SAFETY IN COMPRESSOR CYLINDER DESIGN AND APPLICATION

R. M. Deminski & F. J. Hunter
The Cooper-Bessemer Corp.
Kansas City, St. Louis

Experience has shown that theoretical, empirical and experimental methods of stress analysis must be used in the design of compressor cylinders to insure their freedom from mechanical failure. Each method is related to the other and the developments in one method are continuously used to adjust and make more accurate the results obtained from the other methods. The theoretical methods involve the use of many of the accepted formulas of stress analysis. The empirical methods use formulas and calculation methods that have either been developed or adjusted by results obtained through experimental stress checks. The final and most reliable tools in stress analysis involve the use of test bar results, strain gage tests and, most important, fatigue tests.

The act of designing

The designer, in considering the aspect of stress in compressor design, will be guided by the values of tensile and yield strengths made available by laboratory checks on test bars. These values, along with the proper safety factors, are used to develop allowable theoretical stress levels for a given material. With the general configuration of the cylinder being dictated by the purpose for which the cylinder is to be used, the thickness of the various sections of the cylinder will be determined with the aid of the theoretical and empirical formulas and the allowable stress levels previously described. After the design is completed, a complete and thorough stress analysis is made prior to the manufacture of the cylinder. At this stage we are still relying on the ability of the designer to transfer his previous experience to his empirical calculations and to properly interpret the results he obtains.

The next step is to build the cylinder and its components and subject them to various laboratory checks that will indicate the accuracy of the design with regard to stress.

Fatigue test

The most reliable laboratory check is the fatigue test. In this test a full scale and complete compressor cylinder assembly is charged with oil; and with the aid of proper valving, controls and pumps it is subjected to cyclic type loading. Simultaneously, the maximum cycle pressure is maintained at a constant in the discharge passage, while the suction passage is maintained at atmospheric pressure. The cyclic load varies between zero and the pre-selected maximum at a rate of 400 cycles per minute. The initial maximum value is determined by stress analysis. If no failure occurs in six million cycles, the maximum pressure level is raised and the test is repeated. This process continues until a failure occurs in less than six million cycles. When this point is reached the endurance strength is known. A suitable safety factor must be applied to the determined endurance strength to arrive at an allowable working pressure for the cylinder. Should the allowable working pressure be less than that originally designed for, then a redesign and retesting is necessary.

Strain gage testing

On smaller items that do not lend themselves to fatigue testing, strain gage testing proves to be a valuable tool in determining all working stresses. The stresses thus determined may be compared to those obtained from empirical formulas and the empirical formulas may then be adjusted to make future stress calculations on similar items more accurate.

Safety in specialized work

The considerations of safety and reliability take on added meaning when applying compressors to specialized process work. The compressor must be capable of handling the process gas without danger of explosion, or other hazard, and it must reliably produce a gas of desired purity. Attempts to meet with these requirements have led to the birth and development of the non-lube type compressors for virtually every gas that has ever been compressed.

The first non-lubricated designs were based on the use of carbon sealing and bearing elements and were used for the purpose of compressing hydrogen. These cylinders have been developed over the years to the point where trouble-free compression of hydrogen is assured, and without contamination or catalyst poisoning by lubricants. This construction was gradually applied to other gases, but not with universal success.

Initial attempts to resolve the short life problem were centered around the use of special forms of carbon materials and particularly filled carbons. While some of the materials produced longer life than the materials presently considered as standard, the cost generally made their use impractical. With a very complete study of materials nearly completed, variations in design were investigated. Principally, variations in sealing arrangements with carbon parts were considered.

This attack has yielded the first significant increase in performance and oddly enough at a reasonable cost level. These increases are not under closely controlled and ideal laboratory conditions, but under the rigors of actual field operation. They are being obtained from the operation of a 4,000 psi, five-stage air compressor that has a first stage suction of 43 psia. Life expectancy of the carbon parts at the 4,000 psi level is continuously increasing. From the data obtained from the other four stages of this machine, life expectancy at the discharge pressure levels of each stage can very well be predicted for other machines.

