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BASIC DESIGN MANUAL
SPD0028ERN

COMPRESSOR

REV. : 2

DATE : 2010. 11. 20



SAMSUNG ENGINEERING CO., LTD.

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1. SCOPE

This manual covers the selection of compressors and the knowledges of compressors which the basic design engineers must have.

Since compressors are very expensive equipment, the engineers must take good care in the making of entries in the data sheets.

2. FORMAT

The forms provided for compressors are as follows:

Reciprocating compressors	Form No.
Centrifugal compressors	Form No.
Fans and blowers	Form No.

3. COMPRESSOR SELECTION

3.1 General

The matters to be considered in compressor selection can be classified into the following three major factors for the purpose of reasonable and logical selection:

- (1) Operating cost
- (2) Environment problem
- (3) Initial capital cost

However, the problems of cost and construction may have to be left to a decision or recommendation by the rotary machine engineers or vendor.

The procedure discussed herein is intended for selection based on the capacities and compression ratios in general cases.

- (1) Calculate the inlet volume from the compressor suction conditions.
- (2) Decide the discharge pressure.
- (3) Calculate the overall compression ratio R_c (overall).
- (4) Plot the inlet volume and compression ratio of Fig.3-1.

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

- (5) Two to three 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.

In cases where the selection is difficult, discussions should be held with the rotary machine engineers to make a final decision.

The detailed procedure are discussed in Chapter 4.

3.2 Compressor Selection Flow

Provision of compressors is determined based on the process flow diagram.

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The basic procedure for this is as shown in Fig. 3-2.

3.3 Procedure

Chapters 4 to 6 give the detailed procedures for providing the data sheets and for estimating the approximate power consumptions.

4. MATTERS COMMON TO RECIPROCATING AND CENTRIFUGAL COMPRESSORS

4.1 Determination of Properties Pertaining to Compression

Compression coefficient (Z factor), ratio of specific heats ($C_p/C_v = k$) and molecular weight are three major physical properties for compression which must be clarified.

Molliers diagrams should be used if available.

4.2 Determination of Suction Conditions

The following conditions at the compressor suction flange shall be determined:

(1) Temperature

(2) Pressure

In cases where suction is to be taken from atmosphere, correction must be made for the elevation.

In the case of reciprocating compressors with snubbers, one percent of the absolute suction pressure shall be regarded as loss.

(3) Humidity

$$V_w = V_d \left(\frac{P_1}{P_1 - P_v} \right) = V_d \left(\frac{P_1}{P_1 - \beta_v \gamma (RH)} \right)$$

Where,

V_w : Wet gas volume (m^3/hr) (ft^3/m)

V_d : Dry gas volume (m^3/hr) (ft^3/m)

P_1 : Total pressure of the system (P_v) (RH)

P_v : Saturated vapor pressure at the temperature of the system

RH : Relative humidity (fraction)

} Absolute pressures in the same unit

(4) Flow rate

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

(5) Properties of gases

The properties of gases shall be described, for example corrosive.

(6) Fluctuation in conditions

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

5. RECIPROCATING COMPRESSOR

5.1 Determination of Discharge Conditions

(1) Pressure

For pressure drops in piping, see Section 12-1. Line Sizing. Pressure drops in control valves (PC, FC, etc.) shall be met as required.

(2) 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.

$$T_o = T_i \left(\frac{P_o}{P_i} \right)^{(k-1)/k}$$

Where, T₀ : Outlet temperature (°K) (°R)

T_i : Inlet temperature (°K) (°R)

P₀ : Outlet pressure (absolute pressure)

P_i : Inlet Pressure (absolute pressure)

K : Cp/Cv at the mean inlet and outlet conditions

(3) 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.

However, these limitations do not apply to air compressors smaller than 500 HP or less than 8.5 kg/cm²G (125 psig).

- (a) The rated discharge temperature for any hydrocarbon-lubricated compressor handling air or other oxygen bearing gas which will support combustion of the lubricant shall not exceed 177°C (350°F)

In such service a high-discharge-temperature alarm or shutdown, set at 191°C (375°F) maximum, is recommended. If the discharge pressure exceeds 21 kg/cm²G (300 psig), lower temperature limits should be considered.

- (b) The rated discharge temperature for hydrocarbon-lubricated compressors handling gases which will not support lubricant combustion shall not exceed 191°C (375°F)
- (c) The rated discharge temperature of non-lubricated compressors or compressors lubricated with synthetic lubricants shall not exceed 204°C (400°F).
- (d) In cases where the fluid to be handled is considered to undergo undesirable problems,

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such as decomposition and polymerization, the upper limit temperature shall be decided judging from the compositions of the fluid.

5.2 Estimation of Number of Compression Stages

$$R_c(\text{overall}) = \frac{P_f}{P_i}$$

Where,

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

P_f : Final discharge pressure (kg/cm²A) (psia)

P_i : Inlet pressure (kg/cm²A) (psia)

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

However, if the adiabatic temperature rise does not exceed the limitation shown in Para. 5.1 since the ratio of specific heats is small, a larger compression ratio can be used.

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

$$R_c = \left(\frac{P_f}{P_i} \right)^{\frac{1}{\gamma}}$$

Where, R_c : Compression ratio per stage

γ : Number of compression stages

5.3 Inter-stage Cooler

(1) Cooling temperature

If condensate is to form in the inter-stage coolers and is to be removed, the flow rate and properties of the fluid will vary.

Since the purpose herein is to calculate the brake horsepower approximately, no such condensation is taken up herein.

The inlet temperature of the cylinder after the inter-stage cooler shall be assumed to be a temperature which can be lowered by the cooling medium temperature.

Namely, the inter-stage cooler outlet temperature shall be the cooling water temperature plus 10°C.

- (2) Compression ratio across stage with inter-cooling pressure drop in the inter-stage cooler shall be regarded to be 0.2 to 0.35 kg/cm² (3 – 5 psi). Re-calculations shall be conducted by using this value.

$$P_f = P_i R^\gamma - (\Delta P_n) R^{\gamma-1} - (\Delta P_n) R^{\gamma-2} - (\Delta P_n) R^{\gamma-3}$$

Where, R : Compression ratio

P_f : Final pressure at multistage set of cylinders

P_i : Initial suction pressure, absolute units

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ΔP_n : Pressure drop across interstage coolers

γ : Number of compression stages

In the case of multi-stage compressors, the compression ratio shall be calculated by the trial-and-error method.

Refer to Appendix A for example.

5.4 Brake Horsepower

(1) Basic equation

$$\text{BHP} = \frac{10^4}{4,462.4} \left(\frac{k}{k-1} \right) P_i V_i \left(\left(\frac{P_{id}}{P_i} \right)^{\frac{k-1}{k}} - 1 \right) L_o F_L Z \quad (\text{for metric system})$$

$$\text{BHP} = \frac{144}{33,000} \left(\frac{k}{k-1} \right) P_i V_i \left(\left(\frac{P_{id}}{P_i} \right)^{\frac{k-1}{k}} - 1 \right) L_o F_L Z \quad (\text{for British system})$$

where,

BHP : Actual brake horsepower (hp)

k : Ratio of specific heats

Z : Compressibility factor

P_i : Suction pressure (kg/cm²A) (psia)

P_{id} : Discharge pressure (kg/cm²A) (psia)

V_i : Suction volume (m³/min) (ft³/min)

L_o : Loss factor, comprised of losses due to pressure drop through friction of piston rings, rod packing, valves, and manifold (See Fig. 5-1)

FL : Frame loss for moter-driven compressors only, values range from 1.0 to 1.05

In the case of multi-stage compressors, the brake horsepower shall be calculated for each stage and shall be totaled.

