10

RECIPROCATING COMPRESSOR PERFORMANCE AND SIZING FUNDAMENTALS

From the preceding chapters we recall that most reciprocating compressors encountered in process, gas or oil production, and gas transmission applications use double-acting cylinders. This simply means that compression occurs on both the outward and inward strokes of the piston. This is accomplished using a packed piston rod firmly attached to a crosshead. The crosshead, in turn, is attached to the connecting rod via a wrist pin.

Reciprocating compressors offer the following advantages to the user:

- Flexibility in design configuration
- Good efficiency even in small sizes, at high pressures, and at part loads
- Operating flexibility over a wide range of conditions for a given configuration

Our objective is to introduce the reader to the basics of how to calculate reciprocating compressor performance and to present a methodology of estimating compressor size and power requirements. The methods given here are approximate by necessity, and the reader is encouraged to communicate with compressor vendors if more accurate results are required.

We begin by reviewing basic capacity and horsepower calculations and demonstrating the effect of design variables on the results of the calculations. Basic equations are presented that will enable readers to estimate capacity and horsepower along with frameload. Further information is given that will allow estimation of compressor size given some general information on compressor cylinders, strokes, rotative speeds, horsepower capacity, and rod load capability. The standard nomenclature used by the majority of U.S. compressor manufacturers has been selected for our calculations.

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

10.1 THEORETICAL MAXIMUM CAPACITY

The theoretical maximum capacity of a reciprocating compressor cylinder is given by

$$Q = 0.0509 \frac{P_s}{T_s} \frac{Z_{\text{std}}}{Z_s} (\text{DISP})[1 - \text{CL}(R^{1/N} - 1)] \tag{10.1}$$

where Q = capacity, million standard ft^3/day (ref. 14.7 psia, 520°R)
P_s = suction pressure, psia (flange)
T_s = suction temperature, °R
Z_{std} = compressibility factor at standard conditions
Z_s = compressibility factor at suction conditions
DISP = cylinder displacement, ft^3/min
CL = cylinder clearance volume as decimal fraction of displaced volume
R = pressure ratio across cylinder (flange to flange)
N = isentropic volume exponent at operating conditions (specific heat ratio for ideal gas)

The critical portion of Eq. (10.1) is the theoretical volumetric efficiency, defined as

$$\text{VE} = 1 - \text{CL}(R^{1/N} - 1) \tag{10.2}$$

This equation describes the variation of a given compressor capacity as a function of residual clearance volume, pressure ratio, and gas. The trends are:

- Decreases with increasing clearance
- Decreases with increasing pressure ratio
- Increases with increasing volumetric exponent

The other variables in the capacity equation are related to fluid density at the compressor cylinder inlet. Capacity increases with increasing inlet density.

Figures 10.1 and 10.2 demonstrate the variation of compressor capacity with clearance and pressure ratio. The compressor chosen for these and most of the other figures that follow has the following specifications:

- Bore = 20 in. (double acting)
- Stroke = 15 in.
- Clearance = 15%
- Rod diameter = 3 in.
- Rotative speed = 327 rpm
- Gas = methane
- P_s = 15 psia
- T_s = 560°R (100°F)

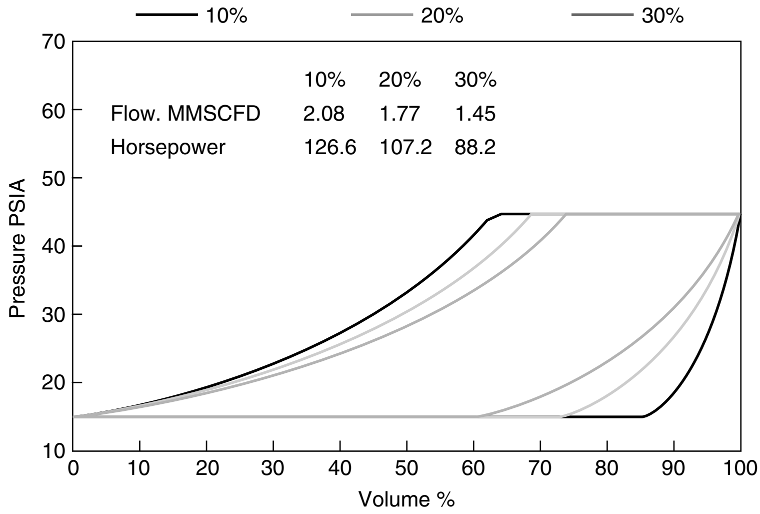

FIGURE 10.1 Effect of clearance. *(Dresser-Rand Company, Painted Post, N.Y.)*

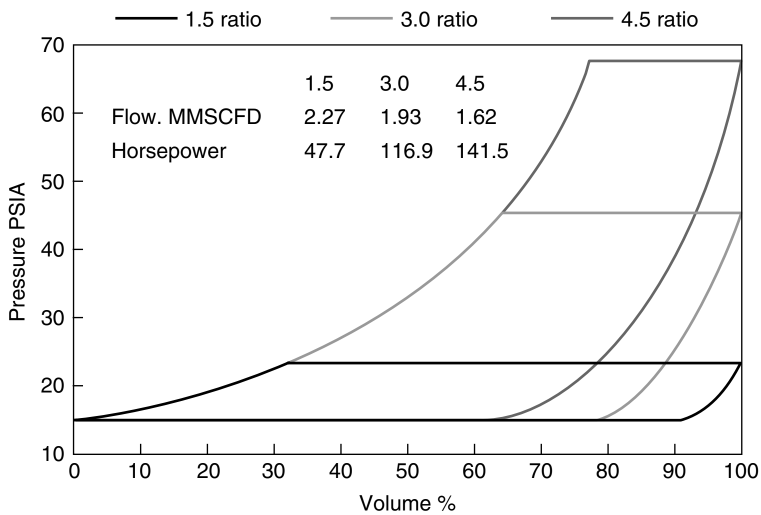

FIGURE 10.2 Effect of compression ratio. *(Dresser-Rand Company, Painted Post, N.Y.)*

10.2 CAPACITY LOSSES

Real compressors with valves, piston rings, packing, heat transfer, and attached piping do not pump ideal capacity. There are a number of loss factors that generally reduce the capacity. This section covers these factors and how they vary with design and operating parameters. The *p–V* diagrams reproduced in Fig. 2.43 graphically demonstrate the effect of most of the loss factors on capacity.

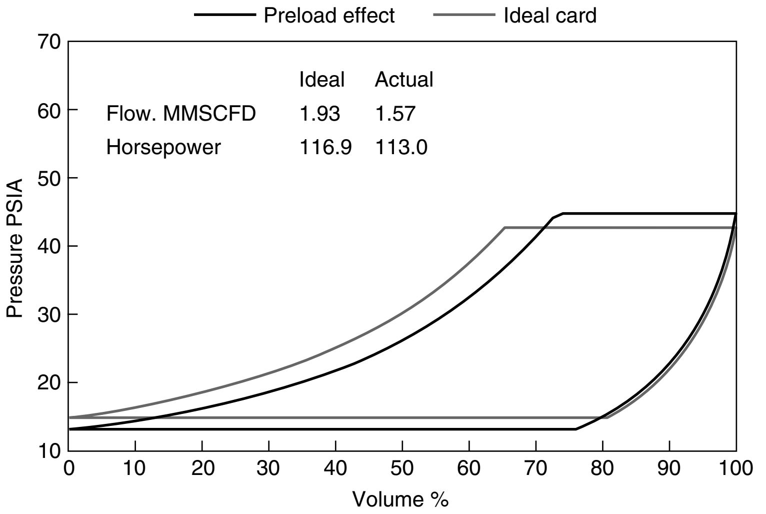

FIGURE 10.3 Effect of preload. *(Dresser-Rand Company, Painted Post, N.Y.)*

10.3 VALVE PRELOAD

As mentioned earlier, reciprocating compressor valves are essentially spring-loaded check valves. Manufacturing tolerance and reliability considerations cause the designer to introduce a positive preload. This means that the compressor must develop a small pressure drop across the valve in the direction of flow before the sealing element will begin to move. The effect of this is to increase the pressure ratio across the compressor cylinder. The pressure trapped in the cylinder will be higher than discharge pressure at minimum volume and lower than suction pressure at maximum volume. The net effect is twofold: (1) the pressure ratio across the cylinder is higher than expected, which decreases capacity by decreasing volumetric efficiency; and (2) the gas density in the cylinder at maximum volume is lower than expected while the density at minimum volume is higher than expected, resulting in reduced capacity.

