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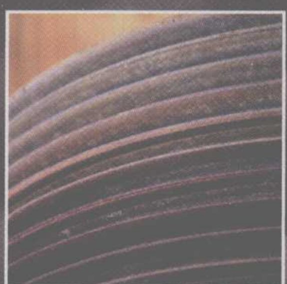
COMPRESSOR HANDBOOK



PAUL C. HANLON

COMPRESSOR HANDBOOK

Paul C. Hanlon Editor

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PREFACE

Compressors fall into that category of machinery that is "all around us" but of which we are little aware. We find them in our homes and workplaces, and in almost any form of transportation we might use. Compressors serve in refrigeration, engines, chemical processes, gas transmission, manufacturing, and in just about every place where there is a need to move or compress gas.

The many engineering disciplines (e.g. fluid dynamics, thermodynamics, tribology, and stress analysis) involved in designing and manufacturing compressors make it impossible to do much more than just "hit the high spots," at least in this first edition.

This is such a truly broad field, encompassing so many types and sizes of units, that it is difficult to cover it all in one small volume, representing the work of relatively few authors. Possibly, more than anything else, it will open the door to what must follow—a larger second edition.

In compressors, the areas of greatest concern are those parts with a finite life, such as bearings, seals and valves, or parts that are highly stressed. Treatment of these components takes up a large portion of the handbook, but at the same time space has been given to theory, applications and to some of the different types of compressors.

Much in this handbook is based on empirical principals, so this should serve as a practical guide for designers and manufacturers. There are also test and analysis procedures that all readers will find helpful. There should be something here for anyone who has an interest in compressors.

Paul C. Hanlon

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CHAPTER 1
COMPRESSOR THEORY

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1.1 NOMENCLATURE

		Units (See note below)
a	Speed of Sound in Gas	ft/sec
a, b	Constants in Equation of State (Pressure Form)	
A, B	Constants in Equation of State (Compressibility Form)	
B	Bore	Inch
CL	Fixed Clearance as a Fraction of Swept Volume	
c_P	Specific Heat at Constant Pressure	$ft.lb_f/lb_m.R$
c_V	Specific Heat at Constant Volume	$ft.lb_f/lb_m.R$
e	Specific Internal Energy (i.e. Internal Energy per unit mass)	$ft.lb_f/lb_m$
E	Internal Energy	$ft.lb_f$
F	Flow Area	$Inch^2$
h	Specific Enthalpy (i.e. Enthalpy per unit mass)	$ft.lb_f/lb_m$
H	Enthalpy	$ft.lb_f$
HP	Horsepower	
J	Joule's Equivalent	$ft.lb_f/BTU$
k	Ratio of Specific Heats ($= c_P/c_V$)	
m	Mass Flow Rate	lb_m/sec
M	Mass	lb_m
n_T	Isentropic Temperature Exponent	
n_V	Isentropic Volume Exponent	
N	Compressor Speed	rpm
P	Pressure	lb_f/in^2 Abs
PW	Power	$ft.lb_f/min$

q	Heat Transfer Rate	BTU/sec
Q	Heat Transfer	BTU
R	Gas Constant	$ft.lb_f/lb_m.R$
s	Specific Entropy	$ft.lb_f/lb_m.R$
S	Stroke	Inch
T	Temperature	Rankine
u	Gas Velocity	ft/sec
U_P	Piston Velocity	ft/min
v	Specific Volume (Volume per unit mass)	ft^3/lb_m
V	Volume	$Inch^3$
W	Work Done on Gas during Process	$ft.lb_f$
WD	Work Done on Gas During one compressor cycle	$ft.lb_f$
Z	Compressibility (sometimes called Supercompressibility)	
ΔP	Pressure Drop	lb_f/in^2
ρ	Density	lb_m/ft^3
θ	Crank Angle	Degree
λ	Integration Constant in expression for Average Pressure Drop	

