ENGINEERING STANDARD

FOR

PROCESS DESIGN OF COMPRESSORS

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APPENDICES:

0. INTRODUCTION

"Process Design of Pressure Reducing/Increasing Machineries and or Equipment" are broad and contain various subjects of paramount importance. Therefore a group of process engineering standards are prepared to cover the subject.

This group includes the following standards:

STANDARD CODE	STANDARD TITLE
IPS-E-PR-740	"Process Design of Pumps"
IPS-E-PR-745	"Process Design of Vacuum Equipment (Vacuum Pumps and Steam Jet Ejectors)"
IPS-E-PR-750	"Process Design of Compressors"
IPS-E-PR-755	"Process Design of Fans and Blowers"

This Engineering Standard Specification covers:

"PROCESS DESIGN OF COMPRESSORS"

1. SCOPE

This Engineering Standard Specification covers the minimum requirements, basic reference data and necessary formulas for process calculations and proper selection of compressors to be used in the OGP industries.

Compressors are dealt within four groups; axial, centrifugal, reciprocating and rotary, and each covered in separate section. Vacuum pumps, jet ejectors and air compressors are discussed in IPS-E-PR-745 and IPS-E-PR-330 respectively.

2. REFERENCES

Throughout this Standard the following standards and codes are referred to. The editions of these standards and codes that are in effect at the time of publication of this Standard shall, to the extent specified herein, form a part of this Standard. The applicability of changes in standards and codes that occur after the date of this Standard shall be mutually agreed upon by the Company and the Vendor/Consultant/Contractor.

API (AMERICAN PETROLEUM INSTITUTE)

API Std. 617, 5th. Ed., Apr. 1988	"Centrifugal Compressors for General Refinery Services"
API Std. 618, 3rd. Ed., February 1986	"Reciprocating Compressors for General Refinery Services"
API Std. 619, 2nd. Ed., May 1985	"Rotary Type Positive Displacement Compressors for General Refinery Services"
API Publication No. 2564, March 1974	"Conversion of Operational and Process Measurement Units to the Metric (SI) System"

GPSA (GAS PROCESSORS SUPPLIERS ASSOCIATION)

"Engineering Data Book", Vo1. 2, 1987

IPS (IRANIAN PETROLEUM STANDARDS)

IPS-E-PR-330	"Process Design of Production & Distribution of Compressed Air Systems"
IPS-M-PM-170	"Centrifugal Compressors for Process Services"
IPS-M-PM-190	"Axial Flow Centrifugal Compressors"
IPS-M-PM-200	"Reciprocating Compressors for Process Services"
IPS-M-PM-220	"Positive Displacement Compressors, Rotary"

3. DEFINITIONS AND TERMINOLOGY

3.1 Terms used in this Standard are defined as per Section 1.4 of API Standard 617 for centrifugal and axial compressors, Section 1.4 for API Standard 618 for reciprocating compressors and Section 1.4 of API Standard 619 for rotary compressors, unless otherwise stated in this Section.

3.2 Inlet Cubic Meters per Hour (Im^3/h)

Refers to flow rate determined at the conditions of pressure, temperature, compressibility and gas composition, including moisture, at the compressor inlet flange (substitution to API Std. 617, 1.4.18).

3.3 Actual Cubic Meters per Hour (Am^3/h)

Refers to the flow rate at flowing conditions of temperature and pressure at any given location. Because this term describes flow at a number of locations, it should not be used inter-chanegably with inlet m^3/h (substitution to API Std. 617, 1.4.19).

3.4 Standard Cubic Meter per Hour (Sm^3/h)

Refers to the flow rate at any location corrected to a pressure of 101.325 kPa and at a temperature of 15°C with a compressibility factor of 1.0 and in a dry condition.

3.5 Normal Cubic Meters per Hour (Nm^3/h)

Refers to a flow rate at any location corrected to the normal atmospheric pressure and a temperature of 0°C with a compressibility factor of 1.0 and in dry conditions.

3.6 Specific Volume

Is the volume per unit mass or volume per mole of material.

4. UNITS

This Standard is based on International System of Units (SI), except where otherwise specified.

5. GENERAL

5.1 Compressors are generally divided into three major types, dynamic, positive displacement and thermal. Categorization of compressors by "Gas Processors & Supplies Association (GPSA)", is shown in Fig. A.1 of Appendix A.

5.2 The type of compressor to be used shall be the most suitable for the duty involved. See the compressor coverage chart in Fig. A.2 of Appendix A.

5.3 Adequate knock out facilities including demister pads where necessary shall be provided to prevent dam- age by liquid carry over into the compressor.

5.4 Compressors handling SO_2, HCl or other gases which are corrosive in the presence of water, shall not employ water as a cooling medium unless the water circuit is positively isolated from the gas side, e.g., by separate water jackets. It is not sufficient to rely on gaskets or seals for isolation.

Similar restrictions shall apply to the use of glycol as a coolant for machines handling corrosive gases plus hydrogen as the hydrogen can react with glycol to form water. The use of oil as a cooling medium will be acceptable as an alternative in special cases.

5.5 Rotodynamic compressors are to be provided with anti-surge equipment. The response time for the control equipment shall be such as to prevent surge during any anticipated process condition, due consideration being given to the speed at which process changes or upsets can move the compressor operation towards surge.

For the more complicated installations with multiple stages and sidestreams, or multiple units (in series or parallel) or variable speed units, an analysis of the stability of the anti-surge control system is also necessary.

5.6 Type Selection Criteria

The choice of the type of compressor, whether axial, centrifugal, reciprocating or rotary, depends primarily on the required flow to be compressed, the density of the gas in conjunction with the total head (for a given gas, this is the compression ratio) and the duty which has to be performed. Table A.1 of Appendix A outlines the compression limits for the four types of compression equipment.

5.6.1 Axial compressors

Axial compressors can handle large volume flow and are more efficient than centrifugal compressors. However, centrifugals are less vulnerable and hence more reliable, have wider operating ranges and are less susceptible to fouling.

Axial compressors should be considered only for air, sweet natural gas or non-corrosive gases. Axial compressors shall be in accordance with Iranian Petroleum Standards IPS-M-PM-190, for "Axial Flow Centrifugal Compressors".

5.6.2 Centrifugal compressors

Providing a centrifugal compressor can handle the required flow with a reasonable efficiency, then this type is the preferred choice because it has the potential to operate continuously for long periods, if properly designed and assembled. If the flow at discharge conditions is 300 m^3/h or more, then a centrifugal compressor shall always be considered.

Centrifugal compressors shall be designed in accordance with API Std. 617 as amended by Iranian Petroleum Standards IPS-M-PM-170, for "Centrifugal Compressors for Process Services".

5.6.3 Reciprocating compressors

Where the required flow is too small for a centrifugal compressor, or where the required head is so high that an undesirably large number of stages would be necessary, then generally the choice should be a reciprocating compressor.

As a reciprocating compressor cannot fulfill the minimum requirement of continuous uninterrupted operation for a two-year period, due to fairly high maintenance requirements, a full-capacity spare shall be provided as general rule for reciprocating compressors in critical services. Alternatively, three half-capacity machines may be specified, two running in parallel with the third unit as a spare. Reciprocating compressors shall be in accordance with API Std. 618 as amended by IPS-M-PM-200, for "Reciprocating Compressors for Process Services".

5.6.4 Rotary compressors

Rotary compressor shall be considered only where there is proven experience of acceptable performance of this type of compressor in the duty concerned and only where there are advantages over a reciprocating compressor.

The application of oil flooded screw compressors for instrument air and of dry running rotary screw compressors, sliding vane compressors and rotary lobe compressors for process duties, requires the explicit approval of the Company.

Rotary-type positive displacement compressors shall be in accordance with API Std. 619 as amended by IPS-M-PM-220, for "Positive Displacement Compressors, Rotary".

5.7 Atmospheric Pressure

The absolute pressure of the atmosphere at the site should be considered as the "absolute pressure" in the compressor calculations. The value of the absolute pressure is taken as 101.342 kPa at sea level and declines with increasing altitude as shown in Table A.2 of Appendix A.

5.8 Specification Sheets

Process information required to complete specification sheets for compressors are presented in Appendix B.

6. CENTRIFUGAL COMPRESSORS

6.1 General

6.1.1 The centrifugal (radial flow) compressor is well established for the compression of gases and vapors. It has proven its economy and uniqueness in many applications, particularly where large volumes are handled at medium pressures.

6.1.2 Centrifugal compressors shall conform to API Std. No. 617 for all services handling air or gas, except machines developing less than 35 kPa (0.35 bar) from atmospheric pressure, which may be classified as fans or blowers.

6.1.3 Performance

6.1.3.1 Compressors shall be guaranteed for head, capacity, and satisfactory performance at all specified operating points and further shall be guaranteed for power at the normal operating point.

6.1.3.2

a) The volume capacity at the surge point shall not exceed the specified percentage of normal capacity at normal speed, and normal (unthrottled) suction conditions. The rise in pressure ratio from normal capacity to the surge point at normal speed shall not be less than that specified.

b) The head developed at 115% of normal capacity at normal speed shall be not less than approximately 85% of the head developed at the normal operating point.

6.1.3.3 The head-capacity characteristic curve shall rise continuously from the rated point to the predicted surge. The compressor, without the use of a bypass, shall be suitable for continuous operation at any capacity at least 10 percent greater than the predicted approximate surge capacity shown in the proposal (API Std. 617, 2.1.3).

6.1.3.4 For variable speed compressors, the head and capacity shall be guaranteed with the understanding that the power may vary ±4%.

6.1.3.5 For constant-speed compressors, the specified capacity shall be guaranteed with the understanding that the head shall be within ±5% and -0% of that specified; the power shall not exceed stated power by more than 4%. These tolerances are not additive.

6.1.4 The compressor manufacturer shall be responsible for checking the "k" (ratio of specific heats) and "Z" (compressibility factor) values specified against the gas analysis specified.

6.1.5 Compressor mach numbers shall not exceed 0.90 when measured at any point.

6.2 Design Criteria

6.2.1 This Section of Standard covers information necessary to select centrifugal compressors and to determine whether the selected machine should be considered for a specific job.

