COMPRESSOR HANDBOOK

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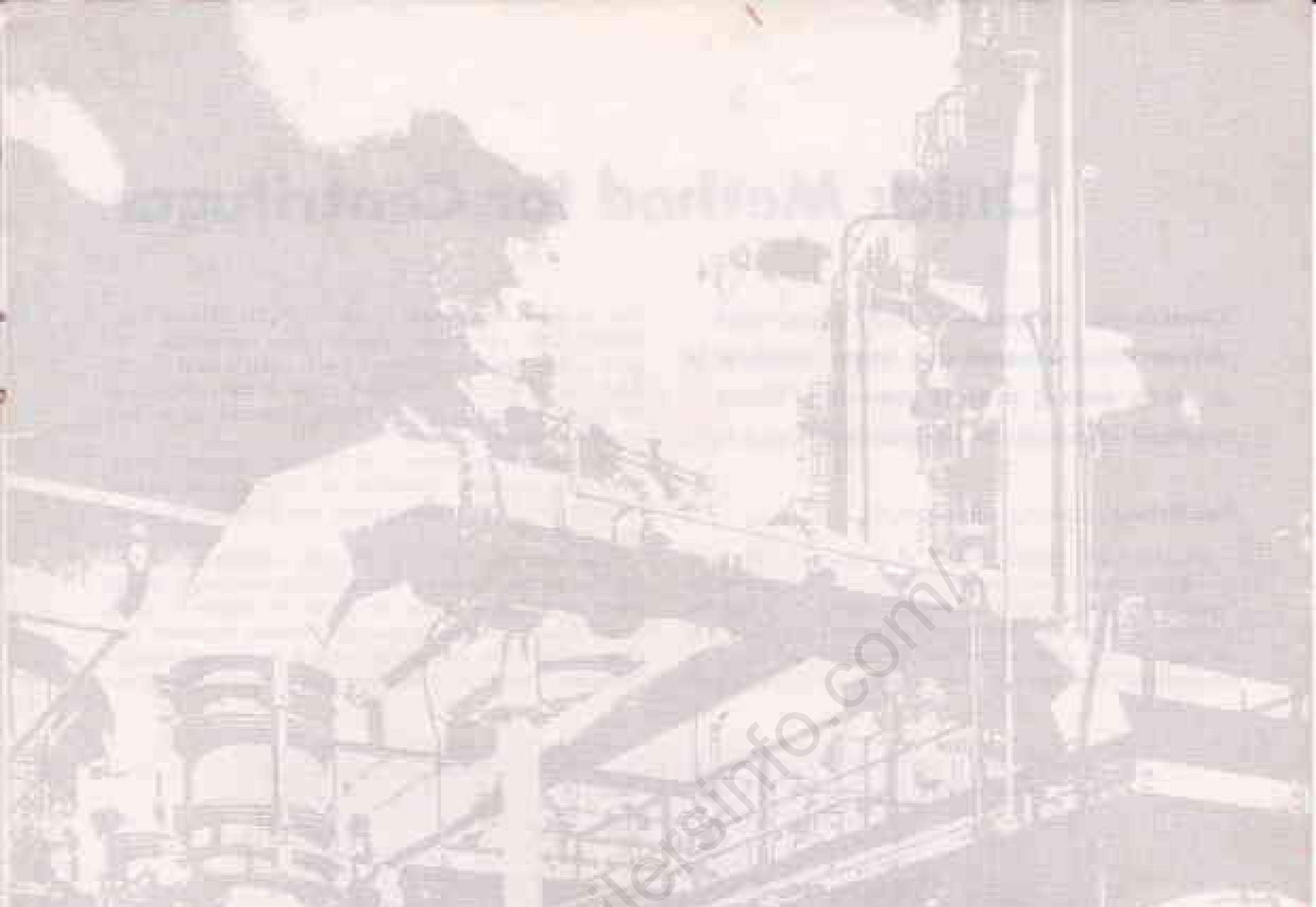

COMPRESSOR HANDBOOK

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Quick Method for Centrifugal

Centrifugal compressor size, price, and driver requirements are often needed in a hurry during project planning. Chart method streamlines estimating procedure

Don Hallock, Elliott Co., Jeannette, Pa.

IN THE EARLY planning stage of process unit projects, quick estimates of compressor size, price, and driver requirements are desirable. Typical questions asked by process engineers are: We want to increase the discharge pressure of our existing process. Would it cost more to supercharge the existing compressor with a high flow, low pressure compressor, or to "top" the existing compressor with a low flow, high pressure compressor? We plan to buy new compressor capacity and would like the flexibility of multiple part-flow units. What premium do we pay for this flexibility? Which approach lends itself best to available turbines?

The method described here has been designed to answer these questions thus filling the difficult void between conception of a compressor need and preparation of formal inquiries.

Selections can be made (largely without slide rule) for the entire range of conventional multistage equipment, by which is meant the family of machines having heavy horizontally split casings, enclosed impellers running under 1,000 feet per second tip speed, and vaneless diffusion. This type of machine has gained universal ac-

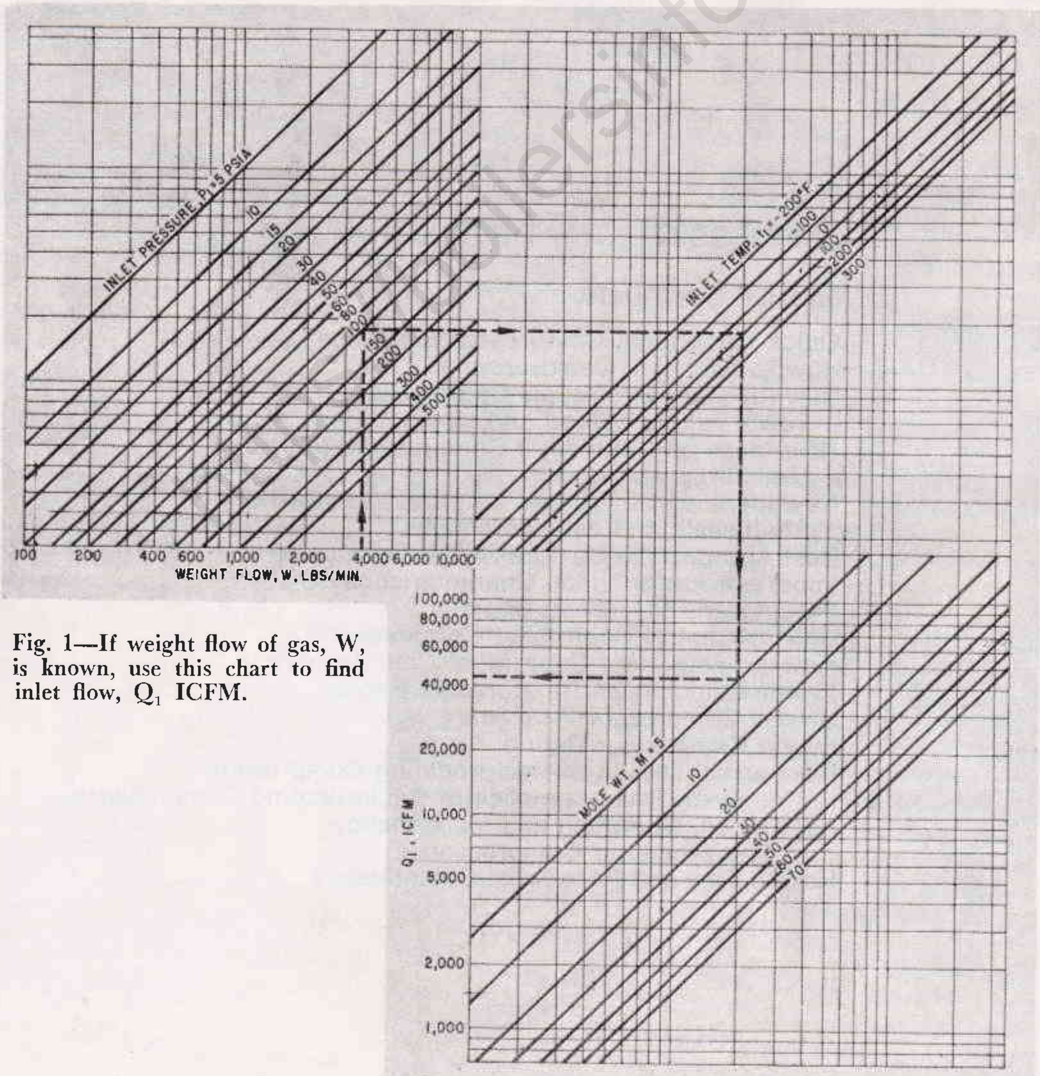


Fig. 1—If weight flow of gas, W, is known, use this chart to find inlet flow, Q_1 ICFM.

Compressor Estimates

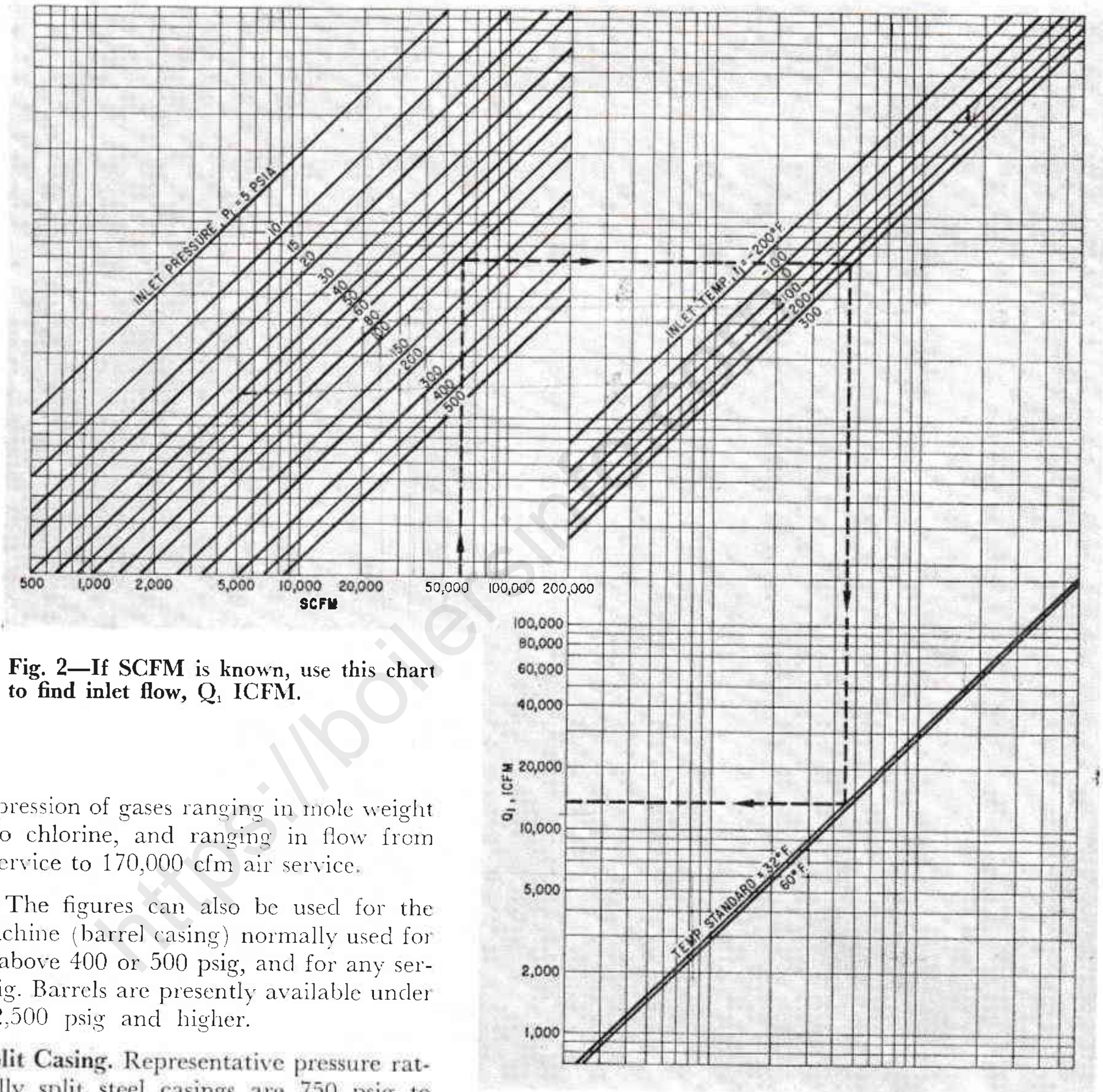


Fig. 2—If SCFM is known, use this chart to find inlet flow, Q_1 ICFM.

ceptance for compression of gases ranging in mole weight from hydrogen to chlorine, and ranging in flow from 700 cfm recycle service to 170,000 cfm air service.

Barrel Casing. The figures can also be used for the vertically split machine (barrel casing) normally used for hydrogen service above 400 or 500 psig, and for any service above 750 psig. Barrels are presently available under 10,000 cfm for 2,500 psig and higher.

Horizontally Split Casing. Representative pressure ratings of horizontally split steel casings are 750 psig to 8,000 cfm, 600 psig to 15,000 cfm, and 300 psig to 60,000 cfm. Cast iron pressure ratings are 40 to 50 percent of the foregoing steel ratings. Above 60,000 cfm, casings are generally available in cast iron only, with ratings between 50 and 100 psig.

Flow Limits. Although flow limits for centrifugals are dictated somewhat by the specific conditions involved and by the available hardware of different manufacturers, generally positive displacement equipment is indicated below 500 cfm, and above 200,000 cfm axial equipment is indicated. The flow regions from 500 to 2000 cfm can be considered a "gray area" between positive displacement and centrifugal, and from 50,000 to 200,000 cfm a "gray area" between centrifugal and axial.

Gas flows other than air above 50,000 or 60,000 cfm have not appeared until recently, but increases in plant size are now pushing flows above this figure, particularly on propane or propylene refrigeration service. Pressure limitations of large casings and physical size of oil seals are moving compressor vendors to double-flow (two gas streams in parallel), multibody selections for applications above these flows.

Selection Accuracy. The selection charts are based generally on Elliott Co. equipment, but the user will find that they apply reasonably well for any conventional machine. The charts are constructed mostly in log-log form, so that consistent accuracy is maintained over the entire flow range. Accuracy of 10 to 15 percent is attainable, sufficient for most estimating purposes.

It must be emphasized that the charts are *not* applicable for any type of positive displacement machine, for axial machines, or for the recent generation of high tip speed, vaned diffusion centrifugal equipment sometimes used for air service. Also, the charts are of necessity based on uncooled compression at constant weight flow.

To handle cooled compression (air over 50 psig, chlorine, ethylene plant feed gas), or side load compression (any refrigeration process having economizers or extraction), it is necessary to divide the total compression into a series of uncooled, constant weight flow compressions. The charts are then applied separately to each of these compression requirements. This procedure will be explored more fully after the use of the charts is described.

Use of Charts

The following quantities must be known:

1. W—weight flow in lbs. per min., or scfm—standard cu. ft. per min.
2. P_1—inlet pressure in psia
3. R_p—pressure ratio (discharge psia ÷ inlet psia)
4. t_1—inlet temp., °F
5. M—mole. weight
6. K—ratio of specific heats

Determine Inlet CFM, Q_1. If W is known, use Figure 1, proceeding through P_1, t_1 and M to find Q_1.

If SCFM is known, use Figure 2, proceeding through P_1, t_1 and "temperature standard" to find Q_1.

Determine Head *H*. On Figure 3, enter R_p and proceed through K, t_1 and M as shown. If head H exceeds 80,000 to 90,000, more than one compressor body will be required.

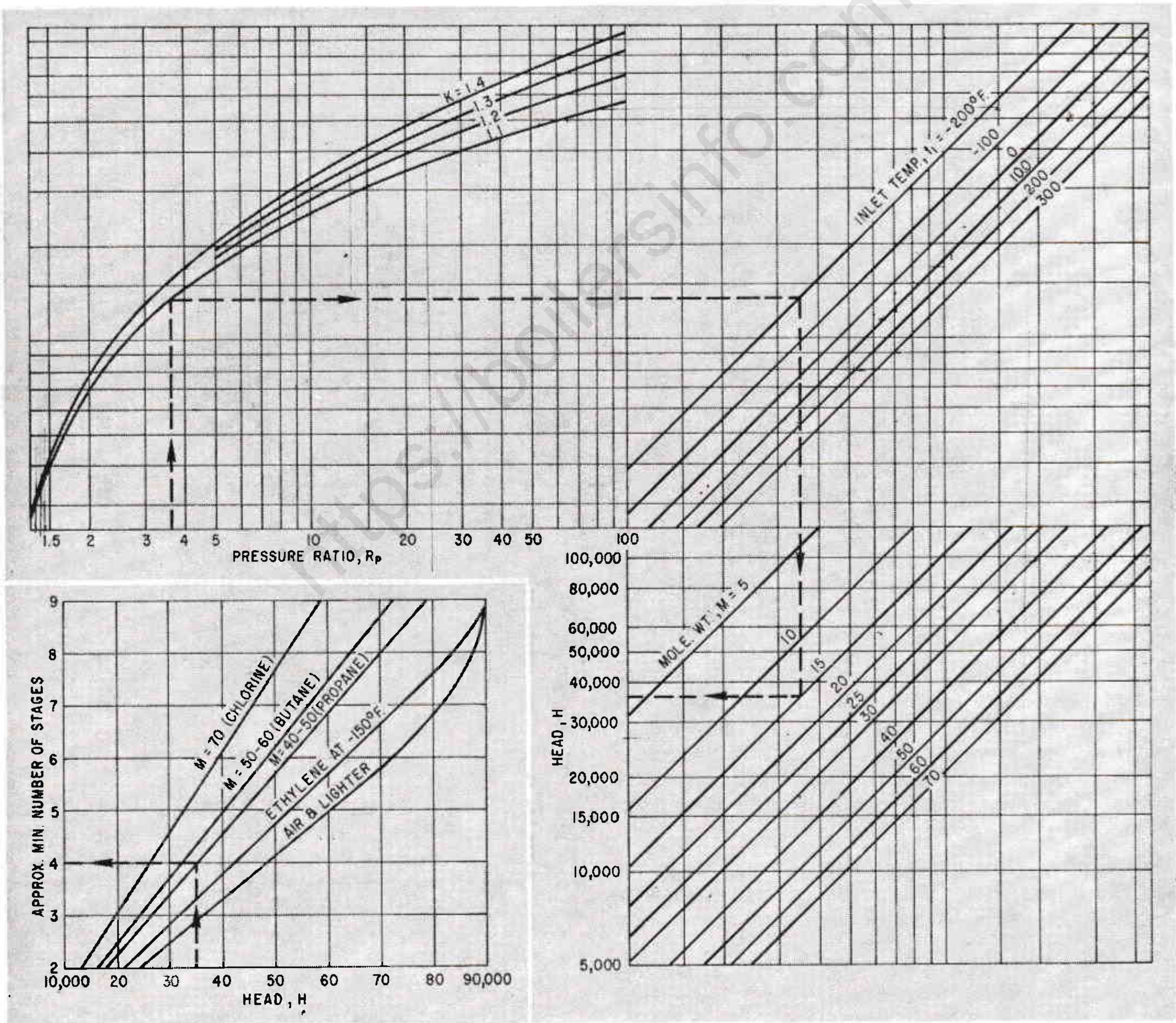


Fig. 4—Enter this chart with H found on Figure 3 to find number of stages required.

Fig. 3—Enter this chart at R_p, the pressure ratio (discharge/inlet, psia) to find Head, H.

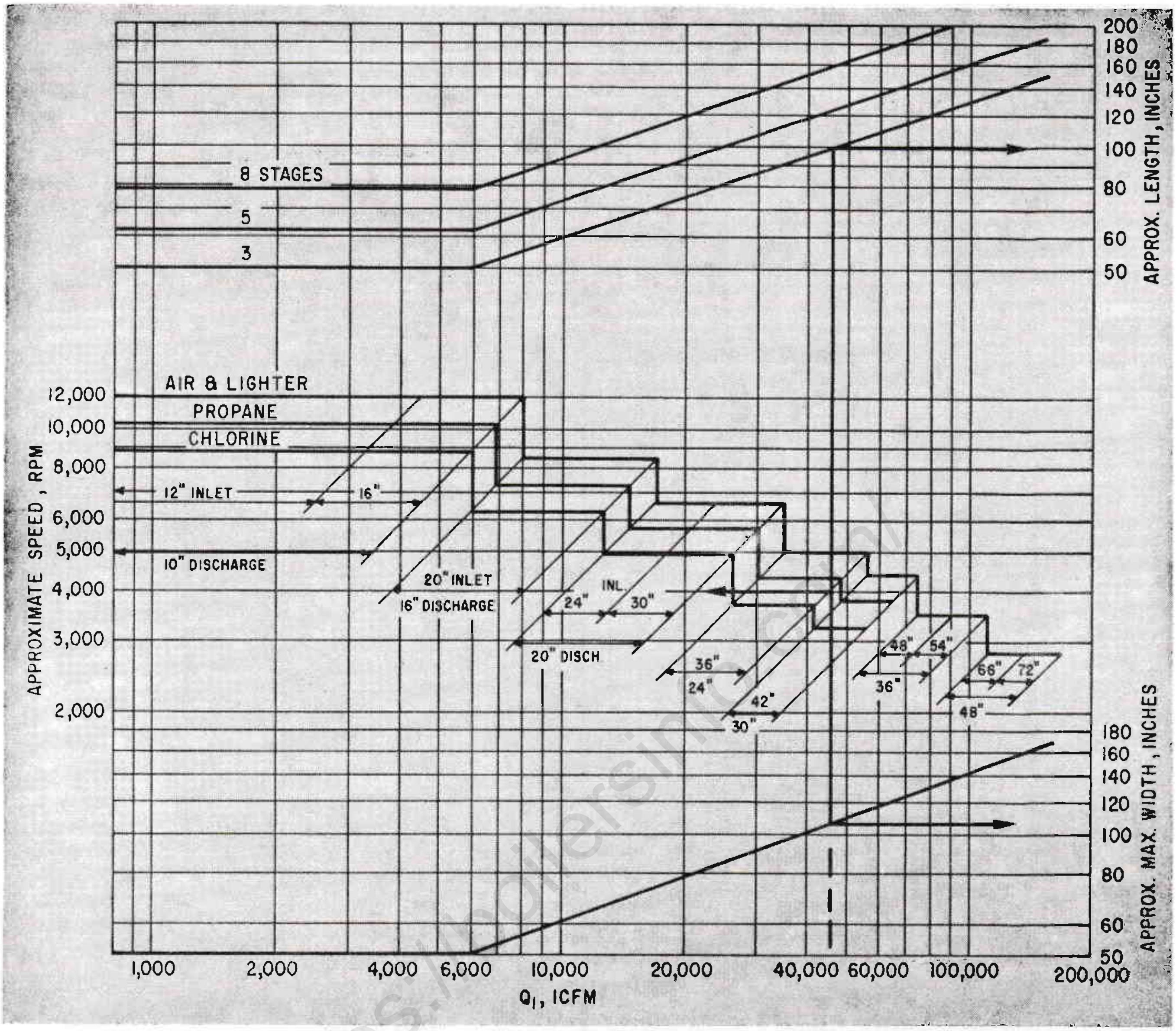


Fig. 5—Enter this chart at Q_1 found from Figure 1 or 2 and find speed, width, length and flange sizes.

Determine Number of Stages Required. On Figure 4, enter head H and proceed through M to read the number of stages required. Round this off to the next higher even number.

Determine Speed and Size of Machine. On Figure 5, enter Q_1 and read maximum width in inches. Proceed to the stepped lines and read rpm and flange sizes. Proceed through number of stages and read length of machine in inches. In the example shown, the ICFM is 45,000 and the gas is between propane and chlorine in mole weight. The speed is shown to be 4,000 rpm and the flanges are 36 and 24 inches. A slightly higher flow requires 3,500 rpm and 42 and 30 inch flanges.

Determine Approximate Price of Machine. On Figure 6, enter Q_1, proceed through H and read dollars. Correct this figure by the multipliers shown. The resulting price does not include lube system, driver, baseplate or special features. Because of changing market conditions and the wide difference in customer specifications, the accuracy of this particular curve must be considered no better than 15 to 20 percent, and must be considered subject to change.

Determine Horsepower Requirement. On Figure 7, enter W, proceed through Q_1 and H and read HP. If W is not known, work backward from Q_1 on Figure 1 to find W before using Figure 7.

For uncooled, constant weight flow compression such as alkylation, wet gas, recycle, or air under 50 psig, the foregoing is sufficient to determine price, size, and driver requirement. For cooled or variable weight flow compression, proceed as follows:

Cooled Compression. Assume one cool and two compression sections, each section handling a pressure ratio equal to the square root of the overall pressure ratio.

- Determine discharge temperature t_2 from Figure 8, proceeding through R_p, Q_1, K, and t_1.

CENTRIFUGAL COMPRESSOR ESTIMATES . . .

• Assuming this t_2 is satisfactory, proceed through all the figures for each of the separate sections. Speed and width of the compressor will be dictated by the first sections. Total HP is the sum of the sections. Price is based on first section Q, and total H of both sections. The price factor for cooler openings on Figure 6 must be included.

• If one cool does not depress t_2 sufficiently, or if still more horsepower saving is desired, try two cools or more. R_p per section for a two-cool, three section arrangement is the cube root of the over-all R_p; for a three-cool, four section arrangement, it is the fourth root. Bear in mind that more than one set of cooler openings is seldom available on a single compressor body. When more than one cooler is chosen, therefore, more than one compressor is dictated.

Considerable judgment is required in choosing the number of coolers to use. Once temperature limits are satisfied, the use of additional coolers becomes a matter of economics between compressor and cooler cost and horsepower evaluation.

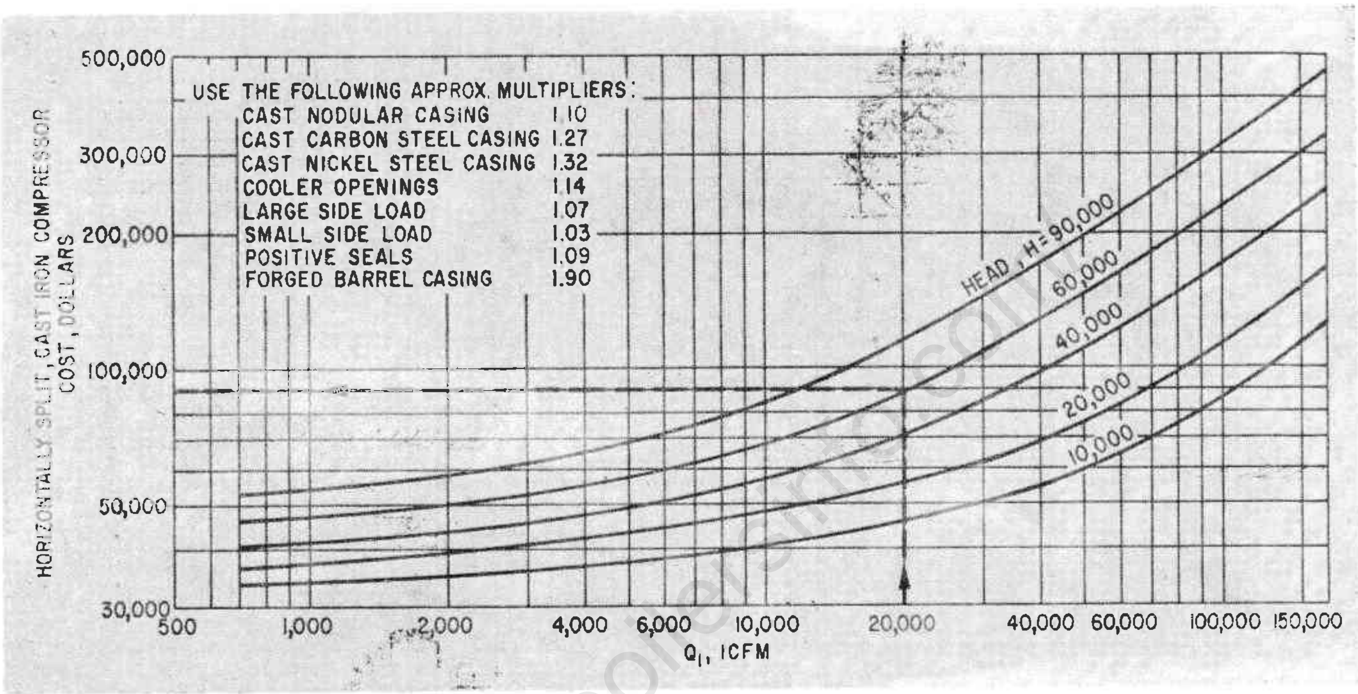


Fig. 6—Enter this chart at Q_1 found from Figure 1 to head H from Figure 3 and find compressor cost in dollars.

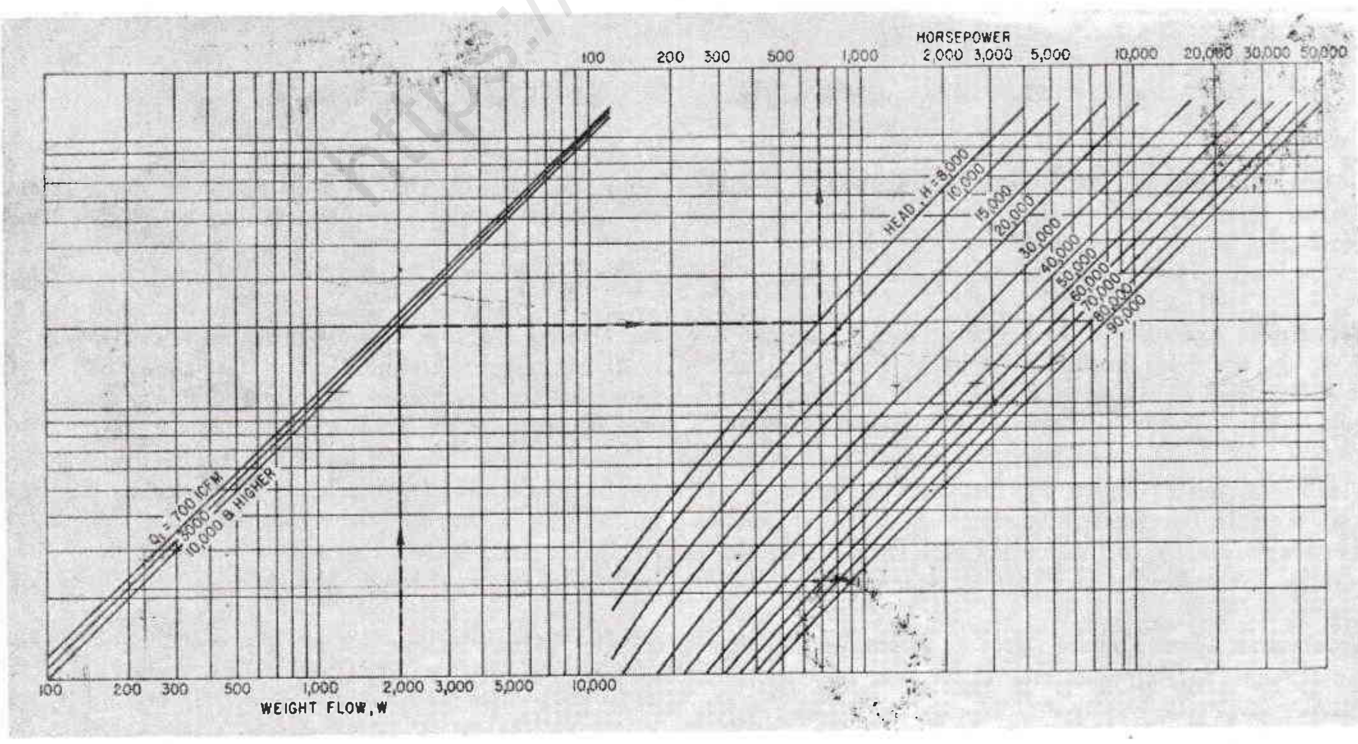


Fig. 7—Enter this chart at weight flow of gas, W, and proceed to find compressor horsepower required.

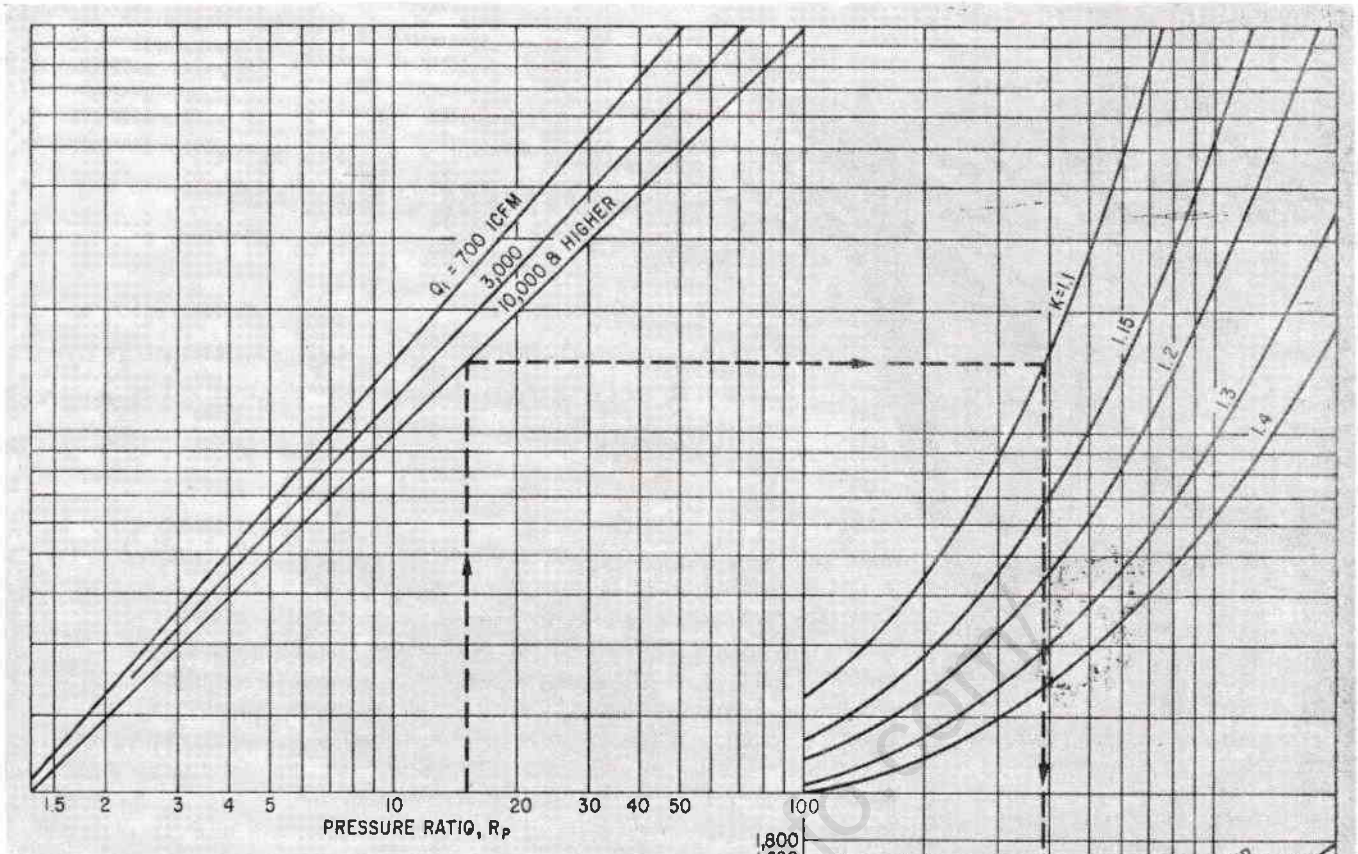
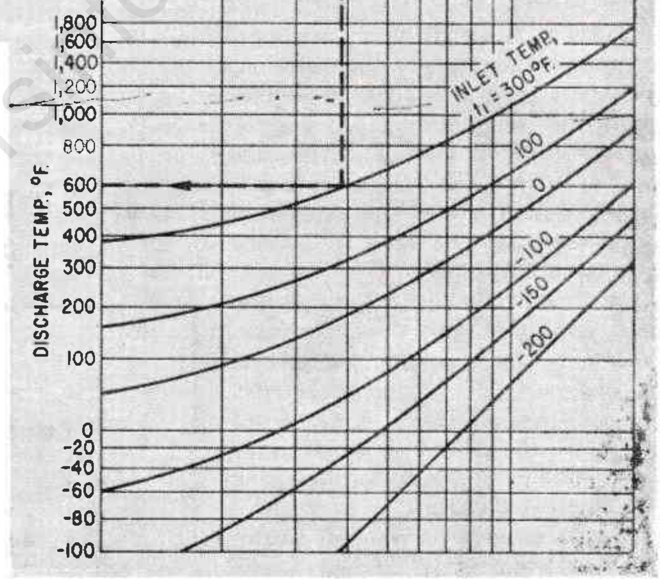


Fig. 8—Discharge temperature can be found on this chart.



Variable Weight Flow. For applications having side flows either in or out, it is necessary to consider each constant flow compression section separately. Mixture temperature to the second section after the first "in" side flow must be calculated by finding the discharge temperature of the first section from Figure 8, multiplying by the first section weight flow, adding in the product of the side stream temperature and weight flow, and dividing by the sum of the weight flows. With mixture t_1, P_1, W, M and K known, the figures can now be used for the second section, and so on through the machine.

M and K of the side stream will generally be the same or quite close to those of the inlet, so mixture calculations for these quantities will normally be unnecessary. For extraction side flows, the second section inlet conditions are the same as the first section discharge conditions, except for W.

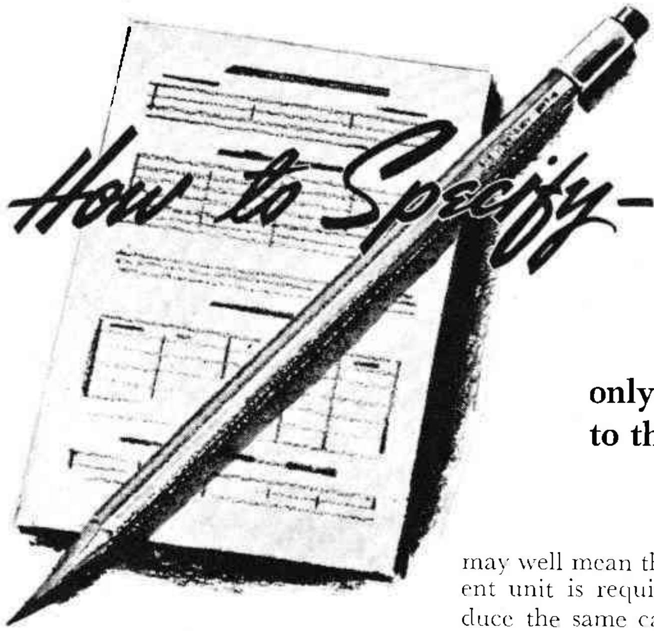
Normally the first section will "see" the largest Q_1, in which case the first section Q_1 will dictate the size and speed of the machine. An occasional refrigeration process, however, will show the second section Q_1 to be the greatest. In this case that Q_1 will dictate machine size and speed.

To determine the number of stages required, add the stages for each compression section and add in a blank stage for each large side load. It is impossible to give criteria for exactly what constitutes a "large" side load, but experience has shown that a typical propylene unit will require a blank stage for the first side load only, whereas a typical ethylene machine may require two blank stages. If the total number of stages, including blanks, exceeds nine, a second machine will probably be required.

# #

About the author

DON HALLOCK *is a senior application engineer with Elliott Co., Jeannette, Pa. His specialized field is concerned with compressors. Mr. Hallock holds a B.S. degree in mechanical engineering from Cornell University and joined Elliott following graduation in 1953.*



How to Specify— Compressors

A compressor will perform properly only if accurate and realistic design data is given to the manufacturer. Here is a guide to specifying.

John E. Stryker
Compressed Air and Gas Institute
Cleveland

THE MOST IMPORTANT considerations in specifying a centrifugal compressor for refinery applications are a thorough knowledge of the gas itself, the compression ratio, and the quantity to be handled. The hydraulic performance, mechanical design, materials of construction, type of seal and control are all directly dependent on this information.

While the foregoing is an important and essential phase of the application, it is only about half of the over-all problem and the selection of the type, size and characteristics of the driver completes the unit. This article, however, deals with the factors affecting the compressor only.

The centrifugal compressor manufacturer is somewhat limited in the assistance he can render by the quality and quantity of the performance data, conditions and requirements submitted and, consequently, the buyer shares heavily in the responsibility for a suitable selection. It should always be kept in mind that since the hydraulic design is predicated on the following few conditions and gas characteristics, namely, intake pressure, intake temperature and intake capacity, molecular weight, compressibility factor, ratio of specific heats and compression ratio each value should be established only after a thorough and complete study. A seemingly small change in any of the gas properties may well mean that an entirely different unit is required in order to produce the same capacity and pressure.

The required performance is generally established by a nominal set of conditions termed "Design," "Normal," "Rated," or "Guarantee," which is the best estimate of what is anticipated and is usually on the conservative side. As a safety margin, and overload performance is frequently specified as "Maximum" and has the order of magnitude of 105 percent rated speed (corresponding to approximately 110 percent of rated head at the rated capacity) and 110 or 115 percent of rated power (corresponding to approximately 110 or 115 percent excess weight flow at the rated head). The judical choice of these values will be reflected in the ability to perform the required service, first cost of the equipment, economy of operation, degree of flexibility of operation, and adaptability to future increased or decreased base load. Proper emphasis must be placed on a realistic determination of the size of compressor; pyramiding safety factors may result in a costly unit operating at light loads while skimping may result in a unit which operates inefficiently at overload and seriously limits the process output.

Fluid to Be Compressed—Whenever it is possible to do so, the gas stream should be free of liquids, corrosive components, abrasive material, particles likely to deposit in the impellers or diaphragms, and compounds which polymerize at conditions encountered anywhere within the unit. Actually, the compressor itself is inherently a centrifuge and will effectively remove liquids and much foreign materials from the gas stream, but this is usually not a desirable manner of operation due to the problem of disposal of this material which tends to clog the passages. The effective removal of all or most such material ahead of the compressor will contribute to a minimum of maintenance and long trouble-free operation.

Centrifugal compressors are, of course, basically gas handling units but within limits may lend themselves to rather severe service by means of special designs, special materials, and special operating procedures, all of

Meet the Author

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which may require adjustment to the particular local conditions. Very often, for cases of non-ideal conditions, the determining factor of whether a particular unit will give satisfactory or unsatisfactory service is the degree to which an operating technique and schedule is developed. This requires a certain amount of ingenuity, patience, experimenting, and then strict and unceasing adherence to the successful routine.

For cases where it is not possible or practical to eliminate these undesirable components, certain design features may be employed to minimize their ill-effects. Departures from conventional materials and construction entail compromises in performance and increase in the cost. The following will serve to illustrate what can be done along these lines:

Abrasive Dust—Impeller may be designed with heavier metal thickness, resistant metals may be selected, and even chrome plating or other surface coatings have merit in some cases. If the unit handles air from an atmosphere containing appreciable dust an intake filter can be justified.

Deposits are likely to build up in impellers with resultant unbalance and restriction to flow. Impellers with radial type vanes are fairly self-cleaning by virtue of the centrifugal force being in the same direction as the blade and the flow. Backward bent vanes are less satisfactory under these conditions as the centrifugal effect tends to hold the particle against the underside of the vane. Impellers of the unshrouded radial or "paddle wheel" type have also a scraping action on the casing wall which is quite effective in preventing build-ups and in addition is accessible for manual cleaning. Figure 1 shows this type of construction. Deposits within the impeller and casing passages have in some cases been removed by washing, i.e., atomizing water or a suitable solvent, but such a procedure should involve a minimum of injection and careful, frequent, and regular internal inspection of the unit.

Corrosive Constituents—Depending on the nature of the contaminant, amount of moisture present, and temperature encountered, standard materials of construction may not be adequate, and impellers, shaft sleeves, and other rotating parts may require bronze, stainless steels, or monel materials. Generally, the stationary parts may be carbon steel or cast iron, as these parts can tolerate without distress a much greater degree of corrosion than the highly stressed, cynamically balanced rotor. For severe conditions the casing may be designed initially with a "corrosion allowance" such that the casing may be serviceable over a long period of time before reaching a predetermined retirement thickness. In the case of conditions below the dew point at the intake and in the presence of some corrodents it is feasible to use special resistant materials in the first stage and sometimes even in the second stage. The heat of compression will, of course, raise the gas temperature above the dew point in the succeeding stages and permit the use of standard materials. In some cases of corrosive constituents, it may be necessary to avoid copper bearing metals, aluminum or lead.

Liquids—Entrained liquid in a gas stream already saturated will increase appreciably the power required to drive the compressor, may cause serious damage to the rotating element and thrust bearing if the moisture enters in slugs, or cause pitting and erosion of the impellers due to the continuous impingement of droplets of liquid against the rapidly rotating impellers. Also, as the mixture is compressed and the temperature increases as the moisture evaporates any residue may deposit within the unit. If moisture separators cannot be used, then every effort should be made to drain the intake pipe ahead of the unit, the intake head (if the intake connection is up) and the individual stages.

Seals—The type of fluid being compressed and the pressure level will determine to a great extent the shaft seal requirements. Such a seal is provided where the shaft passes through the casing to atmosphere or through the bearing housing to atmosphere on the drive end of compressors having the bearing integral with the casing and at the intake pressure level.

Labyrinth Seal—For air compressors and others handling gas of a non-toxic, non-inflammable and possibly a non-obnoxious nature at moderate pressures, slight leakage of the fluid from the casing is of no consequence and the conventional labyrinth seal is used. This seal is shown in Figure 2. The packing box is a soft non-galling aluminum or bronze alloy and sufficient clearance is provided so that the shaft will not be in contact with the labyrinths.

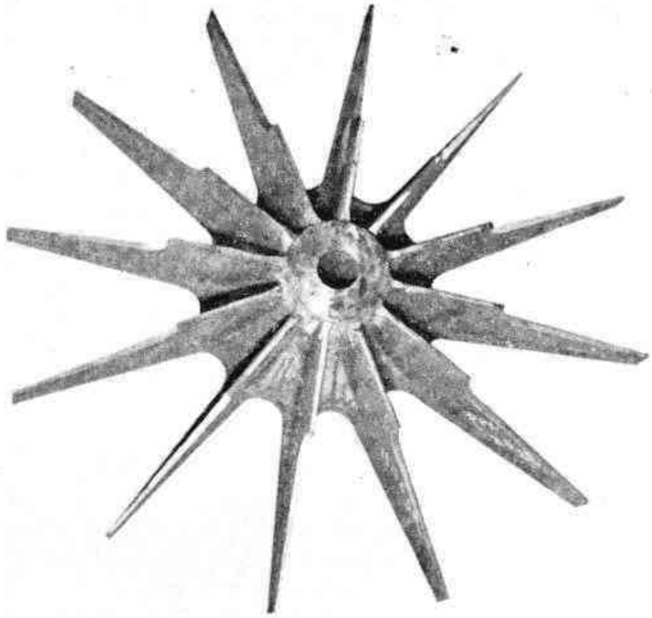
Ejector Seal—Where leakage of gas to atmosphere cannot be tolerated, a seal similar to the conventional labyrinth described above can be used if provided with an injection (or ejection) point at an intermediate location. A source of "inert" sealing gas or air can be injected at this point at a pressure greater than that inside the casing causing flow into the unit and outward to atmosphere. In this manner, the sealing gas only will leak to the atmosphere and, of course, will slightly dilute the product stream or by connecting this intermediate connection to a steam or gas ejector, a negative pressure can be produced within the seal and the net result will be a slight loss of some product gas and a mixture of this product gas, atmospheric air, and the ejector motive fluid to be disposed of possibly back into the compressor intake or to a flare line.

Similar seals using carbon rings instead of metalic labyrinths will accomplish exactly the same function with less leakage but the carbon seal is most suitable for units handling clean gas or where clean sealing gas may be injected between two carbon rings. This seal is not adaptable to compressors handling dirty gas where the ejector system is used because of the flow of dirty gas across the inner carbon ring.

To insure good sealing under variable conditions and to minimize the quantity of injection gas or motive gas for the ejector a differential control may be used to maintain a fixed differential above or below the pressure being sealed against.

Oil Seal—For cases where the above described "dry" type seals are not adaptable, the more elaborate oil seal is available. It will produce a positive seal preventing the leakage of gas from or atmospheric air into the casing. In general, due to its appreciable expense and maintenance requirements, usage of the oil seal is limited to applications where the pressure level is high, where no leakage is tolerable or where for reasons of unavailability of sealing gas, dilution of the product gas, etc., the other types of seals are not adequate. The seal oil system consists of a reservoir, main and standby oil pumps, coolers, filters, and control valves. Depending on the gas, contamination of the oil may be a problem to a greater or lesser degree not only from the standpoint of foreign matter from the gas and sludge or solids produced by combination of gas with the oil but also from dilution with condensibles from the gas stream. Consequently, it is sometimes necessary to employ oil conditioning equipment and for some designs the oil which has come in contact with the gas is thrown away rather than being returned to the system.

FIGURE 1—"Paddle wheel" type of impeller construction.



Compression Ratio and Quantity to Be Handled—Centrifugal compressors must be designed to produce a pressure at least equal to the maximum system pressure encountered. Unlike the positive displacement unit, which will discharge against any pressure (providing the compressor is mechanically suitable and the driver is capable of delivering the required power), the centrifugal will have no output unless designed to produce a pressure exceeding the pressure level inside the system to which it is connected. If the system pressure is higher than that produced by the compressor and no check valve is used gas will flow back through the unit and out its intake.

The following general comments may be helpful in an evaluation of maximum possible deviations from the specified performance requirements:

- One performance point only is guaranteed.
- Other performance points are "expected" but not guaranteed.
- The shape of the performance curve and pumping point are also "expected" but not guaranteed.
- The guarantee requires meeting the specified capacity and discharge pressure without tolerance and meeting the power within a 4 percent tolerance.
- European practice has been less stringent on the guarantee and a tolerance in capacity and pressure is permitted in addition to that in power.

It should, however, be appreciated that independently of the legal obligations of the performance guarantee it is entirely realistic to assume that the unit will be capable of meeting all of the quoted conditions.

Pressure and Quantity—The process system resistance curve, i.e., pressure vs. quantity frequently is most difficult to determine with accuracy. Yet for a centrifugal which has essentially a constant pressure variable capacity characteristic the compression ratio is quite critical and among the few difficulties encountered in applications many are inadequate pressure. A careful analysis is required of the pressure drops in intake filters, intake piping and orifice plates as well as that in the discharge piping, all valves, fittings, coolers, heaters, separators and the like. For cases where the output is controlled at constant pressure or a hydrostatic head or the equivalent is involved or where there is a combination of this and friction drop, the problem is simplified due to the greater certainty of the controlled factor. The compressor must, however, be selected to have sufficient margin in compression ratio to offset the possible accumulation of errors in evaluating the system characteristics.

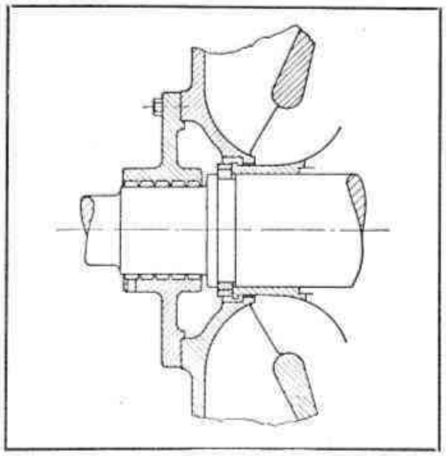


FIGURE 2—Labyrinth shaft seal.

With the above in mind, it becomes apparent that once the system is defined, it still remains to specify for the machine the compression ratio at the various conditions and gas properties encountered. It should be understood that compressors are rated on the intake flange to discharge flange compression ratio based on total pressures (static plus velocity head) and no allowance exists for pressure losses in the intake or discharge piping. The unit must be designed to develop the required discharge pressure and capacity for the most severe combination of circumstances, i.e., minimum intake pressure, maximum intake temperature, minimum molecular weight, maximum ratio of specific heats. If all of these conditions can not occur at the same time then, of course, only the worst set which can exist simultaneously need be considered.

It is obvious now that there is some degree of uncertainty from two standpoints (a) the actual system characteristic and (b) the actual

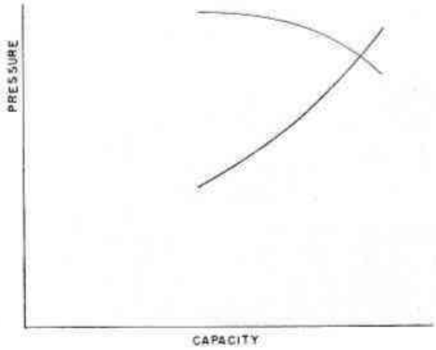


FIGURE 3—Typical pressure-volume relationship for centrifugal compressor and system frictional resistance curve.

pressure produced by the compressor when handling the process gas at the existing conditions. Note system and compressor characteristics plotted in Figure 3. The only point at which this unit will operate when discharging into this system is the intersection of the two curves.

Size and Quantity—At this point it perhaps would be helpful to discuss briefly the compressor size in terms of increased or decreased capacity and/or increased or decreased pressure. In Figure 4 curve *A* represents a particular size of unit. To increase the capacity at the same compression ratio curve *B* would apply and to increase the compression ratio at the same capacity curve *C* would apply. All three of these units are of basically different size. Curve *B* might represent a unit of the same number of stages as *A* but with impellers of greater width and diameter running at a lower speed in a larger casing. Curve *C* might represent a unit of twice the number of stages as *A* and with impellers of the same diameter and about the same width and running at the same speed. For the unit having performance curve *A*, an increase in speed (if the unit and driver are suitable for the higher speed) would result in curve *A′* and a decrease in speed curve *A″*. The compressor changes in pressure and capacity simultaneously but only within rather narrow limits. To obtain the most efficient and economical unit for any particular set of conditions, the centrifugal is designed with the minimum number of stages required to produce the rated head. In other words centrifugals are rated at or near their maximum allowable speed and if the compression ratio is shy it is seldom possible to compensate for this by increased speed. Finally the effect of the power wheel or of variable intake guide vanes on the size of unit as compared with variable speed is shown in Figure 5. Curve *A* represents 100 percent speed as well as the radial position of the guide vanes. As the vane angle is changed, curves *D, E,* and *F* are produced. The curves become steeper, the pumping limit improved and the unit becomes substantially smaller. Figure 6 illustrates a multistage compressor with power wheel and guide vanes including some of the constructional details. In the case of variable intake guide vanes the direction of flow entering the impeller is changed so as to cause prerotation and thereby decrease the pressure rise through the impeller. The power wheel is in effect a turbine wheel located so as to be an extension of the first stage impeller inlet and is arranged with variable guide vanes at its periphery. When the vane direction is changed from the radial a pressure drop is created and the flow is directed into the power wheel in such a manner as to drive it and impart torque to the shaft reducing the net power required to produce a lower discharge pressure. With the guide vanes in the radial position the performance is substantially the same as for an equivalent unit without the power wheel.

Excess Capability—From the foregoing it can be seen that virtually no possibility exists for increasing the pressure of a centrifugal at the same capacity for any constant set of intake conditions. Similarly, while the capacity may be increased at a sacrifice of pressure, the system characteristics usually require more, not less, pressure at an increased quantity of flow.

As a result of the uncertainties described above as well as the relatively fixed nature of compressor and process characteristic, the compressor must be built with some excess capability. The exact amount of this margin is governed by the degree of accuracy to which the particular process conditions are known. Another consideration is possible increased production. The capacity of any plant is limited of necessity to the capacity of one or more elements of the equipment. If indications are that the plant may have some extra capacity the centrifugal should be amply sized since the additional cost involved is moderate compared to the major expense involved in rebuilding or supplementing an existing unit for more capacity or pressure. In all cases the driver must be selected for the horsepower required for the maximum conditions because any limit in power will be as serious as a small compressor.

Several possibilities exist for matching the compressor to the process but in each case the result is to get the curve of the unit to intersect the process curve at the desired quantity

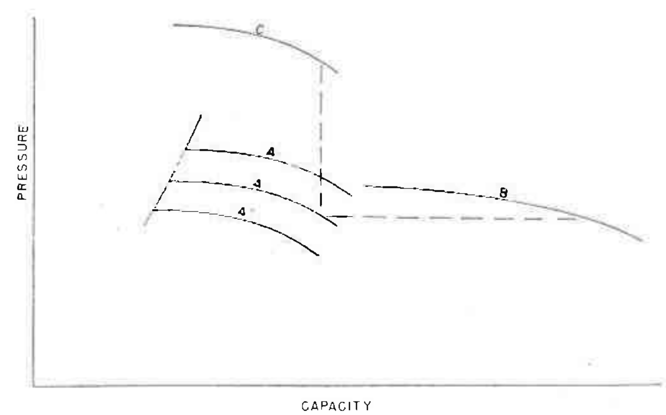


FIGURE 4—Performance curves for three compressors of different basic size.

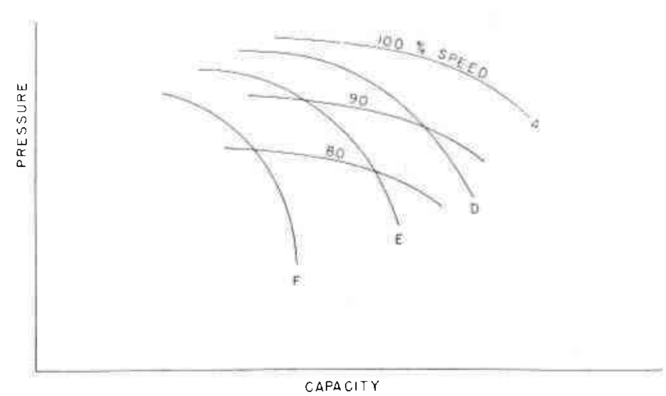


FIGURE 5—Variable intake guide vane performance curves superimposed on variable speed curves.

of flow. If the intersection occurs at too low a capacity the desired capacity cannot be realized, if (by coincidence) the intersection occurs at exactly the desired capacity that is an ideal situation in one sense although it leaves no reserve for varying conditions and finally if this intersection occurs at a greater capacity than desired, then either the compressor or process characteristic must be altered. The centrifugal may, if the driver is of the variable speed type, be slowed down as shown in Figure 7 (a) to a speed at which the characteristic curve just provides the necessary pressure at the desired capacity. Similarly this curve can be lowered by means of a power wheel or variable intake guide vanes, Figure 7 (b). In each of these cases the margin in compressor size is eliminated in an economical manner, i.e., the power consumed, Figure 7 (a) and (b), is that required to provide only the pressure level utilized. The process curve, on the other hand, may be raised by throttling across a valve at the discharge of the compressor as per Figure 7 (c). Thus the intersection is effected at the desired capacity while the compressor curve remains unchanged, i.e., the power consumed corresponds to that required to produce a pressure exceeding the normal process pressure level and consequently the excess pressure represents a continuous waste of some power. Note that the relative merits of variable speed, variable intake vanes or power wheel depend on many factors but the only significant comparison here is between any one of these means of adjusting the compressor characteristic curve and throttling at the discharge. It is true that intake throttling is somewhat more economical than discharge throttling depending in part on the particular design of the unit; however, in both cases of throttling practically speaking the loss is complete. For any proposed installation an analysis of the economics will indicate whether throttling or other provisions are justified.

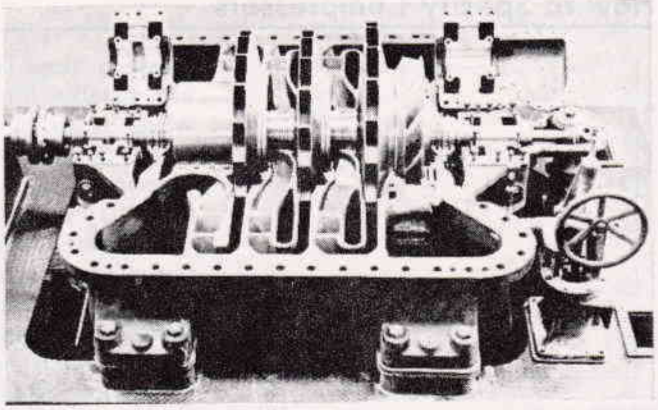


FIGURE 6—(A) View of compressor with casing upper half removed showing power wheel and guide vanes.

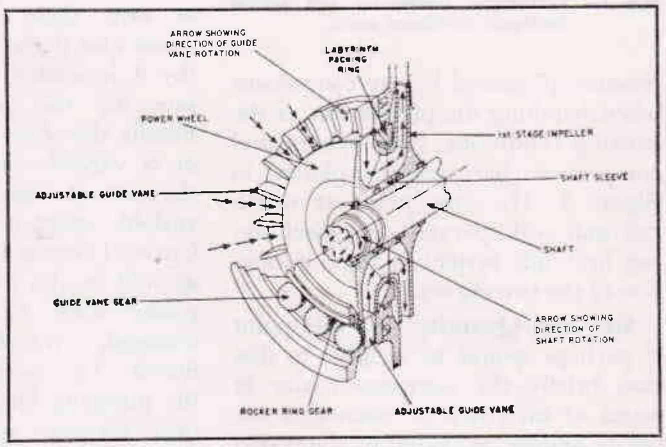


FIGURE 6 — (B) Sketch of power wheel and vane mechanism with direction of flow indicated.

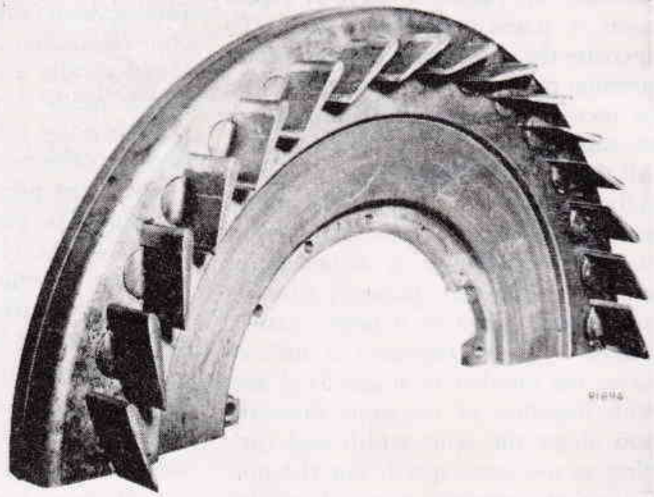


FIGURE 6 — (C) Guide vane assembly with vanes in a partially throttled position.

Supplementing the previous section on "fluid to be compressed" it must be realized that while variable vanes might be desirable from the standpoint of economy due consideration must be given to the effect of the gas on these movable parts. For air or clean gas no problems are to be expected but if the gas is dirty or tends to create build-ups inside the casing the vanes may become frozen in some position in a short time. Under adverse conditions much can be done from the operating and maintenance standpoints to keep the vanes free but in general unless the process gas is fairly clean it is well to consider variable speed or throttling.

In exploring more thoroughly the quantity to be compressed independent of the process pressure-capacity characteristic it is apparent that the centrifugal is flexible in its capacity handling ability. This capacity will vary from the pumping limit (averages approximately 50 to 70 percent of rated flow at intake conditions for a single casing) to the rated flow, and beyond, at substantially the same pressure level. The pumping limit or surge point occurs at or near the peak of the pressure curve and is the lowest capacity for which the unit will deliver a continuous flow of gas at a

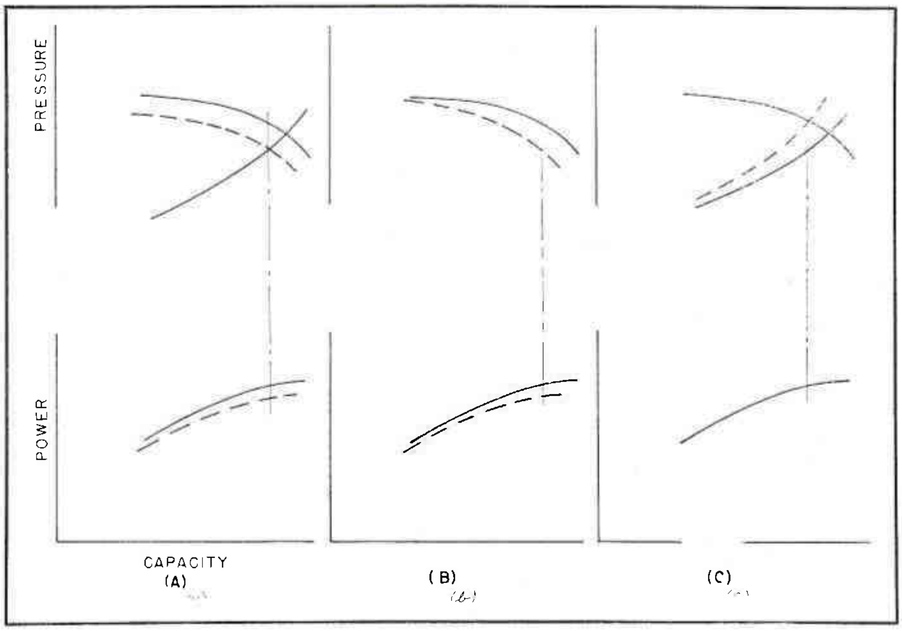


FIGURE 7—(A), (B) and (C) Power consumption for varying compressor characteristics, compared with throttling at the discharge.

steady pressure. In the region below the pumping limit the flow will temporarily reverse creating surges in pressure the frequency and severity of which are a function of the design of compressor, the number of stages, the gas being compressed, the pressure, and the particular process system. Operation in this range is not recommended and provisions must be made to increase the flow to something above the pumping limit. "Minimum Volume" controls are available and can be arranged to recirculate gas back to the intake or for air units waste air to the atmosphere. If recirculation is contemplated provision must be made to prevent overheating the compressor.

Efficiency—Figure 8 shows that for only one capacity does the best efficiency occur. With a fairly flat efficiency characteristic, however, a reasonably good efficiency can be obtained for quite a wide range of capacity on either side of the peak efficiency. It might be added that for various hydraulic designs the efficiency curve will tend to peak more or less sharply but unless the capacity is known precisely and the load constant, a flatter efficiency characteristic is most desirable. In practice the rated capacity is located at or just beyond (higher flow) the peak efficiency point so as to provide as wide a stable operating range as possible. If the unit ultimately operates at an average capacity slightly less than the rated point it will tend toward the peak efficiency.

