Compressors

Selection of gas compressors: part 1

Gas compressors find a wide range of uses, from internal combustion engines and industrial processes to domestic gas supply and refrigeration. In this first article of a nine-part series, Eduardo Larralde and Rafael Ocampo set out the basics, discussing the theoretical laws that govern the behaviour of compressors and the parameters fundamental to their design.

Gas compressors are used in a wide range of applications, such as refrigeration cycles; gas turbines; combustion processes; turbochargers and superchargers in internal combustion engines; pipeline transport of domestic gas; and pneumatic conveyance systems, ejection and air service (pneumatic tools, workshop equipment, instrument drives, cleaning, atomizing, drying and filling/emptying). In industry, compressors also play a very important role in chemical, petrochemical and refining processes.

This series of articles is intended to give engineers in charge of compressor selection and other readers the basic laws regarding compressor design, guidance on the application of different types of compressors, and the procedure for selecting optimum compressor configurations and auxiliaries.

Compression processes

From the thermodynamic standpoint, compression processes can occur in several different ways, namely isothermal, isentropic or polytropic, as expressed in Table 1.1.

The isentropic exponent 'k' is the ratio of the specific heat at constant pressure to the specific heat at constant volume. Its value is easily found from tables of gas properties or suitable software. In contrast, the polytropic exponent value 'n' is influenced by several factors and is rather difficult to calculate.

The compression processes described in Table 1.1 are represented in the p-V diagram shown in Figure 1.1.

Compressor head

Compressor head, as well as flow rate, is a key parameter in the performance of the machine. It represents the work developed by the compressor per unit of weight of the handled fluid. It is expressed in metres or feet (kg·m/kg or lb·ft/lb) and is defined as:

$$H = 101{,}972\int v.dp \tag{1.1}$$

where H is the head in metres (m), v the specific volume (m^3/kg) and p the absolute pressure (MPa). Specific volume (v) can be directly obtained from gas tables or calculated from the inverse of density (ρ).

After substituting and transforming, the following equations are obtained:

$$H_p = 101{,}972[n/(n-1)]p_1v_1[(p_2/p_1)^{(n-1/n)} - 1] \tag{1.2}$$

$$H_p = 101.972[n/(n-1)]ZRT_1[(p_2/p_1)^{(n-1/n)} - 1] \tag{1.3}$$

where H_p is the polytropic head (m); R the gas constant (kJ/kg.K); T_1 the intake temperature (K); and Z the mean compressibility factor. R = 8.3142/MW where MW is the molecular weight of the gas.

Equation 1.2 should be used when the value of the compressibility factor is one. The mean compressibility factor may be used in Equation 1.3 with negligible error in those instances where its deviation is not large, i.e. where the average Z varies between 0.95 and 1.02 or where it remains fairly constant over the range of compression. In other instances, the following formula should be used:

Table 1.1. Types of compression processes

Type of process	Main feature	Equation
Isothermal	No temperature change	$pV = C$
Isentropic	Gas neither absorbs nor emits heat and entropy remains constant (adiabatic reversible)	$pV^k = C$
Polytropic	Continuous real process	$pV^n = C$

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$$H_p = 101{,}972\log(p_2/p_1)\cdot[(p_2v_2 - p_1v_1)/\log(p_2v_2/p_1v_1)] \qquad (1.4)$$

Equation 1.4 is of particular utility for hydrocarbon gases at moderate or high pressures and/or low temperatures.

The correct determination of the polytropic exponent is an extremely important matter if Equations 1.2 and 1.3 are to be successfully applied. To this end, the hydraulic or polytropic efficiency of the compressor is determined from the following equation:

$$\eta = 1{,}000\int v\,dp/\Delta h \qquad (1.5)$$

where η is the hydraulic or polytropic efficiency; and Δh the enthalpy difference (kJ/kg).