(2) Rough calculation by chart

Calculations can be conducted with same degree of accuracy by using charts.

(a) Chart calculation procedure

The value (BHP/MMCFD) shall be plotted from the compression ratio and the ratio of specific heats on Fig. 5-2.

In cases where the compression ratio is under 2.5, the correction factor shall be read from the inlet pressure as a parameter on the figure in the left-upper portion of Fig. 5-2.

Then, the value of BHP/MMCFD shall be multiplied by the correction factor.

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$$\text{BHP} = (\text{BHP/MMCFD}) \text{ C}/10^6$$

Where, C : Flow rate at 14.4 psia and inlet temp. (CFD)

b) Precautions to be taken

- Z factors are not considered
- Mechanical cylinder efficiency is taken to be 95%
- In the case of multi-stage compressors, the brake horsepower for each stage must be summed up

(3) Calculation by Molliers Diagram

Calculations can be worked out most easily and that accurately if the Molliers diagram is available.

(a) Horsepower

The enthalpy at the inlet pressure and temperature shall be calculated.

The enthalpy at the outlet pressure shall be calculated in the unit of entropy.

Horsepower (amount of work) = $h_0 - h_i$

Where, h_i : Enthalpy at the inlet (kcal/kg) (Btu/lb)

h_0 : Enthalpy at the outlet (kcal/kg) (Btu/lb)

(b) Brake horse power (BHP)

$$\text{BHP} = \frac{1}{641.39} (W)(h_0 - h_i) L_0 F_L \quad (\text{for metric system})$$

$$\text{BHP} = \frac{779}{33,000} (W)(h_0 - h_i) L_0 F_L \quad (\text{for British system})$$

Where, W : Gas flow rate (kg/hr) (lb/min)

L_0 : Loss factor

F_L : Fram loss for motor driven unit only

5.5 Entries in P & I Diagram

(1) Utility piping

Entries of cooling water to jackets, inter-stage coolers, etc. and winterizing piping shall be limited to a required minimum.

(2) Pulsation damper

Snubbers or surge drums shall be provided at the discharge and/or suction.

Peak-to-peak amplitudes of pulsation shall be limited by damping to 2% of the mean absolute pressure and the pressure drop to under 1% of the absolute pressure.

Refer to API standard 618 20.

Normally, the compressor design is provided by the compressor manufacturer.

(3) Spill-back

In some cases, spill-back from discharge to suction will be required for flow or pressure

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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%

(4) Cooling water rate

The cooling water rate is to be determined after completion of type selection, but it is estimated roughly at 10 tons/hr including consumptions in the jacket and oil cooler.

6. CENTRIFUGAL COMPRESSORS

6.1 Determination of Discharge Conditions

(1) Discharge pressure

Discharge pressure shall be determined based on the process conditions.

(2) Polytropic exponent, n

The value n should be finally determined based on the ratio of specific heats ($k = C_p/C_v$) of the gas to be handled and the mechanical design.

The method discussed herein is intended merely for rough estimation.

In the case of centrifugal compressors, as the polytropic exponent n is larger than the ratio of specific heats, it cannot be regarded as an adiabatic change in the calculation of discharge temperature and brake horsepower.

(a) If the value of polytropic efficiency e_p can be known from record or vendor information the value n can be calculated by the following equation:

$$e_p = \frac{(k-1)/k}{(n-1)/n}$$

(b) The value n can be known from the chart: the value $(n-1)/n$ can be calculated from the value k and the suction volume (CFM) by using Fig. 6-1.

(3) Discharge temperature

$$T_0 = T_i \left(\frac{P_0}{P_i} \right)^{(n-1)/n}$$

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Where, T_i : Inlet temperature (°K) (°R)

T_o : Outlet temperature (°K) (°R)

P_i : Inlet pressure (absolute)

P_o : Outlet pressure (absolute)

This equation can be used to estimate the efficiency from operating data.

(4) Temperature limitations

The operating temperature should not exceed 232°C (450°F) at any point in the operating range.

In usual cases, the maximum temperatures are close to the surge point.

Should this temperature be exceeded, difficulties will be encountered in the mechanical design.

6.2 Estimation of Number of Compression stages

$$R_c(\text{overall}) = \frac{P_f}{P_i}$$

Where, $R_c(\text{overall})$: Overall compression ratio

P_f : Final discharge pressure

P_i : Inlet pressure

The discharge temperature shall be calculated from the overall compression ratio, and if the discharge temperature exceeds 232°C (450°F), the number of stages shall be increased and the compression ratio (R_c) per stage shall be calculated by the following equation:

$$R_c = \left(\frac{P_f}{P_i} \right)^{\frac{1}{\gamma}}$$

Where, R_c : Compression ratio per stage

γ : Number of compression stages

6.3 Inter-stage coolers

For cooling temperatures and pressure drops, see Para. 5.3.

6.4 Brake Horsepower

(1) Basic equation

$$BHP = \frac{10^4}{4562.4} \left(\frac{n}{n-1} \right) P_i V_i (R_{Ci}^{\frac{n-1}{n}} - 1) \frac{Z}{e_p \beta_{Loss}} \quad (\text{for metric system})$$

$$BHP = \frac{144}{33000} \left(\frac{n}{n-1} \right) P_i V_i (R_{Ci}^{\frac{n-1}{n}} - 1) \frac{Z}{e_p \beta_{Loss}} \quad (\text{for British system})$$

or, disregarding the loss,

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$$\text{BHP} = \frac{W H_p}{4562.4 e_p} \quad (\text{for metric system})$$

$$= \frac{W H_p}{33000 e_p} \quad (\text{for British system})$$

$$H_p = \frac{848 Z T_i}{MW} \left(\frac{n}{n-1} \right) (R_{Ci}^{n-1/n} - 1) \quad (\text{for metric system})$$

$$H_p = \frac{1545 Z T_i}{MW} \left(\frac{n}{n-1} \right) (R_{Ci}^{n-1/n} - 1) \quad (\text{for British system})$$

H_p for British system can be obtained from Fig. 6-2.

where,

R_{ci} : Compression ratio of i stage

V_i : inlet volume at inlet condition (m³/min)

n : polytropic exponent

z : Compression coefficient

Calculations can be conducted with higher accuracy by using the average of the inlet and outlet compression coefficients.

e_p : polytropic efficiency

Loss : Mechanical loss in compressor (normally 0.01 – 0.03)

In the case of centrifugal compressor, gear loss shall be taken into account.

W : Gas flow rate (kg/min) (lb/min)

H_p : Polytropic head (m) (ft)

T_i : Inlet gas temperature (°K) (°R)

MW : Mean gas molecular weight

In the case of multi-stages, the brake horsepower of each stage must be calculated and summed up.