6.2.2 An approximate idea of the flow range that a centrifugal compressor will handle is shown in Table 1. A multi-stage centrifugal compressor is normally considered for inlet volumes between 850 and 340,000 Im^3/h. A single stage compressor would normally have applications between 170 and 255,000 Im^3/h. A multi-stage compressor can be thought of as series of single stage compressors contained in a single casing.

TABLE 1 - CENTRIFUGAL COMPRESSOR FLOW RANGE

NOMINAL FLOW RANGE (INLET m³/h)	AVERAGE POLYTROPIC EFFICIENCY	AVERAGE ISENTROPIC EFFICIENCY	SPEED TO DEVELOP 3048 m HEAD/WHEEL
170 - 850	0.63	0.60	20,500
850 - 12,743	0.74	0.70	10,500
12,743 - 34,000	0.77	0.73	8,200
34,000 - 56,000	0.77	0.73	6,500
56,000 - 93,400	0.77	0.73	4,900
93,400 - 135,900	0.77	0.73	4,300
135,900 - 195,400	0.77	0.73	3,600
195,400 - 246,400	0.77	0.73	2,800
246,400 - 340,000	0.77	0.73	2,500

6.2.3 Effect of speed

6.2.3.1 With variable speed, the centrifugal compressor can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination of variable capacity and variable pressure.

6.2.3.2 Basically, the performance of the centrifugal compressor, at speeds other than design, follows the affinity (or fan) laws.

6.2.3.3 By varying speed, the centrifugal compressor will meet any load and pressure condition demanded by the process system within the operating limits of the compressor and the driver.

6.2.4 Performance calculation

6.2.4.1 Determination of properties pertaining to compression

Compressibility factor (Z factor), ratio of specific heats (C_p/C_v or k value) and molecular mass are three major physical properties for compressor which must be clarified. Mollier diagrams should be used if available.

6.2.4.2 Determination of suction conditions

The following conditions at the suction flange should be determined:

a) Temperature

b) Pressure

In case of air taken from atmosphere, corrections should be made for elevation. Air humidity should also be considered.

c) Flow rate

All centrifugal compressors are based on flows that are converted to inlet or actual conditions (Im³/h or inlet cubic meters per hour). This is done because centrifugal compressor is sensitive to inlet volume, compression ratio (i.e., head) and specific speed (see 6.2.7).

d) Fluctuation in conditions

Since fluctuations in inlet conditions will have large effects on the centrifugal compressor performance, owing to the compressibility of the fluid, all conceivable condition fluctuations must be taken into consideration in determination of design conditions.

6.2.4.3 Determination of discharge conditions

6.2.4.3.1 Calculation method

Discharge conditions of a centrifugal compressor can be calculated by the following procedure.

a) Calculate the polytropic exponent "n":

1) using the equation:

$$\frac{n}{n \; 1} = \frac{k}{k \; 1} ¢ \pm p \qquad \textbf{(Eq. 1)}$$

if η_p (polytropic efficiency) is known from the manufacturer data. η_p can also be estimated from Table 1 (k is the ratio of specific heats).

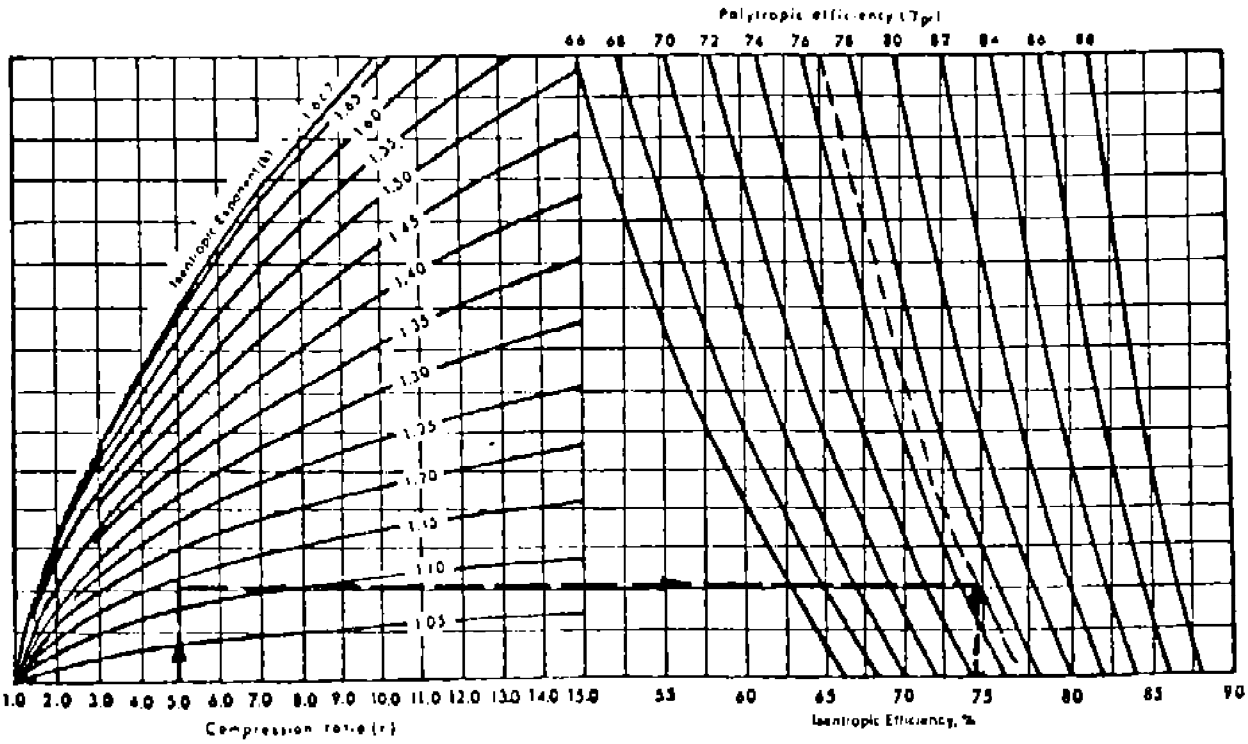
2) if η_{is} (isentropic or adiabatic efficiency) is known, then η_p can be found from Figs. 1 or 2 and the Equation 1 can be used to calculate "n".

3) Fig. 3 is useful for rough estimation of "n".

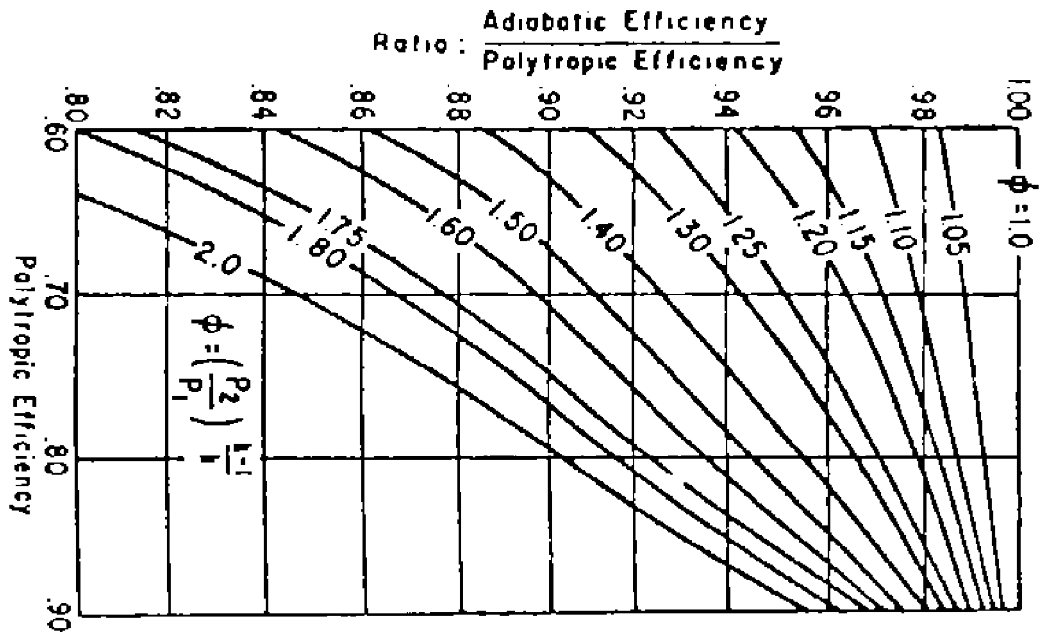
4) "n" can also be calculated iteratively from equation:

$$n = \frac{\log 10 \, (P_2 = P_1)}{\log 10 \, (V_1 = V_2)} \qquad \textbf{(Eq. 2)}$$

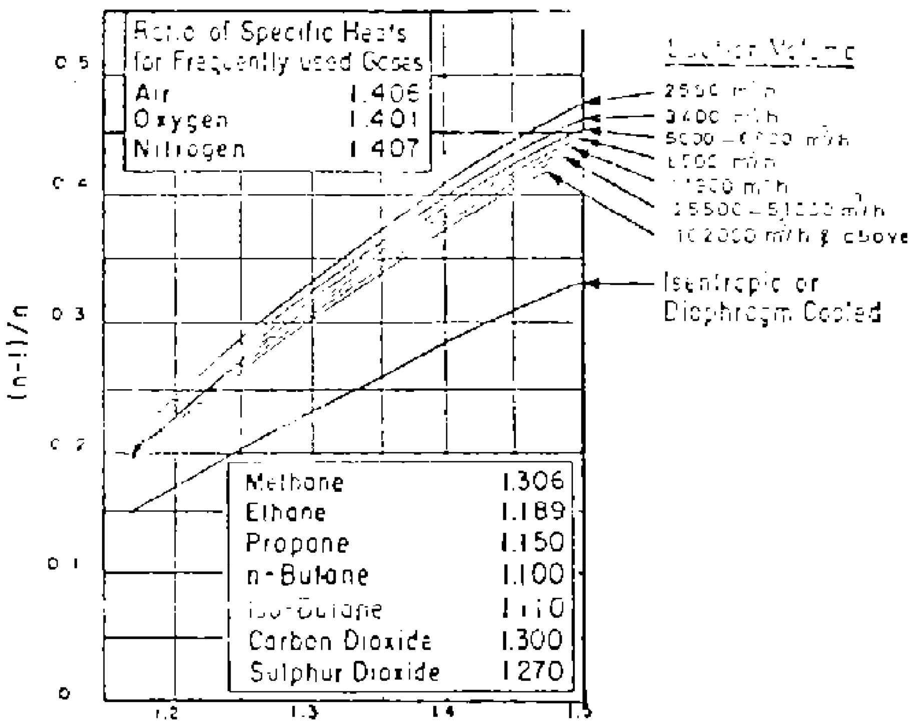
Where V_1 and V_2 are specific volumes (actual) and P_1 and P_2 are absolute pressures at inlet and outlet conditions respectively.



