

COMPRESSORS**JGS 210-120-1-28E**

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1. SCOPE

This standard practice covers the basic design of reciprocating and centrifugal compressors in petroleum refineries, petrochemical and similar plants.

2. WORK PROCEDURE**2.1 Input to the Design****2.1.1 Process Conditions****(1) Source of the data**

The major design data for compressor will be obtained from the following documents;

- Process Flow Diagram (PFD)
- Material and Heat Balance Sheet
- Basic Engineering Design Data (BEDD)
- Piping & Instrument Diagram (P&ID)

2.1.2 Input Data

The required data for compressor design are as follows.

- (1) Service name
- (2) Fluid name
- (3) Flow rate of gas (wet or dry) V_w or V_d (m³/hr)
- (4) Operating temperature (inlet/outlet) T_i/T_o (°C)
- (5) Operating pressure (inlet/outlet) P_i/P_o (kg/cm²A)
- (6) Vapor pressure P_v (kg/cm²A)
- (7) Physical properties of compressed fluid
 - Molecular weight
 - $\kappa = C_p/C_v$ (inlet/outlet)
 - Compressibility factor (inlet/outlet)
 - Relative humidity (% , inlet) RH
$$V_w = V_d \left(\frac{P_i}{P_i - P_v \cdot RH} \right)$$
 - Corrosive material
 - Composition profile
 - Compressor operation, control method
- (8) Mechanical Design Data
 - Driver type (motor or turbine, etc.)
 - Material of construction (if specified from experience)
 - Set pressure of safety valve for reciprocating compressor
 - Mechanical design temperature if applicable
 - Max allowable temperature
 - Utility data
 - Settling out pressure

2.1.3 Format

Data sheets for compressors shall be prepared in the latest JGC standard form 1756-1 ~ 5 and 1786-1 ~ 7 for reciprocating and centrifugal type, respectively, unless otherwise specified, and the design requirements shall be clearly indicated therein. If the compressor is intended for several cases of operation, operating condition for each cases shall be shown.

Conditions in special cases such as regeneration, N₂ drying and start-up shall be shown together with the operating condition.

2.2 Output from the Design

- (1) Major information from compressor vendor
 - P&ID's around compressor including auxiliary facilities
 - Utility consumption
 - Trip and alarm sequence
- (2) Follow-up work based on the information given by compressor vendor
 - Performance curve check
 - Design pressure at discharge lines
 - Surge analysis, if required
 - Utility consumption
 - P&ID's