Figure 10.3 demonstrates the effect. This effect is most pronounced at low suction pressures and decreases to the point where it is negligible at higher pressures. The designer must, however, know the details of the valve design to be used to accurately predict the preload effect on capacity.

10.4 VALVE AND GAS PASSAGE THROTTLING

Compressor valves and cylinder gas passages must see the flow that goes through the bore. Pressure losses are associated with this flow, and the suction losses have an effect on capacity. As each increment of fluid flows into the cylinder bore, it experiences a pressure drop. The process may be closely approximated as isenthalpic. Once each increment of fluid is in the cylinder, it must be recompressed to the pressure that exists at maximum volume. The work required to do this increases the temperature of the fluid so that it is higher at maximum volume than the suction temperature. This reduces the density at maximum volume and therefore the capacity.

The effect of discharge valve and passage flow losses on capacity is determined by whether all of the fluid that should flow out of the cylinder does so by the end of the stroke (minimum volume). There is a similar effect on the suction side. The impact of this on capacity is similar to the effect of valve preload.

Figures 10.4 and 10.5 demonstrate the capacity loss due to throttling. Figure 10.4 shows the same compressor configuration as that used previously. Figure 10.5 reflects the effect of increased rotating speed by shortening the stroke to 5.5 in. and increasing the speed to 892 rpm. The bump at the beginning of the valve event is an inertia effect. It is more pronounced at the higher rotative speed.

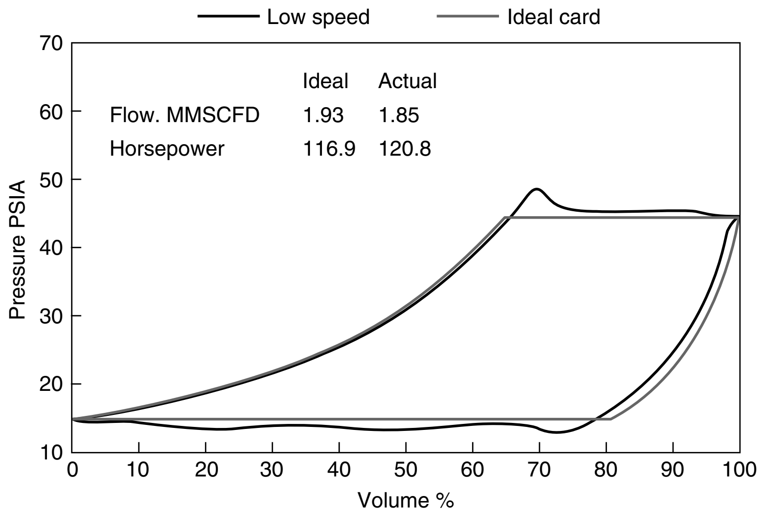

FIGURE 10.4 Effect of valve and passage flow losses, low speed. *(Dresser-Rand Company, Painted Post, N.Y.)*

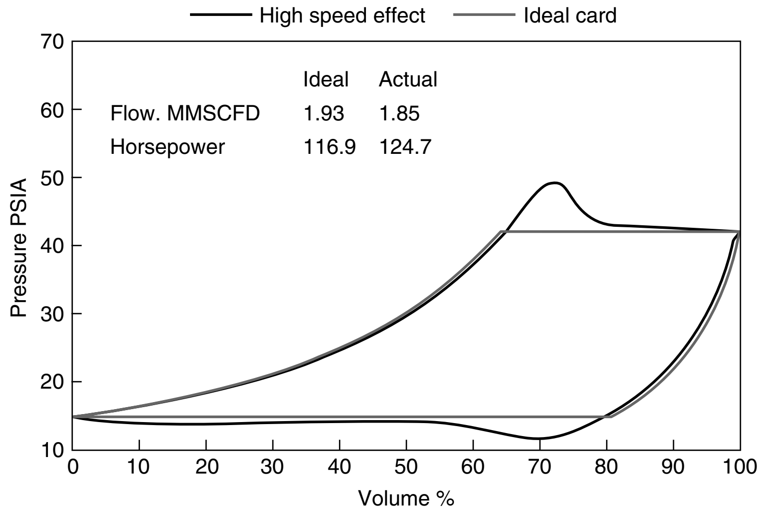

FIGURE 10.5 Effect of valve and passage flow losses, high speed. *(Dresser-Rand Company, Painted Post, N.Y.)*

The throttling capacity loss has an inverse relationship to flow efficiencies built into the valves and cylinder flow passages. Large and open valves and passages mean low pressure losses and low capacity loss. They generally mean higher clearances that also reduce capacity. Thus, low valve losses do not always improve capacity.

To predict the effect of valves on capacity accurately, we must know the details of the design being used. While we generally know the average velocity through the lift area, we must know how efficiently the valve design uses that area. The true loss is determined by what can be termed the *effective flow area*, defined as the product of geometric lift area and flow coefficient. The flow coefficient can vary widely from one valve design to another.

10.5 PISTON RING LEAKAGE

The compressor piston uses seals to minimize leakage. However, ring leakage does occur and has a detrimental effect on capacity. In a double-acting compressor, the effect is twofold: (1) As fluid leaks from the higher-pressure side of the ring, the capacity is reduced by loss of mass in the high-pressure end. This also increases the mass in the low-pressure end, thus decreasing the mass that will flow in through the suction valves. (2) The leakage process is closely approximated as isenthalpic. This has a tendency to increase the temperature in the low-pressure end of the cylinder. The result is lower density at maximum volume, further reducing capacity.

When the piston reverses and the high-pressure end becomes the low-pressure end, the process reverses. Thus, a small amount of gas is essentially trapped in the cylinder. The effect of piston ring leakage is shown in Fig. 10.6 for a double-acting cylinder. A single-acting cylinder will generally show higher leakage than a double-acting cylinder because the time-average pressure drop in one direction is higher. The temperature effect is also there if the inactive end of the cylinder is vented to the suction of the active end. This is normally the case.

