7

APPLICATION TO PRIME MOVERS

7.1 INTRODUCTION

Prime movers (pumps and compressors) are the workhorses of the process industry. They can be split into two generic categories, depending on how they impart energy to the process fluid. These categories are as follows:

- *Kinetic or Dynamic Systems*: Energy is imparted by accelerating the fluid with an impeller and then decelerating the fluid with an inherent rise in pressure. Examples of this type of equipment are centrifugal pumps and compressors, axial and centrifugal fans, and blowers. Bernoulli's theory can be used to understand how these prime movers operate.
- *Displacement Systems*: Energy is imparted by displacing the fluid from a lower pressure to a higher pressure environment with a pushing or rotating action. Examples of this type of equipment are reciprocating pumps and compressors and rotary blowers and compressors.

In order to formulate theoretically correct working hypotheses for problems with pumps and compressors, knowledge of several relationships and definitions are required. These are summarized in this chapter. In addition, several problems are presented to illustrate the use of these concepts.

Problem Solving for Process Operators and Specialists, First Edition. Joseph M. Bonem.

7.2 KINETIC SYSTEMS

As indicated above, Bernoulli's theory is the key to understanding these systems. The application of Bernoulli's theory to the flow of fluids in pipes was discussed in Chapter 5. In this chapter, it was only discussed in terms of the flow of fluids in piping. However, it is also valuable when considering the flow of fluids inside a pump or compressor case. Bernoulli's equation and the definition of terms is repeated as follows:

$$\Delta P/\rho + \Delta(v^2)/2g_c + \Delta z = -w - lw \tag{7-1}$$

where

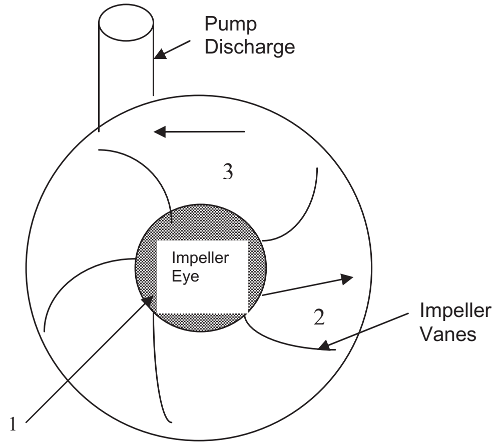
ΔP = pressure difference between two points
ρ = fluid density
$\Delta(v^2)$ = difference in velocity squared between two points
g_c = gravitational constant
Δz = difference in elevation between two points
w = amount of work added by the prime mover
lw = frictional loss in the piping system

When one is considering what happens inside the case of a centrifugal pump or compressor, only the first two terms of equation (7-1) are significant. They clearly indicate what happens to the fluid after it is accelerated by the work of the prime mover to approach the impeller tip velocity. As the fluid enters the outlet section of the pump case, it decelerates and the pressure increases to the discharge pressure. Figure 7-1 shows the typical flow path in a centrifugal prime mover.

Bernoulli's theory as applied to centrifugal prime movers has several implications that must be considered: The head ($\Delta P/\rho$) that a pump or compressor develops is directly related to impeller tip speed squared. High-speed impellers are required to develop high heads for either pumps or compressors. An alternative is multiple-staged impellers. The high-speed impellers and multiple-staged impellers can also be combined in a single pump or compressor.

The operating characteristic of a centrifugal system (pump or compressor) is defined by the equipment's characteristic "head curve." This head curve is developed by the equipment supplier and is provided as part of the equipment purchase. An example is shown in Figure 7-2. In this figure, head and horsepower are shown as a function of flow rate. The stability limit is also shown. Several generalized points can be made about this figure.

Operation to the left of the stability limit will result in flow instabilities, as flow surges forward and then backward through the prime mover. The stability is usually well defined for compressors and blowers. However, for pumps, it is usually 25–40% of the BEP (best efficiency point). If flow reversal



1. Fluid enters the eye of the impeller by pressure flow from the pump suction flange.

2. It is accelerated by centrifugal force to a velocity approaching the tip velocity of the impeller.

3. As the fluid approaches the discharge, the internals of the pump casing allow it to slow down with the inherent increase in pressure.

Figure 7-1 Flow path in centrifugal equipment.

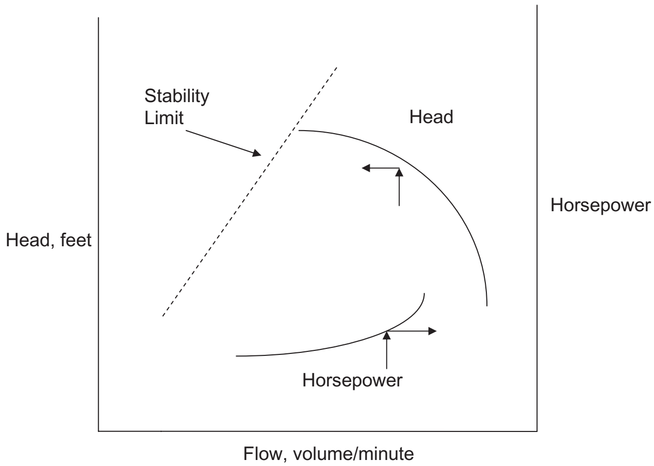


Figure 7-2 Characteristic centrifugal pump or compressor curve.

occurs, it can damage the equipment. Operation of equipment in parallel can sometimes result in operation within the surging region. This happens because the parallel pieces of equipment have different head curves, or, often, there are slight differences in the mechanical characteristics of "identical" pieces of equipment.

The BEP is the point on the "head curve" at which the hydraulic efficiency is at a maximum. Note that this efficiency does not include motor efficiency. While the head curve is usually developed using water or air, it is valid for any fluid if the correct units are utilized for flow and head. These units are defined later.

As shown in Figure 7-2, horsepower requirements normally peak at the "end of the curve" (maximum flow rate). The driver for the prime mover may or may not be provided with end of the curve protection. If the driver is not designed with end of the curve protection in mind, operating at this point will normally cause the driver to be overloaded. If the driver is a steam turbine, it will slow down, causing the pump to appear to be operating "off the pump curve." If the driver is an electric motor, it will shut down; the electrical load causes the motor protection device to react and shut down the motor before the motor fails due to overload.

7.3 PUMP CALCULATIONS

Before considering exact calculation procedures for pumps, a few definitions are required. These definitions are applicable to positive displacement pumps as well as centrifugal pumps.

- *Cavitation*: A condition that occurs if $NPSH_R > NPSH_A$. If this situation occurs, some of the liquid being pumped will vaporize between the pump suction flange and the pump impeller. This will cause the pump to operate off the head curve, and damage may occur to the impeller.
- *Off the head curve*: This is a condition at which the actual operating point, as defined by the flow rate in gallons/min (gpm), and pressure rise, as described by feet of fluid flowing, is below the pump curve, as defined , for example, in Figure 7-2.
- *$NPSH_R$*: Net positive head required. This is the head, in feet, required to overcome the pressure loss between the pump suction flange and pump impeller eye. The pump supplier will specify this. A typical $NPSH_R$ versus flow rate curve is shown in Figure 7-3. Note that this curve is usually developed with water, but is valid for any fluid.
- *$NPSH_A$*: Net positive head available. This is the difference, in feet of head, between the actual pressure at the pump suction flange and the vapor pressure of the liquid being pumped. If the liquid has been stored under a nitrogen, air, or inert gas blanket, some question may arise regarding the actual vapor pressure of the liquid. The most conservative approach is to assume that the vapor pressure is equivalent to the pressure in the storage vessel. If this assumption is used, then the $NPSH_A$ will be the elevation of the drum above the pump impeller less the frictional pressure loss (expressed in feet), assuming there is no change in the suction piping diameter.

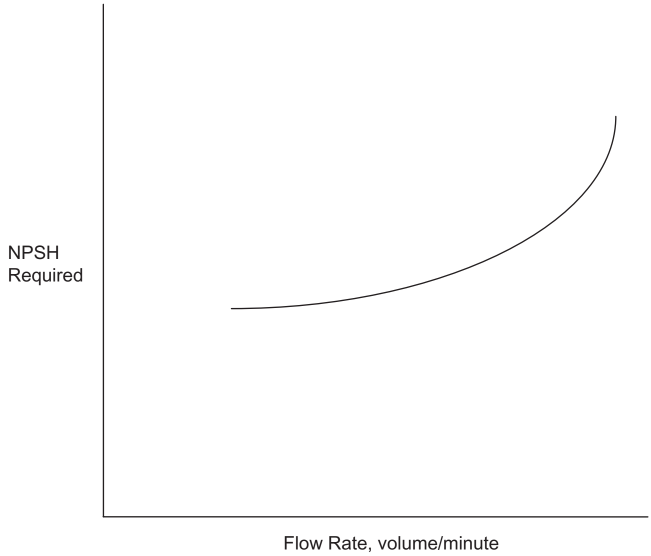


Figure 7-3 Typical NPSH required for centrifugal pump.

The equations below can be used in combination with a pump curve supplied by the manufacturer, similar to Figure 7-2, to determine essential pump operating characteristics. When using the pump curve for a centrifugal pump, the differential pressure across the pump can be calculated as follows:

$$\Delta P = H \times \rho/144 \quad (7\text{-}2)$$

where

ΔP = pressure rise across the pump, psi
H = pump head at the given flow rate, ft
ρ = pumped fluid density, lb/ft^3

As indicated earlier, pump curves are usually developed using water as the operative fluid. They are valid for any fluid as long as the density used for calculations is that of the fluid being pumped. Thus in equation (7-2), the pressure developed by a centrifugal pump can be calculated from the head curve supplied by the pump manufacturer, using the head developed at the desired flow rate and the density of the flowing fluid.

The hydraulic horsepower of a pump can be estimated as follows:

$$\text{BHP} = H \times F \times 100/(E \times 33{,}000) \quad (7\text{-}3)$$

where

BHP = energy delivered to the fluid, horsepower
F = flow rate through the pump, lb/min
E = hydraulic pump efficiency, %

It should be noted that this calculation only includes the hydraulic efficiency. Other efficiencies (mechanical and/or electrical) must also be considered.

On rare occasions, it will be necessary to determine the temperature change across the pump. For example, pumps with very low flow rate, high recirculation rate from pump discharge to pump suction, and high heads may cause significant heat to be generated and conducted to the fluid recycling to the pump suction line. This higher temperature at the pump suction flange may reduce the $NPSH_A$. The reduction in $NPSH_A$ is due to the increase in vapor pressure as the temperature is increased. The fluid temperature increase across the pump can be estimated as follows:

$$\Delta T = BHP \times (100 - E) \times 2545/(100 \times 60 \times F \times C_P) \tag{7-4}$$

where

C_P = fluid specific heat, BTU/lb-°F

This calculation will often indicate that a cooler is required in the fluid recycle line. Even in high head pumps with low flow and no recirculation, heat generation can be a problem. The heat generated by the pump inefficiency will heat up the fluid, which, in turn, will reject some of the heat to the pump case. As the pump case is heated, the $NPSH_A$ will decrease as the vapor pressure at the pump impeller eye increases.

7.4 CENTRIFUGAL COMPRESSOR CALCULATIONS

As indicated earlier, a head curve similar to that shown in Figure 7-2 is valid for centrifugal compressors as well as pumps. The compressor flow is normally expressed as actual cubic feet/minute (ACFM) at the compressor inlet conditions. The ACFM can be calculated by dividing the gas flow rate in lb/min by the gas density in lb/ft^3. Equation (5-3) in Chapter 5 provides the correct approach for estimating the density of a gas. In addition, the calculation of head is much more complicated for a compressor than for a pump. This is because gases are compressible as opposed to noncompressible liquids. The specific volume and density of a liquid, for all practical purposes, can be considered constant. This is not true for gases. As the pressure on a gas is increased, the gas density increases and the specific volume decreases.

