SECTION 13
Compressors and Expanders

Compressors

Depending on application, compressors are manufactured as positive-displacement, dynamic, or thermal type (Fig. 13-2).

Positive displacement types fall in two basic categories: reciprocating and rotary.

The reciprocating compressor consists of one or more cylinders each with a piston or plunger that moves back and forth, displacing a positive volume with each stroke.

The diaphragm compressor uses a hydraulically pulsed flexible diaphragm to displace the gas.

**FIG. 13-1**

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AVR =</td>
<td>actual volumetric rate (m^3/h) (i.e. at process conditions)</td>
</tr>
<tr>
<td>(A_P)</td>
<td>cross sectional area of piston, ((mm^2))</td>
</tr>
<tr>
<td>(A_L)</td>
<td>cross sectional area of piston rod, ((mm^2))</td>
</tr>
<tr>
<td>(B_P)</td>
<td>brake or shaft power (kW)</td>
</tr>
<tr>
<td>(C)</td>
<td>cylinder clearance as a percent of cylinder volume</td>
</tr>
<tr>
<td>(C_p)</td>
<td>specific heat at constant pressure, (kJ/(kg \cdot °K))</td>
</tr>
<tr>
<td>(C_v)</td>
<td>specific heat at constant volume, (kJ/(kg \cdot °K))</td>
</tr>
<tr>
<td>(D)</td>
<td>cylinder inside diameter, (mm)</td>
</tr>
<tr>
<td>(d)</td>
<td>piston rod diameter, (mm)</td>
</tr>
<tr>
<td>(E_P)</td>
<td>extracted power of expander (kW)</td>
</tr>
<tr>
<td>(F)</td>
<td>an allowance for interstage pressure drop, Eq 13-4</td>
</tr>
<tr>
<td>(G_{hp})</td>
<td>gas power, actual compression power, (kW), excluding mechanical losses, (W)</td>
</tr>
<tr>
<td>(H)</td>
<td>head, (N \cdot M/kg)</td>
</tr>
<tr>
<td>(h)</td>
<td>enthalpy, (kJ/kg)</td>
</tr>
<tr>
<td>(IVR)</td>
<td>inlet volumetric rate (m^3/h), usually at suction conditions</td>
</tr>
<tr>
<td>(k)</td>
<td>isentropic exponent, (C_p/C_v)</td>
</tr>
<tr>
<td>(MC_p)</td>
<td>molar specific heat at constant pressure, (kJ/(kmole \cdot °K))</td>
</tr>
<tr>
<td>(MC_v)</td>
<td>molar specific heat at constant volume, (kJ/(kmole \cdot °K))</td>
</tr>
<tr>
<td>(M)</td>
<td>molecular mass</td>
</tr>
<tr>
<td>(N')</td>
<td>speed, (rpm)</td>
</tr>
<tr>
<td>(N_m)</td>
<td>molar flow, moles/h</td>
</tr>
<tr>
<td>(n)</td>
<td>polytropic exponent or number of moles</td>
</tr>
<tr>
<td>(P)</td>
<td>pressure, (kPa) (abs)</td>
</tr>
<tr>
<td>(P_c)</td>
<td>critical pressure, (kPa) (abs)</td>
</tr>
<tr>
<td>(PD)</td>
<td>piston displacement, (m^3/h)</td>
</tr>
<tr>
<td>(P_L)</td>
<td>pressure base used in the contract or regulation, (kPa) (abs)</td>
</tr>
<tr>
<td>(pP_c)</td>
<td>pseudo critical pressure, (kPa) (abs)</td>
</tr>
<tr>
<td>(P_{R})</td>
<td>reduced pressure, (P/P_c)</td>
</tr>
<tr>
<td>(pT_c)</td>
<td>pseudo critical temperature, (K)</td>
</tr>
<tr>
<td>(P_V)</td>
<td>partial pressure of contained moisture, (kPa) (abs)</td>
</tr>
<tr>
<td>(p)</td>
<td>pressure, (kg/m^2)</td>
</tr>
<tr>
<td>(Q)</td>
<td>inlet capacity (IVR) (m^3/h)</td>
</tr>
<tr>
<td>(R)</td>
<td>universal gas constant (= 8.314 \frac{kJ}{kmole \cdot K})</td>
</tr>
<tr>
<td>(S)</td>
<td>entropy, (kJ/(kg \cdot K)) or number of wheels</td>
</tr>
<tr>
<td>(SVR)</td>
<td>standard volumetric rate (m^3/h) measured at 101.35 kPa and 15°C</td>
</tr>
<tr>
<td>(stroke)</td>
<td>length of piston movement, (mm)</td>
</tr>
<tr>
<td>(T)</td>
<td>absolute temperature, (K)</td>
</tr>
<tr>
<td>(T_c)</td>
<td>critical temperature, (K)</td>
</tr>
<tr>
<td>(T_R)</td>
<td>reduced temperature, (T/T_c)</td>
</tr>
<tr>
<td>(t)</td>
<td>temperature, °C</td>
</tr>
<tr>
<td>(V)</td>
<td>specific volume, (m^3/kg)</td>
</tr>
<tr>
<td>(VE)</td>
<td>volumetric efficiency, percent</td>
</tr>
<tr>
<td>(W)</td>
<td>work, (N \cdot M)</td>
</tr>
<tr>
<td>(w)</td>
<td>weight flow, (kg/h)</td>
</tr>
<tr>
<td>(X)</td>
<td>temperature rise factor</td>
</tr>
<tr>
<td>(y)</td>
<td>mole fraction</td>
</tr>
<tr>
<td>(Z)</td>
<td>compressibility factor</td>
</tr>
<tr>
<td>(Z_{avg})</td>
<td>average compressibility factor (= \frac{(Z_s + Z_d)}{2})</td>
</tr>
<tr>
<td>(\eta)</td>
<td>efficiency, expressed as a decimal</td>
</tr>
</tbody>
</table>

**Subscripts**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>avg</td>
<td>average</td>
</tr>
<tr>
<td>d</td>
<td>discharge</td>
</tr>
<tr>
<td>is</td>
<td>isentropic process</td>
</tr>
<tr>
<td>L</td>
<td>standard conditions used for calculation or contract</td>
</tr>
<tr>
<td>p</td>
<td>polytropic process</td>
</tr>
<tr>
<td>S</td>
<td>standard conditions, usually 101.35 kPa (abs), 15°C</td>
</tr>
<tr>
<td>s</td>
<td>suction</td>
</tr>
<tr>
<td>t</td>
<td>total or overall</td>
</tr>
<tr>
<td>1</td>
<td>inlet conditions</td>
</tr>
<tr>
<td>2</td>
<td>outlet conditions</td>
</tr>
</tbody>
</table>
Rotary compressors cover lobe-type, screw-type, vane-type, and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other such as lobes or screws, or that displace a fixed volume with each rotation.

The dynamic types include radial-flow (centrifugal), axial-flow, and mixed flow machines. They are rotary continuous-flow compressors in which the rotating element (impeller or bladed rotor) accelerates the gas as it passes through the element, converting the velocity head into static pressure, partially in the rotating element and partially in stationary diffusers or blades.

Ejectors are "thermal" compressors that use a high velocity gas or steam jet to entrain the inflowing gas, then convert the velocity of the mixture to pressure in a diffuser.

Fig. 13-3 covers the normal range of operation for compressors of the commercially available types. Fig. 13-4 summarizes the difference between reciprocating and centrifugal compressors.

**RECIROCATING COMPRESSORS**

Reciprocating compressor ratings vary from fractional to more than 20,000 hp per unit. Pressures range from low vacuum at suction to 30,000 psi and higher at discharge for special process compressors.

Reciprocating compressors are furnished either single-stage or multi-stage. The number of stages is determined by the overall compression ratio. The compression ratio per stage (and valve life) is generally limited by the discharge temperature and usually does not exceed 4, although small-sized units (intermittent duty) are furnished with a compression ratio as high as 8.

Gas cylinders are generally lubricated, although a non-lubricated design is available when warranted; example: nitrogen, oxygen, and instrument air.

On multistage machines, intercoolers may be provided between stages. These are heat exchangers which remove the heat of compression from the gas and reduce its temperature to approximately the temperature existing at the compressor intake. Such cooling reduces the actual volume of gas going to the high-pressure cylinders, reduces the horsepower required for compression, and keeps the temperature within safe operating limits.

Reciprocating compressors should be supplied with clean gas as they cannot satisfactorily handle liquids and solid particles that may be entrained in the gas. Liquids and solid particles tend to destroy cylinder lubrication and cause excessive wear. Liquids are non-compressible and their presence could rupture the compressor cylinder or cause other major damage.

**Performance Calculations**

The engineer in the field is frequently required to:
1. determine the approximate horsepower required to compress a certain volume of gas from some intake conditions to a given discharge pressure, and
2. estimate the capacity of an existing compressor under specified suction and discharge conditions.

The following text outlines procedures for making these calculations from the standpoint of quick estimates and also presents more detailed calculations. For specific information on a given compressor, consult the manufacturer of that unit.

For a compression process, the enthalpy change is the best way of evaluating the work of compression. If a P-H diagram is available (as for propane refrigeration systems), the work of compression would always be evaluated by the enthalpy
change of the gas in going from suction to discharge conditions. Years ago the capability of easily generating P-H diagrams for natural gases did not exist. The result was that many ways of estimating the enthalpy change were developed. They were used as a crutch and not because they were the best way to evaluate compression horsepower requirements.

Today the engineer does have available, in many cases, the capability to generate that part of the P-H diagram required for compression purposes. This is done using equations of state on a computer. This still would be the best way to evaluate the compression horsepower. The other equations are used only if access to a good equation of state is not available.

Section 13 continues to treat reciprocating and centrifugal machines as being different so far as estimation of horsepower requirements is concerned. This treatment reflects industry practice. The only difference in the horsepower evaluation is the efficiency of the machine. Otherwise the basic thermodynamic equations are the same for all compression.

The reciprocating compressor horsepower calculations presented are based on charts. However, they may equally well be calculated using the equations in the centrifugal compressor section, particularly Eqs. 13-25 through 13-43. This also includes the mechanical losses in Eqs. 13-37 and 13-38.

