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BASIC THERMODYNAMICS OF RECIPROCATING COMPRESSION

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AUTHOR BIOGRAPHY

Greg Phillippi is the director of process compressor marketing and sales for Ariel Corporation in Mount Vernon, Ohio. Greg has earned degrees in mechanical engineering (Ohio Northern University 1978) and master of business administration (Ashland University 2000). He began his career as a design engineer with Cooper Energy Services in 1978 and in 1985 accepted a design engineer position with Ariel Corporation. In 2000 Greg accepted a position with ACI Services, Inc. in Cambridge, Ohio where he had responsibility for marketing, sales and engineering. In January 2004 he returned to Ariel in his present role. Greg has significant experience in the design, application, marketing and sales of reciprocating compressors for the oil and gas industry. He is a current member of both the API Standard 618 Sixth Edition Task Force and Sub-Committee on Mechanical Equipment and is a former member of the Gas Machinery Research Council Project Supervisory Committee and Gas Machinery Conference Advisory Committee.

ABSTRACT

This tutorial provides a straightforward explanation of how a reciprocating compressor works relying heavily upon the pressure-volume diagram (P-V diagram) as a point of reference. The discussion begins with a thorough review of the P-V diagram. Next, how capacity is determined and calculated is discussed, which requires an explanation of volumetric efficiency. How much power is required to compress a certain volume of gas is then explained involving detailed discussions of adiabatic, valve loss and friction power. The tutorial will then delve into discussions of compression efficiency and how changes in pressure and rotating speed affect the P-V diagram. The tutorial should be an excellent introduction for the inexperienced and a solid refresher for the experienced.

INTRODUCTION

A reciprocating compressor is a fairly simple machine and easy to understand the basic mechanics. This tutorial focuses on what occurs in the compression chamber of a compressor cylinder, and more specifically on the P-V diagram (Figure 1). How much gas is compressed and how much power is required for that compression are both derived directly from the P-V diagram. Hence, understanding the P-V diagram is critically important and forms the foundation for a deeper understanding for how a reciprocating compressor works.

PRESSURE - VOLUME DIAGRAM

A reciprocating compressor is a positive displacement machine in that a volume of gas is drawn into a compressor cylinder's compression chamber where it is trapped, compressed and pushed out. The P-V diagram is a plot of the pressure of the gas versus the volume of the gas trapped in the compression chamber. In Figure 1, P_S, suction pressure, represents the pressure of the gas at the inlet to the compressor cylinder. P_D, discharge pressure, represents the pressure of the gas at the outlet from the compressor cylinder. V_1 represents the maximum volume of gas trapped in the compression chamber and V_3 the minimum. The difference between V_1 and V_3 is known as piston displacement, or how much volume is displaced in one stroke length of the piston:

$$\text{Piston Displacement} = V_1 - V_3$$

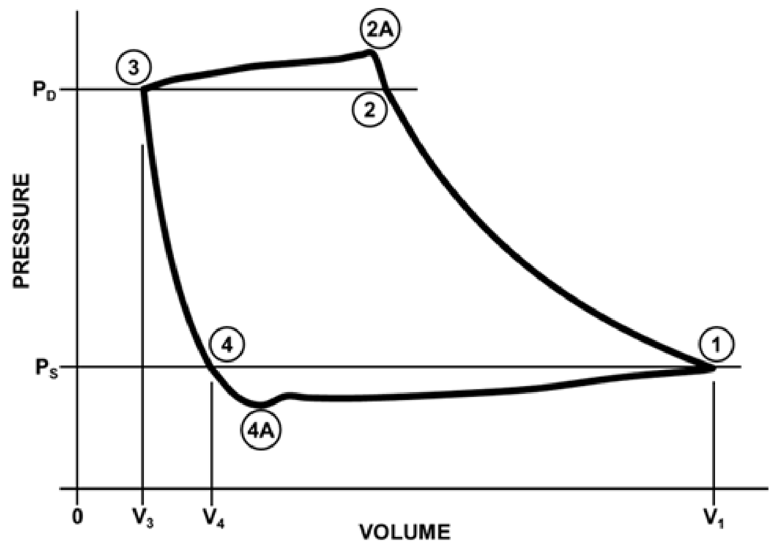

Figure 1
Example pressure-volume diagram

Referring to Figure 1, the P-V diagram is made up of four basic segments or events, 1-2, 2-2A-3, 3-4, and 4-4A-1, and each will be explained. Position 1 has the maximum volume (V_1) of gas trapped in the compression chamber and that gas is at suction pressure and temperature. At Position 1 all of the compressor valves are closed and the piston is at rest. All of the compressor valves are simple spring-loaded check valves and are not actuated by any outside means. As the crankshaft rotates the piston will move and the volume inside the compression chamber decreases (moving from Position 1 to 2). From Position 1, as the volume decreases, the gas pressure increases (the gas is trapped in the compression chamber with the volume decreasing). When the pressure inside the compression chamber becomes slightly higher than discharge pressure (P_D, at position 2), the discharge compressor valve will open. This completes the first segment of the diagram and is called the compression event.

At Position 2, the discharge valve opens and, as the piston continues to move, gas at discharge pressure and temperature is pushed out of the compression chamber through the discharge compressor valve and into the discharge gas passage of the cylinder. At Position 3, the piston comes to rest, the discharge compressor valve closes and this segment ends. Segment 2-2A-3 is called the discharge event. Moving from Position 1 to 3 the piston has moved through one stroke length and one-half revolution of the crankshaft.

Position 3 represents the minimum volume of gas trapped inside the compression chamber (V_3). Note this is not zero volume.

At Position 3 the piston will reverse and travel in the opposite direction. As it moves the volume inside the compression chamber increases and the trapped gas (V_3) will increase in volume and decrease in pressure - the gas will expand. Segment 3-4 is called the expansion event.

At Position 4, the pressure inside the compression chamber will be slightly less than suction pressure (P_S) causing the suction compressor valve to open. With the suction valve open, the compression chamber is open to the suction gas passage and as the piston continues to move, the volume continues to increase and the compression chamber fills with gas at suction pressure and temperature. The piston returns to Position 1, comes to rest, and the process repeats. Segment 4-4A-1 is called the suction event.

This completes the basic reciprocating compression process. One cycle around the P-V diagram represents one revolution of the crankshaft and two stroke lengths of the piston, one from Position 1 to 3 and another from Position 3 to 1. Four events make up the P-V diagram - compression, discharge, expansion and suction.

CAPACITY

The volume of gas compressed by this P-V diagram is the difference in volume between Positions 1 and 4:

$$\text{Capacity} = V_1 - V_4$$

This volume is influenced by the compression ratio (R_C) and the magnitude of V_3. Compression ratio is:

$$R_C = \frac{P_D}{P_S}$$

with absolute pressures

As compression ratio increases the capacity decreases as less gas is drawn into the compression chamber and compressed. As V_3 increases there is more gas that must expand thus reducing capacity. V_3 is referred to as the fixed clearance volume.