While there are multiple references to compression calculations in the literature of this field, two of the references used in the preparation of this book were written by Lyman F. Scheel, *Gas Machinery* and *New Ideas on Centrifugal Compressors, Part 1*. The calculation of head for a compressor depends on the relationship between the pressure and volume of the gas. This relationship is given by the thermodynamic equation:

$$P \times V^k = \text{constant} \tag{7-5}$$

where

P = pressure at any point in the compression

V = volume occupied by the gas being compressed at the pressure P

Equation (7-5) determines the exact relationship between pressure and volume. When applying this relationship to compression (centrifugal or reciprocating), the actual value of k depends on which of three thermodynamic assumptions is used. These assumptions are as follows:

1. *Isothermal*: This assumption surmises that the compression occurs at constant temperature. The constant temperature assumption is impossible to achieve in practice.
2. *Adiabatic*: This assumption requires that no heat be added to or removed from the system. This path can be approached in a piston compressor where the discharge and suction compressor valves have exceptionally large areas. The stipulation of exceptionally large area valves is necessary, since the suction and discharge pressures are measured before and after the compressor valves. These compressor valves are flat ribbon valves that are integral to the compressor. However, if the pressures could be measured inside the valves, no stipulation of exceptionally large area suction and discharge valves would be required. In this idealized compression, $k = C_p/C_v$. C_p/C_v is the ratio of the specific heats at constant pressure and constant volume. For an ideal gas, k equals 1.4. If compression is both adiabatic and reversible, it is referred to as isentropic compression. For a compression to be reversible, there must be no friction loss occurring. This type of compression occurs at constant entropy. For this special compression, a Mollier diagram determines the compression path. This description of isentropic compression is included for reference only since, like isothermal compression, it rarely occurs in practice.
3. *Polytropic*: This is an empirical assumption for evaluating the compression path in a compressor. However, it is the normal technique used to evaluate compression head. For polytropic compression, the k value is replaced by n, which is obtained either from the compressor manufacturer or from plant test data. This will be discussed later. The polytropic assumption of compression path is the assumption most frequently used for solving plant operating problems.

The polytropic and adiabatic head can be determined from the same basic equation, shown below:

$$H = 1545 \times T_S \times Z \times (R^\sigma - 1)/M\sigma \quad (7\text{-}6)$$

where

H = polytropic or adiabatic head, ft
T_S = suction temperature, °R (it should be noted that this suction temperature must be in absolute temperature units of degrees Rankin. This can be determined by adding 460 to the value in degrees Fahrenheit)
Z = average (suction and discharge) compressibility
R = compression ratio (this term is the discharge pressure divided by the suction pressure, in absolute units such as psia)
M = gas molecular weight
σ = polytropic or adiabatic compression exponent

There are three different ways to determine the value of "σ." They are as follows:

1. The compressor manufacturer may supply the value based on test data.
2. The compressor vendor may supply either the polytropic or adiabatic compression efficiency. In this case, the compression exponent (σ) can be calculated as follows:

$$\sigma = (k - 1) \times 100/(k \times E) \quad (7\text{-}7)$$

where

E = either the adiabatic or polytropic compression efficiency, %
k = ratio of specific heats, C_p/C_v. This value is 1.4 for an ideal gas

3. The polytropic compression exponent can be calculated from plant data using the following relationship:

$$T_D = T_S \times R^\sigma \quad (7\text{-}8)$$

where

T_D = absolute discharge temperature
T_S = absolute suction temperature

It should be recognized that the relationship shown in equation (7-8) gives the compression exponent at any point in time. If the compressor needs mechanical repairs, this value may be higher than predicted by the compressor manufacturer or higher than it was in previously collected plant data. Since the compression exponent calculated from plant data does vary with the mechanical condition of the compressor, this value could be used as a daily monitoring tool for critical compressors. For example, an increase in the compression exponent might indicate that the clearances in a centrifugal compressor had increased to the point at which excessive amounts of gas are recirculating to the compressor suction. An increase in the compression exponent for a reciprocating compressor might indicate that the suction and/or discharge valves should be replaced.

Once the head and flow rate are known, the fluid horsepower requirements for the compressor can be easily calculated, using equation (7-9) below:

$$\text{BHP} = F \times H \times 100/(33{,}000 \times E) \tag{7-9}$$

where

F = flow rate, lb/min

H = polytropic or adiabatic head, ft

E = either the adiabatic or polytropic compression efficiency, %

A cautionary word is of value at this point. The method used to determine compression efficiency (adiabatic or polytropic) should be consistent with the method used to determine the compression exponent. This will normally be the polytropic efficiency and the polytropic compression exponent. It should also be noted that equation (7-9) includes no mechanical or electrical efficiency that will be associated with belts, gears, or motors.

7.5 DISPLACEMENT SYSTEMS

The term "displacement systems" refers to the prime movers which displace a fixed amount of fluid (liquid or gas), essentially independent of the differential pressure, across the pump or compressor. Typical equipment items that belong within this category are reciprocating pumps/compressors and rotary pumps/compressors. While it might be argued that this type of equipment is no longer used in modern plants, an examination of different processes will show that this type of equipment does indeed have a place in a modern process plant. Examples of the use of this type of equipment are as follows:

- Reciprocating compressors for which the head requirements are high and the volume rate is low to moderate. An example of this is the high-pressure compression step in autoclave and tubular low density polyethylene plants.

- Proportioning pumps with high head requirements or where there is a need to control flow by pump adjustments as opposed to using a flow controller. These might include additive or catalyst pumps.
- Pumps or compressors which must transfer a fixed rate of material regardless of the discharge pressure. This would not be possible with a typical centrifugal system, since the flow rate would decrease as the discharge pressure increased.

There are important concepts which must be understood in order to comprehend this class of prime movers. It should be realized that, with this type of equipment, energy is imparted to the fluid by displacement of a fixed volume of that fluid. The mass flow rate will depend on the fluid suction conditions and physical dimensions of the equipment. That is, larger pump or compressor cylinders will allow displacement of a larger volume of fluid, but the mass of fluid moved also depends on the density at suction conditions.

The calculations described in the previous section are also applicable to this type of system. However, head will not be dependent on flow rate, there will be no head versus capacity curve, and the pump or compressor head will depend on the suction and discharge pressures only. Figure 7-4 shows a typical flow pattern for a reciprocating pump/compressor.

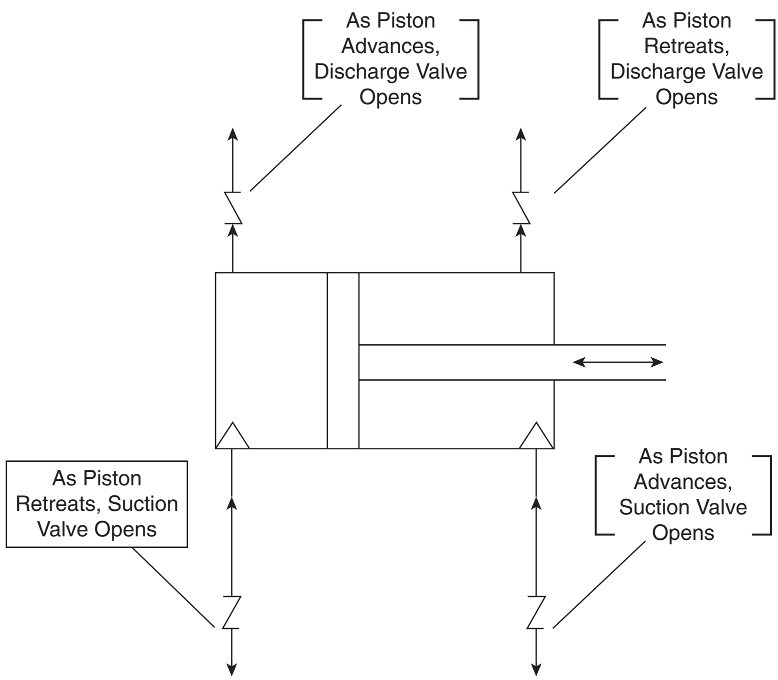


Figure 7-4 Reciprocating flow path in double acting equipment.

Some definitions will aid your ability to evaluate and understand this type of equipment. They are as follows:

- *Volumetric Efficiency*: The actual volume of fluid displaced relative to the volume of the cylinder of a reciprocating pump/compressor or rotating pockets of a rotary pump/compressor. For a liquid, this efficiency approaches 100%. However, for a gas, it is approximately 70%. The differences are due to the compressibility of gases. As the pressure in the cylinder or rotating pocket decreases from discharge pressure to suction pressure, the residual gas in the cylinder/pocket expands to partially fill the cylinder/pocket and reduce the volumetric efficiency of the compressor.
- *Leakage*: This is an additional loss in volumetric efficiency caused by leakage through clearances. An example of this leakage in a reciprocating pump or compressor is the flow of material between the piston and cylinder wall or across the suction and discharge check valves.
- *Horsepower Load Point*: This is a unique feature of a positive displacement compressor. It is the point on a plot of horsepower versus suction pressure at which the required fluid horsepower is at a maximum. On one side of this point, increasing the suction pressure increases the mass flow, which overrides the decrease in compression ratio and causes an increase in the required horsepower. On the other side of this point, the increasing suction pressure results in a decreased compression ratio, which more than compensates for the increase in mass flow rate and results in a decreased horsepower. This concept is discussed in more detail later.
- *Clearance*: This is the part of the cylinder in a reciprocating pump/compressor that is not displaced completely by the piston. Figure 7-5

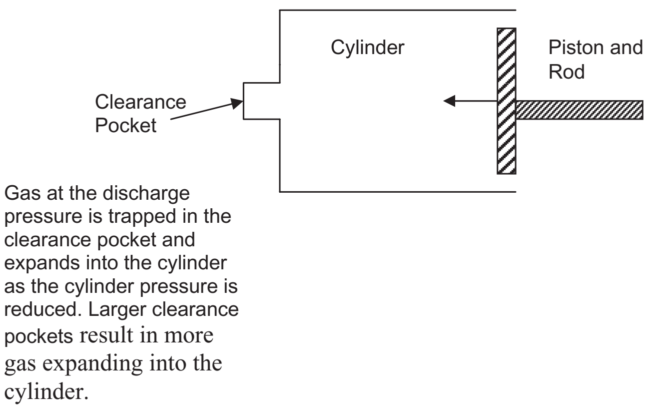


Figure 7-5 Clearance pocket reciprocating compressor.

shows a typical sketch of a "clearance pocket." Because of the compressibility of gases, this becomes more important for gases than for liquids. These are often used to reduce the volumetric efficiency of a compressor and, hence, the required horsepower.

7.6 DISPLACEMENT PUMP CALCULATIONS

Many of the calculations for displacement pumps are identical to those for centrifugal pumps. For example, equations (7-2) and (7-3) can be used to calculate the head/pressure rise relationship and BHP. The main differences in calculations for the two types of pumps are calculation of the fluid flow rate and $NPSH_A$.

For a single-acting reciprocating pump, the following relationship can be used to estimate the volumetric flow rate.

$$V = \Pi \times D^2 \times L_S \times S \times E_V/(4 \times 100) \tag{7-10}$$

where

V = pump capacity, ft^3/min
D = diameter of the pump cylinder, ft
L_S = length of the pump stroke, ft
S = pump speed, RPM
E_V = pump volumetric efficiency, %

There are several points to consider when examining equation (7-10). The pump volumetric capacity is independent of pressure rise and/or pump head, except as the discharge pressure decreases the volumetric efficiency. This is different from the relationship between flow and head for a centrifugal pump.

Equation (7-10) is for a single acting pump, that is, one which displaces the cylinder volume one time for each stroke. The pump that is shown in Figure 7-4 is a double acting pump. It displaces the cylinder two times for each complete stroke. The capacity of a double acting pump will be somewhat less than two times that of a single acting pump of the same dimensions. The actual volume displaced on the return stroke of the piston will be slightly less, due to the volume occupied by the piston and piston rod. Equation (7-10) can be modified for more complicated pumps.

The pump design (reciprocating, plunger, or rotary), the mechanical condition of the check valves, and the internal pump clearances primarily determine the volumetric efficiency. In addition, for high differential pressure pumps, liquid compressibility must also be considered. If the liquid is compressible, the liquid trapped in the clearances will expand when the pressure decreases from discharge pressure to suction pressure. This will cause the volumetric efficiency to decrease. There will also be more leakage across the check valves with high pressure pumps.

The determination of the amount of $NPSH_A$ is more complicated for reciprocating type pumps than it is for centrifugal or rotary pumps. For any type of pump which is pumping a fluid at its boiling point, the $NPSH_A$ is simply the difference in elevation head between the liquid level in the suction drum and the pump suction less the frictional pressure drop. This concept is valid for reciprocating pumps as well, except the frictional pressure drop must be determined at the actual flow rate rather than the average flow rate. The actual flow rate will be greater than the average flow rate due to the cyclic action of the reciprocating pump. In addition, since the fluid in the pump suction line must at some point in the cycle be accelerated from zero velocity to the maximum velocity, there is a pressure loss due to this energy requirement. Fortunately, these two pressure losses occur at different times, so their values are not directly additive. For example, the maximum acceleration head occurs at zero flow rate. Since the flow rate is zero, this is the point of minimum frictional loss. Conversely, the maximum frictional head loss occurs at the maximum rate when there is no acceleration required. For most low-viscosity fluids, the acceleration head dominates. It can be estimated as follows:

1. Calculate the maximum flow rate in the suction pipe.

$$V_P = K \times D^2 \times L_S \times S/(60 \times D_P^2) \tag{7-11}$$

   where

   V_P = peak velocity in suction pipe, ft/sec
   D = diameter of the pump cylinder, ft
   L_S = length of the pump stroke, ft
   S = pump speed, RPM
   D_P = diameter of the suction pipe, ft
   K = a factor that depends on the pump design

   The K value for a double acting pump is approximately Π/2 or about 1.57. K values for other style of pumps can be obtained from the pump vendors.