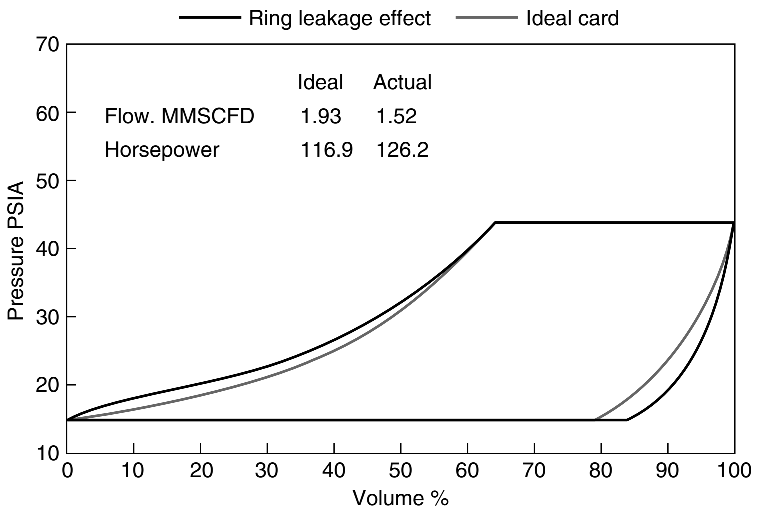

FIGURE 10.6 Effect of piston ring leakage. *(Dresser-Rand Company, Painted Post, N.Y.)*

The effects of design and operating parameters on ring leakage are such that leakage:

- Decreases with increasing rotative speed
- Decreases with increasing bore
- Increases with decreasing molecular weight
- Increases with pressure ratio
- Decreases with the number of rings
- Is higher with nonlube construction

10.6 PACKING LEAKAGE

Compressors that use piston rods have packing. As in the case of piston rings, the designer attempts to minimize the leakage, but a small amount will occur unless the packing a buffer pressure has been introduced that can negate leakage from inside the cylinder. When the packing leaks, the effect on capacity is limited to the loss of fluid from the packed end of the cylinder. The effect is shown in Fig. 10.7.

The effects of operating and design parameters cause packing leakage to:

- Increase with increasing pressure level
- Increase with rod diameter
- Increase with decreasing molecular weight
- Decrease with increasing rotative speed

10.7 DISCHARGE VALVE LEAKAGE

Figure 10.8 demonstrates the effect of discharge valve leakage. Fluid leaks back into the cylinder bore from the discharge passage. This not only distorts the *p–V* diagram but also

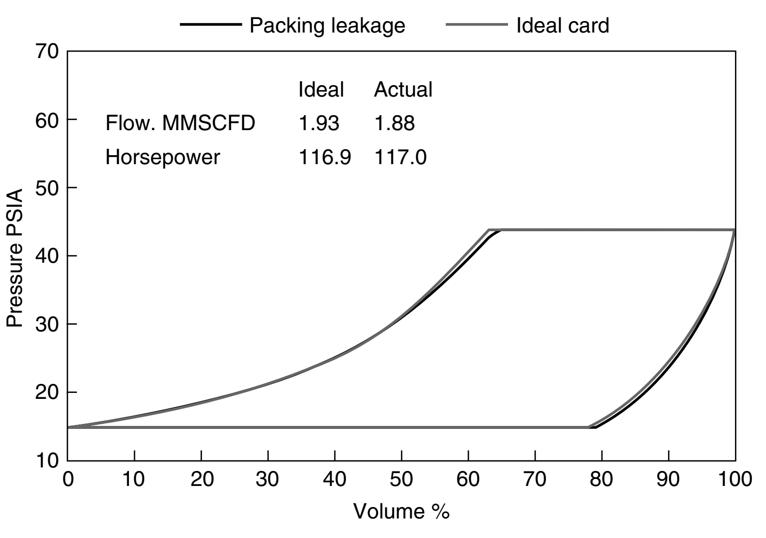

FIGURE 10.7 Effect of packing leakage. (*Dresser-Rand Company, Painted Post, N.Y.*)

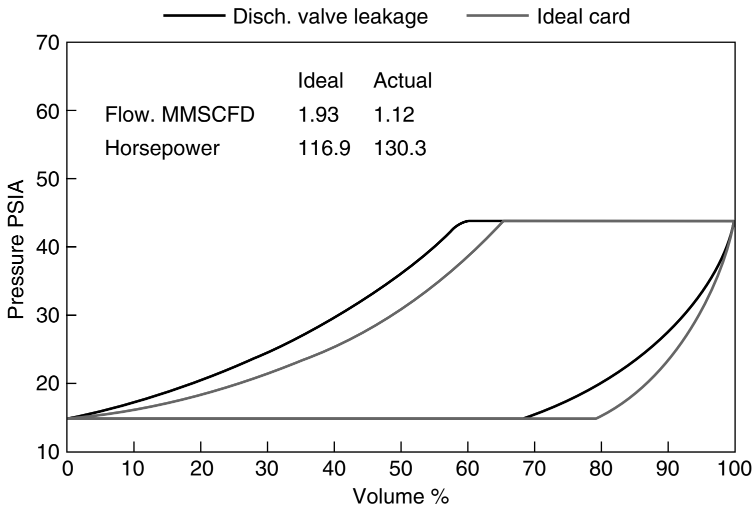

FIGURE 10.8 Effect of discharge valve leakage. (*Dresser-Rand Company, Painted Post, N.Y.)*

lets hot gas back into the cylinder. Both effects decrease capacity. As in the case of other leakage effects, the designer attempts to minimize valve leakage. The effect is so dramatic that zero leakage is a worthwhile goal.

Operating and design parameters have the following effect on discharge valve leakage:

- Increases with increasing pressure ratio
- Increases with decreasing molecular weight
- Decreases with rotative speed
- Decreases when plastic sealing elements are used

10.8 SUCTION VALVE LEAKAGE

Figure 10.9 shows the effect of suction valve leakage. Fluid leaks from the cylinder bore to the cylinder suction passage. The effect on capacity is similar to discharge valve leakage, and zero leakage is a worthwhile goal. The effects of operating and design parameters are identical to those of discharge valve leakage.

10.9 HEATING EFFECTS

Heating effects on capacity can be divided into two categories: (1) external heat transfer between the surroundings and the fluid prior to entering the cylinder, and (2) internal heat transfer between the cylinder walls, cooling medium, and the fluid while it is in the cylinder.

Heat transfer to or from the fluid prior to entering the cylinder bore has a direct effect on capacity by raising or lowering the temperature of the fluid trapped in the cylinder at maximum volume. As explained earlier, raising the temperature lowers the capacity, while lowering the temperature has the opposite effect. In fact, many operators intentionally lower inlet temperature to increase compressor throughput. This should be done with caution because there may be effects such as increased condensation that have undesirable side effects.