Teflon now used

While the explorations continue in the field of carbon for more reliable materials and designs, the application of filled Teflons and the basic mechanics of non-lubricated compression in general have come under very close scrutiny. Filled Teflons, unfortunately, were initially used as a panacea, a general cure-all for all non-lubricated operations. As might be expected, many failures resulted. Some of the failures resulted from improperly processed materials, some from improper engineering, and still others from attempts to use the material beyond its physical capability. These reasons can be grouped into the classification of inexperience.

Film theory

To develop a more accurate approach to future non-lubricated material selections, a study of the mechanics of non-lubricated operation was launched. The most popular theory advanced to describe the action that occurs between the metallic surfaces and the piston rings wear rings and packing rings is that of the film theory. This theory proposes that the sealing and bearing elements, during break-in, deposit a film on the metallic surfaces which then permits a material of low friction to run against itself. Another theory, with some analytical background, proposes that as the seal and bearing materials wear, they and their filler materials and/or the base materials present in the cylinder liner will react with each other and/or the gas being handled to produce chemical compounds that will exhibit qualities that may be either abrasive or lubricating in nature. Test results have indicated to some extent that this theory may be valid.

Minimal lubrication

Regardless of the reason why a non-lubricated cylinder operates, it is generally conceded that the cost and maintnenance of non-lubricated equipment is high. Although in some cases the safety feature of this type of operation justifies the high cost, there are those processes where lubricants may contaminate the product, but do not constitute a hazard. In these instances it would be well to consider the various materials that are available for use with minimal lubrication. Minimal lubrication, depending on the conditions of compression, may take one of several forms. It may be gained directly by the reduction of lubricator output or by the carry-over of lubrication from previous stages, from packing lubrication, from process piping and from the crankcase when direct lubrication has been removed from the cylinder. The piston rings and packing rings that are used with these various forms of minimal lubrication are fabricated from various low-friction type materials. An example of one such material, one that has seen fairly extensive use, is molybdenum disulfide impregnated Bakelite. Several articles have been published by P. J. Chandler of the Arkansas Louisiana Gas Company describing the use of this material.

The dangers

While many benefits may be derived from minimal lubrication, the effect of any adverse operating conditions is generally magnified when changing from full lubrication to minimal lubrication. A temporary but abnormal liquid carry-over could destroy the extra thin oil film and thus cause scoring of liners and rods. Carry-over of dirt into the compressor is more serious since there is generally insufficient oil to flush the unit clean. A broken valve may induce temperatures sufficiently high to vaporize or carbonize the small amount of oil present which may also tend to score liners.

The purpose of plastics, carbons and other related materials in non-lubricated compressor cylinders is to provide a material with good sealing and wearing characteristics that will not contaminate the gas stream or cause a hazard of any type. Perhaps this same goal can be attained through the use of more conventional materials lubricated with liquids, powders or other elements that would be compatible with the gas and the process in general. The most obvious example would be the use of water as a lubricant.

Oil separation

While we have mentioned non-lubricated and minimal operation, and also suggested other forms of lubrication that may be worthy of investigation, we feel that normal methods of lubrication with oil separation equipment should also be considered as a very suitable alternate for oil-free process gas. In many instances the presence of oil during the compression process is neither a hazard nor a contaminant. If this oil can satisfactorily be removed before the gas passes on to a critical area in the system, then no harm is done. This, for example, is the case if the oil can be removed from the ammonia synthesis gas prior to coming in contact with the catalyst.

While the mechanical and efficiency considerations of oil removal equipment are beyond the normal scope of compressor design considerations, the cost of oil removal equipment must be weighed against the premium cost of non-lubricated equipment and the additional maintenance that is generally associated with it. There exists today equipment that is capable of removing from a gas stream 100% of all oil particles above 3 microns in diameter and 99.6% of all particles below 3 microns. Equipment of this type used in conjunction with an activated carbon or similar filter should yield a high oil removal efficiency provided adequate attention is given to the lubricating oil characteristics. A high MW oil that has been vacuum treated to eliminate the light ends suggests itself.

