c = \Delta P_{or,s} \quad (49)$$

$$C_{13} \frac{P_8 T_d}{P_d T_8} (P_d - P_8) = C_{13} \frac{P_8 T_d}{P_d T_8} = \Delta P_{or,d} \quad (50)$$

However, White arrived at the same result by assuming that the inlet temperature and molecular weight were constant; he later offered corrections for these. Here, we have shown that such corrections are unnecessary.

As in White's case, here the surge limit line is unaffected by variations in suction pressure.

If R_c is greater than 2.5, $(R_c^\sigma - 1)/\sigma$ may be approximated by $C_{14}(\Delta P_c - 1) + a$ and the surge limit line becomes:

$$C_{15}\Delta P_c + a_2 P_8 = \Delta P_{or,s} \quad (51)$$

When changes in inlet pressure are negligible:

$$C_{15}\Delta P_c + b_1 = P_{or,s} \quad (52)$$

where b_1 is a constant.

Note that the above equations apply to centrifugal compressors without intercoolers. With intercoolers, it is necessary to consider each section as a separate compressor or to use an approximation that considers the unit as a whole.

Eqs. (49), (51), and (52) are recommended for calculating the surge limit line, as they contain the fewest number of variables, all of which can be easily measured. Sometimes, the surge limit line is obtained from Eq. (37). When this is done, it is necessary to neglect variations in molecular weight, and this can lead to inaccuracies, even on air compressors, since the humidity of ambient air varies.

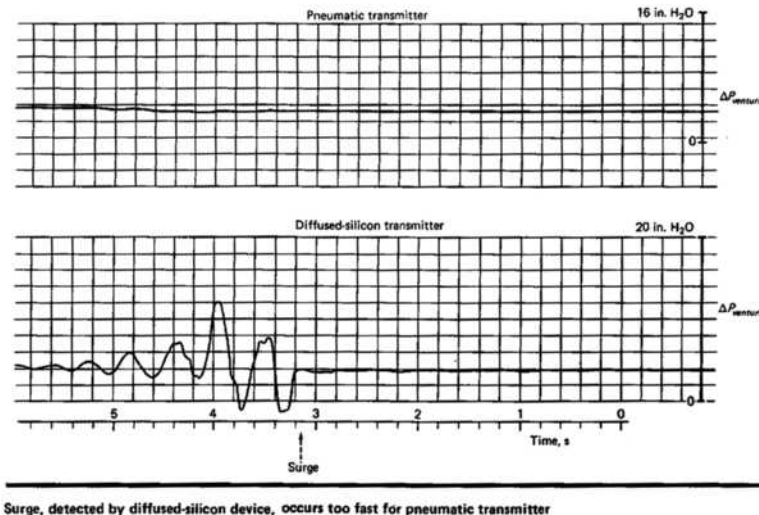
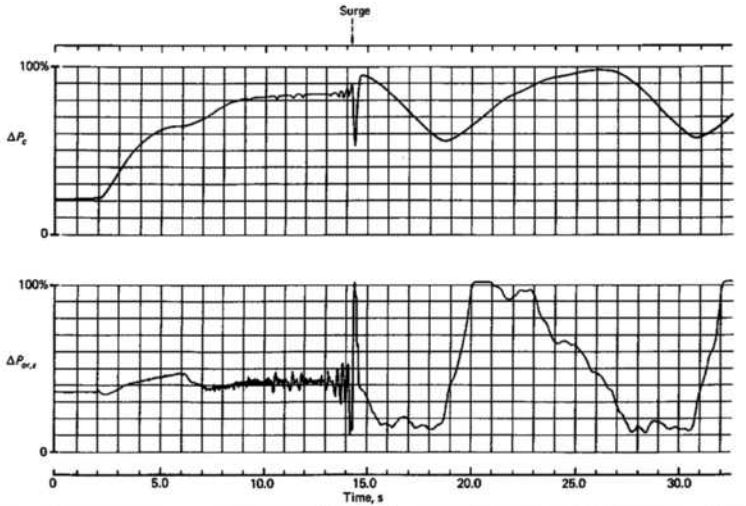


Figure 3.17. Detection of surge



Flow drops precipitously before surge cycles begin, then reverses quickly

Figure 3.18. Explanation of flow drops

SURGE CONTROL LINE

The three common forms of the surge control line are shown in Figure 3.14. The optimal position of this line is parallel to the surge limit line (Figure 3.14a). To minimize recirculation, the surge control line should be set as close to the surge limit line as possible. Setting the control line with a slope less than that of the limit line (Figure 3.14b) can lead to excess recirculation at high pressures, and surge at low pressures during stopping and startup. The third method is to select a minimum safe volumetric flow and set a vertical control line (Figure 3.14c). This can lead to excess recirculation at low pressures and surge at high pressures. Many systems measure the flow in the discharge without correcting for suction conditions. This gives maximum recirculation with minimum surge protection and is not recommended.

If it is desired to keep the control line parallel to the limit line, how close together can the two lines be? This depends upon how accurately the surge control line is set and how well it accounts for changes in inlet temperature and pressure, and molecular weight. Also critical to the location of the control line is the effectiveness of the anti-surge system in handling upsets. This effectiveness depends on the control strategy chosen and the nature of the anti-surge system, its transmitters, controller, and anti-surge valve.

To set the surge control line equidistant from the limit line, Eq. (49) can be used. The control line is displaced to the right by some fixed amount, b_a :

$$C_{12}\Delta P_c + b_a = \Delta P_{or,s} \quad (53)$$

Equation (53) applies when R_c is less than 2.5. For values of 2.5 and greater, Eq. (51) may be used. The control line is displaced by some fixed amount, b_b :

$$C_{15}\Delta P_c + a_2 P_s + b_b = \Delta P_{or,s} \quad (54)$$

INSTRUMENTATION AND VALVES

How fast must the anti-surge system be to detect the onset of surge and effectively stop it?

It is often thought that a very rapid response is not necessary; also, many believe that protection against large upsets is unnecessary. However, upsets that are both fast and large are not rare, and these can be caused by factors such as jammed check valve, operator errors, and shutdowns of process equipment triggered rapidly by protective systems.

We have found that surge cycles can occur faster than is thought possible and that often surge goes undetected. This is because conventional pneumatic controllers are too slow to detect this phenomenon. Only fast transmitters can cope with the high speeds of compressor transients. We recommend a diffused-silicon electronic transmitter. We also recommend pressure-differential transmitters with a lag not greater than 0.035 second as necessary to maintain surge control and to calibrate the surge limit line. Also, recorders (used for calibrating and testing the anti-surge system) must have a chart speed not less than 25 mm/s.

Figure 3.17 compares the response of a diffused-silicon transmitter with that of a pneumatic device. Pressure differential was measured across a venturi. At about 3.2 second, surge began, and the pneumatic transmitter was too slow to detect flow fluctuations.

The slowness of most transmitters and recorders explains why records will show no surge, yet upon inspection, compressors reveal changed clearances or damaged impellers, seals, or bearings. Such undetected surges will gradually alter the compressor, resulting in decreased efficiency. Some electronic transmitters are dampened and are too slow for use here; others are not stable at high speeds.

Figure 3.18 shows how quickly surge can begin. Tests were made on a centrifugal air compressor that supplied a blast furnace. Here, surge occurred at about 14 second. Occasionally, there are pre-surge oscillations (incipient surge). Flow drops quickly before surge occurs. As surge begins, flow drops precipitously—typically from a set point level to a reverse condition in about 0.05 second, regardless of the pressure or compressor speed (see Figure 3.17).

ANTI-SURGE VALVE AND CONTROLLER

For proper surge control, the anti-surge valve must be large and fast acting. It should be capable of recirculating 100 percent of the design flow rate.

When flow is decreasing and operation is moving close to surge, it is necessary to effect a quick increase in flow. The rate at which flow can be increased depends on the response of the valve and its size. The larger the valve, the greater the effect of its opening. The valve should be able to recirculate the entire flow rate because sometimes complete blockage of the system occurs, and all of the flow must be recirculated.

Field tests show that the full stroke of the anti-surge valve should be from 0.5 to 1.5 second. New valves can meet this requirement; existing valves can be speeded up with boosters. Boosters increase the time of response by increasing the pneumatic signal for the valve actuator.