Flow Conditions—For the centrifugal compressor the quantity of flow is dependent on the impeller capacity per revolution. It is, therefore, wise to express the flow in terms of volume at intake conditions in order to avoid any possible misunderstandings as to the exact conditions which are to be the basis of design. While the conversion calculations from weight flow or "standard" conditions are reasonably well known and understood it occasionally occurs that the data provided is not clearly defined or complete and may lead to flow quantities which differ significantly from the true value. The same general philosophy applies equally well to the determination of the gas physical properties and no one is in a better position than the user to state the intake capacity and gas properties accurately. In the last analysis the compressor manufacturer has control only over the design and the unit will perform properly only if subjected to the conditions for which it was built.

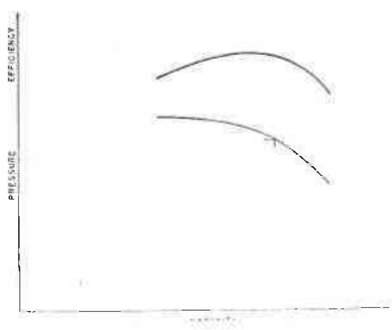


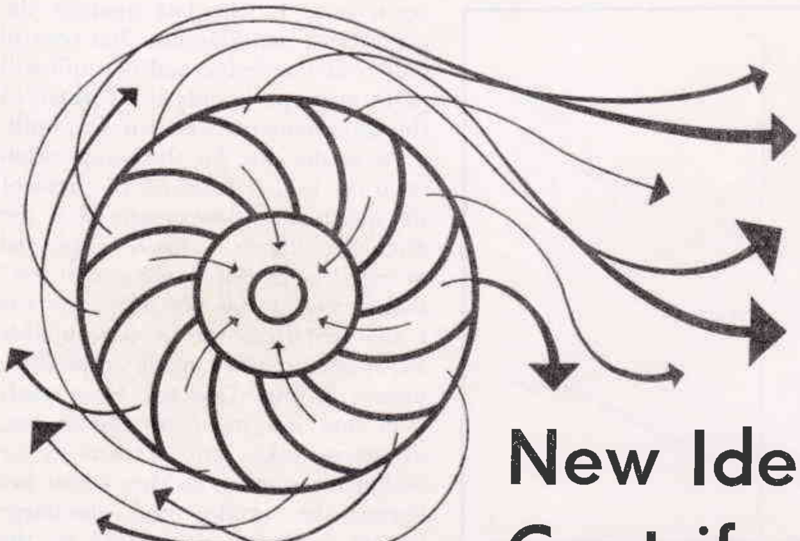
FIGURE 8—Compressor efficiency curve.

As in the case for the compression ratio the unit is rated on the basis of the quantity of flow measured at the discharge flange (usually expressed in capacity at the intake conditions) and if any of this net output of gas is diverted from the process suitable adjustment must be made in the compressor rating. Leakage from shaft seals and internal recirculation are, of course, taken into account in the design in as much as they occur between the intake and discharge flanges and are chargeable to the compressor. Gas used as motive power for shaft seal ejectors, for example, must be accounted for in the determination of the compressor capacity rating.

Sidestreams can be accommodated in many designs but exactly the same considerations apply and in fact the problem of determining precisely the quantities of flow and compression ratios is even more complex. Allowances must be made to insure pressure levels compatible with the entrance or exit of the required quantities at intermediate points. Where sidestreams of relatively high flow rates leave the unit any percentage deviation in this quantity could result in tremendous percentage of overload or underload of the impellers handling the balance of the flow.

For compressors operating in series with components removed during cooling between units or with other gas entering between units every effort must be made to accurately identify conditions and gas properties at the entrance of each unit. For single state units the behavior of the gas is less significant than for multistage units and similarly several casings in series will magnify greatly the effect of inaccurate data.

In conclusion a thorough and accurate knowledge of the process gas properties and contaminants, its quantity and the pressure requirements of the system will all contribute substantially to the selection of a centrifugal compressor which will adequately perform the required service with economy, flexibility, and reliability. ##



New Ideas on Centrifugal Compressors

This three part series describes the following: Part 1: power rating method and sizing; Part 2 metallurgy and design; Part 3: shaft seals and balance pistons

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There are three basic reasons for using centrifugal compressors in hydrocarbon processing plants instead of reciprocating compressors.

- **Environmental.** They occupy less space; they operate with minimum attention and are quieter.
- **Lower Operating Cost.** They will run 12 to 30 months without mechanical repair. The maintenance cost is about one-third the cost of piston compressor operation, excluding the drivers.
- **Lower Initial Investment.** The lowest cost turbodriver is either a gear-motor or a steam turbine. The cost of such a driver and centrifugal compressor is approximately equal to the least costly type of synchronous motor driven reciprocating compressor in the 2,500 hp category. The centrifugal machinery cost is about two-thirds that of the piston machinery in the 5,000 hp bracket. The cost advantage of the turbomachinery is enhanced as the power demand is increased.

The reciprocating machine and motor drivers are priced on a cost per horsepower basis. Centrifugal machines and steam turbines are sold on a lump sum, plus a small unit cost per hp for the driver. The cost of a 12,000-hp centrifugal compressor may not be much more than the same casing rated for 1,000-hp service. Confronted with such an economy, there is an unnderstandable demand on the part of the hydrocarbon processors for turbomachinery. The purpose of this part of the series is to review the prevalent *state of the art* as to the compressors capability and a method of power rating and sizing the machine.

Compression Head. The compression head is the intangible measurement of the energy density imparted to a gas stream by a compressor. It may also be defined as the enthalpy added to the gas by the compressor. It may be observed by the increase in the gage pressure as the gas passes through the machine. Some engineers choose to evaluate the compression head as an isothermal function. A great many European compressors are evaluated on this basis. It presumes that the gas leaves the compressor at the entering temperature. This is an obvious false premise.

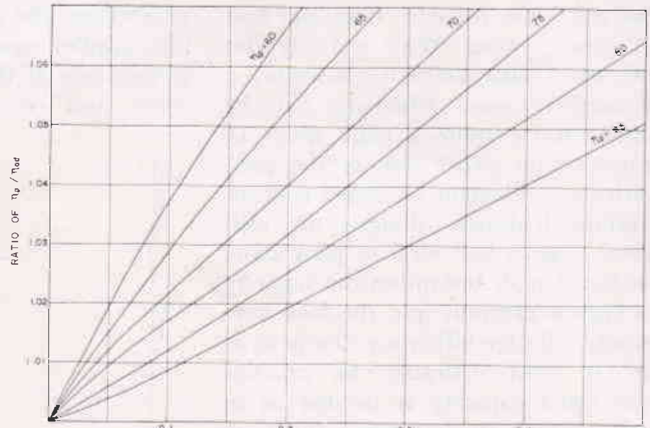

Fig. 1-1—Ratio of polytropic to adiabatic efficiency, with reference to the adiabatic exponential function and the polytropic efficiency.

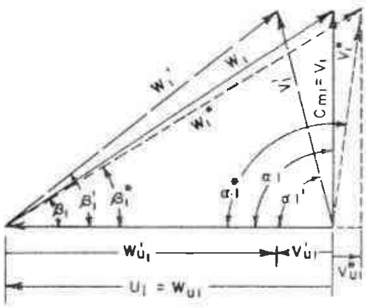

Fig. 1-2a—Vector diagram depicting typical suction velocities. C_m is the meridional velocity, U is the peripheral velocity of the impeller, V is the absolute velocity of gas flow, and W is relative velocity of the gas flow. α_1 is the eye entrance angle.

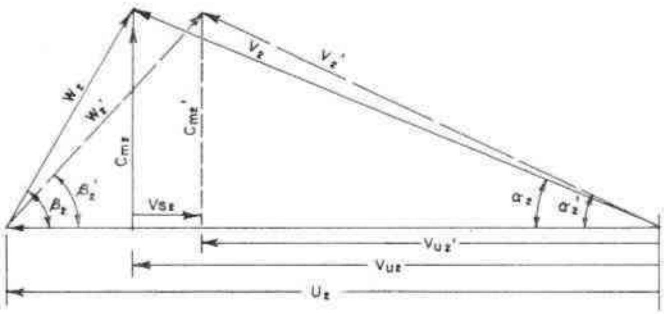

Fig. 1-2b—Vector diagram depicting typical discharge velocities. β_2 is the back-lay angle of the vane at the impeller tip.[4]

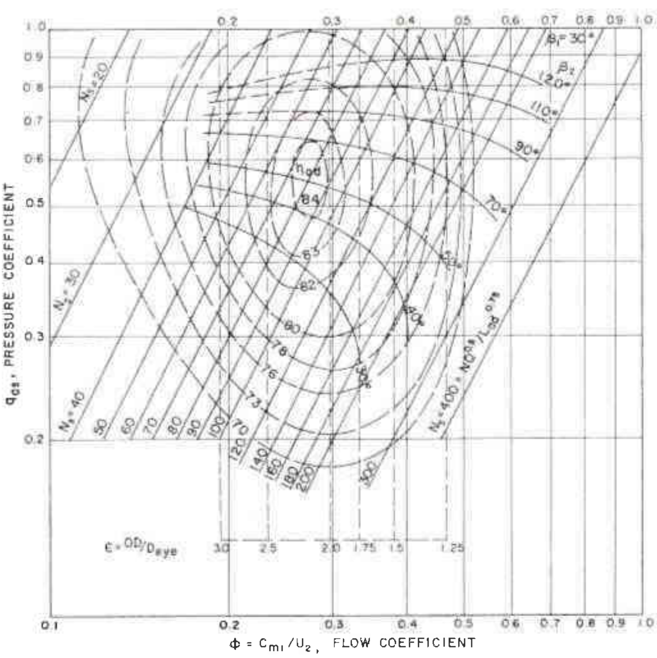

Fig. 1-3—Eckert chart giving pertinent adiabatic performance data for impellers having various vane angles.

Isothermal Head. The isothermal head is determined from the equation:

$$L_{iso} = 144\ (\text{Log}_e\ R_c)\ P_o\ (V_{so}),\ \text{ft.-lb./lb.} \qquad (1\text{-}1)$$

The application of this head to the simple hp equation given below, produces the isothermal efficiency for this instance.

$$\eta_{iso} = L_{iso}\ (w)/hp_{dy}\ (550) \qquad (1\text{-}2)$$

The resultant efficiency is related to the reference head. If the adiabatic head is used the resultant efficiency is adiabatic. The symbols used in the above and subsequent equations are explained in the nomenclature.

The isothermal head and efficiency have no particular significances or reference values. It is 6 to 9 percent less than the adiabatic efficiency. It is about 10 percent less than the polytropic efficiency.

Adiabatic Head. The adiabatic head is determined from the equation:

$$L_{ad} = (R_c^{\sigma} - 1)\ 1545\ (T_o)\ Z_a/\overline{m}\ (\sigma) \qquad (1\text{-}3)$$

The adiabatic efficiency is determined from Equation 1-2 as described above. Any reference of this order should bear the same significance as to the degree of perfection attained in design. The adiabatic efficiency reflects this proficiency.

The performance of a piston compressor with an abundance of valve area, or where the valve losses are evaluated, is as close to adiabatic behavior as existing instrumentation is able to register. From such experience we can state that intrinsic compression of a piston machine is adiabatic. It is the most efficient process known for compressing gases. The closer that the power and temperature rise of a centrifugal compressor approaches that of the intrinsic adiabatic piston compressor values, the more efficient it becomes. The temperature rise for identical compressor service is greater for a centrifugal compressor than it is for an intrinsic piston compression. In addition to the adiabatic temperature rise, the gas experiences the friction of high-velocity obstructions and resistances that exist in the inducer eye, the impellers, the diffusers, the exit connections, the shaft-seal leakage and the disk friction. These inefficiencies add heat to the gas directly proportional to their magnitude.

Adiabatic versus Isentropic. The adiabatic head and efficiency are often referred to as the isentropic values. The terms adiabatic and isentropic are used interchangeably. The thermodynamic definition of an adiabatic process requires that no heat be added or withdrawn from a facility where a change of state occurs. It may or may not be reversible to qualify as an adiabatic process. If the process is reversible, it is truly an isentropic operation. An isentropic change of state occurs at constant entropy. This identifies isentropic analysis with Mollier charts and tables of gas properties. For lack of a better mark of distinction, we refer to such enthalpy calculations as isentropic and the exponential R_c values as an adiabatic process.

Polytropic Head. The polytropic head is determined from the equation:

$$L_p = (R_c^{\sigma'} - 1)\ 1545\ (T_o)\ Z_a/\overline{m}\ (\sigma') \qquad (1\text{-}4)$$

The polytropic efficiency can be determined from the Equation 1-2 relationship. The unique and only justification for the polytropic efficiency is its identity with the discharge temperature.

$$T_e = T_o R_c^{\sigma'} \text{ and} \tag{1-5}$$

$$\sigma' = (k - 1)/k\,\eta_p = \sigma/\eta_p \tag{1-6}$$

The polytropic efficiency can be determined for any operating point, where the suction and discharge temperature and the k value of the gas are known. This ratio on the Rankine scale and the knowledge of the gas k value applied to Equation 1-6, give the polytropic efficiency. The usual connotation of the word *efficiency* refers to the degree of perfection.

If the adiabatic process is the ideal, then the adiabatic efficiency is the only valid efficiency. The isothermal and polytropic efficiencies are *pseudo* values. The former evaluates as a nonheating compressor which is impossible. The latter evaluates the machine as a heating device, rather than as a gas compressor.

Efficiency Conversion Chart. Fig. 1-1 presents a simplified correction factor for converting polytropic efficiences to adiabatic efficiency. It was prepared from ratios of compression of 1.2 to 3.7; for air, natural gas, LPG and hydrogen mixtures. The correction values are not absolute, but are close approximations. The integrated radical $(R_c^{\sigma} - 1)$ is used as the abscissa and the ratio of the polytropic to the adiabatic efficiency.

The *isoeffic* (constant efficiency) lines cover the normal range of adiabatic efficiencies. For an illustration, an air compressor is operated at $R_c = 2$, the integrated radical $(R_c^{\sigma} - 1)$ is 0.22, where $\sigma = (k - 1)/k = (1.40 - 1.0)/1.40 = 0.286$. Then if the polytropic efficiency was 65 percent, the correction ratio is 1.057 and efficiency would be $(65/1.057) = 61.5$ percent. If the polytropic efficiency was 85 percent, the adiabatic efficiency would be $(85/1.022) = 83$ percent. These correction ratios are obtained from the equation:

$$R\eta = \eta_p\,(R_c^{\sigma/\eta_p} - 1)/(R_c^{\sigma} - 1) \tag{1-7}$$

The equivalent isothermal efficiency for the above 83 percentage case can be determined by the inverse proportion of the respective heads and efficiencies. The isothermal head is:

$$L_{iso} = 144\,(\text{Log}_e\,R_c)\,P_o\,v_{so}\text{ ft.-lb./lb.} \qquad \text{(repeating 1-1)}$$

$$L_{iso} = 144\,(0.693)\,14.7\,(13.1) = 19{,}200\text{ ft.-lb./lb.}$$

The adiabatic head is:

$$L_{ad} = (R_c^{\sigma} - 1)\,1545\,(T_o)\,Z_a/\overline{m}\sigma \qquad \text{(repeating 1-3)}$$

$$L_{ad} = (0.22)\,1545\,(520)\,1.00/29\,(0.286) = 21{,}200\text{ ft.-lb./lb.}$$

The equivalent isothermal efficiency is: 83 (19,200/21,200) = 75 percent.

Example Head Calculation. The following example illustrates the derivation and application of the three reference heads and efficiencies. A shop performance test demonstrated that a centrifugal compressor was handling 36,200 cfm of ambient air at 14.7 psia and 60° F to 29.4 psia. The driving motor has an efficiency of 93.5 percent and demands 2,000 kw of power for the test. The discharge temperature is 215° F and the mechanical efficiency of the compressor had been previously determined to be 98 percent. The dynamic hp is: (2000/0.746) 0.935 (0.98) = 2,450 hp.

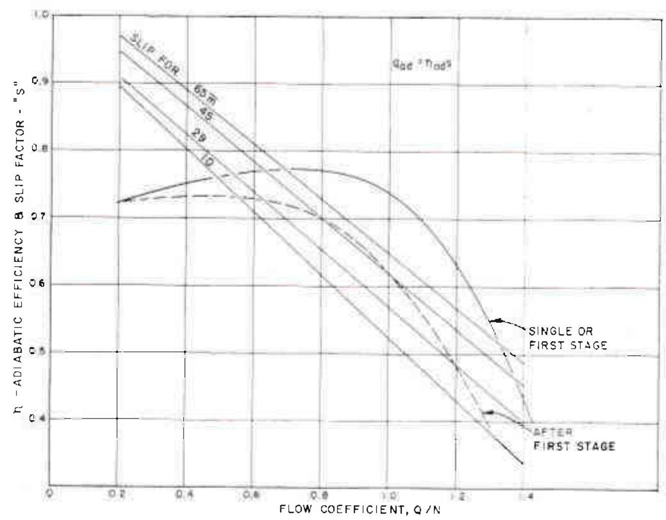

Fig. 1-4—Typical basic performance curve of a commercial centrifugal impeller, depicting the pressure coefficient, efficiency and slip plotted against the flow coefficient Q/N. The solid lines represent single stage performance. The broken lines represent multistage factors.

This value and the discharge temperature represent the vital data pertinent to the plant process. The respective efficiency used to project this information is only incidental.

The weight flow is:

$$v_{so} = 10.73\,T_o\,Z_o/\overline{m}\,P_o \tag{1-8}$$

$$v_{so} = 10.73\,(520)\,1.00/29\,(14.7) = 13.1\text{ cf lb.}$$

$$w = 36{,}200/13.1\,(60) = 46\text{ lb./sec. (pps).}$$

The isothermal head for $R_c = 2$ (29.4/14.7) was determined to be 19,200 ft.-lb./lb. in the previous paragraph. Transposition of the fundamental power equation produces the isothermal efficiency in the following manner:

$$hp_{dy} = w\,L_{iso}/550\,\eta_{iso} \tag{1-9}$$

$$\eta_{iso} = w\,L_{iso}/hp_{dy}\,(550)$$

$$\eta_{iso} = 46\,(19{,}200)/2{,}450\,(550) = 65\text{ percent}$$

The adiabatic head for R_c was determined to be 21,200 ft.-lb./lb. The adiabatic efficiency equation has the same form as given above.

$$\eta_{ad} = w\,L_{ad}/hp_{dy}\,(550) \tag{1-10}$$

$$\eta_{ad} = 46\,(21{,}200)/550\,(2450) = 72\text{ percent.}$$

The isentropic efficiency (η_i) should equal the adiabatic efficiency. The enthalpy difference between suction conditions of 60° F and 14.7 to 29.4 psia at constant entropy should be 21,200/778 or 27.2 Btu/lb.

$$hp_{dy} = W\,(H_e - H_o)/2544\,\eta_i \tag{1-11}$$

$$\eta_i = 46\,(3{,}600)\,27.2/2{,}544\,(2{,}450) = 72\text{ percent.}$$

The fundamental design of the impeller is represented by the vector diagram of Fig. 1-2. The Euler equation,

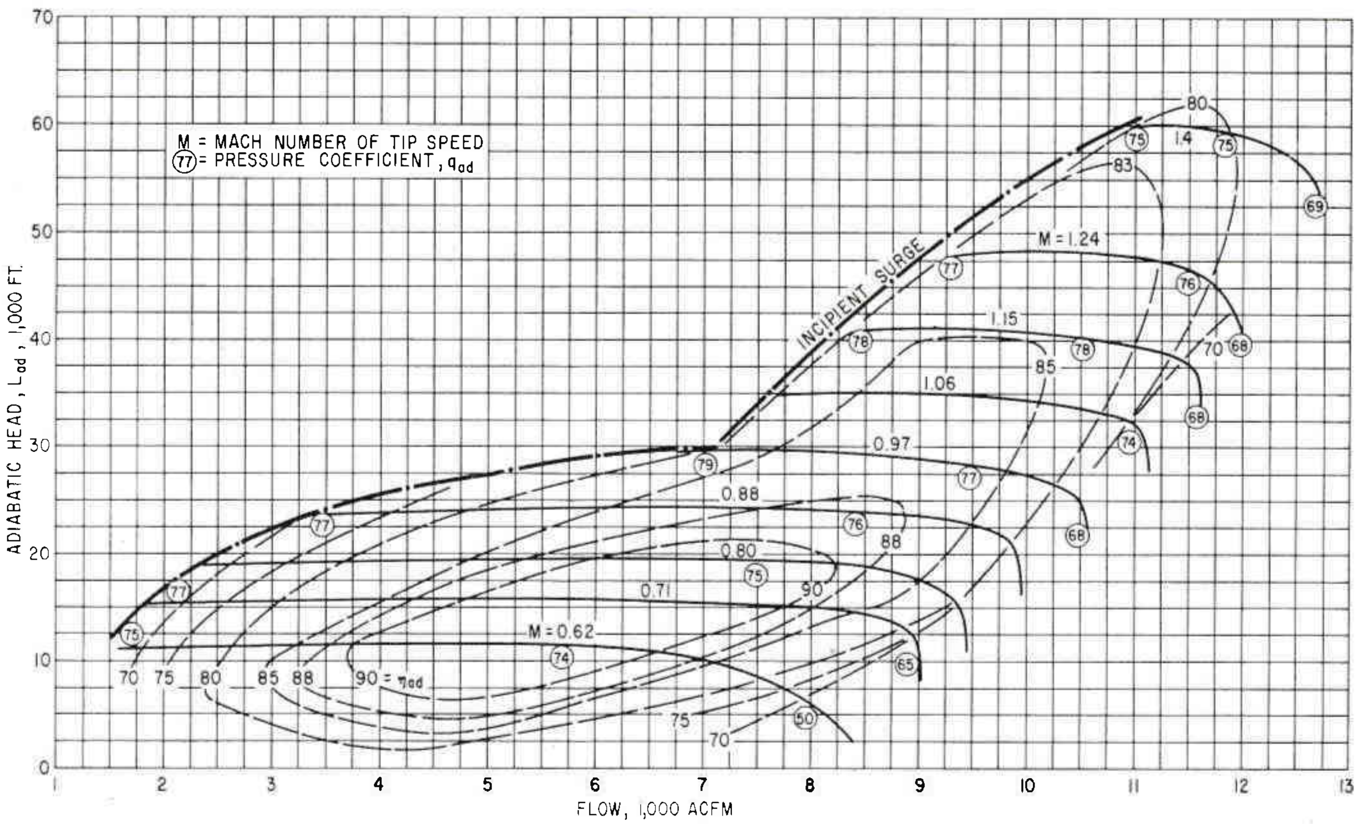

Fig. 1-5—Performance curve of highly perfected NACE impeller.[3]

Equation 1-12 resolves these vectors into an energy density, gas head:

$$L_{eu} = (V_{u2}\, U_2 - V_{u1}\, V_1)/g \qquad (1\text{-}12)$$

The Euler efficiency will approach the adiabatic equation if the flows typify the actual flow and the Euler efficiency will be:

$$\eta_{eu} = w\, L_{eu}/550\, hp_{dy} \qquad (1\text{-}13)$$

In addition to the vector analysis, there are the flow losses through the suction eye, the vanes, the diaphragms, the scroll or diffuser, the guide vanes, the seal leakage and disk friction which contribute to the inefficiencies of the compressor and raising the discharge temperature. The reason for making a shop performance test is to confirm the accuracy of the design premise.

Temperature. The polytropic efficiency is an over-all thermal evaluation of the design. The discharge temperature is calculated thus:

$$T_e = T_o\, R_c^{\sigma'} \qquad (1\text{-}14)$$

$$\sigma' = (k - 1)/k\, \eta_p = \sigma/\eta_p \qquad (1\text{-}15)$$

$$R_c^{\sigma'} = 2.00^{\sigma'} = 675/520 = 1.298$$

$$\sigma' = \log 1.298/\log 2.00 = 0.376$$

$$\eta_p = \sigma/\sigma' = 0.286/0.376 = 76 \text{ percent.}$$

Without the discharge temperature, the power requirement should have reflected the machines efficiency and the polytropic efficiency can be developed from Fig. 1-1. The polytropic efficiency is usually 2 to 4 percent greater than the adiabatic efficiency. Taking the abscissa as 0.22 (from previous data) and estimating the isoeffic η_p as 75 percent, the R_η is 1.035 and the resolved polytropic efficiency is 75.5 percent. If the *isoeffic* is assumed to be 70 percent, the resolved $\eta_p = 76.6$. If the isoeffic is assumed to be 80 percent, the resolved $\eta_p = 75.5$. This trial and error technique resolves $\eta_p = 75.9$. The polytropic efficiency has merit in establishing a realistic discharge temperature. It remains constant at optimized specific speeds and specific diameters for any gas providing there is no abnormal leakage and the Mach number and the Reynolds affect are equal or adequately corrected. The polytropic head is as useless and misleading as the isothermal head. But most manufacturers use the term in describing performance of their machinery.

It is determined from the equation:

$$L_p = (R_c^{\sigma'} - 1)\, 1545\, T_o\, Z_a/\bar{m}\sigma' \qquad (1\text{-}16)$$

$$L_p = (0.298)\, 1545\, (520)/29\, (0.376) = 22{,}000 \text{ ft.}$$

$$\eta_p = 46\, (22{,}000)/550\, (2450 = 75 \text{ percent.}$$

Caution. There are two important points to remember concerning the various reference efficiences.

1. The realistic discharge temperature can only be determined from the polytropic efficiency as applied to Equation 1-14 or to the ΔH in a Mollier chart solution.

2. The realistic power can be determined from any reference efficiency, if these efficiencies are referenced to

the same real power data. The adiabatic references are the most professional and useful.

Design Criteria. The maximum obtainable efficiencies of turbomachines, together with optimum design geometry, based on the state-of-the-art knowledge can be presented as a function of the similarity parameters, specific speed, N_s, and specific diameters, D_s, for constant Reynolds and Mach numbers.[2] The specific speed and specific diameters are:

$$N_s = N\,(Q^{0.5})/L^{0.75} \qquad (1\text{-}17)$$

$$D_s = D\,(L^{0.25})/Q^{0.5} \qquad (1\text{-}18)$$

$$U = d\,(N)/229 \qquad (1\text{-}19)$$

$$d = 1{,}760\,(q_{ad}/0.55)\;L^{0.5}/N \qquad (1\text{-}20)$$

where $D_s N_s = 147$. The last two equations (1-19 and 1-20) are presented for quickly resolving important criteria, such as the diameter of the impeller (d inches or D feet); U is the peripheral tip speed (fps); Q is the flow in cubic feet per second (cfs); L is the head; N is rpm and q_{ad} is the adiabatic head coefficient.

The latter equation is more frequently stated as:

$$q_{eu} = g\,(L_{eu})/U^2 \qquad \text{or}$$

$$q_{ad} = g\,(L_{ad})/U^2 \qquad (1\text{-}21)$$

$$\bar{q}_{ad} = 11{,}850/N_s^2\,(D_s^2) \qquad (1\text{-}22)$$

The suffix (eu) refers to the Euler equation, Equation 1-12 and g is 32.2 ft./sec.^{2} gravitational acceleration. The head coefficient may have numerous reference bases, such as the adiabatic given in Equation 1-21, where L_{ad} is derived from Equation 1-3. The Euler coefficient is sometimes referred to as the geometric or theoretical pressure coefficient, q_{th}. There are design data available for correcting the geometric (vector analysis) to the realistic q_{ad} values. These factors correct for the number of vanes, the slip, the various inefficiencies of the guide vanes, the rotor, the scroll or diffuser and the discharge velocity conversion.

Mach Numbers. All significant velocities in aerodynamic design are referenced as decimal Mach numbers. Velocities of equivalent Mach numbers have approximately equal resistances in terms of percent of the system pressures. The following Table 1-1 illustrates the behavior.

TABLE 1-1

Gas	Hy-drogen Mixture	Natural Gas	Air	Propane	Butane
$\overline{m}$	6.2	18.2	29	**44**	58
k	1.38	1.26	1.40	**1.13**	1.09
Sonic, fps	2,400	1,350	1,120	**760**	700
0.2 M, Velocity	480	270	224	**152**	140
Velad, ft., ($V^2/2g$)	3,580	1,130	780	**300**	305
Ft./psi	8,750	3,010	1,890	**1,240**	945
Δ psi/Velad	0.41	0.38	0.41	**0.29**	0.32
Equivalent C_m, fps	431	242	202	**136**	126

The *velad* is the gas head (ft.) required to support the 0.2 M velocity, $V^2/2g$, or 480 (480)/64.4 = 3,580 ft. The *head per psi* is determined from the specific volume (144 v_{so}), when divided into the velad, gives the pressure

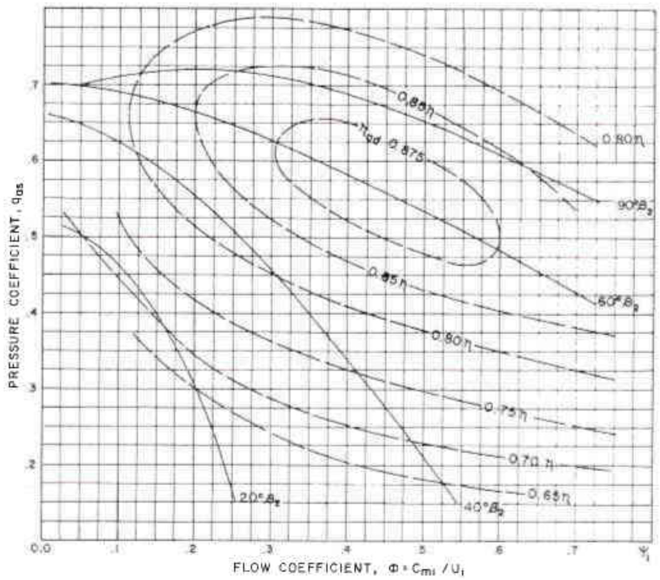

Fig. 1-6—Characteristic centrifugal compressor performance. No discharge losses are included.[2]

drop, Δp *per velad*. The flow-coefficient (ϕ_2) at the point of *surge* is shown on Fig. 1-3 to be about 0.2. If the tip speed U_2 is taken as 0.9 Mach, the meridional suction eye velocity (C_{m1}) for an air compressor would be: $C_{m1} = 0.2\,(0.9)\,1120 = 202$ fps. The same equivalent *surge* would be realized if the impeller had a tip speed of 2,160 fps and the *eye* velocity was 431 fps. If the tip speed remained the same 1,020 fps as for the air case, the flow-coefficient for the greater flow of 431 fps, ϕ_2 would be 0.42 and the impeller would experience the opposite flow-characteristic where *choke* exists. The *eye* velocities for the other gases are given on the bottom line of Table 1-1 for 0.2 and $U_2 = 0.9$ Mach.

Flow Characteristics. The design conditions consist of adiabatic (or polytropic) head and the flow, in actual cfm for a given operation. Fig. 1-4 illustrates a typical master impeller characteristic curve which has been developed by thorough testing. The abscissa is plotted as a unit of flow per revolution, usually cfm/rpm, Q/N or other similar flow references. The solid line represents a single-stage performance. The dash line shows the loss from the return diaphragm in multi-stage operation. The pressure-coefficient establishes the head from Equation 1-21. The dotted line is the product of q_{ad} and η_{ad}. The crest of this curve represents the Best Operating Point (BOP). Impellers are selected so that the design point is as close to this crest as practical. The performance represented by the $D_s - N_s$ intersection on Fig. 1-4 are BOP. The BOP performance for Fig. 1-3, 1-5 and 1-6 are obviously the zones of the highest efficiency.

The pump head (and q_{ad} values) will increase some 5 to 8 percent from BOP as the flow is retarded to the point of *stall* or *surge*. The *surge* zone follows the N_s lines of 40 to 60 on Fig. 1-3. It is well identified on Fig. 1-5. It follows the flow coefficient of 0.10 to 0.15 on Fig. 1-6. The other extreme from *stall* is the maximum flow condition of *choke* or *stonewall*. This flow represents the

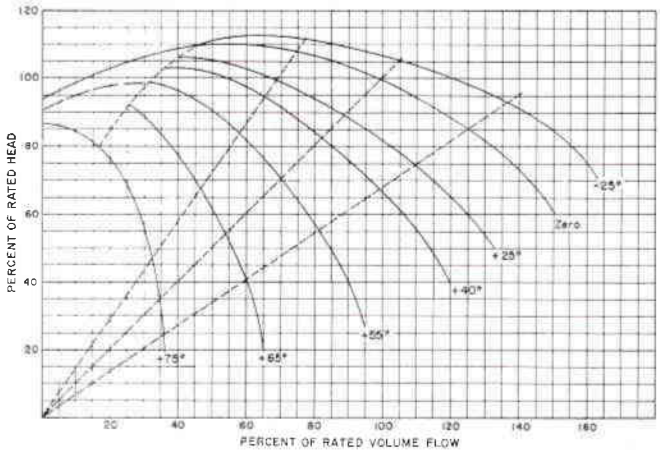

Fig. 1-7a—Effect of inlet guide vanes on the pressure head of a centrifugal compressor.[4]

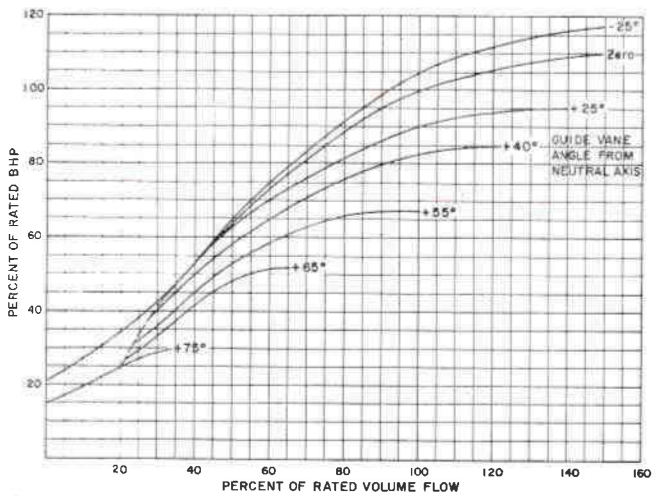

Fig. 1-7b—Effect of inlet guide vanes on the power requirement for centrifugal compressors.[4]

maximum flow that can be drawn through the impeller eye. It is the flow experienced to the right of $\phi_2 = 0.35$ to 0.50 in Fig. 1-3. The abrupt *knee* breaks on the H-Q curves, illustrate the *choke* affect most emphatically. Particular notice should be made of the high q_{ad} values shown in the small circles, ranging from 0.79 at *surge* to 0.77 at BOP and 0.68 at the *knee*. This represents the degree of perfection that NACA has attained.[3]

Slip. Gas that is approaching the eye of an impeller will take on a prerotation in the direction of the impeller turning if the casing does not contain inlet guide vanes. The prerotation tends to reduce the compressor capacity and increase the slope of the head curve. Moveable Inlet Guide Vanes (IGV) are sometimes installed to unload the compressor by exaggerating this characteristic. The reverse action of the IGV can create a counter-rotation which has the effect of supercharging each individual impeller so equipped. This tends to reduce the H-Q and the power curve. These effects are illustrated in Fig. 1-7.[4] The impact of the gas molecules against the back side of a rotating vane distorts the radial flow pattern. The effect of this distortion on the ideal relative velocity vector, W_2 in Fig. 1-2b causes the β_2 to be reduced to β_2' and W_2 to increase to W_2'. When this vector is resolved with the tip speed U_2, the resultant absolute velocity is V_2'. The tangential component of the ideal absolute vector is C_{u2} and for the distorted flow is C_{u2}'. The difference between these two vectors is the *Slip,* V_{s2}. The meridional velocity leaving the tip annulus is C_{m2}', which is presumed to equal C_{m2} and C_{m1}.

Fig. 1-2a shows the same flow behavior for the gas entering the impeller. U_1 is the peripheral speed of the impeller eye. C_{m1} is the meridional vector and represents the average impeller eye velocity. The flow-coefficient ϕ is C_{m1}/U_2 and is applied as the abscicca on Fig. 1-3. There are occasions when a similar *gulp* factor (ϕ') is taken as C_{m1}/U_1 and is so applied in Fig. 1-6. The blade angle at the eye, β_1 includes the relative vector W_1 and the eye velocity U_1. When the guide vanes direct the meridional flow normal to the eye velocity U_1, the horizontal component W_{u1} is equal to U_1. When the inlet guide vanes are adjustable and the gas is given a prewhirl (rotation in the direction of rotation), the relative velocity follows the angle β_1' and vector W_1'.[5] The absolute vector is V_1' and the horizontal component and slip is V_{u1}'. When the incoming gas is given a counter-rotation with regard to the impeller, the flow follows angle β_1^* and the relative vector W_2^*. The absolute velocity is V_1^* and the supercharging head beneficient is V_{u1}^*. The magnitude of the head correction can be evaluated from the Euler equation by applying the following guide angle A degrees of deviation from normal.

$$L_{eu} = [V_{u2}\, U_2 - 1 + A\, (U_1\, C_{m1} \sin A)]/g \qquad (1\text{-}23)$$

The counter-rotation should be treated as a negative quantity, when corrected by the equation minus sign it becomes additive.

The *slip* is largely affected by the number of vanes (n). The effect on the Euler equation for an impeller having back-lay vanes is:

$$L_{eu} = [U_2\, V_{u2}\, (1 - 2/n) - U_1\, V_{u1}]/g \qquad (1\text{-}24)$$

The equation for a radial vane impeller is:

$$L_{eu} = (1 - 2/n)\, U_2^2/g \qquad (1\text{-}25)$$

Flow Coefficient. The two heavy lines striking horizontally across Fig. 1-8 marked ε 1.6 and 1.4, represent the approximate ratio of the OD to the eye diameter. Where N_s is taken as 100 and D_s as 1.47, ε is 1.4 (presuming the impeller is designed for end suction with no shaft extension through the eye and is 16.8 inches in diameter). The suction eye is 18.8/1.4 = 12 inches and has 0.785 square feet area. Presume further that the tip speed U_2 is 1,000 fps and U_1 is (1,000/1.4) = 715 fps. An optimized flow coefficient ϕ_2, value from Fig. 1-3 for a 50-degree β_2 back-lay vane impeller is 0.275. The meridional velocity is: 0.275 (1,000) = 275 fps. The impeller design capacity is: 275 (0.785 square feet) = 216 fps or 13,000 acfm. The flow coefficient ϕ_1 is: (275/715) = 0.385. This number is a reasonable value corresponding to a 50-degree back-lay impeller in Fig. 1-6.

Fig. 1-9 shows impellers having back-lay vanes.

Application Example Problem. The design flow rate is 18,000 mols per hour of 22 mol weight gas, having a k value of 1.26 and both Z values are unity. The gas is compressed from 85° F and 40 psia to 51.6 psia. Develop

•the approximate size impeller, connections, speed, efficiency and power.

Solution. The specific volume is:

$$v_s = 10.73\ (559)\ (1.00)/22\ (40.0) = 6.82 \text{ cf/lb.} \quad (1\text{-}8)$$

$$w = 18{,}000\ (22)/3600 = 110 \text{ pps.}$$

$$Q = 110\ (6.82) = 750 \text{ cfs or } 45{,}000 \text{ acfm.}$$

$$Q^{0.5} = 27.4.$$

$$\sigma = (1.26 - 1.0)/1.26 = 0.206.$$

$$L_{ad} = [(51.6/40)^{0.206} - 1]\ 1545\ (545)\ (1.0)/22\ (0.206).$$

$$L_{ad} = 10{,}000 \text{ ft.-lb./lb.} \quad (1\text{-}3)$$

An inspection of the Baljé and Eckert charts (Fig. 1-3 and 1-6) shows that 100 N_s is an optimum design point and 1.47 D_s is a matching ordinate for a reasonable q_{ad} value of 0.55.

$$L^{0.75} = 1000,\ L^{0.5} = 100 \text{ and } L^{0.25} = 10.$$

$$N = N_s\ L^{0.75}/Q^{0.5} = 100\ (1{,}000)/27.4 = 3640 \text{ rpm.} \quad (1\text{-}17)$$

$$D = D_s\ Q^{0.5}/L^{0.25} = 1.47\ (27.4)/10 = 4.03 \text{ ft.} = 4.03 \text{ or } 48.5 \text{ in.} \quad (1\text{-}18)$$

$$dia = 1760\ (0.55/0.55)\ 100/13{,}640 = 48.5 \text{ in.} \quad (1\text{-}20)$$

The last equation is a simple check of the equation above. The tip speed U_2 is: 48.5 (3640)/22.9 = 770 fps. The flow coefficient is read as 0.32 on Fig. 1-3 at the intersection of N_s = 100 and q_{ad} = 0.55. The meridional velocity C_{m1} entering the eye is: 0.32 (770) = 246 fps. The eye area is: 750 (144)/246 = 440 square inches or 23.6 inches in diameter. The effect of the tip speed Mach numbers on the adiabatic efficiencies is shown in Fig. 1-5.

The efficiencies (η) in Fig. 1-8 relate the L_{ad} to the total inlet pressure and the static outlet pressure. The conditions further presumes that the meridional flow entering the impeller eye (C_{m1}) is equal to the velocity leaving the machine (C_{m3}) and that the Reynolds number is 10^6 or greater. The usual practice in multi-staged compressors is to relate the compression head to the total discharge pressure (η_t). These values are greater than η, especially for N_s values > 60 and for D_s values < 2.0. The difference between η and η_t is that the former discharge pressure does not include the discharge nozzle velocity head which may exceed the suction nozzle velocity that is presumed to be equal to C_{m1}. The process gas line velocities are usually about 0.04 Mach or 40 to 60 fps for common (20 to 30 $\overline{m}$) gases. The nozzles on multi-stage compressors are operated at velocities of 75 to 150 fps at design loads. The discharge nozzle is two-thirds the diameter of the suction nozzle on multi-stage casings. The suction and discharge nozzles are usually the same size on single or two-stage cantilevered type compressors.

The discharge temperature for this example at 80 percent adiabatic efficiency is: 545 $(51.6/40)^{0.206/0.8} = 582°$R or 122° F. The discharge volume is: 750 cfs (1.0677/1.29) = 620 cfs, v_{se} = 5.65 cf/lb.

The process suction line would be sized at 50 fps: 600 (144)/50 = 1,790 square inches or 43 inches.

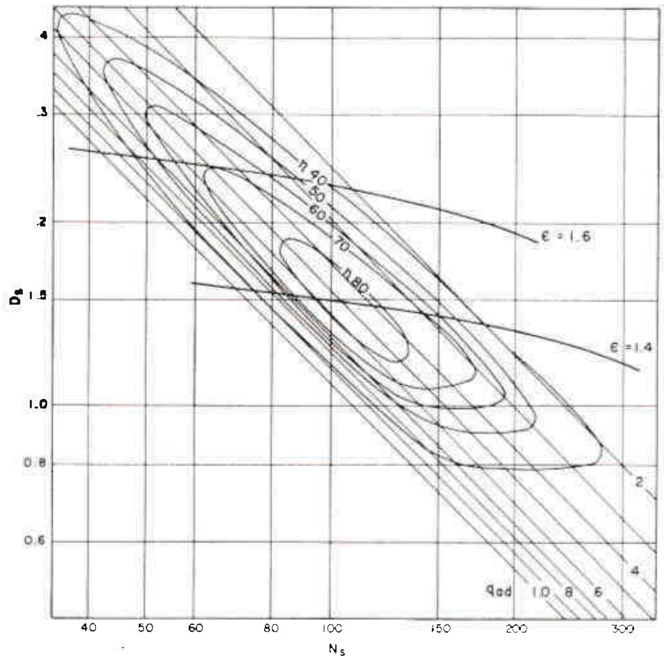

Fig. 1-8—Balje', generalized performance curves, orienting the adiabatic-dynamic efficiency and the pressure coefficient as a function of the specific speed and specific diameter.

The compressor casing for this volume has 30 by 20-inch nozzles. The suction velocity is: 750 (144)/685 = 157 fps. The discharge velocity is: 620 (144)/300 = 298 fps. The discharge static pressure correction is calculated from the difference in velocity (velads): $(298 - 157)^2/64.4$ = 305 feet. The energy density on the discharge line is 305 ft.-lb./lb. short of the objective 10,000 ft.-lb./lb.

The conditions for using the efficiency values from Fig. 1-8 required C_{m1} to be equal to C_{m3}. The corrected efficiency from conditions of STATIC DISCHARGE to TOTAL DISCHARGE conditions is: (10,000 - 305) 80.0/10,000 = 77.5 percent.

This condition can be corrected by providing a 27-inch discharge nozzle on the compressor casing, or by installing a 20 x 30 divergence tube 100 inches long to cover the excessive discharge velocity. The power required to operate the 3,640 rpm single-stage compressor is:

$$\text{Dynamic } hp = w\ L_{ad}/550\ n \quad (1\text{-}26)$$

$$= 110\ (10{,}000)/550\ (0.775) = 2{,}580$$

The mechanical losses from the bearings, etc., which do not affect the temperature of the gas, are not included in Fig. 1-8. It is proposed that the mechanical losses are well within the frictional hp of the equation: fph = (dynamic hp)$^{0.4}$

The frictional hp for this case would be $(2580)^{0.4}$ = 25. The driver should include this value, plus a 10 percent power tolerance, which totals 2,865 bhp at 3,640 rpm.[6]

Corrections. The data plotted on Fig. 1-8 included as much test and operating performance as available. The curves were developed from loss analysis, simplifying assumptions and averaging. The correction for the high discharge velocity is an example of how other corrections may be applied. The Baljé efficiencies include the normal interstage leakage and disk friction. Subsequent parts of

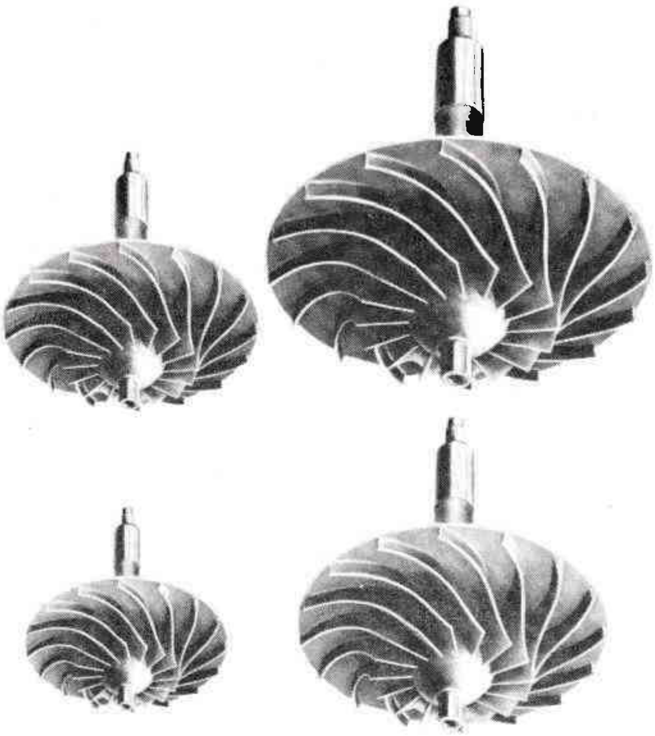

Fig. 1-9—Investment cast stainless steel (17-7PH) impellers having back-lay vanes and without a shroud. The performance for a similar impeller is shown in Fig. 1-5. (Courtesy of Chicago Pneumatic Tool Co.)

this series will illustrate how to cope with the abnormal leakage, disk friction evaluation, multi-staging losses, materials, ultimate speeds, control features and economics.

NOMENCLATURE

- *acfm* Flow rate, actual cubic feet per minute
- *acfs* Flow rate, actual cubic feet per second
- *bhp* Brake horsepower
- C Characteristic design factor
- C_m Eye velocity, fps
- *cfm* Flow rate, cubic feet per minute
- *cfs* Flow rate, cubic feet per second
- d Diameter of impeller, inches
- d_e Diameter of impeller eye, inches other specific references are identified by suffix given by text application
- D Diameter of impeller, feet
- D_s Specific diameter
- F Force as in thrust, etc., pounds or temperature °F
- *fhp* Frictional horsepower
- *fps* Velocity, feet per second
- g Acceleration of gravity, 32.2 ft./ sec./ sec.
- H Enthalpy, Btu per pound.
- hp_{dy} Dynamic horsepower
- k Ratio of specific heats at mean compression temperature
- L Feet of pipe line or feet of gas head, usually adiabatic
- L_{ad} Feet of adiabatic gas head, ft.-lb./lb.
- L_{eu} Feet of Eulers' gas head, ft.-lb./lb.
- L_{iso} Feet of isothermal gas head, ft.-lb./lb.
- L_p Feet of polytropic gas head, ft.-lb/lb.
- $\overline{m}$ Molecular weight
- M Mach number
- *MMscfd* Million standard (14.7 and 60° F) cubic feet per day
- N Revolutions per minute, rpm
- N_s Specific speed
- n Number of elements
- *ppm* Pounds per minute
- *pps* Pounds per second
- P Pressure, psia
- Q Quantity flow rate, cubic feet per second, cfs
- q_{ad} Pressure coefficient
- R_c Ratio of compression, P_2/P_1
- R Universal gas constant, $1{,}545/m$
- S Stress in material, psi
- T Absolute temperature, °R
- U Tip speed, fps
- v_s Specific volume, cf/lb.
- v_{so} Suction specific volume, cf/lb.
- *velad* Velocity head, $V^2/2g$
- V Velocity, feet per second
- w Flow rate, pounds per second (pps)
- W Flow rate, pounds per hour
- $\overline{w}$ Flow rate, molal pounds per hour
- X Unity gas constant as related to the adiabatic head
- Z Gas compressibility factor

GREEK LETTER SYMBOLS

- α Entrance angle of impeller eye
- β Exit angle at impeller tip
- Δ Difference in pressure, temperature, length, etc.
- δ Specific weight, lb./cf.
- ε Ratio of d/d_e
- η Efficiency, usually adiabatic unless specified as isothermal, polytropic or otherwise by suffix abbreviations
- Θ Valve loss, percent of system pressure
- μ Kinematic viscosity, ft.²/sec.
- σ Exponential function of ratio of specific heats (k) (k-1)/k
- τ Sonic velocity, fps
- ϕ Flow coefficient

The suffix *o*, 1, 3 and add numbers represent suction condition. The lower case *e* and even numbers represent exit conditions. The exceptions which are made in the text have precedence over this statement.

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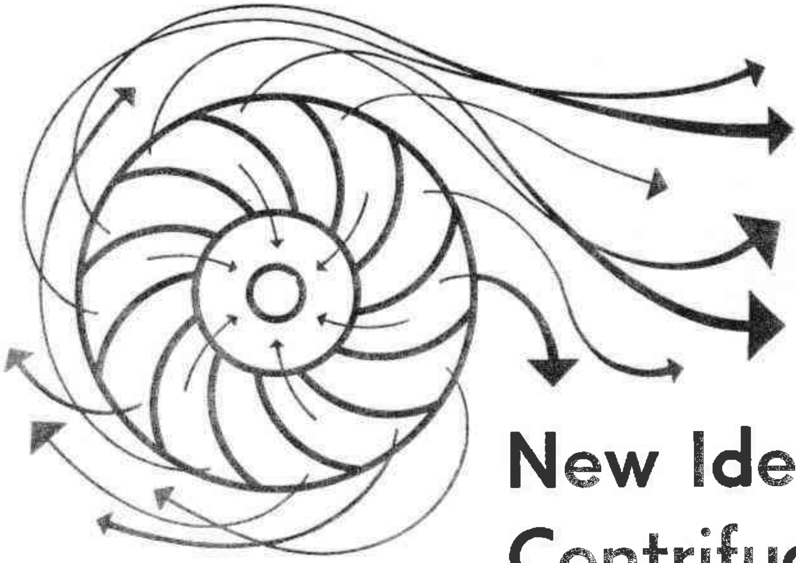
Indexing Terms: Adiabatic-6, **Centrifugal-9**, Compressors-9, **Design-8**, Diameter-7, Efficiency-7, Flow-6, **Horsepower-7**, Impellers-9, Isentropic-6, Isothermal-6, Pressure-6, Rating-8, **Speed-6**, **Surging-6**, Temperature-6.



New Ideas on Centrifugal Compressors

This three part series describes the following: Part 1: power rating method and sizing, Part 2: metallurgy and design, Part 3: shaft seals and balance pistons

Lyman F. Scheel, Earhart & Associates, Los Angeles

Part 1 presented a method of evaluating the compression head and the range of inlet volume flow to provide efficient performance. The head, speed and impeller diameter are determined from the equations of Part 1.

$$L_{ad} = (R_c^{\sigma} - 1)\ 1545\ T\ Z/\overline{m}\sigma, \text{ ft.-lb./lb.} \qquad (1\text{-}3)$$

$$U = d\ (N)/229, fps \qquad (1\text{-}19)$$

$$d = 1{,}760\ (q_{ad}/0.55)\ L^{0.5}/N \qquad (1\text{-}20)$$

In Part 1, it was stated that the head limitation per stage is 10,000 ft.-lb./lb., which is a tip speed of 800 fps when q_{ad} is 0.503. This limitation is largely provoked by the means of securing the shroud. This part will exploit the metallurgical and design requirements.

The equation for determining the stress developed in the rotating impeller is:

$$S' = \delta\ C\ U^2/g, \text{ lbs./sq. ft.} \qquad (2\text{-}1)$$

The densities of several likely materials used in impeller design are:

TABLE 2-1—Density of impeller materials.

Material	lbs./cu. ft.
Bronze, Inconel, Monel	550
Stainless and Carbon Steels	485
Titanium Alloys	300
Aluminum Alloys	170

The relative effective strength of material for making impellers is best expressed as the quotient of the yield strength divided by the density.

Fig. 2-1 lists 16 impeller materials giving this strength-density ratio versus operating temperature. Titanium far out-classes all other materials. Aluminum alloys out-class the stainless steels, except type 410. The above equation is modified to produce the effective tip speed U:

$$U = k\ (S'/\delta)^{0.5}, \text{ fps} \qquad (2\text{-}2)$$

where $k = (32.2/C)^{0.5}$ and $S_{psi} = S'/144$

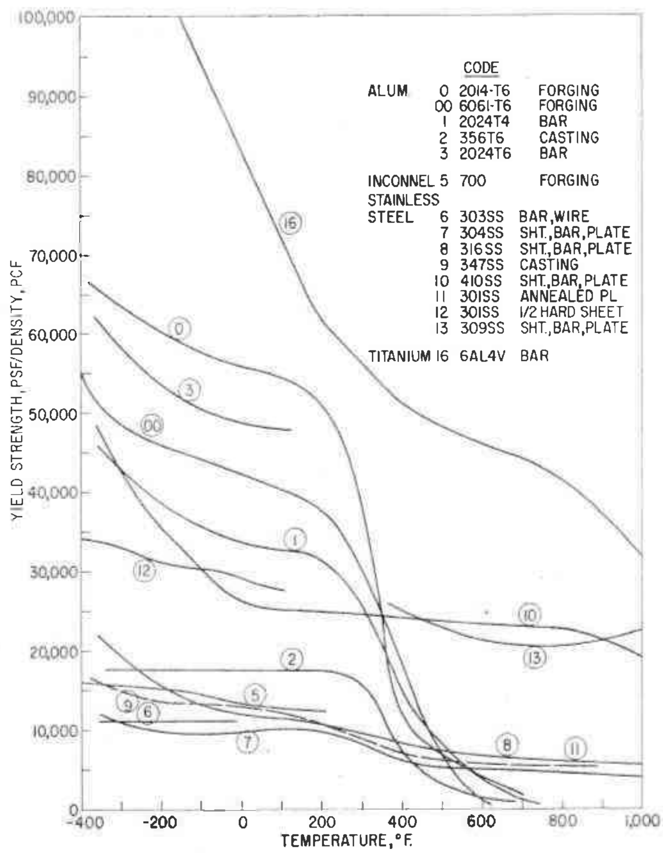


Fig. 2-1—Strength-density ratio versus operating temperature for various alloys.

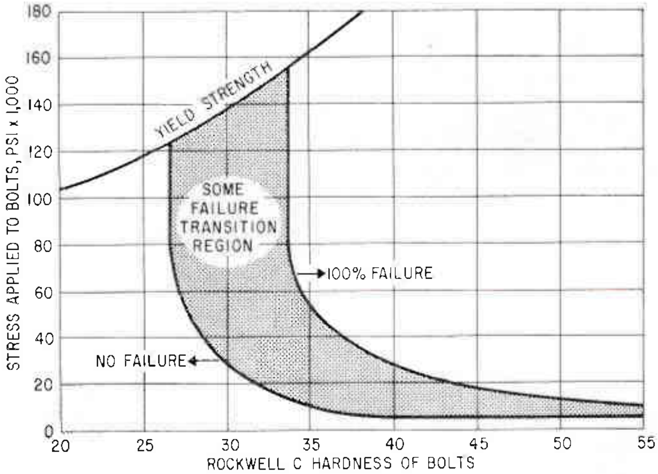


Fig. 2-2—Cracking susceptibility of AISI 4140 bolts in H_2S—H_2O for one year at 40° C and 250 psi. (From Warren & Beckman, Corrosion, Vol. 13, No. 10, (1957).

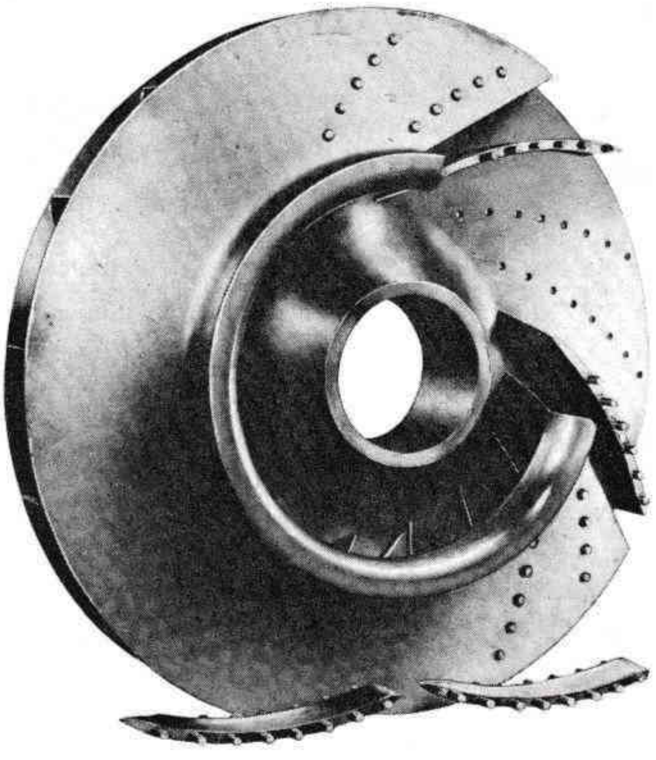


Fig. 2-3—Sectional view of Class 1, riveted construction impeller having back-lay vanes. (Courtesy of Allis Chalmers Mfg. Co.)

TABLE 2-3—Tip speeds at 60 percent yield point.

Materials	(Feet per second) 3	5	9
Aluminum 6061-T6 at 500° F.	550	650	1,450
Aluminum 6061-T6 at 100° F.	1,200	1,450	3,200
Aluminum 6061-T6 at −350° F.	1,550	1,850	4,100
Type 316 S S at 100° F.	650	800	1,750
Type 410 S S at 100° F.	950	1,150	2,500
Titanium 6 AL 4V at 500° F.	1,300	1,500	3,600
Titanium 6 AL 4V at 100° F.	1,600	1,900	4,200
Titanium 6 AL 4V at −350° F.	2,100	2,500	5,500

An evaluation of the C and k design factors for various configurations is given below:

TABLE 2-2—Classification of impeller types.

Class	Description of Impeller	C	k
1.	Back-lay vanes, with riveted shroud	1.3	5.0
2.	Radial vanes, with riveted shroud	1.0	5.7
3.	Back-lay vanes, with welded shroud	0.9	6.0
4.	Radial vanes, with welded shroud	0.8	6.3
5.	Radial milled vanes, no shroud	0.6	7.3
6.	Axial blades, b/d = 0.25, fir-tree	0.7	6.8
7.	Axial blades, b/d = 0.15, fir-tree	0.5	8.0
8.	Axial blades, b/d = 0.10, fir-tree	0.32	10
9.	Axial blades, b/d = 0.05, milled	0.12	16

The b/d in Table 2-2 relates the axial blade length to the outer diameter.

The design factors in Table 2-2 utilize 50 to 75 percent of the minimum yield strength.

The above factors were generalized from many sources.[2,6,7,8,9] One reference states that "higher admissible tip speeds may be attained with rotor geometries which are carefully stress balanced by rigorous design analysis."[2] Some 200 aluminum impellers, ranging from 1 to 1,600 pounds and from 12 to 60 inches in diameter were spin tested to 70,000 rpm and tip speeds of 3,000 fps. Most of the impellers elongated (diametrically) 0.1 to 0.5 percent before failing.[10]

This elongation usually functions as a brake. When the impeller makes contact with the casing, it drags to a stop.

All industrial impellers are given a *spin-test* of at least 115 (some 121) percent of maximum continuous design speed before assembly. When the compressor is motor driven at constant speed, the impeller integrity is thereby assured to be operating below 75 percent of the minimum yield strength. Steam turbine driven compressors can be operated at 110 percent of design speed. The turbine governor is set to trip at 110.3 percent. The square of this increased speed times the 75 percent minimum yield, produces an impeller stress of 91.4 percent of the minimum yield strength. This 8.6 percent is an uncomfortably small safety margin to allow between the *spin-test* stress and the stress that a turbine governor may permit, particularly at startup when the turbine driver has the potential for much greater speed. The spin-test should be 115 percent of the turbine governor trip speed to maintain the working stress about 60 percent of the minimum yield strength. Table 2-3 was prepared to show the tip speed at approximately 60 percent of the yield point for several designs, materials and temperatures.

It is not uncommon to use 80 percent of the minimum yield strength at the coupling end of the shaft. The shaft is not exposed to corrosive conditions at this point. Inside of the casing, the shaft size is fixed by the critical speed rigidity requirements. The internal shaft stress is between 3,000 and 5,000 psi.

Carbon alloy steels and stainless steels of equal strengths have equivalent performance at ambient temperatures. The performance of ferritic type 410 SS is equivalent to aluminum at cryogenic temperatures. Austenitic 18-8 stainless steels are better than aluminum at temperatures above 250° F. Carbon steels are impractical to use at cryogenic temperatures. Martensitic nickel alloy steels should be used for this purpose. We know that some riveted Class 1 impellers, made of ferritic type steels, are spin-tested to 1,000 fps. They experience sufficient distortion in the shroud that the periphery must be remachined and balanced. Class 3 impellers are spin-tested to 1,250 fps without distortion. It was recently reported that a Class 9 ferritic steel impeller withstood 2,750 fps spin test without elongation. We also know that jet engines having Class 6, 7 and 8 impellers operate at blade velocities in excess of 1,800 fps during *take-off* in every jet engine powered aircraft. They cruise at 1,200 fps and have a remarkable safe record doing both operations. The metallurgy problem of turbomachines are minor compared to the design inadequacies.

Corrosion engineers of NACE have made an extensive study of the metallurgy problems that have beset some 300 refinery compressors for the past 25 years.[8] The mean working stress on the impellers of these units is 37,500 psi, by Equation 2-1. Ten percent of these machines are working on a stress less than 25,000 psi and 15 percent exceed 50,000 psi. Seventy percent of these machines were relatively trouble free from the startup. One-third of the remaining units, or 10 percent of the total, still have troublesome maintenance problems. There were eight impeller failures. One impeller failed at the periphery and could be classified as a metallurgy failure. It was diagnosed as sulfide stress cracking caused by excessive hardness. Another companion impeller definitely failed at a sharp cornered key slot. The other six failures were the result of similar design inadequacies and showed evidence of stress concentrations. In another instance loose shroud rivets were welded without annealing. This mistake was the cause for three impeller failures.

Clean Seal Oil. Two-thirds of all troubles involved the shaft seals. Twenty-five percent of the trouble was from dirty lube oil. The deluxe lube oil consoles of recent years have undoubtedly reduced this problem. Fig. 2-4 illustrates a comprehensive lube oil console. The balance of the maintenance concerns the shaft seals, equally divided between the oil film type and the labyrinth. The latter are usually made of aluminum and set in the casing. The knife edges surround the shaft. Where the temperature exceeds 250° F, a soft stainless steel labyrinth is recommended. European manufacturers usually machine the labyrinth edges on the shaft and set the knives on a soft babbit or aluminum half-cylinders secured in the casing. There was only one valid case of corrosion in a fluid catalytic cracking unit which was handling sour hydrogen sulfide gas. In this instance the 11-13 chrome impellers were upgraded to 17-7PH. AISI 4140 is the most prevalent steel alloy used for impellers and shafts for all services. Corrosion engineers recommend that the heat treatment be held to 27 Rockwell C, which still permits a yield stress of 127,000 psi. Fig. 2-2 shows the effect of hardness on the yield strength in a hydrogen embrittlement environment.[8]

Impeller Attachment. A half-sectional profile of an impeller disk is a right angle triangle shape having a peripheral angle of about 16 degrees. The profile base includes a hub which projects ½ to 1 inch beyond the shaft. The design hub is significant in that it constitutes the impeller suction eye configuration, plus it transmits the driving torque from the shaft, provides the necessary disk rigidity and fixes the impeller balance position. The USA manufacturers use square keys staggered 90 degrees to complement the balance of multi-stage machines. The impeller hubs are reamed to a *push* fit of 0.001-inch clearance or to a *drive* interference fit of 0.003 inch. Cantilever single and dual-stage machines generally do not have a key and are given an approximate 0.009-inch interference fit. The shaft is drilled so as to require a 2-ton hydraulic jack to remove the impellers. European practice usually avoids keys and uses a heat shrink of about 0.006 inches. This can be accomplished by an atmospheric steam hose applied to the impeller which gives 140° F of expansion or 0.001 inch per diametrial inch.