2. Calculate the frictional pressure drop, $(h_F)_{MAX}$, in the suction line using conventional techniques, as described in Chapter 5, based on the peak velocity, calculated using equation (7-11).

3. Calculate the acceleration head required to accelerate the liquid in the suction line from zero velocity to peak velocity. This can be determined as follows:

$$(h_A)_{MAX} = 1.35 \times L_P \times S \times V_P/307 \tag{7-12}$$

where

$(h_A)_{MAX}$ = maximum acceleration head loss, ft
L_P = actual suction pipe length, ft
S = pump speed, RPM
V_P = peak velocity in suction pipe from equation (7-11), ft/sec

4. The most conservative approach is to determine the suction piping loss by combining the two types of head loss as follows:

$$(h_T)^2_{MAX} = (h_F)^2_{MAX} + (h_A)^2_{MAX} \tag{7-13}$$

where

$(h_T)_{MAX}$ = maximum suction piping head loss to be used to determine the $NPSH_A$

However, as indicated earlier, these two head losses almost always occur at separate times. Thus the use of equation (7-13) may predict overly conservative head losses. The most reasonable estimate of maximum head loss in the suction piping is the larger of the two losses, that is, either the friction head loss or the acceleration head loss.

While equation (7-12) appears to be empirical, it can be easily derived from primary principles of engineering by realizing that the entire mass in the suction line must be accelerated to the maximum suction line velocity as the suction stroke of the pump begins. The value of 1.35 is an experience-based, empirical factor. It is added to allow for the nonsinusoidal motion of the piston and the increased acceleration loss due to elbows and pipe fittings.

Additional information about estimating the $NPSH_A$ for reciprocating pumps can be found in a magazine article entitled *Reciprocating Pumps* by Terry L. Henshaw.

Another aspect that must be considered when dealing with displacement pumps is their "on and off" nature. Rather than the continuous flow that a centrifugal pump provides, the flow will vary from zero to a maximum rate. In some specialized applications, it may be important to assure that there is always some minimal amount of flow from a displacement type of pump. An example of this might be the need to maintain a continuous flow of catalyst to a polymerization reactor. Stopping the flow of catalyst to the reactor, even for a few seconds, might result in plugging of the catalyst injection nozzle. This continuous flow can be obtained in a reciprocating pump system by one of two methods. A more complicated duplex or triplex pump can be utilized. These pumps allow for almost continuous flow of the material being pumped. Another alternative is to install a flow surge bottle in the discharge piping of a simplex reciprocating pump. This vessel, which is equipped with an internal bladder and pressured with nitrogen, can be designed so that the discharge flow to the process approaches an average flow, rather than a peak flow followed by a period of zero flow.

7.7 CALCULATIONS FOR POSITIVE DISPLACEMENT COMPRESSORS

As indicated earlier, the head calculations for both types of compressors are identical. Equations (7-6) through (7-9) can be used to evaluate both head and horsepower requirements for a positive displacement compressor. The actual suction and discharge pressure for a reciprocating compressor may be slightly different than that measured, since the actual pressure is measured in the suction and discharge piping rather than directly at the compressor cylinder during the suction and discharge strokes. In between the compressor cylinder and where the suction and discharge pressures are measured are the ribbon valves. The ribbon valves are pipe-diameter-sized, flat devices which contain multiple strips of metal. These serve as check valves to isolate the compressor cylinder from the suction and discharge pressures at appropriate points in the compressor cycle. For example, when the piston advances, causing a buildup of pressure in the cylinder, the suction ribbon valve closes, keeping gas from flowing back into the suction piping. If the compressor suction and discharge valves are in good mechanical condition, this pressure difference will not be significant. However, if the valves are partially restricted, the pressure at the compressor will be different than that measured. Leakage can also occur in these valves. As indicated earlier, equation (7-8) can be used to monitor the status of these valves for critical compressors.

The flow rate from a positive displacement compressor depends on both the volume of the cylinder or rotating pocket and the volumetric efficiency. As indicated earlier, the volumetric efficiency of a compressor is much lower than that of a pump. This is because gases are more compressible than liquids. The volumetric efficiency of a positive displacement compressor can be estimated as follows:

$$E_V = 100 \times (1 - L - C \times (R^{1/k} - 1)) \tag{7-14}$$

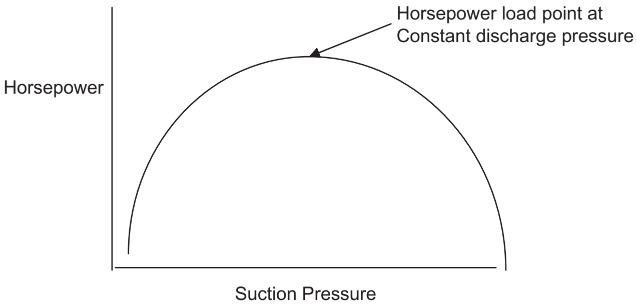


Figure 7-6 Positive displacement compressor load point.

where

E_V = volumetric efficiency, %

C = residual gas remaining in the reciprocating compressor clearance pocket or rotating pocket after discharge, fraction of displacement volume

L = leakage of gas around the piston or rotating element, fraction of displacement volume (roughly 2% of the compression ratio, expressed as a fraction or 0.02)

R = compression ratio

Horsepower load point is a unique feature of a positive displacement compressor. As shown in Figure 7-6, it is the suction pressure at which the BHP reaches a maximum.

To the right of the horsepower load point, the compression ratio and head decrease, but the mass flow increases. To the left of the load point, the compression ratio and head increase while the mass flow decreases. Thus, for a positive displacement compressor, driver overload can occur when the suction pressure is either rising or falling. A single-stage reciprocating compressor, with a constant discharge pressure, constant volumetric efficiency, and an approximate clearance pocket volume of 10%, will reach a maximum horsepower loading at an approximate compression ratio of:

$$R = 1.1 \times (n + 1) \tag{7-15}$$

where

R = compression ratio

n = constant used to define the polytropic compression exponent $\sigma = (n - 1)/n$

Multiple-stage compressors are more complicated, but the concept that decreasing the compression ratio does not always decrease the horsepower requirements is still valid.

7.8 PROBLEM-SOLVING CONSIDERATIONS FOR BOTH SYSTEMS

Compressors

For critical compressors, the suction and discharge temperatures should be used to monitor compressor performance. As indicated previously, equation (7-8) can be used on a daily basis to monitor the performance of a critical centrifugal or positive displacement compressor. If the compression exponent (α) begins to increase, it is an indication that the compressor is becoming less efficient. Another approach that uses the same input is to calculate the "leakage" from the discharge pressure to the suction pressure. An increase in this leakage occurs due to mechanical wear caused by increased clearances in the compressor. This leakage can be calculated as follows:

$$L = [(T_D/(P_D/P_S)^{\sigma}] - T_S)/(T_D + \Delta T_H - T_S) \qquad (7\text{-}16)$$

where

L = discharge to suction leakage, fraction of volumetric flow
T_D = discharge temperature, °R
T_S = suction temperature, °R
ΔT_H = Joule Thompson cooling effect, °R
P_D = discharge pressure, psia
P_S = suction pressure, psia
σ = compression exponent

The Joule Thompson cooling effect is the amount of temperature change that would be expected as the pressure is lowered from discharge pressure to suction pressure. It is available in various handbooks.

A similar approach can be utilized in monitoring the steam turbines that are often used as drivers on large pumps and compressors. The efficiency of a steam turbine can be determined by comparing the actual change in enthalpy with that predicted assuming an isentropic (constant entropy) expansion. That is:

$$E_T = 100 \times (H_I - H_O)/(H_I - H_E) \qquad (7\text{-}17)$$

where

E_T = turbine efficiency, %
H_I = inlet steam enthalpy, BTU/lb
H_O = outlet steam enthalpy, BTU/lb
H_E = outlet steam enthalpy with isentropic expansion, BTU/lb

The values of enthalpy can be determined from steam charts. These steam charts are available in various publications. While there is a sample problem given later in this chapter, the basic approach for utilization of equation (7-17) is as follows:

1. Knowing the inlet steam temperature and pressure, look up the inlet enthalpy (H_I) and inlet entropy (S_I).
2. Knowing the outlet steam temperature and pressure, look up the outlet enthalpy (H_O).
3. Knowing the inlet entropy and the fact that isentropic expansion (no frictional loss) occurs at constant entropy, look up the constant entropy outlet enthalpy (H_E) based on the outlet steam pressure.

The constant entropy outlet temperature will be different from the actual outlet temperature. It is also possible that this constant entropy expansion may result in an outlet condition that yields a mixture of vapor and liquid. Recognize that this calculated outlet condition is only used so that the turbine efficiency can be calculated. It has no practical use beyond the determination of steam turbine efficiency.

For critical turbines, efficiency can be calculated and monitored on a daily basis. This will allow problem solvers to spot mechanical problems before they become so severe that an immediate repair is required. If problems can be uncovered at an early stage, it is likely that the repair of a steam turbine or compressor can be coordinated with another plant repair downtime.

For centrifugal compressors, the compressor performance curve supplied by the compressor manufacturer can be used to analyze problems. To do this accurately will require careful planning and consideration. The following items should be included in any planned problem-solving activity associated with compressors:

- Make sure that all field instruments have been calibrated before taking any data.
- Determine the kinetic head as described in equation (7-6).
- Calculate the ACFM as accurately as possible. If necessary, adjust the metered flow rate for differences in pressure, temperature, and molecular weight between the meter specification sheet and actual conditions.
- Make sure that the gas composition is known, since it can affect variables such as molecular weight, calculated head, temperature difference between the suction and discharge, and flow rate in lb/hr.

If problem solving involves a plant test on a centrifugal compressor, beware of increasing the speed at constant volumetric flow. Since the surge point normally increases with speed, the compressor could go into surge if the volumetric flow rate is maintained constant when speed is increased.

Similar guidelines are appropriate for positive displacement compressors, except that there will not be a head versus flow curve as there is for a centrifugal compressor. It will be important to have instruments calibrated and to know the gas composition so that the actual capacity can be compared to the predicted capacity. The calculated compression exponent can also be compared to those supplied by the compressor manufacturer.

As indicated earlier, restrictions between the pressure measurement point (discharge or suction) can cause the actual pressures in the compressor cycle to be different from the measured pressures. This could cause the compression exponent calculated from inlet and outlet temperatures to be higher than anticipated. In a reciprocating compressor, the most likely cause of this difference in pressures is ribbon valves that are partially plugged. These valves can also cause a loss of compression capacity if they are partially plugged or if they are leaking. This problem can be monitored and detected using equation (7-8). Chronic valve malfunction is often due to liquid or solid entrainment.

Pumps

One of the most common problems causing poor pump performance is an inadequate available NPSH. While there may be pump manufacturers who claim that their pumps have minimal required NPSH, it should be recognized that all pumps have significant NPSH requirements. The problem solver should look with suspicion on any claims that the NPSH requirements are less than those shown in Table 7-1.

In resolving a centrifugal pump problem, the centrifugal pump performance curve supplied by the pump manufacturer should be used to analyze the problem. If the pump is operating as predicted by the performance curve, then the problem is related to high flow rates or piping limitations. If the pump is not operating on the performance curve, then one of the following five problems may be occurring:

1. The pump clearances may have worn so that a large amount of liquid is recirculating from the discharge to the suction. This will cause the pump to actually be pumping more fluid than the amount shown by flow meters, which will result in a discharge pressure which is lower than anticipated.

Table 7-1 Minimum NPSH requirements

Pump Type	NPSH Required, ft
Centrifugal	6
Centrifugal with booster	1 to 2
Positive displacement	4

2. The $NPSH_A$ is not sufficient. If there is not sufficient NPSH available, some of the liquid will vaporize in the pump inlet, causing the actual amount of liquid being pumped to decrease.
3. A vortex is being formed in the suction side storage tank due to the high velocity in the suction line. The vortex causes vapor to be sucked into the pump suction line. This will cause the pump performance to be below the predicted value, even if the NPSH is sufficient. This problem can be remedied with the installation of a vortex breaker or larger suction line.
4. The pump impeller is the wrong size. Essentially, all centrifugal pumps can be fitted with impellers of different sizes. Often, during routine maintenance, a new impeller of the wrong size is installed in the pump. It should also be noted that most pump curves show multiple impeller sizes. Therefore, before the problem solver can accurately consider a pump problem, he needs to know the size of the impeller that is installed in the pump under consideration.
5. The pump is being driven by a steam turbine and it is not running at the design speed. While this is obviously a steam turbine or steam supply problem, it is often presented to the problem solver as a pump problem.

