

EXXON EXXON ENGINEERING	COMPRESSORS COMPRESSION CALCULATIONS PROPRIETARY INFORMATION - For Authorized Company Use Only	DESIGN PRACTICES	
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SCOPE

This section presents the procedures and calculation methods necessary to determine the process design requirements of a compression service. The information in this section is generally applicable to all compressor types commonly applied in process plants although some calculation methods and information specific to a particular compressor type are also included.⁽¹⁾ This section includes:

1. Compressor calculation flowchart.
2. Methods to determine the number of process stages required for a compression service.
3. Compressor inlet volume flow rate calculation (at condition).
4. Information to estimate the efficiency of various compressor types.
5. Calculations to estimate the inlet and outlet pressures for each process compression stage.
6. Calculations for process compression stage outlet temperature.
7. Outlet temperature limits for various compressor types and services.
8. Calculations for head, efficiency, power, pressure ratio and volume ratio.
9. Information to estimate the characteristics of a process compression stage.
10. Information about computer programs to automate and improve the compression calculations.

REFERENCES

COMPUTER PROGRAMS

CPEE-41 CompCalc - PC Program for Compressor Network Simulation and Design
 O2666 Reciprocating Compressor Selection

TECHNICAL LITERATURE

ASME Performance Test Code 10, *Compressors and Exhausters*, 1965, Reaffirmed 1987.
 Schultz, J. M., *The Polytropic Analysis of Centrifugal Compressors*, ASME J. of Engineering for Power, January 1962, p. 69-82.
 Huntington, R. A., *Evaluation of Polytropic Calculation Methods for Turbomachinery Performance*, ASME J. of Engineering for Gas Turbines and Power, October 1985, p. 872-878.

BACKGROUND

A variety of compressor types may be applied within process plants. However centrifugal, reciprocating, axial, and helical screw compressors are the most common. These types include both dynamic compressors (centrifugal and axial) and positive-displacement (reciprocating and helical screw) compressors which require somewhat different methods to calculate or predict their performance. However, each of these compressors and their calculations share common considerations, as follows:

- Each compressor type raises the pressure of a gas by adding energy to the gas. This energy is known as **Head** and has units of energy per unit mass of gas (e.g., ft-lb/lb_m or kJ/kg).
- The capability of a compressor to raise gas pressure is dependent on the volume of gas entering the compressor at its current inlet conditions. This volume rate is known as its **Actual Inlet Volume Flow** rate and has common units of ACFM (actual cubic ft per minute) and ACMH (actual cubic meters per hour).
- The **Efficiency** of a compressor is measured by the ratio of the head found for a reversible (idealized) compression process between the same pressure levels and the enthalpy change of the gas for the actual compression process.
- The **Gas Power** of a compression stage is the product of total work done (reversible work/efficiency) to raise the gas pressure and the mass flow rate of gas being compressed. Gas Power includes all thermodynamic losses such as internal leakage, gas friction within the compressor gas path but excludes mechanical losses.

The complexity of the compressor calculations ranges from very simple where the compressed gas acts as a **Perfect Gas** to difficult for non-ideal gases or those that condense at cooling steps within the compression process. The more difficult cases are better suited to partial or full solution via computer.

Note:

- (1) API compressor data sheets specify pressure in psi, bar, and kPa. In this section, bar is the metric unit used for pressure. See [Section XI-A, Nomenclature](#), for parameter symbols and units not defined in text.

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DEFINITIONS

Head

Reversible work necessary to raise the pressure of a gas.

Ideal Gas

A gas for which the relation between pressure, temperature and volume can be expressed as:

$$PV = RT$$

over the range of interest of pressure and temperature.

Isentropic Efficiency, η_s

Ratio of Isentropic Head vs. the enthalpy rise from the initial pressure and temperature to the final pressure and temperature.

