Gas Machinery

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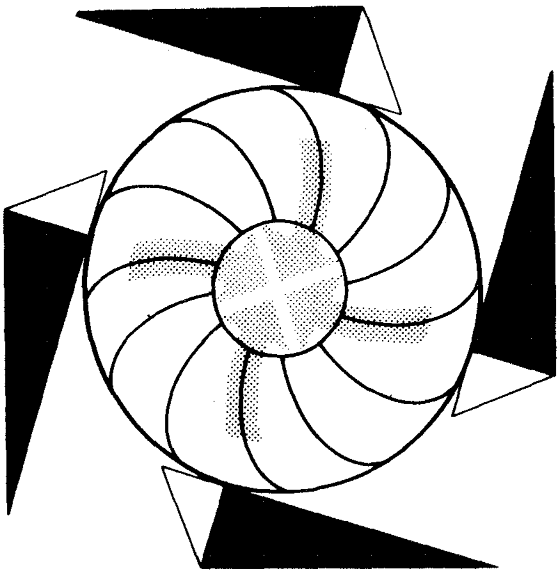
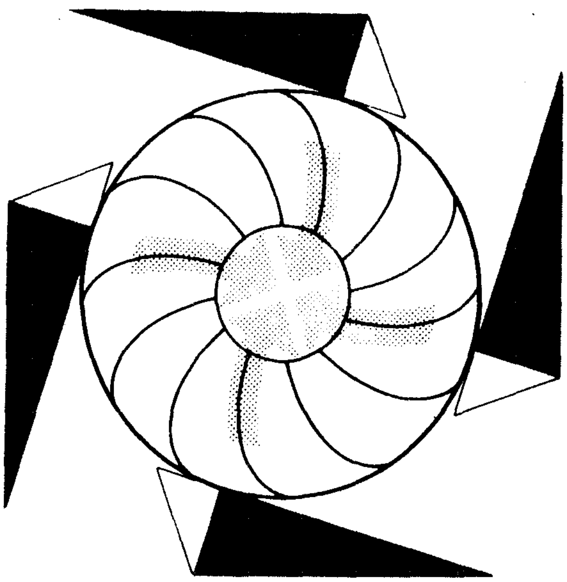


Dedication

This textbook is dedicated to the engineer who wants to make an independent and comprehensive evaluation of the performance capabilities of gas machinery.

The author searched the first 30 years of his career for a more concrete procedure and a common method of evaluating gas compressor capabilities. He has spent the past 15 years analyzing and describing a new unilateral technique which produces realistic compressor performance data.



Preface

I was motivated to write this text because comprehensive technical information concerning the performance of gas machinery is lacking. The contents include the following innovations and features:

1. A series of computer programs to resolve most every piston and centrifugal compressor application.
2. A comprehensive analysis for all types of turbomachinery.
3. The design limitations and metallurgy for rotors.
4. Specific diameters and specific speeds projected to optimum efficiency for fans, blowers and all forms of turbomachinery.
5. Comprehensive methods for projecting the performance data of axial and radial fans, lobe blowers, sliding-vane, spiral-axial, helical-screw and liquid-liner applications.
6. A series of calculation sheets with supporting equations for quick sizing and power appraisal of piston, rotary and turbomachinery.
7. An evaluation of the pulse intensity experienced with piston machinery and a method of sizing the attenuation chambers.
8. Shaft seal leakage and balanced piston evaluations.
9. Simplified charts for evaluating piston compression efficiency.
10. An evaluation of the minimum capabilities of centrifugal compressors in refrigeration service versus piston compressors.

The objective of this text is to advance a technology that produces substantial performance data for all types of gas machinery. These methods are rooted in fundamental aerodynamics and thermodynamics, using empirical data as secondary support material.

The method of rating piston compressors has been successfully applied for over 40 years. The intrinsic valve loss concept has been infallible for the past 11 years. The Baljé system of *specific* performance has proven to be an invaluable tool for over 10 years.

The hypothetical equations concerning the performance of rotary machines needs to be sustained by experience. These data should prove useful in advancing this technology and the reference literature. The analysis which relates the charge and exhaust head losses to the rotor tip speed poses a novel solution, similar to the NPSH requirement of a process type centrifugal pump.

The top of the acknowledgment and gratitude list must go to Dr. O. E. Baljé. His classical treatise on *Design Criteria of Turbomachines*, ASME Papers, numbers 60-WA-230 and 231, formed the matrix of the turbomachinery chapters.

The next parties to whom I am indebted are P.M. Huemmer and C.W. Pace, the ranking officers of Ehrhart & Associates, a division of Procon, affiliate of Universal Oil Products. Without their indulgence and appreciation of my objective, this book would not have been written.

Thanks to my fellow workers and contemporaries, R.F. Neerken, H.M. Rubenstein, J.P. Buchwold, E.S. Perkins and D.R. Jacobson who proofread and appraised the contents. R.F. Cline was invaluable in organizing the computer programs.