There are two ways in which the thermodynamic calculations for compression can be carried out — by assuming:
1. isentropic reversible path — a process during which there is no heat added to or removed from the system and the entropy remains constant, $p v^k = \text{constant}$
2. polytropic reversible path — a process in which changes in gas characteristics during compression are considered, $p v^n = \text{constant}$

FIG. 13-3
Compressor Coverage Chart

FIG. 13-4
Comparison of Reciprocating and Centrifugal Compressors

The advantages of a centrifugal compressor over a reciprocating machine are:
- Lower installed first cost where pressure and volume conditions are favorable,
- Lower maintenance expense,
- Greater continuity of service and dependability,
- Less operating attention,
- Greater volume capacity per unit of plot area,
- Adaptability to high-speed low-maintenance-cost drivers.

The advantages of a reciprocating compressor over a centrifugal machine are:
- Greater flexibility in capacity and pressure range,
- Higher compressor efficiency and lower power cost,
- Capability of delivering higher pressures,
- Capability of handling smaller volumes,
- Less sensitive to changes in gas composition and density.
For a multi-component gas, the mole weighted average value of molar heat capacity must be determined at average cylinder temperature. A sample calculation is shown in Fig. 13-7.

The calculation of \( p_{Pc} \) and \( p_{Tc} \) in Fig. 13-7 permits calculation of the reduced pressure \( p_{r} = p/p_{Pc} \), mix and reduced temperature \( T_{r} = T/p_{Tc} \), mix. The compressibility \( Z \) at \( T \) and \( P \) can then be determined using the charts in Section 23.

If only the molecular weight of the gas is known and not its composition, an approximate value for \( k \) can be determined from the curves in Fig. 13-8.

### Estimating Compressor Horsepower

Eq 13-4 is useful for obtaining a quick and reasonable estimate for compressor horsepower. It was developed for large slow-speed (300 to 450 rpm) compressors handling gases with a specific gravity of 0.65 and having stage compression ratios above 2.5.

**CAUTION:** Compressor manufacturers generally rate their machines based on a standard condition of 14.4 psia rather than the more common gas industry value of 14.7 psia.

Due to higher valve losses, the horsepower requirement for high-speed compressors (1000 rpm range, and some up to 1800 rpm) can be as much as 20% higher, although this is a very arbitrary value. Some compressor designs do not merit a higher horsepower allowance and the manufacturers should be consulted for specific applications.

\[
\text{Brake horsepower} = (22)^\frac{\text{ratio}}{\text{stage}} (#\text{of stages})(\text{MMcfd})(F)
\]

**Eq 13-4**

Where:
- \( \text{MMcfd} \) = Compressor capacity referred to 14.4 psia and intake temperature
- \( F = 1.0 \) for single-stage compression
- \( 1.08 \) for two-stage compression
- \( 1.10 \) for three-stage compression

Eq 13-4 will also provide a rough estimate of horsepower for lower compression ratios and/or gases with a higher specific gravity, but it will tend to be on the high side. To allow for this tendency use a multiplication factor of 20 instead of 22 for gases with a specific gravity in the 0.8 to 1.0 range; likewise, use a factor in the range of 16 to 18 for compression ratios between 1.5 and 2.0.

Curves are available which permit easy estimation of approximate compression-horsepower requirements. Fig. 13-9 is typical of these curves.

**Example 13-1** — Compress 2 MMcfd of gas at 14.4 psia and intake temperature through a compression ratio of 9 in a 2-stage compressor. What will be the horsepower?

**Solution Steps**

- **Ratio per stage** = \( \sqrt[3]{9} = 3 \)
- From Eq 13-4 we find the brake horsepower to be:

\[
(22) (3) (2) (1.08) = 285 \text{ Bhp}
\]

From Fig. 13-9, using a \( k \) of 1.15, we find the horsepower requirement to be 136 Bhp/MMcfd or 272 Bhp. For a \( k \) of 1.4, the horsepower requirement would be 147 Bhp/MMcfd or 294 total horsepower.
The two procedures give reasonable agreement, particularly considering the simplifying assumptions necessary in reducing compressor horsepower calculations to such a simple procedure.

### Detailed Calculations

There are many variables which enter into the precise calculation of compressor performance. Generalized data as

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#### FIG. 13-6

**Molar Heat Capacity** \( MC_p \) (Ideal-Gas State), Btu/(lb mol \( \cdot °R \))

*Data source: Selected Values of Properties of Hydrocarbons, API Research Project 44; MW updated to agree with Fig. 23-2

#### FIG. 13-7

**Calculation of** \( k \)

<table>
<thead>
<tr>
<th>Component name</th>
<th>Mol fraction</th>
<th>Individual Mol weight</th>
<th>y, MW</th>
<th>Individual ( MC_p ) @ ( 150°F )</th>
<th>y ( \cdot MC_p )</th>
<th>Component critical pressure ( P_c ), psia</th>
<th>Component critical temperature ( T_c °R )</th>
<th>y ( \cdot T_c )</th>
</tr>
</thead>
<tbody>
<tr>
<td>methane</td>
<td>0.9216</td>
<td>16.04</td>
<td>8.23</td>
<td>8.42</td>
<td>8.46</td>
<td>10.71</td>
<td>13.14</td>
<td>16.34</td>
</tr>
<tr>
<td>ethane</td>
<td>0.0488</td>
<td>26.08</td>
<td>9.68</td>
<td>10.22</td>
<td>10.33</td>
<td>10.71</td>
<td>13.14</td>
<td>16.34</td>
</tr>
<tr>
<td>propane</td>
<td>0.0185</td>
<td>28.05</td>
<td>9.33</td>
<td>10.02</td>
<td>10.16</td>
<td>10.72</td>
<td>13.14</td>
<td>16.34</td>
</tr>
<tr>
<td>i-butane</td>
<td>0.0039</td>
<td>30.07</td>
<td>11.44</td>
<td>12.17</td>
<td>12.32</td>
<td>12.95</td>
<td>13.78</td>
<td>14.63</td>
</tr>
<tr>
<td>n-butane</td>
<td>0.0055</td>
<td>30.07</td>
<td>11.44</td>
<td>12.17</td>
<td>12.32</td>
<td>12.95</td>
<td>13.78</td>
<td>14.63</td>
</tr>
<tr>
<td>i-pentane</td>
<td>0.0017</td>
<td>42.08</td>
<td>13.63</td>
<td>14.69</td>
<td>14.90</td>
<td>15.75</td>
<td>16.80</td>
<td>17.85</td>
</tr>
</tbody>
</table>


---

*For values of \( MC_p \) other than @150°F, refer to Fig. 13-6*
from molar flow (Nm, mols/min)

\[ Q = \left( \frac{379.5 \cdot 14.7}{520} \right) \frac{N_m T_1 Z_1}{P_1 Z_L} \]  

Eq 13-9

From SCFM

\[ Q = \text{SCFM} \left( \frac{14.7}{520} \right) \frac{T_1 Z_1}{P_1 Z_L} \]  

Eq 13-10

From weight flow (w, lb/min)

\[ Q = \frac{10.73}{MW} \left( \frac{w T_1 Z_1}{P_1 Z_L} \right) \]  

Eq 13-11

Volumetric Efficiency

In a reciprocating compressor, the piston does not travel completely to the end of the cylinder at the end of the discharge stroke. Some clearance volume is necessary and it includes the space between the end of the piston and the cylinder head when the piston is at the end of its stroke. It also includes the volume in the valve ports, the volume in the suction valve guards, and the volume around the discharge valve seats.

Clearance volume is usually expressed as a percent of piston displacement and referred to as percent clearance, or cylinder clearance, C.

\[ C = \frac{\text{clearance volume, cu in.}}{\text{piston displacement, cu in.}} \]  

Eq 13-12

For double acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

The term "volumetric efficiency" refers to the actual pumping capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity of a cylinder can be represented by:

\[ VE = 100 - C \left( \frac{Z_2}{Z_1} (r^{1/k}) - 1 \right) \]  

Eq 13-13
Volumetric efficiencies as determined by Eq. 13-14 are theoretical in that they do not account for suction and discharge valve losses. The suction and discharge valves are actually spring-loaded check valves that permit flow in one direction only. The springs require a small differential pressure to open. For this reason, the pressure within the cylinder at the end of the suction stroke is lower than the line suction pressure and, likewise, the pressure at the end of the discharge stroke is higher than line discharge pressure.

One method for accounting for suction and discharge valve losses is to reduce the volumetric efficiency by an arbitrary amount, typically 4%, thus modifying Eq. 13-14 as follows:

\[
VE = 96 - r - C \left[ \frac{Z_s}{Z_d} \left( \frac{r_1}{k} \right)^k - 1 \right]
\]

Eq 13-15

When a non-lubricated compressor is used, the volumetric efficiency should be corrected by subtracting an additional 5% for slippage of gas. This is a capacity correction only and, as a first approximation, would not be considered when calculating compressor horsepower. The energy of compression is used by the gas even though the gas slips by the rings and is not discharged from the cylinder.

If the compressor is in propane, or similar heavy gas service, an additional 4% should be subtracted from the volumetric efficiency. These deductions for non-lubricated and propane performance are both approximate and, if both apply, cumulative.

Fig. 13-10 provides the solution to the function \( r^{1/k} \). Values for compression ratios not shown may be obtained by interpolation. The closest k value column may be safely used without a second interpolation.

Volumetric efficiencies for "high speed" separable compressors in the past have tended to be slightly lower than estimated from Eq 13-14. Recent information suggests that this modification is not necessary for all models of high speed compressors.

In evaluating efficiency, horsepower, volumetric efficiency, etc., the user should consider past experience with different
speeds and models. Larger valve area for a given swept volume will generally lead to higher compression efficiencies.

**Equivalent Capacity**

The net capacity for a compressor, in cubic feet per day @ 14.4 psia and suction temperature, may be calculated by Eq. 13-16a which is shown in dimensioned form:

\[
\text{MMcfd} = \frac{\text{PD} \cdot \text{VE} \cdot \text{Ps} \cdot 10^6}{\text{Z}_{104}}
\]

**Eq 13-16a**

which can be simplified to Eq. 13-16b when \(Z_{14.4}\) is assumed to equal 1.0.

\[
\text{MMcfd} = \frac{\text{PD} \cdot \text{VE} \cdot \text{Ps} \cdot 10^6}{\text{Zs}}
\]

**Eq 13-16b**

For example, a compressor with 200 cu ft/min piston displacement, a volumetric efficiency of 80%, a suction pressure of 75 psia, and suction compressibility of 0.9 would have a capacity of 1.33 MMcfd at 14.4 psia and suction temperature. If compressibility is not used as a divisor in calculating cu ft/min, then the statement "not corrected for compressibility" should be added.

