The Intrinsic Method of Rating Piston Compressors

L. F. Scheel

Consultant for Gas Machinery

Follow this and additional works at: https://docs.lib.purdue.edu/icec


This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
THE INTRINSIC METHOD OF RATING PISTON COMPRESSORS.

Lyman F. Scheel, Fellow Member ASME
Consultant & Author, GAS MACHINERY
San Gabriel, Ca - 91775

INTRODUCTION

This article presents a comprehensive method of evaluating the "compression efficiency" of a piston gas compressor. The losses are related to the maximum gas velocity through the valves. This valve velocity is in turn related to the action of the piston and amplified by the ratio of the piston/valve area. The procedure produces the mean valve resistance as a decimal fraction of the suction and the discharge system pressure. It also projects the magnitude of peak pressure pulses that is anticipated in each header system.

The thesis of this article holds the piston gas compressor to be an adiabatic or isentropic process. When the head is derived from the exponential "k" factor, the process is deemed to be adiabatic. The discharge temperature relates to the visual header Rc and the G exponent.

When the process is evaluated in terms of enthalpy and entropy from Mollier Charts or Gas Tables, it is deemed to be an isentropic function. The latter must be reversible and the Chart and Tables are completely reversible, hence the process must be isentropic.

The applicant should not be concerned over these thermodynamic terms. If there were simpler words they would have been used. The example given in Table I illustrates that the most complex technique only requires Rc raised to the G power. This is a simple manipulation on a log-log slide rule. With the event of a desk satellite computer just around the corner, it is possible to make a complete report concerning the compressor sizing and rating in a matter of two minutes, without the aid of a slide-rule or a stenographer.

A normal adiabatic discharge temperature signifies well seated valve elements and a good ring seal in the piston. It also indicates optimum compression performance. Super-adiabatic temperatures indicate the magnitude of valve and ring leakage. It also indicates the degree of inefficiency.

References 1, 3, 4, 5 and 7 contain data which demonstrate that the temperature of compression does relate to the ratio of compression (Rc), using visual line connection pressures. This is true for ratios of compression less than 4-6. Greater compression causes lower volumetric efficiencies which reduce the cylinder mass flow. The influence of the heat leak from the machinery and the piping connection become manifest and the discharge temperatures are sub-adiabatic.

P-V DIAGRAM

Figure 1 shows typical pressure (ordinate)-volume (abscissa) diagram. The adiabatic or isentropic features are described as follows. The compression line is A3. Above point C, the pressure continues to rise to point B, but the temperature does not rise beyond point C. This is an isothermal behavior caused by the escape of the cylinder gas into the discharge line through the opening of the discharge valves. This flow peaks at condition B. This peak pulse (dp) is the damaging surge which permeates the discharge piping system. The area of the cap-like figure above the discharge pressure represents the flow resistance of the valves and the cylinder channels. The average resistance is the quotient of the area.########/Qc and designated Qd. The discharge valves are closed at point C. The interstitial residue is known as the clearance gas and is represented graphically as CP2. It expands isentropically to F1, recovering the energy represented by area CEP. The suction valve starts to open at point F. The flow continues isothermally to point D where the suction valves are wide open. The cylinder filling operation continues with reduced differential head until the cylinder pressure reaches point A. At this point, the cylinder and the suction line pressures are in equilibrium at a pressure of F1. The slipper-like figure AFD represents the energy spent in drawing the gas into the cylinder. The heel of the slipper or point D represents the peak pulse (sp) which permeates the suction piping system. The average height of the slipper is 8s. The peak suction pulse, sp, is usually twice the average pulse 8s. The
The discharge pulse is related to the suction pulse by the approximate relation:
\[ \Theta_d = 4 \frac{\Theta_s}{R_0 C_k} \]  
\[ \Theta_s = (a - U_2) \frac{F}{T_1} 10^4 \]  

Another factor which confirms that the piston compression is an adiabatic function is demonstrated by a replotted indicamgram on log/log chart. Fig 2 is a replotted of a piston-parallelograph (GROSBY) indicator card. The PV slope of the compression line AG is 1.28. This \( k \) value is the ratio of specific heats which is unique for each gas. This would be typical for a "wet" natural gas. The adiabatic pressure of cold water, the heat leak of the clearance gas to the jackets have been flushed with an abundance of cold water, the steepness of the CF slope is slightly reduced from 1.28 to 1.255. This is equivalent to a 15 percent heat leak of the clearance gas to the coolant. There is no evidence of such heat leak from the compression cycle. A greater mass flow, a greater piston speed and a less favorable heat transfer condition contribute to this fact. Ref 7.

Table 1 gives the Input Data and a Piston Sizing Calculation Sheet for a typical application. The problem requires the selection of the cylinder bore, the comprehensive compression efficiency and method of correction, the thrust and power evaluation. The data given under the \( \text{TVIN CYL} \) column is for the parallel 18.5 in. cylinders having 10% clearance and 7.5 in. Piston/Valve area. The mechanical loss is the equal to the SCR of the Gas hp per cylinder, which totals 55 hp in lieu of 44 used for the single cylinder. The area of the twin pistons is determined by taking half of the single cylinder area.

