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## BASIC PRACTICES IN COMPRESSORS SELECTION

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### ABSTRACT

Selection of each major type of Reciprocating, Rotary, Centrifugal and axial Compressor is briefly reviewed. It is explained how the non-dimensional scales of Specific Speed and Specific Diameter can be used to select any of the above configuration of Compression machinery at its optimum efficiency range.

The paper presents the approach of Engineers - Constructors in selecting compressors in petroleum, chemical and other industries. The limiting aero, thermo- and rotor dynamic parameters in respect of each type of Compressor are discussed. It is shown how the initial Capital Cost, Operating Cost, environmental and logistics considerations influence the final selection of a Compressor for a given application.

Future trends in the industry are addressed. Areas needing attention by the manufacturers and research workers are identified to further our knowledge and confidence in the selection of Compression machinery.

### INTRODUCTION

The type of compressors used in the petroleum, chemical and other industries falls into two broad categories viz positive displacement type and dynamic type. Fig. 1 presents various sub-categories referred to in accordance with their design, construction and or operation. It is outside the scope of this paper to present details of each of these sub-categories. Fig. 2 explains the broad range of capacity (actual CFM) and discharge pressures (PSIG). Note the overlap between configurations. Final selection of any type of compressor is influenced by economic, operating, logistics and environmental considerations as outlined in Fig. 3. In order to provide general understanding of the selection criteria following parameters are defined.

### DEFINITION OF AERODYNAMIC PARAMETERS

#### Specific Speed

Specific speed is an index number for the impellers or rotors of various types of pumps or compressors. It relates to the flow and head handling capabilities of a turbomachine at a given speed. It enables to relate the optimum design at peak efficiency. It is defined as:

$$N_s = \frac{N Q^{1/2}}{(g_c H)^{3/4}} \quad \text{Where}$$

$N_s =$  Specific Speed, non-dimensional

$N =$  Compressor Speed,  $\frac{\text{RPM}}{60}$

$Q =$  Inlet Flow,  $\text{ft}^3/\text{Sec}$ .

$H =$  Adiabatic head,  $\text{ft lbf/lbm}$

$$g_c = 32.2 \frac{\text{lbm ft}}{\text{lbf Sec}^2}$$

The Specific Speed is indicative of the type of rotating impeller or rotor. From Figure 4 it can be seen that large specific speeds are associated with axial compressors, which handle large flow and low heads and modest speeds.

#### Specific Diameter

Diameter of a rotating wheel in turbomachinery is a measure of volume flow when related with rotational speed. Specific diameter is developed as a non-dimensional scale factor for comparing turbomachinery performance.

Specific Diameter is defined as:

$$D_s = \frac{D(Hg_c)^{1/4}}{\sqrt{Q}}$$

where  $D_s =$  Specific diameter  
 $D =$  Dia of rotor/impeller, ft.

$$H = \text{Adiabatic Head } \frac{\text{ft lbf}}{\text{lbm}}$$

$$Q = \text{Inlet Flow, ft}^3/\text{sec}$$

$$g_c = \text{Gravity Conversion factor, } 32.2 \frac{\text{lbm} \times \text{ft}}{\text{lbf} \text{ sec}^2}$$

It can be noted from the Balje Chart, Figure (4), Ref. 1 & 2, that positive displacement compressors have high Specific Diameter values due to high head low-flow handling capability.

#### Actual Flow ACFM

It is customary to refer to flow rates in SCFM i.e., standard atmospheric conditions 14.7 psia and 60°F. However, compressors handle the flow that is available at the inlet flange at the prevalent pressure and temperature conditions. Actual flow ACFM is computed as follows:

$$VA = \frac{Ps Vs}{Zs Ts} \frac{Za Ta}{Pa}$$

Where  
 Va = Actual Volume @ Pa, Ta ACFM  
 Ps = Standard Reference Pressure, psi  
 Vs = Standard Volume, SCFM  
 Zs = Compressibility factor at standard conditions  
 Za = Compressibility factor at actual conditions

#### Head Coefficient

The head coefficient is a dimensionless factor which relates the adiabatic head capability of a wheel with its peripheral velocity requirement. It is defined as:

$$\psi = \frac{g_c H}{U^2}$$

Where:  $\psi$  = Head Coefficient, non-dimensional

$$H = \text{Adiabatic Head in } \frac{\text{ft lbf}}{\text{lbm}}$$

$$g_c = \text{Gravity constant } 32.2 \frac{\text{lbm ft}}{\text{lbf Sec}^2}$$

U = Peripheral Velocity of the impeller ft/sec.

$$= \frac{DN}{60}$$

N = Rotational Speed RPM

D = Impeller diameter, ft.

