3

RECIPROCATING COMPRESSOR PERFORMANCE AND MONITORING CONSIDERATIONS

3.1 CAPACITY CONTROL

As already discussed, a reciprocating compressor is a positive displacement device. During normal operation it will take in a quantity of gas from its suction line and compress the gas as required to move it through its discharge line. Unlike centrifugal pumps, the reciprocating compressor cannot self-regulate its capacity against a given discharge pressure; it will simply keep displacing gas until told not to. This would not be a problem if we had an unlimited supply of gas to draw from and an infinite capacity downstream to discharge into; however, in the real world of refineries, chemical plants, and gas transmission lines, we find that we have specific parameters within which to work, and that capacity is a unique quantity at any point in time. Thus, we have a real need to control the capacity of the reciprocating compressor.

We also find that in most instances a reciprocating compressor needs to be unloaded for startup. Simplistically, it would seem that starting a positive displacement machine fully loaded would require a driver with 100% starting torque. However, what we find is that a reciprocating compressor can typically have a 3:1 peak-to-mean torque ratio. This peak torque requirement, coupled with breakaway friction, means that the driver now must have as much as a 350% starting torque capability. Again looking at the real world, we find that motors are designed to have 40 to 60% starting torque capability, thus necessitating an unloaded start. Additionally, we can convince ourselves that it is healthier for the compressor to start and continue running unloaded, to give the compressor time to warm up.

From basic thermodynamics and compression theory we know that there are equations describing the performance of a reciprocating compressor cylinder. The equations relate such

A Practical Guide to Compressor Technology, Second Edition, By Heinz P. Bloch

parameters as compression ratios, cylinder clearances, volumetric efficiency, and thus cylinder flow capacity. From our earlier discussion, we recall that

$$\mathrm{VE} = \eta_v = 100 - C(r^{1/k} - 1) \tag{3.1}$$

We can also draw a pressure–volume (*p*–*V*) diagram that visually gives us a feel for how the cylinder is operating. This was shown in Section 1.2. Referring back to this information allows us to understand how some of the capacity control methods work.

3.1.1 Recycle or Bypass

One of the simplest methods of controlling capacity is to recycle, or bypass, the compressed gas back to the compressor suction. This is physically accomplished by piping from the compressor discharge line through some type of control valve and going back to the compressor suction line. To reduce the flow to process, one simply opens up the bypass line and diverts the excess flow back to the compressor suction. In addition to being simple, this system has the advantage of being infinitely controllable (within the limitation of the size of the bypass line). Many, if not most process-type compressors have some sort of recycle line so that operators can fine-tune the flow to process.

A recycle system does, however, have shortcomings, the greatest being its inefficiency. Consider that we are taking gas at an elevated discharge pressure after having invested considerable horsepower in compressing it to that discharge pressure. Now we are going to expand it back down to the lower suction pressure simply so that we can invest more horsepower by compressing it again. As far as the compressor is concerned, it is always running at 100% load and is consuming 100% horsepower, even though the flow actually delivered to process could be a low percentage, or even zero. Another problem is in the actual design of the bypass line. The greater the percentage of bypass, the larger the piping has to be. In addition, depending on the particular gas characteristics, it may be necessary to include a cooler in the bypass line to dissipate the heat generated in compressing and continually recycling the gas. Also, consider the case of multistage compression where multiple bypass lines may be needed. The cost of installing an extensive bypass line could become prohibitive.

The most practical application for the bypass line is for small degrees of fine capacity control or for limited-duration startup unloading, where a simple loop around the compressor can be opened for a short period of time to relieve the initial compression load.

3.1.2 Suction Throttling

Although not used very widely, suction throttling is another method of controlling the capacity of a reciprocating compressor. The technique is to reduce the suction pressure to the compressor by limiting or throttling the flow into the cylinder. By referring back to our *p*–*V* diagram, we see that if all else is held constant, reducing the suction pressure creates a narrow card, indicating a lower volumetric efficiency and thus less flow. Additionally, the density of the gas will be reduced at this lower pressure, thus helping to reduce the mass flow delivered.

Suction throttling has its limitations. It takes a fairly dramatic reduction in suction pressure to give any sizable reduction in capacity. Additionally, as the suction pressure is reduced and the discharge pressure held constant, the compression ratio is increased. This causes higher discharge temperatures and higher rod loads.

3.1.3 Suction Valve Unloading

Probably the most common method of controlling compressor capacity is via suction valve unloading. The technique here is to physically keep the cylinder from compressing gas by maintaining an open flow path between the cylinder bore and the cylinder suction chamber. The cylinder will take in gas normally; however, instead of completing the normal cycle of compression and discharge, the cylinder will simply pump the gas still at suction pressure back into the suction chamber via this open pathway. There is absolutely no gas discharged to process. Additionally, since no compression is occurring, virtually no horsepower is consumed other than passageway losses.

There are three basic types of suction valve unloading. The first, and oldest, is the *finger-type unloader*, shown in Fig. 3.1. These unloaders consist of a series of small fingers that are housed in the valve crab assembly and actuated via a push rod from an outside actuator. To unload the valve, the fingers are lowered so that they depress the valve-sealing components and thus hold the valve in the open position. The pathway between the cylinder bore and the gas passage is then through these open suction valves. Finger-type unloaders will typically be mounted on each suction valve so that the flow area of the unloaded pathway is maximized. Also, since the fingers are simply holding open the existing suction valves, no special valve design is required. Actuation of finger-type unloaders can be manual using a handwheel and screw or lever arrangement to lower the fingers, or using a small air cylinder on the top of the unloader stem.

One of the biggest problems associated with finger-type unloaders is the potential for damaging the valve-sealing elements with the fingers. If one visualizes the force generated by pneumatically actuated fingers as they are driven down against the valve-sealing components, one can see how this could contribute to premature valve failure.

An alternative to the finger-type unloader is the *plug-type unloader* (Fig. 3.2). Here, instead of acting on the valve-sealing components themselves, we have a passageway bored through the middle of the valve. In normal operation this passageway is sealed with

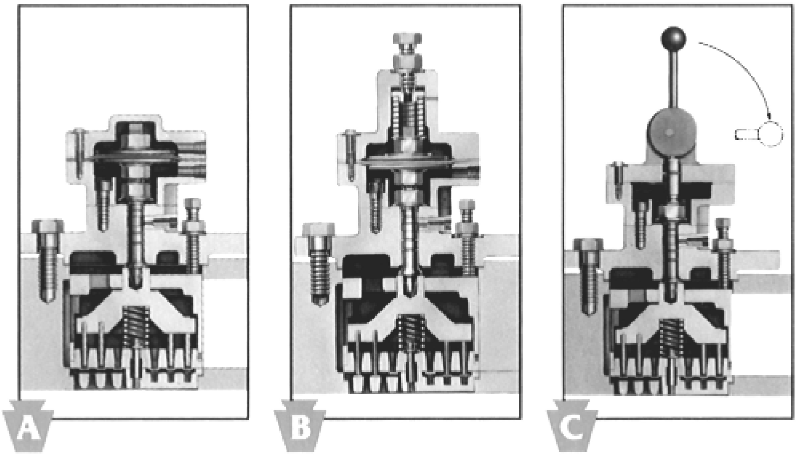


FIGURE 3.1 Finger-type unloaders, pneumatically operated: (A) direct-acting (air-to-unload); (B) reverse-acting or fail-safe (air-to-unload) which automatically unloads the compressor in the event of control air failure; (C) manual operation. *(Cooper Cameron Corporation, Cooper-Bessemer Reciprocating Products Division, Grove City, Pa.)*

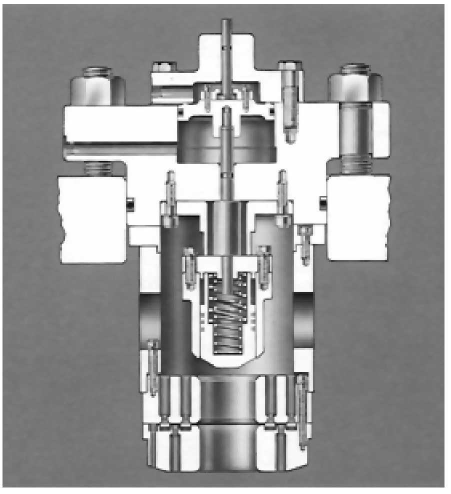


FIGURE 3.2 Outside operated plug-type unloader. Actuating air cannot mix with the gas being compressed. *(Dresser-Rand Company, Painted Post, N.Y.)*

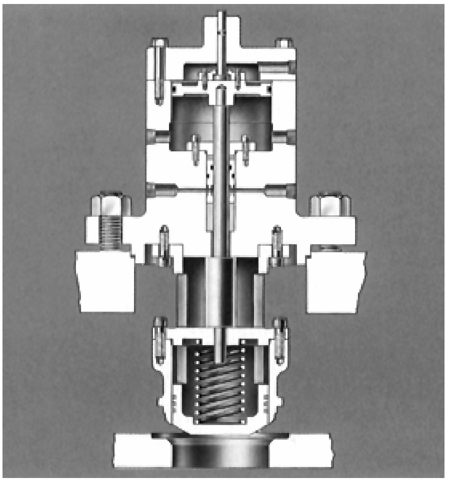


FIGURE 3.3 An outside-operated port-type unloader requires the use of only one unloading device per cylinder end and is typically used for lower-molecular-weight gases. Air cannot mix with the gas being compressed. *(Dresser-Rand Company, Painted Post, N.Y.)*

a plug. To unload the cylinder, we simply remove the plug and allow the gas to flow in and out of this passageway, with no compression taking place.

The plug-type unloader offers a major benefit over the finger-type unloader because it does not work on the valve-sealing components as does the finger-type unloader. However, since there is now a passageway through the center of the valve, we have reduced the normal effective flow area of the valve. In the case of a heavy gas, this could mean higher consumed horsepower because of the reduced valve flow area and increased pressure drop through the valve. Again as with the finger-type unloader, each suction valve would typically require a plug-type unloader.

The third type of unloader is the *port*, or *passage-type unloader* (Fig. 3.3). This type of device also uses a plug to seal the unloader passageway. However, instead of working in a passageway through the valve, the unloader uses a separate port between the cylinder bore and the gas passage. The port is created by removing one suction valve per cylinder end and replacing it with a large plug assembly. The port-type unloader has many advantages. The flow area of the passageway is large, since the plug is the full diameter of a normal valve rather than just a portion of the diameter. Moreover, when opened, the plug will lift from 1 to 2 in. off its seat. With only one unloader needed per cylinder end, the compressor incorporates fewer devices for maintenance purposes. Additionally, since a port unloader does not work on an active valve, it does not need to be removed for regular valve maintenance. Port unloaders are ideal for low-molecular-weight gas applications, where the total number of suction valves is typically reduced to improve pressure drop across the active valves.

Again, all valve unloader types can be manually operated or actuated by a pneumatic cylinder. When pneumatically actuated, these devices can be designed to load or unload upon either application or removal of air pressure. The advantages of pneumatic operation are the ability to control the capacity of the compressor remotely or even to automate this control.

Since suction valve unloaders keep compression from occurring, they control capacity in discrete steps. For example, a double-acting cylinder can be operated 100% loaded, 50% unloaded by unloading one end of the cylinder, or fully unloaded by unloading both ends of the cylinder.

An interesting alternative to these discrete steps of unloading is the stepless capacity control system offered by Hoerbiger compressor controls. This system uses finger-type unloaders that are pneumatically actuated. However, rather than keeping the valve unloaded continuously, the stepless system actually unloads the valve for only a portion of its stroke.

By allowing compression to occur during only part of the stroke, one obtains partial flow instead of full flow or no flow for a fully unloaded cylinder. Actuation of these unloaders is by a specially designed control panel that monitors process flow requirements and unloads the compressor as necessary. Because of its principles of design, this system is limited in turndown, and applications must be reviewed individually to confirm their suitability.

In general, suction valve unloading is an excellent method of controlling capacity. The devices are simple and easy to maintain and operate. They are efficient and are very good for startup unloading as the starting torque requirements are extremely low. Nevertheless, suction valve unloading does have some drawbacks.

If a cylinder is operated fully unloaded for an extended period of time, gas temperatures in the inlet passage will rise, since the same gas is being worked back and forth through the unloader passageway. This can become a problem, especially where the gas k value is high. The solution is to load the cylinder periodically so that the heated gas will be pumped to process and the cooler inlet gas stream will normalize the temperatures. Typically, this cyclic loading should occur for 10 minutes out of every hour of unloaded operation. This heating problem does not occur at 50% loads, since the active end of the cylinder will scavenge the heated gas and keep the stream of cool suction gas in the suction chambers.

Another area to be looked at when operating suction valve unloaders is the potential for a nonreversing crosshead pin load. For a given cylinder description, there may be conditions where unloading one end of the cylinder, typically the frame end, can cause a nonreversing load. This would prevent lubricant from flowing into the pin clearance. To guard against this, each mode of unloading should be studied for pin load reversal and any unacceptable operating modes highlighted to the compressor operator, or, in the case of an automated system, locked out of the unloader logic.

