

300 Reciprocating Compressors

Abstract

This section discusses engineering principles, types of reciprocating compressors, configurations, and performance characteristics. It contains sufficient information for understanding how to specify and apply reciprocating compressors including auxiliaries and support systems.

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310 Introduction

Section 100 provided general information and fundamental compression equations applicable to all types of compressors. This section presents additional equations, charts, and recommendations specifically related to reciprocating compressors. The machinery covered is limited to crosshead compressors having power ratings from approximately 150 to several thousand horsepower, with speeds ranging from 250 to 1000 RPM. With engineering judgment, much of the information presented can be applied to machines with ratings below 150 HP and speeds to 1500 RPM.

The purpose of Section 300 is to provide information to assist the reader in:

- Making initial estimates of power consumption, number of stages, cylinder sizing, and plot area for a proposed new compressor installation.
- Evaluating retrofits and modifications to re-rate existing compressors for new service conditions.
- Troubleshooting performance problems occurring on existing equipment.

Figure 300-1 shows a cross-section of a typical single-cylinder (one crankthrow) compressor, and includes the basic nomenclature that will be used throughout this section.

320 Engineering Principles

321 Compression Cycle

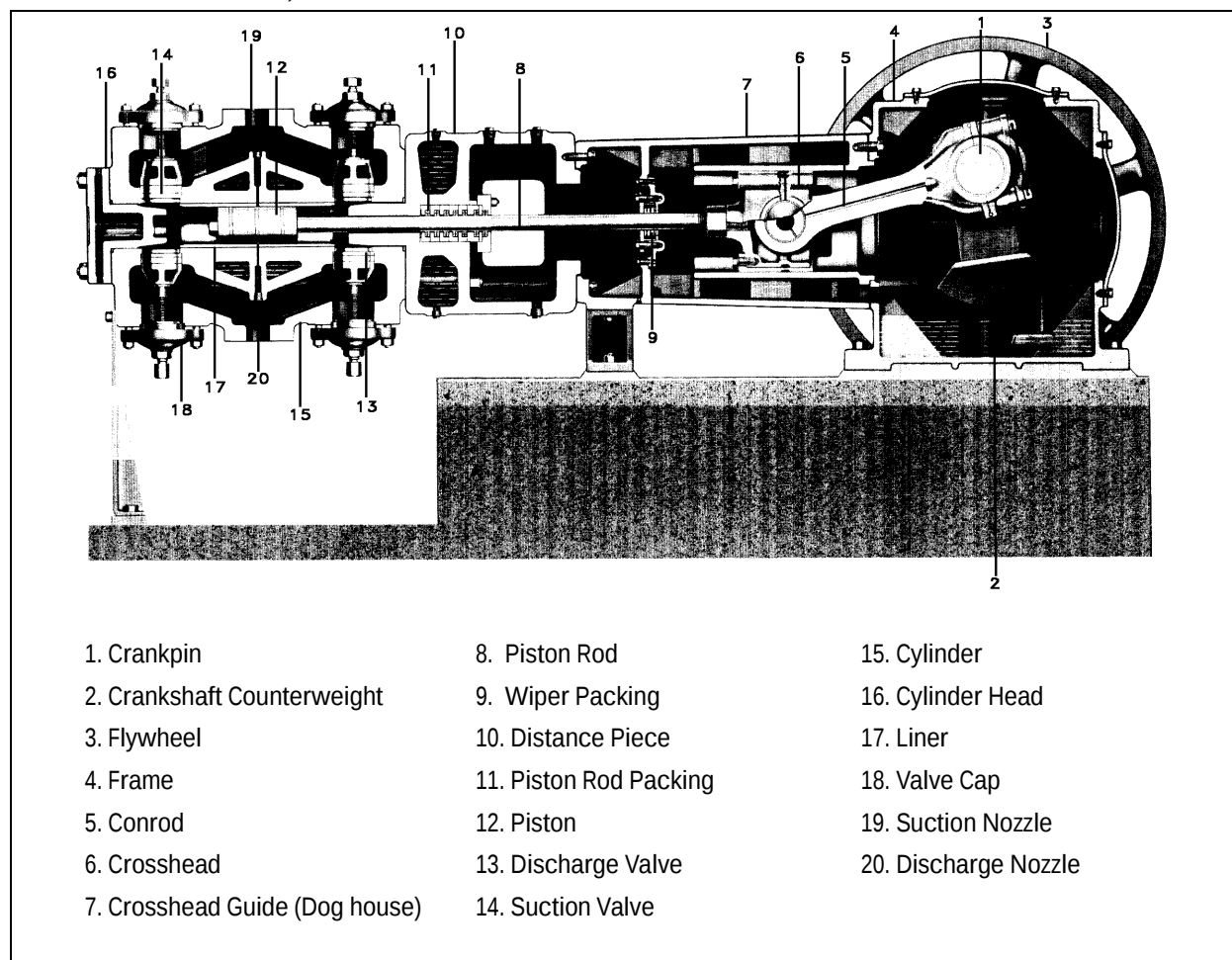
In a reciprocating compressor, compression takes place as a result of the back-and-forth travel or stroke of a piston in a cylinder equipped with inlet and discharge valves. The cycle is usually described by a **Pressure-Volume (PV) diagram**, similar to the one shown in Figure 300-2.

322 Volumetric Efficiency

Volumetric efficiency (E_v) is an important variable in reciprocating compressor calculations, affecting the diameter, stroke, and speed for a given compressor capacity. The compressor cylinder's actual inlet flow (ICFM) is the product of the volumetric efficiency and the cylinder's displacement over time, expressed in cubic feet per minute (CFM).

E_v is related to the clearance volume in the cylinder (the volume to the left of the dotted lines in Figure 300-2). On the PV diagram, it is the volume between points 0 and 3. The total volume displaced by a full stroke of the piston in the single-acting cylinder illustrated in Figure 300-2 is the volume between points 1 and 3. **Clearance** is usually expressed as a percentage of the **displaced volume** as follows:

Fig. 300-1 Typical Reciprocating Compressor (From Ingersoll-Rand Compressor Catalog, 1977. Courtesy of Dresser-Rand)



$$C = \frac{V_c}{V_{cyl}}(100)$$

(Eq. 300-1)

where:

C = percent clearance

V_c = clearance volume, cubic inches

V_{cyl} = displacement volume, cubic inches

Fig. 300-2 Steps in the Cycle of a Reciprocating Compressor (From Compressors: Selection & Sizing, by Royce Brown © 1986 by Gulf Publishing Company, Houston, TX. Used with permission. All rights reserved.)

The "ideal indicator diagram" is followed by a series of cylinder illustrations depicting piston movement and valve position. The figure shows, in diagram form, one complete crankshaft revolution representing a complete compression cycle. To begin the cycle, refer to the figure at:

(a) where the piston is at the lower end of the stroke (bottom dead center) and is at **path point 1** on the indicator diagram. At this point the cylinder has filled with gas at intake pressure P_1 . Note that the valves are both closed.

(b) the piston has started to move to the left. This is the compression portion of the cycle and is illustrated by **Path 1-2**.

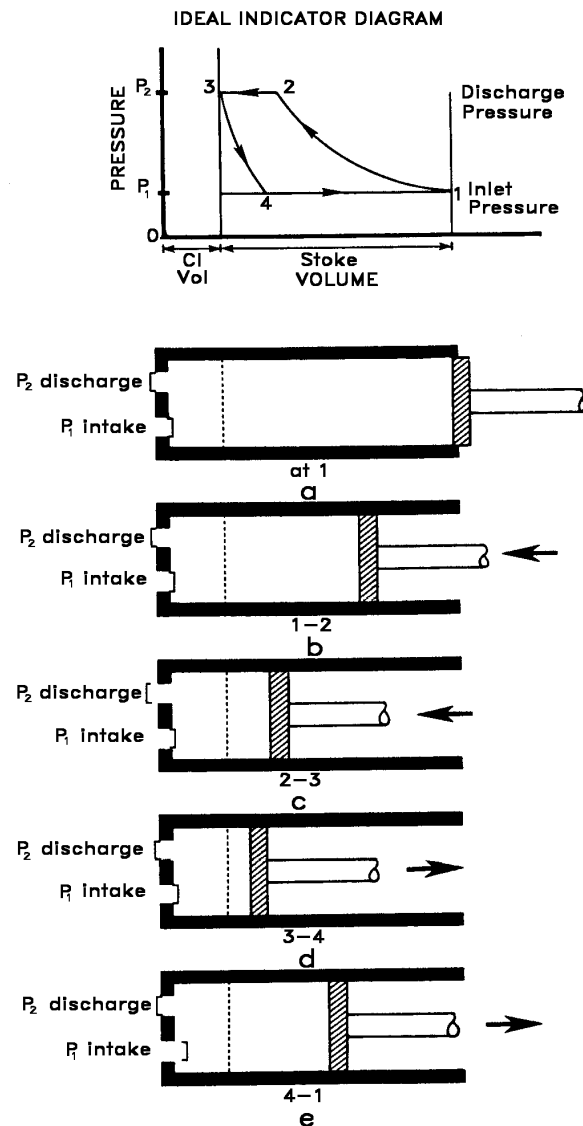
When the piston reaches **point 2** on the indicator diagram, the exhaust valve starts to open. The discharge portion of the cycle is shown at **(c)**. This is shown on the indicator diagram **Path 2-3**. Note that the discharge valve is open during this period while the intake valve is closed. The gas is discharged at the discharge line pressure P_2 .

(c) When the piston reaches **point 3** it has traveled to the upper end of its stroke (top dead center). Physically, at this point in the stroke, there is a space between the piston face and the head. The space results in a trapped volume and is called the clearance volume.

(d) Next in the cycle, the piston reverses direction and starts the expansion portion of the cycle, as illustrated at **(d)** in the figure. **Path 3-4** shows this portion of the cycle. Here the gas trapped in the clearance volume is re-expanded to the intake pressure. Note that the discharge valve has closed and the intake valve is still closed.

(e) At **point 4** the expansion is complete and the intake valve opens. The intake portion of the cycle is shown at **(e)**. This is indicated by **Path 4-1** on the indicator diagram. The cylinder fills with gas at intake line pressure P_1 .

When the piston reaches **point 1** the cycle is complete and starts to repeat.



The **theoretical volumetric efficiency** (in percent) of a cylinder is:

$$E_v (\text{theo.}) = 100 - C \left(r^{\frac{1}{k}} - 1 \right) \quad (\text{Eq. 300-2})$$

where:

$$r = \frac{P_2}{P_1} \text{ or } \frac{P_d}{P_s} = \text{pressure ratio}$$

$$k = \frac{C_p}{C_v}$$

Corrections to E_v for Gas Characteristics

In practice, corrections have to be made to E_v to account for valve losses, non-ideal re-expansion of gas in the clearance volume, internal leakage, compressibility, and other effects. Each manufacturer has his own set of empirical corrections. Therefore, several corrections for E_v are used in the industry.

Several of these corrections were compared for natural gas (S.G. = 0.72) and for propane. They were all within about 5% for pressure ratios between two and four.

However, for hydrogen-rich gases with molecular weights below 10, the corrections varied as much as 17% at a pressure ratio of four. However, the maximum pressure ratio for hydrogen-rich gases is typically about three (to keep the discharge temperature below 300°F). At a pressure ratio of 3, the deviation is less than 10%.

The following equation yields results that are approximately the average of the other formulas when used for more ordinary lubricated-cylinder compressor applications. Also, see the following discussion for limitations and corrections.

$$E_v = 97 - F_r - C \left[\left(\frac{Z_s}{Z_d} \right) r^{\frac{1}{k}} - 1 \right] \quad (\text{Eq. 300-3})$$

where:

E_v = volumetric efficiency, in percent

Z_s = compressibility at suction conditions

Z_d = compressibility at discharge conditions

F = 2.4 for mol. wt. < 10

= 1.4 for mol. wt. 10 to 30

= 1.0 for mol. wt. > 30.1

r = pressure ratio

Equation 300-3 is similar to the formula in the *GPSA Handbook*. The main difference is that F in Equation 300-3 adjusts for internal gas slippage which varies inversely with molecular weight.

Mechanical Corrections to E_v

Equation 300-3 applies to lubricated cylinders. Non-lubricated cylinders have a lower E_v due to greater piston-to-liner clearances, and other factors. For non-lube applications, change the 97 in Equation 300-3 to 94.

Speed also affects E_v . Equation 300-3 applies to slower speed machines up to 600 RPM. For 1000 RPM machines, E_v is about 3% less, and the E_v determined from Equation 300-3 should be multiplied by 0.97.

Applications and Limitations of E_v

Although Equation 300-3 is somewhat complex, it does not necessarily imply a high level of accuracy. It should only be used for estimating when pressure ratios range from two to five. Furthermore, the equation assumes the cylinder design provides ample valve flow area, the valve dynamics are satisfactory, and pressure pulsations are moderate. If any one of these conditions is abnormal, the E_v prediction is questionable.

Note that although E_v is a vital sizing consideration, it does not affect power consumption. E_v becomes an important factor, if the owner is considering a re-rate, or is troubleshooting a compressor capacity problem. Manufacturers may be consulted to determine highly accurate values of E_v based on proprietary information and programs.

Actual Inlet Flow

Actual inlet flow (ICFM) to the cylinders is determined by the equation:

$$\text{ICFM} = Q = V_d E_v \quad (\text{Eq. 300-4})$$

where:

V_d = displacement rate, CFM

E_v = volumetric efficiency expressed as a decimal.

Note When E_v appears in an equation, it is expressed as a decimal value, not a percent.

(Refer also to Section 100, Equations 100-14 and 100-15 for other Q relationships.)

323 Capacity and Power Calculations

Background

The power required by a reciprocating compressor is normally calculated using adiabatic (isentropic) relationships. The common industry practice is to state power in units of “brake horsepower per million cubic feet per day” (Bhp/MMCFD).

It is important to differentiate between:

- **Bhp/MMCFD** - brake horsepower per million cubic feet per day (14.4 psia and actual suction temperature)
- **Bhp/MMSCFD** - brake horsepower per million standard cubic feet per day (14.7 psia and 60°F)

When the capacity is given in MMSCFD, it may be converted to MMCFD as follows:

$$\text{MMCFD} = \text{MMSCFD} \left(\frac{14.7}{14.4} \right) \left(\frac{T_s}{520} \right) \left(\frac{Z_{rc}}{Z_o} \right) \quad (\text{Eq. 300-5})$$

where:

T_s = Actual suction temperature, °R

Z_{rc} = Compressibility at reference conditions of 14.4 psia and actual suction temperature

Z_o = Compressibility at 14.7 psia and 60°F

Note $\frac{Z_{rc}}{Z_o}$ can usually be taken as 1.0 with negligible error

If the capacity is given in SCFM, or pounds per minute, MMCFD can be calculated as follows:

$$\text{MMCFD} = \frac{(\text{SCFM})T_s}{353,742} \left(\frac{Z_{rc}}{Z_o} \right) \quad (\text{Eq. 300-6})$$

$$\text{MMCFD} = \frac{wT_s Z_{rc}}{932 M} \quad (\text{Eq. 300-7})$$

where:

w = weight flow in lbs/minute

M = molecular weight

When the flow rate in MMCFD, and Bhp/MMCFD are known, brake horsepower can be calculated from:

$$\text{Bhp} = \left(\frac{\text{Bhp}}{\text{MMCFD}} \right) (\text{MMCFD}) \quad (\text{Eq. 300-8})$$

MMCFD can be converted to ICFM (inlet cubic feet per minute), or Q as follows (bearing in mind that MMCFD is equal to 10^6 cubic feet/day):

$$\begin{aligned} Q &= (\text{MMCFD})(10^6) \left(\frac{14.4}{P_s} \right) \left(\frac{Z_s}{Z_{rc}} \right) \left(\frac{\text{day}}{1,440 \text{ min.}} \right) \\ &= (\text{MMCFD}) \left(\frac{10^4}{P_s} \right) \left(\frac{Z_s}{Z_{rc}} \right) \end{aligned} \quad (\text{Eq. 300-9})$$

where:

P_s = Suction pressure in psia

(Notice that division of 14.4 by 1440 results in 1/100 which simplifies the arithmetic, and it is for this reason that 14.4 psia is used as a convenient reference pressure for MMCFD.)

Combining Equations 300-4, 300-8 and 300-9:

$$Q = V_d E_v = \frac{\text{Bhp}(10^4)Z_s}{(\text{Bhp/MMCFD})Z_{rc}(P_s)} \quad (\text{Eq. 300-10})$$

where:

V_d = displacement rate, CFM

E_v = volumetric efficiency as a decimal

Note Z_{rc} is often taken as 1.0 in Equations 300-9 and 300-10 as a simplifying assumption. The error is usually small.

A number of formulas and charts have been developed over the years for determining Bhp/MMCFD. (Note that Bhp/MMCFD is often shortened to Bhp/MM.) Most of the recently published charts and equations have been for large motor-driven, low-speed, heavy-duty machines for critical refinery and chemical plant services covered by API 618. These relationships are also generally applied to large low-speed integral gas-engine compressors.

The Bhp/MM curves are used along with several correction factors to account for specific gravity, low-inlet pressure, etc. Additional correcting factors are often applied for smaller frames with higher speeds.

Computer programs have also been developed to calculate the theoretical horsepower and power losses associated with specific types of valves, and mechanical friction. The resulting power predictions are much more accurate than the Bhp/MM curves and the correction factors, and the computer methods continue to be improved. By the mid-1980's, the major manufacturers were using computer methods on most applications.

The manufacturers consider their valve designs and loss factors to be proprietary, and they are, therefore, not available. However, fairly good estimates can be made using updated Bhp/MM curves or formulas with some corrections.

Rough Compressor Horsepower Estimate

Equation 300-11 can be used to obtain a quick and rough estimate (plus or minus 10%) of compressor horsepower. It was developed for large low-speed (300-450 RPM) units handling 0.65 specific gravity gases and having stage compression ratios above 2.5. The equation may produce a result that is as much as 20% low for high-speed compressors (900-1200 RPM) due to the higher valve losses in these machines.

$$\text{Bhp} = (22) (\text{MMCFD}) (f) (n) (r) \quad (\text{Eq. 300-11})$$

where:

- r = compression ratio per stage (should be between 2.5 and 4.0)
- n = number of stages
- f = 1.0 for single-stage compression
- = 1.08 for two-stage
- = 1.10 for three-stage

Note that:

$$r = r_{\text{oa}}^{1/n} \quad (\text{Eq. 300-12})$$

where:

$$r_{\text{oa}} = \text{overall compression ratio}$$

Equation 300-11 will also provide a rough estimate of horsepower for gases with higher specific gravities or for lower stage compression ratios, with modification. For specific gravities between 0.8 and 1.0 use a multiplication factor of 20 instead of 22 in the equation. Likewise, use a factor of 16 to 18 for stage compression ratios between 1.5 and 2.0.

More Accurate Power Estimates

Figure 300-3 summarizes the equations and charts available to determine power for four classes of reciprocating compressors as defined in the chart. A Reciprocating

Compressor Performance Calculation Worksheet, CMP-EF-873, is included in the Standard Drawings and Forms section. A sample problem using the worksheet is also included in Appendix B.

Fig. 300-3 Classes of Reciprocating Compressors and Reference Data

Class	Typical HP Range	Nominal Speed Range, RPM	Nominal Stroke Range, inches	Equation Reference	Figure Reference
Class A: API 618 type heavy-duty process gas machine	500-10,000	250-600	10-20	300-11 300-12 300-13 300-14 300-15 300-16	300-4 300-5 300-6 300-7
Class B: Often called "separable" are normally used in oil and gas field operations.	150-3000	600-1000	5-6	300-17 300-18	300-8
Class C: Similar to Class A, but the frames are smaller. Used in both process and oil field services.	20-300	250-600	5-13	(multiply results from Class A by 1.03)	
Class D: Smaller light-weight versions of Class B. Usually applied in the oil field.	10-200	800-1500	2.5-4	(multiply results from Class B by 1.05)	

Equation 300-13 along with correction factors may be used to obtain a more accurate estimate of compressor horsepower.

$$\frac{\text{Bhp}}{\text{MMCFD}} = \left(\frac{46.9}{Z_{rc}} \right) \frac{\left(r^{\frac{k-1}{k}} - 1 \right)}{\frac{k-1}{k}} \left(\frac{Z_s + Z_d}{2} \right) \quad (\text{Eq. 300-13})$$

where:

$$k = \frac{C_p}{C_v} \text{ at 14.7 psia and average compression temperature } \left(\frac{T_s + T_d}{2} \right)$$

Equation 300-13 includes a correction for a mechanical efficiency of 93%. (In other words, if the compressor frame and cylinders had zero mechanical losses or 100% mechanical efficiency, the 46.9 constant in Equation 300-13 would be 43.6.) The 93% is an arbitrary average. Actual mechanical efficiency can vary significantly

with machine selection. See Section 330, “Compressor Types and Performance Characteristics.”

Figures 300-4, 300-5, and 300-6 are graphic representations of Equation 300-13, less compressibility effects. These figures are similar to figures found in the *GPSA Handbook* and other handbooks, and may be used instead of Equation 300-13 to determine the uncorrected BHP/MMCFD.

Corrections to Equation 300-13. There are several potential factors which may be accounted for in order to obtain a more accurate horsepower estimate. See Equation 300-20 for the total adjusted horsepower. The corrections are described as follows.

Valve Loss Corrections. Valve losses tend to increase with increasing gas density. Molecular weight (specific gravity) and inlet temperature affect density. These effects can be taken into account with an addition to the basic Bhp/MM. For Class A compressors, this correction is as follows:

$$\text{Corrected Bhp/MM} = \text{Basic Bhp/MM} + \text{Bhp/MM}_{(SG)} \quad (\text{Eq. 300-14})$$

where:

$$\text{Bhp/MM}_{(SG)} = \text{valve loss correction}$$

When Specific Gravity ≥ 1.0

$$\text{Bhp/MM}_{(SG)} = \frac{4,320(SG)}{T_s Z_{rc}} \quad (\text{Eq. 300-15})$$

When Specific Gravity < 1.0

$$\text{Bhp/MM}_{(SG)} = \frac{3,624(SG) + 696}{T_s Z_{rc}} \quad (\text{Eq. 300-16})$$

The corrections in Equations 300-15 and 300-16 are appropriate for cylinders having good valve flow characteristics. However, if the valves are not properly selected for the compressor application, or if the machine's service conditions differ significantly from the original design conditions, the corrections may not provide an accurate prediction of compressor power. Excessive pressure pulsations can also affect the accuracy of predictions.

Class B compressors, with higher speeds, often have higher valve losses. Methods of predicting horsepower for Class B machines vary from simply adding 5% to the Bhp/MM, regardless of the gas, to a set of additions related to specific gravity. The latter method is the one used by the *GPSA Handbook* (Refer to the “References” section of this manual).

Fig. 300-4 Basic Bhp/MM for Gas Compressors 150 HP and Above

Note: This chart applies to Class A machines, as defined in Figure 300-3, and also applies to Integral-Engine Compressors. Refer to Section 323 for further instructions, and for corrections to this chart for specific gravity, moisture, and non-lubricated cylinders.

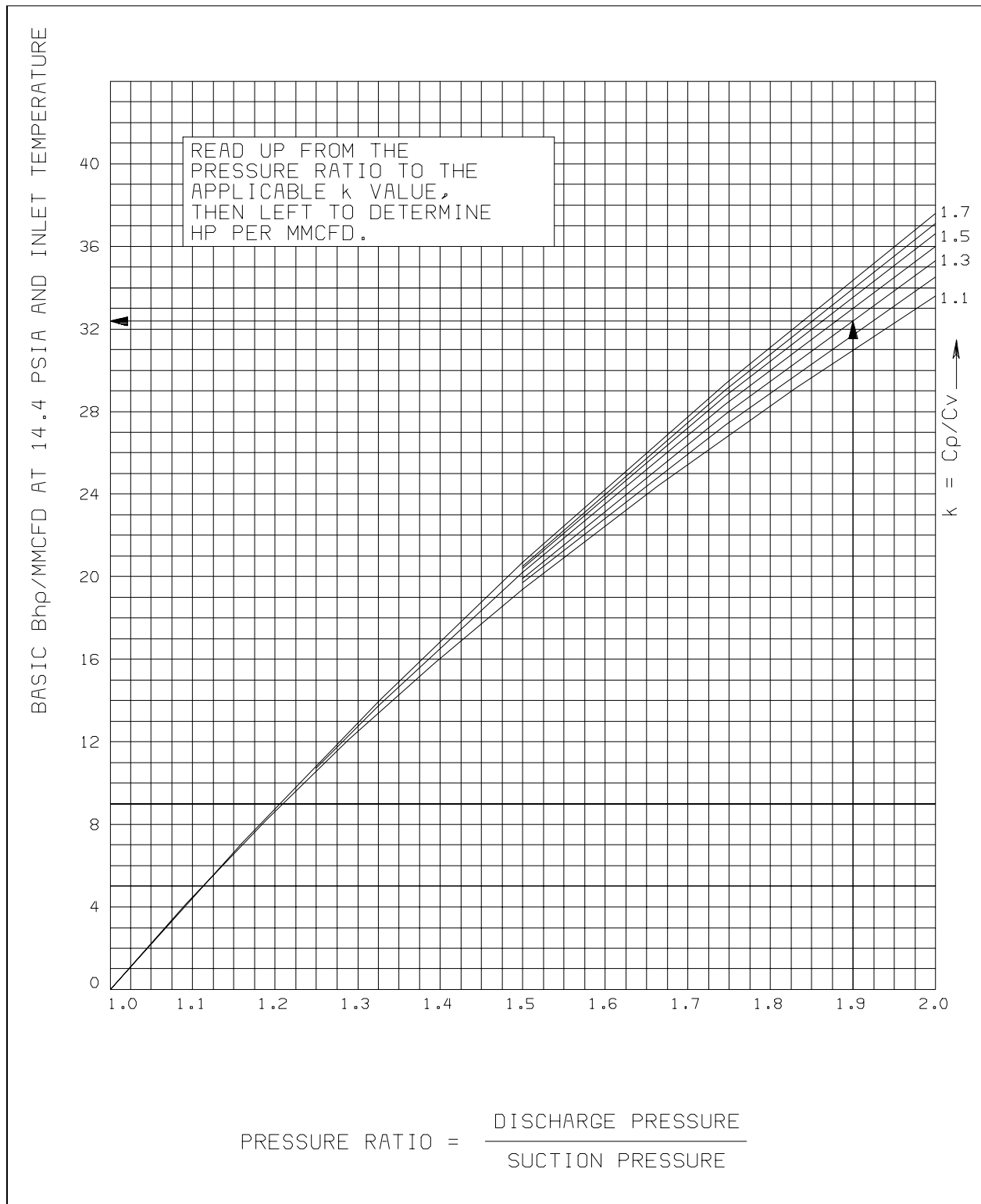


Fig. 300-5 Basic Bhp/MM for Gas Compressors 150 HP and Above

Note: This chart applies to Class A machines, as defined in Figure 300-3, and also applies to Integral-Engine Compressors. Refer to Section 323 for further instructions, and for corrections to this chart for specific gravity, moisture, and non-lubricated cylinders.

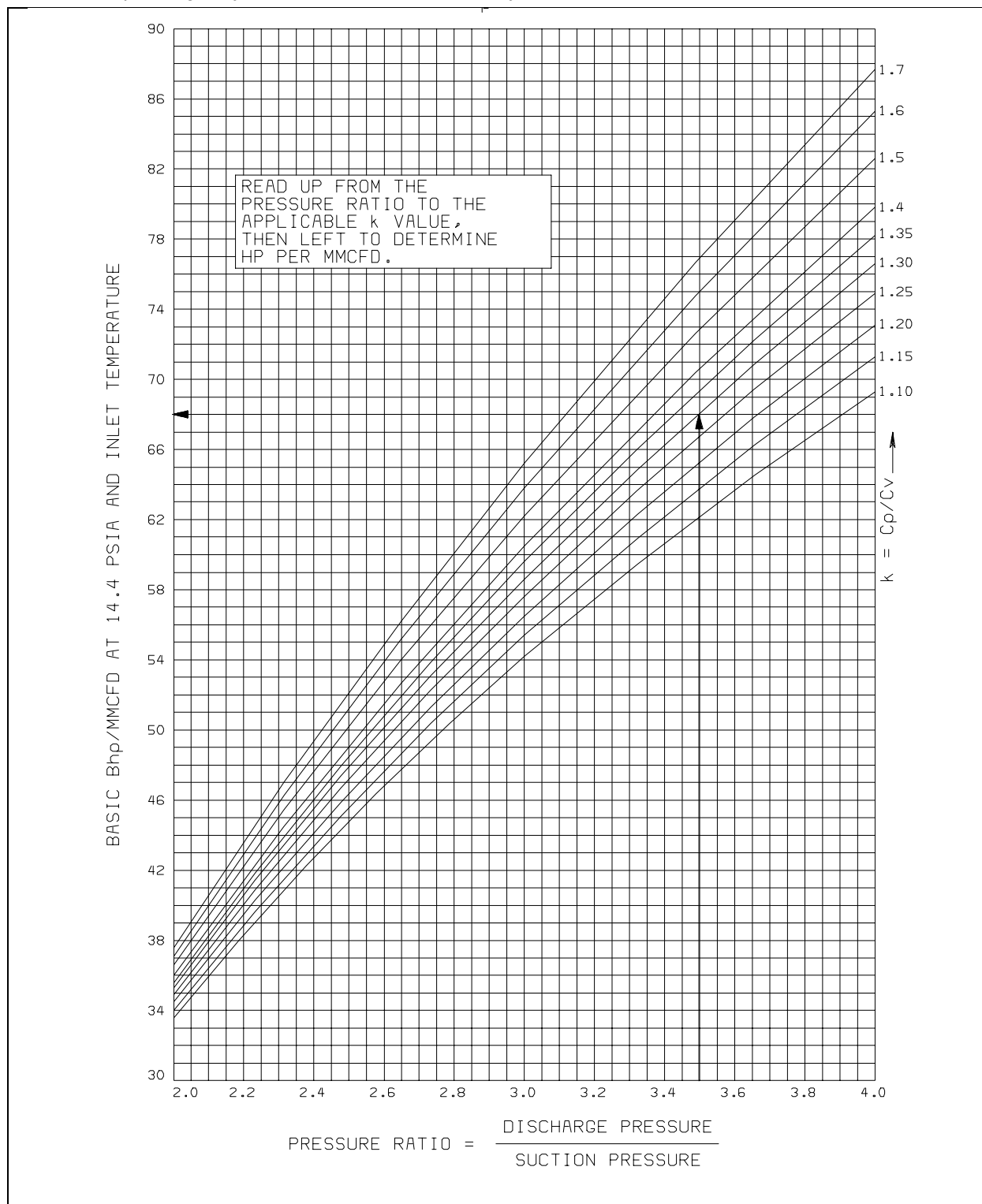
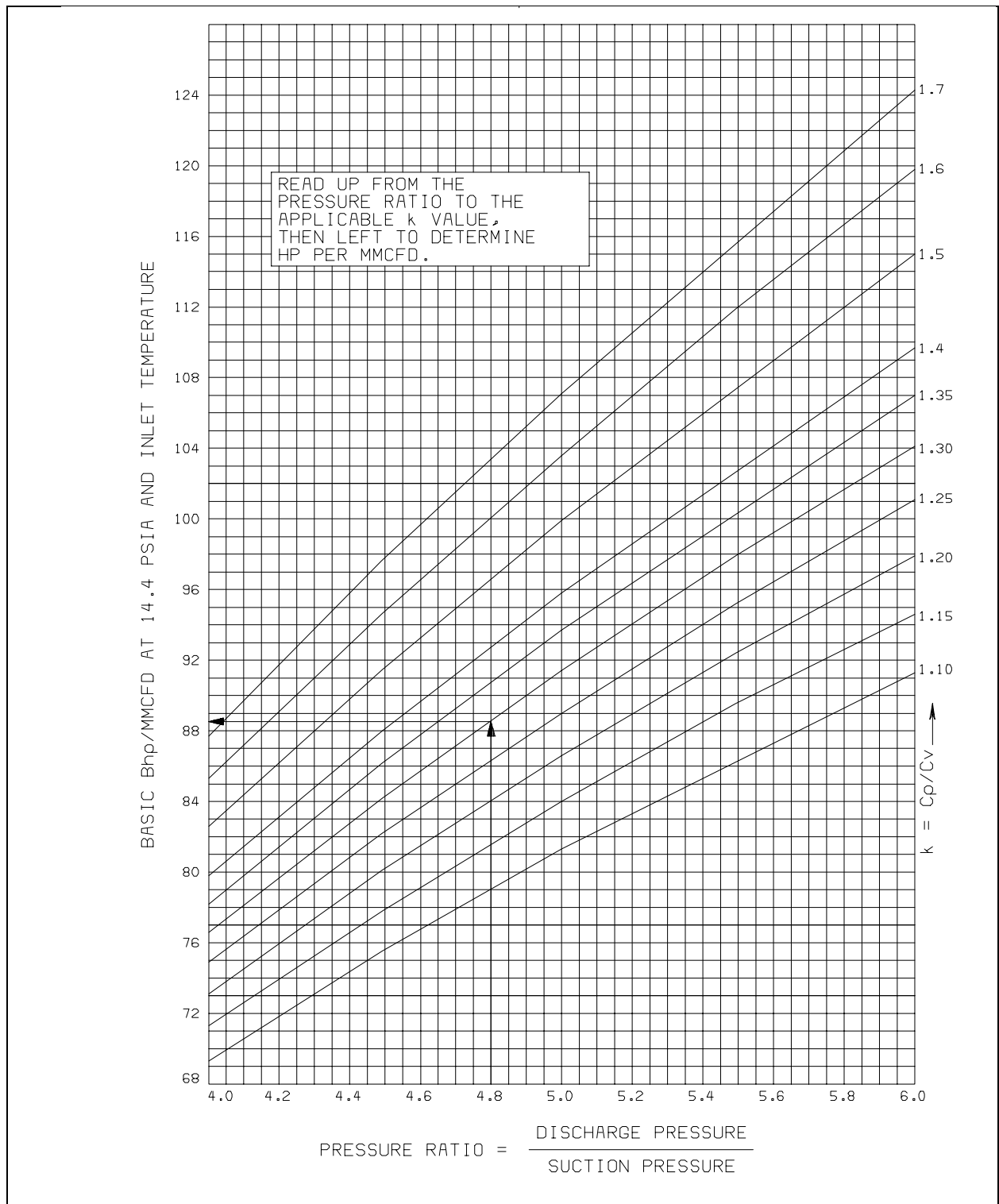


Fig. 300-6 Basic Bhp/MM for Gas Compressors 150 HP and Above

Note: This chart applies to Class A machines, as defined in Figure 300-3, and also applies to Integral-Engine Compressors. Refer to Section 323 for further instructions, and for corrections to this chart for specific gravity, moisture, and non-lubricated cylinders.