The change of enthalpy during compression is:

$$\Delta h = 1{,}000[k/(k-1)]p_1v_1[(p_2/p_1)^{(n-1/n)} - 1] \qquad (1.6)$$

Leading to:

$$\eta = [(k-1)/k]/[(n-1)/n] \qquad (1.7)$$

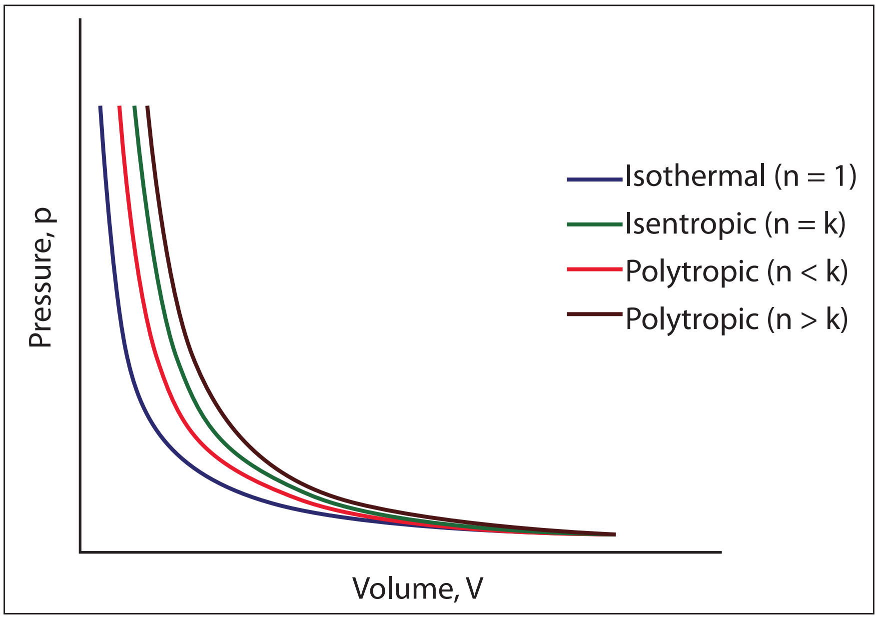

Figure 1.1. p-V diagram for different compression processes.

From Equation 1.7 and knowing or assuming the polytropic efficiency, the polytropic index can be calculated. Polytropic efficiency is generally a function of the compressor capacity at suction conditions for a given compressor and can be determined through testing. Medium-size centrifugal compressors using 2D impellers can reach a polytropic efficiency between 72% and 80%. Large machines using 3D impellers can reach 83% and large axial compressors 85%.

Figure 1.2 shows general approximated values of polytropic efficiency as a function of suction capacity for multistage centrifugal compressors equipped with 2D impellers. Obviously these values can vary according to the specific design

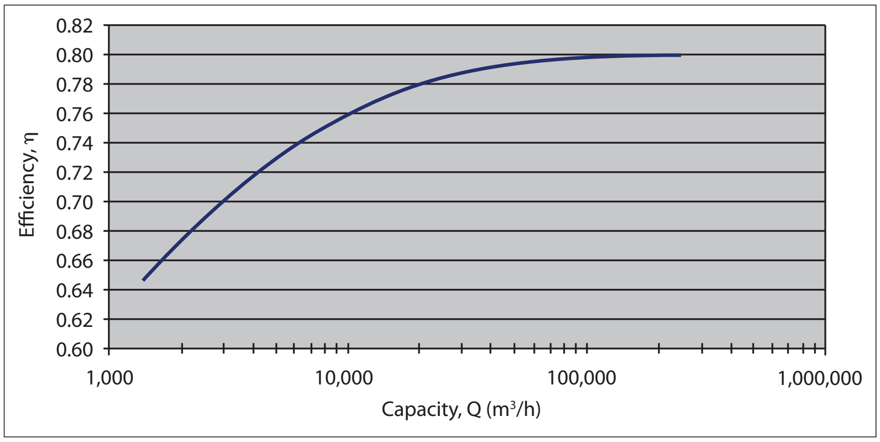

Figure 1.2. General approximated values of polytropic efficiency for centrifugal compressors.

and construction of the compressor, particularly the impellers, so the curve in Figure 1.2 should be used just as a guide to start the calculation procedure. When performing economic analysis for long periods of time, the efficiency values derived from Figure 1.2 should be reduced by several percentage points, mainly due to the influence of the wear of the labyrinth seals.

In order to facilitate calculations, charts usually give the value of $(n-1)/n$ after entering the isentropic exponent 'k' for different values of suction capacity or polytropic efficiency. An example will be presented in Figure 7.1, in part 7 of this series of articles.

For positive displacement compressors, the compression process is nearly isentropic and the corresponding equation can be applied with fairly good results. The same is true for centrifugal compressors using cooled diaphragms:

$$H_a = 101{,}972[k/(k-1)]p_1v_1[(p_2/p_1)^{(k-1/k)} - 1] \quad (1.8)$$

$$H_a = 101.972[k/(k-1)]ZRT_1[(p_2/p_1)^{(k-1/k)} - 1] \quad (1.9)$$

where H_a is the isentropic head in metres.