In many instances the gas sales contract or regulation will specify some other measurement standard for gas volume. To convert volumes calculated using Equation 13-16 (i.e. at 14.4 psia and suction temperature) to a PL and TL basis, Eq 13-17 would be used:

\[
\text{MMscfd at PL, TL} = (\text{MMcfd from Eq 13-16}) \left( \frac{14.4}{\text{PL}} \right) \left( \frac{\text{TL}}{\text{Ts}} \right) \left( \frac{\text{ZL}}{\text{Zs}} \right)
\]

**Eq 13-17**

**Discharge Temperature**

The temperature of the gas discharged from the cylinder can be estimated from Eq 13-18, which is commonly used but not recommended. (Note: the temperatures are in absolute units, °R or K.) Eqs 13-31 and 13-32 give better results.

\[
T_d = T_s \left( r^{(k-1)/k} \right)
\]

**Eq 13-18**

Fig. 13-11 is a nomograph which can be used to solve Eq 13-18. The discharge temperature determined from either
Eq 13-18 or Fig. 13-11 is the theoretical value. While it neglects heat from friction, irreversibility effects, etc., and may be somewhat low, the values obtained from this equation will be reasonable field estimates.

**Rod Loading**

Each compressor frame has definite limitations as to maximum speed and load-carrying capacity. The load-carrying capacity of a compressor frame involves two primary considerations: horsepower and rod loading.

The horsepower rating of a compressor frame is the measure of the ability of the supporting structure and crankshaft to withstand torque (turning force) and the ability of the bearings to dissipate frictional heat. Rod loads are established to limit the static and inertial loads on the crankshaft, connecting rod, frame, piston rod, bolting, and projected bearing surfaces.

Good design dictates a reversal of rod loading during each stroke. Non-reversal of the loading results in failure to allow bearing surfaces to part and permit entrance of sufficient lubricant. The result will be premature bearing wear or failure.

Rod loadings may be calculated by the use of Eqs 13-19 and 13-20.

Load in compression = \( P_d A_p - P_s (A_p - A_r) \)

\[ = (P_d - P_s) A_p + P_s A_r \]

**Eq 13-19**

Load in tension = \( P_d (A_p - A_r) - P_s A_p \)

\[ = (P_d - P_s) A_p - P_d A_r \]

**Eq 13-20**

Using Eqs. 13-19 and 13-20, a plus value for the load in both compression and tension indicates a reversal of loads based on gas pressure only. Inertial effects will tend to increase the degree of reversal.

The true rod loads would be those calculated using internal cylinder pressures after allowance for valve losses. Normally, the operator will know only line pressures, and because of this, manufacturers generally rate their compressors based on line-pressure calculations.

A further refinement in the rod-loading calculation would be to include inertial forces. While the manufacturer will consider inertial forces when rating compressors, useful data on this point is seldom available in the field. Except in special cases, inertial forces are ignored.

A tail-rod cylinder would require consideration of rod cross-section area on both sides of the piston instead of on only one side of the piston, as in Eqs 13-19 and 13-20.

**Horsepower**

Detailed compressor horsepower calculations can be made through the use of Figs. 13-12 and 13-13. For ease of calculations, these figures provide net horsepower, including mechanical efficiency and gas losses. Figs. 13-14 and 13-15 are included for modifying the horsepower numbers for special conditions.

Proper use of these charts should provide the user with reasonably accurate horsepower requirements that are comparable to those calculated by the compressor manufacturer. For more detailed design, the engineer should consult a compressor manufacturer.

Volumes to be handled in each stage must be corrected to the actual temperature at the inlet to that stage. Note that moisture content corrections can also be important at low pressure and/or high temperature.

When intercoolers are used, allowance must be made for interstage pressure drop. Interstage pressures may be estimated by:

1. Obtaining the overall compression ratio, \( r_t \).
2. Obtaining the calculated ratio per stage, \( r_s \), by taking the \( s \) root of \( r_t \), where \( s \) is the number of compression stages.
3. Multiplying \( r_s \) by the absolute intake pressure of the stage being considered.

This procedure gives the absolute discharge pressure of this stage and the theoretical absolute intake pressure to the next stage. The next stage intake pressure can be corrected for intercooler pressure drop by reducing the pressure by 3-5 psi. This can be significant in low pressure stages.

Horsepower for compression is calculated by using Figs. 13-12 and 13-13 and Eq. 13-21.

\[ B_{hp} = (B_{hp/MMcfd}) \left( \frac{P_L}{14.4} \right) \left( \frac{T_s}{T_L} \right) (Z_{avg}) (MMcfd) \]

**Eq 13-21**

\( B_{hp/MMcfd} \) is read from Figs. 13-12 and 13-13 which use a pressure base of 14.4 psia.
FIG. 13-11
Theoretical Discharge Temperatures
Single-Stage Compression
Read r to k to $t_s$ to $t_d$

\[ T_d = T_s \left( \frac{P_d}{P_s} \right)^{\frac{k-1}{k}} \]

which is equivalent to

\[ T_d = t_s + 460 \left( \frac{P_d}{P_s} \right)^{\frac{k-1}{k}} - 460 \]

Note:
- Pressure drop across inlet and discharge valves is assumed to be nil. Allowance should be made for a higher-than-indicated compression ratio if this is not the case.
The curves predict the approximate performance of large slow speed (300-450 rpm) integral compressors. The horsepower requirement for higher speed machines should be adjusted based on the user's experience or manufacturer's data. For small high speed compressors (1000 rpm and higher), the horsepower adjustment could range from essentially zero up to an additional 10-12% (with a corresponding increase in discharge temperatures). In some cases, it may be as much as 20% greater.
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Figs. 13-12 and 13-13 are for standard valved cylinders. Caution should be used in applying conventional cylinders to low compression-ratio pipeline compressors. For low ratio pipeline compressors a high clearance type cylinder permits valve designs with higher efficiency. The compressor manufacturer should be consulted for Bhp curves on this type cylinder.

Fig. 13-14 provides a correction for intake pressure. The correction factor, as read from the curve, is used as a multiplier in the right hand side of Eq 13-21 to obtain the corrected brake horsepower.

Fig. 13-15 provides a correction factor for gas specific gravity. The correction factor is used as a multiplier in the right-hand side of Eq 13-21 to obtain the corrected horsepower.

Data presented in Figs. 13-12 and 13-13 are for slow speed integral compressors rather than the high speed separable compressors. To adjust the horsepower for the high speed unit, the values obtained from Figs. 13-12 and 13-13 may be increased by the following percentages:

<table>
<thead>
<tr>
<th>Gas Specific Gravity</th>
<th>Percent horsepower increase for high speed units</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5-0.8</td>
<td>4</td>
</tr>
<tr>
<td>0.9</td>
<td>5</td>
</tr>
<tr>
<td>1.0</td>
<td>6</td>
</tr>
<tr>
<td>1.1</td>
<td>8</td>
</tr>
<tr>
<td>1.5 and propane</td>
<td>10</td>
</tr>
<tr>
<td>refrigeration units</td>
<td></td>
</tr>
</tbody>
</table>

Because of variations by different manufacturers in specifying valve velocities for high speed as opposed to slow speed compressors, a given unit may differ from the horsepower corrections shown. Experience with compressors from a specific manufacturer will serve to guide the user and give confidence in utilization of the correction factors shown. For applications which are outside typical ranges discussed here, compressor manufacturers should be consulted.

**Example 13-2** — Compress 2 MMscfd of gas measured at 14.65 psia and 60°F. Intake pressure is 100 psia, and intake temperature is 100°F. Discharge pressure is 900 psia. The gas has a specific gravity of 0.80 (23 MW). What is the required horsepower?

1. Compression ratio is
   \[
   \frac{900 \text{ psia}}{100 \text{ psia}} = 9
   \]

   This would be a two-stage compressor; therefore, the ratio per stage is \( \sqrt{9} \) or 3.

2. \( 100 \text{ psia} \times 3 = 300 \text{ psia} \) (1st stage discharge pressure)
   \( 300 \text{ psia} \rightarrow 5 = 295 \text{ psia} \) (suction to 2nd stage)

   Where the 5 psia represents the pressure drop between first stage discharge and second stage suction.

   \[
   \frac{900 \text{ psia}}{295 \text{ psia}} = 3.05 \text{ (compression ratio for 2nd Stage)}
   \]

   It may be desirable to recalculate the interstage pressure to balance the ratios. For this sample problem, however, the first ratios determined will be used.
3. From Fig. 13-8 a gas with specific gravity of 0.8 at 150°F would have an approximate k of 1.21. For most compression applications, the 150°F curve will be adequate. This should be checked after determining the average cylinder temperature.

4. Discharge temperature for the 1st stage may be obtained by using Fig. 13-11 or solving Eq 13-18. For a compression ratio of 3, discharge temperature = approximately 220°F. Average cylinder temperature = 160°F.

5. In the same manner, discharge temperature for the second stage (with r = 3.05 and assuming interstage cooling to 120°F) equals approximately 244°F. Average cylinder temperature = 182°F.

6. From the physical properties section (Section 23), estimate the compressibility factors at suction and discharge pressure and temperature of each stage.

1st stage: \(Z_s = 0.98\)

\(Z_d = 0.97\)

\(Z_{avg} = 0.975\)

2nd stage: \(Z_s = 0.94\)

\(Z_d = 0.92\)

\(Z_{avg} = 0.93\)

7. From Fig. 13-12, Bhp/MMscfd at 3 compression ratio and a k of 1.21 is 63.5 (1st stage).

From Fig. 13-12, Bhp/MMscfd at 3.05 compression ratio and a k of 1.21 is 64.5 (second stage).