While the GPSA method will provide fairly good predictions for average cases, inlet temperature is not accounted for, and applications with higher pressure ratios tend to be overcorrected. Corrections similar to those for low-speed compressors are also available. Following are additions to the basic Bhp/MM which have been used in the industry for Class B compressors:

When Specific Gravity ≥ 1.0

$$\text{Bhp/MM}_{(SG)} = \frac{6,027(SG)}{T_s Z_{rc}} \quad (\text{Eq. 300-17})$$

(The correction calculated is applied in Equation 300-14.)

When Specific Gravity < 1.0

$$\text{Bhp/MM}_{(SG)} = \frac{5,672(SG) + 335}{T_s Z_{rc}} \quad (\text{Eq. 300-18})$$

Figures 300-7 and 300-8 may be used in place of Equations 300-15, 300-16, 300-17, and 300-18 for valve loss corrections for Class A and B machines. They neglect compressibility effect ($Z_{rc} = 1$).

Moisture Correction. When the gas being compressed contains water vapor, the vapor content should preferably be included in the gas analysis. When it is included, there is no need to correct the Bhp/MM for moisture. If it is not included, a moisture correction should be made. The moisture correction is typically used in air compressor calculations. The moisture correction factor is:

$$MF = \frac{P_s}{P_s - P_v} \quad (\text{Eq. 300-19})$$

where:

MF = Moisture correction factor

P_s = Suction pressure, psia

P_v = Vapor pressure of water at suction temperature (from steam tables/charts).

The moisture factor must be calculated for each stage. After the first stage, it is usually assumed that the gas is saturated (relative humidity = 100%) with water vapor as it leaves the intercooler. Note that there will be water drop-out at each intercooler.

Correction for Non-lubricated Cylinders. Another correction factor is applied when the cylinders are not lubricated. The commonly used non-lube correction factor (NLCF) is a multiplier of 1.05. This factor is widely accepted in the industry.

Fig. 300-7 Addition to Basic Bhp/MM to Correct for Valve Losses in Class A Compressors, as defined in Figure 300-3.

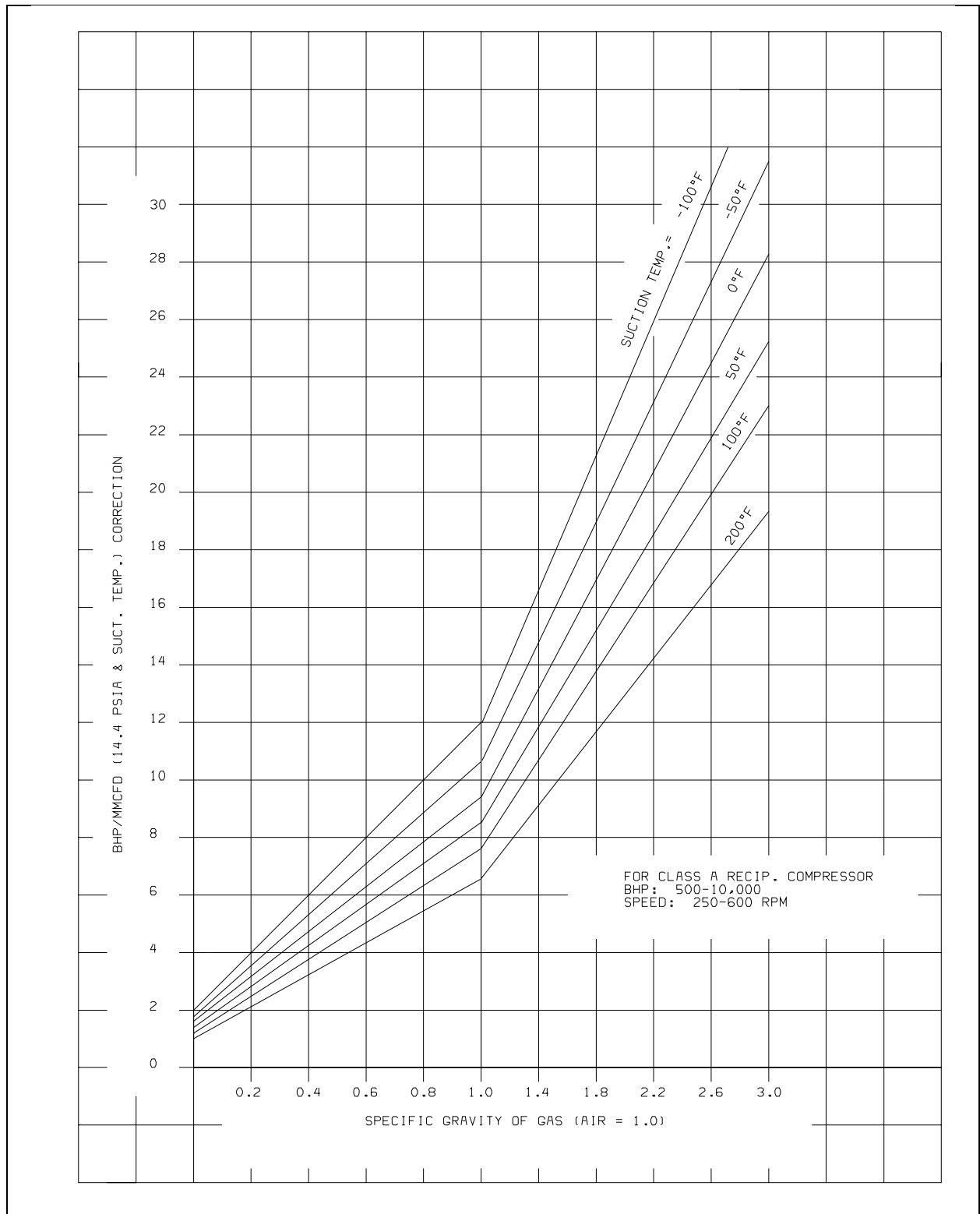
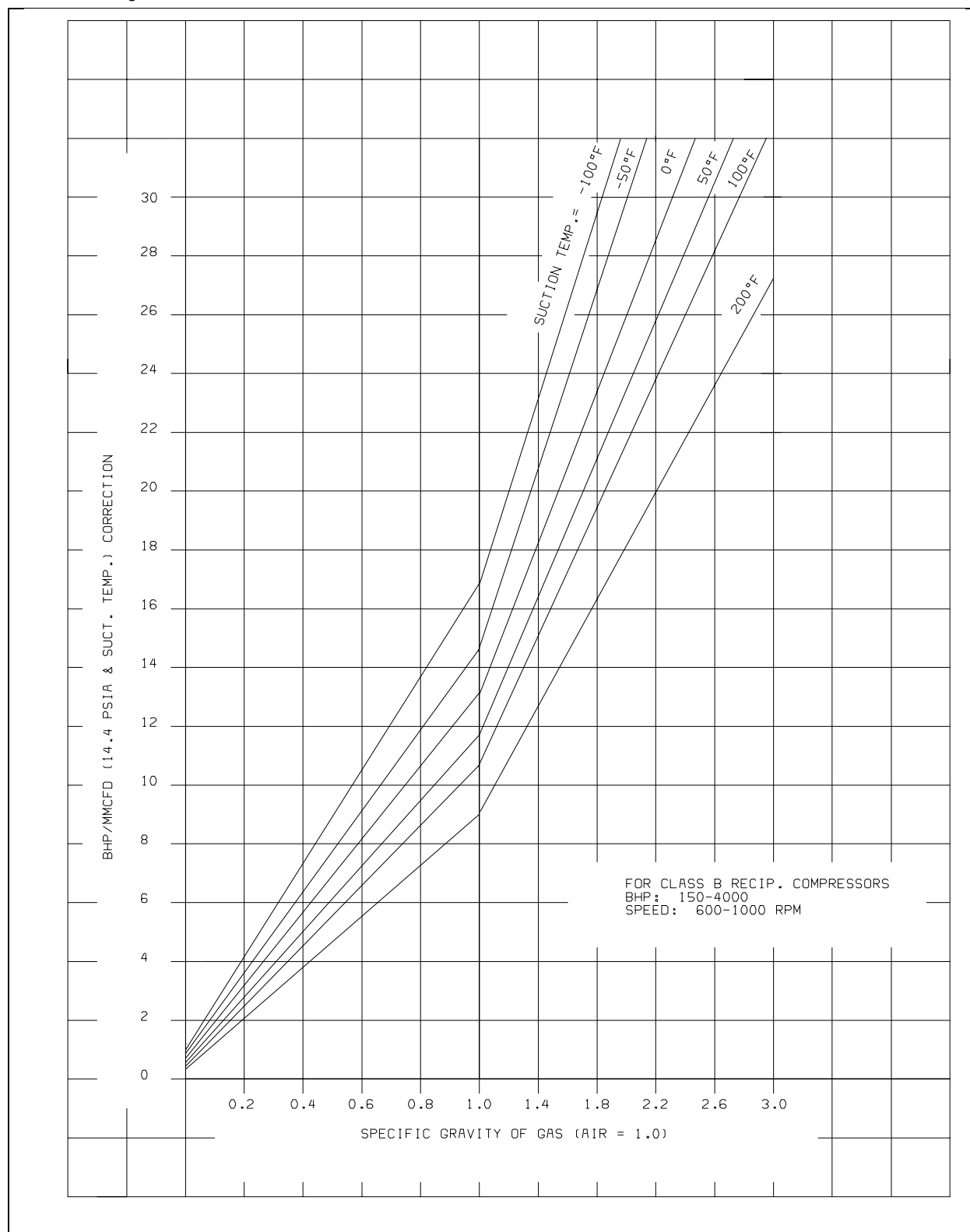


Fig. 300-8 Addition to Basic Bhp/MM to Correct for Valve Losses in Class B Compressors, as defined in Figure 300-3.



Although there is some evidence that a well-engineered non-lube design will only require 1–2% more power, using 1.05 is recommended unless you know the correction should be less.

Total Corrected Power. The total adjusted Bhp/MM per stage is then:

$$\begin{aligned} \text{Total adjusted } \frac{\text{Bhp}}{\text{MM}} \\ = \left(\text{Basic } \frac{\text{Bhp}}{\text{MM}} \right) (\text{MF})(\text{NLCF}) + \frac{\text{Bhp}}{\text{MM}_{(\text{SG})}} \end{aligned} \quad (\text{Eq. 300-20})$$

The result of Equation 300-20 can then be applied in Equation 300-8 to determine the Bhp for each stage. The total Bhp for the machine is the sum of the Bhp's for the stages.

Power Estimates for Class C and D Compressors. A quick estimate for Class C Compressors can be made by adding 3% to the horsepower calculated on the basis of a Class A machine. Similarly, for Class D machines, add 5% to the power determined for Class B units.

In making performance estimates, it is customary to subtract 1% from the initial suction pressure, and add 1% to the final discharge pressure. These allowances are for the pressure drop through the pulsation dampers.

For interstage pressure drop, use 3% to account for pulsation dampers, piping, knockout drum (separator), and intercooler. If Equation 100-36 is used, replace 0.98 with 0.97 in the denominator. If actual values for pressure drops are known, they should be used instead of the foregoing estimates.

324 Discharge Temperature

Properly cooled compressor cylinders stay reasonably close to the theoretical adiabatic (isentropic) process with respect to discharge temperature. Therefore, the adiabatic discharge temperature equations in Section 100 may be used to predict the discharge temperature with fairly good accuracy. These equations can be rewritten as follows so that Fahrenheit temperature may be used:

$$t_d = \left[(t_s + 460) r^{\frac{k-1}{k}} \right] - 460 \quad (\text{Eq. 300-21})$$

where:

t_s and t_d are in Fahrenheit.

This equation is applicable for machines with ideal cylinder cooling, and may be used as a first approximation for all applications. If more accurate estimates are

needed, refer to Section 360 of this manual, “Application and System Considerations” where cylinder cooling and deviations from Equation 300-21 are discussed.

Refer to Figure 300-9 for a graphic representation of Equation 300-21.

325 Number of Stages

The number of stages is governed by the following factors:

1. Allowable discharge temperature.
2. Rod loading.
3. Existence of a fixed sidestream pressure level (where flow is added to or withdrawn from main flow of compressor).
4. Allowable working pressure of available cylinders.

Discharge temperature is the most important factor affecting the number of stages. Class A and B reciprocating compressors are generally limited to 300°F for most gases in upstream and downstream plants. API 618 further limits the discharge temperature of hydrogen-rich gases to 275°F. These limits restrict the stage pressure ratios. It is often necessary to increase the number of stages so that intercoolers can be added to keep the discharge temperature within limits, while achieving the required overall pressure ratio. Knowing the discharge temperature limit, Equation 300-21 can be rewritten to find the allowable pressure ratio as follows:

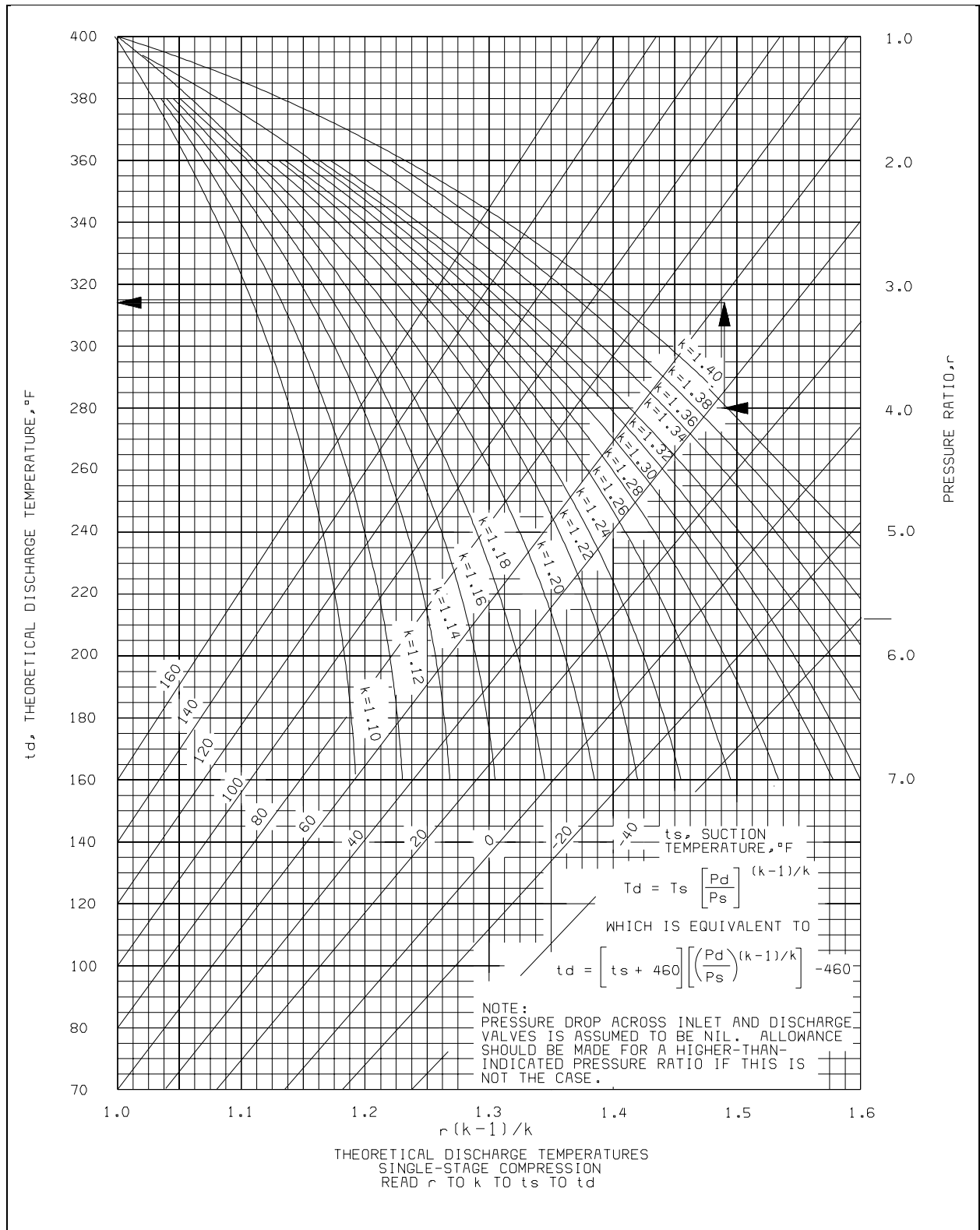
$$r = \left(\frac{t_d + 460}{t_s + 460} \right)^{\frac{k}{k-1}} \quad (\text{Eq. 300-22})$$

Figure 300-9 may also be used to find r (pressure ratio) corresponding to a given discharge temperature limit.

Adding intercoolers to a centrifugal compressor tends to save horsepower. With reciprocating compressors, however, there will seldom be any benefit in adding intercoolers beyond those needed to maintain discharge temperature limits. The reasons are: (1) reciprocating compressors are already highly efficient, and adding an intercooler adds pressure drop which offsets the power savings, and (2) addition of a stage requires additional cylinder(s), pulsation dampers, knockout drum and piping.

The **rod-load** limit can affect the number of stages since the combined rod loading is related to the differential pressure across the cylinder. Increasing the number of stages obviously reduces the differential pressure of each stage. Quite often a rod loading problem can be solved by using two cylinders for one compression stage. In this case, the differential pressure would remain the same, but piston area, upon which the differential pressure acts, would be reduced. Rod loading is covered in further detail in Section 350, “Selection Criteria.”

Fig. 300-9 Theoretical Discharge Temperatures Single-Stage Compression (Read r to k to t_s to t_d) (Courtesy of GPSA)



Sometimes a compressor application has **more than one suction or discharge pressure level**. For example, in an oil field gas system, the compressor may take different quantities of gas from the separator at two pressures, say 40 and 250 psig. This machine could also be required to deliver a portion of the gas at 1000 psig for gas lift, and the remainder at 2500 psig for injection back into the formation. In this case, these pressures would set the interstage pressures so that the sidestreams are accommodated. Note also that two stages might be required between the 40 and 250 psig levels (depending on suction temperature and k value) to stay below the discharge temperature limits.

Allowable Working Pressure. Occasionally a given pressure ratio might be achieved in one stage with satisfactory discharge temperature and rod loading, but an actual cylinder does not exist to handle both the capacity (ICFM) and pressure. In these situations, it is necessary to use two stages, or use two smaller single-stage cylinders depending on hardware and economics.

326 Cylinder Sizing

Cylinder displacement is easily calculated with basic geometry. There are three cylinder configurations to consider:

1. single-acting
2. standard double-acting
3. double-acting with tail rods (see Section 351)

Refer to Figure 300-2 for an illustration of a **single-acting** cylinder. Figure 300-10 illustrates **double-acting** configurations. Following are displacement and displacement rate formulas.

Single-Acting

Cylinder displacement can be calculated:

$$V_{\text{cyl}} = \frac{\pi}{4} D^2 S = 0.785 D^2 S \quad (\text{Eq. 300-23})$$

where:

V_{cyl} = displacement, cubic inches

D = piston diameter, inches

S = stroke, inches

The displacement rate can be calculated:

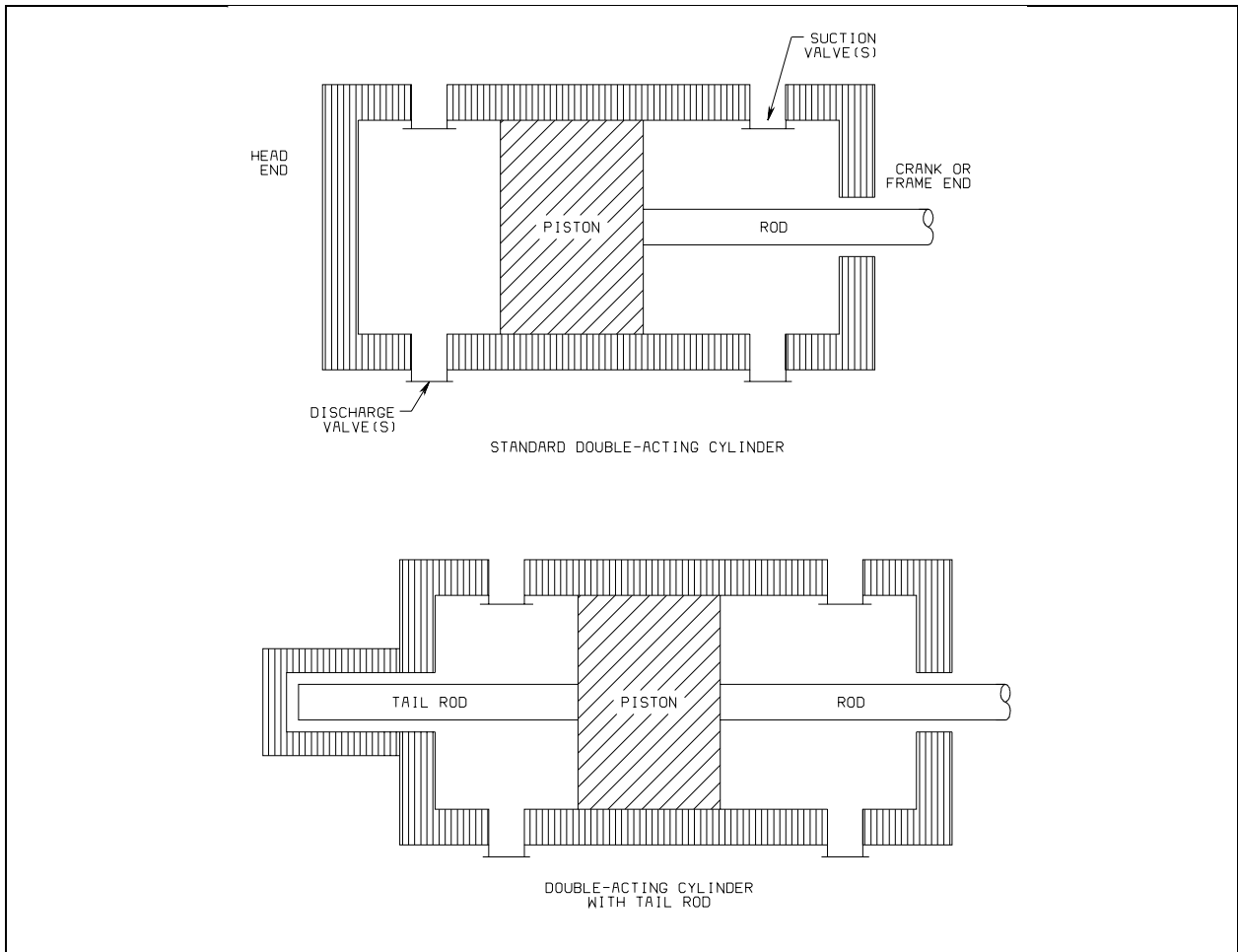
$$V_d = \left(\frac{\pi}{4} \right) \frac{D^2 S N}{1,728} = \frac{D^2 S N}{2,200} \quad (\text{Eq. 300-24})$$

where:

V_d = displacement rate, CFM

N = RPM

Fig. 300-10 Double-Acting Cylinders



Double-Acting

Cylinder displacement is calculated:

$$V_{cyl} = 0.785 (2D^2 - d^2) S$$

(Eq. 300-25)

where:

D = piston diameter, inches

d = rod diameter, inches

Cylinder displacement rate is calculated:

$$V_d = \frac{(2D^2 - d^2)SN}{2,200} \quad (\text{Eq. 300-26})$$

Double-Acting with Tail Rod

Cylinder displacement is calculated:

$$V_{\text{cyl}} = 1.571 (D^2 - d^2) S \quad (\text{Eq. 300-27})$$

Cylinder displacement rate is calculated:

$$V_d = \frac{(D^2 - d^2)SN}{1,100} \quad (\text{Eq. 300-28})$$

Estimating Cylinder Size to Accommodate a Given Flow Rate

Compressor cylinders are made in certain classes defined by:

1. Maximum allowable working pressure (MAWP)
2. Stroke
3. Number of valves per cylinder
4. Diameter range

The diameter in a class is varied by changing the wall thickness of the inner cylinder, or liner. Each class of cylinder has essentially the same **clearance volume** regardless of the diameter. However, the **percent clearance** varies with diameter. Volumetric efficiency also varies since it is related to clearance.

For example, assume that a 12-inch-stroke cylinder (double-acting) has a 3-inch rod, a MAWP of 1000 psig, a speed of 360 RPM, a diameter range of nine to 11 inches, and a clearance of 15% when the diameter is 10 inches. Further assume that the application has a pressure ratio of three, a molecular weight of 20, and a k value of 1.25. Figure 300-11 shows how percent clearance and volumetric efficiency vary with diameter for this cylinder class.

The percent clearance varies from about 8% to well over 30% among the many classes of cylinders available. There is no rule of thumb to relate percent clearance to diameter with much accuracy. However, for rough estimates use:

- 20% for diameters up to eight inches
- 15% for eight to 20 inches
- 12% in excess of 20 inches

Fig. 300-11 Example: Percent Clearance and Volumetric Efficiency Variance

Dia. ⁽¹⁾	Displ. rate, V _d , CFM	Displ., V _{cyl} , in ³	Cl. vol., in ³	% Clearance	E _v , %
9	300	1442	270	18.7	66.5
9-1/2	337	1616	270	16.7	69.3
10	375	1800	270	15.0	71.7
10-1/2	415	1993	270	13.6	73.6
11	458	2196	270	12.3	75.5

(1) Practice varies among manufacturers, but cylinders can usually be furnished in diametral increments of 1/8 inch up to about 12-inch diameter, 1/4-inch increments from 12 inches to 20 inches, and 1/2-inch increments beyond 20 inches.

The common approach to cylinder sizing is to make an educated guess at E_v and then solve for the displacement rate using Equation 300-4. Cylinder diameter can then be calculated using Equations 300-24, 300-26, 300-28, as appropriate. This approach may have to be repeated two or three times to arrive at a combination that satisfies a given inlet flow quantity, Q .

As cylinder diameters do not come in an infinite number of increments, it is customary to select the next largest increment. In multistage machines, depending on the size of increments, oversizing of an initial stage is sometimes balanced by slightly undersizing the subsequent stage, assuming the interstage pressure level is not fixed (by a sidestream for example).

330 Compressor Types / Performance Characteristics

331 Types

There are two basic types of reciprocating compressors distinguished by the style of piston and the linkage between the piston and crankshaft. One type covers the lighter-duty machines having **trunk-type** (automotive-type) single-acting pistons lubricated by crankcase oil, with no crossheads. These machines are typically used for air compression to 125 psig, although cylinders are available for working pressures to 6000 psig, for various gases. They operate at speeds in the range of 1200 to 1800 RPM and have ratings to about 125 HP. This type of machine is not often used for petroleum gas service, therefore, it is not described further in this manual.

The other type of reciprocating compressor is a heavy-duty **crosshead-type** machine where each piston is usually double-acting, and is connected to the crankpin by a piston rod, crosshead, and connecting rod. The cylinders are lubricated by a force-feed lubricator. Refer to Figure 300-1 for a cross-section of a typical single-cylinder reciprocating compressor.

Crosshead-type machines may be categorized into four broad classes:

- **Class A** – API 618 type of heavy-duty process gas machine.

- **Class B** – Often called “separable,” are normally used in oil and gas field operations.
- **Class C** – Similar to Class A, but the frames are smaller. Used in both process and oil field services.
- **Class D** – Smaller, light-weight versions of Class B. Usually applied in the oil field.

The manual concentrates generally on Classes A and B, but much of the information is also generally applicable to Classes C and D. (Also see Figure 300-3.)

332 Performance Characteristics

One of the inaccuracies involved in estimating power from Bhp/MM equations, or curves, stems from friction horsepower losses. For example, say a compressor has a 12-inch stroke frame with a maximum rating of 4000 Bhp. Also assume that the frame and cylinders have a mechanical efficiency of 93% (basis of Equation 300-13). Friction horsepower would be 280 HP (4000×0.07) for the frame. Now, say this frame is only loaded to 2000 Bhp for a certain application. The friction HP is about the same (i.e., 280 HP) so now that friction loss is 14% ($280 \div 2000$) of load and the mechanical efficiency is 86%.

This example may be somewhat out of the ordinary, but it points out the fact that significant error can arise from not accounting for frame loading. It is impractical to develop a loading factor, because it varies from one design to the next. Friction HP also varies with the number of throws and sizes of cylinders. The vendor should be contacted if there is serious concern about the friction HP contribution to overall compressor power.

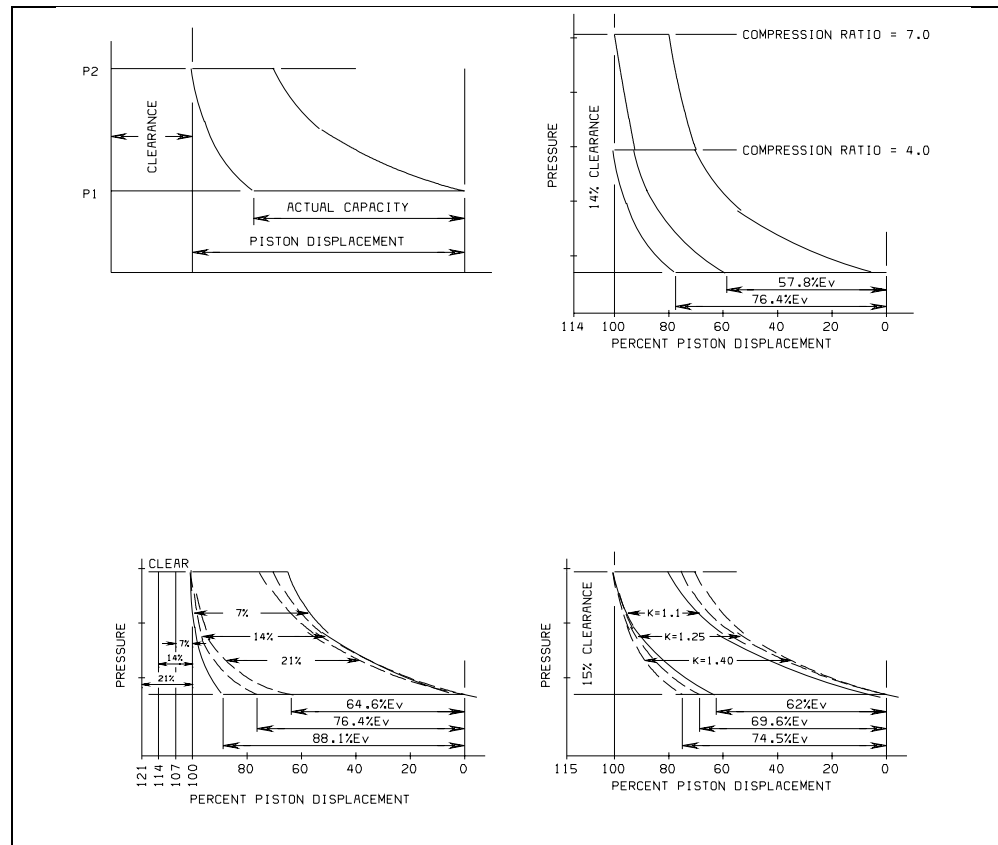
Figure 300-12 shows a theoretical PV Diagram and the effects of changes in clearance, pressure ratio, and k value on the area (work) of the compression cycle. The figure shows that increasing the clearance decreases the volumetric efficiency; hence, the ICFM is reduced. Therefore, clearance is an appropriate variable to use to control the capacity.

Capacity Control by Unloading

Reducing the capacity is commonly called **unloading**, and devices that cause capacity reduction are called **unloaders**. Capacity control is often stated in terms of loading steps. Five-step unloading, without further clarification, means nominal capacity steps of 100, 75, 50, 25, and 0%; three-step means 100, 50, and 0%.