(2) Molliers diagram

The molliers diagram for compressed fluid, if available, should be used.

(a) Adiabatic head

The inlet gas enthalpy shall be calculated from the inlet gas conditions.

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The enthalpy at the outlet pressure shall be calculated in the unit of entropy.

$$\Delta h(ad) = h_0(ad) - h_i$$

$$H(ad) = \begin{cases} \Delta h(ad) \times 427 & \text{(for metric system)} \\ \Delta h(ad) \times 778 & \text{(for british system)} \end{cases}$$

Where,

- h_i : Gas enthalpy at the inlet condition (kcal/kg) (Btu/lb)
- $h_0(ad)$: Gas enthalpy as adiabatic change at the outlet pressure (kcal/kg) (Btu/lb)
- $H(ad)$: Adiabatic head (kg m/kg) (ft lb/lb)

(b) Polytropic head (H_p)

$$H_p = \frac{H(ad) \cdot e_p}{e_{ad}}$$

Where,

- e_p : Polytropic efficiency
(whose calculation is explained in (2) of Para. 6.1)
- e_{ad} : Adiabatic efficiency

$$e_{ad} = \frac{R_{Ci}^{k-1/k} - 1}{R_{Ci}^{n-1/n} - 1}$$

Where, R_{Ci} : Compression ratio of i stage

H_p : Polytropic head (m) (ft)

This can be plotted from the values of k and e_p in Fig.6-3.

(c) Brake horsepower (BHP)

The brake horsepower can be calculated by the following equation, disregarding loss:

$$\begin{aligned} BHP &= \frac{W H_p}{4562.4 e_p} && \text{(for metric system)} \\ &= \frac{W H_p}{33000 e_p} && \text{(for British system)} \end{aligned}$$

Where, BHP : Brake horsepower (kW×0.7457)

W : Gas flow rate (kg/min) (lb/min)

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6.5 Operation Controls

(1) Speed control

Speed control is used in the case of steam turbine driven compressors.

In other word, the application of steam turbine is preferable in the case where speed control is required.

(2) Throttling of inlet

In cases where constant speed drivers are used, this means is used most generally.

The inlet gas density can be reduced by throttling.

However, care for surging should be taken.

6.6 Required Cooling Water Consumption

Roughly speaking, the following consumptions are considered:

Oil cooler	10 tons/hr
Seal oil cooler	5 tons/hr

6.7 Affinity Laws

The affinity laws express the relationship between the head, capacity, speed and size of centrifugal compressors. In general these relations can be applied to inlet volume conditions for good preliminary designs, but all final designs apply these laws to the actual discharge volumes from the impeller.

(1) Speed

$$V_2 = V_1 \left(\frac{\text{rpm}_2}{\text{rpm}_1} \right)$$

Flow rate

$$H_2 = H_1 \left(\frac{\text{rpm}_2}{\text{rpm}_1} \right)^2$$

Head

$$\text{BHP}_2 = \text{BHP}_1 \left(\frac{\text{rpm}_2}{\text{rpm}_1} \right)^3$$

Horsepower

Where sub-1 presents the first, and sub-2 the second condition of operation.

(2) Impeller diameters (similar) impeller

For two geometrically similar, same family, wheels with the same specific speed and operated at the same rpm:

$$H_2 = H_1 \left(\frac{D_2}{D_1} \right)^2$$

$$V_2 = V_1 \left(\frac{D_2}{D_1} \right)^3$$

$$\text{BHP}_2 = \text{BHP}_1 \left(\frac{D_2}{D_1} \right)^5$$

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(3) Impeller diameters (changed)

When an impeller diameter is reduced, but the speed is held constant:

$$H_2 = H_1 \left(\frac{D_2}{D_1} \right)^2$$

$$V_2 = V_1 \left(\frac{D_2}{D_1} \right)$$

$$BHP_2 = BHP_1 \left(\frac{D_2}{D_1} \right)^3$$

(4) Effect of temperature

$$BHP_2 = BHP_1 \left(\frac{T_2}{T_1} \right)$$

7. REFERENCE

- (1) Oil & Petrochemical, January 15, 1971, M.A. Brightwell
- (2) Applied Process Design for chemical and Petrochemical plants Vol. 3, Gulf Publishing Company
- (3) API 618

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APPENDIX

Appendix A: Interstage Pressure and Ratios of Compression

(1) Example 1: Two stages of compression

Question:

For two stages of compression, what should be the pressure across the cylinders if the intercooler and piping pressure drop is 3 psi?

Suction to first stage: $P_1 = 0$ psig (14.7 psia)

Discharge from second stage: $P_2 = 150$ psig (164.7 psia)

Answer:

Compression ratio per stage without cooling:

$$R_2 = \sqrt{164.7/14.7} = 3.35$$

(a) No intercooling

$P_1 = 14.7$ psia

Discharge from first stage : $P_{n1} = 3.35 \times 14.7 = 49.2$ psia

Suction to second stage : $P_2 = 49.2$ psia

Discharge from second stage : $P_{n2} = 3.35 \times 49.2 = 164.7$

(b) With intercooling

The following equation can be used:

$$P_{n2} = P_1 R_c^2 - \Delta P_n R_c$$

Where, ΔP_n : Pressure drop across interstage coolers (psi)

The compression ratio per stage with intercooling R_c can be solved using the following values.

$$P_1 = 14.7$$

$$P_{n2} = 164.7$$

$$\Delta P_n = 3.0$$

Then, $R_c = 3.45$

$$P_{n1} = 3.45 \times 14.7 = 50.7$$

$$P_2 = 50.7 - 3.0 = 47.7$$

$$P_{n2} = 3.45 \times 47.7 = 164.7$$

(2) Example 2: Three stages of compressions

Question:

For three stages of compression, what should be the pressure across the cylinders if the intercooler and piping pressure drop is 3 psi?

Suction to first stage: $P_1 = 0$ psig (14.7 psia)

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Discharge from second stag: $P_{n3} = 380 \text{ psig (394.7 psia)}$

Answer :

Compression ratio per stage without cooling:

$$R_2 = \sqrt[3]{394.7/14.7} = 2.99$$

(a) No intercooling

$$P_{n1} = P_2 = 2.99 \times 14.7 = 44.0$$

$$P_{n2} = P_3 = 2.99 \times 44.0 = 131.8$$

$$P_{n3} = 2.99 \times 131.8 = 394.7$$

(b) With intercooling

The following equation can be used:

$$P_{n3} = P_1 R_c^3 - \Delta P_n R_c^2 - \Delta P_n R_c$$

Using the following values, the compression ratio per stage with intercooling can be obtained by means of trial and error calculation.