3.1.4 Clearance Pockets

Clearance pockets (Fig. 3.4) are also a common way of reducing the capacity of a compressor. From compressor theory, we know that the volumetric efficiency of a cylinder is dependent on its clearance or nondisplaced volume. We again recall that

$$\mathrm{VE} = \eta_v = 100 - C(r^{1/k} - 1) \tag{3.2}$$

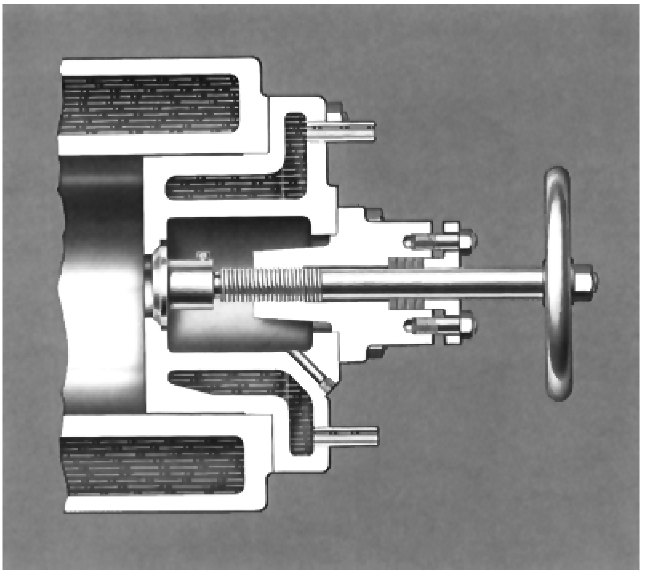


FIGURE 3.4 Manual fixed-clearance pocket valve is generally located in the outer head of a cylinder, as shown. This type of control is used for applications that require limited and infrequent capacity changes. *(Dresser-Rand Company, Painted Post, N.Y.)*

This equation illustrates that as clearance is increased, the volumetric efficiency decreases, thus reducing the amount of flow to process. It should be noted here that the degree to which the volumetric efficiency is affected by increased clearance is governed by the compression ratio of the service. In practical terms we see that when the compression ratio is 2.5 or higher, an increase in clearance will actively reduce the volumetric efficiency. However, for ratios of 2.0 or less, it takes a fairly major increase in clearance for any reasonable reduction in volumetric efficiency. As a generalization, clearance pockets are ineffective and not useful with low compression ratios (1.5 or less).

The design of a clearance pocket is simple and can be visualized from Figs. 3.4 and 2.22. It is essentially an empty volume, typically in the outer head of the cylinder, with a valved passage to the cylinder bore. During normal operation, the valve is closed and the cylinder operates at full capacity. For reduced-capacity operation, the valve is opened and the cylinder capacity is reduced by the effect of this added clearance on the volumetric efficiency. Typically, clearance pockets are of a fixed volume and sized to reduce flow precisely to a predetermined level. It is not uncommon to use multiple fixed-volume clearance pockets to allow for numerous discrete reduced-capacity control steps.

For example, a large cylinder may be executed with two fixed-volume clearance pockets, one small and one large. Using the small and large pockets independently and then together, one can obtain three reduced flow modes of operation from two clearance pockets. It is also possible to put clearance pockets on the frame end of a cylinder so that many different clearance volume combinations are feasible. Fixed-volume clearance pockets, just as valve unloaders, can be actuated manually or pneumatically.

Figure 3.5 depicts an option available with clearance pockets. Instead of executing them with a fixed volume, they can be configured for a variable volume. A variable-volume

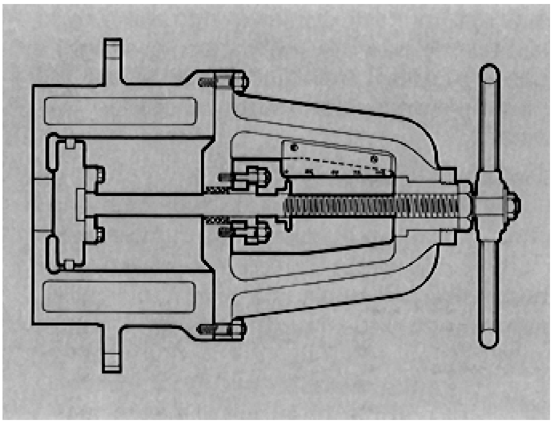


FIGURE 3.5 Manually controlled variable-volume clearance pocket. This clearance pocket provides capacity reduction in an infinite number of steps over a given range. This pocket can also be automatically actuated by a hydraulic system that varies the position of the piston in the pocket. *(Dresser-Rand Company, Painted Post, N.Y.)*

clearance pocket is typically mounted on the outer head of a cylinder and is basically a cylinder itself with a piston moved back and forth to increase or reduce the volume of the pocket. An advantage of the variable-volume clearance pocket is that instead of having fixed steps of unloading, stepless turndown is achieved simply by adjusting the pocket. This is particularly useful in meeting ever-changing process conditions, since the pocket can be adjusted in the field as required. Historically, variable-volume clearance pockets have been actuated manually by means of a very large handwheel to move the piston back and forth. When the correct volume is achieved, the pocket can be locked in place by means of a second locking wheel.

The handwheel method sounds fine until one considers the force against the pocket piston created by the process gas pressure inside the cylinder. At times it might take the strength of two operators to turn the handwheel and adjust the pocket. Also, as corrosion or deposits are formed from the process gas, the piston may tend to stick in one position and may no longer be useful for adjustment.

An improvement over the manually controlled pocket is the hydraulically controlled variable-volume clearance pocket. Here, the manual handwheel has been replaced by a hydraulic cylinder. The cylinder is coupled directly to the pockets piston and uses hydraulic pressure from a support package to move the piston back and forth. Not only can the hydraulic cylinder create a higher force than the typical field operator, but it also allows the pocket to be automated so that the pocket will adjust itself to field conditions.

All of the capacity control methods described so far have assumed that the compressor is running at a constant speed. Since the reciprocating compressor is a positive displacement device and does not rely on sophisticated gas dynamics for its compression, it is possible to vary the speed to adjust the throughput directly. One typical variable-speed driver is the steam turbine. Whenever steam is economically available in plants, it is not uncommon to have a large steam turbine coupled to a reciprocating compressor through a gear speed reducer. By varying the speed of the turbine, the capacity of the compressor can be controlled directly. However, because of the complexity of the drivetrain (large flywheel, soft couplings, multireduction gearboxes) the system is limited in its turndown by torsional problems.

Typically, a turbine gear drive arrangement is limited to a $\pm 10\%$ speed variation. Operation outside of the approved speed range could lead to massive failure of the drive system. New technology makes it feasible to use variable-speed ac motors for running compressors. (Dc drive systems have been available for some time but are limited to low-horsepower applications.) Modern control technology allows ac motors of all sizes to operate with fully variable speeds. However, since this technology is relatively new to the reciprocating compressor industry, there are still a number of questions to be addressed: torsional response; pressure pulsations at reduced rotative speeds; feedback into the electrical grid due to torque pulsations from the compressor; economic feasibility; whether variable speed should be employed analogous to bypasses in the past (i.e., simply to trim flow between major steps of operation). All these areas need to be considered before variable-speed ac motors are specified as drivers for reciprocating compressors. As of 2005, economic and reliability concerns were too formidable to allow large reciprocating compressors to be operated at variable speed.

3.2 MORE ABOUT CYLINDER JACKET COOLING AND HEATING ARRANGEMENTS

During their normal compression cycle, reciprocating compressor cylinders typically generate considerable amounts of heat. The heat comes from the work of compression plus the friction of the piston rings against the cylinder wall. Unless some of this heat is dissipated, undesirably high operating temperatures will occur. Most cylinders intended for process gas operation are designed with a jacket (shown in Figs. 2.20 through 2.22) to allow this heat to be removed by some cooling medium.

There are a number of advantages in dissipating this heat.

1. By lowering the cylinder wall operating temperatures, one can reduce losses in capacity and horsepower due to the suction gas being preheated by warm cylinder gas passages. The cooler the inlet gas, the denser it is, and greater mass flow per unit volume will result. Removing heat from the gas during compression lowers its final discharge temperature and reduces the power required for compression.
2. Dissipating the heat from the cylinder and reducing the inlet gas temperature creates a better operating climate for the compressor valves, yielding longer valve life and reduced formation of deposits.
3. A jacketed cylinder filled with coolant will maintain a more even temperature throughout the cylinder and reduce hot spots, which could cause uneven thermal expansion and undesirable deformation of the cylinder.
4. A lower cylinder wall temperature leads to better bore lubrication. Lubricants will break down less on a cool wall than they would on a hot wall, and better lubrication leads to extended ring life and less maintenance.

However, care is needed not to reduce the cylinder operating temperatures too much. Consider the problems created by introducing a warm saturated gas into a cylinder with cold metal sections. Condensation will occur in the bore; thus, washing the lubricant from the cylinder walls will cause accelerated wear of the piston and rider rings. Even worse, a large quantity of condensed liquid could collect in the inlet gas passage and be introduced into the cylinder as a slug of liquid. This could lead to at least broken valves and perhaps a broken cylinder. To avoid this condensation problem, it is considered good practice to use a cylinder coolant temperature approximately 6°C (10°F) warmer than the inlet gas temperature.

3.2.1 Methods of Cooling

When evaluating a process compressor for cooling, a cylinder will fall into one of four general categories.

1. *Noncooled.* For a cylinder operating in cryogenic service where gas temperatures are typically below −60°C (−75°F), no cooling is required. In fact, no cooling medium is suitable for providing uniform, acceptable cylinder temperatures. For applications like these, cylinders are often designed with no cooling jacket at all and will simply be insulated from the ambient air in an attempt to avoid severe temperature differentials or frost formation on the cylinder exterior.

2. *Static cooling*. Static cooling is used for applications where gas discharge temperatures are below 88°C (190°F) and mean temperatures are low [below 60°C (140°F)]. This type of cooling is also used where there will be no unloaded cylinder operation that could create abnormally high temperatures.

In a static system the cylinder water jacket is simply filled with a cooling medium such as a water–glycol mixture. No attempt is made to circulate the mixture. A small reservoir vented to atmosphere is provided to allow for thermal expansion.

3. *Thermosyphon cooling*. A cylinder may be thermosyphon-cooled where discharge temperatures are moderate [88 to 90°C (190 to 210°F)], mean temperatures are in the range of 60 to 66°C (140 to 150°F), and where there will be no extended periods of fully unloaded operation that could increase operating temperature. This cooling method was illustrated in Fig. 2.32.

A thermosyphon system is similar to the static system; however, there is now a small section of pipe connecting the top cooling medium outlet to the bottom of the cylinder. The idea here is that as the warm water in the radiative sections cools, it will flow to the bottom of the cylinder, creating a slight circulation through the cylinder jackets.

4. *Full-circulation cooling***.** For applications where gas mean and discharge temperatures are in excess of the previously stated limits or where extended periods of fully unloaded operation are anticipated, the cylinder requires that a coolant be circulated through its jacket to dissipate the heat buildup. This was discussed previously (see Fig. 2.31).

For large process cylinders, it is common to have water-cooled cylinder heads as well as a jacket around the cylinder bore. These sections are connected by external jumper pipe. As mentioned earlier, the temperature of the coolant should be controlled so that it is maintained approximately 6°C (10°F) higher than the gas inlet temperature. In addition, the flow should be controlled so that the temperature rise across the cylinder circuit is between 3 and 11°C (5 and 20°F). Flow through a cylinder is controlled by means of a globe valve on the discharge of the jumper piping around the cylinder. It is customary to throttle the coolant outlet to ensure that the cylinder is constantly flooded with coolant. A thermometer and sight flow indicator are located immediately upstream of the discharge globe valve to assist the operator in adjusting the coolant flow to maintain the correct temperature rise across each cylinder.

In calculating coolant flow required for a given cylinder, we find that the coolant temperature rise across the cylinder and the coolant flow rate are inversely proportional. In other words, a high flow rate of water will pick up only a slight increase in temperature. Conversely, a slow trickle of water will rise appreciably in temperature. In optimizing the balance of flow rate and temperature rise, it is important to remember to keep temperature rises moderate [3 to 11°C (5 to 20°F)]. This will ensure a fairly even cylinder temperature and will keep the flow rate within certain limits. Too low a flow rate will allow silt or other entrained particulates to drop out in the water jacket, eventually leading to a blockage of flow. On the other hand, too high a flow rate can create a prohibitively high pressure drop through the coolant circuit. Typically, velocities between 4 and 8 ft/s based on the size of the coolant jumper pipe have been used as guidelines.

Actual calculation of coolant flow is, at best, an estimate based on empirical formulas and, at worst, a black art. An old rule of thumb for determining coolant flow in gpm is to divide cylinder horsepower by the allowable temperature rise. This method is simple but neglects accounting for most of the critical parameters. Coolant temperatures, gas temperatures, cylinder size, and frictional factors are all important parameters in calculating cylinder coolant flow requirements. Calculation methods today are based on empirical formulas that use these parameters and have been found to have good field correlations.

Untreated water can be used, provided that the temperatures are acceptable and that the water is filtered before it goes through the cylinder jackets. Nevertheless, refineries typically use cooling tower water, which is temperature controlled, filtered, and treated. Coolant discharging from a shell-and-tube intercooler is a good source of coolant since it has been preheated by the compressed gas stream. This should reduce the risk of incurring moisture condensation and liquid slugging.

The ultimate source of coolant is from a closed cylinder jacket coolant console dedicated specifically to an individual compressor. Figure 3.6 depicts such a console. The system can be tailored specifically to the compressor in question. Motor-driven pumps (main and backup) provide circulation. Coolant temperature can be kept high enough by the use of a reservoir heater, kept cool enough by a shell-and-tube or radiative-style cooler, and controlled by an automated temperature control circuit. Instrumentation is added to monitor and protect the system.