FIXED CLEARANCE

The fixed clearance volume (V_3 in Figure 1) is the volume of gas remaining in the compression chamber at the discharge end of the stroke that must expand from discharge pressure (P_D) to suction pressure (P_S) before the suction compressor valve can open and the suction event begin. Figure 2 is a cross-section of a typical compressor cylinder assembly with some of the fixed clearance volume highlighted in red.

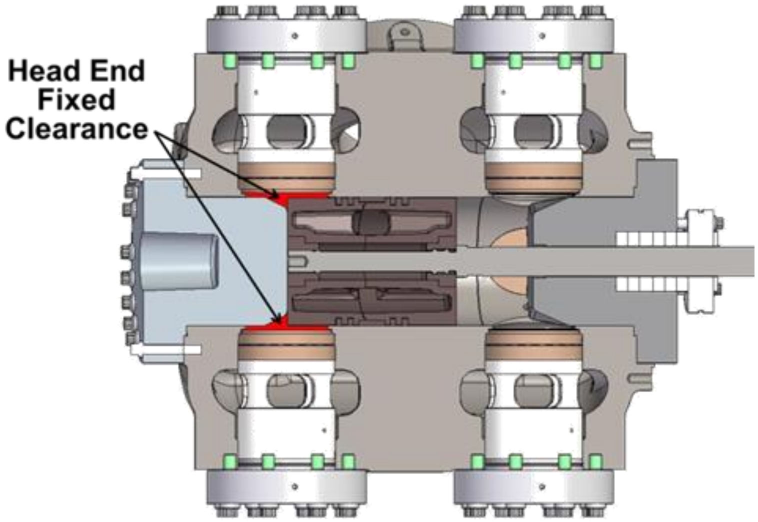

Figure 2
Compressor cylinder cross-section highlighting fixed clearance volume

The cavity between the compressor valve and cylinder bore (red highlight) might represent 70 percent of the total fixed clearance volume. There is also some volume between the head and the bore, between the piston and the bore and between the piston and head. All of this volume adds together to form the total fixed clearance volume.

Fixed clearance (CL) is expressed as a percentage of the piston displacement (referring to Figure 1):

$$\mathrm{CL} = \frac{V_3}{V_1 - V_3} \times 100\%$$

VOLUMETRIC EFFICIENCY

Volumetric efficiency (VE) is the percentage of stroke that fills with gas at suction pressure and suction temperature. In equation form (referring to Figure 1):

$$\mathrm{VE} = \frac{V_1 - V_4}{V_1 - V_3} \times 100\%$$

Some notes about VE:

- VE represents the capacity.
- VE is <u>NOT</u> suction valve open time. The suction valves do not have to be open for the full VE.
- A higher number for VE does not mean it is “better” as might be the case for energy efficiency. VE simply represents capacity. The influence of VE on energy efficiency is through the relationship of VE to average piston velocity (average velocity of gas through valves).

The equation for VE as derived from thermodynamics (or from the P-V diagram) is:

$$VE = 100 - CL\left[\left(\frac{Z_S}{Z_D}\right)\left(\frac{P_D}{P_S}\right)^{\frac{1}{K}} - 1\right]$$

Where:
VE = Volumetric efficiency, %
CL = Fixed clearance, %
Z_S = Compressibility factor @ P_S & T_S
Z_D = Compressibility factor @ P_D & T_D
P_D = Discharge pressure, absolute
P_S = Suction pressure, absolute
K = Adiabatic exponent, "K-Value"

A VE equation that might be used in compressor selection software might look like this:

$$VE = 100 - R_C - CL\left[\left(\frac{Z_S}{Z_D}\right)(R_C)^{\frac{1}{K}} - 1\right]$$

Note the term "-R_C" that has been added. Instead of subtracting from 100%, this equation subtracts from 100 - R_C. This term is intended to account for the fact a real running compressor does not conform to pure thermodynamic theory. For example the seals surrounding the compression chamber, specifically the compressor valves, piston rings and packing, are not perfect. There is always some internal gas leakage. This means real VE will never agree with VE from theory. So a "fudge factor" must be used, and "-R_C" is just such a factor. Every compressor manufacturer has a unique method for adjusting the VE equation and "-R_C" is just one simple method. Compression ratio (R_C) typically varies from 1.3 to 3.5 so the -R_C term reduces VE (capacity) by 1.3% to 3.5%.

There can be a concern with VE being too low. Figure 3 shows a P-V diagram (in red) that has low VE. The P-V diagram is very narrow and the discharge event is very short. This raises the possibility that the discharge valves may not have enough time to open and close properly and can cause the discharge valves to fail prematurely.

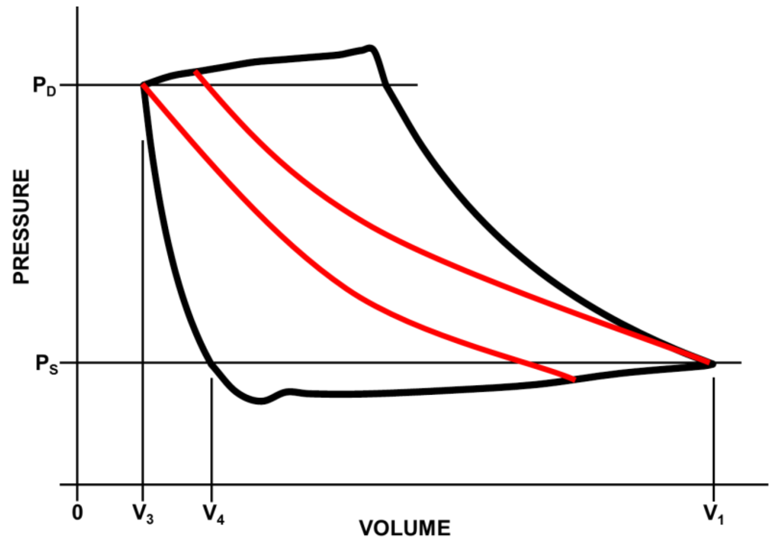

Figure 3
Pressure-volume showing "low" volumetric efficiency

POWER

The power required to drive a reciprocating compressor can be divided into three pieces, adiabatic, valve loss and friction and each will be discussed separately.

The power required to compress a volume of gas is represented by the area enclosed by the P-V diagram, or:

$$\text{Work} = \int PdV$$

The compression and expansion events are modeled thermodynamically as adiabatic processes, meaning it is assumed that no heat is transferred to or from the gas during these events. An adiabatic thermodynamic process is an isentropic (constant entropy) process. The area of the P-V diagram in Figure 1 bounded by 1-2-3-4-1 is the adiabatic power.