Gas leakage problem

Very often associated with non-lubricated hydrogen compressors and in some cases with lubricated hydrogen and helium compressors, is the problem of leakage of a hazardous or valuable gas. Leakage of these gases is a particular problem because of their low molecular weight. To protect against such leakage, when it is impractical to use copper gaskets, special composition gaskets have been developed. When required, each cylinder produced for service with one of these gases is subjected to a leakage test after a normal hydrostatic test. A standard test procedure is used when a customer specifies no particular method. In most instances, where size permits, the cylinder in question is submerged in water and then charged with helium to a pressure level that is 110% of the normal discharge pressure (Figure 1). An alternate method relies on a check for leaks around gasket joints and bolts with soap suds after the helium has been admitted to the cylinder. While both methods are 100% reliable, the exact location of a leak, should one occur, is more readily located with the latter method.

Operating environment

Having studied some of the physical and mechanical considerations of compressors that affect safety and reliability, next look at the influence of operating environment: By environment is: 1. The physical condition of the gas as it reaches the compressor cylinder; 2. The manner in which the gas enters and leaves the cylinder; 3. The conditions imposed upon the cylinder itself.

The ideal condition of a gas in a normal lubricated compressor cylinder would be described as clean and dry. This is a condition probably attained only in a closed loop operation, or in a high purity process stream. An example of such would be a refrigeration loop in which the same gas is continuously circulated, or an oxygen compressor where the gas, by necessity, is pure. In a process plant, particularly during the early months of operation, dirt that has been trapped in piping and the overall system during installation causes the gas to be less than ideal. Similarly, wear and corrosion particles are sometimes carried through a cylinder from some part of the process system and once again renders the gas unfit for the cylinder. A particular problem with dirt is noted in the air compressor. With the exhaust of process systems and other matter that contaminates the atmosphere in the region of the air compressor inlet, the free air entering this machine, even with filtration, is not ideal. It is neither ideal from the abrasive standpoint nor from the safety standpoint if this machine is used as a primary compressor in a separation plant.

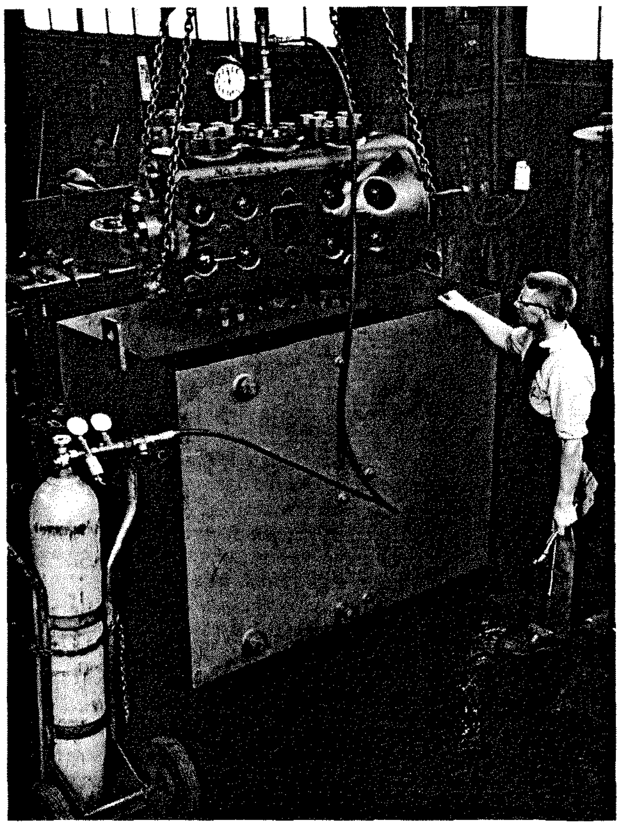

Figure 1. Where size permits the cylinder is submerged in water and then charged with helium.

The matter of a dry gas is major consideration in evaluating cylinder lubrication. Excessive liquid carry-over not only flushes lubricants from the cylinder, but it will also account for valve problems and in extreme cases cause cylinder failures. Valve breakage in some synthesis gas circulators has been traced to excessive liquid carry. This problem was remedied through the simple expedient of added fixed clearance to the cylinder. This clearance makes it possible for the cylinder to purge itself of these liquids over a period of several strokes rather than in a single revolution of the crank. Thus, at no time is the piston attempting to force an excessive quantity of liquid through a discharge valve. Generally then, we must consider that a compressor cylinder is neither handling a clean nor a dry gas.