Suffixes

C	Critical Pressure or Temperature
D	Discharge from the Compressor Cylinder
eq	Equivalent (Area)
in	At Entry to a Control Volume
o	Stagnation Value
out	At Exit from a Control Volume
R	Reduced (Pressure or Temperature)
SW	Swept (Swept Volume is Maximum minus Minimum Cylinder Volume)
1, 2, 3, 4	At Corresponding Points in the Cycle (Fig. 1.2)
1, 2	Before and after process

NOTE:. The basic equations given in this section can be used with any consistent system of units. The units given above are not consistent and the numerical factors required to use the equations with the above units are given at the end of each equation in square brackets. If the above units are used, the equations can be used as written. If an alternate, consistent, system of units is used, the numerical factors at the ends of the equations should be ignored.

1.2 THEORY

1.2.1 Gas Laws

By definition, compressors are intended to compress a substance in a gaseous state. In predicting compressor performance and calculating the loads on the various components, we need methods to predict the properties of the gas. Process compressors are used to compress a wide range of gases over a wide range of conditions. There is no single **equation of state** (an equation that allows the density of a gas to be calculated if the pressure and temperature are known) that will be accurate for all gases under all conditions. Some of the commonly used ones, starting with the most simple, are discussed below.

The simplest equation of state is the perfect gas law:

$$Pv = \frac{P}{\rho} = RT\left[\frac{1}{144}\right]$$

This equation applies accurately only to gases when the temperature is much higher than the critical temperature or the pressure much lower than the critical pressure. Air at atmospheric conditions obeys this law well.

To predict the properties of real gases more accurately, the perfect gas law is often modified by the addition of an empirical value "Z", called the **compressibility**, or sometimes the **supercompressibility**, of the gas. The value of Z is a function of the gas composition and the pressure and temperature of the gas. The modified equation is:

$$\frac{p}{\rho} = ZRT\left[\frac{1}{144}\right]$$

This equation is accurate if, and only if, Z is known accurately. Z can be estimated with reasonable accuracy in many cases using the **Law of Corresponding States** which states that the value of Z as a function of the reduced pressure and temperature is **approximately** the same for all gases. That is:

$$Z = fn\,(P_R, T_R) = fn\left(\frac{P}{P_C}, \frac{T}{T_c}\right)$$

A curve of Z as a function of reduced pressure and temperature is shown as Fig. 1.1. This gives reasonable results for most gases when the gas state is not close to the critical point or the two phase region.

It is frequently useful to have an equation to predict Z. This allows calculation of other properties such as entropy, enthalpy and isentropic exponents that are needed to predict compressor performance. The use of an equation rather than charts is also convenient when a computer is used to perform the calculations. Many equations are available: one of the most simple, the **Redlich-Kwong Equation of State** is given below. Other equations are more accurate over a wider range of gases and conditions, but are more complex. Some of these are discussed in Refs. 2 and 3. The Redlich-Kwong equation of state is:

$$P = \left(\frac{RT}{v - b} - \frac{a}{v^2 + bv}\right)\left[\frac{1}{144}\right]$$