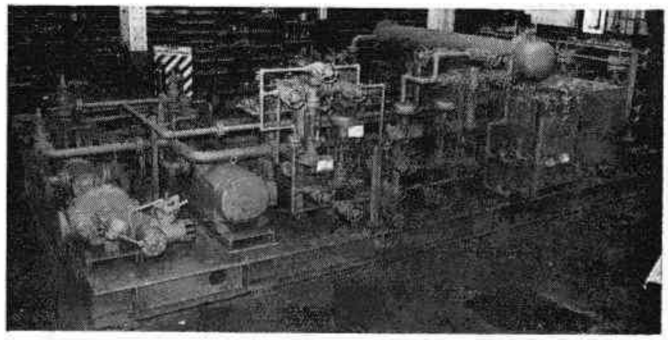


Fig. 2-4—Modern combination seal and lube oil console for major centrifugal compressor. (Courtesy of Cooper-Bessemer Co.)

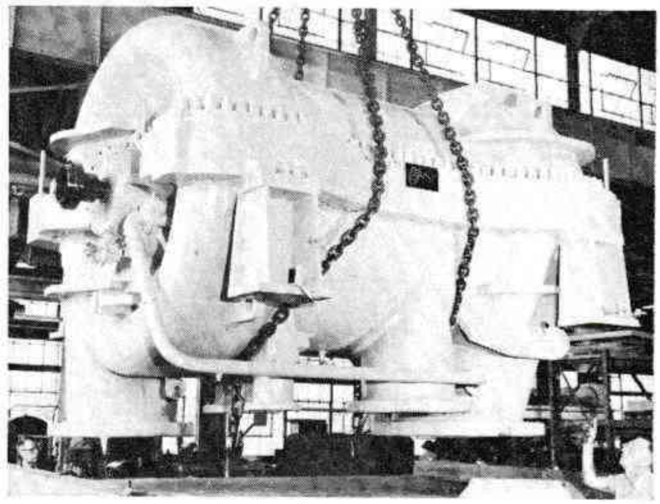


Fig. 2-5—This large centrifugal compressor is made from weldment sections. The unique design and application won the 1967 Lincoln Welding Award. The unit has capacity in excess of 100,000 acfm and 20,000 hp. (Courtesy of Elliott Co., Div. Carrier Corp.)

The strength of an interference fit is determined from the following equations. The radial pressure P_c on the shaft contact surface is:

$$P_c = \Delta E (d_s^2 - d_i^2)(d_h^2 - d_s^2)/2d_s^3 (d_h^2 - d_i^2) \qquad (2\text{-}3)$$

Where the shaft diameter (d_s) is 5.5 inches and (d_i) is the internal bore or zero when solid, the impeller hub (d_h) is 7 inches and E is the 30,000,000 psi the elasticity modulus for steel. The interference is delta (Δ).[9] The tangential stress S_t is determined from:

$$S_t = P_c (d_h^2 - d_s^2)/(d_h^2 + d_s^2) \qquad (2\text{-}4)$$

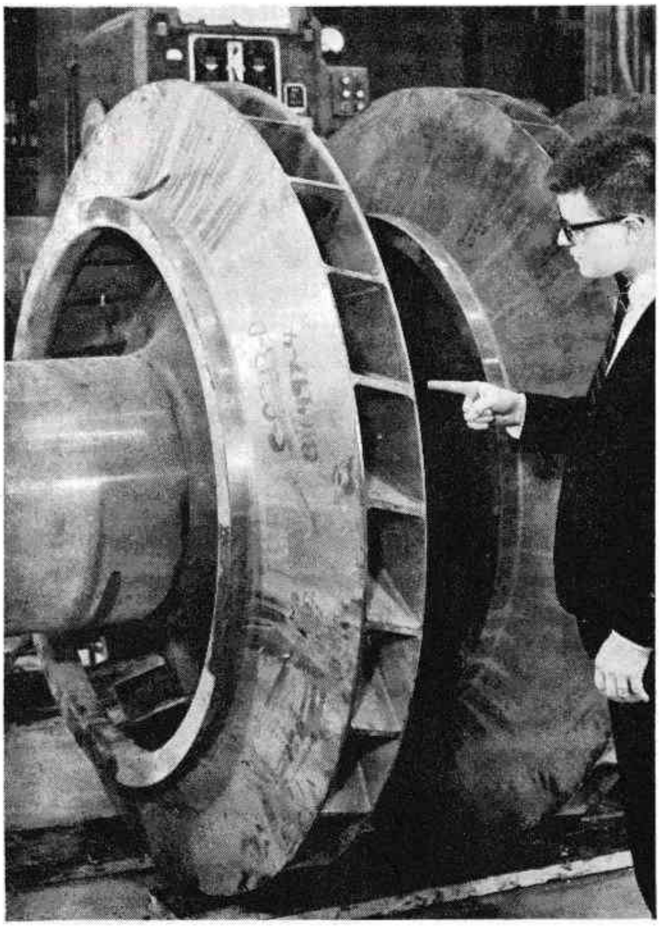


Fig. 2-6—Class 3 impeller for twin flow, three stage impeller for large capacity, low head centrifugal compressor. (Courtesy of Elliott Co.)

Where the tangential stress is limited to 8,000 psi and applied to the above equation, the surface contact pressure P_c is 1,900 psi. When this P_c is applied to Equation 2-3 it requires an interference of 0.00264 inch.

The force required to remove the impeller from the shaft is:

$$F_s = f \pi d_s L_s P_c \tag{2-5}$$

Where f is the coefficient of friction (0.12), L_s is 4 inches (the length of the axial contact of the impeller on the shaft), F_s = 0.12 (3.14) 5.5 (4) 1,900 = 15,700 pounds.

The torque transmitting capability is: 15,700 (2.75/12) = 3,600 ft.-lb.

The power potential without slipping is: 3,600 (6,000 rpm)/5,250 = 4,100 hp.

The torsional stress in the hub is usually modest and seldom exceeds 10,000 psi. The size of the shaft is predicated upon the critical speed characteristics. The nominal stress is:

$$S_t = 5.1 \text{ (Torque, in.-lb.)/dia.}^3 \tag{2-6}$$

For the 5.5-inch diameter shaft just cited, the torsional stress would only be 1,210 psi.[9] A sharp cornered keyway with a snug shaft fit can develop stress densities two to three times the nominal stress. A loose fit would compound this stress concentration. The use of two close fitted *feather* keys has merit. The thickness of a *feather* key is one-fourth of the width (peripheral) dimension. All keyways should have well rounded fillets. After studying the corrosion engineers' report on refinery turbomachinery problems and from discussion with the manufacturers, it is the writer's conviction that an impeller attachment specificaton is needed to guard against loose attachments, sharp cornered and overstressed keyways.[8]

A second conviction is that all impellers of sophisticated design should be an investment casting or fabricated by arc-welding and/or milled.

Materals for Casings. The API Standard 617 set up the following requirements for steel casings.[6]

- Air or nonflammable gas at a design pressure over 250 psig.
- Air or nonflammable gas at a calculated discharge temperature over 450° F at any point within the range of the maximum continuous speed.
- Flammable or toxic gas at a design pressure over 75 psig.
- Flammable or toxic gas at a calculated discharge temperature over 350° F at any point within the range of the maximum continuous speed.

Metallurgy.

- Cast steel for the above categories operating under 500° F and 1,050 psig are satisfied by the ASTM Spec. A216 Grade WC-8. Weldments are equally acceptable when in full compliance with the ASME Section VIII Code.
- Operating temperatures and pressures greater than 500° F and 1,050 psig require forged steel which complies with ASTM Spec. A-235 or A-237.
- For operating temperatures below −20° F, the steel shall have an impact strength of not less than 15 ft.-lb. (ASTM E 23). ASTM Spec. A-352 Grade LC-B is satisfactory for −50° F and Grade LC-3 is good for −150° F. These martensitic steels contain 2.25 percent and 3.5 percent nickel respectively.

Cast iron is acceptable for all other applications except those specifically defined above. The applicable cast iron specification is ASTM-A278. Horizontal-split cast steel or weldment cases are used for gas pressures up to 650 psig, discharging less than 350° F and having a molecular weight greater than 16. Pressures above 650 psig and less than 1,050 psig require a cast steel or weldment *barrel* design. Pressures in excess of 1,050 psig require a forged steel barrel design. Where the partial presure of hydrogen or helium in the gas handled exceeds 250 psig, the barrel design should be required. The flat joint of the horizontal-split case cannot contain these low molecular weight gases. Weldments are proving popular in lieu of cast steel casings because of the improved delivery. A faulty steel casting can set the schedule back two months. This can happen all too often and accounts for most late deliveries of cast steel casings. The manufacturer has complete control of the production schedule for weldment casings. They can be fabricated for approximately the same cost (see Fig. 2-5).

Nodular (ductile) iron offers an alternative to cost steel and weldment casings at a lower price. There appear to

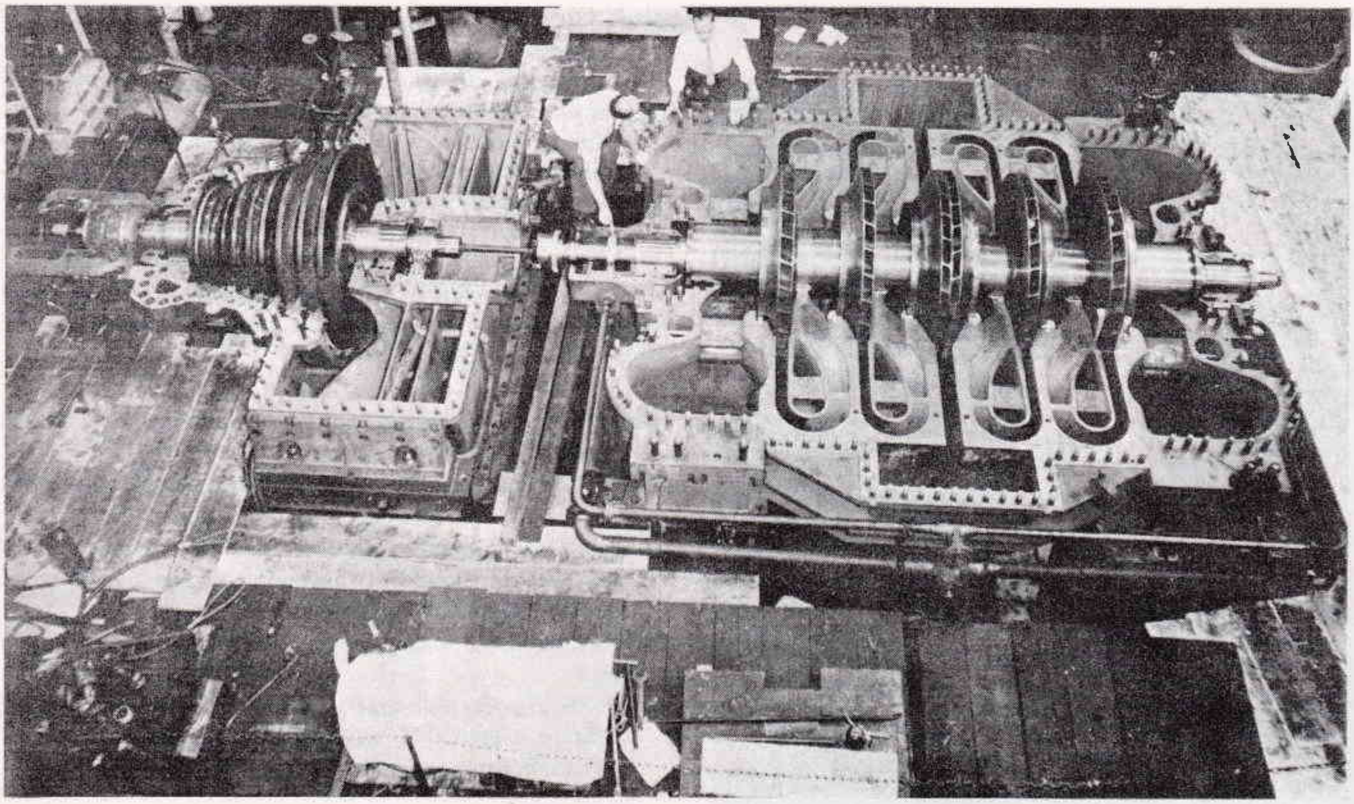


Fig. 2-7—The impeller in Fig. 2-6 forms the two first stages of this twin, three-stage compressor. The steam turbine driver on the left illustrates the four categories of axial flow rotors. The Class 6 blade height over the rotor diameter ratio of 0.25 is nearest to the compressor. The Class 9, 0.05 b/d rotor is on the extreme left hand end of the shaft. Class 7 and 8 have intermediate b/d ratios. (Courtesy of Elliott Co.)

be certain prejudices concerning the use of nodular iron. It can be cast with the same confidence that cast iron can be delivered. Weld repairs can be made as readily as cast steel and better than with cast iron. ASTM Spec. 395-61 Class 60-45-15 provides 15 percent elongation. ASTM Spec. 439-62, Class D-2C and D-5 provides 20 percent elongation. This ductility is equivalent to cast steel and its alloys at normal operating temperatures. At elevated temperatures as experienced in a major fire, the superior strength of cast steel is granted. However, by the time the superiority of the cast steel is evident, the system pressure should be vented by the protective devices or by a ruptured vessel in the system.

Testing. Mechanical Running Tests should be performed as specified in API Std. 617. Some form of performance tests should be made and witnessed. The test procedure proposed by O'Neill and Wickli is recommended to resolve the test data.[11] Any head-capacity curves that can be developed from these tests should be useful in establishing the credibility of the machine's performance. These tests are especially valuable when the compressor design exceeds the normal parameters.

NOMENCLATURE

- b Axial blade length, inches
- C Characteristic design factor
- d Diameter of impeller, inches
- E Modulus of elasticity of steel
- F Forces as in thrust, etc., pounds of temperature °F
- g Acceleration of gravity, 32.2 ft./sec./sec.
- k Ratio of specific heats at mean compression temperature
- L Feet of pipe line or feet of gas head, usually adiabatic
- L_{ad} Feet of adiabatic gas head, ft.-lb./lb.
- m Molecular weight
- N Revolutions per minute, rpm
- P Pressure, psia
- Q Quantity flow rate, cubic feet per second, cfs
- q_{ad} Pressure coefficient
- R_c Ratio of compression, P_2/P_1
- S Stress in material, psi
- T Absolute temperature, degrees Rankine
- U Tip speed, fps
- Z Gas compressibility factor
- δ Specific weight, lb./cf.
- σ Exponential function of ratio of specific heats (k), $(k-1)/k$

The Suffix *o*, 1, 3 and add numbers represent suction condition. The lower case *e* and even numbers represent exit conditions. The exceptions which are made in the text have precedence over this statement.

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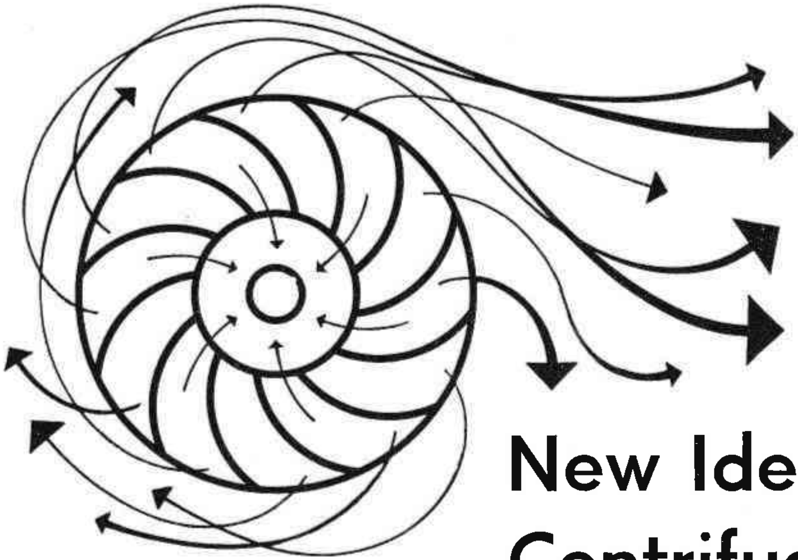
[6] API Standard 617, "Centrifugal Compressors for General Refinery Services."
[7] Prescott, John, "Applied Elasticity," Dover Publications.
[8] Neill, Heller & Muller, "Corrosion Experiences of Refinery Compressors," NACE, March 22, 1968.
[9] Marks & Baumeister, Handbook for Mechanical Engineers, McGraw-Hill Book Co., New York, 1967.
[10] Anderson, R. G., "High-Speed Rotating Parts," Machine Design, Oct. 17, 1967.
[11] O'Neill, P. P., and Wickli, H. E. "Predicting Process Gas Performance of Centrifugal Compressors from Air Test Data," ASME Paper No. 61-PID-6.
Note: Literature cited items 1-5 may be found in Part 1.

Indexing Terms: Balancing-7, Casings-9, Centrifugal-4, Classifications-6, Compressors-4, Density-6, Factors-6, Failures-7, Hardness-6, Hubs-9, Impellers-9, Lambrinth-9, Materials-8, Mechanical Properties-6, Metallurgy-8, Physical Properties-6, Properties/Characteristics/-6, Sealing-6, Speed-7, Strength-6, Stresses-7, Torque-7, Yield Strength-6.



New Ideas on Centrifugal Compressors

This three-part series describes the following: Part 1: power rating method and sizing, Part 2: metallurgy and design, Part 3: shaft seals and balance pistons

Lyman F. Scheel
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THE DEVELOPMENT OF CENTRIFUGAL compressors for high-pressure services has been greatly limited by the capabilities of the shaft seals. This part will describe the various shaft seals and their performance in typical process compressor service.

Shaft Seals. The shaft seals may be divided into the following categories:

- Labyrinth
- Restrictive carbon rings
- Mechanical face contracting rings
- Floating bushing, oil film, and
- Dynamic or centrifugal pumping chokes

The first two seal categories are usually operated dry, when the casing pressure is less than 100 psig and the outer terminal exhausts to the atmosphere.

Labyrinth seals are used extensively for interstage shaft and impeller eye sealing. Fig. 3-1 illustrates the simplest form of labyrinth seal as it is applied to the shaft. Most labyrinth seals are machined from bronze, babbitt or aluminum and fit in the casing in the form of a horizontal split cylinder. Other designs use aluminum rings cut from coil strip about 0.020-inch thick. The labyrinth rings are caulked into small grooves machined in the casing. An alternate type of coil strip blading has a J-section form. The hook form is pressed into the groove, instead of using the more tedious caulking procedure. When the temperature exceeds 250° F, Monel or stainless steel strips are used. The labyrinth seals are fit as snug as possible or given a slight negative clearance.

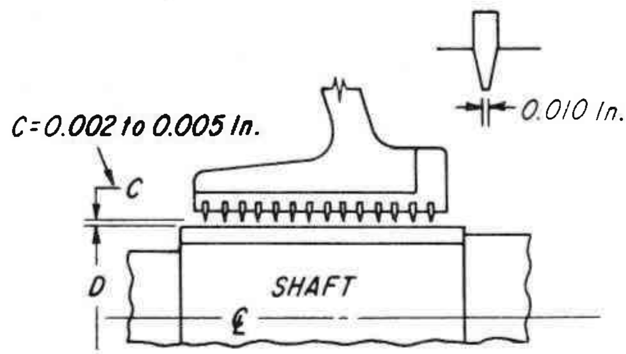


Fig. 3-1—Straight labyrinth seal as used on an interstage shaft, on suction eye seals, and on low-pressure outside seals. Staggered and tapered diameters can reduce the leakage as much as 40 percent.

The clearance usually provided for turbomachinery shaft seals, using labyrinth blading, is about 0.002 inch. The clearance is considered excessive when it exceeds 0.030 inch. Labyrinth seals on casings operating at pressures in excess of 50 psig have 8-20 blades. Lower pressure casings contain 3-6 blades. Leakage tolerance is frequently stated as approximately 0.5 percent of the compressor capacity. This is a rather crude and irrelevant gage. The leakage can be estimated with reasonable accuracy by the following procedure. The velocity of the gas through the annulus of the smooth shaft and sharp blades as shown in Fig. 3-1, is:

$$V = 100\,[\Delta P\, v_{sm}/\,(1.5 + 1.5/(n-1))]^{0.5}\ (fps) \qquad (3\text{-}1)$$

When ΔP is the pressure differential, v_{sm} is the mean specific volume and n is the number of blades or barriers. A staggered or stepped pattern of labyrinth will reduce the leakage by 40 percent.

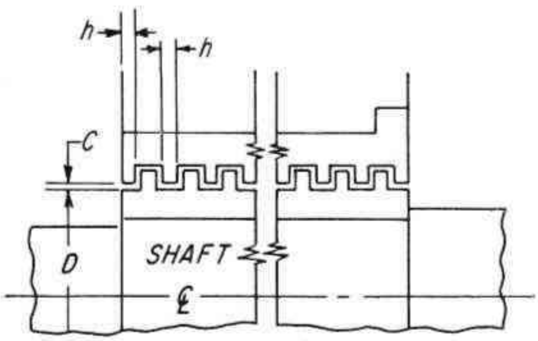


Fig. 3-2—Interlocking labyrinth seals require clearances of 0.014 plus 0.004 per diametrical inch. Clearances greater than 0.030 inch are considered excessive.

If the sharp blades are replaced with castellated, square concentric lands similar to Fig. 3-2 except that the shaft has no interlocking features, the velocity is derived from this equation.

$$V = 100\,[\Delta P\, v_{sm}/(1.5 + (fh/\,2C) + (n - 1))]^{0.5}\ (fps) \qquad (3\text{-}2)$$

Where f is the Reynolds number (R_e) frictional factor, C is the radial clearance and h is the axial length of each barrier, both expressed in inches.

$$R_e = 2\,C\,V/12\,\mu \qquad (3\text{-}3)$$

Where μ is the kinematic visosity in centistokes, ft.^{2}/sec. and V is velocity, fps.

$$f = 0.33/R_e^{0.25}\ \text{(turbulent flow)} \qquad (3\text{-}4)$$

$$= 64/R_e \qquad (\text{laminar, } R_e < 2300)\ \text{(Lit. Cited 12)}$$

Fig. 3-2 shows an interlocking labyrinth consisting of square concentric lands. The velocity flow for such a resistance is:

$$V = 100\,[\Delta P\, v_{sm}/(1.5 + (fh/2\,C) + 2\,(n - 1))]^{0.5}\,(fps) \qquad (3\text{-}5)$$

Where the labyrinth is omitted and the smooth sealing surface is fit against the straight shaft, the velocity is:

$$V = 100\,[\Delta P\, v_{sm}/(1.5 + fh/2\,C)]^{0.5}\,(fps) \qquad (3\text{-}6)$$

The flow Q (cfs) is the product of the annulus area (A_s) expressed in square feet and the respective velocities, $Q = A_s\,V$.

Example. The following example will illustrate the application of the above equations. A FCC Main Air Blower handles 133,000 cfm or 167 pps of air from 60° F and 14.5 psia to 46.0 psia. The unit is driven by a 17,500-hp motor at 3,600 rpm and the adiabatic efficiency of the five-stage compressor is 65 percent. The compressor performance and shaft leakages are shown below:

TABLE 3-1—Performance and Leakage of 17,500-hp Main Air Compressor

STAGE	1	2	3	4	5
Discharge Pressure, psig	18.2	22.8	28.6	36.8	46.0
(T_1 = 520° R); T_2 =	568	620	676	740	805
Shaft, ΔP	1.8	2.2	2.7	3.7	4.2
Eye, ΔP	2.8	3.5	4.4	6.1	6.9
Mean Pressure, P_m	16.4	20.5	25.2	32.7	41.4
Specific Volume, v_{sm}	12.8	11.2	9.8	8.4	7.2
Number of Barriers, n	1/1.5″	6/B	6/B	10/B	6/0.25″
Equation Applied	3-6	3-1	3-1	3-1	3-1
Velocity at Shaft	270	164	172	141	117
Shaft Leakage, cfs	(1.62)	0.72	0.75	0.62	0.50
Shaft Leakage, pps	(0.126)[a]	0.063	0.078	0.072	0.069
Velocity at Eye Seal	335	205	220	180	232
Eye Leakage, cfs	2.95	1.80	1.94	1.58	2.04
Eye Leakage, pps	0.23	0.16	0.20	0.19	0.29
Total					1.352

Note: a .Outboard shaft seal leakage flow is inward and does not contribute to inefficiency.

C = 0.020, h = 0.25, shaft seal diameter d = 10 inches, A_s = 0.0044 square foot.
B = Sharp blade labyrinth
$R_e = 0.04\ (300\ \text{fps})/12\ (3)\ 10^{-4} = 3{,}300^{0.25} = 7.5$
$f = 0.33/7.5 = 0.044,\ fh/2\,C = 0.28$

The suction eye is 20 inches in diameter and the area

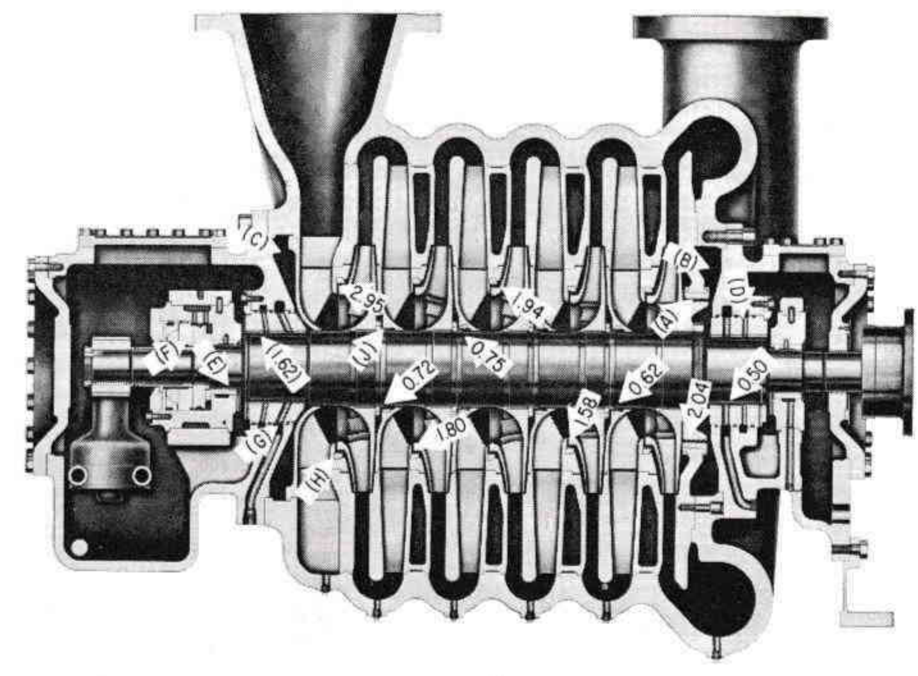


Fig. 3-3—Various labyrinth seal applications and leakages (cfs) are shown in a typical five-stage centrifugal compressor.

(A) The seal between the discharge and equalizer chamber.

(B) Leakage is 0.50 cfs with seal described in Table 3-1.

(C) Suction chamber draws in 1.62 cfs of outside air.

(D) An injector or inductor port. Sweet gas can be introduced or an eductor can withdraw gas from this port insuring against lube oil contamination when handling sour gas.

(E) Radial sleeve bearing.

(F) Kingsbury type thrust bearing.

(G) Can be used as a drain as shown or as a source of feeding sweet gas.

(H) Depicts the suction eye seal where the leakage is 2.95 cfs.

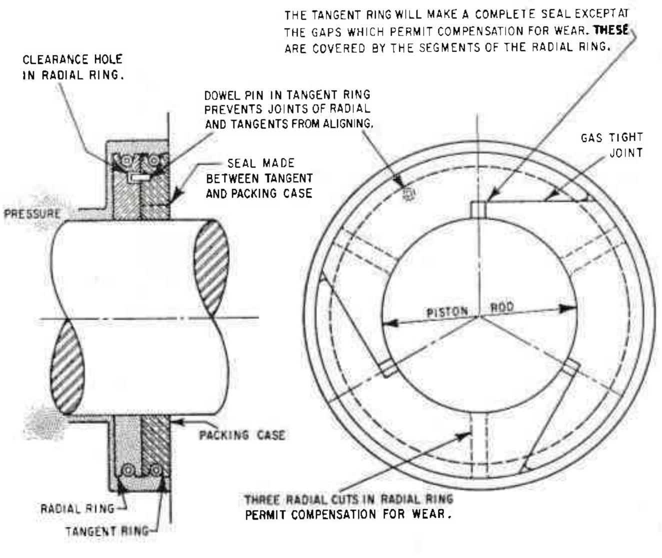
(J) Shows the shaft seal between the first stage impeller back and the second stage suction, where the leakage is estimated to be 0.72 cfs.

is 0.088 square foot. The pressure drop through the eye seal is estimated to be 75 percent of the stage pressure differential. The differential across the shaft seal is estimated to be 50 percent of the stage differential.

Fig. 3-3 shows a similar five-stage centrifugal compressor in sectional view with all of the labyrinth seals tagged with respective leakage.

The cfs leakage through the various seal sections is shown for the respective points in Fig. 3-3. The total internal leakage is 1.352 pps. The external leakage through the inboard driving end and the discharge end seal is 0.069 pps or 54 scfm. The outboard suction shaft-seal leakage into the machine is 0.126 pps or 100 scfm. The temperature of leakage gas is that of the higher pressure condition. It may be considered to be an isothermal flow or at best, the flow is at constant enthalpy, where the Joule-Thomson effect may show a small temperature drop. The hot gas leakage will blend with the incoming gas, raising the temperature of the mixture by the mass-heat ratio. For the example cited, the discharge temperature is 400° F, ΔT_t is 340° F, (total rise from 60° F) and ΔT_s is 68° F per stage. The effect of the leakage on the discharge temperature is: 167 (340) plus 1.283 (68) divided by 167 is 340.5° F. If the leakage was five percent of the total intake, the temperature would be 343° F. By the same equation and an abnormal temperature rise, the quantity of the contributing leakage can be evaluated. This recycling and abnormal heating directly affects the dynamic or total adiabatic losses. The discharge shaft seal leakage is considered to be a portion of the mechanical losses, along with the gear and bearing friction.

The effect that the labyrinth leakage has on the dynamic efficiency can be appreciated by the following evaluation: The sum of the total internal leakage is divided by the product of the total flow and the number of stages. For this example the leakage loss is: 1.352/167(5) = 0.0016 or 0.16 percent. This loss is consistent with the above temperature rise. The mechanical loss concerned with the discharge seal is: 0.069/167 = 0.00041 or 0.04 percent. In large process compressors, the discharge seal is balanced with the suction system. There is no mechanical loss involved in permitting the suction gas leakage to the ambient. There is just a loss of commodity. The venting loss of the high-pressure gas back to the suction seal chamber is charged to the internal dynamic losses.



ONE PAIR RADIAL TANGENT PACKING RINGS

Fig. 3-4—Restrictive ring shaft seal side view of the gland is shown at the left. A front view assembly of a tangential cut and a radial cut segmental carbon ring.

Restrictive Carbon Rings. This type of shaft-seal has been used extensively for the past 50 years on steam turbines. The steam encountered by the carbon rings is relatively clean and moist at the exhaust condition.

The latter offers the necessary lubricant to minimize the wear and to sustain the carbon bond. Fifteen years of such service is commonplace. Fig. 3-4 illustrates a typical carbon ring seal element. It shows an assembly of the two, three-piece carbon ring elements. The first ring is cut in three segments with tangetial separations. These segments are backed with identical size carbon rings having three radial separations. The solid sections of one segment will overlay the alternate cuts and wear gaps. Both rings are secured to the shaft by means of garter springs. These springs also take-up the wear. The clearances used for carbon ring shaft are less than used for labyrinth. The rings are permitted external radial clearance to adjust for shaft *run-out*. The gas leakage can be estimated from this equation:

$$Q = (2)\, 10^{-5}\, d\, [\Delta P\, v_{sm}/(3 + 3\,(n - 1))]^{0.5},\ cfm \qquad (3\text{-}7)$$

Balancing Pistons. The sealing chamber for the high-pressure shaft is equalized with the suction seal chamber by means of a balance line (see Fig. 3-3, Item C to D). The high-pressure chamber includes a labyrinth seal (Item A) to break down the compressor discharge to the suction pressure. The back of the impellers is exposed to approximately 85 percent of the total head developed for each respective stage. The shroud face is exposed to about 67 percent of the discharge pressure. The effect of these forces on a typical five-stage, 32.5 psig, 17,500-hp air compressor is shown in Table 3-2. The accumulative thrust to the suction end is 55,000 lbs. A 15-inch Kingsbury type thrust bearing can provide 155 square inches of supporting surface. The thrust load would be 355 psi, which is about ¼ of the ultimate pressure that the bearings should experience. There are several protective measures which may be taken to minimize excessive thrust loads. The last stage impeller *stack-up* position can be reversed so as to reduce the total longitudinal thrust to 11,000 lbs. or 71 psi on the 15-inch Kingsbury bearing. A balancing piston or *drum* offers a third method of cop-

TABLE 3-2—Unbalanced Thrust Forces Acting on Impellers

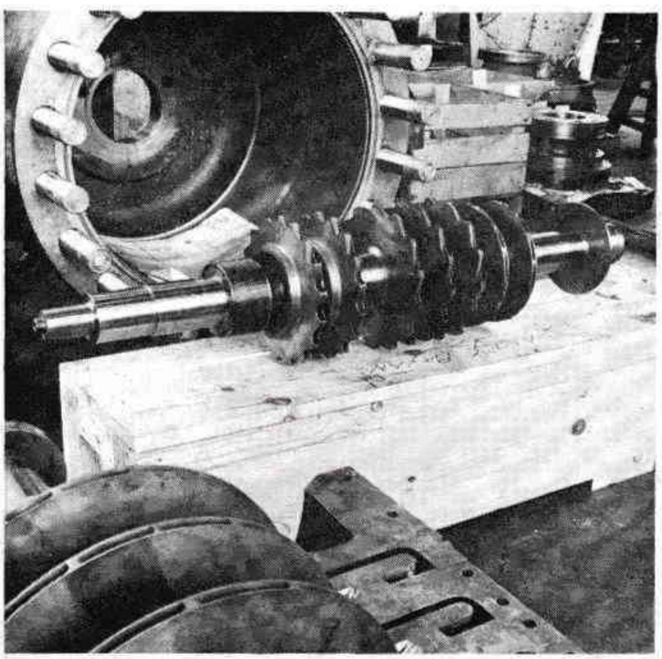
Stage	1	2	3	4	5
Suction Temp., R	520	568	620	676	740
Suction Pressure, psia	14.5	18.2	22.8	28.6	36.8
Specific Volume, cu. ft./lb.	13.3	11.6	10.1	8.75	7.45
Suction Volume, cfs	2,220	1,940	1,690	1,460	1,240
Suction Eye, in.	30	30	30	26.7	26.7
Differential Pressure, psig	3.7	8.3	14.1	22.3	32.5
Back Disk Pressure, lbs.	3,900	8,700	14,800	23,400	34,000
Shroud Pressure, lbs.	1,400	3,000	5,200	8,200	12,000
Thrust	2,500	5,700	9,600	15,200	22,000

ing with excessive thrusts. The piston is attached to the shaft at the discharge end. It fits into a cylinder and is sealed with labyrinth rings.

The above compressor has 40-inch impellers and a net back disc area of 1,230 square inch. The first three stages have 30-inch eye diameters and 550 square inches net thrust area. The last two have 26.7-inch eye diameter and 700 square inch thrust area. The discharge pressure is 32.5 psig. The total longitudinal thrust is 55,000 pounds. Reversing position of the fifth impeller reduces the net thrust to 11,000 pounds.

The outboard end of the drum is exposed to a pressure several psi over suction pressure to permit the high-pressure leakage to flow back through the equalizer line to the suction chamber. The drum diameter is approximately equal to the suction eye diameter. Using a 29-inch drum for this case, the thrust capabilities are: 660 square inch less 30 square inches for a 6⅛-inch shaft (32.5-1.5 back pressure) = 19,500 pounds. The balancing drum would reduce the thrust load for the 15-inch Kingsbury bearing to 39,500 pounds or 229 psi. Where the pressure levels are higher and the speeds are greater, the benefits of the balancing drums are more pronounced.

There are numerous other devices used to reduce the impeller back pressure. A series of 0.200-inch ribs machined on the back of the impeller are so effective as a *pump-out* device that the 85 percent back-pressure factor is reduced to 15 percent. Deep scallops cut out of the disc between vanes reduces the back-pressure factor to 20 percent (see Fig. 3-5). Pressure equalizer holes drilled through the eye hub, about every 3 to 4 inches apart will reduce the back-pressure to 22 percent. The equalizer hole diameters should be about 2 percent of the impeller diameter. Several thin 0.060-inch ribs can reduce the back-pressure to 25 percent. A shroudless disc has a thrust factor of 30 percent of the differential pressure on respective front and back impeller areas.



Courtesy Cooper Bessemer Co.

Fig. 3-5—Deep scallops cut out of the disc between vanes reduces the back-pressure factor to 20 percent.

Liquid-Film Seal. The shaft seals considered thus far were concerned with relatively low pressure compressors operating at discharges of less than 125 psig, 250° F, and the seals are operated dry. There are exceptions where lube-oil is applied to carbon-ring seals by drop-feed lubricators. This is done to reduce the frictional heat of the higher pressure seal contacts and where the lube-oil is not a nuisance. Where the compressed gas is inexpensive and expendable, the pressure range of labyrinth seals is extended beyond these arbitrary limitation, otherwise the *oil-film* type seals are applied. The casing discharge pressure is reduced with a labyrinth seal to suction pressure and equalized to that suction seal chamber. The *oil-film* shaft seal consists of two sleeves which are supplied 2-3 gpm of seal oil at a midpoint about 5 psi greater than the gas chamber pressure. The inner sleeve facing the

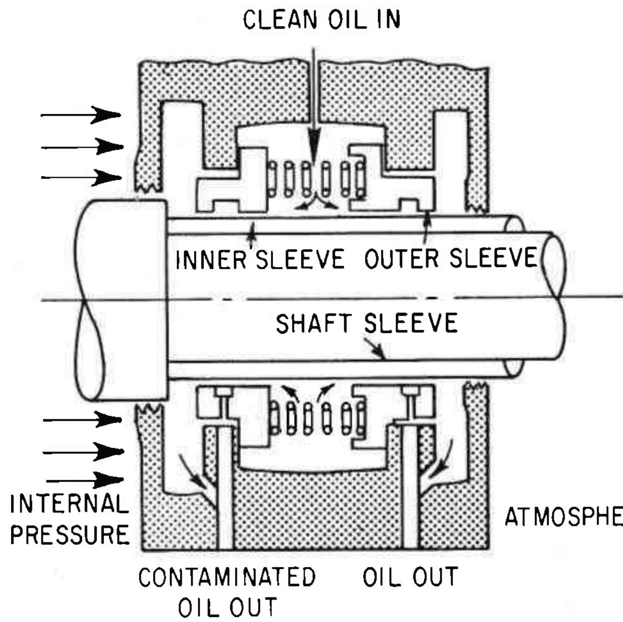


Fig. 3-6—The liquid shaft seal contains two "L" section sleeves which are flushed with seal oil at 5 psi above the process gas pressure. The oil is supplied from an overhead reservoir.

gas is shorter than the outer sleeve. It is also fit to within 2 mils clearance which will only permit a leakage of 1-3 gallons per day. This leakage is collected and gas separated by means of a trap draining the internal chamber. When the process gas carries undesirable contaminates, the trapped oil is drained to waste. The bulk of the oil circulation passes through the outer sleeve which is somewhat longer than the inner sleeve to break down the greater pressure drop. The circulation maintains the seal at or about 120° F with a temperature rise of about 5° F. The design of both sleeve cylinders is such that an adequate degree of radial movement is permitted and the sleeves may follow the shaft play. The entire assembly is sealed into the casing with suitable O-rings and gaskets to avoid gas leakage.[13] (see Fig. 3-6.)

Dynamic Seals. There are variations of the oil-film seal which use the dynamic pumping action of the inner sleeve. Instead of the straight sleeve, a conical sleeve is substituted which has rather large clearances over the

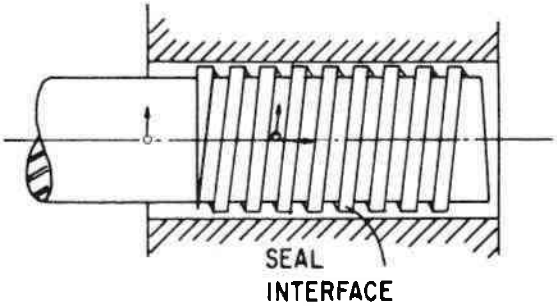


Fig. 3-7—This dynamic seal functions on the Archimedes principle.

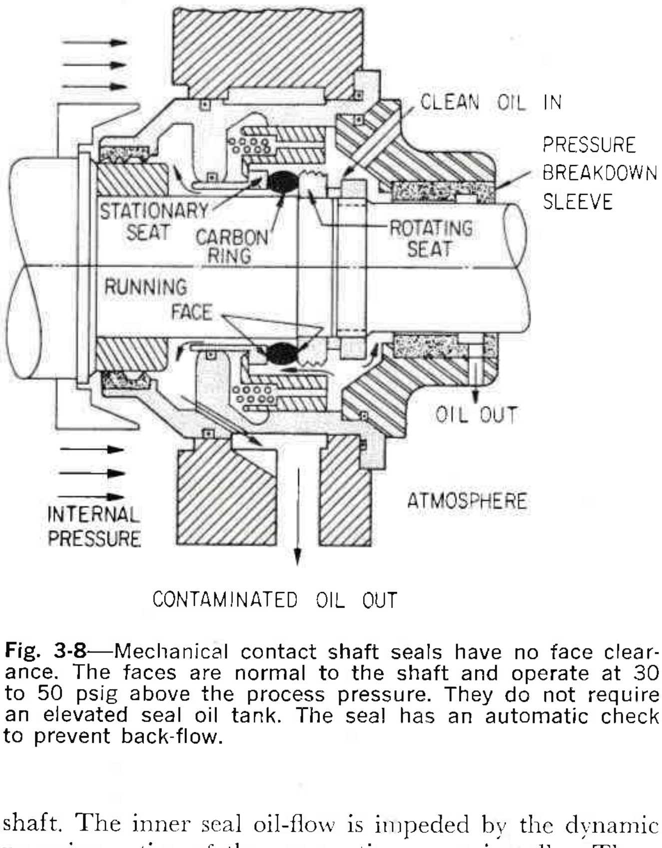
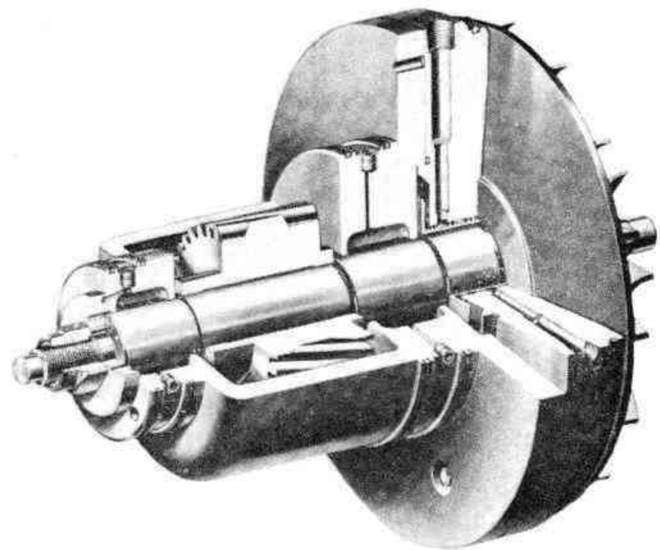


Fig. 3-8—Mechanical contact shaft seals have no face clearance. The faces are normal to the shaft and operate at 30 to 50 psig above the process pressure. They do not require an elevated seal oil tank. The seal has an automatic check to prevent back-flow.

shaft. The inner seal oil-flow is impeded by the dynamic pumping action of the cone acting as an impeller. There are other dynamic types of seals developed principally for gas turbines which operate at low pressures (15 psig to high vacuum and temperatures up to 1,400° F. Their leakage is only a few pounds per week and they have a life of one to five years. One of the more interesting seals is shown in Fig. 3-7. It depends upon the action of an Archimedes screw pump.[14]

Mechanical Contact Seal. The difference between this seal and the oil-film seal is that the clearance is reduced



Courtesy Chicago Pneumatic Tool Co.

Fig. 3-9—Cartridge module for a four-stage, sunflower type compressor. The central pinion gear engages the main bull-gear as one of the four stages. The unique hydrostatic bearing system and the shaft seals are also in evidence.

to zero and the sealing faces are normal to the shaft instead of longitudinal. The seal requires a greater differential of 30 to 50 psig above the internal casing pressure. The features of the seal are shown in Fig. 3-8. The unattached floating carbon ring is presumed to run at half shaft speed. The mating surfaces are lapped to two microns. The inner gland oil leakage is less than the oil-film seal. The oil supply system is less complex and it has an automatic closing device. The seal will contain the internal gas pressure when there is a loss of seal-oil pressure or at shutdown. Small auxiliary pistons are activated by internal casing pressure which maintains a contact between the stationary seat and the carbon ring to prevent gas escape to the atmosphere The higher differential pressure on the seal oil permits higher compressor discharge pressure surges without gas blowing past the seals. It also eliminates the need for the overhead surge tank. The oil-film seal system must contain a 90 gallon (or larger) pressurized auxiliary oil reservoir to supply the necessary seal oil to produce a gas shut-off, equivalent to the automatic feature of the mechanical contact seal. This reservoir oil is also used to provide minimum bearing lubrication during shutdown.[13]

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Indexing Terms: Balancing-9, Carbon-9, Centrifugal-4, Clearances-6, Comparison-8, **Compressors-4**, Contact-9, Dynamic-9, Efficiency-7, Evaluation-8, **Films-9**, **Flow-6**, Labyrinth-9, Leakage-7, Metallurgy-6, Pistons-9, Rings-9, **Seals-9**, **Shafts-4**, Sleeves-9, Velocity-7.

Best Approach to Compressor Performance

When a centrifugal compressor is specified for nonideal gases and the manufacturer's data is based on air, how do you predict design and off-design performance? This calculation procedure solves the problem

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THE SPECIFICATION of a centrifugal compressor and prediction of design and off-design performance are parts of a problem that continually arise in process design studies. To make these calculations less difficult, a computer program was written which handles very general situations including multistage compression, inter- and intra-stage leakage, multicomponent gas streams, departures from perfect gas behavior, intercooling when necessary, and allowances for manufacturers' data on the basis of air to permit design for hydrocarbons and other vapors.

The equations used for the individual compression stages of the compressor are summarized in the sections which follow. The assumptions and limitations implied in the general efficiency and head coefficient correlations used, are discussed. The equations given are more general than those usually appearing in the literature.[1,4,6,8,9,12]

COMPRESSOR PERFORMANCE EQUATIONS

Outlet Temperature and Pressure. The temperature and pressure at the outlet of the compression stage are related to the hydraulic efficiency (polytropic efficiency) and head coefficient by the following set of equations:

$$T_2 = T_1 + \frac{(0.0012854)}{(60)^2 C_p}\left[\frac{1}{\varepsilon} + \left(\frac{\partial \ln Z}{\partial \ln T}\right)_p\right]\frac{\pi^2 D^2 N^2 \Psi}{2 g_c} \qquad (1)$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^m \qquad (2)$$

where

$$m = \frac{1}{\frac{ZR}{MC_p}\left[\frac{1}{\varepsilon} + \left(\frac{\partial \ln Z}{\partial \ln T}\right)_p\right]} \qquad (3)$$

The efficiency and head coefficient* are defined by

$$\varepsilon = \frac{(0.0012854)\, W_p}{H_2 - H_1} \qquad (4)$$

$$\Psi = \frac{2\,(60)^2 g_c W_p}{\pi^2 D^2 N^2} \qquad (5)$$

where the polytropic head, W_P, is defined

$$Wp = (144)\int \frac{dP}{\rho} \quad \text{(Polytropic path)} \qquad (6)$$

The properties appearing in Equations (1) and (3) are to be averaged over the compression path. The temperature equation, Equation (1), requires an efficiency which is an appropriate average over the path of the compression, but the pressure equation is only valid for a constant efficiency path.

Dimensionsless Numbers. Dimensional analysis of the compressible fluid flow equations[7] indicate those variables and groups of variables which are important in specifying the performance of a centrifugal compression stage. Applicable equations are the equation of of continuity, the momentum equation, the energy balance, and the equation of state. These equations must be solved subject to specified boundary conditions.

The solution to the equations is controlled by the magnitude of the following dimensionless groups:

The Grashof number

$$N_{Gr} \equiv \frac{g\beta \Delta T L^3}{\nu^2} \qquad (7)$$

The Reynolds number

$$N_{Re} \equiv \frac{UL}{\nu} \qquad (8)$$

The Prandtl number

$$N_{Pr} \equiv \frac{C_p \mu}{\lambda} \qquad (9)$$

The Eckert number

$$N_{Ek} \equiv \frac{(0.0012854)\, U^2}{g_c C_p \Delta T} \qquad (10)$$

If the reference temperature, ΔT, in the energy balance equation is taken to be the absolute temperature, T, the Eckert number is related to the Mach number

$$N_{Ek} = \frac{(k-1)\, N_{Ma}}{\left[1 + \left(\frac{\partial \ln Z}{\partial \ln T}\right)_p\right]^2} \qquad (11)$$

* The definition of the head coefficient, Equation (5), involves a factor of 2 which is omitted in some sources.

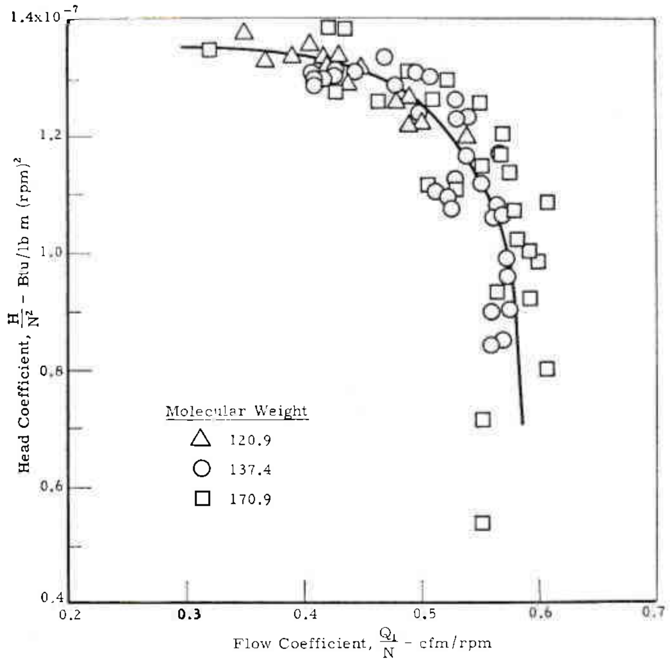

Fig. 1—Head correlations for a variety of speeds.

In the above list of dimensionless groups, the Eckert number (Equation 10), may be replaced by

The Mach number

$$N_{Ma} \equiv \frac{U}{V_s} \qquad (12)$$

The speed of sound, appearing in the denominator of Equation (12), is the speed at which a small amplitude disturbance will propogate through the gas in the absence of absorption losses, i.e., for an adiabatic reversible or isentropic process. The speed of sound is given by the relation:

$$V_s = \sqrt{144\, g_c \left(\frac{\partial P}{\partial \rho}\right)_s} \qquad (12a)$$

The relations of thermodynamics may be used to transform this equation to a form which allows direct use of the equation of state:

$$V_s = \sqrt{144\, g_c\, k \left(\frac{\partial P}{\partial \rho}\right)_T} \qquad (12b)$$

The final relationship for the speed of sound in a nonideal gas is:

$$V_s = \sqrt{\frac{144\, gc\, kZRT}{M\left[1 - \left(\frac{\partial \ln Z}{\partial \ln P}\right)_T\right]}} \qquad (12c)$$

The calculation makes use of Equation (12c) evaluated at the inlet of a compression stage.

In the computer program, the characteristic velocity (U) is taken to be the impeller tip speed. In addition to the above groups, the equation of state and the boundary conditions are implicit in determining the solution.

Two compression stages will have the same solution in terms of appropriate dimensionless variables provided certain side conditions are satisfied. First of all, the boundary conditions in the two situations must be identical. Second, both compression stages must be geometrically similar, that is, the ratios of all dimensions must be identical, and third, the equation of state for the two gases undergoing compression must be identical.

Thermodynamic Variables. Solutions to gas motion equations are generally given in values of gas thermodynamic properties as functions of spatial variables in the impeller and diffuser. However, the solutions of interest are certain functions of these thermodynamic variables rather than their complete spatial variation. In particular, the solution may be thought of as the ratio of useful work to the total work or total enthalpy rise, and the ratio of useful work or pressure head to the kinetic head of the impeller. This first variable is the polytropic head divided by the actual work input and is referred to as the hydraulic efficiency. The second variable is the head coefficient. These variables have been previously defined by Equations (4) and (5). These ratios will depend on the boundary conditions and in particular on the ratio of the fluid velocity to the tip velocity of the impeller. This ratio is the volumetric flow coefficient (ϕ)

$$\phi \equiv \frac{Q}{\pi DNA} \qquad (13)$$

and the formal relations for the efficiency and head coefficient and the flow coefficients are:

$$\varepsilon = \varepsilon\,(\phi; N_{Gr}, N_{Re}, N_{Pr}, N_{Ma}; \text{ equation of state parameters, geometric variables})$$

$$\Psi = \Psi\,(\phi; N_{Gr}, N_{Re}, N_{Pn}, N_{Ma}; \text{ equation of state parameters, geometric variables})$$

The volumetric flow used in defining the flow coefficient may be an inlet flow, an outlet flow, or any average of inlet or outlet flows. Several alternate approaches have been reported in the literature.[2,3,4,9,11] Better correlations appear to result from a use of intermediate flows, but correlations in this form are not always available from the manufacturers. The example given is based on inlet volumetric flows.

For impellers which are geometrically similar, the area *(A)* is uniquely related to the impeller diameter *(D)*. However, the value of *A* has been left free so that it may be adjusted to improve the efficiency and head coefficient correlations or to put them in a convenient form, particularly if geometric similitude is not maintained. For example, the flow coefficient may be interpreted as the ratio of the volumetric flow to the manufacturers' design volumetric flow if a nominal area *(A)* is taken to be

$$A = \frac{Q\ (\text{design point})}{\pi DN} \qquad (14)$$

This form is often convenient since the manufacturers' correlations are commonly in terms of percentage flows relative to the design point, i.e., Equation (14) fixes $\phi = 1$ at the design point. A similar definition of a nominal diameter *(D)* may be obtained from Equation (5) for $\Psi = 1$ at the design point.

Efficiency and Head Coefficient Correlation. When the heat loss to the surroundings is negligibly small, the Prandtl number may be dropped from the solution.[7] If the Reynolds number is large ($N_{Re}^2 >> N_{Gr}$), natural convection is no longer of importance; thus the Grashof number may be eliminated from the solution.[7] If the Mach number is smaller than 1, that is, if the speed of sound is not exceeded, at all points along the compression path the performance of the compression stage is insensitive to the Mach number.[9,12,13] For Reynolds numbers greater than 100,000, as is usual, the results are insensitive to the Reynolds number.[2,3,6]

The mechanical design of different impellers for a given frame or frames is not generally based on the principle of geometric similitude. Impellers for the same frame designed for varying volumetric flow capacities will typically have the same impeller diameter with a varying channel width or area of flow. The assumption of geometric similitude is then not strictly applicable. An assumption implied in the example calculation is that a single correlation for head coefficient and for efficiency applies for all of the impellers available in the computation. This assumption will be strictly valid only if the impellers are geometrically similar. However, it is necessary to make this assumption in many cases since more detailed information on the performance of the individual impellers is not always available.

The form of the correlations which is suggested for use reduces to:

$$\varepsilon = \varepsilon(\phi)$$
$$\Psi = \Psi(\phi) \qquad (15)$$

An example correlation of head coefficient as a function of volume coefficient is presented based on the data of C. A. Macaluso[4] for a single stage compressor. Since the actual physical dimensions of the compressor were not included as part of the data, the results are correlated only in terms of ratios of the appropriate quantities. That is, the head coefficient is replaced by the dimensional ratio (in enthalpy units)

$$\Psi \sim \frac{H}{N^2} \qquad \left[\frac{\text{Btu/lb mass}}{(\text{rpm})^2}\right] \qquad (16)$$

and the volume coefficient (based on suction volume) is replaced by the dimensional ratio

$$\phi \sim \frac{Q_1}{N} \frac{cmf}{rpm} \qquad (17)$$

The results for a variety of speeds ranging from 5,400 rpm to 8,250 rpm and volumetric flows of 2,000 cfm to 4,500 cfm are presented in Figure 1. While there is scatter of the results, a single correlation can be presented with an accuracy of 5-10 percent. A point of particular interest is that the results are for compression of different gases (molecular weights of 120.9, 137.4, and 170.9). The surge limit or minimum value of the volumetric flow coefficient and the choking limit or maximum value of the volumetric flow coefficient are reduced to single values of the volumetric flow coefficient when presented on the basis of Figure 1.

The above results and others from the literature[9,11] suggest that correlations for both head coefficient and efficiency as functions of volumetric flow coefficient may be used for a variety of nonideal gases provided the appropriate dimensionless forms are retained. In the absence of information on the particular gas being compressed, manufacturers' information based on air can be used in the computation even though exact similitude would not be obtained.

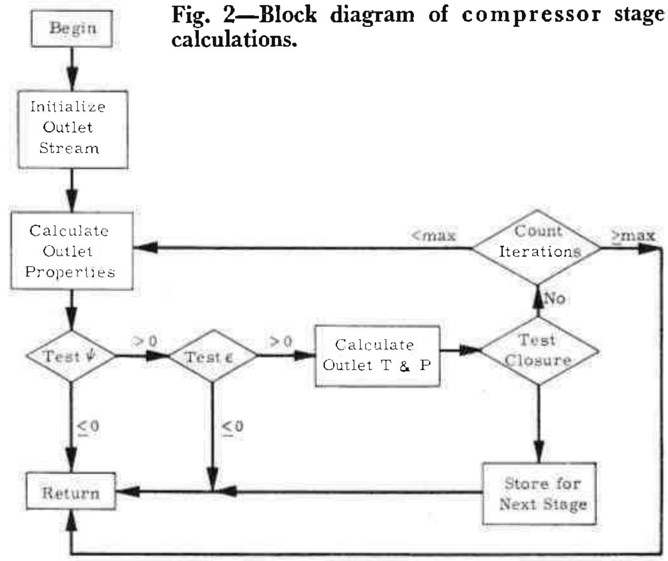

Fig. 2—Block diagram of compressor stage calculations.

Compressor Performance Calculations. The performance calculation for a compressor or series of compressors is accomplished on a wheel-by-wheel basis. The performance of each impeller or compression stage is considered in sequence by application of the general set of performance equations, Equations (1), (2), and (3). These equations allow calculation of the temperature and pressure at the outlet of a compression stage provided that the efficiency and head coefficient are known.

The specific definitions of the coefficients and dimensionless groups used have been presented as Equations (4) and (5). The efficiency and head coefficient are represented as functions of a single variable, the volumetric flow coefficient, Equation (15). The correlation of efficiency and head coefficient with volumetric flow coefficient is often a satisfactory representation of the performance of not only a specific impeller, but also of other impellers of similar design but which are scaled for use at different volumetric flows. This correlation of efficiency and head coefficient provides a convenient condensation of information on the performance of a compression stage for a variety of operating conditions.

There are limitations inherent in this approach, and some of these have been discussed above. The calculation procedure is sufficiently flexible so that additional information, when available, can be included.

Outline of Program Logic. The program logic may be schematically outlined as follows:

1. Read in input data.
2. Calculate recycle (leakage) flows.

[If the compressor calculations are incorporated with other process calculations, recycle flows (not necessarily leakage, as in this example) may depend on the compressor discharge conditions. In that case an over-all material balance may require iterative application of the entire compressor calculation.]

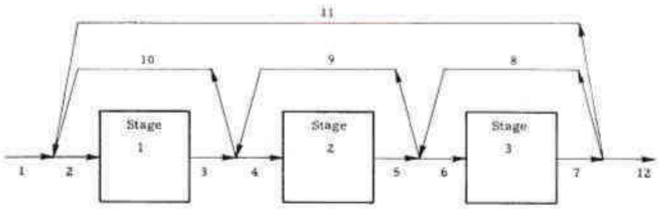

Fig. 3—Flow diagram for a three-stage compressor.

3. Calculate physical properties for the suction.

4. Initialize stage discharge physical properties.

5. Calculate appropriate average flow coefficient.

6. Calculate head coefficient and efficiency.

7. Calculate new discharge temperature and pressure.

8. Compare current and previous discharge temperature. If closure to within a specified tolerance is not achieved, return to (5). If closure is achieved, calculate the next stage starting with (4). After the last stage, continue with (9).

9. Print out final results.

Figure 2 is a simplified diagram of the subroutine which calculates the performance of a single stage.

Use of the Computer Program. The input data required by the program can be divided into the following categories:

Component data—consisting of a specification for vapor components in the form appropriate to the subroutines which calculate the compressibility factor, Z, and other physical properties required for the computations.

The equation of state used in the example is based on a modified version of the Martin and Hou equation of state[5,10]

$$p = \frac{RT}{(v-b)} + \frac{A_2 + B_2T + C_2e^{-KT/T_c}}{(v-b)^2} + \frac{A_3 + B_3T + C_3e^{-KT/T_c}}{(v-b)^3} + \frac{A_4}{(v-b)^4} + \frac{B_5T}{(v-b)^5}$$

Impeller correlation data—specifying a correlation of efficiency and head coefficient of the impeller or impellers with the volumetric flow coefficient. (The correlation may be based on inlet flow, outlet flow, or any average of inlet and outlet flows from a compression stage. Variations in the efficiency and head coefficient with Mach number and Reynolds number may be included, if desired.)

Compressor sizing data—consisting of a list of impeller scale parameters or physical dimensions to be used in the correlations for efficiency and head coefficient. (These usually are a characteristic impeller area and diameter, but other variables which reflect the capacity of the impeller may be used.)

Stream data—consisting of a specification of the inlet stream temperature, pressure, and molar flows for each component.

Miscellaneous specifications—including the rotational speed for each frame, and several bounds and increments which must be specified.

The output from the program is a reiteration of the input data and a statement of the calculated properties for the inlet stream, i.e., the density and the volumetric flow. A listing of parameters and properties for the outlet stream of each compression stage is presented. These include the stream number, the frame number and impeller number, the nominal diameter and area for the impeller, the rotational speed, the temperature, pressure, density, molecular weight, the volumetric flow, and arithmetic average value of the specific heat ratio, the value of the exponent which defines the compression path (given by Equation 3), the value of the average flow coefficient, the Mach number, Reynolds number, head coefficient, efficiency, the head and horsepower. When a stage consists of intercooling, the temperature and pressure of the stream is presented, and, if phase separation occurs, molar flows of both the vapor and the liquid stream are presented.

In addition to the calculated properties, a warning flag, FLAG, is indicated for each compression stage. Values of FLAG are used to indicate the course of the computation and signal any difficulties which may have been encountered during the computation. The warnings which may be indicated by FLAG are:

- The Mach number exceeds the critical Mach number.
- The average value of the flow coefficient is less than a minimum value.
- The average value of the flow coefficient is larger than a maximum value.
- The number of iterations for the calculation at this stage has exceeded the maximum number of iterations, i.e., the calculation has failed to converge.
- The calculated head coefficient is negative (may result from inappropriate extrapolation of correlation).
- The calculated efficiency is negative (as in head coefficient above).
- Intercooling has resulted in condensation.
- The temperature has exceeded the maximum allowable temperature.

Example Calculation. An example is presented on the application of the above procedure using a large scale digital computer. Calculations are made for a three-stage propylene refrigeration compressor. The shape of the head-capacity and the efficiency-capacity relations was taken to be the same for all three wheels, with the flow coefficient based on inlet flows. The head-capacity relation for the second stage was used as supplied by the manufacturer. The shape of the curves for the first and third stages differ by 2-5 percent at the extremes in the range of 60-140 percent design capacity. An efficiency-capacity relation was not available for the wheels in this machine. The manufacturer was able to supply an efficiency-capacity curve for a similar machine at a specific speed which was close to the design condition at the second stage. This curve was used to calculate

Table 1—Example Calculations Compared With Performance Test

Flow[a] Lb.-Mole/Hr.	RPM	SUCTION Temp., °F	SUCTION Press., psia	SUCTION cfm[b]	DISCHARGE (3rd Stage) Temp., °F	DISCHARGE (3rd Stage) Press., psia	DISCHARGE (3rd Stage) cfm	DISCHARGE (3rd Stage) Brake, hp[c]	PERFORMANCE TEST Press., psia	PERFORMANCE TEST Brake, hp
3250	8950	78	80	3563	231	312	1093	2949	308	2820
3500	8950	78	80	3837	225	304	1197	3025	298	2870
3750	8950	78	80	4111	218	292	1325	3078	286	2930
4000	8950	78	80	4385	211	276	1487	3107	268	2980

a–100 Percent Propylene. b–For feed flow, temperature, and pressure. c–1.02 x Compression hp.

an efficiency-capacity relation for the range in capacity of from 60-140 percent of the design points. Again, the wheels may differ by approximately 5 percent at the extremes of the range. The flows in the recycle (leakage) streams were taken as 1.5 percent at each stage and 6.5 percent from the discharge of stage 3 to the suction of stage 1. Figure 3 is a schematic diagram indicating the interstage flow, stream 12, and the intrastage flows, streams 8, 9, and 10.

The results of the computations are summarized in Table 1 and are compared with a performance test.

The discharge pressures of 312-276 psia and power requirements of 2949-3107 hp are somewhat above the 275-200 psia and 2,400-2800 hp ranges from the original design specifications. However, the calculated values are close to those from the actual performance tests for this machine. The calculated discharge pressures (third stage) are within 1-3 percent of the pressures actually achieved with even closer agreement for the brake horsepower. The agreement is excellent considering that manufacturers' head and efficiency data for air has been used to calculate performance for propylene under conditions where it is a nonideal gas.