EFFICIENCY CONVERSION
Fig. 1



RELATIONSHIP BETWEEN ADIABATIC AND POLYTROPIC EFFICIENCIES
Fig. 2



Ratio of Specific Heats, $k = \frac{C_p}{C_v}$

(n-1)/n VERSUS RATIO OF SPECIFIC HEATS
Fig. 3

This equation applies with good accuracy for single wheels and the overall multistage compressor.

b) Calculate discharge temperature T_2, (kelvin) from equation:

$$T_2 = T_1\,(P_2/P_1)^{(n-1)/n} \qquad (T_1 \text{ and } T_2 \text{ are absolute temperatures}) \qquad \textbf{(Eq. 3)}$$

These values are for polytropic compression in an uncooled compressor with no diaphragm cooling, no liquid injection and no external coolers. In the cases of internal cooling, the adiabatic exponent "k" approximates the actual condition.

In such cases:

$$\Delta T = T_1 \cdot \frac{\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]}{\eta_{is}} \qquad \textbf{(Eq. 4)}$$

Where:

ΔT	is the temperature increase, in (°C).

Note:

The operating temperature should not exceed 190°C (375°F) at any point in the operating range, otherwise, difficulties will be encountered in the mechanical design, higher temperatures up to 232°C (450°F) are subject to Company's approval.

c) Calculate adiabatic (isentropic) head H_{is} (meters):

$$H_{is} = \frac{Z_{ave}\,R.\,T_1}{g.M(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{(Eq. 5)}$$

Where:

R	is gas constant, in (8314.3 J/kmol.K);
Z_{ave}	is average inlet and outlet compressibility factors;
T_1	is inlet absolute temperature, in kelvin (K);
M	is molecular mass, in (kg/kmol);
g	is acceleration of gravity, in (9.80665 m/s²).

d) Calculate polytropic head H_p:

$$H_p = \frac{Z_{ave}\,R.\,T_1}{g.M(n-1)/n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \textbf{(Eq. 6)}$$

Note:

Polytropic and isentropic heads are related by:

$$\frac{H_p}{H_{is}} = \frac{\eta_p}{\eta_{is}} \qquad \textbf{(Eq. 7)}$$

e) Calculate gas horse power in kilowatt (hp):

$$Ghp = \frac{W.\ H_p}{6119.099\ \eta_p} \qquad \text{(Eq. 8)}$$

or;

$$Ghp = \frac{W.\ H_{is}}{6119.099\ \eta_{is}} \qquad \text{(Eq. 9)}$$

Where:

W	is mass flow rate, in (kg/min);
H_p	is polytropic head, in (m).

f) Estimate head per stage:

1) Use the following equation based on molecular mass for calculation of maximum head per stage:

$$H_{max}/stage = 4572 - 457.2\ (M)^{0.35} \qquad \text{(Eq. 10)}$$

2) Find uncorrected number of stages S_N:

$$S_N = \frac{H_p}{H_{max}/\ stage} \qquad \text{(Eq. 11)}$$

3) Correct number of stages by choosing the next upper integer, S.

4) Find H', the head per impeller:

$$H' = \frac{H_p}{S} \qquad \text{(Eq. 12)}$$

g) Estimate speed and wheel diameter:

1) Use Fig. 4 to find the size number;
2) find the approximate wheel diameter, D, from Table 2;
3) choose pressure coefficient, æ and find peripheral velocity, u, from Fig. 5;
4) calculate N from the equation.

$$N = \frac{59809.42}{D}\sqrt{\frac{H'}{\mu}} = 19108.33 \times u / D \qquad \text{(Eq. 13)}$$

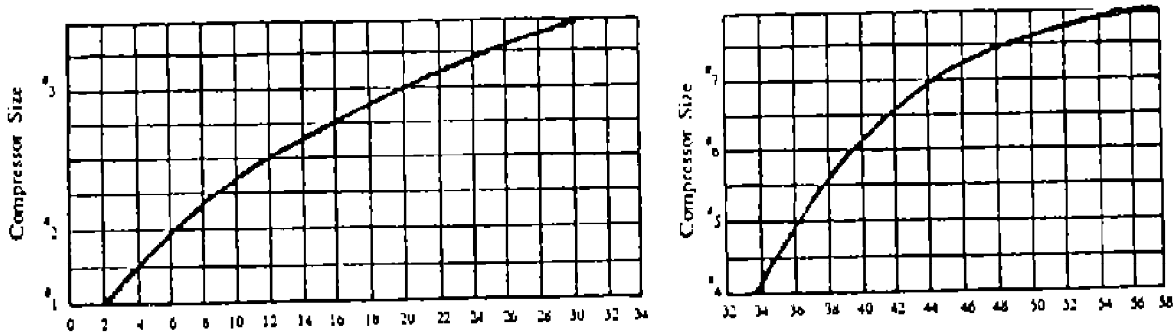
Where:

N = r/min (rpm)

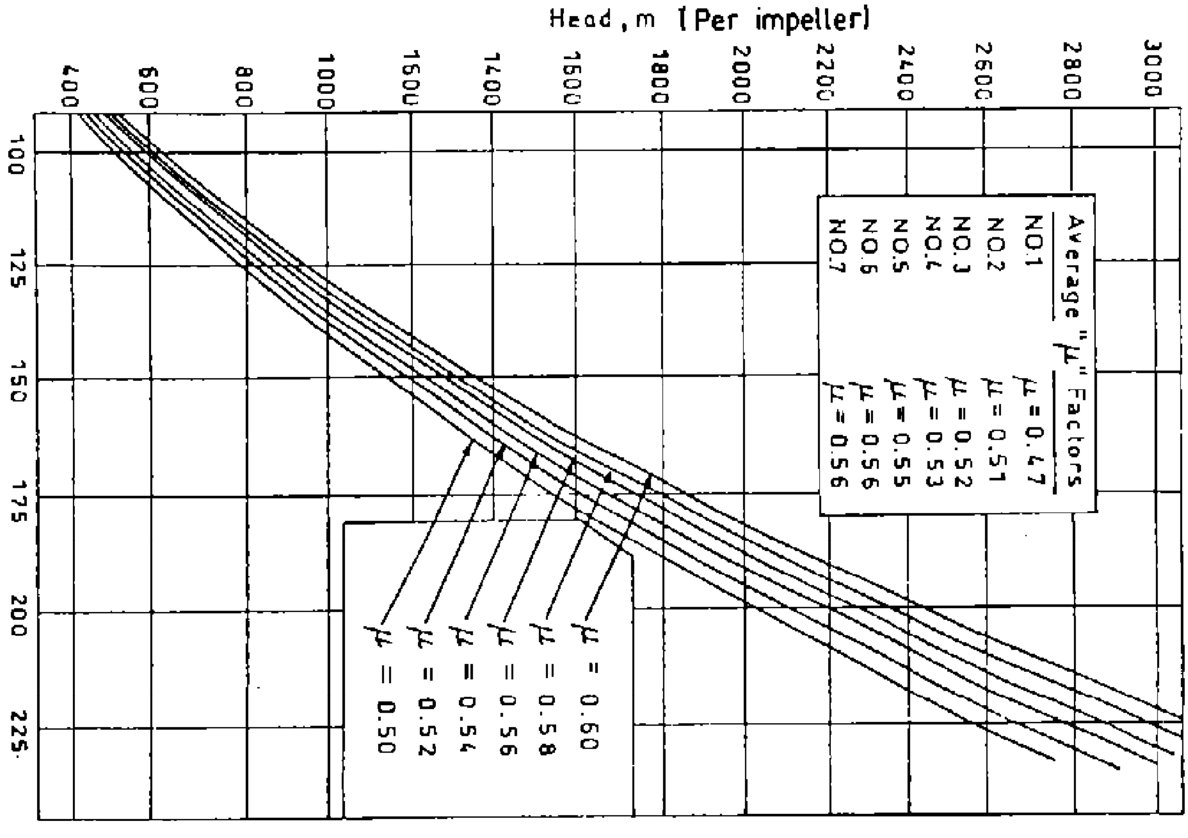
H_{total}	is total compressor head (meters of fluid);
S	is No. of stages S, can also be found by dividing H_p by H';
D	is impeller diameter, in (mm);
H'	is head per stage, in (meters of fluid);
μ(mu)	is pressure coefficient, values range from 0.5 to 0.6 (average 0.55, use Fig. 5);
u	is peripheral velocity, in (m/s).

