THEORY AND PRACTICE OF NATURAL GAS COMPRESSION

Danny Miller

Compressor Systems, Inc.

ABSTRACT

The purpose of this paper is to provide practical information on the selection of natural gas compressors. No claim is made that this selection procedure is unique -- this information is drawn from experience and data from many sources. The intent was to keep the commercial aspect out of the paper and to try to provide a valuable source of information to be used by anyone.

The paper will present practical design calculations used to size natural gas compressor packages for field application. The first part of the paper will be a combination of definitions and concepts that are involved in compression and the second part will concern itself with the sizing of the compressor package.

DEFINITIONS

Atmospheric Pressure is the pressure exerted by the earth's atmosphere as measured by a barometer. This pressure at sea level is equivalent to the pressure exerted by a column of mercury, one inch square and 29.92 inches high, and at sea level it is normally 14.7 psia.

At elevations above sea level, the pressure exerted by the earth's atmosphere is less than 14.7 psia. It is very important when sizing a compressor that the elevation be known, because it affects the pressure conditions (especially in low-pressure applications) and the available compressor horsepower if the elevation is over 3000 feet.

Gauge Pressure is the pressure above atmospheric pressure as shown on the dial of a pressure gauge. This pressure is expressed in pounds per square inch, and is abbreviated "psig." When submitting an inquiry for a compressor, always list the pressure conditions as gauge, when possible.

Absolute Pressure equals gauge pressure plus atmospheric pressure and is abbreviated "psia."

Example: 20 psig (gauge pressure) is equivalent to 34.7 psia (absolute pressure) at sea level and 33.16 psia at 3000 feet elevation. See Figure 1.

Ratio of Compression is the ratio of the absolute discharge pressure to the absolute suction pressure. It is normally abbreviated "R."

Example: Find the ratio of compression when compressing a gas from 34.7 psia to 114.7 psia:

$$R = \frac{\text{absolute discharge pressure (psia)}}{\text{absolute suction pressure (psia)}} = \frac{114.7}{34.7} = 3.3$$

Ratio of compression is normally limited at 5 to 1 on single stage and 4 to 1 on two or more stages; however, we have gone to higher ratios when the above factors permit. The limiting factors are rod load, maximum working pressure of the compressor cylinder, and the resulting discharge temperature.

Absolute Temperature. Absolute zero is the temperature at which all molecular motion ceases; minus 460° on the Fahrenheit scale. Absolute temperature is the reading of a thermometer in degrees Fahrenheit plus 460°F, and is expressed in degrees R (Rankine).

Example: Find the absolute temperature equivalent to 80°F:

$$^{o}R = 80^{o} + 460^{o} = 540^{o}$$

Always use absolute temperatures in compressor calculations.

Ambient Temperature is the surrounding temperature, normally 100°F in the Southwest and California, and could be considered lower in northern areas.

Approach. Many customers request that the compressed gas be cooled. With oil field compressors, it is normally more economical to cool with air than to cool with water. Coolers that use air as the coolant economically cool to within 10°F, 20°F or 30°F of ambient. A gas cooler designed for 30° "approach" will cool the gas to 130°F when the ambient temperature is 100°F, and to approximately 120°F when the ambient temperature is 90°F.

Boyles' Law. This law states that if the temperature of a gas remains constant, its volume varies inversely with the absolute pressure. This is expressed by the formula:

$$P_1 V_1 = P_2 V_2$$

Charles' Law. This law states that if the pressure of a gas remains constant, its volume varies directly with the absolute temperature. This is expressed by the formula:

$$V_1 T_2 = V_2 T_1$$

By combining Boyles' law and Charles' law, it will be noted that if the volume of a gas remains constant, its absolute temperature varies directly with the absolute pressure. These relationships are expressed by the following formula:

$$P_1 V_1 T_2 = P_2 V_2 T_1$$

Abbreviations of Capacity

CFM = cubic feet per minute
CFH = cubic feet per hour
CFD = cubic feet per day
MCFD = thousand cubic feet per day
MMCFD = million cubic feet per day
MMSCFD = million standard cubic feet per day

Specific Gravity is the ratio of the molecular weight of the gas to the molecular weight of air taken at 28.9, and is abbreviated "SG." The specific gravity of air is 1.0.

Example: Find the specific gravity of a gas having a molecular weight of 18.8:

$$SG = \frac{\text{molecular weight (MW) of gas}}{\text{molecular weight (MW) of air}} = \frac{18.8}{28.9} = 0.65$$

Whenever possible, always list the specific gravity when submitting an inquiry.

"N" Value of Gas is the ratio of specific heat at constant pressure to specific heat at constant volume. The ratio of the specific heats for natural gas is normally 1.26 and for air it is 1.41. See Figure 2.

Maximum Allowable Working Pressure (MAWP) is the maximum continuous operating pressure for which the components of a compressor package have been designed: cooler section, compressor cylinders, piping, etc.

Rated Discharge Pressure is the highest continuous operating pressure to meet the conditions specified by the customer for the intended service. This is always less than the MAWP by at least 10 percent or 15 psi, whichever is greater, for the operation of safety valves. Always keep rated discharge pressure below the working pressure of the weakest component.

Supercompressibility and Superexpandability. If the actual volume a gas occupies at a given temperature and pressure is less than the volume according to the natural gas law, we use the term "supercompressibility" and this reaches a maximum at the critical temperature and pressure of the gas. If the actual volume a gas occupies at a given pressure and temperature is more than the volume according to the natural gas law, we use the term "superexpandability."

SINGLE AND MULTI-STAGE COMPRESSION

With single-stage compression, the total pressure differential of the compressor is handled in one ratio or stage. With multi-stage compression, the gas is compressed to some intermediate pressure in the first stage, passes through an intercooler where the gas is cooled, through a separator where liquids formed by cooling are removed, and into the second-stage cylinder where it is further compressed. As the gas passes through the intercooler, separator, and associated piping, a small pressure drop results. If the final compressor discharge pressure is reached in the second stage, it is a two-stage compressor; if this pressure is reached in the third stage, it is a three-stage compressor, etc.

Whether to use single-stage or multi-stage compression depends on many factors: pinload encountered; cylinder working pressure; resulting discharge temperature; horsepower savings; and economics. Normally, Ariel uses single stage for ratios up to 5:1, and two stages for ratios up to 16:1, if the above factors permit.

Multi-stage compression yields higher volumetric efficiencies than single-stage, requiring less piston displacement and providing lower cylinder cost. Counteracting this saving in cylinder cost is the additional cost for the intercooler, interstage scrubber, and associated piping. Volumetric efficiency will be discussed later. The discharge temperature encountered when compressing natural gas is not excessive. There is less maintenance due to longer valve, packing, and ring life because multi-stage uses lower ratios and temperature.

Single-stage compression should be used when possible. The original cost is less, the difference in maintenance is not excessive, there are capacity and horsepower savings at ratios less than approximately 5.5:1, and there are fewer parts.

CALCULATING HORSEPOWER REQUIRED

Example Calculation of Horsepower Required for Single-Stage Compression (Ex. 1)

Determine the horsepower required to compress 1.9 MMSCFD measured at 14.65 psia and 60^{o}F from 80 psig to 300 psig. Conditions are as follows: Suction temperature is 80^{o}F, ambient temperature is 100^{o}F, elevation of 1500 feet, and specific gravity of 0.64.