Even gas flow

It would be ideal for the compressor industry, in general, if all cylinders were assured of a constant, even gas flow, in and out. Industry is beginning to recognize this need as evidenced by the increasing number of installations that are being equipped with suction and discharge "bottles" located close to the compressor cylinders (Figure 2). There is no doubt that smooth gas flow requires less horsepower and maintenance.

The conditions imposed upon a cylinder itself is an awkward term intended to encompass anything from the conditions of compression for which the cylinder is designed to lubrication, cooling-water flow and other conditions.

Start with conditions of compression. The condition of primary importance that is imposed upon a compressor cylinder is the discharge temperature. Discharge temperature is, first of all, an indication of the work that is being done within the cylinder on the particular gas being handled. There are 34 in. diameter cylinders that have a successful history of operation with summer discharge temperatures that run as high as 375°F ... 24 hours a day. Where special situations may warrant, there is no reason to be disturbed by discharge temperatures in this range. On the other hand, with normal lubricating oils, it is better to limit the discharge temperature to something on the order of 300°F to 315°F. At least one lubricating oil manufact-

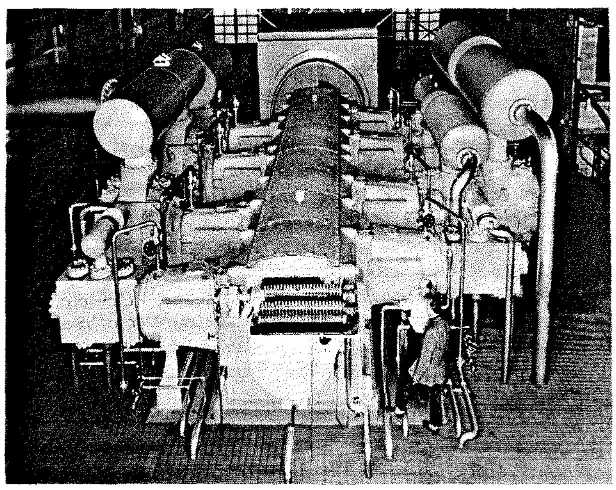

Figure 2. Increasing numbers of installations are being equipped with suction and discharge "bottles" located close to the compressor cylinders.

urer says that 40 weight oil does not lubricate above that temperature. Some installations bear out their contention. The available supply of cylinder cooling water is closely related to the highest discharge temperature that can be handled—and not on the rule of thumb, Btu/hp basis that was the basis of jacket water quantity computation in the past. Cooling water recommendations are now based upon heat rejection figures that are determined from the difference between the average gas temperature and the temperature of the cooling water to the compressor jackets. Depending upon this difference our experience indicates that the Btu/bhp hr. heat rejection can vary from below 200 to something over 700. Water-cooled type compressor cylinders can be operated with dry jackets where the discharge temperature does not exceed 140°F providing the adiabatic temperature rise in the cylinder is not excessive. In situations where the discharge temperature does not exceed 250°F, where the adiabatic temperature rise is reasonable, and where the cylinder does not operate unloaded for more than very brief periods of time, the jackets can be filled with a suitable liquid and vented without circulation of the liquid other than that which may occur as a thermal effect. The incoming cooling water temperature should be maintained at least 10°F above the condensing temperature of any constituent of the gas being handled.

If the maximum discharge temperature of the compressor cylinder is limited to, say, 315°F, automatically a limit is set upon the ratio of compression. Assuming air or ammonia and a suction temperature of 100°F, the maximum Rc is 3.24; with a suction temperature of 90°F, it is 3.97.

The compression ratio for a given cylinder is further limited by the allowable rod loading of the compressor frame. Rod load is a function to the differential pressure and the size of the piston. Rod load is, therefore, a factor in limiting ratios per stage or increasing the number of cylinders per stage for large volumes of gas at low-pressure levels, or for what we might call reasonable volumes at high-pressure levels. Recent developments, such as rolled-thread design are permitting higher rod loads for a particular compressor frame.