8. There are no corrections to be applied from Fig. 13-14 or 13-15, as all factors read unity.

9. Substituting in Eq 13-21:

1st stage:

\[
\text{Bhp/MMscfd} = 63.5 \left( \frac{14.65}{14.4} \right) \left( \frac{560}{520} \right) 0.975 = 67.8
\]

Bhp for 1st stage = 2 MMscfd \times 67.8 = 135.6

2nd stage:

\[
\text{Bhp/MMscfd} = 64.5 \left( \frac{14.65}{14.4} \right) \left( \frac{580}{520} \right) 0.93 = 68.1
\]

Bhp for 2nd stage = (2 MMscfd) (68.1) = 136.2

Total Bhp for this application = 135.6 + 136.2 = 271.8.

Note that in Example 13-1 the same conditions result in a compression power of 285 Bhp which is close agreement.

**Limits to compression ratio per stage** — The maximum ratio of compression permissible in one stage is usually limited by the discharge temperature or by rod loading, particularly in the first stage.

When handling gases containing oxygen, which could support combustion, there is a possibility of fire and explosion because of the oil vapors present.

To reduce carbonization of the oil and the danger of fires, a safe operating limit may be considered to be approximately 300°F. Where no oxygen is present in the gas stream, temperatures of 350°F may be considered as the maximum, even though mechanical or process requirements usually dictate a lower figure.

Packaging life may be significantly shortened by the dual requirement to seal both high pressure and high temperature gases. For this reason, at higher discharge pressures, a temperature closer to 250°F or 275°F may be the practical limit.

In summary, and for most field applications, the use of 300°F maximum would be a good average. Recognition of the above variables is, however, still useful.

Economic considerations are also involved because a high ratio of compression will mean a low volumetric efficiency and require a larger cylinder to produce the same capacity. For this reason a high rod loading may result and require a heavier and more expensive frame.

Where multi-stage operation is involved, equal ratios of compression per stage are used (plus an allowance for piping and cooler losses if necessary) unless otherwise required by process design. For two stages of compression the ratio per stage would approximately equal the square root of the total compression ratio; for three stages, the cube root, etc. In practice, especially in high-pressure work, decreasing the compression ratio in the higher stages to reduce excessive rod loading may prove to be advantageous.

**Cylinder Design**

Depending on the size of the machine and the number of stages, reciprocating compressors are furnished with cylinders fitted with either single- or double-acting pistons, see examples in Figs. 13-16 through 13-18.

In the same units, double-acting pistons are commonly used in the first stages and often single-acting in the higher stages of compression.

Cylinder materials are normally selected for strength; however, thermal shock, mechanical shock, or corrosion resistance may also be a determining factor. The table below shows discharge pressure limits generally used in the gas industry for cylinder material selection.

<table>
<thead>
<tr>
<th>Cylinder Material</th>
<th>Discharge Pressure (psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast Iron</td>
<td>up to 1,200</td>
</tr>
<tr>
<td>Nodular Iron</td>
<td>about 1,500</td>
</tr>
<tr>
<td>Cast Steel</td>
<td>1,200 to 2,500</td>
</tr>
<tr>
<td>Forged Steel</td>
<td>above 2,500</td>
</tr>
</tbody>
</table>

API standard 618 recommends 1000 psig as the maximum pressure for both cast iron and nodular iron.

Cylinders are designed both as a solid body (no liner) and with liners. Cylinder liners are inserted into the cylinder body to either form or line the pressure wall. There are two types. The wet liner forms the pressure wall as well as the inside wall of the water jacket. The dry type lines the cylinder wall and is not required to add strength.

Standard cylinder liners are cast iron. If cylinders are required to have special corrosion or wear resistance, other materials or special alloys may be needed.

Most compressors use oils to lubricate the cylinder with a mechanical, force-feed lubricator having one or more feeds to each cylinder.

The non-lubricated compressor has found wide application where it is desirable or essential to compress air or gas without contaminating it with lubricating oil.
For such cases a number of manufacturers furnish a "non-lubricated" cylinder (Fig. 13-19). The piston on these cylinders is equipped with piston rings of graphitic carbon or plastic as well as pads or rings of the same material to maintain the proper clearance between the piston and cylinder. Plastic packing of a type that requires no lubricant is used on the stuffing box. Although oil-wiper rings are used on the piston rod where it leaves the compressor frame, minute quantities of oil might conceivably enter the cylinder on the rod. Where even such small amounts of oil are objectionable, an extended cylinder connecting piece can be furnished. This simply lengthens the piston rod so that no lubricated portion of the rod enters the cylinder.

A small amount of gas leaking through the packing can be objectionable. Special distance pieces are furnished between the cylinder and frame, which may be either single-compartment or double-compartment. These may be furnished gas tight and vented back to the suction, or may be filled with a sealing gas or fluid and held under a slight pressure, or simply vented.

Compressor valves for non-lubricated service operate in an environment that has no lubricant in the gas or in the cylinder. Therefore, the selection of valve materials is important to prevent excessive wear.

Piston rod packing universally used in non-lubricated compressors is of the full-floating mechanical type, consisting of a case containing pairs of either carbon or plastic (TFE) rings of conventional design.

When handling oxygen and other gases such as nitrogen and helium, it is absolutely necessary that all traces of hydrocarbons in cylinders be removed. With oxygen, this is required for safety, with other gases to prevent system contamination. Deoiling schemes are discussed in Refrigeration, Section 14.

High-pressure compressors with discharge pressures from 5,000 to 30,000 psi usually require special design and a complete knowledge of the characteristics of the gas.

As a rule, inlet and discharge gas pipe connections on the cylinder are fitted with flanges of the same rating for the following reasons:
Practicality and uniformity of casting and machinery,
Hydrostatic test, usually at 150% design pressure,
Suction pulsation bottles are usually designed for the same pressure as the discharge bottle (often federal, state, or local government regulation).

Reciprocating Compressor Control Devices

Output of compressors must be controlled (regulated) to match system demand.

In many installations some means of controlling the output of the compressor is necessary. Often constant flow is required despite variations in discharge pressure, and the control device must operate to maintain a constant compressor capacity. Compressor capacity, speed, or pressure may be varied in accordance with the requirements. The nature of the control device will depend on the regulating variable — whether pressure, flow, temperature, or some other variable — and on type of compressor driver.

Unloading for Starting — Practically all reciprocating compressors must be unloaded to some degree before starting so that the driver torque available during acceleration is not exceeded. Both manual and automatic compressor startup unloading is used. Common methods of unloading include: discharge venting, discharge to suction bypass, and holding open the inlet valves using valve lifters.

Capacity Control — The most common requirement is regulation of capacity. Many capacity controls, or unloading devices, as they are usually termed, are actuated by the pressure on the discharge side of the compressor. A falling pressure indicates that gas is being used faster than it is being compressed and that more gas is required. A rising pressure indicates that more gas is being compressed than is being used and that less gas is required.

A common method of controlling the capacity of a compressor is to vary the speed. This method is applicable to steam-driven compressors and to units driven by internal combustion engines. In these cases the regulator actuates the steam-admission or fuel-admission valve on the compressor driver to control the speed.

Electric motor-driven compressors usually operate at constant speed, and other methods of controlling the capacity are necessary. On reciprocating compressors up to about 100 hp, two types of control are usually available. These are automatic-start-and-stop control and constant-speed control.

Automatic-start-and-stop control, as its name implies, stops or starts the compressor by means of a pressure-actuated switch as the gas demand varies. It should be used only when the demand for gas will be intermittent.
Constant-speed control permits the compressor to operate at full speed continuously, loaded part of the time and fully or partially unloaded at other times. Two methods of unloading the compressor with this type of control are in common use: inlet-valve unloaders, and clearance unloaders. Inlet-valve unloaders (Fig. 13-20) operate to hold the compressor inlet valves open and thereby prevent compression. Clearance unloaders (Fig. 13-21) consist of pockets or small reservoirs which are opened when unloading is desired. The gas is compressed into them on the compression stroke and expands into the cylinder on the return stroke, reducing the intake of additional gas.

Motor-driven reciprocating compressors above 100 hp in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load.

Five-step control (full load, three-quarter load, one-half load, one-quarter load, and no load) is accomplished by means of clearance pockets. On some makes of machines inlet-valve and clearance control unloading are used in combination.

A common practice in the natural gas industry is to prepare a single set of curves for a given machine unless there are side loads or it is a multi-service machine.

**FIG. 13-20**

Inlet Valve Unloader

![Inlet Valve Unloader](Courtesy of McGraw-Hill Book Co.)

**FIG. 13-21**

Pneumatic Valves Controlling Four Fixed Pockets in Compressor for Five-Step Control

![Pneumatic Valves Controlling Four Fixed Pockets in Compressor for Five-Step Control](Courtesy McGraw-Hill Book Co.)

Fig. 13-22 shows indicator cards which demonstrate the unloading operation for a double acting cylinder at three capacity points. The letters adjacent to the low-pressure diagrams represent the unloading influence of the respective and cumulative effect of the various pockets as identified in Fig. 13-21. Full load, one-half, and no load capacity is obtained by holding corresponding suction valves open or adding sufficient clearance to produce a zero volumetric efficiency. No-capacity operation includes holding all suction valves open.

Fig. 13-23 shows an alternative representation of compressor unloading operation with a step-control using fixed volume clearance pockets. The curve illustrates the relationship between compressor capacity and driver capacity for a varying compressor suction pressure at a constant discharge pressure and constant speed. The driver can be a gas engine or electric motor.

The purpose of this curve is to determine what steps of unloading are required to prevent the driver and piston rods from serious overloading. All lines are plotted for a single stage compressor.

The driver capacity line indicates the maximum allowable capacity for a given horsepower. The cylinder capacity lines represent the range of pressures calculated with all possible combinations of pockets open and cylinder unloading, as necessary, to cover the capacity of the driver.

Starting at the end (line 0-0) with full cylinder capacity, the line is traced until it crosses the driver capacity line at which point it is dropped to the next largest cylinder capacity and follow until it crosses the driver line, etc. This will produce a "saw tooth" effect, hence the name "saw tooth" curve. The number of "teeth" depends upon the number of combinations of pockets (opened or closed) required for unloading.

The same method is followed for multi-stage units. For each additional stage another "saw tooth" curve must be constructed, i.e., for a two stage application, two curves are required to attain the final results.

**FIG. 13-22**

Indicator Diagram for Three Load Points of Operation

![Indicator Diagram for Three Load Points of Operation](Courtesy McGraw-Hill Book Co.)
Although control devices are often automatically operated, manual operation is satisfactory for many services. Where manual operation is provided, it often consists of a valve, or valves, to open and close clearance pockets. In some cases, a movable cylinder head is provided for variable clearance in the cylinder (Fig. 13-24).