Step 1: Prepare the following table:

Ps	Pd	R	Ts	Td	BHP/MMCFD	Cap. (MMCFD)	BHP

Where:
- Ps = absolute suction pressure
- Pd = absolute discharge pressure
- R = ratio of compression
- Ts = suction temperature
- Td = discharge temperature
- BHP/MMCFD = brake horsepower required to compress 1.00 MMCFD
- Cap. (MMCFD) = capacity in MMCFD measured at 14.4 psia and suction temperature
- BHP = brake horsepower

Step 2: Find the "N" value using Figure 2.

Step 3: Convert MMSCFD to MMCFD measured at 14.4 psia and suction temperature:

$$MMCFD = 1.90 \times \frac{14.65}{14.40} \times \frac{540}{520} = 2.01$$

Place this figure under Cap. (MMCFD).

Step 4: Determine the absolute suction and discharge pressure at 150 feet elevation (use Figure 1):

Absolute suction pressure = 80 + 13.9 = 93.9 psia. Place this figure under Ps.

Absolute discharge pressure = 300 + 13.9 = 313.9 psia. Place this figure under Pd.

Step 5: Determine ratio of compression:

$$R = \frac{\text{absolute discharge pressure}}{\text{absolute suction pressure}} = \frac{313.9}{93.9} = 3.34$$

Place this under R.

Step 6: Determine discharge temperature based on a suction temperature of 80°F, ratio of 3.34 and "N" value of 1.26:

From Figure 3 or from the formula $T_2 = T_1 \times R^{\frac{N-1}{N}}$, the discharge temperature is 230°F.

Place the suction and discharge temperatures in their respective columns.

Step 7: Determine BHP required to compress 1.00 MMCFD of gas measured at 14.4 psia and suction temperature at a ratio of 3.34:1, when the "N" value is 1.26. From Table 1, the BHP/MMCFD is 73. Place this figure under BHP/MMCFD.

Step 8: Determine the BHP required, knowing the BHP/MMCFD and MMCFD required from the following formula:

$$BHP = \frac{BHP}{MMCFD} \times MMCFD = \frac{73.0}{1.0} \times 2.01 = 146.7$$

Place this figure under BHP.

Step 9: For this application, offer a JG2-1 with F1197 which has 150 BHP available for compression at 1500-ft elevation.

This JG2-1 will deliver:

$$\frac{150.0}{146.7} \times 2.01 = 2.05 \text{ MMCFD at 14.4 psia and } 80^{\circ}F$$

or

$$\frac{150.0}{146.7} \times 1.95 = 1.994 \text{ MMSCFD at 14.65 psia and } 60^{\circ}F$$

The table in Step 1 should now be complete as follows:

Ps	Pd	R	Ts	Td	BHP/MMCFD	MMCFD	BHP
93.9	313.9	3.34	80	230	73.0	2.05	146.7

Example Calculation of Two-Stage Horsepower Required (Ex. 2)

Determine the horsepower required to compress 1.00 MMCFD from 40 psig to 800 psig. Conditions are as follows: Suction temperature is 80^{o}F, ambient temperature is 100^{o}F, approach is 30^{o}F, elevation is 3000 feet, and "N" value is 1.28.

Step 1: Prepare the following table:

S	Ps	Pd	R	Ts	Td	BHP/MMCFD	Tc	BHP/MMCFDc	Cap. (MMCFD)	BHP

Where:
- S = stage
- Ps = absolute suction pressure
- Pd = absolute discharge pressure
- R = ratio of compression
- Ts = suction temperature
- Td = discharge temperature
- BHP/MMCFD = brake horsepower to compress 1.00 MMCFD
- Tc = temperature correction factor
- BHP/MMCFDc = corrected brake horsepower to compress 1.00 MMCFD of gas
- Cap. (MMCFD) = capacity in MMCFD measured at 14.4 psia and suction temperature
- BHP = brake horsepower required

Step 2: Determine the absolute suction and discharge pressures at 3000-ft elevation (use Figure 1):

Absolute suction pressure = 40 + 13.16 = 53.16.
Place this figure under Ps - 1st stage.

Absolute discharge pressure = 800 + 13.16 = 813.16.
Place this figure under Pd - 2nd stage.

Step 3: Determine overall ratio of compression:

$$R = \frac{813.16}{53.16} = 15.3$$

Step 4: Determine square root of overall ratio of compression:

$$\sqrt{15.3} = 3.911$$

Step 5: Multiply ratio obtained above by 1.07, to obtain first-stage ratio. (This is done to distribute horsepower equally between stages if the temperature difference is 50^{o}F at normal ratios. A lower multiplier is used for lower temperature differentials and vice versa). Equal distribution of horsepower prevents excessive rod load, excessive temperature, and results in horsepower savings.

$3.911 \times 1.07 = 4.18$ Place this figure under R - 1st stage.

Step 6: Determine discharge pressure of first stage by multiplying first-stage suction pressure by first-stage ratio:

$53.16 \times 4.18 = 222$ psia Place this figure under Pd - 1st stage.

Step 7: Determine suction pressure of second stage by using a 5 psi pressure drop through the intercooler, separator, and associated piping:

$222 - 5 = 217$ Place this figure under Ps - 2nd stage.

Step 8: Determine ratio of compression of second stage based on 217 psia suction pressure and 813.16 discharge pressure:

$$R = \frac{813.16}{217} = 3.7$$ Place this figure under R - 2nd stage.

Step 9: Determine suction temperature of second stage based on 30°F approach cooling to ambient:

$100^\circ + 30^\circ = 130^\circ F$

Step 10: Determine discharge temperature of each stage based on its respective suction temperature, ratio and "N" value of 1.28 from Figure 3 or from the formula:

$$T_2 = T_1 \times R^{\frac{N-1}{N}}$$

The discharge temperature from the first stage is 260°F and from the second stage, 320°F. Place the suction and discharge temperatures in their respective columns.

Step 11: Determine BHP required to compress 1.00 MMCFD of gas measured at 14.4 psia and suction temperature from Table 1. Based on a 1.26 "N" value, the BHP/MMCFD is 88.0 for the first stage and 79.6 for the second. Place these figures in their respective columns.

Step 12: Determine the temperature correction factor for the second stage based on an 80°F first-stage suction temperature and a 130°F second-stage suction temperature:

$$Tc = \frac{590}{540} = 1.093$$

Step 13: Determine the corrected BHP/MMCFD for the second stage. This is obtained by multiplying the BHP/MMCFD (from Table 1) by the temperature correction factor:

$$BHP/MMCFDc = BHP/MMCFD \times Tc = 79.6 \times 1.093 = 87$$

Since there is no temperature correction factor for the first stage, use the BHP/MMCFD obtained from Table 1 as the BHP/MMCFDc. Place these figures in their respective columns.