Why should the transmitter have a maximum response time of 0.035 second, while the anti-surge valve has a maximum time of 1:5 second? The

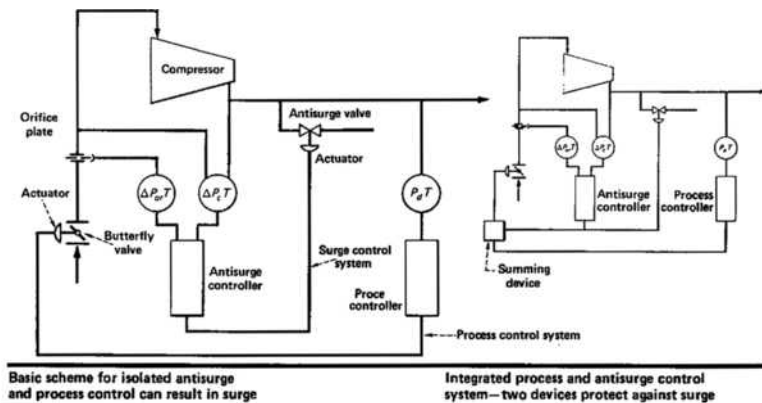


Figure 3.19. Basic scheme for isolated anti-surge and process control can result in surge (a). Integrated process and anti-surge control system—two devices protect against surge (b)

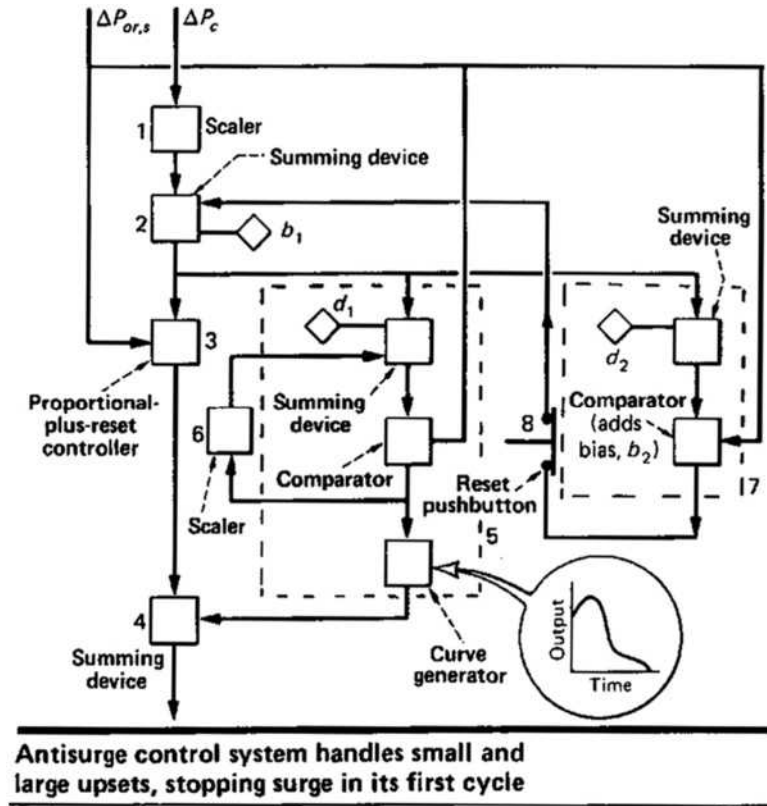


Figure 3.20. Anti-surge control

surge control line is set close to the surge limit line—typically, the distance between them is set at 15 percent of design flow. Thus, a partial valve stroke will be enough to stop the movement toward surge; further opening of the valve will restore the operating point to the surge control line.

The controller must be fast as well. Pneumatic controllers are too slow to reliably prevent surge. Digital controllers should be used with caution. Since these devices look in sequence at each control loop, the scanning time may not be short enough to detect the transients typical of compressor surge cycles. Microprocessors must have a scanning time less than 0.1 second.

CONTROL: ISOLATED ANTI-SURGE LOOP

First, we will look at the anti-surge control system without considering its interaction with the process controller. Then we will integrate the two

systems. In developing our control system, two typical systems will be studied first: a proportional-plus-reset controller and a relay (on–off) controller. Consider an electronic analog anti-surge controller that has a proportional-plus-reset response, with an anti-windup device (Figure 3.19a). The anti-windup device is necessary due to the nature of the proportional and reset functions. Normally, the compressor operates in an area some distance from the control line, resulting in an offset between the measurement and the set point of the controller. As a result, the output signal winds up to its high or low limit.

The proportional band and the operating point will be on opposite sides of the set point or control line; control will not be affected until the measured operating point reaches the control line. Surge can result if the measurement approaches the control line quickly, since overshoot will occur before the controller can unwind.

Anti-windup adjusts the reset loading to shift the proportional band to the same side of the control line that the measurement is on when the controller reaches its output limit. Then, if the control line is approached rapidly, the measurement enters the proportional band and the control starts before the value reaches the control line. Overshoot is reduced.

Derivative control is not recommended, for it can open the anti-surge valve far from the surge control line and can cause system oscillations. Rapid oscillations in flow, even in the safe-operating zone, can cause the valve to open because of the nature of the derivative response.

The response speed of this controller depends on the width of the proportional band and the reset time. These parameters influence the stability of the system. Decreasing the proportional band or increasing the reset time increases the speed of the controller's response, but past a certain point, the system stability will be disturbed—all closed-loop control systems have a stability limit.

The speed of an anti-surge controller's response is limited mainly by the inertia of the compressor and its networks, the transmitters, and the anti-surge valve and its actuator.

The slower these elements are, the slower the controller must be set. The inertia of the compressor and its piping network cannot be changed, but the inertia of the transmitter and the valve and its actuator can be decreased by a proper selection of faster elements.

Once fast transmitters and an anti-surge valve are selected, the proportional band level and reset time are set, based on the parameters of the compressor and its network. These parameters include the volume of the system, the length of the pipes, and the inertia of the compressor. This limitation is common to all closed control loops with feedback. Thus, modulated control responds well to slow upsets and give good control. However,

the valve cannot be opened quickly for fast upsets, due to limitations of system stability.

Now consider using a relay for control. The relay is part of an open-loop system that opens the anti-surge valve to a preestablished level after the compressor's operating point reaches the surge control line. The output of a relay device can change from minimum to maximum at any speed, without upsetting the system's stability. This is because (1) the relay is an on-off device and there is no limit as to how fast the device can be turned on and off and (2) the system is open loop and oscillations cannot be set up in the control system.

However, the relay system has certain disadvantages for anti-surge controllers. The preadjusted output level may be either lower or higher than that required for protection. If it is lower, the compressor's operating point will cross the surge limit; if it is higher, the compressor will be operated with large amounts of recirculation or blowing off when the anti-surge valve is opened. Also, the operator can be unsure as to when to reset the relay without endangering the compressor.

Comparing the closed-loop system to the open-loop system has led to a patented method that combines the advantages of both, while limiting their shortcomings.

Figure 3.20 shows the controller with inputs for Eq. (52), one form of the surge control line. The input corresponding to ΔP_c passes through a scaler (No. 1) and summing unit (No. 2) that transform it to the required form:

$$k_1 \Delta P_c + b_1 \quad (55)$$

where k_1 is the slope of the surge limit line and b_1 is the normal distance between the surge limit line and the surge control line.

The value of $k_1 \Delta P_c + b_1$ is compared with the value of $\Delta P_{or,s}$ by the proportional-plus-reset controller (No. 3). If $\Delta P_{or,s}$ is greater, then the controller's output is set at zero.

When $\Delta P_{or,s}$ is less than the signal given by Eq. (55), the controller begins producing a signal that increases until $k_1 \Delta P_c + b_1$ becomes equal to $\Delta P_{or,s}$.

As a result, the compressor's operating point will be restored to the surge control line. This system will work well if disturbances are small and slow, but cannot open the anti-surge valve quickly enough to prevent surge when disturbances are large or fast.

To handle large disturbances, the output from the summing device (No. 2) is decreased by a fixed amount, d_1 , which is typically set at one half the value of b_1 . This addition is done by the summing device in element 5. The comparator in element 5 compares the value of $\Delta P_{or,s}$ with that of $k_1 \Delta P_c + b_1 - d_1$. When $\Delta P_{or,s}$ is smaller, a signal is sent to the curve

generator. The output of the curve generator follows the shape shown in Figure 3.20; it increases nearly instantaneously, then exponentially decreases to zero. This signal goes to a summer (No. 4), overriding the signal from the proportional-plus-reset controller.

Thus, for slow upsets, the valve will be opened slowly by the action of the proportional-plus-reset controller; for fast upsets, No. 5 will quickly open the valve, overriding the slow response of the controller.

The output from No. 5 decreases exponentially, allowing the proportional-plus-reset element to take over.

Thus, the operating point is restored to the surge control line as in any other anti-surge closed loop. After the output of No. 5 decreases to zero, No. 6 automatically resets the override system.

Field tests have shown that, if the surge control line is set far enough from the surge limit line, then the combination of the proportional-plus-reset response with the relay override signal is sufficient to prevent surge.