Gas Pulsation Control

Pulsation is inherent in reciprocating compressors because suction and discharge valves are open during only part of the stroke.

Pulsation must be damped (controlled) in order to:

a. provide smooth flow of gas to and from the compressor,

b. prevent overloading or underloading of the compressors, and

c. reduce overall vibration.

There are several types of pulsation chambers. The simplest one is a volume bottle, or a surge drum, which is a pressure vessel, unbaffled internally and mounted on or very near a cylinder inlet or outlet.

A manifold joining the inlet and discharge connections of cylinders operating in parallel can also serve as a volume bottle.

Performance of volume bottles is not normally guaranteed without an analysis of the piping system from the compressor to the first process vessel.

Volume bottles are sized empirically to provide an adequate volume to absorb most of the pulsation. Several industry methods were tried in an effort to produce a reasonable rule-of-thumb for their sizing. Fig. 13-25 may be used for approximate bottle sizing.

Example 13-3

Indicated suction pressure = 600 psia
Indicated discharge pressure = 1,400 psia
Cylinder bore = 6 in.
Cylinder stroke = 15 in.
Swept volume = \( \pi (6^2/4) (15) = 424 \) cu in.

From Fig. 13-25:

At 600 psi inlet pressure, the suction bottle multiplier is approximately 7.5. Suction-bottle volume = (7.5) (424) = 3,180 cu in.
At 1,400 psi discharge pressure, the discharge bottle multiplier is approximately 8.5. Discharge-bottle volume = (8.5) (424) = 3,600 cu in.

NOTE: When more than one cylinder is connected to a bottle, the sum of the individual swept volumes is the size required for the common bottle.

For more accurate sizing, compressor manufacturers can be consulted. Organizations which provide designs and/or equipment for gas-pulsation control are also available.

FIG. 13-23

"Saw Tooth" Curve for Unloading Operation

![Diagram](image)

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FIG. 13-24

Sectional View of a Cylinder Equipped with a Hand-Operated Valve Lifter and Variable-Volume Clearance

![Diagram](image)
Having determined the necessary volume of the bottle, the proportioning of diameter and length to provide this volume requires some ingenuity and judgment. It is desirable that manifolds be as short and of as large diameter as is consistent with pressure conditions, space limitations, and appearance.

A good general rule is to make the manifold diameter 1-1/2 times the inside diameter of the largest cylinder connected to it, but this is not always practicable, particularly where large cylinders are involved.

Inside diameter of pipe must be used in figuring manifolds. This is particularly important in high-pressure work and in small sizes where wall thickness may be a considerable percentage of the cross sectional area. Minimum manifold length is determined from cylinder center distances and connecting pipe diameters. Some additions must be made to the minimum thus determined to allow for saddle reinforcements and for welding of caps.

It is customary to close the ends of manifolds with welding caps which add both volume and length. Fig. 13-26 gives approximate volume and length of standard caps.

**Pulsation Dampeners (Snubbers)**

A pulsation dampener is an internally-baffled device. The design of the pulsation damping equipment is based on acoustical analog evaluation which takes into account the specified operating speed range, conditions of unloading, and variations in gas composition.

Analog evaluation is accomplished with an active analog that simulates the entire compressor, pulsation dampeners, piping and equipment system and considers dynamic interactions among these elements.

Pulsation dampeners also should be mounted as close as possible to the cylinder, and in large volume units, nozzles should be located near the center of the chamber to reduce unbalanced forces.

Pulsation dampeners are typically guaranteed for a maximum residual peak-to-peak pulsation pressure of 2% of average absolute pressure at the point of connection to the piping system, and pressure drop through the equipment of not more than 1% of the absolute pressure. This applies at design condition and not necessarily for other operating pressures and flows. A detailed discussion of recommended design approaches for pulsation suppression devices is presented in API.
<table>
<thead>
<tr>
<th>Trouble</th>
<th>Probable Cause(s)</th>
</tr>
</thead>
</table>
| **Compressor Will not Start** | 1. Power supply failure.  
2. Switchgear or starting panel.  
3. Low oil pressure shut down switch.  
4. Control panel. |
| **Motor Will Not Synchronize** | 1. Low voltage.  
2. Excessive starting torque.  
3. Incorrect power factor.  
4. Excitation voltage failure. |
| **Low Oil Pressure** | 1. Oil pump failure.  
2. Oil foaming from counterweights striking oil surface.  
3. Cold oil.  
4. Dirty oil filter.  
5. Interior frame oil leaks.  
6. Excessive leakage at bearing shim tabs and/or bearings.  
7. Improper low oil pressure switch setting.  
8. Low gear oil pump by-pass/relief valve setting.  
9. Defective pressure gauge.  
10. Plugged oil sump strainer.  
11. Defective oil relief valve. |
| **Noise In Cylinder** | 1. Loose piston.  
2. Piston hitting outer head or frame end of cylinder.  
3. Loose crosshead lock nut.  
4. Broken or leaking valve(s).  
5. Worn or broken piston rings or expanders.  
6. Valve improperly seated/damaged seat gasket.  
7. Free air unloader plunger chattering. |
| **Excessive Packing Leakage** | 1. Worn packing rings.  
2. Improper lube oil and/or insufficient lube rate (blue rings).  
3. Dirt in packing.  
4. Excessive rate of pressure increase.  
5. Packing rings assembled incorrectly.  
6. Improper ring side or end gap clearance.  
7. Plugged packing vent system.  
8. Scored piston rod.  
| **Packing Over-Heating** | 1. Excessive lube oil.  
2. Improper lube oil (too light, high carbon residue).  
3. Insufficient lube rate.  
4. Oil carryover from inlet system or previous stage.  
5. Excessive temperature due to high pressure ratio across cylinders. |
| **Excessive Carbon On Valves** | 1. Faulty relief valve.  
2. Leaking suction valves or rings on next higher stage.  
3. Obstruction (foreign material, rags), blind or valve closed in discharge line. |
| **Relief Valve Popping** | 1. Excessive ratio on cylinder due to leaking inlet valves or rings on next higher stage.  
2. Fouled intercooler/piping.  
3. Leaking discharge valves or piston rings.  
4. High inlet temperature.  
5. Fouled water jackets on cylinder.  
6. Improper lube oil and/or lube rate. |
| **High Discharge Temperature** | 1. Loose crosshead pin, pin caps or crosshead shoes.  
2. Loose/worn main, crankpin or crosshead bearings.  
3. Low oil pressure.  
4. Cold oil.  
5. Incorrect oil.  
6. Knock is actually from cylinder end. |
| **Frame Knocks** | 1. Faulty seal installation.  
2. Clogged drain hole. |
| **Crankshaft Oil Seal Leaks** | 1. Worn scraper rings.  
2. Scrapers incorrectly assembled.  
3. Worn/scored rod.  
4. Improper fit of rings to rod/side clearance. |
| **Piston Rod Oil Scraper Leaks** | 1. Lubrication failure.  
2. Improper lube oil and/or insufficient lube rate.  
3. Insufficient cooling. |

Courtesy of Ingersoll-Rand Co.
Troubleshooting

Minor troubles can normally be expected at various times during routine operation of the compressor. These troubles are most often traced to dirt, liquid, and maladjustment, or to operating personnel being unfamiliar with functions of the various machine parts and systems. Difficulties of this type can usually be corrected by cleaning, proper adjustment, elimination of an adverse condition, or quick replacement of a relatively minor part.

Major trouble can usually be traced to long periods of operation with unsuitable coolant or lubrication, careless operation and routine maintenance, or the use of the machine on a service for which it was not intended.

A defective inlet valve can generally be found by feeling the valve cover. It will be much warmer than normal. Discharge valve leakage is not as easy to detect since the discharge is always hot. Experienced operators of water-cooled units can usually tell by feel if a particular valve is leaking. The best indication of discharge valve trouble is the discharge temperature. This will rise, sometimes rapidly, when a valve is in poor condition or breaks. This is one very good reason for keeping a record of the discharge temperature from each cylinder.

Recording of the interstage pressure on multistage units is valuable because any variation, when operating at a given load point, indicates trouble in one or the other of the two stages. If the pressure drops, the trouble is in the low pressure cylinder. If it rises, the problem is in the high pressure cylinder.

Troubleshooting is largely a matter of elimination based on a thorough knowledge of the interrelated functions of the various parts and the effects of adverse conditions. A complete list of possible troubles with their causes and corrections is impractical, but the following list of the more frequently encountered troubles and their causes is offered as a guide (Fig. 13-27).

CENTRIFUGAL COMPRESSORS

This section is intended to supply information sufficiently accurate to determine whether a centrifugal compressor should be considered for a specific job. The secondary objective is to present information for evaluating compressor performance.

Fig. 13-28 gives an approximate idea of the flow range that a centrifugal compressor will handle. A multi-wheel (multi-stage) centrifugal compressor is normally considered for inlet volumes between 500 and 200,000 inlet acfm. A single-wheel (single-stage) compressor would normally have application between 100 and 150,000 inlet acfm. A multiwheel compressor can be thought of as a series of single wheel compressors contained in a single casing.

Most centrifugal compressors operate at speeds of 3,000 rpm or higher, a limiting factor being impeller stress considerations as well as velocity limitation of 0.8 to 0.85 Mach number at the impeller tip and eye. Recent advances in machine design have resulted in production of some units running at speeds in excess of 40,000 rpm.

Centrifugal compressors are usually driven by electric motors, steam or gas turbines (with or without speed-increasing gears), or turboexpanders.

There is an overlap of centrifugal and reciprocating compressors on the low end of the flow range, see Fig. 13-3. On the higher end of the flow range an overlap with the axial compressor exists. The extent of this overlap depends on a number of things. Before a technical decision could be reached as to the type of compressor that would be installed, the service, operational requirements, and economics would have to be considered.

Performance Calculations

The operating characteristics must be determined before an evaluation of compressor suitability for the application can be made. Fig. 13-29 gives a rough comparison of the characteristics of the axial, centrifugal, and reciprocating compressor.

The centrifugal compressor approximates the constant head-variable volume machine, while the reciprocating is a constant volume-variable head machine. The axial compressor, which is a low head, high flow machine, falls somewhere in between. A compressor is a part of the system, and its performance is dictated by the system resistance. The desired system capability or objective must be determined before a compressor can be selected.