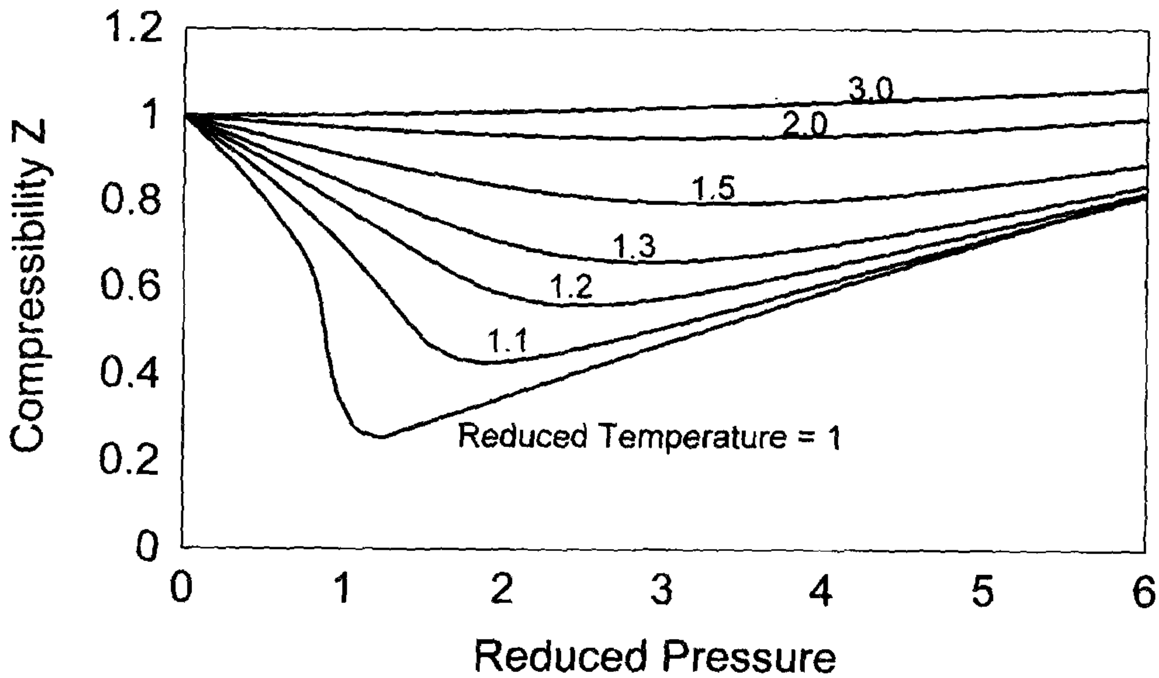


FIGURE 1.1 Compressibility chart (based on the Redlich Kwong equation of state).

$$\text{where } a = 0.42748 \frac{R^2 T_C^{2.5}}{P_C T^{0.5}} \left[\frac{1}{144}\right]$$

$$b = 0.08664 \frac{RT_C}{P_C} \left[\frac{1}{144}\right]$$

or

$$Z^3 - Z^2 + (A - B - B^2)Z - AB = 0$$

$$\text{where } A = 0.42748 \frac{P_R}{T_R^{2.5}}$$

$$B = 0.08664 \frac{P_R}{T_R}$$

Solving the above cubic equation for Z once P_R and T_R are known is equivalent to looking up the value of Z on Fig. 1.1.

Other equations of state commonly used in predicting compressor performance include the Soave Redlich Kwong, Peng Robinson, Benedict Webb Rubin, Han Starling, Lee-Kesler, and API Method equations. Details of these methods can be found in the literature (e.g. Refs. 2 and 3) .

1.2.2 Thermodynamic Properties

To predict compressor performance ways to calculate the enthalpy, internal energy and entropy of the gas are needed. It is also often convenient to use the **isentropic volume exponent** n_V and the **isentropic temperature exponent** n_T.

The isentropic exponents are defined such as to make the following equations true for an isentropic change of state.

$$PV^{n_V} = \text{Constant}$$

$$\frac{P^{\frac{n_T - 1}{n_T}}}{T} = \text{Constant}$$

For a **perfect gas**, the above properties are easily calculated. The following is for a gas that obeys the ideal gas laws and has constant specific heats. Specific properties are those per unit mass of gas.

$$\text{Specific Internal Energy} = e = c_v T$$

$$\text{Specific Enthalpy} = h = c_p T$$

$$n_V = n_T = c_P / c_V = k$$

$$\text{Change of Specific Entropy} = s_2 - s_1 = c_P \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

For a **real gas,** the above properties can be obtained from a Mollier chart for the gas or from the equation of state and a knowledge of how the specific heats at low pressure vary with temperature. Methods for this are given in Refs. 2 and 3.