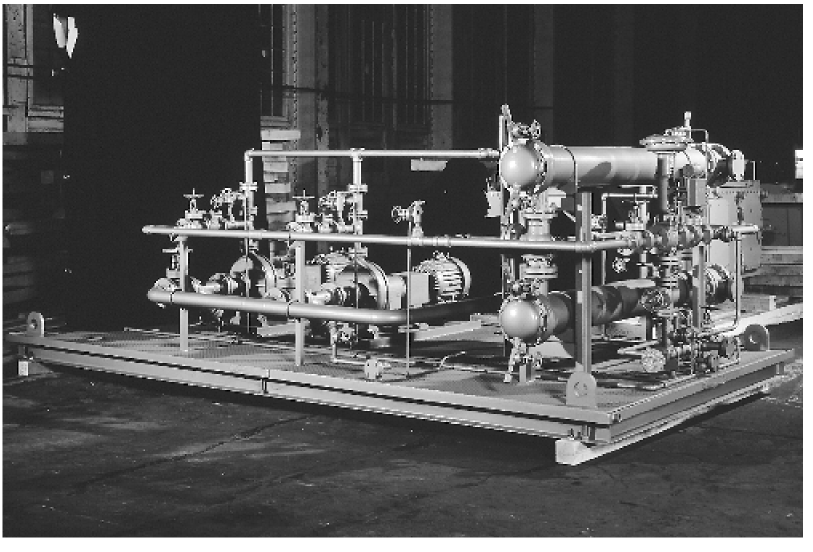


FIGURE 3.6 Packaged jacket water cooling system. *(Dresser-Rand Company, Painted Post, N.Y.)*

3.3 COMPARING LUBRICATED AND NONLUBRICATED CONVENTIONAL CYLINDER CONSTRUCTION

One of the major areas of motion, and thus wear, in a reciprocating compressor is in the cylinder. Considering that during a year's normal operation a piston travels nearly 100,000 miles in the cylinder bore, it can be seen that cylinder lubrication merits closer investigation.

3.3.1 Lubricated Cylinder Designs

Probably 80% of all process reciprocating compressor cylinders are lubricated. Lubricating the cylinder bore makes sense. It reduces friction between the piston rings and the cylinder bore, thus reducing frictional heat and wear of both cylinder bore and piston rings. It lubricates the cylinder valves, helping them to survive the 100 million + cycles they go through in a year's operation. A film of lubricant in the cylinder also helps protect the cylinder components from the effects of corrosive gases.

When using lubricated construction, cylinder designers use the lubricating film to the best advantage. Because the sliding surfaces will be lubricated, harder piston and rider ring materials can be used. Typical materials of construction would be glass and/or molybdenum-filled Teflon (PTFE). Because of its relative hardness, this material has excellent durability, and when lubricated, the wear characteristics of both rings and bore contacted are excellent. Carbon- or graphite-filled PTFE, which has become a rather universal ring material, is also frequently used in lubricated service. Since the piston will be riding on a film of lubricant, the piston can be relatively heavy. It should be noted that although lubricated construction allows a piston to be run directly in the cylinder bore, it has become common practice in the process compressor industry over the past decades to design pistons with rider bands (shown in Figs. 2.33 and 2.34), supporting the pistons in the cylinder bore.

Rider bands can be considered as an expendable support shoe for the piston. As wear occurs, the ring could be readily replaced. For lubricated designs, rider band bearing loads are typically in the 8- to 10-psi range when considering a contact area of 120° of arc.

Piston rod packing rings would also be made of glass and/or molybdenum-filled PTFE for lubricated service. Again, this relatively hard compound shows excellent durability and wear characteristics in lubricated service, without being excessively abrasive to the piston rod surface. As with the piston and rider rings, carbon- or graphite-filled PTFE is also commonly used.

Because the compressor cylinders are operating at elevated pressure, the lubricant must be pumped into the cylinder in a controlled manner. There are a variety of cylinder lubricators designed precisely for this purpose. The most common style of cylinder lubricator is the pump-to-point lubricator (Fig. 3.7).

Pump-to-point lubricators are designed with a fabricated steel box serving not only as the main body of the system, but also as the lubricator sump. Through this box runs a multicammed shaft that is driven either by the compressor crankshaft or by an electric motor. The actual pumping units are located on top of the box. The units are equipped with a suction straw that drops into the sump and a follower that rides on the cams to actuate the pumping plunger. On top of the pumping units is a transparent cap that allows observing the lubricant as it is drawn up through the suction straw and drips down to be pumped to the cylinder. Counting the frequency of drops facilitates monitoring of lubrication flow rates (Fig. 3.8).

A cylinder typically has multiple points of lubrication. As illustrated in Fig. 2.29, there may be several feeds in the main bore, depending on size and pressure. There could also be several feeds in the packing case, and sometimes a feed in a partition packing. Each point

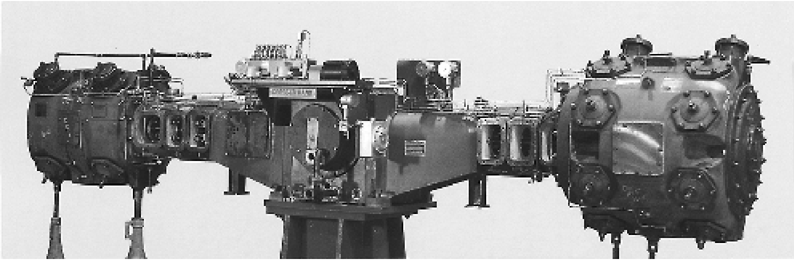


FIGURE 3.7 A pump-to-point lubricator mounted at the center of the frame supplies different amounts of lubricant to the packing and cylinders. *(Dresser-Rand Company, Painted Post, N.Y.)*

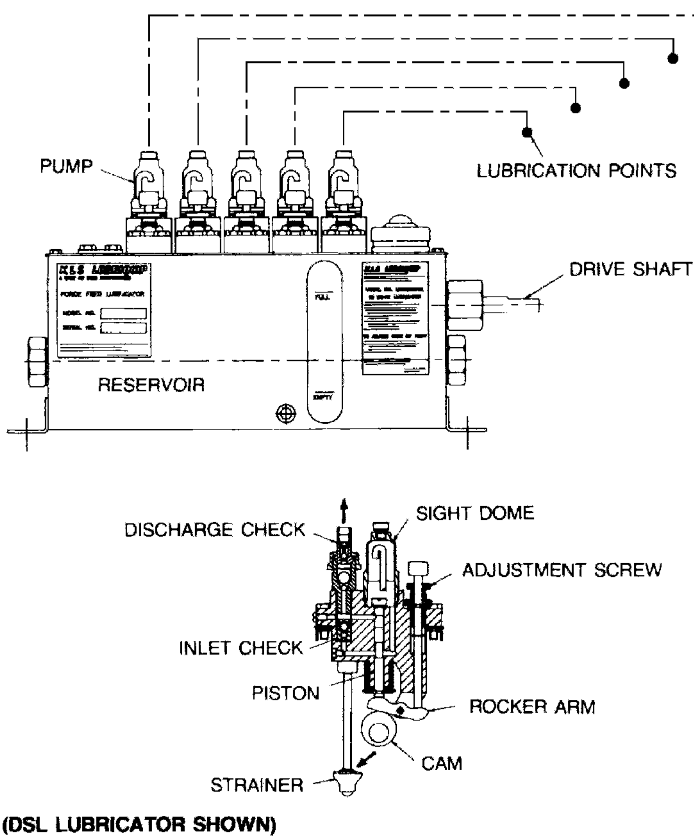


FIGURE 3.8 Pump-to-point lubricator. *(Lubriquip, Inc., Cleveland, Ohio)*

of lubrication is fed by an individual pump from the lubricator, which is why this style is called a *pump-to-point lubricator*. The advantage of using a pump-to-point lubricator is that each pump is adjustable individually. Distribution of lubricant to different areas in the cylinders can thus be tailored to best suit the lubrication requirements.

The other major style of lubricator is the *divider block lubricator* (Fig. 3.9). Divider block lubricators incorporate a single high-pressure injection pump feeding a number of

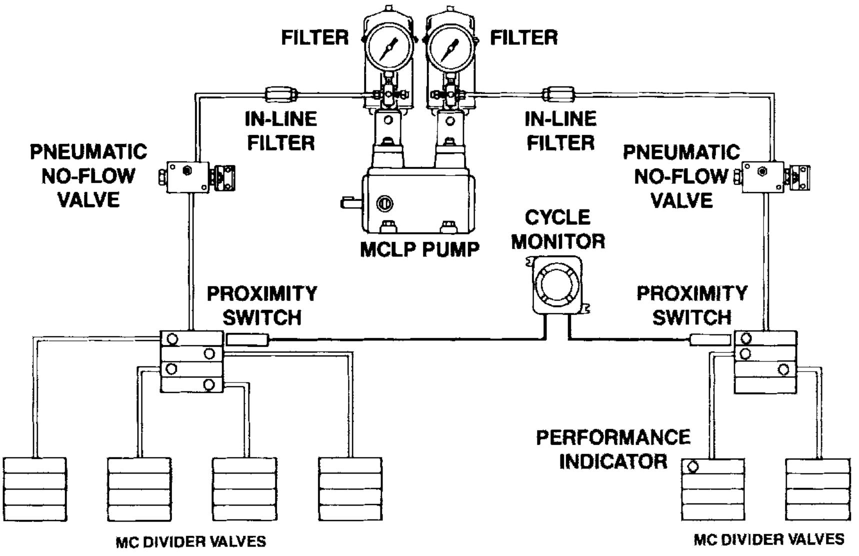


FIGURE 3.9 Divider block lubricator system. *(Lincoln Division of McNeil Corporation, St. Louis, Mo.)*

divider blocks, also referred to as *splitter blocks,* or *divider valves.* The function of a divider block is to divide the flow of lubricant from the pump into multiple streams of various predetermined proportions so that each point can be fed an appropriate amount of lubricant. This style of lubricator is popular because, once constructed, the proportion of flow from the divider blocks is fixed and cannot get out of adjustment in the field. Moreover, divider block lubricator arrangements are easily instrumented to annunciate deficiencies in any point being lubricated. This must be attributed to the cascading motion of the divider blocks, which are interconnected through ports, thus requiring that each block deliver its portion of lubricant before the next block functions.

Should any individual point become clogged or blocked, all oil flow would stop. A single no-flow switch can thus be used to monitor all lubrication points. Additionally, there are cycle monitors that can be used to monitor the rate of cycling of the blocks. If for some reason the blocks take too long to cycle, indicating a reduced flow rate to the cylinder, this device can sound an alarm. One of the disadvantages of divider block lubricators is the difficulty in field-adjusting the differently proportioned flow amounts. Such adjustments may become desirable whenever substantial differences exist in cylinder pressure levels.

3.3.2 Nonlubricated Cylinder Design

There are some processes that will not tolerate oil entrained in a gas stream. Oil separators can be installed in the compressor discharge lines; however, sometimes these are not effective enough for the level of cleanliness required, or there may be safety problems associated with a particular gas contacting the lubricant. In such cases, the only alternative is to use nonlubricated cylinder designs. Probably 20% of all process gas compressors are designed for nonlubricated operation because of process demands. Oil in the gas stream could lead to catastrophic problems in an oxygen compressor or even in a high-pressure air compressor. Also, many chemical processes cannot tolerate the existence of lubricant in their catalysts.

Special consideration must be given to nonlubrication applications. Without an oil film, piston rings do not seal in the cylinder bores as efficiently, thus causing blowby in the cylinder, resulting in a lower delivered gas flow from the cylinder. There are certain operational limits that must be addressed. At higher operating gas pressure, piston rings have much greater force against cylinder walls, thus creating greater wear problems. Except for special applications, nonlubricated construction is limited to pressures below 2000 psi.

Without the lubricating film to reduce wear, we must change out piston ring materials. Typically, a nonlubricated piston or rider ring is made from carbon- or graphite-filled PTFE. The carbon or graphite filling gives it a measure of lubricity. Carbon- or graphite-filled PTFE is also a little softer and less abrasive in cylinder bore contact than is glass and/or molybdenum-filled PTFE. For applications with very dry gases, Morganite Graflon compounds have proven to be very good. For oxygen applications, copper-filled PTFE is used, replacing lead-filled rings due to the environmental problems associated with lead. The oxygen atmosphere causes the copper-filled rings to form a copper oxide layer that gives them a natural lubricity similar to the carbon or graphite filling. Copper filling is preferred over carbon filling because it is less active in an oxygen atmosphere. Of course, this is where noncontacting, labyrinth piston design excels. This is discussed later.

For conventional nonlubricated designs, the cylinder designer must address bearing load in the cylinder. Without the advantage of the lubricating film, the designer must reduce the bearing load on the rider bands typically to 3 to 5 psi, by reducing the piston weight or increasing the rider band area.

Piston rod packing designs for nonlubricated construction will also differ from those used in lubricated construction. Typical ring materials are again a carbon-filled PTFE material. Because of the additional frictional heat generated between the nonlubricated rings and the piston rod, it is wise to have a cored packing case and circulate coolant through it at pressures above 250 psi. In some cases, it may also be necessary to include bronze backup rings in the packing case to aid in heat transfer from the rod into the packing case and cooling media.

3.4 COMPRESSOR VENT AND BUFFER SYSTEMS

Whenever a moving surface must be sealed against a pressure differential, one must expect leakage. In a reciprocating compressor, this leakage typically occurs at the piston rods. Even with state-of-the-art multiring packing cases, it is likely that process gas operating at hundreds or thousands of psi will leak from cylinders past the rod packings. Since this process gas is typically hazardous, flammable, or even toxic, it is certainly undesirable and often illegal simply to allow the gas to leak into the atmosphere. It is thus necessary to capture this leakage gas.