Isentropic Head, H_s

Head calculated along an isentropic path (reversible adiabatic) from the initial pressure and temperature to the desired pressure as follows:

$$H_s = h_{2s} - h_1$$

where: h_1 = $h(P_1, T_1)$, enthalpy at inlet conditions
 h_{2s} = $h(P_2, T_{2s})$, enthalpy at discharge conditions
 T_{2s} = The temperature at which $s(P_2, T_{2s}) = s(P_1, T_1)$
 T_1 = Inlet temperature
 P_1 = Inlet pressure
 P_2 = Discharge pressure
 s = Entropy

Perfect Gas

A gas which conforms to the ideal gas relationship between pressure, temperature and volume and which has a constant specific heat over the range of interest of pressure and temperature.

Polytropic Efficiency, η_{poly}

Ratio of polytropic head vs. the enthalpy rise from the initial pressure and temperature to the final pressure and temperature.

Polytropic Head, H_{poly}

Head calculated along a reversible path along which reversible head and reversible heat are added in constant proportions to raise the pressure and temperature of the gas from the initial values to the desired pressure and actual outlet temperature. A polytropic path is described for all gas conditions by the differential equation: $dH_{poly} = \eta_{poly}dh$. For perfect gases, this is simplified to the familiar: $PV^n = \text{constant}$ along the polytropic path where n is a constant known as the polytropic volume exponent.

Pressure Ratio, r

The ratio of the absolute outlet pressure vs. the absolute inlet pressure of a compression stage.

Volume Ratio, V_1 / V_2

The ratio of the inlet volume flow rate or inlet specific volume of the gas vs. the outlet volume flow rate or outlet specific volume of the gas.

Volumetric Efficiency, η_v

The ratio of the inlet volume flow rate of a reciprocating compressor stage vs. the piston displacement of that stage. Piston displacement is the product of the cylinder swept volumes and the compressor speed in revolutions per unit time (e.g., RPM). Swept volume of a cylinder is the product of piston area and stroke for each acting end of the cylinder.

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COMPRESSOR DESIGN FLOWCHART

A flowchart to complete the process design calculations for a new compressor service is illustrated in **Figure 1**. This flowchart describes the overall steps to determine characteristics of the compression equipment for the new service including: the number of process stages, the efficiency, head and power of each stage, the overall power of the compression equipment, and the physical characteristics of the compressors. This **Compressor Calculations Flowchart** must be used in conjunction with **Section XI-B**, *Compressor Type Selection* and **Sections XI-E, XI-F, XI-H, and XI-I** for the particular type of compressor to be used.

For each of the steps described in the flowchart, a section follows that details the information required, sources for that information and necessary calculations to complete the step. In some cases, the steps will be repeated as the information is refined. In other cases, it may become necessary to re-evaluate the compressor type selection and repeat the process in total. These are expected and normal aspects of the compressor design process.

INTERSTAGE COOLER PRESSURE DROP

Interstage cooling of compressed gas streams is necessary or desired for two primary reasons: 1) to limit the temperature of the gas to some specified maximum, and 2) to reduce the compression power necessary for a service based on an economic evaluation by reducing the head requirement through the lowering of the mean temperature of the compressed gas and/or by reducing the mass flow rate of gas by removing condensable material. Interstage equipment generally consists of an air-fin or shell-and-tube heat exchanger, a knock-out drum fitted with a crinkle-wire-mesh screen (CWMS) and interstage piping that may include a flow meter.

A typical value of pressure drop for the interstage system described above is 7 to 10 psi (0.5 to 0.7 bar). Use other values based on the calculated pressure drop for the interstage equipment, if known. This may especially be required for low-pressure applications [cooler inlet pressure less than 50 psig (3.5 barg)]. However, in other cases use 7 psi (0.5 bar) as an approximation.

INLET VOLUME FLOW RATE

The inlet volume flow rate refers to the actual volume flow rate at the inlet flange of the compressor section. This volume rate may be calculated based on the weight, molar or standard volume flow rate plus gas properties at the inlet pressure and temperature at the flange of the compressor; a nominal pressure drop for an inlet throttle valve (if installed) should be subtracted from the upstream pressure prior to evaluation of the inlet conditions. A typical pressure drop for an inlet throttle valve should be based on a line-size butterfly valve that is 80% open at the design flow rate to the compressor.