An **approximation** that allows isentropic processes to be calculated easily for a real gas if the Z values are known is often useful. Consider an isentropic change of state from 1 to 2.

$$\frac{\rho_2}{\rho_1} = \frac{Z_1}{Z_2}\frac{P_2}{P_1}\frac{T_1}{T_2} = \left(\frac{P_2}{P_1}\right)^{1/n_V}$$

$$\frac{T_1}{T_2} \approx \left(\frac{P_1}{P_2}\right)^{n_T - 1/n_T}$$

$$\therefore \left(\frac{P_2}{P_1}\right)^{1/n_V} = \frac{Z_1}{Z_2}\left(\frac{P_2}{P_1}\right)^{1/n_T}$$

It is found that if the gas state is not too near the critical or two phase region, and is therefore acting somewhat like an ideal gas, then n_T is approximately equal to $k = c_p/c_V$. Then

$$\left(\frac{P_2}{P_1}\right)^{1/n_V} \cong \frac{Z_1}{Z_2}\left(\frac{P_2}{P_1}\right)^{1/k}$$

1.2.3 Thermodynamic Laws

For calculating compressor cycles, the energy equation, relationships applying to an isentropic change of state, and the law for fluid flow through a restriction are needed.

The **Energy equation** for a **fixed mass of gas** states simply that the increase of energy of the gas equals the work done on the gas minus the heat transferred from the gas to the surroundings. For the conditions in a compressor, we can ignore changes in potential and chemical energy. In applications where the energy equation for a fixed mass of gas is used, we can usually also ignore changes in kinetic energy. The energy equation then reduces to:

$$E_2 - E_1 = M(e_2 - e_1) = W - Q[J]$$

If we consider a **control volume,** that is a volume fixed in space that fluid can flow into or out of, we must consider the work done by the gas entering and leaving the control volume, and in many cases where this equation is used, we must consider the kinetic energy of the gas entering and leaving the control volume. The **energy equation** then becomes:

$$E_2 - E_1 = M_{in}h_{o\,in} - M_{out}h_{o\,out} + W - Q[J]$$

where $h_o = h + \frac{1}{2}u^2\left[\frac{1}{32.18}\right]$

$h = e + Pv[144]$

For a steady process, there is no change of conditions in the control volume and $E_2 = E_1$

$$\text{Then } M_{out}h_{o\,out} - M_{in}h_{o\,in} = H_{o\,out} - H_{o\,in} = W = Q[\mathrm{J}]$$

The equations for **isentropic** change of state were given above. They apply to any change during which there are no losses and no heat transfer to the gas. The change of properties can be obtained from a Mollier chart for the gas, or if the gas behaves approximately as a perfect gas, by the equations given above.

$$PV^{n_v} = \text{Constant}$$

$$\frac{P^{\frac{n_r - 1}{n_r}}}{T} = \text{Constant}$$

The law for **incompressible fluid flow through a restriction** is:

$$m = F\sqrt{(2\rho\,\Delta P)}\left[\sqrt{\frac{32.18}{144}}\right]$$

F = Effective Flow Area = Geometric Flow Area × Flow Coefficient

For a **perfect gas**, if the pressure drop is low enough that the flow is **subsonic**, as should always be the case in reciprocating compressors, the pressure drop is given by:

$$m = k\frac{p_1}{a_1}\left(\frac{p_2}{p_1}\right)^{k+1/2k} F\sqrt{\left[\frac{2}{k-1}\left(\left(\frac{p_1}{p_2}\right)^{k-1/k}-1\right)\right]} \quad [32.18]$$

if $\frac{p_2}{p_1} < \left(\frac{2}{k+1}\right)^{k/k-1}$ the flow is sonic and $m = k\frac{p_1}{a_1}\left(\frac{2}{k+1}\right)^{k+1/2(k-1)} F$ [32.18]