TABLE 2 - APPROXIMATE WHEEL DIAMETER vs SIZE NUMBER

Size No.	1	2	3	4	5	6	7
Wheel diameter (mm)	375	450	600	800	1060	1350	1650



CENTRIFUGAL COMPRESSOR SIZE vs CAPACITY
Fig. 4



PERIPHERAL VELOCITY (m/s)
Fig. 5

h) Brake horse power (in kilowatt):

Is determined by addition of power losses due to friction in bearings, seals and speed increasing gears to the gas horse power in kilowatts (Ghp). The equation:

$$Mechanical\ losses = 0.663\ (Ghp)^{0.4} \quad \text{(Eq. 14)}$$

is a good estimation for these losses.

$$Bhp = hp + mechanical\ losses \quad \text{(Eq. 15)}$$

6.2.4.3.2 P-H diagram method

The use of an enthalpy diagram, when available, is the most accurate and an easy method for determining power. Fig. 7 represents a section of a typical P-H diagram. The following procedure should be followed: starting from point 1 (inlet conditions), follow the line of constant entropy to the required discharge pressure P_2, locating the isentropic discharge state point (2_{is}). Now the differential isentropic enthalpy can be calculated from:

$$\Delta h_{is} = h_{2is} - h_1 \quad \text{(Eq. 16)}$$

H_{is} (isentropic head) can then be found:

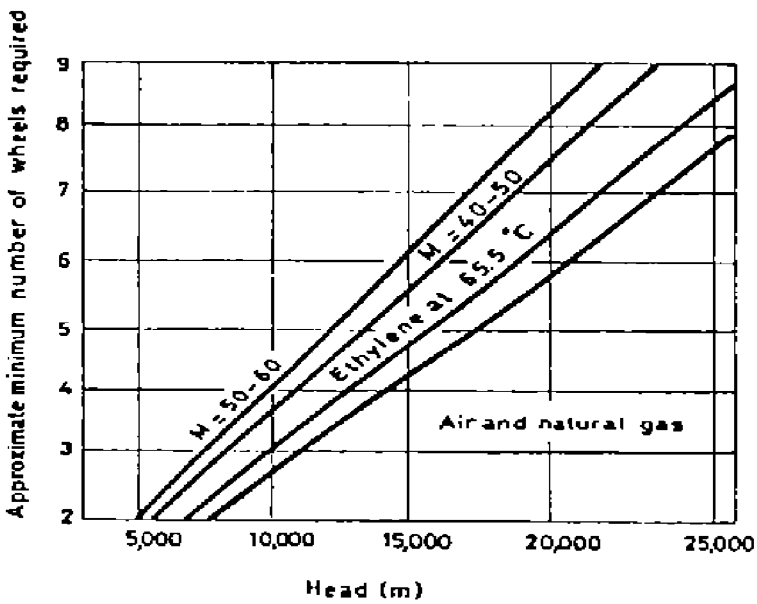
$$H_{is} = \Delta h_{is} \times 101.978 \text{ if } \Delta h \text{ is in kJ/kg} \quad \text{(Eq. 17)}$$

Where H_{is} is in meters of fluid.

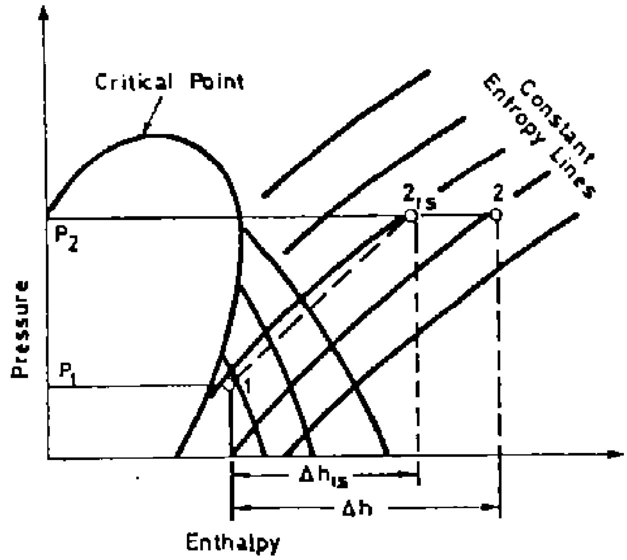
From Eqs. 17, 7, 8 and 9:

$$\Delta h = \frac{\Delta h_{is}}{\eta_{is}} = \frac{\Delta h_p}{\eta_p} \quad \text{(Eq. 18)}$$

Where Δ h is the actual differential enthalpy.



REQUIRED NUMBER OF WHEELS
Fig. 6



P-H DIAGRAM CONSTRUCTION
Fig. 7

To find the discharge enthalpy:

$$h_2 = \frac{\Delta h_{is}}{\eta_{is}} + h_1 \qquad \textbf{(Eq. 19)}$$

The actual discharge temperature can now be obtained from the P-H diagram.

6.2.5 Sonic or acoustic velocity, (the velocity of sound) in any gas may be calculated from:

$$V_S = \left(\frac{k.\,r.\,T.\,Z}{M} \right)^{\pm 1/2} \qquad \textbf{(Eq. 20)}$$

Where:

- V_s is in meters per second;
- k is the ratio of specific heats;
- T is suction absolute temperature of gas, in (kelvin);
- R is 8314.3 N.m/kmol. K.

General design practice avoids using gas velocities near or above the sonic velocity.

6.2.6 The ratio of gas velocity at any point to the sonic velocity of the gas is known as "mach number, (M')".

$$M' = u'/V_s \qquad \textbf{(Eq. 21)}$$

Where u' is gas velocity at any point.

6.2.7 Specific speed

At a given point, the "specific speed", correlates the important performance factors of adiabatic head, capacity and r/min for geometrically similar wheels. The specific speed of all geometrically similar wheels is the same and does not change when the speed of the wheel is changed.

$$N_S = \frac{0.315 \cdot r/min \sqrt{V_1}}{(H_a)^{0.75}} \qquad \textbf{(Eq. 22)}$$

Where:

N_s	is specific speed, in r/min (rpm);
V_1	is flow rate, in m³/h at suction condition;
H_a	is total adiabatic or polytropic head, in meters;
r/min	is actual speed of wheel.

Note:

Centrifugal compressors usually have specific speeds of 1500-3000 at the high efficiency point.

6.2.8 Flow limits

Two conditions associated with centrifugal compressors are surge (pumping) and stone-wall (choked flow).

6.2.8.1 Surge point

At some point on the compressors operating curve there exists a condition of minimum flow/maximum head where the developed head is insufficient to overcome the system resistance. This is the "surge point". When the compressor reaches this point, the gas in the discharge piping back-flows into the compressors. Without discharge flow, discharge pressure drops until it is within the compressor's capability, only to repeat the cycle. The repeated pressure oscillations at the surge point should be avoided since it can be detrimental to the compressor. Surging can cause the compressor to overheat to the point the maximum allowable temperature of the unit is exceeded. Also, surging can cause damage to the thrust bearing due to the rotor shifting back and forth from the active to the inactive side.

6.2.8.2 Stone-Wall or choked flow

Stone-wall-or choked flow occurs when sonic velocity is reached at any point in the compressor. When this point is reached for a given gas, the flow through the compressor can not be increased further without internal modifications.

6.2.9 Interstage cooling

Multistage compressors rely on intercooling whenever the inlet temperature of the gas and the required compression ratio are such that the discharge temperature of the gas exceeds about 150°C. Performance calculations indicate that the head and power are directly proportional to the absolute gas temperature at each impeller.

The gas may be cooled within the casing or in external heat exchangers. In the case of diaphragm water cooling system, AP1 Standard No. 617, Section 2.1.4 should be followed for design parameters.

6.2.10 Control systems

Centrifugal compressor controls can vary from the very basic manual recycle control to the elaborate ratio controllers. The driver characteristics, process response and compressor operating range must be determined before the right controls can be selected.

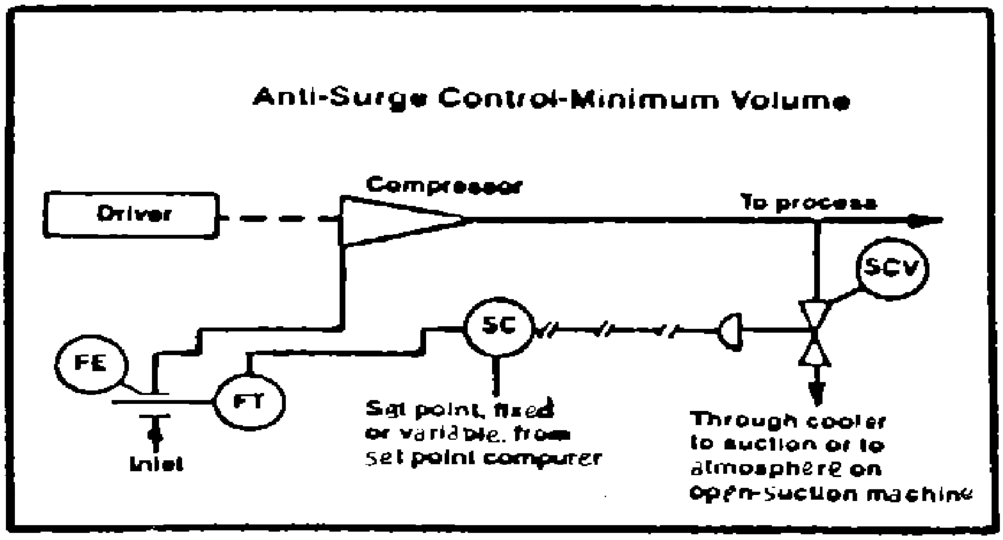
The most efficient way to match the compressor characteristic to the required output is to change the speed in accordance with the Fan (Affinity) laws. In cases where constant speed drivers are used, the inlet gas density can be reduced by throttling or by adjusting the compressor guide vanes. Different features of the centrifugal compressor control systems are described in GPSA, "Engineering Data Book", Volume 2, 1987.

6.2.10.1 Anti-Surge control

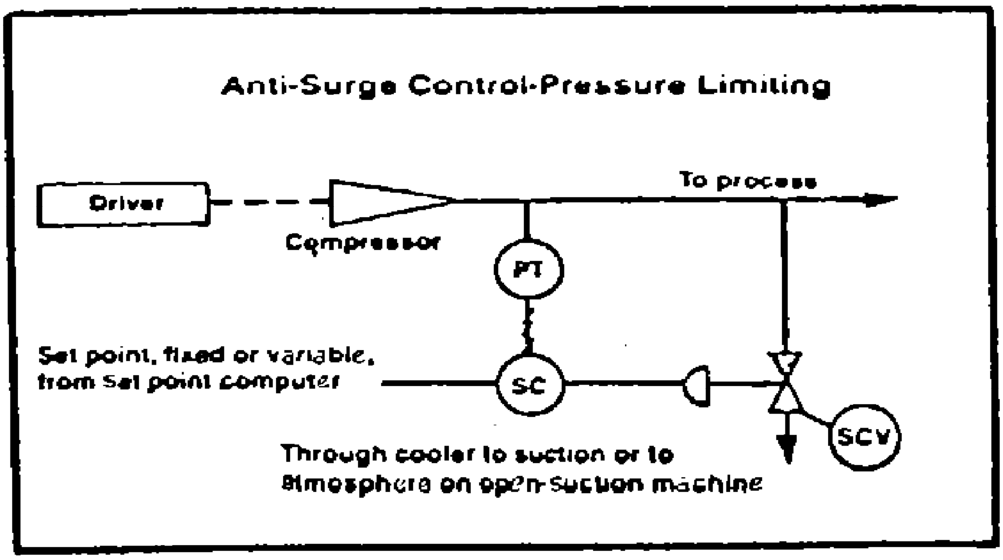
It is essential that all centrifugal compressor control systems be designed to avoid possible operation in surge which usually occurs below 50% to 70% of the rated flow.