Step 14: Determine the total BHP/MMCFD required, which is the sum of the BHP/MMCFDc for each stage:

$$88.0 + 87 = 175$$

Step 15: Determine the BHP required:

$$BHP = \frac{BHP}{MMCFD} \times MMCFD = \frac{175}{1.0} \times 1.0 = 175$$

Step 16: For this use, offer a JGW 2-2 with G342 NA which has 190 BHP available for compression at 3000 ft elevation. This JGW will deliver:

$$\frac{190}{175} \times 1.00 = 1.085 \text{ MMCFD at 14.4 psia and } 80^{o}F$$

The table in Step 1 should now be complete as follows:

S	Ps	Pd	R	Ts	Td	BHP/MMCFD	Tc	BHP/MMCFDc	Cap. (MMCFD)	BHP
1	53.16	222.00	4.18	80	260	88.0	1.000	88.0	1.00	175
2	217.00	813.20	3.70	130	320	79.6	1.093	87.0		
								175.0		

Horsepower Required for Three or More Stages

Use the same method as used for two-stage compression; however, use the cube root of the overall ratio to obtain the basic three-stage ratio, the fourth root for the basic four-stage ratio, etc. Multiply the basic ratio by 1.07 to obtain the first-stage ratio and use balanced ratios for the remaining stages, as this will simplify calculations and allow for the same discharge temperature.

For application involving more than three stages, it is recommended that you determine balanced ratios by obtaining the ratio of the last stage first, the penultimate stage second, etc. using the same ratio for each stage except for the first stage. With this method, reduce the basic ratio obtained from the root of the overall ratio by 3.0 to 5.0 percent, and use this ratio to obtain the ratio of the second, third, fourth, etc. stages.

Calculating Horsepower When Capacity is Unknown

Basically, the same method is used as is used for calculating horsepower for single and multi-stage compression. Suppose that in the previous section, "Example Calculation of Horsepower Required for Single-Stage Compression," the customer requested the maximum capacity of a JGP2/G3306NA instead of 1.65 MMSCFD measured at 14.65 psia and 60^{O}F. Follow Steps 1 through 6 as was done in that section. The subsequent steps would be as follows:

Step 7: Determine the maximum horsepower of a JGP2-1/G3306NA at 2800 ft elevation = 120 HP

Step 8: Determine the maximum capacity, knowing the maximum BHP and the BHP/MMCFD required:

$$\text{MMCFD} = \frac{\text{BHP}}{\text{BHP/MMCFD}} = \frac{110}{84.8} = 1.30 \text{ MMCFD measured at 14.4 psia and } 80^{O}\text{F}$$

Other Design Considerations Regarding Compressor Horsepower

The total BHP available for compression is less than the rated BHP; this is because of the horsepower required to drive the cooler fan. The maximum BHP can be used for the compression BHP if the extra BHP is required to meet the customer's capacity requirement.

At elevations above 500 feet, for naturally aspirated air, Caterpillar uses a 3 percent deration factor for every 1000 feet above 500 feet. At temperatures above 60^{O}F ambient, they use 1 percent for every 10^{O}F above 85^{O}F ambient.

For a G-342NA at 4500-ft elevation and 100^{O}F ambient temperature, the maximum continuous BHP rating may be calculated as follows:

$$\text{Deration for elevation:} \quad \frac{4500 - 500}{1000} \times 3.0\% = 12\%$$

$$\text{Deration for temperature:} \quad \frac{100 - 85}{10} \times 1.0\% = 1.5\%$$

$$\text{Maximum BHP} = 225 \times 0.88 \times 0.985 = 195$$

Subtract horsepower for cooler to get compression horsepower available.

Aside from the fact that many customers request the discharge temperature before cooling, it should be calculated to determine if it is excessive. An excessive discharge temperature is any temperature greater than 300^{O}F.

In most cases, the volume handled by subsequent stages is more than the first, because suction temperature for these stages is higher and volume is directly proportional to absolute temperature. Since volume affects horsepower, it will take more horsepower for these subsequent stages, provided the ratios of compression are equal. Because of this, a temperature correction factor is needed.

A pressure drop is required between stages to account for the loss through the intercooler, separator, and interstage piping. As an average, use 1-1.5 psi to 50 psia, 2 psi from 51 to 100 psia, 3 psi from 101 to 150 psia, psi from 151 to 200 psia, 5 psi from 201 to 300 psia, and 6 psi from 301 to 500 psia. High-pressure applications will require higher pressure drops.

Is it then true that it takes the same horsepower to compress 1.0 MMCFD of gas from 100 to 300 psia as it would take to compress 1.0 MMCFD of this same gas from 1000 to 3000 psia? If supercompressibility or superexpandability are not considered, one can assume that for equal ratios and volumes, the same horsepower is required.

PISTON AND CYLINDER DESIGN CALCULATIONS

Cylinders

A single-acting cylinder is one that compresses gas during either the forward or backward stroke (compression or tension stroke), whereas a double-acting cylinder is one that compresses gas during both the forward and backward strokes. The head end is the end of the cylinder where the gas is compressed on the forward or compression stroke, and the crank end is where the gas is compressed on the backward or tension stroke.

Ariel furnishes both single and double-acting cylinders. Single-acting cylinders are referred to as tandem on the JGI1-2 and JGP1-2 (LP. cyl. outboard). They are referred to as steeple (high-pressure cylinder outboard) on all other frames.

The tandem arrangement (L.P. outboard) gives the option to utilize a variable volume pocket.

Swept Volume

The volume swept by the compressor piston during the forward and/or backward stroke is called swept volume. The speed of the compressor is not considered. The swept volume is expressed in cubic inches and is calculated as follows:

Head end = area of piston (in.) x stroke in inches

Crank end = (area of piston - area of rod) x stroke

Total = [(2 x area of piston) - (area of rod)] x stroke

The swept volume of 5-3/4 in. diameter x 3-in. cylinder having a 1-1/8 in. diameter piston rod is:

$$\text{Total swept volume} = [\,2 \times (5.75/2)^2 \times 3.146 - (1.125/2)^2 \times 3.146] \times 3 = 153 \text{ cubic inches}$$

Piston Displacement

The product of the swept volume and compressor speed in revolutions per minute (RPM) divided by 1728 is called piston displacement and is expressed in cubic feet per minute (CFM). The piston displacement of the 5-3/4 in. x 3-in. cylinder in the swept volume calculation (furnished on a JGP2-2 compressor that operates at 1800 RPM) is:

$$\text{Piston displacement in CFM} = \frac{\text{swept volume x RPM}}{1728} = \frac{153 \times 1800}{1728} = 159.4$$

Compressor Cylinder Clearance and Performance

Compressor clearance is the volume of space in the end of the cylinder, measured when the piston is at the end of its stroke. It includes the space between the piston and the cylinder head, cavities where the valves communicate with the cylinder bore, counterbore, etc., and extra clearance that may be added to meet a particular compressor application.

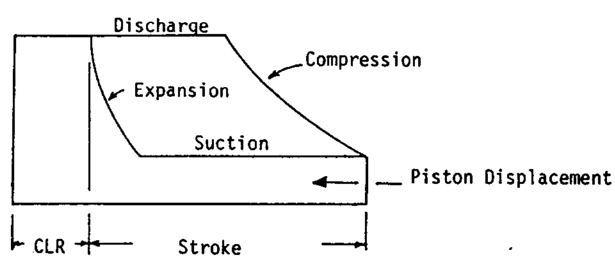
Clearance is unavoidable since: (1) the piston cannot be allowed to touch the head; (2) clearance must be maintained around the valves; and (3) a counterbore is required to remove the piston after it has operated for a period of time.