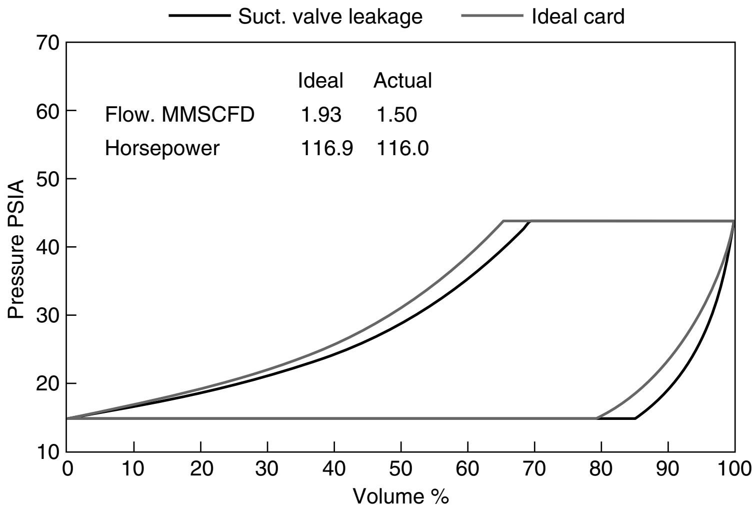

FIGURE 10.9 Effect of suction valve leakage. (*Dresser-Rand Company, Painted Post, N.Y.)*

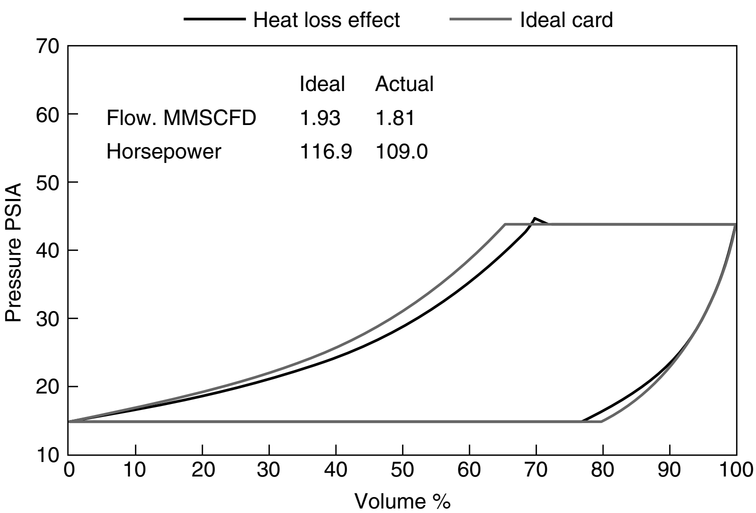

FIGURE 10.10 Effect of internal heat transfer. (*Dresser-Rand Company, Painted Post, N.Y.*)

Heat transfer internal to the cylinder bore can affect capacity both by affecting the temperature of the trapped fluid and the shape of the *p–V* diagrams. Figure 10.10 demonstrates the effect of net heat transfer from the gas while in the bore.

Although heat transfer and its resulting effects on capacity are complicated and extremely dependent on details of design, the following generalizations may be made and debated:

- For a given cylinder design, increasing rotative speed decreases heat transfer effects.
- For a given cylinder design at constant speed, increased fluid mass flow decreases heat transfer effects.

- For a given fluid at a given density, increasing cylinder bore decreases heat transfer effects.
- Increasing the differential temperature between the fluid and surroundings and/or cylinder coolant increases heat transfer effects.

10.10 PULSATION EFFECTS

Reciprocating compressors are unsteady flow machines. This time-varying flow is repeatable from one crankshaft rotation to the next. The resulting pressure variations in the connecting pipework, called *pulsations*, affect capacity. The basic effect is determined by the pressure the pulsations impose on the cylinder bore at maximum and minimum volumes. Like heat transfer, pulsations, can either increase or decrease capacity. The characteristics are:

- Higher pressure in the suction passage at maximum cylinder volume increases capacity.
- Higher pressure in the discharge passage at minimum cylinder volume decreases capacity.
- Lower pressure in the suction passage at maximum cylinder volume decreases capacity.
- Lower pressure in the discharge passage at minimum cylinder volume increases capacity.

Figure 10.11 shows the effect of pulsations for a particular set of circumstances. Predicting pulsations at the time of compressor sizing is virtually impossible. Therefore, the effects on capacity are limited by controlling pulsations to acceptable levels at a later stage of the design cycle.

With all of the potential effects of operating conditions and design features on compressor capacity, how are we to estimate compressor sizes without detailed knowledge? Fortunately, the effect of some parameters increases with speed, while the effect of other parameters decreases with speed. Generally, slow-speed compressor capacity is governed more by

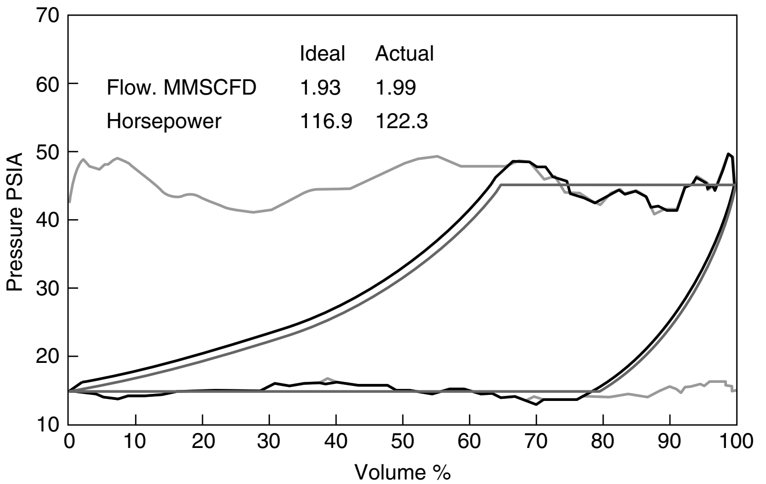

FIGURE 10.11 Effect of pulsation. (*Dresser-Rand Company, Painted Post, N.Y.*)

leakage and heat transfer effects, while high-speed compressor capacity is governed more by valve effects. We can, therefore, write a general equation to use for all reciprocating compressors as a first estimate. Expressed in MMscfd, this equation is

$$Q = 0.0509 \frac{P_s}{T_s} \frac{Z_{\text{std}}}{Z_s} (\text{DISP})[0.95 - \text{CL}(R^{1/N} - 1)] \tag{10.3}$$

This essentially states that any reciprocating compressor has a built-in 5% capacity loss. In some cases, this is a conservative estimate, and in others, it is liberal. This is a starting point only. For nonlube and/or single-acting service, an additional loss of 4% should be used in each case.

10.11 HORSEPOWER

We base the compression efficiency on the theoretical isentropic horsepower:

$$\text{hp} = 43.67Q \frac{N}{N-1} (R^{(N-1)/N} - 1) \tag{10.4}$$

Inspection of this equation indicates that the major effects on isentropic power are the capacity Q and the pressure ratio R: (1) horsepower increases with capacity, and (2) horsepower increases with pressure ratio until the decrease in capacity with increasing ratio becomes the overriding factor.

Figures 10.1 and 10.2 demonstrated that theoretical horsepower depends on volumetric clearance and pressure ratio.

10.12 HORSEPOWER ADDERS

The same factors that affect capacity have an adverse effect on horsepower. It is appropriate to call these *adders*, as opposed to the more popular term, *losses*. The reason is that the horsepower actually consumed by a compressor is almost always higher than isentropic.

Horsepower adders have three effects on the horsepower required for compression. The first effect is to add power to the suction and discharge portions of a p–V diagram. This is caused by valve losses. The second effect is to distort the compression and reexpansion lines. This is caused by leakage and internal heat transfer. The third effect is simply to add the power required to overcome mechanical friction.

Summing up the horsepower adders, we find major effects from the valving, whereas other effects, from leakage and heating, are relatively minor in all but a few applications. These are confined to low-density and low-suction pressure applications such as vacuum pumps. Friction can be treated as a direct adder. With these considerations, we can use the following equation to estimate horsepower requirements of a compressor cylinder and its associated running gear:

$$\text{bhp} = 43.67Q \frac{N}{N-1} [R^{(N-1)/N} - 1] \frac{1}{N_c} \frac{1}{N_m} \tag{10.5}$$

where N_c is the compression efficiency and N_m is the mechanical efficiency.