$$P_1 = 14.7$$

$$P_{n3} = 394.7$$

$$\Delta P_n = 3.0$$

Then, $R_c = 3.087$

$$P_{n1} = 3.087 \times 14.7 = 45.4$$

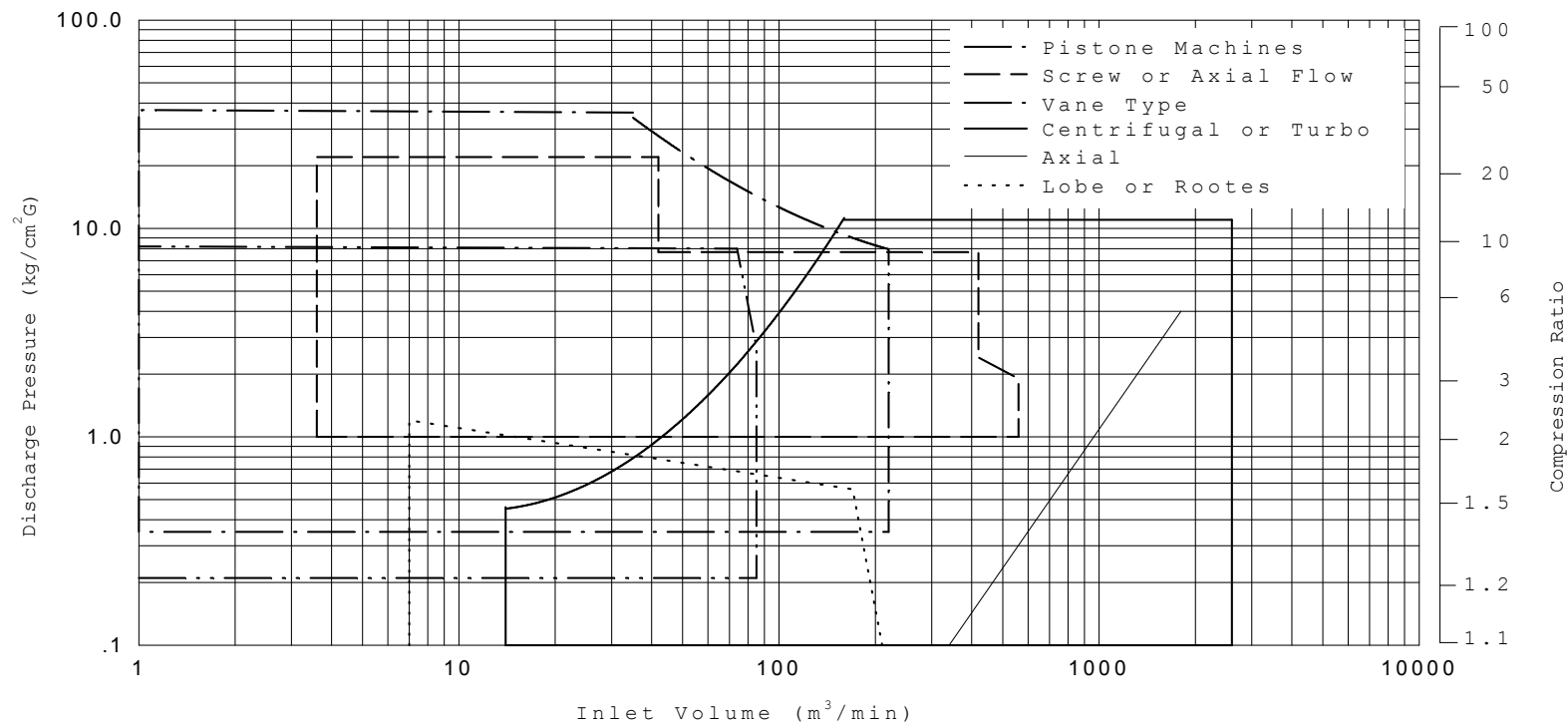
$$P_2 = 45.4 - 3.0 = 42.4$$

$$P_{n2} = 3.087 \times 42.4 = 130.8$$

$$P_3 = 130.8 - 3.0 = 127.8$$

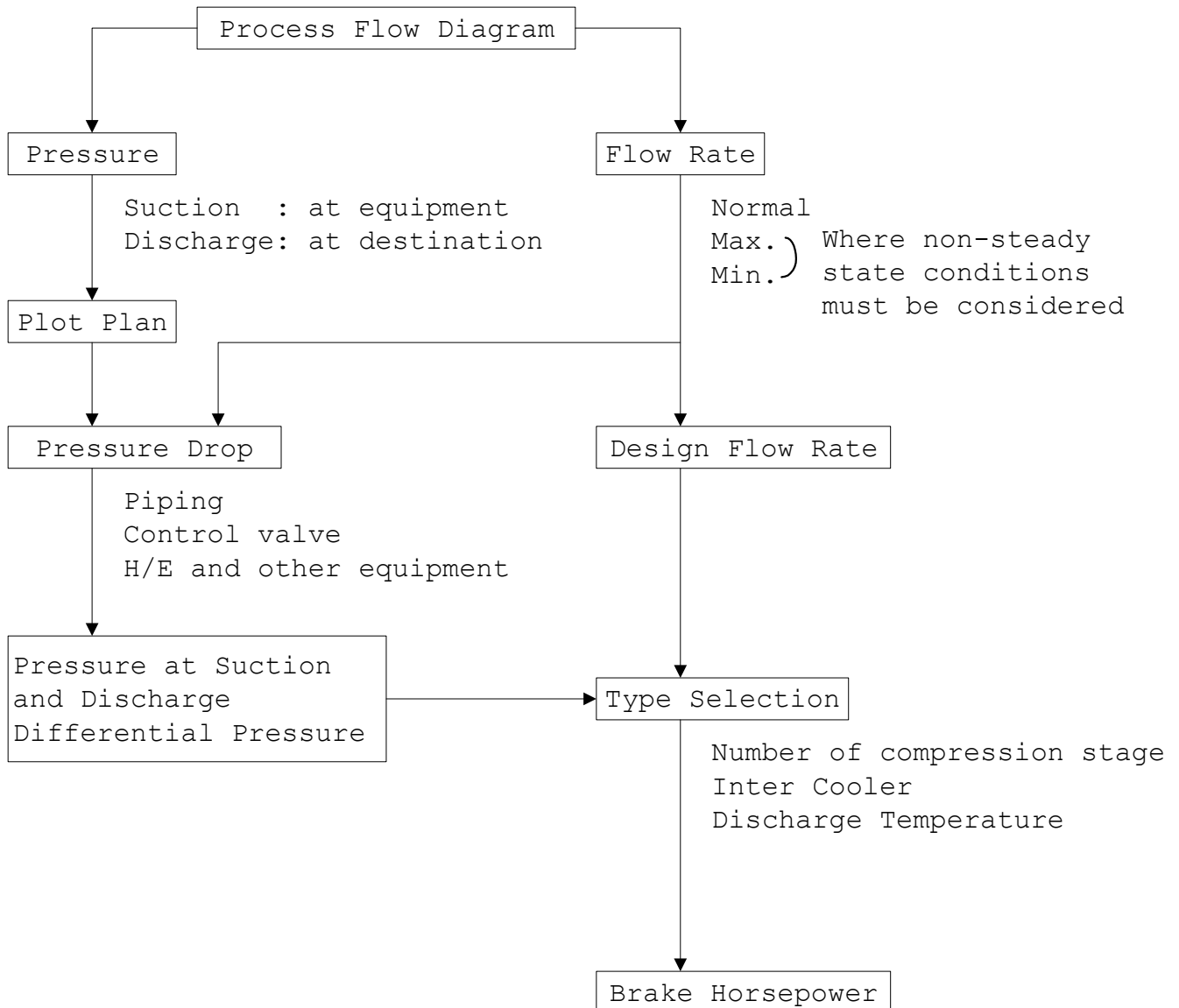
$$P_{n3} = 3.087 \times 127.8 = 394.7$$

Fig. 3-1. Field Activity Normally Rated Varios Type of Compressor