At the end of the compression stroke, the clearance space is filled with gas at maximum discharge pressure. As the piston starts on its return stroke, this gas expands, preventing the entrance of inlet gas until the pressure in the cylinder is slightly less than the suction pressure. Thus, a portion of the stroke is wasted insofar as delivering gas is concerned. The higher the clearance and the higher the compression ratio, the greater the loss due to reexpansion of gas from the clearance space. This reexpansion does not appreciably affect the compressor horsepower since most of the power used in compressing the gas in the clearance space is regained during reexpansion.

Volumetric Efficiency

Because of the cylinder clearance, the gas actually discharged by the cylinder is not equal to the piston displacement. This is measured as percent volumetric efficiency. Volumetric efficiency is the ratio of gas actually delivered, compared to the piston displacement.

Volumetric efficiency can be illustrated by an indicator card:



From the indicator card, it is seen that the higher the clearance and the higher the ratio, the more the reexpansion, and thus the lower the volumetric efficiency. These are not the only two factors that affect the volumetric efficiency -- reexpansion is also affected by the "N" value.

These three factors are considered in the basic volumetric efficiency formula. An example calculation is shown below.

The basic formula for volumetric efficiency is:

$$VE = 100 - \%Clr\,(R^{1/N} - 1)$$

Where:

VE = volumetric efficiency, %
%Clr = percent clearance divided by 100
R = ratio of compression
N = reexpansion "N." Although lower than the "N" in calculating BHP, use the BHP "N" value.

Other factors that affect VE are: loss through the compressor valves; expansion of the inlet heat; compressor piston ring blow-by; and nature of gas being compressed. These factors are included in the volumetric efficiency formula by substituting the value 97 for 100, and the formula for volumetric efficiency used in basic compressor calculations becomes:

$$VE = 0.97 - \%Clr\,(R^{1/N} - 1)$$

The volumetric efficiency of a cylinder having a clearance of 10.5 percent, a compression ratio of 3:1, and an "N" clearance of 1.26 is shown below:

$$VE = 0.97 - \%Clr\,(R^{1/N} - 1) = 0.97 - 0.105\,(3.0^{.794} - 1)$$

$$= 0.97 - (0.105 \times 1.39) = 0.97 - 0.146 = 0.824$$

When the volumetric efficiency is zero or less, volumetric efficiency is negative. In this case, the compressor will not deliver gas, since the reexpansion is such that no gas enters the cylinder on the suction stroke.

Will the volumetric efficiency be negative if the clearance is over 97 percent? Not necessarily, because the VE also depends on the ratio of compression and the "N" value. If the $(R^{1/N} - 1)$ value is less than 1.0, the clearance can be over 97 percent, and if the ratio is very low, a cylinder with clearance well over 100 percent can be used.

Are cylinders ever furnished with clearances over 100 percent? Yes, especially for application involving low ratios and varying discharge pressures. This high clearance is furnished to allow the compressor to operate loaded through the complete range of discharge pressures without changing clearance.

Is there an easier way to determine the volumetric efficiency than by using the above formula? The volumetric efficiency may be approximated by cross-plotting on Figure 4.

It is not detrimental to use a cylinder with low volumetric efficiency. The main consideration is to furnish a cylinder that will deliver the required volume to load the compressor. Whether this volume is achieved with a cylinder that has 80 percent volumetric efficiency or a 50 percent volumetric efficiency is incidental. In fact, in some cases, it is beneficial to have a cylinder with low volumetric efficiency because it could either mean a larger cylinder than required for the application which would enable the compressor to operate the lower suction pressures or large valve area which in turn means less horsepower loss through the valves.

Cylinder Size and Capacity

The capacity of a compressor cylinder can be calculated by using the following formula:

$$\text{Cap.} = VE \times CFM \times \frac{Ps}{PB} \times 1440$$

Where:
- Cap. = cylinder capacity in cubic feet per day
- VE = volumetric efficiency
- CFM = piston displacement in cubic feet per minute
- Ps = absolute suction pressure
- PB = base pressure taken at 14.4 psia
- 1440 = number of minutes in a day

This number can be simplified by dividing 1440 by 14.4 to obtain the following formula:

$$\text{Cap.} = VE \times CFM \times Ps \times 100$$

This simplification is why we use 14.4 psia as the base pressure for the BHP/MMCFD.

Calculation of Cylinder Size (Ex. 3)

The following calculation procedure, for the JGW compressor, illustrates a method for determining cylinder size for two-stage compression using information from Example Calculation No. 1 (Ex. 1).

<u>First Stage Calculation Procedure</u>:

Step 1: Determine the approximate piston displacement required from the following formula, using a volumetric efficiency of 0.70:

$$CFM = \frac{MMCFD}{Ps \times VE \times 100} = \frac{1{,}000{,}000}{53.16 \times 0.70 \times 100} = 269$$

The volumetric efficiency used above is based on a trial-and-error method, and with experience, the approximate volumetric efficiency can be estimated on the first try.

Step 2: From the list of cylinders available for the JGW, select a cylinder whose piston displacement approximates 269 CFM required.

In this case, the 8-1/2 in. by 4-1/4 in. W cylinder has a piston displacement of 329 CFM and a normal clearance of 12.98 percent.

Step 3: Calculate the volumetric efficiency of the 8-1/2 in. by 4-1/4 in. cylinder based on its normal clearance of 12.98 percent and a ratio of 4.17.

Using the volumetric efficiency formula [$VE = 0.97 - \%Clr\ (R^{1/N} - 1)$] the volumetric efficiency with this clearance and at this ratio is 0.7092. Although the volumetric efficiencies (Figure 4) are based on a 1.26 "N" value, they can be used to estimate the volumetric efficiency in this case even though the "N" value for this particular application is 1.28.

Step 4: Calculate the capacity of the 8-1/2 in. by 4-1/4 in. cylinder from the following formula to determine whether it is the correct size:

$$\begin{aligned} \text{Cap.} &= VE \times CFM \times Ps \times 100 \\ &= 0.7092 \times 329 \times 53.16 \times 100 \\ &= 1{,}240{,}365 \end{aligned}$$

Step 5: Since a capacity of 1.00 MMCFD is required and this cylinder will deliver a capacity of 1.240, it is correct for this application.

Second Stage Calculation Procedure:

Step 1: Follow the same procedure as used for the first stage, but introduce the temperature correction factor in the equation to determine the approximate CFM required; thus, the equation becomes:

$$CFM = \frac{\text{Cap.} \times 1.093}{Ps \times VE \times 100}$$

Step 2: Determine the approximate piston displacement required, using a volumetric efficiency of 60 percent, by using the following formula:

$$CFM = \frac{\text{Cap.} \times 1.093}{Ps \times VE \times 100} = \frac{1{,}000{,}000 \times 1.093}{217 \times 0.60 \times 100} = 83.9$$

Step 3: For this application, select the 4-5/8 in. by 4-1/4 in. JGR cylinder which has a piston displacement of 94 CFM and a normal clearance of 14.31 percent.

Step 4: Calculate the volumetric efficiency for this cylinder based on its normal clearance of 14.31 percent and a ratio of 3.7.

The volumetric efficiency obtained from the formula VE = %Clr, etc., is 0.699.