Compression efficiency varies with many factors, and a unique relationship is difficult to define. However, companies such as Dresser-Rand recommend using 0.85 as a first guess for lubricated service. Deduct an additional 0.05 for nonlube and/or single-acting service. Mechanical efficiency is generally accepted to be approximately 0.95, and this value is recommended by most manufacturers.

Readers requiring a more accurate analysis of horsepower may wish to contact the compressor vendors. There are many design variations between vendors that can have significant influence on actual power.

10.13 GAS PROPERTIES

10.13.1 Ideal Gas

As described in Section 1.15, many compression processes can be described using the ideal gas law:

$$PV = mT\,\frac{10.73}{\text{MW}} \tag{10.6}$$

where P = pressure, psia
V = volume, ft^3
m = mass, lb_m
MW = molecular weight, lb_m/mol
T = temperature, °R

This, plus a description of an isentropic process, enables us to describe a compression process theoretically. The equation is

$$PV^N = \text{constant} \tag{10.7}$$

where N is the specific heat ratio.

The discharge temperature may be estimated as the isentropic temperature at the end of the compression process. This is given by

$$T_d = T_s R^{(N-1)/N} \tag{10.8}$$

where T_d = discharge temperature, °R
T_s = suction temperature, °R
R = pressure ratio

10.13.2 Real Gas

No gas is truly ideal. To approximate an ideal gas, we use a concept called *compressibility factor*. This factor is used to represent the difference between a real gas and an ideal gas. Using this factor, we rewrite Eq. (4.1) as

$$PV = ZmT\,\frac{10.73}{\text{MW}} \tag{10.9}$$

where Z is the compressibility factor. This factor is obtained using a suitable equation of state that more accurately describes gas properties.

The isentropic process is still described as

$$PV^N = \text{constant} \tag{10.7}$$

N is now the isentropic volumetric exponent defined by real gas properties.

The discharge temperature is described by

$$T_d = T_s R^{(N_t - 1)/N_t} \tag{10.10}$$

where N_t is the isentropic temperature exponent defined by real gas properties.

10.14 ALTERNATIVE EQUATIONS OF STATE

Over the years, there have been many equations of state proposed. For most gases, equations of state or compressibility methods based on work by Redlich–Kwong, the American Petroleum Institute, Peng–Robinson, Benedict–Webb–Rubin, or Lee–Kessler are recommended. Although detailed discussion of these calculation methods is outside the scope of this book, we should note that every one of these methods falls apart for all gases when operating in the dense phase. This is generally above and to the left of the liquid–vapor dome on the temperature–entropy diagram. When this is the case, we usually resort to National Bureau of Standards *T–S* diagrams for pure gases.

For gas mixtures in the dense phase, it is recommended to use specialized methods such as the National Bureau of Standards DMIX software for defining real properties. An alternative source of real gas properties for unusual mixtures might be a user of these mixtures. Users have often conducted research to define these properties to accurately design processes.

10.15 CONDENSATION

Many fluids that must be compressed contain saturated water and/or hydrocarbon vapors. When this is the case, we must evaluate the quantity of fluid that will condense in the heat exchangers used to remove the heat of compression downstream of the compressors. Evaluating condensed water vapor is relatively easy, and most computerized performance models handle this.

Hydrocarbon condensation can be evaluated by several software packages. Among them are NGPSA (National Gas Processors Association), Process (Simulation Sciences), and Chemshare. Most compressor vendors have one or more of these programs and use them when condensation is suspected.

10.16 FRAME LOADS

Most compressors have limitations on the loading imposed by the compression process. This loading results from the differential pressure across the piston. The loading seen by the frame is dependent on pressures internal to the cylinders. However, at the early stage of compressor sizing, the investigator will only know the pressures at the cylinder flanges. We can get a good idea of what class of machine will be required from calculations based on flange pressures.

For a double-acting compressor, the frame loads are calculated by

$$\text{tensile load} = P_d(A_p - A_r) - P_s A_p + P_a A_r \qquad \text{lb} \tag{10.11}$$

$$\text{compressive load} = P_d A_p - P_s(A_p - A_r) - P_a A_r \qquad \text{lb} \tag{10.12}$$

where P_d = discharge pressure, psia
A_p = piston area, in^2
A_r = piston rod area, in^2
P_s = suction pressure, psia
P_a = atmospheric pressure, psia

Most compressors require the load to reverse so that the load is tensile in one part of the cycle and compressive during the rest of the cycle. Failure to reverse will result in crosshead pin and/or bearing problems. The degree of reversal required for reliable operation depends on details of design. If the ratio of higher load to lower load is 5 : 1 or less, reversal will be adequate to allow proper lubrication of the pin and bearing. Some designs allow higher ratios, but the higher the ratio, the more sensitive the design will be. Caution is recommended in applying this criterion, as crosshead pin reversal is affected by reciprocating inertia as well.

If reversal problems are encountered, special design considerations are in order. As mentioned earlier, single-acting cylinders, tailrods, divided cylinders, or tandem cylinders can be used to overcome reversal problems.

Table 10.1 gives a typical selection of strokes, rod diameters, speeds, brake horsepower per crank, number of cranks, and maximum cylinder bores to use in initial sizing calculations. The frame loads given are less than the maximum allowable, to leave a margin for internal pressures and relief valve considerations. The compressor speeds are given as recommended maximum 60-Hz synchronous speeds for electric motor drives. Also included are horsepower per throw and maximum number of throws available.

10.17 COMPRESSOR DISPLACEMENT AND CLEARANCE

Compressor displacement for a double-acting cylinder is calculated by

$$\text{DISP} = (2A_p - A_r)S\,\frac{\text{rpm}}{1728} \qquad \text{cfm} \tag{10.13}$$

where A_p = piston area, in^2
A_r = piston rod area, in^2
S = stroke, in.
rpm = rotative speed

TABLE 10.1 Typical Frame Sizes and Geometries Available from Major Reciprocating Compressor Manufacturers

Frame Symbol	Frame Load (lb)	Stroke (in.)	Speed (rpm)	Maximum Number of Throws	Bhp per Crank	Rod Diameter (in.)	Maximum Cylinder Bore (in.)
			High-Speed Separable Frames				
A	26,500	5.0	1200	4	480.0	2.00	22.50
B	50,000	6.0	1200	6	1000.0	2.50	26.50
			Electric Drive Frames				
C	10,000	6.0	720	2	75.0	1.50	14.00
D	22,000	12.0	400	4	600.0	2.00	27.50
E	44,000	15.0	360	6	800.0	3.00	42.00
F	72,000	15.0	360	8	1900.0	3.50	42.00
G	90,000	15.0	360	10	2400.0	4.00	42.00
H	145,000	15.0	360	10	3300.0	5.00	42.00
I	170,000	15.0	360	10	4900.0	5.25	42.00
			Integral Engines				
J	80,000	19.0	300	5	1000.0	4.00	17.50
K	105,000	19.0	330	8	1200.0	4.50	17.50

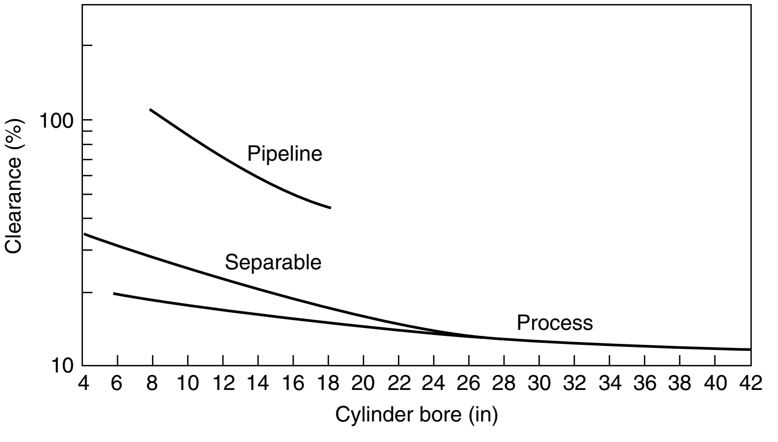

FIGURE 10.12 Type of cylinder vs. clearance. *(Dresser-Rand Company, Painted Post, N.Y.)*

The volumetric cylinder clearance is an important factor in calculating compressor capacity. For initial sizing purposes, it is recommended that the values given in Fig. 10.12 be used to calculate capacity. This figure gives approximate clearance values for three classes of compressors. This allows the user to select from high-speed machines used mainly in gas fields, or medium- and low-speed machines used in process and large enhanced oil recovery projects, and low-speed natural gas transmission pipeline machines.