There are two general types of unloading devices. One type **adds clearance** to the cylinder to reduce the capacity. The other type fully **deactivates** one end of a cylinder. To avoid confusion between the two types, the former devices are often called **clearance pockets** (fixed or variable), clearance bottles, or clearance spacers; whereas, the latter devices are called **unloaders**. These are described further in Section 340.

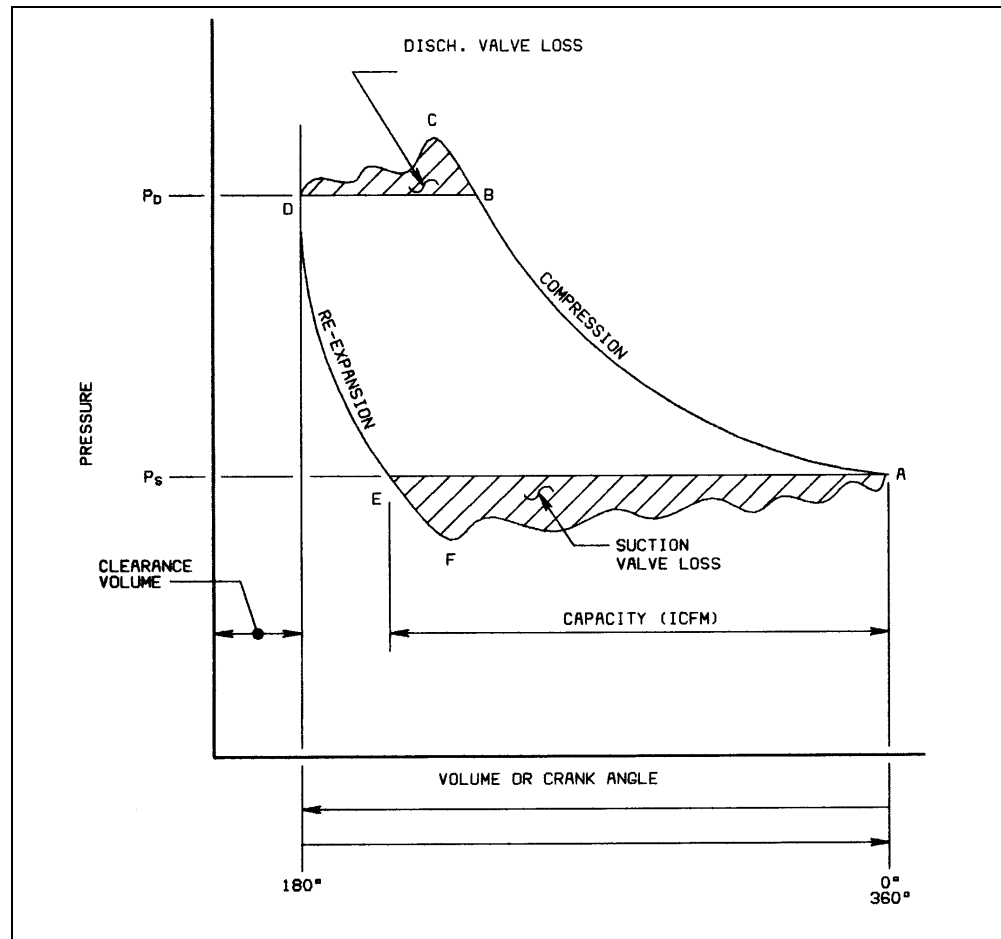
Fig. 300-12 Effect of Various Parameters on P-V Diagram (Note: The original drawing used V_E in place of E_v . The drawing was changed for consistency of units in this manual.) (Courtesy of Dresser-Rand)



There is some parasitic gas power loss in the unloaded end of a cylinder as a result of gas flowing back-and-forth through suction valves held open or through the port of a plug-type unloader. Moreover, when one end of a cylinder is unloaded, the capacity of the loaded end is reduced to some extent by heat build-up caused by the back-and-forth flow in the unloaded end. The heat built-up from the power loss causes a rise in suction temperatures, resulting in a lower weight flow. This effect is usually small but can be as much as 10–15% if the parasitic loss is proportionally high.

Bypassing, suction throttling, and varying speed are other means of capacity control that are external to the compressor. These methods are occasionally used for reciprocating compressors. Speed control is available only on units with variable-speed drivers, such as internal combustion engines.

Reciprocating compressor performance is usually presented in a tabulation of data showing Bhp's and capacities for various pressure conditions and gas compositions. Also, if there are clearance pockets or many unloading steps to accommodate highly variable pressure conditions, curves may be furnished by the vendor (similar to Figures 300-13 and 300-14).

Fig. 300-16 Actual PV Diagram

fully open but will close prematurely. Real valves also present a restriction to flow which, combined with any plates that do not open fully, could cause a greater pressure drop across the valve, thus increasing the power consumption. In addition, volumetric efficiency is reduced.

The compression efficiency of a cylinder depends largely on the valve losses. Manufacturers have made vast improvements in analytical techniques to optimize valve design. Optimized valve designs have effected up to a 15% improvement in Bhp/MM for some applications.

Figure 300-16 is a typical PV diagram showing valve losses. Suction and discharge valve loss is represented by Areas A-F-E and B-C-D. The amount of loss is a function of the flow rate, drag coefficient and mass of the valve elements, valve-spring stiffness, pressure drop, gas pulsations, effective valve flow area, and compressor speed. Therefore, calculation of valve losses is not a straightforward process. It requires a complex computer program with empirical factors related to each specific valve design and cylinder.

Whenever possible, each cylinder should have at least two suction and two discharge valves per end (four suction and four discharge valves total for a

double-acting cylinder). A greater number of valves will reduce the effect of a broken valve on cylinder performance and rod loading. In some small bore cylinders, it may be impossible to provide more than one suction and one discharge valve per end. If this is the case, ask the vendor if clearance can be added to a larger cylinder with more valves to achieve the same inlet capacity. In extreme cases, it may be possible to reduce the stroke on a particular crank throw in order to utilize a larger cylinder. Rod loading may be adversely affected, however.

Valve porting also influences volumetric efficiency by contributing to the clearance volume. If the porting must be enlarged to reduce the flow loss, it is done at the expense of clearance volume and a reduction in volumetric efficiency.

Compressor specifications often refer to the average inlet valve velocity as a general index of valve performance. The average inlet valve velocity can be used to make very generalized comparisons of compressor offerings with respect to valve performance. Generally, the lower the average inlet valve velocity, the lower the power loss as a result of valve losses.

The average inlet valve velocity is calculated from the cylinder displacement and the total valve lift area of all the suction valves in the cylinder. The following formula is used to calculate the average inlet valve velocity:

$$V = \frac{288V_d}{A} \quad (\text{Eq. 300-29})$$

where:

V = average gas velocity, FPM

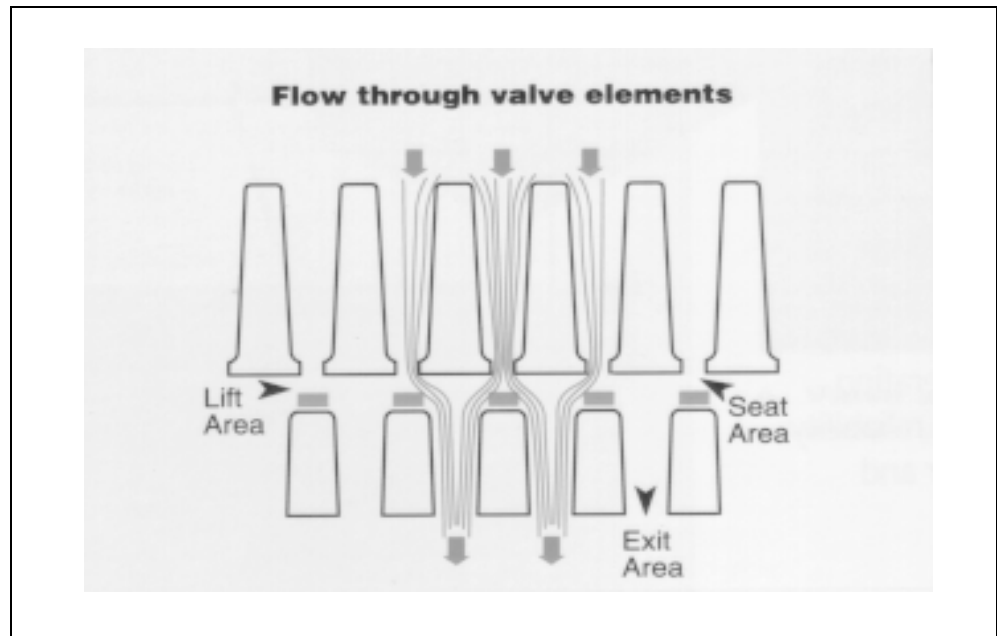
V_d = cylinder displacement rate, CFM

A = product of the actual lift, valve opening periphery, and the number of suction valves per cylinder, Square Inches

Figure 300-17 shows the lift area in a plate type valve. The product of the actual lift and valve periphery is the valve lift area. These values are furnished by the compressor valve manufacturer. When the valve lift area is not the smallest area in the valve flow path, the average inlet valve area is calculated on the basis of the smallest area.

Note The 288 factor in the valve velocity formula in Paragraph 2.7.1 of API 618 is correct. For many years API, GPSA, and other references erroneously used a constant of 144 in the formula. The 144 was wrong because the original formula was based on all suction valves of a cylinder being open at the same time. Actually, only half the valves are open at any given time in the cycle.

In reviewing bids for compressors, look for **large** differences in average valve velocities among the proposed machines. Small differences such as 5000 versus 6000 FPM are probably not significant. But if one vendor consistently quotes velocities twice those of another vendor, find out why. Also, determine whether they are quoting “average” or “effective” valve velocities.

Fig. 300-17 Lift Area of a Plate Valve (*Courtesy of Dresser-Rand*)

340 Machine Components and Cylinder Arrangements

341 Machine Components

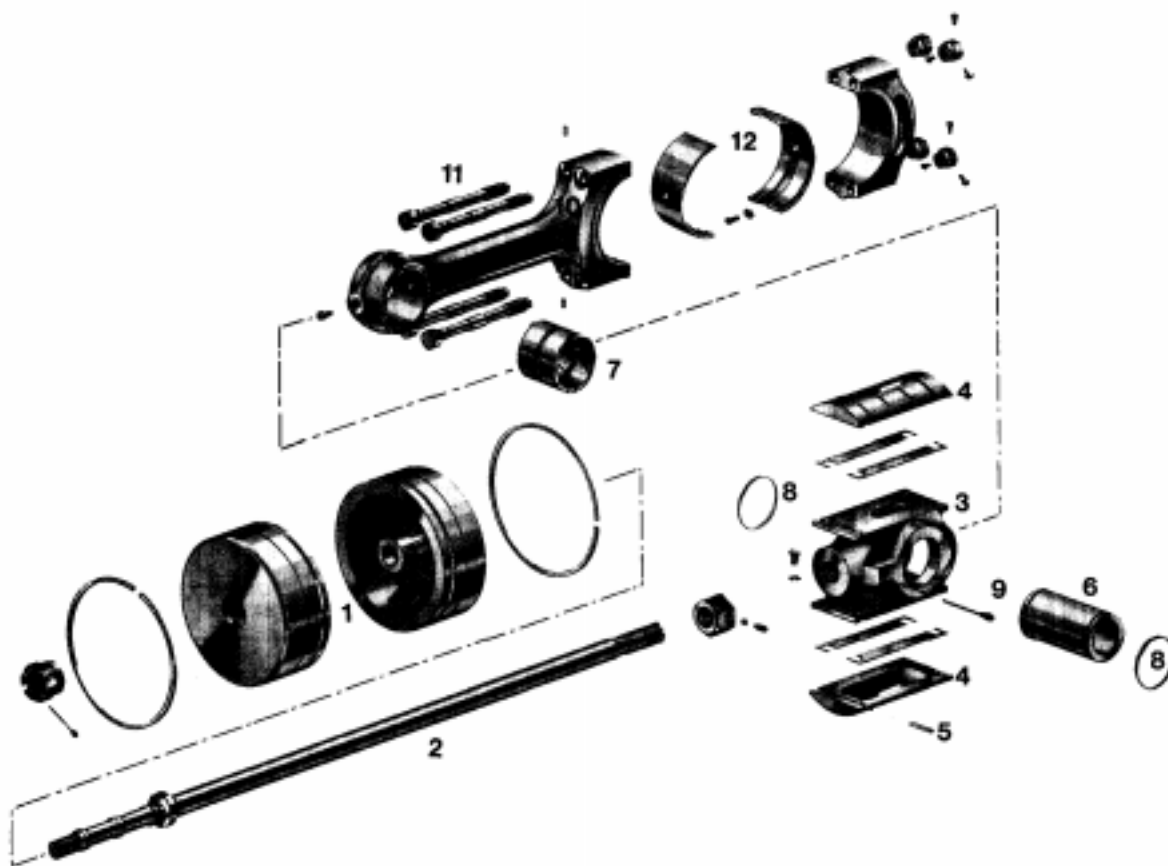
The machine components of a reciprocating compressor are:

- Running Gear
- Pistons
- Piston-to-Rod Attachment
- Crankshafts
- Piston Rod
- Unloaders and Clearance Devices
- Compressor Valves
- Compressor Cylinders
- Piston Rings and Rider Rings
- Packing
- Distance Pieces
- Bearings

Running Gear

The internal moving parts are often called the **running gear**. A typical set of running gear is shown in Figure 300-18. Note that a balancing weight(s) (usually a circular plate, not shown) can be added at the threaded end of the crosshead.

Lubrication of Running Gear. Lubrication is normally accomplished with a pressurized system using the sump of the compressor frame as the reservoir. API 618 describes requirements for the system components. Although many systems do not

Fig. 300-18 Typical Running Gear (Courtesy of Dresser-Rand)

1. **Piston** - two-piece design shown
2. **Piston Rod** - special material and surface finish. Rolled threads are typical
3. **Crosshead** - steel or nodular iron casting
4. **Crosshead Slippers** - Babbitt or aluminum-faced and grooved for lubrication - shim adjustable for piston-rod alignment
5. **Crosshead Slipper Key** - Positive locking device
6. **Connecting Rod** - Heat-treated, high-tensile strength steel forging, rifle drilled for positive crosshead lubrication
7. **Crosshead-pin Bearing** - Removable bushing (usually bronze), grooved for a lube pattern with locking device to prevent turning
8. **Crosshead-pin Retainers**
9. **Dowel Pin** - To prevent turning of piston rod
10. **Connecting Rod** - Heat-treated, high-tensile strength steel forging, rifle drilled for positive crosshead lubrication
11. **Connecting-rod-bolts** - Aircraft quality alloy steel

necessarily meet the requirements of API 618, the diagram included in API 618 shows a typical system schematic arrangement.

Whether a single oil filter or duplex filters are to be furnished is usually a matter of plant preference, based on the criticality of the service and number of machines applied to the service. Unlike centrifugal-compressor systems, twin oil coolers are seldom used in reciprocating compressor systems.

Splash type systems are sometimes used to lubricate the running gear of smaller machines (200 HP or less).

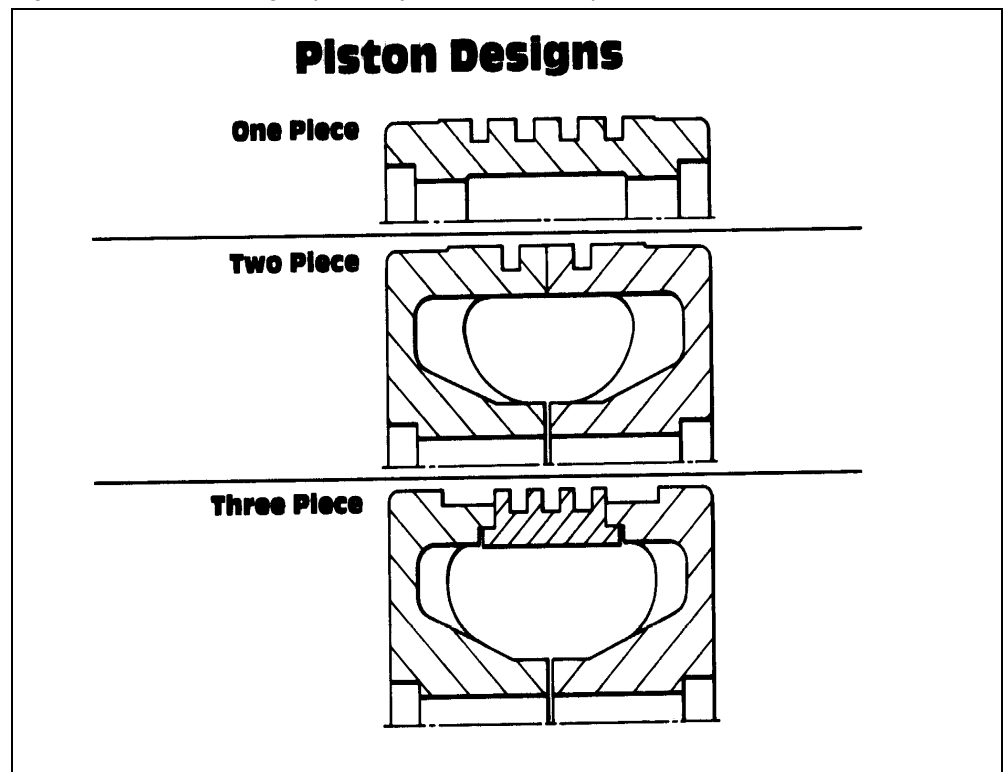
Compressor cylinder and packing lubrication is provided with force-feed mechanical lubricators. There are two types of lubricator systems: the multiple pump type (also called “single-plunger-per-point”), and the distributor type (also called “divider block”).

These lubricator systems are described further in Appendix J.

Pistons

One-piece pistons are made of cast iron, or steel for small diameters and high-differential pressures. They are also sometimes used when it is necessary to add weight for balancing to reduce reciprocating shaking forces. (Figure 300-19 illustrates three styles of pistons.)

Fig. 300-19 Piston Designs (Courtesy of Dresser-Rand)



Two-piece designs are used for ease in casting and weight control. They are made of aluminum or cast iron, and are generally applied for diameters above 10 inches. Aluminum is used to reduce reciprocating mass.

The **three-piece segmental design** incorporates a ring carrier. They are used to facilitate installation of rider rings (wear bands) which, when required, are placed on each side of the carrier. In this way, the rider band can be thicker because it does not have to be stretched over the outside diameter of the piston.

Another piston design worth mentioning is the labyrinth piston for non-lube service. This piston has no rings, and is used only in vertical machines. It has a large number of labyrinth grooves to limit blow-by. The blow-by is slightly more than that of ringed pistons, but the advantage is in low piston maintenance. The labyrinth compressor is manufactured only by Sulzer in Switzerland, and is used in rather special services such as oxygen at relatively low pressures.

Piston-to-Rod Attachment

Rods are attached to pistons with a single extension of the rod through the piston (see Figure 300-18), or with multiple through-bolts as shown in Figure 300-20. The advantages of multiple bolts are:

1. The smaller bolts are much easier to accurately tension.
2. Adequate pre-stress levels are reliably maintained.
3. Loading by the bolts is more evenly distributed in the piston.

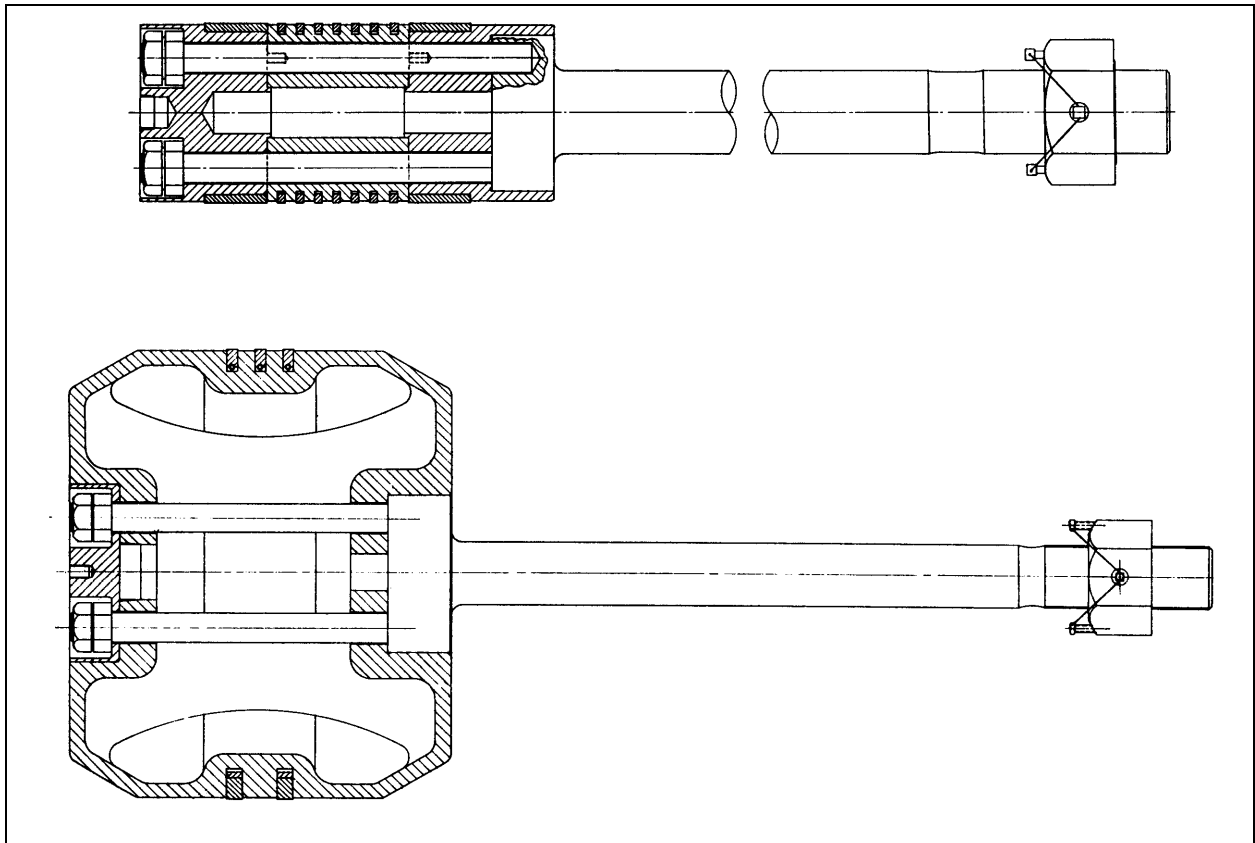
This feature is especially useful for large diameter aluminum pistons with large diameter rods. Such pistons with the single through-bolt attachment sometimes have nut-loosening problems after a number of temperature cycles. Factors contributing to this looseness are the difficulty of properly torquing one large nut, and possible non-squareness of the nut's face with the piston surface.

Crankshafts

Crankshafts are one-piece forgings or castings, although provisions are usually made for removable counterweights.

Reciprocating machines have two kinds of motion - rotational and reciprocating (translational). In **rotational** motion, a rotating force is caused when there is an unbalanced weight at some distance from the center of rotation. The imbalance involved in the rotational motion of one crankthrow of the compressor consists of the weights of the crankpin, crankshaft webs, and a portion (usually about 2/3) of the connecting rod. Counterweights are sometimes used to compensate for the off-centered weights of these components. (Figure 300-21 illustrates these terms.)

The components involved in **translational** motion are the piston, piston rod, cross-head, and the remaining portion (usually about 1/3) of the connecting rod. A fluctuating force results when these parts are accelerated and decelerated as the piston travels back and forth.

Fig. 300-20 Pistons with Multiple Thru-bolts (*Courtesy of the Cooper Cameron Corporation*)

For a single-cylinder compressor, the forces caused by both kinds of motion can be resolved into two sets of forces, primary and secondary, acting both horizontally and vertically.

Primary forces result from the **rotational** motion, and their frequency is that of running speed. **Secondary forces** result from translational motion, and their frequency is two times running speed due to the acceleration and deceleration during each stroke of the piston. Secondary forces act only along the axis of the cylinder.

Now, if a horizontally opposed compressor has two cylinders, a force **couple** can be generated by the unbalanced force of each cylinder acting in opposite directions and separated by the distance between the crankthrows. Figure 300-22 shows a primary couple for a **two-throw** machine having equal reciprocating weights on each throw. It also shows how counterweights can be added to the crank webs to reduce the primary couple.

Pistons on opposite adjacent throws are often not of the same diameter, so their weights are unequal. Figure 300-22 shows the location where a balance weight could be added to equalize the reciprocating weights. Dissimilar piston materials can also be used to equalize the weights.

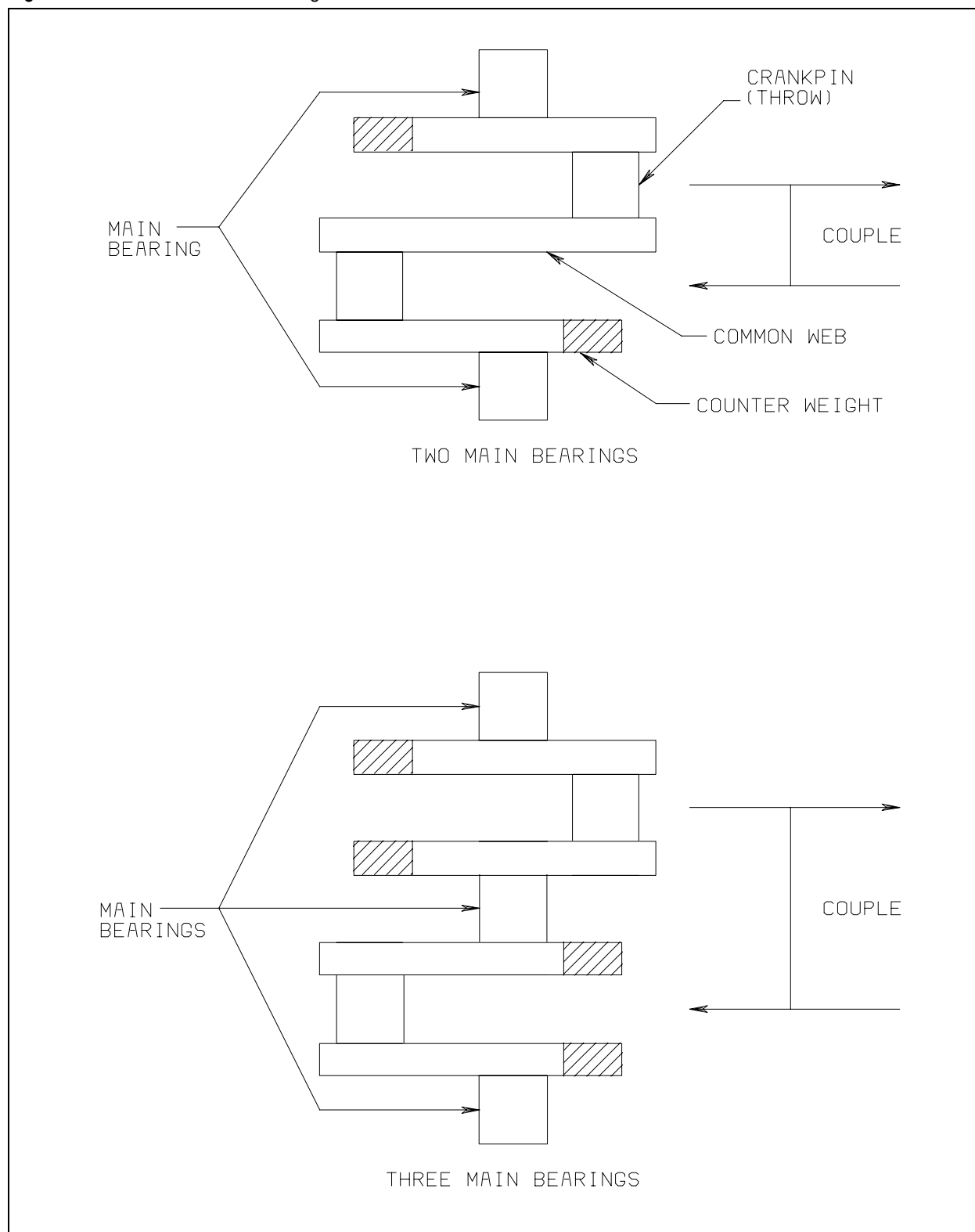
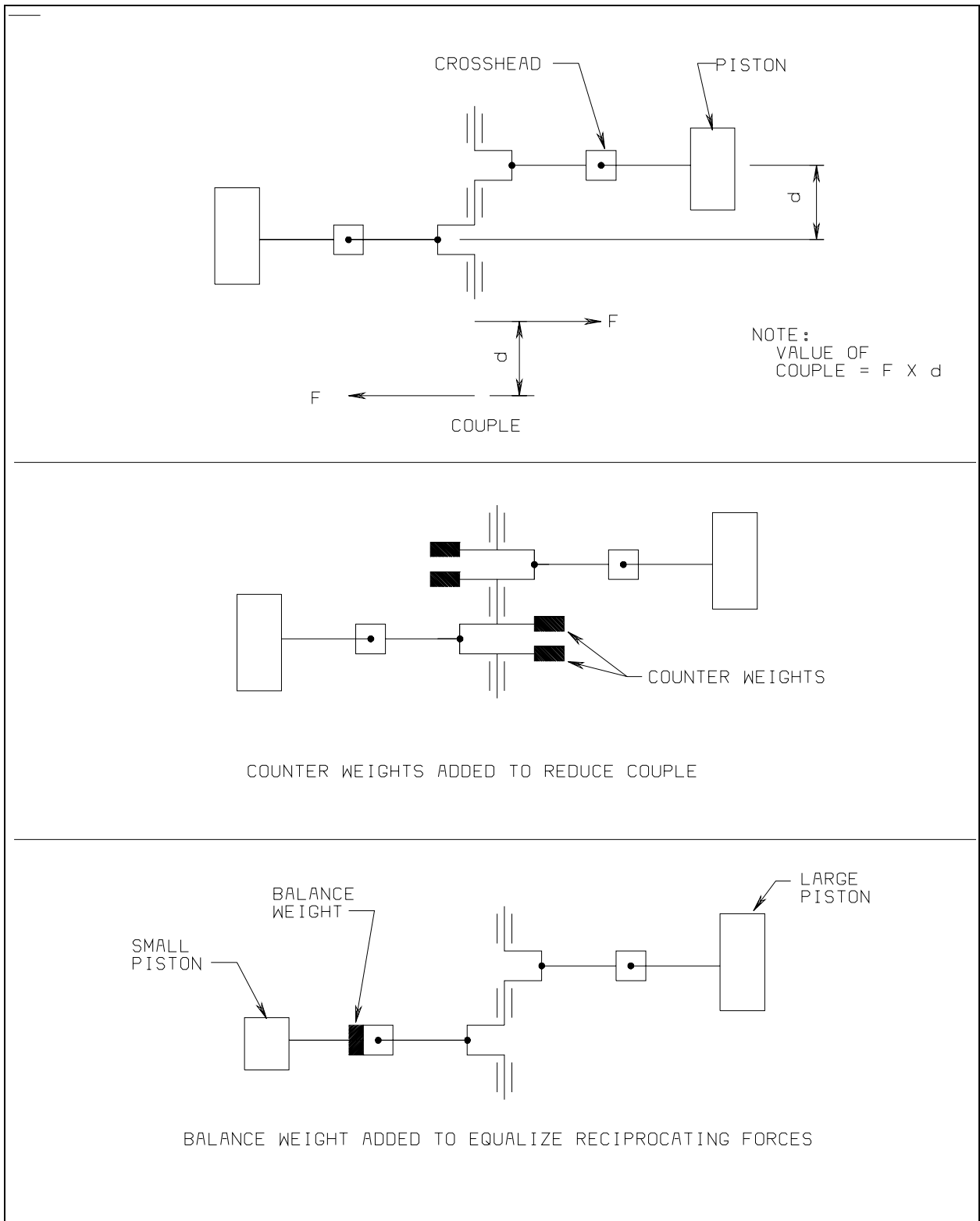
Fig. 300-21 Number of Main Bearings

Fig. 300-22 Counter Weights and Balance Weights

It is seldom practical to fully compensate for forces and couples with counterweights and balance weights. The design becomes more complex where the machine has more than two throws. The resultant magnitudes of the unbalanced forces and couples, then, depend on:

- the number of throws,
- the angular orientation of the crankpins with respect to each other,
- the distance between the throws,
- the difference in reciprocating weights, and
- the amount of counterweighting that can be applied.