Step 5: Calculate the capacity of the 4-5/8 in. by 4-1/4 in. cylinder from the following formula:

$$\text{Cap.} = \frac{VE \times CFM \times Ps \times 100}{1.093} = \frac{0.699 \times 93.95 \times 217 \times 100}{1.093}$$

$$= 1{,}303{,}808$$

Step 6: Since a capacity of 1.093 MMCFD is required (which is the product of the capacity required for the first stage times the temperature correction factor of 1.093), the 4-5/8 in. by 4-1/4 in. cylinder is the proper size since it will deliver a capacity of 1.303 MMCFD.

Why use the JGR cylinder instead of the JGW cylinder on the second stage? The 4-3/4 in. by 4-1/4 in. JGW cylinder has a rated discharge pressure of 620 psig which is below the requirement of 800 psig pressure.

Can't we stretch a point on the 4-3/4 in. by 4-1/4 in. JGW cylinder since its maximum working pressure approximates the second-stage operating pressure? Even though the cylinder is hydrostatically tested to one and one-half times the maximum working pressure, which in this case would be 1020 psig for the 4-3/4 in. by 4-1/4 in. JGW cylinder, it could not be used. Doing so would start a trend of using cylinders of insufficient working pressure which would tend to lead to a dangerous situation.

In the second stage, the temperature correction factor was introduced. As stated previously, the volume is directly proportional to its absolute temperature. Since the gas that passes through the first stage must pass through the second stage, the volume of gas handled by the second stage is greater due to its higher temperature. Thus, the cylinder must be larger.

Both the first and second stage cylinders deliver more gas than is required. Will this not overload the unit and, if so, how is it corrected? In this case of the first stage, yes, the unit will be overloaded; but in the second stage, there is no appreciable overload. The capacity of the first stage is reduced by adding clearance, and the amount of clearance required can be calculated by using the volumetric efficiency formula as follows:

Step 1: Since the volumetric efficiency is directly proportional to the capacity delivered, determine the volumetric efficiency required to deliver the necessary capacity as follows:

$$VE = \frac{\text{capacity required}}{\text{capacity delivered}} \times VE$$

$$VE = \frac{1.000}{1.240} \times 0.709 = 0.572$$

This gives the volumetric efficiency which is required to deliver 1.00 MMCFD.

Step 2: Using the volumetric efficiency formula, determine the amount of clearance required to give a volumetric efficiency of 0.572 where R = 3.911, N = 1.28 from Ex. 2:

$$VE = 0.97 - \%Clr\,(R^{1/N} - 1)$$

$$0.572 = 0.97 - \%Clr\,(3.911^{1/1.28} - 1)$$

$$0.572 = 0.97 - \%Clr\,(1.902)$$

$$\%Clr = 0.2093$$

$$\text{or } 20.93\%$$

Step 3: Since the normal clearance is 12.98 percent and the total clearance is 20.93 percent, 7.95 percent clearance must be added to the cylinder to prevent overloading:

$$20.93\% - 12.98\% = 7.95\%$$

How many cubic inches of clearance are required to obtain 7.95% additional clearance? Cubic inches of clearance is equal to the swept volume times the percent clearance divided by 100. The swept volume of an 8-1/2 in. by 4-1/4 in. cylinder with a 1-1/2 in. piston rod is:

$$SV = [56.74 + (56.74 - 1.76)] \times 4.25 = 475 \text{ cu. in.}$$

$$\text{Cubic inches required} = 475 \times 0.0795 = 37.76 \text{ cu. in.}$$

This clearance can be added in a number of ways such as:

Head end clearance plugs	--	Recommended with VHL cylinders.
Fixed clearance pocket	--	Recommended only when a clearance adjustment is required without shutting down the unit.
Variable clearance pocket	--	Recommended when variable clearance is required without shutting down the unit or when a customer requires full load at all pressures throughout the operating range.

In the preceding example, clearance would be added with a head end variable volume pocket which is standard equipment on CSI packages.

Since clearance must be added to the first-stage cylinder, as in the previous 4-5/8 in. by 4-1/4 in. cylinder case, must clearance be added to the second-stage cylinder also? The volume of gas delivered by the first stage of a multi-stage compressor must necessarily pass through the subsequent stage or stages; therefore, the volumetric efficiency of a multi-stage compressor is the volumetric efficiency of the first-stage cylinder and is affected only slightly by variations in the suction and discharge pressures of the second stage, and even less so by variations in pressures of the third stage, etc.

In other words, the second-stage cylinder can only deliver the capacity of the first stage, and if the second-stage cylinder is oversized, the suction pressure to the cylinder will automatically drop to a point where it will deliver the capacity of the first stage. This small drop in suction pressure produces a change in first-stage discharge pressure; however, it affects the first-stage ratio of compression only slightly because a change in discharge pressure does not have as pronounced an effect on ratio of compression as a change in suction pressure. Naturally, if the second-stage cylinder were excessively large, the pressure drop would be significant and in this case, we would have to add clearance to the second stage. But, since the difference in delivered capacity and required capacity is only slight, the pressure drop will be insignificant.

Consequently, adjustable clearance is normally added only to the first-stage cylinder. If clearance is added to subsequent stages, the volumetric efficiency of the first stage is only slightly altered and this slight alteration cannot compensate for the cost of the additional clearance to the subsequent stages.

Some compressors require variable clearance on subsequent stages for one or both of the following reasons: To overcome rod load problems that might be encountered or to maintain the suction pressure to the subsequent stage as may be required when gas is introduced between stages.

Piston Speed

Piston speed is the speed at which the piston travels, in feet per minute. It is calculated as follows:

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

where: PS = piston speed in ft/min
S = compressor stroke in inches
RPM = compressor speed in RPM

In a reciprocating compressor, the piston travels both forward and backward in one revolution, whether it is a single-acting or double-acting cylinder. Therefore, the stroke, S, is multiplied by a factor of 2 in the above equation.

An example of piston speed for an Ariel JGW compressor is shown below:

$$PS = \frac{S}{12} \times 2 \times RPM = \frac{4.25}{6} \times 1200 = 850 \text{ ft/min}$$

for a 4-1/4 in. compressor stroke and a maximum speed of 1200 RPM.

Valve Velocity

Valve velocity is defined as the velocity at which the gas enters and leaves the cylinder in a given unit of time, and is normally expressed in feet per minute (FPM).

Since this velocity depends on the density of the gas (at specified temperature and pressure conditions), as well as the quantity of gas delivered by the cylinder, determining the actual valve velocity would be a complicated process. Therefore, base compressor valve velocity is based on the piston displacement of the cylinder, which is not affected by the above factors. With this as a criterion, one can compare the valve velocities of one cylinder with those of another.

The basic formula for calculating valve velocity is:

$$\text{Velocity} = \frac{\text{volume per unit of time}}{\text{area of opening}}$$

Using this basic formula for computing the valve velocity of a compressor cylinder, the following formula is derived:

$$VV = \frac{\text{piston area x piston speed}}{\text{FLA x N}}$$

FLA = free lift area of suction or discharge valve in square inches. If free lift area is not the same for both suction and discharge valves, use the average.