Thanks also to the manufacturers who have contributed substance to the text, mainly D. E. Steel of Allis Chalmers and many of his staff; W.F. Hartwick, W.G. McCachen and others from Cooper Bessemer; D.W. Schmitt and others from Joy Mfg.; H.H. Boettcher and T.A. Ammer of Chicago Pneumatic Tool Company.

A last word of appreciation for the feminine touch of the typewriter for Valerie Jelinek and my good wife.

San Gabriel, Calif. 91775

Lyman F. Scheel
June, 1972

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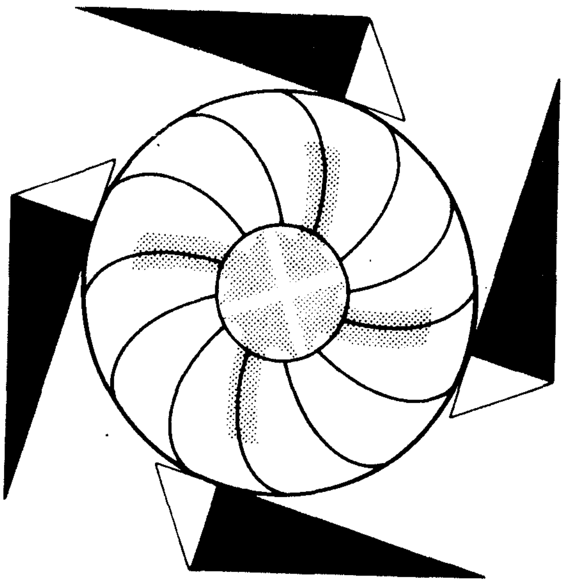


1 A Briefing in Compression

Gas is aeriform; it is without form. It must be confined in a vessel or pipe system in order to have identity. Its mass must be evaluated by a definition of conditions. The American Gas Association, American Standard Association, American Society of Mechanical Engineers and the most credible references in the United States accept the definition that one standard cubic foot (scf) of 60°F dry air at sea level, where the mean barometer is 14.696 psia, must weigh 0.0763 pounds. The European counterpart is the *normal cubic meter* which weighs 2.846 pounds at 14.696 or 760 Torr (*mm*) and zero Centigrade. The British *standard cubic meter* weighs 2.70 pounds at 14.73 psia or 30 inches of mercury barometric and 60°F.

The term *free air* is used to describe the capacity of compressors. It is the net volumetric efficiency corrected displacement at the local ambient conditions. For example, a specific volume of one pound of 60°F dry air at the San Diego Airport contains 13.1 cf; in mile high Denver, it contains 15.4 cf at a 12.5 psia (25.4 inches Hg) barometer. (See Chart 14 for elevation density derating.) The equivalent term for *free air* when concerned with a process gas compressor is actual cubic feet per minute, acfm. The specific volume is determined from the equation:

$$v_s = 10.73\ T(Z/m)P. \qquad (1.1)$$

Equation 1.1 is developed from Avogadro's law. It states that the volume of all gases are proportional to their respective molecular weight, providing the pressure and temperature are held constant. It follows that the molecular weight of air, 28.97, when multiplied by the specific volume, 13.1, produces a volume of 379.5 cf. The volume is common to all gases when the compressibility correction is unity. A vessel containing 379.5 cf of hydrogen gas would contain a net weight of 2 pounds. If filled with propylene, the net weight would be 42 pounds. The Universal Gas Constant is derived from the same mol volume:

$$R = 379.5(14.70)/520(m) = 10.73/m. \qquad (1.2)$$

When the pressure is referred to the pounds per square foot (psf), the gas constant is $1{,}545/m$.

Gas Fluidity

Gas possesses instant fluidity. Any release in pressure from a gas vessel will create a velocity in accordance with the Torricelli (1643) equation:

$$V = F(2gL)^{0.5}, \text{fps}. \qquad (1.3)$$

Table 1.1
Common Nozzle and Orifice Coefficients

Nozzle Description	F-factor
Conoidal mouthpiece	0.98
Short cylindrical nozzle	
Rounded edge	0.92
Sharp edge	0.82
Pitot tube	0.86
Thin orifice plate, sharp edge	
d/D ratio, 0.10 to 0.60	0.65
d/D ratio, 0.61 to 0.74	0.68
d/D ratio limit, 0.75	0.72

The symbol "g" represents the gravitational acceleration, 32.2 ft/sec/sec. The letter "L" represents the head in feet of gas. It is the product of the differential pressure supporting the flow, the specific volume and 144 square inches per square foot. The term gas head has the same significance as a centrifugal pump head; i.e., (144/62.4) = 2.31 feet of water head is equivalent to one psi and so is (144/0.0763) = 1,885 feet of standard air. For example, a nozzle velocity from a 20 psig air tank exhausting into standard ambient conditions is determined in this manner. The average specific volume is

$$v_{sa} = 10.73(520)/29(14.7 + 20/2) = 7.8 \text{ cf/lb.}$$

The head of air is 144(7.8)20 = 22,500 feet. The ideal nozzle spouting velocity is $(64.4 \times 22{,}500)^{0.5}$ = 1,200 fps (by Equation 1.3). The actual velocity would be reduced by the amount of the flow coefficient, F. Several common nozzles and orifice coefficients are given in Table 1.1.