Fig. 13-30 is a typical performance map which shows the basic shape of performance curves for a variable-speed centrifugal compressor. The curves are affected by many variables, such as desired compression ratio, type of gas, number of wheels, sizing of compressor, etc.

With variable speed, the centrifugal compressor can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination variable capacity and variable pressure.

Basically the performance of the centrifugal compressor, at speeds other than design, is such that the capacity will vary directly as the speed, the head developed as the square of the speed, and the required horsepower as the cube of the speed. As the speed deviates from the design speed, the error of these rules, known as the affinity laws, or fan laws, increases. The fan laws only apply to single stages or multi-stages with low compression ratios or very low Mach numbers.

Fan Laws:

$$Q \propto N; \text{ i.e., } Q_{110} = \frac{Q_{100}}{N_{110}} = \frac{Q_{90}}{N_{100}} = \frac{Q_{90}}{N_{100}} = \frac{Q_{90}}{N_{100}}$$

Eq 13-22

$$H \propto N^2; \text{ i.e., } H_{110} = \frac{H_{100}}{N_{110}} = \frac{H_{90}}{N_{100}} = \frac{H_{90}}{N_{100}}$$

Eq 13-23

$$Bhp \propto N^3; \text{ i.e., } Bhp_{110} = \frac{Bhp_{100}}{N_{110}} = \frac{Bhp_{90}}{N_{100}} = \frac{Bhp_{90}}{N_{100}}$$

Eq 13-24

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

Fig. 13-30 depicts typical performance curves with a small compression ratio. The system resistance has been superimposed on the chart: Line A represents typical system resistance of a closed system, such as a refrigeration unit where there is a relatively constant discharge pressure. Line B is an open-end system, such as pipeline application where pressure increases with capacity.

Fig. 13-31 shows a higher compression ratio. The range of stable operation is reduced because of the larger compression ratio. This is indicated by the surge line in Fig. 13-31 being further to the right than in Fig. 13-30.

Estimating Performance

Figs. 13-32 through 13-39 may be used for estimating compressor performance. These curves are suitable for estimating only and are not intended to take the place of a "wheel-by-wheel" selection by the compressor manufacturer, nor should the curves be used to calculate performance using field data in an attempt to determine a variance from predicted performance based on manufacturer's data. Fig. 13-32 is used to convert scfm to icfm. All centrifugal compressors are based on flows that are converted to inlet or actual cubic feet per minute. This is done because the centrifugal wheel is sensitive to inlet volume, compression ratio (i.e., head), and specific speed.

Fig. 13-33 is a useful curve to find inlet (actual) cfm when the weight flow in lb/min is known. Actual cfm and inlet cfm both denote the gas at suction conditions. These terms are often used interchangeably. This curve can be used in reverse to determine mass flow.

Fig. 13-34 is used to determine the approximate discharge temperature that is produced by the compression ratio. Discharge temperatures above the 400°F range should be checked since mechanical problems as well as safety problems may exist. This curve includes compressor efficiencies in the range of 60 to 75%.

Example 13-4 — Given: \( r = 10.0; Q_1 = 10,000 \text{ icfm} \)
\[ k = 1.15; t_1 = 0°F \]

Find: Discharge temperature

Answer: \( t_2 = 230°F \) (approximately) from Fig. 13-34.

Fig. 13-36 gives the approximate horsepower required for the compression. It includes compressor efficiencies in the range of 60 to 70%.

Example 13-5 — Given: Weight flow, \( w = 1,000 \text{ lb/min} \)
\[ \text{head} = 70,000 \text{ ft-lb/lb} \]

Find: Horsepower

Answer: \( \text{Ghp} = 3,000 \) from Fig. 13-36.

Fig. 13-39 predicts the approximate number of compressor wheels required to produce the head. If the number of wheels is not a whole number, use the next highest number.

Calculating Performance

When more accurate information is required for compressor head, gas horsepower, and discharge temperature, the equations in this section should be used. This method applies to a gas mixture for which a P-T diagram chart is not available. To calculate the properties of the gas, see Figs. 13-7 and 13-8. All values for pressure and temperature in these calculation procedures are the absolute values. Unless otherwise specified, volumes of flow in this section are actual volumes.

To calculate the inlet volume:
\[
Q = \frac{(w)(1,545)(T_1)(Z_1)}{(MW)(P_1)(144)} \quad \text{Eq 13-25}
\]

If we assume the compression to be isentropic (reversible adiabatic, constant entropy), then:
\[
H_{is} = \frac{ZRT}{MW(k-1)}/k \left[ \left( \frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \quad \text{Eq 13-26}
\]

Since these calculations will not be wheel-by-wheel, the head will be calculated across the entire machine. For this, use the average compressibility factor:
\[
Z_{avg} = \frac{Z_1 + Z_2}{2}
\]

The heat capacity ratio, \( k \), is normally determined at the average suction and discharge temperature (see Figs. 13-7 and 13-8).
Isentropic Calculation

To calculate the head:

\[ H_{is} = \frac{Z_{avg}RT_1}{MW} \left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \]  
\[ \text{Eq 13-27a} \]

which can also be written in the form:

\[ H_{is} = \frac{1545}{MW} \left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \]  
\[ \text{Eq 13-27b} \]

The gas horsepower can now be calculated from:

\[ G_{hp} = \frac{(w)(H_{is})}{\eta_{is} (33,000)} \]  
\[ \text{Eq 13-28} \]

The approximate theoretical discharge temperature can be calculated from:

\[ \Delta T_{ideal} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \]  
\[ \text{Eq 13-29} \]

\[ T_2 = T_1 + \Delta T_{ideal} \]  
\[ \text{Eq 13-30} \]

The actual discharge temperature can be approximated:

\[ \Delta T_{actual} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1 \]  
\[ \frac{1}{\eta_p} \]  
\[ \text{Eq 13-31} \]

\[ T_2 = T_1 + \Delta T_{actual} \]  
\[ \text{Eq 13-32} \]

Polytropic Calculation

Sometimes compressor manufacturers use a polytropic path instead of isentropic. Polytropic efficiency is defined by:

\[ n \left(\frac{n}{n-1}\right) = \frac{k}{(k-1)} \eta_p \]  
\[ \text{Eq 13-33} \]

(See Fig. 13-37 for conversion of isentropic efficiency to polytropic efficiency.)

The equations for head and gas horsepower based upon polytropic compression are:

\[ H_p = \frac{Z_{avg}RT_1}{MW} \left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1 \]  
\[ \text{Eq 13-34a} \]

which also can be written in the form:
FIG. 13-32
Actual Inlet Cubic Feet Per Minute
$Z = 1$

Example:

Given:
- Standard cfm = 60,000
- Pressure ($P_i$) = 100 psia
- Temperature ($t_i$) = 100°F

Find:
Actual inlet cfm

Answer:
9,700 ICFM
**FIG. 13-33**

Actual Inlet Cubic Feet Per Minute

\[ Z = 1 \]

---

**Example:**

**Given:**
- Weight flow (\( w \)) = 3600 lb/min
- Pressure (\( P_h \)) = 100 psia
- Temperature (\( T_h \)) = 100°F
- Molecular weight = 5

**Find:**
- Actual ICFM

**Answer:**
43,000 ICFM (approximately)
Example:

Given:
- Compression ratio, \( r = 10 \)
- \( Q_1 = 10,000 \text{ cfm} \)
- \( k = 1.15 \)
- Inlet temperature, \( t_1 = 0^\circ F \)

Find:
Discharge temperature

Answer:
\( t_2 = 230^\circ F \) (Approximately)
FIG. 13-35

Head

Z = 1

Example:

Given:
- Compression ratio = 10
- k = 1.15
- Inlet temperature = -50
- Molecular weight = 42

Find:
- Head

Answer:
- 39,000 ft lb/ lb (approximate)
FIG. 13-36
Horsepower Determination

Example: Given:
Weight flow, \( w \) = 1,000 lb/min
Head = 70,000 ft-lb/lb

Find:
Horsepower

Answer:
Hp = 3,000

FIG. 13-37
Efficiency Conversion
Mechanical Losses

<table>
<thead>
<tr>
<th>Casing Size</th>
<th>Max Flow (inlet acfm)</th>
<th>Nominal Speed (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7,500</td>
<td>10,500</td>
</tr>
<tr>
<td>2</td>
<td>20,000</td>
<td>8,200</td>
</tr>
<tr>
<td>3</td>
<td>33,000</td>
<td>6,400</td>
</tr>
<tr>
<td>4</td>
<td>55,000</td>
<td>4,900</td>
</tr>
<tr>
<td>5</td>
<td>115,000</td>
<td>3,600</td>
</tr>
<tr>
<td>6</td>
<td>150,000</td>
<td>2,800</td>
</tr>
</tbody>
</table>

**HP**

\[
H_p = \frac{1545 \ Z_{\text{acfm}} T_1}{\text{MW}} \left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \quad \text{Eq 13-34b}
\]

\[
G_{hp} = \left( \frac{w}{\eta_p} \right) \left( \frac{33,000}{H_p} \right) \quad \text{Eq 13-35}
\]

**Mechanical Losses**

After the gas horsepower has been determined by either method, horsepower losses due to friction in bearings, seals, and speed increasing gears must be added.

**FIG. 13-38**

Compressor Speed

The basic equation for estimating the speed of a centrifugal compressor is:

\[
N = \left( \frac{N_{\text{nominal}}}{\sqrt{s}} \right) \left( \frac{H_{\text{total}}}{H_{\text{max/stage}}} \right) \quad \text{Eq 13-39}
\]

where the number of wheels is determined from Fig. 13-39.

Nominal speeds to develop 10,000 feet of head/stage can be determined from Fig. 13-28. However, to calculate the maximum head per stage, the following equation based on molecular weight (or more accurately, density) can be used.

\[
H_{\text{max/stage}} = 15,000 - 1,500 \times \left( \frac{\text{MW}}{30} \right)^{0.35} \quad \text{Eq 13-40}
\]

This equation will give a head of 10,000 ft for a gas when \( \text{MW} = 30 \) and 11,000 ft when \( \text{MW} = 16 \).

**P-H Diagram**

When a P-H diagram is available for the gas to be compressed, the following procedure should be used. Fig. 13-40 represents a section of a typical P-H diagram.