As indicated for compressors, all instruments should be calibrated and adequately compensated for nondesign conditions before the problem solver begins to collect any data. The problem solver should also be concerned about pumps operating too far from their design point. If a pump is operating at very low flow rates, the resulting instability may cause damage to the pump internals. In addition, high flow can result in excessive $NPSH_R$ or excessive horsepower requirements.

Positive displacement pump problems often are related to either inadequate available NPSH or leaking check valves. The calculation of available NPSH for a reciprocating pump was discussed earlier. Leaking check valves in reciprocating pumps result in a similar loss in capacity as that described for reciprocating compressors.

EXAMPLE PROBLEM 7-1

A centrifugal compressor (C-100L) handling a gas stream that is mostly nitrogen does not seem to have the required process capacity, according to operations personnel. The compressor is driven by a steam turbine. The problem solver associated with this plant was asked to determine the cause of the problem and to recommend actions by which to get the compressor back up to capacity as soon as possible.

Given:

- The compressor curve (Fig. 7-7) is available from the compressor vendor
- Gas molecular weight = 28

Table 7-2 Compressor data

Variable	Run 1	Run 2
Suction pressure, psig	5	5
psia	19.7	19.7
Discharge pressure, psig	28	24
psia	42.7	38.7
Suction temperature, °F	100	100
°R	560	560
Discharge temperature, °F	308	300
°R	768	760
Flow, lb/hr	33,100	41,300

- Gas compressibility (z) = 1
- Gas specific heat ratio ($k = C_p/C_v$) = 1.4
- Plant conditions as shown in Table 7-2. All instruments were calibrated before any of the data was taken.

When analyzing the problem, the problem solver used the five-step procedure described earlier. He recognized in the words of the operations personnel that they believed the compressor was not performing as it should, but they had not compared the compressor's actual performance to its predicted performance.

Step 1: Verify that the problem actually occurred.

The problem solver decided that, in order to verify the problem, he had to review the compressor's performance in detail. Therefore, he included the verification step in the problem statement step.

Step 2: Write out an accurate statement of what problem you are trying to solve.

The problem statement that he developed was as follows:

> Plant personnel indicate that the performance of C-100L is not as efficient as they remember it being in the past. The performance of this compressor is causing a decrease in the production limit which does not seem to be easily overcome. Determine if the compressor is performing as predicted by the compressor manufacturer's performance curve. If it is not performing as predicted, determine the cause of the problem and recommend remedial actions.

Note that the above statement emphasizes the step of determining whether a problem exists, compared to the statement provided by operations personnel which emphasizes determining the cause of the problem. Problem-solving

Table 7-3 Compressor calculation steps

1. He calculated α using equation (7-8):

$T_d = T_s \times R^\sigma$ or
$\sigma = (\ln(T_d/T_s)/\ln R)$

	Run 1	Run 2
Suction temperature, °R	560	560
Suction pressure, psia	19.7	19.7
Discharge temperature, °R	768	760
Discharge pressure, psia	42.7	38.7
Compression ratio	2.17	1.96
Compression exponent (σ)	0.408	0.452

2. He calculated the polytropic head using equation (7-6):

$H = 1545 \times T_s \times Z \times (R^\sigma - 1)/M\sigma$

Polytropic head, ft	28110	24400

3. He calculated the ACFM (actual ft^3/min) at C-100L suction:

Gas density, lb/ft^3	0.0919	0.0919
ACFM	6000	7490

4. He looked up the predicted polytropic head from Figure 7-7:

Polytropic head, ft	32000	31500

5. He compared the predicted versus actual polytropic heads, and expressed these values as a percentage:

(Actual – curve) × 100/curve	–12	–23

6. He calculated the polytropic efficiency using equation (7-7) and knowing the compression exponent from step 1 and the specific heat ratio of 1.4:

$\sigma = (k - 1) \times 100/(k \times E)$

$(k - 1)/k$	0.286	0.286
Polytropic efficiency	70	63

7. He calculated fluid horsepower required using equation (7-9):

$BHP = F \times H \times 100/\ (33000 \times E)$

Horsepower	672	805

steps 3 to 5 cannot be initiated until the alleged poor compressor performance can be verified.

To determine whether C-100L was performing as predicted by the performance curve, the problem solver proceeded and developed Table 7-3.

A reasonable conclusion is that C-100L performs well below the performance curve at the higher flow rate, and performs somewhat below the performance curve at the lower rate. In addition to the poor operation relative to the performance curve, the polytropic efficiency is lower at the higher rates.

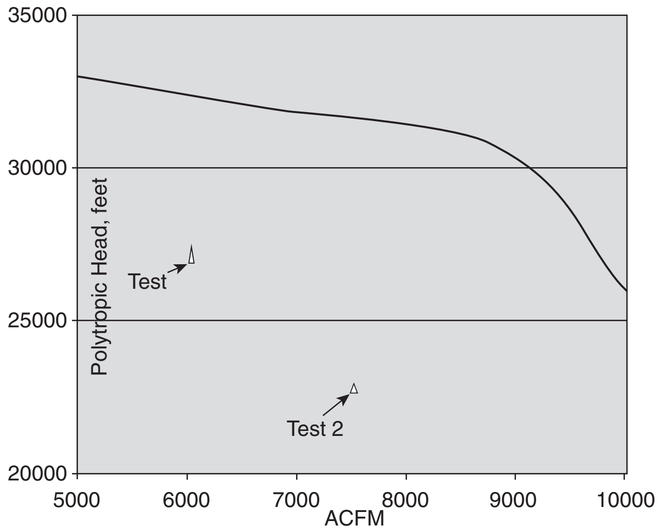


Figure 7-7 Compressor performance curve.

The actual operating points, along with the predicted operating curve, are shown in Figure 7-7. If sufficient data are available, it may be of value to consider when the problem began. The next step is as follows:

Step 3: Develop a theoretically sound working hypothesis that explains as many specifications of the problem as possible.

Referring back to the list of questions given in Chapter 6, the problem solver developed the questions and answers shown below, in Table 7-4.

Note that in formulating answers to the questions, the operator's memory of past performance being better than current performance is not taken as a fact. If the operator's memory was taken as factual, the possibility of design or construction errors could be eliminated. However, memory that is not backed up by hard data is questionable.

Based on the answers to these questions, some possible explanations of the problem were proposed as follows:

1. The suction or discharge piping between the pressure gages and the compressor is too small.
2. There is a restriction in the compressor suction inlet or in the discharge outlet piping.
3. The steam turbine driving the compressor slows as the compressor's horsepower increases.
4. There may be large amounts of compressor internal leakage allowing gases to flow from the discharge to suction side.

Table 7-4 Questions/comments for Problem 7-1

Question	Comment
Are all operating directives and procedures being followed?	All appeared to be correct and are being followed.
Are all instruments correct?	The instruments had allegedly been calibrated.
Are laboratory results correct?	A gas analysis confirmed the molecular weight of the gas.
Were there any errors made in the original design?	The piping might not have been designed for the high rates. However, this would not explain subpar operation at the lower rates.
Were there changes in operating conditions?	No.
Is fluid leakage occurring?	Internal leakage might explain the problem. In addition, the presence of solids might cause a restriction in the suction piping.
Has there been mechanical wear that would explain problem?	Mechanical wear of internals might explain the problem.
Is the reaction rate as anticipated?	Not applicable.
Are there adverse reactions occurring?	Not applicable.
Were there errors made in the construction?	Restrictions in the suction or discharge piping associated with construction might explain the problem.

Hypothesis 3 was eliminated, because the compressor's polytropic efficiency decreases with increasing flow rate. An inadequate steam turbine would not cause the decrease in polytropic efficiency. Hypothesis 4 was treated as a lower priority possibility, even though it would explain a reduced polytropic efficiency. However, it is unlikely that the efficiency would decrease as flow rate increases. The problem solver decided to test both hypotheses 1 and 2 since they are both theoretically sound working hypotheses. Since he was testing both hypotheses, he postponed formalizing the final working hypothesis until he had done additional work.

Step 4: Provide a mechanism to test the hypothesis.

Several methods were developed to test these hypotheses. These tests and the results were as follows:

1. The problem solver calculated the pressure drop in the compressor suction and discharge lines, between the pressure instruments and compressor inlet and outlet flanges. These calculations indicated that the calculated pressure drop was minimal in both lines, thus he decided that hypothesis 1 could be eliminated.

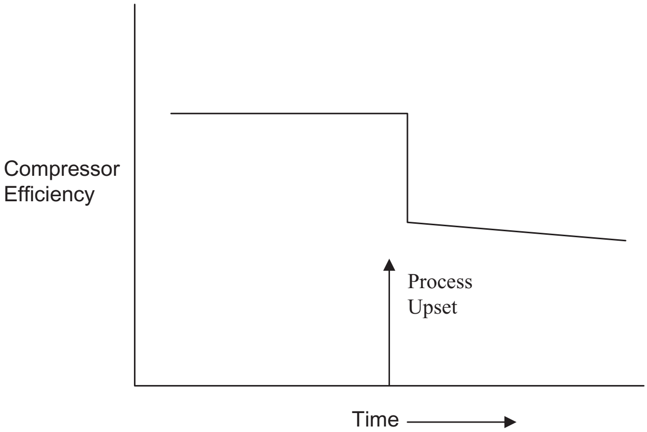


Figure 7-8 Compressor efficiency vs. time.

2. After the pressure drop calculations were made, he arranged for X-rays of the suction and discharge piping to be taken. Based on these X-rays, there appeared to be a significant buildup of solids and/or debris on the bottom of the suction piping before and after the suction pressure instrument.
3. Historical data was reviewed in order to determine the relationship of the compressor's efficiency with time. This relationship is shown in Figure 7-8. It is obvious from this figure that there was a single event in time during which the efficiency dropped dramatically. A review of operating data indicated an upset in the suction knock-out drum that might have caused large amounts of solids to be entrained.

Based on this information, the problem solver developed the following working hypothesis:

> The loss of capacity of C-100L appears to be due to the accumulation of solids in the suction piping. This accumulation of solids occurs after the suction pressure gage. The reduced suction pressure is causing the compressor to appear to operate well below the performance curve at high rates, and 12% below the compressor curve at the lower rates. This reduced suction pressure is also causing an apparent loss in compressor efficiency that is more noticeable at high rates. The loss in performance is more noticeable at high rates because the pressure drop caused by the restriction is greater. The presence of solids in the piping may be related to an upset that occurred in the compressor suction drum.

Step 5: Recommend remedial action to eliminate the problem without creating another problem.

Since the X-rays indicated that the likely presence of solids is the most reasonable cause for the loss of compressor capacity, plans were formulated to shutdown the system and clean out the suction piping. While cleaning out the suction piping did eliminate the current compressor limitation, additional problem-solving effort was required. The problem solver next reviewed the upset that appeared to cause solids to be entrained and developed means to prevent this from happening again.

Lessons Learned Problems should always be investigated in as quantitative a fashion as possible. For example, in this problem, the first step that the problem solver took after developing the problem statement was to determine where the compressor was operating relative to an absolute criterion such as the head-flow curve provided by the compressor's manufacturer. It would have been possible to launch into a hypothesis development period before comparing the actual operation to the predicted head curve; however, it would have given equal validity to erroneous hypotheses and valid hypotheses. For example, it was only when the predicted and actual performances were compared at high and low rates that hypotheses 3 and 4 (that there was internal leakage and that the steam turbine was slowing down) could be eliminated.

While hypotheses 1 and 2 were equally valid and could well have been pursued in parallel, it is almost always faster to make calculations to confirm a hypothesis rather than to arrange for elaborate testing. The calculations of pressure drop in the suction piping indicated that the additional cost and time associated with X-raying the suction piping was indeed of value.

This problem also illustrates the flexibility of the five-step approach. The problem solver did not feel that he had sufficient data to propose a hypothesis until he had done some additional work (step 4) including calculations and X-rays. The 5-step approach should not be used to fit all problem-solving activities into a single exact method.

EXAMPLE PROBLEM 7-2

A process that had been idled for 12 months due to a decrease in demand was now required to operate at full rates. A centrifugal pump (P-25) in the process operated as anticipated until the design flow rate was approached. At that point, it no longer operated as predicted by the pump curve. As the unit returned to full capacity, the level in the accumulator drum upstream of the pump had been decreased to provide for more surge capacity at the higher rates. The pump curve is shown in Figure 7-9. The pump was handling a liquid hydrocarbon at its vapor pressure. A schematic flow sheet for the process is shown in Figure 7-10. Operations personnel requested that the problem solver determine why the pump fails to perform at design conditions.