Viscosity is another factor to consider in reference to fluidity. Viscosity is the resistance of fluid particles to shear and the flow of these molecules against the molecular boundary layer. The tenacity of this attraction is measured by the viscosity and manifest in the Reynolds (R_e) number:

$$R_e = 0.105\ (\text{ppm})/d\mu \tag{1.4}$$

Where ppm is the pounds per minute of flow, d is the diameter in inches and μ is the absolute viscosity in centipoise. The effects of viscosity cause two different conditions of flow. The smooth streamline flow which occurs at low velocities of approximately one fps for water and 4 fps for air is known as *laminar flow* in conventional sized pipe 2 to 10 inches. The upper limit of laminar flow is between a R_e of 2,700 and 4,000. Turbulent flow occurs at greater velocities and R_e values. The fluid particles are retained in symmetric layers by viscous action when in laminar flow. The particles move in heterogeneous fashion when in turbulent flow.

Velocity Head (Velad)

Not only does the pressure drop provide the necessary impetus to support the high velocity through a nozzle, but it also provides necessary motive power to propel fluids through long pipelines. The design of new installations requires a valuation of realistic resistances that may constitute the flow line circuits serving the compressor. Such nondescript pipelines can be assumed to have a resistance of 500(L/D). This parameter would include the equivalent length of 2,000 feet of 3-inch pipeline to 390 feet of 16-inch pipeline. This (L/D) ratio is multiplied by a frictional factor of 0.015 to give the number of velocity heads consumed in supporting the required velocity experienced in most plants where the Reynolds number varies from 300,000 to 3,000,000 using 3 to 16-inch pipelines (1). The term *velocity head* has been coined, ***velad***, and is identified as K. It has become a convenient resistance reference for fittings, pipeline facilities, etc. A list of common resistances are listed in Table 1.2.

A fitting having a resistance of one K consumes one velad at that respective velocity. The velocity of gas flowing in pipeline is readily determined from the equation.

$$V = 3.06(\text{acfm})/d^2,\ \text{fps.} \tag{1.5}$$

The last six items in Table 1.2 are the common accessories serving compressor installations. Presume that several compressor cylinders are operating in series. They are required to handle an airflow of 1,000 lb/min at 100 psia and 300°F to the intercooler. The air is cooled to 100°F and returned for the second stage of compression. The facilities also include a gas separator and a pulse damper with a choke on each end of the circuit. The line selection and pressure drop is developed in this same manner.

The hot gas density is 10.73(760)/29(100) = 2.81 cf/lb. The gas head is 144(2.81) = 405 ft/psi. The volume flow is 2.81 (1,000) = 2,810 acfm, and the velocity is 2,810(3.06)/(12 x 12) = 60 fps in a 12-inch pipeline. The unit of frictional resistance in terms of velads is $V^2/2g$ or (60 x 60)/64.4 = 56 ft/K or per velad. The resistance of the circuit from the cylinder through the intercooler is 500(L/D)0.015 = 7.5K; plus 12K for one surge bottle with a choke tube; plus 17K for an intercooler, which makes a total resistance of 36K. The friction is 56(36)/405 = 5.0 psi using a 12-inch pipeline. The cool gas has a density of 2.07 cf/lb. The gas head is 298 ft/psi and volume flow of 2,070 acfm. The velocity is 63 fps in a 10-inch pipeline, and the frictional resistance is 62 ft/K. The resistance of the 10-inch return connection is 500(12/10)0.015 = 9K plus the separator, 7K; plus another PD = 12K, making the total 28K. The pressure drop is 62(28)/298 = 5.8 psi. This frictional loss is too great. A 12-inch pipeline can be substituted and reduce the friction to 2.55 psi, making the total friction of 7.5 psi, or 7.5 percent of the system pressure (100 psia) is consumed in transmitting the gas from the first stage to the second stage cylinder.

Critical Velocity

The ideal spouting velocity example given earlier is only valid within certain limitations. There is a limiting pressure which establishes the maximum velocity flow. Where P_a is the reduced pressure, presumably 14.7 psia, and P_v is the vessel presure, the maximum velocity is attained at

$$P_v = P_a(2/k + 1)k/(k-1). \qquad (1.6)$$

The limiting ratio for several gases are

Helium, P_v = 14.7(2.04) = 30.0 psia;
Air, P_v = 14.7(1.88) = 27.7 psia;
Natural gas, P_v = 14.7(1.82) = 26.8 psia;
Butane, P_v = 14.7(1.67) = 24.6 psia.