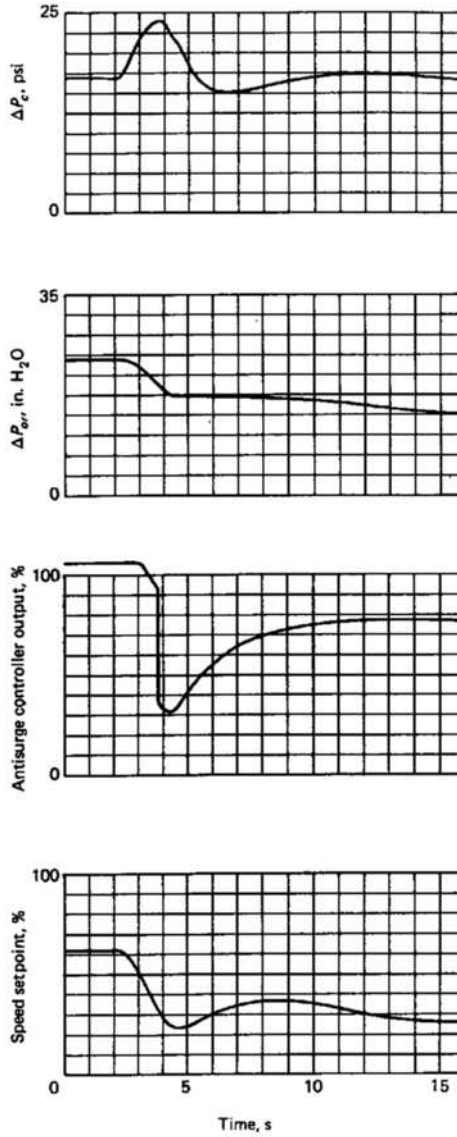
However, if the surge control line is set too close to the limit line, the operating point of the compressor can cross the limit line before the override signal appears: As a result, surge may begin, causing an almost instantaneous flow decrease. Under such circumstances, the deviation of the compressor's operating point from the surge control line increases.

Here, it is necessary to move the surge control line to the right—the task of No. 7. This element, like No. 5, subtracts a fixed amount (d_2) from the signal from No. 2 and compares the sum with $\Delta P_{or,s}$. When the sum is greater than $\Delta P_{or,s}$, a preselected bias, b_2 , is added:

$$k_1 \Delta P_c + b_1 + b_2 = \Delta P_{or,s} \quad (56)$$

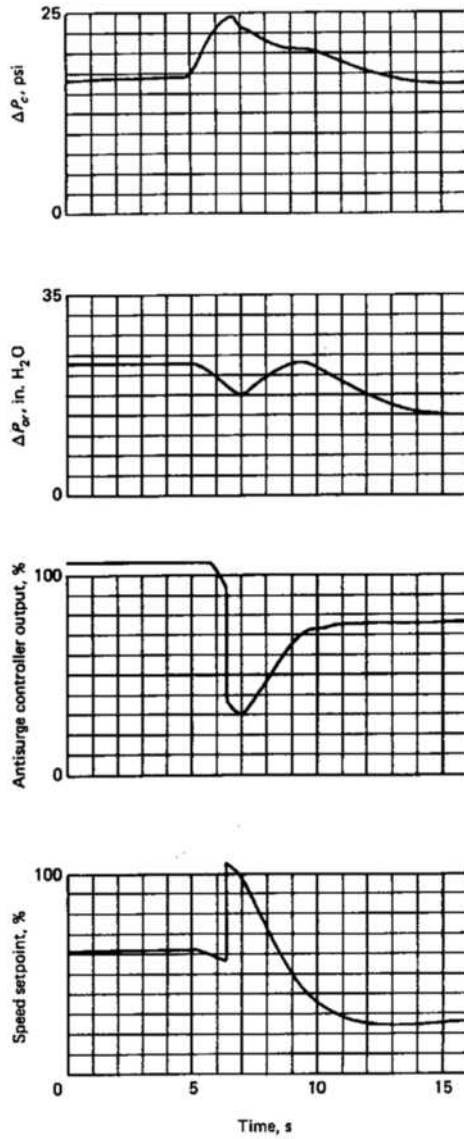
Bias b_2 moves the surge control line to the right. These three elements (3, 5, and 7) can stop surge during the first cycle and keep the operating point at a safe distance from the surge limit line, even if gas consumption decreases to zero. This system protects the compressor against changing operating conditions, not just at design conditions. The reset pushbutton (No. 8) restores the surge control line to its original position.

Test results on this system are shown in Figure 3.21. Here, rather than using a butterfly valve, the compressor was controlled by changing its speed. The speed set point is a record of how the process controller changes the set point of the speed governor. The action of the anti-surge system is seen in the graph of the anti-surge controller output. The horizontal part corresponds to the anti-surge valves being closed; the negatively sloped line corresponds to the opening of the valve by the proportional-plus-reset controller (No. 3 in Figure 3.20); and the vertical drop corresponds to the action of the override controller (No. 5 in Figure 3.20).



Isolated control. Antisurge valve is opened as speed setpoint is lowered

Figure 3.21. Anti surge control effects



Integrated system. Compare output of antisurge controller and speed setpoint with that in Fig. 8

Figure 3.22. Anti surge control effects

INTERACTION WITH PROCESS CONTROL

A process control system is shown in Figure 3.19a. This consists of a pressure transmitter, a pressure controller, and a butterfly valve with an actuator.

By closing the butterfly valve, the process controller forces the compressor's operating point to move toward the surge. If the point crosses the surge control line, the anti-surge controller will open the anti-surge valve.

With both control systems, the deviation in the compressor's operating point from the surge control line depends on the speed of response of the anti-surge controller and of the process controller, and on the dead times of the anti-surge valve and its actuator.

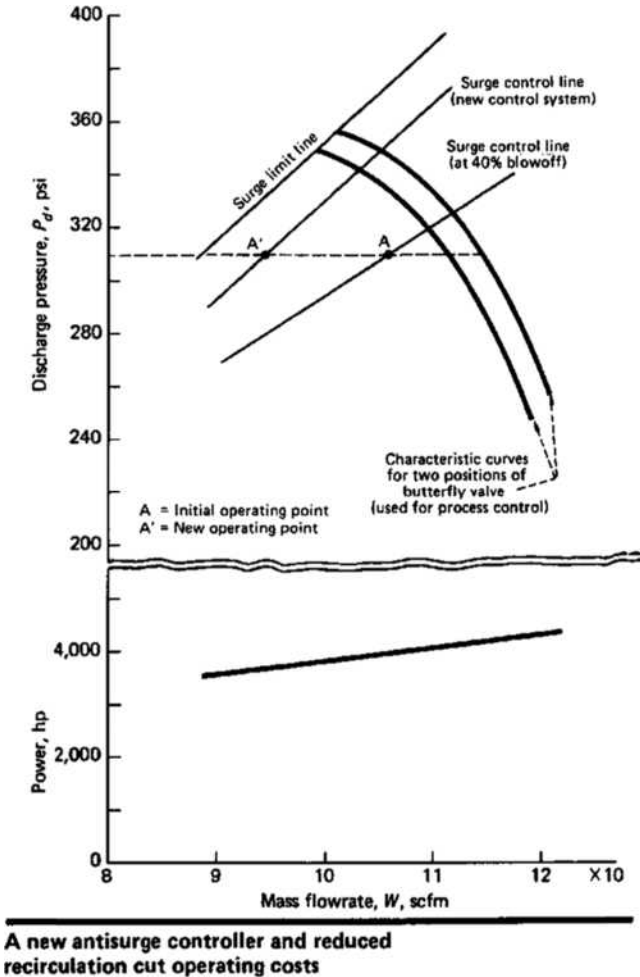


Figure 3.23. Anti surge control effects

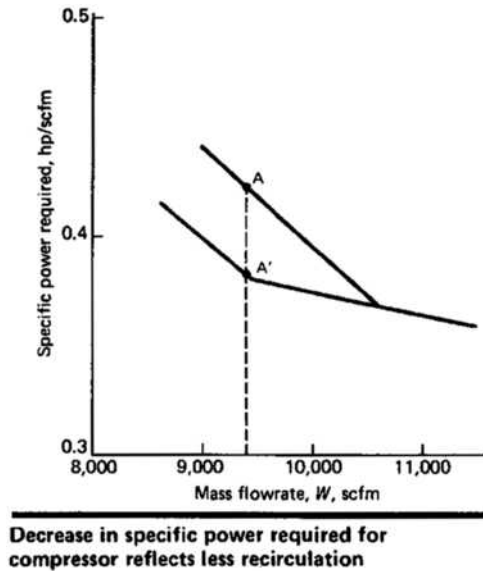


Figure 3.24. Effects of decrease in specific power levels

If the process controller is faster than the anti-surge controller, then this difference may interfere with surge control. The dead time of the anti-surge valve can further add to this problem. Until the output of the anti-surge controller overcomes the dead time, the process controller may push the compressor's operating point over the surge limit line.