How valid is the assumption that the compression and expansion events are adiabatic? For a compressor with a rotating speed of 300 rpm (a slow rotating speed) one P-V cycle takes only 0.2 seconds to complete. Assuming each of the four events of the P-V cycle take equal time, that's 0.05 seconds (or 50 milli-seconds) per event. That's not much time for any significant amount of heat to transfer, therefore lending credibility to the adiabatic assumption. Yes, the gas does get hot as it's compressed but not from heat being transferred to the gas. That heat is the heat of compression.

Inefficiency in the P-V diagram is the pressure drop incurred in moving the gas from the inlet flange of the cylinder into the compression chamber and in moving the gas from the compression chamber to the outlet flange. Overcoming this pressure drop requires energy. This energy is represented by the areas 1-4-4A-1 (suction valve loss power) and 2-2A-3-2 (discharge valve loss power) in Figure 4.

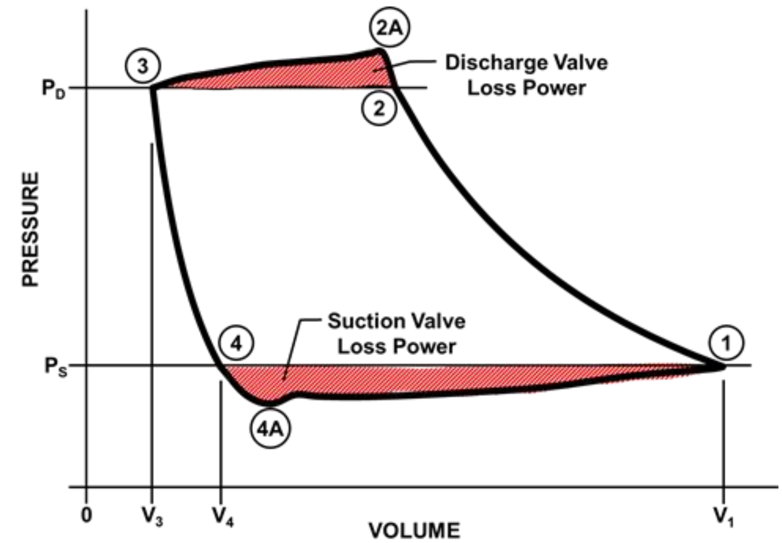

Figure 4
Pressure-volume diagram highlighting suction and discharge valve loss power

This valve loss power represents all of the inefficiency in the P-V diagram. Friction is the remaining inefficiency and will be discussed later. The valve loss power can be expressed by the following relationship:

$$VLP \approx \frac{(MW)(P)(VE)(R_P)(A_{BORE})^3(S \times RPM)^3}{(Z)(T)(N)(A_{VLVPKT})^2}$$

Where:

MW	=	Gas mole weight
P	=	Pressure, suction or discharge
VE	=	Volumetric efficiency
R_P	=	Resistance factor
A_{BORE}	=	Cross-sectional area of the cylinder bore
S	=	Stroke
RPM	=	Rotating speed, revolutions per minute
Z	=	Compressibility factor @ suction or discharge
T	=	Temperature, suction or discharge
N	=	Number of suction or discharge valves feeding a head-end or crank-end compression chamber
A_{VLVPKT}	=	Cross-sectional area of the valve bore
SxRPM	=	Piston speed, fpm As used in this relationship, it is the average piston speed during the valve open time.

This relationship in an even simpler form:

$$VLP \approx \Delta P(A_{BORE})(VE)(S \times RPM)$$

will be discussed in some detail.

The first variable on the right side is pressure drop. Pressure drop is:

$$\Delta P \approx \rho(V)^2$$

Where:
ρ = Density
V = Velocity

Density for a gas is:

$$\rho \approx \frac{P(MW)}{Z(T)}$$

Where:
P = Pressure
MW = Mole weight
Z = Compressibility factor
T = Temperature

The velocity as used here is the average velocity of the gas as it moves through the valve bores as if the valves were not installed. That works out to:

$$V \approx \frac{(A_{BORE})(S \times RPM)}{(N)(A_{VLV\ PKT})} \approx \frac{(D_{BORE})^2(S \times RPM)}{(N)(D_{VLV\ PKT})^2}$$

Substituting this relationship for velocity into the equation for pressure drop:

$$\Delta P \approx \frac{P(MW)(A_{BORE})^2(S \times RPM)^2}{Z(T)(N)(A_{VLV\ PKT})^2}$$

This relationship represents the average pressure drop through the compressor valve bores in the cylinder body - as if the valves were not installed and the valve bores were simple orifices. Of course, what is needed is the pressure drop through the compressor valve. Adding the resistance factor term (R_P) accomplishes this:

$$\Delta P \approx \frac{P(MW)(R_P)(A_{BORE})^2(S \times RPM)^2}{Z(T)(N)(A_{VLV\ PKT})^2}$$

Resistance factor (1) is defined as the ratio of measured pressure drop across a compressor valve to the pressure drop that would be predicted in flowing the same quantity of the same gas at identical upstream pressure and temperature conditions through a round hole (an orifice) having a discharge coefficient equal to one and an area equal to the valve pocket opening. Typical resistance factors range from 30 to 200. Meaning a compressor valve could have 30 to 200 times the pressure drop as an orifice the same diameter as the compressor valve. Note that resistance factor is a dimensionless number as it is pressure divided by pressure. So resistance factor is:

$$R_P = \frac{\text{Compressor Valve } \Delta P}{\text{Orifice } \Delta P}$$

Another term used in the same manner is valve equivalent area (VEA). Valve equivalent area has units of area. Valve equivalent area is the orifice area required to generate the same pressure drop as that through a compressor valve when flowing the same quantity of the same gas at the same pressure and temperature. Compressor and compressor valve manufacturers will use either term (resistance factor or valve equivalent area) to describe the relative efficiency of a compressor valve. One can be converted into the other:

$$\mathrm{VEA} = \frac{A_{\mathrm{VLV\ PKT}}}{\sqrt{R_P}} \quad \text{or} \quad R_P = \frac{\left(A_{\mathrm{VLV\ PKT}}\right)^2}{(\mathrm{VEA})^2}$$

Some further discussion about the "S x RPM" term in the above relationships is required. This term is generally known as piston speed, or the average linear speed at which the piston moves through one stroke length. Average piston speed in feet per minute is calculated by:

$$\mathrm{PS} = \frac{2 \times S \times \mathrm{RPM}}{12}$$

$$\mathrm{PS} = \frac{S \times \mathrm{RPM}}{6}$$

Where:

PS	=	Piston speed, feet per minute
S	=	Stroke, inch
RPM	=	Rotating speed, revolutions per minute

Figure 5 is a plot of instantaneous and average piston speed versus crank angle:

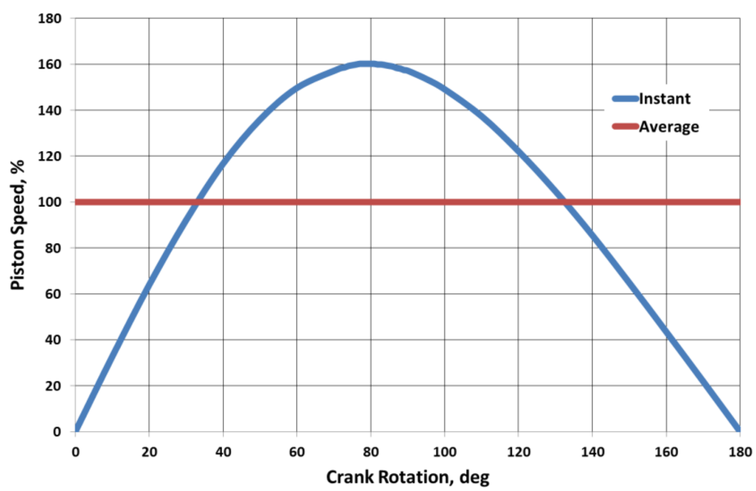

Figure 5
Plot of piston speed (as percent of average) versus crankshaft rotation

Instantaneous piston speed reaches a maximum near the middle of the stroke but not exactly the middle (90° of rotation). Note that the maximum piston speed is about 60% greater than the average.

But the velocity used in the above relationships for valve pressure drop and valve loss power is the average piston speed during the time the compressor valves (suction or discharge) are open as shown in Figure 6:

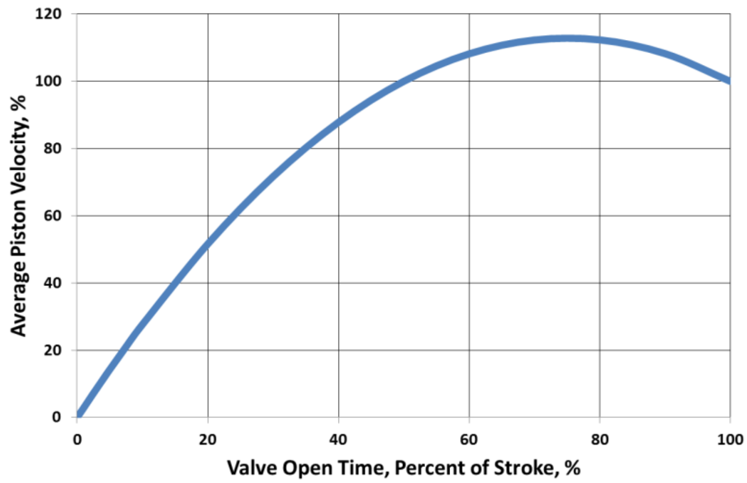

Figure 6
Plot of piston velocity versus compressor valve open time

For example, if the suction valve is open for 40% of the stroke, the average piston velocity would be about 87% of the full stroke average velocity.

If the above relationships are substituted back into the valve loss power equation the following results:

$$\mathrm{VLP} \approx \frac{\mathrm{P}(\mathrm{MW})(\mathrm{R_P})(\mathrm{A_{BORE}})^3(\mathrm{S}\times\mathrm{RPM})^3(\mathrm{VE})}{\mathrm{Z}(\mathrm{T})(\mathrm{N})(\mathrm{A_{VLV\ PKT}})^2}$$

Valve loss power (VLP) and friction represent all of the inefficiency in a reciprocating compressor (not considering pressure drop in getting gas to and from the compressor and possible efficiency losses due to P-V diagram distortion resulting from gas pulsation). The magnitude of typical friction power is ~5%, meaning the majority of the inefficiency is associated with VLP. Some comments about VLP:

- VLP varies with $(\mathrm{S} \times \mathrm{RPM})^3$. This is a large number and therefore significantly impacts VLP.

- VLP varies with stroke and rotating speed and not just rotating speed. Comments are sometimes made that "high speed compressors are inefficient". This is not correct. A more accurate statement is "high piston speed compressors are relatively inefficient". The following table lists several combinations of stroke and rotating speed that result in the same piston speed:

Stroke, inches	Rotating Speed, rev/min	Piston Speed, feet/min
21.0	257	900
19.5	277	900
18.0	300	900
16.5	327	900
15.0	360	900
13.5	400	900
12.0	450	900
10.5	514	900
9.0	600	900
7.5	720	900
6.0	900	900
4.5	1200	900
3.0	1800	900

Everything else being equal (admittedly very difficult) all of these combinations would have the same relative compression efficiency.

- VLP varies directly with mole weight (MW). For example, a hydrogen compressor (MW = 2) would have 89% less VLP as compared to the same compressor compressing natural gas (MW = 18) simply because of the very low mole weight.

- The very basic relationship between the cylinder bore diameter (A_{BORE}^3) and the number and size of the compressor valves (N x $A_{VLV\ PKT}^2$) determines the basic efficiency of a given cylinder. In simple terms - the larger the valves for a given cylinder bore diameter the better the efficiency.

- As the cylinder bore increases in diameter the relative efficiency decreases. This results from simple geometry:

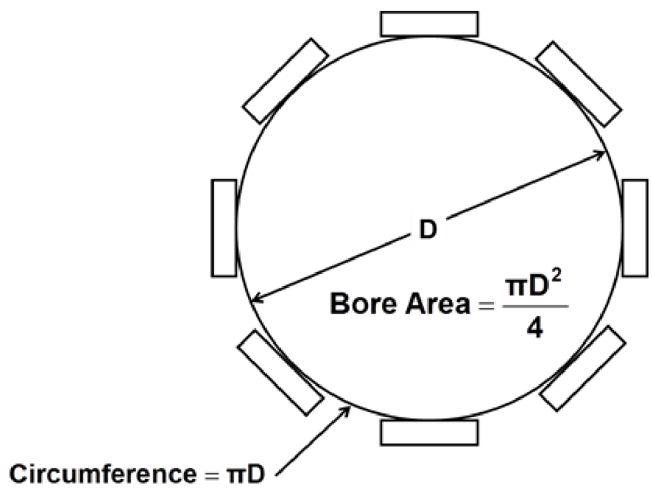

Figure 7
Drawing showing the space available for compressor valves in a typcial compressor cylinder design

The bore area grows by the bore diameter to the second power, but the circumference, which is the space available to locate compressor valves (represented by the rectangles in Figure 7), only grows by the bore diameter to the first power.

Fundamentally, a compressor cylinder is made more efficient by using larger compressor valves for a given cylinder bore diameter (everything else being equal). But something else happens with the compressor valves being larger - the fixed clearance becomes larger. Larger fixed clearance results in lower volumetric efficiency which is lower capacity. A given cylinder diameter with more and/or larger compressor valves will compress less gas, but compress that gas with better energy efficiency (lower power per capacity).