Normally, a packing case will have a vent connection piped from between the last two rings in the packing case (Fig. 2.46). This connection not only serves as a vent to remove gas before it is leaked into the cylinder distance piece but also serves as a drain for excess cylinder lubricating oils. This is why packing vents should always be piped to a point below the piston rod. These vent connections can be piped to a flare stack or gas recovery system or back to gas suction if the pressure is low enough.

Venting the packing case will not, however, give completely ensure against the leakage of gas into the cylinder distance piece. To prevent this gas from leaking into the compressor frame, where it can cause damage to the frame end running gear, cause an explosion, or simply leak to the atmosphere through the frame breather, it is customary to vent the distance piece to a flare stack or recovery system. A more conservative approach is to use a two-compartment distance piece where the cylinder end compartment is buffered with an inert gas and the frame end compartment is vented to flare (Fig. 2.50). By pressurizing the cylinder end compartment to a higher pressure than the flare stack, any leakage past the rod packing will be by the inert gas leaking toward the packing vent rather than the process gas leaking out of the packing case into the distance piece. The vented frame end compartment ensures that no gas will leak into the frame.

In cases where it is imperative to capture as much leaking gas as possible, one can use the same type of purge and vent system in the packing case itself. This was illustrated in Fig. 2.46. Here a buffer of inert gas is introduced between the last two rings in the packing case and vented between the second- and third-to-last packing rings. Since the last several rings in the packing case are now at a very reduced pressure level, one can no longer rely on the gas pressure to adequately seal the rings against the rod and packing case. This problem can be alleviated by using mechanically loaded ring sets in the last two packing cups.

The question of where to vent the leakage is best answered by the end user. Nonhazardous, nonflammable gases may be allowed to leak directly to atmosphere. Some compressors will have a simple gooseneck on top of the distance piece vent connection or use louvered covers on the distance piece doors to allow ambient air to purge any leakage. The most obvious vent destination for a flammable process gas is the flare stack. Flare systems typically run at pressures between 5 and 15 psig. For higher-pressure systems it may be necessary to put an additional sealing ring at the end of the packing case beyond the vent to help encourage the gas leakage to run into the high-pressure flare header instead of into the distance piece and then to the atmosphere.

The most economical place to which to vent is the first-stage suction of the compressor. The process gas leakage is thus recovered and put back into the process. Discretion must be used in taking a packing vent to suction. If suction pressures are much above 25 or 30 psig, it may create a major problem for the packing case. This case would now possibly experience pressurization of the midportion with 30-psig gas, and this might compromise its sealing capabilities.

3.5 COMPRESSOR INSTRUMENTATION

The area of compressor design that offers the greatest degree of freedom and personal input from the equipment specifier is instrumentation. Since compressor instrumentation does not contribute directly to the pumping duty of the machine, it is often neglected or overlooked. The amount of instrumentation on a given unit can range from a few simple pressure and temperature gauges to sophisticated electronic or computer-based systems. Even more important is the possible integration of monitoring and analysis tasks into the process control computer. See Section 3.6 for additional data.

As we analyze this subject, we find that instrumentation serves three basic purposes: to monitor, to protect, and to diagnose. Monitoring instruments are the most basic. They are the gauges, or readout devices, that allow operators to examine the compressor and its process and to determine whether the unit is functioning properly. It is important to install monitoring instruments for all parameters critical to the operation of a compressor.

Protective devices are those that alert operators to an upset condition or keep the compressor from destroying itself. Should operator oversights occur, the compressor protective devices will shut the unit down before the problem reaches disastrous dimensions.

Diagnostic instruments are those that monitor various parameters, integrate their findings, and make a diagnosis as to the health of the compressor. Diagnostics determine not only whether the compressor is running properly or has failed; they also predict an approaching failure or operational problem. Used properly, diagnostic instrumentation helps schedule maintenance shutdowns rather than having the compressor trip unexpectedly.

Reciprocating compressor instrumentation generally covers the following parameters.

1. *Pressure*. Since the basic function of a reciprocating compressor is to elevate gas pressure from one level to another, pressure would appear to be the most basic parameter to look at. For monitoring purposes, the simple pressure gauge tells us at a glance exactly what we need to know. Pressure gauges should have large-diameter dials so that they are easily read. As with any instrument, the scale on the gauge should be selected so that under normal operating conditions, the pointer is approximately midrange. As with all pressure instruments, pressure gauges should be provided with an isolation valve to facilitate replacement or servicing.

Pressure switches are an important component of any instrumentation system. Whenever pressures go beyond the normal operating limits, the pressure switch can activate an alarm, protective shutdown, or both.

Many different pressure switch designs are available: single-pole, double-throw; double-pole, double-throw; single-level switches; multilevel switches; dual switches in a single housing; internally adjusted; externally adjusted; factory adjusted. A typical switch should be constructed ruggedly of materials suitable for the application and should be listed and approved for operation in the area classification for its installed environment. An internal adjustment is convenient to allow for field readjustment in case of changing pressure conditions and to prevent accidental misadjustment from inadvertent outside contact.

Conventional switch logic is to have the contacts normally closed in operation and open for alarm or shutdown actuation so that if the field lines are ever accidentally cut, the circuit opens, the machine shuts down, and the unit will not run without protection. Pressure switches should be installed with suitable block and bleed valves. This allows the switch to be blocked out and bled down so that its set point can be verified. Normally, a pressure gauge or connection for a gauge will be piped next to the pressure switch to assist in calibration.

2. *Temperature.* Temperature is as important as pressure in most processes. A 4½-in. dial-type thermometer is the common temperature indicator for the process industry. When measuring a fluid temperature, it should be installed in a thermowell of suitable material. Thermometers with flexible head mechanisms (every angle design) are convenient because they allow the thermometer face to be adjusted for ease of viewing.

Another method of monitoring temperatures is via a thermocouple or *resistance temperature detector* (RTD). This allows the temperature to be monitored at a remote location on a readout device. Thermocouples and RTDs can also be useful as sensors for protective circuitry. Instead of a monitoring instrument, or perhaps being included in a monitoring instrument, an RTD is a comparative circuit that compares the actual input from the thermocouple to a preset level. When this level is reached, a signal is sent to the alarm or shutdown circuitry to alert operators to the problem.

A more common approach to abnormal temperature protection is a filled capillary temperature switch. Here a gas-filled probe is used to sense the temperature being monitored. The probe is connected to the switch assembly via a protected stainless steel capillary. As the temperature changes, the gas in the probe and capillary expands or contracts, sending a proportional signal back to the switch. When the signal causes the switch to exceed a preset level, a contact is activated. With a filled capillary type of temperature switch, the switch housing itself is normally mounted away from the monitoring point, making it easier to wire and protecting it from vibration or abuse. Capillary lengths are typically 8 to 10 ft. They can be made longer, but 25 ft is probably the upper limit to ensure accuracy and responsiveness.

3. *Vibration.* Because of its design, a reciprocating compressor is subject to vibration. Reciprocating masses, reversing loads, and pulsating gas streams all contribute to the normal vibration level on a compressor. However, if this normal level is exceeded, it indicates that something abnormal is happening and should be investigated. Typical sources of abnormal vibration are pistons hitting a cylinder head from misadjustment or debris in the cylinder, a failed component in the drivetrain, or even an acoustical vibration being transmitted through the gas pipe into the compressor. Any of these sources can have a detrimental and even catastrophic effect on the compressor. To protect against damaging the compressor, many reciprocating compressors have a vibration switch mounted on their frame.

Vibration switches have typically been the mechanical (spring or magnet) type, where increased vibration causes a switch element to be released from the magnet holding it, thus activating the alarm. The setting of these switches is often the subject of considerable debate. Although certain guidelines can be set up to predict how many *g*'s are acceptable and what level is unacceptable, the best way to protect a compressor is to set the switch sensitivity in the field. The highest normally anticipated operating vibration is typically the jolt of the main drive motor starting, or changes in flow rate or flow direction of process gas streams. In recent years, the considerably greater level of sophistication employed in centrifugal compressor technology has touched the field of reciprocating compressor vibration monitoring. Accelerometers, seismic instruments, and noncontacting shaft vibration probes are now available that have enhanced sensitivity and are well suited for the diagnosis of reciprocating machines.

4. *Flow.* It is often advantageous to monitor gas flow in compressor installations. Small gas flows can be monitored using a simple rotameter. Liquid flows are typically indicated by pinwheels or flapper-type sight flow indicators. Major flows such as process gas flows are typically monitored by means of a calibrated flow orifice and its associated instrumentation.

Flow orifices are normally located in the downstream piping. Protection is generally for loss of flow of a critical fluid. The protective devices are almost always based on loss of pressure against a calibrated orifice, which then triggers a pressure switch.

5. *Liquid level.* Monitoring of liquid levels is done with liquid-level gauges. For small reservoirs vented to the atmosphere, such as a compressor crankcase or cylinder lubricator reservoir, a simple protected transparent plate or tube attached to the side of the reservoir is normally acceptable.

For pressurized applications, such as a separator or knockout drum in the gas stream, a more rugged type of gauge is required. Armored reflex or transparent liquid gauge glasses are designed to take the high pressures, mechanical vibrations, and physical abuse seen in a typical plant environment. These glasses should be isolated from their reservoirs by block valves so that the gauge glass can be removed for maintenance or replacement without depressurizing the reservoir. If required, these gauge glasses can be fitted with illuminators to allow viewing in low-light conditions. Two basic switch types are available for abnormal liquid-level protection. The first, a *displacement-type level switch,* uses a float that is raised or lowered by the liquid level in the switch standpipe. When this float goes above or below its set limit, it trips a switch, normally by using a series of magnets. Because this system uses a rising or falling column of liquid, there must be two connections to the switch: one for liquid flow, the other for pressure equalization. As with liquid-level gauges, this liquid-level switch should have isolating valves.

The other type of switch uses a variable capacitance principle. This solid-state instrument has a single probe, normally made of stainless steel or coated with an inert material, that is inserted through a single connection into the reservoir. As the liquid level moves along the probe, the electronic circuitry senses a change in the capacitance of the probe. From this changing capacitance, it determines how much of the probe is being contacted. By mounting the probe vertically in the tank, the switch can monitor the changing level of the liquid and compare it to a preset level. When the liquid reaches this level, a signal can be sent to activate an alarm or shutdown. The advantages of this type of switch are that it requires only a single connection into the reservoir and that it can monitor the liquid level continuously. Additionally, multiple-level alarm points can be set. Another advantage is that solid-state electronics are less sensitive to vibration than are positive displacement switches. If necessary, the probe can even be remotely mounted in a tank and the electronics portion housed in another area.

Another necessity for level protection is *active control* of the liquid level in a separator sump or knockout drum, rather than reling on operator monitoring and manual intervention. There are automatic traps available that monitor liquid level with a float that is linked to a valve mechanism. When the level in this trap gets too high, the float will open the drain valve and keep it open until the float drops to a lower limit. Because these traps are large and heavy, they must be remotely mounted below the sump or knockout drum so that the liquid flows down into them. There also must be a pressure equalization line between the trap and the drum.

Although these automatic traps are functional and self-contained (no external power required), they do have limitations. The valve linkage mechanisms are subject to fouling by lubricating oils or other sludge that may form in the liquids. Location of the traps and routing of the associated piping can be cumbersome, particularly for connecting to suction dampener or separators. Additionally, because these automatic traps are typically made of castings, they are limited in their pressure containment capability, and many users will not allow them in process plants.

An alternative to mechanical traps are electromechanical systems. These systems use a liquid-level switch as described previously, to energize a solenoid-actuated drain valve. When the liquid level reaches a predetermined high point, the switch makes contact, opening the solenoid drain valve. As the liquid is drained (typically, through an orifice, to control the flow rate), the liquid level is lowered until it reaches a low-level set point in the switch. Now, the switch signals the solenoid valve to close, and the cycle is ready to repeat. Regardless of which automated system is used, it is wise to retain manual draining capability.

Having reviewed what instrumentation is available and what kind of parameters to instrument, we need to determine which critical systems merit this instrumentation. Again, since the main purpose of the compressor is to compress gas, the gas system is an obvious choice to instrument. Gas pressure and temperature are important to monitor at both suction and discharge for each stage of compression. Most process applications include a high-temperature alarm switch in the discharge gas stream for each stage. If there is more than one cylinder per stage, the discharge temperature will be monitored and alarmed for each cylinder discharge. Since the discharge pressure of each stage is normally protected by a pressure relief valve, high-pressure discharge switches are seldom seen. However, low first-stage suction pressure switches are not uncommon and can help to keep from overloading a compressor due to low suction pressure or, in essence, excessive differential pressure.

Mechanically speaking, probably the most important system to the compressor itself is the frame lube oil system. Here the most critical parameter is, of course, oil pressure fed to the main bearings. Standard instrumentation would include a pressure indicator for monitoring the pressure, as well as low-pressure alarm and shutdown switches for protection of the frame and running gear. If an auxiliary lube oil pump is supplied, the low lube oil pressure alarm switch can be wired to sound the alarm and start the auxiliary pump simultaneously. The shutdown switch is normally set a nominal 5 psi below the alarm switch. Should the pressure continue to degrade after alarm activation, the compressor will be shut down before damage is done to the bearings.