The velocities attained from these pressures are the *sonic* velocities for each gas. The sonic velocity is determined

$$\tau = 224(kT_o/m)^{0.5}. \qquad (1.7)$$

Consistent flow characteristics are attained where critical or pertinent flow velocities are related to the decimal fraction of sonic velocity for that respective gas. The critical flow equation is

$$\text{acfm} = 73.3d^2F(kT/m)^{0.5}. \qquad (1.8)$$

Specific Heat

The specific heat of a gas is the amount of heat required to raise one pound of gas one degree F.

Table 1.2
Common Resistances

Resistance Description	K-factor
Reducer contraction	
0.75	0.2
0.50	0.3
Reducer enlargement	
0.75	0.5
0.50	0.6
0.25	0.9
Gate valve	
Fully open	0.15
0.25 open	25.0
Elbow	
Long radius	0.15
Short radius	0.25
Miter	1.10
Close return bend	0.5
Swing check or ball valve	2.2
Tee flow through bull-head	1.8
Angle valve, open	3.0
Globe valve, open	5.0
Filters	
Clean	4.0
Foul	20.0
Intercoolers	17.0
Gas separators	7.0
Surge bottles	
No choke tube	4.0
With choke tube	12.0

There are two forms of specific heats for gases, defined as (a) specific heat at constant volume and (b) specific heat at constant pressure. The heating of a gas under constant pressure requires more energy to satisfy the increased volume change. The specific heat at constant pressure, C_p, is always greater than C_v for this reason. Several simple Perfect Gas Law (PGL) relations between C_p and C_v are as follows:

$$k = C_p/C_v = C_{pm}/(C_{pm} - 1.986)$$
$$= C_{pm}/C_{pm} - 0.993\,(Z_s + Z_d) \qquad (1.9)$$
$$C_p - C_v = 1{,}545/m788 - (Z_s + Z_d)0.993/m \qquad (1.10)$$
$$H = C_p\,(T_2^* - T_1) \qquad (1.11)$$
$$\text{Isehp} = WC_p\,(T_2 - T_1)/42.5 = W\Delta H/42.5 \qquad (1.12)$$
$$P_1V_1k = P_2V_2k = \text{constant}. \qquad (1.13)$$

*Where T_2 is isentropic or adiabatic temperature rise.

Table 1.3
Gas Identification by Molecular Structure

Molecular Structure	"k" value	Typical Gas	Mean Temp. Degree
Monatomic	1.67	Argon, helium	220
Diatomic	1.40	Air, hydrogen	180
Triatomic	1.30	Carbon dioxide, steam	170
Polyatomic	1.20	Acetylene, ethane	135
Heavy Organic	1.10	Butane, benzene	105

The last equation is the fundamental adiabatic equation which indicates the realistic change of state. A further extension of the PV behavior follows:

$$T_2/T_1 = (P_2/P_1)^{(k-1)k} = (V_1/V_2)^{(k-1)} \quad (1.14)$$
$$(V_1/V_2) = (P_2/P_1)^{1/k}. \quad (1.15).$$

The molecular structure is the simplest category to identify a gas with an approximate exponential value "k." Table 1.3 lists these common categories. The mean temperature of a 3 R_c compression from an intake of 80°F is shown in the right column (2).

Discharge Temperature

For all practical design and operating purposes, the temperature rise in a piston machine follows the adiabatic Equation 1.14. The Rankine discharge temperature is the product of the Rankine suction temperature and the ratio of compression raised to the $\sigma = (k-1)/k$ exponent; i.e.,

$$t_2 = T_1[(Z_1/Z_2)R_c)]^{\sigma} - 460.$$

Several well-documented references support this premise (2, 6 and 10). If the enthalpy differential and the discharge temperature can be abstracted from a Mollier chart (Isentropic Method), it is recommended over the adiabatic method using Equations 1.16 and 1.17.

During the CNGA Compressor Tests in 1936, it was noted that the actual temperature rise was less than the adiabatic function at R_c greater than 4.0. This same deviation was observed in a high pressure air test. A high vacuum air test showed a similar deviation at R_c values > 2.5. This indicates that the gas density has a significant effect on the temperature rise. The trends are shown in Figure 1.1 and on Chart 13. The deviation from the abiabatic temperature rise is a result of the reduced mass flow because of the smaller volumetric efficiencies at the higher R_c values. The influence of the heat conduction from the attached pipe header system and from the cylinder jackets is more pronounced with the low mass flow than it is at the normal 3 R_c design point. In all three cases the cylinders were cooled with an abundance of relatively cool water. The compression line AB in Figure 1.2 is relatively longer for a plus 4 R_c than a R_c value < 2. This provides a greater time and opportunity to dissipate more heat from the >4 R_c operation.

The vacuum cylinders used in the test were 42 inches in diameter and had 2.4 percent clearance. They showed an ultimate temperature rise of 1.50 at 7.5 R_c. Beyond this asymptotic crest, the heat leak dominated the greater exponential effect. The mole weight of the gas as well as the gas density in the cylinders apparently affect this deviation. In retrospect, these asymptotic crests are consistent with maximum discharge temperatures experienced under high R_c conditions. It is a rare set of conditions where discharge temperature for an air compressor exceeds 450°F, or 380°F for a natural gas operation. These temperatures are consistent with the 1.5 and 1.9 exponential values for the asymptotic crests shown on Figure 1.1.

