KLM Technology Group Project Engineering Standard	KLM Technology Group www.klmtechgroup.com	Page : 1 of 47 Rev: 01 April 2011
KLM Technology Group #03-12 Block Aronia, Jalan Sri Perkasa 2 Taman Tampoi Utama 81200 Johor Bahru Malaysia	**PROCESS DESIGN OF COMPRESSORS** **(PROJECT STANDARDS AND SPECIFICATIONS)**	

TABLE OF CONTENT

SCOPE

This Project Standards and Specifications 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.

REFERENCES

Throughout this Standard the following dated and undated standards/codes are referred to. These referenced documents shall, to the extent specified herein, form a part of this standard. For dated references, the edition cited applies. The applicability of changes in dated references that occur after the cited date shall be mutually agreed upon by the Company and the Vendor. For undated references, the latest edition of the referenced documents (including any supplements and amendments) applies.

1. API (American Petroleum Institute)

API Std. 614	"Lubrication, Shaft-Sealing, and Control-Oil Systems and Auxiliaries" 5th. Ed
API Std. 617,	"Axial & Centrifugal Compressors & Expander Compressors for Petroleum, Chemical and Gas Industry Services" 7th Ed., July 2002
API Std. 618,	"Reciprocating Compressors for Petroleum, Chemical and Gas Industry Services" 4th Ed. 1995
API Std. 619,	"Rotary-Type Positive- Displacement Compressors for Petroleum, Petrochemical, and Natural Gas Industries" 3rd Ed. 1997
API Publication	"Conversion of Operational and Process Measurement Units to the Metric (SI) Units",2001

DEFINITIONS AND TERMINOLOGY

Terms used in this Standard are in accordance with the relevant sections of definition of terms specified in API Standard 617, API Standard 618 and API Standard 619, unless otherwise stated in this Section.

Inlet Cubic Meters per Hour - Refers to flow rate determined at the conditions of pressure, temperature, compressibility and gas composition, including moisture, at the compressor inlet flange

Actual Cubic Meters per Hour - 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.

Standard Cubic Meter per Hour - 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.

Normal Cubic Meters per Hour - 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.

Specific Volume - Is the volume per unit mass or volume per mole of material.

SYMBOLS AND ABBREVIATIONS

SYMBOL/ABBREVIATION	DESCRIPTION
C_p	Specific heat at constant pressure
C_v	Specific heat at constant volume
D	Cylinder inside diameter
d	Piston rod diameter
Ghp	Gas horsepower, actual compression horsepower excluding mechanical losses
H	Head
h	Enthalpy
k	Isentropic exponent, C_p/C_v
MC_p	Molar specific heat at constant pressure
MC_v	Molar specific heat at constant volume
MW	Molecular weight
N	Speed, rpm
Nm	Molar flow, moles/min
n	Polytropic exponent or number of moles
P	Pressure
PD	Piston displacement
Q	Inlet capacity
Qg	Standard gas flow rate
r	Compression ratio, P_2/P_1
s	Entropy
Stroke	Length of piston movement
T	Absolute temperature
t	Temperature

VE	Volumetric efficiency
w	Weight flow
Z	Compressibility factor
Z_{avg}	Average compressibility factor
η	Efficiency, expressed as a decimal
Subscripts	
avg	Average
d	Discharge
g	Gas
is	Isentropic process
L	Standard conditions used for calculation or contract
p	Polytropic process
S	Standard conditions, usually 14.7 psia,60°F
s	Suction
t	Total or overall
1	Inlet conditions
2	Outlet conditions

UNITS

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

GENERAL

- Compressors are generally divided into three major types, dynamic, positive displacement and thermal as shown in Fig. A.1 of Appendix A.
- For typical figures of three type of compressors see Appendix C
- 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.
- Adequate knock out facilities including demister pads where necessary shall be provided to prevent damage by liquid carry over into the compressor.
- Compressors handling SO2, 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.

- 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 antisurge control system is also necessary.

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.

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.

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 the possibility of using centrifugal compressor to be investigated.
   Centrifugal compressors shall be designed in accordance with API Std.

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 twoyear period, due to fairly high maintenance requirements, a fullcapacity 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.

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 lube compressors for process duties, requires the explicit approval of the Company.
   Rotary-type positive displacement compressors shall be in accordance with API Std. 619.

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 101.325
kPa at sea level and declines with increasing altitude as shown in Table A.2 of Appendix A.

Specification Sheets

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

CENTRIFUGAL COMPRESSORS

General

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.
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.

3. Performance
   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 rated point.
   b. 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.