Other lube oil systems instrumentation will normally include pressure indicators at the discharge of all oil pumps, temperature indicators monitoring oil temperatures in and out of the oil cooler, a differential pressure gauge, and perhaps even a differential pressure switch around the oil filters giving indications as to how dirty the filter is and whether it is starting to restrict flow. In some cases, an oil temperature alarm switch is furnished downstream of the oil cooler. Indications of high oil temperature might point to the oil viscosity becoming too low for long-term dependable operation. A liquid-level gauge glass and sometimes a level switch are installed on the frame that acts as the oil reservoir for the compressor. Temperature monitoring of the bearing surfaces is very useful on reciprocating compressors. Thermocouples or RTDs in the bearing caps to monitor main bearing temperatures are not uncommon on large reciprocating process compressors. Thermocouples can also be used in crosshead guides and motor bearings.

Protecting the crankpin or crosshead pin bearings is more difficult. Here, a eutectic device is occasionally installed at the back of the connecting rod bearing cap or in the crosshead pin. The eutectic device contains a fusible element designed to melt at a predetermined temperature and a spring-loaded pin that pops out when the predetermined temperature is reached. When these pins pop out, their motion trips a strategically placed *flapper valve,* venting an auxiliary manifold that, in turn, trips a pressure switch.

Cylinder lubricant is another critical fluid for the compressor that should be instrumented. For a normal pump-to-point lubricator, lubricator drive failure and low reservoir level can be monitored readily. To do this, the manufacturer often adds an extra lubricating pump to the box and pipes it to a pressure switch. This additional pumping unit has a shorter suction straw than that of the other pumps in the box. The theory here is that if the drive system for the lubricator fails, this extra pump, along with all the other pumps, will cease to function; thus, the pressure switch will lose pressure and activate an alarm. Additionally, as the level in the lubricator reservoir drops, this extra pump will be the first to starve. This would also activate the alarm, while the other pumps would continue to supply lubricant. With a divider block lubricator, it is practical to include a pressure indicator in the discharge line from the main pump. A no-flow switch indicating lack of flow from the entire block system can be included on one point. Cycle monitors are available that can monitor the rate of cycling of the divider blocks. To help diagnose failures, each feed point from the various divider blocks can be equipped with a pin indicator so that if an individual line is blocked, the pin will indicate which line caused the shutdown.

Other systems and types of instrumentation are sometimes selected. Rod packing thermocouples and RTDs are not uncommon. Sensing of temperature excursions here can indicate that the packing rings are worn or on the verge of failure. Also, rod drop indicators are becoming more popular in the industry. Their purpose is to monitor the position of the piston rod relative to the packing case, to give an indication of how the wear or rider bands in the cylinder are degrading. As the wear bands become thinner, the piston drops in the cylinder; thus, the rod drops relative to the packing case.

At least two styles of rod drop indicators are available. The contacting type requires that the rod drop down, contacting a soft metal cap over a pneumatic line mounted at the bottom of the packing case flange. As the rod rubs off the soft metal cap, air escapes from the pneumatic line, thus venting pressure from a switch, which in turn activates an alarm.

There is also a noncontacting style or eddy-current device. In this system, a small probe is mounted on the packing case flange over the piston rod. The probe emits an electronic signal, and by evaluating the change in interference with this signal created by changing proximity to the rod, an electronic circuit determines the probe-to-rod distance. By knowing the initial clearance between the probe and the rod and the allowable wear of the rider band, calculating and presetting alarm points is possible. The advantages of this system include elimination of wear-prone contact between the sensing element and rod in the packing travel area. Also, eddy-current devices facilitate continuous monitoring of rider band wear rates.

3.5.1 Electric vs. Pneumatic Switches

A point made earlier was that any electrical switch must be certified for operation in a particular atmosphere. This normally does not present a problem since most switches on the market carry the requisite approval for typical refinery atmospheres. There are some cases, however, where switches are not available for the proper atmosphere or where there is a question of suitability between the electrical device and the medium being instrumented. This is generally where pneumatic switches find application.

Pneumatic instruments do not encounter problems with area classifications or intermittent power availability. Pneumatic systems can also be used for remote indication of parameters by use of a pneumatic transmitter. It is not uncommon to use pneumatic transmitters for remote indication of pressure in hydrogen gas or lubricating oil systems. Obviously, the routing of tubing carrying a flammable gas or fluid into a control panel or control room would be considered hazardous.

3.5.2 Switch Set Points

In determining set points for various protective switches, it is necessary to examine the safety and reliability philosophy of a given plant. In some cases, operating personnel want to be alerted to the fact that there is an upset condition, as with the typical alarm switch. In an oil system, for example, one may set the low oil pressure switch at 25 or 30 psig, low enough to indicate there is a problem that merits operator attention but still within acceptable operating limits. A shutdown switch, on the other hand, needs to be set specifically to protect the equipment from damage. Again, in the case of a typical lube oil system, the low-pressure shutdown switch would be set for 12 to 15 psig. Below this pressure, continued operation could cause damage to the bearings or crankshaft.

It is customary for an equipment manufacturer to suggest set points for switches supplied on a compressor. However, many times these will need to be modified in the field to reflect actual compressor operation.

3.5.3 Control Panels

Most process compressors have associated with them some sort of control panel. This can be a master panel in the control room that monitors several of the critical parameters of the compressor, or it can be a dedicated panel standing adjacent to the compressor.

A dedicated panel will normally include everything required to control that compressor. It will have stop–start buttons for the main drive motor as well as switches to control electric lubricators, electric heaters, and auxiliary pumps for prelubrication or main pump backup. The panel may include various pressure or temperature gauges so that the operator can monitor the compressor and its processes from the panel rather than walking around the compressor. Main motor ampere meters may also be on this panel. One of the major features of a dedicated control panel is an annunciator that will be connected to the various switches on the compressor. As the switches send their signals, the annunciators will sound alarm horns and display what malfunction is occurring. As a shutdown signal is received, the annunciator will shut the compressor down and then indicate what caused the shutdown. Compressor capacity can usually be selected from the panel. This can be done manually by turning a multiposition switch or pressing a series of buttons, or even a pair of raise or lower buttons. Operation at a particular compressor capacity step would be displayed on the panel. Capacity control could also be automated in the panel by means of suction or discharge pressure monitors and a logic system set up to load or unload the compressor as required to maintain a given suction or discharge pressure.

Panels often include programmable controllers and minicomputers. These can be used to control the loading of a compressor or handle many other decision-making tasks. One of the newer developments for control panels is using minicomputers for compressor diagnostics. Here, the computer monitors various parameters on the compressor and forms a database, or operating history. The computer can then look for combinations of values or rates of change in values that help identify an impending failure before it occurs. A typical example of this is the monitoring of valve temperatures.

Monitoring the operating temperature of valves through the use of thermocouples in the valve chambers makes it possible to record the temperature history of each valve throughout the operation of the compressor. If the computer notices an individual valve getting hotter than the average of the other valves or observes the rate of change in temperature increasing, it can alert the operator. This indication of impending valve failure allows the operator to schedule a maintenance shutdown instead of waiting for the valve to disintegrate and potentially damage the compressor.

3.5.4 Valve-in-Piston Reciprocating Compressors

A rather interesting variation of the conventional reciprocating compressor is marketed by Dresser-Rand. Using frames in the 500- to 2000-kW size range, with speeds of 1800 rpm and strokes ranging from 3½ to 5 in. (89 to 127 mm), *valve-in-piston* (VIP) *cylinders* (Figs. 3.10 and 3.11) are available for a number of services. As a point of interest, smaller valve-in-piston machines were used in refrigeration service many decades ago.

The gas compression principles in the VIP cylinder are very straightforward. Two inlet or suction valves are stationary and mounted directly in opposite ends of the cylinder bore. The discharge valves are dynamic and mounted on the piston rod (Fig. 3.10).

As the discharge valves move toward the outer end, the frame end suction valve opens, allowing incoming gas to flow into the void created by the movement of the discharge valve. At the other end, the discharge valve opens as the gas is compressed against the outer end suction valve. Gas flow is direct and simple.

The valves used in the VIP compressor are the same mass-dampened ported-plate PF-style valves used in Dresser-Rand gas field compressors. The VIP cylinder (Fig. 3.11) is a one-piece cast high-strength double-acting cylinder. Basically, this design eliminates the conventional piston because it is both a valve and a piston.

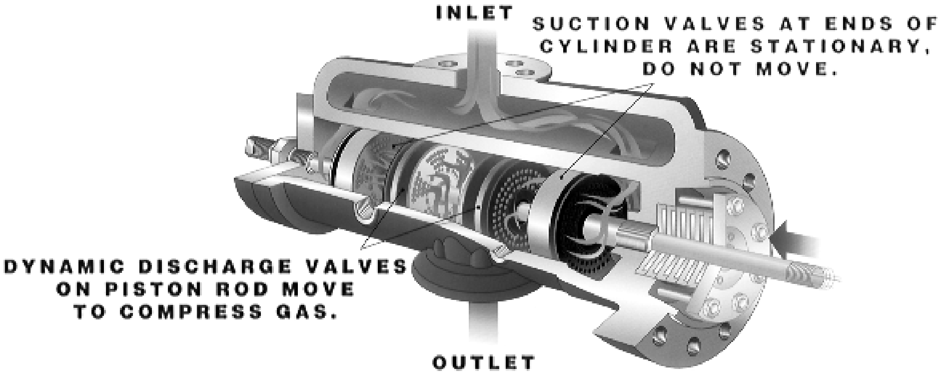


FIGURE 3.10 Valve-in-piston (VIP) compressor cylinder. *(Dresser-Rand Company, Broken Arrow, Okla.)*

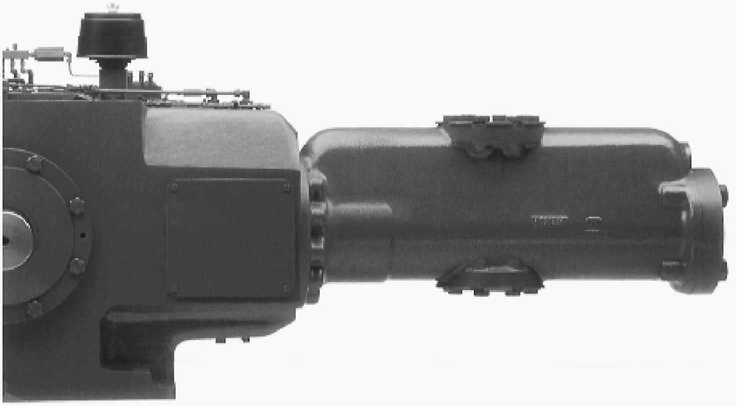


FIGURE 3.11 Conventional compressor frame with unconventional valve-in-piston (VIP) cylinder. *(Dresser-Rand Company, Broken Arrow, Okla.)*

3.5.5 Barrel-Frame Reciprocating Compressors

A departure from traditional reciprocating compressor design is shown in Fig. 3.12 and 3.13. Cooper Compression developed machines with a cylindrical frame structure that eliminates the multiple-tie-bar concept typically found in conventional machines. The cylinder bolts are accessible from the outside of the barrel frame. Compressor bolting tightness is more easily verifiable without necessitating removal of crosshead guide side covers. The stiffer cylindrical frame also eliminates the need for separate support of the crosshead. Quite obviously, a barrel-frame machine can be made for less and saves space as well. By adding a spacer module between two-throw sections, the manufacturer also achieves four- and six-throw machines.

As a concluding note, both VIP and barrel-frame machines are used in upstream oil production operations. Time will tell if they are chosen for downstream processing as well.

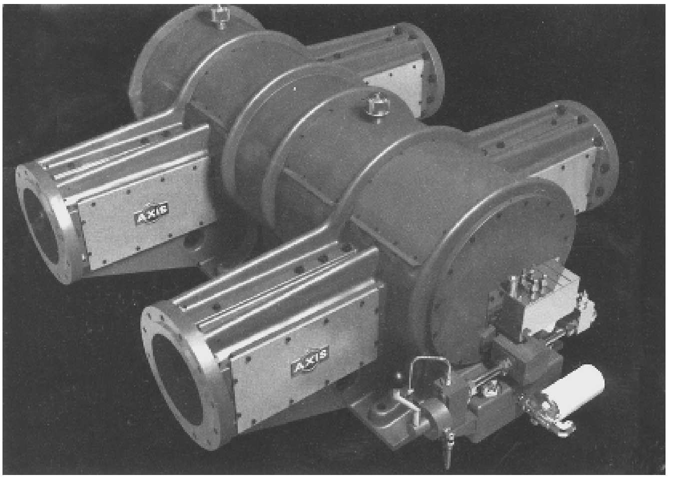


FIGURE 3.12 Barrel frame with integral distance pieces. *(Cooper Compression, Houston, Tex.)*

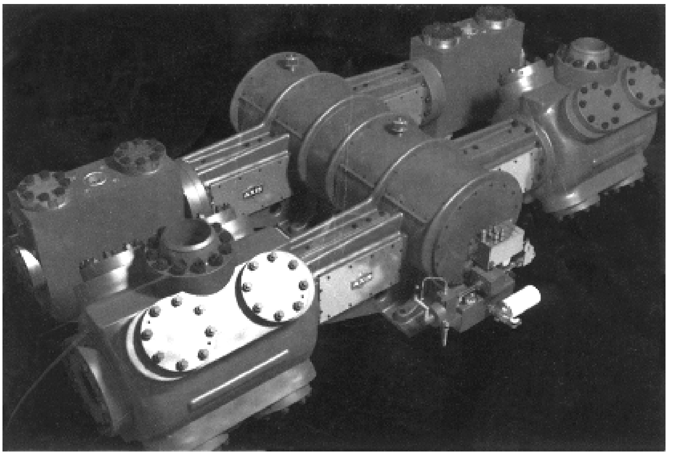


FIGURE 3.13 Barrel frame reciprocating compressor, four-throw version. *(Cooper Compression, Houston, Tex.)*

3.6 CONDITION MONITORING OF RECIPROCATING COMPRESSORS*

Over the years, electronic measurement instruments and analyzers have been used increasingly in condition monitoring and performing maintenance work on machines in industrial facilities. It would be fair to say that in the late 1970s, the "screwdriver behind the mechanic's ear" method was replaced by the piezoelectric accelerometer and the digital FFT (fast Fourier transform) analyzer.