Why This Approach? The approach presented here is intended as a thermodynamically consistent method for performance calculations when nonideal gas phases are important. It is hoped that the presentation will encourage publication of further experimental results which will allow evaluation of this approach and outline more clearly its limitations. The author would like to emphasize to those concerned with design, manufacture, and performance of such compressors that methods and computer capabilities for evaluation of thermodynamic properties of fluids are becoming generally available and should extend the usefulness of many design methods. The physical properties programs[10] used here have been made available to the Physical Properties Project Subcommittee of the American Institute of Chemical Engineers.

About the author

Dr. S. A. Shain is a chemical engineer with Shell Development Co., Emeryville, Calif. He is engaged in a variety of activities relating to fundamental understanding and model simulation of physical and chemical processes. Dr. Shain holds a B.S. degree from M.I.T. and a Ph.D. degree from the University of California (Berkeley). He is a member of AIChE and ACS.

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NOMENCLATURE

A nominal area, sq. ft.
C_p heat capacity at constant pressure, Btu/lb. mass-°R
D nominal diameter, ft.
g local acceleration of gravity, ft./sec. sq.
g_c gravitational constant, 32.16 lb. mass-ft./lb. force-sec. sq.
H enthalpy, Btu/lb. mass
k heat capacity ratio, C_p/C_v
L characteristic dimension, ft.
m exponent, defined by Equation (3)
M molecular weight, lb. mass/lb. mole
N rotational speed, rpm
N_{Ek} Eckert number, defined by Equation (10)
N_{Gr} Grashof number, defined by Equation (7)
N_{Ma} Mach number, defined by Equation (12)
N_{Pr} Prandtl number, defined by Equation (9)
N_{Re} Reynolds number, defined by Equation (8)
P pressure, psia
Q volumetric flow, cmf
R gas constant, equal to 1.987 Btu/lb. mole-°R, or 10.73 psia-cu. ft./lb. mole-°R
S entropy, Btu/lb. mass-°R
T temperature, °R
U characteristic velocity, ft./sec.
V_s velocity of sound, ft./sec.
W_p polytropic head, ft.-lb. force/lb. mass
Z compressibility factor

Greek Symbols

β coefficient of thermal expansion, vol fract/°R
ε polytropic efficiency, defined by Equation (4)
λ thermal conductivity, Btu/sec.-ft.-°R
μ viscosity, lb. mass/ft.-sec.
ν kinematic viscosity, sq. ft./sec.
π constant, equal to 3.14159
ρ density, lb. mass/cu. ft.
ϕ volumetric flow coefficient, defined by Equation (13)
Ψ polytropic head coefficient, defined by Equation (5)

Subscripts

1 refers to compression stage inlet
2 refers to compression stage outlet

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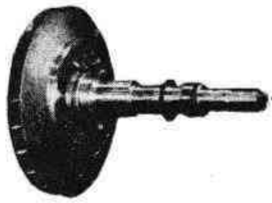

Important Performance Characteristics

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THE PETROLEUM industry has used centrifugal compressors for some time in cracking facilities. More recently they have been employed in large numbers in recovery and reforming operations.

The use of the centrifugal is becoming more prevalent, due to: (1) Larger plants, (2) fitting refinery heat balances with a turbine driver, (3) acceptance of high speeds and (4) low installation and operating costs.

Each of these factors is reflected in the selection of a compressor and its driver.

Type and Ranges of Application—Horizontally and vertically split compressors are used depending on the pressure and type of gas. The pressure dividing line is roughly 800 psi. The pressure is lowered for hydrogen.

The compressors are characterized with constant diameter impellers whose blades curve backwards with respect to the direction of rotation.

They are best suited to medium capacities and pressures. A yardstick for determining the minimum flow is 400 cubic feet per minute measured at exit conditions. Inlet pressure ranges from below atmosphere to as high as 600 psig. Discharge pressures range up to 1000 psig and non-refining applications may require considerably higher pressures.

Pressure-Volume Characteristics—There is one characteristic peculiar to a gas compressor as opposed to a pump. This is the phenomena of surge wherein the compressor does not meet the pressure of the system into which it is discharging. This causes a cycle of flow reversals as the compressor alternately tries to deliver gas and the system returns it. It is due to the compressibility of a gas. Surge may be caused by a system disturbance or insufficient flow.

The point of surge due to flow is a minimum with a single impeller or, alternately, the range of stable operation is a maximum. This range, which is defined as a percent of rated inlet volume at constant discharge pressure, decreases approximately five percentage points with the addition of each impeller. Hence, if a four impeller air compressor has a 50 percent stable operating range, the addition of two impellers will decrease the range to 40 percent. The molecular weight of the gas will also influence this figure. High molecular weights decrease it while lighter gases extend the stable operating range.

The "stonewall" or sonic barrier is generally reached at the eye of the first impeller. A further increase in flow cannot occur past this point.

The efficiency of the compressor normally peaks to the left of the rating point and drops on both sides of the peak toward surge or the stonewall.

The characteristic curve is a summation of the decision made on placing the rating point with respect to surge, efficiency and the stonewall. This curve is shown in Figure 1 with the rating point placed for optimum stability and efficiency. Additional capacity at a reduced pressure is also indicated.

Operation "back on the curve" will increase the efficiency at the cost of a narrower stability range whereas moving the rating point out on the curve will lower the efficiency and increase the stability. Every selection is therefore a suitable compromise of these factors.

Parallel and Series Operation—Most refinery installations operate at high load factors and the range of operation of a single compressor is adequate. However on new processes such as reforming, parallel operating units have been installed as an insurance measure.

The long term trend is toward single compressors with a spare rotor, due to the confidence gained with long periods of successful operation. Single units for catalytic cracking and gas recovery units are examples of this trend.

Performance Calculations—Selection of an uncooled compressor requires the following data:

1. Gas Characteristics
2. Inlet conditions and discharge pressure
3. Type of driver
4. Driver operating conditions
5. Any special consideration or limitations due to process, surrounding atmosphere or parallel units.

The majority of refinery uses involve gases that do not follow ideal gas laws. Therefore, ideal gas behavior is the exception rather than the rule. All performance calculations are based on this premise.

List of Typical Data

	Symbol
1. Gas	
2. Mmscfd at......psia......°F.	
3. Inlet Volume, cfm	Q_s
4. Suction Pressure, psia	P_s
5. Suction Temperature, °R	T_s

	Symbol
6. Discharge Pressure, psia	P_d
7. Barometric Pressure, psia	
8. Molecular Weight	M
9. Average Compressibility	Z
10. Gas Constant	R
11. Isentropic Exponent	α

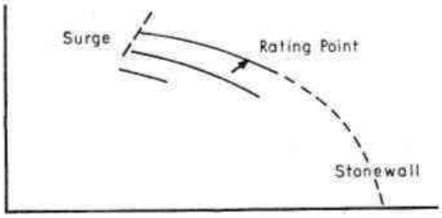


Figure 1 (above). Characteristic curve showing efficiency of compressor.

It will be noted that the user specifies the molecular weight as well as the isentropic exponent of compression. The latter is frequently different from the ratio of specific heats and is required for an accurate determination of the discharge temperature.

Secondly, it is desirable to specify the average compressibility factor. Determination of the number of stages, operating speed and driver selection are dependent on this figure.

	Symbol	Formulae
12. Specific Volume, of 3/lb.	$\bar{v}_s$	$\frac{ZRT}{144P}$
13. Mass rate of flow, lb.	W	Q_s / v_s
14. Hydraplic Efficiency, %	n_h	Mfg. Data
15. Exponent of Compression, n	m	$\frac{\alpha - 1}{n_h} = \frac{n - 1}{n}$
16. Compression Ratio	r	P_d/P_s
17. Head, ft. lb./lb.	H	$\frac{ZRT\ (r^m - 1)}{m}$
18. Gas horsepower,	GHP	$\frac{HW}{33{,}000 \times n_h}$
19. Total horsepower, hp	BHP	1.01 GHP
20. Discharge Temperature, °F.	T_d	$T_s \times r^m$
21. Pressure co-efficient	μ	Assume 0.55
22. Speed, rpm	N	$\frac{1300}{D^*}\sqrt{\frac{H}{\text{Stages} \times \mu}}$
23. Inlet Rating	Q_s/N	
24. Discharge Rating	Q_d/N	

* Impeller Diameter, inches.

The significant items are the head, volume, horsepower and speed.

Head—The polytropic head is used because it allows the use of hydraulic efficiency. This efficiency gives a true picture of compressor losses that is independent of compression ratio.

Some manufacturers use adiabatic head. A convenient curve for the conversion of polytropic to adiabatic head is given in Figure 2.

The polytropic head is related to the hydraulic design of the compressor. This relation is:

$$\text{Head/Impeller} = \frac{(\text{A constant}) \times (\text{Tip Speed})^2}{\text{Gravitational Constant}}$$

This constant is known as the pressure coefficient and is a function of the compressor design. An easy rule of thumb value is 0.55.

Tip speeds are visible manifestations of impellers stresses. Reasonable designs use values of 770 feet per second or less. Substitution of these values in the head equation gives a figure of 10,000 foot pounds per pound.

The total head in the calculation may be divided by this figure and the next largest whole number represents the number of impellers.

Volume—The inlet capacity determines the compressor diameter or size. For a given design it is related to speed. A convenient parameter known as Q/N (i.e. Volume/Speed) is used to size the compressor.

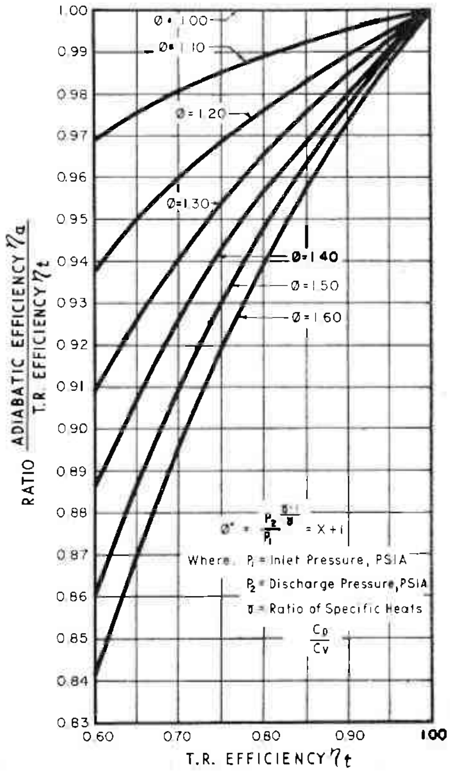


Figure 2 (right). Convenient curve for conversion of a polytropic to adiabatic head.

This parameter is also checked at discharge conditions and must fall within the range of the manufacturers rating conditions.

Speed—The speed is determined from the frame size, and head per impeller. It may be varied due to:

(1) Need or desire for a lower speed,
(2) Parallel operation requiring a steeper characteristic, or
(3) High molecular weight gases and sonic problems.

Each of these variations is met by adding one or more impellers to the frame selected, or by adding a second case.

Horsepower—Gas horsepower is given in the formula:

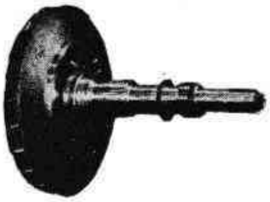
$$GHP = \frac{\text{Head} \times \text{Mass rate of flow}}{33{,}000 \times \text{efficiency}}$$

Friction horsepower from bearings and seals must be added to this figure.

The hydraulic efficiency is a measure of the "losses" inside the compressor such as shock, turbulence, disc friction, etc. It is exclusive of friction horsepower resulting from bearing and seal losses. The friction horsepower which is 1-2 percent of the gas horsepower must be added.

The losses decrease as the size of the compressor increases since the relation of area and volume is less. The losses increase as stages are added.

Conclusion—The foregoing items will enable an engineer to select or better evaluate single compressors, involving such applications as catalytic cracking, gas recovery and catalytic reforming. Two case offerings with a single drive involve only a head correction for a different frame size. ##



Process and Mechanical Design

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MANY GENERAL mechanical design specifications have been established for the present day commercial centrifugal compressor. As better materials are obtained and a better knowledge is acquired of the function of the integral parts of these machines, the limits of good commercial practice are being extended. By virtue of this increased knowledge, it is expected and there is obtained longer trouble-free service life than was experienced in the past.

The service requirements for centrifugal compressors have also become more challenging as new processes are developed and gases which chemically attack ordinary materials must be compressed. Fortunately, as the process needs have changed, the metallurgist has supplied higher strength and more corrosive resistant materials. This makes it possible to handle all but a very few gases and vapors. Even extremely corrosive fluids can be handled if limited service life can be tolerated.

The centrifugal compressor industry has established many standards which define the present day commercial machine.

Compressor shells or casings are usually designed to withstand internal hydrostatic test pressures of one and one-half times the working pressure. Most manufacturers also test the casing at full working pressure with gas under water to observe any leaks by bubble formation. Or, a tracer gas can be used and minute leaks are detected by means of a halide lamp or an electronic leak detector. This latter method is extremely sensitive.

Compressor rotors will differ depending upon the design practices of the various manufacturers and compressor requirements. The individual impellers can be manufactured by several different methods. They can be built up by welding or by riveting the blades to the hubs and covers. Or they can be cast or completely machined from a solid forging. The latter impellers are of the open design.

The tip speed at which impellers are usually operated will range from about 800 feet per second as a maximum for riveted construction up to 1300 feet per second for open impellers machined from a solid forging. These speeds are not absolute limits by any means, and will be exceeded in commercial practice by centrifugal compressors of the future.

For many applications of centrifugal compressors low tip speeds are desired, because a low speed impeller with backward curved blades will produce a wide stable operating range. The high tip speed, radial vane design, on the other hand, will limit the volume flow range over which the compressor can operate without pumping or "surging." As the tip speed of the impeller increases, the centrifugal compressor characteristic approaches that of the axial compressor; a very steep curve with small stable volume range.

To prove the adequacy of the impeller, it is the general practice of the manufacturer to overspeed test the individual impellers and then the complete rotor assemblies. The complete rotors are usually tested at speeds of 110 percent of the maximum allowable continuous operating speed specified for the compressor. During the manufacture, the individual impellers are overspeeded to at least 115 percent of the maximum continuous speed specified for the compressor.

In discussing speeds up to this point, no mention has been made of revolutions per minute. This must follow directly from the tip speed limitations and the volume capacity desired in the centrifugal compressor. The tip speed will determine the pressure rise per stage which is theoretically attainable, but a limit on rpm will determine to a large extent the efficiency and actual pressure rise which is attainable.

For example, the limit on rpm has been established at approximately 10,000 rmp for the past several years. This means that the individual impellers cannot be designed to give satisfactory efficiency for flows less than about 500 actual cfm. Below this flow with impellers limited to 10,000 rpm and large enough to have a reasonable tip speed (pressure rise) the gas flow passages become so narrow that aerodynamic losses become great, and the efficiency becomes quite unattractive. To overcome this, the future small volume compressors must be smaller in diameter and must be operated at higher revolutions per minute.

These higher speed compressors (15,000 to 20,000 rpm) do not present any real mechanical problems for the designer. Bearings and shaft seals have been under study by most manufacturers for some time and higher rotating speeds present no insurmountable problem.

Normal surface running speed for shaft journal bearings is about 140 feet per second for the present day centrifugal compressors. However, journal bearings have been run at twice this speed with no difficulties. One fear that has entered the minds of the operators of higher rotating speed machines is the danger of bearing instability. This is a phenomenon which has recently occurred on some few compressors and their drivers. The instability results primarily when lightly loaded journal bearings are operated at high rotating speeds. The exact nature of the instabilities that result and the methods employed in design to circumvent instability troubles are too involved for this discussion, but most manufacturers of this type of machinery know how to avoid these difficulties. Unfortunately, many vibration troubles have been labeled as bearing instability and it has been a difficult task to remove this stigma even though in most cases it has been proven that the vibration came from other causes.

The shaft seals for centrifugal compressors are designed to fit the service requirements. In applications where a slight amount of gas leakage from the machine can be tolerated, a simple shaft labyrinth can be employed. This type of seal is cheap and is easy to maintain. Preventing the leakage of gas to the atmosphere can be accomplished by using a sealing gas, or by employing a seal pressurized with oil or some other fluid. Each manufacturer of centrifugal compressors can supply a variety of seals and sealing systems to meet the intended service satisfactorily.

In general, centrifugal compressor mechanical design is aimed at creating rotating elements of adequate strength to withstand centrifugal stresses and bearings of a suitable nature which will support the rotating shaft assembly. The casing must be the pressure vessel and furnish support for the journal and thrust bearings. All of this, of course, must be handled in such a way as to provide properly designed inner stage passages. The bearing span and shaft size must be such as to place the natural frequency of the shaft assembly outside the speed range needed to meet the customer's requirements. The mechanical design problems are well understood and can be solved in a straightforward manner.

In this brief discussion few mechanical design limits have been stated, since each manufacturer of centrifugal compressors will have different limiting values which best fit his basic design of the machine. However, it is safe to say that in almost any category, the present day commercial limits will be exceeded in compressors of the near future. The process requirements are demanding new types of machines and the centrifugal compressor is being extended into these new fields as the requirements develop.

Limits of pressure level, types of gases being compressed, volume flows being handled, sealing systems being employed—all of these are constantly changing as process demands require them to change.

Lubrication Systems

There are two general classifications of lubrication systems in a centrifugal compressor: (1) Lubrication of the bearings, which include the main journal bearings and the thrust bearing, must be dealt with in all compressors. (2) Lubrication must also be provided for seals. Where the seals require lubrication, they probably represent the most extreme problems of the whole compressor installation.

Journal bearings and thrust bearings have been used by industry on rotating machines for so long that the lubricating problems and their general requirements are very well established. Even at high rotating speeds, modern journal bearings are nearly fool-proof. Modern high speed bearings, in general, should have oil supplied to them at temperatures probably not in excess of 120 F. and in sufficient quantity so that the heat which they generate may be removed without an oil temperature rise of more than 40 F. There are many designs of bearings which will alter these general conditions, but in general the above limits represent a satisfactory relationship. The bearing is relatively simple in its modern form and the quantity of oil that should be supplied from the lube system is only that necessary for adequate heat removal.

As long as the compressor has no other contact surfaces, and if labyrinths or gas sealing devices are used and no other lubrication is required for seals, the lubrication system can be relatively simple. Many customers now prefer an external main oil pump and an auxiliary oil pump. One is a motor driven unit and the other is usually a steam turbine driven unit. These are operated through adequate pressure controls so that if the main oil pump fails to deliver oil pressure, the auxiliary will start up. Each of these pumps is good for continuous duty and should be able to carry all requirements of the compressor.

There are some users and some manufacturers who prefer that the main oil pump be an integrally driven unit from the main compressor shaft. This has its advantages, but also some disadvantages. Drive gear mechanisms are required. With this arrangement, external oil piping is simplified and in a great many instances the floor space is reduced. This space saving may be important for the customer.

The general trend, however, is toward replacing the internal oil pump, especially if the unit is to be operated continuously or under circumstances where it would be expensive or undesirable to take the compressor out of operation. Service failure of a shaft driven pump may require such an emergency shut-down.

As far as coolers and filters are concerned, it is generally felt that twin coolers and filters should be employed with a suitable switching arrangement. This allows one of these to be cleaned while the other unit is in operation. The only real requirement for these coolers should be that they have removable tube bundles and that they be adequate for the heat rejection service expected of them. Small tubes closely nested are the most efficient from a heat transfer standpoint. The optimum would be ⅛-inch tubes, closely nested together. However, this is not generally practical from the standpoint of clogging from the water side, with a great many of the cooling water sources that are used today.

Therefore, some compromise must be accepted and the oil cooler manufacturers have a variety to offer. It is a general recommendation that nominal ⅝-inch tubes be used to meet the general requirements of minimum floor space, minimum cost, and ease of maintenance, with good reliability. Some users of centrifugal equipment have standardized on ¾-inch prime surface tubes for their oil coolers. There may be special reasons for this and if there are, they should be adhered to. The equipment is available, but it is more expensive and usually consumes more floor space.

Filters in the oiling system are a problem in themselves. It is very difficult to lay down specific recommendations on a general basis as to what style and type of oil filters should be used. Every job has its own problems. Filters are so vitally important that they should be in the system in such quantity that they can be cleaned, re-charged, or whatever is necessary to reactivate them, while the plant is in operation.

The filter problem becomes more acute if the gas handled will react in any manner with the oil when there is any possibility of the oil and gas coming in contact.

The above discussion tends to recommend twin coolers, twin filters, and twin pumps. There are many applications where sparing of auxiliary equipment is not warranted. The requirements should be established for the particular installation by balancing the cost of an emergency shutdowns and the normal frequency of scheduled shutdowns against the cost of the spare auxiliary equipment.

If the compressor is equipped with seals, requiring oil for heat removal or requiring oil for pressure film operation, the auxiliary system and the lube system can become more complicated.

Probably the simplest form is where the requirements of the pressure portion of the seals are sufficiently low that normal pressure lubrication oil can be used. Under those conditions, all that is required is an increase in the size of oil pumps, filters and coolers.

If, however, the seals require higher pressure levels than the journal bearings, then the seal oil problem becomes a matter of choice, either to have completely separate lube and seal oil systems or to integrate them. If it is at all possible, it is more economical from both the cost and space standpoint to use the combined system. There are fewer items of equipment to install, and fewer pumps to keep running.

If the integrated system can be used, the general method would be to supply oil at lubricating oil pressure level for the journal bearings and thrust bearings, and to do all the necessary cooling and filtering at this level. Then from the low pressure system, high-pressure oil for the seals only can be supplied by special high-pressure booster pumps.

If completely separate seal oil and lube oil systems are required, then the major items such as coolers, filters, reservoirs and pumps must all be furnished for both the seal oil and the lube oil systems. The auxiliary items for the high pressure seal oil system (above 250 pounds) become costly and cumbersome.

If the gas handled by the compressor causes contamination, chemical reaction or any other deterioration of the oil, sealing systems have been developed which are of such a nature that they completely isolate the compressor gas stream from the atmosphere and the lube oil. The quantity of oil going toward the gas stream is relatively minute. Under some conditions, this contaminated oil might well be thrown away. It is probably less expensive to do this than to set up a purification system with elaborate filters which may be chemical as well as mechanical and to maintain that system as well as operate it. Oil loss rates for seals currently on the market are between one and eight quarts per day per seal.

The lubrication systems of some compressors and drivers may be combined under satisfactory operating conditions. If the lubrication oil of the compressor is not in contact or contaminated with the gas stream in any way, it is completely acceptable to combine them. This is particularly true when the driver happens to be a motor and gear set, and there is no possibility of water contamination of the oil from the steam turbine source.

However, there are many combined steam turbine and compressor lubrication systems in very satisfactory operation today. These are mainly on air compressors, but the system can be used when gases are compressed if isolating seals are used.

Requirements for Foundations

Foundations for centrifugal compressors are relatively

NOTES

simple. However, even though they may be simple, they are critical, and quite often not enough attention is given to the problem. The first requirement for the foundation for a centrifugal compressor and its driver is that it keep the equipment in perfect alignment. The second and probably equally important function is to carry the load of the unit and distribute it properly onto the floor structure.

There are two general classifications of bases: (1) Reinforced concrete with steel sole plates and (2) cast or fabricated iron or structural steel.

Concrete foundations are suggested first because in general they are the least expensive and at the same time the most satisfactory. They also tend to eliminate vibration problems such as might be caused by resonant or harmonic frequencies of the unit. Steel sole plates adequately anchored in the concrete serve as finished surfaces for alignment of the equipment. Adequate reinforcing steel, at both the top and bottom of the concrete slab, should be used.

The concrete slab should be in the order of 18 inches thick at a minimum, and will seldom be greater than 30 inches thick. It is a simple matter to provide for variations in height between compressor and driver when this concrete foundation is being poured.

Steel foundations can be useful where reduction of weight is a prime consideration, however they are costly and require quite extensive engineering design. It is a simple matter to design them with adequate strength to carry the loads, but vibration studies must be made to make sure that the steel base is not in resonance with any basic frequency of the compressor or driver.

There has been some feeling that when a continuous steel base was purchased under compressor and driver, the unit would be completely assembled and shipped as a unit from the manufacturer. This is not so in most cases because the compressor and the driver are such heavy pieces of equipment that they cannot safely be shipped in that manner. Therefore, the assembly problem remains in the field just as though no steel base was furnished.

As a third factor, many installations have been visited where steel bases have been supplied and the writer has observed that under the steel base is a simple concrete slab carrying the steel base. This concrete slab is raised several inches above the floor line. Since this slab had to be poured, it was necessary to establish dimensions for the depth of the steel base, and location of hold-down bolts between the steel base and the concrete slab. This was almost as much work and expense as using concrete and sole plates for direct mounting of the units. The cost of the structural steel here was an extra item.

If the unit is to be mezzanine mounted on a steel framework with steel floor beams, then the steel base can be very readily justified, as part of the steel work.

Bases of cast iron are not too common today. The cost of the pattern equipment under the present economic atmosphere tends to make the welded steel more acceptable.

Accessibility for Maintenance

The ideal, of course, for accessibility is engine room space which will permit mounting of the compressor and driver at floor level with gas and steam lines coming up through the floor. All lubrication equipment for both turbine and compressor would be located away from the unit so that there would be a clear six-foot aisle all around the unit. In this type of arrangement, an overhead crane or a mono-rail could easily be made available to lift the upper half of the compressor casing as well as the upper half of the drive casing.

However, in many cases, this ideal is either impossible or not practical because of the cost of the space involved. There are many satisfactory substitutes that can and are being effectively used today. Certainly, the equipment should have one clear access aisle with the lube systems and gas and steam piping on the other side. It should be noted that a centrifugal compressor requires only a minimum of access room, the reason being that there are so few parts that require inspection under normal conditions. Further, these parts are generally so small that they can be handled without the use of auxiliary lifting equipment. Inspection normally would consist of removing the journal and thrust bearing parts of the compressor. In almost all designs available, these are externally accessible through either their own covers or specially designed inspection covers on the compressor housing. The basic minimum space requirement therefore, becomes sufficient room for one man to work, if no more can be provided. Most jobs will be one man operations, such as lifting the bearing cap and checking the bearing liner, or inspecting the thrust bearing shoes.

Almost every installation, with reasonable care and planning, can provide access to the centrifugal machine and its driver from each end and on one side. This is quite satisfactory and the space need not be excessively large. The end clearance should be enough to allow a man to work comfortably except in the case of certain specialized equipment which may require movement of an internal casing axially out of the end of the machine. Under these conditions, axial clearance space must be maintained.

Other items requiring maintenance are oil pumps, oil coolers, and filters. Generally, it should be possible to pull tube bundles of all the oil coolers in one direction. Oil pumps should be arranged so that they can be worked on conveniently or at least can be removed from the system and taken to the shop for maintenance work. Oil filters should also be conveniently located so that they can be inspected, and cartridges or whatever is necessary can be changed.

It is strongly recommended that space be allocated in the general machine room layout for the auxiliary equipment with adequate clearance about the major components.

The auxiliary equipment frequently requires as much or more maintenance work than the main compressor and driver. ##

How to Control Centrifugal Compressors

Use performance curves to determine best control method for your particular applications

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Mountainside, N. J.

THE TREND of modern day refinery and petrochemical processes dictates, for economic reasons, the replacing of positive displacement type compressors with centrifugal compressors. These dynamic compressors are essentially a constant pressure, variable volume machine (as contrasted to a positive displacement compressor being a constant-volume, variable-pressure machine) and must be regulated in almost all applications. Before control can intelligently be discussed, the operating characteristics must be understood.

Constant Speed Performance Curves. The most universal centrifugal compressor performance map would plot capacity versus head and versus efficiency. This discussion will concern itself with polytropic compression and, therefore, polytropic head and efficiency.

Figure 1 depicts a typical centrifugal compressor performance map operating at some fixed speed. A typical specific performance map plots capacity in terms of inlet cfm (cubic feet per minute) on the absicissa and brake horsepower and discharge pressure on the ordinate. The map is usually based on constant inlet conditions, such as pressure, temperature, MW and K value of the gas.

In Figure 1, the shape of the head curve is shown for a particular impeller design. Aerodynamic design of centrifugal compressors is beyond this discussion; however, it should be pointed out that centrifugal compressor impellers can be designed to some extent to achieve more nearly the sloping characteristics of that required to suit the systems requirements. For practical reasons, manufacturers usually will offer standardized impeller designs with standardized frame sizes.

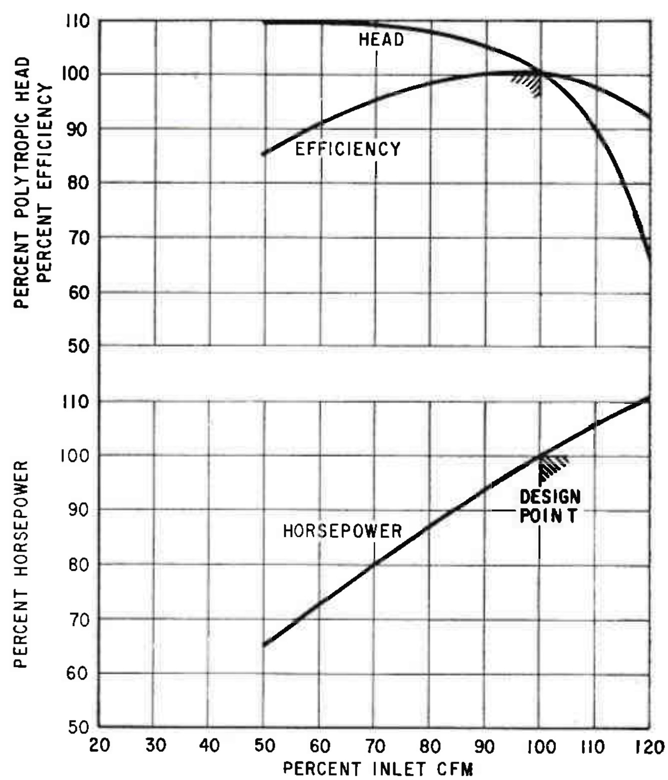


FIGURE 1—Typical constant speed performance curves for a centrifugal compressor.

A given centrifugal compressor design "knows" only its inlet volume and its speed. At a given speed it will develop a particular head for a given flow throughout. Consider the inlet cfm equation:

Equation (1)

$$Q_1 = WV_1 = \frac{W}{\rho_1} = \frac{WZ_1RT_1}{P_1\,144}$$

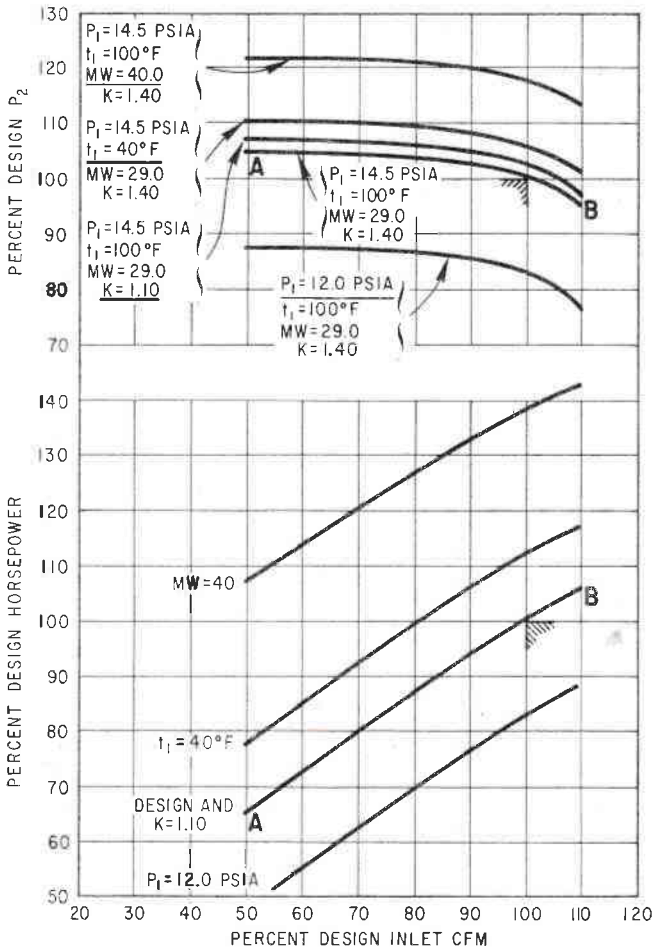


FIGURE 2—Effect of changing gas conditions on centrifugal compressor operating at constant speed.

Where Q_1 = Inlet volume, cfm

W = Total mass thruput, #/min.

V_1 = Specific volume at inlet conditions, ft³/#

ρ_1 = Density at inlet conditions, #/ft³

Z_1 = Inlet compressibility factor

R = Universal gas constant

$$\frac{\text{ft — lb}}{\text{lb — °R}}\left(R = \frac{1545}{\text{Mole Wt}}\right)$$

T_1 = Inlet temperature, absolute, °R (°F + 460)

P_1 = Inlet pressure, absolute, psia

From Equation (1) it can be seen that it is possible to change completely the design inlet gas conditions and still maintain design inlet cfm. For all practical purposes (within the scope of this discussion), the polytropic head and efficiency of a given single stage centrifugal compressor remains the same even though the design gas and gas conditions have changed, as long as design inlet cfm and speed is obtained.

Effect of Changing Gas Conditions. Consider the polytropic head equation:

Equation (2)

$$H_p = Z_m R T_1 \left(\frac{K\eta_p}{K-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{\eta_p K}} - 1\right]$$

Where H_p = Polytropic head, $\frac{\text{Ft — lb}}{\text{lb}}$

Z_m = Mean compressibility factor

R = Universal gas constant,

$$\frac{\text{Ft — lb}}{\text{lb — °}_R}\left(R = \frac{1545}{\text{Mole Wt.}}\right)$$

T_1 = Inlet temperature, absolute, °R

K = Specific heat ratio, c_p/c_v

P_1 = Inlet pressure, absolute, psia

P_2 = Discharge pressure, absolute, psia

η_p = Polytropic efficiency

Since a given compressor design operating at design speed will produce design head when passing design inlet cfm, the discharge conditions, pressure and temperature will be determined solely by the thermodynamic properties of the gas being compressed. To illustrate this further, Figure 2 shows the characteristic curve for a given centrifugal compressor operating at constant speed but under varying inlet conditions.

Curve AB represents the characteristic for a centrifugal compressor designed to handle air at an inlet pressure of 14.5 psia, inlet temperature of 100° F, molecular weight of 29.0 (dry air), and K value of 1.40. This unit would develop 100 percent of design discharge pressure at an inlet capacity of 100 percent design cfm.

If now, the intake air temperature is decreased from 100° F to 40° F, all other conditions remaining the same, the discharge pressure would increase to 106 percent of design. This can be seen by the head equation (2) previously discussed. The horsepower required under these conditions would be more than design by the ratio of the absolute temperature $\left(\frac{460 + 100}{460 + 40}\right)$, since by decreasing temperature we increase density and, therefore, mass flow for a given volume rate (refer to Equation 1 and 3).

Equation (3)

$$GHP = \frac{(W)(H_p)}{(33000)(\eta_p)}$$

Where GHP = Gas Horsepower

W = Total Mass Flow, lbs/Min.

H_p = Polytropic Head, Ft. lb/lb

η_p = Polytropic Efficiency

By lowering the inlet pressure from 14.5 psia to 12.0 psia, all other conditions remaining constant, the discharge pressure would drop to 83 percent of design—by the exact percentage the inlet pressure dropped, since the developed pressure ratio has not changed.

The horsepower again would be reduced by the ratio of the inlet pressure to that of design $\left(\frac{12.0}{14.5}\right)$ since, in this case, the inlet density has been reduced and, therefore, mass flow for a given volume rate, (refer to Equation 1).

By changing the composition of the gas such that the molecular weight increases from 29.0 to 40.0, all other conditions remaining the same, the discharge pressure would increase to 118 percent of design. In this case, the horsepower would increase from design horsepower by the ratio of the molecular weights (40/29).

Lastly, if the ratio of specific heat value, K, is decreased from 1.40 to 1.10, all other conditions remaining the same, the discharge pressure will increase to 102 percent of design with no change whatsoever in compressor horsepower.

Capacity Limitations. There are definite limitations of the stability range of a centrifugal compressor. Limiting the minimum capacity of a given centrifugal compressor is a phenomenon called "surge" which normally occurs at about 50 percent of the design inlet capacity at design speed. This extremely complex phenomenon is probably still one of the most difficult problems in the field of fluid dynamics. To provide a simplified explanation, consider the single stage compressor operating at constant speed, discharging through a throttling valve. By throttling on the discharge valve, we increase the system resistance and, therefore, the head required by the compressor to overcome this resistance. As we continue to throttle this valve, less flow will be capable of flowing through the compressor.

This continues up to the point of maximum head capability of the compressor. At flows below this "surge" limit, the compressor head characteristic curve takes a reverse slope, resulting in decreased head capability. At this condition, the system back pressure exceeds that capable of the compressor delivery, causing a momentary backflow condition. At this time, however, the back pressure has been lowered, enabling the unit to again be capable of delivering flow higher than the flow at which the surge began. If the obstruction to flow downstream of the compressor is unchanged (i.e., same discharge valve position), operation follows back along the head characteristic curve until the peak head delivery is reached again. This cyclic action is what the industry properly calls "surge". To operate at flows below the surge flow, requires controls which will be discussed later.

While the stability range of a centrifugal compressor is commonly indicated from the rated point to the surge point, the unit can operate stably to the right of the rated point. The greater the load demand on a centrifugal compressor, the greater the "fall-off" in delivered pressure. The upper limit of capacity is determined by the phenomenon of "stonewall". Stonewall occurs when the velocity of the gas approached its sonic velocity somewhere in the compressor, usually at the impeller inlet. Shock waves result which restrict the flow, causing a "choking" effect—rapid fall off in discharge pressure for a slight increase in volume throughput. Stonewall is usually not a problem when compressing air and lighter gases; however, in compressing gases heavier than air, the problem becomes more prevalent as the molecular weight increases.

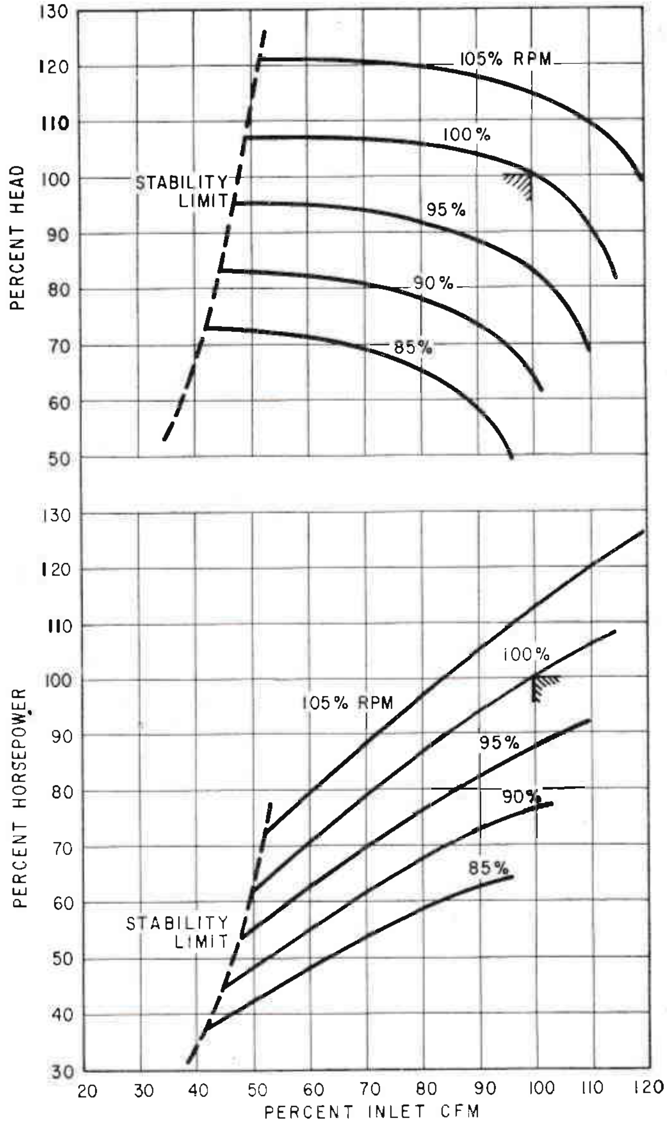


FIGURE 3—Typical variable speed performance curves for centrifugal compressor.

In discussing the operating range of centrifugal compressors, limitation to single stage compressors was purposely made for simplicity. As the number of stages increases, performance maps tend to show a more sloping curve with lesser stable range than shown in Figure 1 or Figure 2, as dictated by the particular application.

Variable Speed Performance Curves. A typical variable speed performance map is shown in Figure 3. With variable speed, the compressor easily can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination variable capacity and variable pressure.

Basically, the performance of the centrifugal compressor, at speeds other than design, are such that the capacity will vary directly as the speed, the head developed as the square of the speed, and the required horsepower as the cube of the speed. As the speed devaites from the design speed, the error of these rules, known as the Affinity Laws, or Fan-Laws, increases.

By varying speed, the centrifugal compressor will

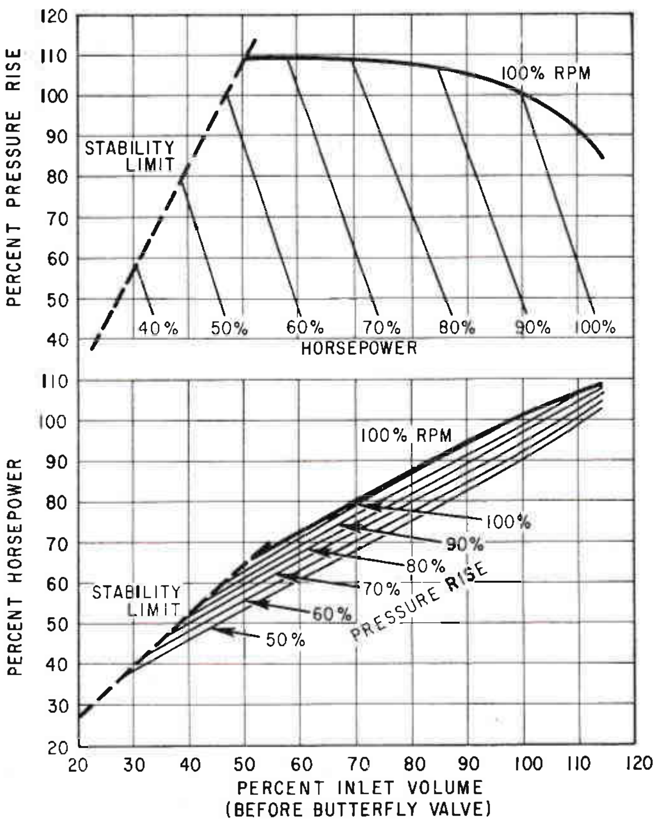


FIGURE 4—Typical constant speed performance curves. Compare the slope of the constant horsepower lines with Figure 5.

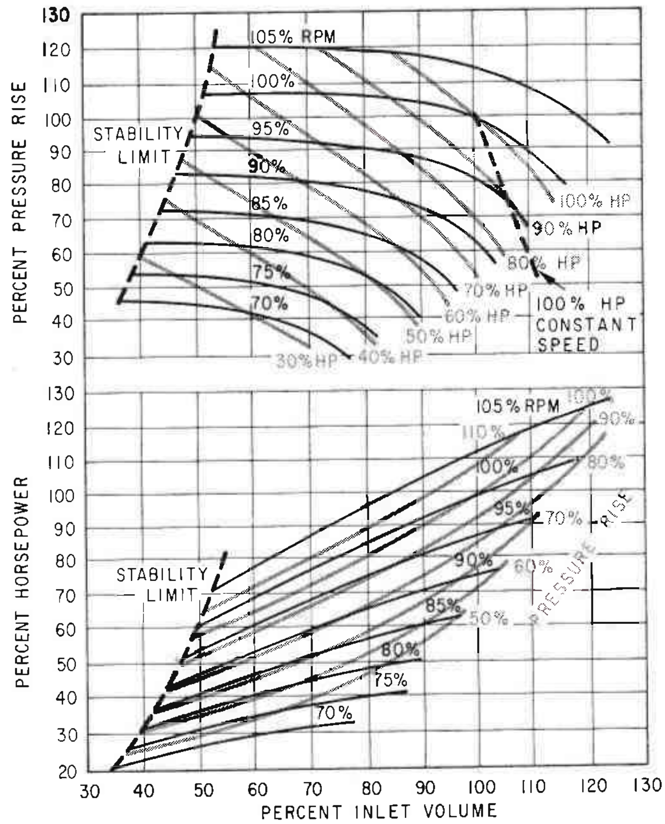


FIGURE 5—Typical variable speed performance curves. Compare the slope of the constant horsepower lines with Figure 4.

meet any load and pressure condition demanded by the process within the operating limits of the compressor and the driver. It normally accomplishes this as efficiently as possible, since only that head required by the process is developed by the compressor. This compares to the essentially constant head developed by the constant speed compressor.

Factors Affecting Decision of Control. For the majority of centrifugal compressor applications, some form of regulation is required. The type of control used depends first on the compressor driver.

For turbine driven compressors, normal control is accomplished by varying the speed. This method of control permits a wide range of operation in a relatively efficient manner. Speed control is more efficient than throttling the flow at constant compressor speed since, by artifically creating resistance, an unrecoverable loss in power results. This can be seen by comparison of Figure 4 and 5 which shows constant speed and variable speed performance curves respectively. Compare the slope of the constant horsepower lines.

For motor driven compressors, the control can become more intricate, especially for the usual constant-speed type motors. For this type driver, means of obtaining control are:

1. Use of a hydraulic or electric coupling between motor and compressor to obtain speed variation. This is not a popular method of controlling speed because of severe coupling efficiency penalties throughout the range of speed operation.

2. Use of butterfly valve at the compressor inlet or at compressor discharge. Throttling at the suction is preferred, since by so doing, the gas density is decreased, thus meaning less mass flow for a given inlet cfm. In other words, throttling at the discharge does not take advantage of the head nor density reduction obtained by inlet throttling which artificially increases the compressor cfm toward the rated point, resulting in lowered horsepower requirements.

3. Use of adjustable inlet guide vanes which adjust the characteristic curve of the centrifugal compressor. The adjustable inlet guide vanes are most effective for conventional compressors developing less than 30,000 feet head or a multistage compressor with three stages or less. Power savings by the use of adjustable inlet guide vanes at the inlet of a compressor over suction throttling can approximate 10 percent for a single stage compressor, 5 percent for a two-stage compressor, and 3 percent for a three-stage compressor. In a sense, the adjustable inlet guide vanes change the aerodynamic design of the first impeller. It does this by prewhirling the gas entering the wheel in the direction of its rotation, thereby developing less head than at design. (An analogy to this "unloading" of the first wheel can be more easily understood when considering a sled being pushed downhill rather than on level ground). This is accomplished with a decrease in design efficiency. However, the ratio of the head to efficiency decreases from its design value, resulting in lowered horsepower.

Although not used commercially in the United States, adjustable inlet guide vanes are available at the inlet of every stage in conventional multistage compressors by some European manufacturers. Economics justify this expensive and sometimes complicated feature in most European countries because of high utility rates.

4. Use of a power wheel at the inlet of the compressor upstream of the first stage impeller. The gas to be compressed expands through a set of movable guide vanes and is directed upon the turbine blades. The power wheel or turbine wheel is theoretically designed to more effectively handle part load conditions than adjustable inlet guide vanes by itself. However, the power wheel has disadvantages, in particular, to the inherent parasitic losses at the wheel at design conditions resulting in lowered efficiency. This control method is not popular in the United States.

5. Use of adjustable diffuser vanes. Like the multiple adjustable inlet guide vanes, the adjustable diffuser vanes are not commercially used in the United States for the same reasons mentioned. It is important to note that devices such as these, when used with gases that are gummy, corrosive, or erosive, lead to heavy maintenance problems.

6. Use of a wound-rotor induction motor obtaining speed variation by varying the resistance in the rotor or "secondary" circuit This is an expensive and relatively inefficient drive and, therefore, not normally used for industrial centrifugal compressor drives.

7. Use of a direct-current motor obtaining speed variation by varying the field current by means of a rheostat. Again, this is a relatively expensive and inefficient drive, having commercial DC availability problems and, therefore, not normally used for centrifugal compressor drives.

Summarizing, the head-capacity control methods most commonly used domestically are:

- Speed control
- Inlet throttling
- Adjustable inlet guide vanes

The other controls mentioned are seldom seen in the United States. European high cost of power and fuel relative to labor and material there justify the more elaborate types of control.

With limitation to the three common types of control, typical forms of regulation can be studied.

System Characteristics. Unfortunately, from a user's viewpoint, manufacturers are sometimes unjustifiably blamed for providing a compressor design that does not function in the process as expected. In many instances, experience proves that the system resistance was not properly understood by the user in specifying the design and operating conditions of the compressor. It is extremely important, therefore, that the system resistance or characteristics be fully known before discussing or recommending a control system.

Figure 6 shows a plot of pressure versus capacity for a constant speed compressor. Shown on this plot

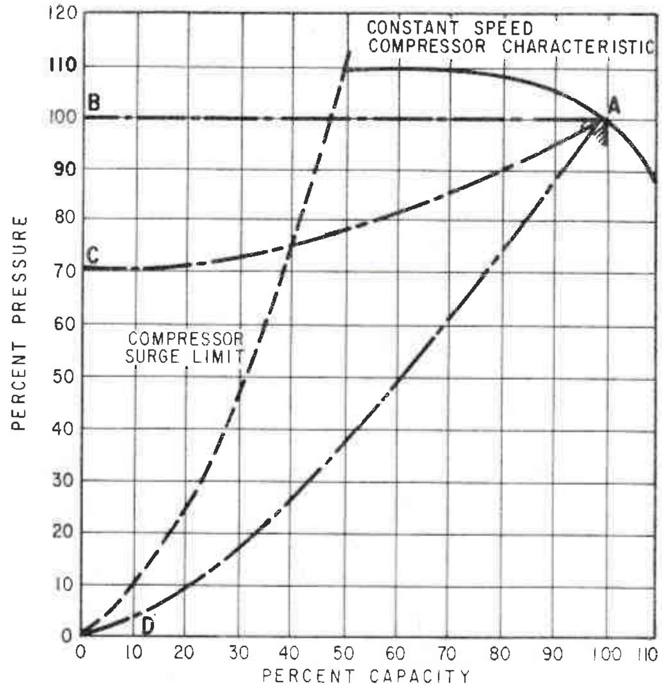


FIGURE 6—Pressure vs. capacity for constant speed compressor.

are three different types of system characteristics. A compressor operating against a fixed head or pressure would have a system characteristic defined by AB. A close-coupled packaged refrigeration compressor such as used in alkylation units would follow a system resembling AB. In contrast to the forementioned system, a compressor discharging into a large system through a long run of pipe would have a system characteristic that follows Curve AD. In this case, all the delivered pressure by the compressor is used to overcome pipe line friction. A natural gas pipe line compressor is an example of this type of system. Most commonly, there are systems which have essentially fixed top pressure, except for some pipe friction in the system. A catcracker air compressor would typify a system represented by Curve AC.

Note in Figure 6 that the compressor head-capacity characteristic curve follows none of the system characteristics. Variation of the compressor output to meet the demand of the system, therefore, requires controls to regulate the volume, pressure, or a combination. In all of the controls, either a high pressure oil system or a source of high pressure air will be required for the operation of a servomotor as the final power medium. Rather than discuss specific details of control mechanisms, schematic diagrams will be used along with a performance curve to illustrate the following types of control:

- Constant pressure control
- Constant pressure control—parallel operation
- Contant weight flow control
- Constant weight flow control—series operation
- Anti-surge control

Constant Pressure Control. Figure 7 shows a constant discharge pressure control system for a turbine-

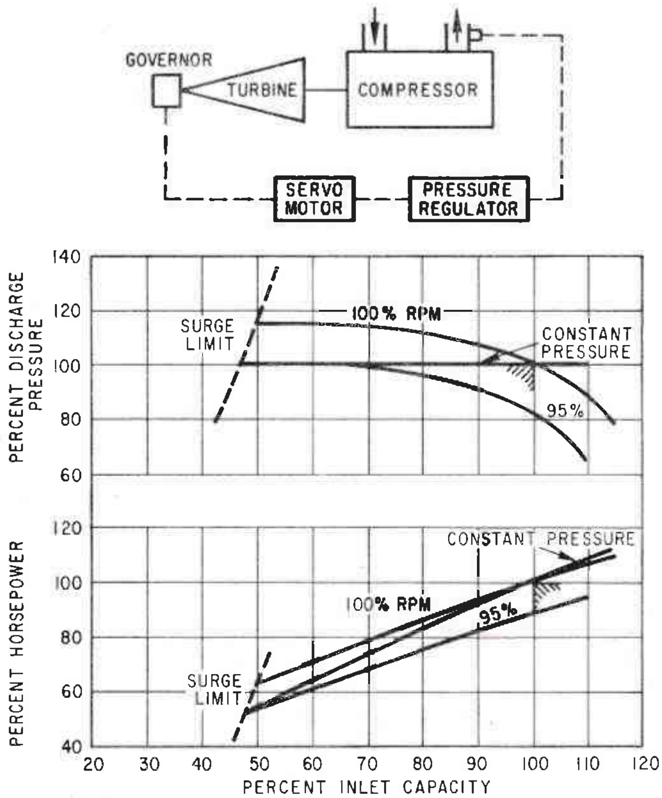


FIGURE 7—Constant discharge pressure control system for turbine driven compressor.

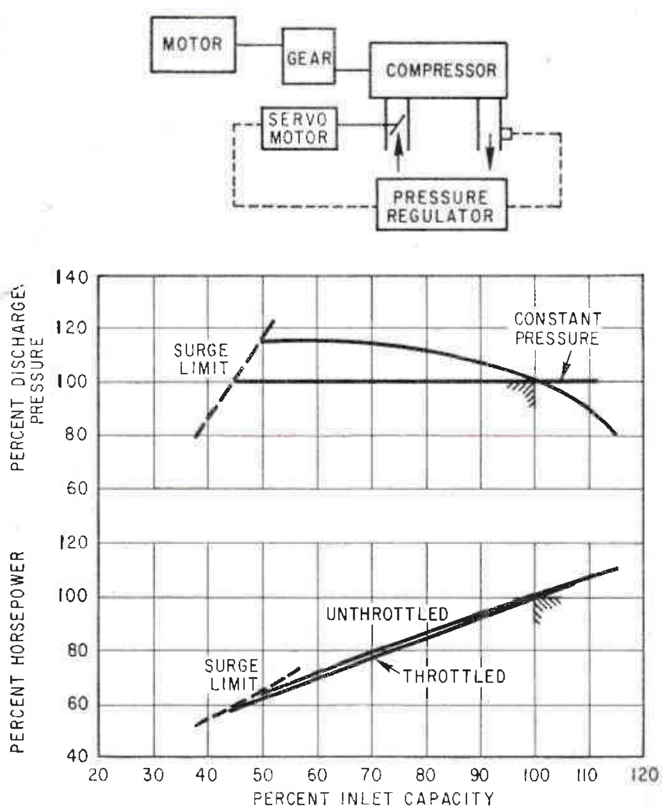


FIGURE 8—Constant discharge pressure control system for motor driven compressor with suction throttling.

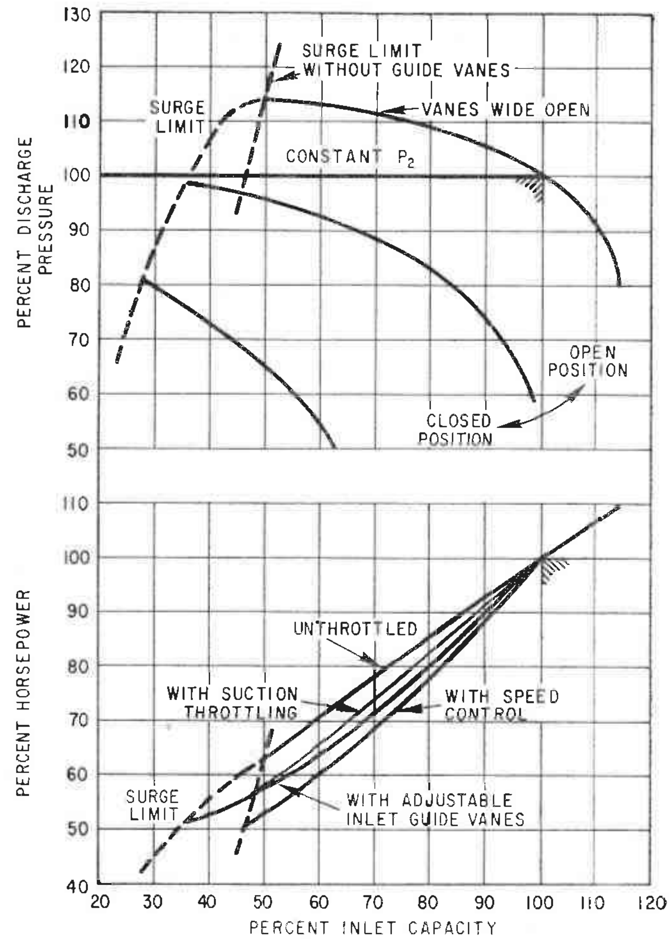
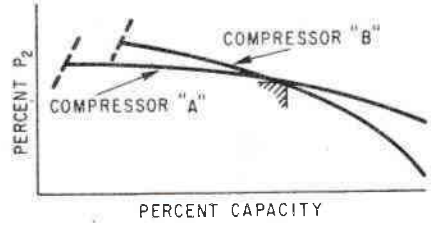


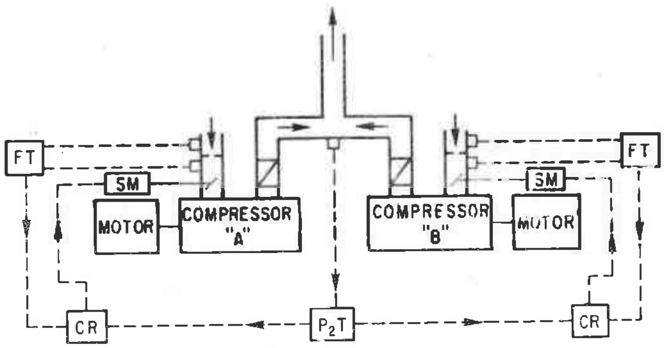
FIGURE 9—Constant discharge pressure control system for motor driven compressor with adjustable inlet guide vanes.

driven centrifugal compressor. The pressure regulator may be air operated or hydraulically operated to position the servomotor. The servomotor in turn is connected to the turbine speed-governing system, or more simply, to the steam valve at the turbine inlet. Constant discharge pressure is maintained by varying the speed of the turbine. Note the performance curve in Figure 7. In this case, the compressor is able to develop constant head by lowering its speed. If there is a 10 percent head rise from design to surge, the approximate minimum speed to maintain constant discharge pressure will be $N_{des}\sqrt{\frac{100}{110}}$ or, roughly 0.95 N_{des}, as determined by the Affinity Laws previously mentioned.

Figure 8 shows a constant discharge pressure control system for a motor-driven centrifugal compressor. In this case, the servomotor positions a butterfly valve located at the compressor inlet. The discharge pressure is held constant by positioning the inlet butterfly valve, thus, throttling out the excess pressure ratio developed at flows lower than design. Note the performance curve in Figure 8. The performance map shows the savings in horsepower by throttling at the inlet of the compressor, as compared to throttling at the discharge.

If the servomotor in Figure 8 were to position adjustable inlet guide vanes rather than an inlet damper, the performance curve would take the shape as shown in Figure 9. For comparative purposes, this





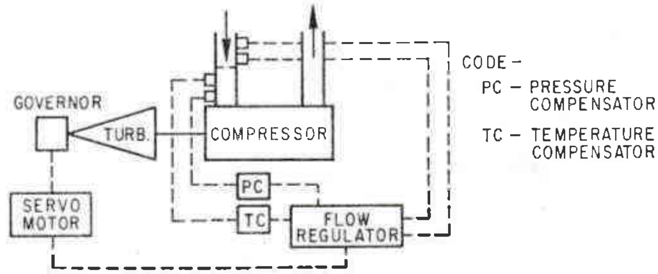
CODE –
FT — FLOW TRANSMITTER
CR — COMPUTER RELAY
P_2T — DISCHARGE PRESSURE TRANSMITTER
SM — SERVOMOTOR FOR INLET DAMPER

FIGURE 10— Constant discharge pressure control system for two compressors operating in parallel having dissimilar operating characteristics.

curve superimposes the curves of Figures 7 and 8. Note the horsepower savings over suction throttling. In contrast, note the horsepower advantage that speed control has over this method of control. Specific percentages are not repeated here because of lack of true meaning. In general, the greater the pressure ratio and number of compression stages, the more pronounced the difference will be between speed control and the other methods

Parallel Operation. Controlling two or more compressors operating in parallel and having identical characteristics would be relatively simple. The system described in Figures 7 and 8 would still apply, except only one pressure regulator would be required for both units. The two or more servomotors obtain a hydraulic or pneumatic impulse signal from the pressure regulator; and, in the case of the turbine driven compressor, would control the turbine speed. In the case of the motor-driven compressor, the servomotors would position the inlet butterfly valves or a set of adjustable inlet guide vanes. In any case, check valves must be installed at the discharge of each compressor to prevent any back flow of fluid to overcome any slight unbalance in the characteristics of the two compressors.

More complicated, but more likely, would be the system involving two or more compressors with dissimilar, or similar but not identical, characteristics. Figure 10 shows a combined performance map for two compressors which have dissimilar operating characteristics. To maintain constant discharge pressure, one compressor would be operating at flow differently than the one in parallel with it. As a result, the control system would have to include a separate flow con-



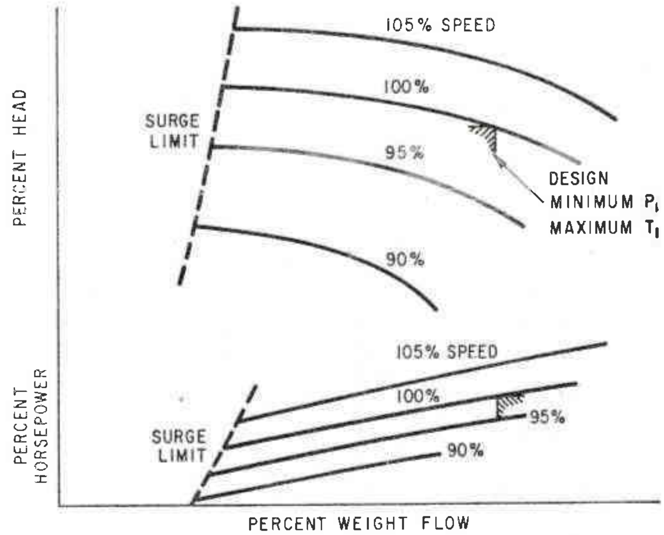


FIGURE 11—Constant weight flow control system for turbine driven compressor.

troller for each compressor unit. Figure 10 shows t system schematically for two motor driven comp sors.

Constant Weight Flow Control. In the case of stant weight flow systems, for a turbine driven pressor, a servomotor actuated by a flow regul would maintain constant weight flow by varying speed of the turbine.

In most systems involving variable system resist (Curve AC, Figure 6) constant weight flow contro some means is used. Figure 11 shows this control s tem whereby inlet pressure and temperature will vary over some known range. In this system, pressure and temperature compensation is included to adjust the signal transmitted by the flow regulator for true constant weight flow control.

The performance curve shown in Figure 11 requires no explanation.

For a motor driven compressor, the system schematically would be similar to Figure 8 with constant weight flow being maintained by positioning an inlet butterfly valve.

Series Operation. If, in Figure 11, an additional compressor body were added in the compression system by means of direct coupling to the first compressor body, the control system would not normally change in any way. That is, as long as one drive controls speed of both compressors, the system can be treated as a single body control problem. (Note: Surge control is not discussed here for simplicity: however, it

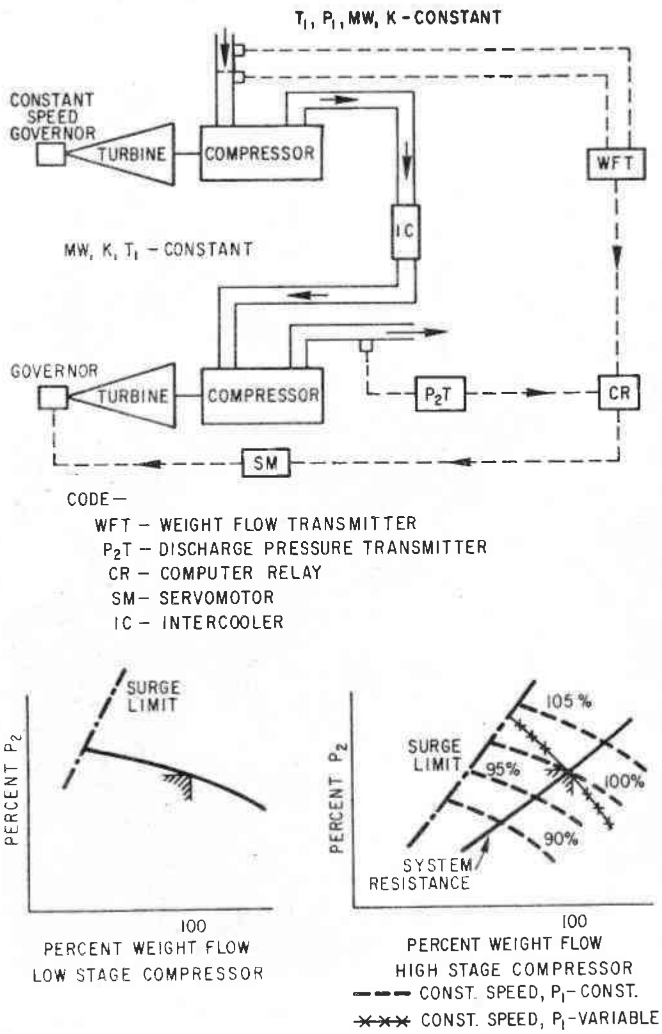


FIGURE 12—Constant weight flow control system for two compressors operating in series.

should be pointed out that by-pass lines should be installed around each compressor, mainly to facilitate start-up).