      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.
   c. 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.
   d. For variable speed compressors, the head and capacity shall be guaranteed with the understanding that the power may vary ±4%.
   e. 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.
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.
5. Compressor mach numbers shall not exceed 0.90 when measured at any point.

Design Criteria

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.
2. An approximate idea of the flow range that a centrifugal compressor will handle is shown in Table 1. A multistage 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

Speed to develop 3048 m head/wheel	Average isentropic efficiency	Average polytropic efficiency	Nominal flow range (inlet m^3/h)
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

3. Effect of speed

   a. 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.

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

   c. 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.

   d. If speed is constant then Characteristic operating curve will be also constant. The following factors will increase suction pressure resulting in change of discharge pressure:

      - Molecular weight of gas increases
      - Suction pressure increases
      - Inlet temperature decreases
      - Compressibility factor decreases
      - Ratio of specific heats, k decreases

4. Performance calculation

   a. Determination of properties pertaining to compression

      Compressibility factor (Z factor), ratio of specific heats (Cp/Cv or k value) and molecular mass are three major physical properties for compressor which must be clarified.

   b. Determination of suction conditions

      The following conditions at the suction flange should be determined:

i) Temperature

ii) Pressure

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

iii) 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.

iv) 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.

c. Determination of discharge conditions

i) Calculation method

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

- Calculate the polytropic exponent "n":
  - o Using the equation:

$$\frac{n}{n-1} = \frac{k}{k-1} \times \eta_p \qquad \text{Eq. (1)}$$

if η_p (polytropic efficiency) is known from the manufacturer data.

η_p can also be estimated from Table1 (k is the ratio of specific heats).

  - o if ηis (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".
  - o Fig. 3 is useful for rough estimation of "n".
  - o "n" can also be calculated iteratively from equation:

$$n = \frac{\log_{10}(P_2/P_1)}{\log_{10}(V_2/V_1)} \qquad \text{Eq. (2)}$$

Where V1 and V2 are specific volumes (actual) and P1 and P2 are absolute pressures at inlet and outlet conditions respectively.