The surge limit line can be reached by either reducing flow or decreasing suction pressures. An anti-surge system senses conditions approaching surge, and maintains the unit pressure ratio below the surge limit by recycling some flow to the compressor suction. A volume-controlled anti-surge system is shown in Fig. 8. As the flow decreases to less than the minimum volume set point, a signal will cause the surge control valve to open, to keep a minimum volume flowing through the compressor.

A pressure-limiting anti-surge control system is shown in Fig. 9. A process pressure increase over the pressure set point will cause the blow-off valve to open. The valve opens as required to keep the pressure limited to a minimum of gas flowing through the compressor.



ANTI-SURGE CONTROL-MINIMUM VOLUME
Fig. 8



ANTI-SURGE CONTROL-PRESSURE LIMITING
Fig. 9

7. AXIAL COMPRESSORS

7.1 General

7.1.1 Axial compressor is usually a single inlet, uncooled machine consisting essentially of blades mounted on a rotor turning between rows of stationary blades mounted on the horizontally split casing.

7.1.2 All requirements and recommendations specified in this Section are amendments or additions to those of Section 6 of this Standard.

7.1.3 Performance guarantee

a) Compressors shall be guaranteed for head, capacity and satisfactory performance at all specified operating points and further shall be guaranteed for power at the normal operating point.

b) For variable-speed compressors, the head and capacity shall be guaranteed with the understanding that the power may vary ±4%.

c) For constant-speed compressors, the specified capacity shall be guaranteed with the understanding that the head shall be specified for 100.0 and 105.0 percent; the power consumption shall not exceed stated power by more than 4%. These tolerances are not additive.

d) For compressors handling side loads or for two or more compressors driven by a single drive, the required performance guarantee for each compressor "section" shall be agreed upon by the Company and the Vendor.

7.2 Design Criteria

7.2.1 Performance

7.2.1.1 The minimum head rise to surge of an axial machine should be specified. The normal operating point shall be at least 10% removed in flow from surge point.

7.2.2 Gas velocities

General guideline for good design practice indicates an axial velocity for air of 91 to 137 meters per second. For other gases, the axial velocity range is in direct proportion to the speed of sound of the gas compared to air. The internal shape of the machine is usually arranged to give constant gas velocity as the gas travels through.

7.2.3 Volume

The size is determined by the inlet volume. The lower volume limit is approximately 8500 m^3/h but the upper limit practically does not exist, units have been built to handle well above 1,700,000 m^3/h.

8. RECIPROCATING COMPRESSORS

8.1 General

8.1.1 The reciprocating compressor is a positive displacement unit with the pressure on the fluid developed within a cylindrical chamber by the action of a moving piston. It may consist of one or more cylinders each with a piston or plunger that moves back and forth, displacing a positive volume with each stroke.

8.1.2 Reciprocating compressors shall conform to API 618 for all services except portable air compressors, and standard utility air compressors of 400 kW or less with not more than 900 kPa (9 bar) discharge pressure. This latter group will generally be purchased as packaged units.

8.1.3 Reciprocating compressors normally should be specified for constant-speed operation to avoid excitation of torsional and acoustic resonances. When variable-speed drivers are used, all equipment shall be designed to run safely to the trip speed setting.

8.1.4 When considering the use of a single frame for cylinders on different services particular attention shall be given to the means of independently controlling the different process streams. Care shall also be taken to ensure that the frame, transmission and driver can accept the wide variety of loadings that occur during all operating modes including start-up and shut-down.

8.1.5 Speed ranges

Low speed unit-up to	330 r/min	(rpm).
Medium speed unit	330 to 700 r/min	(rpm).
High speed unit-over	700 r/min	(rpm).

Generally high speed units are preferred for units under 1865 kW. For larger units the choice is between low and medium speed.

8.1.6 Capacity control

8.1.6.1 Capacity control for constant-speed units normally will be obtained by suction valve unloading (depressors or lifters), clearance pockets, a combination of both pockets and unloaders, or bypass. Operation of controls shall be automatic. Unless stated otherwise, five-step unloading shall provide capacities of 100, 75, 50, 25 and 0 percent; three-step unloading shall provide capacities of 100, 50, and 0 percent; and two-step unloading shall provide capacities of 100 and 0 percent.

8.1.6.2 Capacity control on variable-speed units generally is by speed control.

8.1.6.3 Clearance pockets may be either the two-position type (pocket either open or closed) or the variable-capacity type. If not specified, the Vendor shall propose on the data sheet the type recommended for the Purchaser's or Company's operating conditions.

8.1.6.4 When unloading for startup is necessary, unloading arrangement shall be stated on the data sheet or shall be mutually agreed upon between the Purchaser's and or the Company's and the Vendor.

8.2 Design Criteria

8.2.1 This Section covers information necessary for process engineers to determine the approximate power required to compress a certain volume of gas at some intake conditions to a given discharge pressure, and estimate the capacity of an existing reciprocating compressor under specified suction and discharge conditions.

8.2.2 Reciprocating compressors are furnished either single-stage or multi-stage. The number of stages is determined by the overall compression ratio.

8.2.3 On multistage machines, intercoolers may be provided between stages. Such cooling reduces the actual volume of gas going to the high pressure cylinders, reduces the power required for compression, and keeps the temperature within safe operating limits.

8.2.4 Reciprocating compressors should be supplied with clean gas as they cannot satisfactorily handle liquids and solid particles that may be entrained in the gas.

8.2.5 In evaluating the work of compression, the enthalpy change is the best way. If a P-H diagram is available, the work of compression should always be calculated by the enthalpy change of the gas in going from suction to discharge conditions.

8.2.6 The k value of a gas is associated with adiabatic compression or expansion. The change in gas properties at different states is related by:

$$P_1 . V_1^k = P_2 . V_2^k = P_3 . V_3^k \qquad \textbf{(Eq. 23)}$$

For a polytropic compression the actual value of "n" (polytropic exponent) is a function of the gas properties such as specific heats, degree of external cooling during compression and operating features of the cylinder. Usual reciprocating compressor performance is evaluated using adiabatic C_p/C_v.

8.2.7 "Power Rating" or kilowatt rating of a compressor frame is the measure of the ability of the supporting structure and crankshaft to withstand torque and the ability of the bearings, to dissipate frictional heat.

8.2.8 "Rod Loads" are established to limit the static and internal loads on the crankshaft, connecting rod, frame, piston rod, bolting and projected bearing surfaces.

8.2.9 Performance calculation

8.2.9.1 Determination of properties pertaining to compression

Compressibility factor (Z factor), ratio of specific heats (C_p/C_v or k value), and molecular mass are three major physical properties for compression which must be clarified. Mollier diagrams should be used if available.

The k value, may be calculated from the ideal gas equation:

$$k = \frac{C_p}{C_v} = \frac{M.C_p}{MC_p - R} \qquad \textbf{(Eq. 24)}$$

Where:

MC_p is molar heat capacity at constant pressure in (kJ/kmol.K);
R is gas constant, 8.3143 kJ/kmol.K.

Method presented in Tables A.3 and A.4 of Appendix A can be used for calculation of k value of hydrocarbon gases and vapors.

8.2.9.2 Determination of suction conditions

The following conditions at the suction flange should be determined:

a) Temperature

b) Pressure

In case of air taken from atmosphere, corrections should be made for elevation. Air humidity should also be considered.

c) Flow rate

Since the maximum and minimum flow rates are important parameters in compressor selection in some cases, studies must be conducted carefully.

For the purpose of performance calculations, compressor capacity is expressed as the actual volumetric quantity of gas at inlet to each stage of compression on a per hour basis (Im³/h).

1) Inlet volume

From mass (weight) flow W, (kg/h):

$$Q = \frac{R}{M}\left[\frac{W.T_1.Z_1}{P_1.Z_L}\right] \qquad \textbf{(Eq. 25)}$$

Where:

R	is gas constant;
M	is molecular mass, in (kg/kmol);
$\boldsymbol{P_1}$	is absolute pressure, in (kPa);
$\boldsymbol{T_1}$	is absolute temperature, in (K);
$\boldsymbol{Z_1}$	is gas compressibility at inlet condition;
$\boldsymbol{Z_L}$	is gas compressibility at standard condition.

From (Sm³/h):

$$Q = Sm^3/h\left(\frac{101.34}{288.15}\right)\left[\frac{T_1.Z_1}{P_1.Z_L}\right] \qquad \textbf{(Eq. 26)}$$

From (Nm³/h):

$$Q = Nm^3/h\left(\frac{101.34}{273.15}\right)\left[\frac{T_1.Z_1}{P_1.Z_L}\right] \qquad \textbf{(Eq. 27)}$$

2) Piston displacement

Piston displacement is equal to the net piston area multiplied by the length of the piston sweep in a given period of time:

For a single-acting piston compressing on the outer end only:

$$PD = \frac{S(r/min)(D^2)\pi \times 60}{4 \times 10^9} = 47.124 \times 10^{-9} \times S(r/min)\, D^2 \qquad \textbf{(Eq. 28)}$$

For a single acting piston compressing on the crank end only:

$$PD = \frac{S(r/min)(D^2 - d^2)\pi \times 60}{4 \times 10^9} = 47.124 \times 10^{-9} \times S(r/min)\,(D^2 - d^2) \qquad \textbf{(Eq. 29)}$$

and for a double acting piston (other than rod tail type):

$$PD = \frac{S(r/min)(2D^2 - d^2)\pi \times 60}{4 \times 10^9} = 47.124 \times 10^{-9} \times S(r/min)\,(2D^2 - d^2) \qquad \textbf{(Eq. 30)}$$

Where:

PD	is piston displacement, in (m^3/h);
S	is stroke length, in (mm);
D	is cylinder inside diameter, in (mm);
d	is piston rod diameter, in (mm);
r/min	is compressor speed, in (rotations per minute).