This interaction between an isolated surge control system and the process controller occurs not only with the control of discharge pressure but also with the control of suction pressure and flow.

INTEGRATED SYSTEM

System performance can be improved by changing from an isolated anti-surge loop to an integrated process control and protective system (Figure 3.19b).

In this system, the butterfly valve is controlled by the output of the summing device. This applies equally to a compressor that is under variable speed control. The output of this device depends on the output of the anti-surge controller as well as on that of the process controller. The operating point will, after reaching the surge control line, follow this line rather than the line of constant pressure.

With integrated process control and anti-surge protection, the effect of the dead time of the anti-surge valve will be minimized, since two devices

are used to protect the compressor from surge—the anti-surge valve and the butterfly valve. Test results of the integrated system are shown in Figure 3.22, with a variable-speed compressor.

SETTING AND TESTING EQUIPMENT

The surge limit line can be located precisely, since the control system stops surge during its first cycle.

To calibrate the surge limit line, the slope of the surge control line, k_1 , is set at zero to avoid the response by the proportional-plus-reset controller. The distance between the surge limit and surge control lines, b_1 , is set at 15 percent of the flow rate, and b_2 is set at 75 percent. This is because the control line is vertical and a value of $b_1 = 15$ percent will induce surge. The distance between the surge control line and the control line of the override anti-surge relay element, d_1 , is set at zero, so that override will start immediately. The level of the override signal opening the anti-surge valve must be set at 100 percent; the time of its exponential decrease must be not less than 3 minutes. This ensures that the valve is kept open.

If the discharge line of the compressor is closed completely, surge will be induced. At the beginning of the first surge cycle, the override anti-surge relay element opens the anti-surge valve at maximum speed. At the same time, the comparator (No. 7; Figure 3.20) moves the surge control line to the right, making the distance between the surge control and surge limit lines $b_1 + b_2$ equal to 90 percent, which is safely out of the surge zone. This will keep the anti-surge valve open, and surge will be stopped. The outputs of the transmitters, ΔP_c and $\Delta P_{or,s}$, will be obtained at the moment surge occurs.

The ratio $\Delta P_{or,s}/\Delta P_c$ yields k_1 . If the control line is not a straight line, this test can be repeated to generate different values of k .

Setting d_1 depends on the speed of response of the proportional-plus-integral part of the anti-surge controller. If the discharge line is closed slowly, the override anti-surge relay should not operate. Experience indicates that d_1 may be between 2 percent and 5 percent of $\Delta P_{or,s}$.

The distance b_1 between the surge limit and the surge control line is selected so that the severest disturbances (e.g., the fastest possible complete closing of the discharge line) will not cause surge. This distance can be narrow, because the improved anti-surge controller operates at such a high speed. In most cases, b_1 is set between 5 percent and 10 percent of $\Delta P_{or,s}$. Careful selection of k_1 and b_1 will reduce the energy expense for any compressor that operates with recirculation or blowoff.

EXAMPLE OF ENERGY SAVINGS

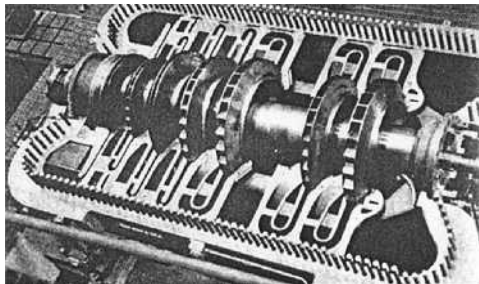
A 4,000 hp air compressor was being operated at constant 40 percent blowoff (see Figure 3.23). The specific power required by this machine is shown in Figure 3.24. The flow rate was 9,400 scfm at 310 psi.

The unit had a pneumatic surge control system that was supplied by the manufacturer. In addition to constant blowoff, damage was frequent—the machine had to be rebuilt twice during the year. The control system described here was installed at a cost of about \$15,000. About half the cost was for instruments, the other half for installation. After installation, the anti-surge valve was closed and recirculation was kept to a minimum. A valve booster was not needed.

Assuming operation on a yearly basis, the savings in energy costs are calculated, with electricity costs of \$0.024/kWh:

$$\begin{aligned}
 \text{Savings} &= 0.746 \frac{\text{kW}}{\text{hp}} \times 8,760 \frac{\text{h}}{\text{yr}} \\
 &\quad \times (0.4225 - 0.3825) \frac{\text{hp}}{\text{scfm}} \times \frac{\$0.024}{\text{kWh}} \times 9,400 \text{scfm} \\
 &= \$59,000 / \text{yr}
 \end{aligned}$$

CAN YOU RERATE YOUR CENTRIFUGAL COMPRESSOR?



Do you have a centrifugal compressor that you would like to use above its capacity or pressure rating? Here is how to determine if it is possible.

Compressor users often want to obtain increased production, which can take the form of increased capacity, or increased pressure rise, or both.

One way to obtain an increase in production lies in the purchase of new equipment; however, one can often rerate (i.e., rebuild) an existing

compressor to provide the required increase. Even if the compressor requires a new rotor and all internal stationary hardware, one can usually still salvage the compressor casing.

Most often, compressors are rerated to obtain greater throughput, with only a minor change in pressure levels. Under these conditions, one can probably reuse a significant amount of existing hardware.

One problem always appears when considering a rerate. No way exists to change the casing size or the bearing centerline distance. One cannot change the nozzle locations, and only a limited number of impellers will fit in any section of the compressor.

When he designs new equipment, the manufacturer starts with a “clean sheet of paper” and can do almost anything to satisfy the requirements of the process. However, the manufacturer must forego this luxury when rebuilding an existing unit.

When considering a rerate, the project engineer must first study the process and determine the flow required to handle the scheduled production increase. What pressure levels will he require of the compressor? How will the compressor’s operation affect the overall process? He must determine the answers to these and other questions before considering the compressor for possible rerate.

A frustrating thing can happen to a project engineer. He designs the process and determines the new required capacity and associated pressure levels at which he expects his compressor to operate, only to learn from the manufacturer that the compressor cannot meet intended requirements. He then heads back to the drawing board and attempts to answer the pertinent questions until he arrives at a set of numbers compatible with both the compressor and the process.

The project engineer could often save a great deal of time if he understood the rerate capabilities of his compressor. If he could figure out for himself what his compressor could deliver, he could determine the probability of reaching his scheduled production increase, without communicating with the manufacturer, until he derived a feasible set of numbers. He might save days, or even months, depending on the extent of the process change and its relation relative to the compressor casing. He would know better than to try to put “10 pounds of coal into a 5-pound bag.”

Actually, the engineer need possess only a minimal amount of compressor knowledge and spend a minimal amount of calculation time in order to determine the rerate feasibility of a compressor. He must consider the following:

1. Capacity: Will the size of the nozzle accept the projected increase in flow?

2. Horsepower: Will the motor, motor gear, turbine, or turbine gear handle the increased horsepower?
3. Pressure: Can the casing accommodate the intended pressure levels? Can the compressor aerodynamically meet these levels?
4. Speed: Will the compressor handle the required speed within the API (American Petroleum Institute) critical-speed limitations?

We will now review these four considerations, presenting them, as above, in the order of their importance.

RERATE FEASIBILITY: CAPACITY

The most important consideration for capacity lies in nozzle sizes. One must determine whether the nozzle size will pass the required volume flow with a reasonable pressure drop. An existing compressor possesses fixed nozzle sizes—the geometric size of the nozzle sets the maximum feasible volume flow possible. Hence, we can reduce the analysis to the consideration of inlet velocities.

Although inlet-velocity limits vary with conditions, a good rule of thumb sets the limit at a maximum of 140 ft/s for air and lighter gases.

Owing to the inverse proportionality between the inlet-velocity limit and the square root of the molecular weight of the gas, one should generally limit heavier hydrocarbons such as propane (mol. wt. = 44.06) to approximately 100 ft/s.

Equation (57) determines the actual inlet velocity:

$$V_a = 3.06Q / D_2 \quad (57)$$

Since velocity limitations set the maximum capacity, the maximum velocity of 140 ft/s can limit one when rerating a compressor. With geometrically sound inlet piping, the engineer may find higher inlet velocities possible.

Manufacturers often furnish compressors with inlet nozzles other than the main inlet. They design some compressors with side-load nozzles, some with interstage cooling nozzles, and some with a combination of both. Use the above inlet-velocity criteria for all inlet nozzles, and check all inlet nozzles for gas velocities.