An examination of the compression curve AB on Figure 1.2 suggests that the discharge temperature factor could be (P_4/P_1) in lieu of the visual (P_2/P_1) value. The reference material supports the latter, visual R_c basis. An appreciation of the valve action

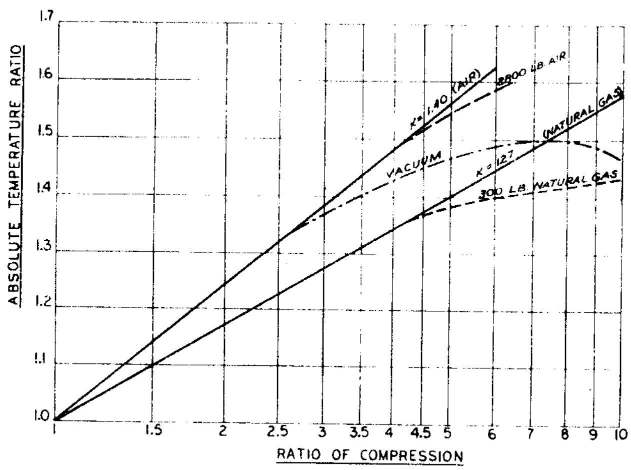
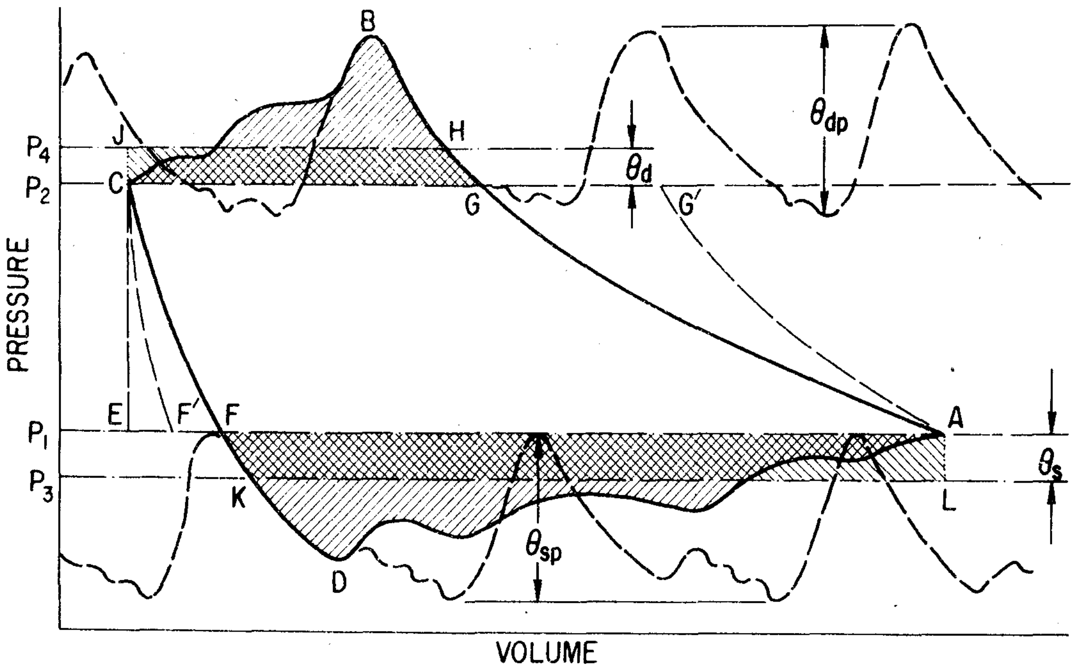


Figure 1.1. Temperature of compression factors vs. ratio of compression. (From Gas and Air Compression Machinery, *Courtesy of McGraw-Hill Book Company.)*



$\theta_s = a^2 m U^2 10^{-4}/T$; $\theta_{sp} = 2\theta_s$

$\theta_d = \theta_s / R_c^{\sigma}$; $\theta_{dp} = 4\theta_s / R_c^{1/K}$; $B = (1+\theta_d)/(1-\theta_s)$

COMPREHENSIVE $R_c = P_4/P_3 = B(P_2/P_1) = BR_c$

THE DASH CURVES ILLUSTRATE THE OSCILLOSCOPE PRESSURE WAVES TAKEN AT THE CYLINDER FLANGE

Figure 1.2. A pressure-volume diagram illustrating the gas behavior inside a piston compressor.

offers an explanation for the discharge temperature rise conforming to the visual R_c. The valve action was once thought to be a matter of pneumatic leverage. The discharge valve would open when the product of the area under the discharge valve seat passage and the internal pressure in the cylinder exceeds the product of the full disk or strip area and the pressure in the cylinder chamber. To unseat the element under these conditions would require internal pressures 30 to 90 percent greater than the discharge pressure. Such pressure differentials are unrealistic. An evaluation of realistic pulse pressures is given in Chapter 3.