But there is an application where it is possible, and highly advantageous, to use larger compressor valves with their associated higher fixed clearance. When the compression ratio is low the addition of fixed clearance does not cause a significant drop in capacity. This is shown in Figure 8:

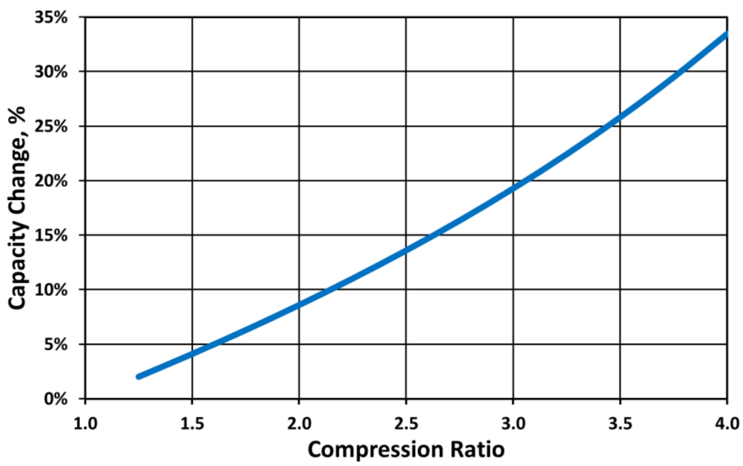

Figure 8
Plot of change in capacity versus compression ratio

This plot shows the percentage capacity change resulting from a change in the fixed clearance volume from 20% to 30% (a ten percentage point increase). Note that at low compression ratios the capacity change is very small. For example, at a compression ratio of 1.5 the resulting capacity change is only about 4%, but at a compression ratio of 3.0 it is about 19%. In fact manufacturers take advantage of this by designing compressor cylinders specifically for low compression ratio applications. These cylinders have large valves for the cylinder bore diameter and are very efficient. But they cannot be used at compression ratios much above 1.7 or so due to this loss of capacity because a larger diameter cylinder would be required to compress the required volume of gas (more expensive) thus requiring more rod load (potentially yet even more expensive).

DEACTIVATED END POWER

A double-acting compressor cylinder is one where the compression process occurs on both sides of the piston or ends of the compression chamber. It is possible to deactivate one end and only compress gas with the other end. It is beyond the scope of this tutorial to explain how this is done mechanically. For a thorough discussion of deactivated end power please see the second reference.

The deactivated end has a P-V diagram (red in Figure 9) and therefore consumes power.

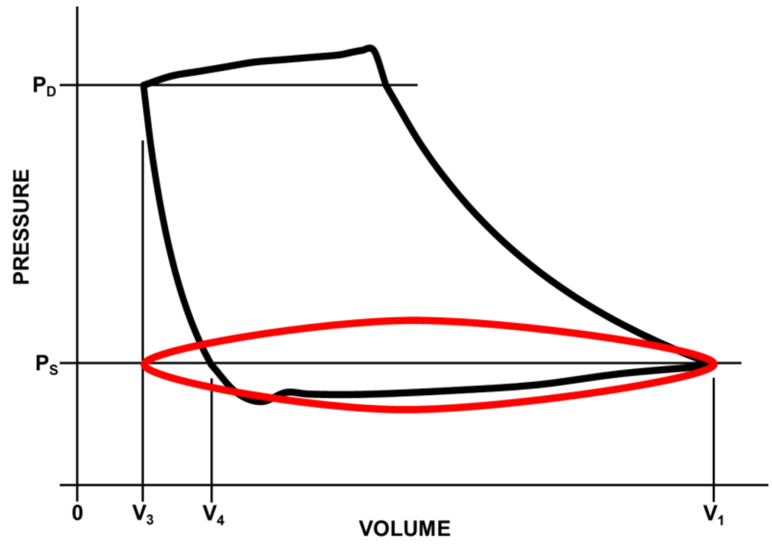

Figure 9
Pressure-volume diagram for a deactivated end

Figure 9 shows two P-V diagrams - one of an active end (black) and one deactivated (red). The deactivated end has zero capacity as its discharge compressor valves never open, but it does consume power. This power manifests itself as heat.

Figure 10 is an infrared image showing three identical compressor cylinders all with the same suction and discharge pressure but in three different unloading configurations (each compressing a different capacity).

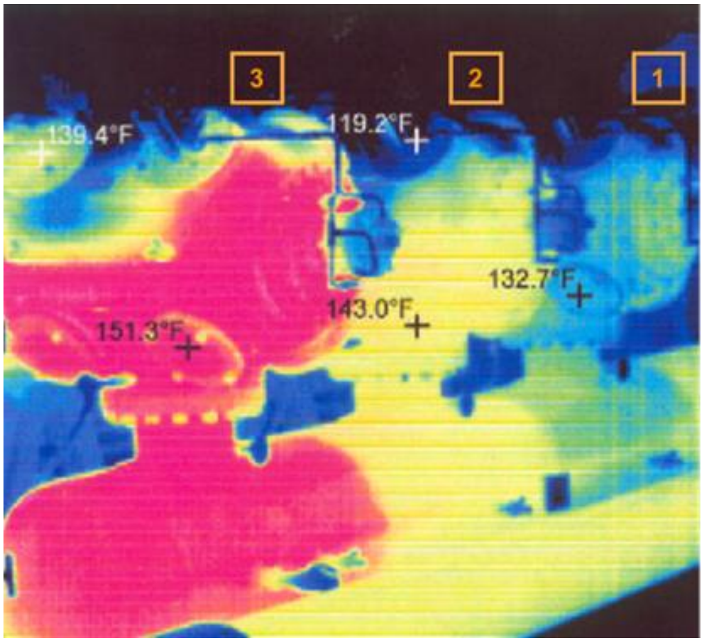

Photo courtesy of Energy Imaging 580-388-4385

Figure 10
Infrared photo comparing loaded and unloaded compressor cylinders

Cylinder #1 is double acting and is at 100% capacity. Cylinder #2 has a head end clearance pocket open and is at 80% capacity. Cylinder #3 has the head end deactivated and is at 50% capacity. Note the discharge temperature of each cylinder as measured on each discharge valve cap. Cylinder #1 has a discharge temperature of 132.7 ºF and cylinder #3 has 151.3 ºF. Also note that cylinder #2 has a suction temperature of 119.2 ºF (cylinder #1 can be assumed to have the same suction temperature as #2 as it's not shown) and cylinder #3 has 139.4 ºF. This shows that the deactivated head end in #3 is essentially pre-heating the suction gas prior to compression in its crank end (the active end) resulting in a higher discharge temperature.

FRICTION

Friction is the last piece of the "power pie" that has to be discussed. A reciprocating compressor is a mechanical device and as such encounters and must overcome friction. Friction is accounted for very simply:

$$BP = \frac{IP}{M.E.}$$

Where:
BP = Brake power
IP = Indicated power
M.E. = Mechanical efficiency, typically 95% to 97%

A definition of indicated power is:

$$\begin{array}{l} \text{Adiabatic Power} \\ +\text{Suction Valve Loss Power} \\ +\text{Discharge Valve Loss Power} \\ \hline \text{Indicated Power} \end{array}$$

Indicated power is all the power derived from the P-V diagram.