Once the design engineer decides that the compressor can pass the desired capacity, he can investigate the effect of the increase on the horsepower required.

RERATE FEASIBILITY: HORSEPOWER

The project engineer must include power requirements as the second major consideration in a rerate-feasibility analysis. This holds especially true in the case of motor-driven compressors, since customers usually do not purchase oversized motors along with the original equipment. However, one can usually rebuild gears and turbines to provide greater power capacity. Motor drives may require buying a new motor, which may lead, in some cases, to foundation problems.

The compressor will require power approximately proportional to the increase in weight flow desired. This means that an increase in weight flow of 20 percent will dictate an increase in horsepower of at least 20 percent (or 1.2 times the original power). Moreover, the manufacturer will usually suggest at least an additional 10 percent horsepower availability to take care of overload should this occur. Therefore, a 20 percent weight-flow increase will result in an increase of approximately 32 percent in power requirements ($1.2 + 10\%(1.2) = 1.32$), should the project engineer wish to design a bit conservatively (see Figure 3.25). Note also that in the case of a motor-gear drive, general practice requires the allowance of an additional 2 percent horsepower for gear losses.

If the driver cannot deliver the required horsepower to the compressor, the engineer must either buy a new driver or relax the scheduled production increase until he can reduce the required weight flow to within the capabilities of the driver. In cases where critical power requirements prevail, driver capabilities formulate the starting point of a rerate-feasibility analysis.

RERATE FEASIBILITY: PRESSURE

The project engineer must next consider the hydrostatic test pressure. The compressor maker, at the time of manufacture, hydrostatically tests the casing to 150 percent of the maximum expected operating pressure. Compressor aerodynamics may set the maximum operating pressure, or the process may set it by the existence of a relief valve in the system.

If the expected pressure levels of the new process exceed the nameplate maximum pressure, one should make a new hydrostatic test. **CAUTION:** Sometimes, in a compressor made up of two or more sections (as with inter-stage cooling, side loads, or extractions), the test engineer will test the compressor sectionally, with the inlet section at a lower pressure than the outlet section. In such a case, the project engineer must review both sections for the possibility of exceeding the maximum, sectional, safe-operating pressure.

He must then investigate the compressor's aerodynamic ability to deliver the required head. He can more easily do so in terms of polytropic head—the amount of work done by the compressor (which differs from the input work by the polytropic efficiency). Equation (58) relates the polytropic head to the desired pressure ratio:

$$H_P = ZRT[n / (n - 1)][r_p^{(n-1)/n} - 1] \quad (58)$$

Although the polytropic efficiency can vary from 68 percent on small impellers to 83 percent on ideal impellers, for a feasibility analysis one can assume a polytropic efficiency of 70 percent. If the engineer analyzes the role of the polytropic efficiency in Eq. (58), he will find that the error in the “ $n/(n - 1)$ ” multiplier tends to counterbalance the error in the “ $(n - 1)/n$ ” exponent, thus considerably reducing error in the calculated polytropic head.

Equation (58) can give both the original polytropic head and the rerate polytropic head requirement. With the original rated speed known, the “Fan Law,” Eq. (59), determines the approximate rerate speed:

$$N_{\text{rerate}} = N_{\text{original}} (H_{P_{\text{rerate}}} / H_{P_{\text{original}}})^{1/2} \quad (59)$$

The speed calculated from Eq. (59) may turn out too high for safe operation (see the considerations of rotational speed discussed in the next section). However, the manufacturer can sometimes obtain increased polytropic head by adding an impeller, or replacing the existing impellers with ones of lower backward lean. The manufacturer will ultimately have to determine the feasibility of obtaining the required head.

The project engineer can make a quick estimate of the maximum head capability of his compressor. If the compressor contains a “blank” stage (stage space with no impeller), Eq. (60) will show the approximate maximum head capability with no change in speed:

$$HP_{\text{max}} = HP_{\text{original}} [(a + b) / a] \quad (60)$$

One can then apply the “Fan Law” along with Eq. (60) to determine the approximate polytropic head capability of the compressor. This procedure should yield a polytropic head within 10 percent of the maximum compressor capability.

The above procedure for estimating the required polytropic head, although presented for straight through units, will also work for units with side loads, extractions, or interstage cooling, provided one uses the sum of the head requirements for each section as the total head.

Determine the head for Section 1, based on the operating parameters of Section 1 (i.e., the inlet temperature, inlet pressure, discharge pressure, etc.); the head for Section 2, based on the operating parameters for Section 2; and so on. The sum of all sectional head requirements yields the total head requirement for the rerate. One can determine the total head requirement for the original rated point in a similar manner and derive the approximate rerate speed from Eq. (59) as before.

One important effect of an increase in head requirement remains to be analyzed, that is, the head effect on the power requirement. Horsepower is directly proportional to polytropic head:

$$GHP = \frac{WHp}{33,000\eta_p} \quad (61)$$

As can be seen, an increase of 20 percent in the required polytropic head will raise the power requirement 20 percent if the efficiency remains constant. For example, if an increase of 20 percent in the weight flow is coupled with one of 20 percent in the polytropic head, the power requirement will rise by 44 percent ($1.20 \times 1.20 = 1.44$). To this figure, an additional 10 percent horsepower should be added for overload. A 2 percent gear loss should also be included, where appropriate.

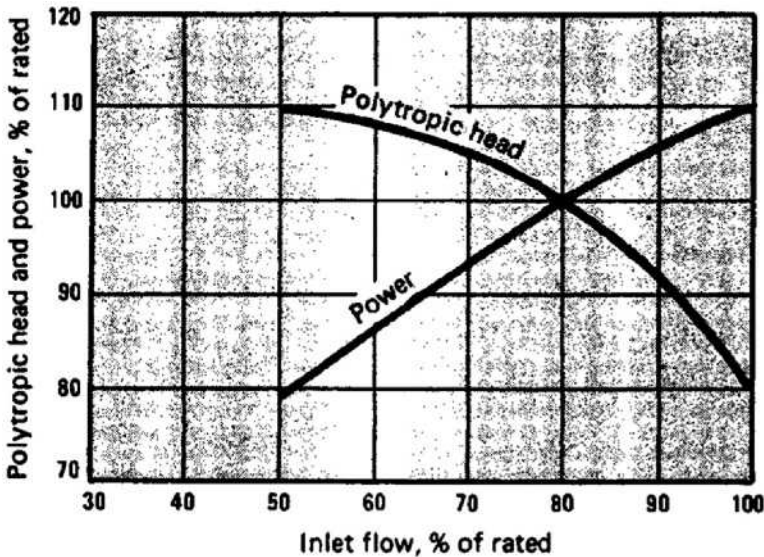


Figure 3.25. A centrifugal compressor: typical performance

RERATE FEASIBILITY: SPEED

The final consideration is that of rotational speed. Two major criteria must be satisfied:

1. The rotational speed must be slow enough to not overstress the impellers.
2. The rotational speed should fall within certain limits of the compressor's first and second critical speeds.

The first criterion above is best viewed from the standpoint of mechanical tip-speed limitations. The mechanical tip speed can be determined by:

$$u = \pi dN/720 \quad (62)$$

Mechanical tip-speed limits vary with manufacture, size, impeller material, and type of construction. A typical limit on mechanical tip speed might be 900 ft/s. When using the "Fan Law" of Eq. (59), the rotational speed that results in a tip speed of 900 ft/s for the largest impeller should be considered the upper limit (providing that this speed falls within API critical-speed limitations).

The project engineer will not know what effect the rebuild will have on the compressor critical speeds when conducting the feasibility analysis, except possibly through experience with such effects. Since the bearing centerline distance is fixed and most rebuilds can be accomplished without a change of bearings, it can generally be assumed that the critical speeds will not be affected (at least, as a first approximation).

SAMPLE PROBLEM

For the purpose of discussion, let us consider a "straight through" centrifugal compressor on a dry-air process, having the following nameplate data: inlet capacity = 11,000 ICFM; inlet temperature = 90°F; rated inlet pressure = 14.5 psia; rated discharge pressure = 55 psia; rated power input = 1,700 hp; rated speed = 8,100 rpm; maximum continuous speed = 8,500 rpm; first critical speed = 4,800 rpm; rated molecular weight = 28.97; $K = C_n/C_v = 1.4$; maximum discharge pressure = 65 psia.

An investigation of the data files on the compressor reveals that the second critical speed is equal to 10,800 rpm and that the smallest wheel

diameter is 22 in. Also the cross-sectional drawing of the compressor indicates that the inlet nozzle diameter is 20 in.