**FIG. 13-39**

For the given inlet conditions, the enthalpy can be shown as point 1 on the P-H diagram. Starting from Point 1 follow the line of constant entropy to the required discharge pressure \( P_2 \), locating the isentropic discharge state point \( \left( 2, s \right) \). With these two points located the differential isentropic enthalpy can be calculated from the following equation:

\[
\Delta h_{\text{is}} = h_{2s} - h_1 \quad \text{Eq 13-41}
\]

To convert to isentropic head, the equation is:

\[
H_s = \Delta h_{\text{is}} \left( 778 \text{ ft} \cdot \text{lb/lb} \right) \quad \text{Eq 13-42}
\]

To find the discharge enthalpy:
\[ h_2 = \frac{\Delta h_{ls}}{\eta_s} + h_1 \quad \text{Eq 13-43} \]

The actual discharge temperature can now be obtained from the P-H diagram. The gas horsepower can be calculated using Eq 13-28 and Eq 13-35.

From Fig. 13-39 and Eqs. 13-39 and 13-40, the speed and number of wheels can be estimated.

To convert to polytropic head it will be necessary to assume a polytropic efficiency. See Fig. 13-28 for an efficiency corresponding to the inlet flow. Fig. 13-37 will give a corresponding adiabatic efficiency. The polytropic head may now be determined from Eq. 13-36.

When a P-H diagram is available, it is the fastest and most accurate method of determining compressor horsepower and discharge temperature.

**Centrifugal Refrigeration Compressors**

Compression ratio per wheel will vary on the order of 1.5 to 2.75 per wheel depending on the refrigerant and speed.

Due to the ease of applying external side loads to centrifugal machines, it is quite common to flash refrigerant from the condenser en route to the evaporators and/or to accept side loads from product being cooled by refrigerant at higher pressures than the lowest evaporator level.

Since side-loading is the practice rather than the exception, it is common to let the centrifugal compressor manufacturer obtain the desired performance characteristics from the following data: evaporator temperature levels; refrigeration loads required in MM Btu/hr; heat rejection medium (air or water); and type of driver.

Refrigeration compressors are also discussed in Section 14.

**GENERAL**

**Flow Limits**

Two conditions associated with centrifugal compressors are surge (pumping) and stone-wall (choked flow).

At some point on the compressor’s operating curve there exists a condition of minimum flow/maximum head where the developed head is insufficient to overcome the system resistance. This is the surge point. When the compressor reaches this point, the gas in the discharge piping back-flows into the compressor. Without discharge flow, discharge pressure drops until it is within the compressor’s capability, only to repeat the cycle.

The repeated pressure oscillations at the surge point should be avoided since it can be detrimental to the compressor. Surging can cause the compressor to overheat to the point the maximum allowable temperature of the unit is exceeded. Also, surging can cause damage to the thrust bearing due to the rotor shifting back and forth from the active to the inactive side.

"Stonewall" or choked flow occurs when sonic velocity is reached at any point in the compressor. When this point is reached for a given gas, the flow through the compressor cannot be increased further without internal modifications.

**Interstage Cooling**

Multistage compressors rely on intercooling whenever the inlet temperature of the gas and the required compression ratio are such that the discharge temperature of the gas exceeds about 300°F.

There are certain processes that require a controlled discharge temperature. For example, the compression of gases such as oxygen, chlorine, and acetylene requires that the temperature be maintained below 200°F.

The thermal stress within the horizontal bolted joint is the governing design limitation in a horizontally split compressor case. The vertically split barrel-type case, however, is free from the thermal stress complication.

Substantial power economy can be gained by precooling the gas before it enters the interstage impellers. Performance calculations indicate that the head and the horsepower are directly proportional to the absolute gas temperature at each impeller.

The gas may be cooled within the casing or in external heat exchangers.

Two methods of cooling within the casing are used — water cooled diaphragms between successive stages and direct liquid injection into the gas.

Diaphragm cooling systems include high-velocity water circulation through cast jackets in the diffuser diaphragms. The diaphragm coolers are usually connected in series.

Liquid injection cooling is the least costly means of controlling discharge temperatures. It involves injecting and atomizing a jet of water or a compatible liquid into the return channels. In refrigeration units, liquid refrigerant is frequently used for this purpose. Injected liquid also functions as a solvent in washing the impellers free of deposits. Nevertheless, the hazards of corrosion, erosion, and flooding present certain problems resulting in possible replacement of the compressor rotor.

External intercoolers are commonly used as the most effective means of controlling discharge temperatures. The gas is discharged from the compressor casing after one or more stages of compression and, after being cooled, is returned to the next stage or series of stages for further compression.
Intercoolers usually are mounted separately. When there are two or more compressor casings installed in series, individual machines may or may not be cooled or have intercoolers. In some cases, it may be advantageous to use an external cooler to precool gas ahead of the first wheel.

**Journal and Thrust Bearings**

Radial journal bearings are designed to handle high speeds and heavy loads and incorporate force-feed lubrication. They are self-aligning, straight sleeve, multi-lobe sleeve, or tilting pad type, each sized for good damping characteristics and high stability.

Tilting pad bearings have an advantage over the sleeve type as they eliminate oil whip or half-speed oil whirl which can cause severe vibrations.

Bearing sleeves or pads are fitted with replaceable steel-backed babbit shells or liners.

Axial thrust bearings are bidirectional, double faced, pivoted-shoe type designed for equal thrust capacity in both directions and arranged for force-feed lubrication on each side.

Thrust bearings are sized for continuous operation at maximum differential pressure including surge thrust loads, axial forces transmitted from the flexible coupling and electric motor thrust.

On units where the thrust forces are low, a tapered land thrust bearing may be used but must be selected for proper rotation direction. At times a combination of pivoted-shoe and tapered land is recommended.

Compressor designs with impellers arranged in one direction usually have a balance drum (piston) mounted on the discharge end of the shaft to minimize axial loads on the thrust bearing.

**Shaft Seals**

Shaft seals are provided on all centrifugal compressors to limit, or completely eliminate, gas leakage along the shaft where it passes through the casing.

With the wide range of temperature, pressure, speed, and operating conditions encountered by compressors, there can be no one universal seal, or seal system, to handle all applications.

Basically, the designs of seals available are: labyrinth (gas), restrictive ring (oil or gas), liquid film (oil), and mechanical (contact) (oil or gas).

A mechanical (contact) seal, Fig. 13-42, has the basic elements similar to the liquid film seal. The significant difference is that clearances in this seal are reduced to zero. The seal operates with oil pressure 35 to 50 psi above internal gas pressure as opposed to 5 psi in the liquid film seal.

The mechanical (contact) seal can be applied to most gases, but finds its widest use on clean, heavier hydrocarbon gases, refrigerant gases, etc.

A mechanical gas seal uses the process gas as working fluid to eliminate the seal oil system.

The liquid film seal, Figs. 13-43 and 13-44, was also developed for the severe conditions of service but requires higher oil circulation rate than the mechanical (contact) type.

The seal consists of two sleeves which run at close clearance to the shaft with a liquid injected between the sleeves to flow to the seal extremities. The sleeves are lined with babbit or a similar non-galling material which is compatible with the properties of the compressed gas and the sealing liquid.

The sealing liquid, usually a lubricating oil, is introduced between the two rings at a controlled differential pressure of about 5 psi above the internal gas pressure, presenting a barrier to direct passage of gas along the shaft. This fluid also performs the very important functions of lubricating the sleeves and removing heat from the seal area.

**Lubrication and Seal-oil Systems**

On all centrifugal compressors that have force-feed lubricated bearings, a lubrication oil system is required. When oil-film or mechanical (contact) seals are used, a pressurized seal-oil system must be provided.

Each system is designed for continuous operation with all the elements (oil reservoir, pumps with drivers, coolers, filters, pressure gauges, control valves, etc.) piped and mounted on a flat steel fabricated base plate located adjacent to the compressor. The compressor manufacturer normally supplies both systems in order to have overall unit responsibility.

---

**FIG. 13-41**

Journal and Thrust Bearing Assembly

[Diagram of journal and thrust bearing assembly]

*Courtesy of Ingersoll-Rand Co.*
Depending on the application, lubrication and seal-oil systems may be furnished as combined into one system, or as one lubrication system having booster pumps to increase the pressure of only the seal oil to the required sealing level. In service involving heavily contaminated gases, separate lube-oil and seal-oil systems should be used.

The lubrication system may supply oil to both compressor and driver bearings (including gear) couplings (if continuously lubricated), as well as turbine governor, trip and throttle valve, and hydraulic control system.

A single lubricant shall be used in all system equipment, usually an oil, having approximate viscosities of 150 Saybolt Universal seconds (SUS) at 100°F and 43 SUS at 210°F.

In addition to all the elements of a common pressurized lubrication system, the seal oil system requires a collection system for the oil. Depending on the gas composition, a degassing tank may be installed in the seal oil trap return line to remove the oil-entrained gas prior to return of the seal oil to the common oil reservoir. The flow past the outer sleeve passes through an atmospheric drain system and is returned to the reservoir. The relatively low flow through the inner sleeve is collected in a drain trap or continuous drainer and may be returned to the reservoir or discarded, depending upon the degree and type of contamination which occurred while it was in contact with the internal gas.

Compressors using only liquid film seals should be provided with a seal-oil system which incorporates an overhead surge tank. The surge tank provides seal-oil capacity for coastdown in case of a compressor shutdown.

In combined seal-oil and lube-oil systems when large amounts of contaminants are present in the process gas, the
seal-oil design may call for buffer gas injection to form a barrier between the compressed gas and the seal oil.

Fig. 13-45 shows clean sweet buffer gas being injected into the center of a labyrinth seal preceding the oil film seal with seal oil supplied between the two sleeves. Part of the seal oil flows across the inner sleeve and mixes with buffer gas and then drains into the seal oil trap. The other part of the seal oil flows across the outer sleeve, mixes with the bearing lube oil drain flow, and returns to the common lube- and seal-oil reservoir.