The lapped finish of the valve seat and the valve element is seldom equal or better than 10 RMS. It is reasonable to assume that a mean clearance of 16 micro-inches exists between these two components. This clearance provides an area of 0.02 percent of a normal valve area having a lift of 0.080 inch. The minimal blow-by leakage volume is 0.02 percent of the normal flow. The actual head against the closed valves is the full R_c which would make leakage velocity approach the sonic velocity or about 10 times the nominal open valve velocity. The valve leakage would be about 0.2 percent of the cylinder capacity under these conditions. With a minute flow always passing under the valve, the slightest favorable unbalance in pressures would cause the valve to lift. It is presumed that the adiabatic compression function "per se" terminates at point G, Figure 1.2. Greater pressures above line CG are in effect providing the necessary energy to expell the gas (isothermally) into the cylinder discharge channel. The reduced pressures below AF, Figure 1.2, represent the velocity heads that are expended to charge the cylinder.

There is a new arbitrary pressure drop correction that is being applied to the suction pressure of a piston compressor. These corrections may range from one to 10 psi. This presumedly represents the "spring-load" applied to the valve disks. The purpose of the springs was to damp the valve flutter. The force to move the spring 0.080 inch seldom exceeds 2 pounds which would limit the resistance to a fractional psi. The writer has only known of one instance where the compressor diagram toe (point A, Figure 1.2) was less than the system pressure. The latter was 24 inches of mercury, and the toe was 27 inches. These measurements were admittedly on the nebulous side. The valve spring tension was designed for dense gases. When the stiff springs were replaced with standard light-tension springs, the normal capacity was recuperated.

This arbitrary "spring loss" should not be misconstrued to be the equivalent of pulse differential ($\theta_s P_1$), the valve loss factor, expressed as a decimal fraction of the system pressure, times that pressure, P_1. (See Equation 2.10 and Equation 3.1.) It would be sheer coincidence that the arbitrary spring loss was equal to $\theta_s P_1$. The latter is a comprehensive evaluation of the contributing factors. It is used to appraise the extraneous power required in excess of the minimal adiabatic. The comprehensive valve loss is not a constant value, nor does it affect the cylinder charge pressure or the position of Point A, Figure 1.2. When such a pressure is applied, the correction is charging density and the capacity is reduced. Such correction is not consistent with the past 45 years of experience in rating gas compressors.

There is one further condition which can increase the temperature rise above the normal adiabatic function in a piston machine. A broken piston ring and/or a scarred valve disk or seat will permit a back-flow of 2 percent or more of the cylinder capacity. The hot discharge gas leaks isothermally from the compression cycle to the suction cycle. This dilution warms the suction gas which extends the discharge temperature.

An illustrative example follows. Assume that R_c = 3.0, k = 1.28 and σ = 0.2185. The incoming gas is 60°F (520 R); all cylinders are discharging at 202°F (662 R), except one which is 232°F (692 R). Determine the X percent of blow-by:

$$R_c^{\sigma} = 3.0^{0.2185} = 1.272$$

Normal discharge should be 1.272(520) = 662 R. The state of conditions follows:

$$[(1.00 - X)520 + X(692)]\ 1.272 = 1.00(692)$$
$$[520 + (692 - 520)X]\ 1.272 = 692$$
$$219\ X = 30$$
$$X = 14 \text{ percent leakage.}$$

This method of evaluating valve leakage is also valid for larger percentages. Using the same conditions, except that the discharge temperature is 363° F or 823R = T_x

$$[T_O + (T_x - T_O)X]\ R_c^{\sigma} = T_x \quad (1.16)$$
$$[520 + (823 - 520)X]\ 1.272 = 823$$
$$385\ X = 823 - 662$$
$$X = 42 \text{ percent leakage.}$$

Compression Efficiency

The area of "cap" BCGB on Figure 1.2 and the "sole" area AFDA represents the power added to the basic adiabatic power diagram AGCFA. It appears to be completely logical to add the area GHJCG, which has an area equal to BCGB to represent the added discharge burden. Likewise, the suction pressure is reduced to P_3, and the cross-hatched area AFKLA is equal to the area AFDA. The alternate and conventional method is to steepen the slope of curve AG and CF until the area AG′CF′A is equal to the full ABCDA. Reference 6 gives a case where the real "k" value of 1.28 was supplanted with a quasi "n" value of 1.375. A complete energy balance revealed that the Joule's mechanical equivalent heat unit must be 642 ft-lb per Btu in lieu of the universally accepted Joule's mechanical heat equivalent of 778 ft-lb per Btu (7). A comprehensive method of solving for the compresssion efficiency is given in the next chapter.