Had the clearance been greater, the new \( \Theta_v \) would have been lower and the bore somewhat larger. The method of cylinder load control is obvious from lines 1, 4 and 5. Fig 11, 12 and 13 are also useful for adjusting the cylinder load. Most cylinders are equipped with some means of adding supplementary clearance. The constant "10" in line 5 is the proximate rod displacement. In the instance of a small bore, high pressure cylinders which require a much larger piston rod, the constant "10" can be readily adjusted to suit.

The piston thrust load for 2000 hp cylinders is usually to 60,000 pounds or a piston rod tensile load of 10,000 psi. A single cylinder 26.2 inches in diameter nearly doubles these two criteria. Two parallel 18.5 inch cylinders satisfy the rod load requirements. The manufacturer stipulates the magnitude of these criteria. If the rod capacity is fixed at 40,000 pounds, then a third parallel cylinder must be provided.

Equations 1 and 2 are applied to lines 9 and 10. The first comprehensive compression efficiency is 77% and requires 1944 Bhp or 20% more power than is required by the Twin cylinders. The significant difference being the "a" ratio of 13 compared to the 7.5 "a" ratio for the TWIN cylinders. Figures 4 through 9 show the influence of the "a" ratios, the piston speed and the mol weight, on the Intrinsic "M" factors and the compression efficiency. The significance of the mol weight is best revealed in Fig 8. The influence of the piston speed and the reciprocal of the MACH number, has a linear effect on the compression efficiency. This is shown in Fig 9.

The symbols that are used in this article and not described in the Nomenclature, are given below. \( \text{mcfd} \) refers to the weight of gas flow in pounds per minute, \( Q \) refers to \( \text{acfm} \), \( R_5 \) indicates the speed of rotation, rpm; \( R_5 \) indicates the piston rod diameter and \( R_6 \) is the piston stroke in inches. \( U \) is \( \frac{A_2 R_5}{360} \), fps. \( Q_d \) is the cylinder displacement and is approximately equal \( Vd \) of \( \frac{2\pi}{3} \) (rpm/800). The latter allows for the piston speed. \( A_p \), \( A_r \), \( L_1 \), \( L_2 \) and Edy refer to the area of the piston and the piston load, the thrust load in pounds, rod tension in psi and the dynamic efficiency.

![Fig. 1 Compressor indicator P-V diagram illustrating valve loss effect on power requirement.](image-url)

\[ \text{Visual } R_0 = \frac{P_2}{P_1}; \text{ Comprehensive } R_0 = \frac{P_2}{P_3} = B R_0 \]

\[ B = \frac{(1 + \Theta_3)}{(1 - \Theta_2)} \]
The principal reason for developing equations (1) and (2) was to establish the ratio of internal pressures actually experienced by the piston. To accomplish this, it is necessary to know the magnitude of the discharge valve loss to add to the visual discharge pressure, and the suction valve loss to subtract from the suction visual or stagnation pressure.

The suffix "x" represents the operation being changed. Fig. 8 shows the marked effect that five widely different gas densities have on the "B" resistance factor.

The term compression efficiency has never had a precise definition. It was most generally understood to be the ratio of adiabatic power divided by the actual power, exclusive of mechanical friction losses. The mathematical expression in equation (4) is the simplest form for giving the compression efficiency. It can be stated as the line B less one, divided by the intrinsic "B" less one, where "B" is the algebraic
MECHANICAL EFFICIENCY

The advent of the large integral gas engine and balanced opposed compressor frame introduced a 95-percent mechanical efficiency in 1951. The average power carried in each cylinder of the early units was about 400 hp. Prior to this, the twin units carried about 100 hp per cylinder and the mechanical efficiency was well-established at 90 percent. The tests reported in reference (4) showed a mechanical efficiency no less than 97.5 percent. The SQR of these cylinder loads approximates the mechanical friction and gives substance to the mechanical efficiency equation 5. It is reasonable to expect the friction to decrease on an incremental basis rather than follow a flat percentage.

$$\eta = \frac{(hp - b\cdot 0.5)}{hp}$$
VALVE VELOCITIES

There is no optimum valve velocity. A valve velocity of 8000 fpm may be satisfactory for air-compressor service. Butane gas applied to the same machine could account for an over-loaded drive. A light gas mixture of hydrogen in the same compressor would provoke excessive valve maintenance from the unloaded fluctuating action. The valve velocity expressed as a Mach number has a more comprehensive significance. The Mach number is the decimal fraction of a given velocity as related to the sonic velocity of that gas.

Sonic velocity ($T$) = $224 (KT/A)^{0.5}$ (18)

The sonic velocity for 60 F air is 1220 fps, butane 695 fps, and hydrogen 6.5 mol weight mixture is 2350 fps. A valve velocity of 8000 fps has an air Mach number of 0.119, 0.192 for butane, and 0.057 for hydrogen. Resistances equivalent to the 8000-fpm air velocity would be realized at 4950 fps for butane and 16,800 fps for the hydrogen mixture.