Drivers — Centrifugal compressors can be driven by a wide variety of prime movers including electric motors, steam turbines, gas combustion turbines, and gas-expander turbines. Each driver has its own design parameters. A motor drive presents limitations in operation of the compressor due to constant and low speed. The constant speed restriction is minimized by suction or discharge throttling. The low speed restriction is corrected by introduction of a speed increasing gear. A steam turbine, on the other hand, has variable speed capability that allows more control of the compressor capacity or discharge pressure, and its high speed permits the compressor to be directly connected to the driver. In the case of a single-shaft gas turbine, the power output is limited at a reduced speed.

Drivers are discussed in Section 15.

CONTROL SYSTEMS

Centrifugal compressor controls can vary from the very basic manual recycle control to elaborate ratio controllers. The driver characteristics, process response, and compressor operating range must be determined before the right controls can be selected.

The most efficient way to match the compressor characteristic to the required output is to change speed in accordance with the fan laws (affinity laws, see Eqs 13-22 to 13-24):

\[
\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \left(\frac{H_1}{H_2}\right)^{\frac{1}{2}}
\]

One of the principal advantages of using steam or gas turbines as drivers for compressors is that they are well suited to variable-speed operation. With such drivers, the speed can be controlled manually by an operator adjusting the speed governor on the turbine or, alternatively, the speed adjustment can be made automatically by a pneumatic or electric controller that changes the speed in response to a pressure or flow signal.

Pressure Control at Variable Speed

The control system operates as follows:

The pressure transmitter (PT) in Fig. 13-46 senses the process pressure. It converts this signal to a signal proportional to the process pressure and sends it to the pressure controller (PC).

The pressure controller amplifies the transmitter signal and sends a modified signal to the final element. Depending on system requirements the controller may require additional correction factors called reset and rate.

The final element in this case is speed control. This mechanism varies the turbine-governor speed setting over a predetermined range.

As the load decreases, pressure will rise. An increase in process pressure above the set value will cause the signal to reach the governor and reduce the speed, maintaining the desired system pressure.

Volume Control at Variable Speed

If the nature of the process requires constant volume delivered, then the arrangement shown in Fig. 13-47 would be used.

Here, the flow transmitter (FT) senses the process flow, converts the signal to a signal proportional to the process flow, and sends it to the flow controller (FC).

The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Reset and rate correction factors may be needed.

The final element is speed control, which is accomplished by a mechanism that varies the turbine-governor speed setting. An increase in flow over set point would cause a signal to reach the governor and reduce the speed to maintain the desired system flow.

When using electric motors as constant speed drivers, the centrifugal compressor is normally controlled by a suction throttling device such as butterfly valve or inlet guide vanes. Throttling the suction results in a slightly lower suction pres-
Pressure Control at Constant Speed

The control system shown in Fig. 13-48 has the pressure signal sensed and amplified in a similar manner as described in the scheme for variable speed control (Fig. 13-46).

The final element is a suction throttle valve (STV) that reduces the flow of gas into the compressor.

A process pressure increase over a set value would cause a signal to reach the suction throttle valve (STV) and would partially close the valve in order to reduce the inlet pressure.

Volume Control at Constant Speed

The control scheme for this arrangement is shown in Fig. 13-49.

The flow transmitter (FT) senses the process flow using an orifice or venturi as the primary flow element (FE), converts this to a signal that is proportional, and sends this signal to the flow controller (FC). The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Reset and derivative controller actions may be required.

The final element is the compressor guide-vane mechanism. The guide vanes are adjusted by means of a positioning cylinder. This cylinder is operated by a servo-valve (SRV) that receives a signal from the flow controller.

Here, an increase in flow above the set point causes a signal to reach the final element, which will result in the dosing of the guide vanes to decrease flow.

Adjustable Inlet Guide Vanes — The use of adjustable inlet guide vanes is the most efficient method of controlling a constant speed compressor. The vanes are built into the inlet of the 1st stage, or succeeding stages, and can be controlled through the linkage mechanism either automatically or manually.

The vanes adjust the capacity with a minimum of efficiency loss and increase the stable operating range at design pressure. This is accomplished by pre-rotation of the gas entering the impeller which reduces the head-capacity characteristics of the machine. Fig. 13-50 illustrates the effect of such control at various vane positions.

Anti-surge Control

It is essential that all centrifugal compressor control systems be designed to avoid possible operation in surge which usually occurs below 50% to 70% of the rated flow.

Compressor surge is a large pressure and volume fluctuation that takes place when attempting to operate at a higher pressure ratio than design maximum. The surge limit line (see Fig. 13-50) can be reached from a stable operating point by either reducing flow or decreasing suction pressures. An anti-surge system senses conditions approaching surge, and maintains the unit pressure ratio below the surge limit by recycling some flow to the compressor suction. Care must be taken to cool this recycle stream.

Volume, pressure rise, or pressure ratio may be used as control parameters to sense an approaching surge condition. Such a condition will be established by the characteristic curve of the compressor.

A volume-controlled anti-surge system is shown in Fig. 13-51. The flow transmitter (FT) senses the process flow using an orifice or venturi as a primary flow element (FE). It converts the signal to one that is proportional to the process flow and sends it to the surge controller (SC).

The surge controller compares the transmitted signal to the set-point signal. If the set-point signal is exceeded, the controller amplifies the signal difference and sends this modified signal to the final element. Reset and rate correction factors may be needed.
The final element is a surge control valve (SCV). The valve releases pressure buildup at the discharge of the compressor.

As flow decreases to less than the minimum volume set-point, a signal will cause the surge control valve to open. The valve opens, as required, to keep a minimum volume flowing through the compressor.

For a pressure-limiting anti-surge control system, see Fig. 13-52. The pressure transmitter (PT) senses the process pressure. It converts this to a signal that is proportional to process pressure and sends it to the surge controller (SC).

The surge controller compares the transmitted signal to the set-point signal. If the set-point signal is exceeded, the controller amplifies the signal difference and sends this modified signal to the surge control valve. Reset and rate correction factors may also be required. The blow-off valve relieves a pressure buildup on the discharge end of the compressor.

A process pressure increase over the pressure set-point signal will cause the blowoff valve to open. The valve opens as required to keep the pressure limited to a minimum volume of gas flowing through the compressor.

A set-point computer may be required on multiparameter compressor-control systems where the surge set point may change. Suction temperature, suction pressure, and speed are examples of parameters that cause the surge point of the compressor to change.

In the design of a compressor system, any changes that may occur from a single operating point in the overall process must be carefully considered. In many instances, automatic controls are needed to maintain the highest degree of system performance. In any event, the choice of manual or automatic controls is dictated by the operating pattern of the centrifugal compressor.

The user, compressor manufacturer, and contractor should work closely together to determine the minimum control system required. Various phases of operation, such as start-up, shut-down, initial runs, and normal runs, should be investigated to make certain that a workable system has been designed.

**Vibration Control System**

This control system may be provided to monitor the driver behavior at the shaft bearings for detection of excessive lateral vibration and axial movement and for protection against possible machinery failure through alarm and/or shutdown devices.

The system may protect not only the compressor but also the driver, such as a steam or gas turbine, that usually runs at the same high speed as the compressor. When a speed increasing or reducing gear unit is furnished between the compressor and driver, also consider monitoring vibration at the gear shaft bearings.

The main system components are: variation transducer(s), signal amplifier(s) with d-c power supply, and vibration monitor and/or analyzer.

Vibration transducers fall into three categories: displacement probe, velocity pick-up, and accelerometer.

The displacement probe is most commonly used for equipment with high value, as it can measure shaft vibration relative to bearing housing. Output signal from each transducer is small and, therefore, it must be amplified before being transmitted to a vibration monitor or analyzer.

**Fig. 13-53** shows a vibration severity chart for use as a guide in judging vibration levels as a warning of impending trouble.

For more information on vibration monitoring systems, see API Standard 670, Noncontacting Vibration and Axial Position Monitoring System, and API Standard 678, Accelerometer-Based Vibration Monitoring System.
OPERATIONAL CONSIDERATIONS

Rotor Dynamics and Critical Speeds

The demand for smooth-running turbomachinery requires careful analysis of rotor dynamics taking into account bearing performance, flexibility, critical speed, and rotor response.

Equally important is to analyze the dynamic behavior of the compressor for sudden changes in load due to start-up, shut-down, or loss of power supply.

Successful rotor design is the result of accurate calculation of critical speeds. A critical speed occurs at a condition when the rotor speed corresponds to a resonant frequency of the rotor-bearing support system. Under no circumstances should the compressor be allowed to run at a critical speed for a prolonged length of time as the rotor vibrations amplified by this condition can cause machinery failure.

Critical Speed Map

A critical speed map is one of various methods used to predict the operational behavior of the rotor. First, the critical speeds for a given rotor geometry are calculated for a range of assumed bearing support stiffness values. The result is a map like that shown in Fig. 13-54. The bearing stiffness characteristics are determined from the geometry of the bearing support system, and cross-plotted on the critical speed map.

The map depicts the values of the undamped critical speeds and how they are influenced by bearing stiffness. The intersections of the bearing stiffness curve and the critical speed lines represent the undamped critical speeds. The intersection points generally indicate margins between the criticals and the operating speed range.

However, the use of this map is very limited because it is based on a simplified undamped, circular synchronous analysis with no cross-coupled or unbalance effects. It is a good trending tool showing a machine's basic dynamic characteristics. It may not accurately depict peak response frequencies.

The critical speed map is used extensively because it enables determination of bearing or support stiffness by correlating test-stand data.

Unbalance Response Analysis

This method predicts rotor-bearing system resonances to greater accuracy than the critical speed map. Here, bearing support stiffness and damping are considered together with synchronous vibration behavior for a selected imbalance distribution. A computer is normally required to solve the resulting differential equations. Satisfactory results depend on the accurate input of bearing stiffness and damping parameters.

Several runs are usually made with various amounts and locations of unbalance. The plot of results of a typical unbalance response study is shown in Fig. 13-55. Each curve represents the rotor behavior at a particular station or axial location such as those corresponding to the midspan, bearings, and overhangs.

No rotor can be perfectly balanced and, therefore, it must be relatively insensitive to reasonable amounts of unbalance.

The unbalance-response results predict the actual amplitudes that permit calculations of the unbalance sensitivity.

This is expressed in mils of vibration amplitude per ounce-inch or gram-inch of unbalance.

The peaks of the response curves represent the critical speed locations. Fig. 13-56 shows limits of placement of critical speeds as specified in the API Standard 617, Centrifugal Compressors for General Refinery Services.