It should be borne in mind that the exponent and its corrections only effect the slope of AG and CD. The exponent σ applied to R_c in the denominator is partially cancelled by the σ factor in the numerator of the adiabatic head equation:

$$L_{ad} = (R_c^{\sigma} - 1)1{,}545TZ/\sigma m. \quad (1.17)$$

The significance of the exponent correction has been abused. The valve losses are of much greater consequence, and their resistance affects are ignored. Equation 1.9 and 1.10 show the manner in which the compressibility corrections are applied as a pseudo polytropic function. In lieu of using the average compressibility as proposed by R. S. Ridgway, many application engineers prefer to use a correction factor of $(Z_s + Z_d)2Z_s$. This is subterfuge to inflate the power requirement. Ridgway's averaging method has been entirely satisfactory for a quarter of a century in solving the gamut of applications (8). The calculated (PGL) volumes are nebulous along the saturated vapor line. Mollier charts offer more reliable data. This also applies to heavy hydrocarbons operating at *ultra-high pressures.*

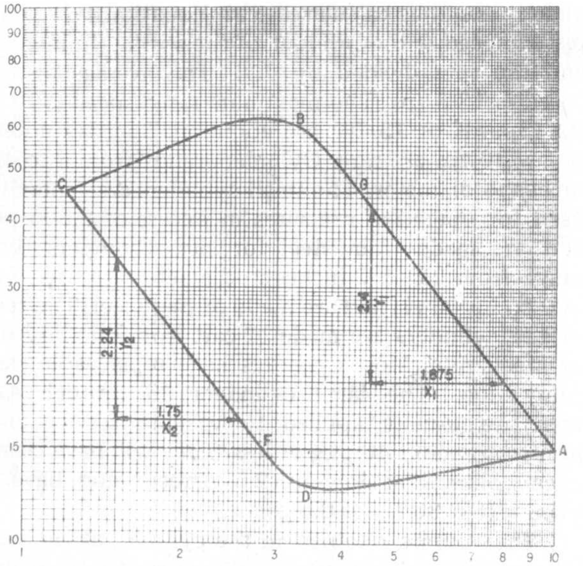


Figure 1.3. Compressor P/V diagram plotted on log/log chart.

These few cases would constitute the exception rather than form the rule. It is regretable that issues of this order which have little significance on the general procedures can create such a flurry of interest, while the gas technology in general is so lax in other comprehensive fundamentals.

Thermodynamic Behavior

Considerable dissertation has been given in the previous section concerning the four thermodynamic modes of P—V change. The first premise presumes that the temperature remains constant during a change of conditions where $P_1 V_1 = P_2 V_2 =$ Constant, for the isothermal mode. An instance where isothermal flow can be found is in the ideal throttling from an elevated pressure to a lower pressure through a control valve. The expansion of the gas after the compressor valves are open, points ≫ to B to C, and from points F to D to A in Figure 1.2, are other examples.

Adiabatic Behavior

The adiabatic (or isentropic) mode of compression requires the least power, except the isothermal, which is not a realistic mode. The adiabatic mode constitutes the best reference for all efficiency statements. Equations 1.13, 1.14 and 1.16 govern the realistic *adiabatic* change of state. These ideal changes presume that no heat is added or extracted. Piston friction is presumed to be negligible. When the added concession of reversibility is included in a cycle, it is known as an *isentropic* process. The piston compressor is the closest approach to an ideal adiabatic process that can be found in nature or in fact. When the power is calculated from the gas head using the Universal Gas Constant, the head is referred to as the *adiabatic* head and the power as the *adiabatic* horsepower. This is the second mode.

When the gas head is determined from enthalpy table or Mollier charts, complete reversibility is presumed. Power calculated from the enthalpy is identified as *isentropic* horsepower and constitutes the third mode. This terminology is used to provide each term with a more specific meaning (3, 4 and 5).

Polytropic Behavior

Figure 1.2 can be transposed to a log/log graph and have the general appearance of Figure 1.3. The slope of Y_1/X_1 is the compression line AG. The slope of Y_2/X_2 is the expansion line CF. Both of these slopes on an ideal diagram are equal to the ratio of specific heats, "k." The *cap* BCG and the *sole* AFD represent the isothermal flow through the valves. The area of the rectangle represents the power of compression. The slopes of AG and CF are the unadulterated true behavior of an ideal compressor. Leakages up to 2 percent are not apparent and consequently are negligible. Where the dominate leakage is past the piston rings, the compression and the expansion slopes are decreased. This should reduce the "k" value, but actually the "warm-up" dilution of the suction charge has the effect of a larger "k" valve. Where the dominate leakage is through the suction valve, the affects are the same. Where the discharge valve leakage is dominate, both slopes are increased as well as apparently increasing the "k" value.

Consider the preceding thermal-leakage example and apply that data to Figure 1.3. Whereas the slope "n," Y_1/X_1, would be 1.350 and $\sigma' = 0.259$ to match the abnormal discharge temperature of 232° F.