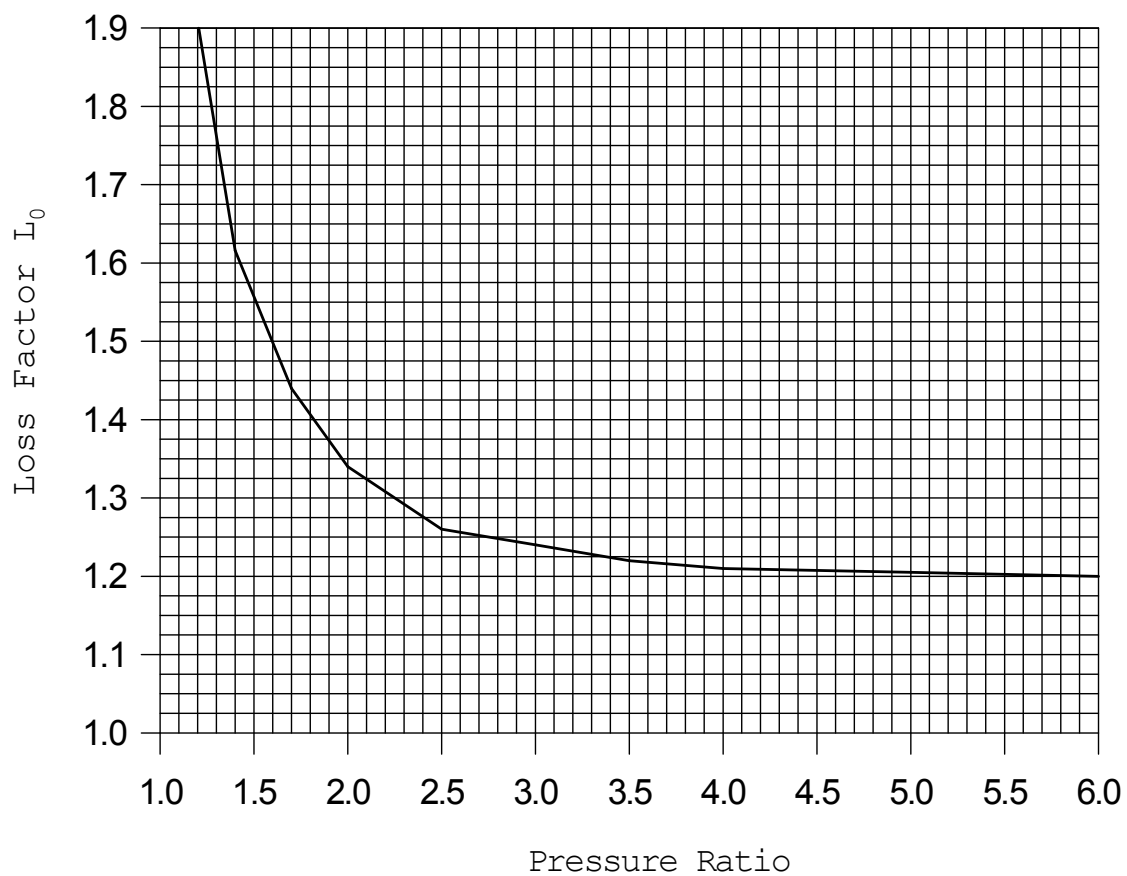
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Fig. 3-2. Compressor Selection Flow



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Fig. 5-1 Loss Factor Curve



Actual Horsepower = (Adiabatic HP)(Loss Factor, L_0)

For $C_p/C_v = 1.20$

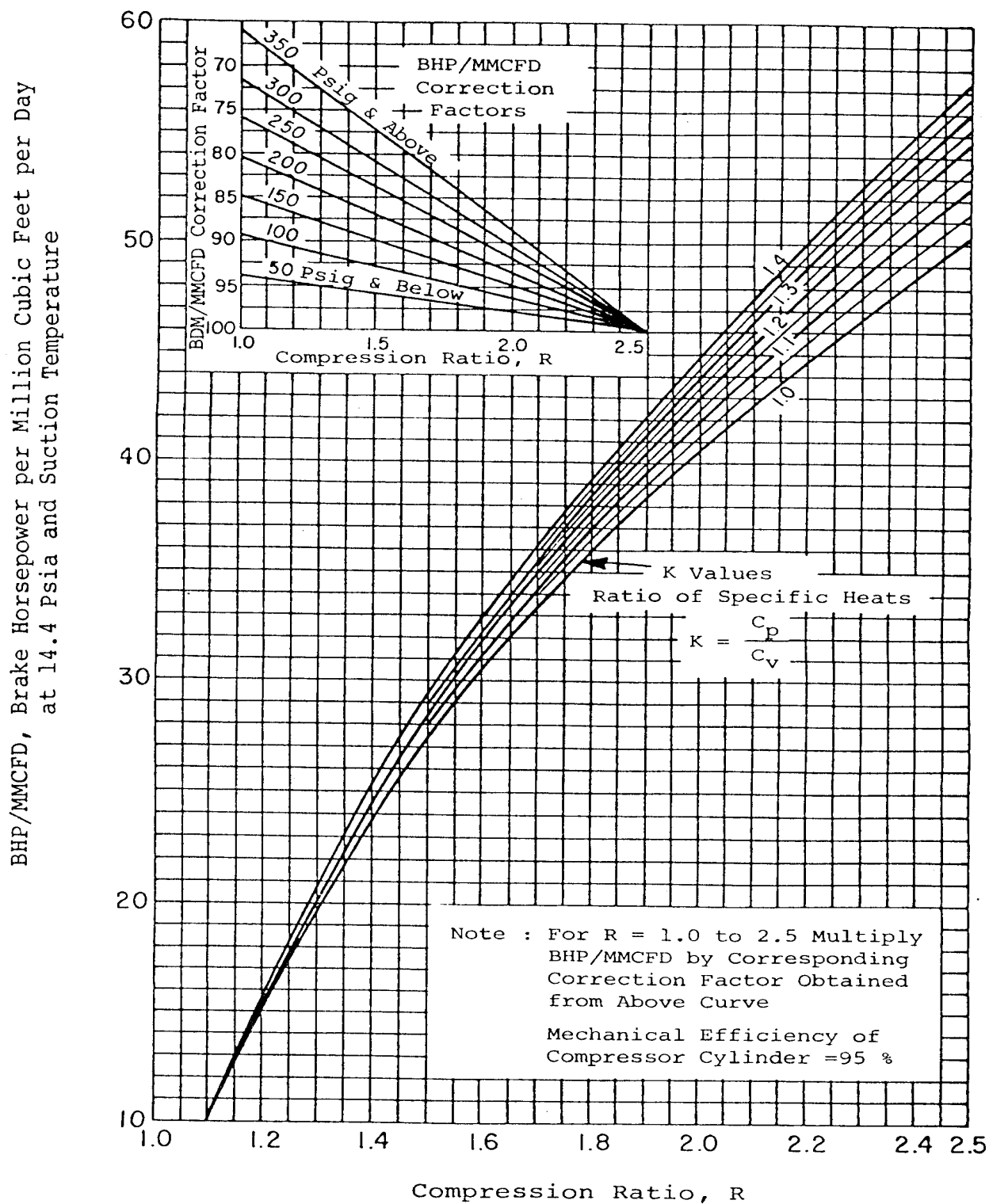
Adiabatic Hp/MMcfd + 262 ($R_c^{0.2/1.2} - 1$)