3) Cylinder clearance

In a reciprocating compressor the piston does not travel completely to the end of the cylinder at the end of the discharge stroke. Some clearance volume is necessary and it includes the space between the end of the piston and the cylinder head when the piston is at the end of its stroke. It also includes the volume in the valve ports, the volume in the suction valve guards and the volume around the discharge valve seats.

Clearance volume is usually expressed as percent of piston displacement and referred to as percent clearance, or cylinder clearance, C.

$$C = \frac{clearance\ \ volume}{piston\ \ displacement} \times 100 \qquad \textbf{(Eq. 31)}$$

For double acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

4) Volumetric efficiency

The term "volumetric efficiency" refers to the actual pumping capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity of a cylinder can be represented by:

$$VE = 100 \quad r \quad C \quad \frac{Z_s}{Z_d}(r)^{1=k} \quad 1 \qquad \textbf{(Eq. 32)}$$

Where:

VE	is volumetric efficiency;
r	is compression ratio, P_2/P_1;
C	is percent clearance;
Z_s & Z_d	are gas compressibility at suction and discharge.

Note that volumetric efficiencies as determined by the above equation are theoretical in that they do not account for suction and discharge losses. One method for accounting for losses is to reduce the volumetric efficiency in the following manner:

4%	for valve losses,
5%	for gas slippage (for non-lubricated compressors),
4%	for heavy gases (propane and similar).

5) Actual capacity (actual flow rate)

This is the volume of gas measured at intake to the first stage of a single or multistage compressor, at stated intake temperature and pressure.

$$V_a = PD \times VE/100\ m^3/h \qquad \textbf{(Eq. 33)}$$

6) Equivalent capacity

The net capacity of a compressor, in m³/h at 101.325 kPa and suction temperature, may be calculated by equation:

$$Q_{eq} = \frac{PD.\ VE.\ P_s}{100 \times 101.325 \times Z_{ave}} \qquad \textbf{(Eq. 34)}$$

Where:

Q_{eq}	is equivalent capacity at 101.325 kPa (abs), in (m³/h);
PD	is piston displacement, in (m³/h);
VE	is volumetric efficiency (see 4 above);
PS	is suction pressure, in [kPa (abs)];
Z_{ave}	is average compressibility.

If compressibility is not used as a divisor in calculating m³/h, then the statement "not corrected for compressibility" should be added. The figure 101.325 kPa (abs) is used in Eq. 34, because it is the common base pressure for the compression power charts (see 8.2.9.4.2).

8.2.9.3 Determination of discharge conditions

8.2.9.3.1 Discharge temperature

The temperature of the gas discharged from the cylinder can be estimated from the equation:

$$T_d = T_s\,(r^{(k-1)/k}) \qquad \textbf{(Eq. 35)}$$

Although the result of this equation is theoretical value and heat from friction, irreversibility effects etc., are neglected, use of it has been recommended by many sources. Polytropic exponent "n" may be used instead of "k" in the above equation and will give better results.

8.2.9.3.2 Limitations on discharge temperature

Limitations on the discharge temperature are as shown in API Std. 618, 2.3.

8.2.9.3.3 Estimation of number of compression stages

a) Number of compression stages is a function of compression ratio per stage (R_{cs}), these parameters are related by the equations:

! 1 = -

$$R_{cs} = \frac{P_f}{P_1} \qquad \textbf{(Eq. 36)}$$

and

!

$$R_{co} = \frac{P_f}{P_1} \qquad \textbf{(Eq. 37)}$$

Where:

R_{co}	is overall compression ratio;
P_1 & P_f	are absolute initial and final pressures respectively;
γ *(gamma)*	is number of compression stages.

b) The maximum ratio of compression permissible in one stage is determined considering and limited by the discharge temperature (see 8.2.9.3.2), or by rod loading (see 8.2.8), particularly in the first stage. Economic considerations are also involved, because a high ratio of compression will mean a low volumetric efficiency.

c) In multi-stage operation, equal ratio of compression per stage shall be used, unless otherwise stated by process design. This will result minimum power requirement.

8.2.9.3.4 Interstage cooling

a) Interstage cooling operations affect the cumulative power required to do the work of compression.

Note that if condensate forms in interstage coolers and is to be removed, the flow rate and properties of the fluid will vary.

b) Cooling water system, where necessary, shall be designed as per Section 2.1.3 of API Standard 618.

c) Compression ratio across stage with intercooling.

Pressure drop in the interstage cooler shall be regarded to be 20 to 35 kPa. Ratio of compression per stage (R) may be calculated by:

$$P_f = P_1 . R^{\gamma} - (\Delta P_1) . R^{\gamma-1} - (\Delta P_2) . R^{\gamma-2} - (\Delta P_3) . R^{\gamma-3} \ldots\ldots \qquad \textbf{(Eq. 38)}$$

Where:

ΔP_1, ΔP_2, etc., are interstage cooler pressure drops of different stages.

Other variables are defined earlier.

Number of terms on right side of this equation should be equal to the number of stages. This equation shall be solved by trial and error method for the case of multi-stage compression.

8.2.9.4 Determination of power required

Brake horse power is the actual power input at the crankshaft of the compressor drive. It does not include the losses in the driver itself, but is rather the actual power which the driver must deliver to the compressor crankshaft. There are three methods for determination of power required for compression. These methods are described in the following section.

8.2.9.4.1 Calculation method

a) Single stage compression

Use basic equation to determine brake horse power:

$$Bhp = \frac{P_1 . V_1}{3600\,(k-1)/k} \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] (L_o)\,(F_L)\,(Z_1) \qquad \textbf{(Eq. 39)}$$

Where:

Bhp	is brake horse power,in kilowatt (kW);
P_1	is suction pressure, in (kPa);
P_2	is discharge pressure, in (kPa);
V_1	is suction volume, in (m³/h), at suction conditions;

L_o	is loss factor, comprised of losses due to pressure drop through friction of piston rings, rod packing, valves, and manifold (see Fig. 10);
F_L	is frame loss for motor driven compressors only, values range from 1.0 to 1.05;
Z_1	is compressibility factor, based on inlet conditions.

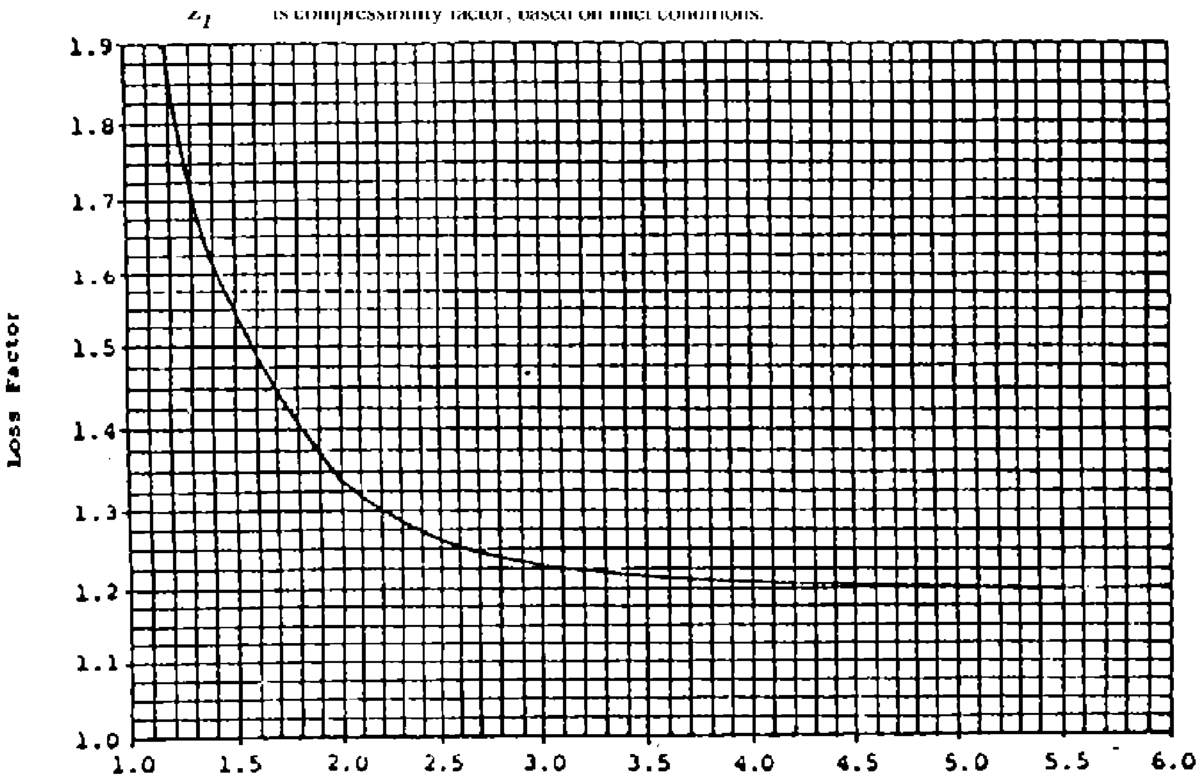


Fig. 10

b) Multi-Stage compression

Multi-Stage power is the sum of the power requirements of the individual cylinders on the compressor unit.

$$\text{Actual Bhp} = \frac{F_L \cdot k/(k-1)}{3600}\left\{P_1 \cdot V_1\left[\left(\frac{Pn_1}{P_1}\right)^{(k-1)/k}-1\right]L_{o1}+P_2 \cdot V_2\left[\left(\frac{Pn_2}{P_2}\right)^{(k-1)/k}-1\right]Lo_2+\ldots\ldots+P_f \cdot V_f\left[\left(\frac{P_f}{P_f}\right)^{(k-1)/k}-1\right]Lo_f\right\}$$

(Eq. 40)

Where:

P_{ni}	is discharge pressure of stage i, in (kPa);
P_i	is inlet pressure to stage i, in (kPa);
V_i	is inlet volume to stage i, in (Am³/h);
$L_{o1}, L_{o2}, ...$	are loss factors designated by cylinder stages. Correction for compressibility
L_{of}	may be incorporated as described for the single stage cylinder.