The pseudo polytropic efficiency for the greater slope is $\sigma/\sigma' = 0.2185/0.259 = 84.5$ percent. This ratio develops the spot polytropic efficiency, but it does not represent a consistent margin required between the basic adiabatic power and the actual power requirement. It more accurately represents the affect of a 14 percent internal leakage. The effect of further exponential corrections can only increase the slope of the compression and expansion curves. The slopes can be raised toward the perpendicular to include an area equivalent to the "cap" BCG and the "sole" ADF to create a new rectangle AG′ CFA′. This would only serve this one particular special case, and the results could not be projected to other instances with any degree of confidence.

Comprehensive methods of evaluating the magnitude of the valve (cap and sole area) losses are developed in subsequent chapters. It is important to know that all piston and rotary displacement compressions are essentially adiabatic processes. The abnormal discharge temperature rises ($<4R_c$) are most likely the pseudo polytropic effect of internal by-passing. The only true polytropic compression is experienced in an axial or centrifugal compressor. The super-adiabatic heating effects are the result of intrinsic friction at high velocities, >0.4 MACH.

The polytropic efficiency is an important device to project the discharge temperature of a centrifugal compressor. In fact, the abnormal amount of heating experienced over the adiabatic is the method used to evaluate the efficiency of centrifugal compressors. This data is largely reproducible (10). Again referring to the leakage example, assume that it is a centrifugal compressor and has a discharge temperature of 253°F. The polytropic discharge multiplier is 520/(460 + 258) = 1.38. The exponent is $l_n 1.38/l_n 3.0 = 0.292 = \sigma'$. The polytropic efficiency is σ/σ' or 0.2185/0.292 = 75 percent.

The rotary displacement compressors are also adiabatic machines that have a pseudo polytropic effect on the discharge temperature. Again, repeating the advise that it is better to evaluate the machine as an adiabatic function and examine the discharge temperature as an index of the quantity of internal by-passing.

Capacity

The volume sweep of the piston is termed the cylinder displacement. It is measured in cubic feet per minute (cfm). It is determined from the equations:

$$\text{cfm} = 0.00091LN(D^2 - 0.5d^2) \tag{1.18}$$

$$\text{cfm} = 0.327U(D^2 - 0.5d^2) \tag{1.19}$$

$$\text{cfm} = 4.2\,D^2\,(U/13.3 \text{ fps}). \tag{1.20}$$

Where L = length of stroke D is the diameter of the cylinder and d is piston rod diameter, all in inches. U is the average piston speed in feet per second (fps.). Equation 1.20 offers a quick displacement with acceptable accuracy for 10-inch and larger cylinders having 2 to 4-inch piston rods. The capacity of cylinders having bores smaller than 8 inches should be calculated with Equation 1.19. The piston speed is the product of the stroke (in inches) and the rpm divided by 360. The average piston speed is 13.3 fps. The maximum rated speed is 18.3 fps. Speeds of 15 fps are common. Speeds quoted below 10 fps are rare. Cylinders range from 4 to 42 inches in diameter. The small cylinders are generally used for operating pressures in excess of 2,500 psi and are made from steel forgings. Cylinders that range from 12 to 6 inches in diameter are used for 800 to 3,000 psi and are made of cast steel. The rest of the cylinders are made of cast iron, ductile iron or Meehanite. Most cylinders have replaceable cast iron liners. The large cylinders—36 through 42-inch—are usually applied to vacuum service and limited to about 40 psig or less. They will have clearance volumes as low as 2.5 percent and have a blank-off at 30 R_c or 0.5 psia.

The volume retained in the cylinder at the end of the stroke is called the clearance gas. It is expressed as a percentage of the volume sweep. The minimum clearance for a typical cylinder is about 10 percent. The clearance can be estimated by totaling the volume of void space within the cylinder. The clearance between the piston and the head is about 0.100 inch at the head and crank end dead centers. Each valve occupies a volume equal to its port area and one inch deep. The sum of all valve volumes in each end plus the piston clearance divided by the piston sweep (piston area times stroke) equals the percent of clearance. For example, a 20 x 15-inch cylinder has a cross-sectional area of 314 square inches and a sweep of 4,710 cubic inches. The cylinder has four valves in each end that are 8 inches in diameter. The 8-inch valve area is 50 square inches or a total of 200 cubic inches. The estimated clearance is (200 + 31.4)/4,710 = 4.9 percent. The clearance is increased by the manufacturer when the cylinder displacement is too large for the design conditions. This is accomplished by either turning down the piston thickness or adding to the cylinder head spacer. The clearance gas trapped at the end of the stroke expands in reversible mode as illustrated by the curve CD on Figure 1.2. The amount of gas pumped with each stroke is represented by the ratio of lines AF/AE. This is known as the volumetric efficiency. The equation that best depicts this function is

$$E_v = 100 + C\,(1 - \Lambda R_c 1/k) \text{ or}$$

$$E_v = 100 + C - C\Lambda R_c{}^{1/k}\,(Z_s/Z_d). \tag{1.21}$$

The decimal fraction of Lambda (Λ) represents the percentage of clearance gas leakage that escaped past