The equations given above assume a single-phase gas. If the compressor inlet stream contains gas and liquid (i.e. wet gas), these equations must be modified[1]. The use of Equations 1.8 and 1.9 has the same restrictions as Equations 1.2 and 1.3 regarding the values of the compressibility factor. Besides, when dealing with non-perfect gases, the isentropic index may vary as the compression process develops. When the values of k at the beginning and at the end of the compression differ by a small amount, the two figures can be averaged. For other instances it will be necessary to pick an appropriate equation of state or to establish a single compression exponent (γ) through the calculation of the final temperature by using a Mollier diagram and the use of the following equation[2]:

$$\gamma = \ln(p_2/p_1)/[\ln(p_2/p_1) - \ln(T_2/T_1)] \quad (1.10)$$

Not every engineer in charge of compressor selection achieves coincidence when selecting the type of compression process, the type of efficiency (i.e. isentropic, isothermal or polytropic) and the formulae to be used in calculations of performance. Some prefer the isentropic process for any type of air compressor, for single-stage centrifugals and for dry screw compressors. Some engineers even choose isothermal compression for calculations concerning piston compressors with intensive cooling or oil-injected screw compressors. There are manufacturers who use the isentropic cycle for their whole range of centrifugal compressors. Whatever the case, the efficiency type must correspond to the compression process selected.

A polytropic process is more difficult to analyse than a system assumed to be isentropic. The difficulty arises from the fact that heat flows in and out of the system, and this added energy changes some of the basic gas properties, specifically the ratio of specific heats. For a polytropic process, each new calculation requires a new value for this ratio[3].

However, the polytropic compression process is generally selected for rotodynamic compressors as it is better suited to handling the wide range of gases used in industry[3,4], while the performance of positive displacement compressors is calculated using the isentropic cycle[5,6].

Required power

The power required to compress a gas is calculated via the expression:

$$GKW = w\Delta h/3{,}600 \quad (1.11)$$

where GKW is the gas power (kW) and w the gas mass flow rate (kg/h).

For polytropic compression, substitution yields:

$$GKW_p = wH_p/(367{,}200\eta) \quad (1.12)$$

And for isentropic compression:

$$GKW_a = wH_a/367{,}200 \quad (1.13)$$

Also by substitution, the general expressions for gas power required in polytropic and isentropic compression processes respectively can be obtained:

$$GKW_p = [n/(n-1)]\cdot[wZRT_1/(3{,}600\eta)]\cdot[(p_2/p_1)^{(n-1/n)} - 1] \quad (1.14)$$

$$GKW_a = [k/(k-1)]\cdot[wZRT_1/3{,}600]\cdot[(p_2/p_1)^{(k-1/k)} - 1] \quad (1.15)$$

By substituting the Equation of State:

$$GKW_p = 0.2777[n/(n-1)]\cdot[p_1Q_1/\eta]\cdot[(p_2/p_1)^{(n-1/n)} - 1] \quad (1.16)$$

$$GKW_a = 0.2777[k/(k-1)]\cdot[p_1Q_1]\cdot[(p_2/p_1)^{(k-1/k)} - 1] \quad (1.17)$$

where Q_1 is the gas volume flow rate at suction conditions, (m^3/h).

For centrifugals, the rated power at the compressor shaft (KW) is:

$$KW = GKW/\eta_m \quad (1.18)$$

where η_m is the mechanical efficiency.

Table 1.2. Usual mechanical efficiency ranges for compressors

Centrifugal	Without speed reducer/multiplier	98–99%
	With speed reducer/multiplier	96–97%
Piston	Directly coupled large compressors with low pressure rates	90–98%
	Small compressors with high pressure rates	84–89%
Belt transmission		92–98%