Given:

- The pump curve shown in Figure 7-9
- Plant test data shown in Table 7-5
- Liquid specific gravity = 0.7

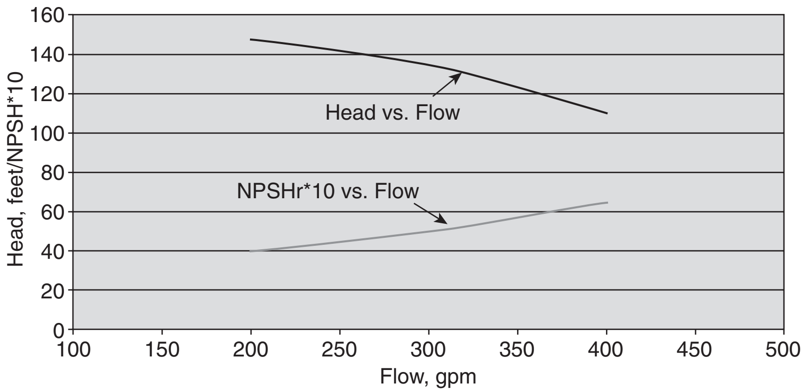


Figure 7-9 Head and NPSHr for Problem 7-2.

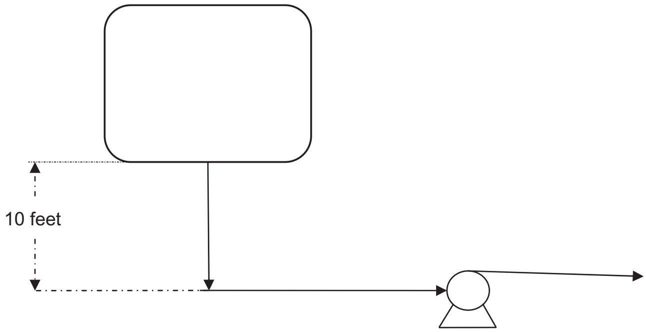


Figure 7-10 Schematic flow for Problem 7-2.

Table 7-5 Plant test data

Test	Run 1	Run 2	Run 3
Flow rate, gpm	200	300	400
Drum pressure, psig	40	40	40
Pump discharge, psig	88	83	65
Drum liquid level, ft	3	3	4

- Hydrocarbon vapor pressure = 40 psig
- Pressure drop in pump suction line = $0.0000528 \times F^{1.8}$, where F = flow rate in gpm

In analyzing the problem, the problem solver used the five-step procedure described earlier. He combined the first two steps as shown below into a problem statement.

Table 7-6 Performance calculations

	Run 1	Run 2	Run 3
1. He calculated the pressure at the pump suction using the relationship given between flow and pressure drop.			
Suction line pressure drop, psi	0.732	1.52	2.55
Elevation head, psi	3.94	3.94	4.25
Suction pressure, psig	43.2	42.4	41.7
2. He calculated the pump differential head using equation (7-2) and knowing the specific gravity relative to water was 0.7 (43.7lb/ft^3).			
Discharge pressure, psig	88	83	65
Differential pressure, psig	44.8	40.6	23.3
Differential head, ft	148	134	77
3. He compared the actual to predicted differential pressure, expressed as a percentage.			
Curve differential head, ft	148	135	110
Error, %	0	0.7	30

Step 1: Verify that the problem actually occurred.

Step 2: Write out an accurate statement of what problem you are trying to solve.

The problem statement that he developed was as follows:

> P-25 has been reported to be operating off the pump curve as the design flow rate is approached. It is likely that this problem was only noticed as the design rates were required after a 12 month period of low rate operation. The level in the accumulator was decreased when the unit was returned to full rates. Since this pump is creating a major production limitation, work to investigate the problem is planned. The first step will be to confirm that P-25 operates on the pump manufacturer's pump curve until the design rate is approached. If the pump does not operate on the curve at all rates, determine the cause of the inadequate performance.

To confirm whether P-25 is operating on the vendor pump curve, the problem solver did the calculations shown in Table 7-6.

The problem solver concluded that P-25 was operating close to the pump curve until the flow rate reached 400 gpm. The next step was:

Step 3: Develop a theoretically sound working hypothesis that explains as many specifications of the problem as possible.

Again, the questions given in Chapter 6 were helpful to the problem solver in formulating possible hypotheses. These questions and the appropriate comments developed by the problem solver for this example problem are shown below, in Table 7-7:

Table 7-7 Questions/comments for Problem 7-2

Question	Comment
Are all operating directives and procedures being followed?	All appeared to be correct and being followed. The operating directive for the level in the drum had recently been reduced.
Are all instruments correct?	The instruments had allegedly been calibrated.
Are laboratory results correct?	Not applicable.
Were there any errors made in the original design?	The piping might not have been designed for the high rates or the increasing $NPSH_R$ as the flow increase might not have been considered.
Were there changes in operating conditions?	No, except for the increased flow and reduced level.
Is fluid leakage occurring?	Internal leakage might explain the problem.
Has there been mechanical wear that would explain problem?	Deterioration of wear rings might explain the problem.
Is the reaction rate as anticipated?	Not applicable.
Are there adverse reactions occurring?	Not applicable.
Were there errors made in the construction?	Not applicable; there had been no recent construction.

Based on this series of questions, two possible hypotheses were developed, as follows:

1. The suction piping is too small or is restricted, causing insufficient NPSH at the pump inlet flange.
2. Deterioration of internal pump wear rings might be causing internal leakage, so that the pump is recirculating large quantities of fluid. This will cause the pump to appear to be operating off the performance curve.

Following the principle of doing calculations prior to making recommendations to do mechanical work, the following calculations shown in Table 7-8 were done.

Since at the highest rate (400 gpm), the $NPSH_R$ (7 ft including the safety factor) is greater than the $NPSH_A$ (5.6 ft) the problem solver developed the following working hypothesis: "The failure of P-25 to operate on the pump curve at high rates is due to the fact that the NPSH is not sufficient. "

Step 4: Provide a mechanism to test the hypothesis.

The problem solver developed two mechanisms to test the hypothesis. The suction line could be replaced, to reduce the pressure drop in the suction line.

Table 7-8 Pump calculations for Problem 7-2

	Run 1	Run 2	Run 3
Vapor pressure of liquid, psig	40	40	40
Suction pressure at pump, psig from Table 7-6	43.2	42.4	41.7
NPSH available, psi	3.2	2.4	1.7
NPSH available, ft	10.5	7.9	5.6
NPSH required, ft			
From curve	4	5	6.5
Safety factor	0.5	0.5	0.5
Total	4.5	5.5	7

A larger suction line that reduced the pressure drop to less than 1 psi would increase the $NPSH_A$ to a value above that required by the pump. Another possible mechanism to test the hypothesis would be to raise the level in the drum. If the level were increased by at least 1.5 ft, the $NPSH_A$ should be adequate.

Step 5: Recommend remedial action to eliminate the problem without creating another problem.

The remedial action depends on which of the hypothesis testing mechanisms is used. If the suction line is replaced and the problem is eliminated, this mechanism becomes the remedial action. If raising the drum level is a successful test, this may or may not be a remedial action. Raising the drum level reduces the amount of surge volume that can be used. A careful study would be required to determine if this action creates other problems.

Lessons Learned While the available data did not allow the problem solver to eliminate the hypothesis that there was a mechanical problem with the pump that allowed it to recirculate large amounts of fluid, he could use calculations to show that the hypothesis of inadequate available NPSH was a correct hypothesis. Thus if the pump had no mechanical problems at all, it would still not perform as predicted by the pump curve. Therefore, the problem solver elected to develop solutions to the problem of the lack of available NPSH.

Often, in a process unit, operating directives that are changed and seem to work well at reduced rates are not adequate when the rates are increased to design levels. In addition, designs that work at less than design rates may not work at full rates. In the specific example problem, the reduction in the drum level seems to be the cause of the lack of available NPSH. However, additional study would be required before raising the drum level would be considered an acceptable solution.

EXAMPLE PROBLEM 7-3

(This problem is provided to illustrate a calculation technique.) A reciprocating compressor (C-100A) was operating in a mode that resulted in an overload condition on the electric motor. The compressor was equipped with adjustable clearance pockets so that the clearance as a fraction of displacement could be varied. What clearance pocket setting should be used to obtain the maximum flow rate without overloading the electric motor?

Given:

• Clearance pocket setting, % of displacement	4 or 6 or 10
• Suction pressure, psig	15
• Discharge pressure, psig	85
• Compressibility factor (z)	1
• Molecular weight	42
• Polytropic compression exponent	0.32
• Suction temperature, °F	100
• Ratio of specific heats	1.4
• Polytropic efficiency	90
• Mechanical efficiency	95
• Piston displacement, ft^3/min	300
• Motor rating, HP	55

The following procedure, shown in Table 7-9, can be used to analyze this problem:

Table 7-9 Calculation procedure

1. Calculate the compression ratio remembering to use psia:

$$R = 99.7/29.7 = 3.36 \tag{7-18}$$

2. Calculate the polytropic head using equation (7-6):

$$H = 1545 \times T_S \times Z \times (R^{\sigma} - 1)/M\sigma$$
$$H = (1545 \times (100 + 460) \times 1 \times (3.36^{0.32} - 1))/(42 \times 0.32) \tag{7-19}$$
$$= 30500 \text{ ft}$$

3. Calculate the volumetric efficiency for each clearance, using equation (7-14):

$$E_V = 100 \times (1 - L - C \times (R^{1/k} - 1))$$

The leakage factor (L) can be evaluated as 2% of the compression ratio, as follows:

$$L = 2 \times R/100 = 0.07 \tag{7-20}$$

Clearance, %	4	6	10
Clearance factor (C)	0.04	0.06	0.10

(*Continued*)

Table 7-9 (*Continued*)

Leakage factor (L)	0.07	0.07	0.07
$R^{1/k} - 1$	1.38	1.38	1.38
Volumetric efficiency, %	87	85	79
4. Calculate the mass flow rate and BHP. Use equation (7-9) to calculate the BHP:			
$\text{BHP} = F \times H \times 100/(33{,}000 \times E)$			
Volumetric flow rate, ft^3/min	261	255	237
From piston displacement and volumetric efficiency			
Gas density, lb/ft^3	0.208	0.208	0.208
Mass flow rate, lb/min	54.3	53	49.3
BHP	58.7	57.3	53.3

Note the BHP includes both mechanical and polytropic efficiency.

Conclusion: The 10% pocket setting must be used to avoid overloading the motor.

EXAMPLE PROBLEM 7-4

The compressor described in Problem 7-3 began losing capacity. The flow meter showed about 35 lb/min. Estimates of motor loading, based on ampere readings, indicated that the compressor required significantly less than 50 BHP. All operating conditions were identical to those specified in Problem 7-3. In addition, the pocket clearance was set at 10%. When the problem solver was asked to consider this problem, he used the five-step approach discussed earlier.

Step 1: Verify that the problem actually occurred.

Verification that the problem was actually occurring was relatively simple, since both the flow meter and the ampere reading indicated that the compressor's output was less than its rated capacity.

Step 2: Write out an accurate statement of what problem you are trying to solve.

The problem solver wrote out the following problem statement:

> C-100A seems to be operating well below its design capacity. The flow meter shows 35 lb/hr where the flow should be close to 50 lb/hr. In addition, the horsepower loading is well below the anticipated load of 50 BHP as determined by an ampere reading. The compressor unloading pockets are set so that the clearance is about 10%, which is normal. It is not known when the decrease in capacity actually occurred, but it seems to have been a gradual decrease. Determine what has caused the loss of capacity on C-100A (the reciprocating compressor), why it occurred, and what can be done to eliminate the problem.

Step 3: Develop a theoretically sound working hypothesis that explains as many specifications of the problem as possible.

The questions given in Chapter 6 were used to help formulate possible hypotheses. These questions and the appropriate comments developed by the problem solver for this example problem are shown below, in Table 7-10.

Several possible hypotheses were developed by the problem solver, as follows:

1. The valves in the compressor are bad, causing the loss in capacity.
2. The flow meter is erroneously low.
3. The worn belt drive between the electric motor and compressor is slipping, causing the compressor to operate at a much lower speed than design.
4. The molecular weight of the gas being compressed is different than design.
5. The actual suction pressure is much lower than that shown by the instrumentation.

The problem solver developed a table to attempt to sort out the various hypotheses by comparing the theoretical impact of the hypotheses to the actual observations. This is shown in Table 7-11.

Table 7-10 Questions/comments for Problem 7-4

Question	Comment
Are all operating directives and procedures being followed?	All appeared to be correct and being followed.
Are all instruments correct?	All instruments except the flow meter had been calibrated.
Are laboratory results correct?	Analysis of gas being compressed had not been obtained for several months.
Were there any errors made in the original design?	Not applicable, since the compressor capacity loss was a recent occurrence.
Were there changes in operating conditions?	No.
Is fluid leakage occurring?	Internal leakage might explain problem.
Has there been mechanical wear that would explain problem?	Deterioration of compressor valves might explain the problem. In addition, the belt on the electric motor was beginning to look worn.
Is the reaction rate as anticipated?	Not applicable.
Are there adverse reactions occurring?	Not applicable.
Were there errors made in the construction?	Not applicable.