N = number of suction or discharge valves per end

The valve velocity of a 4-3/4 in. by 4-1/4 in. Ariel cylinder operating at 1200 RPM and having a piston rod diameter of 1-1/2 in., one suction and discharge valve per end, FLA per valve of 1.860 is calculated as:

$$\text{HE Vel.} = \frac{\text{piston area x piston speed}}{\text{FLA x N}} = \frac{17.7206 \text{ x } 850}{1.860 \text{ x } 1}$$

$$= 8098 \text{ FPM}$$

$$\text{CE Vel.} = \frac{(\text{piston area - rod area})}{\text{FLA x N}} \text{ x piston speed}$$

$$= \frac{15.9535 \text{ x } 850}{1.860 \text{ x } 1} = 7291 \text{ FPM}$$

$$\text{Avg. Vel.} = \frac{8098 \text{ x } 7291}{2} = 7694 \text{ FPM}$$

Compressor Rod Load

Compressor rod load is the safe operating load limit as calculated by the manufacturer. It is not necessarily the load on the rod that will limit this; it can be based on the webbing, frame, crosshead, bearings and bushings or a combination of any of these.

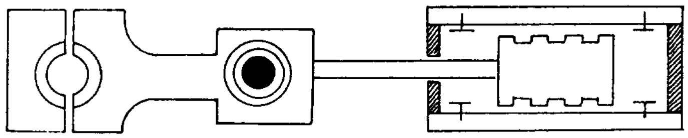
Rod load compression = AHE x Pd - ACE x Ps

Rod load tension = ACE x Pd - AHE x Ps

Example rod load calculation:

Ps = 213.4 AHE = 12.56 sq. in.

Pd = 613.4 ACE = 10.49 sq. in.



Piston is moving ⟶ Compression rod load:

RLc = AHE x Pd - ACE x Ps

RLc = (12.56 x 613.4) - (10.49 x 213.4) = 7704 - 2238 = 5466 RLc

Piston is moving ⟵ Tension rod load:

RLt = ACE x Pd - AHE x Ps

RLt = (10.49 x 613.4) - (12.56 x 213.4) = 6435 - 2680 = 3755 RLt

DISCHARGE AND SUCTION PRESSURE

How does one design a compressor with a constant discharge pressure and a varying suction pressure? Always design the unit for the lowest suction pressure and add clearance for operation at the higher pressure. This will allow the compressor to operate loaded through the complete pressure range.

How does one design a compressor that has both varying suction pressure and varying discharge pressure? As a normal rule, design the unit to handle the capacity from the low suction to the low discharge pressure which is the lowest ratio. It will be found that if a cylinder is designed for these conditions, it will deliver ample capacity to handle the compressor capacity at the other conditions.

SUPERCOMPRESSIBILITY AND SUPEREXPANDABILITY

Deviation from the ideal gas law occurs when the volume of a gas at a particular temperature and pressure differs from the volume the gas would occupy at these same temperature and pressure conditions according to the ideal gas law. The amount that a gas deviates is the ratio of the actual volume of the gas to the volume of the gas according to the ideal gas law. If the actual volume a gas occupies at a given temperature and pressure is less than the volume according to the natural gas law, we use the term "supercompressibility" and this reaches a maximum at the critical temperature and pressure of the gas. If the actual volume a gas occupies at a given pressure and temperature is more than the volume according to the natural gas law, we use the term "superexpandability." In our work we are normally concerned with supercompressibility, as superexpandability normally occurs when handling monoatomic gases such as hydrogen, oxygen, nitrogen, etc.

Supercompressibility becomes an important factor in gas compression when handling heavy hydrocarbons at normal pressures or light hydrocarbons at pressures above 1500 psig. Heavy hydrocarbons deviate at normal pressures. Although light hydrocarbons deviate above 50 psig, their deviation does not become critical until we reach pressures over 1500 psig. For this reason, it is suggested that when one has an inquiry involving either heavy hydrocarbons or light hydrocarbons at high pressures, it be submitted to Ariel in Mount Vernon, Ohio, for quotation as supercompressibility affects not only the horsepower but also the cylinder capacity.

APPLICATIONS REQUIRING SPECIAL HANDLING

Heavy hydrocarbons-valve losses are higher than encountered when handling light gases and these gases deviate at normal pressure.

In high-pressure applications, pressures above 1500 psig, supercompressibility in an amount that will seriously affect compressor performance will occur even when handling light hydrocarbons.

Pressure losses through the valves become a very important factor at low compression ratios, say below 2:1. In this case, the unit must be designed with a compressor cylinder that has a relatively low valve velocity in order to reduce valve losses to a minimum.

With vacuum compressors, the suction pressure passes through the range of pressures from atmospheric down to the minimum or normal operating pressure. At atmospheric pressure or the initial pressure, the BHP/MMCFD is lowest and the capacity the highest. At the minimum pressure or final pressure, the BHP/MMCFD is the highest and capacity the lowest. The product of the BHP/MMCFD and capacity indicates the BHP required; however, the maximum BHP is not at either the initial or final suction, but somewhere in between. This point or peak load must be calculated and the unit sized to handle this BHP load.

ACKNOWLEDGMENT

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Table 1
Brake Horsepower per MMCFD for Gas with 1.26 "N" Value

C.R.	H.P.	C.R.	H.P.	C.R.	H.P.	C.R.	H.P.	C.R.	H.P.
1.10	10.2	2.10	46.1	3.10	68.5	4.10	86.7	5.10	103.4
1.12	11.2	2.12	46.6	3.12	68.8	4.12	87.0	5.12	103.7
1.14	12.2	2.14	47.1	3.14	69.2	4.14	87.3	5.14	104.0
1.16	13.3	2.16	47.6	3.16	69.6	4.16	87.6	5.16	104.3
1.18	14.3	2.18	48.1	3.18	70.0	4.18	88.0	5.18	104.6
1.20	15.3	2.20	48.6	3.20	70.4	4.20	88.4	5.20	104.9
1.22	16.3	2.22	49.1	3.22	70.7	4.22	88.7	5.22	105.2
1.24	17.3	2.24	49.6	3.24	71.1	4.24	89.0	5.24	105.5
1.26	18.4	2.26	50.1	3.26	71.5	4.26	89.3	5.26	105.8
1.28	19.5	2.28	50.6	3.28	71.9	4.28	89.7	5.28	106.1
1.30	20.5	2.30	51.1	3.30	72.3	4.30	90.1	5.30	106.5
1.32	21.3	2.30	51.5	3.32	72.7	4.32	90.4	5.32	106.8
1.34	22.1	2.34	51.9	3.34	73.0	4.34	90.7	5.34	107.1
1.36	22.9	2.36	52.3	3.36	73.3	4.36	91.0	5.36	107.4
1.38	23.7	2.38	52.8	3.38	73.7	4.38	91.4	5.38	107.7
1.40	24.6	2.40	53.2	3.40	74.0	4.40	91.8	5.40	108.1
1.42	25.4	2.42	53.6	3.42	74.4	4.42	92.1	5.42	108.4
1.44	26.2	2.44	54.0	3.44	74.8	4.44	92.4	5.44	108.7
1.46	27.0	2.46	54.6	3.46	75.2	4.46	92.7	5.46	109.0
1.48	27.8	2.48	55.0	3.48	75.6	4.48	93.1	5.48	109.3
1.50	28.6	2.50	55.5	3.50	76.0	4.50	93.5	5.50	109.7
1.52	29.2	2.52	55.9	3.52	76.3	4.52	93.8	5.52	110.0
1.54	29.9	2.54	56.3	3.54	76.6	4.54	94.1	5.54	110.3
1.56	30.6	2.56	56.8	3.56	77.0	4.56	94.4	5.56	110.6
1.58	31.3	2.58	57.3	3.58	77.4	4.58	94.7	5.58	110.9
1.60	32.0	2.60	57.8	3.60	77.8	4.60	95.1	5.60	111.3
1.62	32.6	2.62	58.2	3.62	78.1	4.62	95.4	5.62	111.6
1.64	33.2	2.64	58.6	3.64	78.4	4.64	95.7	5.64	111.9
1.66	33.8	2.66	59.1	3.66	78.8	4.66	96.0	5.66	112.2
1.68	34.5	2.68	59.6	3.68	79.2	4.68	96.4	5.68	112.5