Recommendations

Most reciprocating compressor cylinders that are in operation today are oil lubricated. Recommendations for compressor cylinder lubrication become more detailed as the years go by due both to new service conditions and new lubricating oils. For compressor break-in, it is usually recommended that a mineral oil with 5% to 8% acidless tallow and a viscosity of 100-120 SSU at 210°F. be used. Generally, for discharge temperatures of 300°F and below, our recommendations center around SAE 30 and SAE 40 oils of naphthenic or mixed base of the type that produces loose and flaky carbon. This recommendation is modified depending upon the type of gas, its condition and the service. For instance, due to the presence of moisture, it is recommended that air compressors use an oil containing 3% to 5% acidless tallow. This can be a blend. If acid is present in the air, a corrosion inhibitor should be added.

High temperature discharge

In air compression service, where the discharge temperature exceeds 300°F, it is recommended that a fire resistant, synthetic cylinder lubricant that has suitable viscosity characteristics for lubrication at the maximum discharge temperature be used. There are some fifteen or more of these lubricants on the market. Monsanto Pydraul AC and Celanese Corporation's Cellulube 220 are two that we have seen used successfully. A word of caution ... this type of lubricant is an excellent paint remover and will also attack some types of gasket material. In addition some, if not all, synthetic lubricants are slightly toxic and should be used accordingly.

For synthesis gas service it is generally recommended that an SAE 30 or 40 oil as previously described be used. Some operators feel that they secure better results in high pressure stages by compounding with a turbine oil containing some acidless tallow. Experience in one plant showed where operating cylinder discharge temperatures on the hyper cylinders exceeded 325°F an approximate 60 weight straight turbine oil with about 5% tallow proved to be the most satisfactory lubricant.

Recommendations as to operating (as opposed to break-in) quantity of oil for a double-acting cylinder begin with the formula:

$$GPD = \frac{\text{Bore (inches)} \times \text{stroke (inches)} \times \text{RPM}}{385{,}000}$$

and this is exclusive of the oil required for rod packing. The rod packing is rgearded as a separate cylinder with the rod diameter as the bore.

Conservative recommendations

For the benefit of the plant operators who desire to hold the quantity of oil in the gas to a minimum, it is known that after the cylinder glaze has been established the quantity from the above formula can be cut back considerably, but the minimum amount must be determined from practical experience in the particular plant. There is at least one plant where no cylinder lubrication is used on the hyper stage. The stage goes through break-in with conventional bronze piston rings and packing to produce the glaze. The unit is then shut down, Teflon rings installed, and the cylinder lubricant is cut back gradually to zero. We wish to point out that after restart, in this particular instance, however, the packing is lubricated and there is a migration of some lubricant into the cylinder. The particular cylinders are operating at the 5,000 psig discharge pressure level.

In this particular non-lubricated cylinder approach, the setting of the glaze is the most important factor. Generally speaking, however, some lubrication is important for longest continuous duty operation of any compressor cylinder. Under conditions imposed upon a compressor cylinder, rotative and piston speeds must also be considered. Both are increasing. While these factors are relatively fixed, and to a certain degree inflexible, it may be well to realize that their level is to a great extent dictated by the compressor user. As attempts are made to handle a given horsepower on a smaller frame and with fewer cylinders, piston and rotative speeds are driven upwards. The resulting savings in floor space is in some cases offset by increased maintenance and down time.

Not too many years ago 700 to 800 feet per minute piston speeds were sort of a rule of thumb for synthesis gas and air compressor cylinders and as low as 400 feet per minute for recirculators. Today synthesis gas units are operating with piston speeds as high as 900 feet per minute, recirculators as high as 700 feet per minute. Depending upon stroke, rotative speeds vary from 250 rpm to 450 rpm for synthesis, air and ammonia refrigeration units. Higher rotative speeds are used for other gases, but generally these are limited to compressors below 1000 hp.

DISCUSSION

Anonymous—It is common practice to get gases under pressure from the low temperature plants at lower-than-normal temperature. I would like to know what is the level at which this lower temperature can be compressed with safety.