Table 7-11 Hypothesis comparison

	Would Hypothesis Explain Observations of	
Hypothesis (Number)	Low Ampere Reading	Low Flow Rate
Bad valves (1)	Yes	Yes
Flow meter (2)	No	Yes
Slipping belt (3)	Yes	Yes
Molecular weight (4)	No(1)	?
Low suction pressure (5)	?(2)	Yes

(1)For a reciprocating compressor, the horsepower load is essentially independent of molecular weight. This is true because head is indirectly related to molecular weight (see equation (7-6)), and the mass rate is directly related to the molecular weight. For example, if the molecular weight of a gas being compressed doubles, the polytropic head will be reduced by a factor of 2 and the amount of gas being compressed will increase by a factor of 2. This assumes that the suction and discharge pressures are constant and the only change is in the molecular weight of the gas.

(2)Whether or not a low suction pressure can explain the decrease in horsepower requirements depends upon which side of the load point the compressor is operating at normal conditions. See Figure 7-6 for a typical load versus suction pressure curve. For this specific compressor, it is likely that decreasing the suction pressure will have minimal impact on the horsepower requirements.

Since "slipping belts" (hypothesis 3) will almost always be heard, even above the noise in a compressor house, the problem solver proposed the residual theoretically sound working hypothesis as follows: "The poor performance of C-100A is due to leakage through the suction and/or discharge valves."

Step 4: Provide a mechanism to test the hypothesis.

The actual mechanism for testing the hypothesis involved determination of the "optimum technical depth" required, as described in Chapter 3. Some points that were considered in determining the "optimum technical depth" were as follows:

- How urgent was the problem? Was the loss of capacity causing a loss in plant production, or could gases be diverted to another location?
- Was a spare compressor available? If a spare compressor were available, the required degree of confidence that the solution was correct would be less than it would be if no spare were available.
- Would a compressor shutdown require an entire unit shutdown? If a unit shutdown was required, the required degree of confidence would be much higher.
- Could inlet and outlet temperatures be easily measured, and was historical data available? If these temperatures could be easily obtained and compared to historical data, it would increase the degree of confidence in the solution.

The two possibilities were that the current and historical inlet and outlet temperatures were easily available. In this case, the compression exponent could be calculated using equation (7-8), as shown below:

$$T_D = T_S \times R^{\sigma} \tag{7-8}$$

Then archived data could be used to calculate the historical compression exponent (σ). The current data could be compared to historical data and if a change were obvious, it could be concluded that this was a valid test of the hypothesis. If no historical data is available, equation (7-7) can be used to calculate the theoretical compression exponent from the vendor's performance curve.

$$\sigma = (k-1) \times 100/(k \times E) \tag{7-7}$$

This theoretical compression exponent can then be compared to the actual exponent based on plant operating data. If compressor inlet and outlet temperatures are not readily available, it may be possible to use infra-red techniques discussed in Chapter 11 to approximate compressor temperatures.

In this particular problem, a spare compressor was not available. An analysis of the inlet and outlet temperatures indicated that the deterioration of performance had been a gradual decline. This analysis is shown in Table 7-12.

Step 5: Recommend remedial action to eliminate the problem without creating another problem.

Unfortunately, it was necessary to recommend a compressor shutdown to replace the worn valves. Since there was no spare compressor and the com-

Table 7-12 Analysis of compression exponent for Problem 7-4

		Days After Last Valve Replacement			
Variable	Design	0	30	60	90
Suction					
Temperature, °F	70	70	70	70	70
°R	530	530	530	530	530
Pressure, psig	15	15	15	15	15
psia	29.7	29.7	29.7	29.7	29.7
Discharge					
Temperature, °F	320	320	331	340	360
°R	780	780	791	800	820
Pressure, psig	85	85	85	85	85
psia	99.7	99.7	99.7	99.7	99.7
Compression ratio	3.36	3.36	3.36	3.36	3.36
Compression exponent	0.32	0.32	0.33	0.34	0.36

pressor was an integral part of the unit, it was necessary to shutdown the entire unit. As seems to be the rule, the shutdown occurred at a bad time from a business perspective.

Lessons Learned Replacement of valves on a reciprocating compressor is a "normal maintenance" item. However, since this compressor was in a critical service, the problem solver developed a system to allow inclusion of the compression exponent in a daily monitoring system. His recommendation to do this allowed optimization of the timing for a compressor downtime. In addition, it allowed him to follow the valve performance closely and to potentially correlate valve performance with process upsets or changes that are causing rapid valve wear.

The importance of understanding technology is also illustrated by this problem. The original hypothesis that a change in molecular weight of the gas being compressed was eliminated because a reciprocating compressor's horsepower is essentially independent of molecular weight. In addition, the possibility that a low suction pressure is causing the problem was eliminated since, with this specific reciprocating compressor, suction pressure has minimal affect on horsepower load. These conclusions are not intuitively obvious and can only be developed through an understanding of compression technology.

EXAMPLE PROBLEM 7-5

(This problem is provided to illustrate a calculation technique.) Determine the $NPSH_A$ for a reciprocating pump given the following:

• Pump type	double acting
• Cylinder diameter	0.5 in
• Stroke length	4 in
• Speed	60 RPM
• Suction line diameter	3/4 in
• Suction line length	25 actual ft
• Suction drum pressure	40 psig
• Liquid elevation	10 ft above pump suction
• Fluid vapor pressure	40 psig
• Fluid specific gravity	0.65 relative to water
• Fluid viscosity	0.2 centipoise

As indicated in Section 7.6, there are two kinds of pressure drop in the suction line of a reciprocating pump. There is the typical type of frictional loss, but this occurs at maximum flow rate which is greater than the average flow rate. In addition, the entire volume of fluid in the suction piping must be accelerated from a nonflow condition to the maximum velocity. As indicated in this section, these two frictional losses occur completely out of phase with each other. That is, the frictional loss associated with the maximum flow rate occurs when there is no acceleration. And the acceleration loss from zero velocity to full line velocity occurs when there is no frictional loss. Thus, the pressure loss in the pipe is generally taken as the larger of the two calculated values. The calculations for estimating these pressure losses are based on equations (7-11) and (7-12). The calculation procedure is as follows:

1. Calculate the maximum flow rate in the suction pipe:

$$V_P = K \times D^2 \times L \times S/(60 \times D_P^2) \tag{7-11}$$

$$= \pi \times (0.0417^2) \times 0.33 \times 60/(2 \times 60 \times (0.0625^2) \tag{7-21}$$

$$= 0.23 \text{ fps}$$

2. Calculate the frictional pressure drop by conventional means. This calculation is based on equations (5-24) and (5-25) as shown below:

$$\Delta P = 0.323 \times f \times S \times U^2 \times L/d \tag{5-24}$$

$$f = -0.001 \times \ln(D \times U \times S/Z) + 0.0086 \tag{5-25}$$

$$f = -0.001 \times \ln(0.75 \times 0.23 \times 0.65/0.2) + 0.0086 \tag{7-22}$$

$$f = 0.0092$$

$$\Delta P = 0.323 \times 0.0092 \times 0.65 \times (0.23^2) \times 25/0.75 \tag{7-23}$$

$$\Delta P = 0.0034 \text{ psi}$$

$$h_F = 0.012 \text{ ft}$$

3. Calculate the head required to accelerate the fluid:

$$h_A = 1.35 \times L_P \times S \times \mathrm{V_P}/307 \tag{7-12}$$

$$= 1.35 \times 25 \times 60 \times 0.23/307 \tag{7-24}$$

$$= 1.5 \text{ ft}$$

4. Select the larger of the two values to determine the actual NPSH for the system.

In this specific problem, the acceleration head will be the critical pressure loss. Actual NPSH at the pump suction will be 10 ft less 1.5 ft, or 8.5 ft. This available NPSH will normally be adequate for most reciprocating pumps.

EXAMPLE PROBLEM 7-6

A centrifugal process gas compressor, C-5A, driven by a steam turbine, no longer had sufficient capacity for the service that it was designed for. The speed of the turbine was controlled by a steam control valve in the incoming steam.

Table 7-13 Design and current operations

Variable	Design	Current Operations
Suction pressure, psig	5	5
Discharge pressure, psig	45	35
Suction temperature, °F	100	100
Discharge temperature, °F	301	264
Flow rate, lb/hr	25000	25000
Gas composition		
Propylene, mol %	95	95
Nitrogen, mol %	5	5
Calculated molecular weight	41.3	41.3
Compressibility	1.0	1.0
Specific heat ratios		
Propylene	1.21	1.21
Nitrogen	1.4	1.4
Mixture	1.2195	1.2195
Turbine speed, RPM		
Maximum	10000	10000
Design	9000	
Actual		8400
Inlet steam conditions		
psig	200	190
Temperature, °F	500	485
Outlet steam conditions		
psig	25	25
Temperature, °F	280	295
Steam flow rate, lb/hr		Out of service
Maximum	16000	
Design	14700	
Steam control valve		
Position, %	80	100

There had allegedly been no changes in operating conditions except for a decrease in steam pressure from 200 psig to 190 psig. This decrease in steam pressure was part of an overall optimization of the plant utility system. In order to increase the steam pressure slightly, operations personnel removed the steam meter orifice that was used to measure the steam flow to the steam turbine. Operations personnel had requested technical help to prove that the steam pressure should be increased back to 200 psig.

The design and current operating conditions are shown in Table 7-13.

The maximum design of the steam turbine included a slight safety factor to allow operations at 10000 RPM with the steam control valve 90% open. As noted in the above table, even with the steam valve 100% open, it is not possible to reach 9000 RPM. The design conditions are those required for the process gas compressor.

The problem solver approached the problem using the 5-step procedure as shown below.

Step 1: Verify that the problem actually occurred.

As the first step in the process of verifying that the problem was real, the problem solver had the appropriate instruments checked. He also used independent sources to confirm the flow and pressure meters. While the compressor output had not been followed on a daily basis, key variables shown in Table 7-13 were available from computer archives. All of these indicated that there was a real problem.

Step 2: Write out an accurate statement of what problem you are trying to solve.

The problem solver wrote out a description of the problem that he was trying to solve as follows:

> Currently, C-5A does not seem to have the desired capacity, even though capacity was adequate in recent history. The steam turbine is operating at a slightly lower speed than that in the design values and the steam control valve is fully open. There has been a reduction in inlet steam pressure from 200 psig to 190 psig. However, it is not clear that this is the cause of the lack of capacity. Determine why C-5A does not have sufficient capacity and provide recommendations for improving the performance of C-5A.

Step 3: Develop a theoretically sound working hypothesis that explains as many specifications of the problem as possible.

The questions given in Chapter 6 were helpful in formulating possible hypotheses. These questions and appropriate comments for this example problem are shown in Table 7-14.

Several hypotheses were proposed as follows:

- When the steam system was optimized, the process designer did not adequately consider the impact of the steam pressure on the steam turbine.
- There is excessive leakage around the wear rings on the process gas compressor. This excessive leakage would cause a compressor efficiency lower than that of the design, as well as a reduced gas rate.
- The steam turbine steam jets may have deteriorated. This would cause the steam turbine to have a lower efficiency and thus extract less horsepower per pound of steam than it was designed to do.

Table 7-14 Questions/comments for Problem 7-6

Question	Comment
Are all operating directives and procedures being followed?	All appeared to be correct and being followed except for the lower steam pressure.
Are all instruments correct?	All instruments except the steam flow meter had been calibrated.
Are laboratory results correct?	Laboratory results indicating that the gas being compressed was 95% propylene and 5% nitrogen were confirmed.
Were there any errors made in the original design?	The original design was okay, but there was a question about whether the change in steam pressure had received adequate consideration.
Were there changes in operating conditions?	Yes. The inlet steam pressure was reduced.
Is fluid leakage occurring?	Internal leakage might explain the problem.
Has there been mechanical wear that would explain problem?	Deterioration of the steam turbine nozzles might explain the problem.
Is the reaction rate as anticipated?	Not applicable.
Are there adverse reactions occurring?	Not applicable.
Were there errors made in the construction?	Not applicable.

Table 7-15 Calculation of polytropic compression exponent

Variable	Design	Operating
Suction pressure, psig	5	5
psia	19.7	19.7
Discharge pressure, psig	45	35
psia	59.7	49.7
Suction temperature, °F	100	100
°R	560	560
Discharge temperature, °F	301	264
°R	761	724
Calculated compression ratio	3.03	2.52
Polytropic compression exponent	0.277	0.278

At this point, the problem solver had developed three theoretically correct working hypotheses, all of which could be evaluated further by additional calculations. Since two of the hypotheses would require compressor shutdowns to inspect and the compressor was not spared, he elected to pursue additional calculations as part of step 4.