If, however, each compressor body were driven by a separate drive, such as shown in Figure 12, a simple solution would be to operate the low stage compressor unit at constant speed, letting the discharge pressure rise or fall from design point, in accordance with the compressor's characteristic curve. The high stage compressor would then be speed controlled to maintain constant weight flow. Due to the system resistance, the combination of final discharge pressure and weight flow can indicate the operating speed, as shown on the performance map in Figure 12.

Anti-Surge Control. In some compressor applications, operation is practically always at design capacity. In this case, the surge control would merely consist of a manual bleed valve at the discharge of the compressor. In the case of compressing air, a non-toxic or inexpensive gas, the system which is the simplest possible is shown schematically in Figure 13.

More practically, however, operation at other than design condition would normally require some form of

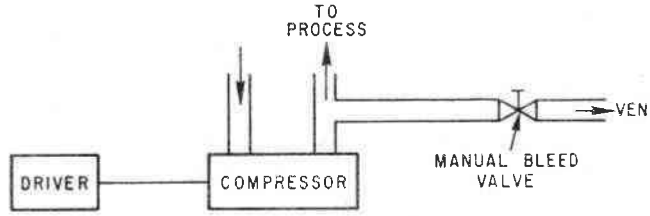


FIGURE 13—Manual surge control system.

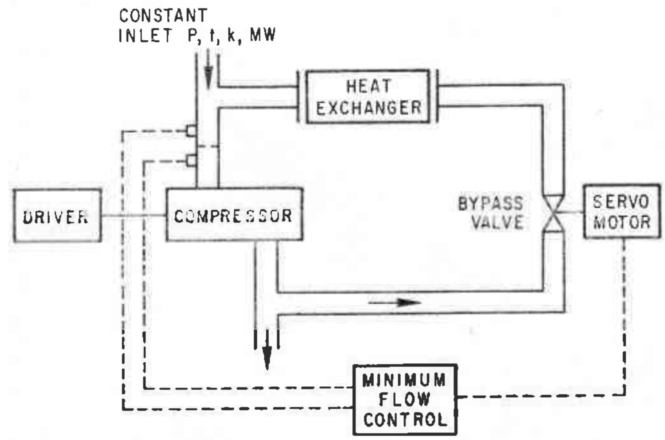


FIGURE 14—Automatic anti-surge control with recirculating bypass.

automatic anti-surge control. Consider the control essentially a minimum flow regulator which, through a servomotor, operates the surge valve as required to maintain stable operation. Again for compressing air, a non-toxic or inexpensive gas, the surge valve can vent to atmosphere. However, in many applications, the gas is expensive or toxic and, therefore, it becomes desirable to re-circulate the flow necessary to insure stable flow through the compressor.

Figure 14 shows a system in which automatic surge control is applied to a compressor handling an expensive gas. Note the bypass cooler which is required to remove the heat of compression before mixing with the process gas at the inlet. This prevents compressor performance fall-off which would occur should the inlet temperature rise above design value.

Centrifugal compressors can easily be controlled because of their inherent performance characteristics. If you keep in mind the thermodynamic factors affecting compressor performance, a control system can be designed to fit the process in which the compressor functions. ##

About the Author

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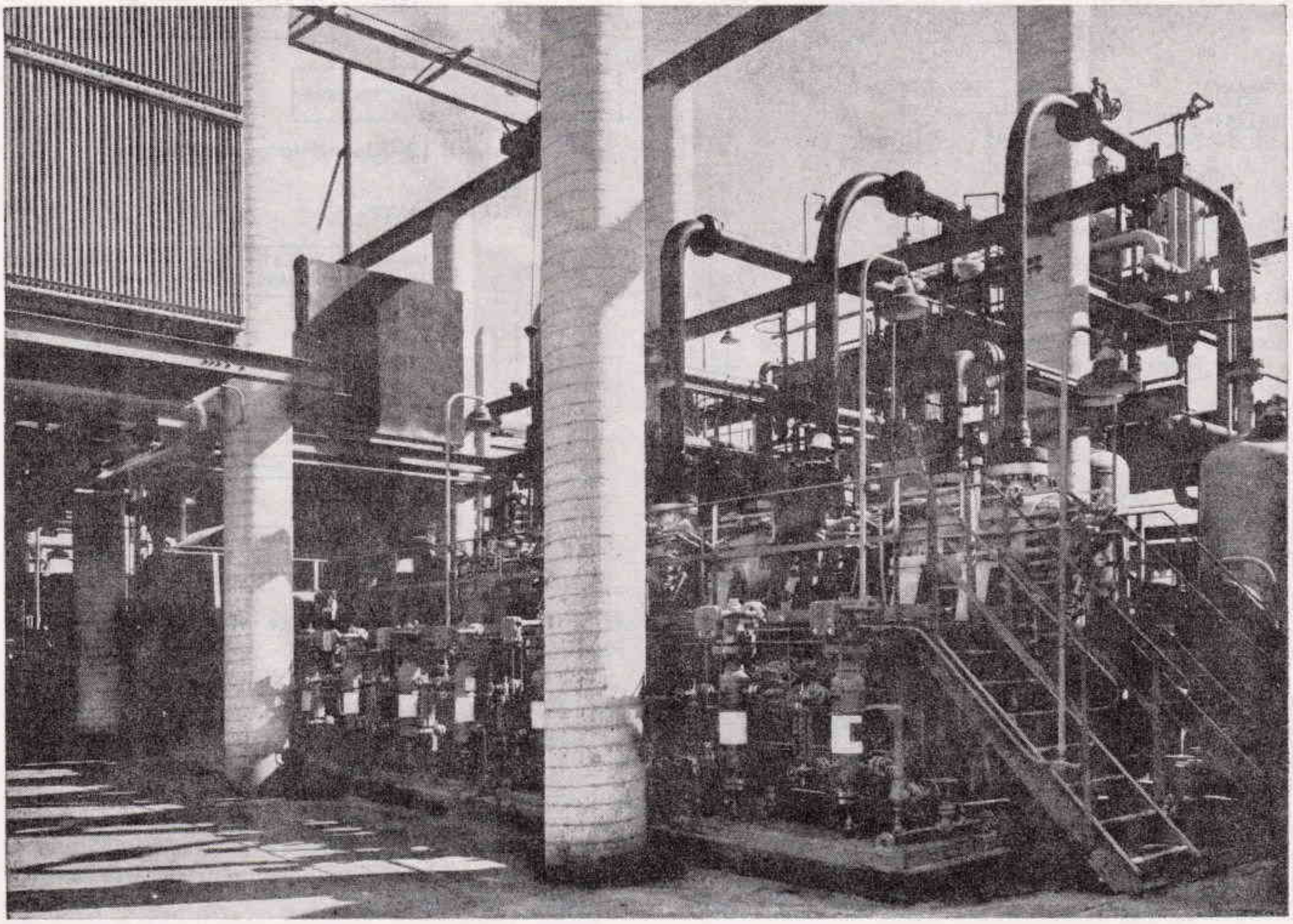

Gas turbine-driven compressor train for 600 ton/day ammonia plant located at Amoco Chemical Co., Texas City, Texas.

How to Instrument Centrifugal Compressors

Unless capacity control is provided for centrifugal compressors, the system may become unstable, causing surging and upsets in the compressor and process

Richard E. Daze, The M. W. Kellogg Co., New York

The performance characteristics of the centrifugal compressor are such that unless adequate and proper capacity control is provided, through instrumentation, the compressor circuit may become unstable, causing surging and upsets in the compressor and process. The objective therefore of any centrifugal compressor control system is to achieve smooth capacity regulation and to prevent the compressor from surging, even though the process flow drops below the surge limit of the compressor.

This article will describe some of the methods used to control the capacity of centrifugal compressors and illustrate typical instrumentation for several compressor services.

System Characteristics. To determine the optimum compressor control and instrumentation requirements for a given application, the performance characteristics of the compressor and of the connected system must be

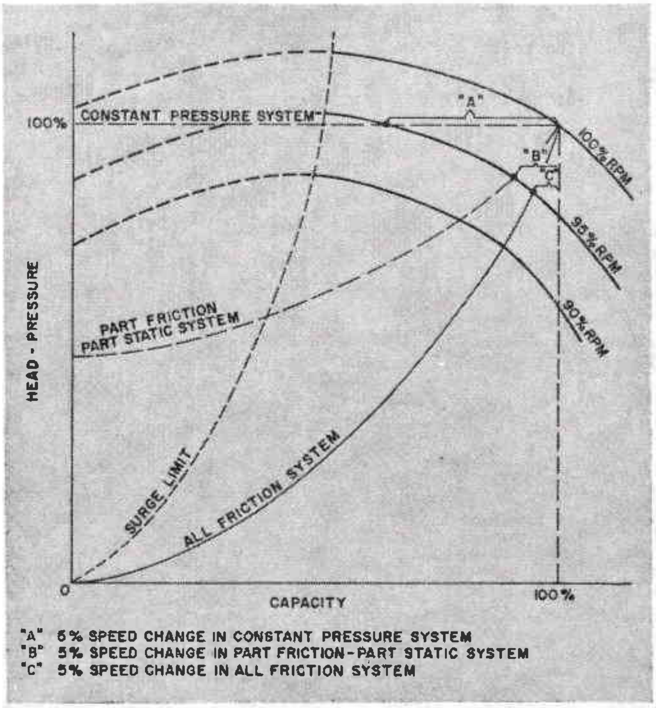

Fig. 1—The effect of compressor speed on system flow. A 5-percent decrease in speed of a compressor in a constant pressure system will effect a much larger reduction in capacity than a 5-percent decrease of compressor speed in a friction circuit.

established Basically the centrifugal compressor is a variable capacity, nearly constant pressure ratio machine. In contrast, the reciprocating compressor is generally a constant capacity, variable pressure ratio machine. The operating point of the compressor is always determined by the intersection of the compressor pressure vs. capacity curves and the system pressure vs. capacity curve.

There are three types of system characteristic curves which are normally encountered. They are:

- **All flow resistance** as in gas pipe lines and approached in recycle circuits,
- **Constant pressure** as in systems employing back-pressure regulator or other flow sensitive throttling device (Condensing systems, such as propane or ammonia refrigeration are usually of the constant pressure type),
- **Combination of the above** which apply to the more general cases, where part of the total pressure drop is constant, and part is due to flow resistance.

The various types of system curves are shown together with a typical compressor performance curve in Figure 1.

Variable Speed Control. Speed variation is a simple and effective method of controlling the capacity of a centrifugal compressor. It has many advantages. Variable speed can provide an infinite "family" of compressor head-capacity curves over the driver speed range. Figure 2 shows that capacity control by speed variation is most efficient as it requires the least part load horsepower compared to other control methods.

From the curves in Figure 1, it is seen that the effect of speed on the capacity of the compressor varies with the type of system resistance. If the system pressure is constant, a small speed change will effect a large capacity change as shown by increment "A". On the other hand, the same speed change in a flow resistance circuit would produce a smaller capacity change as shown by increment "C".

The steam turbine is particularly suitable for variable speed duty. The turbine governor is usually provided with an air head so that a 3-15 psig instrument air signal from the process controller can be applied. The air signal resets the governor to maintain a new set speed. With the turbine under governor control, a change in speed caused by a change in steam conditions will be corrected by the governor when it senses a speed change. NEMA has defined the control characteristics of turbine governors and Table 1 gives the specification for various class governors. For most compressor control applications in process services, a NEMA Class "C" turbine governor is suitable.

Constant Speed Capacity Control. Many compressor installations today are based on constant speed because:

1. Driver speed is constant—such as with electric motors.
2. Limited speed variation—such as may occur with single shaft gas turbines.
3. Multiple compressor services require constant speed.

Oftentimes, several compressor casings in different process services are arranged for drive-through by a single driver. In the case of a gas turbine drive, this is done to match the compressor loads to the available gas turbine horsepower, since there are large horsepower increments between gas turbine ratings. With the multiple service arrangements, a change in speed would affect more than one compressor service, which may be undesirable. Hence, with multiple compressor arrangements, the driver—whether steam or gas turbine—is usually run at constant speed.

To affect capacity control under constant speed conditions several methods are available, namely:

TABLE 1—Speed Governor Standards

NEMA* Class	Adjustable Speed Range (%)	Maximum Steady State Speed Regulation (%)	Maximum Speed Variation Plus or Minus (%)	Maximum Speed Rise (%)	Trip Speed (% Above Rated Speed)
A	10	10	0.75	13	15
	20	10	0.75	13	15
	30	10	0.75	13	15
	50	10	0.75	13	15
	65	10	0.75	13	15
B	10	6	0.50	7	10
	20	6	0.50	7	10
	30	6	0.50	7	10
	50	6	0.50	7	10
	65	6	0.50	7	10
	80	6	0.50	7	10
C	10	4	0.25	7	10
	20	4	0.25	7	10
	30	4	0.25	7	10
	50	4	0.25	7	10
	65	4	0.25	7	10
	80	4	0.25	7	10
D	30	0.50	0.25	7	10
	50	0.50	0.25	7	10
	65	0.50	0.25	7	10
	80	0.50	0.25	7	10
	85	0.50	0.25	7	10
	90	0.50	0.25	7	10

- Discharge throttling
- Suction throttling
- Variable inlet guide vanes

Discharge Throttling. Throttling of the compressor discharge will cause the compressor to follow the pressure-capacity curve as shown on the typical performance curve in Figure 1. This method is not commonly used as it is inefficient and wastes power.

Suction throttling is simple and more efficient than discharge throttling because:

- Less pressure has to be throttled on the suction to accomplish the same result as discharge throttling. If a compressor has a compression ratio of say, 8, a throttling of 1 psi on the suction side is equal to throttling 8 psi on the discharge side. By throttling the suction, the actual CFM to the compressor is being increased even though the up stream CFM or weight flow is reduced. This allows the compressor to operate further out on its head capacity curve—closer to its maximum efficiency point—so that a much smaller amount of head or pressure must be throttled than with discharge throttling. Since less head or pressure is throttled, less power is wasted. The difference in the power requirements between discharge throttling and suction throttling is illustrated in Figure 2.
- The increased CFM to the compressor because of suction throttling also has the effect of lowering the surge point of the compressor system when compared to the upstream flow. Figure 2 illustrates the reduction of the surge point by suction throttling in contrast to discharge throttling. Suction throttling is accomplished by providing a butterfly valve or suction damper in the inlet piping to the compressor. These devices are usually operated by means of an air diaphragm and equipped with a clutch and handwheel for manual control.

Variable inlet guide vanes have a two-fold action:

1. The suction pressure is reduced (as in suction throttling) thereby creating a greater inlet CFM.

2. The flow is directed through adjustable vanes, so that pressure energy is efficiently converted to velocity energy. This velocity energy can, via adjustment of the vanes, be directed to cause prerotation of the gas, with or against compressor rotation. By adjusting the gas rotation relative to the impeller rotation, the first-stage impeller can be effectively loaded or unloaded.

Although variable inlet guide vanes can be put at the inlet of all stages of the compressor, it is more common to have only the first stage provided with this feature. The effect of the variable first stage flow performance has a significant effect on the performance of the subsequent stages so that the over-all compressor performance will be markedly changed. The basic effect of variable inlet guide vanes is to alter the impeller velocity diagrams. This can affect a change in impeller performance with a minimum of throttling, therefore, it is a more efficient method than suction throttling.

By using variable inlet guide vanes, the efficiency will be improved over a wider operating range. However, the peak efficiency may be slightly lowered because of fluid drag even when the inlet guide vanes are wide open.

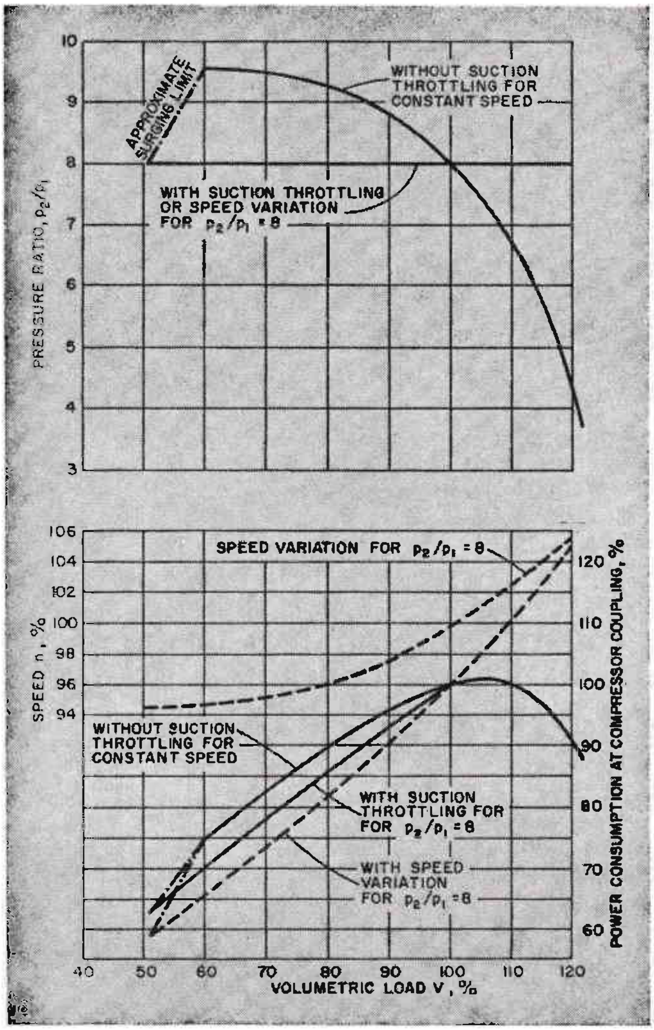

Fig. 2—Capacity control by speed variation.

Should the gas be dirty, there may be a tendency for the vanes to stick if any solids or deposits build up.

The variable inlet guide vanes are relatively expensive compressor features. They are justified only if the compressor is expected to operate at part loads for considerable periods of time and the resulting savings in driver power costs are significant.

Any of the above methods can be used for variation of the compressor capacity, in accordance with the process demands. However, in every case a minimum flow condition will be reached which may produce surging of the compressor.

Surge. It is an inherent characteristic of the centrifugal compressor that its performance becomes unstable at some minimum flow point. This is mainly caused by the drooping head characteristic between the highest pressure point and shut-off as shown by the dotted portion of the head-capacity curve in Figure 1. The surge point depends on such factors as the number of stages, the relative wheel loading, blade angle, method of control, etc. Compressors with few wheels—say three or four—may have this surge point at about 50 percent of design flow while compressors with many stages—say eight to ten—may have their surge point at about 85 percent of design

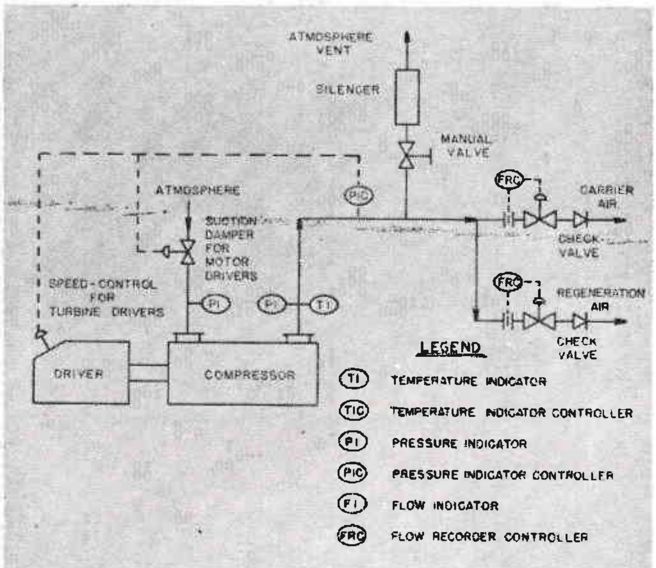

Fig. 3—Air blower for fluid catalytic cracking unit.

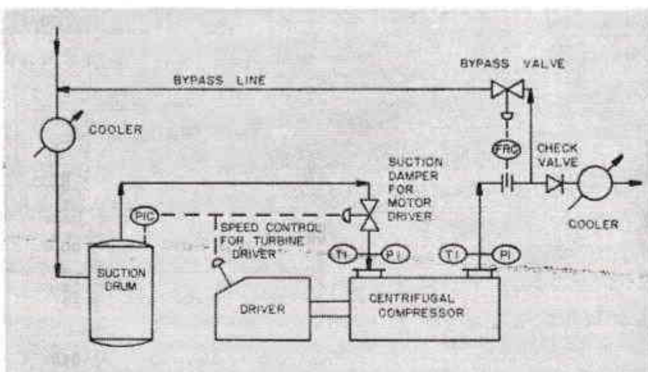

Fig. 4—Single case for gas recovery compressor in FCC unit.

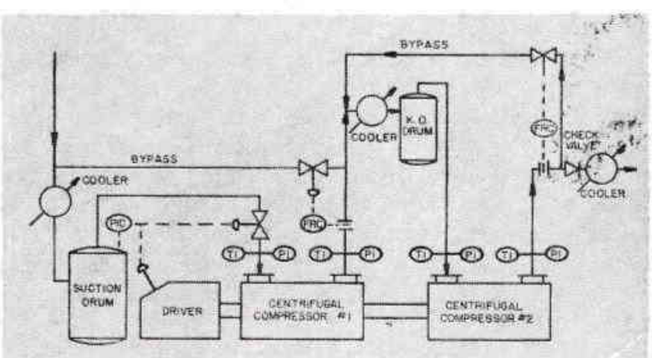

Fig. 5—Two-case gas recovery compressor for FCCU.

flow. The greater the number of stages, the higher the surge point.

Since the turn-down range of centrifugal compressors is from 15 percent to 50 percent, it is essential that the compressor installation be provided with a manual or automatic flow controlled minimum bypass system to prevent the compressor circuit from going into surge.

With a flow controller measuring total compressor flow, the controller can be set at a predetermined minimum—say, the estimated surge volume plus 5 or 10 percent—so that when the process flow falls below the minimum set point, the kickback or recycle valve will open, returning gas to the compressor suction and thereby assuring a continuous minimum flow to the compressor inlet. Since the gas has been heated by the compression it is important that the bypassed gas be cooled to approximately the normal inlet gas temperature by means of a process cooler, a suction cooler, a special bypass cooler or by liquid injection into the bypass stream. If the gas were not cooled, a continuous temperature build-up would occur as the hot discharge gas mixes with the suction gas. Any increase over the normal gas inlet temperature will cause the compression ratio put up by the compressor to fall off.

In the more complex installations, the compressor may have one or more inlet or extraction side streams. If the side stream flow is of such a magnitude that its failure would cause the compressor to surge, then the side streams must be flow controlled to insure that the predetermined minimum flow will always be entering the compressor. Cooling of the recycled gas to the side stream inlets is often necessary.

It is important to have the bypass line take off between the compressor outlet and the discharge check valve. This arrangement will allow the compressor to operate within its own recycle loop even though the compressor discharge pressure is insufficient to lift the check valve.

On compressors with multiple connections, each side stream should be examined for the effect that boil-offs from evaporators might have on a compressor that has been shut down. If a flow of gas would cause reverse rotation of the compressor, the side stream should be provided with a check valve.

TYPICAL APPLICATIONS

Outlined above are some of the basic principles of compressor control; however, each specific application has its own peculiarities. Many of the applications such as air blowers for FCCU service, gas recovery, reformer recycle, and refrigeration cycles that are encountered quite frequently, lend themselves to establishing a basis of minimum instrumentation. Some of the more common arrangements are described below.

Air Blower For FCC Unit. The air blower in a fluid catalytic cracking unit operates against a relatively constant discharge pressure set by the pressure in the catalyst regeneration system. If both the carrier air and regeneration air are supplied by a single air blower, the air requirements for each service must be individually flow controlled and the compressor discharge pressure controlled as shown in Figure 3. If all the air from the blower is used at a single pressure level then the compressor speed or suction damper may be placed on flow control. Since the operating conditions on the air blower are relatively constant, the minimum flow bypass valve is usually manually operated.

Gas Recovery Compressor of FCC Unit. The gas recovery compressor of an FCC unit operates against an essentially constant discharge pressure system. Figure 4 shows the instrumentation for a typical installation. A pressure controller on the suction drum actuates the compressor control system to maintain constant suction pressure. If the flow drops below a predetermined minimum, the flow controller will automatically open the bypass valve insuring sufficient compressor flow to stay in the stable region. It should be noted, in Figure 4, that the hot discharge gas is cooled by means of a heat exchanger on the suction side of the system. If the discharge cooler had been used to cool the gas, some of the heavy ends of the gas would condense out. Hence, a lighter mol weight gas would be returned to the compressor suction. The pressure ratio of the compressor would be reduced if the molecular weight of the inlet gas was lower than the design molecular weight. Figure 5 shows a two-case arrangement driven by a single driver. Each case is provided with its own minimum flow kickback to insure there will be no significant change in molecular weight of the bypassed gas.

Refrigeration Compressor For Alkylation Unit. The refrigeration compressor in an alkylation unit operates against a constant discharge pressure as set by the liquefication temperature of the gas in the condenser. A side stream or economizer connection is oftentimes provided on the compressor to reduce the horsepower requirements. The economizer usually "floats" on the flash drum at some intermediate pressure. Figure 6 shows a typical alkylation compressor installation. The temperature in the reactor is kept constant by maintaining a constant pressure. A pressure controller actuates the compressor control system to maintain a constant pressure to the centrifugal compressor. If the flow falls below a predetermined minimum, the flow controller will cause the bypass valve to open insuring sufficient compressor flow. The discharge gas recycled back to suction may be cooled by the addition of a liquid refrigerant quench. The side stream flow, while of a significant quantity, is not usually so large that its failure would cause the compressor to surge. No special instrumentation is provided for the side stream flow.

Reformer Recycle Compressor. The reformer recycle compressor operates in essentially a flow resistant circuit. That is, the pressure drop through the furnaces, exchangers, piping and reactors is almost all resistance. Referring to Figure 1, we can see that a compressor operating in an all resistance circuit is not likely to surge since the system curve is always to the right of the compressor surge curve. This characteristic makes the recycle compressor circuit inherently stable and no flow controller is normally required for minimum flow protection. The flow is usually manually set either by fixing the speed in the case of a variable speed drive or by manually adjusting the flow control valves in the case of a constant speed drive. During the course of a run as the catalyst activity decreases, periodic adjustments are made to either the speed setting or to the manual flow control valves. Figure 7 illustrates a typical reformer recycle installation. A manually operated bypass valve is provided for use at startup.

Refrigeration System. One of the more complex compressor applications is the low temperature refrigeration circuit. Figure 8 illustrates a typical refrigeration installation. Since the gas is condensed, the compressor operates essentially at a constant discharge pressure. Low level

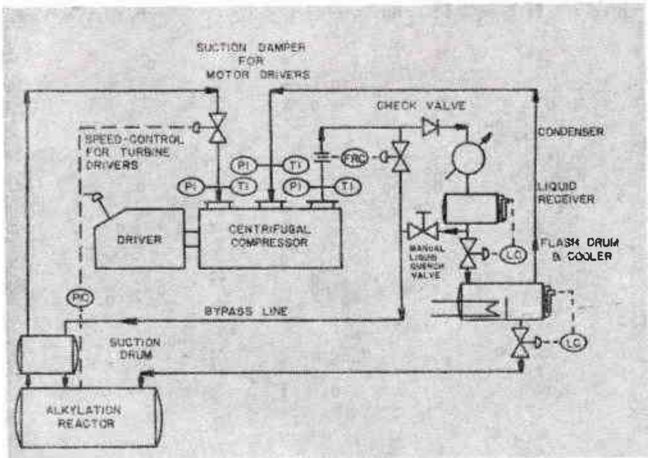

Fig. 6—Alkylation refrigeration compressor.

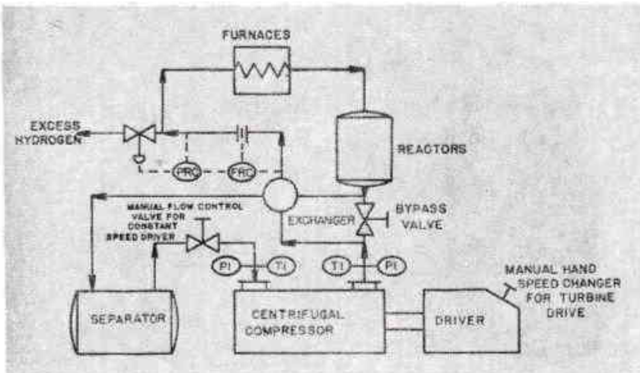

Fig. 7—Reformer recycle gas compressor.

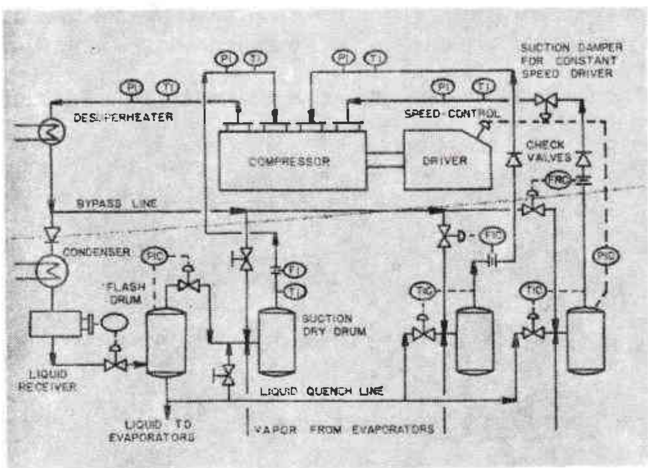

Fig. 8—Refrigeration system.

evaporator temperatures are held constant by maintaining constant suction pressure in the suction drum. A pressure controller on the suction drum actuates the compressor control system. In most cases, the sizes of the side stream flows are of the same order of magnitude as the suction streams. In order to maintain a good flow balance to all compressor impellers, the minimum flow controllers are installed in the suction side streams to the compressor. These flow controllers actuate bypass valves to supply gas flow to whatever stream has fallen

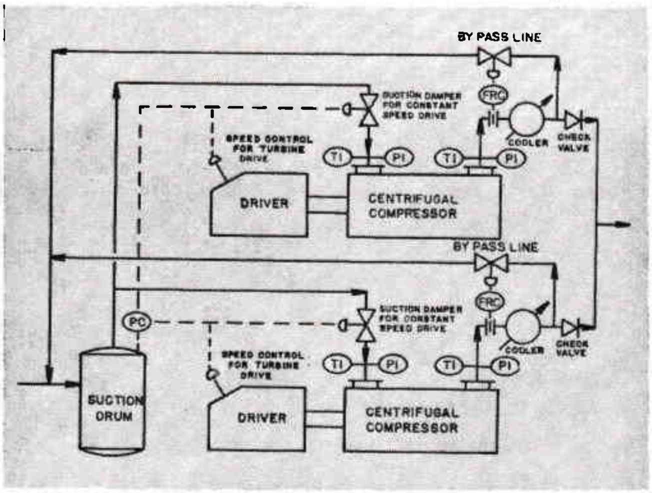

Fig. 9—Parallel operation of centrifugal compressors.

below its predetermined minimum. However, the bypassed gas is at a high temperature level relative to the gas being compressed and unless it is cooled, the warm gas to the compressor suction will adversely reduce its pressure ratio.

Liquid refrigerant quench is used to maintain the gas temperature near the design gas inlet temperatures. At the compressor suction, the liquid quench is usually temperature controlled to maintain a relatively constant gas temperature. At successively higher inlet stages, whether the streams are flow and/or temperature controlled will depend upon the relative size of the side load compared to the flow of the main stream. Sufficient flow and temperature controllers must be put on automatic control to keep the compressor out of surge during normal operation. Additional manual flow and temperature controls may be provided to allow for starting up or other unusual operating conditions.

Parallel Operation. Centrifugal compressors operating in parallel are oftentimes used when:

- The process flow quantities are very large, indicating that more than one compressor is required to fulfill the process operating condition.
- Continuity of plant operation is of the utmost importance.
- Plant expansion requires adding compression facilities.

Oftentimes in revamping units, the compression capacity is increased by the addition of a centrifugal compressor. If the added compressor has a larger or smaller capacity than the original machine, its performance characteristics will obviously be different. Even duplicate machines may have different performance characteristics.

To insure stable operation and to prevent the effects of one machine surging to upset the rest of the system, minimum flow controllers must be provided for each individual machine. Figure 9 shows a typical arrangement. This arrangement provides for individual bypass lines and allows each unit to be started up separately, recycling gas within its own closed loop before being placed in parallel operation with the other compressor. If the flow to either of the compressors drops below its predetermined minimum, the bypass for that particular compressor will open keeping the machine out of surge and not upsetting the other compressors.

The above illustrations only indicate the minimum process instrumentation to insure stable operation. In addition, other instrumentation is normally provided, such as high liquid level alarm and shutdown on suction drums, high discharge gas temperature alarms and shutdown, low lubê and seal oil pressure alarms and shutdowns and others, depending upon the installation. Sufficient instrumentation has been indicated on the schematic illustrations to run a field performance check for comparison with the compressor manufacturer's predicted performance, if a gas analysis is available.

Summary. While the control and instrumentation of each application must be studied individually, there are certain basic elements common to most installations which must be considered:

- The system characteristics must be established. It should be remembered that the operating point of the compressor is always determined by the intersection of the system pressure vs. capacity curve and the compressor pressure vs. capacity curve.
- Sufficient automatic flow control equipment must be provided to keep the compressor out of surge during normal operation. In addition, manual controls may be required for startup, shutdown or abnormal operating conditions.
- Recycled gas must be cooled if it is returned to the compressor inlet. Care must be taken to insure that cooling of the gas does not change its characteristics (i.e., mol weight).
- The bypass line must be taken off at a point between the compressor and the discharge check valve. With this arrangement the compressor can be recycling gas without having to lower the system pressure to a value lower than the compressor discharge pressure.
- Pressure, temperature, and flow indication points should be provided at each compressor inlet or outlet connection so that the data obtained from these instruments together with a gas analysis can be used to check compressor performance.

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Power Calculations for Nonideal Gases

Ideal gas laws may be used within very narrow ranges for any gas. Outside these ranges, check physical properties to find compressor power requirements

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LEST THE TITLE of this article mislead some, all gases are nonideal. Ideal gas relations may be applied within narrow ranges for any gas, with accuracy varying according to the relative pressures and temperatures, and the amount of compression involved. As the liquid phase is approached, ideal-gas relations break down for all substances.

When the departure from ideality is small, the compressibility factor can be considered a function of relative pressure and temperature, and inserted as an additional term in the expression for work derived from ideal gas relations. For greater departures, charts are useful, but often they do not provide the demanded accuracy. More and more, reliance is being placed on the computer to carry out the tediously detailed calculations required for best possible accuracy.

Taking "ideal" gas relations as a starting point, the next step is to determine under what conditions these are adequate, and when more complex relations are needed. The latter may be tables, charts, diagrams, or equations. The application of computers is also discussed.

Power for Ideal Gases. Power requirement of a compressor working on an ideal gas can be calculated by well-known methods. For example, when the unit of power is horsepower, pressure (P) is psia and volume (V) in cfm, the expresion is

$$Pwr = \frac{144\ P_1\ V_1}{33{,}000}\left(\frac{n}{n-1}\right)\left[R^{\left(\frac{n}{n-1}\right)} - 1\right] \qquad (1)$$

where the subscript 1 refers to inlet conditions, R is the pressure ratio (P_2/P_1) and n is the exponent of polytropic compression. If we know n, the actual power can be computed directly. If we know "adiabatic efficiency," we substitute the exponent for isentropic compression (k) for (n), and divide the resulting "ideal power" by efficiency. Final temperature is found from:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Another relationship enables us to find n when polytropic or stage efficiency is known:

$$\frac{n-1}{n} = \eta_p \left(\frac{k-1}{k}\right)$$

When there are several stages of compression, the difference between polytropic and adiabatic efficiency becomes significant. This is because the departure from ideal compression results in an increase in temperature, and therefore in volume, at the outlet of each stage. The succeeding stage must, therefore, do more work than if the stages ahead of it were ideal. When polytropic efficiency is known, adiabatic can be found from:

$$\eta_a = \eta_p/1 + \frac{s-1}{s}\left[\frac{1}{\eta_p}\left(\frac{1-\rho^{\left(\frac{n-1}{n}\right)}}{1-\rho^{\left(\frac{k-1}{k}\right)}}\right) - 1\right]$$

where s is number of stages and p is overall.

Power for Nonideal Gases. These formulas are only approximations. How good these approximations are depends on the gas compressed and the conditions before and after compression. In many cases, the actual power requirement is less than calculated, but in some cases it is more. The ratio of actual to ideal power is roughly equal to the "compressibility factor," defined as:

$$Z = PV/RT$$

For an ideal gas, $Z = 1.0$, while for real gases it is usually slightly lower than 1.0. The variation is shown graphically in Figure 1, based on charts prepared by Nelson and Obert.[5,6] Note the parameters. The abscissa is "reduced pressure" or the ratio of actual pressure to critical, hence dimensionless. Lines are also drawn for each "reduced temperature" which is similarly, the dimensionless ratio of absolute temperature to absolute critical temperature. For a given temperature, the departure of Z from unity is very nearly proportional to relative pressure, for low values of the latter.

The same chart could be drawn for actual pressures and temperatures of a given gas, but it would be useful only for that particular gas. As drawn, it is approximately correct for a wide variety of gases.

At a relative temperature of about 2.5, the value of Z remains about constant at unity. This temperature is known as the Boyle point—the point where Boyle's law

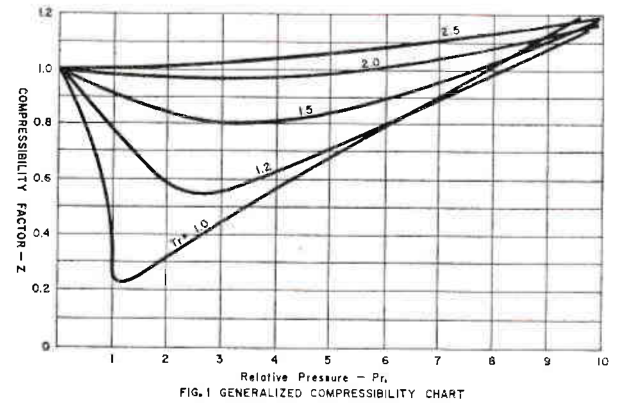

Fig. 1—General compressibility chart shows variation in Z factor.

is exactly true even at fairly high pressures. This explains why nitrogen, at usual ambient temperatures, can be considered an ideal gas, while ethylene with a higher critical pressure (therefore lower relative pressure at any given pressure) shows considerable departure. Nitrogen's critical temperature is 126° K, so that at 300° K (80° F) its relative temperature is 2.38; ethylene, with a critical temperature of 283° K, has a relative temperature of only 1.06 at 300° K.

Above the Boyle point, Z increases above 1.0 as pressure rises. The rate of increase is maximum at a relative pressure around 5.0. Above this, the behavior of the gas again approaches the ideal gas law.

If we were to set limits where ideal gas laws apply within engineering accuracy, we might select an arbitrary permissible departure of 1 percent, and define areas where this departure is not exceeded. Thus, Z lies between 0.99 and 1.01 at all relative pressures below 0.3, where relative temperature is also above 2.0. The range of relative pressures can be extended to about 1.0 for relative temperatures above 2.5; if temperature remains between 2.4 and 3.0, pressure can be increased up to 2.5.

While these may seem rather narrow limits when expressed as relative quantities, critical pressures of most gases are so high that they include many practical cases. An example of this is nitrogen up to 10 atmospheres (150 psia) at temperatures above —5° F, and up to 34 atm (500 psia) at temperatures over 107° F. Since high pressures are usually associated with higher temperatures, these limits include most operations outside the refrigeration and cryogenics fields.

When these limits are exceeded, the first step is to use the Z factor as an additional multiplier in Equation 1. This is reasonably accurate when Z is greater than 0.95, and can be used for rough calculations perhaps down to 0.9. At lower values, a more accurate calculation is available using tables or charts.

Using this method, the values of n and k for ideal gases are used. Sometimes these are also used to find the final temperature. Though this procedure is not correct, frequently the temperature need not be known as accurately as the power requirement. If it is, Edmister[3] provides a method which can be used with good accuracy.

Martin[1] gives the following generalized (i.e., applicable to a large number of gases) expression for Z.

$$Z = 1 + \left(\frac{0.188}{T_r} - \frac{0.468}{T_r^2} - \frac{0.887e^{-5T_r}}{T_r^2}\right) P_r$$

This relation is accurate at values of P_r less than about 0.1. It can be used with small error for higher relative pressures with the limit being higher for higher relative temperatures.

Tables of Properties. The steam tables have been in use for many years by turbine engineers to define the properties of steam. Similar tables have been published for hydrocarbon and refrigerant gases. When tables are used, the procedure is to look up an initial state point as defined by at least two state properties such as pressure and temperature (or moisture content in the case of wet vapors). Other state properties such as enthalpy and entropy are then found.

The next step may be to define an ideal end point or one having the same entropy as the initial condition but a pressure defined by the problem. Again other conditions including enthalpy are found and the change in work content is calculated by subtraction. This method is not well adapted to use of polytropic efficiencies or exponents.

With the difference in enthalpy for an ideal compression, usually given in Btu per pound, the expression for power corresponding to Equation (1) is:

$$Pwr = \frac{V_1(\Delta h)}{42.42\ v_2(\eta_c)}$$

where Δh is enthalpy change, v_1 the specific volume in cubic feet per pound, and η_c the adiabatic efficiency.

Charts and Diagrams. Because use of tables involves tedious interpolation, engineers more commonly refer to diagrams on which the state properties are shown graphically. In the case of steam tables, the Mollier diagram in which enthalpy and entropy are the coordinates, is most often used. For hydrocarbon gases, several charts have been published, among the most recent a series that appeared in this magazine.[2] Generally pressure and enthalpy are the coordinates.

Using these charts the procedures are substantially equivalent to those employed in the case of tables. Since these charts are often made to rather small scales, extreme care is necessary in reading them and too much confidence should not be placed on the precision possible by this method.

Equations of State. Before digital computers were generally available, equations of state were useful mainly for correlating experimental data as a step in the preparation of tables of properties. Equations that describe the properties of real fluids with sufficient accuracy over a wide range of conditions, are too complex for use in ordinary computations.

The computer however makes it possible to use such equations. In fact, they can be put into medium size computers that do not have adequate memory capacity for storage of complete tables and interpolation procedures.

Equations fall into two categories: (1) those that apply to a single fluid to a high degree of accuracy, and (2) those that apply to all fluids but with limited accuracy. The Keenan-Keyes equations for steam and the Benedict-Webb-Rubin equations for hydrocarbon gases fall into the first class. The van der Waals' type of equation, such as the Redlich-Kwong,[6] fall into the second category. An accurate equation of the second type would solve all our problems, as separate equations are not being devised fast enough to keep up with the needs. The common occurrence of mixtures intensifies the problem.

The Benedict-Webb-Rubin equations take the following form:

$$P = RTd + (B_0RT - A_0 - C_0/T^2)d^2 + (bRT - a)d^3 + a\alpha d^6 + \frac{cd^3}{T^2}\left[(1 + \gamma d^2)\, e^{-\gamma d^2}\right]$$

$$S = S_0 - R\ ln\ dRT - (B_0R + 2\ C_0/T^3)d - bRd^2/2 + 2\ cd^2/T^3 \left[\frac{1 - e^{-\gamma d^2}}{\gamma d^2} - \frac{e^{-\gamma d^2}}{2}\right]$$

$$H = H_0 + (B_0RT - 2\ A_0 - 4\ C_0/T^2)\ d + (2\ bRT - 3a)\ d^2/2 + 6\ a\alpha d^5/5 + cd^2/T^2 \left[3\ \frac{1 - e^{-\gamma d^2}}{\gamma d^2} - \frac{e^{-\gamma d^2}}{2} + \gamma d^2 e^{-\gamma d^2}\right]$$

in which P = pressure
T = temperature
d = density
R = gas constant
$A_0, B_0, C_0, a, b, c, \alpha, \gamma$ are constants for a given gas or mixture of gases.

These equations can be rewritten in reduced form:

$$Z = 1 + (B_0 + B_1/T_r + B_3/T_r^3)/V_r + (C_o + C_1/T_r + C_3/T_r^3)/V_r^2 + C_3C_3''/T_r^3/V_r^4 + A_5/T_r/V_r^5$$

$$\frac{S - S_0}{R} = \frac{2C_3'}{C_3''T_r^3}\left(1 - e^{-C_3''/V_r^2}\right) - (B_0 - 2\ B_3/T_r^3)/V_r - \left(\frac{C_0}{2} + \frac{C_3'\, e^{-C_3''/V_r^2}}{T_r^3}\right)/V_r^2$$

$$\frac{H - H_0}{RT} = \frac{3C_3'}{C_3''T_r^3}\left(1 - e^{-C_3''/V_r^2}\right) + \left(B_0 + \frac{2B_1}{T_r} + \frac{4B_3}{T_r^3}\right)/V_r + \left(C_0 + \frac{3C_1}{2T_r} - \frac{C_3'}{2T_r^3}\, e^{-C_3''/V_r^2}\right)/V_r^2 + C_3''C_3'/T_r^3\, e^{-C_3''/V_r^2}/V_r^4 + \frac{6\ A_5}{5T_rV_r^5}$$

where $Z = PV/RT$
T_r is reduced temperature
V_r is $V\ P_c/RT_c$
$C_3 = C_3'\, e^{-C_3''/V_r^2}$
$B_0, B_1, B_3, C_0, C_1, C_3', C_3'', A_5$ are constants for a given gas or mixture.

In this form, the eight coefficients (B_0 thru A_5) have a much narrower range of numerical values [4] than in the original equation, and attempts to correlate them with known variables may be expected to have greater success. For example, correlating with critical temperature, we might use:

$$B_0 = 0.12469$$
$$B_1 = .00028812\ T_c - .40192$$
$$B_3 = -.00031\ T_c - .05697$$
$$C_0 = .000078812\ T_c + .013927$$
$$C_1 = -1.5196 \times 10^{-7}\ T_c^2 - .000063503\ T_c - .0093604$$
$$C_3^1 = .14342 \times 10^{-6}\ T_c^2 - .72169 \times 10^{-5}\ T_c + .034472$$
$$C_3'' = -.00005399\ T_c + .061697$$
$$A_5 = .00008433$$

These values do not yield accuracy equivalent to the values specifically calculated for a particular gas, but they are useful when specific values are not available.

Use of Computers. A computer program, to be of greatest use, should be applicable to all gases and their mixtures, and should be useful for finding all properties of a mixture at any state point defined. Such a program would become a part of larger procedures that would calculate complete processes and cycles.

To economize on computer time, the program should first check to see if ideal-gas relations are applicable and use them if they are. If they are not, it should ascertain the degree of departure and use equations sufficiently accurate for the occasion.

When the gas is definitely not ideal, it may still be desirable to calculate its ideal conditions as a first approximation. One of the difficulties is that accurate expressions for thermodynamic properties are usually available only as functions of one set of parameters. For example, the Benedict-Webb-Rubin equations express pressure, enthalpy and entropy as functions of temperature and specific volume. When the latter are among the unknowns, the solution must be obtained by successive approximation. Care must be used in selecting a method of refining the approximations, to be sure it is convergent. Divergent and oscillating iterations are often encountered.

When the perfect gas relations are used to find a first approximation, the temperature and specific volume are then used as input to the exact relation. For example, if enthalpy and entropy were the given quantities, the corresponding ideal-gas temperature and volume would first be computed. From these, new values of enthalpy and entropy for a real gas would be found, differing somewhat from the given values.

Reversing these differences, new values of enthalpy and entropy would be devised, presumed to be the ideal-gas values corresponding to the real temperature and specific volume. With these, the process would be repeated until the calculated real-gas enthalpy and entropy agreed within reasonable limits with the given values. This method works quite well on computers, which do not mind the tedious repetition necessary to provide an answer.

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Unique Compressor Problems

Case histories on foundation and impeller resonance, thrust bearing loading, and two-phase flow describe centrifugal compressor problems experienced by today's HPI engineer

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UNIQUE CENTRIFUGAL problems and the solutions developed are described in the following case histories. They are typical of those problems being experienced in industry today. However there is always something unique about each centrifugal compressor problem that requires a unique or different solution.

The Problem of Resonance. In one recent installation, a vibration problem was encountered on a centrifugal compressor and a two-motor drive train mounted on an elevated concrete structure. Vibration measurements on the supporting columns revealed high lateral sensitivity to the design running speed of 3,596 rpm. This was confirmed in part during initial test work, when the supporting columns were struck by a heavy timber. Reed tachometer measurements indicated a natural frequency of 3,550 rpm. Also, recordings of vibration levels on the foundation during deceleration consistently showed an increase to occur at approximately 3,500 rpm. A typical plot of a rundown recording is shown in Fig. 1. As a point of further confirmation, measurement taken at one-foot intervals of column height revealed that the point of maximum deflection in the columns occurred at close to the expected 5/9 L point for resonant columns. Since the foundation sensitivity manifested itself in the form of shaft deflection at the coupling between the two motors, a short-term repair was made by installing timbers as shown in Fig. 2. This reduced vibration levels on the motors and compressors by 50 percent or from five mils to two mils. The lateral deflection of the columns was reduced also. During a subsequent downtime, additional

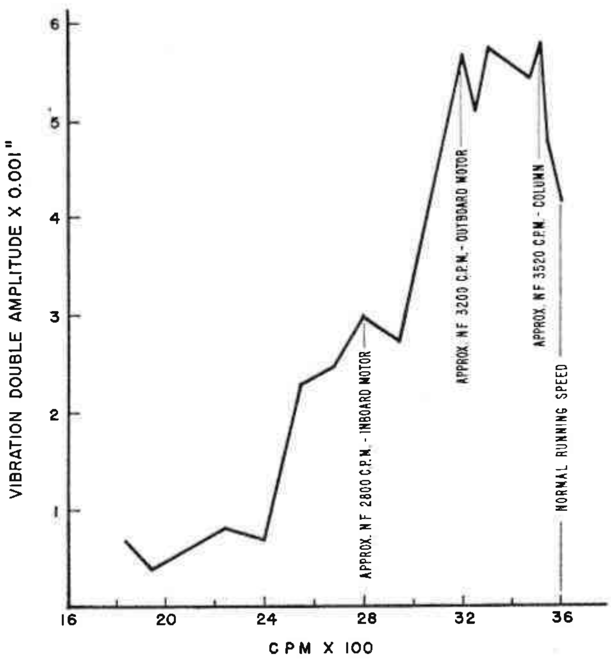

Fig. 1—A typical plot of compressor foundation vibration levels.

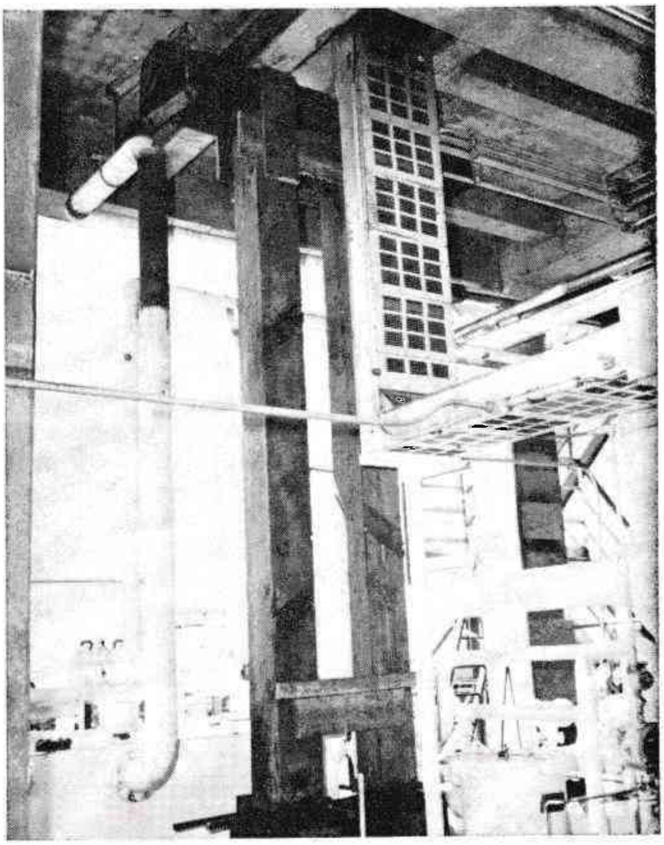

Fig. 2—Temporary repair of foundation using timbers.

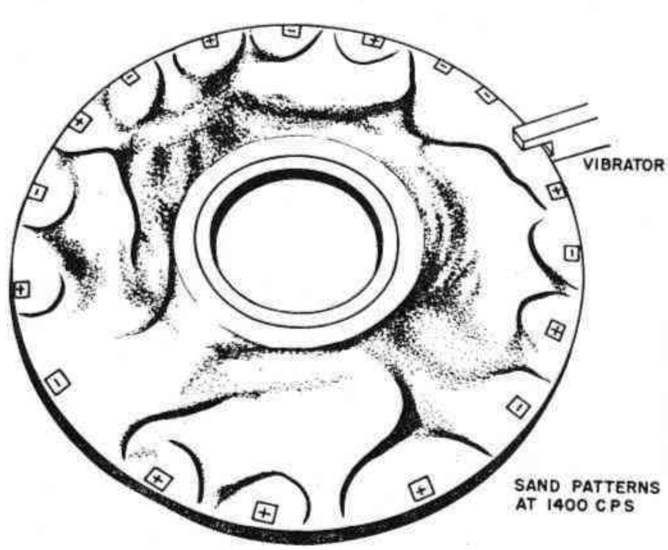

Fig. 3—Sand patterns on first stage impeller.

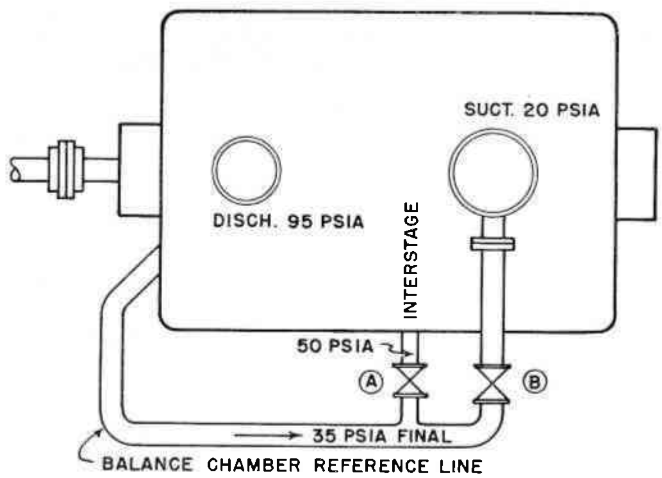

Fig. 4—Balance piston chamber piping arrangement.

mass was added to the foundation in the form of larger cross section in the columns and deeper transverse beams. Vibration levels continued to be two mils or less on all shafts.

Another resonance problem occurred on the same installation. During inspection of the spare rotor prior to its installation, a test was made to determine the natural frequencies of the impellers. This was done to better understand the over-all frequency response range of a prototype impeller. One of the frequencies explored was the product of the number of variable inlet guide vanes and rps since this would be the environment in which the first-stage impeller would run. As shown in Fig. 3, the sand pattern shows a concentration to occur at each vane. The manufacturer agreed that this sensitivity could cause fatigue-type cracks on the disc side of this impeller. The problem was resolved by replacing the variable inlet guide vanes with a butterfly valve, thus eliminating the source of the problem. This was confirmed by an inspection of this impeller after eight months of operation which failed to show any distress in this area.

Thrust Bearing Loadings. One of the continuing problems in the application of centrifugal compressors is proper thrust bearing loading. This force is the result of differential pressure acting on the compressor impeller surfaces and is usually carried by a thrust bearing. Our present specification limits the thrust load on new compressors to 150 psi. Because of more general use of strain gages, the application engineer now has a tool which can be used to determine thrust loads when the equipment is new and to check any deterioration of internals after extended service. Also, the use of shaft position indicators and wear buttons has almost eliminated radical thrust bearing failures when they have been applied. Several case histories of thrust bearing problems follow.

Case 1. A centrifugal compressor in alkylation plant service had a history of frequent thrust bearing failures. After strain gages were installed on the thrust bearing support ring, the normal thrust load was found to be approximately 420 psi. However, during surge, the load would approach 600 psi. The rotor had no restraint in the form of seals that could help absorb the axial loading caused by the mass of the rotor moving during surge. Because of internal configuration, it was not possible to increase the size of the thrust bearing without machining. This would have required an extended downtime. Fortunately, the vendor had referenced the balance piston chamber pressure back to an interstage point rather than to suction. The configuration of this line was changed such that it could be referenced to the original interstage point or back to the main suction (See Fig. 4). After being returned to service, the valves were manipulated such that the original line (Valve A) to interstage was finally closed and the desired balance piston chamber pressure was achieved by partially closing the valve to the main suction (Valve B). After the precalculated pressure had been reached, strain gage readings confirmed a thrust load of approximately 200 psi. No thrust bearing failures have occurred since.

Case 2. In a more recent installation in propane refrigeration service, the compressor was equipped with strain gages in the thrust bearing support ring as well as a proximity-type shaft position indicator. "Zero" readings on the strain gages were taken prior to startup. After startup, the shaft position indicator showed the shaft to be running against the inactive shoes. Also, the strain gages did not show any change in output. A calculation of the expected thrust with the observed pressures on the suction, discharge, and three side loads indicated the thrust load to be approximately 150 psi and in the inactive direction. During a subsequent downtime, a test valve was installed in the balance piston chamber reference line and sealed open (See Fig. 5). After the unit was returned to service and normal flows had pressures established, the test valve was slowly closed. A table with the resulting data follows:

	1	2	3	4	5
Upstream Pressure, psig	36	44.5	50.5	52.5	44.5
Downstream Pressure, psig	35	34.5	34.5	34.5	34.5
Oil Temperature In. °F	122	122	122	120	119
Oil Temperature Out, °F	167	163	156	156	153
Shaft Position	−0.013	−0.0125	−0.008	−0.005	−0.003

At this point, the test was stopped since the valve was

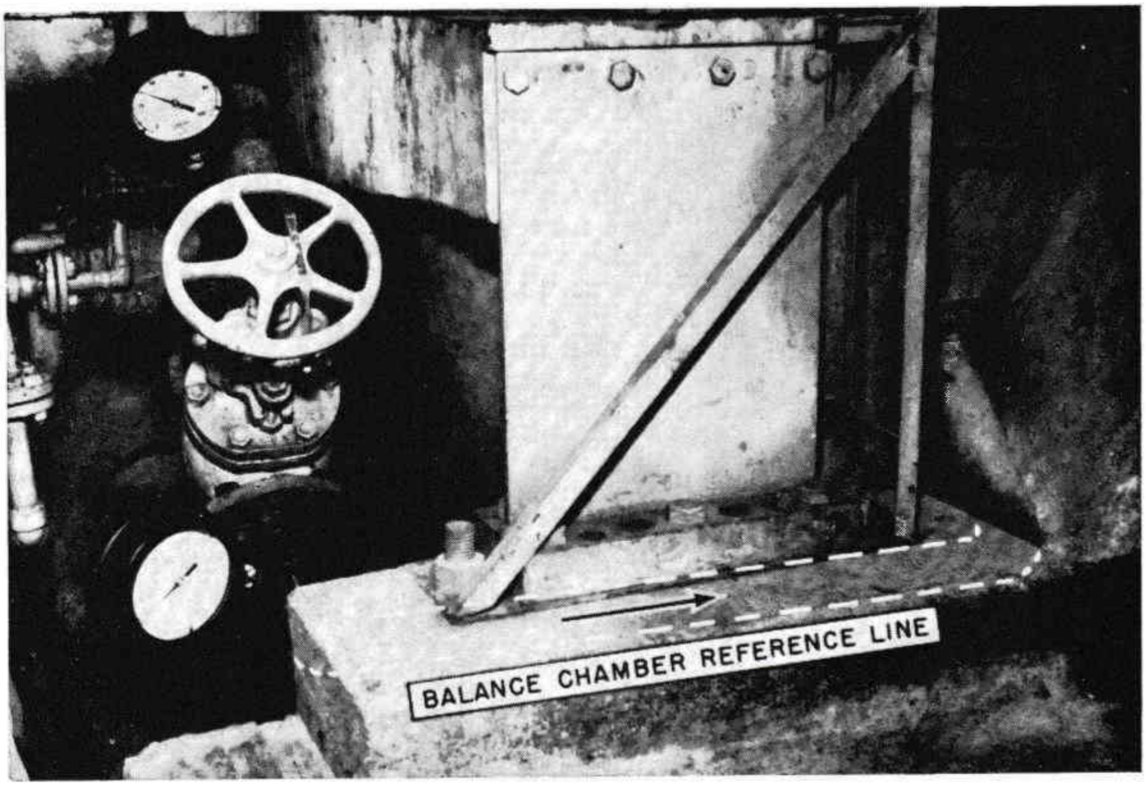

Fig. 5—Test valve installed in balance piston chamber reference line.

almost closed. The data does show that as the valve was being closed, not only did the shaft move toward the active shoe but unloading of the inactive shoe was taking place as indicated by the significant change in the return oil temperature. The vendor is presently considering a redesign of the balance piston.

Two-Phase Flow. Many application engineers have considered the problems relating to two-phase flow in a centrifugal compressor. In some cases, liquid is injected for cleaning or antifouling purpose or to limit the discharge temperature. Tests run with liquid injection showed that substantial rates could be tolerated. In one case, it was interesting to note that the calculated polytropic efficiency changed one percent for each 2° F change in discharge temperature caused by liquid injection. However, in most cases, the presence of liquid is the result of inadequate knockout facilities in suction piping. An illustration of this is given below.

The piping to two centrifugal compressors is shown in Fig. 6. A question of capacity led to a test of both compressors. Although both had identical suction pressures/temperatures and a common discharge pressure, Unit B had a discharge temperature 9° F cooler than Unit A. Further, although the manufacturer had predicted a polytropic efficiency of 72 percent, the test indicated Unit A to be 74 percent efficient while data on Unit B indicated it to have an efficiency of 78 percent. Further examination of the suction facilities revealed that with the increased gas rates being experienced, the gas velocity in the suction knockout drum was now substantially above the required level for separation. Further, it was conjectured that because of this arrangement, the bulk of the liquid present was being centrificated into Unit B. Since the test, an inspection has revealed an accelerated loss of metal on the first-stage impeller caused by the passage of liquid. Plans have been made to increase the size of the knockout facility during the next downtime.

The high-pressure or second-stage portion of this in-

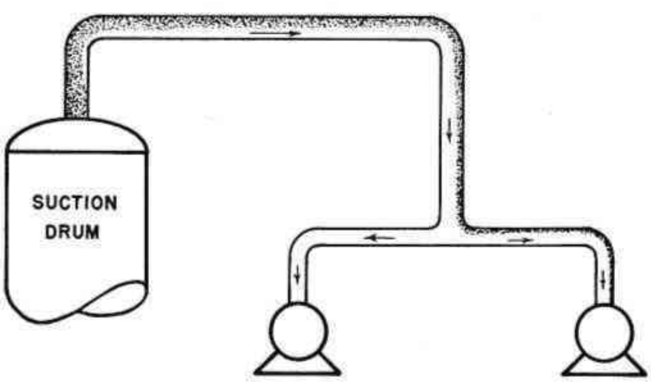

Fig. 6—Piping arrangement from suction drum to two centrifugal compressors.

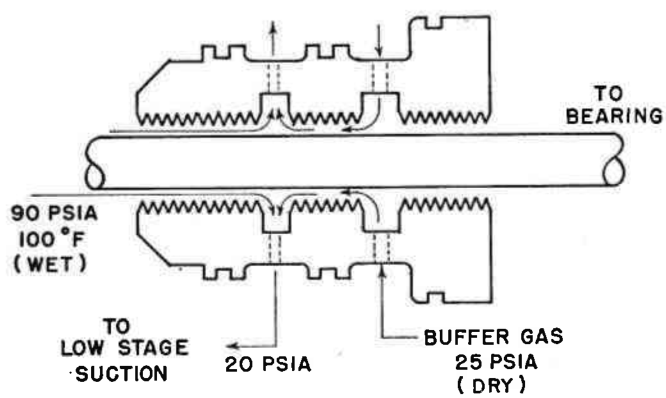

Fig. 7—Seal system for second stage of centrifugal compressors.