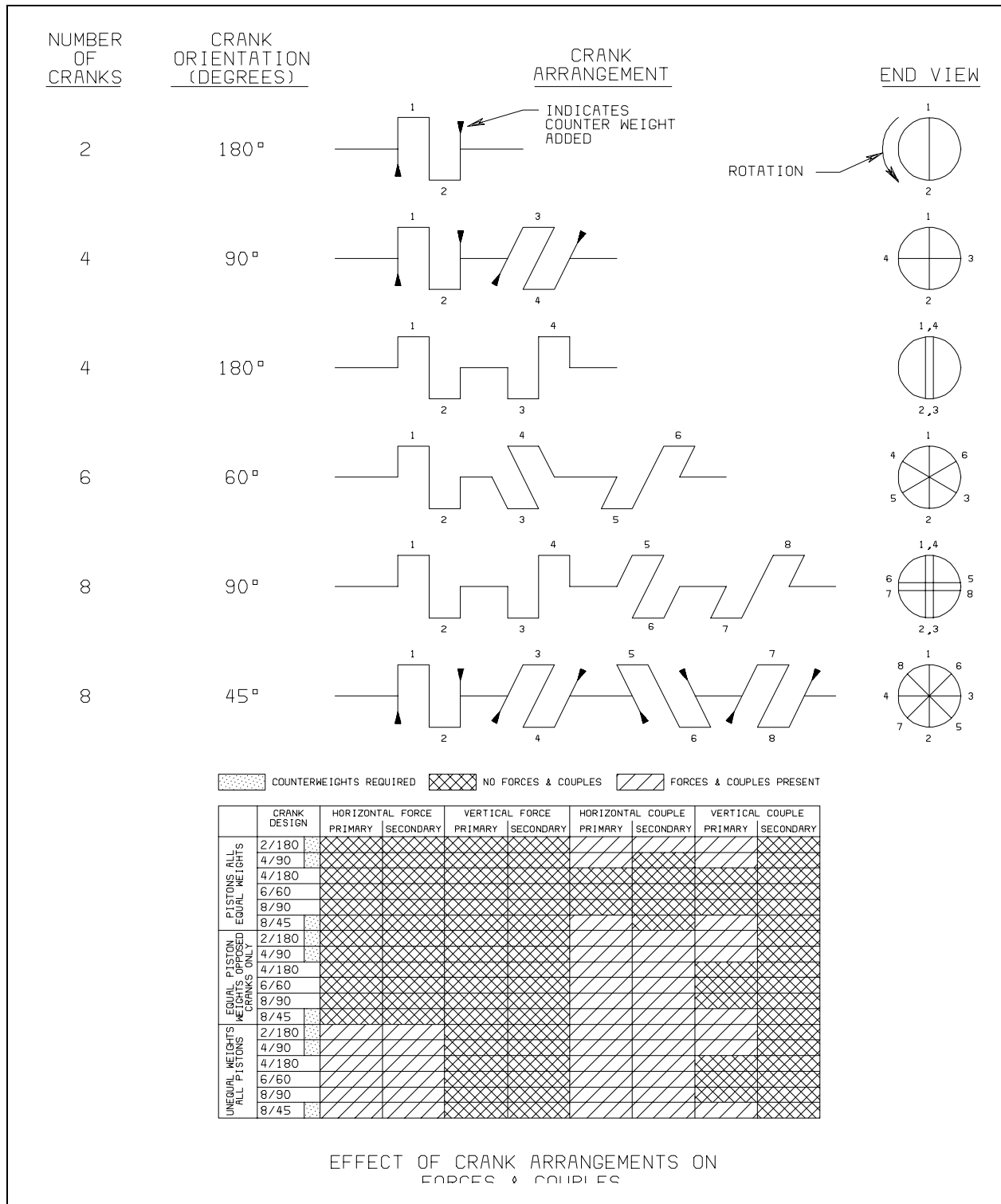
It is probably possible to balance the reciprocating weights on a pair of adjacent throws, but to have identical weights for all throws of a machine with several stages is seldom practical.

Figure 300-22 shows the simple case of a two-throw machine with a crankpin orientation of 180 degrees. As the number of throws increases, the effect of crankpin orientation on forces and couples gets quite complicated. Figure 300-23 qualitatively shows these effects for some of the more common crank arrangements with equal and unequal reciprocating weights.

Selecting the best arrangement from Figure 300-23 is not always the complete answer to the matter of shaking forces and couples. For example, for a four-throw machine, the 180 degree “flat” crankshaft is obviously the best choice from the standpoint of shaking forces and couples, and it eliminates the need for counterweights. However, the “flat” crankshaft causes all cylinders to be compressing at the same time. Hence, the torque-effort diagram (see Section 100, Figure 100-35) of the compressor may have undesirable oscillations from the standpoint of the driver. The “flat” crankshaft might require the addition of a large flywheel effect to the driver system, larger crankshaft diameter, and a special coupling to attenuate the oscillations. Therefore, the 90-degree arrangement might be more economical for an application that is not sensitive to shaking forces and couples. On the other hand, if the application is offshore, or onshore with undesirable soil conditions, the “flat” crankshaft will likely be the best choice.

A question that often arises concerns the number of main bearings. There are two configurations in Figure 300-21. One has two main bearings and a common web between the throws; the other has three main bearings, both for a two-throw crankshaft.

The advantage of the two-bearing design is that the throws are closer together, so the magnitude of the couple is less than that of the three-bearing design. With the three-bearing design, however, the crankshaft is more uniformly supported so that shaft deflection and bending stresses are less than that of the two-bearing design. The three-bearing arrangement has more places to attach counterweights to reduce the primary couple, but can the added complexity be justified? Opinions vary, but the fact remains that both designs are common, and both work.

Fig. 300-23 Effect of Crank Arrangements on Forces & Couples (typical only) (Courtesy of Dresser-Rand)

Note that with a main bearing between each throw, the frame can have odd and even numbers of throws. When two throws have a common web, only even numbers of throws are possible.

Piston Rods

The piston rod (Figure 300-18) is a principal component in the running gear. The primary design factor regarding piston rods is a phenomenon loosely called “**rod loading**” described in detail in Section 351. As the rod is subjected to high alternating stresses (compressive and tensile), its surface finish must be extremely smooth to avoid stress intensification, which could lead to fatigue failure. The threads on the rod where it attaches to the crosshead are critical for the same reason.

For hardness and surface finish specifications for commonly used rod materials, refer to API 618 and/or Section 800. For clean non-corrosive gas service, AISI-4140 carbon steel is a typical rod material. For this material, guidelines for the allowable stress at the root area of the threads at the crosshead are:

Thread Type	Stress, psi
Cut	6000
Cut-and-ground	7500
Rolled	8000

Cut threads have a higher stress intensification factor.

These guidelines may be used by dividing the vendor’s maximum allowable continuous rod load rating (in pounds) by the thread root area (square inches). Obviously, if the rod material is not 4140, the guideline stress values should be adjusted by the ratio of tensile strengths of the two materials. Rolled threads are preferred for all applications, but may not be available on some of the smaller machines.

Some new piston rods are provided with metal coatings for certain services. Metal coatings are also used to recondition rods. A summary of Company experience with piston-rod reconditioning is included in Section 830.

Unloaders and Clearance Devices

Unloaders and clearance devices are used to:

- Unload the compressor for startup;
- Maintain fixed suction or discharge pressure in a process;
- Control capacity (flow rate);
- Maintain optimum loading on driver under varying compressor operating conditions; and
- Maintain operating pressures, temperatures, or piston rod loads within allowable limits under varying compressor operating conditions.

Unloaders are applied to the **suction side** of a compressor cylinder to deactivate one or both ends of the cylinder. Unloaders are installed on suction valves only, so

the gas circulating through the valve will be cooler than if the unloaders were installed on discharge valves. Metal fatigue and attendant valve failure occurs more quickly at elevated temperatures. The two types of unloaders, **plug** and **finger**, are shown in Figure 300-24.

There are two kinds of plug-type unloaders (as shown in Figure 300-24, Items A and B). Item A is fitted on a suction valve. Item B is applied to an individual gas port on the suction side of the cylinder. With either kind, when the unloader is actuated, an opening is maintained during both the suction and discharge strokes. With this opening, the gas is no longer trapped, but is free to flow back to the suction side during the compression stroke. Normally, only one plug-type unloader is required on a cylinder end. Either kind can be operated manually or automatically with air or gas as the motive fluid.

The finger type suction valve unloader (Item C of Figure 300-24) is applied to all suction valves on a cylinder end. This type is sometimes called a “suction valve depressor”. When actuated, the fingers push down on the suction valve plate(s), and hold them open. Again, the gas flows back to suction during the compression stroke.

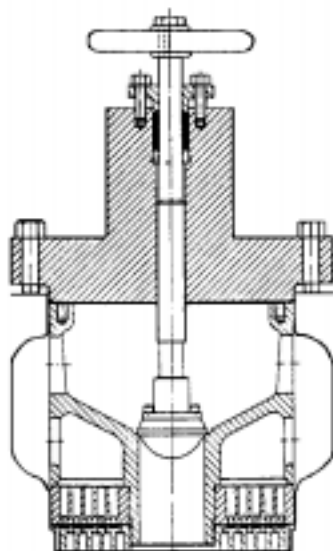
The plug-type tends to be more reliable than the finger-type. Valve plate failure can be caused by the concentrated loads where the fingers push on the plate. The disadvantage of using plug-type unloaders is that there is less overall available valve area in the cylinder, resulting in higher valve losses.

One European compressor valve manufacturer markets a “stepless” controller for their valve unloaders. In this scheme, the finger type unloader holds the suction valve plate open for a specific length of time corresponding to the desired capacity. Since the suction valves are not allowed to close at the normal time (at or near bottom dead center), in effect, the displacement of that end of the cylinder is reduced. Therefore, it is theoretically possible to provide an infinite number of capacity steps depending on valve timing.

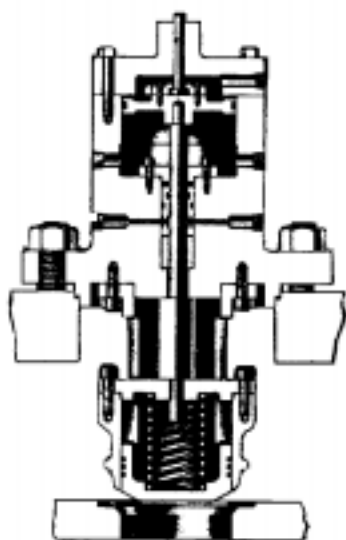
The stepless system has been applied to quite a few installations in Europe, but it has not gained much favor in the domestic petroleum applications due to its complexity. One USA manufacturer offers a similar stepless control system, but it is typically applied only to storage (pump-in, pump-out) compressors having continually varying suction and discharge pressures.

There are two kinds of clearance devices: **clearance pockets** and **clearance spacers**. Pockets can be **fixed** or **variable volume**. (Figure 300-25 shows examples of fixed and variable volume pockets.) Fixed volume pockets can be either manually or pneumatically (air or gas) operated.

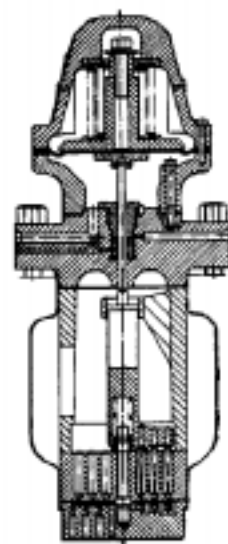
Variable volume pockets are normally only manually operated. Hydraulic actuation has been tried, but the results have generally been unfavorable. Variable volume pockets are applied only to the head end of a cylinder, because it is physically impossible to locate them on the crank end. Occasionally, variable-volume pockets may be capable of more adjustment than required for the load step. Opening the pocket too far can cause overheating problems because the volumetric efficiency becomes too low. External mechanical stops can be installed to limit handwheel travel to the length required in the capacity control design.

Fig. 300-24 Unloaders

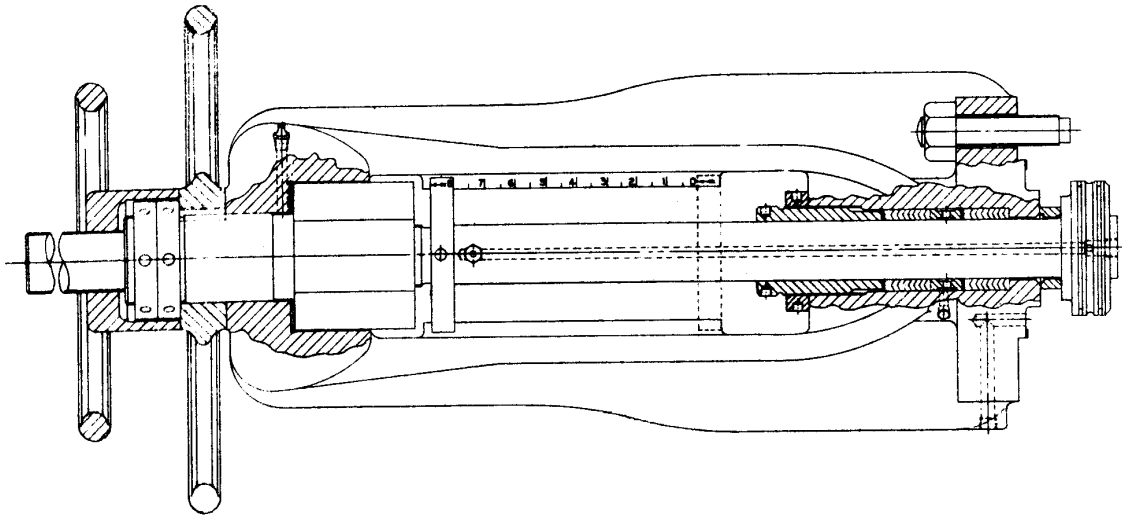
A. Plug-type suction valve unloader—manual operation
(Courtesy of the Cooper Cameron Corporation)



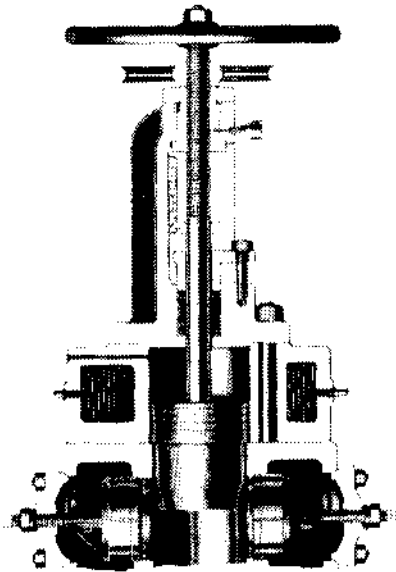
B. Plug-type suction valve unloader—
pneumatic operation
(Courtesy of Dresser-Rand)



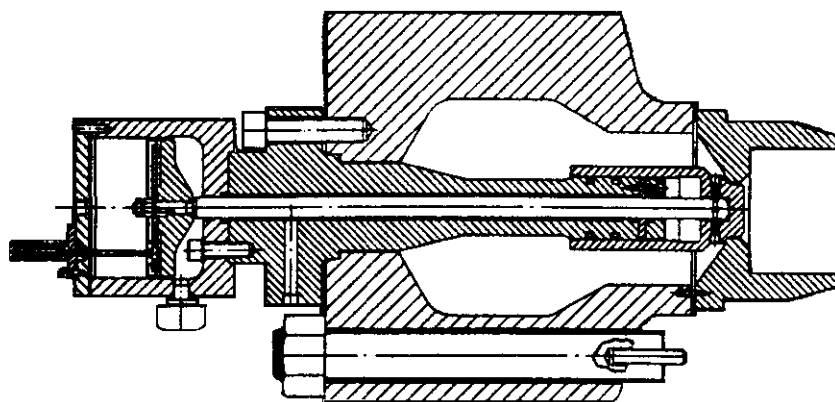
C. Finger Type Suction valve unloader—
pneumatic operation
(Courtesy of Dresser-Rand)

Fig. 300-25 Clearance Devices (1 of 2)

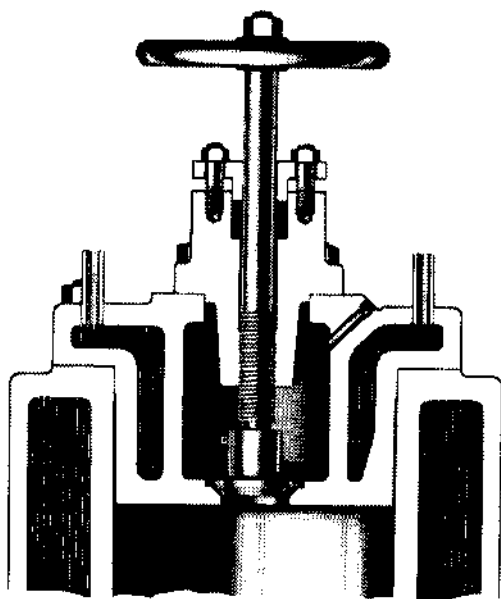
A. Manual variable volume clearance pocket—high pressure
(Courtesy of the Cooper Cameron Corporation)



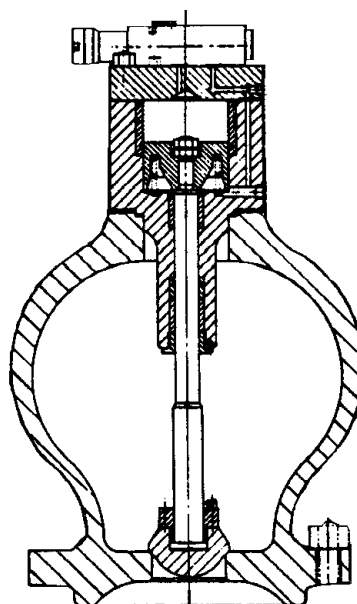
B. Manual variable volume clearance pocket—low pressure
(Courtesy of Dresser—Rand)

Fig. 300-25 Clearance Devices (2 of 2)

C. Pneumatically operated fixed volume clearance pocket
(Courtesy of the Cooper Cameron Corporation)



D. Manual fixed volume clearance pocket
(Courtesy of Dresser-Rand)



E. Pneumatically operated fixed volume clearance pocket
for valve cap
(Courtesy of the Cooper Cameron Corporation)

Fixed volume pockets are normally applied to the head end for the same reason, although fixed pockets can be applied to suction valve caps on either the head or crank ends. Fixed-clearance pockets often look like an ell-shaped piece of pipe, pointed upwards to prevent liquid accumulation. This type of pocket can be installed by the manufacturer, or as a field retrofit.

Valve cap pockets require a special suction valve with an opening at the center of the seat so that the pocket volume is in direct communication with the cylinder's interior when the pocket is opened. Installation of valve cap pockets on discharge valves is not recommended because of the possibility of collection of oil or liquids in the pockets.

Clearance spacers are sometimes used to increase the clearance volume. A spacer can be installed between the cylinder head and cylinder body. Also, a compressor valve can be designed with a two-piece yoke (hold-down for valve). By using one piece of the yoke as a spacer, the valve seat and guard can be moved farther away from the cylinder bore, thus increasing the clearance. A shutdown is required to change clearance with spacers.

Compressor Valves

Valves are manufactured by the original equipment manufacturer (OEM), and by non-OEM suppliers. The compressor will almost always be supplied with OEM valves, but non-OEM valves are often retrofitted to solve a particular problem, or when a unit is re-rated. Occasionally, non-OEM valves are specified on new machines to match existing equipment or to solve known problems. For more information on various types of compressor valves, see Section 354.

Compressor Cylinders

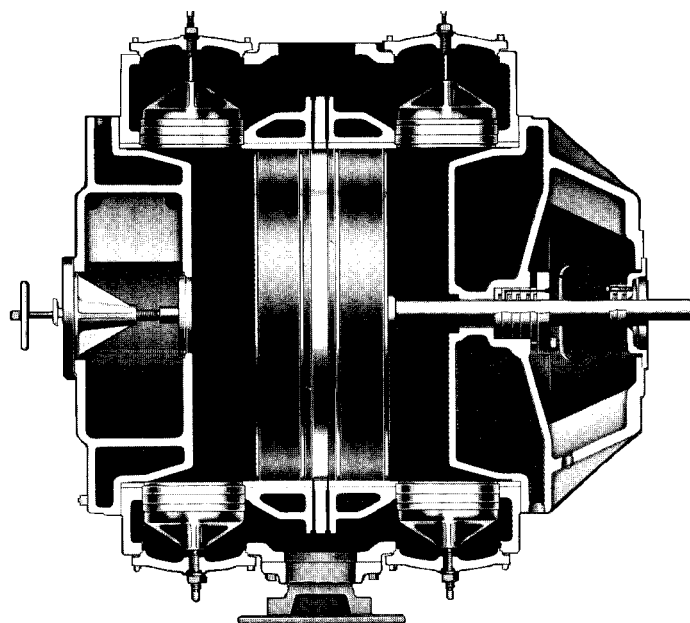
Cylinder design is dependent mainly on the working pressure and diameter range. Cylinder bodies can be castings of gray iron, nodular (ductile) iron, or steel. Forged and fabricated steel cylinders are also available for higher pressures. Figure 300-26 shows a variety of cylinders. All of these cylinders have liners. Many of the cylinders offered for high-speed compressors (Class B and D) do not have liners.

Note that cylinders are available in the tandem arrangement having two sizes of pistons. This rather uncommon arrangement allows two cylinders to be placed on one crankthrow. Usually both pistons are single-acting, but in some designs only one of the two will be double-acting.

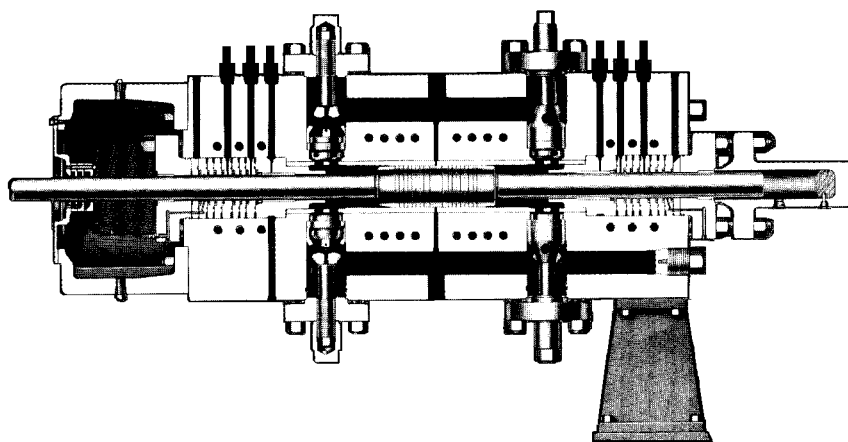
Piston Rings and Rider Rings

The purpose of **piston rings** is to prevent the blow-by of gas from one end of the piston to the other. **Rider rings** or **wear bands** support the weight of the piston, help guide the piston in the bore, and prevent rubbing of the piston on the cylinder wall. Some designers use nylon buttons in the piston skid to prevent contact with the cylinder on trunk-type pistons.

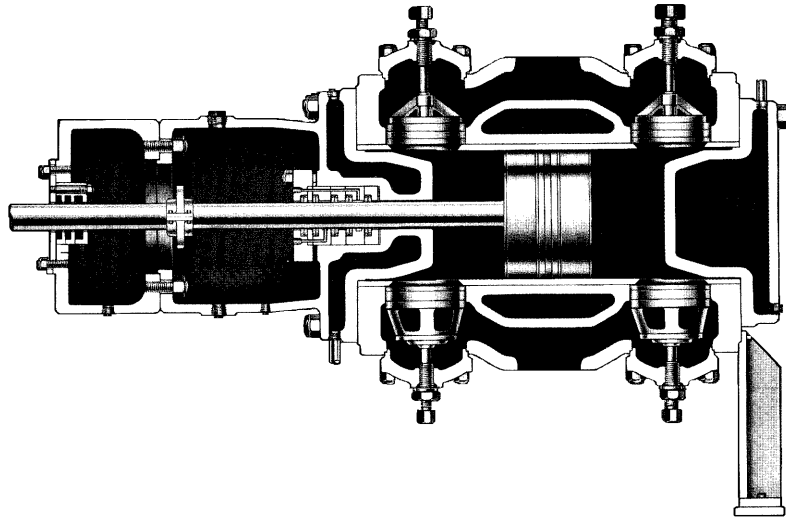
For many years, piston rings were made from non-abrasive, relatively soft metallic materials. Cast iron was the most common material, later largely replaced by

Fig. 300-26 Compressor Cylinders (1 of 2)

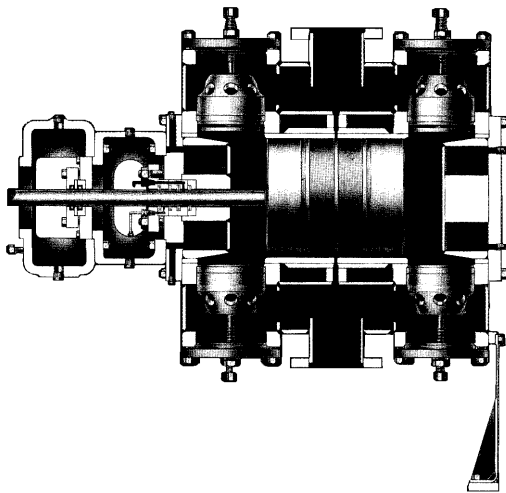
Large Cast Iron Cylinder for Low Pressure
(Courtesy of the Dresser-Rand)



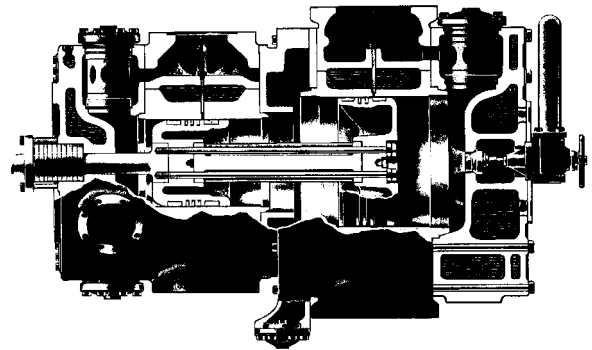
Forged Steel Cylinder with Tall Rod for High Pressure
(Courtesy of Dresser-Rand)

Fig. 300-26 Compressor Cylinders (2 of 2)

Non-lube Cast Iron Cylinder with Two-compartment Distance Piece for Medium Pressure
(Courtesy of Dresser-Rand)



Fabricated Steel Cylinder
for Medium to High Pressure
(Courtesy of Dresser-Rand)



Cast Iron Tandem Cylinder with
2-Volume Head End Pocket
(Courtesy of the Cooper Cameron Corporation)

bronze. Metallic rings were favored because of their good heat transfer characteristics.

However, much development of non-metallic piston rings and rider rings occurred when non-lube applications became common in the 1950's.

Carbon-graphite was tried first, but was found to be brittle and did not have suitable wear characteristics. Phenolic and laminated plastics such as Bakelite and Micarta were used when temperatures were low. Although it is relatively weak with poor heat transfer properties, PTFE (teflon) with various filler materials eventually became the favored material, because of its excellent low-friction characteristics. Today PTFE is used almost exclusively for lubricated, as well as nonlubricated services. Bronze is still used on rare occasions for clean and dry lubricated service when good heat transfer is needed.

The shapes of piston and rider rings are shown in Figure 300-27. Some designs call for an inner ring or expander ring (not shown) to be fitted under the piston ring to energize the piston ring and keep it against the cylinder wall as wear takes place. However, the most popular and safest design employs gas pressure to energize the piston ring.

The "angle" cut is generally preferred, and is the most commonly used. For smaller lower-pressure cylinders, the "step" cut is used, although care must be taken in the design to avoid joint breakage. The "seal" cut provides the best seal, but is more expensive.

Pressure in the cylinder acts on the piston rings, and assuming that the ring does some sealing, there will be a pressure drop from one side of the ring to the other. This pressure difference results in a net "pressure induced force" holding the ring against the side of the piston groove and outward against the cylinder bore (refer to Figure 300-28).

Figures 300-29 through 300-31 provide some typical dimension ranges for piston rings and piston clearance. The latter is governed mainly by the coefficient of thermal expansion of the piston material. In general, the ring should not protrude from the piston groove by more than 25% of its thickness.

Rider rings and piston rings are almost always of the same material. Rider rings must be designed so that they do not act as a piston ring. Otherwise, wear will occur too rapidly. Solid rider rings are not prone to outward expansion, but cut rider rings must be vented with holes or slots to bleed off pressure. Figure 300-32 and Figure 300-33 are examples of typical thicknesses for solid and cut rider rings versus cylinder diameter.

Rider-ring width is determined by the bearing pressure. Figure 300-34 shows piston ring and rider ring arrangements on the rider ring. The bearing pressure is generally limited to five psi for PTFE in non-lube services and 10 psi for lubricated cylinders (see API 618). These pressures are based on the weight of the piston plus one-half the weight of the rod divided by $0.87 DW$ (where D is the piston diameter, and W is the width of all rider rings on the piston).

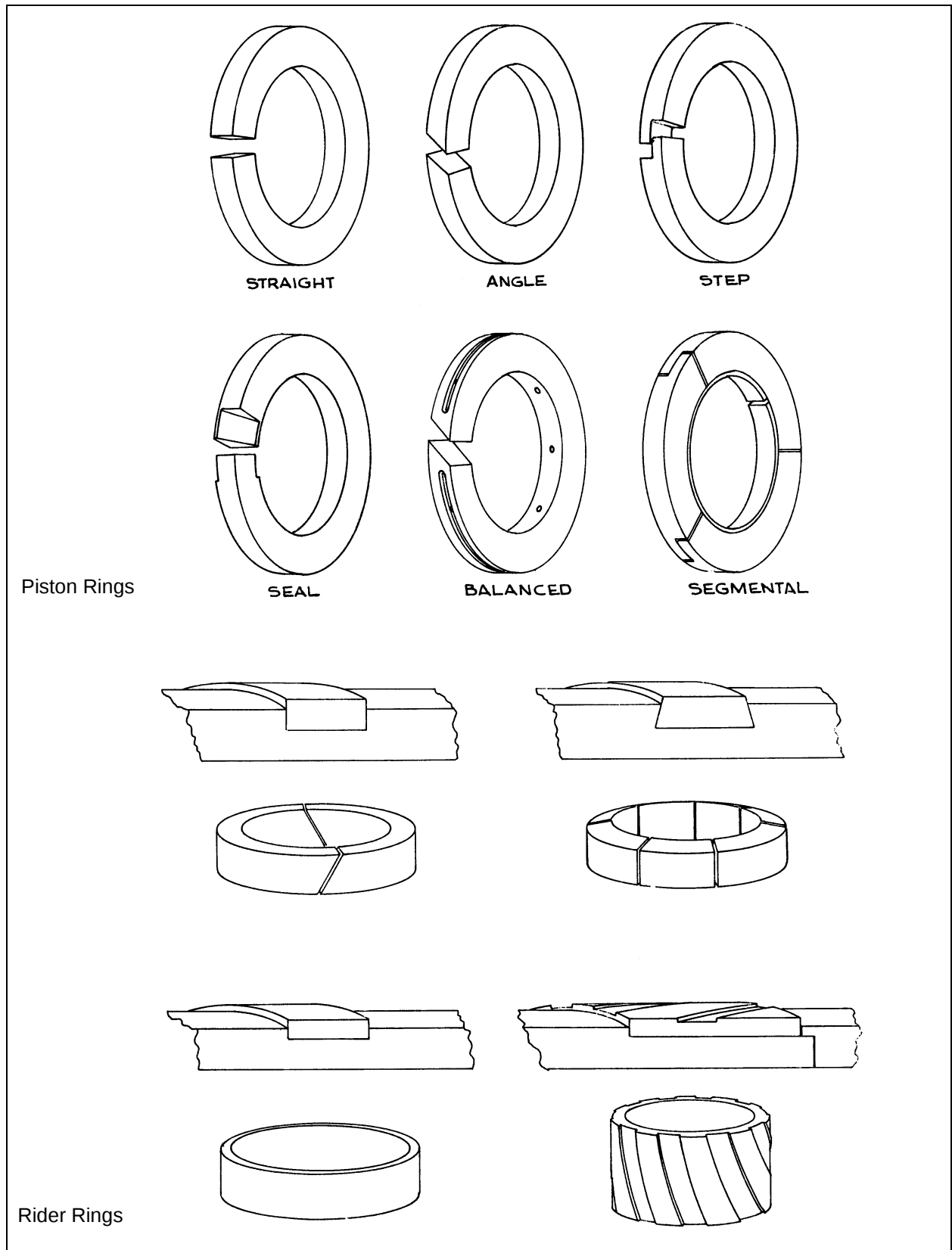
Fig. 300-27 Piston Rings and Rider Rings *(Courtesy of the Cooper Cameron Corporation)*