Step 4: Provide a mechanism to test the hypothesis.

The problem solver made the following calculations to help determine if one of these proposed hypotheses was both possible and supported by calculations.

1. He determined the polytropic compression exponent using equation (7-8) as shown in Table 7-15:

$$T_D = T_S \times R^{\sigma} \tag{7-8}$$

or

$$\sigma = \ln(T_D/T_S)/\ln R$$

Since the polytropic compression exponent is the same for the design and operating conditions, the problem solver concluded, based on equation (7-7) shown below, that the compressor efficiency was the same as the design efficiency.

$$\sigma = (k-1)\times 100/(k \times E) \tag{7-7}$$

2. He calculated the steam turbine efficiency using equation (7-17) and the data in Table 7-13, as shown below:

$$E_T = 100 \times (H_I - H_O)/(H_I - H_E) \tag{7-17}$$

The steam turbine efficiency is simply the actual enthalpy change, divided by the enthalpy change at 100% efficiency, multiplied by 100, as shown in equation (7-17) earlier and equation (7-25) below:

$$\begin{aligned}\text{Steam turbine efficiency (design case)} &= 100 \times 91/142 \\ &= 64\%\end{aligned} \tag{7-25}$$

It is of value to review how the thermodynamic properties in Table 7-16 were actually developed. The inlet and outlet steam enthalpies were taken from steam enthalpy tables, which are given in multiple publications. These values were used to determine the actual enthalpy change across the turbine. The value shown in the table as "At 100% efficiency" assumes isentropic (constant entropy) steam expansion. The most efficient steam turbine process is one that occurs at constant entropy. The outlet steam conditions are then based on the outlet pressure and maintaining the same entropy as that of the inlet steam conditions. The outlet enthalpy for isentropic expansion is then determined from steam charts or steam tables, as shown in Table 7-17.

The actual steps involved in developing Table 7-17 are as follows:

Table 7-16 Calculation of steam turbine efficiency

Variable	Design	Operating
Inlet steam conditions		
psig	200	190
psia	214.7	204.7
Temperature, °F	500	485
Enthalpy, BTU/lb	1267	1260
Outlet steam conditions		
psig	25	25
psia	39.7	39.7
Temperature, °F	280	308
Enthalpy, BTU/lb	1176	1191
Enthalpy change, BTU/lb		
Actual	91	69
At 100 % efficiency	142	137
Steam turbine efficiency, %	64	50
Calculated steam rate, lb/hr	14700	15800

Table 7-17 Calculation of outlet enthalpy for an isentropic (100% efficiency) expansion

	Design	Operating
Inlet steam conditions		
psig	200	190
psia	214.7	204.7
Temperature, °F	500	485
Enthalpy, BTU/lb	1267	1260
Entropy, BTU/lb-°R	1.615	1.612
Outlet steam conditions		
psig	25	25
psia	39.7	39.7
Entropy, BTU/lb-°R	1.615	1.612
Vapor entropy	1.6763	1.6763
Liquid entropy	0.3919	0.3919
% vapor	95.2	95.0
Vapor enthalpy, BTU/lb	1169.7	1169.7
Liquid enthalpy, BTU/lb	236.03	236.03
Outlet enthalpy, BTU/lb	1124.9	1123.04
Enthalpy change		
Isentropic expansion, BTU/lb	142	137

1. From the inlet steam conditions of temperature and pressure, determine the enthalpy and entropy of the steam.
2. Remembering that the isentropic expansion (100% efficiency) is, by definition, one that occurs when there is no change in entropy, determine the condition of the outlet steam at constant entropy. In this case, the entropy

at 25 psig is between the liquid and vapor entropy. Thus there will be some liquid in the steam leaving the turbine.

3. The actual % vapor can be determined for an isentropic expansion in the design case using the formula below:

$$\begin{aligned}\%\text{ vapor} &= 100 \times (1.615 - 0.3919)/(1.6763 - 0.3919)\\ &= 95.2\% \end{aligned} \tag{7-26}$$

4. Knowing the % vapor, the enthalpy of the outlet steam if the expansion were isentropic can be calculated as shown below:

$$\begin{aligned}\text{Outlet enthalpy} &= 236.03 + 95.2 \times (1169.07 - 236.03)/100\\ &= 1124.9 \text{ BTU/lb}\end{aligned} \tag{7-27}$$

5. The enthalpy change, or work for an isentropic expansion, can then be calculated as follows:

$$\text{Work} = 1267 - 1124.9 = 142.1 = 142 \text{ BTU/lb} \tag{7-28}$$

As shown in Table 7-16, the current efficiency of the steam turbine appeared to be less than design. Thus it seemed unlikely that the current capacity loss was associated with the steam pressure change. Since archived data was available, the problem solver reviewed the historical data to determine when the loss of efficiency occurred and whether it was a onetime loss or a gradual loss. The problem solver used the techniques shown above and developed Figure 7-11. This figure clearly showed that the steam turbine problem was not related to the drop in steam pressure, but was a gradual decay in efficiency that began well after the change in steam pressure occurred.

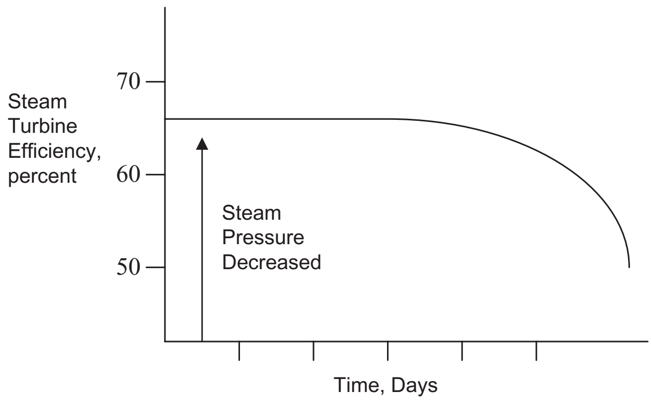


Figure 7-11 Steam turbine efficiency vs. time.

Table 7-18 Compressor horsepower calculations

Variable	Design	Operating
Suction pressure, psig	5	5
psia	19.7	19.7
Discharge pressure, psig	45	35
psia	59.7	49.7
Suction temperature, °F	100	100
°R	560	560
Discharge temperature, °F	301	264
°R	761	724
Flow rate, lb/hr	25000	25000
Polytropic efficiency	65	65
Polytropic compression exponent	0.277	0.278
Calculated polytropic head	27200	22100
Fluid horsepower	530	430

The problem solver had one more question to answer: Would an increase in steam pressure, back to 200 psig, ameliorate the loss of turbine capacity? To answer this question, he calculated the current horsepower load and compared it with the design horsepower load.

Equations 7-6 and 7-9 shown below and the data given in Table 7-13 were used to calculate the polytropic head and fluid horsepower. These equations were then used to develop Table 7-18.

$$H = 1545 \times T_S \times Z \times (R^{\sigma} - 1)/M\sigma \tag{7-6}$$

$$\sigma = (k - 1) \times 100/(k \times E) \tag{7-7}$$

$$\text{BHP} = F \times H \times 100/(33{,}000 \times E) \tag{7-9}$$

He then considered whether the increase in steam pressure from 190 psig to 200 psig would cause the compressor to return to normal. To consider this change in steam pressure, the problem solver first estimated the current steam rate, knowing the delivered fluid horsepower (430 BHP) and the enthalpy change across the turbine. He assumed that steam conditions were returned to the higher pressure, the steam rate increased slightly due to the higher pressure, and that the turbine efficiency remained the same (50%). He then calculated the horsepower that would be delivered to the process, as shown in Table 7-19.

Based on the calculations shown in Table 7-19, the problem solver concluded that raising the steam pressure would not significantly increase the capacity of the process gas compressor. It should be noted that since the turbine efficiency is known, it is not necessary to develop the outlet steam conditions of temperature and enthalpy. However, these conditions could be developed, if desired, using a similar approach as described earlier.

Table 7-19 Calculation of BHP delivered to compressor with increased steam pressure

	Operating Conditions		
Variable	Design	Current	Proposed
Inlet steam conditions			
psig	200	190	200
psia	214.7	204.7	214.7
Temperature. °F	500	485	500
Enthalpy, BTU/lb	1267	1260	1267
Outlet steam conditions			
psig	25	25	25
psia	39.7	39.7	39.7
Temperature, °F	280	308	TBD
Enthalpy, BTU/lb	1176	1191	TBD
Enthalpy change, BTU/lb			
At 100% efficiency	142	137	142
Steam turbine efficiency	64	50	50
Actual	91	69	71
Steam rate	14700	15800	16300
Work from turbine			
MBTU/hr	1.3377	1.09	1.157
Horsepower	525	428	455

He then developed the following hypothesis:

> It is believed that the steam turbine steam jets have suffered mechanical damage, which has resulted in the gradual deterioration of their efficiency. This would cause the steam turbine to have a lower efficiency and thus to extract less horsepower per pound of steam than it was designed to do. Increasing the steam pressure from 190 psig to 200 psig would have minimal impact on the turbine.

Step 5: Recommend remedial action to eliminate the problem without creating another problem.

Since all indications were that there was some sort of mechanical damage to the internals of the steam turbine and that the turbine efficiency was continuing to deteriorate, the problem solver had no choice but to recommend that the steam turbine be shut down for repairs. In addition, the time relationship indicated that the repairs should be done as soon as possible. Since issues with safety did not appear to be involved, the actual timing of the shutdown was left to the discretion of the management team.

The management team was pleased with the detailed analysis of the problem because:

1. The initial apparent cause of the problem (the steam pressure reduction) would have been difficult to reverse since it involved several processes in the plant.
2. The relationship of the turbine efficiency with time that was developed was helpful in determining that the steam turbine should be repaired very quickly.

Lessons Learned The value in doing calculations and time related figures is clearly illustrated by this real life problem. Before the problem solver could determine which piece of equipment was the source of the problem, he had to determine the performance of both the compressor and turbine. He determined this by comparing the actual efficiency to the design efficiency. Knowing that the compressor efficiency was essentially the same as the design and that the steam turbine efficiency was less than design allowed him to conclude that the operating problem was related to the steam turbine. In addition, an analysis of the steam turbine supply pressure allowed him to conclude that decreasing the steam pressure was not the cause of the problem and that increasing it back to the original setting would not resolve the performance discrepancy.

If these calculations had not been done, there would be three possible hypotheses that could all have been treated as valid. Without the calculations, the most easily identified change would be the reduction in steam pressure. It would also appear to be the easiest route to improving performance. If the problem solver had not completed the discussed calculations, it is likely that increasing the steam pressure would have been chosen as the route to improve performance. Valuable time that could have been spent in doing a better job in assessing the problem would have been spent raising the steam pressure.

EXAMPLE PROBLEM 7-7

A new process had recently been put into operation. While the startup had gone very well, there was a continuing compressor problem: the main recycle gas compressor would mysteriously shutdown. The 1500-BHP, two-stage reciprocating compressor was driven by an electric motor. This motor was provided with a shutdown device that was triggered if the horsepower load was 3% greater than the motor rating. Thus the compressor motor would shutdown if the load exceeded 1550 BHP.

The compressor was provided with a "first out" indicator which was used to determine which process variable caused the electric motor to shut down. The first out indicator included process variables such as oil pressure, high discharge temperatures, high suction temperatures, and low suction pressure.

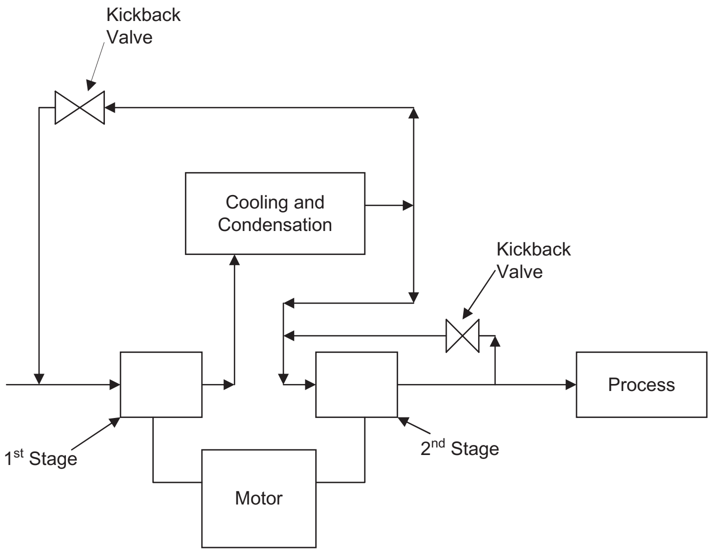


Figure 7-12 Schematic flow for Problem 7-7.