C.R.	H.P.	C.R.	H.P.	C.R.	H.P.	C.R.	H.P.	C.R.	H.P.
1.70	35.2	2.70	60.1	3.70	79.6	4.70	96.8	5.70	112.8
1.72	35.8	2.72	60.5	3.72	79.9	4.72	97.1	5.72	113.1
1.74	36.4	2.74	60.9	3.74	80.2	4.74	97.4	5.74	113.4
1.76	37.0	2.76	61.4	3.76	80.5	4.76	97.7	5.76	113.7
1.78	37.7	2.78	61.9	3.78	80.9	4.78	98.0	5.78	114.0
1.80	38.4	2.80	62.3	3.80	81.3	4.80	98.4	5.80	114.3
1.82	38.9	2.82	62.7	3.82	81.6	4.82	98.7	5.82	114.6
1.84	39.4	2.84	63.1	3.84	81.9	4.84	99.0	5.84	114.9
1.86	39.9	2.86	63.5	3.86	82.3	4.86	99.3	5.86	115.2
1.88	40.5	2.88	63.9	3.88	82.7	4.88	99.6	5.88	115.5
1.90	41.0	2.90	64.4	3.90	83.1	4.90	100.0	5.90	115.8
1.92	41.5	2.92	64.8	3.92	83.4	4.92	100.3	5.92	116.1
1.94	42.0	2.94	65.2	3.94	83.7	4.94	100.6	5.94	116.4
1.96	42.5	2.96	65.6	3.96	84.1	4.96	100.9	5.96	116.7
1.98	43.1	2.98	66.1	3.98	84.5	4.98	101.3	5.98	117.0
2.00	43.7	3.00	66.3	4.00	84.9	5.00	101.7	6.00	117.3
2.02	44.1	3.02	66.9	4.02	85.2	5.02	102.0		
2.04	44.6	3.04	67.3	4.04	85.5	5.04	102.3		
2.06	45.1	3.06	67.7	4.06	85.9	5.06	102.6		
2.08	45.6	3.08	68.1	4.08	86.3	5.08	103.0		

SINGLE STAGE BHP = (22) x (CR) x VOLUME
MULTI STAGE BHP = (22) x (ICR) x (NO. STAGES) x (VOLUME)
ICR is Ideal Compression Ratio
VOLUME is in MMCF

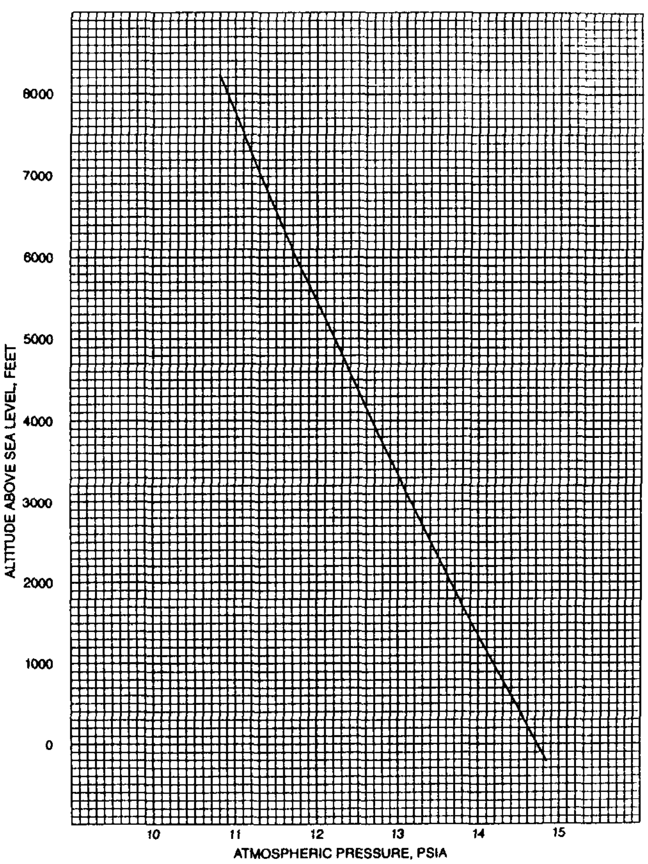

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Figure 1 - Altitude vs atmospheric pressure

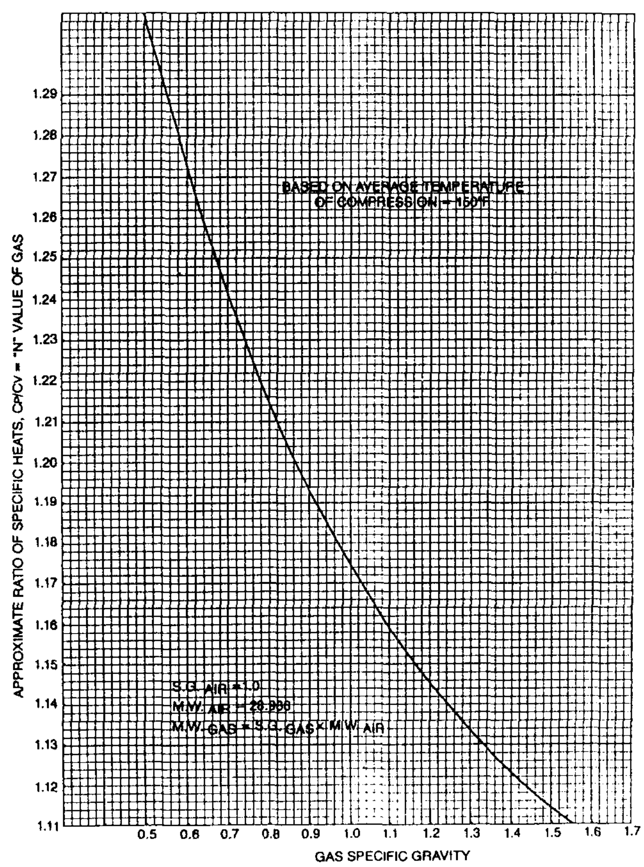

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Figure 2 - Specific heat (N-values vs. gas gravity)