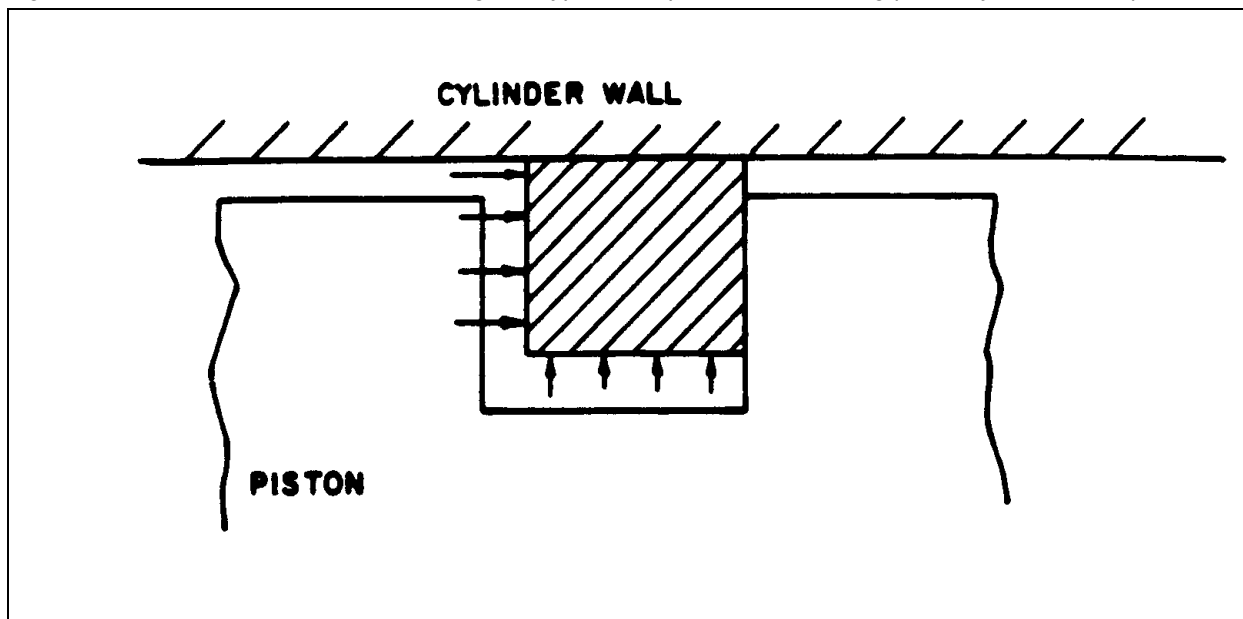
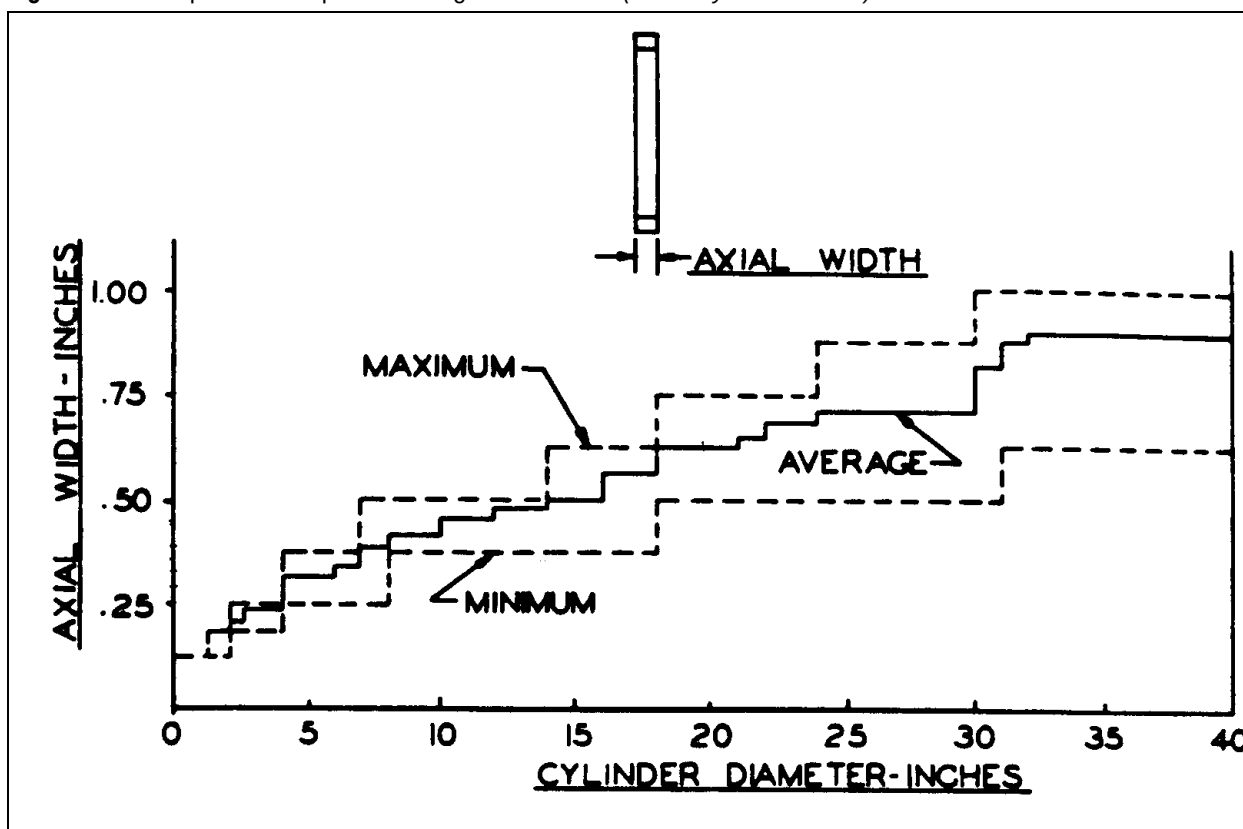
Fig. 300-28 Pressure Induced Forces Acting on a typical Compression Piston Ring (*Courtesy of the ASME*)**Fig. 300-29** Compilation Compression Ring—Axial Width (*Courtesy of the ASME*)

Fig. 300-30 Compilation Compression Ring—Radial Thickness (Courtesy of the ASME)

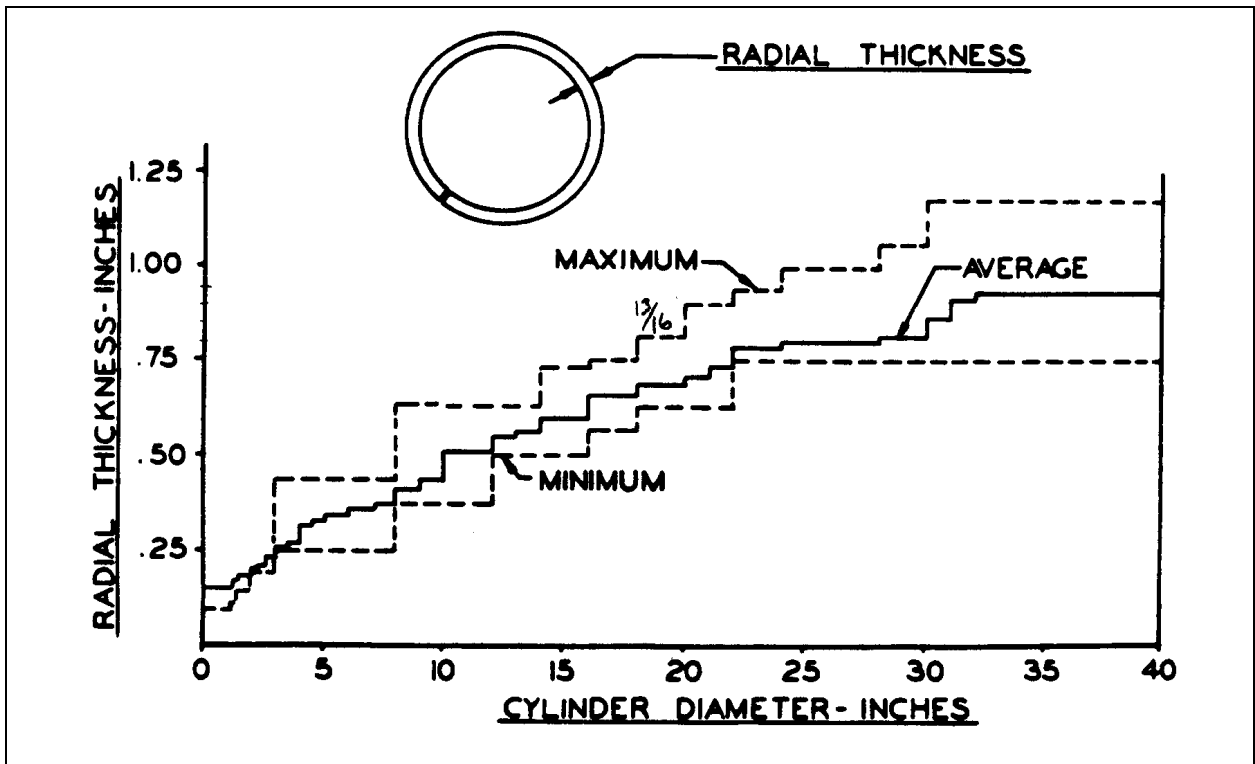


Fig. 300-31 Compilation Piston to Cylinder Clearance—A-B (Courtesy of the ASME)

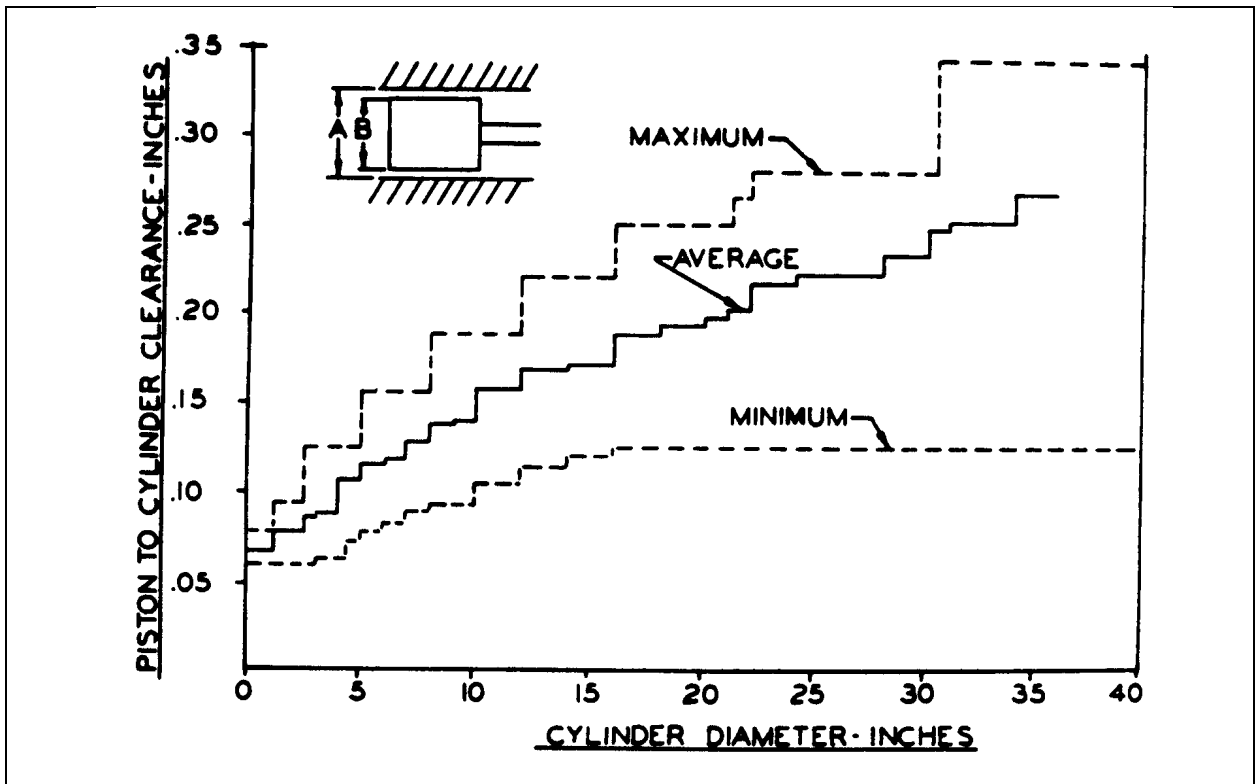


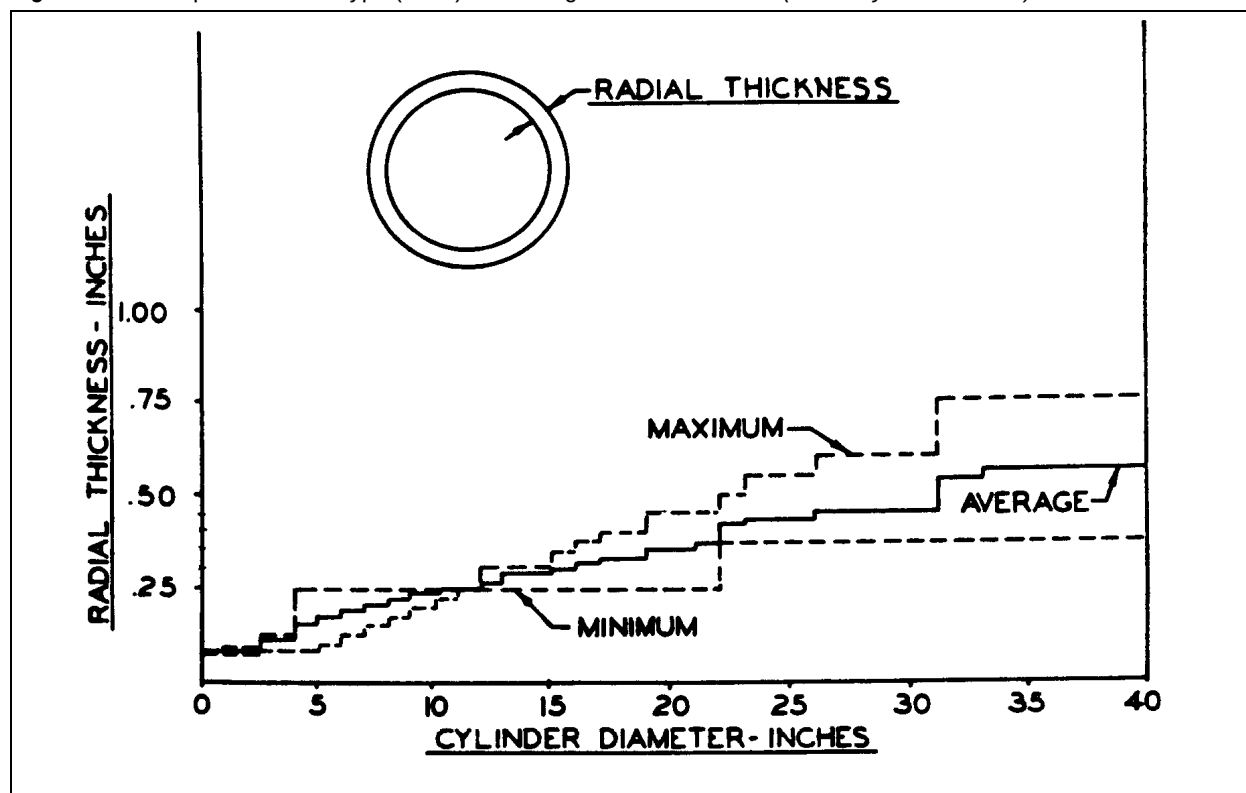
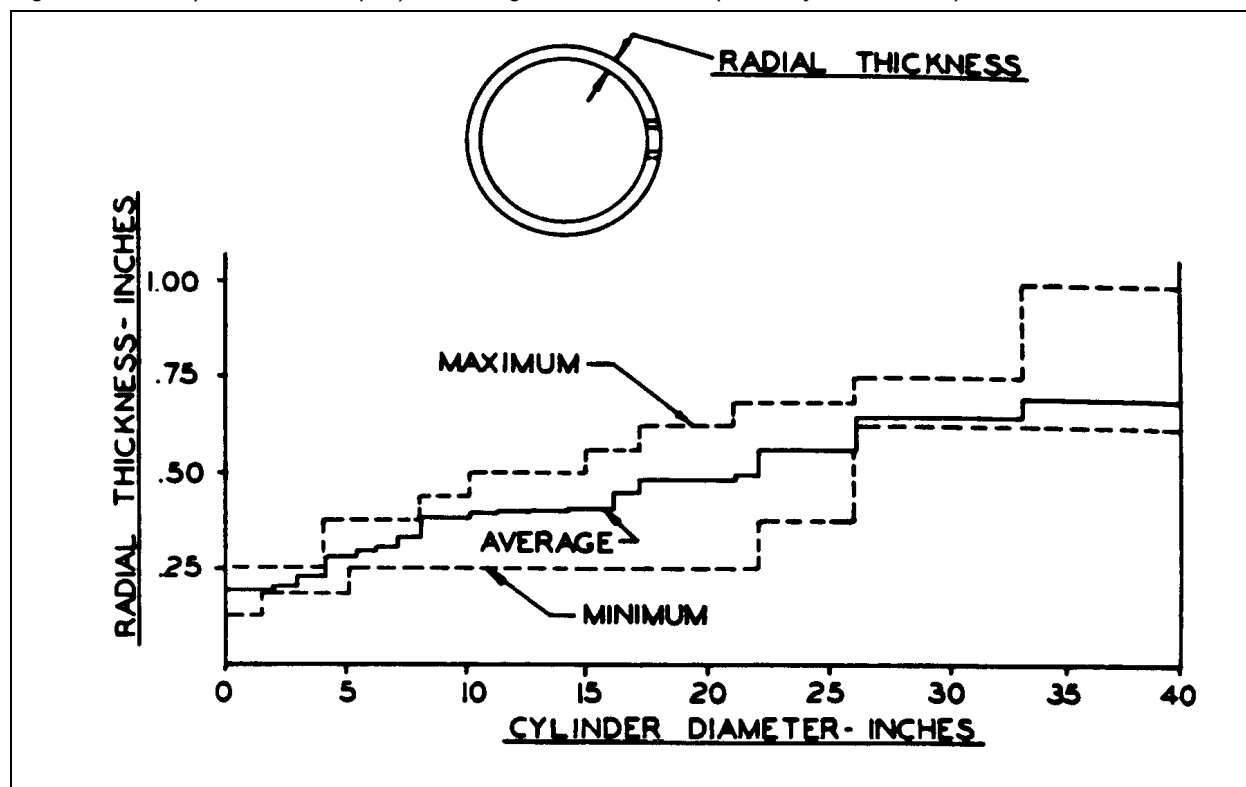
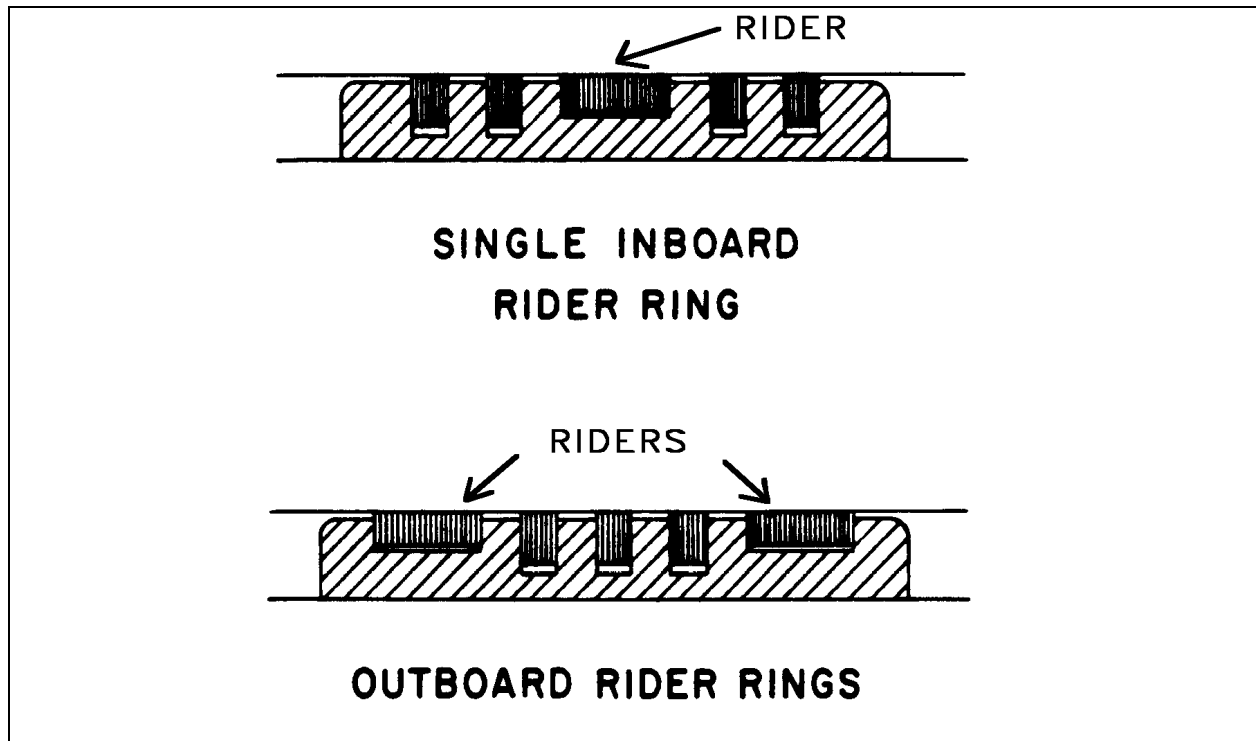
Fig. 300-32 Compilation Band-type (Solid) Rider Rings Radial Thickness *(Courtesy of the ASME)***Fig. 300-33** Compilation Joined (Cut) Rider Ring Radial Thickness *(Courtesy of the ASME)*

Fig. 300-34 Typical Arrangements for Piston & Rider Rings *(Courtesy of the ASME)*

Packing

On double-acting cylinders (and some single-acting), the piston rod must be sealed to reduce gas leakage from inside the cylinders. This seal is called **pressure packing**. It is of the **full-floating** design so that the packing rings follow any lateral motion of the piston rod.

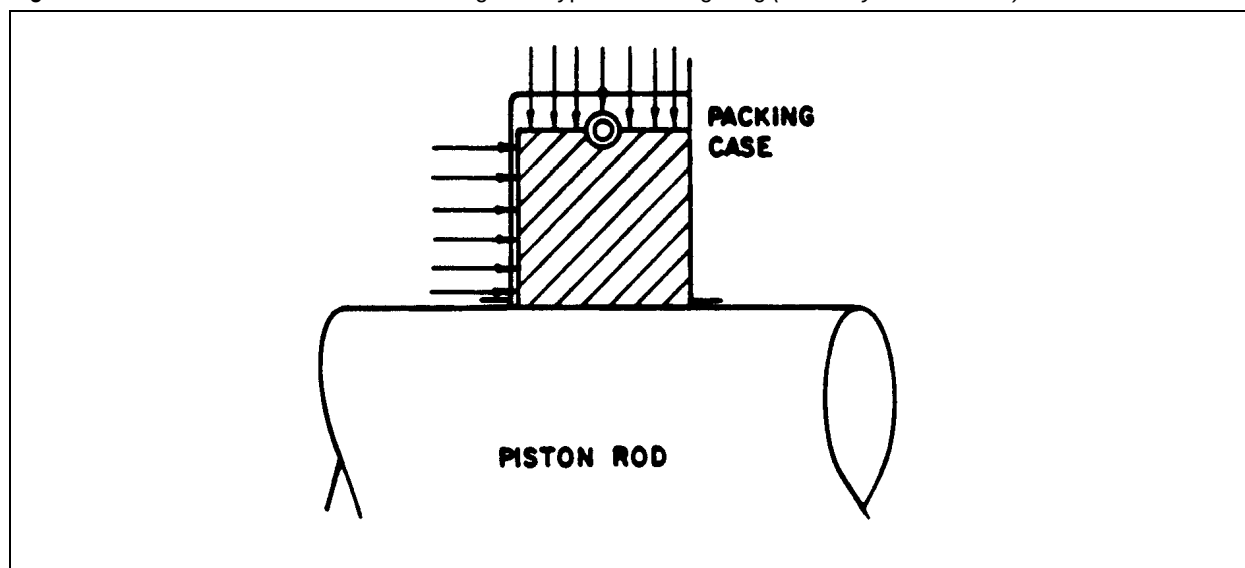
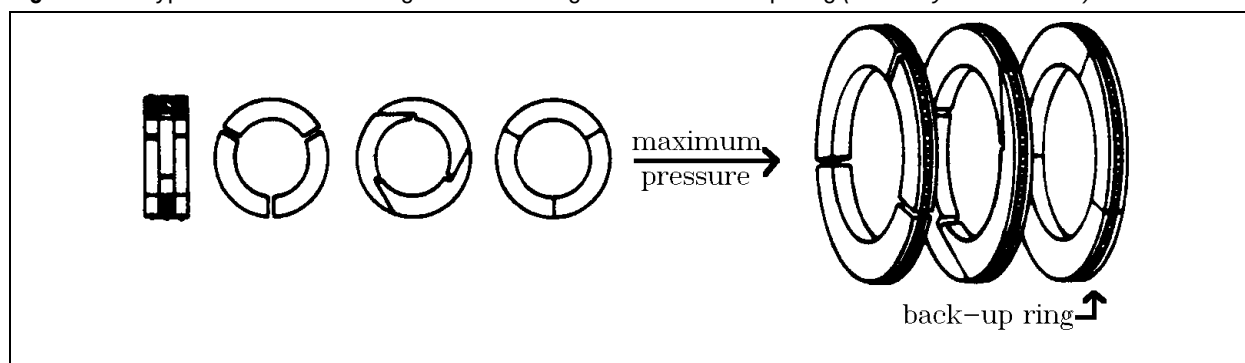
As with piston rings, PTFE is used extensively for packing rings. Figure 300-35 shows the forces on a packing ring. Figure 300-36 shows a typical arrangement of packing rings. The back-up ring limits deformation of the packing ring, and is usually not required below 500 psi. The back-up ring is sometimes made of bronze for better heat dissipation.

Packing cases with vent and buffer arrangements are shown in the Appendix of API 618. Figure 300-37 shows a packing case with passages for coolant. Guidelines for when cooled packing is applied are included in the text of API 618.

Wiper packing is also shown on the diagrams in API 618. The purpose of this packing is to minimize leakage of crankcase oil.

Distance Pieces

The **distance piece** (refer to Figure 300-1) separates the cylinder from the cross-head guide (doghouse). It provides a housing for both wiper packing and pressure packing. The distance piece prevents entry of compressed gas into the crankcase, and provides access for maintenance of the packing and piston rod.

Fig. 300-35 Pressure Induced Forces Acting on a Typical Packing Ring (*Courtesy of the ASME*)**Fig. 300-36** Typical Radial Cut & Tangent Cut Packing Sets with Back-up Ring (*Courtesy of the ASME*)

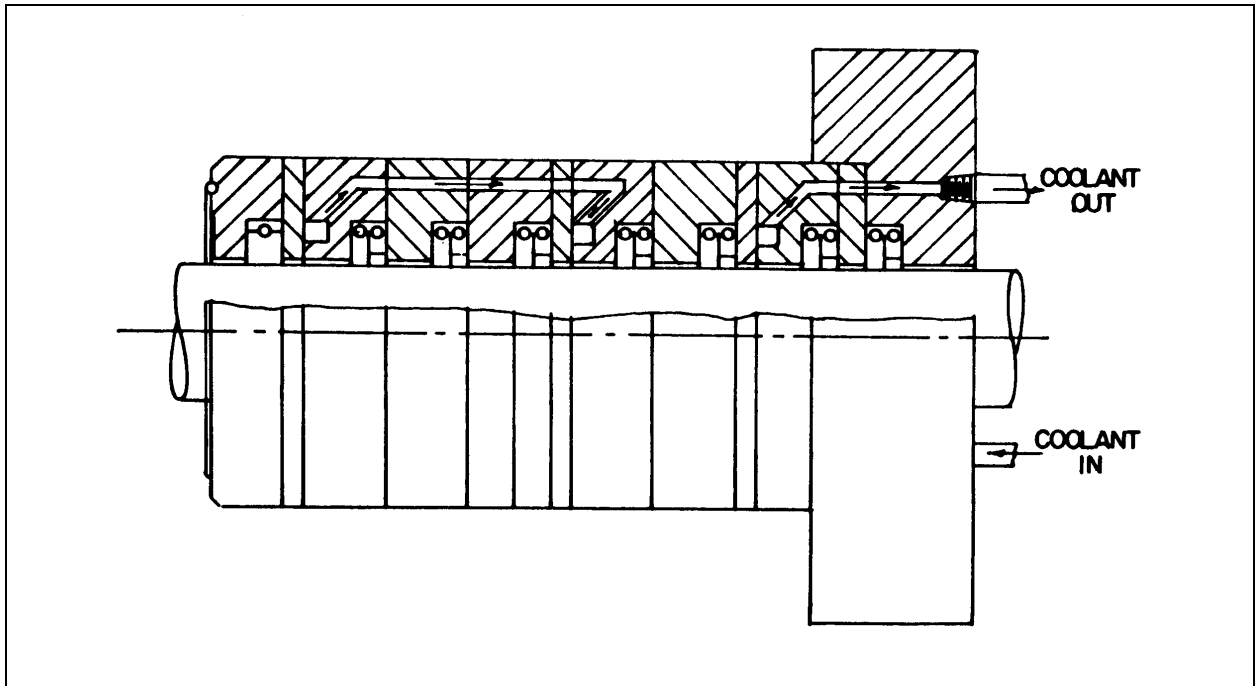
Four types of distance pieces (with corresponding diagrams) and their applications are covered in API 618. Vent and purge connections are shown on the API 618 diagrams. Further information on venting and purging is provided in Section 360, "Application and System Considerations."

Bearings

Main and connecting rod bearings for the larger heavy-duty frames are split-sleeve precision insert type (refer to Figure 300-18). The most common materials are cast iron/babbitt or steel/babbitt. Occasionally, aluminum bearings are used. Aluminum bearings require better oil filtration as they are sensitive to dirt.

Although sleeve bearings are the most common, API 618 allows rolling-element (anti-friction) main bearings for Class A machines up to a rating of 200 HP. Rolling-element bearings are used in some Class B machines for ratings in excess of 1000 HP.

Refer to the *General Machinery Manual* for additional information on both types of bearings.

Fig. 300-37 Pressure Packing Case with Coolant Passages (*Courtesy of the ASME*)

342 Cylinder Arrangements

Refer to Figure 300-1 for a single-cylinder compressor. However, most reciprocating compressors are multi-stage (refer to Figure 300-39). Even single-stage machines are often multi-cylinder for better balance. There are numerous arrangements of the cylinders. Figure 300-38 covers most of the possible arrangements. The horizontally opposed machine shown in the lower right-hand corner of this Figure is the most widely used configuration. Figures 300-39 and Figure 300-40 show other views of typical horizontally opposed machines.

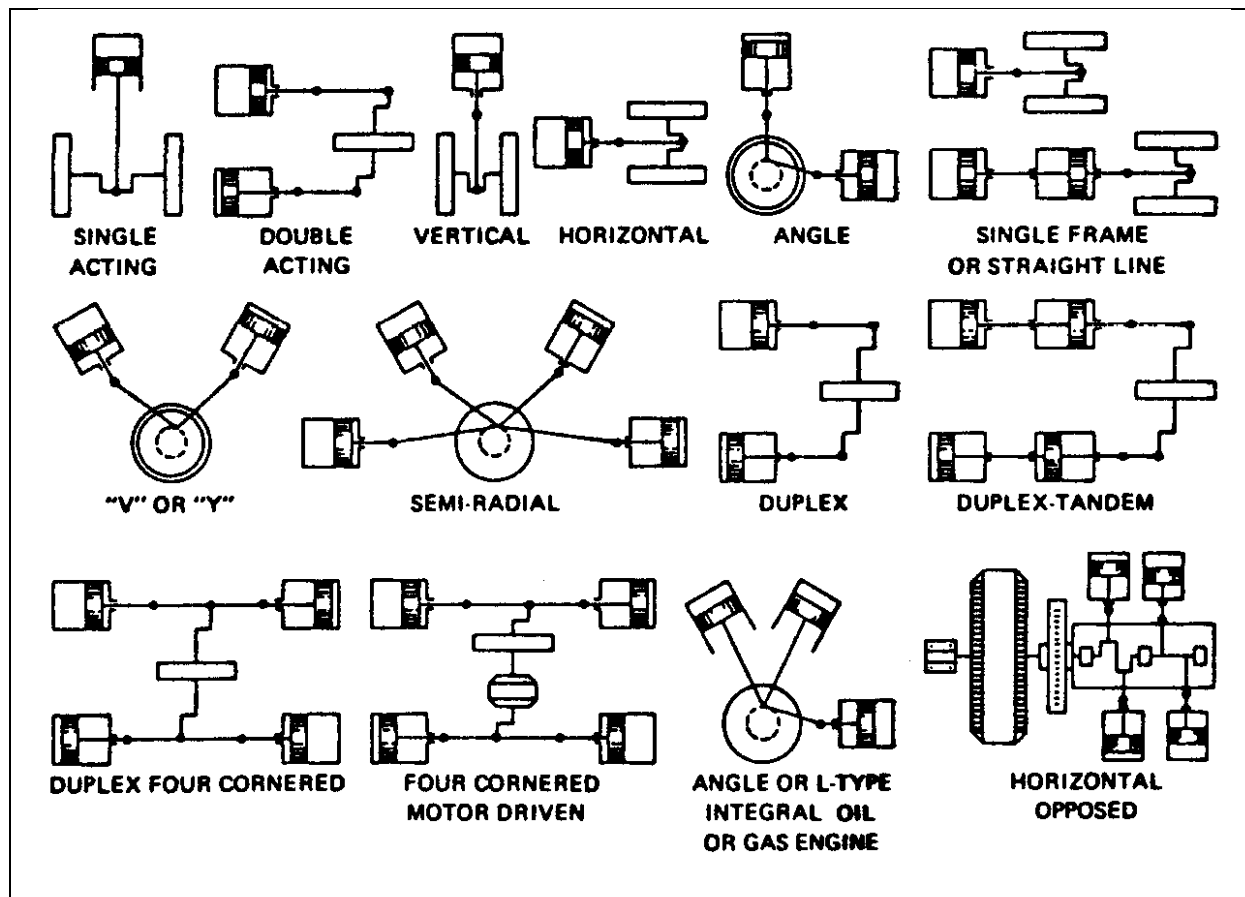
The integral gas engine shown in Figure 300-38 is still popular for many oil field duties. A cutaway view is shown in Figure 300-41. Figure 300-42 illustrates an integral-engine compressor.