High suction pressure was not included in the first out indicator. However, there was a high suction pressure alarm which would provide a warning prior to the pressure increasing to the point at which the safety valve would release. The indicator panel also had a category called "other reason." Unfortunately, the indicator almost always showed "other reason" for the shutdowns being experienced.

The process gas load was not constant. The gas load varied as the reactor conversion changed; if the reactor conversion decreased, the gas load increased. The process design of the facilities assumed that the conversion would remain constant. Decreases in the conversion were associated with the presence of impurities in the reactor feed or changes in the catalyst flow rate to the reactor.

The compressor was provided with computer-controlled unloaders and recycle capabilities to maintain a constant suction pressure on both stages. If the gas load was such that both the recycle valves were closed and the unloaders were in a position to compress the maximum amount of gas, the suction pressure would increase until the gas density increased to the point that the compressor had adequate capacity. A schematic sketch of the facilities is shown in Figure 7-12. The key variables for the compressor are given in Table 7-19.

Operations personnel have requested help from the technical organization to determine what was wrong with the first out indicator. The problem solver approached the problem using the 5-step procedure, as shown below.

Table 7-20 Compressor operating data for Problem 7-7

First-stage suction pressure, psig	3
Second-stage suction pressure, psig	60
Interstage pressure drop, psi	2
Discharge pressure, psig	275
Compressibility factor (z)	1
Molecular weight	42
First-stage suction temperature, °F	100
First-stage discharge temperature, °F	283
Second-stage suction temperature, ° F	110
Second-stage discharge temperature, °F	280
Interstage condensation, lb/hr	0
Ratio of specific heats	1.21
Polytropic efficiency	90
Mechanical efficiency	95
Piston displacement, ft^3/min	5000
Motor rating, HP	1550 (includes a 3% overload factor)

Step 1: Verify that the problem actually occurred.

There was no doubt that the compressor was shutting down. However, it wasn't obvious that there was anything wrong with the "first out" indicator. There were times when it showed indications besides "other reason." In addition, simulated signals were used when the system was out of service to confirm that all other indicators worked. The problem solver expanded the original problem scope and stated it as shown in step 2.

Step 2: Write out an accurate statement of what problem you are trying to solve.

The statement written by the problem solver was as follows:

> C-122 has been shutting down for no apparent reason. The "first out" indicator shows the cause is "other reason." At about the same time as the mysterious shutdowns, the suction pressure tends to increase. However, it has been impossible to determine whether this causes the shutdowns or is the result of the shutdowns. Investigate the shutdowns of C-122, the recycle gas compressor, to determine what is causing the unknown shutdowns. There may be a problem with the first out indicator or there may be an unknown reason for the compressor shutdowns. When the cause is determined, provide recommendations to eliminate the problem.

Step 3: Develop a theoretically sound working hypothesis that explains as many specifications of the problem as possible.

The questions given in Chapter 5 were used to help formulate possible hypotheses. These questions and appropriate comments for this example problem are shown below, in Table 7-21:

Table 7-21 Questions/comments for Problem 7-7

Question	Comment
Are all operating directives and procedures being followed?	All appeared to be correct and being followed, though there were times when the low-stage suction pressure was not well controlled.
Are all instruments correct?	All pressure instruments had been calibrated.
Are laboratory results correct?	Not applicable.
Were there any errors made in design?	Since this was a new process, this was a consideration. However, the major concerns were the first out indicator and the capability of the compressor to handle swings in recycle gas rates with subsequent changes in suction pressure.
Were there changes in operating conditions?	No, except for the suction pressure swings.
Is fluid leakage occurring?	Internal leakage might explain the failure of the compressor to handle the gas flow under some conditions.
Has there been mechanical wear That would explain problem?	Leaking compressor valves might explain the problem.
Is the reaction rate as anticipated?	Normally yes. However, decreases in reaction rate cause increased recycle gas rates.
Are there adverse reactions occurring?	Not applicable.
Were there errors made in the construction?	Construction errors in the first out system might explain the problem.

Based on the above table, the problem solver formulated the following hypotheses.

1. There is a compressor mechanical problem, such as bad valves. This problem is not apparent at low flow rates, but becomes apparent at high rates. This mechanical problem causes the compressor to have insufficient capacity at high gas rates. As the suction pressure increases, the compressor shuts down for some unknown reason.
2. The compressor is shutting down, for some unknown reason, when the suction pressure rises slightly due to normal process variability.
3. There is an intermittent instrumentation failure with the first out system. This failure is causing some problem, such as low oil pressure, to shutdown the compressor but not show up on the display panel.

Note that there is similarity between the first and second hypothesis. The first hypothesis implies that there is a mechanical condition that can be repaired

Table 7-22 Calculations of compressor efficiency

Variable	Design	Current
Low-stage suction pressure, psig	3	3
psia	17.7	17.7
Low-stage discharge pressure, psig	62	62
psia	76.7	76.7
High-stage suction pressure, psig	60	60
psia	74.7	74.7
High-stage discharge pressure, psig	275	275
psia	289.7	289.7
Low-stage suction temperature, °F	100	100
°R	560	560
Low-stage discharge temperature, °F	278	283
°R	738	743
High-stage suction temperature, °F	110	110
°R	570	570
High-stage discharge temperature, °F	276	280
°R	736	740
Compression exponent (from temperatures)		0.193
Polytropic efficiency, %	92	90
Polytropic head, ft	67200	67400
Mass flow rate, lb/min	620	620
Total horsepower, BHP	1440	1480

and the problem will be eliminated. The second hypothesis indicates that the mysterious shutdowns are due to process variability and that there is nothing mechanically wrong with the compressor.

When faced with the need to decide which of these hypotheses to pursue, the problem solver recognized that it might be difficult to trace an intermittent instrument failure associated with the first out system. On the other hand, it would be possible to quickly do some calculations to confirm whether or not either hypotheses 1 or 2 were theoretically correct. He used the following relationships and the basic data shown in Table 7-20 to develop Table 7-22.

$$T_D = T_S \times R^{\sigma} \tag{7-8}$$

The problem solver used equation (7-8) with the suction and discharge temperatures in absolute temperature units and calculated a compression exponent of 0.193. He then used the compression exponent (σ), the specific heat ratio, and equation (7-7), shown below, to calculate the polytropic efficiency. The ratio of specific heats is given in Table 7-20.

$$\sigma = (k-1) \times 100/(k \times E) \tag{7-7}$$

Table 7-23 Calculated horsepower load versus suction pressure

Variable	Case 1	Case 2	Case 3
LS suction pressure, psig	3	4	5
psia	17.7	18.7	19.7
LS discharge pressure, psig	62	62	62
psia	76.7	76.7	76.7
HS suction pressure, psig	60	60	60
psia	74.7	74.7	74.7
HS discharge pressure, psig	275	275	275
psia	289.7	289.7	289.7
LS suction temperature, °F	100	100	100
°R	560	560	560
LS discharge temperature, °F	283	275	267
°R	743	735	727
HS suction temperature, °F	110	110	110
°R	570	570	570
HS discharge temperature, °F	280	280	280
°R	740	740	740
Compression exponent	0.193	0.193	0.193
Polytropic efficiency	90	90	90
Polytropic head	67400	65900	64500
Mass flow rate, lb/min	620	655	690
Total horsepower	1480	1523	1575

As shown in Table 7-22, the polytropic efficiency is slightly lower than the design efficiency (90 vs. 92). This slightly low efficiency is likely within the accuracy of the calculations. The mass flow and calculated total horsepower are also shown. The problem solver concluded that the hypothesis that the compressor required mechanical repairs was not a valid hypothesis. He then considered the question would process upsets such as a sudden increase in the recycle gas rate be sufficient to overload the compressor. He developed a spreadsheet to allow calculation of the horsepower load at various low-stage suction pressures. The results of these calculations are shown in Table 7-23.

The problem solver believed that his calculations clearly indicated that the reason for the mysterious compressor shutdowns was associated with a slight increase in process gas rates that caused the compressor suction to increase to the point at which the compressor motor was overloaded. This was in spite of the intuitive thought process that implied that the horsepower load should decrease as the compression ratio decreases. The 1500 BHP motor with a 3% overload factor would likely shutdown if the compressor suction pressure exceeded 4.5 psig. He expressed his hypothesis as follows:

> It is believed that the mysterious recycle compressor shutdowns are caused by spikes in the recycle gas rate that cause the motor to overload as the suction pressure is increased.

While it might seem that this hypothesis could be confirmed by a simple analysis of the low-stage suction pressure and compressor amperes versus time, the exact point of shutdown was not obvious. As the compressor suction pressure rose and the compressor shutdown, the pressure continued to rise rapidly. It was impossible to determine whether the shutdown occurred when the pressure reached 4, 4.5, 5, or 6 psig.

Step 4: Provide a mechanism to test the hypothesis.

The problem solver considered that the best option to test his hypothesis was to run a test at reduced production rates. A test was run at a low enough production rate that the recycle gas never got high enough to cause the suction pressure to go above 4 psig. During this test, there were no mysterious compressor shutdowns.

Step 5: Recommend remedial action to eliminate the problem without creating another problem.

While this test was successful at preventing the mysterious compressor shutdowns, it was obviously not a permanent solution. Experience with the process over the early startup period indicated that the recycle gas rate was likely to increase to 15% above the steady state design rate during decreases in reactor conversion. The problem solver looked at an increase of 15% in gas rate and concluded that the suction pressure would increase from 3 psig to 5.7 psig if this occurred. At 5.7 psig, the calculated horsepower load would be 1610 BHP. This would be well above the motor acceptable load with its 3% overload factor. He discussed the situation with the motor manufacturer and found out that operating the motor continuously at 10% overload would be expected to shorten the life of the motor by a slight amount. Since the 10% overload would only occur during times when the reactor conversion dropped, a larger overload switch was installed.

Lessons Learned While this problem solution may seem obvious, it should be recognized that the initial assessment was that there was a problem with the first out instrumentation system. If this idea had been followed through with no consideration of other possible hypotheses, there would have been a significant delay in solving the problem since the problem was thought to be an intermittent failure. In addition, while the 3% overload rating may seem too conservative, without the calculations done by the problem solver it would not be known whether the 10% overload rating would cover the range of loads to be expected. The calculations also helped to steer the problem solver away from the conclusion that there was a mechanical problem with the compressor.

This problem also illustrated that what might seem to be intuitively correct (lowering the compression ratio by increasing the suction pressure will decrease the horsepower load) is not always true. While it is true that lowering the compression ratio by increasing the suction pressure lowers the polytropic head, it also increases the mass flow rate. Both polytropic head and mass flow rate are important in determining the horsepower load. The exact relationship between suction pressure and horsepower load can only be determined by calculations.

NOMENCLATURE

BHP	Energy delivered to the fluid, horsepower
C	Residual gas remaining in the positive displacement compressor clearance pocket or rotating pocket after discharge, fraction of displacement volume
C_P	Fluid specific heat, BTU/lb-F°
D	Diameter of the pump cylinder, ft
D_P	Diameter of the suction pipe, ft
E	Hydraulic pump efficiency or adiabatic/polytropic compression efficiency, %
E_T	Turbine efficiency, %
E_V	Pump or compressor volumetric efficiency, %
F	Flow rate, lb/min
g_c	Gravitational constant
H	Pump head or adiabatic/polytropic head, ft
H_I	Inlet steam enthalpy, BTU/lb
H_O	Outlet steam enthalpy, BTU/lb
H_E	Outlet steam enthalpy with isentropic expansion, BTU/lb
$(h_A)_{MAX}$	Maximum acceleration head loss for a reciprocating pump, ft
K	A factor for reciprocating pumps that depends on the pump design
k	Ratio of specific heats, C_p/C_v
L	Leakage of gas around the piston or rotating element, fraction of displacement volume. It is roughly 2% of the compression ratio expressed as a fraction
L_S	Length of the pump stroke, ft
L_P	Actual suction pipe length, ft
lw	Frictional loss in the piping system
M	Gas molecular weight
n	Constant used to define the polytropic compression exponent $\sigma = (n - 1)/n$

P_D	Discharge pressure, psia
P_S	Suction pressure, psia
R	Compression ratio
S	Pump speed, RPM
T_D	Absolute discharge temperature, °R
T_S	Absolute suction temperature, °R
V	Reciprocating pump capacity, ft^3/min
V_P	Peak velocity in suction pipe of a reciprocating pump, ft/sec
w	Amount of work added by the prime mover
Z	Average (suction and discharge) compressibility
ΔP	Pressure difference between two points or pressure rise across a pump
ΔT_H	Joule Thompson cooling effect, °R
$\Delta(v^2)$	The difference in velocity squared between two points
Δz	Difference in elevation between two points
ρ	Pumped fluid density, lb/ft^3
σ	Polytropic or adiabatic compression exponent