8.2.9.4.2 Power determination by chart

Detailed compressor power calculations can be made through the use Figs. 4.7 through 4.10 of GPSA, "Engineering Data Book", Chapter 4, SI Edition.

8.2.9.4.3 Power calculation by Mollier diagram

Power calculations can be worked out most easily and that accurately if the P-H (Mollier) diagram is available.

The procedure is as follows:

a) The enthalpy at the inlet pressure and temperature shall be calculated. The enthalpy at the outlet pressure shall be found from the diagram following the line of constant entropy:

$$\text{amount of work} = h_2 - h_1 \qquad \textbf{(Eq. 41)}$$

Where h_1 and h_2 are enthalpies at inlet and outlet conditions respectively in (kJ/kg).

b) Brake horse power in kW (Bhp) is calculated from the equation:

$$Bhp = 2.78 \times 10^{-4} \times W\,(h_2 - h_1)\,(L_o)(F_L) \qquad \textbf{(Eq. 42)}$$

Where:

Bhp is in kilowatts;
W is mass flow rate of gas, in (kg/h).

8.2.10 Reciprocating compressor control devices

Output of compressors must be controlled (regulated) to match system demand. Compressor capacity, speed, or pressure, may be varied in accordance with the requirements. The nature of the control device will depend on the regulating variable; whether pressure, flow, temperature, or some other variable; and on type of compressor driver.

8.2.10.1 Capacity control

The most common requirement is regulation of capacity. Many capacity controls, or unloading devices, as they are usually termed, are actuated by the pressure on the discharge side of the compressor.

A common method of controlling the capacity of a compressor is vary the speed. This method is applicable to steam-driven compressors and to units driven by internal-combustion engines.

On reciprocating compressors up to about 75 kW, two types of control are usually available. These are automatic-start-and-stop control and constant-speed control.

8.2.10.1.1 Step control

Motor-driven reciprocating compressors above 75 kW in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load.

8.2.10.1.2 Manual contorl

Although control devices are often automatically operated, manual operation is satisfactory for many services.

8.2.10.2 Control by spill-back

In some cases, spill-back from discharge to suction will be required for flow or pressure control. This flow rate depends on the design of the unloader.

Minimum flow rates are as shown below.

Unloader Design					Spill Back Flow Rate	
5-stage	100	75	50	25	0%	25%
4-stage	100	75	50	—	0%	25%
3-stage	100	—	50	—	0%	50%
2-stage	100	—	—	—	0%	100%

8.2.11 Gas pulsation control

Pulsation is inherent in reciprocating compressors because suction and discharge valves are open during only part of the stroke.

Pulsation must be damped (controlled) in order to:

a) Provide smooth flow of gas to and from the compressor;
b) prevent overloading or underloading of the compressors, and;
c) reduce overall vibration.

8.2.11.1 Pulsation dampeners (snubbers)

A pulsation dampener is an internally-baffled device. The design of the pulsation dampening equipment is based on acoustical analog evaluation which takes into account the specified operating speed range, conditions of unloading, and variations in gas composition. Detailed discussion of recommended design approaches for pulsation suppression devices is presented in API Std. 618.

9. ROTARY COMPRESSORS

9.1 General

9.1.1 Rotary compressors are positive displacement gas (or vapor) compressing machines. Rotary compressors cover lobe-type, screw-type, vane-type and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other such as lobes or screws, or that displace a fixed volume with each rotation.

9.1.2 Rotary compressors shall conform to API Std. No. 619 for all services handling air or gas, except those machines which this Standard does not cover.

9.1.3 Performance

9.1.3.1 Compressor shall be guaranteed for satisfactory performance at all specified operating conditions.

9.1.3.2 Compressor performance shall be guaranteed at the normal operating point unless otherwise specified. At this point no negative tolerance is permitted on capacity and power may not exceed 104% of the quoted power.

9.1.4 Cooling water

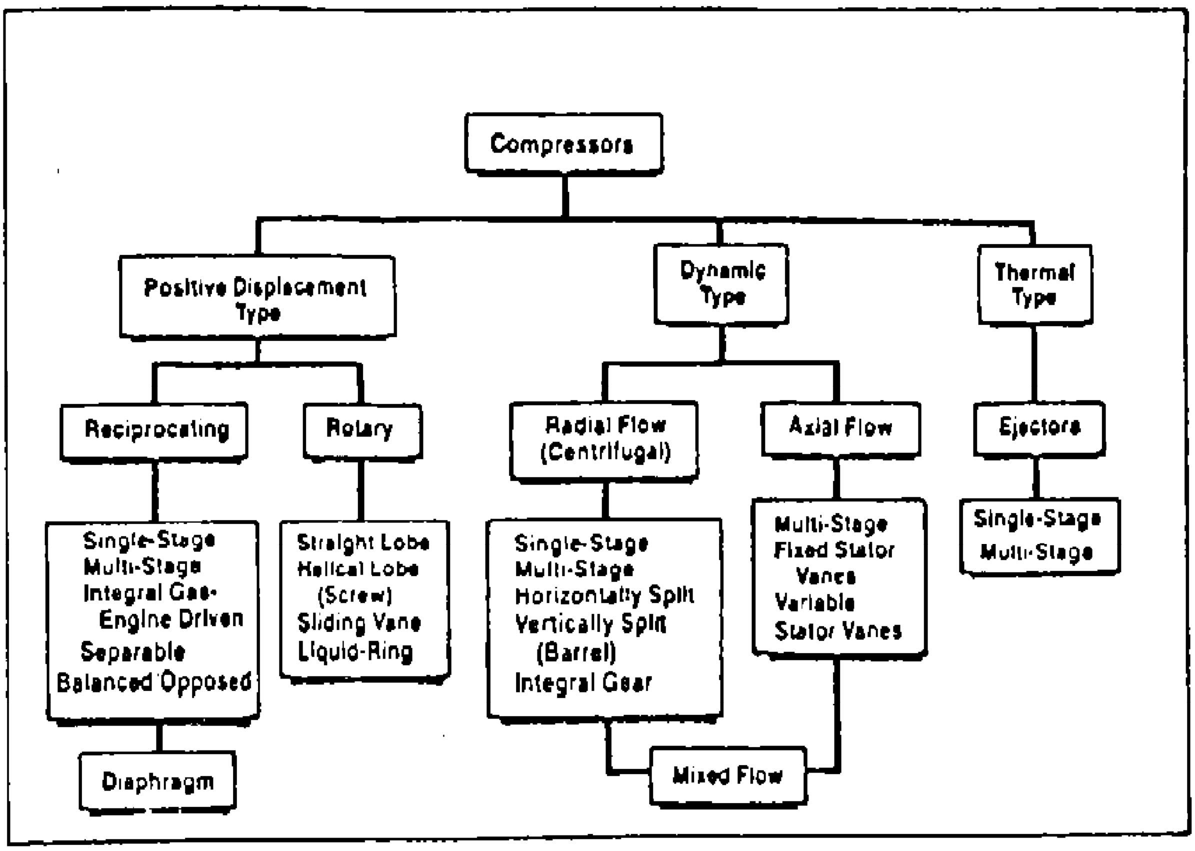
The compressor cooling water jacket shall be designed for the specified cooling water pressure but not less than 618 kilopascals (absolute). The maximum pressure drop shall be 70 kPa and provisions shall be included for complete draining and venting of the jackets (Modification to API Std. 619 2nd. Ed., 2.1.3). The cooling water design conditions shall be in accordance with API Std. 619, 2nd. Ed., 2.1.3.

APPENDICES

APPENDIX A

TABLE A.1 - GENERAL COMPRESSOR LIMITS

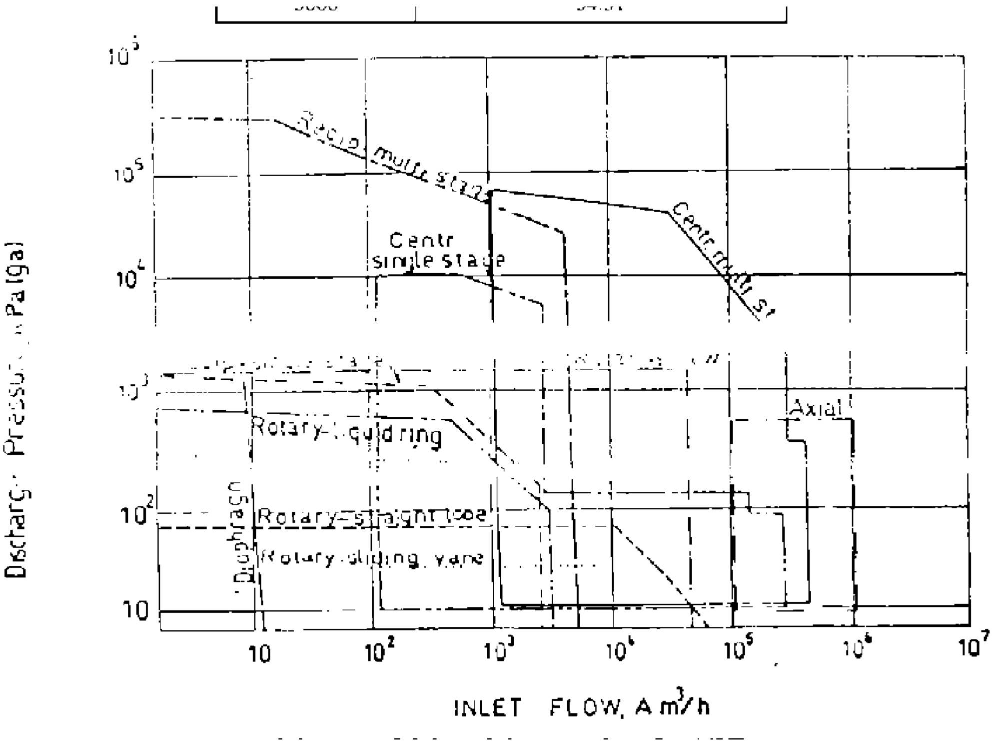
COMPRESSOR TYPE	APPROX. max. COMMERCIALLY USED DISCH. PRESS. kPa	APPROX. max. COMPRESSION RATIO PER STAGE	APPROX. max. COMPRESSION RATIO PER CASE OR MACHINE
Reciprocating	240,000-345,000	10	As required
Centrifugal	20,600-34.500	3-4.5	8-10
Rotary displacement	690-896	4	4
Axial flow	550-896	1.2-1.5	5-6.5