10.18 STAGING

As was brought out earlier in this book, the first decision to be made is how many stages of compression to use. This depends on many factors, and the evaluation is based on the following considerations:

- Discharge temperature
- Process considerations (sidestreams)
- Overall efficiency
- Frame loads
- Volumetric efficiency

The two factors that have the greatest effect on compressor reliability are discharge temperature and volumetric efficiency. When discharge temperatures exceed 300°F (149°C) on compressors using mineral oil for cylinder lubrication, we find that reliability decreases. This is due primarily to breakdown of lubricating oil and the resulting deposits causing high wear and valve problems. Diester-based synthetic lubricants would provide increased reliability and should be given serious consideration.

We had seen earlier that whenever the volumetric efficiency gets too low, some of the factors that affect capacity and horsepower become quite large compared to the ideal conditions. This and compressor valve reliability on the discharge side causes us to limit the discharge volumetric efficiency at any operating condition to a minimum of 0.1, or 10%. The discharge volumetric efficiency is given by

$$\mathrm{VED} = \frac{\mathrm{VE}}{R^{1/N}} \tag{10.14}$$

In many applications that could be handled with a single-stage compressor, power savings are available by using a two-stage approach with an intercooler between stages. The intercooler removes the heat of compression after the first stage. This increases the gas density, which reduces the bore requirement for the second stage. The result is that lower power is required in the second stage. However, the savings are not as good as one would calculate on an isentropic basis. There are interstage pressure drops and a second set of valve effects to take into account.

Remember that the equations presented earlier relate to cylinder flange pressures. It will be necessary to add pressure drops for initial suction, interstages, and final discharge to account for pulsation vessel, cooler, and piping losses. It is customary to use 1% of line pressure for each element in the system. For example, for initial suction pressure, assume a 1% loss to account for a suction vessel.

For an interstage system close-coupled to the cylinders, use 1% for the first-stage discharge vessel, 1% for the intercooler, and 1% for the second-stage suction vessel, for a total of 3%. If the intercooler is remote mounted, use an additional 1% for the piping, for a total of 4%. For the final discharge, use 1% for the vessel, 1% for the aftercooler (if used), and 1% for piping if the aftercooler is remote-mounted.

10.19 FUNDAMENTALS OF SIZING

The following step-by-step method is recommended for compressor sizing. Although it may appear that sizing is a matter of following cookbook rules, considerable judgment and experience are required to do it well. To perform reciprocating compressor screening and sizing calculations, the following information must be given:

1. Capacity requirement (convert to MMscfd)
2. Initial suction pressure and temperature
3. Cooling medium temperature and desired approach temperature (determines interstage suction temperatures)
4. Final discharge pressure
5. Sidestream capacity; temperature and pressure (if used)
6. Gas analysis
7. Any special application information (relates to particular user requirements)

10.19.1 Number of Stages

Calculate the total pressure ratio R_t. If it exceeds 5, two or more stages are probably required. Assume no more than an average of 3 as the ratio per stage unless the discharge temperature allows. To determine the number of stages, use

$$R_s = (R_t)^{1/N_s} \tag{10.15}$$

where R_s = stage pressure ratio
R_t = total ratio
N_s = number of stages

Use Eq. (10.7) and the suction temperatures given to check for acceptable discharge temperatures. If the discharge temperatures are not acceptable, increase the number of stages until they are. Remember to add the appropriate pressure losses previously mentioned.

10.19.2 Approximate Horsepower

Using the capacity and number of stages, use Eq. (10.5) to calculate the horsepower. This can be done by calculating each stage independently using the appropriate isentropic volumetric exponent. This may change for each stage, depending on pressures, temperatures, and gas analysis. Adding the individual stages together will give the total horsepower.

Using Table 10.1 allows us to determine what frame size is needed based on horsepower per throw and the total number of throws available. If the individual stage horsepowers and/or the total horsepower exceed what is available, we will need to use more than one throw per stage and/or more than one machine.

Another consideration at this point is the number of capacity steps demanded required by process considerations. If more than five are required, we will generally need more than one cylinder per stage. If the process designers want multiple steps, challenge them. Multiple steps will drive the cost of the compressor up and in many cases will increase compressor size while decreasing efficiency and reliability.

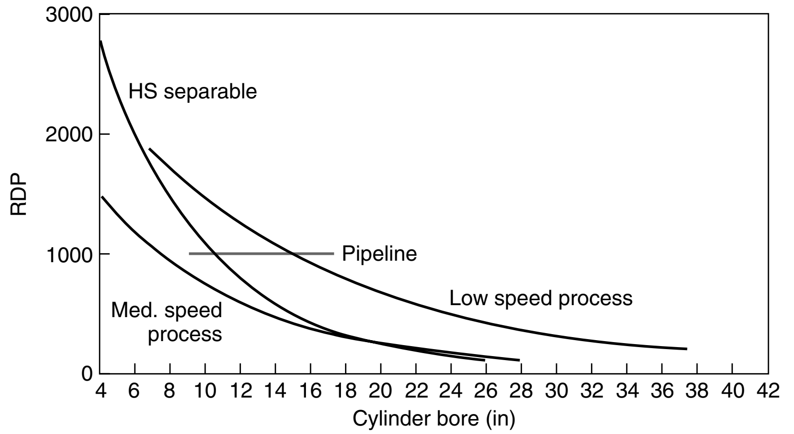

FIGURE 10.13 Size of cylinder vs. rated discharge pressure. (*Dresser-Rand Company, Painted Post, N.Y.)*

10.19.3 Cylinder Bore Requirements

Using Fig. 10.12 and stroke and speed information from Table 10.1, determine the cylinder bores for each stage. This is done by trial and error using Eqs. (10.3) and (10.13) to calculate cylinder bores required for various clearances, as given in Fig. 10.12. This may best be accomplished by guessing a bore above what is required and one below, calculating the capacity, and plotting capacity vs. bore. The bore required can then be determined by the capacity required. Remember that the capacity required must be divided by the total number of cylinders used for each stage. A useful hint is that the compressor vendor can usually add clearance to a given cylinder to meet specific operating conditions. This is usually preferred to selecting nonstandard bore diameters.

Figure 10.13 indicates pressure ratings for various cylinder bores and classes of machine. This information is for cast cylinders only. Higher-pressure forged steel cylinders are available. Check this against the discharge pressure for each stage. If any pressure ratings are exceeded, either increase the number of cylinders per stage or unbalance the pressure ratios on various stages to meet the requirements. The latter move means backing up to our earlier segment dealing with determination of stages, checking temperatures, and proceeding from there as before.