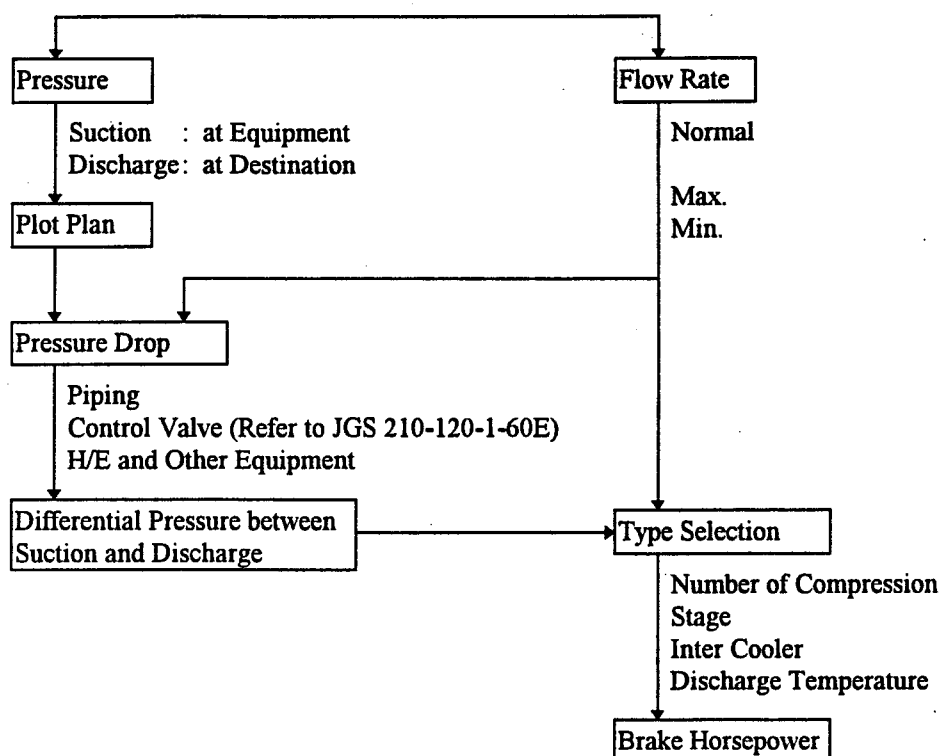
2.3 Work Steps

- (1) Main work of process engineer for compressor design consists of:
 - (a) Estimation of power consumption for economical study
 - (b) Selection of compressor type
 - (c) Preparation of compressor data sheet
 - (d) Preparation of P&ID
 - (e) Follow-up work after detailed design
 - Surge check
 - Performance curve check including surge pressure
 - Provision of antisurge, or spill back piping, if required
 - Incorporation of auxiliary piping to P&ID
 - Incorporation of utility consumption to utility summary
- (2) For hydraulic calculation, refer to JGS 210-120-1-37E "Hydraulic Design".
- (3) The estimation result shall be verified and clarified by vendor's design of compressor for project execution and P&ID preparation.

3. DESIGN**3.1 Compressor Selection**

- (1) The main characteristic of reciprocating and centrifugal compressors are as follows.
 - (a) Reciprocating compressor
 - Suitable for small, medium flow capacity
 - Applicable for high discharge pressure and high pressure ratio
 - Severe limitation of discharge temp. (up to 121 ~ 149°C)
 - Spare machine is necessary for continuous operation
 - Severe limitation for foreign particles
 - Pressure fluctuation
 - Low speed
 - (b) Centrifugal compressor
 - Suitable for large flow capacity
 - About 3,000 m head per one impeller
 - About max. 10 stages per one casing (Depending on the size of machine)
 - Relatively mild limitation of discharge temp. (up to 150 ~ 220°C)
 - Suitable for continuous operation, usually no spare machine
- (2) The type selection of compressor depends on its capacity and compression ratio. The relationship among the type of compressor, capacity and compression ratio is shown in attached Fig.-3.1a for the guideline of the type selection.

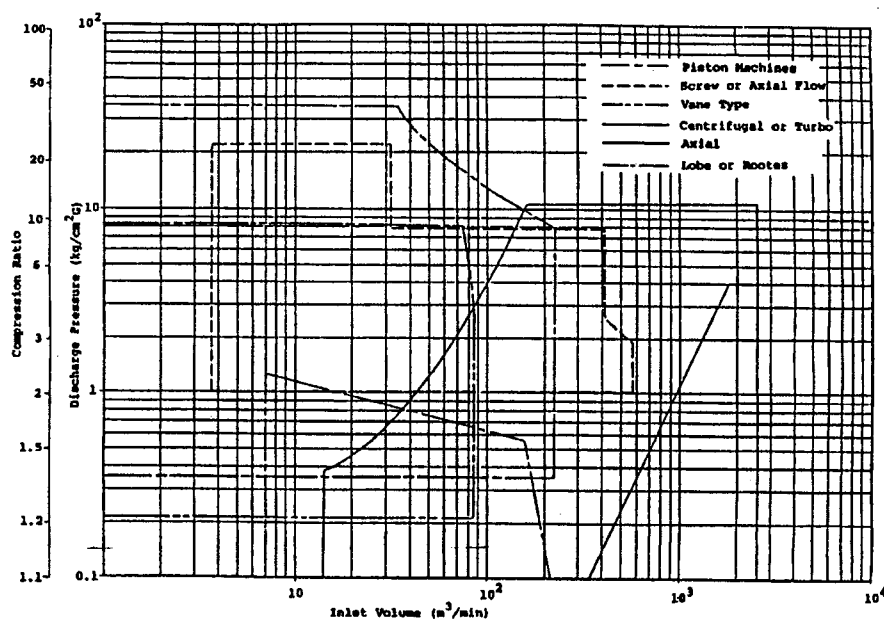
Fig. 3.1a



The steps of selection are generally as follows:

- Calculate the suction volume from the compressor suction conditions.
- Decide the discharge pressure.
- Calculate the overall compression ratio R_c (overall).
- Plot the suction volume and compression ratio on Fig. 3.1b.