350 Selection Criteria

When new compressors are being purchased, obviously the first consideration is installed cost. However, with compressors, it is particularly important to evaluate the present worth of the overall project for a chosen life cycle. Evaluated cost includes consideration of energy consumption as well as other direct costs. (Refer to **Life Cycle Costs** in Section 154.)

In addition to costs, the technical merits of proposed machines must be weighed. Rod loading and discharge temperature are important. Other major factors include reliability, number of stages, piston speed, rotating speed, number and type of valves, driver, "footprint" size (plot area) and weight, shaking forces (foundation requirements), and noise. The following is a brief discussion of these factors.

Fig. 300-38 Cylinder Arrangements Used for Various Reciprocating Compressors. Dots on Connecting Rods Indicate Crossheads (From *Hydraulics Pneumatics, "Air Compressors, Conditioning, Costs and the Crunch"* ed. N.R. Stull © 1976)



351 Rod Loading

Rod Loading is a general term having many interpretations. It is also called "pin loads," "frame loads," and "combined rod loads." Excessive rod loading can result in rod or crosshead failure and possible destruction of the machine.

API 618 gives definitions for the terms "combined rod loading," "gas load," "inertia force," and "rod reversal." **Combined rod loading** is the important term as it is the net effect of gas and inertia loading. **Rod reversal** occurs when the force on the rod changes from tension to compression, or vice versa. It is a critical lubrication factor for the crosshead pin and bushing. If there is no reversal, the pin and bushing surfaces do not separate; hence, oil is prevented from lubricating these surfaces. This can result in catastrophic damage to the crosshead, connecting rod, bushing, and crosshead pin.

Frame loading is the result of reaction forces from the gas-pressure resultant forces. Frame loading stresses static items such as crosshead guides, distance pieces, the frame itself, and the bolting between these items.

Fig. 300-39 Low-Speed Horizontally-Opposed Compressor (*From Compressed Air & Gas Data ed. by A.W. Loomis © 1982 by Ingersoll-Rand. Courtesy of Dresser-Rand*)

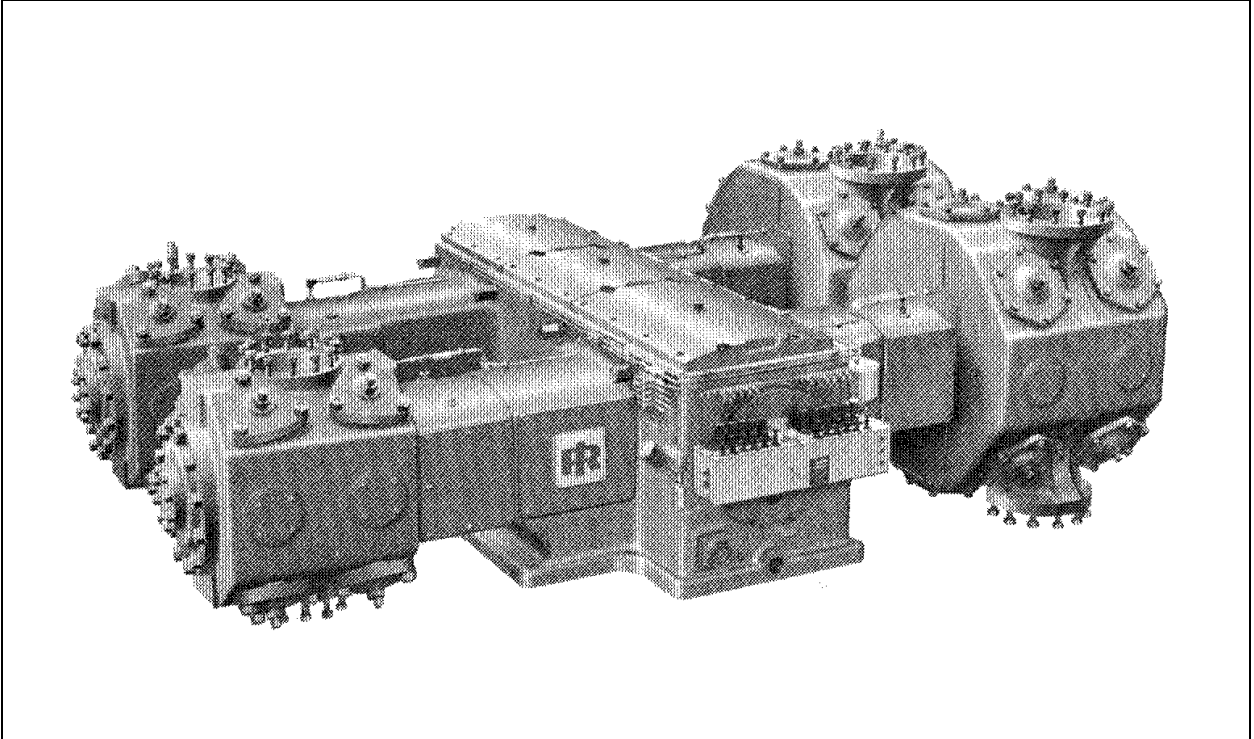


Fig. 300-40 Package High Speed Engine (*Courtesy of Demag Delaval*)

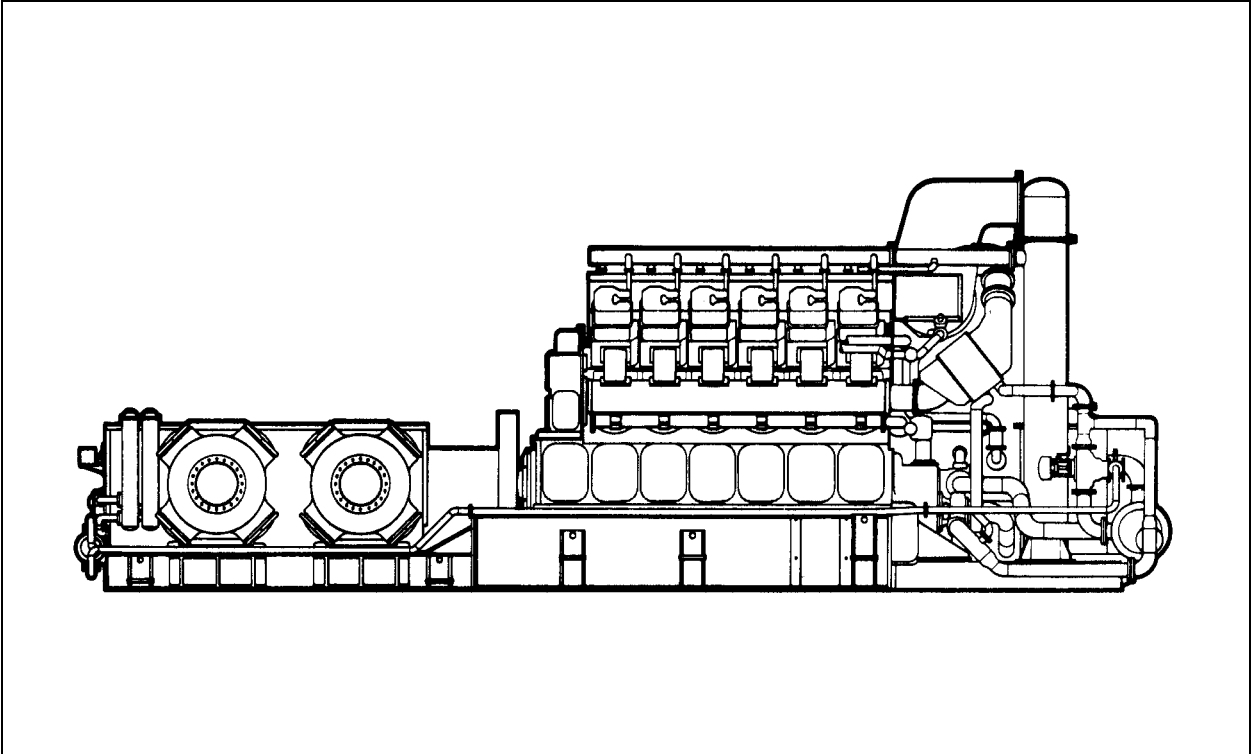
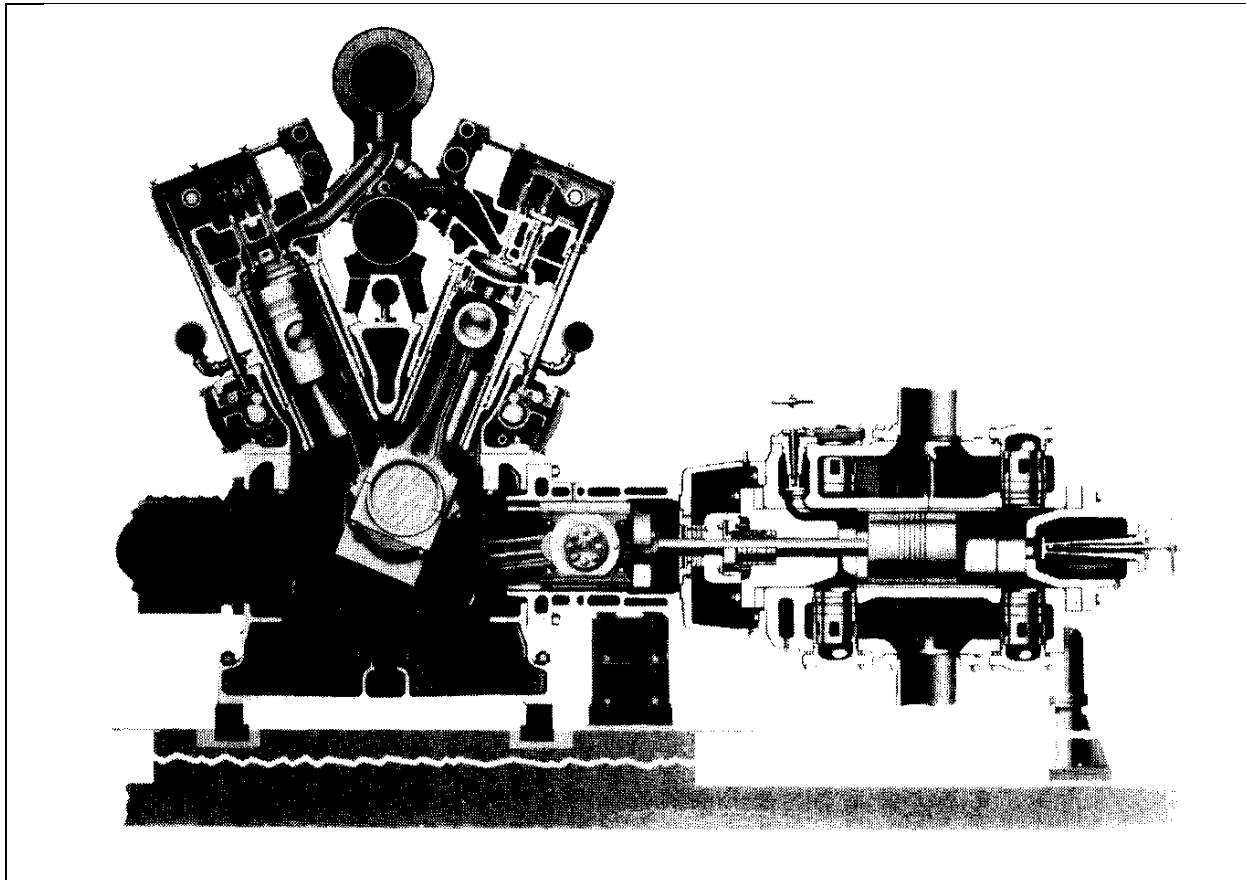


Fig. 300-41 A “V”-angle Arrangement of an Internal Gas-engine Driven Compressor (*From Compressed Air & Gas Data ed. by A.W. Loomis © 1982 by Ingersoll-Rand. Courtesy of Dresser-Rand*)



Rod loading is somewhat of a misnomer in some cases. A manufacturer's advertised maximum allowable continuous rod load (MACRL) rating is based on the weakest link in the running gear/frame system. The piston rod is not always the weakest link. It may be the crosshead pin or bushing, for example.

Years ago, quite often only gas loads were considered in evaluating proposed compressors. In most cases, a quick comparison of gas loading with the MACRL was adequate. But occasionally, inertia forces augmented the gas loading at certain crank angles so that the MACRL was exceeded.

Rod reversal becomes a problem at low pressure ratios. It can also be a problem at high-pressure ratios for single-acting cylinders, half loaded cylinders, or when the bore of double-acting cylinders is not much larger than the piston rod. Tail rods can be used to alleviate the latter situation.

Figure 300-43 shows gas, inertia, and combined loading for pressure ratios of 4.00 and 1.05. The 4.00 ratio examples show the maximum combined loading is less than the gas loading (the inertia force helped the situation). Such is not always the case, however. It depends on the differential pressures throughout the stroke, and the weights of the reciprocating parts. It is impossible to provide a “rule of thumb” for this phenomenon.

Fig. 300-42 Integral-engine Compressor *(Courtesy of the Cooper Cameron Corporation)*

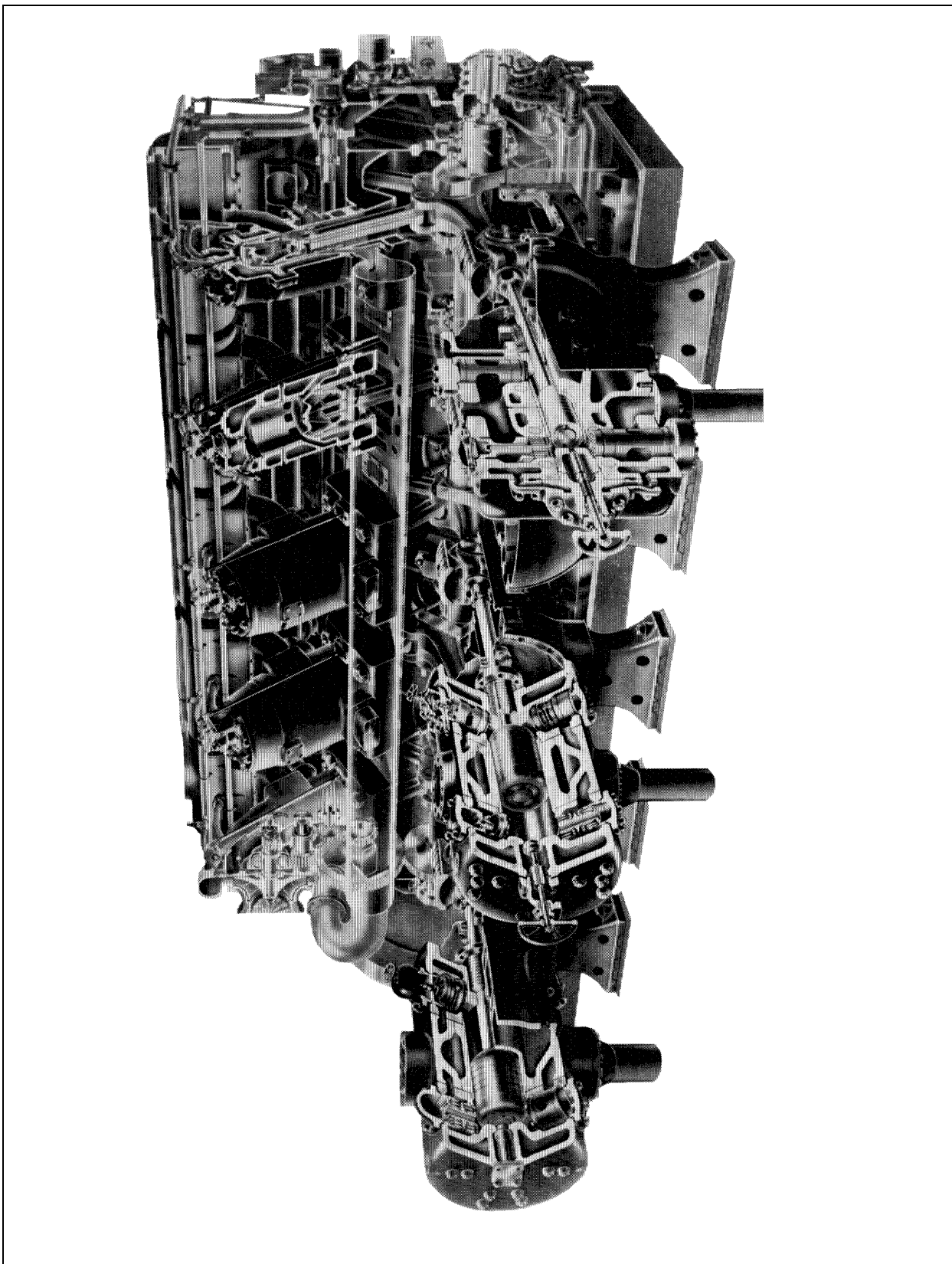


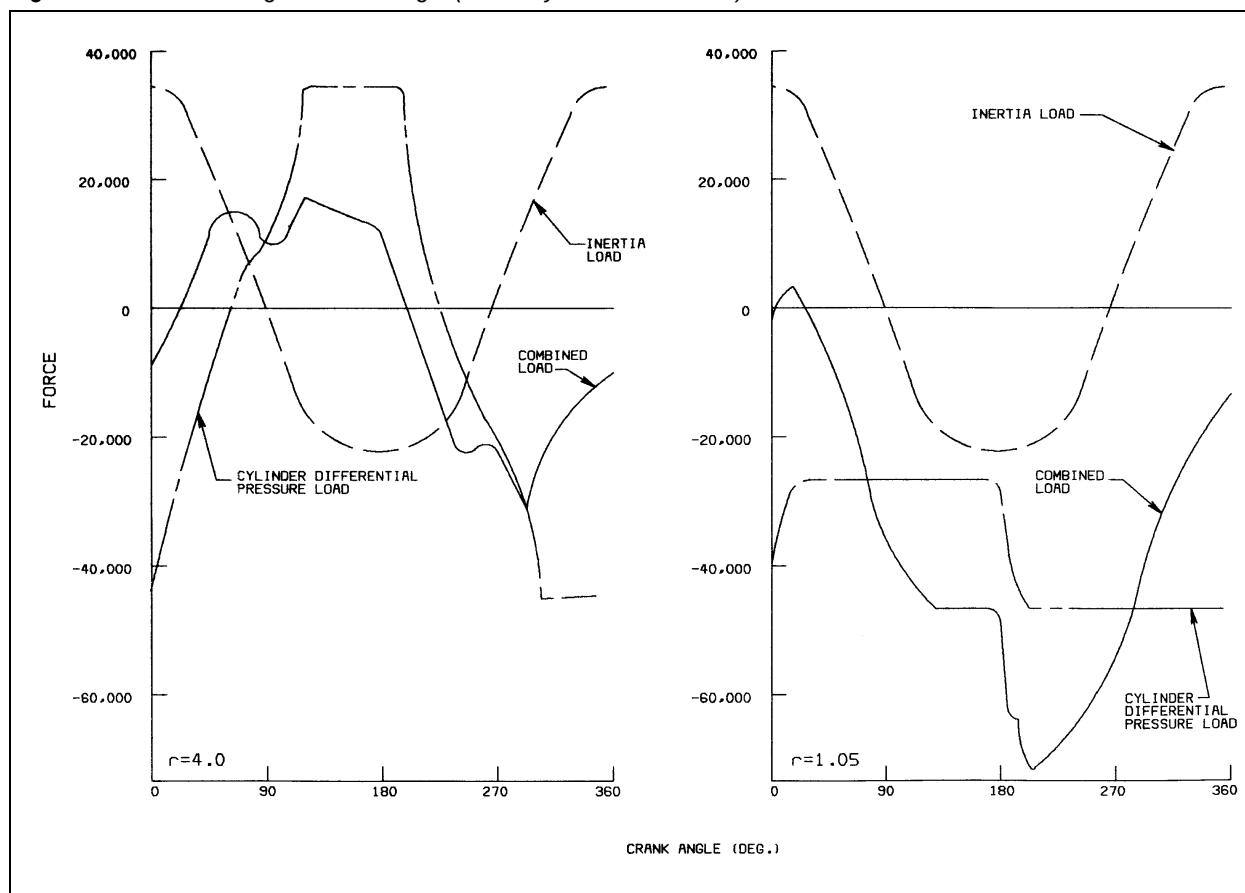
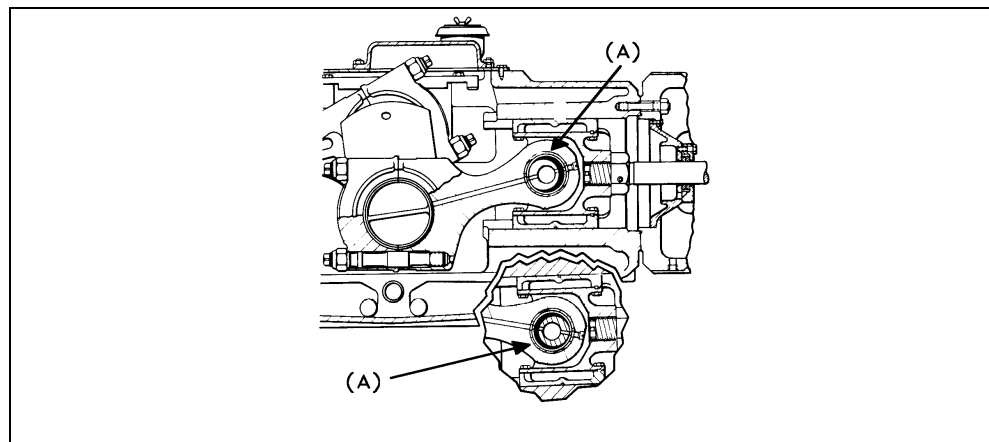
Fig. 300-43 Rod Loading vs. Crank Angle (*Courtesy of Dresser-Rand*)

Figure 300-43 also illustrates a cylinder with a very low pressure ratio (1.05). In this case, the maximum combined load is greater (negative) than either the gas or inertia loads. Also the combined load only reversed for 15-20 degrees of crankshaft rotation. (Note that API 618 suggests a minimum of 15 degrees of reversal). Figure 300-44 shows the ideal reversal (see dark areas labeled "A") at the crosshead pin.

Fig. 300-44 Load Reversal at Crosshead Pin (*Courtesy of Dresser-Rand*)

A subtlety often overlooked is that the actual differential pressure that exists at a certain crank angle **inside** the cylinder is higher than the theoretical differential pressure. Refer to Figure 300-15 and note that the vertical distance between points C and F is significantly greater than discharge minus suction pressure. Pressure pulsations can make matters worse. It should be noted that, at any one instant, the upper part of the diagram in Figure 300-15 would correspond to the head end, while the lower part would correspond to the crank end.

When purchasing new compressors, you should insist that the vendor make a study of combined rod loading versus crank angle using internal cylinder pressures with: (1) normal operating conditions, and (2) with any broken valve, with relief valve setting as discharge pressure.

Calculated gas loads can be very misleading, but sometimes they can be used as a preliminary index to estimate the frame size or number of cylinders per stage. These calculations can be made using pressures at the cylinder flanges and the full area of the piston on the head end and the net area (piston area minus rod area) for the crank end. Loading should be calculated for tension and compression. When one end of a cylinder is unloaded, that end will have suction pressure in it at any crank angle.

352 Discharge Temperature

Discharge temperature is an important factor, as it must be limited to prevent material damage. Additionally, higher discharge temperatures mean higher **differential temperatures** which cause greater differences in thermal expansion of components made of dissimilar materials. This in turn, causes higher stresses and larger deviations from design clearances.

300°F is a good all-around limit for process and oil-field compressors. The limit often applied to machines compressing hydrogen-rich gases is 275°F.

Although temperatures as high as 350–375°F have been used with a fair amount of success, 300°F provides a better margin of safety for several design factors, thus improving reliability and availability. The widely used PTFE piston ring and packing materials last longer and perform better at lower temperatures. With a calculated or observed temperature of 300°F, there are likely localized temperatures 50 degrees higher or more.

The 300°F limit sometimes causes a machine to have a greater number of stages than a higher temperature limit. More stages reduce rod loading by reducing the differential pressure. Another benefit of lower discharge temperatures is a reduced tendency toward deposits of solids on the discharge valves. This is likely to occur in dirty gas and heavy hydrocarbon services.

The penalties of lower discharge temperatures are cost, more working parts, and increase in footprint size. However, experience has shown that the benefits outweigh the penalties in the long run. There may be isolated cases where higher discharge temperatures may be justified, but these cases should be carefully studied.

353 Rotating Speed and Piston Speed

Rotating Speed

Higher rotating speed increases the number of cyclic events (such as stress reversals) that occur in a given period of time. For instance, the number of valve openings and closures increase directly with speed. Valve life is related to the number of cycles of operation, as are the lives of other components. For a given stroke, an increase in rotating speed directly increases the average piston speed.

Piston Speed

Average piston speed is an index of the rubbing speed and wear of the piston and its rings. Average piston speed, in feet per minute (FPM), may be calculated with the following formula:

$$\text{A.P.S.} = \text{SN}/6 \quad (\text{Eq. 300-30})$$

where:

S = stroke in inches

N = RPM

For critical service heavy-duty Class A compressors, a speed limit of 360 RPM is recommended. The average piston speed should be limited to about 860 FPM for lubricated machines, and 700 FPM for non-lubricated machines.

Class B, high-speed machines used in medium- to light-duty applications in the oil field typically have upper limits of 1000 RPM. Machines will work at these levels, but significant improvement in reliability can be achieved by running these units at 900 RPM or slower. Although the slower speeds result in larger cylinder sizes and increased rod loading, this disadvantage is usually overbalanced by the reduced wear.

354 Compressor Valves

Valves are highly stressed wearing parts that account for the majority of compressor downtime. Reciprocating compressors represent 1% of the machinery in Chevron Refineries but account for 12% of the maintenance budget. The Company spends approximately five million dollars per year for maintenance of reciprocating compressors. A significant percentage of that cost is attributable to compressor valves.

Although liquid, dirt, or process contaminants often cause valve failures, design factors are often a major contributor. Furthermore, valve design can also reduce the effects of contamination in some cases. Valve life in some severe services has been as short as 4 days when the wrong combination of valve lift and materials was specified.

Although new materials and a better understanding of valve dynamics have greatly improved reliability, valves continue to have a major impact on overall compressor availability. For example, a large refinery compressor shut-down for valve replacement reduced the plant feedrate by 40%, which cost \$65,000 (1990 dollars) in lost production. Even small improvements in valve life which postpone valve repairs can have an a large impact on plant profitability.

There are many types of compressor valves. Almost all are spring loaded and gas actuated. Reciprocating compressors generally used one of the three basic valve configurations: plate, strip, or poppet. These are described in the following paragraphs.

Plate Valves

Figure 300-45 shows a typical plate valve. This type of valve is actuated by unbalanced pressures on either side of the valve. The valve plates or elements are held against the ports in the valve seat by spring force. The gas pressure overcomes the spring force, the elements lift away from their seats and stop against the guard, opening the valve.

Fig. 300-45 Plate Valves (*Courtesy of Dresser-Rand*)



Concentric Ring Valves. The plate valve shown in Figure 300-45 is a concentric ring valve. This type of valve can be fitted with plastic elements, which is an advantage in corrosive services. The concentric ring valve can be used over the widest range of compressor applications and can withstand the most extreme operating conditions. Concentric ring valves have been used for pressures as high as 60,000 psi with differential pressures greater than 10,000 psi and temperatures in excess of 500°F.

Ported Plate Valves. Ported plate valves are very similar to concentric ring valves except that the individual rings are joined to form one or two larger plates. Their chief advantages are ease of manufacture and simpler assembly. Ported plate valves are used primarily for high speed gas field compressors. Ported plate valves are shown in Figure 300-46.

Strip Valves. Strip valves include channel and flexible element valves. They are used primarily for air service, either lube or nonlube. A disadvantage is that they cannot be used with plastic elements and are therefore not very tolerant of dirt or liquids in the gas stream. They have good flow areas and are relatively inexpensive.

The maximum discharge pressure for these valves is about 1,500 psi. The highest recommended operating temperature is 350°F and the maximum pressure differential is approximately 500 psi. Strip valves may be used in compressors with rotating speeds up to 1,800 rpm.

One type of strip valve, called a “feather” by the original manufacturer, employs a single component to serve as the sealing element and the return spring. Application of the feather valve is generally limited to operating pressures less than 1000 psi, differential pressures no greater than 400 psi, and maximum temperatures of 350°F. A feather valve is shown in Figure 300-47.

Fig. 300-46 Ported Plate Valves (Courtesy of Dresser-Rand)

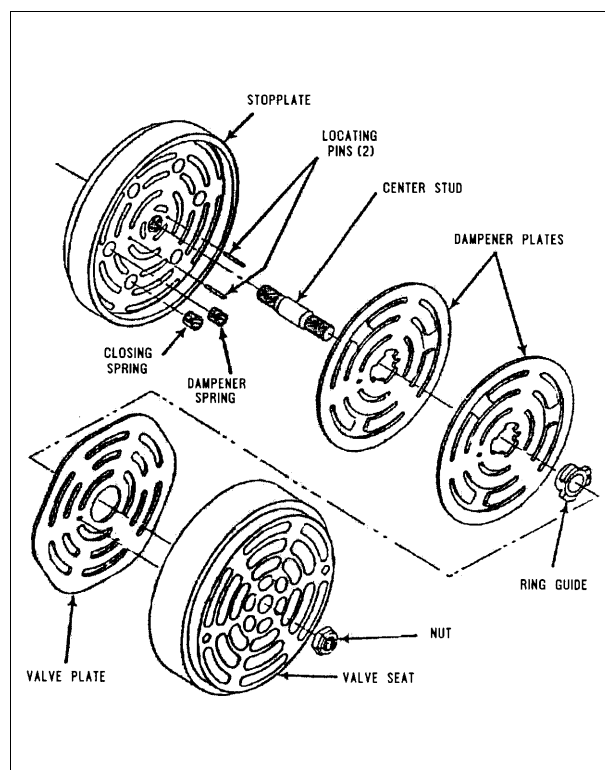
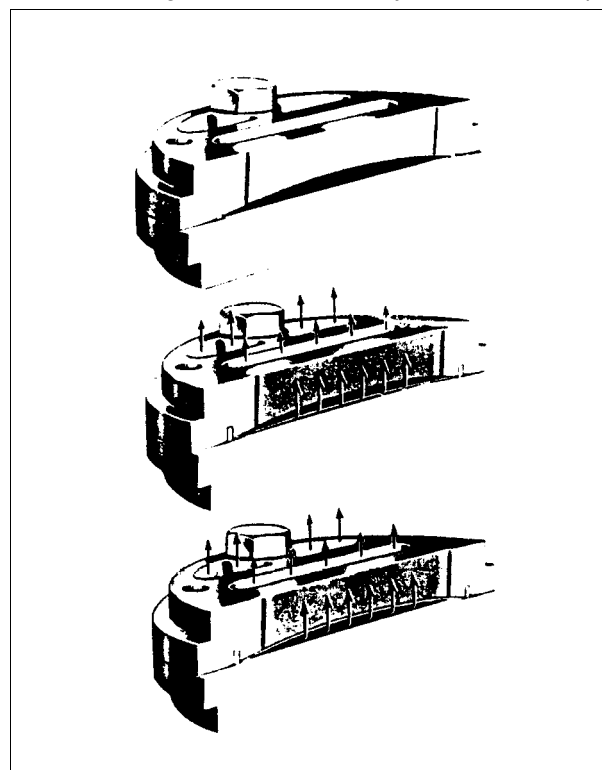


Fig. 300-47 Feather Valves (From Compressed Air & Gas Data ed. by A.W. Loomis © 1982 by Ingersoll-Rand. Courtesy of Dresser-Rand)



Another type of strip valve, called a channel valve, is shown in Figure 300-48.

Poppet Valves. Poppet valves have an effective lift area approximately 50% greater than that provided by the same size concentric ring valve. Poppet valves can operate with lifts as great as 1/4 inch and are used extensively in the natural gas transmission industry. They have recently been applied in other process applications such as carbon dioxide injection for tertiary oil recovery. By design, they add built-in clearance, which causes a capacity penalty, especially with high specific heat gases.

The poppet valve utilizes a mushroom-shaped element made from a variety of materials. The sealing element material determines the range of application. Valves with metallic poppets can withstand pressures up to 3000 psi and temperatures to 500°F. However, metallic poppets are seldom used due to inertial effects.

Nonmetallic poppets are limited to 450°F and 800 psi, with compressor speeds up to 1,800 rpm. Typically NYLON, TORLON, and now PEEK are used for the poppet material because of their light weight and conformability to the valve seat. Several OPCOs are using poppet valves with good success. Refer to Figure 300-49.

Fig. 300-48 Channel Valve (*From Compressed Air & Gas Data ed. by A.W. Loomis © 1982 by Ingersoll-Rand. Courtesy of Dresser-Rand*)



Fig. 300-49 Poppet Valve (*Courtesy of Hoerbiger Corporation of America, Inc.*)



Valve Motion

The motion of valve elements during the compressor cycle are affected by two forces, as shown in Figure 300-50. The first is an aerodynamic force due to flow of gas around the element. The second is the spring force resisting this aerodynamic force.

Fig. 300-50 Forces Acting on a Valve Plate (Courtesy of Hoerbiger Corporation of America, Inc.)