Rather than debate ambiguous velocities, it is more prudent to evaluate the valve losses in terms of compression efficiency, which can be reduced to operating power costs. The horizontal iso-efficiency (line of constant efficiency) in Figs. 4 through 6 convey this information. Fig. 9 resolves the same advice for gas in terms of Mach numbers and average piston speeds.

Modern process-type cylinders are usually provided with piston/valve "a" ratios of 8 to 12. Early 1930 models were equipped with "a" ratios of 20 and greater. An "a" ratio of 5 or less requires a high-lift, poppet-type valve. A lift of 0.080 in. is common for disk-type valves in 1000-psig service. This lift may be reduced to 0.050 in. for light gases at 2000 psig and greater pressures. When the mol weight is less than 10, a lift of 0.030 in. has demonstrated a reduction in valve maintenance without causing a power penalty. Lifts of 0.100 in. are common for 100 psig and lower pressures. Nylon poppet-type valves with 0.250 lift have rendered excellent service at speeds of 600 rpm in 1000-psig service.

\[ \theta = \text{valve loss, decimal fraction of system pressure} \]
\[ \Lambda = \text{clearance-gas leakage factor, usually 1.10} \]
\[ \Delta = \text{difference in psi, temp, or } R \]
\[ \sigma = \text{adiabatic exponent } (k - 1)/k \]
\[ \Sigma = \text{horsepower per million cubic feet} \]
\[ \Sigma = \text{horsepower per molal ppm} \]

NOTE: The suffix 1 and s denotes the suction, T, and Z condition. The suffix 2 and d denotes the discharge, F, T, and Z condition. Other special suffixes are described as applied.

REFERENCES

POWER REQUIREMENT

Fig. 10 has been prepared for rapid calculation of the power requirement and compression efficiency. The following equation gives the adiabatic horsepower per molal pound per minute of throughput.

\[
\bar{E} = 24.3 \left( \frac{R_0^2 - 1}{\gamma} \right)
\]

Fig. 10 Adiabatic horsepower per molal pound per minute flow at 60°F inlet. Note: Circled numbers are visual ratio of compression

---

NOMENCLATURE

- \( a \) = piston/valve area ratio
- \( acfm \) = actual cubic feet per second, flow
- \( bhp \) = brake horsepower
- \( B \) = intrinsic \( R_0 \) factor
- \( C \) = cylinder clearance, percentage
- \( cfm \) = flow, cubic feet per minute
- \( C_p \) = mean molal heat capacity, Btu per molal pound per deg \( F \)
- \( d \) = diameter of compressor cylinder, in.
- \( f \) = resistance coefficient, dimensionless
- \( ft \) = feet
- \( fpm \) = velocity, feet per minute
- \( fps \) = velocity, feet per second
- \( E_v \) = volumetric efficiency
- \( F \) = degree Fahrenheit
- \( g \) = acceleration of gravity, \( 32.2 \) ft/sec/sec
- \( H \) = enthalpy, Btu/lb
- \( k \) = ratio of specific heats at mean temperature compression
- \( hp \) = horsepower
- \( M \) = molecular weight of gas
- \( Mcfd \) = thousand cubic feet per day
- \( MMscfd \) = million standard (14.7 and 60°F) cubic feet per day
- \( n \) = effective polytropic specific heat ratio
- \( psi \) = pressure, pounds per square inch
- \( psia \) = pressure, pounds per square inch absolute
- \( ppm \) = pounds per minute
- \( ppm/lb \) = molal pounds per minute
- \( pps \) = pounds per second
- \( P \) = pressure, psia
- \( rpm \) = revolutions per minute
- \( R_c \) = ratio of compression, \( P_2/P_1 \)
- \( R \) = universal gas constant, \( 1545 \) ft-lb/deg R
- \( T \) = absolute temperature, degrees Rankine
- \( v \) = average piston speed, fps
- \( V \) = velocity, feet per second
- \( V_0 \) = specific volume, cf/lb
- \( velad \) = velocity head, \( V^2/2g \)
- \( X \) = unity gas constant as related to the adiabatic head
- \( Z \) = gas compressibility factor
- \( \eta \) = efficiency factor

Fig. 12 Volumetric efficiency for dry natural gas, \( k = 1.28, n = 1.25, \Lambda = 1.1 \)

Fig. 13 Volumetric efficiency for LPG and heavy hydrocarbon gases, \( k = 1.13, n = 1.116, \Lambda = 1.1 \)
\[ \theta_s = 2.4 \quad \int a^2 \, m \, u^2 \, 10^{-5} / P_1; \theta_{sp} = 2 \, \theta_s \]
\[ \theta_d = \theta_s / R_c \quad \theta_{dp} = \Delta \theta_s / R_c \quad B = (1 + \theta_d) / (1 - \theta_s) \]

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 COMPRESSOR INDICATOR P-V DIAGRAM ILLUSTRATING VALVE LOSS EFFECT
FIG. 2

COMPRESSOR INDICATOR P-V DIAGRAM PROJECTED ON LOG-LOG PAPER. The ratio of specific heats "k" value of gas = \( \frac{Y}{X} = \frac{2.4}{1.875} = 2.24/1.75 = 1.28 \) for natural gas.