The head coefficient is also called pressure coefficient, and it can be analytically shown to combine all blade angle, slip and efficiency

effects into one value. For centrifugal compressors the head coefficient varies from 0.48-0.54 based on the extent of backward-curved blades. Impellers with radial blades have higher values to 0.63. Higher pressure coefficients can also be obtained by increasing number of blades in the impeller. However, cost and pressure losses limit the number of blades that can be used in an impeller.

#### Flow Coefficient

The flow coefficient is a non-dimensional unit which takes flow (Q) into consideration for a given tip speed (U) and impeller diameter. It is defined as:

$$\phi = \frac{700Q}{N D^3}$$

Where:  $\phi$  = Flow coefficient, non dimensional

Q = Inlet CFM at actual P, T, ft<sup>3</sup>/min. or ACFM

N = Speed, RPM

D = Impeller Diameter, inches

For centrifugal compressors, the flow coefficient ranges between .01-0.1. The minimum discharge flow coefficient is around .008 - .01. The peak efficiency occurs around mid range 0.04 -.05.

#### Mach Number

Mach number is a ratio of velocity of gas to the velocity of sound in the gas at operating pressure and temperature. The Velocity of Sound in a gas is calculated thus:

$$U_s = \sqrt{K g_c R T}$$

Where:  $U_s$  = Velocity of sound in gas, ft/sec

K = Specific heat ratio  $\frac{Cp}{Cv}$

R = Gas Constant  $\frac{\text{ft-lb}}{^{\circ}\text{R lbm}^{\circ}\text{f}}$

T = Gas Temperature  $^{\circ}\text{R}$

$g_c$  = Gravity Constant,

$$32.2 \frac{\text{lbm ft}}{\text{lbf Sec}^2}$$

For simplicity, the velocity of gas is referred to as the velocity of the impeller at the tip. It should be noted that the absolute velocity of gas leaving the impeller is different from the tip speed.

Mach Number is defined as  $M = \frac{U}{U_s}$

Where  $U$  is the impeller velocity at a given station e.g., a tip or eye.

For turbomachinery, the Mach Number presents the aerodynamic stability of flow. It is mandatory to render the inlet flows subsonic i.e. Mach Numbers less than unity (0.6-0.7) to avoid choking of compressors.

#### AERODYNAMIC PARAMETERS FOR COMPRESSORS SELECTION

O. E. Balje presented a rationalized approach to describe the aerodynamic characteristics of compression machinery. Figure 4 is derived from his article, "A Study on Design Criteria and Matching of Turbomachines, Part A & B.", Journal of Engineering for Power, Jan. 1962,. Whereas, the generalized Specific Diameter and Specific Speed relationship with a superimposed contour of adiabatic efficiency are indicative of the Best Operating Point (BOP) for a given configuration, it should be noted that these are analytical guidelines only. It serves a good check to determine broadly which type of compression the process demands viz. reciprocating, rotary, centrifugal or axial along with their respective regimes of peak efficiency and stable operation. The paper provides an extensive analytical background in the development and limitations of these  $D_s$  vs  $N_s$  curves. The other parameters usually considered are Mach Number (for choking limitations), head and flow coefficients to optimize the compressor selection

It is not the intent of this paper to delve into the details of such analytical aerodynamic treatment. However, for purpose of utilizing Balje's rationalized approach, reference is made to R. F. Neerken's comprehensive paper "Compressor Selection for the Process Industries". In this article, the use of Balje's analysis as a tool to check a given selection is demonstrated through Figure 4.

Table 1 provides a summary of Compressor performance limits.

Following are the basic guidelines in compressor selection.

#### WHEN TO USE POSITIVE DISPLACEMENT COMPRESSOR

- a. Low inlet flow rates below 500 ACFM, below which the centrifugal compressor selection becomes difficult due to smaller wheels and narrow flow passages. Reciprocating compressors usually range up to 3000 ACFM capacity. Note that some manufacturers offer small centrifugal compressors up to 350 ACFM.
- b. When discharge pressures are in excess of 5000 psig.
- c. When the adiabatic head per impeller exceeds

12,000 ft.lbf/lbm and the total head demands more than 10-12 impellers per casing.

- d. For capacity turndowns below 40% at constant speed. Below these flows, centrifugal compressors have limitation due to surge and power loss due to excessive recycle.
- e. When the intended service requires unloaded compressor operation for extended periods with nominal power loss e.g. refrigeration.
- f. In plants where lower initial cost is required.
- g. When higher compression efficiency at partial and rated capacity is required.
- h. When the rotor Specific Speed below 20 and Specific Diameter (1.0-20.0) render the centrifugal and axial compressor selection impractical due to aerodynamic limitations.
- i. When the process has wide fluctuation in gas composition. Centrifugal compressor selection may be impractical if the molecular weight variations demand changes in the impeller characteristics. Such design change can only be affected by changing compressor speed and or internals.
- j. When spare train of compressors is available and 3 years continuous uninterrupted operation is not mandatory.
- k. When two or more separate process streams are to be compressed independently by the same compressor frame. Use multi-throw reciprocating compressor to meet such requirement.