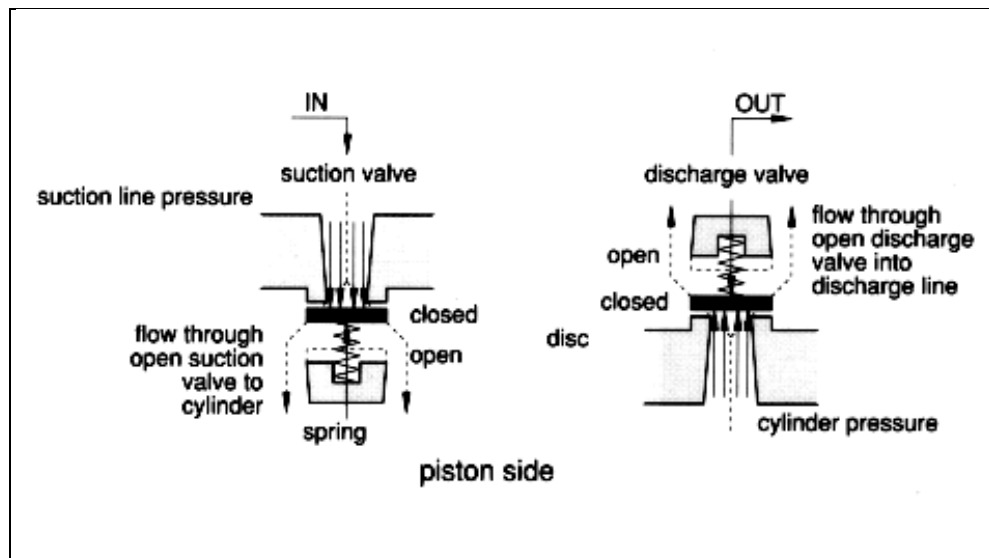


Plate or strip motion affects compressor performance and reliability. Not only must the valve open and close at the correct time, but it must open completely and seal tightly upon closing. In addition to being able to open fully and close completely, the valve element must not strike the seat or guard with excessive force, or short valve life will result.

Each valve is a damped spring/mass system with natural frequency. If the natural or resonant frequency is forced by some driving frequency, such as the pulsations from the compressor pistons, rapid cycling of the valve element will occur. The element generally bounces between the guard and the seat. This is commonly called “flutter”. Flutter can greatly reduce the life of springs and valve elements.

The manufacturer modifies the valve lift to suit the gas specified. For example, an air compressor might be furnished with a lift of .100 inches. The same compressor valve applied in a low mole weight service such as hydrogen, might use a lift of .032 inches.

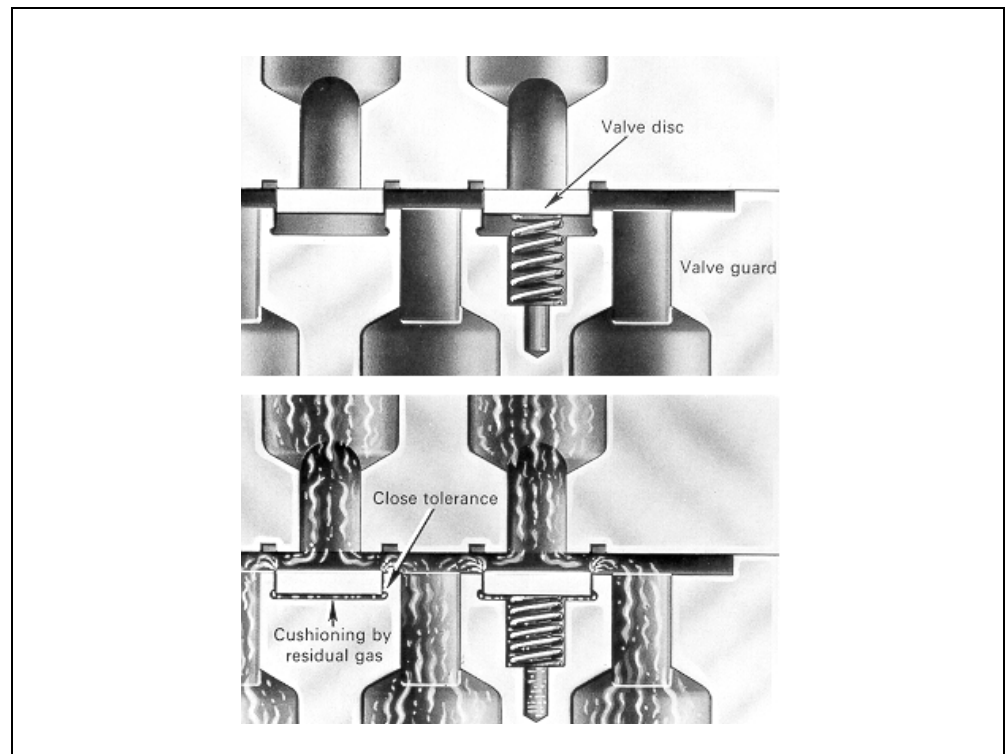
The problem with a higher lift valve in the hydrogen service is that hydrogen lacks the damping properties of air. As a result, the valve elements would experience high impact forces and might bounce. Broken valve elements might result.

Valve Element Failure

When the valve element is stopped abruptly at the guard or seat, it may not land evenly. Typically, one edge contacts first, which induces a bending moment. This flexing will eventually cause fatigue failure of the valve element. Several steps can be taken to reduce the flexing. Reducing valve lift is one remedy, but frictional power loss may increase. Providing balanced spring arrangements that allow the valve element to return to the seat as evenly as possible and/or providing a gas cushion can reduce fatigue failures of the valve element.

The gas cushion is produced by a close clearance between the valve element and a groove in the valve seat, which also locates the valve element. As the gas trapped in the space between the underside of the valve element and the walls of the groove escapes along the edges of the element, the element is cushioned as it returns to the seat, as shown in Figure 300-51.

Fig. 300-51 Gas Damped Valve Plate (Courtesy of Cooper Energy Services Div., Cooper Cameron Corporation)



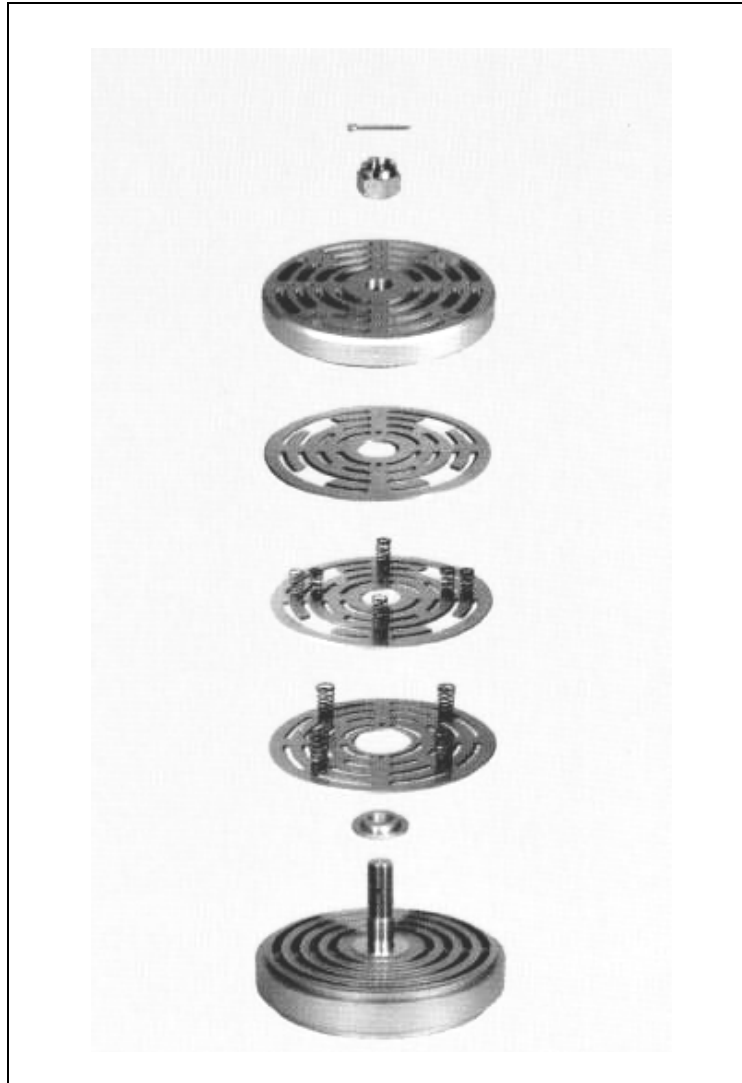
Another type of damping, mass damping, is used in the “damped plate” design. A disk ported the same as the valve element is attached to the valve element, usually at its center. This damping plate adds inertia to the valve element and therefore retards the impact velocities of the valve element against the guard and seat. This type of damping is useful for high speed compressors where valve velocities are high. It is also more desirable than gas damping, where the gas stream is dirty or heavily lubricated. A damped plate valve is shown in Figure 300-52.

Allowable Plate Impact Velocity

Excessive valve plate impact velocity at the valve seat or at the valve guard contributes heavily to valve plate and valve spring failures.

One manufacturer's operating experience has shown that, in order to eliminate valve plate or spring failures resulting from improper valve dynamics, the plate impact velocities in Figure 300-53 should not be exceeded.

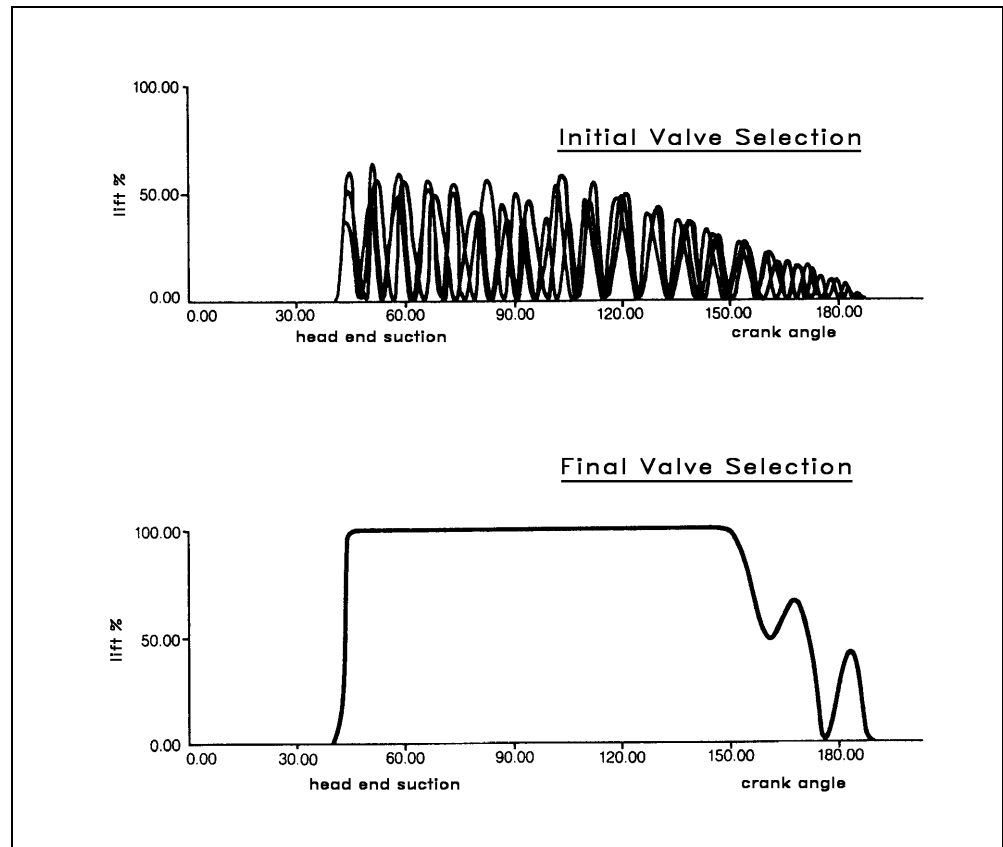
Most vendors have computer programs that mathematically model valve element motion, aerodynamic drag coefficients, and flow through the valves during the

Fig. 300-52 Damped Plate Valve (Courtesy of Hoerbiger Corp. of America)**Fig. 300-53** Maximum Allowable Plate Impact Velocity (inch/sec)

Valve Design	At Valve Guard	At Seat
Std. Plate	150	75
Damped-Plate	200	100

compression cycle. These programs assess the valve dynamics and optimize the design in terms of lift, spring stiffness, and the damping for a specific application. The programs have a fairly good record of success, and continue to be improved with time. Figure 300-54 shows the results of a typical computer analysis (note the greatly reduced valve flutter).

If practical, before purchasing, have the vendor run a computer study for new machines, or for any valve upgrades planned for existing machines. It is highly recommended to discuss actual field experience with users of any particular type of valve proposed. Avoid valves which are not being used in similar services.

Fig. 300-54 Results of Computer Study of Valve Dynamics (*Courtesy of Dresser-Rand*)

355 Drivers

Figure 300-55 presents a generalized curve of speed versus horsepower for Class A machines based on one vendor's experience. Figure 300-56 shows the distribution of driver types for Class A compressors, again based on one vendor's experience.

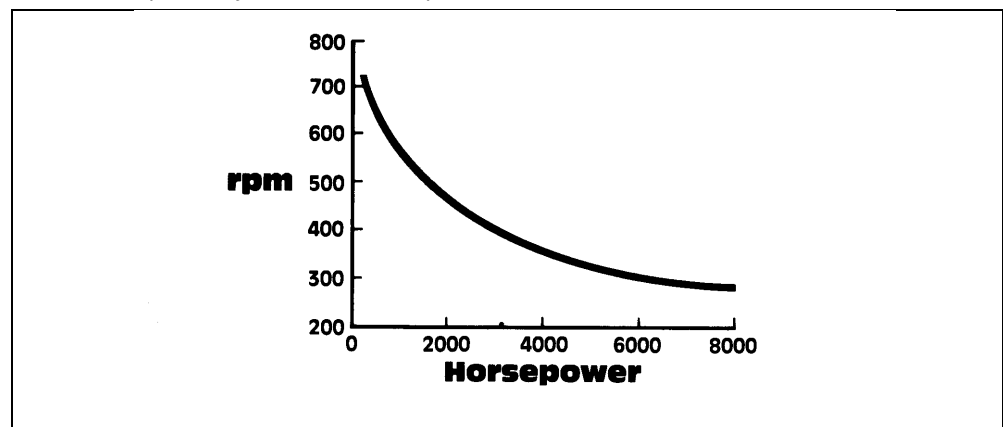
Fig. 300-55 Reciprocating Process Compressors—Rotative Speed vs. Horsepower (*Courtesy of Dresser-Rand*)

Fig. 300-56 Reciprocating Process Compressors Driver Practice (*Courtesy of Dresser-Rand*)

Type Driver	% of Installations	Criteria for Selection
Low Speed Synchronous	80	Efficiency—Price—Power Factor Correction
Low Speed Induction	16	Simplicity—No Excitation Required
Induction Motor/Gear	1	Low Torque per KVA in Starting. Lower Maintenance. Explosion Resistant Enclosure
Steam Turbine/Gear	3	Availability of Low Cost Steam
Coupled Internal Combustion Engine	Negligible	

Refer to the *Driver Manual* for detailed application information, and use the following general comments for any preliminary work:

1. Synchronous motors are 3–4% more efficient than induction motors of the same speed, and usually offer power factor improvement for the electrical system. **High-efficiency induction motors should not be used with reciprocating compressors, because the oscillating torque causes excessive heat.** Rather, use standard induction motors.
2. Steam and combustion gas turbines can be applied to reciprocating compressors, although it is not a frequent practice (see Figure 300-57). A torsional analysis is essential to be certain the train will not operate close to a torsional critical speed.
3. Spark-ignited gas engines are available to drive Class B machines up to about 2000 HP at 900–1000 RPM.
4. Belt-drive systems can be used up to about 200 HP.

356 Footprint and Weight

General footprint and weight data were presented in Section 100. If more detailed information is needed, obtain it from vendor bulletins or directly from the vendor.

357 Noise

Noise data must be obtained from the vendor. Compressor vendors generally cannot meet the 85 dbA noise limit recommended in many Company specifications. They usually guarantee 90 dbA. Acoustic treatment may be required if it is necessary to meet 85 dbA.

358 Efficiency and Economics

Efficiency directly affects power or fuel cost, and this cost will be the dominant factor in the total life cycle cost. The reciprocating machine is very efficient not only at full load, but also at partial load. Figure 300-58 shows the typical full-load efficiency of Class A machines compressing hydrogen.

Fig. 300-57 Typical Turbine Drive (*Courtesy of Dresser-Rand*)

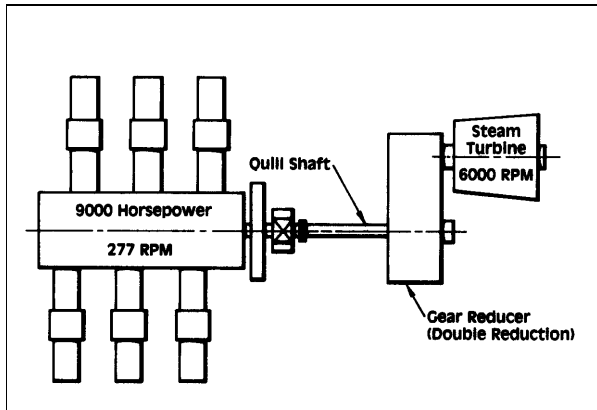
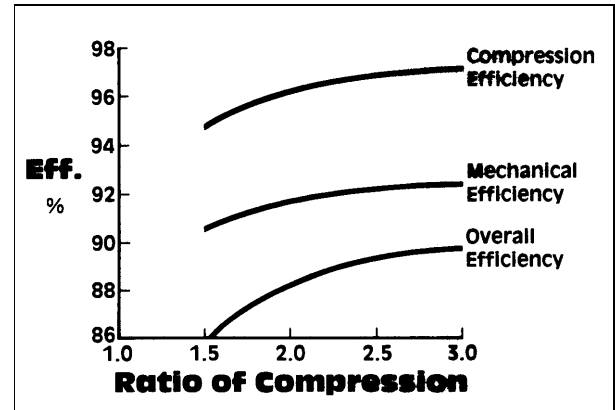


Fig. 300-58 Typical Efficiency Reciprocating Process Compressors—Hydrogen (*Courtesy of Dresser-Rand*)



It is impractical to provide general cost estimating factors such as \$/HP or \$/ICFM. Such factors can be very misleading owing to wide variances in power levels, pressures, etc., in the range of applications. (See Section 154.) Cost estimates for the machinery should be obtained from the vendor.

360 Application and System Considerations

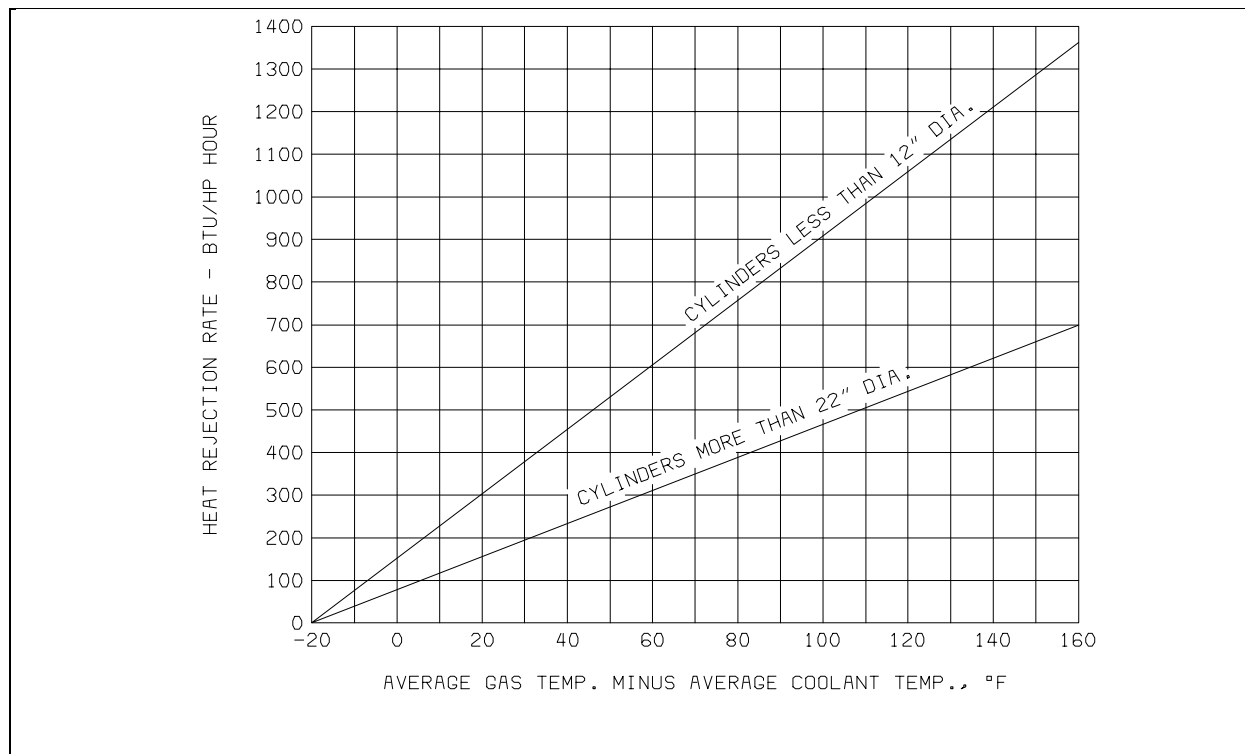
361 Cylinder Thermal Performance

Figure 300-59 shows heat rejection rates for forced liquid cooling of cast iron cylinders. Cast steel or forged steel cylinders have lower heat rejection rates, mainly because the walls are thicker, and the jacketed areas are smaller. The following values may be used for estimating purposes for average conditions:

- cast iron cylinders 700 BTU/Bhp-hr
- cast steel cylinders 250 BTU/Bhp-hr
- forged steel cylinders 200 BTU/Bhp-hr

The coolant inlet temperature should be at least 10°F above the suction temperature to prevent condensation. The coolant temperature-rise across each cylinder should be at least 10°F, but no more than 20°F.

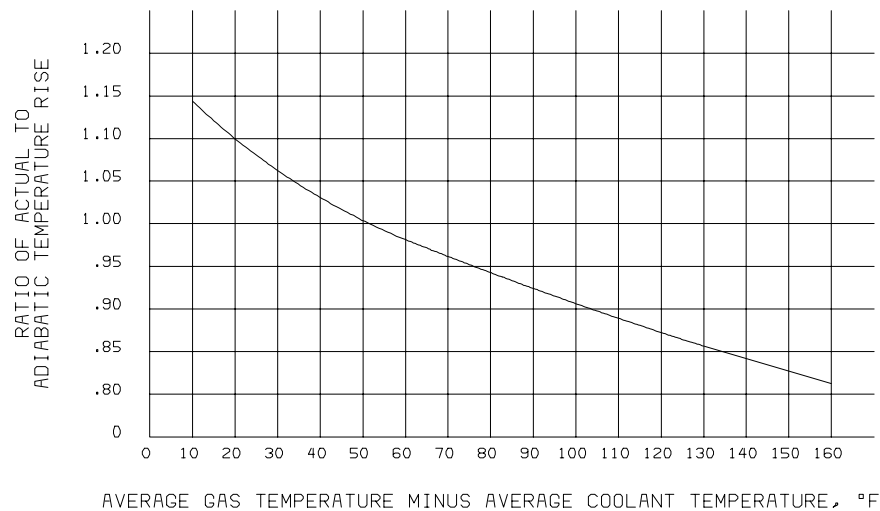
Fig. 300-59 Gas Temperature Rise Characteristics for Operation with Thermally Circulated Coolant and Dry Jackets
(From Compressed Air & Gas Data ed. by A. W. Loomis © 1982 by Ingersoll-Rand. Courtesy of Dresser-Rand)



The following relationships should be treated as guidelines. They will not provide exact answers for all applications. Consult the vendor if it is necessary to know the discharge temperature more accurately.

The average gas and coolant temperatures affect the actual discharge temperature of cylinders with forced cooling. If the average gas temperature less the average coolant temperature is between 50°F - 60°F, the discharge temperature should be close to that predicted by Equation 300-25. For other temperature differentials, refer to Figure 300-60 to determine an estimate of the deviation from adiabatic discharge temperature.

Two other types of cooling systems are applied to cylinders: **thermosyphon** and **no coolant** circulation (static filled jackets). Refer to API 618 for criteria to apply these systems. Figure 300-61 gives temperature relationships for these systems.

Fig. 300-60 Gas Temperature Rise Characteristics with Forced Coolant Circulation

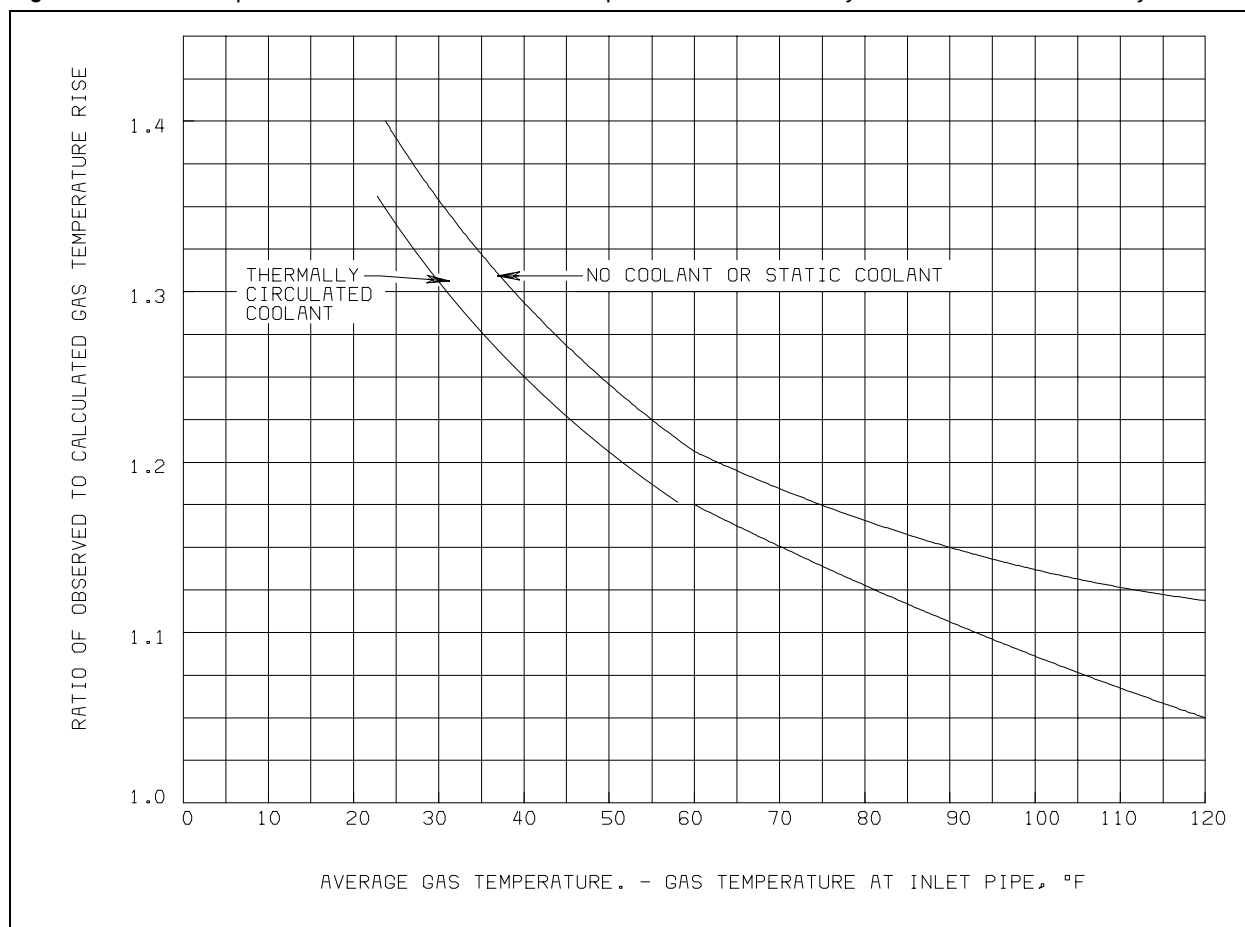
$$\text{AVERAGE GAS TEMP.} = \frac{t_2(\text{CALC}) + t_1}{2}$$

$$\text{AVERAGE COOLANT TEMP.} = \frac{t_{\text{OUT}} + t_{\text{IN}}}{2}$$

WHERE: t_1 = GAS SUCTION TEMPERATURE
 $T_2(\text{CALC})$ = CALCULATED GAS ADIABATIC DISCHARGE TEMPERATURE
 t_{IN} = JACKET COOLANT INLET TEMPERATURE
 t_{OUT} = JACKET COOLANT OUTLET TEMPERATURE

362 System Resistance

Because a reciprocating compressor can usually be unloaded in 25% capacity steps down to zero throughput, it is possible to overlook the fact that suction or discharge pressure (or both) will normally vary with each step (unless these pressures are fixed by process controls). At lower capacity steps, the suction pressure could be much higher than it is at the full capacity step. This condition could cause relief valves to pop during operation unless the higher pressure is accounted for in the system design. Accordingly, a system pressure profile calculation should be made for each capacity step as well as for full load. The Compressor Pressure Profile Worksheet (CMP-EF-876) along with the *Fluid Flow Manual* will help in making and recording these calculations.

Fig. 300-61 Gas Temperature Rise Characteristics for Operation with Thermally Circulated Coolant and Dry Jackets

363 Capacity Control

Capacity may be controlled on the basis of suction pressure, discharge pressure, flow or a combination of these variables. Capacity can be varied by use of unloaders, clearance pockets, and/or by speed variation when the driver is a variable-speed machine.

Whenever possible, constant-speed operation is recommended in order to avoid possible excitation of torsional or acoustical resonances. This is of particular importance on complex compressors with three or more stages, or when the unit(s) are to be located on an offshore platform. (The acoustical simulation becomes increasingly difficult with variable-speed machines, as does torsional vibration analysis.) The complex steel structure of an offshore platform has numerous vibratory natural frequencies which can respond to mechanical or acoustical excitations generated by the compressor. The risk of encountering such resonances are greater with variable speed compressors.

364 Parallel Operation

For machines operating in parallel, avoid allowing one machine to take more than its share of the load. Constant-speed, identical reciprocating compressors may be well matched, but deterioration can occur on one machine due to a broken valve, or worn rings. This would cause the other machine to take on more load in situations where the suction pressure varies with flow rate.

For example, three 50% compressors are piped in parallel to a large receiver vessel feeding fuel gas to a combustion gas turbine, with fuel gas flow varying between 70 and 100% of the rated flow. Two compressors would be running, with one on standby. One unit would be on “base load”, and the other would be the “swing” unit taking all the load variation. This load variation could be accommodated by automatically actuating suction-valve unloaders to maintain the receiver pressure within a set range. Clearance pockets could be used in combination with the unloaders to reduce the control fluctuations.

When the system has no receiver, a bypass with control valve is required if it is necessary to regulate the flow to quantities between those provided by the capacity steps. Otherwise, the control systems set-up could be the same as described above.

There are numerous possibilities for control system arrangements for parallel operation which are beyond the scope of this manual. Consult a control systems engineer to optimize the system and work out the details.

365 Power Margins

The rated horsepower of motor drivers should be at least 10% greater than the highest power required by the compressor (including any power transmission losses) at any of the specified operating conditions. For turbine drivers, a 20% margin is recommended to provide for deterioration in performance and variation between actual and specified steam conditions. (Moreover, the 20% margin has been necessary on smaller units to effectively handle the cyclic compressor torque.)

Internal combustion engines should be rated in accordance with DEMA standards. The DEMA continuous duty rating must be derated for site elevation, maximum ambient temperature, inlet and exhaust pressure drops, and for shaft-driven accessory loads. In addition, it is desirable to have a 10% margin between the site-derated continuous duty rating and the maximum compressor power requirement. With engines, the power margin is not as straight-forward as with other drivers owing to the different levels of conservatism of the manufacturers. A power margin is desirable because such factors as emission control devices and the cooling water temperature to the turbocharger aftercooler can affect engine capability. Engineering judgment will usually have to be applied.

Engines should also meet the DEMA requirement for the capability to carry a 10% overload for two hours out of any 24-hour period. It is also a good idea to review the engine vendor's experience at the proposed rating with other owners of that model.

366 Foundation Mounting and Grouting

Refer to the *General Machinery Manual (Section 300)* for more information on foundations, anchor bolts, and grouting.