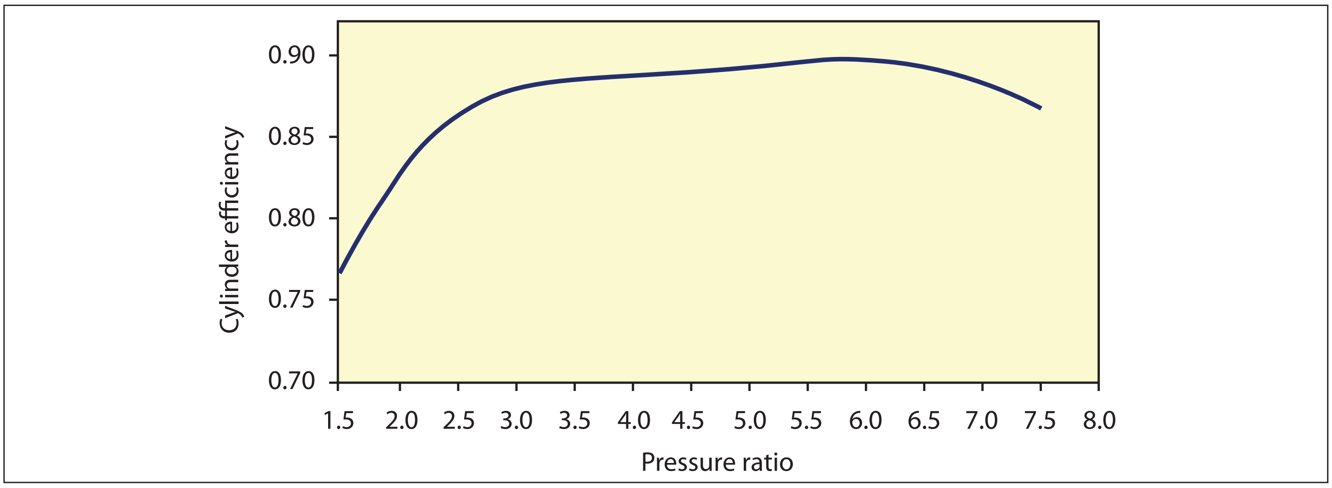

Figure 1.3. Cylinder efficiency for piston compressors.

For reciprocating machines, the mechanical efficiency should be multiplied by the cylinder efficiency (η_c), which is introduced to correct the ideal conditions by considering incremental cylinder sizes and allowable rod loading[2]:

$$KW = GKW/(\eta_m \eta_c) \quad (1.19)$$

Typical ranges for mechanical efficiency are presented in Table 1.2.

Approximated values for cylinder efficiency can be obtained via the pressure ratio by using the curve in Figure 1.3. Compressor power is also influenced by gas specific gravity and inlet pressure. Correction factors can be found in the specialist literature[2].

Discharge temperature

Outlet gas temperature is calculated from the equations:

$$T_2 = T_1(p_2/p_1)^{(n-1/n)} \text{ for polytropic compression} \quad (1.20)$$

$$T_2 = T_1(p_2/p_1)^{(k-1/k)} \text{ for isentropic compression} \quad (1.21)$$

where T_1 and T_2 are the inlet and discharge temperature, respectively, in K.

From the design point of view, the discharge temperatures are generally limited as follows:

- Reciprocating: 150°C
- Centrifugals and axial: 195°C
- Integrally geared: 250°C
- Dry screw: 288°C

However, maximum discharge temperature can be limited by several factors. For reciprocating machines, the maximum predicted discharge temperature must not exceed 150°C and must not exceed 135°C for hydrogen-rich service (MW of 12 or less). Compressor cylinders with gas discharge temperatures of 118°C or less tend to experience longer wear life for cylinder parts[7]. It is generally recommended that the discharge temperature be far lower than these limits. Several design solutions, such as the use of multiple-stage compression and interstage cooling or intensive cooling of the gas during the compression, can be used to reduce the discharge temperature.

Reference conditions for gases

The *normal* or *standard* condition (Nm^3/h: normal cubic metres per hour; sm^3/h: standard m^3/h; scfm; standard cubic feet per minute) varies according to the branch of industry and the specifying authority. More frequently used values for standard conditions according to ISO/CAGI/PNEUROP are: pressure 1 bar, temperature 293 K and

0% relative humidity (dry). However, values established by API are: pressure 1.014 bar (1 ata), temperature 288.5 K and 0% relative humidity (dry)[8].

The *real* condition (m^3/h; am^3/h: actual m^3/h; acfm: actual cubic feet per minute) refers to the gas pressure and temperature existing at the compressor inlet.

For a given compressor running at a given speed, the flow rate remains constant regardless of the temperature, atmospheric pressure or altitude. Several mistakes have been made because this convention has not been taken into account. The conditions that apply to the state of the gas must be known in order to make the necessary corrections through the characteristic equation of a perfect gas. It should also be remembered that, for the same geographic zone, the higher the altitude the lower the barometric pressure and therefore the lower the quantity of air (mass flow rate) admitted by the compressor.