1.2.4 Compression Cycles

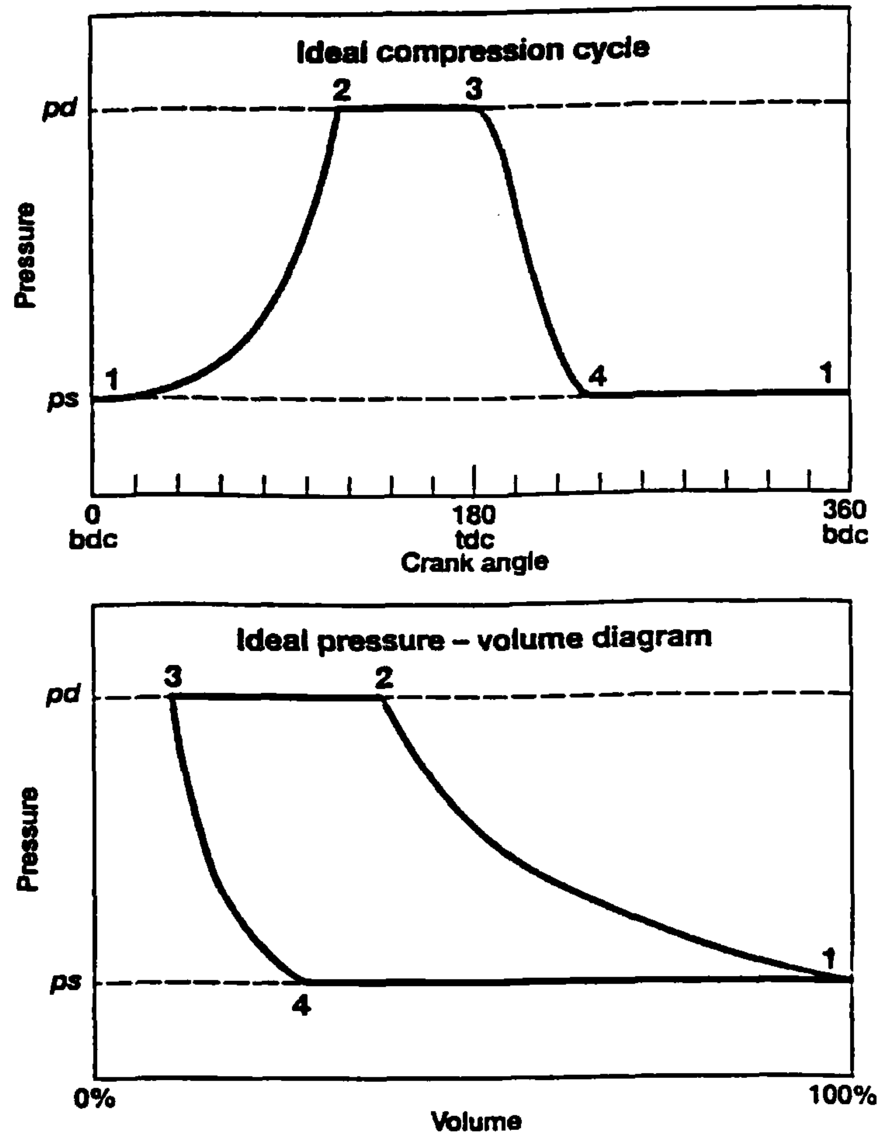
The work supplied to a compressor goes to increasing the pressure of the gas, to increasing the temperature of the gas and to any heat transferred out of the compressor. In most cases, the requirement is to increase the pressure of the gas using the least possible power. If the compression process is adiabatic, that is, there is no heat transfer between the compressor and the outside, then the least work will be done if the process is isentropic. This implies that there are no losses in the compressor and which is an unachievable goal, but one that can be used as a base for the compression efficiency. The **isentropic efficiency** of a compressor is defined as the **work required** to compress the gas in an **isentropic process** divided by the **actual work** used to compress the gas. The efficiency of a compressor is most often given as the isentropic efficiency.