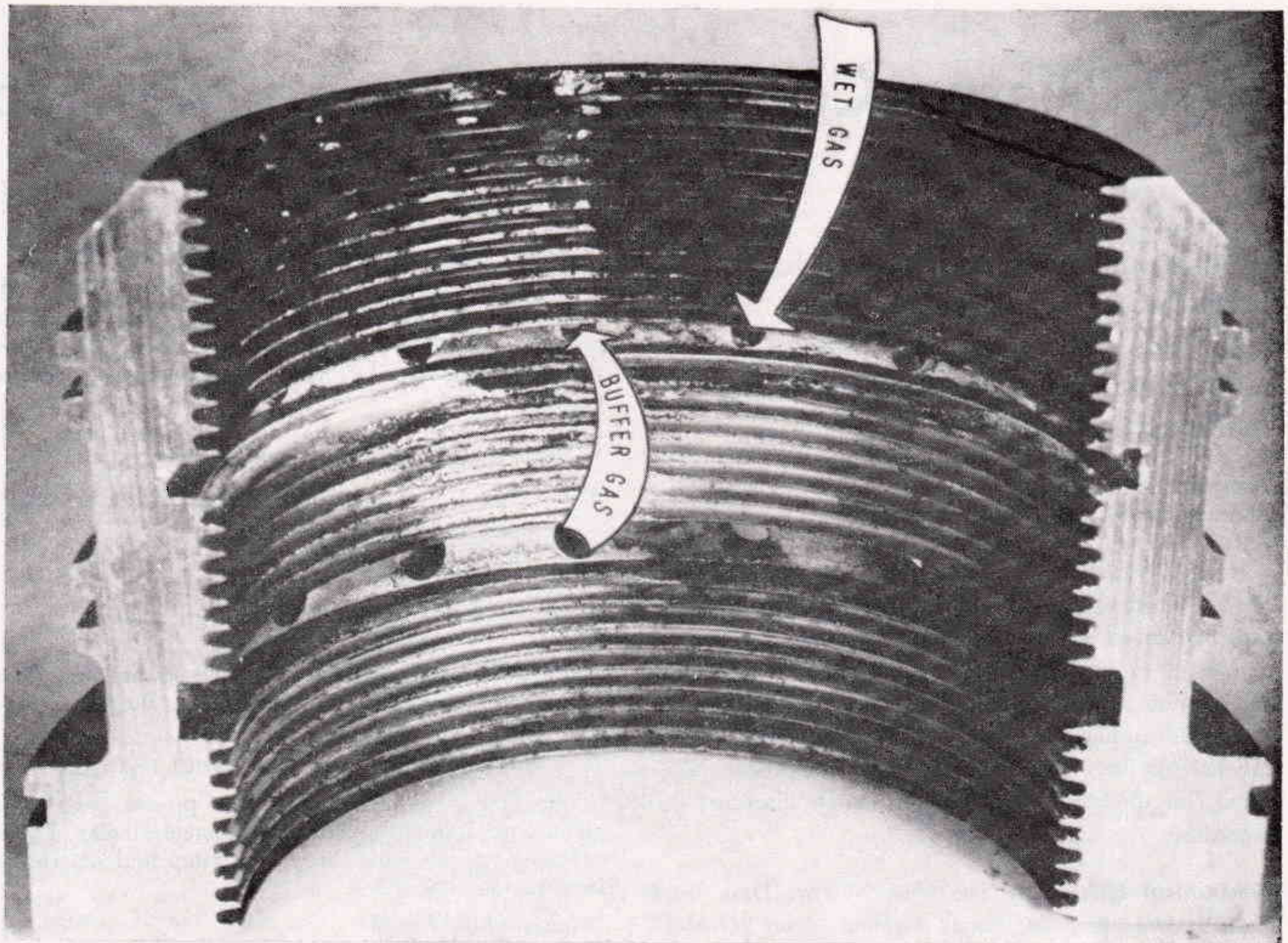

Fig. 8—The suction seals were badly cut.

stallation (two additional compressor casings) had a different type of problem that was attributed to two-phase flow. The sealing system used is shown in Fig. 7. The high-stage suction pressure is 90 psia. The sour/wet gas pressure is broken down to approximately 20 psia and bled to the low-stage suction (the other casing in the train). Sweet or buffer gas is injected to prevent the sour/wet gas from contaminating the lube system. After some 18 months of operation, a lowering of the lube oil viscosity in the system was observed. An analysis of the system revealed that little or no buffer gas was being injected because of an increase in sour gas reference pressure. After a proper differential had been established, the viscosity problem was improved considerably. Shortly after, both compressors were inspected. It was found that the discharge end seals were in excellent condition (gas conditions at discharge of 195 psia and 210° F). However, the suction end seals on both compressors were badly cut as shown in Fig. 8. The condition of the labyrinth was attributed to erosion caused by liquid present in the sour/wet gas. Further, a flash calculation of the gas being compressed confirmed that the two-phase condition existed at not only the suction (six percent liquid) but also at interstage in the labyrinth (two percent liquid). It was concluded that the seals had eroded to a point where some bypassing of the seal gas was occurring and the liquid was entering the bearing chamber. Also, the buffer gas temperature (80° F) was not high enough to reflash the liquid whose boiling point was calculated to be 90° F. Since that time, a buffer gas heater has been installed in an attempt to keep the combined interstage temperature above the boiling point of the liquid in the sour gas.

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Indexing Terms: Bearings-9, Compressors-9, Foundations-9, Gages-10, Loading-7, Maintenance-8, Meters-10, Operating-8, Strain Gages-10, Surging-6, Thrusts-6, Vanes-6, Vibrations-7.

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New Piston Compressor Rating Method

Confidential guarantee curves have been the prime basis for rating reciprocating compressors. This unique method lets you make a sound mathematical approach

Lyman F. Scheel,
Ehrhart and Associates, Inc., Los Angeles

HERE IS A NEW method of rating piston gas compressors. It is based on aerodynamics and thermodynamics rather than on custom and empiricism. The system is unique in presenting a mathematical evaluation of the *compression* and the *mechanical efficiency*. It has been successfully applied over the past eight years in producing rational solutions to the most difficult compressor problems. The application to a typical example illustrates the procedure.

Mechanical Efficiency. The 1966 NGPSA Data Book contains five charts for visual selection of an *"Over-all Efficiency"* which includes a 93 percent mechanical efficiency as well as the *compression efficiency*. Earlier editions contained similar charts which included a 95 percent mechanical efficiency. This 2 percent difference in efficiency maneuvers a subtle \$10,000 price increase for a 3,500-hp integral gas engine compressor. The previous 95 percent mechanical efficiency was adequate for a 400-hp cylinder and perhaps overgenerous for greater powered cylinders. A frictional allowance equal to the square root of the cylinder hp has been found to be quite reliable. For example, a 1,000-hp cylinder would have a 97 percent mechanical efficiency, $1-(1000)^{0.5}/1000 = 0.9683$. A 100-hp cylinder would have a 90 percent mechanical efficiency.

When the API Standard 618 committee introduced the NO NEGATIVE capacity tolerance in the guarantee clause 55, the price increased another \$15,000 for the same engine. Likewise, another 5 percent power penalty or \$25,000 is added for nonlube features. There is serious doubt that 5 percent frictional heat from a 1,000-hp cylinder can be transmitted through a set of Teflon or carbon rings without being consumed. A sliding coefficient of 0.03 could account for about one percent, nonlube frictional loss.

HOW TO ENHANCE COMPRESSOR RATING

The simplest method to introduce credibility into guaranteed hp is to evaluate the valve losses. This requires the following data to be included in the compressor specifications.

- The net adiabatic hp at stated operating conditions
- The sum of the peripheral flowing edges and the permissible lift of the valve element
- The average piston speed
- The brake hp required from the prime mover.

Adiabatic Horsepower. The simple power pump equation is the easiest method of determining the hp for a gas compressor:

$$bhp = ppm\ (\text{adiabatic head})/33{,}000\ (\eta_d)\ \eta_m \qquad (1)$$

The flow is usually expressed in process parlance as pounds per minute (ppm). The adiabatic head (L_{ad}) is equivalent to the head lift for a pump and determined from Equations 2 and 3.

$$L_{ad} = X_\sigma\ (BR_c^{\sigma} - 1) \qquad (2)$$

$$X_\sigma = 1{,}545\ T_1 - uT_1\ Z_a/m\sigma \qquad (3)$$

$$\eta_d = (R_c^{\sigma} - 1)/(BR_c^{\sigma} - 1) \qquad (4)$$

$$\eta_m = (bhp - bhp^{0.5})/bhp \qquad (5)$$

$$\sigma = (k-1)/k \qquad (6)$$

$$B = (1+\theta_d)/1 - \theta_s) \qquad (7)$$

$$R_c = P_d/P_s \qquad (8)$$

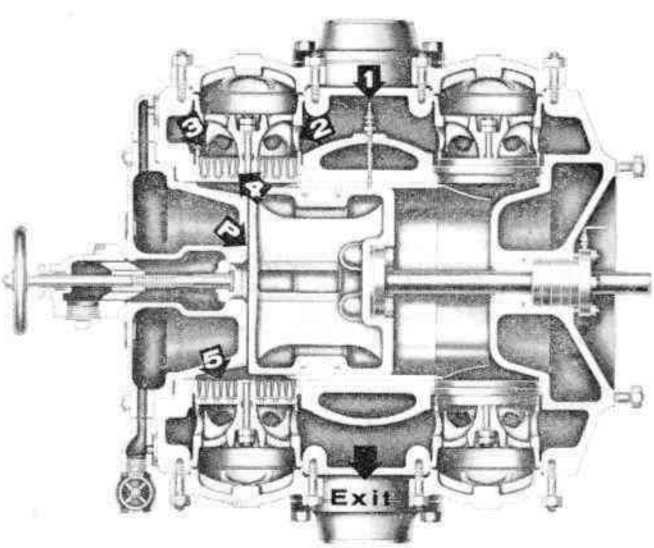

Fig. 1—The five velocity changes involved in charging a compressor cylinder. (Photo courtesy of Cooper-Bessemer Co.)

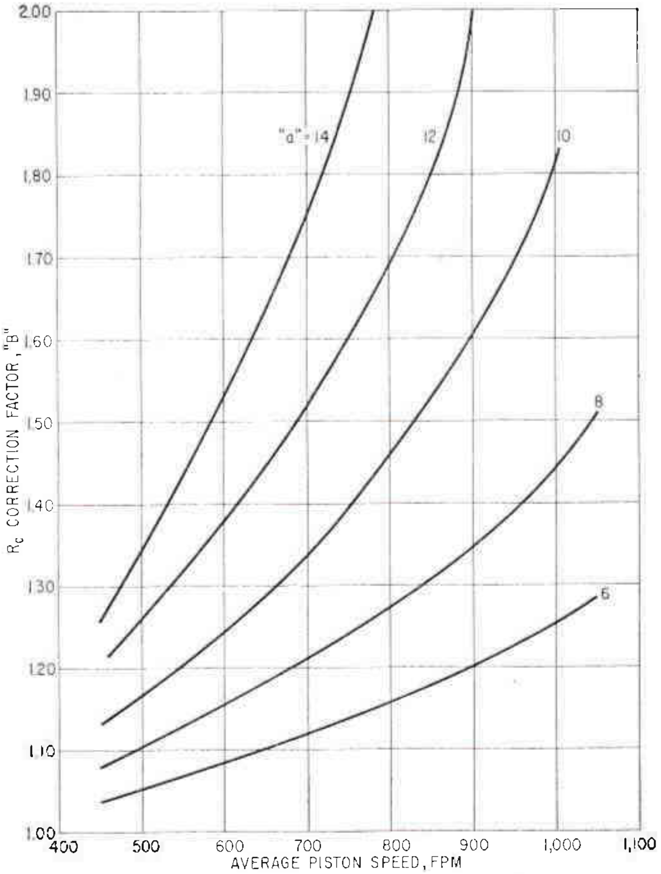

Fig. 2—Correction B factor to convert line R_c to intrinsic R_c for butane, m = 58, t_1 = 60° F, R_c = 3 (a is the piston/valve area ratio).

The various abbreviations are explained in the glossary, except the intrinsic correction factor "*B*". It extends the normal R_c to represent the actual effective R_c within the cylinder. The symbol Θ_d represents the mean psi required to exhaust the cylinder displacement into the header. The symbol Θ_s represents the suction valve loss experienced in filling the cylinder. Both are expressed as a decimal fraction of the respective system pressure. The derivation of an equation to evaluate Θ_s is given below:

Aerodynamic Valve Analysis. The vector flow path through a typical cylinder is shown in Fig. 1. The arrows depict abrupt changes in velocity that are experienced in charging the cylinder.

- V_1 is the line velocity of approximately three times the average piston velocity (U) and the velocity coefficient f_1 is assumed to be 0.3 velads. (Velocity Head) = $V^2/64.4$)
- V_2 is the cylinder channel flow to each valve port at an approximate velocity of $4U$ and the f_2 value is taken as 0.4 velads.
- V_3 is the valve guard velocity of $5U$ and the frictional resistance factor is evaluated as 0.6 velads.
- V_4 is the lifted valve element average velocity of $12U$ and the resistance f_4 value is 4.0 velads.
- V_5 is the valve seat velocity of $5U$ and the resistance f_5 value is 0.7 velads.
- Inside of the cylinder where the gas follows the piston action, the velocity is U and the f value is 1.0 velad.

Substituting the equivalent velads (velocity-heads) for the respective piston velocities, the following equation is evolved:

$$\Theta_s = [(0.3 \times 9)\,U^2 + (0.4 \times 16)\,U^2\,(0.6 \times 25)\,U^2 + (4 \times 144)\,U^2 + (0.7 \times 25)\,U^2 - U^2]/288V_{s1}(g)$$

$$\Theta_s = 616\,U^2/288V_{s1}\,(g)$$

Substituting $10.73\ T_1/m\,P_1$ for V_{s1} and dividing the sum of the suction valve resistances by P_1, so as to express the pressure drop in terms of a decimal percentage of the suction pressure we have: $\Theta_s = 616\,m\,U^2/10^5\,T$. The largest and controlling resistance is the valve element loss of 576 U^2 velads, referred to the average piston speed. It can be equated to fa^2 where f is the valve resistance factor of four and a piston/valve area ratio a factor of 12. The remaining 40 U^2 only represents 6 percent of the total resistance. Any deviation or correction in this evaluation is unlikely to effect the general premise. The equation can be modified to carry all secondary losses as equivalent to 40 U^2. It has been shown that the piston speed during the central 60 percent of the piston travel, is 1.5 times the average full stroke speed.[7] Since most of the valve flow occurs during the central travel, the arithmetical average speed U can be increased by the square of 1.5 or 2.25. The above equation can be regrouped and the controlling valve resistance identified as fa^2, we have:

$$\Theta_s = 2.25\,(40 + f_a^2)\,U^2\,m/10^5\,(T_1) \qquad (9)$$

When the resistances fa^2 are less than a 100, it is advisable to consider the 40 U^2 as 6 percent of the normal loss of a typical cylinder and apply that effect to the constant, 2.25/0.94 = 2.4. The equation then reads:

$$\Theta_s = 2.4\,(f_a^2)\,U^2\,m/10^5\,(T_1) \qquad (10)$$

The only condition that is changed in the above equation to make it applicable to the discharge valve, is the temperature. This is corrected by:

$$\Theta_d = \Theta_s/R_c^{\sigma} \qquad (11)$$

Valve Lifts. The common lift for disk valves operating in 1,000 psig service is 0.080 inch. This is reduced to 0.050 for pressures in excess of 2,000 psig. When high pressure gas has a molecular weight less than 10, a lift of 0.030 inch may minimize the valve maintenance. Lifts of 0.100 inch are common for 100 psig and lower pressures. Nylon poppet type valves with 0.250 inch lift have rendered excellent service at speeds of 600 rpm in 1000 psig service.

Valve Areas. Modern compressor cylinders are usually provided with a piston/valve a ratio of 8 to 12. Early 1930 model cylinders equipped with strip valves had a ratios as high as 20. An a ratio less than 5 generally requires a high-lift, poppet type valve. A set of reference charts, Figs. 2 through 5 are included, whereby B factors can be readily extracted. All charts are based on 3 R_c. The decimal fraction of a similar series of B factors at 1.5 R_c are only 8 percent greater than the 3 R_c decimal fractions. The decimal portion of the B factor can be extrapolated as described in Equation 12. The suffix c represents the operation being changed.

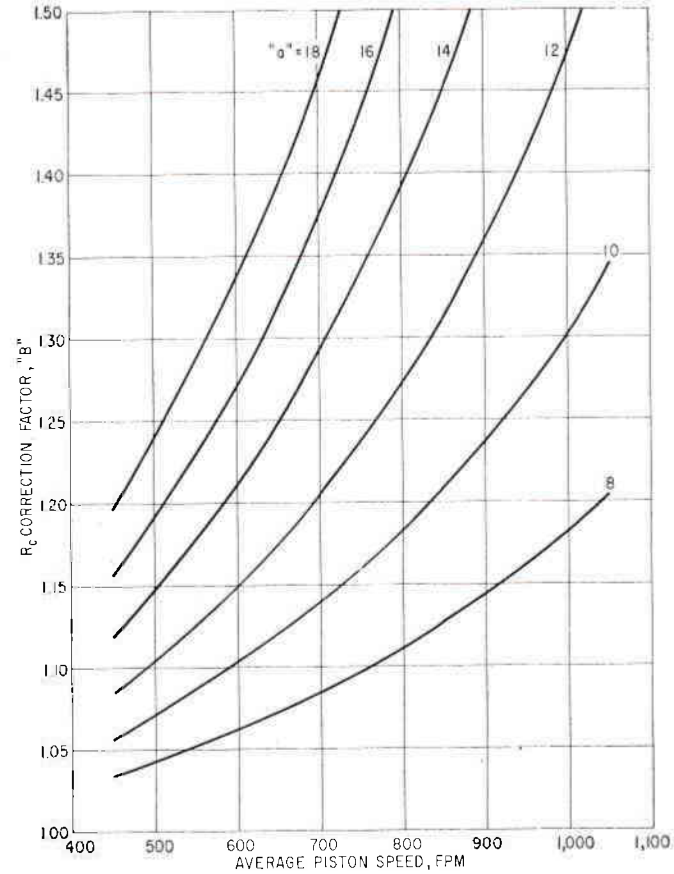

Fig. 3—Correction factor B to convert line R_c to intrinsic R_c for air, m = 29, R_c = 3, t_1 = 60° F (a is the piston/valve area ratio).

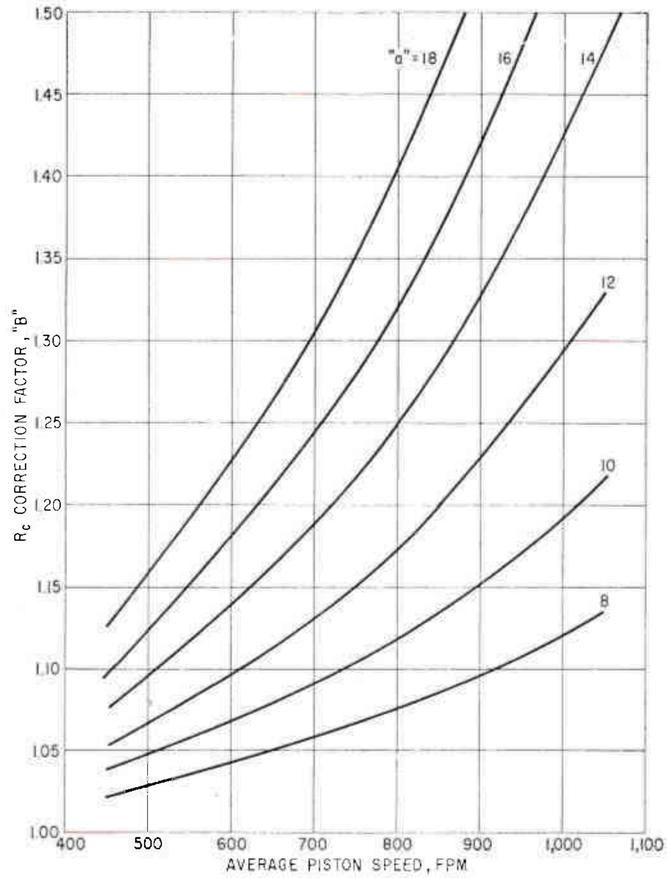

Fig. 4—Correction B factors to convert line R_c to intrinsic R_c, for natural gas, m = 19, R_c = 3, t_1 = 60° F (a is piston/valve area ratio).

$$B_c = (B - 1)\ (520/T_c)\ (m_c/m)\ (U_c^2/U^2) + 1. \quad (12)$$

Compression Efficiency. The term *compression efficiency* has never had a precise definition. The manufacturers confidential guarantee curves have been the prime documents used to rate gas compressors. The issuance of the two adiabatic hp charts in the 1966 NGPSA Data Book has set a precedent in acknowledging the adiabatic power as the basic value. The *compression efficiency* therefore becomes the complement of the valve losses. Equation 4 becomes a mathematical expression for *compression efficiency*. It is defined as the ratio of $(R_c^\sigma - 1)$ to $(BR_c^\sigma - 1)$, where R_c is the compression ratio at the cylinder flanges and B represents the algebraic sum of the valve resistances. The specific heat ratio k values as determined for the proximate mean temperature are entirely adequate for compressor performance ratings. The accuracy of the other factors emphasized in this article is of greater consequence than the k value refinements.

$$k = C_{pm}/(C_{pm} - 1.986) \quad (13)$$

Sample Problem. Given: An 18-inch bore, double acting cylinder with 20-inch stroke, has a piston/valve a ratio of 12 and operates at 240 rpm. The cylinder clearance is 11.5 percent. The gas handled has a molecular weight of 17.7 and a k value of 1.275. The suction is 113.5 psia, Z_s is 0.998 at 100° F and compressed to 283.5 psia where Z_d is 0.989.

Determine the hp requirement:

$\sigma = (1.275 - 1.0)/1.275 = 0.216$

$U = 2\ (20)\ 240/12\ (60) = 13.3$ fps; $U^2 = 178$

$R_c = 283.5/113.5 = 2.50$; $R_c^\sigma = 1.219$

$\Theta_s = 2.4\ (4)\ (144)\ (17.7)\ 178/(560 \times 10^5) = 0.078$

$\Theta_d = 0.078/1.219 = 0.064$

$B = (1 + 0.064)/(1 - 0.078) = 1.064/0.922 = 1.154$

$BR_c = 1.154\ (2.5) = 2.885$; $BR_c^\sigma = 1.258$

Compression Efficiency, $\eta_d = 0.219/0.258 = 85$ percent

Cylinder displacement is 4.25 d^2, when U is 13.3 fps or 800 fpm. (14)

Displacement is: 18 (18) 4.25 = 1377 cfm. (Lit. Cited 7, p. 70)

$E_v = 100 + C - ACR_c^{1/n}$ (Lit. Cited 7, p. 74-77) (15)

$E_v = 111.5 - 1.1\ (11.5)\ 2.09 = 85$ percent

Capacity is 1,377 (0.85) = 1,171 acfm.

$v_s = 10.73\ (560)/(17.7)\ 113.5 = 2.99$ cf/lb.

Capacity is: 1,175/2.99 = 392 ppm or 392/17.7 = 22.1 molal ppm 392 × 380 (cf/lb mole) 1,440/17.7 × 10^6 = 12.12 MMscfd

Adiabatic gas constant: $X_\sigma = 1{,}545\ (560)\ 0.983/17.7\ (0.216) = 225{,}000$ (Equation 3)

Intrinsic adiabatic head is:

$L_{ad} = X_\sigma\ (BR_c^\sigma - 1) = 225{,}000\ (0.258) = 58{,}000$ ft-lb/lb. (Equation 2)

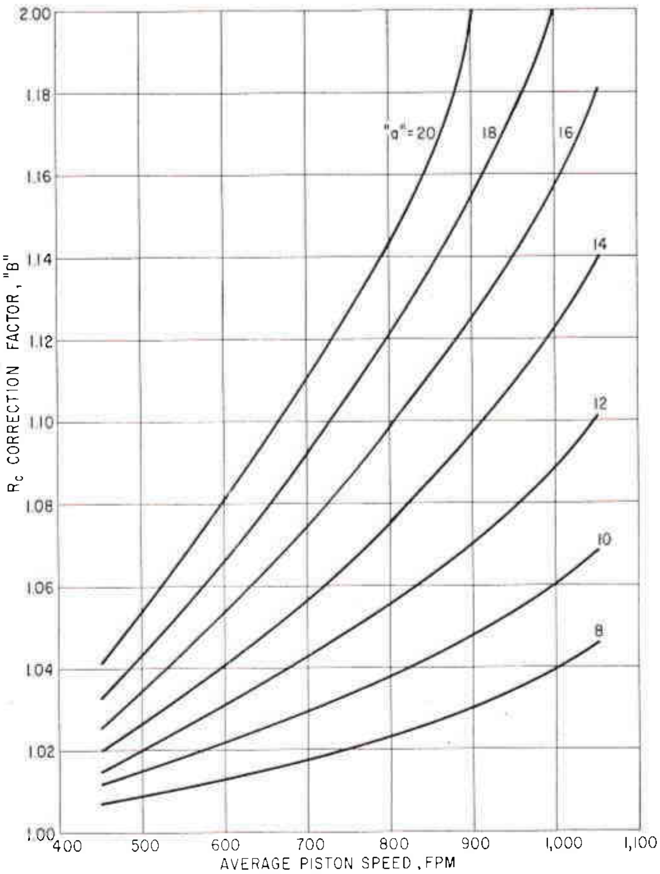

Fig. 5—Correction B factors to convert line R_c to intrinsic R_c for a hydrogen mixture, $m = 6.5$, $t_1 = 60°$ F, $R_c = 3$ (a is the piston/valve area ratio).

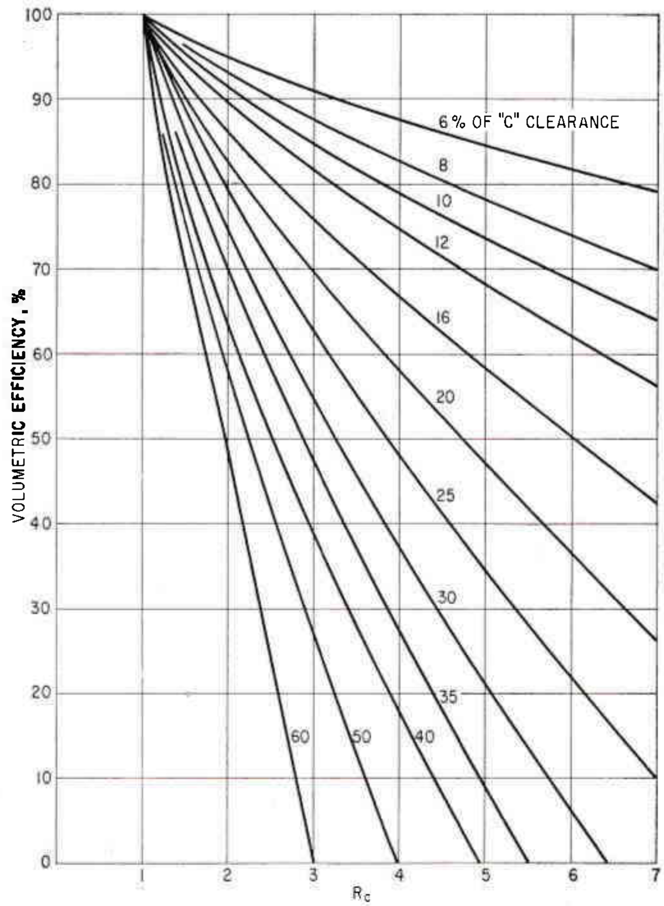

Fig. 6—Volumetric efficiency for air and diatomic gases, $k = 1.40$, $n = 1.35$, $\Lambda = 1.10$, $\eta = 1.1$, $E_v = 100 + C - C\Lambda R_c^{1/n}$.

Intrinsic gas hp = (58,000) 392/33,000 = 690 hp

Frictional hp = $690^{0.5}$ = 26; bhp = 716

Mechanical efficiency η_m = 96.3 percent

Note: The gas data used in this problem is the same as Sample Calculation No. 1 NGPSA 1966 Data Book.

The 18 x 20 cylinder size is five times larger than the 9 x 13 used in the sample calculation.

The power required for the equivalent smaller cylinder is 143.2 bhp.

The sample required 156.1 bhp or 9 percent more power. It includes a 72 percent *over-all* efficiency, less 93 percent mechanical or 77.5 percent compression efficiency. This efficiency would reflect a piston/valve a ratio of 16.4 in lieu of the given value of 12 or 330 rpm in lieu of the given 240 rpm.

Equations 14 and 15 are taken from Lit. Cited 7. Λ is 1.10, which includes a 10 percent valve and piston ring leakage loss effecting only the trapped clearance gas. The heat rejection from the clearance gas of a typical cylinder has a polytropic factor of 1.10 and reduces the k value of 1.275 to an n value of 1.240 and 85.0 E_v. The extreme range of this polytropic factor is 1.25 which reduces n to 1.23 and E_v to 84.8 percent. The 1.25 factor is suitable for a slow speed, 100-hp cylinder having an abundance of cold water. The cylinder clearance is not usually precise at the purchasing stage. The three volumetric efficiency Figs. 6, 7 and 8, provide quick and accurate E_v selections for preliminary sizing. The foregoing example may be simplified by using Fig. 4 and the NGPSA adiabatic hp chart. The B factor for 800 fpm and 12 a ratio is 1.174. The actual mole weight and temperature corrections reduce B to: (1.174 − 1.0) (17.7/19.0) (520/560) + 1 = 1.151. This compares favorably with the calculated value of 1.154. The intrinsic compression BR_c is 2.88. The hp/MMcfd factor for 2.5 R_c and 2.88 R_c are 44.2 and 52.0. The compression efficiency is: 44.2/52.0 = 85 percent.

These hp/MMcfd factors multiplied by 0.557 give the hp/molal ppm. The intrinsic gas hp for compressing 22.1 molal ppm at 100° F is: 52(0.557) 22.1 (560/520) = 690 hp which checks the more elaborate calculation.

Enthalpy charts can be applied with equal ease. Presume an isobutane refrigeration system takes suction at 24 psi and 40° F. The discharge is 84 psia. The enthalpy is raised from 115 to 137 or 22 Btu per pound. The line R_c is 3.5. The piston speed is 600 fpm and piston/valve a ratio is 10. Fig. 2 shows B to be 1.243 at 60° F. The intrinsic BR_c is 4.32.

The compression efficiency is:

$$(3.5^{\sigma} - 1)/(4.32^{\sigma} - 1) = 0.107/0.126 = 85 \text{ percent,}$$
$$\text{where } k = 1.09 \text{ and } \sigma = 0.0825$$

If exponential calculations are troublesome, the same efficiency can be taken from the NGPSA adiabatic hp values. The power factor is 57.5 for $3.5R_c$ and 68.0 for $4.32R_c$. The compression efficiency is: 57.5/68 = 85 percent. The power required to compress 1000 ppm is: 1000

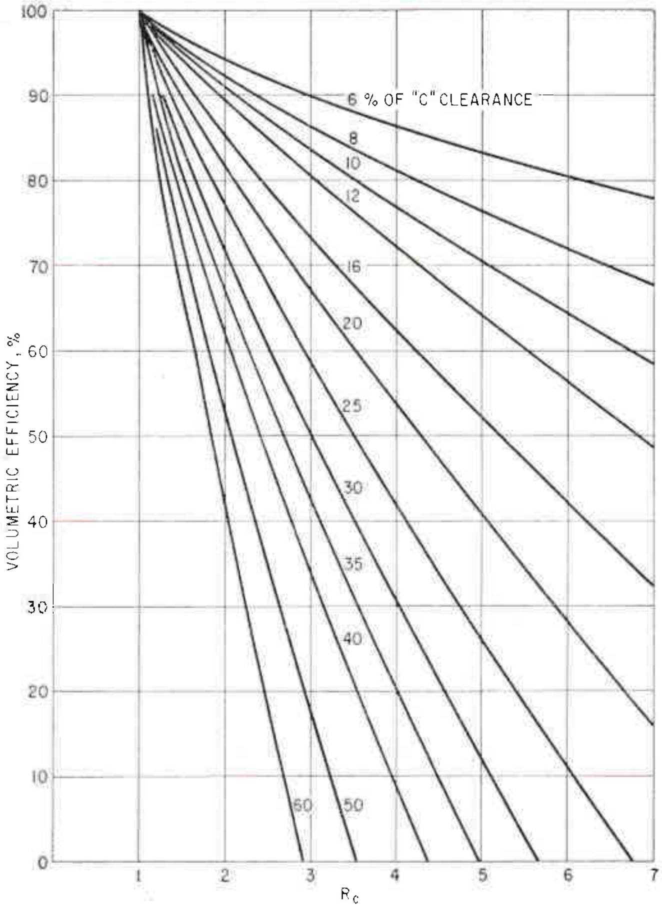

Fig. 7—Volumetric efficiency vs. R_c for dry natural gas, $k = 1.28$, $n = 1.25$, $\eta = 1.1$, $E_v = 100 + C - C\Lambda R_c^{1/n}$.

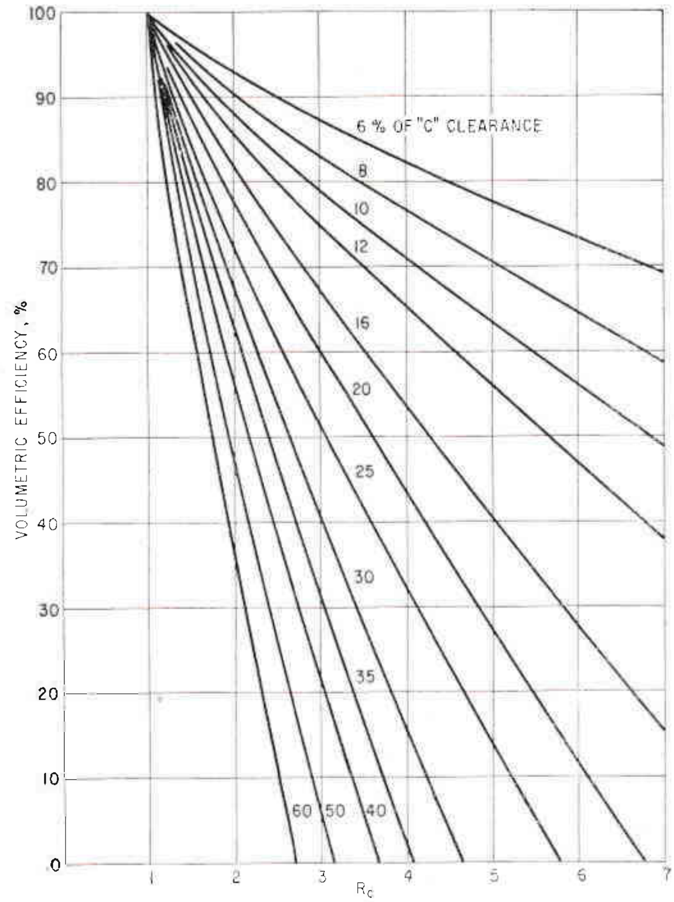

Fig. 8—Volumetric efficiency vs. R_c for LPG and heavy hydrocarbon gases, Sp. Gr. = 1.55, $k = 1.13$, $n = 1.116$, $\eta = 1.1$, $\Lambda = 1.1$.

$(22)/42.5(0.85) = 608$ gas hp. The frictional hp is: $(608)^{0.5} = 25$ and the bhp is 633.

NOMENCLATURE

a	Piston/valve area ratio
$acfs$	Actual cubic feet per second, flow
B	Intrinsic R_c factor
C_{pm}	Mean molal heat capacity, Btu per molal pound per °F
d	Diameter of compressor cylinder, inches
f	Resistance coefficient, dimensionless velads
E_v	Volumetric efficiency
k	Ratio of specific heats at mean temperature
m	Molecular weight of gas
ppm	Pounds per minute
P	Pressure, psia
R_c	Ratio of compression, P_2/P_1
R	Universal gas constant, $1545/m$
T	Absolute temperature, ° Rankine
U	Average piston speed, fps
V_s	Specific volume, cf/lb.
V	Velocity, feet per second
X	Unity gas constant as related to the adiabatic head
Z	Gas compressibility factor
η	Efficiency factor
θ	Valve loss, decimal fraction of system pressure
Λ	Clearance-gas leakage factor usually 1.10
σ	Adiabatic exponent, $(k - 1)/k$
Σ	Horsepower per million cubic feet (14.4 and 60°F) per day

The suffix 1 and s denotes the suction P, T and Z condition. The suffix 2 and d denotes the discharge P, T and Z condition. Other special suffix are described as applied.

About the author

LYMAN F. SCHEEL *is a senior mechanical engineer with Ehrhart and Associates, Inc., Los Angeles. He is the author of the well known book, Gas & Air Compression Machinery. Previously Mr. Scheel was with The Ralph M. Parsons Co. and acted as a consultant to C F Braun & Co., the Fluor Corp., Solar, and other major concerns. He also worked for a number of years as division foreman for Union Oil of California. He specializes in the evaluation, bid analysis and review, and selection plus application of pumps, compressors and turbines for refineries and chemical plants. He is a member of the Western Gas Producers and Oil Refiners Association and ASME.*

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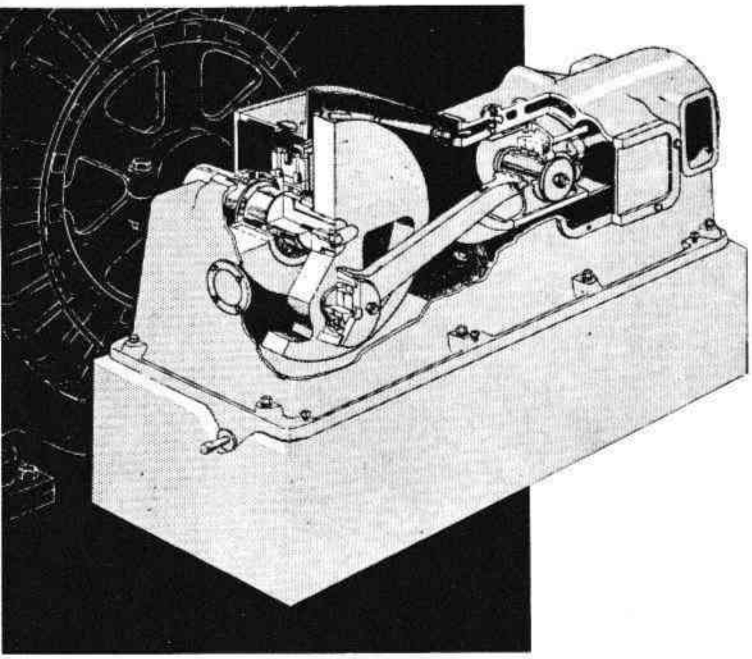
Indexing Terms: Capacity-6, Charts-10, Compressibility-6, Compressors-9, Constants-6, Costs-7, Density-6, Displacement-6, Efficiency-6, Enthalpy-6, Flow-6, Horsepower-10, Nonlubricated-6, Physical Properties-6, Properties/Characteristics/-6, Rating-7, 8, Ratios-6, Reciprocating-9, Resistance-6, Selection-8, Sizing-8, Speed-6.

NOTES



Mechanical Design—It's easier to select and operate a compressor if you know its design & characteristics.

T. O. Kuivinen
The Cooper-Bessemer Corporation
Mount Vernon, Ohio

THE RECIPROCATING compressor has been well established for a wide range of applications. Speeds may range from 125 to 514 revolutions per minute. Piston speeds range from 500 to 950 feet per minute, the majority being 700 to 850 feet per minute. The nominal gas velocity through valving is usually in the range of 4500 to 8000 feet per minute. Discharge pressures may range from vacuum pumps to 30,000 pounds per square inch.

The reciprocating compressor fundamentally is a fixed capacity machine with variable capacity obtained by altering the speed of the prime mover. Capacity may be varied on multiple cylinder machines by making head or outer ends and crank or inner ends inoperative by utilizing devices that lift suction valves open manually or automatically. Additional clearance volume may be provided, which can be cut in or cut out, to vary the compressor capacity as required.

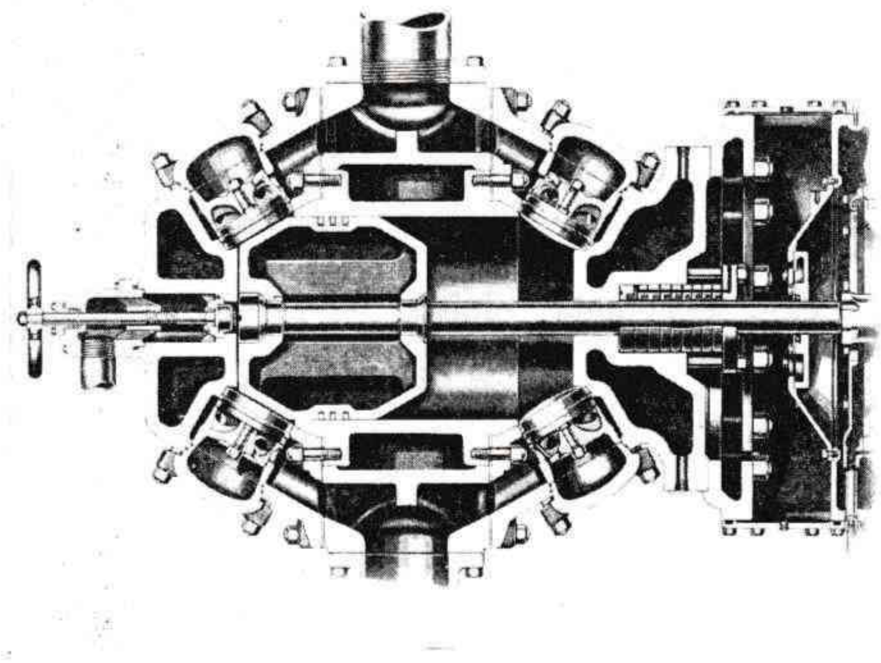


FIGURE 1—Compressor cylinder for medium pressure service. Multiple valves are used to keep clearance volume at a minimum.

For some applications, it is often desirable to include a large clearance volume to maintain approximately constant horsepower over a wide range of pressure conditions. Thus, economical operation is attained by operating the prime mover at its rated load for a broad range of operating pressures.

During design, all parts are given thorough stress analysis to determine that operating stresses are within the endurance strengths of the materials used. Adequate factors of safety are incorporated to cover expected variations in manufacture, materials, and customer usage of the compressor.

Many of these designs are fatigue tested by submitting the actual cylinder assembly to pulsing pressure supplied by a hydraulic pump. Tests were carried out in increments involving at least 6 million cycles of pressure pulses at each pressure until some part failed. From these tests the maximum allowable working pressure for a suitable factor of safety is determined.

Construction—Larger diameter lower pressure cylinders are usually constructed as shown in Figure 1. Pistons are of medium strength specification cast iron or of aluminum alloy. Either type are light construction with ribbing to maintain minimum reciprocating weight. Cylinder heads, cylinders, and cylinder liners are made of specification cast iron. Cylinder heads are fitted with multiple valves to keep clearance volume at a minimum. Piston rods are usually of medium carbon steel. The area on which the packing runs is hardened for long wear life (usually by induction heating and water quenching).

Piston rings are usually cast iron but may be special bronze, multiple piece cast iron, or multiple piece thermosetting plastic. Selection of this material is based on the gas and the type of the compressor service. Piston rod packing is usually multi-stage metallic using materials similar to the piston rings. Practically all compressor cylinder manufacturers supply piston rod packing engineered and manufactured by specialists in that field.

Design—Suction and discharge valves cover a greater area of individuality in design than other parts of a compressor cylinder. In general multiple disc plate valves or strip valves seating on multiported cast iron seats are used. The valve plates open against very light springs in a valve guard. Narrow seating areas and light springs minimize pressure differentials required to open the valves, to keep compression work loss to an absolute minimum. Valves are held in place by cages and caps that in turn are restrained by a circle of studs or bolts.

The outer head of a compressor cylinder is usually equipped with an additional clearance volume chamber that becomes effective by manually opening a built-in valve. In this way clearance space of the outer compression space is altered to reduce the capacity. The volume is usually determined while engineering the compressor cylinder application. Compressor cylinders are sometimes fitted with a fixed clearance pocket cast right in the head. Sometimes a selective clearance volume bottle is attached to the head. Sometimes a variable volume, controlled by manual or hydraulic positioning of an auxiliary piston within the clearance chamber, is provided.

For services of around 1000 psi, a cylinder design as shown in Figure 2 is often used. The higher pressures naturally require heavier walls in the body and head castings. Valves are arranged at right angles to the main bore.

Cylinders as shown in Figure 2 are designed with greater built-in clearance. Added clearance is obtained by using the unique double deck valve construction. When pressure ratios are low, cylinders of this type (equipped with double deck valves

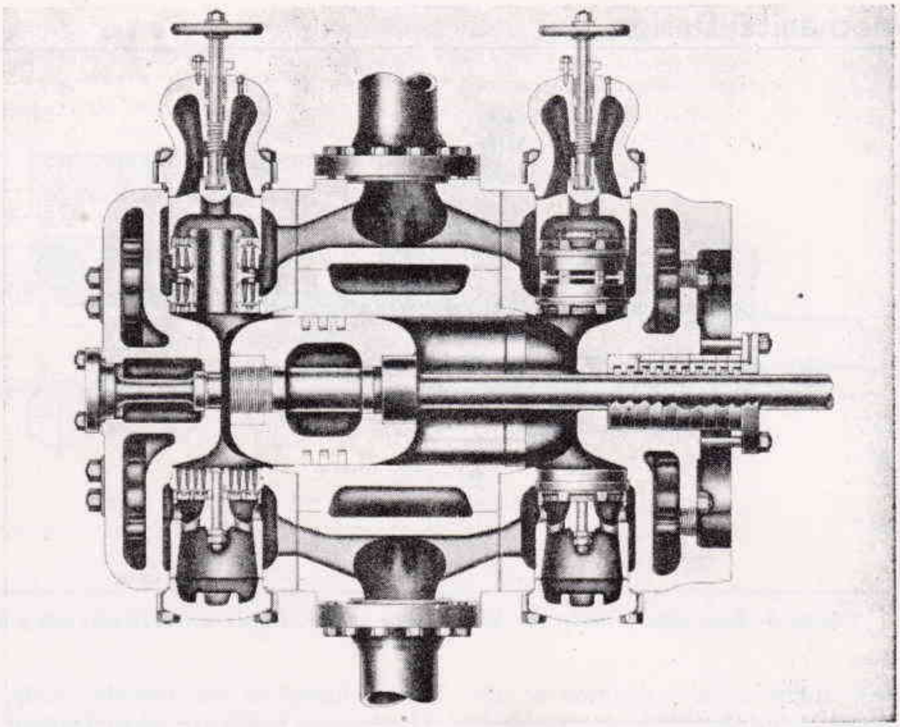


FIGURE 2—Compressor cylinder for 1000 to 1500 psi service. Heavy walls are required and valves are at right angles to the bore.

and large clearance volumes) are self unloading for a wide range of pressure variations. Thus the compressor driver remains at or near full load through a larger range of pressures and capacities with good economy in fuel consumption or motor efficiency.

For higher pressure service, the designer soon runs out of space for all components when he attempts the design with even the best grades of specification cast irons. A few years ago a new material, nodular iron, was developed that fills this need. Nodular iron is produced in the foundry similarly to high strength iron. Patented processing produces an iron having the free graphite uniformly distributed in spherical nodules. The nodular iron material is stronger since the notch effect caused by graphite flakes has been eliminated. Furthermore, nodular iron material is ductile.

Ductile iron's tensile strength compares with mild steel castings. Its ductility, though less than steel, runs from three to ten per cent elongation depending on whether the metal is

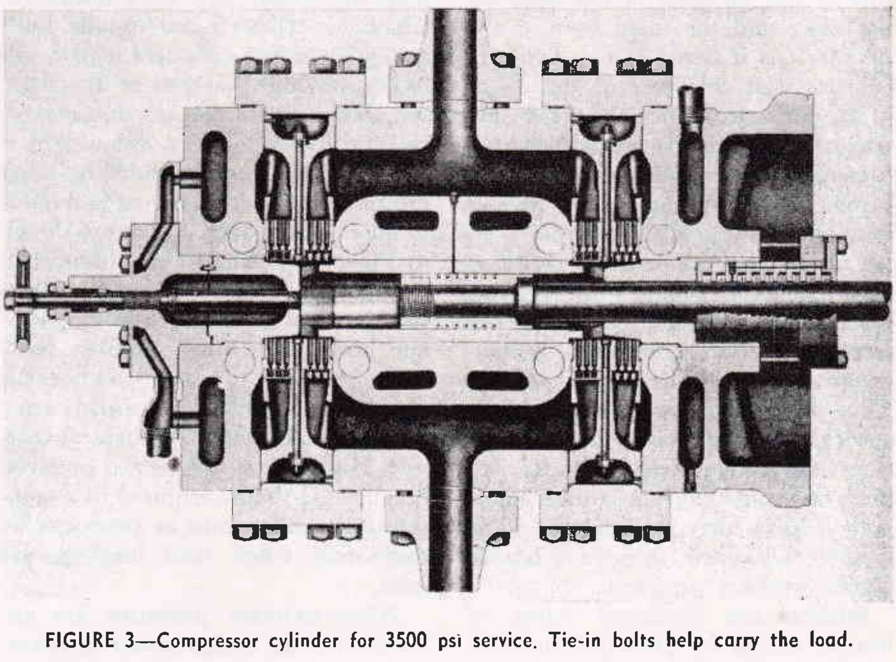


FIGURE 3—Compressor cylinder for 3500 psi service. Tie-in bolts help carry the load.

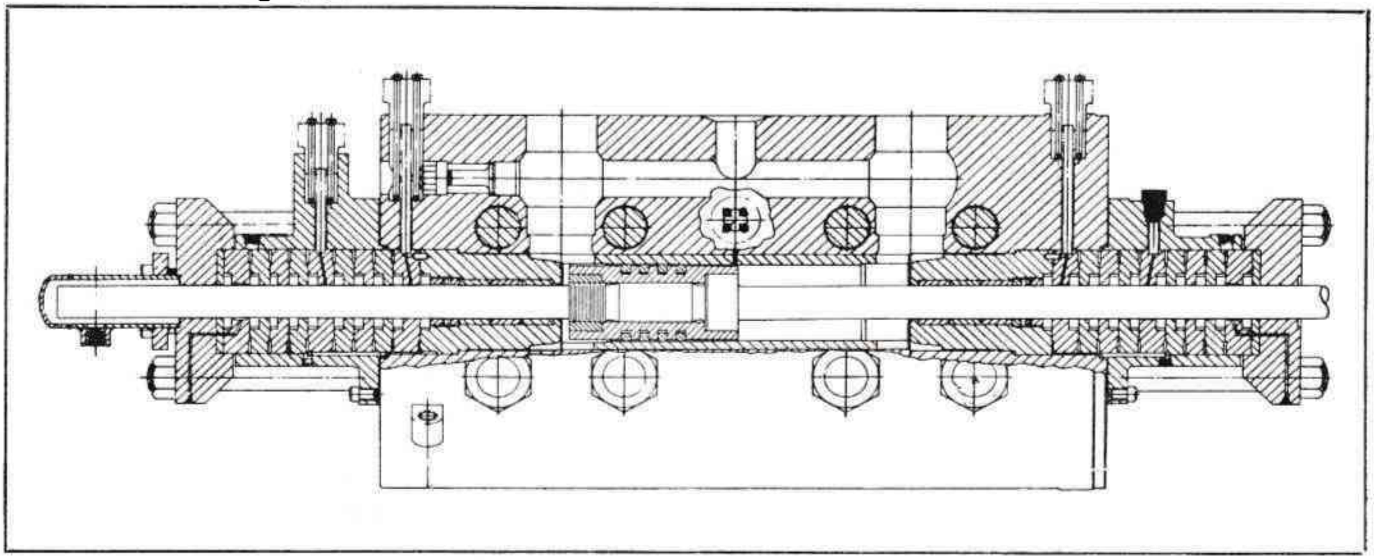


FIGURE 4—Recirculator cylinder for high pressure service. Double acting cylinder with a tail rod of the same diameter as the piston rod.

used in the as-cast condition or annealed by subsequent heat treatment. Many designs exactly like Figure 2 have been giving excellent service up to 1500 psi using nodular iron castings. The versatility of this new high strength ductile foundry product should continue to promote usage at even higher pressures as acceptance increases.

Steel castings are used quite universally in higher pressure cylinders. Since steel castings are more difficult to produce than iron castings compressor cylinder designs are limited to those employing fewer valves.

As the designer attempts to produce designs to withstand higher pressures he discoveres the maximum stress occurs at the junction of the bore for the valve and the main bore of the cylinder. As it becomes impossible to increase wall thickness at this point in an effort to reduce the stress, the design shown in Figure 3 has been produced. The cylinder body is a steel casting fitted with a shrunk in cast iron liner. Eight alloy steel tie bolts are installed at the junction points of valve pocket bores and cylinder bore. Controlled preset of these tie bolts at assembly places the material in compression stress at the junction point. Thus, the internal pressure, when in service, starts to stress the cylinder from a negative level so that maximum resultant stress is reduced. The bolts help to carry the load providing greater endurance strength to handle higher working pressures.

Suction and discharge valves, of heavier design for pressures involved, are mounted in the cylinder body. The cylinder heads are inserted within the body and firmly seated on metal ring gaskets by the cylinder head studs. Flat head gaskets are not suitable for the higher pressures. The usual clearance volume unloaders are provided when required.

Many of the high pressure chemical processes require recirculation in the system. This often means handling relatively small volumes at high pressures and extremely low pressure ratios. Single acting cylinders result in heavy bearing loads on the driving frame. Double acting cylinders also give high unbalance between the outer end and the inner end because the piston is hardly larger in diameter than the piston rod. As a result, recirculator cylinders are usually built as double acting cylinders with a tail rod of the same diameter as the piston rod. Extra crossheads are unnecessary because the piston and rod weight is low. The rod is easily guided by using bronze bushings in the inner and outer cylinder heads. Such a design is shown in Figure 4. Usually the pressures in this design are in the range where steel forgings are required in the cylinder body and in the heads. Steel forgings can be produced in materials of greater tensile and endurance strength than obtainable in steel castings. Higher cost, however, is involved because the shapes required in a compressor cylinder must be produced by machining when steel forgings are used.

When extreme pressures are encountered, the design choice of necessity becomes the single-acting plunger-type of cylinder. Special materials are involved to get plunger surfaces hard enough to endure packing pressure. Nitrided steels are most commonly used. Cylinders are forged steel with iron liners with extremely careful attention paid to design details in order to minimize stress concentrations. Special valve designs, usually operating in line with the cylinder axis, are required to endure the dense gas being handled. All parts are more heavily constructed, and many parts are made of very high strength steels.

Figure 5 shows a design for 15,000 psi service. The plunger attaches to a spherical seated aligning device driven from the driving gear crosshead. Without such an aligning means, it would be difficult to maintain the operation of the metallic packing that seals the plunger.

Many of the gases being compressed in various processes react with some of the materials ordinarily used in compressors. The engineering of such applications begins with consultation and determination of corrosive and other effects. Valves may be made of stainless steels, piston rods may be stainless steel coated or nitrided. Liners may be of special alloys like Ni-Resist iron. Piston rings and piston rod packings might be ordinary cast iron, bronze, thermosetting phenolic plastic, or babbit-faced iron. Sometimes it is necessary to plate gas passages with a suitable coating, but this is quite rare. The combination metallurgist-designer team is essential to solution of such special applications.

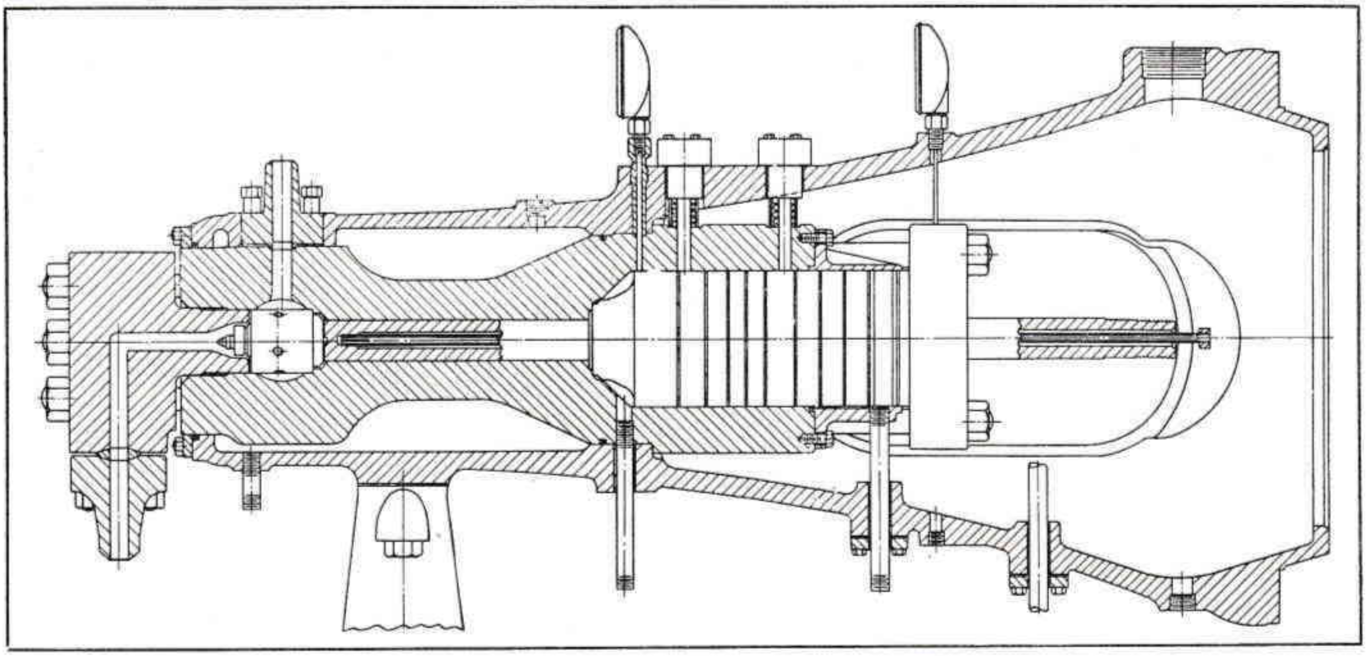


FIGURE 5—Single-acting cylinder for 15,000 psi service. Plunger attaches to a spherical seated aligning device.

Certain processes require the delivery of compressed gas or air that is absolutely devoid of any lubricant. If such is the case, it is often more expedient to construct the compressor for non-lubricated operation rather than attempt to deliver absolutely dry gas by extraction of lubricant carried over in the gas after compression.

The usual solution to this problem is to construct the piston of multiple disc spacers of dense fine grain graphite. Sectional piston rings, backed up by an expander spring, of the same type of graphite operate in the ring grooves formed between the piston pieces. The piston rod packing is also made of this same carbon material. Piston rods, hardened as usual, and cylinder liners are machined, ground, and honed to a very fine micro-finish. A very close grained iron is used for the cylinder liner.

Best wear life of carbon parts is attained when there is present in the gas a very slight amount of vapor. The well known platforming process is served with such designs as shown in Figure 6.

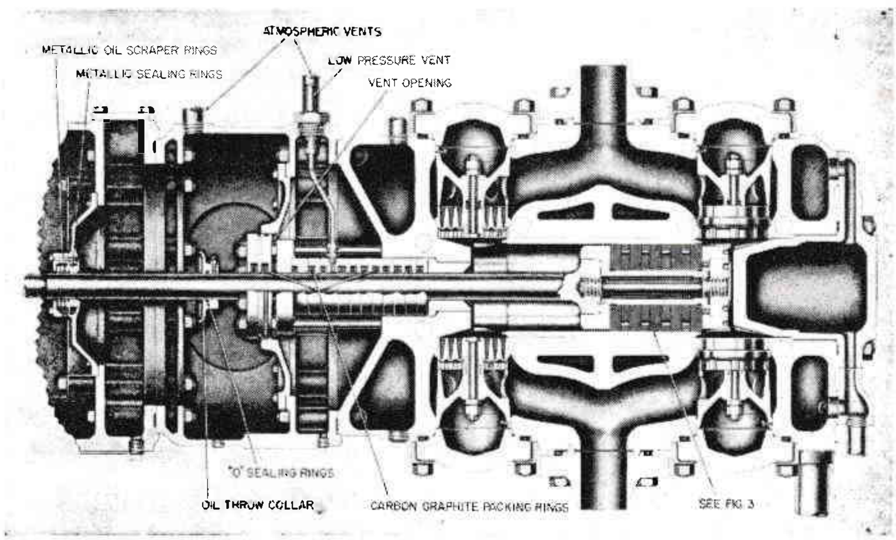


FIGURE 6—Non-lubricated compressor cylinder for moderate pressure. Best wear life of carbon parts if the gas has some vapor.

Lubrication—Reciprocating compressor cylinders are lubricated by the force feed mechanical lubricator. The lubricator is equipped with multiple units of individual plunger pumps with the unit having a built-in oil supply sump. A metered quantity of oil is forced into the cylinder at one or more points through check valves and to each packing as required. Such lubricators may be mechanically driven by the compressor or by an individual electric motor. The electric motor is more often used with motor-driven compressors to provide positive lubrication before the compressor motor is started.

Correct lubrication is vital to efficient operation of the compressor cylinder. In many cases the cylinders may be lubricated with the same oil as used to lubricate the engine or the compressor shaft and connecting rod bearings.

There are many cases where special consideration of lubrication problems is required. When the discharge temperature is high, the lubricant must have a high flash point when compressing air. Oxygen compressors must not use petroleum products nor vegetable oils for lubricants. Certain synthetic lubricants are suitable. When the compressor cylinder is handling gases carrying gasoline vapors or other condensible hydrocarbon vapors, a vegetable oil, or some combination of vegetable and petroleum oils may be used. A soap lubricant is sometimes recommended.

If gases are highly dehydrated, there

is a tendency to absorb the lubricant. A heavier bodied oil must, therefore, be used. Some gases have a chemical affinity for petroleum oil. In this case a special lubricant must be used, depending upon the gas.

In some processes, gases are involved which cannot be contaminated with lubricating oil. A lubricant must be used which, if carried into the gas, will not produce any ill effects. For example, an alcoholic soap lubricant has been found successful in compressing propane and carbon dioxide.

Compressor Drivers—The particular gas being compressed may not necessarily dictate the method used to drive the compressor. If the plant arrangement and availability of fuel dictates using an internal combustion engine, then compressor cylinders are generally integral as a part of an engine-compressor unit. Electric motor drives are frequently used. Some installations have available process steam; so, geared steam turbines become the logical choice. In the latter two cases, the compressor cylinders are attached to a compressor frame carrying its own crankshaft, connecting rods, and crossheads.

Quite a range of unit arrangements have been installed. These seem to run in all combinations from single cylinder compressor units to eight cylinder units and from one to six stages. The compressor manufacturers generally engineer each application to have an individual crank, connecting rod, and cross head for each compressor cylinder. Sometimes the best solution for multi-stage units requires placing some of the cylinders in tandem on a single connecting rod and crosshead.

It becomes apparent that the number of cranks, size of compressor pistons and phase relationship of multiple crank units influences the nature and magnitude of inertia forces resulting from the reciprocating motion. The compressor manufacturers keep these forces minimized by suitable arrangement of angles between the various cranks, by counterweighting, and by matching piston weights as near as possible.

Foundations—When engine-compressor units are installed, foundation requirements are controlled by the engine. A foundation designed as wide as possible and with sufficient mass will operate satisfactorily without excessive transmission of inertia forces to surrounding equipment and structures.

Reinforced concrete is the most universally used foundation construction. If the proposed location requires consideration of using friction piling, then the foundation construction must be carefully engineered to cope with the nature of the inertia forces involved. Piling structures must avoid resonance with the frequency of the reciprocating inertia forces and be of ample strength and suitable distribution to avoid forced vibration from these forces.

Motor or turbine driven compressor unit foundations are engineered to suit the compressor requirements. Breadth rather than depth is desired in the concrete mass. High foundations should be avoided so as to avoid rocking frequencies of the foundation that might be in resonance with inertia force frequencies.

A well engineered installation contributes greatly to successful long life performance of reciprocating compressors. A poorly engineered installation invariably results in many problems that preclude good performance no matter how well engineered the compressor unit itself might be.

Vibration—With multiple crank compressor units there are the problems of torsional vibration resulting from the harmonics of the varying pressures within the various compressor cylinders. The natural frequency of the compressor shafting may be controlled by suitable engineering of the coupling between the compressor and the driver. It can be controlled by selection of counterweights, and by adjustment of the mass of the driver or compressor pistons. The engineer correlates these factors to obtain a desired natural frequency to keep serious critical speeds removed from the operating speed range. Compromise enters the picture if the desired speed range conflicts with a feasible solution of the problem.

Along with the torsional vibration problem, it is necessary to investigate torque variation, caused by the compressor cylinder pressures, when motor drives or geared steam turbine drives are used. This must be done for all conditions of unloading.

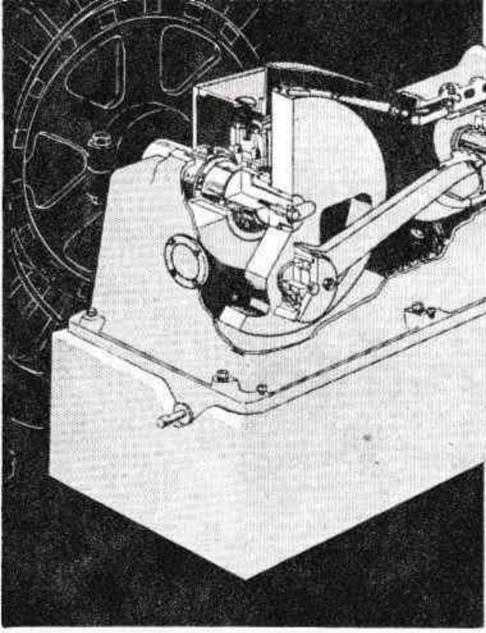
When alternating current motors are used as drivers, suitable WR^2 (flywheel effect) must be incorporated to keep current pulsation (caused by torque variation) within the tolerable limits established by motor manufacturers. Otherwise overheating and inefficient operation occurs. In the case of synchronous motors, the pulsation may be large enough to pull the motor out of synchronism and trip it off the line.

When geared turbine or other geared drives are used, torque variation resulting from compressor pressures or harmonics thereof may exceed the tolerance prescribed by the gear manufacturer. Excessive torque variation results in high tooth stresses and runs the risk of gear failure. Any torque reversal is intolerable because of gear tooth separation resulting in severe impact stress and impact noise. Short gear life in such cases is inevitable. High maintenance costs and unwanted down time are the penalties.

##

Meet the Author

THOMAS O. KUIVINEN, born in Ashtabula, Ohio, graduated from Ohio State University with a bachelors degree in mechanical engineering in 1929. Shortly after graduation, he was employed by The Cooper-Bessemer Corporation and since then his major activities have been in the fields of applied mechanics, design, and consultation. Ohio State conferred on him a professional degree of mechanical engineer in 1937. Then in 1939 he was assigned to the chief engineer's staff and in January, 1954 was made chief engineer, Technical Division. He is a member of the Society of Automotive Engineers and in 1954 was appointed an associate member of the executive committee of The Oil and Gas Power Division of The American Society of Mechanical Engineers.



Performance Characteristics—

Check this to be sure that the compressor will operate according to its design.