Initially, simple data collection with manual assessment played the lead role. But progress was rapid and advances in the field of sensor technology and electronic data capture followed

*Contributed by Prognost Systems GmbH, Rheine, Germany. With special thanks to Thorsten Bickmann and Eike Drewes.

quickly. Indeed, many scientific analytical methods are now available to a wide spectrum of users involved in maintenance and reliability engineering. As an example, a range of well-known procedures for condition analysis (e.g., fast Fourier transforms) is still integrated in this effort and has certainly made it easier for the user to diagnose equipment condition.

Economic reasons prompted condition monitoring instrument developers initially to focus on the large turbomachinery market. This market, of course, addresses the needs of turbines and dynamic compressors. For condition assessment and monitoring of reciprocating compressors, special and segmented analyses were developed later. They had to meet the special requirements imposed by the unique and well-known characteristics of reciprocating machines.

Ever-increasing demands on the availability of machines resulted in a change from periodic (off-line) to continuous (online) condition monitoring. With the major expansion in digital process control systems and local area networks, networkable integrated analysis systems are needed today. These must collect electronically measured data covering the entire compressor. Moreover, these devices must often be able to present the data to a group of maintenance engineers via central visualization monitors. Those responsible for maintenance can now use their workplace computers to access the monitoring system directly through a networkable visual display. With expanding possibilities in telecommunications and increasing business automation, remote monitoring is fast becoming the standard in condition monitoring.

Suffice it to say that condition monitoring of reciprocating compressors has firmly established itself in modern maintenance engineering.

3.6.1 Maintenance Strategies

The requirements imposed on modern condition monitoring are essentially prescribed by four technical and economic objectives:

1. Machine monitoring to ensure the safety of the plant
2. Avoidance of production downtime from unscheduled shutdowns; hence, targeted planning of maintenance and shutdown events
3. Optimal use of the known or measurable wear potential of engine parts (condition-based maintenance)
4. Efficiency monitoring

Depending on the machine's production environment, these four principal aims may well demand different priority rankings. Quite evidently, operational safety will play a more important role in a 2.5-MW hydrogen compressor than it would in a 150-kW air compressor. In contrast, keeping maintenance cost to a minimum is probably the more important objective in the small air compressor case.

3.6.2 Justification for Machine Monitoring

Continuous operational condition monitoring implies that machine condition data are collected and evaluated in real time. Of special interest here are data that could rapidly change and are telltale signs of unexpected degradation of the condition of the machine. Vibration amplitude excursions indicate rapid changes and will give an early warning of disaster. A timely shutdown will minimize possible consequential damage and the resulting extended downtime.

Avoiding production downtime due to unscheduled shutdowns saves money. Along with the direct expense of a maintenance procedure (e.g., material and labor costs), machine stoppage can cause production interruptions and downtime costs that often exceed direct maintenance expenditures. High downtime costs are usually experienced by installations without redundancy or with non-spared equipment. Here, the total cost of maintenance equals material costs plus labor costs and the value of lost production.

Both frequency and duration of shutdowns can be reduced with carefully targeted planning of maintenance work. Such detailed planning allows streamlined labor and spare parts allocation. But intelligent allocation requires information on machine condition to project wear progression and probable time to failure. In condition-based maintenance, machines are shut down only if their condition demands it. Parts are changed only if a damage criterion is reached. In this way the total anticipated survivability of parts in terms of remaining life and wear reserves should be exploited. Thus, achieving a reduction in material costs may be possible only if one has reasonably accurate data on the machine's condition. One obviously gauges the existing wear potential in traditional wear parts (e.g., piston rings or packing) and thereby optimizes machine operating life [1].

Efficiency is another indicator of the condition of a machine. The primary purpose of efficiency monitoring in reciprocating compressors is to record changes in process or machine parameters that influence energy transfer to the compressed gas. Low efficiency causes gas temperature rise and increases power consumption. The efficiency of a reciprocating compressor may be determined, for example, by finding how compression power is related to the driver's power consumption.

3.6.3 What to Monitor and Why

The objectives described in Section 3.6.1 are general and not restricted to reciprocating compressors; they apply to numerous industrial machines. However, designing an effective strategy for the condition monitoring of systems of reciprocating compressors, their mechanical features, and perhaps their most vulnerable parts must be analyzed in terms of maintenance frequency. This was done by Dresser-Rand in 1997 through a survey of 200 operators and designers on the causes of unscheduled shutdowns of reciprocating compressors. The results (Fig. 3.14) clearly indicate the relative maintenance intensity of certain component groups. These statistics show that eight component groups were responsible for 94% of unscheduled shutdowns. Valve defects are obviously responsible for most of the unscheduled maintenance events. More recent experiences indicate similar results, even though the absolute involvement ratio of valves has dropped slightly, due to the use of new materials [2].

Today, monitoring systems are marketed for every component listed in the statistics for rotating machinery. The specific requirements of a reciprocating compressor must be considered when choosing a system. For example, analyses of entire subassemblies or operating point-specific threshold checks are appropriate to detect valve damage. As of 2006, the three most important methods are (1) measurement of the valve pocket temperature, (2) p–V diagram analysis, and (3) vibration analysis.

Valve Pocket Temperatures Measuring the gas temperature in the valve pocket is the simplest and most cost-effective method of valve condition monitoring. The gas temperature in the valve chamber can be measured with a temperature probe, observing the upflow of the suction valve and the downflow of the discharge valve. If there is an obvious increase in temperature at one valve, one can assume that there is damage (e.g., a leak) at this point (see Fig. 3.15).

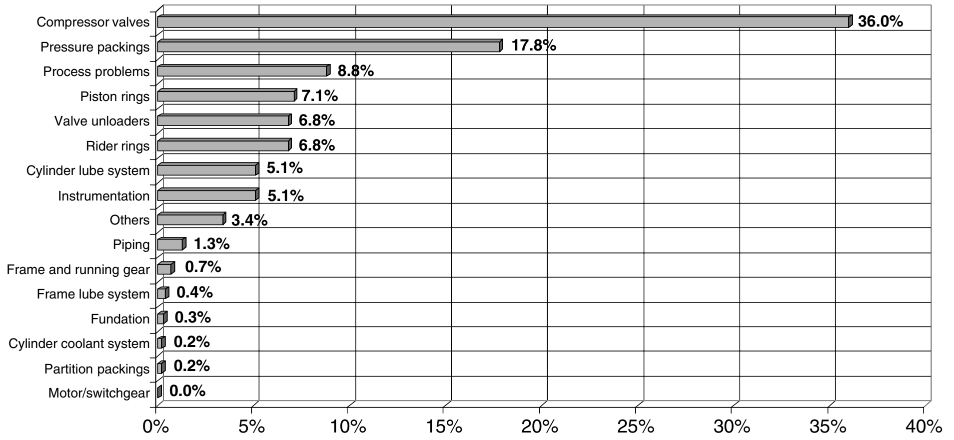
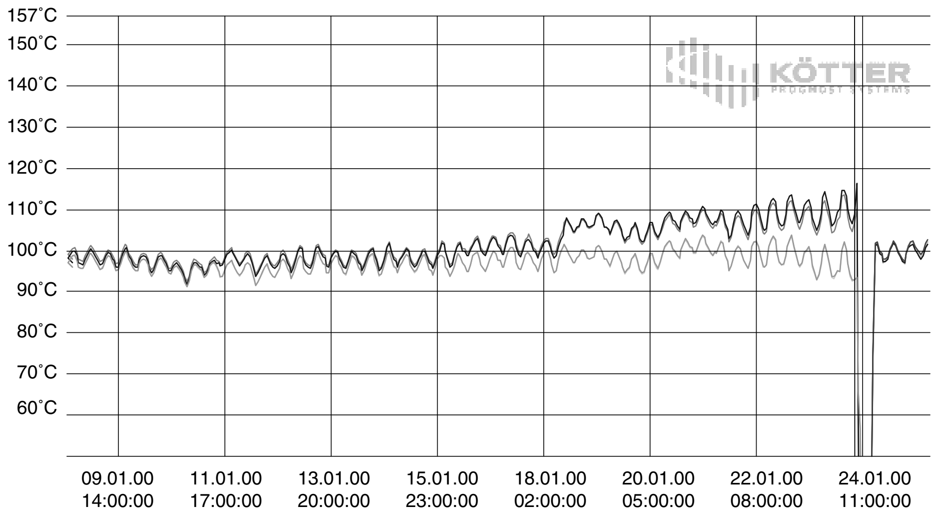


FIGURE 3.14 Primary causes of unscheduled reciprocating compressor shutdown. *(Dresser-Rand Company, Olean, N.Y.)*



	Machine	Measuring point	Description	TAG-Name		Value	Unit
	C-412 A	VT 1. Discharge, CE	temperature discharge valve 1st stage CE	T777		0 - 1000	Deg C
	C-412 A	VT 1. Discharge, CE	temperature discharge valve 1st stage CE	T773		0 - 1000	Deg C
	C-412 A	VT 1. Discharge, HE	temperature discharge valve 1st stage HE	T776		0 - 1000	Deg C
	C-412 A	VT 1. Discharge, HE	temperature discharge valve 1st stage HE	T772		0 - 1000	Deg C

Machine: C-412 A Long Term Trend

FIGURE 3.15 Head and crank end discharge valve pocket temperatures, showing distortion caused by a broken valve plate on a crank end valve. (*Prognost Systems GmbH, Rheine, Germany*)

This method has the advantage of being inexpensive and simple to install in virtually all types of valves. Generally speaking, the measuring points are integrated directly to the process control system and can therefore be fed into larger monitoring systems. The occasionally difficult interpretation of plain measurement data in multistage machines or where there are varying pressures on the intake and discharge sides is noted. However, these problems can be countered by adopting operating condition-dependent threshold monitoring.

With moderate investment in technical effort and cost, an examination of valve pocket temperatures does, in many instances, provide important information on the pressure-retaining capability or *sealing condition* of the valves.

p–V Diagram Analysis In *p–V* diagram analysis the dynamic pressure change inside a cylinder is measured. For this monitoring task, special pressure sensors with a frequency sensitivity of about 5 kHz are needed. These sensors are installed in a bore connecting directly into the cylinder; the composition of hydrocarbon gases may impose specific requirements on these pressure sensors.

Examination of *p–V* diagrams ranks among the most useful and important methods of determining overall valve condition. Recorded signals even contain clues as to the condition of specific sealing elements in the cylinder area. Valve leakage causes characteristic changes in the time rate of change of pressure. The measured time rate of pressure change (see also Chapter 2) is converted into a *p–V* diagram for which characteristic values are calculated at certain fixed points. Threshold values such as valve losses, polytropic exponents, or crank angle at which suction pressure is reached are monitored. Deviation amplitudes and the geometry of relevant excursions on the diagram identify the defect location as either the suction or discharge valve.

Cylinder pressure is a key condition indicator, as it reflects the real situation inside a cylinder. The user obtains clear local and function-oriented information on component condition and can also identify the precise effects on the compression process of the machine. Understanding the extent of a capacity reduction caused by component damage leads to an informed decision on whether repairs are necessary and economically justified.

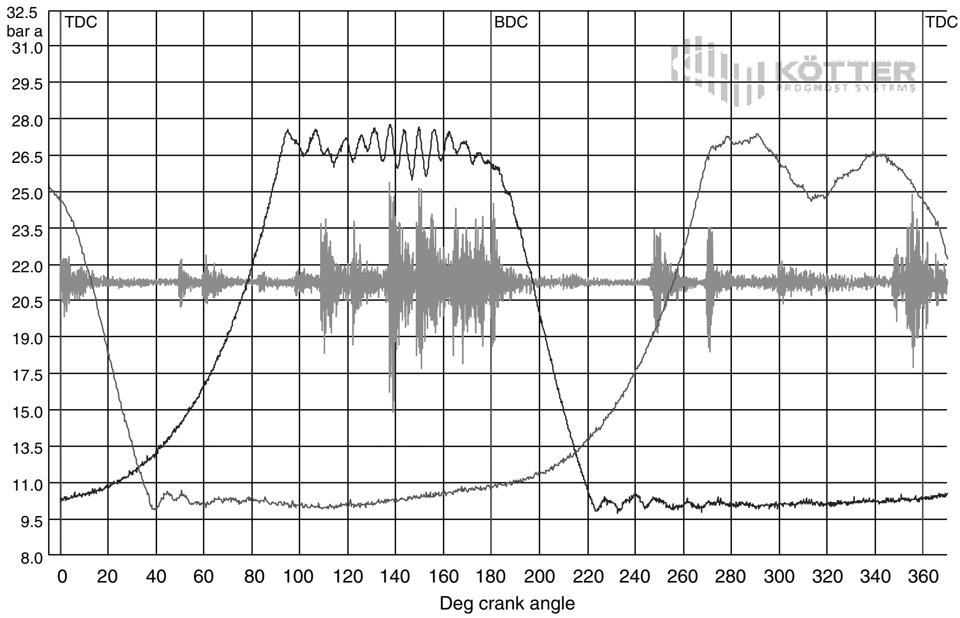
p–V diagram analysis with automatic data formatting requires meticulous operating point-dependent monitoring. The various compressor operating or load conditions must be taken into account and differentiation made between operation- and condition-dependent changes in data. In particular, this applies to machines that regulate capacity with stepwise valve unloading. The energy absorbed by the compression process can be read directly from the area of the measured *p–V* diagram, making it possible to determine the efficiency directly.