Brake power is then the total power required to be input to the compressor to get the indicated power to the gas (for the P-V diagram).

Friction then accounts for somewhere in the range of 3% to 8% (maybe more on some low power applications) of the total input power. Friction is the crankshaft turning in the bearings, driving oil pumps, windage, crossheads sliding in the crosshead guides, packing rubbing against the piston rod, piston rings and wearbands rubbing against the cylinder bore and other items. Friction manifests itself as heat, for example causing the crankcase oil to get hot.

COMPRESSION EFFICIENCY

Compression efficiency is defined as:

$$EFF = \frac{AP}{BP} \times 100\%$$

Where:
EFF = Compression efficiency
AP = Adiabatic power
BP = Brake power

Figure 11 is a plot of compression efficiency versus compression ratio for a given compressor cylinder when compressing two different gases, hydrogen and nitrogen:

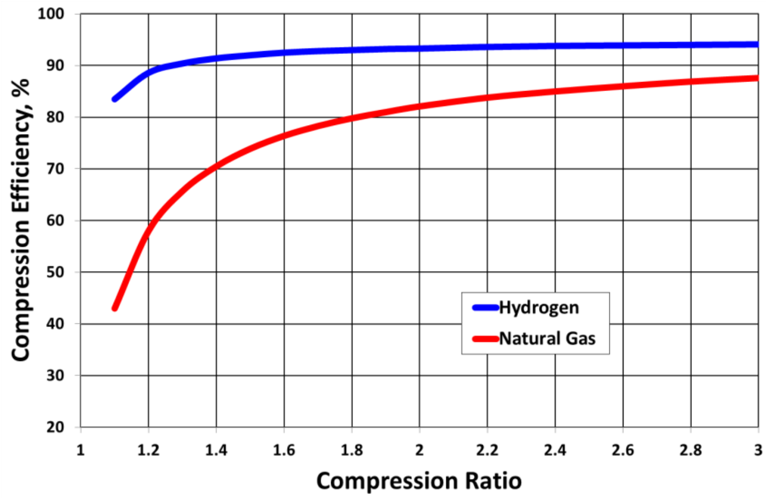

Figure 11
Plot of compression efficiency versus compression ratio

Note how efficiency increases as compression ratio increases. The efficiency curve will have this shape for any reciprocating compressor cylinder. Also note how much higher the efficiency is for hydrogen (with a very low mole weight of two) as compared to natural gas (with a medium mole weight of 18). Everything else being equal, compressing hydrogen will have one ninth (11%, 89% less) the valve loss power, hence much better efficiency.

VARYING CONDITIONS

Figures 12, 13 and 14 are P-V diagrams showing the changes that result from increased discharge pressure, decreased discharge pressure and increased suction pressure.

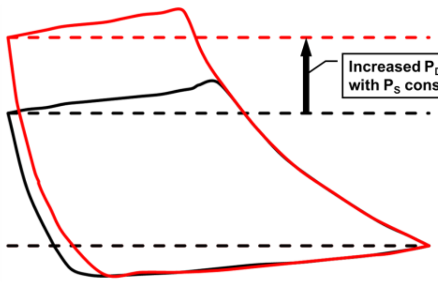

Figure 12
Change to pressure-volume diagram when discharge pressure increases

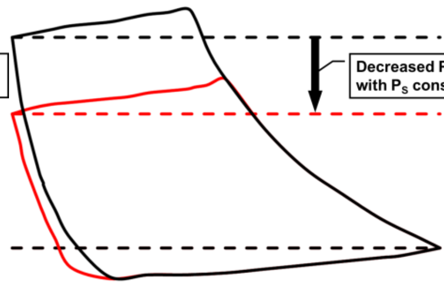

Figure 13
Change to pressure-volume diagram when discharge pressure decreases

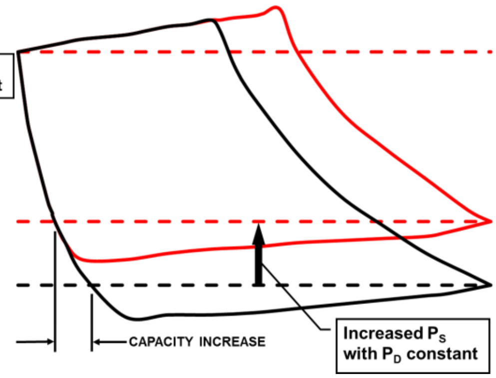

Figure 14
Change to pressure-volume diagram when suction pressure increases

Figure 12 shows how the P-V diagram changes when the discharge pressure increases with the suction pressure staying constant. The volumetric efficiency (capacity) decreases and the power (area enclosed by the P-V diagram) increases. The P-V diagram with the higher compression ratio will have a better efficiency.

Figure 13 shows the discharge pressure decreasing. Here the volumetric efficiency increases and the power decreases. The diagram with the lower discharge pressure, therefore lower compression ratio, will have a lower efficiency.

Figure 14 shows the suction pressure increasing. The volumetric efficiency increases, but it is not at all clear how the power changes just by looking at the P-V diagram as it gets shorter but also wider. The diagram with the higher compression ratio will have a higher efficiency.

In fact, as the pressures increase or decrease the power changes as a curve (Figure 15):

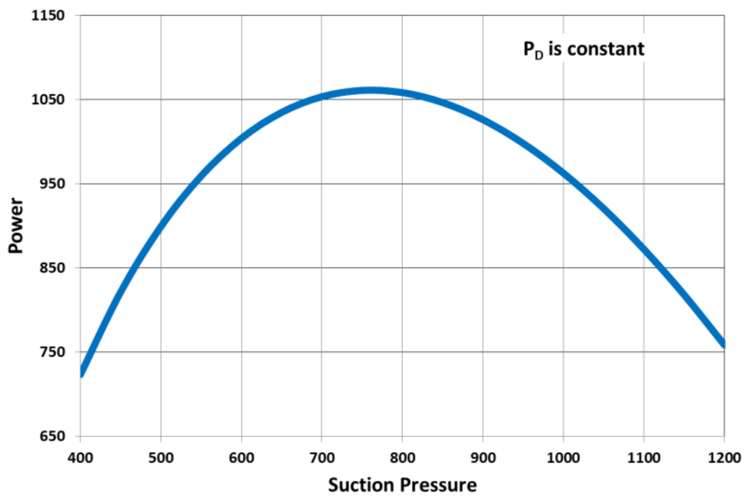

Figure 15
Plot of power versus suction pressure with constant discharge pressure

Figure 15 shows that as compression ratio decreases (suction pressure increases with the discharge pressure constant) the power will increase to a point but then decrease. The P-V diagram changes in height and width at different rates as the pressures change leading to the curve.