Howard M. Boteler
Worthington Corporation
Houston

THROUGHOUT the various branches of the petroleum and petrochemical industries one finds a great variety of reciprocating compressors performing a myriad of chores, each essential to the particular process to which applied. Some of the newer units may still be performing the jobs originally assigned them, operating under original contract conditions. Many others have been assigned new duties, some of them perhaps several times.

Frames and Cylinders—A reciprocating compressor consists essentially of a frame (or frames) to which are attached the necessary compressor cylinders and driver, plus the necessary auxiliaries such as intercoolers where required (either machine mounted or remotely located) interconnecting gas piping, jacket water piping, etc. Every compressor manufacturer has a group of standard frames, each having definite design limitations on speed and load carrying capacity. Load carrying capacity involves two considerations, namely horsepower and forces created by pressure differential across the pistons. The latter is generally referred to as frame load.

Horsepower limitations are determined by the ability of the crankshaft and crank webs (or crank discs) to withstand the necessary torque or turning effort without overstress, and by the ability of the bearings to dissipate the heat of frictional loading.

Frame load limitations are established to prevent the stresses in the frame, piston rods, connecting rods, bolting and other frame parts from exceeding safe conservative values, and to eliminate the imposition of excessive pressure per projected unit bearing area. Frame load for each compressor crank is equivalent to the sum of the differential loads of all pistons attached to that crank. The load for each piston is the difference in the products of net piston areas and pressures exerted on the piston faces. Loadings should be determined for both directions of piston movement. The loadings are not the same in each direction and where the ratio of piston rod area to piston area is high, they may be vastly different. An undesirable negative loading may sometimes result and provision must be made, such as a tail rod addition, to remove the unbalance.

Essential Information—Before a truly complete analysis of a compressor performance problem can be made certain essential data concerning the problem must be available. Such information must come from the prospective user in the inquiry. Figure 1 is an inquiry data sheet indicating facts which must be known for a full analysis. If the prospective user will furnish all of the information requested in this form, a much more intelligent analysis of the problem can be made than would otherwise be possible. A glance at the form will show numerous essential and revelant data. How essential the information is can perhaps best be realized by understanding how it is used.

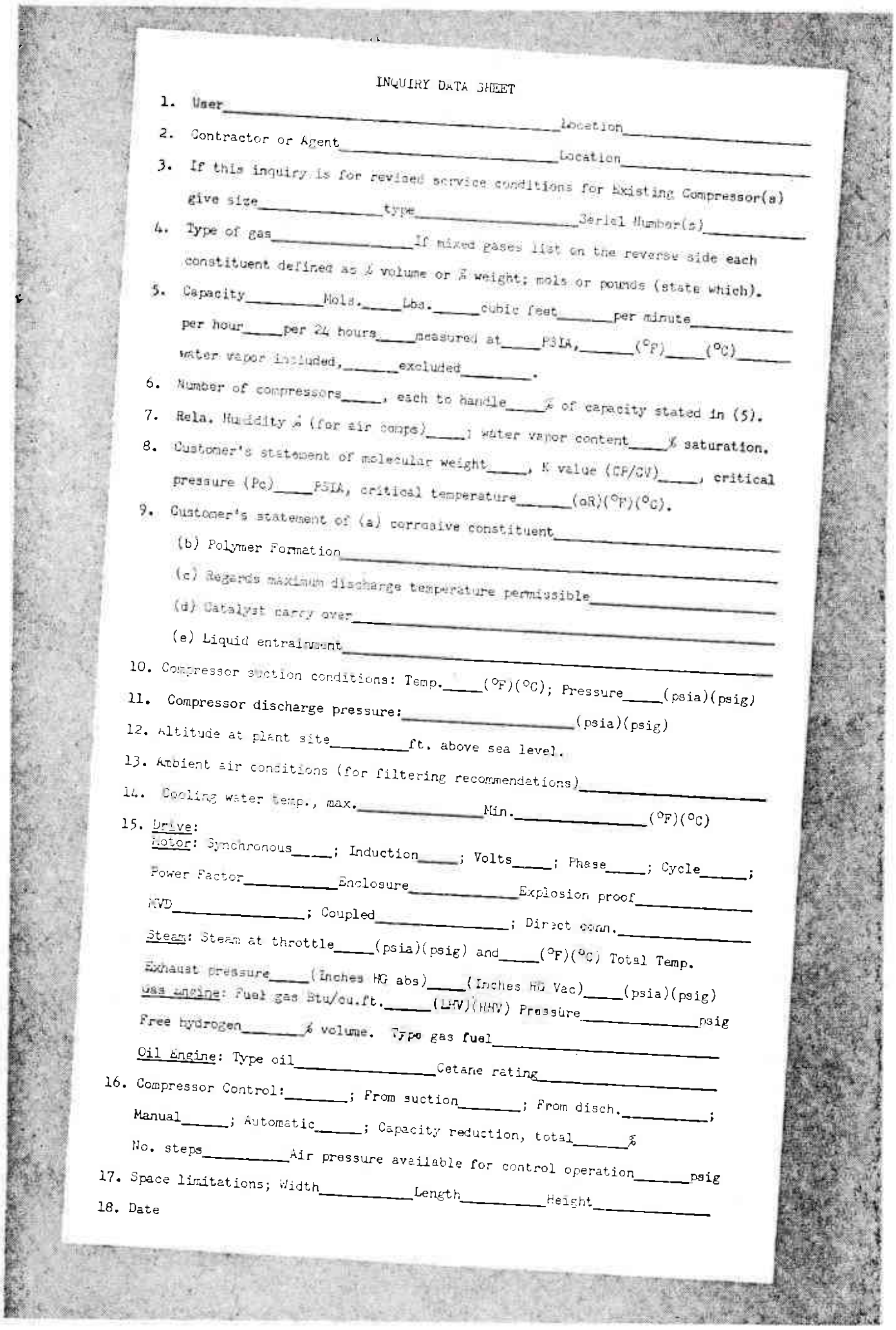
To better appreciate this, however, it is necessary to become familiar with certain terms and definitions used in the industry. Figure 2 sets forth the more common ones. These have perhaps appeared in print on many previous occasions, but are repeated here for convenience so that their true significance may be better understood.

The inquiry data determines the types and size of compressor and driver needed if new equipment is requested or if desired, whether an existing compressors and its driver are suitable for the stated service conditions. Or whether modification can be made to an existing machine to permit its use safely under the stipulated conditions. The procedure is briefly this:

1. The compression characteristics of the gas are determined. For single component gases the necessary information can be found in published tables. For mixed gases these characteristics are determined from the gas analysis. The physical constants to be determined are the K value (C_p/C_v) the molecular weight, the critical pressure (Pc) and the critical temperature (Tc). The critical pressure and temperature as determined from the analysis of mixed gases are sometimes termed pseudo-criticals.

2. The total ratio of compression is determined. This is the ratio of final absolute discharge pressure to absolute initial suction pressure. It must now be decided whether the compressor is to have 1, 2, 3 or more stages of compression. Experience will indicate quite readily how many are needed. The primary consideration is the expected discharge temperature. A sufficient number of stages, with cooling between them, must be used to prevent the discharge temperature of any of them exceeding practical limits.

Another consideration is the relationship of volumetric efficiency and ratio of compression. Gases having the



INQUIRY DATA SHEET

1. User____________________ Location__________
2. Contractor or Agent____________________ Location__________
3. If this inquiry is for revised service conditions for Existing Compressor(s) give size__________ type__________ Serial Number(s)__________
4. Type of gas__________ If mixed gases list on the reverse side each constituent defined as % volume or % weight; mols or pounds (state which).
5. Capacity______ Mols.____ Lbs.____ cubic feet____ per minute______ per hour____ per 24 hours____ measured at____ PSIA,______ (°F)____ (°C)______ water vapor included,______ excluded______.
6. Number of compressors____, each to handle____% of capacity stated in (5).
7. Rela. Humidity % (for air comps)____; water vapor content____% saturation.
8. Customer's statement of molecular weight____, K value (CP/CV)____, critical pressure (Pc)____ PSIA, critical temperature______ (oR)(°F)(°C).
9. Customer's statement of (a) corrosive constituent__________
   (b) Polymer Formation__________
   (c) Regards maximum discharge temperature permissible__________
   (d) Catalyst carry over__________
   (e) Liquid entrainment__________
10. Compressor suction conditions: Temp.____ (°F)(°C); Pressure____ (psia)(psig)
11. Compressor discharge pressure:__________ (psia)(psig)
12. Altitude at plant site________ ft. above sea level.
13. Ambient air conditions (for filtering recommendations)__________
14. Cooling water temp., max.__________ Min.__________ (°F)(°C)
15. Drive:
    Motor: Synchronous____; Induction____; Volts____; Phase____; Cycle____;
    Power Factor__________ Enclosure__________ Explosion proof__________
    MVD__________; Coupled__________; Direct conn.__________
    Steam: Steam at throttle____ (psia)(psig) and____ (°F)(°C) Total Temp.
    Exhaust pressure____ (Inches HG abs)____ (Inches HG Vac)____ (psia)(psig)
    Gas Engine: Fuel gas Btu/cu.ft.______ (LHV)(HHV) Pressure__________ psig
    Free hydrogen______% volume. Type gas fuel__________
    Oil Engine: Type oil__________ Cetane rating__________
16. Compressor Control:______; From suction______; From disch.________;
    Manual______; Automatic______; Capacity reduction, total______%
    No. steps________ Air pressure available for control operation______ psig
17. Space limitations; Width________ Length________ Height________
18. Date

FIGURE 1—Inquiry Data Sheet—List of facts to be determined for analysis of compressor performance.

higher K values will show a higher volumetric efficiency for the same cylinder clearance and ratio of compression than low K value gases. An extremely low volumetric efficiency for normal service is not desirable since an excessively larger cylinder is needed to obtain the desired capacity. This results in a heavy differential load (frame load) and may require a heavier frame than would otherwise be needed.

It is customary in the refinery and chemical process industries for the project engineer, for process reasons or otherwise, to establish the number of stages of compression, and for each, the suction pressure, suction temperature and discharge pressure with due consideration given to pressure drop and cooling between stages. If this information is not contained in the inquiry the manufacturer will ordinarily assume nearly equal ratios per stage to be satisfactory and will assume appropriate intercooler and inter-connecting piping pressure losses between stages. He will likewise assume suction temperatures for the second, third, etc. stages consistent with the available cooling water temperatures as shown in the inquiry.

This part of the procedure shows why it is essential that not only the suction pressure, but all interstage pressures, if given, as well as final discharge pressure be fully identified as either gage or absolute. Where gage readings are specified the feet elevation above sea level at the plant site must be known so that absolute pressures, mandatory in selection procedure, might be determined.

3. The required capacity per compressor is determined and is converted to cubic feet per minute at inlet conditions of the respective stages. The proper corrections for temperature, pressure and compressibility must be made. Where the desired capacity is stated as dry gas, the inlet volume of each stage must be increased by the ratio of the total absolute pressure to absolute partial pressure of the dry gas at stage inlet. This step of the procedure again emphasizes the necessity for clarity and completeness in the inquiry data. The desired capacity rate must be explicit and the conditions of pressure and temperature at which measured *must* be given.

4. A rough check is made to determine the horsepower required. For this purpose generalized charts are used. The factors involved are the total actual capacity of each stage (*cfm* at inlet), the ratio of compression per stage, the specific gravity and the K value (C_p/C_v) of the gas. Again this emphasizes the requirement for complete gas characteristics, clearly stated pressures, temperatures, and capacities.

The total horsepower required for the compressor (sum of horsepowers per stage) determines the required frame size. The kind of drive, whether steam, motor, gas, etc. as stated in the inquiry will dictate the type of frame best suited. There is usually an established length stroke and speed for the frame selected. However, for motor or steam turbine drive it may be necessary or advantageous to modify one or both to better match driver characteristics. Having established these details, the piston speed, which in feet per minute is equal to $\frac{\text{stroke in inches}}{12} \times 2 \times \text{rpm}$, as needed in Step 5, can be calculated.

TERMS AND DEFINITIONS

1. D = Piston displacement, expressed as cubic feet per minute, is equal to square feet of net piston area × length of stroke in feet × number of compression strokes per minute.
2. V_s = Actual capacity, cubic feet per minute, compressed and delivered, expressed at the conditions of total temperature and total pressure prevailing at compressor inlet.
3. VE = Volumetric Efficiency = $\frac{V_s}{D}$
4. THP = theoretical horsepower is the horsepower required to compress isentropically the gas delivered by the compressor through the specified pressure range. This is often referred to as the adiabatic horsepower.
5. CIHP = Compressor indicated horsepower is the actual work of compression developed in the compressor cylinder(s) as determined from the indicator card.
6. CE = Compression efficiency is the ratio of theoretical horsepower to compressor cylinder indicated horsepower = $\frac{THP}{CIHP}$
7. BHP = Brake horsepower is the measured horsepower input at the crank shaft.
8. SIHP = Steam indicator horsepower is the power developed in the steam cylinder(s) of an integral or coupled steam engine driven unit, as determined from the steam cylinder(s) indicator card, to perform the required gas compression duty.
9. ME = Mechanical efficiency is the ratio of compressor cylinder indicated horsepower to brake horsepower or to steam indicated horsepower = $\frac{CIHP}{BHP}$ or $\frac{CIHP}{SIHP}$
10. OE = Over-all efficiency is equal to (CE) × (ME) or $\frac{THP}{BHP}$ or $\frac{THP}{SIHP}$
11. P_s = Suction pressure, pounds/square in absolute.
12. T_s = Suction temperature, degrees Rankine = °F + 459.6.
13. P_d = Discharge pressure, pounds/square in absolute.
14. T_d = Discharge temperature, degrees Rankine = °F + 459.6.
15. R = Ratio of compression = P_d/P_s.
16. C_p = Specific heat at constant pressure is the number of BTU required to raise the temperature of 1 pound of perfect gaseous fluid one degree Fahrenheit, pressure remaining constant.
17. C_v = Specific heat at constant volume is the number of BTU required to raise the temperature of 1 pound of perfect gaseous fluid one degree Fahrenheit, volume remaining constant.
18. K = The ratio of specific heats = C_p/C_v.
19. T_c = Critical temperature is the temperature, degrees R, of a gas above which it cannot be liquefied.
20. P_c = Critical pressure and is the absolute pressure per square inch required to liquefy a gas at its critical temperature.
21. Pr = Reduced pressure is equal to P/P_c.
    Prs = Reduced pressure at suction pressure = P_s/P_c.
    Prd = Reduced pressure at discharge pressure = P_d/P_c.
22. Tr = Reduced temperature is equal to T/T_c.
    Trs = Reduced temperature at suction temperature = T_s/T_c.
    Trd = Reduced temperature at discharge temperature = T_d/T_c.
23. F = Compressibility factor which expresses the deviation from the perfect gas law. This is sometimes given the symbol "Z".
24. F_s = Compressibility factor at compressor inlet pressure and temperature.
25. F_d = Compressibility factor at compressor discharge pressure and temperature.
26. f = Ratio of compressibility factors = F_d/F_s.
27. C = Cylinder clearance volume and is the volume in one cylinder end in excess of the piston displacement per stroke. It is usually expressed as a percentage of stroke displacement.
28. TV = Terminal volume is the space occupied at discharge pressure and temperature of a unit volume of gas entering the cylinder at suction pressure and temperature.

FIGURE 2—Terms and definitions used in this article as applying to reciprocating compressors.

5. The approximate compressor cylinder sizes can now be determined. The required capacity of each has already been established (Step 3). From appropriate charts or formulae the volumetric efficiency of each stage of compression is developed. This involves the ratio of compression, the *K* value of the gas and the compressibility factors at inlet and discharge conditions. A trial selection must be made, assuming cylinder clearance, and the size and clearance readjusted to match cylinder patterns and/or designs available.

This procedure again emphasizes the requirement for exacting inquiry data. If facts for establishing critical constants are missing the cylinders could be sized incorrectly, sometimes appreciably.

6. The frame loadings can now be checked to be sure that all are within the design limits established for the selected frame. If all are too high a heavier frame must be used. If one of several are high it may be possible to readjust the ratios of compression to bring all loadings within the rating. Failing this, a heavier frame becomes necessary.

7. Assuming the frame requirements have been met, the selection, so far as stroke, speed, number of stages and sizes of cylinders is now complete. There remains one more detail —a more accurate check of the horsepower required. In Figure 2 will be noted the terms "compression efficiency" and "mechanical efficiency." In Step 4 generalized horsepower charts were used. Such charts are termed generalized because they are based on average values of mechanical and compression efficiencies. It should be emphasized here that it is impossible to include in any one horsepower chart, such as the widely used "bhp per million" chart, other than average values for these two factors. Therefore, after the selection has been made and all factors involved in determining the compression and mechanical efficiencies are known, the horsepower can be quite accurately established.

The selection is now complete with full performance characteristics known for the conditions of service for which selected.

Multiple Service—Quite often an inquiry will list one or more alternate service conditions, such as a range in suction pressure or discharge pressure or both. Or perhaps, operation on alternate gas streams may be desirable, where pressures and capacities may or may not be identical. In such cases the requirements must be analyzed and the selection based upon the most severe conditions. It may be necessary to select cylinder sizes for one service, to obtain the desired capacity, and to base frame selection on the alternate service because of high horsepower and/or frame loadings required by the alternate.

PERFORMANCE CALCULATIONS

The Indicator Card—Figure 3 lists some useful formulae used in compressor selection and performance calculations. From these formulae the variables involved can readily be seen.

PERFORMANCE FORMULAE RECIPROCATING COMPRESSORS

1. **Ratio of Compression** = ... $\frac{P_d}{P_s}$

2. **Theoretical Volumetric Efficiency** = ... $1.00 - \left(\frac{R^{\frac{1}{k}}}{f} - 1\right) C$

3. **Theoretical Means Effective Pressure** = ... $\frac{KP_s}{K-1}\left(R^{\frac{k-1}{k}} - 1\right)$

4. **Theoretical (isentropic) Horsepower** = ... $.004364 \times V_s \times (M.E.P.)$

5. **Theoretical Discharge Temperature** = ... $T_s\left(R^{\frac{k-1}{k}}\right)$

6. **Theoretical Terminal Volume Expressed as Percent of V_s** = ... $\frac{f}{R^{\frac{1}{k}}}$

Note: All pressures and temperatures are in obsolute units. For nomenclature refer to Figure 2.

FIGURE 3—Performance Formulae—Used to determine compressor characteristics.

The formulae given are all theoretical. Certain modification of results obtained must be made to arrive at actual results. For instance, the actual volumetric efficiency is always lower than that derived from the formula shown because allowance must be made for entrance losses due to wire drawing through the cylinder ports and valves and for increase in volume due to heating during the inlet period. In addition, compensation must be made for leakage past piston rings and cylinder valves and for packing vent losses. Obviously, leakage is primarily a function of pressure differential and type of cylinder construction. Non-lubricated cylinders (carbon piston rings, etc.) can be expected to develop greater slippage than those receiving lubrication, as the oil effectively seals surface contacts in most cases. However, also of importance is the type of gas being handled. Certain gases are simply harder to retain than others and the problem is closely connected with molecular weight and viscosity. There are no hard and fast compensating factor that can be given to cover all these variables. It is customary to allow 3 percentage points for wire drawing and heating entrance losses, so that 0.97 may be substituted in the formula for 1.00. Slippage losses may vary from 2 to probably a maximum of 5 percentage points for ordinary applications involving oil lubricated cylinders and from perhaps 4 to perhaps 10 percentage points for non-lubricated construction. Thus a more practical formula might be:

Volumetric Efficiency = $.97 - \left(\frac{R^{1/k}}{f} - 1\right)C - L$, where L is the slippage loss just discussed.

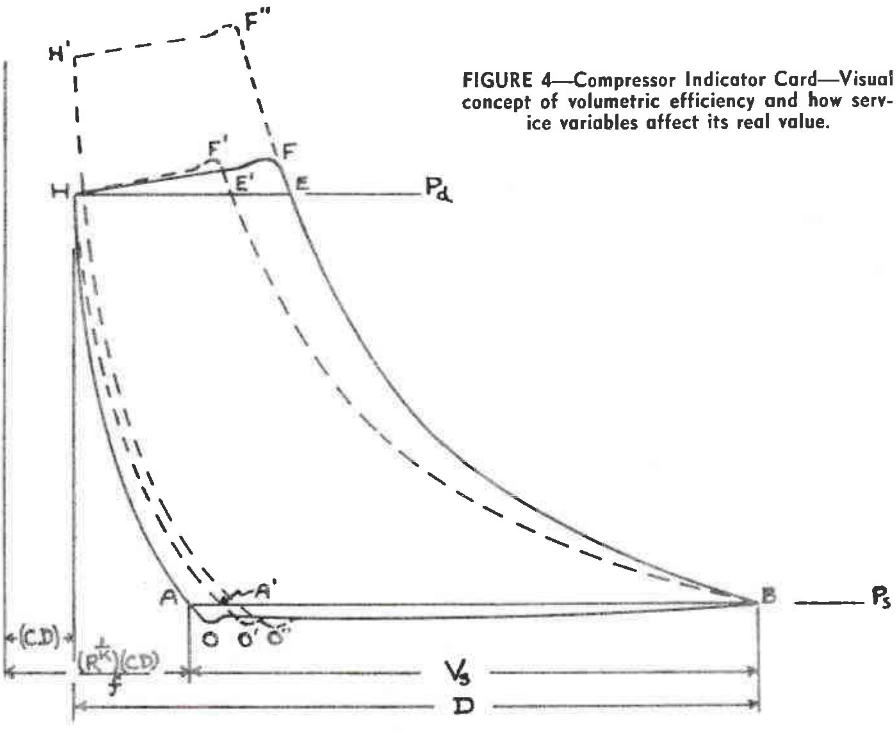


FIGURE 4—Compressor Indicator Card—Visual concept of volumetric efficiency and how service variables affect its real value.

A visual concept of volumetric efficiency can be had by reference to Figure 4. The indicator card shows what takes place within the compressor cylinder. The shape or appearance of the card varies with pressure conditions, cylinder clearance, gas characteristic and other factors.

In Figure 4, diagram ABEH represents behavior of a perfect gas under ideal conditions. P_s and P_d represent the suction and discharge pressures at the inlet and discharge flanges of the cylinder respectively. Movement of the piston from left to right represents the suction stroke. Point B represents the extreme point of piston travel on the suction stroke.

At B, the direction of piston travel is reversed and as the piston moves from right to left on the discharge stroke, compression of the gas trapped within the cylinder proceeds along line BE. At point E, pressure within the cylinder is equal to discharge pressure. Point E, therefore represents effortless opening of the discharge valve. From this point to the end of the discharge stroke at H, the discharge valve remains open while gas is being delivered to the discharge system.

It will be noted, however that some of the gas has been compressed into the clearance spaces within the cylinder and remains in the cylinder when the discharge valve closes. These clearance spaces are found between the valves and the piston, in the passageways in the valves themselves, in head cutouts under the valves for gas passages, linear clearance between the piston and head faces, etc.

As the piston reverses direction at H and again starts on the suction stroke, the gas at discharge pressure in the clearance space begins to expand. Volume increase and pressure reduction continue as the piston moves. Finally, the pressure in the cylinder is equal to suction pressure, P_s, and the effortless suction valve opens at A, permitting the inflow of fresh gas for that portion, AB, of the suction stroke.

Since the displacement is a function of the length of stroke and since the length of the indicator card is proportional to the length of stroke, we can in the diagram, substitute displacement, D, for length of indicator diagram. We can also represent that portion AB of the suction stroke where in gas is flowing into the cylinder by V_s, the actual capacity at intake conditions. $\frac{V_s}{D}$, then is the apparent volumetric efficiency. The value indicated here is usually higher than the actual because the reduction caused by leakage loss is not fully effective at the point of valve opening. There is some leakage from the clearance spaces past the rings and suction valves, as the gas expands. This creates a faster than normal pressure decline and would tend to move point A to the left, for an apparent increase in volumetric efficiency. On the other hand, the expansion from H to A is not completely isentropic and therefore the temperature and volume are greater than theoretical. This slows the actual pressure drop and would tend to move point A to the right for a decrease in volumetric efficiency.

The increase in volume during the intake, AB, because of heat picked up by the gas, would slow the rate of flow from the suction system. In addition, leakage past the rings and valves during the discharge stroke reduces the volume delivered to the discharge system. Since the definition of actual capacity takes these losses into consideration, so must the volumetric efficiency. Further, the actual valve opening does not occur at A. The pressure in the cylinder must be sufficiently below the pressure at the inlet flange to overcome the effect of unbalanced valve area and initial spring resistance before the valve can open. The actual opening is, therefore at point O rather than at point A. Since none of these losses are reflected in the apparent or indicated volumetric efficiency, it is usually higher than the actual. The deviation will vary, however five percentage points might be considered the expected average.

The equation for theoretical volumetric efficiency shown in Figure 3 can be explained also by reference to the indicator diagram; Figure 4. For the expansion we can write the equation

$$\frac{P_1V_1}{T_1F_1} = \frac{P_2V_2}{T_2F_2}$$

Rearranging

$$V_2 = \frac{T_2F_2P_1V_1}{T_1F_1P_2}$$

Substituting subscripts d (Discharge) and s (Suction) for subscripts 1 and 2 respectively.

$$V_2 = \frac{T_sF_sP_dV_d}{T_dF_dP_s}$$

But:

$V_2 = V_{sc} =$ the volume of the clearance when expanded isentropically from pressure P_d to pressure P_s.

Also:

$$\frac{T_s}{T_d} = \frac{1}{R^{\frac{k-1}{k}}}$$

$$\frac{F_s}{F_d} = \frac{1}{f}$$

$$\frac{P_d}{P_s} = R$$

$$V_d = CD$$

Substituting:

$$V_{sc} = \frac{1}{R^{\frac{k-1}{k}}} \times \frac{1}{f} \times R \times CD = \frac{\left(R^{\frac{1}{k}}\right)\left(CD\right)}{f}$$

$$\text{Volumetric Efficiency} = \frac{V_s}{D}$$

$$\text{But } V_s = (D + CD) - \frac{(R^{1/k})\ (CD)}{f}$$

Substituting

$$\text{Volumetric Efficiency} = \frac{D + CD - \frac{(R^{1/k})\ (CD)}{f}}{D}$$

$$= 1 + C - \frac{(R^{1/k})\ (C)}{f} = 1 - \left(\frac{R^{1/k}}{f} - 1\right) C$$

Brake Horsepower—The theoretical horsepower naturally does not include fluid or mechanical friction losses. The fluid losses are grouped in the term Compression Efficiency while the mechanical ones are covered by the term Mechanical Efficiency.

The major factor involved in determining the compression efficiency is the valve loss or pressure drop through the inlet and discharge valves. These fluid losses are a function of gas density and valve velocity. The suction and discharge pressures and the molecular weight establish the density. The valve velocity is fixed by the valve area available in the selected cylinders and the piston speed. Valve velocity is normally stated in feet per minute and is the ratio of piston area to valve area per cylinder corner multiplied by the feet per minute piston speed. It is likewise equal to $\frac{144D}{A}$ where D is the piston displacement of one cylinder, in cubic feet per minute, and A is the valve area in square inches per cylinder corner. By valve area per corner is meant the area of the suction *or* the discharge valves in one end of the cylinder.

Valve losses are usually read from graphs plotted from research data. Applicable corrections for density are designated. There is at present, so far as is known, no precise general formula available for calculating these losses because of the numerous variables in valve designs in use, such as spring loadings, unbalanced areas and flow patterns.

A better understanding of the losses involved in compression efficiency may also be obtained by reference to the indicator diagram. If it were possible to get the gas into and out of the cylinder without fluid losses, the indicator card ABEH could be realized. This card may be said to represent the ideal or theoretical horsepower requirements. But fluid losses are present. Therefore the actual inlet pressure in the cylinder is below that at the cylinder inlet flange. Likewise the pressure in the cylinder during the delivery interval EH is above that at the cylinder discharge flange.

A more practical card would be AOBFH. Here the area AOB, fluid losses through the inlet ports and valves, and the area EFH fluid losses through the discharge valves and ports are included in the card area. Since the horsepower is proportional to the card area, the larger area by reason of fluid loss inclusion, means greater horsepower.

There are other considerations. During compression, BF, a small portion of the gas continually slips past the piston rings and suction valves. Work has been done on this gas, yet it is not delivered to the discharge system. Also slippage past the discharge valves allows gas which has already been delivered to the discharge system to return to the cylinder. Recompression and redelivery take place.

Unless leakage is abnormal the theoretical location of point E is not

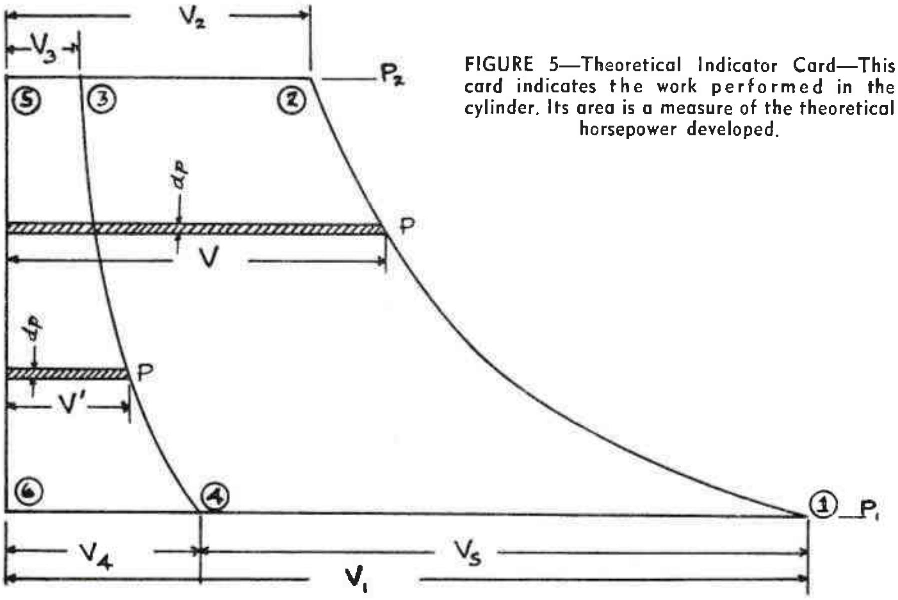


FIGURE 5—Theoretical Indicator Card—This card indicates the work performed in the cylinder. Its area is a measure of the theoretical horsepower developed.

greatly altered. Yet an over-all loss has occurred, first because more gas must be taken into the cylinder to compensate for piston ring and suction valve slip, secondly because work is performed on this lost capacity, and finally because leakages back through the discharge valves must be recompressed and redelivered to the discharge system.

There is still another factor, the cooling effect of cylinder jacketing. Removal of heat by the jacket water would shrink the volume during compression and would tend to move point F to the left reducing the power required. This is a true saving in power expended. Unfortunately it is not of any great significance, except in small cylinders handling rather low density gas through high ratios of compression where the jacket surface is large in proportion to the amount of work performed and heat generated. Further, cold jacket water should be avoided where the compressor is handling readily condensible gases which enter the cylinders at or near the dew point. In these cases it is actually desirable to superheat the gases during the inlet and early compression portions of the cycle, by hot jacket water, to prevent condensation within the cylinders. Condensation, liquid of any kind, and wet saturated gases are extremely detrimental to lubrication. In many cases the same factors adversely affect non-lubricated construction, especially valve service life.

The compression efficiency can be said to equal the ratio of the area of the theoretical card to that of the actual card, the actual card including the fluid entrance and exit losses as above outlined. A reasonable allowance must also be made for the slippage losses. These losses may ordinarily be considered as a reduction in compression efficiency equivalent to ½ to ¾ of the percentage point losses indicated in the volumetric efficiency formula. Again these losses must be predicated on experience and dictated by cylinder construction features, type of gas handled and operating conditions.

Horsepower Formula Development—The indicator diagram illustrates excellently the derivation of the formula shown herein for theoretical horsepower. To prevent confusion a second diagram for this purpose is shown in Figure 5. Path 1-2-5-6 is the compression cycle of a single stage compressor with zero clearance, while path 1-2-3-4 represents single stage compression with cylinder clearance. The energy represented by the area within the paths 1-2-3-4, compression with clearance, is the energy bounded by paths 1-2-5-6 minus the energy bounded by paths 3-5-6-4. This latter is the energy recovered from the expansion of the gas in the clearance space.

The energy of the zero clearance cycle is

$$\int_{P_1}^{P_2} V dp \quad \ldots\ldots\ldots\ldots (1)$$

The energy from expanding clearance gas is

$$\int_{P_1}^{P_2} V^1 dp \quad \ldots\ldots\ldots\ldots (2)$$

The net energy therefore $= \int_{P_1}^{P_2} V dp -$

$$\int_{P_1}^{P_2} V^1 dp \quad \ldots\ldots\ldots\ldots (3)$$

For isentropic compression and expansion

$P_1V_1^k = PV^k$ from which

$$V = P_1^{\frac{1}{k}} V_1 \left(P^{-\frac{1}{k}} \right) \ldots\ldots\ldots (4)$$

$$\text{and } V^1 = P_1^{\frac{1}{k}} V_4 \left(P^{-\frac{1}{k}} \right) \ldots\ldots (5)$$

Substituting valves of V and V^1 from Equations 4 and 5 into Equation 3

$$\text{Net E} = P_1^{\frac{1}{k}} V_1 \int_{P_1}^{P_2} P^{-\frac{1}{k}} dp$$

$$- P_1^{\frac{1}{k}} V_4 \int_{P_1}^{P_2} P^{-\frac{1}{k}} dp \ldots\ldots (6)$$

$$= P_1^{\frac{1}{k}} V_1 \left[\frac{P^{\frac{k-1}{k}}}{\frac{k-1}{k}} \right]_{P_1}^{P_2} - P_1^{\frac{1}{k}} V_4 \times \left[\frac{P^{\frac{k-1}{k}}}{\frac{k-1}{k}} \right]_{P_1}^{P_2}$$

$$= \left(\frac{k}{k-1} \right) \left(P_1^{\frac{1}{k}} V_1 \right) \left[P_2^{\frac{k-1}{k}} - P_1^{\frac{k-1}{k}} \right]$$

$$- \left(\frac{k}{k-1} \right) \left(P_1^{\frac{1}{k}} V_4 \right) \left[P_2^{\frac{k-1}{k}} - P_1^{\frac{k-1}{k}} \right]$$

$$= \left(\frac{k}{k-1} \right) P_1^{\frac{1}{k}} \Big) \left(V_1 - V_4 \right)$$

$$\times \left[P_2^{\frac{k-1}{k}} - P_1^{\frac{k-1}{k}} \right] \ldots\ldots\ldots (7)$$

But $(V_1 - V_4) = V_s$, which is the actual capacity, CFM, measured at intake conditions. Substituting $V_s = (V_1 - V_4)$ in Equation 7.

Net E =

$$\left(\frac{k}{k-1} \right) \left(P_1^{\frac{1}{k}} \right) \left(V_s \right) \left[P_2^{\frac{k-1}{k}} - P_1^{\frac{k-1}{k}} \right]$$

$$\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

Multiplying each side of equation 8 by

$$\frac{P_1^{\frac{k-1}{k}}}{P_1^{\frac{k-1}{k}}}$$

Net Energy =

$$\left(\frac{k}{k-1} \right) \left(P_1 V_s \right) \left[\frac{P_2^{\frac{k-1}{k}}}{P_1^{\frac{k-1}{k}}} - 1 \right]$$

$$\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

But $\frac{P_2}{P_1} = R$; and substituting in Equation 9

Net Energy =

$$\left(\frac{k}{k-1}\right)\left(P_1V_s\right)\left(R^{\frac{k-1}{k}}-1\right)\cdots\cdot(10)$$

The energy obtained from solving Equation 10 is in foot pounds per minute provided P is expressed as pounds per square foot absolute. Since in compressor work it is customary to use pounds per square inch as normal units, it is necessary to multiply by 144 and the energy formula then becomes:

$$W = 144\left(\frac{k}{k-1}\right)\left(P_1V_s\right)\left(R^{\frac{k-1}{k}}-1\right) \cdots\cdots(11)$$

where

W = Foot pounds of energy per miinute.
$K = C_p/C_v$
$P_1 = P_s$ = PISA suction pressure
V_s = Actual capacity, CFM,/minute measured at inlet condition
R = Ratio of compression = $P_2/P_1 = P_d/P_s$

The theoretical horsepower then becomes:

$$HP = \left(\frac{144}{33000}\right)\left(\frac{k}{k-1}\right)\left(P_sV_s\right) \times\left[R^{\frac{k-1}{k}}-1\right]\cdots\cdots(12)$$

No attempt will be made here to explain the development of the horsepower formula including compressibility factors as it is felt that such is beyond the intended scope of this paper.

Variables of Performance—Volumetric Efficiency as represented by formula shown in Figure 3 and also by the relationship V_s/D, Figure 4, the indicator card, is obviously affected by:

(a) The cylinder clearance.
(b) The ratio of compressibility at discharge pressure and temperature to that at suction pressure and temperature.
(c) The K value of the gas being compressed.
(d) The ratio of compression.

By inspection of the formula it is noted that

(a-1) Increase in clearance lowers the volumetric efficiency. Thus it is common practice to use clearance pockets (additional clearance) to reduce capacities.
(b-1) Decrease in f lowers the volumetric efficiency. For perfect gases $f = 1$. For most hydrocarbon mixtures $f < 1$ while for certain diatonic gases especially hydrogen and nitrogen $f > 1$ under certain conditions of pressure and temperature.
(c-1) Decrease in K value lowers the volumetric efficiency. Thus, the volumetric efficiency of a compressor handling propane ($K = 1.15$) would be lower than were the same cylinder compressing hydrogen ($K = 1.40$).
(d-1) Increase in the ratio of compression lowers the volumetric efficiency.

A change in any of the above variables opposite to that indicated results in a higher volumetric.

The same phenomena can be visualized from the indicator card, figure 4.

(a-2) When clearance is increased there is more gas trapped in the clearance spaces. The numerical value of $\frac{(R^{1/k})\ (CD)}{f}$ is therefore increased and V_s, actual intake capacity is decreased as is V_s/D, the apparent volumetric efficiency. This would result in the expansion line taking some path, such as HA'O' resulting in delayed opening of the suction valves. The compression line would be less steep and might be represented by BE'F'. The delivery interval F'H is also shorter. This is a further indication of reduced capacity.
(b-2) A decrease in f would have the same direction tendency upon the location of O and E as does an increase in clearance (a-2).
(c-2) Same tendency as (a-2) and (b-2).
(d-2) Here P_d would be higher than shown on the diagram or P_s would be lower, or both.

In either case the numerical value of $\frac{(R^{1/k})\ (CD)}{f}$ would increase. For instance if suction pressure remained constant and discharge pressure increased the compression would be carried to F''. Thus, a shorter delivery interval F''H' would prevail. Expansion of clearance gases would begin at H' and end at O'', where inlet would begin. Thus V_s, V_s/D and F''H' would be reduced below similar values prevailing at the lower P_d.

THEORETICAL HORSEPOWER as represented by the formula in Figure 3 and by the area of the indicator diagram Figure 5, is affected by

a) The suction pressure, P_s.
b) The capacity, V_s.
c) The K value of the gas.
d) Cylinder clearance.
e) The ratio of compression.
f) The ratio, f, of compressibility factors.

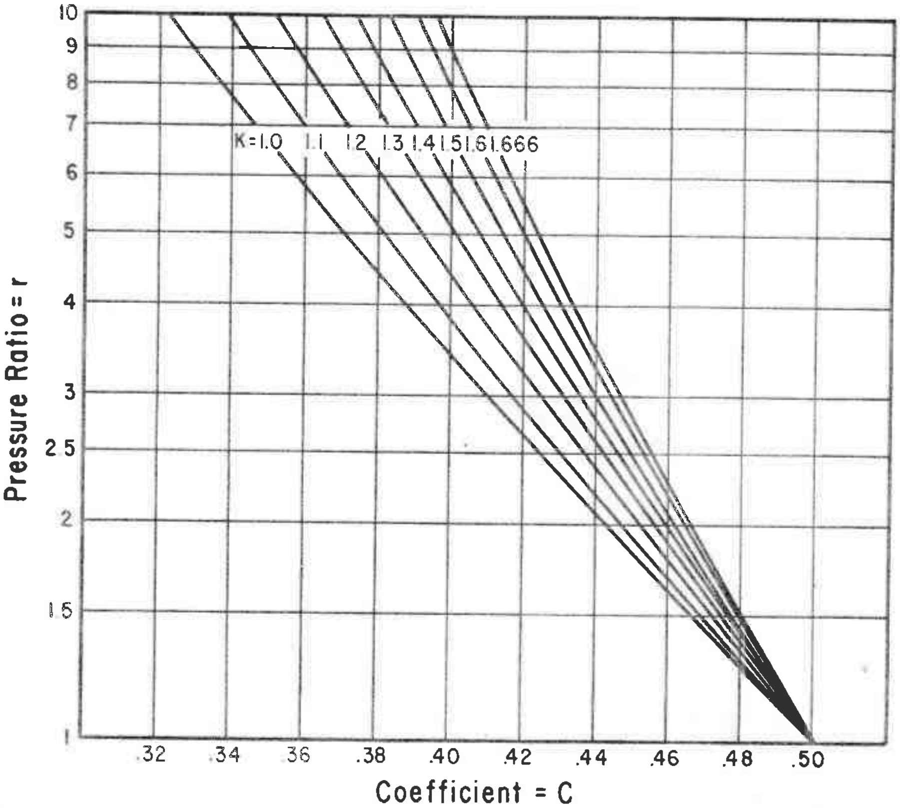
By reference to formula.

a) Other factors remaining constant the horsepower change is directly proportional to the change in suction pressure. For instance if the suction pressure is increased 50 percent the horsepower increases in the same proportion.
b) Other factors remaining constant the horsepower required changes in direct proportion to the change in V_s. For example if the capacity is reduced 50 percent, horsepower is likewise reduced 50 percent.
c) The variation brought about by change in K is three-fold. First, the capacity is altered. Secondly, the value of $\frac{K}{K-1}$ differs and thirdly, the value of $R^{\frac{k-1}{k}} - 1$ changes. The net proportionate increase or decrease is the product of the ratio of altered to original values of the affected phrases in the formula. Suppose for example the K value change is such that V_s increases 5.13 percent. This would mean a corresponding decrease in $\frac{K}{K-1}$ of 33.3 percent but an increase in $R^{\frac{k-1}{k}} - 1$ of 60.7 percent. The ratio of altered to original horsepower then is $1.0513 \times .667 \times 1.607 = 1.125$. Thus there is a net change of $+12\frac{1}{2}$ percent in power required.

This change would result if a compressor with 10 percent cylinder clearance compressing a gas of K value 1.24 through 4 ratios of compression were changed to air service ($K = 1.41$), suction pressure unchanged.
d) The cylinder clearance does not appear in the horsepower formula but as noted under volumetric efficiency, item (a), a change in clearance results in altered volumetric efficiency and capacity. The effect of clearance change on horsepower is in direct proportion to the change in capacity brought about by the change in clearance. Should the clearance change reduce the capacity 25 percent the horsepower likewise would be 25 percent lower.
e) Should the ratio of compression vary, so will the capacity (see volumetric efficiency, item (d). Likewise, formula phrase $R^{\frac{k-1}{k}} - 1$ would vary. Assume the compressors described in the preceding paragraph were required to operate at 5 ratios of compression instead of at 4 ratios. The K value is 1.24 and the suction pressure and cylinder clearance are unaltered. The decrease in volumetric efficiency would be approximately 8.0 percent. The change in $R^{\frac{k-1}{k}} - 1)$ would be an increase of 18.3 percent. The net change therefor would be ($.920 \times 1.183$) or 8.8 percent increase in power required.
f) The effect of compressibility on the horsepower required per cubic foot capacity measured at intake conditions does not appear in the formula. It is small for average operating conditions. In high pressure units, however effects of sizeable proportions are sometimes found. For example, consider a cylinder handling carbon dioxide for a urea plant. Suction pressure is 1720 PSIA at 115 F. The discharge pressure is 3440 PSIA. From published data we find the critical pressure of carbon dioxide to be 1073 PSIA and the critical temperature to be 548° R. At suction conditions the reduced pressure, $Prs = \frac{1720}{1073} = 1.6$. The reduced temperature, $Trs = \frac{575}{548} = 1.05$. From a standard compressibility chart (1 psia basis), the compressibility factor, F_s, at intake conditions is .312.

The theoretical discharge temperature, using the formula from Figure 3, is 675 R(215 F.). Therefore, at discharge conditions of 3440 PSIA and 215 F.; $Prd = \frac{3440}{1073} = 3.2$. $Trd = \frac{675}{548} = 1.232$. Referring again to a standard compressibility chart, the compressibility factor, F_d, at discharge conditions = .575. Therefore $F_d/F_s = f = 1.842$.

Referring now to Figure 6 and following horizontally from a Ratio of



Procedure:

1. Determine compressibility factors F_1 and F_2 from compressibility charts
2. Calculate $f = \frac{F_2}{F_1}$
3. From given r and k determine coefficient (c) from this chart
4. Change in horsepower (deviation) = C (f — 1) HP (Ideal Isentropic HP)
5. Corrected horsepower is algebraic sum of Ideal BHP and deviation, Item 4 above.

FIGURE 6—Horsepower Deviation Coefficient.

Meet the Author

HOWARD M. BOTELER received a B.S. degree in mechanical engineering from the University of Maryland in 1923. He joined the Worthington Corporation that same year and spent several years in research and development, and field engineering. Two years later, Boteler was placed in charge of the research department, and two years after that was made manager of the Service and Erection department. In 1932, he was transferred to sales at Buffalo, N. Y. Here he was responsible for the application of reciprocating compressors for refrigeration cycles. In 1954, Boteler was transferred to Houston where he assumed his present position as engineering and sales consultant with respect to reciprocating compressor applications in the Southwest region.

Compression of 2, to the intersection with the diagonal line representing K = 1.3 (C_p/C_v for carbon dioxide) and reading vertically downward the coefficient "C" equal to .456 is found. Using the explanation given below the chart in this figure, the increase in theoretical horsepower will be C (f — 1) = .456 × 8.42 = .384 which means an increase of 38.4 percent in theoretical horsepower due to the effect of compressibility.

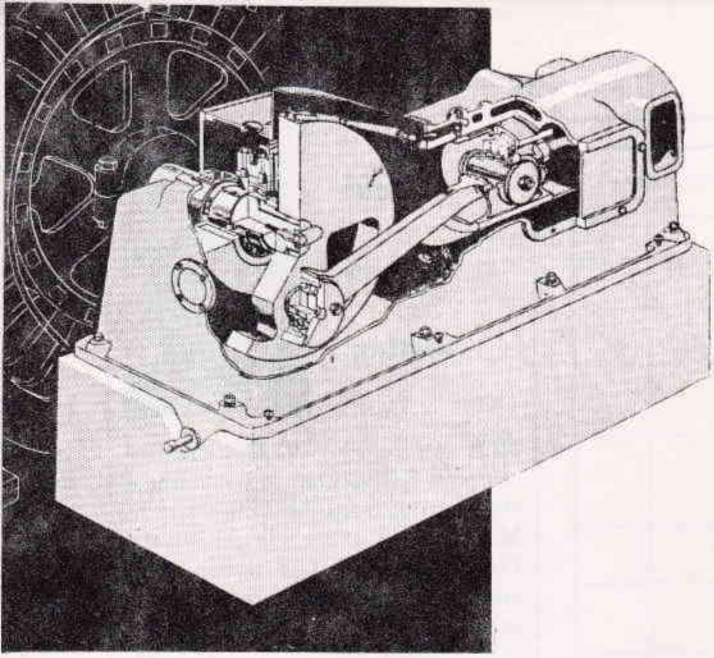
This example is probably one of the most dramatic which might have been selected but it has been used to exemplify the possibility of incurring overload unless all conditions surrounding the inquiry are known and evaluated. The converse is sometimes true and the driver may be appreciably underloaded. However, driver underload does not preclude operation whereas overload may. The possibilities must be checked to be reasonably certain. Compressibility factors are important.

Dirty Corrosive Gases and Wet Gases—The user should not hesitate to give the manufacturer a complete and frank statement of the expected qualities of the gas to be compressed. If the gases have a tendency to polymerize when subjected to normal discharge temperatures (250 F. to 350 F.) such fact should be made known. Additional stages of compression, beyond ordinary practice, may be needed. The user should attempt to give his version of temperature limitation to assist the manufacturer in his selection.

Every precaution should be taken to prevent catalyst dust, or dirt of any kind, entering the compressor. However, if previous experience indicates, regardless of practical safeguards used, that dust inclusion cannot be prevented this fact should be made known. While no one can guarantee 100 percent satisfactory compressor service under such conditions, yet a true statement and discussion of the facts could aid in the selection of more suitable and longer lasting materials for such adverse operating conditions.

The same might be said about corrosive constituents in the gases. Here the manufacturer's suggestions on predrying and preheating the inlet gas may prove invaluable. Likewise, provisions might be made in the cylinder jacketing to keep the gas dry while it is in the cylinder. Many troublesome corrosive problems have been cured in this manner.

Entrained liquid should never be permitted to enter a compressor cylinder. Neither should the jacketing arrangement and jacket temperature cause condensation within the cylinder. Adequate separators or scrubbers with reliable and positive drain features must be used before successful operation can be expected. This liquid removal equipment should be adjacent to the cylinder inlet to preclude post scrubber condensation. Once in the cylinder, readily condensable gases should be heated, rather than cooled, by the cylinder jackets. Hot jacket water is preferable to no water because its use does tend to maintain more uniform cylinder casting temperatures, thus minimizing distortion. A simple thermosyphon jacket system may suffice in many cases when provided with a coolant having a suitable boiling point. ##



Installation, Operation and Maintenance — These experience-proven tips will save costly errors.

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PROPER INSTALLATION and handling of compressors is not acquired so much from a text as it is from experience. The sharp operator can profit considerably from the mistakes and experiences of others.

This discussion will be confined to the compressor only and not its prime mover except where the relationship cannot be disregarded. The article will also dwell lightly upon or omit entirely those things which are commonly accepted, and those installation practices which every manufacturer's service man will follow. Thus the presentation will be devoted principally to pointing out those things which are often omitted or neglected in many installations.

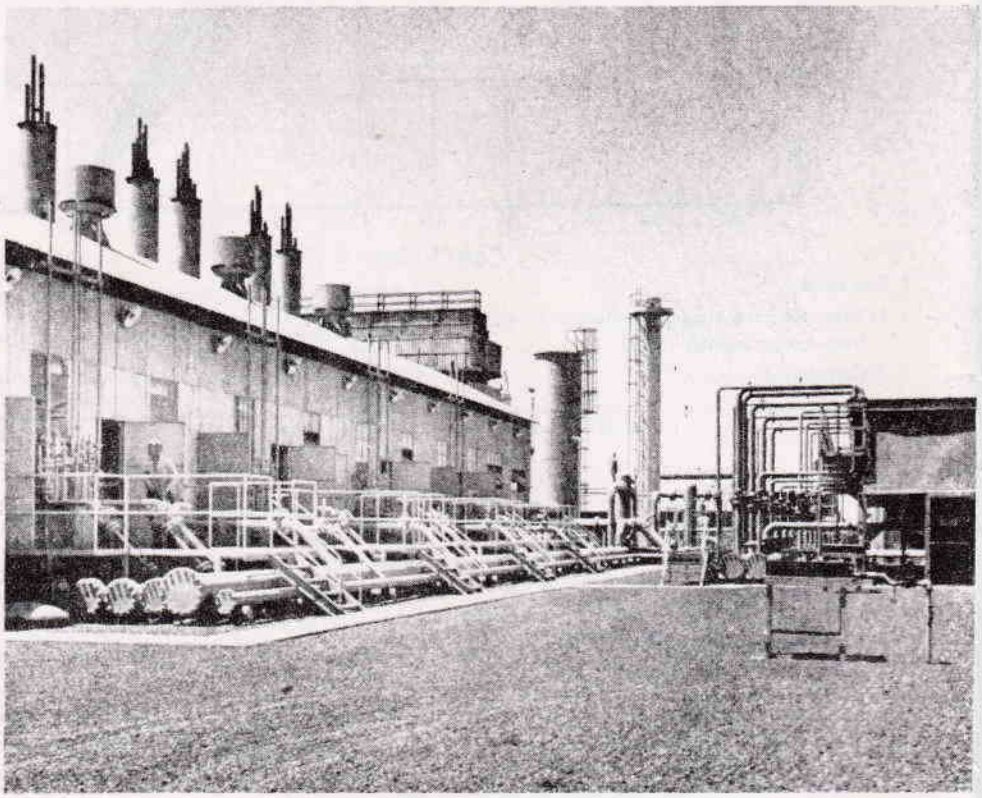

FIGURE 1—Gas headers outside bridged by separate walkway from each other.

INSTALLATION

Selection of Site—The proper location for the installation involves such factors as safety, good soil, accessibility and proximity to cooling water. The compressor house location must be influenced by safety, particularly if the prime movers are internal combustion engines. Convenience to the gas being compressed, good soil for foundations, accessibility to a roadway for transport, and proximity to cooling water are influencing factors, although fuel and water for engines or electric power for motors can be brought to the site rather readily.

Foundations have a bearing upon the selection of the site. If new equipment is being installed, the foundation problem will probably be much less than with old equipment. Most manufacturers are giving more and more attention to balance of reciprocating machines, and care should be exercised when compressor equipment is being selected, to favor better balance. The site probably will be required to furnish a better foundation if gas engines are used for prime movers than if electric motors are supplied.

Header Location—After the site has been selected the arrangement of plant and equipment must be planned. All of these items are so interrelated that it will be impossible to start with one factor and progress logically through the sequence of items involved in the plant. Piping will probably be the major factor at the site, the location of the headers being all important. This location must be planned with considerable care. It must be convenient to the area of gas supply, and provide ample room for the necessary scrubbers. Convenience seems to dictate running the headers parallel to the long axis of the plant (Figure 1). If pulsation dampeners are used, the tuned length

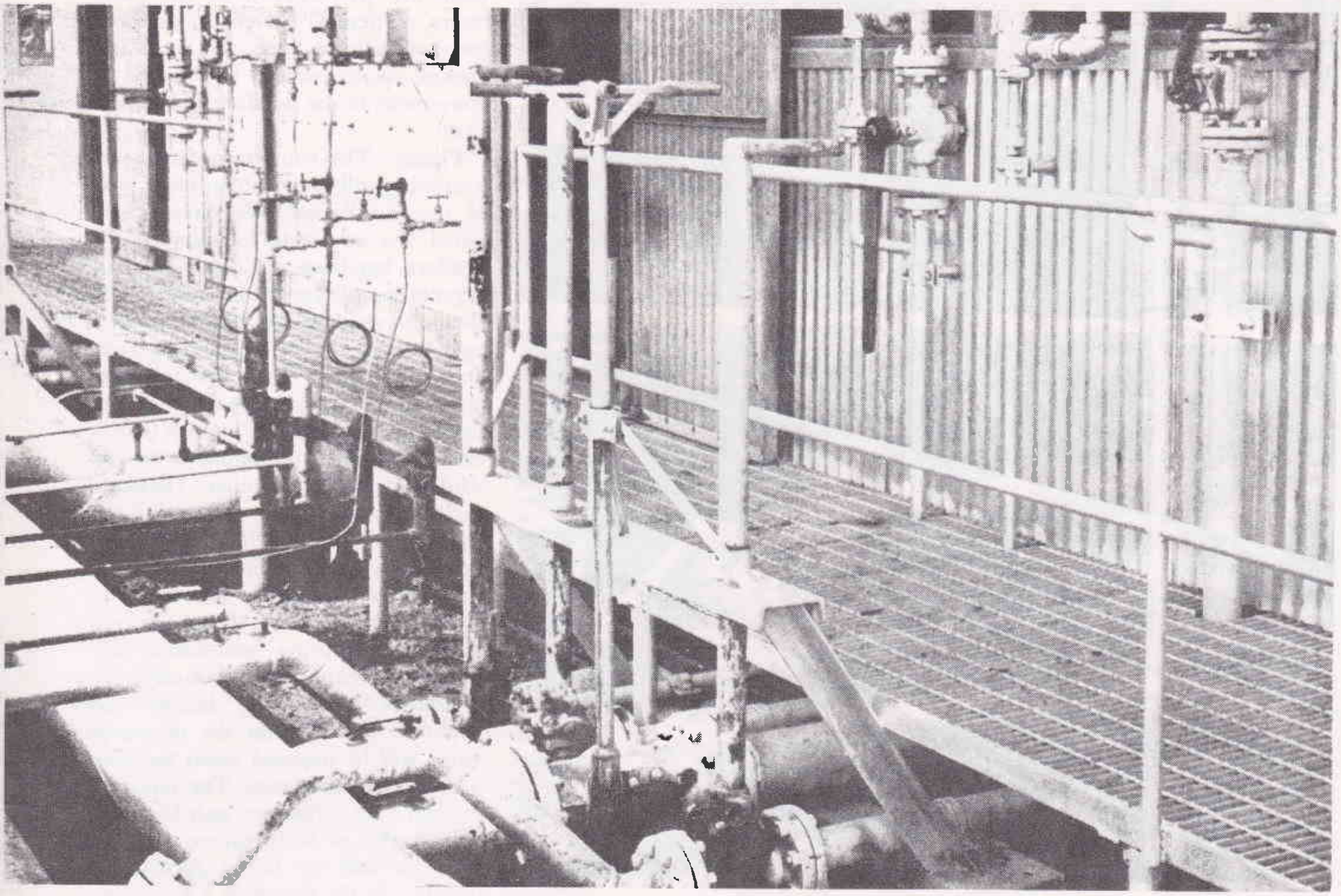

FIGURE 2—Longitudinal walkway for access to stop valves.

of the resistance tubes will frequently dictate a minimum distance out from the plant to the headers.

The gas headers were usually placed in a trench in the earlier plants, but more recently either the concrete trench has been eliminated entirely, or placed above grade so that drainage can be obtained. Height of the plant floor is thus dictated by the location of the headers, or by drainage of the subfloor under the gas surge chambers which are usually placed directly under the compressor cylinders, inasmuch as short connections between the cylinders and surge chambers are essential.

Arrangement of Units—The arrangement of units within the building is often the subject of considerable controversy. Engine-driven, direct-connected units are usually found to be most convenient when they are arranged with their crankshafts forming a single line, end to end. Occasionally to save space machines are set in a double row with the engine ends together, back to back. The compressor cylinders face the outside of the plant, and headers are run on both sides of the building. Operators usually regret this decision because the runway between engine sides is consequently narrow, dark and hot, and maintenance is difficult.

Some operators prefer to install their units crosswise of the plant, carrying the compressor piping to the building side in a trench. This arrangement was designed to permit crankshaft removal through the ends of the crankcase. However, such a design is questionable as shafts are never removed in many instances, and if this is ever necessary, such infrequent engine dismantling will not pay for the many other inconveniences. Where electric driven opposed compressors are used, with cylinders on both sides of the frame, setting the units with their crankshafts crosswise of the plant will be the preferred method. The piping from the cylinders on both sides of the unit can then be carried in basement trenches to the headers on one side of the building only.

Regardless of how the units are arranged, however, the principal item to remember is not to attempt to make a saving on the installation by skimping on the floor area, or getting the units too close together. On an eight-cylinder engine with a rail overhead, four pistons must be set on the space at each end of the engine during overhaul and worked upon, hence it will pay good dividends to allow plenty of room between units. Also, plenty of space should be allowed both in back of the machine for crankcase work, and on the compressor side for pulling pistons and rods.

Thought should be given during layout time for the possible addition of future units, whether or not it

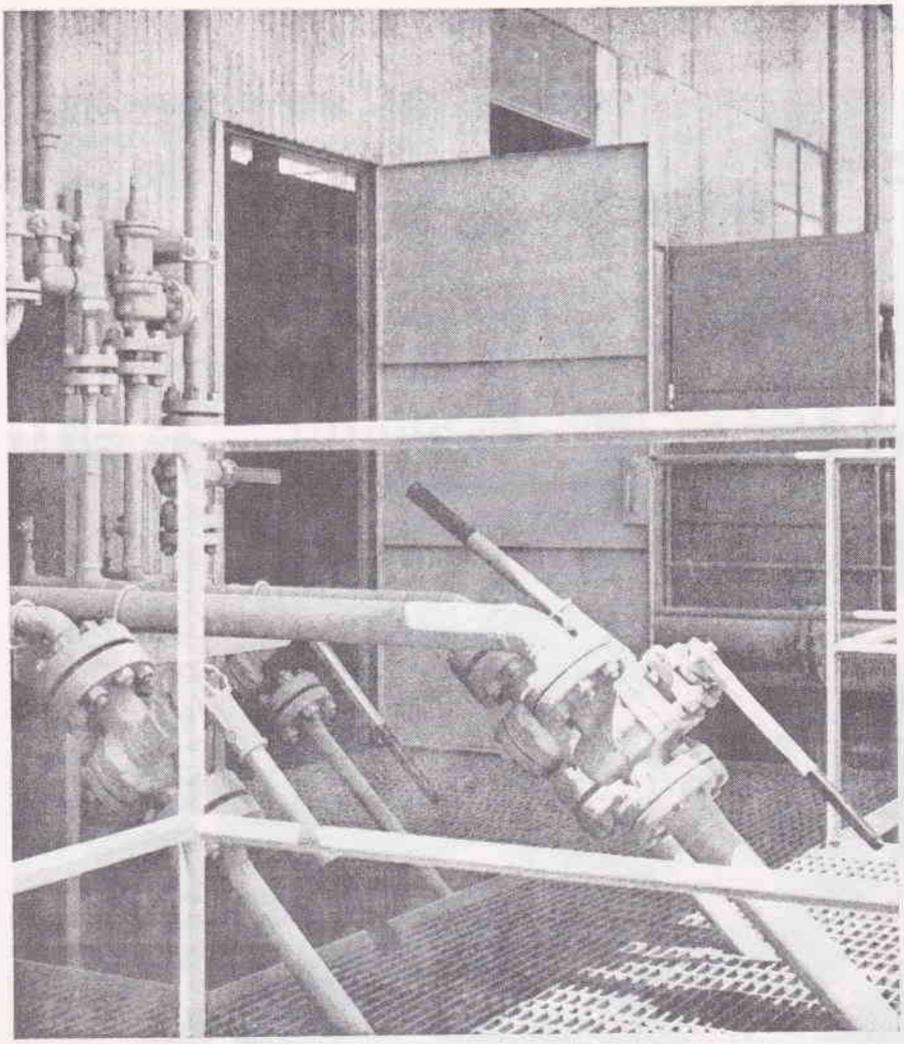

FIGURE 3—Novel method of installing high pressure stop valves. Safety valves securely anchored and very accessible.

seems unlikely at the outset. This must include space for the headers also. If auxiliaries are located in the building they must be either at the end where no expansion is contemplated, or in the center of the plant for easier access.

Compressor Building—Occasionally, in mild climates, someone will question the necessity of a building. This structure should never be omitted since, although rains may be infrequent, the water will find its way into the crankcases and distance pieces, and both rains and winds will seriously hamper the proper attention necessary for good operation and maintenance. Then, too, it is essential that a rail for a crane be placed over the compressor cylinders and another over the engine heads. Or a bridge crane can straddle the full width of the plant for only a slight increase in cost, and with much more usefulness. These can be incorporated into the structure. Care should be taken to see that the craneways extend through the end of the building for loading and unloading compressor equipment onto trucks.

The building should be well provided with windows for good lighting, as this will pay out in better maintenance and operation. Enough of the windows should open so that good ventilation is possible. Doors should be provided opposite each compressor for quick access to the valves located outside at the sides of the walkway.

It will be found much more satisfactory and convenient to make the floor of open grating wherever a trench or basement is underneath, rather than of floor plate. The grating is so much lighter that one man can remove it with ease, whereas floor plate is a back-breaker. Further, it has been found that the possibility of constant observation of surge chambers and piping underneath is quite advantageous. When this area is visible it never accumulates trash. The safety men should be called in to spot their fire protection in the early stages of design. An office, or room for records to be kept, also wash and change rooms, should be located as convenient to the building site as possible.

Piping—The compressor and service piping will constitute a major part of the installation. We have briefly noted the advisable location of the headers, but have not discussed their arrangement. Considerable study should be made of the header anchorages to care for stresses and thermal expansion of the lines. For this purpose piping contractors have developed pipe "hold downs" which permit either side or longitudinal deflection while restraining the other. These are used in conjunction with solid anchorages. Where side deflection of headers by lead-out lines will permit, solid anchorages can be installed near the center of each longitudinal header, and spring hold-downs may be used on either side. It is often advisable to "cold spring" headers and lead-out lines so that the minimum stress will be imposed when the lines are up to temperature. The pipe will expand .001 inches per inch for every 150 degrees of temperature rise, and this can add up to an appreciable distance in the length of a long compressor house. We note these items merely to call the reader's attention to the fact that this is a problem for an expert, and due recognition should be given to it.