However, it is possible to construct a compressor with an isentropic efficiency greater than 100%. The work done in a **reversible isothermal** process is less than that done in an isentropic process. In a reversible isothermal process, the temperature of the gas is maintained at the suction temperature by reversible heat transfer as the compression proceeds. There must, of course, be no losses in this process. Many compressors have a final discharge temperature that is much lower than the isentropic discharge temperature, and the power required is reduced by this. However, the power required is almost always still greater than the isentropic power and so the isentropic efficiency is universally used to rank compressors.

1.2.5 Ideal Positive Displacement Compressor Cycle

As an example of a positive displacement compressor, consider a **reciprocating compressor** cylinder compressing gas from a suction pressure P_S to a discharge pressure P_D. In compressor terminology, the ratio P_D/P_S is known as the **compression ratio**. This can be contrasted to reciprocating engine terminology where the compression ratio is a ratio of volumes.

For a reciprocating compressor, the **ideal** compression cycle is as shown on Fig. 1.2. The cycle is shown on pressure against crank angle and pressure against cylinder volume coordinates. The cycle can be explained starting at point 1. This represents the point when the piston is at the dead center position that gives the



Ideal pressure-crank angle and pressure-volume diagrams.

FIGURE 1.2 Ideal compressor cycle.

maximum cylinder volume. The gas in the cylinder is at the suction pressure P_S. As the piston moves to decrease the cylinder volume, the mass of gas trapped in the cylinder is compressed and its pressure and temperature rise. In the ideal case, there is no friction and no heat transfer and so the change is isentropic and the change of pressure and temperature can be calculated from the known change of volume using the above equations for isentropic change of state.

At point 2, the pressure has increased to equal the discharge pressure. In the ideal compressor, the discharge valve will open at this point and there will be no pressure loss across the valve. As the piston moves to further decrease the cylinder volume, the gas in the cylinder is displaced into the discharge line and the pressure in the cylinder remains constant.

At point 3, the piston has reached the end of its travel, the cylinder is at its minimum volume and the discharge valve closes. As the piston reverses and moves to increase the cylinder volume, the gas that was trapped in the **clearance volume** (sometimes called the **fixed clearance**) at point 3, expands and its pressure and temperature decrease. Again there are no losses or heat transfer and the change of pressure and temperature can be calculated using the expressions for isentropic change of state.

At point 4, the pressure has decreased to again equal the suction pressure. The suction valve opens at this point. As the piston moves to further increase the cylinder volume, gas is drawn into the cylinder through the suction valve. When the piston again reaches the dead center, point 1, the cylinder volume is at its maximum, the suction valve closes, and the cycle repeats.

The **work required per cycle** and hence the **horsepower** required to drive the compressor can easily be calculated from the pressure against volume diagram or from the temperature rise across the compressor.

The work done on the gas during a small time interval during which the cylinder volume changes by dV is equal to P dV and the work done during one compressor cycle is the integral of this for the cycle. That is, the work done equals the area of the cycle diagram on pressure against volume axes (Fig. 1.2). Note that the equivalence of work done per cycle and diagram area holds for real as well as ideal cycles. That is, the magnitude of losses that cause a horsepower requirement increase can be measured off the **indicator card** as the pressure vs. volume plot is often called. (If the pressure on the indicator card is in psi and the volume in cubic inches, the work done as given by the card area will be in inch lb. and must be divided by 12 to give the work done in ft. lb.)

Once the work done per cycle is known, the horsepower can be calculated. If the work done is in ft. lb., and the speed in rpm:

$$HP = WD\ N/33{,}000$$

If the heat transfer from the gas in the cylinder can be measured or estimated, the work done per unit time can be calculated from the energy equation.

$$\text{Work Done per Unit Time} = m(h_2 - h_1) + q[J]$$

For a cycle with **no heat transfer** with a **perfect gas,** Q is zero and $h = c_p\ T$, then

$$\text{Power, } PW = mc_p(T_2 - T_1)[60]$$

Now for an **ideal cycle** and a **perfect gas**, the compression is isentropic and the discharge temperature T_2 can be calculated from the pressure ratio and the suction temperature T_I using the isentropic relationship.