The desired rerate is an increase in the inlet capacity to 12,300 ICFM and an increase in the discharge pressure to 60 psia, with all other inlet conditions unchanged:

1. Compute the inlet velocity, based on the new inlet-volume flow using Eq. (57):

$$V_a = 3.06 \frac{Q}{D^2} = 3.06 \left[\frac{12,300}{(20)^2} \right] = 94 \text{ ft/s}$$

Since this is an acceptable inlet velocity, the proposed capacity is feasible.

2. Since the rated inlet conditions have not changed, the increase in weight flow will be proportional to the increase in volume flow, and therefore, the power requirement due to the change in volume flow will increase by the same proportion:

$$\begin{aligned} \frac{GHP_{\text{rerate}}}{GHP_{\text{original}}} &= \frac{W_{\text{rerate}}}{W_{\text{original}}} \frac{Q_{\text{rerate}}}{Q_{\text{original}}} = \frac{12,300}{11,000} = 1.12 \\ GHP_{\text{rerate}} &= 1.12 GHP_{\text{original}} = 1.12(1,700) = 1,910 \text{ hp} \end{aligned}$$

Note that, up to this point, the driver will have to be capable of:

$$(1.1)(1,910) = 2,100 \text{ hp}$$

plus 2 percent excess horsepower if a gear is involved.

3. Since the nameplate maximum discharge pressure is 65 psia, the compressor will not have to be hydrostatically retested, providing that the process will not allow the value of 65 psia to be exceeded.
4. The approximate polytropic head can now be calculated for both the original and the rerate condition from Eq. (58):

Original:

$$\begin{aligned} n / (n - 1) &= [K / (K - 1)] \eta_p = [(1.4 / 1.4 - 1)](0.76) = 2.66 \\ H_p &= ZRT(n / n - 1)[(P_2 / P_1)^{(n-1)/n} - 1] \\ &= (1.0)(1,545 / 28.97)(550)(2.66)[(55 / 14.5)^{1/2.66} - 1] \\ &= 50,700 \end{aligned}$$

Rerate:

$$\begin{aligned} H_p &= (1.0)(1,545 / 28.97)(550)(2.66)[(60 / 14.5)^{1/2.66} - 1] \\ &= 55,000 \end{aligned}$$

The new required speed can be determined from the “Fan Law,” Eq. (59):

$$\begin{aligned} N_{\text{rerate}} &= N_{\text{original}} \sqrt{H_{p_{\text{rerate}}} / H_{p_{\text{original}}}} \\ &= 8,100 \sqrt{55,000 / 50,700} = 8,440 \text{ rpm} \end{aligned}$$

The smallest wheel diameter is 22 in.; therefore, from Eq. (62):

$$u = \frac{\pi d N}{720} = \frac{(\pi)(22)(8,440)}{720} = 810 \text{ ft/s}$$

The new rotational speed results in a satisfactory mechanical tip speed. API states that the second critical speed must be 20 percent above the highest operating speed. Assuming that the new rerate speed is the highest for the new process, the second critical speed must be at least:

$$(1.2)(8,440) = 10,130 \text{ rpm}$$

The second critical speed (10,800 rpm) is higher than that required and therefore the rotational speed is feasible.

5. The total increase in gas horsepower can now be determined. The new horsepower will be proportional to the increase in the polytropic head and weight flow (in this case, volume flow):

$$\begin{aligned} \frac{GHP_{\text{rerate}}}{GHP_{\text{original}}} &= \left(\frac{H_{p_{\text{rerate}}}}{H_{p_{\text{original}}}} \right) \left(\frac{Q_{\text{rerate}}}{Q_{\text{original}}} \right) \\ &= \left(\frac{55,000}{50,700} \right) \left(\frac{12,300}{11,030} \right) = 1.21 \\ GHP_{\text{rerate}} &= (1.21)(1,700) = 2,060 \text{ hp} \end{aligned}$$

The driver must therefore be capable of:

$$(1.1)(2,060) = 2,270 \text{ hp}$$

plus 2 percent if a gear is involved.

Since the inlet velocity, maximum operating pressure, and required rotational speed are within satisfactory limits, the rerate is feasible.

Nomenclature

<i>a</i>	Number of impellers in original rotor
<i>b</i>	Number of “blank” stages in original configuration
<i>d</i>	Tip dia. of largest impeller, in.
<i>D</i>	Inside dia. of nozzle flange, in.
<i>GHP</i>	Gas horsepower
<i>Hp</i>	Polytropic head, ft-lb/lb _m
<i>K</i>	Ratio of specific heats, dimensionless
<i>N</i>	Rotational speed, rpm
<i>n/(n - 1)</i>	$[K/(K-1)]\eta_P$
<i>Q</i>	Flange volume flow, ft ³ /min, based on rated inlet conditions (i.e., inlet pressure, temperature, compressibility factor and molecular wt)
<i>r_P</i>	Pressure ratio, $P_{discharge}/P_{inlet}$
<i>R</i>	Gas constant, 1,545/mol. wt.
<i>T</i>	Inlet temperature, °R
<i>u</i>	Mechanical tip speed, ft/s
<i>V_a</i>	Actual inlet velocity, ft/s
<i>W</i>	Weight flow, lb _m /min
<i>Z</i>	Average compressibility factor
<i>η_P</i>	Polytropic efficiency

SUMMARY

We have analyzed the four major considerations of a rerate-feasibility analysis: (1) capacity, (2) horsepower, (3) pressure, and (4) speed. The nozzles must be large enough to pass the required flow—or, to state this another way, the maximum capacity will be limited by the size of the inlet nozzle. The driver must be capable of delivering the required power. The casing must be capable of handling the pressure levels, both mechanically and aerodynamically. The required speed must be within certain mechanical limitations and should meet the API critical-speed requirements. Once the project engineer is satisfied that the above criteria can be met, a request for a quotation can be forwarded to the manufacturer with greater confidence.

EASY WAY TO GET COMPRESSION TEMPERATURES

An engineer or operator can quickly find the temperature of gas discharging from a compressor with this nomograph. All he needs to know is the suction pressure and temperature, the discharge pressure, and the ratio of specific heats for the gas.

The basic equation for compressor discharge temperature is:

$$T_2 = T_1 (P_2 / P_1)^{(n-1)/n}$$

where: T_2 = absolute discharge temperature, °F + 460

T_1 = absolute suction temperature, °F + 460

p_2 = discharge pressure, psia.

P_1 = suction pressure, psia.

n = ratio of specific heat at constant pressure to specific heat at constant volume

p_2/p_1 = compression ratio

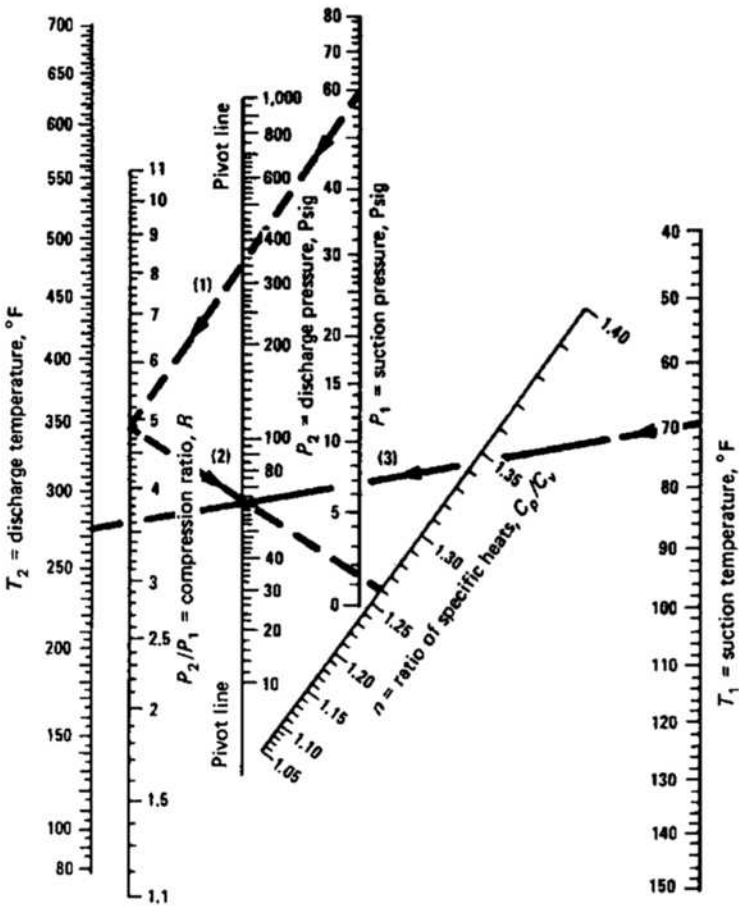
The ratio of specific heats, n , can be calculated, read from tables, or estimated from the chart.