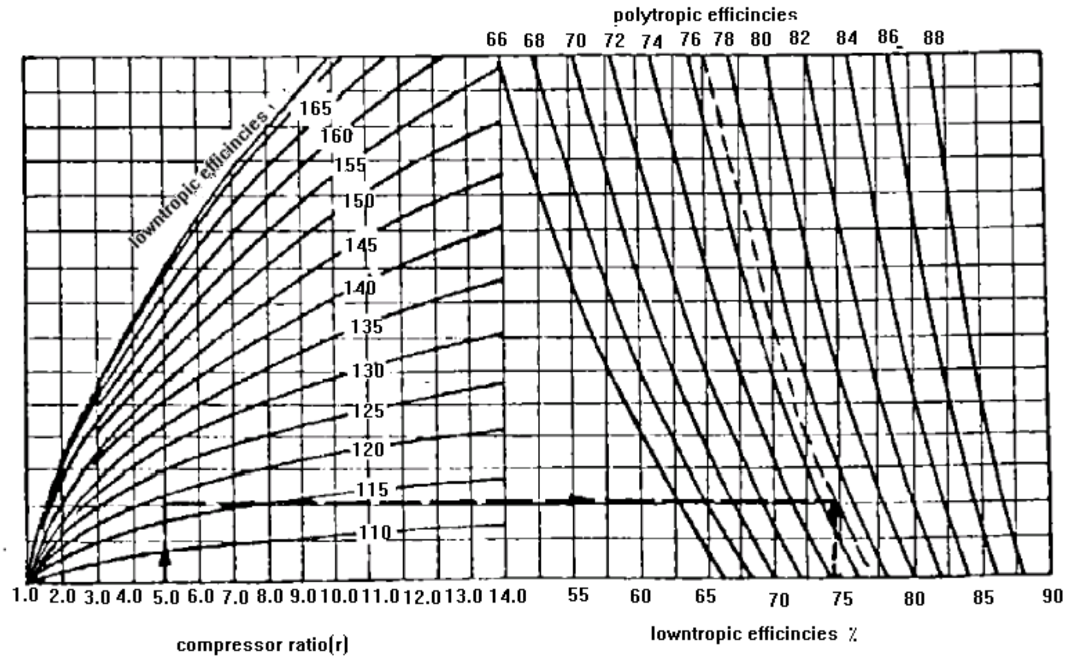

Fig. 1-Efficiency Conversion

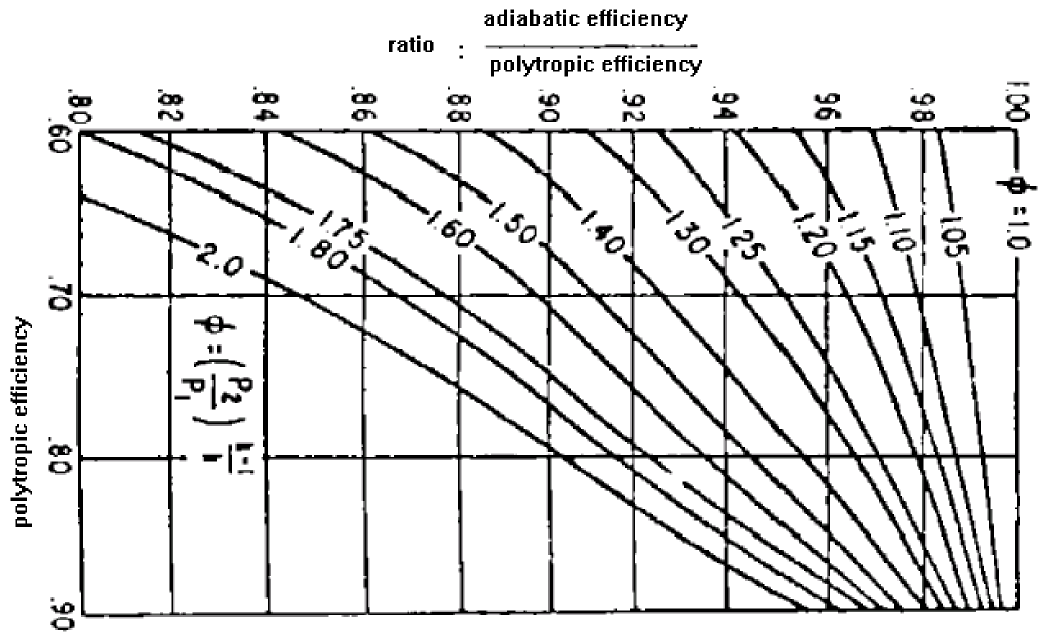

Fig. 2-Relationship Between Adiabatic and Polytropic Efficiencies

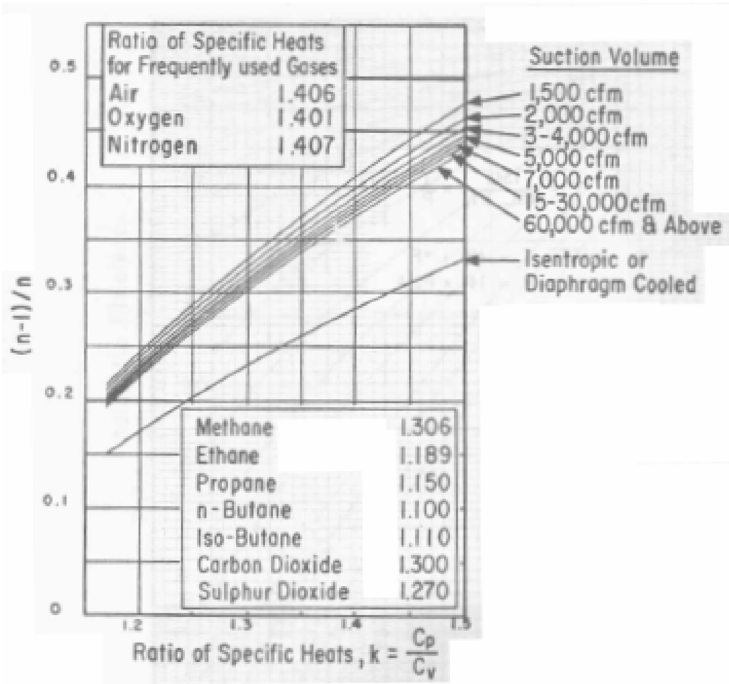

Fig. 3- Ratio of Specific Heats, K = $\frac{C_p}{C_V}$

(n-1)/n Versus Ratio of Specific Heats

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

- Calculate discharge temperature T_2, (Kelvin) from equation:
  - o $T_2 = T_1 (P_2 / P_1)^{(n-1)/n}$ Eq. (3)

    (T_1 and T_2 are absolute temperatures)

    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:

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

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.

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

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

Where:

R is gas constant, in (8314.3 J/kmol.K);

Z_{avg} is average inlet and outlet compressibility factors;

T_1 is inlet absolute temperature, Kelvin (K);

M is molecular mass, (kg/kmol);

g is acceleration of gravity, (9.80665 m/s^2).

- Calculate polytropic head H_p:

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

Note:

Polytropic and isentropic heads are related by:

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

- Calculate gas horse power in kilowatt (hp):

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

or;

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

Where:

W is mass flow rate, (kg/min);

H_p is polytropic head, (m).