While power changes following a curve capacity always increases with decreasing compression ratio (Figure 16):

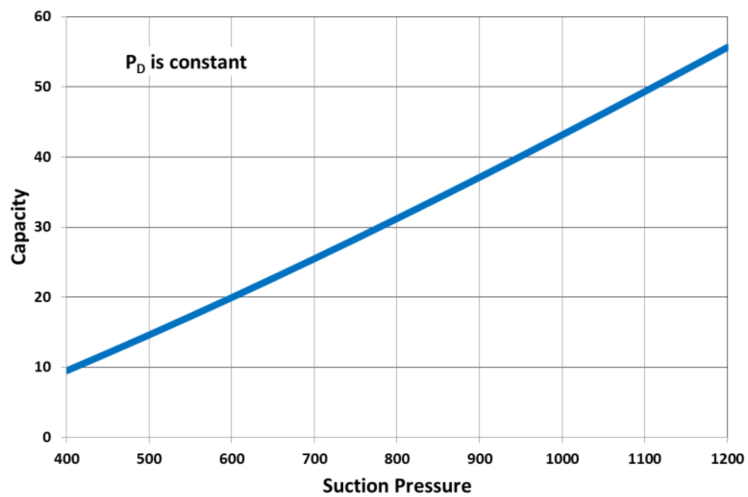

Figure 16
Plot of capacity versus suction pressure with constant discharge pressure

Figure 17 shows the effect of a change in rotating speed on the P-V diagram:

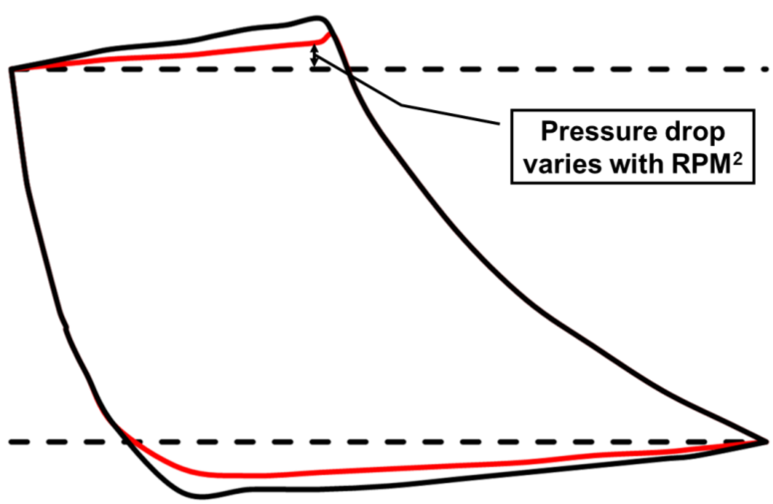

Figure 17
Pressure-volume diagram showing the effect of a change in rotation speed

Valve pressure drop will change with the square of the rotating speed change. The volumetric efficiency does not change, but the capacity will because the number of cycles in a given period of time will change.

THERMODYNAMIC PROPERTIES

The makeup of the gas being compressed (the gas analysis) effects the shape of the P-V diagram and therefore capacity and power. Two important thermodynamic properties are compressibility factor (Z) and adiabatic exponent (K-value).

Compressibility factors are essentially fudge factors and defined by:

$$PV_{IDEAL} = MRT \text{ for ideal gas}$$
$$PV_{REAL} = ZMRT \text{ for real gas}$$
or
$$Z = \frac{V_{REAL}}{V_{IDEAL}}$$

Where:

P = Pressure
V = Volume
M = Mass
R = Universal gas constant
T = Temperature
Z = Compressibility factor

Compressibility factor is typically less than 1.0.

The following P-V diagrams (Figures 18 and 19) show how changes in compressibility factor can change the shape of the P-V diagram.

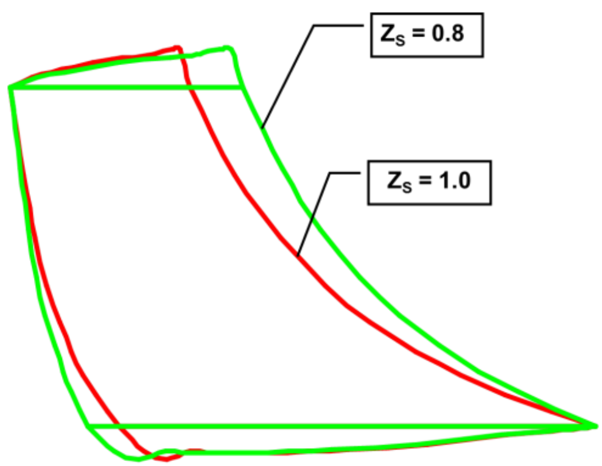

Figure 18
Pressure-volume diagram showing effect of change in suction compressibility factor

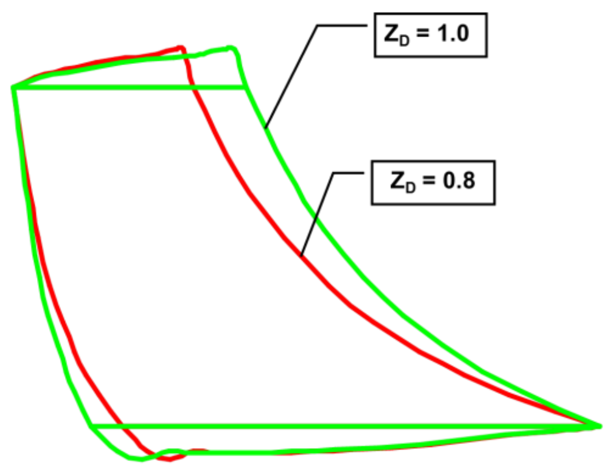

Figure 19
Pressure-volume diagram showing effect of change in discharge compressibility factor

Be aware that these examples assume all other parameters are constant, so only the referenced compressibility factor is changing. Another caution is that in many of the equations discussed above a ratio of compressibility factor is used and very often that ratio is close to 1.0. Nonetheless it is important to always use a gas analysis and get these thermodynamic properties correct. Figure 20 is a plot of compressibility factor for carbon dioxide for three temperatures over a range of pressure. Note that as the pressure and temperature approach the critical point (1071 psia and 87.8 °F) the compressibility factor decreases rapidly. Small changes in pressure can cause large changes in compressibility.

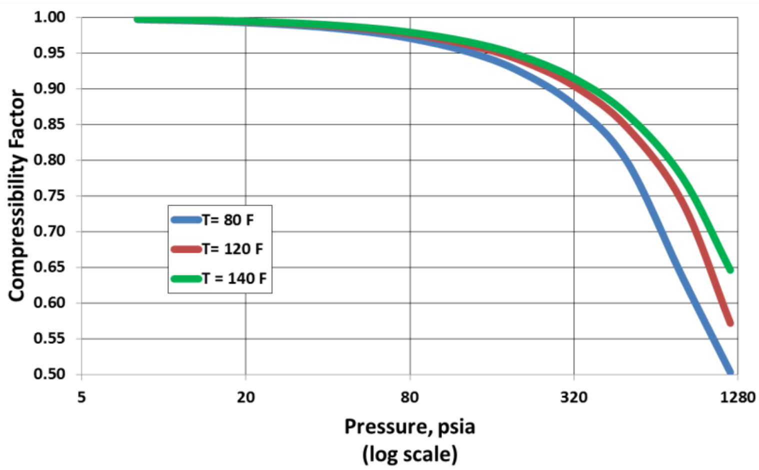

Figure 20
Plot of carbon dioxide compressibility factor versus pressure for three different temperatures

Adiabatic exponent (K-value) is the most important thermodynamic property. As K-value increases the P-V diagram gets wider and the volumetric efficiency (capacity) and power increase. Figure 21 shows how K-value affects the P-V diagram.