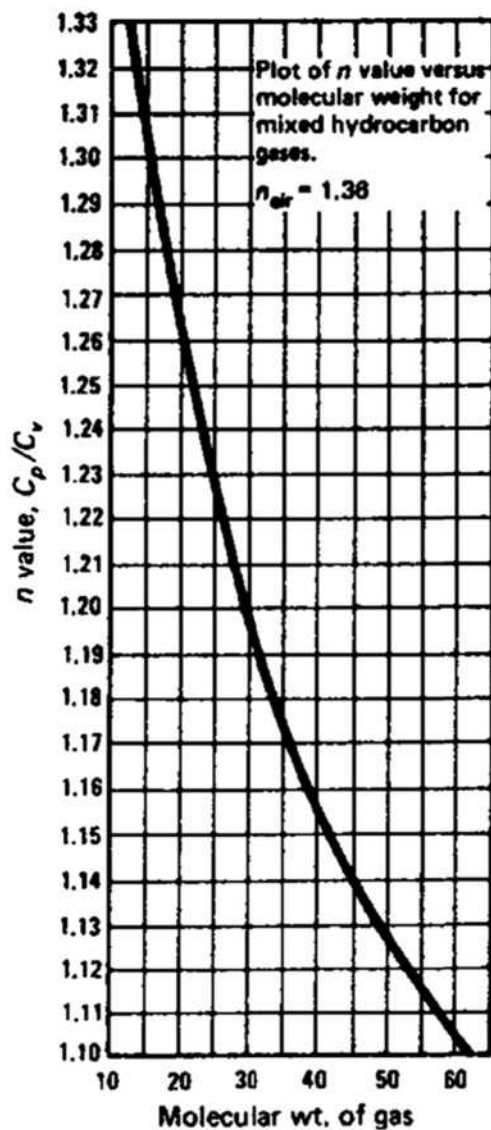
If the compressor suction or discharge pressure is beyond the range in the nomograph, the nomograph can still be used by calculating the compression ratio and taking that as the starting point. It should be noted that the absolute values of temperature and pressure in the equation have been converted to °F and psig.

Example:

A hydrocarbon gas whose molecular weight has been given as 21 is fed to a compressor at 60 psig and 70°F, then compressed to 350 psig. What is its temperature leaving the compressor? Read $n = 1.26$ on the plot of molecular weight versus n . Connect 60 psig on the P_1 scale with 350 on the P_2 scale (note that these pressures are psig on the scale and not psia) and read the compression ratio, p_2/p_1 , as 4.88. Connect this with 1.26 on the n scale, note the intersection on the pivot line, and align this intersection with 70°F on the T_1 scale to read 275°F on the T_2 scale.

Nomograph Determines Gas Compression Temperature





CHAPTER 4

LUBRICATION AND COMPRESSORS



LUBRICATING AIR COMPRESSORS

Choosing the proper lubricant for plant air compressors pays off in terms of long life and trouble-free operation.

The lubrication system in a compressor is a critical necessity in keeping the machine running. It does this by reducing friction, transferring heat to the cooling system, sealing against air leakage, and hushing away dirt and debris.

The lubrication systems of present-day air compressors must be able to provide these vital functions in atmospheres that are usually polluted with dirt particles as well as with corrosive fumes that pass right through the inlet air filter and enter the compression chamber.

In addition, modern air compressors are much smaller, and run at speeds undreamed of a few decades ago, making lubrication requirements all the more critical.

The packaged, integral-gear, centrifugal, plant air compressor has become important over the past 15 years. Here, the lubrication system has to deal with individual pinion gears that run at speeds of 30,000 to 50,000 rpm.

A more recent compressor is the oil-flooded rotary-screw machine. It has brought forth still a different set of requirements, because the oil and air pass through the compression cycle together, with the oil picking up all the dirt and contaminants that come through the inlet air filter.

Suppliers of lubricants have had to introduce more sophisticated products in recent years to meet the requirements of this new generation of air compressors, operating under increasingly adverse environmental conditions. The oil industry has not only developed new lubricants to meet these requirements but has actually been able to extend the service life between oil changes in most cases.

LUBRICATION PROBLEMS

Many factors should be considered when your oil suppliers visit your plant to advise you on the proper lubricant for a particular compressor. Among the significant factors are inlet air temperature, air contamination, condensation, operating temperature, discharge temperature, and pressure.

Inlet air contamination is a major factor in compressor wear, oil oxidation, and discharge-valve deposits. Recent years have seen an amazing increase in the rate of buildup of industrial air contaminants, which are often odorless and invisible. Even on the clearest day, considerable amounts of vapors and fumes come through an inlet air filter in a typical industrial plant. It surprises most people to learn that the human body can tolerate far more air contamination than can a modern air compressor.

Condensation seldom occurs in a compressor during normal operation. However, when the relative humidity of inlet air is high, and the cylinder temperature during the suction stroke is below that of the inlet air, condensation will occur. For instance, to prevent condensation, the cylinder walls in the first stage of a two-stage unit have to be kept at about 100°F when the inlet air is at 80°F. The inexperienced operator who runs too much cold water through the cooling jackets may cause water to condense within the cylinders and wash away the oil film. This promotes the formation of rust, which in turn causes abrasion during compressor operation.

High operating temperatures reduce the viscosity of an oil, thereby reducing the oil-film thickness. This film reduction creates a marginal condition where metal-to-metal contact can occur. This is why the viscosity index of the oil purchased is so important. Prolonged operation at temperatures in the range of 140° to 160°F also increases the oxidation rate of the lube oil. Oxidation can change a thin oil to a thick, carbonaceous, sticky mass in a relatively short time. Hence, the oil-change period must be strictly maintained.

Air discharge temperature directly affects the quantity of deposits on the discharge valves and downstream piping. High discharge temperatures cause rapid oil oxidation; the oxidized residue plates out on the valves. This buildup eventually interferes with normal valve motion, causing valve leakage. Then, preheated or recompressed air increases the discharge temperature, further speeding oxidation and promoting more deposits. This cycle can lead to component failure and, at the extreme, to incandescent hot spots.

Another warning: If someone claims he has the ideal oil for all reciprocating compressors, don't believe him. A single-stage reciprocating machine has a theoretical discharge temperature of 400° to 450°F, while a two-stage unit would discharge at about 250° to 300°F. Some reciprocating units are splash lubricated; others are force-feed lubricated. Some are air cooled; others water cooled. Some use one grade of oil for all moving parts, while others use different oils for cylinders and running gear. There are many variables.

ROTARY-SCREW COMPRESSORS

Oil-flooded rotary-screw compressors have different requirements from reciprocating ones. Because air and moisture sucked in at the inlet carry through with the oil, and are separated after the compression cycle, the oil must have an excellent water-separation characteristic. Since the oil is continually cooled in a screw compressor, the internal temperatures can be kept quite low. However, this is dangerous; as has been said, condensation presents a severe hazard. Thus, most manufacturers recommend that discharge air temperature be maintained between 170°F and 200°F. This concept of deliberately increasing the discharge temperature of a screw machine by

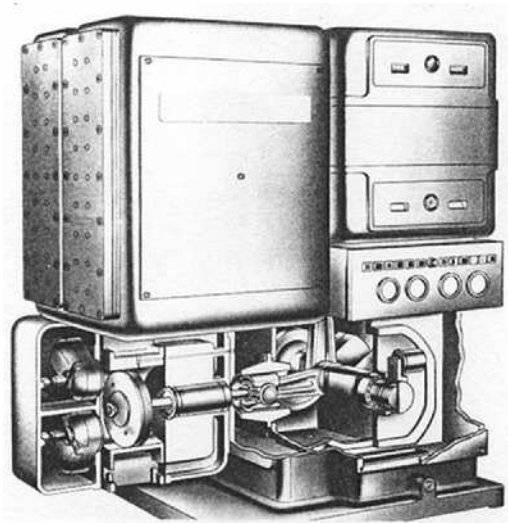
reducing oil cooling is difficult for some operators to accept, but it must be insisted upon to prevent condensation and consequent bearing damage.