Fig. 3.1b



Use discharge pressure instead of the compression ratio in case of air compressors to be designed to draw air from the atmosphere.

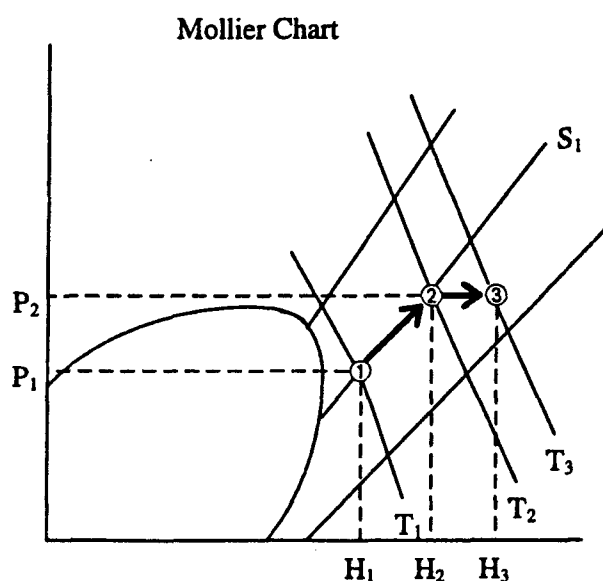
- (e) Several types are available as shown on the figure, but their selection should be made judging from the service conditions and / or experiences in similar cases.
- (3) In cases where the selection is difficult, discussions should be made with the rotary machine engineers to make a final decision.

3.2 Process Evaluation

3.2.1 Isentropic Calculation

Process conditions around compressors are calculated by the molliers diagram as single stage, isentropic operations. Computations require that entropy data be available, either in equation or tabular form, or may be generated along with the thermodynamic rules.

The Mollier chart shown below illustrates the calculation path.



Point 1 represents the inlet condition with pressure P_1 , temperature T_1 , entropy S_1 , and enthalpy H_1 . Starting from this point, a line along with constant entropy is followed to point 2 (P_2, T_2, H_2, S_1). This point represents the discharge conditions for a compressor with an adiabatic efficiency of 100%.

The adiabatic enthalpy change can be calculated.

$$\Delta H_{ad} = H_2 - H_1 = [(HEAD)_{ad}/427]$$

Where:

$$\Delta H_{ad} = \text{Kcal/Kg}$$

$$(\text{HEAD})_{\text{ad}} = m$$

427 = Kg.m/Kcal

The actual enthalpy change may be computed by application of the adiabatic efficiency (typically 70 ~ 75%).

$$\Delta H_{ac} = H_{ad}/\eta_{ad}$$

Where:

η_{ad} = Adiabatic efficiency (usually offered by vendor)

The discharge enthalpy may now be determined.

$$H_3 = H_1 + \Delta H_{ac}$$

The temperature, T_3 , corresponding to P_2 and H_3 is now determined with Mollier Chart or an equilibrium flash calculation.

The adiabatic gas-horsepower is computed as theoretical work.

$$(\text{BHP})_{\text{ad}} = \frac{(\Delta H_{\text{ad}}) (427) (\text{G})}{1.568 \times 10^8} = \frac{1}{3.672 \times 10^5} (\Delta H_{\text{ad}}) (\text{G})$$

Where:

$$(BHP)_{ad} = KW$$

$$\Delta H_{ad} = Kcal/kg$$

$$G = kg/hr$$

$$427 = kg \cdot m/Kcal$$

$$1.568 \times 10^8 = kg \cdot m/hr/KW$$

The actual gas KW, $(BHP)_{ac}$, is determined as actual work.

$$(BHP)_{ac} = (BHP)_{ad} / \eta_{ad}$$

3.3 Reciprocating Compressor

3.3.1 Determination of Process Conditions

(1) Pressure :

Suction Pressure P_s is usually controlled by spill back to suction drum.

Discharge Pressure P_d is determined from the downstream process or control condition such as pressure control or condensing temperature.

(2) Flow rate

Flow rate is independent of pressure.