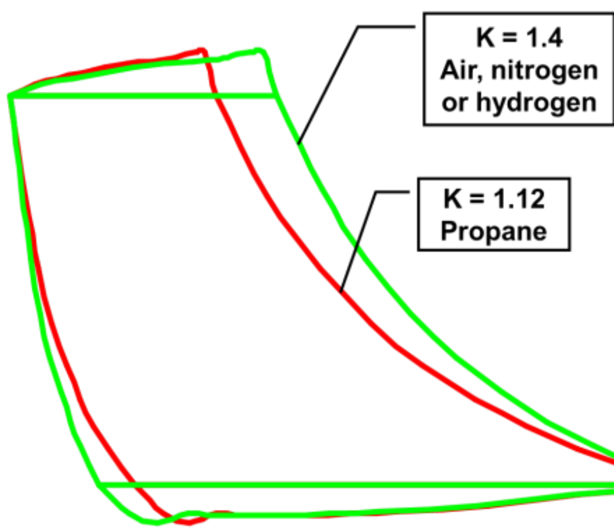

Figure 21
Pressure-volume diagram showing effect of a change in adiabatic exponent (K-value)

K-value is defined by the relationship:

$$\frac{T_{D\text{-}ADIABATIC}}{T_S} = \left(\frac{P_D}{P_S}\right)^{\frac{K-1}{K}}$$

K-value defines an isentropic (constant entropy, adiabatic) path on the temperature - entropy diagram (Figure 22):

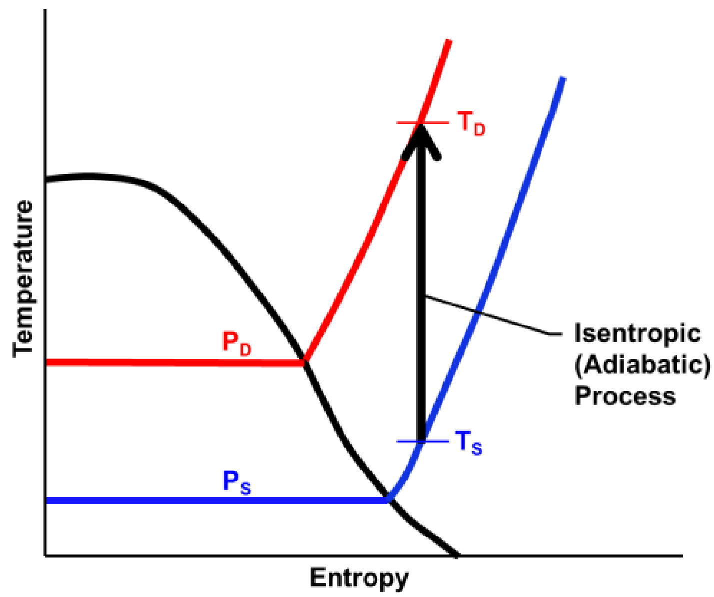

Figure 22
Temperature-entropy diagram explaining the adiabatic exponent

If the gas behaves as an ideal gas (follows PV = MRT) then K-value can be calculated as the ratio of specific heats:

$$K = \frac{C_P}{C_V}$$

Where:
K = Adiabatic exponent
C_P = Specific heat at constant pressure
C_V = Specific heat at constant volume

Calculating K-value using this ratio can lead to errors for some gases. Carbon dioxide is an example.

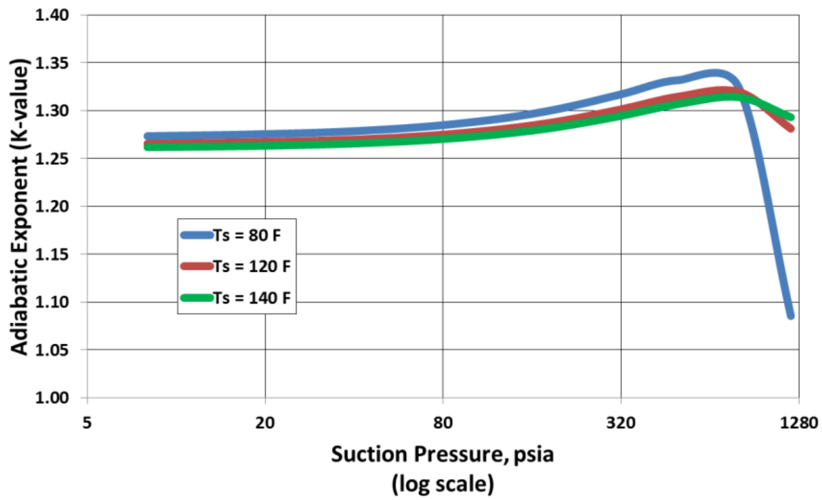

Figure 23
Plot of adiabatic exponent versus suction pressure at three temperatures

Figure 23 shows K-value versus suction pressure for carbon dioxide assuming a compression ratio of 3.0 at suction temperatures of 80 °F, 120 °F and 140 °F. The ratio of specific heats for carbon dioxide is 1.275 at 14.7 psia and 60 °F. At lower pressures this would be accurate and sufficient. But when the suction pressure gets to about 100 psia K-value starts to increase and using 1.275 would introduce some significant error. Note that as the suction pressure approaches the critical point the K-value increases and then decreases rapidly. Small changes in pressure cause relatively large changes to K-value leading to changes in the P-V diagram which changes power and capacity.

CONCLUSION

As stated at the beginning of this paper, the purpose of this tutorial was to provide a straightforward explanation of how a reciprocating compressor works relying heavily upon the pressure-volume diagram. A thorough understanding of the reciprocating compressor's pressure-volume diagram forms the foundation on which more detailed explanations can be developed. Deeper more detailed explanations of gas pulsation, torsional and mechanical vibration, and many other reciprocating compressor topics will originate with the P-V diagram making it the most fundamental building block.

REFERENCES

Hartwick, W., 1968, *Efficiency Characteristics of Reciprocating Compressors*, ASME Technical Paper Number 68-WA/DGP-3.

Hartwick, W., 1975, *Power Requirements and Associated Effects of Reciprocating Compressor Ends Deactivated by Internal Bypassing*, ASME Technical Paper Number 75-DGP-9.