TYPES OF LUBRICATING OILS

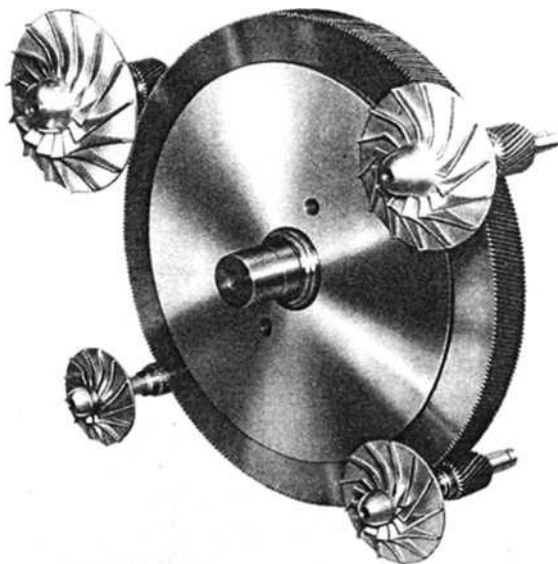
Most of the premium-grade lubricants advertised to the public are the so-called Pennsylvania-crude-based oils, which are excellent for automotive service. They come from a large natural pool under parts of Pennsylvania, Ohio, and West Virginia, and are unique in the United States. These are paraffin-based oils, as opposed to the naphthenic-based oils produced elsewhere in the country. However, paraffin-based lubricants form a hard, varnish-like residue when used in compression cylinders; naphthenic oils form a lighter, huffy, carbon residue that is more easily cleaned off.

Discharge-valve deposits can be the principal problem in some reciprocating compressor installations. When this is the case, it is essential to use a naphthenic oil and minimize discharge temperature. If the air compressor must also be operated under high humidity, high ambient temperatures, and intermittent-duty conditions, then condensation and resulting sticky valves or rings may occur. A compounded or oxidation-inhibited naphthenic-based oil is best suited for condensation conditions where valve deposits are a chief lubrication problem.

Paraffinic oils can be used in air compressors that do not have to contend with valve deposits. They are more resistant to thinning out as temperature increases than are straight naphthenic oils. That is, they have a better viscosity index.



Packaged compressor using a pressurized lube system.



Bull gear and pinions of centrifugal compressor.

Bearing protection is one of the primary functions of a frame or crankcase lubricant. The critical period for any bearing lubricant is when bearing failure begins. The bearing surfaces during critical periods become irregular, and high-localized pressures occur. Naphthenic oils under high pressure become viscous. This provides an effective medium for carrying the bearing metal until its surface is smooth and the load is evenly distributed.

Synthetic lubricants were developed to overcome the two most frequent complaints by users of petroleum oils—carbonaceous deposits that necessitate frequent valve cleaning and danger of fire.

DANGER OF FIRE

The so-called fire triangle—oxygen, fuel, and an ignition source—exists in any air compressor. Air provides the oxygen, and petroleum oils provide the fuel. Normal discharge temperatures are never high enough to trigger a fire or cause an explosion. The ignition temperature of most air-cylinder lubricating oils is above normal compressor operating temperatures.

However, carbonaceous deposits from lube oils do collect on valves, heads, and discharge piping. As has been stated, this may cause leaking discharge valves, so that hot air bypasses and continues to build up heat until autoignition can occur. This is practically always the result of inadequate maintenance, such as dirty fins on air-cooled compressors, dirty

intercoolers, water jackets clogged with scale, broken or leaking valves, and the like. Therefore, the best oils should be used in a limited amount. Keep the compressor clean. Inspect valves frequently. Remove scale formations in the cooling-water jackets.

Phosphate-ester synthetic lubricants were developed to minimize compressor fires. Synthetics have distinct fire-resistant qualities in that they have an auto-ignition point that is about 50 percent higher than that of most petroleum oils and have excellent film strength.

There are some disadvantages to synthetics: Their viscosity indexes are generally not as high as those of petroleum lubes, and they are more vulnerable to water washing in the presence of condensate. They are much more expensive than petroleum products, and the cylinder feed rate must usually be substantially higher. These synthetics are powerful solvents. They can remove all paint on the inside of air passages, and this carries into the valves, clogs oil lines, fouls intercoolers, and raises general havoc inside the compressor.

These synthetics attack normal gasketing material, and fumes must be directed away from an electric motor, for they can destroy electrical insulation. Never use a phosphate-ester synthetic lubricant without first checking with the compressor manufacturer.

Diester-based lubricants, while not offering quite the same explosion protection as phosphate esters, have many other properties that are advantageous, particularly a long service life between oil changes. Then, there are the so-called super-oils, which are petroleum oils with additives. They provide a somewhat longer service life than standard petroleum oils, at a somewhat lower cost than the synthetics.

WHICH TYPE OF OIL IS BEST?

The next task is selecting the right oil for your needs. Consider first the centrifugal compressor. This probably presents the least demanding lubrication problem. Remember, since the centrifugal air compressor is a nonlubricated machine, there is no oil in the air compression chambers; lubrication is confined to the driving gear.

Here the requirement is for good maintenance of film strength under load, with added inhibitors to prevent rust, sludge formation, and foaming.

Oil-flooded screw compressors, on the other hand, are very sensitive to both the quality and the condition of the lube oil. There are no high-temperature conditions to contend with, but the oil is repeatedly exposed to the air being compressed, plus all the contaminants that pass through the inlet filter. Since a part of the oil that floods the compression

chamber is bypassed to go through the bearings, moisture collecting in the unit can be critical to bearing life.

Since operators are not always alert to the dangers of condensation—contamination or oil degradation in an oil-flooded screw compressor—one manufacturer offers an oil-sampling program and urges its use. Small sample bottles of oil from the compressor are mailed to the compressor manufacturer on a regular basis, and an analysis is immediately returned to the user. This gives him adequate warning and enables an operator to schedule a maintenance check before a bearing failure shuts him down.

Because of the very high temperatures incurred in a piston compressor, sometimes 300° to 400°F, this may present the toughest lubrication requirement. Synthetic lubricants and super-oils have been mentioned. Their selection, compared with a conventional petroleum oil, becomes largely a matter of economic choice; that is, are the long-life advantages worth the extra cost?

SETTING THE OIL FEED RATE

Cylinder lubricators on piston compressors have adjustable feed rates. The oil droplets can easily be counted as they pass through a sight glass. The question most often raised is, how many drops per minute should be used? Attempts have been made to relate drops per minute to cylinder diameter, piston speed, etc. However, it must be noted that such an approach may be dangerous; the correct oil quantity for one machine may be too much or too little for another.

A new compressor should be broken in with a very heavy feed rate (three or four times normal) for the first 500 hours of operation, using a cylinder oil two Society of Automotive Engineers (SAE) grades heavier than normal, until the piston and cylinder have acquired a glaze. This break-in routine should be followed using only petroleum oil, even if there is to be a switch to synthetic cylinder oils for regular operation.

Then the lubricator feed rate should be reduced in small steps, shutting down and inspecting the cylinder bore after a few hours of operation at each step, until the top inside the surface has just a slight oil film. Puddles of excess oil lying in the bottom of the cylinder indicate too heavy a feed rate; dry spots on the top are a signal of too little. An experienced operator knows that there is much more art than science in arriving at just the right lubricator setting.

Lubricating oil is the lifeblood of a compressor. Its selection and use should not be left to chance. Manufacturers' publications, such as *Maintenance of Reciprocating Compressors* and the *Compressed Air & Gas Data* book, as well as operating manuals for the particular compressor at hand, should be studied. Then, call in the expert—your oil supplier.

ABOUT THE AUTHOR

James Watterson has a PhD in mechanical engineering and is currently serving as independent training consultant with JMW, United Arab Emirates, since January 2003, and as part-time senior lecturer/lead trainer—commissioning and process start-up in the oil and gas refineries with ADNOC since February 2011. He trains groups of maintenance, operations, instrumentation, and planning engineers in groups of 8 to 20. This includes public courses and in-house courses. Usually spanning 5 days, these courses are highly interactive and use many types of software for vibration analysis, compressors—turbines. There are also many 3D animated videos for most rotating equipment. Coupled with this, the author also provides self-assessment questions. The feedback from these companies has been very positive, with the author receiving many emails of gratitude from the course participants and from the training departments. Previously he was managing director at JW Management and Marketing Consultancy from September 1985 to October 2000.

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A Simple Guide to Understanding Compressors

James M. Watterson

This straightforward guide to compressors seeks to unveil a lot of myths surrounding compressors. In this book, we will be looking at most types of compressors, including the centrifugal compressors, the air compressors, and of course the most troublesome of all compressors, the reciprocating compressors.

Having a compressor with minimal operating problems does not only depend on the selection of the right type and size for your job. Detailed specifications of all auxiliary equipment and operating conditions, as well as keeping constant vigilance over the engineering and installation is imperative. The Simple Guide will explain in a simple yet definitive manner which compressor type is best used for which job and what it can produce.

James M. Watterson has a PhD in mechanical engineering and is currently serving as independent training consultant with JMW, United Arab Emirates, since January 2003, and as part-time senior lecturer/lead trainer—commissioning and process start-up in the oil and gas refineries with ADNOC since February 2011. He trains groups of maintenance, operations, instrumentation, and planning engineers. Previously he was managing director at JW Management and Marketing Consultancy from September 1985 to October 2000.



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