10.19.4 Frame Load

Using the bores calculated and Eq. (10.11), check that the frame loads meet the limits in Table 10.1 and the reversal requirements explained earlier. If frame loads are exceeded, more cylinders will be required on the stages, violating the maximums. The alternative may be to unbalance the pressure ratios if one stage exceeds the limits by a small amount. If the ratios are unbalanced, return to the segment explaining how stage requirements are arrived at, recheck the temperatures, and proceed.

If the frame load reversal criteria are not met, the remedies suggested in Section 10.16 may be tried. Remember that if a single-acting cylinder is used to correct reversals, the horsepower per throw in Table 10.1 is limited to 60% of the indicated value. If a divided cylinder is used, the horsepowers of the two stages must be added together. In any event, the frame load equation, Eq. (10.11), and displacement equation, Eq. (10.13), must be modified to reflect the geometric situation. Again, if the reversal criteria are marginally violated, it may be possible to unbalance ratios to correct it. Return to the segment explaining how stage numbers are determined, check temperatures and proceed.

10.19.5 Vendor Confirmation

If all of the criteria described above are met, you have selected a compressor that will fit the application. The next problem is to get confirmation from the compressor vendors that this selection is valid. Remember that we have outlined approximate methods and approximate information. Many design variations are available that may affect this preliminary sizing. What you have is a reasonable estimate of what you will get.

10.20 SIZING EXAMPLES

We are now ready to look at two examples of compressor sizing. They are intended to demonstrate the principles and methodology presented in the preceding pages. The equations used to make the calculations are either given directly or the equation numbers are given in parentheses after the result. It should be noted that some of the equations given earlier may be rearranged to find the unknown value. This is particularly true of the equation for piston displacement, Eq. (10.13).

It is recommended that the reader follow through these examples in detail to gain a feel for the methods used and the influence on performance introduced by varying the important parameters. We have elected to use only three significant figures in the calculations since this exceeds the possible accuracy of these methods. Obviously, the accuracy of more detailed performance calculation methods, including those of PC-based computer programs, would be considerably greater.

Example 10.1

1. *Given:*

Capacity required = 20 MMscfd
Suction pressure = 75 psia
Suction temperature = 100°F
Air-to-gas coolers
Max. ambient = 110°F
Approach = 30°F
Discharge pressure = 500 psia
Barometer = 14.7 psia
No sidestream
Gas is pure methane

Design requirements:

1% initial suction pressure drop
3% interstage pressure drop
2% final pressure drop

2. Determine the number of stages.

Total ratios = 500/75 = 6.67 [from Eq. (10.15)]
Stage ratio (two stages) = 2.58

Correcting for pressure drops (from Section 10.18):

First suction = 75 × 0.99 = 74.3 psia
First discharge = 75 × 2.58 × 1.03 = 199 psia
Second suction = 199 × 0.97 = 193 psia
Second discharge = 500 × 1.02 = 510 psia
First-stage ratio = 199/74.3 = 2.68
Second-stage ratio = 510/193 = 2.64

Checking discharge temperatures [from Eq. (10.8)]:

First stage = 235°F
Second stage = 282°F

3. Determine the approximate horsepower.

First stage (N_c = 0.85, N_m = 0.95) [from Eq. (10.5)]: bhp = 1190
Second stage (N_c = 0.85, N_m = 0.95): bhp = 1170
Total bhp: 2360

From Table 10.1 we select the higher power, separable frame, and plan on using four throws.

4. Determine the bore sizes.

First stage (two cylinders):
Rated discharge pressure = 199 psia (minimum)
Maximum bore (Fig. 10.13) = 21.5 in.
Clearance (Fig. 10.12) = 13%
Displacement = 3000 cfm/cyl [from Eq. (10.13)]
Capacity (two cylinders) = 32.4 MMscfd [from Eq. (10.3)]
New displacement = 3000 ×20/32.4 = 1850 cfm/cyl
Approximate piston area = 224 in^2 [from Eq. (10.13)]
Bore = 16.9 in.

We will use a 17-in. bore:

Clearance (Fig. 10.12) = 19%
Displacement = 1870 cfm/cyl [from Eq. (10.13)]
Capacity (two cylinders) = 18.4 MMscfd [from Eq. (10.3)]
New displacement = 1870 × 20/18.4 = 2030 cfm/cyl
Approximate piston area = 246 in^2 [from Eq. (10.13)]
Bore = 17.7 in.

We will use a 17.75-in. bore:

Clearance (Fig. 10.12) = 18%
Displacement = 2040 cfm/cyl [from Eq. (10.13)]
Capacity (two cylinders) = 20.4 MMscfd [from Eq. (10.3)]

For rough sizing, this is accurate enough.

Second stage (two cylinders)

Rated discharge pressure = 510 psia (minimum)
Maximum bore (Fig. 10.13) = 15 in.
Clearance (Fig. 10.12) = 20%

Since we have the first stage sized, we can quickly estimate the second stage using the suction density ratio.

Estimated displacement = 2040 × 74.3 × 600/(193 × 560) = 841 cfm/cyl
Approximate piston area = 103 in^2 [from Eq. (10.13)]
Bore = 11.45 in.

We will use a 11.5-in. bore:

Clearance (Fig. 10.12) = 23%
Displacement = 845 cfm/cyl [from Eq. (10.13)]
Capacity (two cylinders) = 19.1 MMscfd [from Eq. (10.3)]
New displacement = 845 × 20/19.1 = 885 cfm
Approximate piston area = 109 in^2 [from Eq. (10.13)]
Bore = 11.8 in.

We will use a 12-in. bore:

Clearance (Fig. 10.12) = 22%
Displacement = 922 cfm/cyl [from Eq. (10.13)]
Capacity (two cylinders) = 21.1 MMscfd [from Eq. (10.3)]

At this point, we could further iterate on cylinder clearance and find that a 12-in. bore with 25% clearance would produce 20.1 MMscfd. However, for our purposes, we have the information we need to complete our job.

5. Check the frame loads and reversals.

First stage:

Tensile load = 30,000 lb [from Eq. (10.11)]
Compressive load = 31,100 lb [from Eq. (10.12)]

Second stage:

Tensile load = 33,400 lb [from Eq. (10.11)]
Compressive load = 36,700 lb [from Eq. (10.12)]

These frame loads are well within the 50,000-lb application limit we set for this frame, and the reversals are excellent. There is the potential for saving money on this application if a lighter frame were available to do this job.

Our final results are:

First stage (two cylinders): 17.75-in. bore × 6-in. stroke
Second stage (two cylinders): 12-in. bore × 6-in. stroke
Driver capability: minimum 2360 hp at 1200 rpm

Example 10.2

1. *Given:*

Capacity = 45.4 MMscfd
Suction pressure = 93.3 psia
Suction temperature = 110°F
Cooling water = 100°F
Approach temperature = 10°F
Final discharge pressure = 1940 psia
Sidestream capacity = 20.4 MMscfd
Sidestream pressure = 208 psia
Sidestream temperature = 110°F

Gas analysis:

Mainstream 83.2% H_2, MW = 6.83, $T_c = -327$°F, $P_c = 262$ psia
Sidestream 82.4% H_2, MW = 7.79, $T_c = -315$°F, $P_c = 266$ psia

Design conditions:

Max. piston speed = 850 ft/min
Initial pressure drop = 1%
Interstage pressure drop = 3%
Final pressure drop = 1%
Max. discharge temp. = 250°F
Atmospheric pressure = 14.4 psia

2. Determine the number of stages. The first stage is determined by the sidestream.

First-stage suction pressure = 0.99 ×93.3 = 92.4 psia
First-stage discharge pressure = 1.03 × 208 = 214 psia
Pressure ratio = 214/92.4 = 2.32

Gas properties:

N_v = 1.33 (API)
N_t = 1.32
Z_s = 1.00
Discharge temperature = 239°F [from Eq. (10.10)]

The rest of the machine is now considered.