Use the following calculations to determine the inlet volume flow rate to the compressor:

CUSTOMARY	METRIC
$Q_1 = \frac{\text{SCFM}}{378.7} \cdot \text{MW} \cdot V_1$ (ACFM)	$Q_1 = \frac{\text{SCMH}}{23.65} \cdot \text{MW} \cdot V_1$ (ACMH)
$Q_1 = \text{Mol} \cdot \text{MW} \cdot V_1$ (ACFM)	$Q_1 = \text{Mol} \cdot \text{MW} \cdot V_1$ (ACMH)
$Q_1 = W \cdot V_1$ (ACFM)	$Q_1 = W \cdot V_1 \cdot 3600$ (ACMH)

where:

$V_1 = \frac{Z_1 R T_1}{144 P_1}$ (ft ³ /lb)	$V_1 = \frac{Z_1 R T_1}{100 P_1}$ (m ³ /kg)
$R = \frac{1545.31}{\text{MW}}$ $\left(\frac{\text{ft} \cdot \text{lb}_f}{\text{lb}_m \cdot ^\circ\text{R}} \right)$	$R = \frac{8.31427}{\text{MW}}$ $\left(\frac{\text{kJ}}{\text{kg} \cdot ^\circ\text{K}} \right)$

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COMPRESSOR EFFICIENCY ESTIMATES

Figures 2 to 4 provide estimates for the polytropic efficiency of uncooled centrifugal and reciprocating compressors. For cooled reciprocating compressors, which include most applied to process services, multiply the value from **Figure 4** by 1.05 to account for the cylinder cooling effectiveness. For Helical Screw compressors, use a polytropic efficiency estimate of 75% unless a specific efficiency is available from a vendor quotation.

DISCHARGE TEMPERATURE ESTIMATES

The discharge temperature of any compressor section is predicted based on the pressure rise of the section, the efficiency of the compressor section, and the properties of the compressed gas. The discharge temperature may be estimated as:

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^m$$

where: T_1 and T_2 are absolute temperatures in either °R or °K.

P_1 and P_2 are the absolute inlet and outlet pressure of the compressor section.

and m is the Polytropic Temperature Rise Exponent.

The Polytropic Temperature Rise Exponent is exactly given by the following formula at a known pressure and temperature:

$$m = \frac{Z R}{C_p} \left(\frac{1}{\eta_{poly}} + X \right)$$

where: $X = \frac{T}{Z} \left(\frac{\partial Z}{\partial T} \right)_p$ which is one of the Schultz compressibility derivative functions.

The values of Z , C_p and X can be calculated with the CompCalc PC program Gas Properties section. Z and C_p may also be obtained from a flow sheet program calculation (PRO/II, ASPEN, COPE, etc.) with X estimated based on the generalized chart included in PTC-10 (1965) and reproduced as **Figure 6**.

For Real Gases, the value of m will change along the compression path and therefore an average value should be calculated at the inlet and roughly estimated outlet conditions to more accurately predict the compressor section discharge temperature.

$$m_{avg} = \frac{m_1 + m_2}{2}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{m_{avg}} \quad (^\circ\text{R or } ^\circ\text{K})$$

Repeat the evaluation of m_2 at the calculated value of T_2 and repeat the discharge temperature calculation as necessary to achieve the desired accuracy.

Ideal and Perfect Gases: For compressed gases that may be approximated as Ideal or Perfect Gases, the evaluation of the Polytropic Temperature Rise Exponent may be simplified. For Ideal Gases, the compressibility factor is constant. Therefore, the value of X is zero and the temperature rise exponent may be expressed as:

$$m = \frac{Z R}{C_p \eta_{poly}} \quad \text{For Ideal Gases only}$$

For Perfect Gases, C_p is also constant and is equal to $ZR + C_v$. By substitution and re-arrangement, this expression becomes:

$$m = \frac{k - 1}{k \eta_{poly}} \quad \text{For Perfect Gases only}$$