Capacity and volumetric efficiency

The capacity of a compressor is the real quantity of gas delivered but measured at the inlet pressure and temperature and expressed as units of volume per unit of time (usually m^3/h or cfm). Volumetric efficiency is defined as the ratio of the real capacity of the compressor (Q) to its piston displacement (v_d):

$$\eta_v = Q/v_d \qquad (1.22)$$

The capacity of a piston compressor is given by Equation 1.23 for single-acting cylinders and by Equation 1.24 for double-acting cylinders:

$$Q = 15\pi D^2 L N \eta_v \qquad (1.23)$$

$$Q = 15\pi(2D^2 - d^2) L N \eta_v \qquad (1.24)$$

where Q is the compressor capacity (m^3/h); D the cylinder internal diameter (m); d the piston rod diameter (m); L the piston stroke (m); and N the rotating speed (rpm).

Multistage compression

Optimum interstage pressures can be obtained by the formulation and optimization of performance as well as by investment in compressor and interstage facilities. Interstage pressures based just on the compressor without respect to the interstage facilities are not justified. Multistage compression (Figure 1.4) has the following advantages:

- Greater volumetric efficiency than that of a single-cylinder machine of the same clearance and the same total pressure ratio.
- Lower final temperatures.

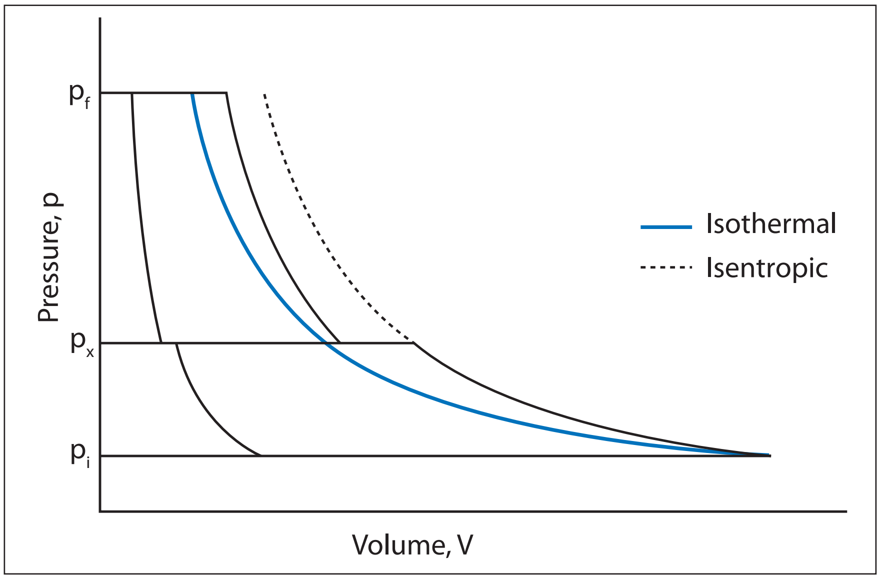

Figure 1.4. p-V diagram for multistage compression, where p_i and p_f are the initial and final pressures, respectively, and p_x is the interstage value.

- The possibility of using an intercooler to lower the gas temperature between stages, which yields a power saving because the combined compression process over all stages is near an isothermal line (represented by the blue line in Figure 1.4).

When the same amount of work is done in each cylinder this minimizes the work to compress a particular mass of gas. For ideal conditions the optimum intermediate pressures for any number of compression stages can be obtained:

$$(p_d/p_s) = (p_f/p_i)^{1/z} \qquad (1.25)$$

where p_d and p_s are the discharge and suction pressure for each stage, respectively; p_f and p_i are the compressor's final and initial pressure, respectively; and z is the number of stages.

For the particular case of a two-stage compression, the interstage pressure p_x is:

$$p_x = (p_f p_i)^{1/2} \qquad (1.26)$$

Obviously, a pressure drop in the intercooler should be spread either side of the ideal values obtained from these equations. It will be demonstrated in parts 8 and 9 of this series that the optimum interstage pressure for real conditions is different from that obtained via the ideal approach. Preliminary optimum pressure drop values are:

- Intercooler: 0.045–0.075 MPa
- Pulsation dampeners and suppression devices: total pressure drop less than 1% of the inlet pressure.

Part 2 of this series will deal with the classification of compressors and will also review the main technical features of each type. ■

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Contact

Eduardo Larralde
Email: larralde@quimica.minbas.cu
Rafael Ocampo
Email: rocampod@gmail.com