There are three methods of mounting reciprocating compressors on foundations:

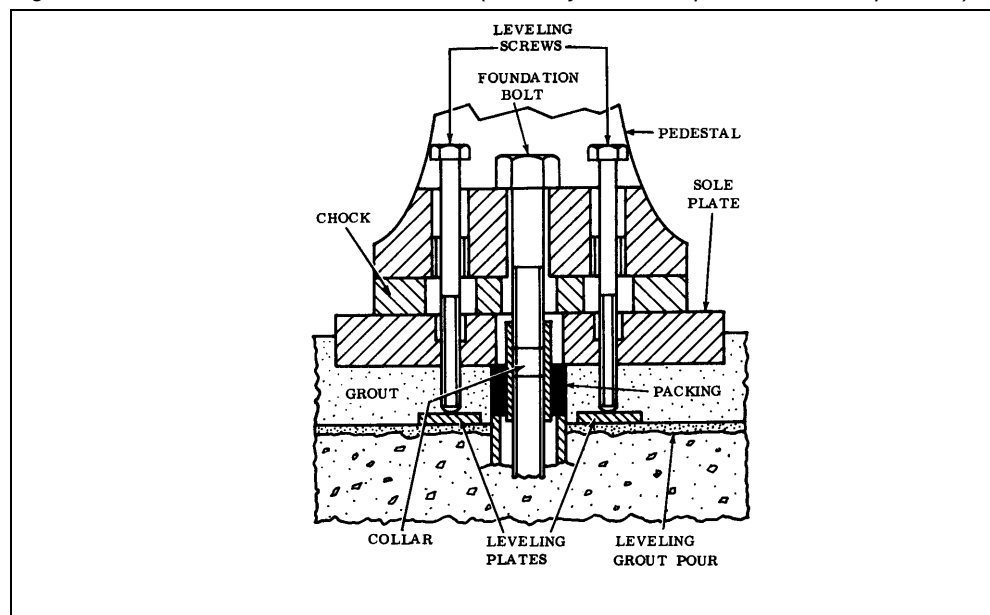
1. Direct grouting
2. Mounting on sole plate (rails)
3. Mounting on a base plate (skid)

Direct grouting of the frame to the foundation is used on smaller machines, such as air compressors. These machines are short and not likely to experience frame distortion due to differential foundation displacement.

If longer machines with several crank throws are directly grouted, the heat from the frame will cause thermal distortion of the foundation. The frame then conforms to the foundation, causing misalignment of the crankshaft. Misalignment can lead to crankshaft failure; therefore, **sole plate mounting** is used. With this method there is air space between the frame and the foundation. Chock blocks and/or shims are used between the sole plate and the frame. If foundation distortion occurs over the years, the frame can be re-chocked or re-shimmed to restore alignment.

Figure 300-62 shows a typical sole plate and chock arrangement for a crosshead pedestal. A similar arrangement would be used under the frame, although the foundation bolt is usually one-piece. **Note that the foundation bolt sleeve is packed to prevent entry of grout. This feature is important.**

Fig. 300-62 Crosshead Pedestal Sole Plate (*Courtesy of the Cooper Cameron Corporation*)



Many Class B units are typically **skid-mounted**. In this case, the frame is mounted on a base plate, and the base plate is grouted to the foundation. When the base plate becomes quite large, sole plates are sometimes used under the base plate.

Except for very small units, **epoxy grout** is recommended for reciprocating machines. Although more expensive than **cementitious grout**, epoxy grout has superior strength, is oil-resistant, and has negligible shrinkage. Some cementitious grouts include an additive that expands during the curing process to reduce shrinkage. However, this process has been unreliable in many cases, and the grout deteriorates with constant exposure to lube-oil. One recommended rule-of-thumb is to use epoxy grout on all reciprocating machines rated at 500 HP or more, or that weigh 5000 pounds or more.

367 Shaking Forces (Foundation Requirements)

Refer to Section 366, “Foundation Mounting and Grouting”. Carefully review the vendor's unbalanced forces and couples with the engineering group designing the foundation. This aspect of the compressor installation is critical, especially if it is offshore or on very poor soil. It may be necessary to provide guidance to the vendor when shaking forces are crucial. It is possible to alter the proposed design to improve the balance of forces and couples.

370 Piping and Arrangements

Refer to the *Piping Manual* for detailed information on piping design and configuration. A few of the fundamental guidelines are presented here.

Reciprocating compressors are sensitive to dirt. Therefore, the suction piping to each stage (from the knockout drum to the suction pulsation damper) should be “pickled” (chemically cleaned). Temporary strainers should be installed and left in place for several days during initial operation. These strainers are usually of the truncated-cone or conical type, and made of perforated steel plate with a double overlay of stainless steel wire mesh screen (often 100 mesh over 30 mesh). The screen is attached to the outside of the strainer basket so that the flow encounters the screen first. The strainer is typically located at the inlet to the suction pulsation damper.

Liquid is also a significant problem. Suction piping must be configured so that liquids cannot be trapped in low spots. Liquids that collect in “pockets” in the piping can “slug” the compressor causing extensive damage. The suction line immediately attached to the separation device (knockout drum, scrubber) should be slightly sloped so that liquid drains back to the vessel. Inadequacy of liquid separators is a common complaint in operating plants. They may be too small, unable to take slugs, or located too far from the compressor. The latter is a very common failing. This aspect of plant design should be studied very carefully. Very often the design is based on preliminary dew point calculations for a given gas, but later the composition changes. Accordingly, the design should include a generous safety factor.

Where there is any possibility of liquid condensation, the suction lines should be heat-traced. A second line of defense is to include a liquid separation chamber in the pulsation damper (see API 618). Pulsation dampers can be heat-traced with a “plate coil” for steam, or with electrical heaters. Bayonet heaters can also be supplied.

The distance between the pulsation damper and the compressor cylinder should be held to a minimum.

It used to be common practice to install intercoolers on top of the compressor cylinders. Each cooler would span across two cylinders, with pulsation-damping volume built into the coolers. This arrangement made a compact installation, but provided little defense against liquids, and compromised maintenance access. It also required that the suction valves of one of the two cylinders be placed in the undesirable location on the bottom of the cylinder. Currently, on-compressor mounting of coolers is no longer used, except perhaps for bone-dry gases.

Piping design should be analyzed to assure that forces and moments exerted on the cylinder flanges are within the vendor's tolerances. Cold springing should not be allowed. (Cold springing is forcing pipe to the machine in a cold condition in order to afford relieved stresses as the pipe heats to operating temperature.)

371 Instrumentation and Control

Compressor control systems use suction pressure, discharge pressure, flow, or an external process signal to regulate capacity. Control systems are typically electronic or pneumatic. Capacity of the compressor is adjusted by:

1. Actuating unloaders and clearance pockets
2. A control valve in a bypass
3. Changing speed

The control system can be manual or automatic. Many systems today are programmable. This section will briefly describe the control devices and instruments usually supplied with the compressor. Overall control system design is not covered (refer to the *Instrumentation and Controls Manual* for information on system design).

372 Control Devices

Capacity Control

Five-step unloading means capacity control in approximately 25% steps from zero to 100%. Therefore, if two 50% compressor units are used, capacity control is available in 12.5% steps. Suction valve or plug-type unloaders can only deactivate each cylinder to zero or 50%. If the 25% intermediate step is desired, it is necessary to use clearance pockets. It is possible to obtain five-step unloading in increments other than 25%.

When the pressure ratio is quite low (less than approximately 1.7) the volume of the clearance pocket becomes very large. The actual volume is dependent on the piston

displacement and “k” value of the gas. Refer to the formula for volumetric efficiency. Also, API 618 recommends that volumetric efficiency remain above 40% since performance prediction is generally unreliable below that value. (However, values as low as 25% have been used when less precise capacity control is acceptable.)

Note also that when the cylinder bore is not much larger than the piston rod, unloading the crank end of the cylinder results in a capacity reduction much less than 50%.

Three-step (100%, 50%, 0%) capacity control is used on some general purpose machines such as air compressors. Three-step control requires more cyclic actuation of the unloaders than five-step control. Therefore, three-step control is more detrimental to machine components (particularly valves).

When precise capacity turndown is required, a bypass with a control valve is necessary. Depending on the system requirements, the bypass may only be across the first stage. But, more often the bypass spills back across all stages unless the differential pressure is too high to be handled by a single control valve. The take-off point for the bypass must be downstream of a heat exchanger so that cooled gas will be spilled back to the suction. If there is no exchanger in the discharge, the bypass must branch into the suction line upstream of an exchanger. Alternatively, a cooler may be placed in the bypass line. In any case, the bypass should tie-in upstream of a suction knockout drum so that any condensate resulting from the expansion cannot enter the compressor.

When a bypass is used in combination with step control, five-step operation is more efficient than three-step operation.

373 Instruments

Loadless Starting

Most motor-driven compressors are equipped with suction valve or plug-type unloaders on both ends of all cylinders on the frame to permit loadless starting. Loadless starting is not mandatory, but facilitates startup, and reduces disturbance in the electrical system. It is also a convenient and less punishing feature for all types of drivers.

Alarms and Shutdowns

Figure 300-63 is a typical list of alarm and shutdown functions for the compressor frame and cylinders. Local preferences may call for additions to or deletions from this list.

The data sheets in API 618 list many other functions. These data sheets also serve as a good checklist.

Refer to the *Driver Manual* for recommended alarm and shutdown parameters for drivers.

Fig. 300-63 Typical List of Alarm and Shutdown Functions

Function	Alarm	Shutdown
Low lube-oil pressure	x	x
High discharge temp. (each cyl.)	x	x
High oil filter diff. pressure	x	
High cyl. jacket water temp.		x
Low cyl. jacket water pressure	x	
Low lubricator flow		x
High vibration	x	x
High liquid level, gas separator	x	x

If a vibration detection device is used on the frame, an accelerometer type detector is recommended to provide continuous measurement. Ball-and-seat or magnetic-type switches are unreliable. Consult the vendor regarding the best location for the device. Sometimes it is necessary to install two devices (one for transverse vibration, and one for axial vibration).

Note that whatever alarms and shutdowns are chosen, it is important to make sure they are installed with facilities to allow testing.

Gages

Location of the gages is generally a matter of plant preference. They can be mounted locally on the compressor or piping, or mounted in a local panel. Gages mounted on the compressor or attached piping may be subject to vibration. Check with plant operating and maintenance personnel regarding the best location for these gages. (Figure 300-64 is a typical list of gages.)

Monitoring

Monitoring systems of the continuous type are not widely used on reciprocating compressors. However:

Eutectic bearing temperature safety devices are available for main and connecting rod bearings. These devices are spring-loaded and they vent control air to alarm or shut down the unit when high temperature melts the eutectic material. At one large Company installation, such devices prevented a major failure. Main bearing thermocouples or RTD's are also available.

One device that is gaining more acceptance is a rod-drop monitor. This device can be a eutectic sensor that melts when rubbed by the rod allowing a signal to be transmitted for alarm or shutdown. This function can also be accomplished with a proximity probe.

Various lubricator flow and drive failure sentries are also available for cylinder and packing lubricator systems.

Fig. 300-64 List of Gages

Function	Temperature-Gage	Pressure Gage
C. W. outlet, each cyl.	x	
C. W. supply	x	x
Lube-oil to frame	x	x
Lube-oil out		x
Lube filter diff press	x	
Suction gas, each stage	x	x
Each interstage gas		x
Disch. gas, each stage	x	
Final disch gas		x
Packing (500 psi and higher)		x
Lube cooler, inlet/outlet	x	
Supply to each cooled packing	x	x
Outlet of each cooled packing	x	

380 Foundations

This sub-section provides a basis for establishing the dynamic forces to be used by civil engineers in foundation design calculations. Soil mechanics, natural frequency calculations, bearing pressure, concrete strength, and other design factors are not covered here. Refer to the *Civil and Structural Manual* for such information. Foundations, anchor bolts, and grouting are discussed in the *General Machinery Manual*, and Section 366 of this manual.

In addition to knowing the dimensions and weights of the machinery to be supported, engineers designing the foundation must know the magnitude, direction, and frequency of the dynamic forces that the machinery will exert on the foundation.

The importance of foundations to a compressor installation cannot be overemphasized. Foundations attenuate vibratory forces generated by the machinery, and reduce transmission of these forces to the surrounding plant and equipment. Foundations also keep the machinery in alignment.

To perform these essential functions throughout the life of the installation, the foundation must be sized to support the weight of the machinery while imposing a tolerable bearing pressure on the soil or structure. It must be properly designed so that the system, consisting of the foundation, soil, machinery, and piping, is not at or near a resonant condition. It is particularly important on offshore structures, which may be susceptible to resonance from the machinery vibration.

The purchaser of the machinery is normally responsible for the design of the foundation. **The vendor or manufacturer of the machinery will seldom take this**

responsibility because his expertise is not in this field. It would not be in his best interest to accept the risks associated with the design. Additionally, the vendor does not have specific knowledge about the soil conditions at the site.

381 Reciprocating Compressor

Unbalanced Forces and Moments

Reciprocating machines generate **primary** and **secondary forces** and **couples** as a result of unbalanced rotating masses and unbalanced reciprocating masses which accelerate and decelerate each revolution (see Section 300). These forces and couples react at the main bearings, and the resultant forces are transmitted to the foundation via the frame and bolting. These forces and moments are sometimes called **shaking forces**. **Primary** refers to the frequency of running speed; **secondary** means the frequency of twice running speed.

Forces arising from differential gas pressure acting on the piston and piston rod have no effect on the unbalanced forces and moments transmitted to the foundation.

The magnitude of these forces and moments can sometimes be extremely large, and they must be accounted for in the foundation design. There is no “rule of thumb” to predict the magnitudes as reciprocating machine designs vary widely in terms of the number and sizes of cylinders and crankshaft configuration. Data Sheet CMP-DS-875 may be sent to the vendor with the Company's quotation request, and preliminary values are provided by the vendor so that the forces and moments of the various machines proposed can be evaluated. Preliminary foundation design can begin with this information, and be refined as the actual forces are furnished by the vendor.

Other Considerations

Any natural frequency of the foundation should be at least 30% above or below the primary and secondary frequencies. For instance, on a **450 RPM** unit, ideally any natural frequency should not be anywhere in the range of 315 to 1170 cycles per minute (cpm). Sometimes it is impractical to keep natural frequencies out of a range that wide. Accordingly, it may be possible to place a natural frequency between the primary and secondary frequencies. A natural frequency in the range of 585–630 cpm would satisfy the criterion of 30% separation margins in the above example.

For reciprocating units, as a rule of thumb, the weight of the foundation should be a minimum of five times the combined weight of the frame, cylinders, and driver.

382 Pulsation Suppression

Background

Simple **volume bottles** were the earliest method, and are still used in some applications for pulsation suppression. Many individuals and companies had their own methods of sizing bottles. These bottles generally worked satisfactorily; however, some installations experienced intolerable piping vibration and failures.

In the early 1950's, the Southern Gas Association appointed Southwest Research Institute to investigate the problem. The result was the development of an analog computer to simulate the acoustical interaction of one or more compressors and the associated piping. At first, the analog results were analyzed only from the standpoint of pressure pulsation amplitudes and frequencies. Later, the piping systems were analyzed for mechanical interaction between the gas pulsations and piping vibration.

Due to this research, in many cases pulsation dampers with proprietary internals replaced volume bottles except in low pressure applications. These dampers are also called **snubbers** or **suppressors**.

Design Approaches

Numerous approaches are currently employed to avoid excessive vibrations that can occur in the connected piping, instrumentation, etc. These include:

1. Relying on vendors to design and supply adequate suppression dampers along with the compressor, without stating any specifications. Vendors may use digital calculations or proprietary empirical correlations. This is normally 5–7% of line pressure peak-to-peak pulsation at the bottle outlet unless otherwise specified.
2. Relying on vendors to design and supply suppression dampers, based on a specified maximum pulsation amplitude at the outlet of the damper. The specified pulsation may be 1–5% of line pressure and determined by estimating pulsation affects on the piping system.
3. Specifying the size of pulsation dampers based on empirical correlations within the Company.
4. Specifying that the vendor obtain an analog study of the pulsation dampers and associated piping, for machines not specified by API 618.
5. Specifying one of the three design approaches as outlined in API 618, as summarized below.

Most manufacturing and many producing locations employ analog studies (from SWRI) when purchasing new equipment. These studies are expensive, however, and may not always be appropriate. Analog studies have the disadvantage of not being very flexible. If changes in piping, vessels, or operating conditions are made after the analog study, the entire analog must be redone at additional cost. For example, Warren Petroleum, which employs a large inventory of integral and high-speed separable compressors used in varying applications, uses all of the previous approaches, as they determine to be applicable.

API 618 thoroughly specifies the requirements for pulsation suppression devices in terms of pressure drop, pulsation level, and mechanical design. Three design approaches are listed in API 618:

Design Approach-1 does not include a simulation study. Empirical factors and experience are applied, and everyone involved hopes for the best.

Design Approach-2 calls for a simulation study which reveals the pulsation amplitudes and frequencies, but does not predict how much the piping is going to vibrate.

Design Approach-3 is an extension of Design Approach 2 where the structural design of the piping is reviewed to determine its vibration tendency with respect to excitation by the gas pulsations.

Some compressor and damper vendors have digital computer programs to simulate compressor systems. They are faster, and may be less expensive than analog studies, and have proven to be reliable. Digital programs also have the advantage of saving a file which can be easily modified when changes are made.

The Company is a member of the SGA-SWRI program for analog simulation. Many systems have been successfully analyzed for the Company using SWRI's services. For large, critical projects, it is recommended that SWRI continue to do this analytical work, because although digital programs have some advantages, SWRI has the overwhelming amount of simulation experience.

Moisture Removal

In many reciprocating compressor installations the pulsation dampers have integral moisture removal sections to provide a final line of defense against liquid carry-over. There are two schools of thought. Pulsation dampers can be designed with these moisture removal sections, or they can be designed so that liquids positively cannot collect. Most refinery pulsation dampers have moisture removal sections. Many oil field dampers do not have moisture removal sections.

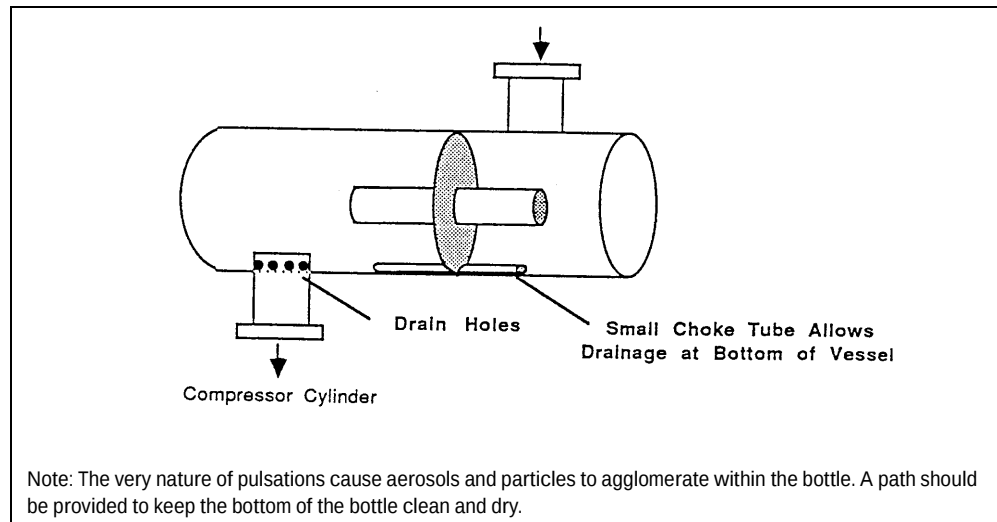
Designing pulsation dampers without moisture removal sections takes great care when the damper includes internals for pulsation attenuation (choke tube and baffle). The baffle needs to have a hole at the bottom to prevent liquid from accumulating on either side. However, a hole in the baffle can make the damper less effective as an acoustic filter. The way around this is to design the damper with an additional choke tube at the bottom of the damper and acoustically model the damper in this configuration. Another problem is the lip created by the nozzle projecting into the bottle. Figure 300-65 illustrates the design aspects.

Additional Notes

Some have questioned the bottle-sizing equation in API 618, and would prefer to use other, less-expensive procedures. Nevertheless, the API equation is based on a combination of theory and statistics, and is recommended as a conservative practice. If an exact equation were available, acoustical simulation studies would not be needed.

Note that for pulsation control, bigger is generally better. On the other hand, there can be interference problems and access to cylinders can be limited. There is a point of diminishing return where a bigger bottle does not gain that much on pulsation attenuation.

To guard against high late charges after the purchase order has been committed, it is a good idea to ask the vendor to quote a "dollar-per-pound" price addition in his

Fig. 300-65 Pulsation Damper Without Moisture Removal Section

proposal for pulsation damper changes. In this way, if the simulation study reveals the necessity for changes to the dampers, there will be a firm-price basis for them.

In setting up the simulation study, be sure to consider all operating conditions.

Beware of variations in molecular weight. For example, some hydrogen compressors handle nitrogen during regeneration. The acoustic velocities of these two gasses differ by nearly a factor of four (4400 fps versus 1200 fps @ 100°F). If the machine is an addition to a plant, the existing machines should be included if they are in the same system. Evaluate combinations of unloading steps, speeds, and variation in gas composition. Sometimes only the extremes of the conditions need to be simulated, but SWRI should be the judge of which conditions can be ignored.

The simulation is not a design tool. The piping design must be completed before simulation can be started. It can, however, indicate trouble spots which can be corrected, and the simulation is then re-run. In view of this procedure, it is advisable to have a Company piping designer or other representative present during the simulation.

390 Rates and Materials of Construction

391 Capacity Changes

The following methods can be used to change the capacity of the machine:

- Change speed
- Increase or decrease clearance of cylinders
- Re-bore unlined cylinders and change pistons
- Re-line cylinders and change pistons
- Replace cylinders

Speed change is rarely a viable option. Obviously, increasing speed will not be an option if the frame is already running at the maximum rated speed. Change in the speed of a motor-driven unit requires replacement of the driver. If the driver is a variable speed machine, there is a chance that something can be done, but the system must be carefully studied. The torque and power ratings of the driver, compressor, couplings, and gear (if any) should be checked, and a torsional analysis of the system conducted. A review of the pulsation dampers and piping design may also be required.

For small changes in capacity, it may be possible to alter the clearance of the cylinders. It will usually be very difficult to significantly increase capacity in this manner unless the cylinders were originally over-clearanced. However, it might be possible to use this method in combination with other methods. Ordinarily, capacity can be reduced quite easily by adding clearance, with spacers or clearance pockets, or by reducing piston length.

Unless the cylinders are at maximum diameter, unlined cylinders can be re-bored, sometimes by a substantial amount, to increase displacement. Similarly, liners can be replaced in lined cylinders to increase or decrease displacement. There is no guideline for the amount of displacement change that can be achieved. Consult the manufacturer. If changes are contemplated, a computer study of the combined rod loads should be made. Although a large change would not be expected, the manufacturer should review the torque-effort diagram.

In rare cases, where it is desired to increase capacity and power, the crankshaft may be a limiting factor. Crankshafts have maximum horsepower-per-throw and total horsepower (torque) ratings.

There is no way to upgrade the maximum allowable working pressure of a cylinder, unless it was under-rated for the original application. It is worthwhile asking the manufacturer whether the cylinder's present nameplate is the actual maximum rating.

392 Valve Upgrades

Before considering valve upgrades, the overall compressor system must be evaluated. For example, to cure a liquid problem, the suction vessel must be looked at. Check the vessel sizing and the damper boot sizing. Check for insulation and heat-tracing if condensation is a possibility. Always keep the cylinder jacket water at least 10 to 20°F above the inlet gas temperature. Most liquid slugging occurs at startup when the compressor is cold. Make sure there are no dead legs where liquids can accumulate. All of these problems should be corrected before making a decision to upgrade the valves.

Valves do not pass liquids well. Liquid slugs have been known to cause broken pistons and, in at least one case, actually separate the compressor cylinder from the crankcase. The usual evidence of liquid slugging is severely damaged plates or no plates at all; only pieces. Springs can also collapse, although this is sometimes difficult to detect, since by the time the cylinder is opened the water or hydrocarbon liquid may have drained or evaporated.

Lube oil accumulation from over-lubrication can have similar effects. This is especially prevalent with some compressors equipped with plug type unloaders on the suction or top side of the cylinder. In one case, a piston assembly and valves were severely damaged when the compressor was loaded after being operated unloaded for about 30 minutes. Oil had accumulated in the valve pocket due to the design of the internal suction passages.

There are many new valve designs available from OEM and non-OEM sources. Valve problems are often design related, and they can be solved by a change in materials, or valve type. For example, one Company compressor experienced over 50 valve failures over a three-year period. Although the gas was wet, changing from strip to peek plate valves dramatically reduced the failure rate.

Before recommending replacement of existing compressor valves, ask the vendor to conduct a valve motion study on the existing application. Be sure the valve design is such that neither the valve guard nor the assembly bolting can fall into the cylinder, even if the valves assembly bolting breaks or comes loose.

Ask the prospective supplier how his valves and unloading system deviate from API 618. Confirm that the existing valves are being built to specifications according to dimensions, material hardness, etc. **Whether valves are repaired by a contractor, or in-house, established specifications and procedures are a prerequisite to reliable performance.**

Non-metallic valve plate materials have been developed for relatively high temperatures. These materials are only about one-sixth the weight of steel. Also, non-metallic materials can be contoured to reduce the drag coefficient for the flow around the plate. These designs have shown a great deal of improvement in valve life at some installations. However, existing valves should not be indiscriminately replaced with these new designs.

It is important to determine whether the manufacturer designs and manufactures the unloader system or only the valves. Be careful of a manufacturer who supplies only valves but relies on others for the unloader mechanism. The unloader system is an integral part of most compressors and must be designed in conjunction with the valves.

Also, determine the manufacturer's capability in materials engineering, finite element analysis, and non-destructive examination. Find out whether the manufacturer can perform mechanical testing including tensile, hardness, and impact tests.

393 Suction System Modifications

Caution should be exercised when modifying reciprocating compressor suction systems because changes may alter the acoustic response. Unacceptable levels of vibration, high piping and nozzle stresses, and compressor valve problems may result from the addition of knockout vessels and coalescers, or from piping changes. Chances are often good that problems will not result, but there is a very real risk, and your project may be the statistic. In new equipment installations, an acoustic study is the tool used to mitigate this risk.

Installing a coalescer or other piece of equipment in a reciprocating compressor suction line changes the acoustic length of the line, or creates two new acoustic lengths where there was previously one. An acoustic study is a design review of these lengths to determine if any acoustic resonances will occur and if they will coincide with the mechanical natural frequencies in the piping system. In some situations it is prudent to revisit this work when making field changes.

API 618 gives guidance as to when an acoustic study is recommended for new machinery installations. An acoustic study should be considered when any of the following are true:

- Two or more compression stages
- Three or more cylinders per stage
- Final discharge pressure exceeds 1000 psig
- Driven equipment horsepower is 500 Bhp or greater
- Service alternates between gases of significantly different molecular weights
- Interaction is anticipated between compressors of 150 Bhp and greater

Acoustic studies are usually not performed for machines of less than 150 Bhp.

The API recommendations provide a good basis for deciding whether to perform an acoustic study when altering a system, but risk assessment should also play a part in decision making as these studies can be quite expensive. Generally, risk of having harmful pulsations increases as compressor running speed decreases and as the gas acoustic velocity increases (usually as molecular weight decreases). As these parameters change, acoustic lengths get very long and fall out of the normal range of field piping lengths. Also, the intended use of the compressor plays a part in risk assessment. The cost of production losses should be weighed against the cost of performing an acoustic study as part of an alteration.

All things considered, a packaged high-speed compressor in a producing field gas application would be considered low risk because it is high-speed, high molecular weight, with low to moderate production losses. Conversely, a large hydrogen booster compressor in a refinery would be considered high risk for acoustic problems because it is low speed, low molecular weight, with high production losses.

394 Materials of Construction

The following is a description of materials available for some of the principle components.

Crankshafts

Crankshafts are most commonly one-piece forgings or castings. (One exception is the Worthington “Cub” which has a two-piece, coupled crankshaft.) Forged steel is used exclusively for Class A machines. Steel or nodular iron castings are sometimes used for other machines up to 1500 HP. The advantages of a casting is that counterweights can be an integral part of the shaft. However, the Company prefers forged steel for ratings of 200 HP and higher. A typical material designation is ASTM A688 Class F.

Piston Rods

The most common material is heat-treated AISI 4140 steel with a maximum Rockwell C hardness of 40 at the core and a minimum of 50 at the surface. If stress corrosion is a design factor, this material is annealed to a hardness of 22C maximum (core) and 50C minimum (surface). AISI 8620 with the same hardness provides higher working stresses for stress corrosion applications.

Rods of 17-4 PH stainless steel are used for corrosive services. Core and surface hardnesses are 40-50C for standard applications. When stress corrosion is present, the through-hardness is limited to 33C.

Crossheads

Crossheads are available in cast gray iron, nodular iron, or steel. The Company prefers cast steel for all high-horsepower applications, but has accepted nodular iron as an alternate on a few applications. It is recommended that cast gray iron be allowed only on smaller machines with ratings less than about 200 HP.

Connecting Rods

Connecting rods should be forged steel. A typical material designation is ASTM A235.

Compressor Cylinders

Maximum allowable working pressures (MAWP) corresponding to materials are typically limited to the values shown in Figure 300-66.

Fig. 300-66 Limitations for MAWP Based on Cylinder Materials

Material	MAWP, PSIG API 618	MAWP, PSIG Oil Field ≤8 Inch Diameter	8 Inch Diameter
Gray Cast Iron, ASTM A278	1000	1600	1000
Cast Nodular Iron ASTM A395	1000 ⁽¹⁾	2500	1500
Cast Steel, ASTM A216	2500	2500	2500
Forged Steel, ASTM A668	>2500	7500	—

(1) May be quoted to 1500 PSIG as separate option

Note that nodular iron may be used for Class A machines above 1000 PSIG only in special cases. Nodular iron is an excellent engineering material, but homogeneity of the material throughout the casting can sometimes be a problem. Thus, the yield strength may not be as high as anticipated. API 618 calls for specimen testing and other NDE in an effort to ensure the quality of nodular iron castings.

Compressor Valves

Valve materials must be selected for both durable, long-term operation and compatibility with the gas being handled. Extensive research and field testing contribute to the improvement of valve materials.

Figure 300-67 shows materials for valve guards and seats. Valve plates are available in various types of stainless steels and thermoplastics, as shown in Figure 300-68. Figure 300-69 shows the wide variety of spring materials available, from music wire to Inconel. Other super alloys, such as Elgiloy and Haynes 25, are being used to avoid hydrogen embrittlement for springs.

Fig. 300-67 Typical Guard and Seat Materials for Compressor Valves

Material	Application
1141	Light Duty Non-Corrosive Service
Heat Treated 1141	Light to Medium Non-Corrosive Service
Ductile Iron	Light to Medium Service - Resistance to some Chemical Attack
4140	Medium to High Strength - Resistance to some Chemical Attack
Heat Treated 4140	High Strength Service - Resistance to Some Chemical Attack
400 Series Stainless Steel	Corrosive Service
300 Series Stainless Steel	Extreme Corrosive Service
17-4 PH Stainless Steel	

Fig. 300-68 Typical Valve Plate Materials

Material	Application
Glass Filled Nylon Thermoplastic	Good Impact and Corrosion Resistance. 270°F Temperature Limit
Peek - Polyetheretherketone	High Strength - High Temperature (up to 375°F)
Linen Based Phenolic Laminate	Clean Gas. Low Compression Ratios. 225°F Temperature Limit
Laminated cloth Based Phenolic	High Temperature Applications up to 400°F - Available for Ported Plate Application only
410 Stainless Steel	Moderate Corrosion Resistance, Good Impact Resistance
17-7 PH Stainless Steel	Moderate Corrosion Resistance, Good Impact Resistance
Inconel X-750	High Corrosion Resistance and High Strength Properties in High Temperature Applications

Fig. 300-69 Typical Valve Spring Materials

Material	Application
Music Wire	Low Corrosion Resistance. Good Durability in Clean Gas Environments and Low Temperatures
302 Stainless Steel	Moderate Corrosion Resistance. Average Durability in Moderate Temperatures
17-7 PH Stainless Steel	Excellent Corrosion Resistance, High Strength Properties in Moderate/High Temperatures (700°F Max)
Inconel X-750	High Corrosion Resistance, High Strength Properties in High Temperature Applications (1100°F Max)

Quality assurance and accurate dimensions are equally important. Surface finish and parallel face surfaces are the most serious considerations for metallic plates. Dimensional stability of thermoplastic plates in humid and high temperature environments is essential.

One drawback to the use of thermoplastic plates is their affinity for moisture, called hygroscopicity. Newer materials, such as PEEK and TORLON, have lower absorption rates (.06%) than nylon glass composites (1%). Also, some of these materials have a lower coefficient of thermal expansion. For example, Dresser-Rand's "HiTemp" valve material has a thermal expansion factor of 1.2×10^{-6} in/in-F, almost half that of nylon glass composites. A low thermal expansion factor makes the plate more resistant to deformation at higher temperatures and better able to hold dimensional integrity.

Quality Assurance on Replacement Parts

A number of processes are used to manufacture valve plates. Among them are die stamping, electron discharge machining, laser cutting and water jet cutting. Each process has advantages and disadvantages with regard to the plate's fatigue strength.

Laser cutting, for example, causes a heat affected zone, which can be minimized if the plate is cut at optimum speed. A recast layer will be found on the cut edge as it comes from the laser; this must be removed. One manufacturer does this by tumbling the plates in a stone media. This method satisfactorily removes the recast layer and also forms a radius on the edge of the plate surface.

Metal valve disks or plates should be suitable for installation with either side sealing and should be finished on both sides to 16 microinches RMS. Edges should be suitably finished to remove stress risers. Valve seats and sealing surfaces should also be finished to 16 microinches.

The ends of coil-type valve springs must be squared and ground to protect the plate against damage from the spring ends.