Loss Curve applies for all C_p/C_v

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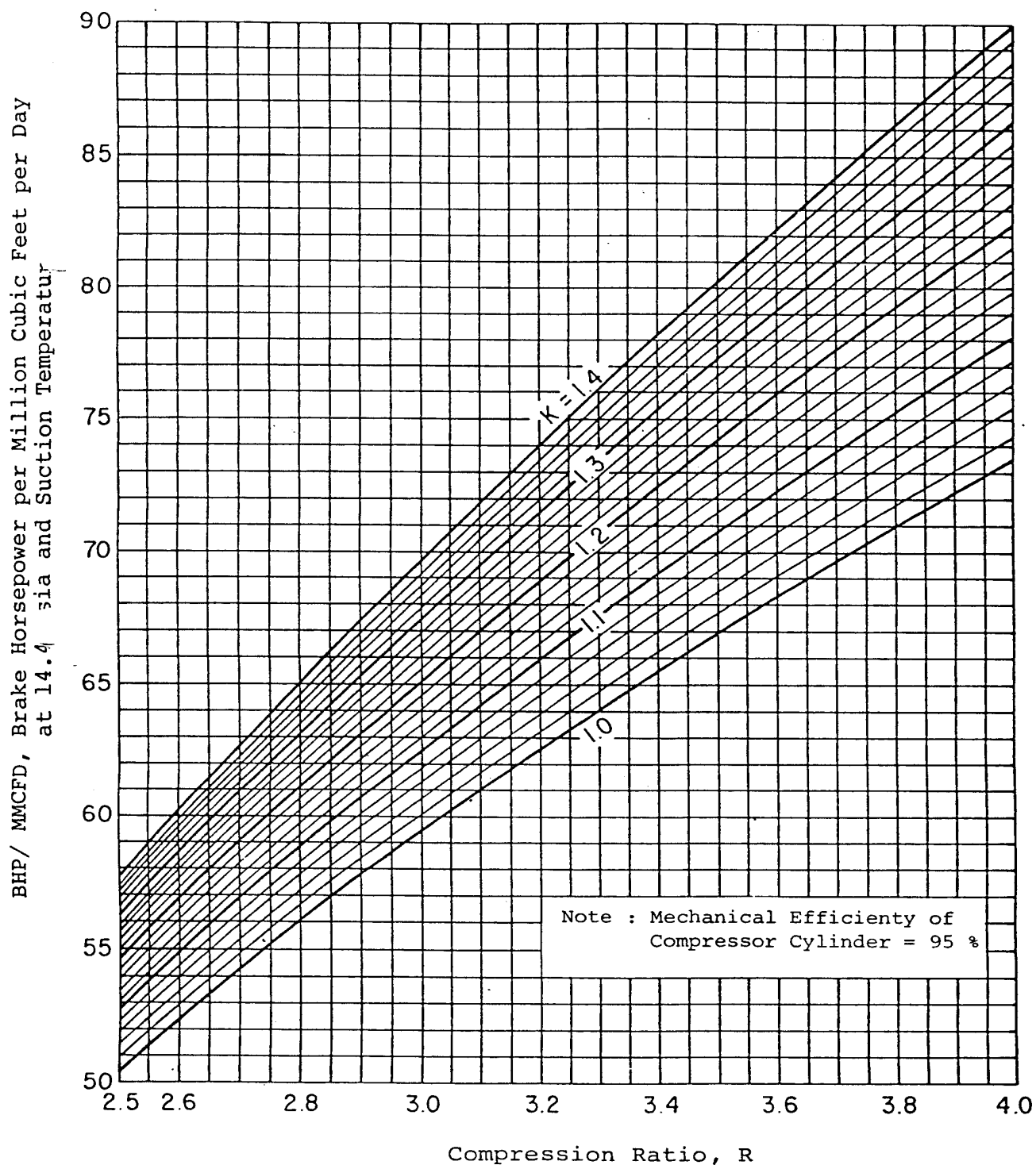
Rc	Actual Bhp/MM See Fig. 5-2.	Adiabatic Hp/mm	Loss Factor
1.1	10	4.19	2.39
1.2	15.4	8.07	1.91
1.4	24.4	15.1	1.616
1.7	34.9	24.2	1.44
2	43	32.1	1.34
2.5	54.5	43.2	1.26
3	65.1	52.6	1.24
3.5	74.1	60.8	1.22
4	82.5	68.1	1.21
5	98.5	80.7	1.205
6	113	91.1	1.2

Fig. 5-2. Brake Horsepower Required to Deliver One Million Cubic Feet of Gas per Day



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Fig. 5-2. Brake Horsepower Required to Deliver One Million Cubic Feet of Gas per Day (continued)



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Fig. 5-3 Brake Horsepower Required to Deliver One Million Cubic Feet of Gas per Day