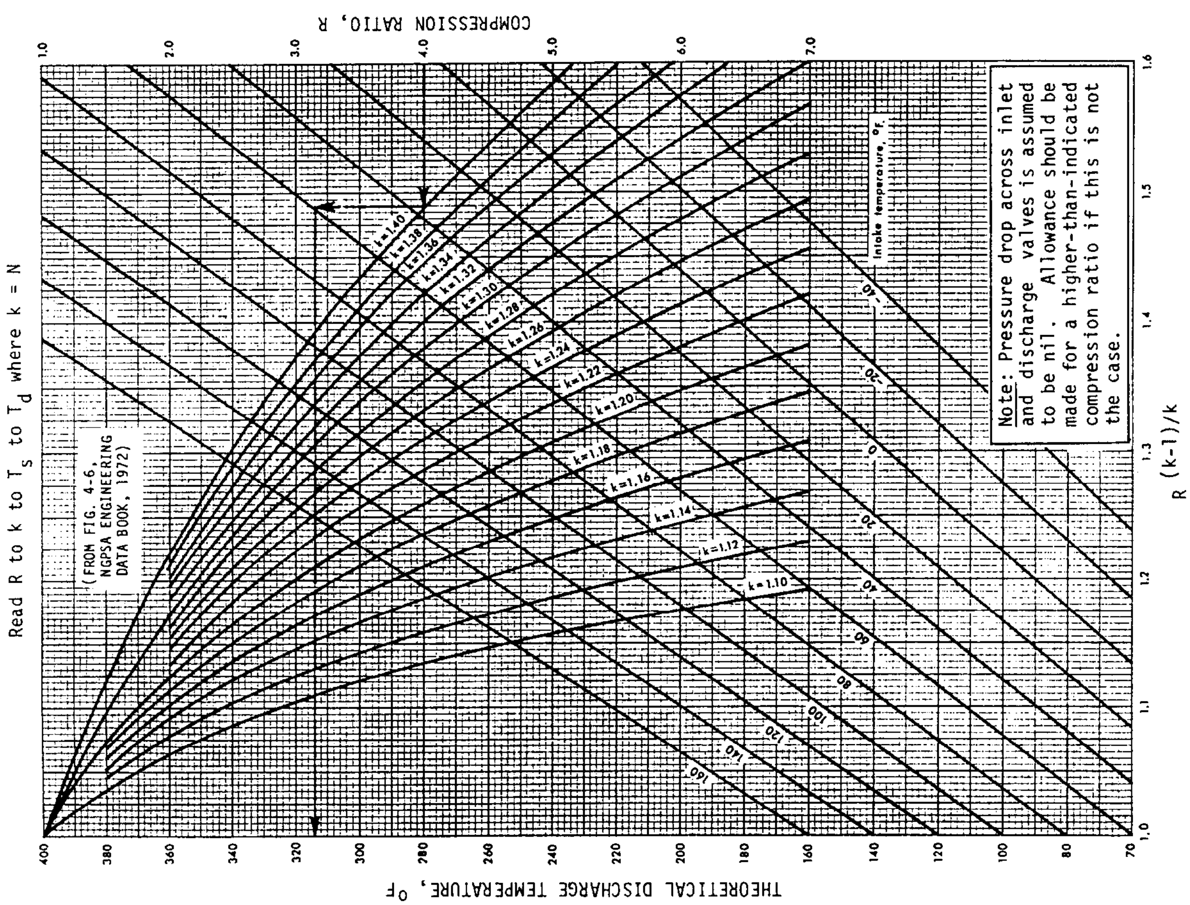

Figure 3 - Theoretical discharge temperatures for single-stage compression

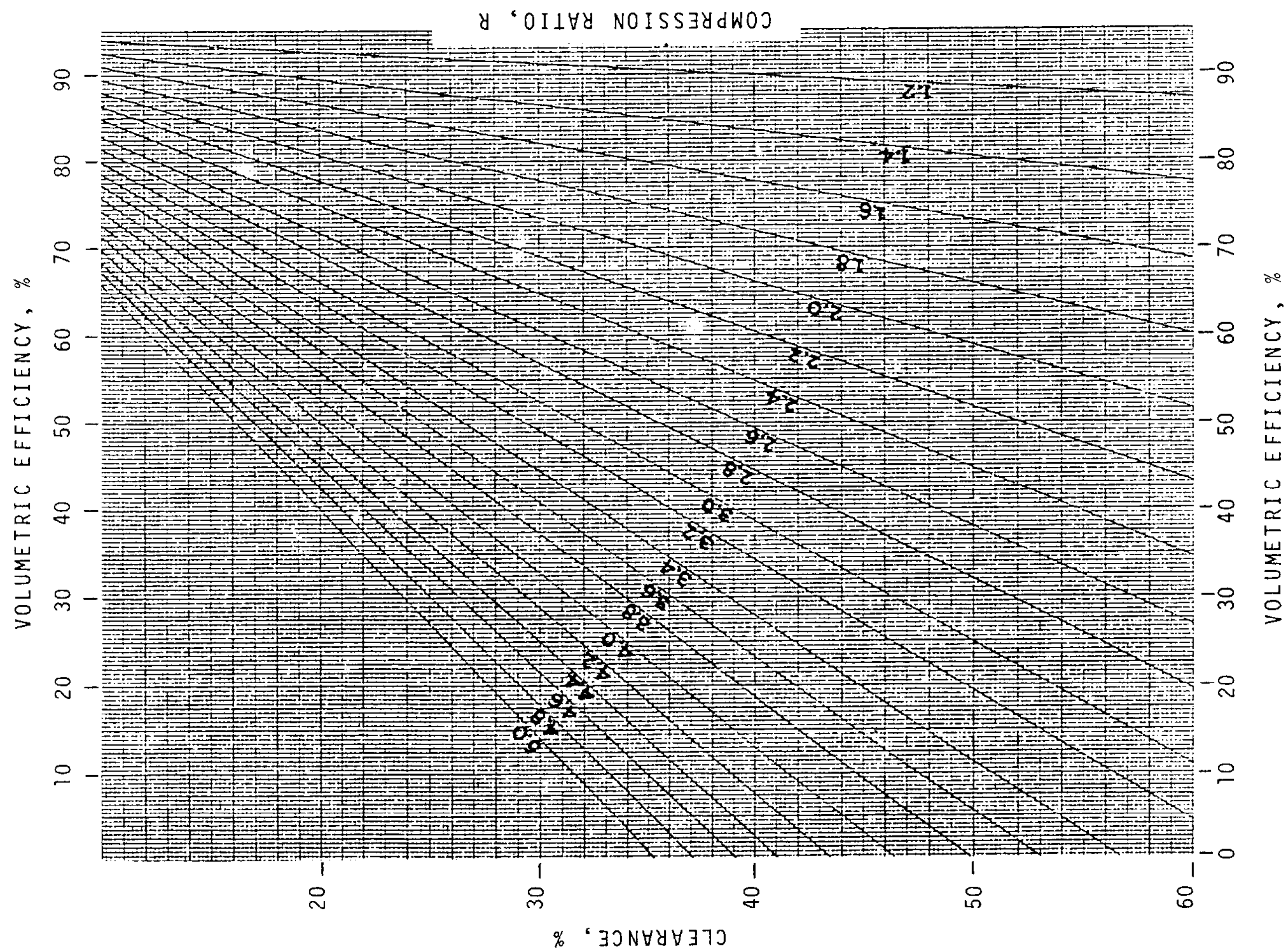

Figure 4 - Clearance vs. volumetric efficiency (Worthington)