Good *p–V* analysis is one of the central and most comprehensive methods for monitoring changes in condition in piston seal tightness. Although instrumentation retrofits on older machines can be relatively expensive, the quality and volume of information often makes it very easy to cost-justify the retrofits. Modern means of condition monitoring should certainly be incorporated in new reciprocating process compressors.

Vibration Analysis Additional information on valve condition is provided by acceleration amplitudes measured at the compressor cylinders. This mode of monitoring involves installing accelerometers on the cylinders, and frequencies up to 30 kHz are of interest here. Acceleration is, of course, the time rate of change of vibration velocity, which, in turn, reflects valve movement and valve component displacement with respect to time. Sensor locations must be selected such that vibrations from as many valves as possible can be measured.

Above all, this provides information on the valve opening and closing processes, where large vibration peaks occur. It identifies *valve flutter,* which often results in dramatic reductions in valve life. Among other possibilities, valve flutter frequently occurs in (rarely used) variable-speed situations or in reciprocating compressors whose stage pressure has been altered. Figure 3.16 illustrates a typical vibration excursion attributed to valve flutter.

As the opening and closing of the suction/discharge valves takes place only at certain intervals at different points on the crank revolution, the vibrations measured at the cylinder—apart



	Machine	Measuring position	Description	TAG-Name	Value	Unit
	HDS-1238 B	Cyl 1	Vibration signal	X771		m/s^2
	HDS-1238 B	Cyl 1, CE	Pressure signal	P776		bar
	HDS-1238 B	Cyl 1, HE	Pressure signal	P775		bar

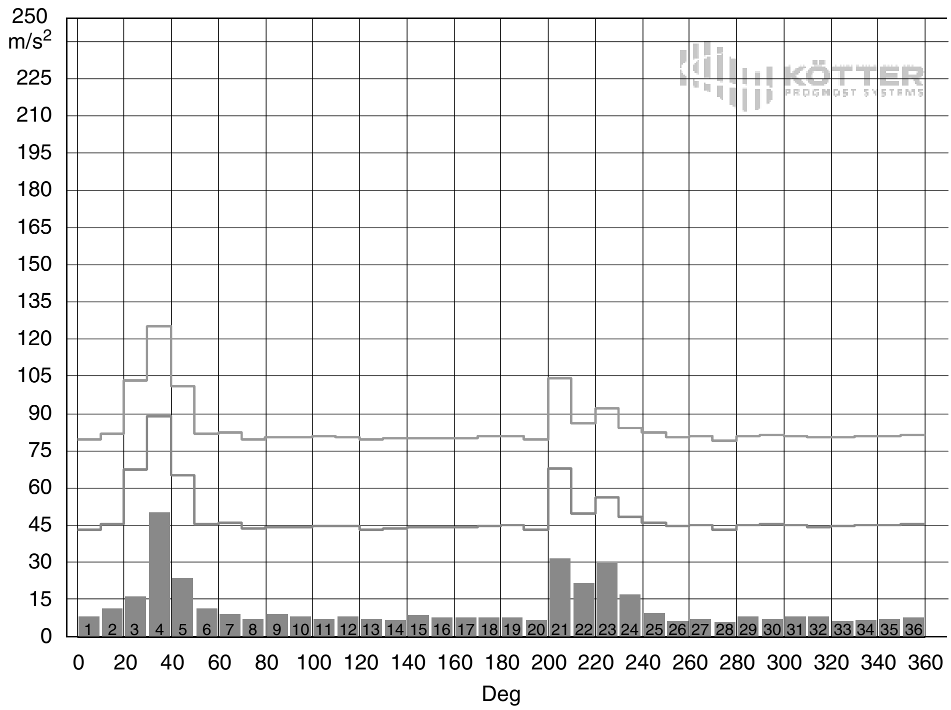
FIGURE 3.16 Cylinder vibration and indicated pressure course measured with valve flutter occurring at the crank end discharge valve. (*Prognost Systems GmbH, Rheine, Germany*)

from the peaks of valve action—also show wide areas with relatively low amplitudes. To provide effective vibration monitoring and to be able to separate the low-vibration areas and the vibration peaks caused by valve action and to monitor them separately, a special analysis approach is required. Here the vibration signal of a crank revolution of 360° is split into 36 segments of 10° each. Characteristic values are calculated for each segment (e.g., for peak and root-mean-square (RMS) values). Each segment is allocated its own threshold value (see Fig. 3.17).

The particular advantage of segmented vibration analysis centers on aspects of integrated diagnosis that generate condition information directly from the signal measured. Each segment is allocated one phase of the work cycle: say, gas intake vs. gas compression. When a threshold is exceeded, direct inferences can be made regarding valve function and component condition.

Vibration analysis on the cylinder is especially effective in combination with pressure monitoring so as to immediately recognize and then quickly prioritize the interpretation of threshold violations. When installing in accelerometer, the mounting points must be checked for signal quality to ensure that peaks are transmitted when valves open and close. Usually only one sensor is mounted per cylinder, so the installation costs are slightly less than for p–V diagram analysis.

Piston Rod Packing The second most frequent reason for unscheduled shutdowns, after deficiencies with the compressor inlet and outlet valves, relates to packing problems. The packing components seal the cylinder at the piston rod passage opening. Two different



	Machine	Measuring position	Description	TAG-Name	Value	Unit
	GB-1561	Cyl 1	Vibration peak values	X98451		m/s^2

FIGURE 3.17 Segmented vibration signal with two-stage threshold setting. (*Prognost Systems GmbH, Rheine, Germany*)

approaches are commonly used to monitor packing wear: (1) measurement of leakage gas flow and temperature, and (2) *p–V* diagram analysis.

Gas leakage flow and temperature are possible indicators of packing wear. Monitoring is accomplished by installing either a temperature probe or a flow gauge in the leakage vent. Obviously, a measurable increase in leakage gas volume or temperature is caused by packing wear or lack of tightness. Leakage gas quantity monitoring is widely used in new compressors, although temperature measurements are valuable as well. However, quantifying leakage losses facilitates economic assessment of the timing of corrective measures. Special passageways may be required to install suitable sensors. The measuring points are then either direct-connected (linked) to the monitoring system or become part of the digital process control system.

To some extent, *p–V* diagram analysis serves to analyze the condition of the packing. Leakage increases cause discrete changes in the indicated pressures and the instantaneous timing of the attendant pressure excursions. Figure 3.18 shows the pressure behavior on the crank-end side of the cylinder, one with tight packing and the other with leaking packing. The changes in the area of reverse expansion are not difficult to identify.

Piston Rings and Rider Bands To determine wear of the piston and rider bands (occasionally called *rider rings*), a *rod drop analysis* is widely used. This involves the piston rod drop being measured continuously by a proximity sensor. In the course of its life time, piston ring wear leads to measurable piston rod drop. For this purpose an eddy-current probe (induction

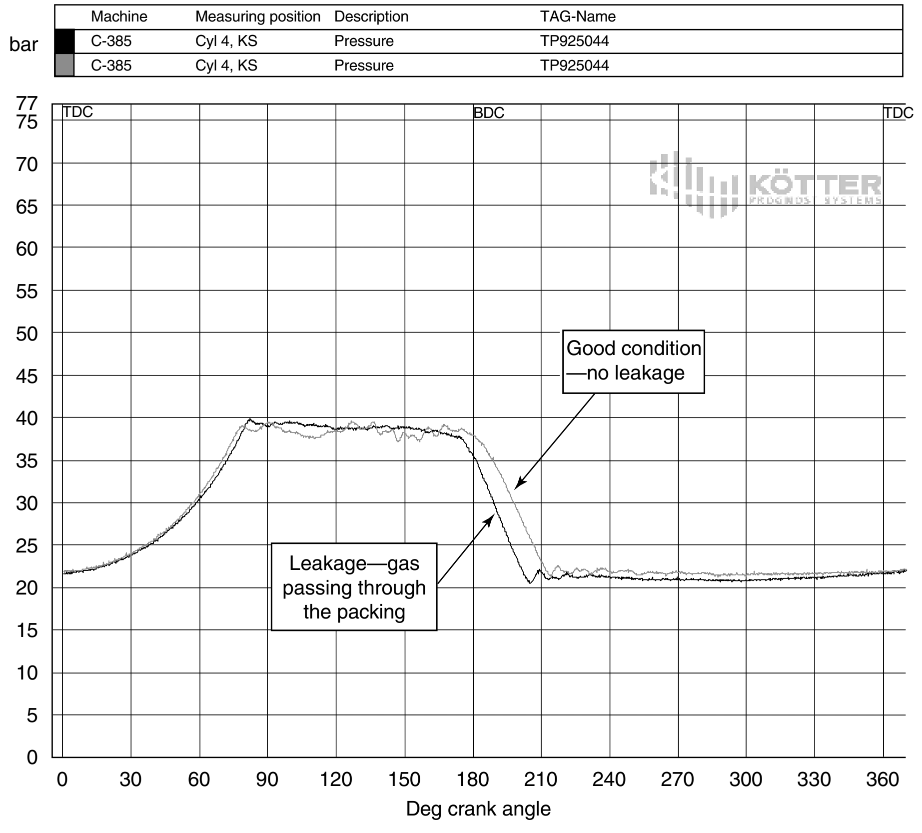
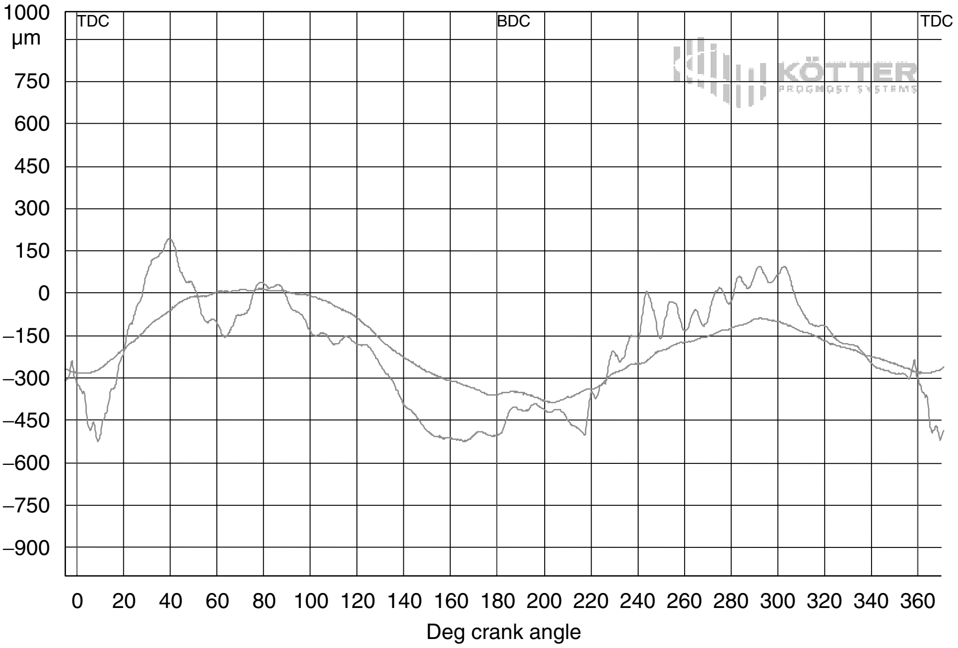


FIGURE 3.18 Effect of leaking packing on the pressure course indicated. (*Prognost Systems GmbH, Rheine, Germany*)

proximity sensor) is fitted to the packing. For signal analysis, individual interval values at specific points on the piston head are measured. Also, specific signal segments can be interrogated as needed or analyzed over the entire signal range. These analyses are particularly useful to spot interruptions caused, for instance, by particles or lubricant residue on the piston rods. Once identified as such, the total signal content can be further interpreted and its relevance assessed. In addition, when there is damage to the piston rod connections to the piston or to the crosshead, information can be gained from the total rod drop signal as the piston rises. Figure 3.19 contrasts the course of a rod-drop signal in good condition to the signal when the piston rod–crosshead connection is broken. Due to the loose connection, the position signal shows strong vibrations in certain areas.

Determining rider ring wear requires monitoring of the measured values over a long period of time in the form of a trend. Figure 3.20 represents the trend values of four segments of a piston rod analysis over a period of about 10 weeks. The values shown give the distance measured from the piston rod to the sensor in the following crank angle areas: 0 to 10°, 80 to 90°, 170 to 180°, and 350 to 360°. The exponential increase in wear is clearly identifiable.

On inspection, the piston rings were found to be badly worn. A more detailed study of the cylinder lubricant (see Fig. 3.14 with 5.1% of the causes) found a partially blocked lube oil inlet pipe. The reason for the particularly rapid rod drop could thus be linked to inadequate cylinder lubrication.