The stop valves are normally placed in the lead-in and lead-out lines between the headers and the cylinder surge chambers, and these valves will usually be found to be most convenient when arranged close to the building wall. For access to them, a walkway is either run for the length of the building (Figure 2), or a platform and steps are provided opposite each compressor, with the valve extension handles being brought to the walkway railing. The safety valves should be located at an accessible spot close to the building so that their discharge lines can be carried individually above the building eaves. These vent lines must be quite firmly anchored if the valve is in high pressure service (Figure 3). When it is not possible to discharge the "pops"

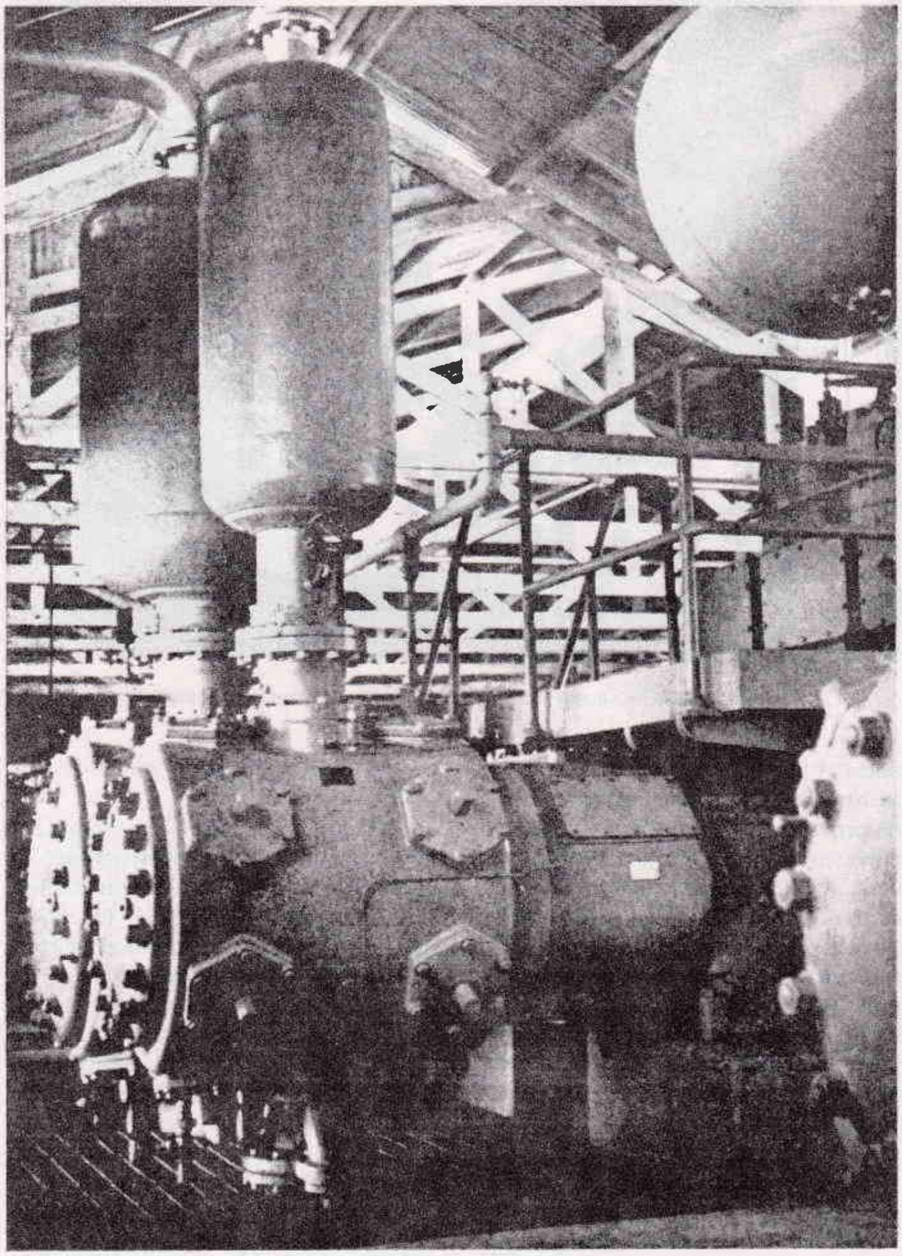

FIGURE 4—Vertical suction surge chambers.

in this manner, and vents must go to a common vent line, it must be amply sized for the worst possible condition.

Another point to watch in the compressor piping is the cylinder suction connections. Suction flanges are often rated at a lower working pressure than the discharge flanges on any cylinder, hence, with the compressor down, the suction stop valve closed, and the discharge open, that section of intake line between the stop valve and the cylinder can conceivably be subjected to discharge pressure by leakage of one suction and one discharge compressor valve. To protect against this possibility, a safety valve must be installed on the suction line, or the suction flange and piping must be adequate to safely handle the discharge pressure.

The point of having suction headers and piping extremely clean cannot be over-emphasized, as it will make so much difference in the amount of compressor valve trouble which will be encountered. Anyone who has experienced the incessant valve trouble caused by dirty lines will agree with the foregoing statement. We have found it wise to go to the extent of sand blasting all of the pipe used in fabricating headers, lead-in lines and surge chambers, and sealing them to protect against corrosion during construction. After the pipe work is completed, and just prior to startup, all lines should be blown down with 100 psig air in an adequate volume to give a good blow, with piping disconnected at the outer ends. If any of the piping or surge chambers have lain idle through inclement weather, they should be treated with great suspicion and thoroughly cleaned before using.

Heavy hardware cloth should be used to back up screen wire and made into steel ringed plates which can be inserted in the suction lines between flanges near the cylinders. In addition, screens should be placed over the compressor suction valves to remove all small particles. These screens should be removed after 2 or 3 weeks of operation.

Pulsation—Pulsation filters are commonly used today on suction and discharge lines adjacent to the compressor cylinders for two reasons: to remove disturbing pulsations from the plant pipe lines, and to prevent starvation and hence capacity loss in the case of the suction lines, and excessive pressures, resulting in high horsepower and some capacity loss, in the case of the discharge lines. Such filters are offered by several concerns and are readily obtainable. Some operators have been lucky about their pipe line pulsations and are still not sold on the advisability of installing filters. On high-pressure installations where the gas density is high, there seems to be no alternative, and the early installations without them usually experienced considerable trouble at the welded connections to the surge chambers.

If pulsation filters are to be omitted on either high or low pressure lines, then adequate surge chambers must be provided on both the suction and the discharge. The surge chambers will tend to prevent starvation of the cylinder on the suction side. There have been instances where cylinder capacity has been reduced by as much as 25 percent by operating without a suction surge chamber. Similarly, the discharge surge chamber will prevent excessive momentary discharge pressures which can result in severe overloads and also somewhat reduce cylinder capacities. These surge chambers may serve several cylinders on the same machine. Their capacity must not be less than seven times the cylinder displacement (bore × stroke) for every cylinder served. They must be relatively close to the cylinder which they serve, else, in combination with the "gas jacket" around the cylinder, and the "too long" connecting line, they may constitute a filter themselves which may be improperly tuned, and some disturbing pulsations may result. However, don't let the fear of this unusual situation tend to favor omitting the surge chambers. Their omission will almost invariably cause shortage or overload conditions, even though it may not be recognized as such.

Discharge surge chambers normally are located on sleepers under the floor. Most high-pressure suction surge chambers are in a similar location for convenience in connecting to cylinder manifolds. Low-pressure suction surge chambers usually are placed across the top of the cylinders so that the cylinder connections can be extremely short. This also permits easy removal. Suction surge chambers may be placed vertically where space is limited (Figure 4). All surge chambers should be equipped with drains and pressure taps for checking pressure losses. Last but not least—don't place them where they cannot be removed for inspection or possible repair!

Scrubbers should be placed on all suction lines. These vessels should be equipped with a separating element to remove liquid, and should not be merely a "knockout drum." The latter does not do a satisfactory job of separating entrained vapor. Scrubbers should be equipped with a gage glass, automatic drain, high level alarm and a shut down device to shut the compressors down on high liquid level. Scrubbers should be located at an accessible spot convenient to the gas headers.

Unless the engines are supercharged, it may be wise to silence the air intakes to the scavenging cylinders of two cycle engines if noise is objectionable. This can be done with pulsation filters. It will be wise to locate the air intakes on such units at some distance from the building wall.

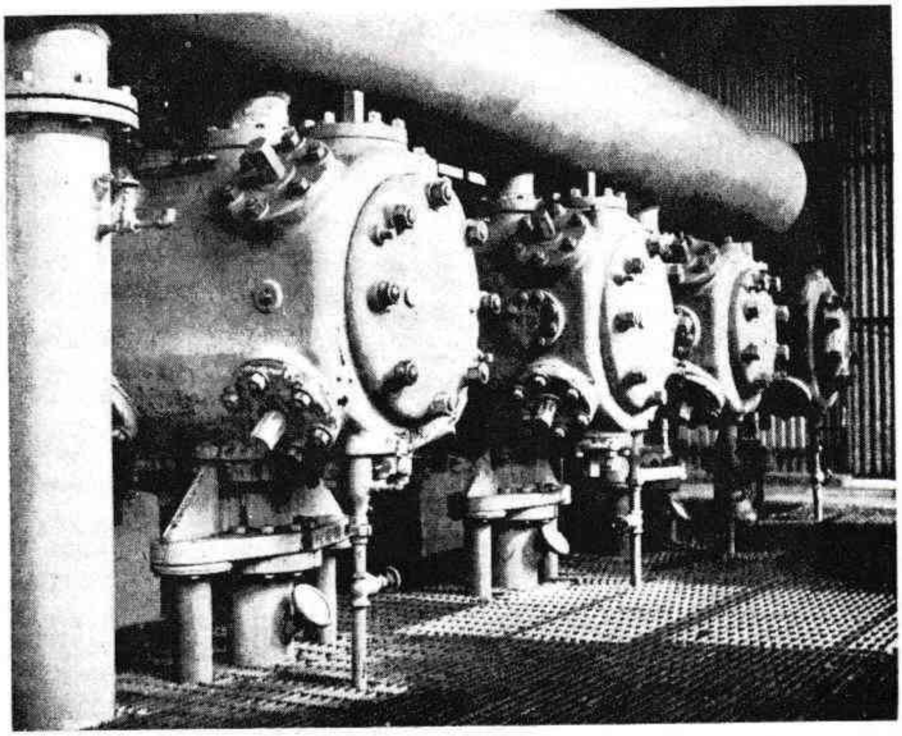

FIGURE 5—Special flange fitting below cylinders to permit cold supports.

Unloading—On each compressor discharge (or for several cylinders in parallel on the same machine) a vent to atmosphere or a vent line should be provided for unloading. The safety valve usually discharges to this same vent. Unless some unusual situation demands that no gas be lost from the system, this type of unloading has some advantages over the bypass system. If, for any reason it cannot be used, then a bypass line and valve must connect the discharge to the intake, inside of the stop valves. Either system permits unloading during the startup and shutdown operations of the compressor.

Grout—In installing the compressor, the manufacturer will supply instructions for grouting and levelling, and they need not be repeated here. We would note, however, that if the crankcase is equipped with an oil pan or sump which projects below the rest of the crankcase, it is usually better not to grout around this pan. This can be worked out either by installing a piece of fiber board between the pan and foundation around the top rim, or placing a piece of fire hose around this section which is pressured with water during the grouting operation, then removed. The more recent practice of following the wet grout with a dry grout to prevent shrinkage has worked out excellently. Any wedges placed for levelling should be removed after 24 hours, else the frame will ride the wedges later on.

Cylinder Supports—The manufacturer will furnish full instructions on assembling the machine and installing cylinders. However, close attention should be paid to the method of supporting the compressor cylinders. It is imperative that the hot discharge line should not go downward to a rigid base, or base ell, which will result in an upward thrust on the cylinder when the line heats up. Cylinders are forced out of line, and occasionally distance pieces are cracked by such methods of installation. In earlier periods, the base ell was the standard method of supporting cylinders. They were usually held up by jackscrews, and manufacturer's service men were careful not to run these jackscrews down until after the discharge line had come up to temperature. Later, however, when the unit was down and cold, the fact that the jackscrews did not reach the support plate was usually noticed by some attentive mechanic, and the damage was done.

Customer pressure has caused the manufacturers to correct this trouble in various ways. Some have put lugs on the sides of the cylinders to which cast legs fit which support the cylin-

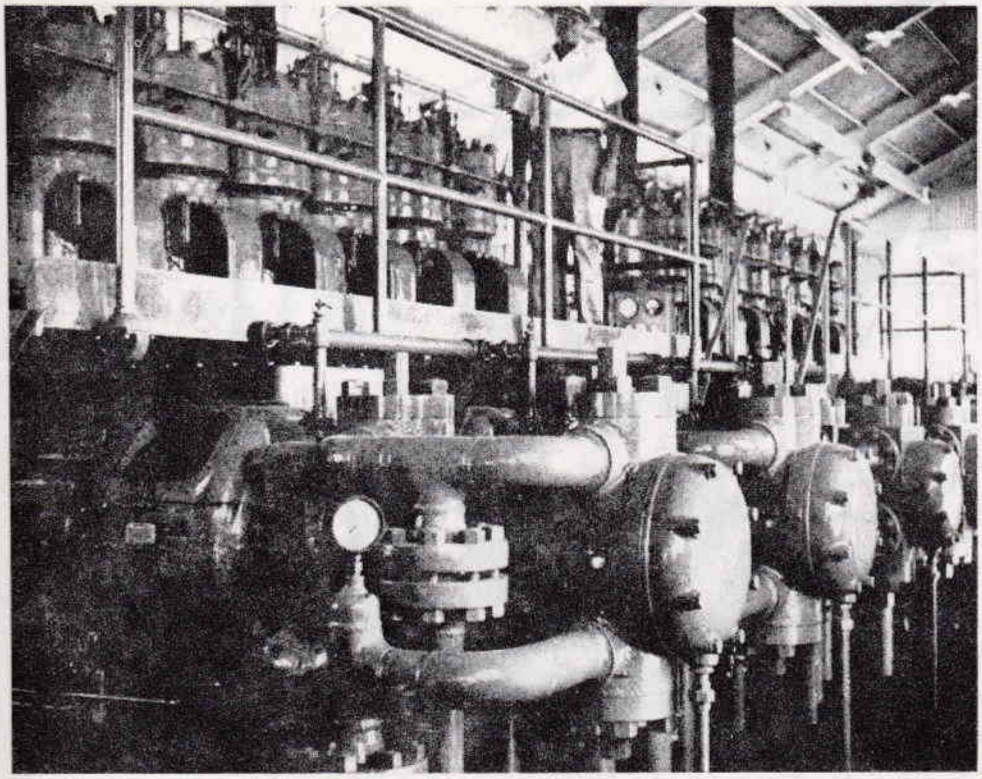

FIGURE 6—Method of locating thermometer in discharge line.

der, straddling the discharge trench. Others have made a wide flange fitting for the discharge, the projections on which are supported by 3-inch or 4-inch pipes which serve no other purpose but to support the cylinder, and are therefore always at room temperature (Figure 5). These methods serve the purpose excellently, but are completely nullified if the contractor fails to recognize the principles involved, and runs the discharge lines directly downward into base ells or surge chambers or any fittings which are rigid. Discharge lines should drop down into an ell which floats free and directs the line into a surge chamber. The ell can then bend under thermal stresses and relieve the cylinder of this strain. Within the last month we had occasion to look over a new installation where the contractor had defeated the provisions for good alignment provided by "cold supports" by installing rigid discharge spools directly into firmly anchored surge chambers directly below the cylinders on more than half of the units.

While on the subject of installation we would warn against the use of an impact wrench in running nuts. The impact wrench is very convenient when dismantling machines, but should never be used for setting up, due to the ease with which bolts and studs are overstressed. The new torque-limiting air wrenches are a different matter, and bid fair to combine a number of desirable qualities into one. Overstressing nuts has caused so much distortion in the past that we have found it wise to supply 1000 foot-pound torque wrenches for assembly of machine parts.

Jacket Water—In arranging to "service pipe" the jacket water, care should be taken to insure that the outlet water flows upward to the outlet header. If the outlet line from any cylinder drops downward it will form an air or gas trap which usually results in blocking off that cylinder, and diverting its cooling water through the others. If it becomes necessary to drop lines or headers downward prior to reaching the surge drum, then air expelling traps must be installed at all of these points to remove the gas which occurs whenever a gasket leaks, or the air which is sucked in by the centrifugal pump or trapped in the system after any dismantling. We have found it most convenient to install a vertical standpipe or surge drum of 20- or 24-inch pipe sufficiently high to collect all jacket waters, and provide an ample head for the centrifugal jacket-water pump. A permanent ladder is provided for this open top tank, and all outlet waters are directed into it at any elevation. The operators make a daily routine of climbing the ladder and checking the bubbles. This normally permits spotting gasket leaks and correcting them before they become serious. In piping the compressors it is good practice to install an industrial thermometer at each water outlet, both to be able to regulate the volume throughput, and to indicate trouble due to overheating.

Compressor manufacturers usually overestimate the heat rejection rate of the cylinder to the jacket water. Rates of 500 Btu per bhp-hr., and 175 times the ratio of compression are two of the most common factors supplied for this heat rejection. In our observation, both are considerably higher than any cylinder rejection rate we have noticed. In fact, it would seem that 100 times the compression ratio would be ample and much closer to the expected rate than the figures noted above. A few of the gas transmission line companies have installed cylinders with no water jackets at all, and put the major portion of the heat into the gas stream, radiating a small portion to the air. Such cylinders fit in well with their program, for normally the ratio of compressions on such stations is below 3. We know of instances where operators have filled the water jackets with oil and left the top openings unplugged. This equalizes the jacket temperature around the cylinder, but forces the rejected heat into the gas being compressed. It causes a small increase in the horsepower of compression, but could make the occasional gasket leaks difficult to find. Such practices are usually confined to ratios of compression below 3.

In water cooling the compressor cylinders, a temperature rise of 10 degrees will be found quite satisfactory. The temperature level may be kept around 120 to 130 degrees, and even higher if the gas is particularly wet, in order to prevent condensation in the cylinder and consequent high ring and wall wear. Such temperatures are quite favorable to dry air cooling of the jacket water, particularly in high ambient temperatures.

Dial Thermometer—During installation, a minimum of instrumentation will include several items which are often omitted. Foremost of these is a heavy duty dial thermometer in each compressor discharge line, adjacent to the cylinder (Figure 6). This is for the purpose of supplying a ready index of the performance of the compressor valves and rings. Those who have not used such indicating thermometers have no idea of their usefulness in pointing out the situation when valves are going bad or valve gaskets are beginning to leak. This is particularly valuable on high pressure cylinders where valve or gasket leaks for only a short period of time either damage the valve beyond repair, or make the repair difficult. Similarly, when cylinder gasket recesses become slotted by high-pressure gas blowing through, it is a long and tedious job resurfacing them. A slight increase in temperature indicates trouble long before such trouble can be detected by the feel of the valve covers, and operators are usually enthusiastic about such a "trouble detector." A heavy duty dial thermometer without gears should be used to prevent destruction from vibration. Such equipment stands up well in this service.

The metallic packer should also be equipped with a thermometer to indicate its performance. If the manufacturer has not done so, the packer flange should be drilled and tapped, radially, for the installation of either a dial thermometer, or the bulb for a remote indicating instrument. The latter is required if the gas being compressed is poisonous, requiring the distance piece to be normally closed. This makes a very neat and convenient installation, as the temperature gages may all be mounted together on a single instrument panel, and any packer trouble will be pointed out rather forcefully. There have been instances where we have placed two such tapped wells in the packer flange and have placed a thermally-operated engine shut-down device in the second one. Such installations are for plants which operate unattended and we do not care to burn up the packer and ruin the piston rod in the event of packer trouble. We have seen instances where other operators have provided such protection in attended plants.

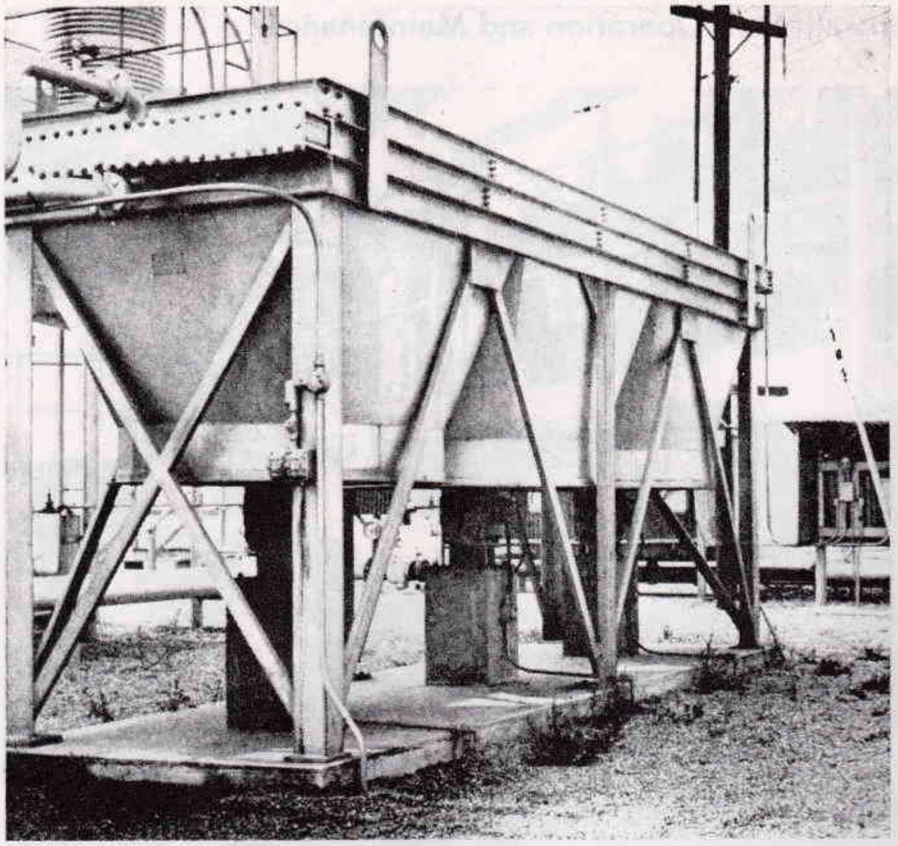

FIGURE 7—Single section three-fan unit for cooling jacket water.

Packers—While on the subject of packers, it has proven quite profitable to install a grease gun fitting and check valve on a tee fitting to the oil line leading to the inner lubrication point on the packer. This is for the purpose of breaking-in a new packer. During startup the grease gun can be filled with a heavy oil, and supplementary injections should be made frequently until both the thermometer in the packer flange and lack of audible blow-by indicate that the packer is seating properly.

Double acting wipers should always be used around the piston rod. Originally, single acting wipers were installed with the thought of stripping the crankcase oil from the rod which had splashed onto it, and preventing its loss to the distance piece. Occasionally, however, a heavier oil than is used in the crankcase, or an oil compounded with tallow is used on the rod, and when this oil is carried into the crankcase it results in contamination, causing frequent crankcase changes. On an electrically-driven machine, crankcase oil changes will be very infrequent unless contamination occurs.

Unless a poisonous gas is being handled, the distance piece covers should be left off on the "operating" side, so that operators can see the packer and its thermometer at a glance, and feel the rod. Recently the author had occasion to work with a gas dangerously high in hydrogen sulfide, and the covers were left on. This method of operation proved so unsatisfactory that the covers were left attached by one bolt, but were rolled out of the way. They could be rolled back into place if necessary in a moment. Any slight leakage was carried away by the vent lines which were tied into a vacuum collecting system. This method of operation proved quite satisfactory, and the thermometers were an excellent index for the operation of the packers.

Lubrication—When ordering lubricators it is possible to have them broken up into several compartments, usually at no extra cost. This may be found of great convenience at a later date if it is advantageous to use oils of a different specification for the various lubrication points.

The lube oil should be filtered. If the compressor is electric driven and not a part of an engine driven machine, then an edge type filter will usually fulfill this requirement. The crankcase oil will usually require a little cooling, and for this the manufacturer will supply a small shell and tube heat exchanger. This may not always be economical to use, particularly if the cooling is to be furnished by dry air coolers. It may well require the jacket water to be cooled cooler than necessary in order to provide the temperature gradient to bring the oil temperature down to the required level, thus calling for more heat exchange surface in the jacket cooler than is necessary.

Therefore the possibility of cooling the oil directly in extended surface coils in the dry air cooler should be explored. This is far more attractive in a direct connected engine driven machine than it is in a compressor only, as there is so much more heat to extract in the former. Crankcase oil temperatures in the order of 150-160 degrees are usually quite acceptable. High oil temperatures are to be avoided principally because of bearing fatigue, the strength of the babbitt decreasing at the fourth power of the temperature. This usually becomes a temperature limiting factor ahead of any destruction to the lubricating oil.

A low oil pressure shutdown switch should always be used to save the bearings and sometimes the shaft in the event of an oil pump failure.

Intercooling—Gas intercooling can be done in shell and tube exchangers near the compressors, or in atmospheric sections placed in a cooling tower, or in dry air coolers with extended surface tubing where the air is circulated by fans. The choice depends upon the individual situation, the availability of cold water, or unused space in a cooling tower.

Dry air coolers are becoming more and more widely used for the ease of installation and the low maintenance involved, also for the simplicity of temperature control. Another factor to be remembered is that they give their full salvage value when it is desired to move them to a new location. Vertical shaft fans directing the flow of air upward will be found more satisfactory, as these are insensitive to wind direction. A number of small fans in parallel will be found preferrable to one large fan (Figure 7), as this will permit automatic startup and shutdown of one or more of the fans by providing automatic thermal control on the outlet temperature of the gas.

Inasmuch as design factors must provide for the highest ambient temperature expected, such control will prove to be quite economical and satisfactory. If the installation is to be near the ocean, copper fins will be preferred over aluminum. The jacket water and lube oil sections can comprise one unit with the oil sections placed beneath or beside the water sections. If the sections have been designed properly using the proper factors, the outlet temperatures of the two will follow each other roughly together as the ambient goes up or down, hence the control point for shutting down the fan motors may be placed in the water outlets, and the oil should never wander too far from the rather wide limits which are acceptable.

OPERATION

Starting—For the initial startup of the plant the procedure is, of course, quite different and more cautious than subsequent ones. After making sure that the compressor crankcase is extremely clean, fill it with lube oil, usually an SAE 30 grade, and pump oil to the pressure system with the hand pump which the manufacturer usually provides on such machines, both to make sure that the bearings are flooded and that the lines are free of air. Through the valve holes and with the piston backed out of the way, watch the oil holes in the cylinder walls while the lubricator is being hand cranked to insure that the oil comes freely and that all the air has been expelled from the lines. The assembly shoulld be once again checked over to make sure that the compressor pistons have been spaced properly (usually with ⅓ of the clearance on the frame end and ⅔ on the head end to provide for expansion of the rod).

Be positive that the compressor valves have been installed properly. Many compressor valves are interchangeable between suction and discharge, hence it is easy for the helper to get the valve in backwards. As there is no safety valve which will protect the cylinder from reversed discharge valves, a nice longitudinal crack usually reveals this error. Check the inlet valve screens and make sure that the valve covers have been pulled down evenly (a torque wrench is preferred) with the jackscrew loose when this is done. Then run the jackscrew down firmly onto the crab or guard to hold the valve tight onto its seat.

Check all the lines through the plant to insure against any closed valves or flange blinds, start the jacket water through the cylinder, check the safety shut-down and overspeed trip for their proper position, make sure that suction and discharge stop valves are closed and that the bleeder or bypass is open, and the unit should be ready for starting.

The unit should be run for not over two minutes, and careful attention should be paid to listening for knocks. During this period the lubricator, set to a heavy feed, should be checked. The unit should then be shut down, the covers removed and all the bearings carefully checked for excessive temperature. This procedure should be repeated for 5- then 10- then 30-minute intervals, after which, if nothing abnormal is apparent, the unit may be idled for several hours. During this time the packers should be watched and frequent gun injections of oil pumped into the packers. In starts after the first one, the unit should be idled for around ten minutes to permit the oil and parts to attain a running temperature before the load is applied. Whether it is initial or subsequent startups, the load is applied to the high pressure cylinders first in most multi-stage plants, dropping down in pressure to the low cylinders which puts all stages on the line before the main stream is brought into the system.

If the cylinder is equipped with a bleeder on the discharge, the load is applied by first cracking the discharge valve until the gas backing through it can be heard whistling through the vent line. It is then safe to close the vent and immediately open the discharge valve fully. If a bypass is used, it is closed before the discharge stop valve is opened to keep the intake safety valve from popping. If the working pressure of the suction piping adjacent to the cylinder is higher than the discharge pressure, then the opening of the discharge may be completed, if desired, before the bypass is closed. The load is applied by gradually opening the intake valve.

In this connection it is often wise to compute the rod load when the cylinder is pumping against discharge pressure with atmospheric pressure on the intake. In rare instances this will be above the allowable, and, although this is usually not something to be too concerned about, due to the generous factors of safety provided by the manufacturer, the condition should not be permitted to exist for any period longer than is necessary.

For the compressor alone, or if it is electric motor driven, we see no advantage to be gained by applying the load a little at a time, such as is recommended for engine driven machines. Full load can be placed on a new cylinder whenever the running gear has shown itself to be in operable shape and has worn itself in for a few hours. When the unit is engine driven, the startup of a new machine should be entirely different beyond the points covered in the foregoing, for the sake of the new engine. Although this is a treatise on compressors, and not engines, we shall note such a startup briefly, inasmuch as the two are so closely interrelated. It has been the custom to start such engine-driven machines, after their running gear was proved, by operating several hours at ¼ load, several hours at ½, several more at ¾ load, then to apply the full load. This is principally for the benefit of seating the engine piston rings. Discussions with ring engineers have indicated that they would prefer to see such rings seated by short applications of full load, rather than continued applications of part load. For example, two 10-minute applications of full load with 15-minute intervals between for cooling, followed by two 20-minute applications of load interspersed with the same quarter hour idle intervals, and so on, seem to be preferred. Actual trial of this type of break in for the engine piston rings has worked out quite well.

Stopping—In shutting down, the cycle is reversed. The suction stop valve is closed first, followed by the closing of the discharge, as simultaneously as possible with the opening of the vent. If the two cannot be worked together, completely close the discharge first, then open the vent. In the event the suction was not closed properly, the discharge safety valve will pop. If a bypass is used instead of a vent the same procedure can be used, substituting the bypass instead of the vent in the above text.

If the suction piping and flanges are designed to stand the discharge pressure, then the bypass may be opened at will after the suction is closed and before the discharge is closed. If, however, the suction piping is rated at a lower working pressure and is protected with a safety valve, then the bypass should not be opened until the discharge stop valve is closed, else the suction safety valve will open. Although there is nothing wrong with this safety valve opening, it is a nuisance.

If the compressor is electric motor driven, the machine may be shut down as soon as the load is removed. If engine driven, the engine should be allowed to run idle for at least 10 minutes after the load is removed so that the parts affected by combustion may readjust their temperatures gradually.

Cylinder Oil—Cylinder and piston rod lubrication is usually no problem if the gas is dry and pressures are nominal. A good 30 grade oil usually will suffice. Oil additives seem to be of no particular advantage to the compressor cylinder or packer. Wet gases or high pressures demand different treatment. In some cases the use of oils of around 60 grade will be all that is necessary to take care of this situation. Steam cylinder oils, compounded with tallow, make an excellent lubricant for such purposes, but their liabilities usually result in ultimately removing them from service. Any slight failure of the double acting wipers permits the compounded oil to be carried into the crankcase, causing the viscosity of the crankcase oil to skyrocket. Further, such compounded oils produce an excessive carbon residue, which is highly undesirable on the discharge valves.

When compressing plant vapors, it is frequently of material advantage to have the condensate uncontaminated with anything which will throw it off color, so that it may be pumped to a white line without having to be rerun. The use of the white viscous oils will permit this to occur with good lubrication, and with considerable saving in steam. The quantity of oil required for cylinder lubrication is surprisingly low. After the rings have seated, cylinders handling dry gases at nominal pressures can usually be cut to around 5 million square feet per gallon feed. We indicate this and subsequent oil figures with great reluctance, for some will find they do well with half this amount, while others could not get along without doubling it. Such figures are only to suggest an order of magnitude. Wet gases and high pressures can demand double this oil rate.

Too much oil will often result in excessive carbon deposit on the discharge valves, particularly if the ratio of compression is on the high side. The adequacy of lubrication can be checked at the outset by frequently removing an intake valve and feeling the cylinder wall for an almost imperceptible oil film. Later it is wise, after a sufficient time has elapsed to make it significant, to insert a micrometer through the valve opening and measure the wear rate of the liner as a check on the lubrication. The new style lubricator tops which are equipped with glycerin or similar visible liquid, through which the lubricant may be pumped under pressure, make the measurement of feed rather difficult. With this type of equipment, the total number of drops per minute for all feeds within one compartment must be counted, and the total oil per day fed from that compartment can be divided in the proportion as the number of drops for any feed bears to the total drops.

Packer lubrication is usually the same as that for the cylinder in quality, though in certain cases, where the oil used in the one location is not the best for the other, we have found it advisable to use different oils for the two, from two different lubricator compartments. This is the unusual case. We have described the "grease gun" fittings on the inboard oil feed previously, and the use of these to lubricate the packer, far in excess of the capacity of the lubricator, will be found highly useful during the break-in period. After break-in, on gases at nominal pressures, lubrication may often be cut as low as 5 million lineal feet per gallon.

Valves—Valves constitute perhaps the major feature of the compressor. The design of the valve must be such that the pressure drop through it will be very small, so called "valve velocities" will be low which means that a sufficient number of valves and the lift area of each must be great enough to keep the velocity of the gas through the valve down within acceptable limits. In reviewing compressor offerings, this is an item which must not be overlooked, and further, it must not be assumed that all manufacturers calculate the valve velocity the same way. This may seem like a surprising statement, but if the reader will divide the cylinder displacement in cubic feet per minute (of both ends of the cylinder) by the lift area of all the suction valves in square feet, he can work out his own valve velocity in lineal feet per minute which good practice likes to limit to 7500.

In order to keep valve velocities (and the resulting losses) down, large cylinders, and many of the pipe line cylinders which compress through low ratios are frequently made with double deck valves. This permits doubling the valve area within the same physical limitations of available cylinder wall area.

Manufacturers normally supply their valves with steel discs, hardened and lapped to a fine finish, with springs to match the lift of the disc and its mass. Independent suppliers pressure the customer to replace the steel discs with those of one of the various varieties of phenolic resin, claiming longer life and lower cost. Sometimes this claim can be met, but all too often the user finds himself installing discs which are materially thicker than those which were originally supplied, resulting in a greatly decreased lift of the valve, and consequent raising of the valve velocity, and a general increase in losses. If thicker discs are used, the valve body must be machined a corresponding amount so that the design lift will be preserved.

In an earlier section we have warned about checking the proper installation of the valves. Earlier compressor valves were made reversible, so that a suction could be made into a discharge readily. This possibility is seldom found in currently made compressors, for the reason that in one of the positions, the center bolt which held the assembly together would fall into the cylinder if it became loose. This disconcerting phenomenon became the source of too many complaints, and designers set to work developing valves on which the center bolts backed away from the cylinder barrel instead of into it, should they come loose. In most cases this was done at the sacrifice of valve interchangeability, which was a doubtful asset in the first place. However, even with this change, it is usally possible to install valves backwards in the cylinder if the mechanic's mind wanders during the operation.

Gaskets—Solid copper gaskets work well on low-pressure cylinders. Where these will not stand the seating pressure, soft steel gaskets seem to be preferred. A few prefer the aluminum gaskets, and these seat well, but have no hoop strength, hence it is necessary to have them made very accurately so that their outside diameters will be as big as the recesses into which they fit, so that the recesses will back them up and make it unnecessary for the gaskets to withstand any pressure in bursting. Gasketing is usually a real problem on high-pressure cylinders, and there are some who are still optimistic enough to believe they can lap the valve into the recess with lapping compound and operate without any gaskets whatever. Slight gasket leaks in high pressure cylinders can quickly slot the recess, causing an appreciable amount of tedious hand work in the repair process, hence it behooves the operators to watch the discharge gas thermometers and feel the valve covers frequently.

Rings—High discharge temperatures can also be caused by rings which are not seating, or which are badly worn. When such trouble is indicated, the valves should be checked first, then, if the trouble is not corrected, the rings should be examined to insure that they are seating, or do not have excessive wear or end gap, or that the grooves are not "veed" in the piston, and that the liner is not scored or out-of-round. At any sign of cylinder trouble it is always wise to make a capacity check of the calculated throughput against the metered throughput. If no external trouble is noticed it is still the wise thing to do to make routine capacity checks at least semi-annually. A "payout" can normally be obtained through proper maintenance if the cylinder is found to deviate from its calculated capacity by more than 10 percent.

High ratios of compression will produce high discharge gas temperatures, such temperatures being readily calculated, or read from charts prepared for this purpose and found in various handbooks. If high ratios of compression are contemplated, the volumetric efficiency of the cylinder should be calculated to insure that the cylinder will not be shut off, that is, that the efficiency will be well above zero. High clearance in cylinders produces a low volumetric efficiency. In every case, the "rod loading" produced by the differential pressure across the piston should be calculated and checked against the maximum allowable loading from the manufacturer's design calculations. Gas discharge temperatures above 300 F. are to be frowned upon, as they accelerate the carbon deposit on the discharge valves, and often build up deposits of carbon in the discharge fittings which eventually results in excessive pressure drop, causing higher horsepower loads and reduced cylinder capacity. Hot cylinder jackets should be regarded with suspicion, as frequently this is an indication of air or gas in the jacket water which is interrupting the water circulation.

Safety Valves—The safety shutdowns should be tested at least monthly to insure that they are always in operable condition. It will be safer to actually make them function as they are supposed to than to develop a synthetic test which leaves any part of their function to conjecture. The safety valves around the cylinder should also be tested at similar intervals. It will be found most convenient to post a schedule for such testing of each item which can be filled out and signed by the responsible party. This will then become a matter of record, which may prove invaluable in the event of accident.

MAINTENANCE

Maintenance on the compressor consists of principally valves, rings and packers. It is wise to know when to anticipate trouble, so that it can be corrected at the operator's convenience, rather than make its own demands upon the operator's inconvenience. It will take some time, keen observation, and good record keeping to work out a maintenance schedule which will permit the maximum safe run on any of the equipment. Compressor operation on dry gases at normal pressures can usually be estimated at a one year run, and may possibly be extended as long as two. High pressure cylinders must usually be checked over in approximately six months. These times are merely to suggest an order of magnitude. Each operator should work out the optimum inspection times for himself for each class of service.

Valve Maintenance—When they need it, valve seats may be ground. If the discs show slight wear, they can in some cases be turned over. Occasionally it may pay to reface valve discs on a magnetic chuck, using a grinder. If this is done, a maximum of around .005-inch is all that should be removed from the disc. Check the valve springs, and replace whenever necessary. Check the completed job with distillate after assembly. Keep a stern watch on the discharge valves for carbon accumulations. This may prove to be the bottleneck which requires first attention.

Ring Maintenance—Check the piston rings for radial wear, which will usually show up by excessive end gap. Three piece segment rings with an internal expander are the type most universally used today. Various mixtures of bronze rings seem to be preferred for quick seating and freedom from scuffing. They may wear more rapidly than iron rings in some instances, but their ease of seating and freedom from scuffing will usually be considered to more than make up for any shorter life experienced. Occasionally phenolic resin rings will work out wonderfully well under difficult circumstances, and at a reduced cost. We have known of such instances. We have also known of instances where, after months of seemingly no wear, they suddenly wore out over night, and the expanders then proceeded to ruin the liners. They are a great help where sour gases are encountered.

The sides of the ring grooves should always be checked carefully for being true, particularly in high-pressure pistons, as this is probably the first point of ring trouble. Grooves have a tendency to "vee" causing blowby and rapid ring wear. Experience has shown it wise to anticipate this trouble and lay in rings which are .025-inch over width. Grooves can be trued up to this oversize, and later on can be turned again in still more increments of .025-inch each. Side clearance in the ring grooves in high pressure cylinders should not exceed .002-inch when a new fit is being made.

Cylinder Liner Maintenance—Cylinder liners should be reconditioned when they are scuffed, out of round, or are worn excessively. If the barrels would wear true, it might be another story, but they invariably wear more at certain points than at others, hence it is wise to recondition when wear reaches the order of around .002-inch per inch of cylinder diameter. After using the practice for many years of reboring to standards of oversize, we have found it better in the overall picture to replace the liner, or spray it back to standard size wherever possible. This permits further use of the piston and does away with the necessity of carrying so many oversizes of piston rings on the shelves. Pistons normally wear much more slowly than liners, and this fortunate circumstance can be made use of. Unfortunately, not all cylinders are lined, and most of them which are not will not stand sufficient cut to install a liner of standard diameter without jeopardizing the maximum working pressure. Many operators are using the metal spray process to good advantage for this work. Pistons may be built up this way, too, although it is not good practice to run a sprayed piston against a sprayed liner. Good success has resulted with spray up to 1000-pound operating pressure, though the higher the pressure the greater the risk. Metal spray work is only as good as the technician who does the work. Our experience has indicated good spray men to be few and far between. If the reader has had sour experiences with spray work, let him regard his workman with a certain amount of skepticism. It is not meant to infer that all spray jobs come out perfect, but in the hands of a good man, better than 90 percent should work out well if the application is a proper one and is well done.

Piston Rod Maintenance—Allowable piston rod wear is an inverse function of pressure, rods at normal pressures seeming to stand a wear of .004 to .006-inch per inch of diameter with good packer operation. Rods hardened to a Rockwell C of around 50 usually show such a low wear rate that the extra cost of the hardening appears well worthwhile. Successful operation of sprayed rods has resulted at pressures up to 1000 pounds. This is found to be a very useful method of reconditioning rods, as it does not call for the rod to be removed from the piston. The removal of the rod is quite a chore in some machines, involving considerable labor, and occasionally resulting in a broken piston, where the interference fit was rather heavy.

The porous nature of the sprayed surface is particularly conducive to good lubrication. Rods can be chromed and ground, but this is a rather expensive procedure. It does produce a good durable surface. When replacing the rod, make sure that the crosshead adjustment is such that the rod runs level through the packing. Make sure, also, that the piston is tight on the rod.

Packer Ring Maintenance—The majority of rings sold for packers are of bronze. Like the piston rings, they seat readily and go to work easily. Various types of phenolic resin rings have been found successful in certain applications, and are quite valuable with sour gases. The manufacturer will specify the packer to be used for the various pressures, the variation occurring in the number of sealing pairs. Most pairs of such rings consist of one radial cut 3-piece ring towards the pressure, and one sealing ring away from the pressure, in the same cage. These sealing rings are usually made either with tangent cuts, in 3 pieces, or they consist of three radial cut rings with 3 small pieces or bridges to seal the cuts, which are usually referred to as "6-piece bridge" rings. From observations of wear characteristics we have a personal preference for the bridge ring, though we appreciate that the trend is the other way. The tangential ring is much easier for the manufacturer to make. Manufacturers differ in the number of breaker rings used, but there should never be less than two in any packer. The garter springs used around the rings are normally made of a stainless material.

The packing should be serviced before the ends of the sealing pairs butt and generous end clearances given. At this time check the wear on the sealing faces of the cages with a straight edge, and grind true if any wear is visible. Replace the sealing pairs of rings when they become thin radially. One side of each ring will be stencilled for mating. Make sure that this stencilled side is not against the seal face of the cage. If packers seal well, their running period may be six months in high pressure service, to perhaps as long as two years in low pressure units. The theory of the breaker rings is quite different from that of the sealing pairs. They are normally made in three radial cut segments, of single thickness sufficient to occupy the whole cage. The designer will provide only enough end clearance so that when the surface against the rod is worn in, the segments will butt. These rings tend to reduce the pressure and duty on the sealing rings, but what is more important, if they are operating properly, they reduce or slow down the back flow of the high pressure gas, trapped in the sealing cages during the discharge stroke, while the suction stroke on the frame end of the cylinder is taking place. When the breaker rings are not functioning properly, the high pressure gas trapped in the sealing cages, in its rush to get back into the cylinder whose pressure has been so suddenly reduced, blow the sealing rings apart momentarily, resulting in rather rapid spring breakage. When packers are found with springs around the sealing rings broken, the packer rings should be checked for adjustment.

Crank Bearing Maintenance—Crank bearings should be checked often enough to insure that the clearance does not become excessive. Even with gas engine pistons working on the same shaft, the much higher loading of the compressor rods will normally make these cranks the most rapidly wearing of all bearings in the machine. In the general case, inspections at six-months intervals is usually worth while. The manufacturers recommendations should be followed in the adjustment of these and the other bearings on the machine. Earlier it was noted that the rod loading limitations should not be exceeded. Many manufacturers will still further curtail the loading when the cylinder is operating single acting, for the sake of the crank bearing which gets no reversal with such operation, and therefore may not be so well lubricated. Single acting limitations should be noted before such operation is contemplated.

Check Cylinder Clearance—When discrepancies between calculated and metered capacities are not corrected by working over compressor valves and rings, it may be found advisable to check the actual cylinder clearance. The manufacturer usually calculates this value, and occasionally he is wrong. Such a measurement is somewhat of a chore, though it can be done with water by sealing all the valves but one intake, and blocking around the piston with grease. The trouble may also lie in the lack of a suction or discharge surge chamber, or the use of one too small. Pulsations can readily cause such discrepancies, and to determine whether or not such effects are troubling the cylinder, the indicator is about the only useful trouble-shooting instrument. The principal difficulty in using the indicator will lie in obtaining an accurate indicator motion. For this it will be best to use an instrument with a reducing motion incorporated within it. Motion from the crosshead should be carried to the indicator with a steel tape, using a sufficiently heavy spring against the tape to insure that there is not the slightest amount of whip in the tape. If necessary, watch the tape with a stroboscope to insure against whip or lost motion. Try to avoid the use of wheels for the same reason. See that the indicator string does not pull at an angle to the tape, and that the tape runs in line with the crosshead. Indicator cards accurately taken will reveal all the pressure disturbances to the cylinder, as well as showing the performance of the rings and valves. In most instances the difficulty in obtaining indicator cards has been amply paid for in accurately locating trouble.

Experience is the priceless ingredient in good compressor operation. It is hoped that the reader may pick up a few points in the foregoing from the experience of others which may prove profitable. ##

Meet the Author

ROBERT S. RIDGWAY has been chiefly concerned with internal combustion and engineering since he graduated from California Institute of Technology in 1924. Since joining Standard Oil Company of California in 1925, he has worked with the Natural Gasoline department, and has held the position of mechanical engineer since 1928. As Standard of California has expanded its natural gas and gasoline activities, Ridgway has been concerned with problems arising in both gas engine and compressor operation, maintenance, installation, and design. Several other articles by Ridgway have been previously published in PETROLEUM REFINER.

For quick estimates . . .

How to Size and Price Axial Compressors

For cat cracker regenerators, axial compressors are being used instead of centrifugals because of comparable first costs and lower operating costs. Here's how to make an estimate

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Allis-Chalmers Mfg. Co., Milwaukee, Wis.

WITHIN THE PAST TWO YEARS several "firsts" were recorded in applications of the axial compressor. Most notable among these firsts is the use of axials for supplying air to catalyst regenerators in refinery catalytic cracking units.

The reasons for the trend to increased axial compressor use in industry are twofold. First of all, the axial, being most suited for higher volume applications, falls in step with the larger volume demands of modern process industries. Secondly, the axial compressor offers many operating advantages over the centrifugal compressor which was more commonly used previously. Principally, the axial offers high efficiency, smaller foundation requirements in weight and space, and more efficient drive selection because of its higher speeds and lower power requirements.

With these basic advantages, the axial should therefore be given consideration in any planned modernization or expansion. The purpose of this discussion is to offer a simple, if preliminary, method of estimating axial compressor sizing, pricing, and performance.

How an Axial is Estimated. The following step by step procedure for sizing an axial is intended to serve for approximate purposes only on applications requiring air with an inlet pressure of 14.7 psia and a temperature of 100° F. To serve as a guide in following this procedure, let's take a random example of a compressor needed to handle 70,000 cfm of air at above mentioned conditions and required to boost the air pressure to 57 psia, such as would be required for a large cat cracker.

Step 1. With the volume and required discharge pressure given, select the compressor size and number of stages from Figure 1.

Example: With 70,000 cfm and a discharge pressure of

Axial Compressor Theory

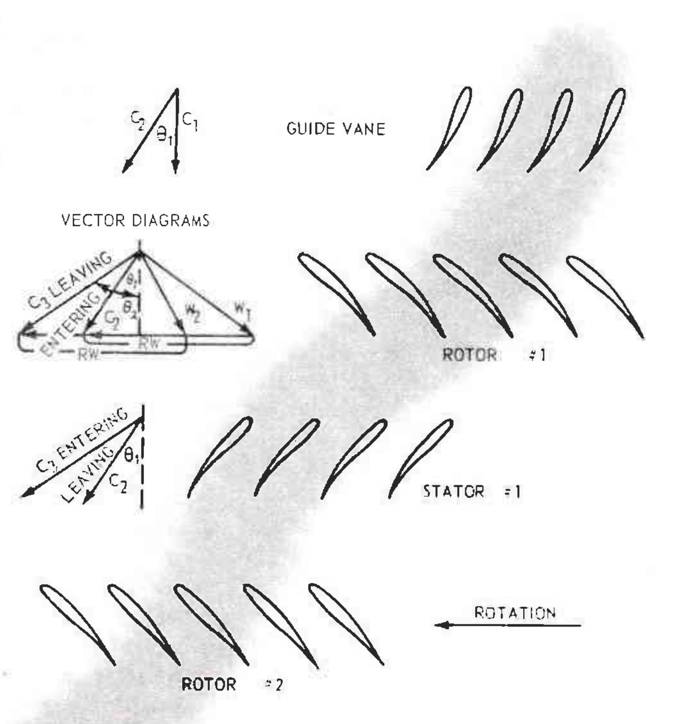
The axial compressor is a dynamic type of machine, identified by the use of moving and stationary blading to accomplish the velocity-pressure conversion for pressure increase. In general, axial compressor design is based on the theory of 50 percent reaction. This means that half of the pressure rise is accomplished in the rotor blade and half in the stator blade.

As air or gas flows through the rotating blades, static pressure and kinetic energy both increase. Each row of stationary blades converts the kinetic energy to pressure, acting as a diffuser for the air or gas flowing out of the preceding row of rotating blades. Also, the stationary blades act as nozzles to guide the air or gas into the next row of rotating blades.

The figure indicates the flow path through an axial compressor. The air or gas enters the stationary guide vane row with an absolute velocity C_1. The guide vane row turns the flow through the angle Θ_1 to an absolute velocity of C_2 for proper entrance to the first row of rotor blades turning with a rotational velocity of RW. This gives a velocity relative to the rotor blades of W_1.

The gas leaves the blade with a relative velocity of W_2 and, again applying the rotational velocity RW, gives an absolute velocity of C_3 leaving the rotor blade and entering the first stator row. The stator row then returns vector C_3 back to vector C_2 in order to give the proper relative velocity entrance into the second row of rotor blades. This cycle is repeated through all the stages. After passing through the final row of stator blades, a stationary straightener vane row removes the whirl initially introduced by the guide vane row by changing vector C_2 back to vector C_1.

Each stage consists, therefore, of one rotating and one stationary row. A nine-stage machine has nine rows of rotor blades and nine rows of stator blades plus the inlet guide vane and straightener vane row. The number of stages is dependent upon the desired pressure rise for a given set of conditions.



AIR PATH THRU AN AXIAL COMPRESSOR

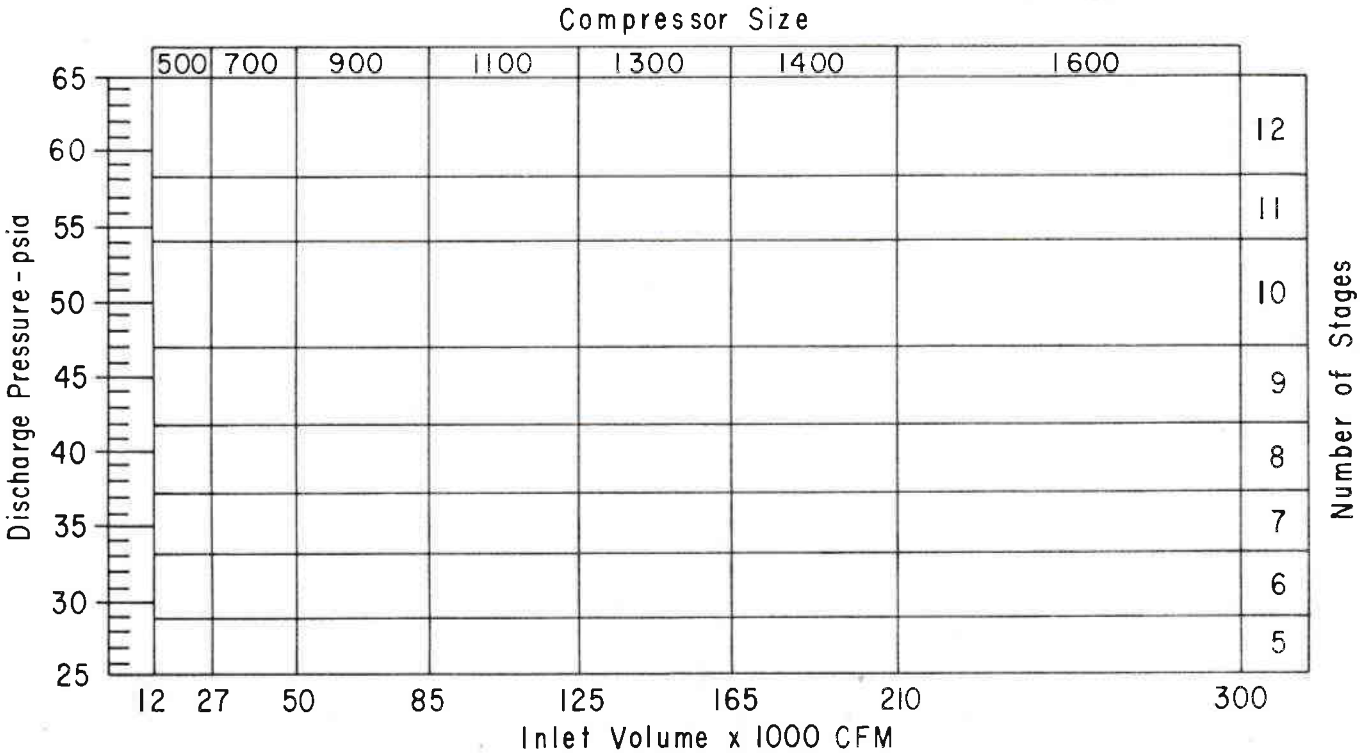
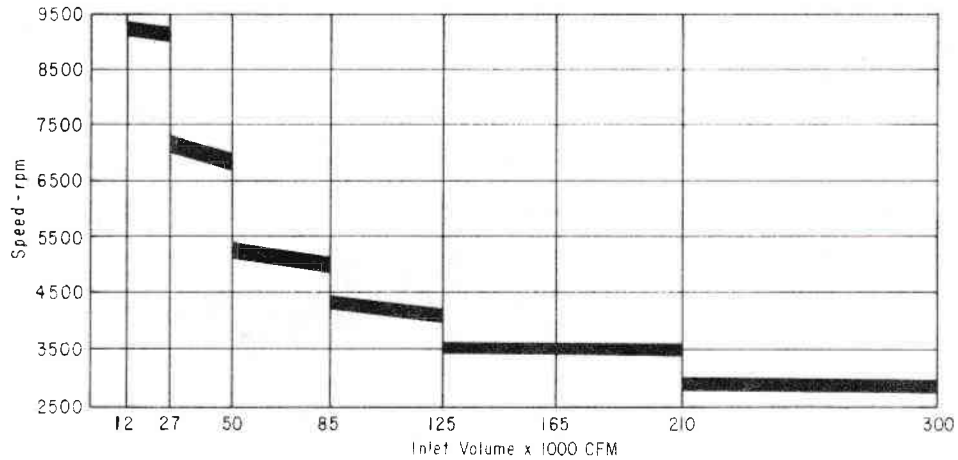


FIGURE 1 (above)—Use this chart to select the compressor size.

FIGURE 2 (right)—Use this chart to find the compressor speed.



57 psia, the compressor size would be 900 with 11 stages.

Step 2. With a known volume, obtain the approximate speed from Figure 2.

Example: With 70,000 cfm, the speed would be between 5,000 and 5,100 rpm.

Step 3. With a known required discharge pressure in psia, obtain the hp required per 100 cfm from Figure 3.

Example: With a discharge pressure of 57 psia required, the hp per 100 cfm would be 12.65 and 12.65 × 700 equals total hp requirement of 8,855.

Step 4. With known compressor selection and number of stages, obtain dimensions from Figure 4.

Example: The 900 compressor with 11 stages would have overall box dimensions of 151 inches long, 118 inches wide (of base) and 96 inches high. Compressor weight is approximately 40,500 pounds.

Step 5. The approximate price of an axial compressor with a motor-gear (Figure 5) or with steam turbine

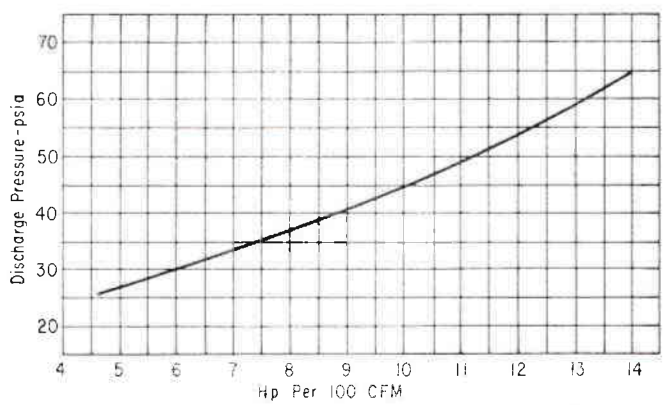
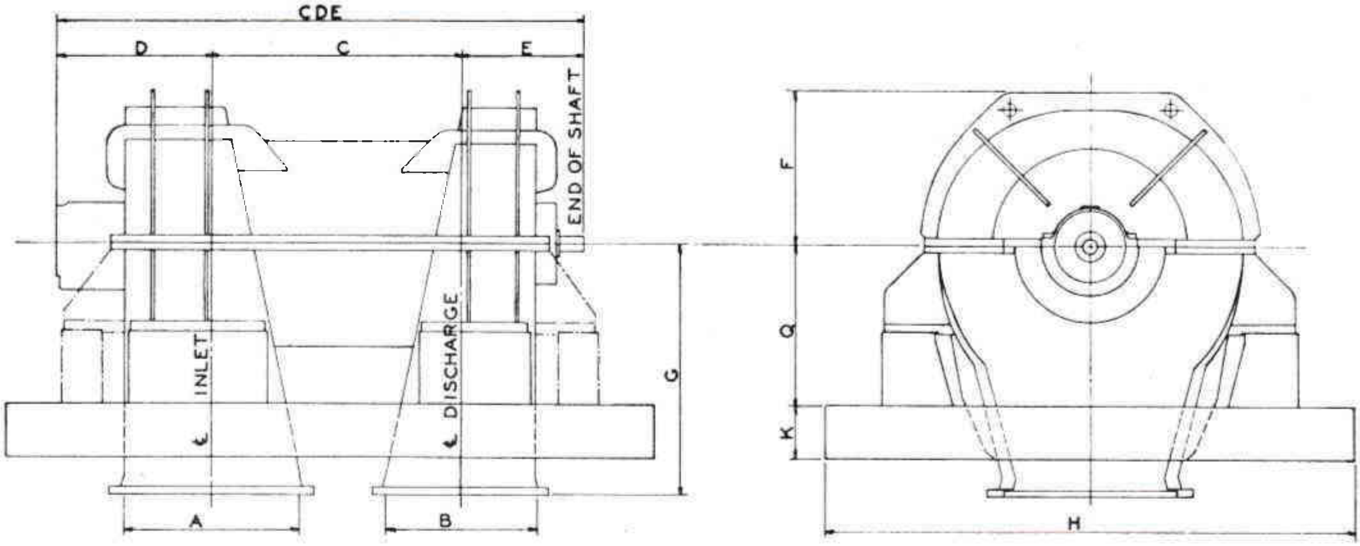


FIGURE 3—Use this chart to find the horsepower required.



TABULATION FOR FIVE STAGE AXIAL COMPRESSOR													FOR EACH ADDITIONAL STAGE	
CASING SIZE	A	B	C	D	E	F	G	H		K	Q	CDE	COMPR. WT. ONLY – LBS.	ADD TO "C" & "CDE"	ADD TO "COMPR. WT."
500+	24	18	20½	32	21	18	33	52		12	36	73½	5000	3	500
700+	36	30	51	32	25	31	60	72		12	27	108	20000	4½	1000
900	48	36	63	32	20	36	60	118		12	36	115	31500	6	1500
900*			72½									124½	40000	7½	2000
1100	60	42	69	32	28½	40	72	124		12	36	129½	34300	6½	1600
1100*			76									136½	46800	7½	3400
1300	66	54	76	32	26	45	78	124		12	36	134	62000	6	2000
1300*						60							69000	7	3000
1400	78	54	76	32	26	45	84	138		12	36	134	62000	8	3000
1700	80	72	104½	32	32	54½	84	192		18	42	168½	125000	8	5000

* MOVEABLE BLADES
\+ END MOUNTED (— — —)

FIGURE 4—Use this table to find the compressor dimensions and weight.

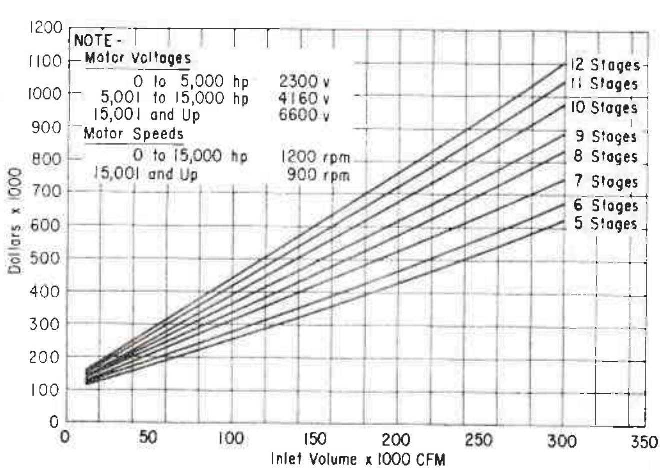


FIGURE 5—Axial compressor price with motor-gear drive.

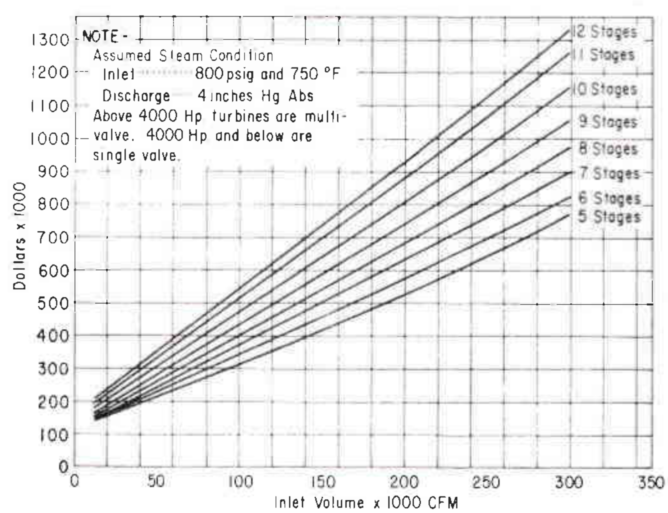


FIGURE 6—Axial compressor price with steam turbine drive.

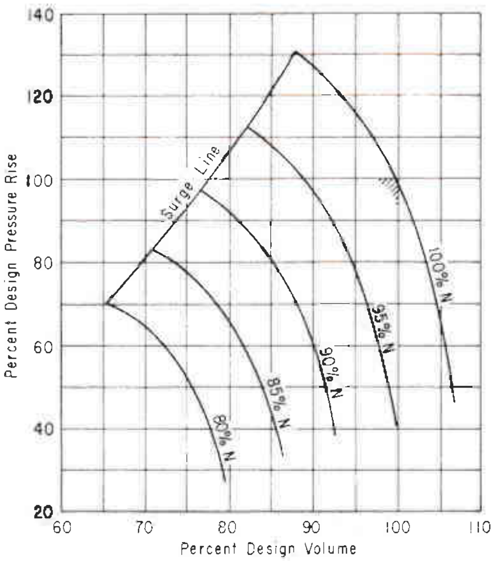


FIGURE 7—Typical axial compressor performance curve.

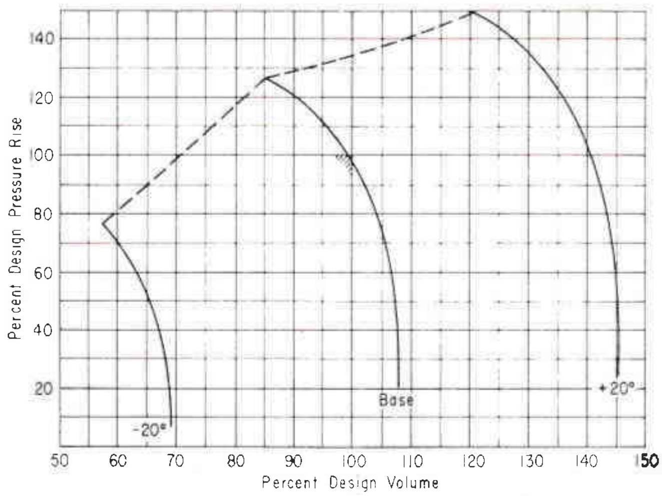


FIGURE 8—Expected performance of axial compressor with full stator blade control at constant speed.

(Figure 6) drive may be obtained. Figure 5 prices include: axial compressor, synchronous motor with 0.8 P.F., suitable speed increasing gear, lubrication system, and base plate for the compressor and drive. Figure 6 prices include axial compressor, multi-valve steam turbine, base plate and lubrication system.

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Schildwachter

Example: The 70,000 cfm axial compressor previously selected with motor-gear drive would have a price of $325,000 as indicated in Figure 5. The price with turbine is $400,000 (shown in Figure 6).

Axial Compressor Performance. The typical axial performance curve is shown in Figure 7. Note the wide variation in pressure with relatively small capacity change for a constant speed line. This makes the axial ideally suited for a base load machine.

Also, the steep curve and high pressure rise at part load points allows parallel operation with other machines. The machines do not have to be matched too closely in design pressure ratio since the axial's discharge pressure will adjust to that of the parallel machine without danger of surge.

In some applications a wider volume range is required than that offered by the basic axial compressor. In these cases adjustable inlet guide vanes or full or partial stator blade control on axial compressors is available to provide the wider range of operating conditions. Notice in Figure 8 that at 100 percent speed the capacity of the unit can be varied appreciably by adjusting all stator blades.

If less variation in operating range is required, partial adjustment of the stator blades, or merely inlet guide vane control may suffice. Any of these methods of range extension can be achieved with the machine in operation. Adjustment can be done manually or the unit can be provided with an automatic control system to change adjustment as the process requirements vary.

With the above material, an approximate evaluation of the pros and cons of an axial compressor may be made. This material must, of course, be considered only approximate and many other factors must be taken into account before arriving at any conclusions. For example, foundation costs, power costs, operating costs, etc., all must be noted. ##