TYPES OF COMPRESSORS
Fig. A.1

TABLE A.2 - ATMOSPHERIC PRESSURE vs ELEVATION

ALTITUDE (meters)	AVERAGE ATMOSPHERIC PRESSURE [kPa (abs)]
0	101.34
100	99.97
200	98.84
300	97.93
400	96.60
500	95.44
600	94.54
700	93.49
800	92.04
1000	90.03
1200	87.77
1400	85.51
1600	83.42
2000	79.41
2500	74.58
3000	70.06
3500	65.54
4000	61.40
4500	57.71
5000	54.31



COMPRESSOR COVERAGE CHART

Fig. A.2

TABLE A.3 - CALCULATION OF k

EXAMPLE GAS MIXTURE		**DETERMINATION OF MIXTURE MOL. MASS**		**DETERMINATION OF MC_p. MOLAR HEAT CAPACITY**		**DETERMINATION OF PSEUDO CRITICAL PRESSURE pPc AND TEMPERATURE pTc**			
Component Name	Mole Faction y	Individual Component Mol. Mass M	y × M	Individual Component MCp at 70°C	y × MC_p at 70°C	Component Critical Pressure Pc, kPa (abs)	y × Pc	Component Critical Temperature Tc k	y × Tc
Methane	0.9216	16.04	14.782	37.471	34.533	4640.4	4276.5	190.6	175.6
Ethane	0.0488	30.07	1.467	58.395	2.850	4944.4	241.3	305.6	14.9
Propane	0.0185	44.10	0.816	82.858	1.533	4256.4	78.7	370.0	6.8
i-Butane	0.0039	58.12	0.227	109.397	0.427	3749.0	14.6	406.9	1.6
n-Butane	0.0055	58.12	0.320	109.497	0.602	3658.6	20.1	425.2	2.3
i-Pentane	0.0017	72.15	0.123	134.379	0.228	3333.2	5.7	460.9	0.8
Total=	1.0000	M =	17.735	MC_p =	40.173	pP_c =	4036.9	pTc =	202.1
$MC_v = MC_p - 8.3143 = 31.859$						$k = MC_p/MC_v = 40.173/31.859 = 1.261$			

Note:

For values of MC_p other than at 70°C refer to Table A.4.

TABLE A.4 - MOLAR HEAT CAPACITY MC_p (IDEAL-GAS STATE) kJ/(kmol.K)

Gas	Chemical formula	Mol. mass	-20°C	10°C	20°C	40°C	70°C	100°C	120°C	150°C
Methane.....	CH_4	16.043	34.387	35.248	35.573	36.288	37.741	39.178	40.291	41.964
Ethyne(Acetylene)	C_2H_2	26.038	40.340	42.781	43.580	44.991	47.009	48.750	49.760	51.207
Ethene(Ethylene)	C_2H_4	28.054	38.824	41.944	42.999	45.108	48.332	51.292	53.305	56.243
Ethane..........	C_2H_6	30.070	47.643	50.944	52.099	54.472	58.395	62.095	64.699	68.542
Propene(Propylene)...	C_3H_6	42.081	56.700	61.492	63.083	66.285	71.204	75.775	78.863	83.429
Propane..........	C_3H_8	44.097	65.099	70.660	72.577	76.495	82.858	88.822	92.919	98.842
1-Butene(Butylene)..	C_4H_8	56.108	74.635	82.004	84.407	89.191	96.462	103.079	107.466	113.931
cis_2_Butene.....	C_4H_8	56.108	68.734	75.515	77.776	82.297	89.321	95.807	100.261	106.862
trans_2_Butene....	C_4H_8	56.108	78.399	84.683	86.742	90.924	97.425	103.389	107.481	113.539
iso_Butane.....	C_4H_{10}	58.123	84.808	92.720	95.432	100.857	109.397	117.320	122.739	130.514
n-Butane.........	C_4H_{10}	58.123	86.540	93.683	96.245	101.388	109.497	117.072	122.077	129.627
iso-Pentane......	C_5H_{12}	72.150	103.852	113.734	117.091	123.910	134.379	144.000	150.468	160.007
n-Petane.........	C_5H_{12}	72.150	106.869	116.675	118.707	125.073	135.149	144.422	150.688	159.955
Benzene...........	C_6H_6	78.114	68.023	77.084	80.020	88.349	95.658	104.409	109.956	118.139
n-Hexane..........	C_6H_{14}	86.177	125.418	137.217	141.127	148.925	160.885	171.855	179.342	190.282
n-Heptane.........	C_7H_{16}	100.204	145.325	159.068	163.631	172.672	186.645	199.444	208.036	220.694
Ammonia...........	NH_3	17.0305	35.665	35.665	35.665	35.665	35.673	35.717	35.707	35.707
Air...............		28.9625	29.047	29.093	29.101	29.139	29.193	29.280	29.341	29.431
Water	H_2O	18.0153	33.398	33.488	35.547	33.622	33.823	34.041	34.190	34.461
Oxygen............	O_2	31.9988	29.170	29.280	29.327	29.440	29.637	29.855	30.004	30.275
Nitrogen..........	N_2	28.0134	29.093	29.093	29.101	29.139	29.143	29.186	29.215	29.305
Hydrogen..........	H_2	2.0159	28.354	28.716	28.791	28.942	29.059	29.103	29.175	29.220
Hydrogen sulfide...	H_2S	34.08	33.458	33.865	34.007	34.271	34.694	35.085	35.398	35.805
Carbon monoxide.....	CO	28.010	29.089	29.135	29.135	29.135	29.193	29.280	29.341	29.431
Carbon dioxide......	CO_2	44.010	34.972	36.418	36.870	37.774	39.114	40.289	41.074	42.109

APPENDIX B

B.1 Process Specification Sheet for Rotadynamic Compressors:

The following information should be prepared by process engineer to be included in the specification sheet.

a) Process Requirements

- Flow at 1.013 bar (abs) and 0°C,	m^3/h,	(Normal & Design).
- Flow at Suction Conditions,	m^3/h,	(Normal & Design).
- Suction Temperature,	°C ,	(Normal & Design).
- Suction Pressure	kPa(abs),	(Normal & Design).
- Discharge Pressure	kPa(abs),	(Normal & Design).
- Discharge Temperature Limitation (if any)		(Normal & Design).
- Compression Ratio		(Normal & Design).
- Approx. C_p/C_v (at Suction)		(Normal & Design).
- Compressibility Factor (at Suction)		(Normal & Design).
- Mass Flow	kg/h,	(Normal & Design).
- Estimated Polytropic Head	m,	(Normal & Design).
- Estimated Bhp Required	kW,	(Normal & Design).
- Estimated Gear Loss	kW,	(Normal & Design).
- Recommended Driver kW	kW,	(Normal & Design).
- Compressor Speed	r/min,	(Normal & Design).

b) Service

- Approximate Gas Composition (vol% or mol%).
- Average Molecular Mass.
- Relative Density (Specific Gravity).
- Corrosiveness & Remarks.

c) Site Informations

- Elevation of Plant Site from Sea Level,	m.
- Minimum Winter Temperature,	°C.
- Maximum Summer Temperature,	°C.
- Normal Barometer,	kPa.

d) Available Utilities

- Cooling water:	max. Inlet Temperature,	°C
	max. Outlet Temperature,	°C.
	Pressure,	kPa.
	Fouling Factor.	
- Instrument Air Pressure,		kPa.
- Electric Power for Instruments, volts, Phase, Hz.		

e) Compressor Location (Outdoor, Indoor).

f) Instrument Graduation System.

g) Remarks on Control System.

(to be continued)

APPENDIX B (continued)

B.2 Process Specification Sheet for Positive Displacement Compressors

The following information should be prepared by process engineer to be included in the compressor specification sheet.

a) Process Requirements

- Flow at 1.013 bar (abs) and 0°C,	m^3/h, Normal.
- Flow at 1.013 bar (abs) and 0°C,	m^3/h, Rated.
- Flow at Suction Conditions,	m^3/h, Rated.
- Suction Temperature ,	°C.
- Suction Pressure,	kPa (abs).
- Discharge Pressure,	kPa (abs) Normal
- Discharge Pressure,	kPa (abs) Rated.
- Differential Pressure,	kPa, rated.
- Compression Ratio.	
- Approx. C_p/C_v (at Suction).	
- Compressibility Factor (at Suction).	
- Mass Flow,	kg/h, Normal.
- Mass Flow,	kg/h, Rated.
- Estimated Power Required,	kW, Normal.
- Estimated Power Required,	kW, Rated.
- Estimated Gear loss,	kW.
- Recommended Driver Bhp.,	kW.
- Compressor Speed Limitation (if any),	r/min.
- Piston Speed Limitation, (if any),	m/s.

b) Service

- Approximate Gas Composition	(vol% or mol%).
- Average Molecular Mass.	
- Relative Density (Specific Gravity).	
- Corrosiveness and Relevant Remarks.	

c) Site Informations

- Elevation of Plant Site From See Level,	m.
- Minimum Winter Temperature,	°C.
- Maximum Summer Temperature,	°C.
- Normal Barometer,	kPa.

d) Available Utilities

- Cooling water:	max. Inlet Temperature,	°C.
	max. Outlet Temperature,	°C.
	Pressure,	kPa.
	Fouling Factor.	

(to be continued)

APPENDIX B (continued)

- Instrument Air Pressure,		kPa.
- Fuel Gas for Engine :	mol. mass.	
	Rel. Density.	
- Calorific value,		kJ/m³, Gross
- Pressure,		kPa.
- Electric Power for Instruments,		volt, Phase, Hz.

e) Compressor Location, Outdoor, Indoor.

f) Instrument Graduation System.

g) Remarks on Control System.