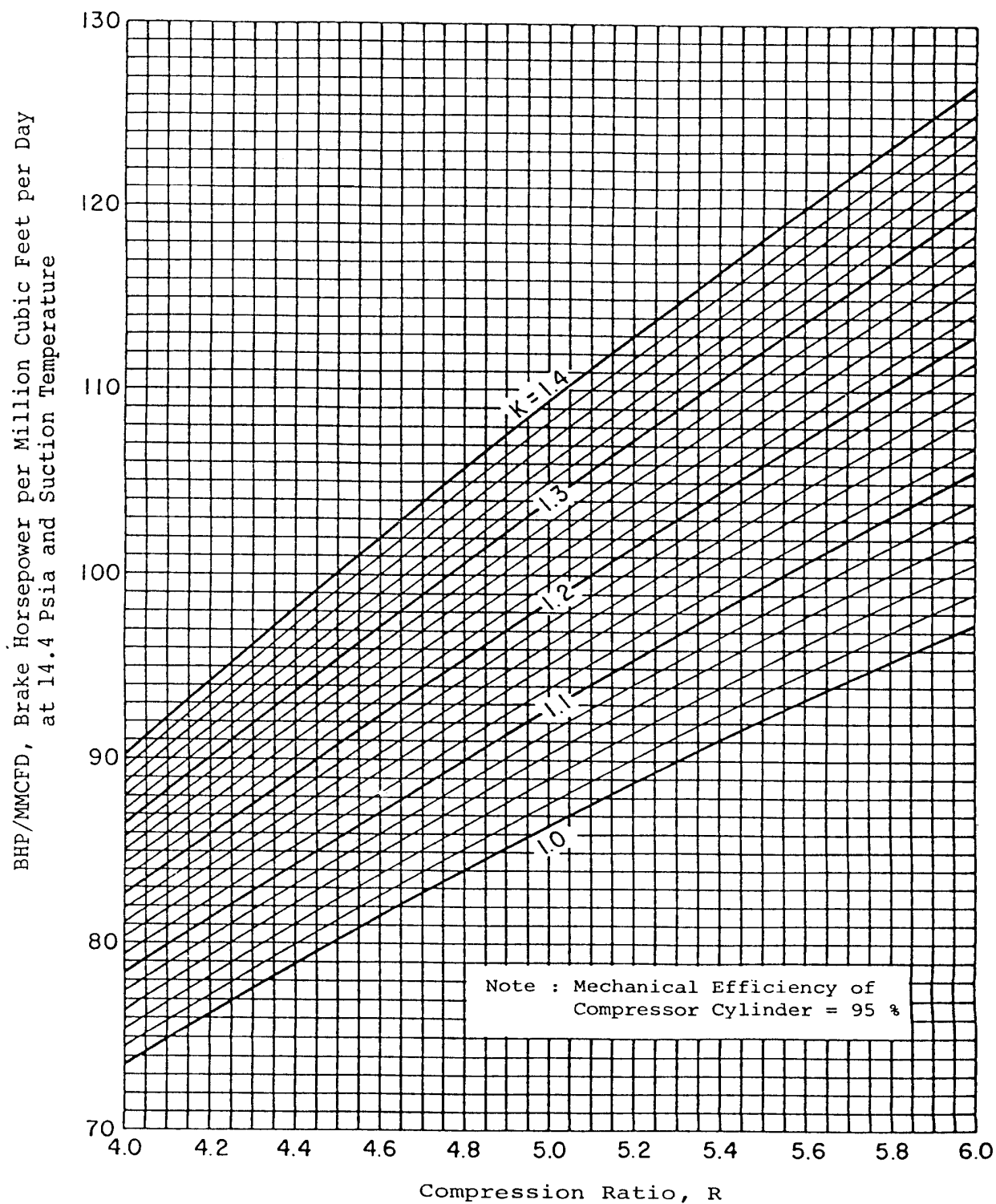


Fig. 6-1. Ratio of Specific Heats vs $(n-1)/n$

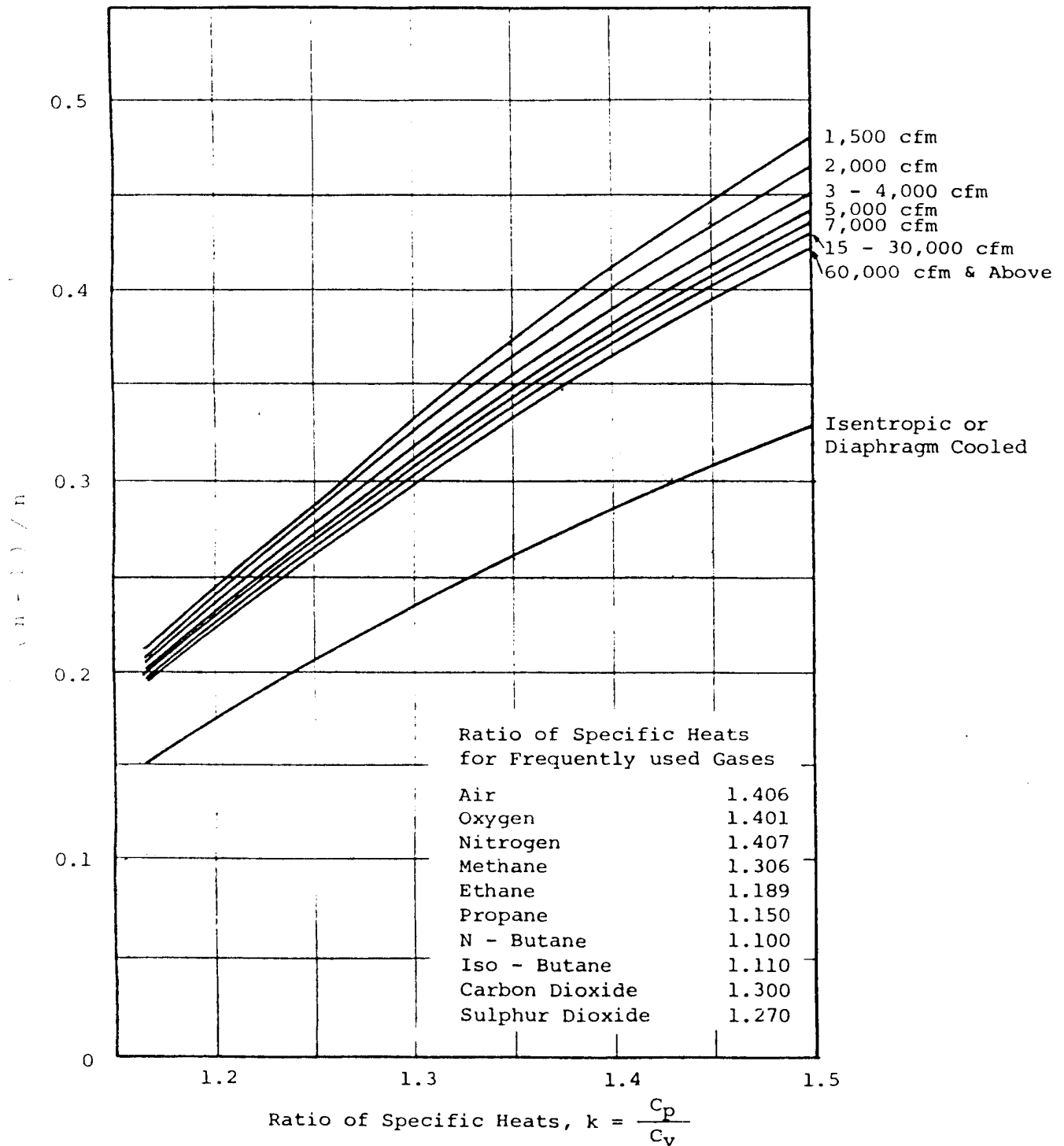


Fig. 6-2. Approximate Head Selection

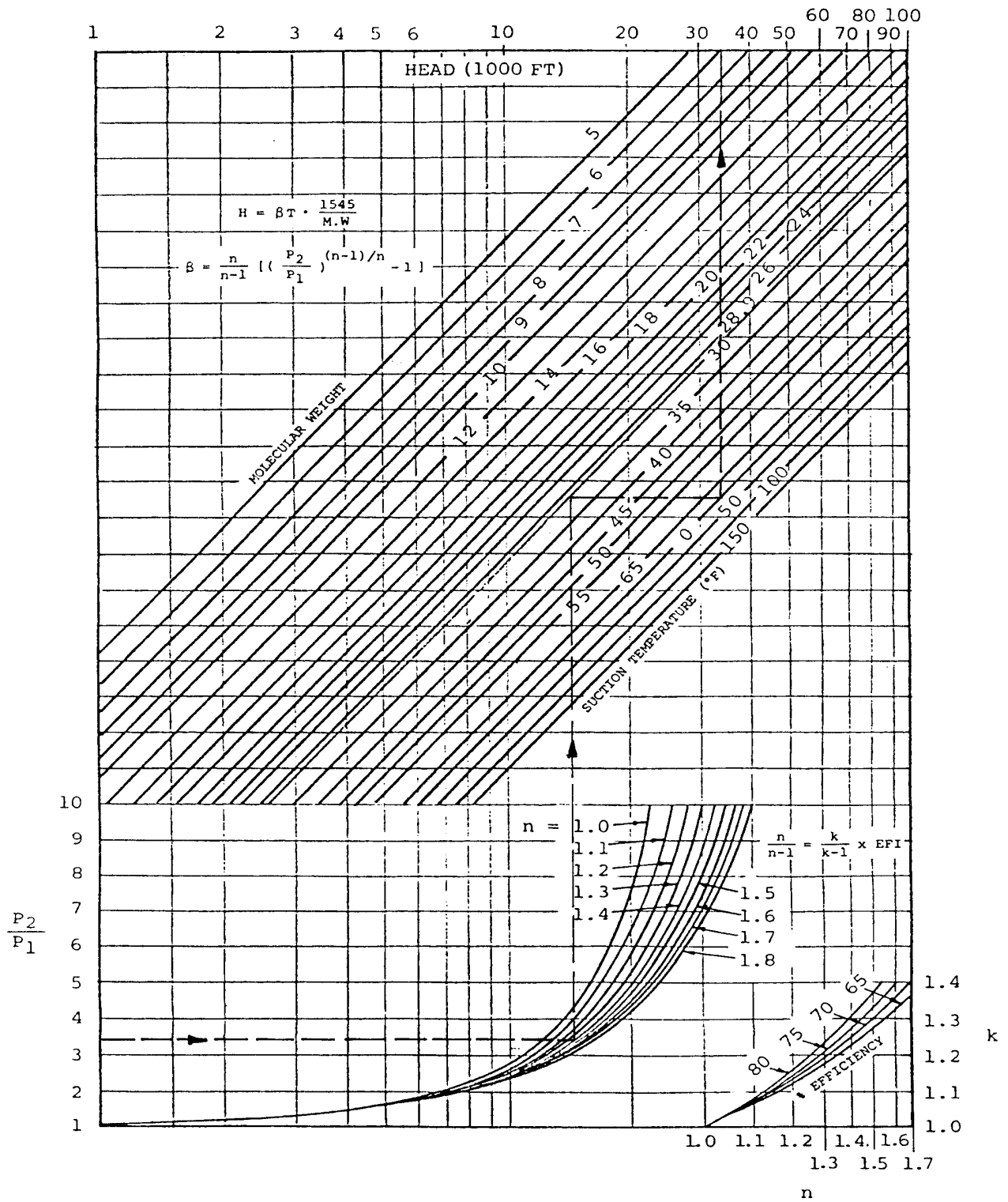
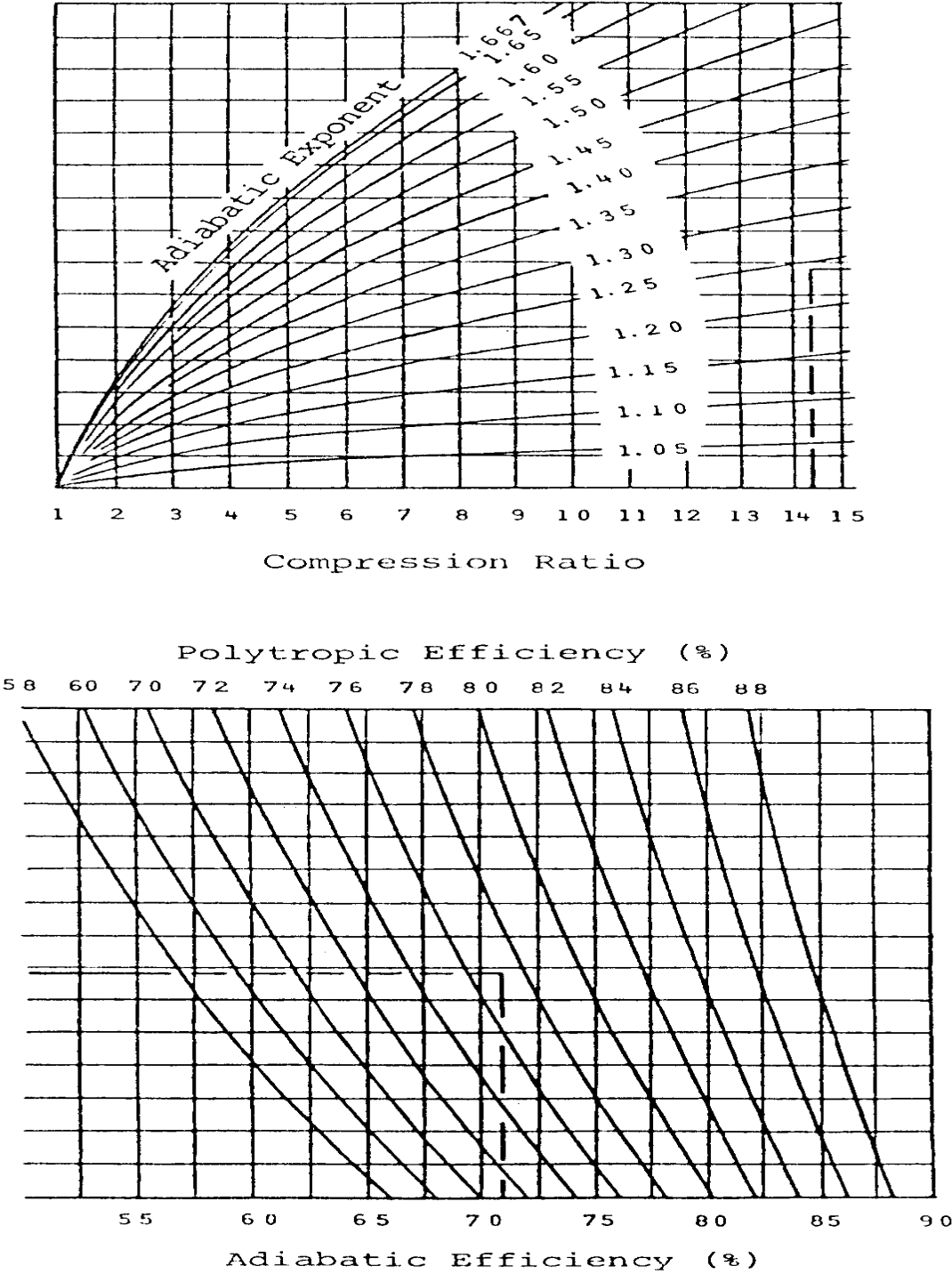


Fig. 6-3. Adiabatic Efficiency Curve



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8. REVISION HISTORY

Rev. No.	Date	Scope	Description
0	98. 4.17	All pages	- First Edition - Team : Process Eng'g Team - Made by : J.H.Park J.E.Lim - Checked by : B.I.Lee S.B.Jo J.C.Ro - Approved by : S.G.Oh
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