Total ratio R_t = 1940/208 = 9.33
Stage ratio (two stages) = 3.05 [from Eq. (10.15)]

Second-stage gas properties:

N_v = 1.32 (API)
N_t = 1.30
Z_s = 1.00
Second-stage discharge temperature = 277°F [from Eq. (10.10)]

The discharge temperature is too high. We will use three stages.

Stage ratio (three stages) = 2.11
Second-stage suction pressure = 208 psia
Second-stage discharge pressure = 1.03 × 208 × 2.11 = 452 psia
Second-stage ratio = 452/208 = 2.17
Second-stage discharge temperature = 222°F [from Eq. (10.10)]
Third-stage suction pressure = 0.97 × 452 = 438 psia
Third-stage discharge pressure = 1.03 × 2.11 × 438 = 952 psia
Third-stage ratio = 952/438 = 2.17

Third-stage gas properties:

N_v = 1.35
N_t = 1.31
Z_s = 1.01
Third-stage discharge temperature = 225°F [from Eq. (10.10)]
Fourth-stage suction pressure = 0.97 × 952 = 923 psia
Fourth-stage discharge pressure = 1.01 × 1940 = 1960 psia
Fourth-stage pressure ratio = 1960/923 = 2.12

Fourth-stage gas properties:

N_v = 1.42
N_t = 1.32

$Z_s = 1.02$
Fourth-stage discharge temperature = 224°F [from Eq. (10.10)]

All stages meet the temperature criteria.

3. Determine the approximate horsepower. Using a compression efficiency of 0.85 and a mechanical efficiency of 0.95, we find, from Eq. (10.5):

First-stage horsepower = 2570
Second-stage horsepower = 3390
Third-stage horsepower = 3450
Fourth-stage horsepower = 3420
Total = 12,800

The piston speed maximum of 850 excludes using 15-in. stroke at 360 rpm. Therefore, we will use 15-in. stroke at 327 rpm (818 ft/min). The 145,000-lb frame is just short on horsepower per throw (2930 at 327 rpm), so we will try the 170,000-lb frame with four throws.

4. Determine the cylinder bore requirements.

First stage:

Minimum rated discharge pressure (RDP) = 214 psia
Maximum bore (Fig. 10.13) = 37 in.
Clearance (Fig. 10.12) = 12%
Displacement = 6040 cfm [from Eq. (10.13)]
Capacity = 42.1 MMscfd (note that $Z_{std} = 1.0$) [from Eq. (10.3)]

This is less than the required capacity. We will need two cylinders and a minimum five-throw frame.

New displacement = 6040 × 45.4/42.1 = 6510 cfm
Piston area (one cylinder) = 584 in^2 [from Eq. (10.13)]
Bore = 27.3 in.

We will use 28 in.

Clearance = 13%
Displacement (two cylinders) = 6870 [from Eq. (10.13)]
Capacity (two cylinders) = 47.3 MMscfd [from Eq. (10.3)]

We will use this bore and add clearance to match capacity.

Second stage:

Minimum RDP = 452 psia
Maximum bore (Fig. 10.13) = 26 in.
Clearance (Fig. 10.12) = 13%

Displacement = 2950 cfm [from Eq. (10.13)]
Capacity = 46.4 MMscfd [from Eq. (10.3)]

This is less than the required capacity. We will need two cylinders and a minimum six-throw frame.

New displacement (two cylinders) = 2950 × 65.8/46.4 = 4180 cfm
Piston area (one cylinder) = 379 in^2 [from Eq. (10.13)]
Bore = 22.0 in.

We will use 22.0 in.

Clearance (Fig. 10.12) = 14%
Displacement (two cylinders) = 4190 cfm [from Eq. (10.13)]
Capacity = 65.2 MMscfd [from Eq. (10.3)]

This is within 1% of the required capacity, and we will use this bore.

Third stage: The approximate displacement is determined by the second-stage displacement times the density ratio.

Approximate displacement = 4190 × 208/438 = 1990 cfm
Piston area = 361 in^2 [from Eq. (10.13)]
Bore = 21.4 in.

A minimum RDP of 952 psia is required. The maximum bore from Fig. 10.13 is 16 in. We will need two cylinders and a minimum seven-throw frame.

Piston area (one cylinder) = 186 in^2 [from Eq. (10.13)]
Bore = 15.4 in.

We will use 15.5 in.

Clearance (Fig. 10.12) = 15%
Displacement (two cylinders) = 2020 cfm [from Eq. (10.13)]
Capacity = 65.2 MMscfd [from Eq. (10.3)]

This is within 1% of the required capacity, and we will use this bore.

Fourth stage: The approximate displacement is determined by the third-stage displacement times the density ratio. We will follow the lead set by the other three stages and use two cylinders requiring an eight-throw frame.

Approximate displacement (two cylinders) = 2020 × 438/952 = 929 cfm
Piston area (one cylinder) = 92.6 in^2 [from Eq. (10.13)]
Bore = 10.9 in.

A minimum RDP of 1960 psia is required. This is above the RDP for a 10.9-in. bore. We will have to use a forged steel cylinder to get the required RDP. We will use an 11-in. bore:

Clearance (Fig. 10.12) = 18%
Displacement (two cylinders) = 956 cfm [from Eq. (10.13)]
Capacity (two cylinders) = 65.7 MMscfd [from Eq. (10.3)]

This is within 1% of the required capacity, and we will use this bore.

5. Check the frame loads.

First stage:
Tensile load = 70,600 lb [from Eq. (10.11)]
Compressive load = 76,600 lb [from Eq. (10.12)]

S*econd stage:*
Tensile load = 83,300 lb [from Eq. (10.11)]
Compressive load = 96,900 lb [from Eq. (10.12)]

Third stage:
Tensile load = 76,700 lb [from Eq. (10.11)]
Compressive load = 106,000 lb [from Eq. (10.12)]

Fourth stage:
Tensile load = 56,400 lb [from Eq. (10.11)]
Compressive load = 118,000 lb [from Eq. (10.12)]

These loads are well below the 170,000-lb maximum we selected. We can use the 145,000-lb frame with adequate margin. The reversals are also within acceptable limits.

6. Make the selection. Our compressor has the following characteristics:

Frame: eight-throw, 15-in. stroke, 145,000-lb frame load
First-stage cylinder: 28-in. bore, two required
Second-stage cylinder: 22-in. bore, two required
Third-stage cylinder: 15.5-in. bore, two required
Fourth-stage cylinder: 11-in. bore, two required

Driver: 327-rpm synchronous motor with a minimum power capability of 12,800 hp. A 13,000-hp motor would be selected to give adequate margin for relief valve setting.

Although this basically manual selection approach at first seems tedious, the responsible engineer will quickly become proficient. Most important, there is no substitute for going the nonautomated, noncomputerized route when it comes to acquiring a thorough knowledge of the numerous interrelating factors that lead to intelligent equipment selection.