	Machine	Measuring position	Description	TAG-Name	Value	Unit
	V-760	Roddrop 1	Rod drop signal 1	SN 576029		µm

FIGURE 3.19 Comparison of normal rod drop signal with a distorted curve due to a loose connection between the crosshead and the piston rod. (*Prognost Systems GmbH, Rheine, Germany*)

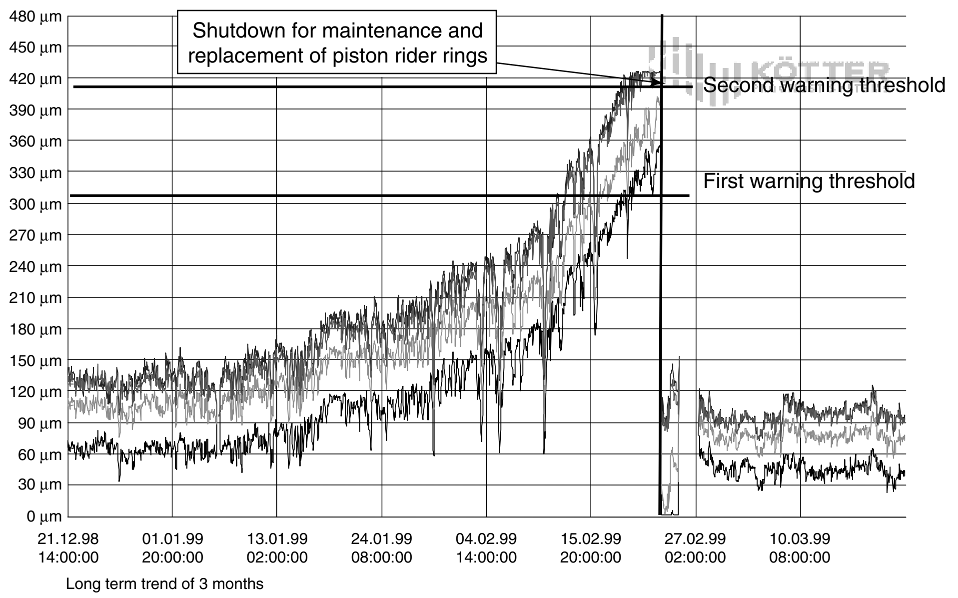
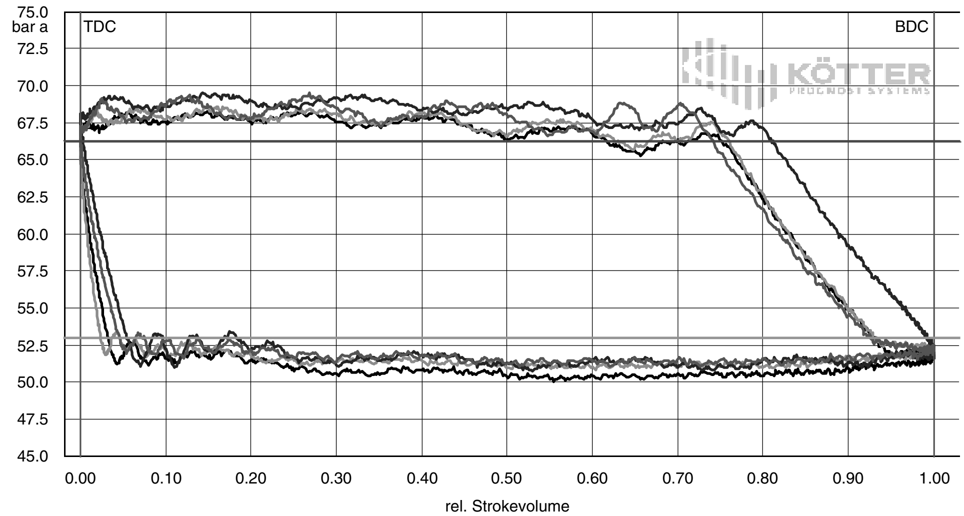


FIGURE 3.20 Rod-drop values of segments 1, 9, 18, and 27 (1-hour average) within the last eight weeks prior to the shutdown for replacement of the worn rider rings. (*Prognost Systems GmbH, Rheine, Germany*)



	Machine	Measuring position	Dataname	TAG-Name	Value	Unit
	K004	Cyl. 1, HE	p-V-diagram	03PI201		bar a
	K004	Cyl. 1, CE	p-V-diagram	03PI200		bar a
	K004	Cyl. 2, HE	p-V-diagram	03PI203		bar a
	K004	Cyl. 2, CE	p-V-diagram	03PI202		bar a
	K004	SP Cyl. 1,HE	suction pressure Cyl. 1, HE	03PC305	52.9	bar a
	K004	DP Cyl. 1,CE	discharge pressure Cyl. 1, CE	03PI304	66.2	bar a

FIGURE 3.21 *p–V* diagrams of a one-stage two-cylinder compressor with one defective valve unloader. (*Prognost Systems GmbH, Rheine, Germany*)

Valve Unloaders Throughput or capacity variation of reciprocating compressors is important where production adjustments and energy savings are of greatest interest. Although the reverse flow principle of capacity control has been known for decades, innovations in valve unloaders and suction valve design made it practical to use this control mode for reciprocating compressors.

Effective function monitoring of these control devices is possible with *p–V* diagram analysis. Measuring the time rate of pressure change reflects cylinder loading or cylinder capacity as a function of time. If *p–V* diagrams are analyzed or compared in conjunction with crank angle position, any control errors can be quickly spotted and associated with a given cylinder. Figure 3.21 shows the four *p–V* diagrams of two dimensionally identical double-acting cylinders equipped with valve unloaders. The lifting device on one of the cylinders is obviously not working properly. In many instances more information on the proper adjustment of a lifting device can be gained from vibration monitoring at the cylinder.

Running Gear A number of monitoring methods are employed to determine the condition of compressor running gear components. As a rule, temperature measurements are used for wear monitoring of crankshaft and connecting rod bearings. Temperature probes and their installation are not associated with high costs. These probes or transducers are frequently supplemented by measurements of lube oil flow and temperature.

Traditional means of vibration monitoring are employed to gauge the overall mechanical condition of reciprocating compressors and to effect precautionary shutdowns so as to avoid consequential damage. To that end, vibration sensors or accelerometers are mounted in the crank or crosshead slide areas. Essentially the same technical requirements pertain to sensor frequency resolution and measuring modes as were outlined earlier for cylinder vibration measurements.

In analyzing vibration processes, segmented analyses have proven their worth in terms of rapid diagnosis. This means that vibration peaks (e.g., at characteristic load change points) can be monitored effectively. Figure 3.22 shows the measured vibration peaks in the area of the crosshead slide that occur at the time of rod load reversal, typically when there is increased bearing play in the crosshead bolt.

To avoid consequential damage, it is important to conduct continuous real-time analysis and quickly alert operating staff or shut down the machine. These requirements are met by special continuous monitoring, which subjects the vibration signals of each crank revolution—individually and segmented—to a threshold check (safety monitoring). In addition, frequency analyses (FFT) of specific signal sections (e.g., of the load exchange areas) can provide information on structural and mechanical changes.

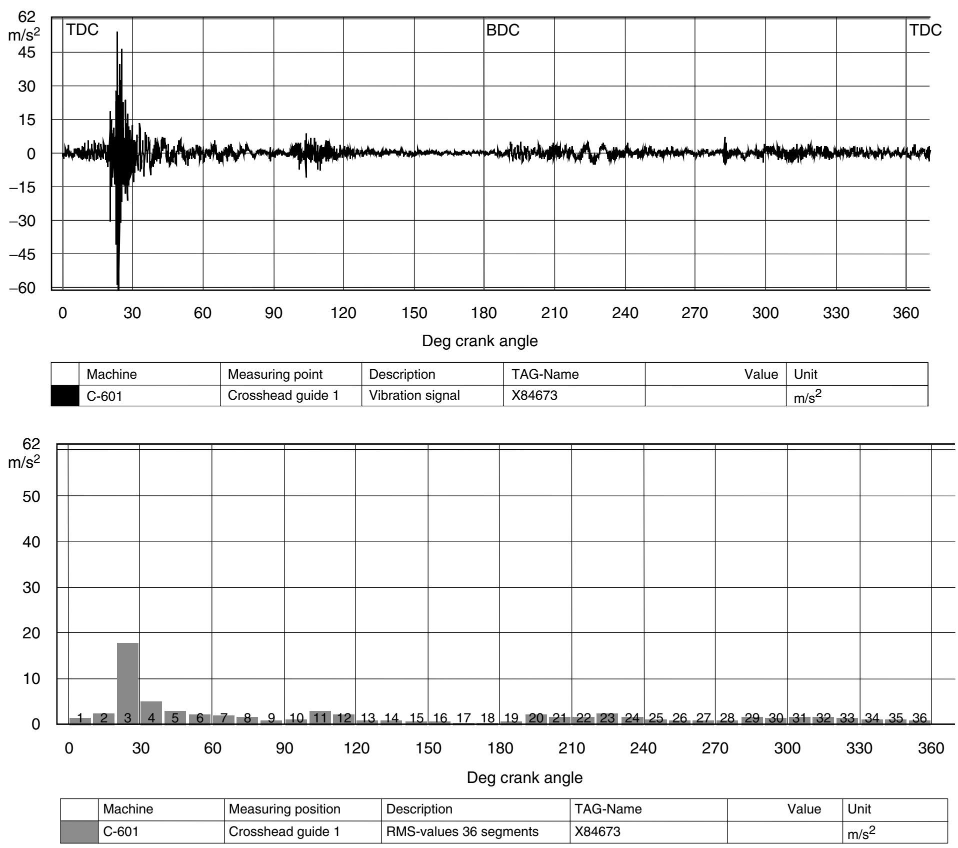
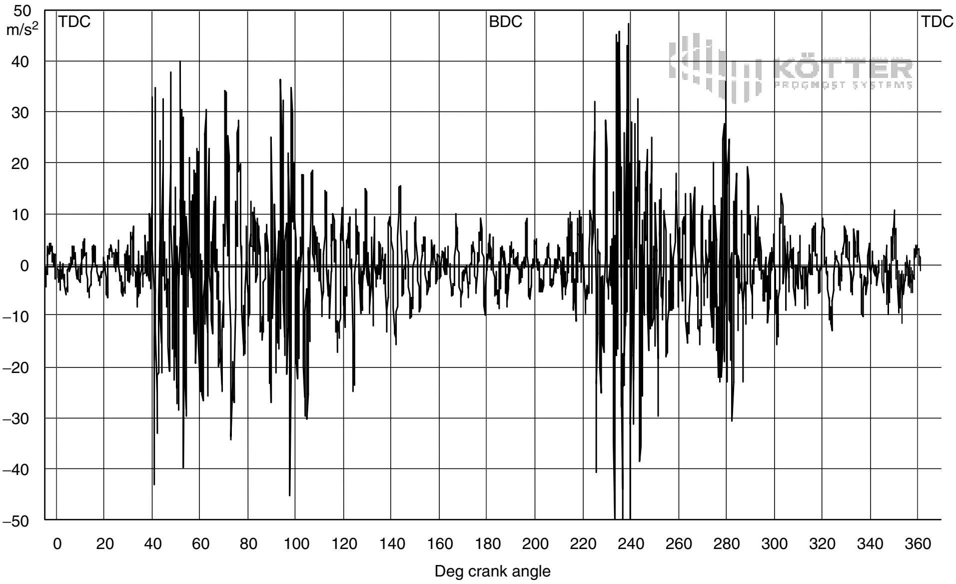


FIGURE 3.22 Measured vibration and related segmented RMS values showing a peak at the rod load reversal. (*Prognost Systems GmbH, Rheine, Germany*)



	Machine	Measuring Point	Description	TAG-Name	Value	Unit
	HDS-421	Crosshead guide 1	Vibration signal	HDS411		m/s^2

FIGURE 3.23 Vibration signal measured at a crosshead slide with damage at the crosshead pin bolt lock. (*Prognost Systems GmbH, Rheine, Germany*)

Figure 3.23 shows the vibration signals of a four-crank horizontal opposed compressor measured on the crosshead slide that triggered an alarm. Inspection of the machine revealed a defect in the crosshead bolt that had already destroyed the pin bolt lock. With the rapid alarm it was possible to avoid the crosshead bolt moving out and causing substantial consequential damage to the crank mechanism.

The Present and the Future Experience in operating reciprocating compressors has highlighted the focal point of maintenance work. Above all, the main areas for using condition monitoring center on seal components (valves, packings, piston rings) and the machine running gear. Methods and systems for analysing the condition of these components have been known and tested for many years [2,3].

Certain analytical methods, such as *p–V* diagram analysis or vibration analysis, make it possible to assess several components with one recorded signal. But to get a more precise understanding of the condition of one group of components, other measurement methods must supplement the analysis. The real-time capture of transient, instantaneous, and rapidly changing vibration or more slowly progressing changes in temperature is being pursued. Integrating these condition-based values in a modern computer-linked monitoring system now offers the maintenance team a single analysis and diagnostic interface.

Systems such as the Prognost-NT have served the reciprocating compressor market since the early 1990s and have done very well. As time progresses, mathematical models will be used increasingly to display the mechanical and physical processes of large reciprocating compressors. Even more detailed error recognition and diagnosis systems will be possible, and a variety of technical options will present themselves to the user.

REFERENCES

1. Ulrich Klein, *Schwingungsdiagnostische Beurteilung von Maschinen und Anlagen* [Vibration diagnostic assessment of machines and plants], Verlag Stahleisen GmbH, Düsseldorf, Germany, 1998.
2. Stephen M. Leonard, *Increasing the reliability of reciprocating compressors in hydrogen services*, presented at the NPRA Maintenance Conference, New Orleans, L., May 20–23, 1997.
3. Johannes Nickol, *Kolbenkompressoren in Prozeßanlagen: 24.000 Laufstunden Verfügbarkeit, Utopie oder Realität?* [Reciprocating compressors in process installations: availability of 24,000 hours, utopia or reality?], presented at the First EFRC Conference, Dresden, Germany, Nov. 4–5, 1999.