(3) Temperature :

The rated discharge temperature will be indicated by the vendor, but in the case of reciprocating compressors, even if it is regarded as an adiabatic change, no large error will arise.

$$T_d = T_s \left(\frac{P_d}{P_s} \right)^{\frac{\kappa-1}{\kappa}}$$

Where :

T_d = Discharge temperature (K)

T_s = Suction temperature (K)

P_d = Discharge pressure (absolute pressure)

P_s = Suction pressure (absolute pressure)

κ = C_p / C_v at the mean inlet and outlet conditions

(4) Limitations on discharge temperature :

Limitations on the discharge temperature are as shown below. If they are exceeded, the compression ratio per stage must be reduced and intercoolers must be provided.

$$T_d \leq 149^\circ C \text{ (General hydrocarbon, Lubricating)}$$

$$\leq 135^\circ C \text{ (MW} \leq 12 \text{ H}_2 \text{ Rich)}$$

(General hydrocarbon, $P_d < 70 \text{ kg/cm}^2\text{G}$, Non Lubricating)

(O_2 or Air, Lubricating)

$$\leq 121^\circ C \text{ (General hydrocarbon, } P_d \geq 70 \text{ kg/cm}^2\text{G, Non Lubricating)}$$

3.3.2 Number of Compression Stages

$$R_c \text{ (overall)} = \frac{P_d}{P_s}$$

Where :

$R_c \text{ (overall)}$ = Overall compression ratio

P_d = Final discharge pressure ($\text{kg/cm}^2\text{A}$) (psia)

P_s = Inlet pressure ($\text{kg/cm}^2\text{A}$) (psia)

In usual cases, the compression ratio per stage shall be under 5 to 6.

Number of compression stages (n) is estimated based on the two equations as below.

(1) Limitation of discharge temp. (T_d)

$$(BHP)_{ac} = (BHP)_{ad} / \eta_{ad} \text{ (Refer to 3.2.1)}$$

$$N = \left(\frac{\kappa-1}{\kappa} \right) (\log P_d - \log P_s) / (\log T_d - \log T_s)$$

$$\geq \left(\frac{\kappa-1}{\kappa} \right) (\log P_d - \log P_s) / (\log 422 - \log T_s) (\because T_d \leq 422K)$$

(2) Limitation of compression ratio for each stage

In reciprocating compressor compression ratio per stage is usually under 5.

$$N \geq \log P_d - \log P_s / \log 5$$

Taking the number of compression stages N, the compression ratio per stage can be formulated as follows :

$$R_c = \left(\frac{P_d}{P_s} \right)^{\frac{1}{N}}$$

Where :

R_c = Compression ratio per stage

N = Number of compression stages

Consideration of inter stage cooler should be referred to APPENDIX-1.

3.3.3 Brake Horsepower

(1) Basic equation (in case when κ and η are available)

$$(\text{BHP})_{ac} = \frac{1}{3.672 \times 10^5 \times \eta} \cdot G \cdot \frac{\kappa}{\kappa - 1} \cdot (R / \text{MW}) \cdot T_s \left(\left(\frac{P_d}{P_s} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right) \cdot Z$$

Where :

$(\text{BHP})_{ac}$ = Actual brake horsepower (KW)

κ = Ratio of specific heats

P_s = Initial suction pressure ($\text{kg}/\text{cm}^2\text{A}$)

P_d = Final pressure ($\text{kg}/\text{cm}^2\text{A}$)

G = Gas flow rate (kg/h)

R = Gas constant $847.826 \text{ kg}\cdot\text{m}/\text{Kmol K}$

Z = Compressibility factor = $0.5 \cdot (Z_d + Z_s)$

T_s = Suction temp. (K)

η = Compressor efficiency, assumed as follows

= 0.8 (Lubricating)

= 0.75 (Non Lubricating)

MW = Molecular weight

Z_d = Compressibility at discharge

Z_s = Compressibility at suction

In the case of multi - stage compressors, the above equation is approximation. The brake horsepower shall be calculated for each stage and shall be totaled.

(2) Calculation by Molliers Chart :

Refer to para. 3.2.1.

3.3.4 Control Around Reciprocating Compressor

(a) Suction valve unloading

Stepwise capacity control 0%, 50%, 100% with forced suction valve open. Long period unload operation shall be avoided because of temperature rise.