HUNTER—Cooper-Bessemer: The answer to that question, I think, would almost involve a paper in itself. One of the programs that is under way involving low temperature suction to compressor cylinders is the proposed liquefied methane program. The big question is the one of the viscosity of low temperature oils that are available. Most of the low temperature oils that are available today are satisfactory to about a minus 50° suction temperature on normal service. However, there is also the question of a design temperature versus an operating temperature. When we are operating at a design temperature of a minus 50°, we can't overlook the possibility that during actual operation the temperature may be pulled down below that. Now the lubricating oil people have come up with an oil that is supposed to be good for temperature down to minus 100°F. This may be a solution to even lower temperatures. As for the materials of construction, we will go to minus 50° and possibly below with cast iron. Of course, beyond that you get into the area of nickel irons. I think that for any specific application, you will have to look at the gas and determine precisely how far you want to take it. Generally, for a conventional low temperature oil, I would like to see the temperatures limited to say a minus 30°. This allows you an extra 20° so that particularly on a refrigerating cycle, if the temperatures happen to vary a little bit, you are not in immediate danger.

BUDDENBURG—Collier Carbon and Chemical Corp: In your section on oil removal, you described what I presume is an agglomeration type unit for oil removal. Then in your lubrication section, you advised the use of tallow with the lubricant. We have some reason to believe that these are not compatible recommendations. We tried the tallow, at the recommendation of your Los Angeles agent, and we found that the efficiency of our oil removal dropped off by about eighty percent and it was restored only when we took the tallow out.

HUNTER—Cooper-Bessemer: I have no doubt that you have problems along that line. There may be oil removal equipment capable of handling the tallow, but I am not enough of an expert on that type of equipment to be qualified to pass comment on it.

SOMMERS—Pennsalt Chemicals: I am sure almost everybody here, at some time or another has had trouble with broken piston rods. In my own experience, I guess I know perhaps of about a dozen that have broken from time to time and, in most cases, the fracture always was of such a nature to indicate that the material was hard and brittle, not at all similar to the condition that existed at the time it was installed. I would like to ask the various compressor manufacturers who are present here what particular types of steel is recommended to stand up over a long period of time on piston rod service?

DEMINSKI—A particular material would, of course, depend on the pressure level at which you are operating. Our material for piston rods, generally up to 4,000—6,000 psi level is a 1045 steel and we attain probably a forty-five to fifty-five Rockwell 'C' hardness only in the area of the rod where the piston rod packing rings will ride. The area of the rod adjacent to the piston is left relatively soft. We feel this prevents the type of failure that you are speaking of. This is the type of material we recommend for probably 75 percent of our cylinders which would be the lower pressure cylinders.

STEVENS—I wonder if the speaker would care to say a word about removable liners in cylinders? Does he figure that they are being used as widely as before?

DEMINSKI: Our present arrangement is to use liners in every cylinder with the exception of our short stroke units. That is our 5-in. and 7-in. stroke machines. We like liners. It benefits the builder and it benefits the compressor user. It assists in delivery. With a liner, you can cast a very uniform section. At the present time, we are using centrifugally cast liners which yield a much more refined grain structure for the wearing surface. We feel there is definitely an advantage. The most obvious advantage is being able to recondition a bore by re-lining rather than replacing a cylinder.

WALKER—Would Mr. Hunter care to comment on how low they can go on suction temperature in a water-cooled cylinder provided that the compression ratio is maintained above 3.

HUNTER: In terms of water cooled cylinders, I personally get nervous with anything below freezing. I realize it can be done because the heat of compression would probably keep the water from freezing. At any time I get thirty-two degrees suction or below, I am just conservative enough that I would just as soon have ethylene glycol in the water. But otherwise, I don't know of any limitation on going down in temperature.

SOMMERS: For whatever it may be worth, I operated an ordinary Norwalk 2—stage 200 H.P. compressor many years ago for a total period of ten years, where the suction temperature was minus 50°C, and 60 psig. There were always particles of solid CO_2 snow which were carried over with the gas so the actual temperature was maintained at minus 50°C, or below. The discharge was at about room temperature. This machine, which was a standard machine, operated very satisfactorily with ordinary oil lubrication, without cooling water on the cylinder. The cylinder wear was absolutely minimum.

JONES—Air intake temperatures in winter surely are below the 32°F. that has just been mentioned, and I am sure that there are a great many people operating compressors consuming large quantities of air in such climates. I am sure that they don't usually use ethylene glycol. I find some difficulty reconciling this with the comments made by the speaker.

HUNTER: I am just indicating the way I feel about cooling water. Now, it is true air compressors are operated water-cooled without ethylene glycol in the jackets. That is a matter of preference, but glycol will protect against freeze-up with accidentally reduced water flow.