#### WHEN NOT TO USE POSITIVE DISPLACEMENT COMPRESSORS:

- a. Large inlet flows in excess of 3000 ACFM.
- b. Process does not permit cylinder lubricant carryover and non lubricated cylinders are not compatible to process.
- c. When gas flow rates render valve velocities unsatisfactory for valve operation due to lack of momentum.
- d. When process does not permit any pressure fluctuations, even below API permissible values.
- e. When the gas turbine driver RPM are too high in low HP range, say 1000-2000, rendering unrealistic gear ratios, say 5:1 and above. Higher gear ratios impose double reduction, higher pinion velocities and noisier operation.
- f. Space restraints and structural requirements

demand compact machinery with no out-of-balance forces on the foundations, e.g. offshore platforms.

- g. Higher pressure ratios above 5:1 per casing restrain rotary compressor selection due to rotor deflections, low volumetric efficiency and speed limitations.
- h. When larger capacities dictate higher piston velocities above normally accepted limits.
- i. When use of cast-iron is prohibited in the process plant for fire hazard reasons, and the use of steel cylinders and casing is not economical.
- j. When the tip speeds exceed the following criteria:

Helical Screw	MACH NO: 0.3-0.4
Spiral Axial Screw	MACH NO: 0.12
Lobe	MACH NO: 0.05

#### WHEN TO USE CENTRIFUGAL COMPRESSORS

- a. When equipment reliability is prime and no spare train of equipment is allowed.
- b. For Specific Speed range 30-3000, Specific Diameter range (0.3-6.0) and flow coefficient (.01-0.1).

#### WHEN NOT TO USE CENTRIFUGAL COMPRESSORS

- a. Below 500 inlet ACFM. Check design with caution.
- b. When the discharge volume rate reduces to 300-400 ACFM range rendering very narrow impellers and diffuser gas passages difficult to produce.
- c. Capacities in excess of 50,000 ACFM and discharge pressures less than 100 psig which are in axial range. Larger inlet flows demand proportionately larger inlet area (to keep inlet Mach Number low hence bigger impeller eye and the resultant wheel diameter).
- d. Discharge temperatures in excess of 450°F limit centrifugal compressor due to material limitation of "O" rings in the seals and carburizing of the seal oil.
- e. For sour hydrocarbon service containing H<sub>2</sub>S, CO<sub>2</sub>, H<sub>2</sub>O requiring impeller tip speeds in excess of 830-860 ft/sec range.
- f. When the gas is raw and slugs of liquid may carry over to compressor despite knock-out facilities.

#### WHEN TO USE AXIAL COMPRESSORS

Axial flow compressors are generally used in air service e.g.: Fluid Catalytic Cracker (FCC) units, blast furnace, sewage aeration, etc.

An Axial flow compressor is recommended:

- a. When large inlet capacities (50,000) ACFM and above render the centrifugal configuration uneconomically bulky.
- b. When higher compression efficiencies (5-10%) above centrifugal configuration are a deciding factor in driver selection and running costs.

It can be explained that the aerodynamic design of axial blades, straight axial flow path and adjustability of stator guide vanes contribute to axial compressor's efficient operation over a wide capacity range with a constant speed driver.

- c. When the aerodynamic parameters i.e., specific speed and specific diameter fall in (1,000-10,000) and (0.3-2.0) range, respectively. Refer to Figure 4.
- d. When a mechanical seal is not required and nominal compressed air or gas leakage across the labyrinth is acceptable.
- e. When the compression temperatures/casing exceed 450°F, dry labyrinth seals with buffer gas permit compression temperatures up to 700°F, whereas the non-metallic components of mechanical seals such as "O" rings and the seal oil, limit compression temperature to 400-450°F.
- f. When part-load efficiency (below 40% being usual centrifugal compression limitation) is a premium, as multi-stage stator blading can be adjusted with effective capacity control even at constant speed.

Note that after first 5-6 stages, variable stator blading has diminishing effect of volume control as increased density in later stages demands excessively wide angle turning of blades for throttling the flow.

#### WHEN NOT TO USE AXIAL COMPRESSORS

- a. When inlet air/gas stream carry erosive dust particles detrimental to axial blades and guide vanes.
- b. Below 50,000 ACFM at which capacities centrifugal compressors are competitive.
- c. When low initial cost is desired, and no premium is assigned to efficiency.
- d. When side load admissions are required. The compact rotor with aerfoil stages stacked

design is not readily adaptable to intercooler connections or for additional side process loads. Exceptions to this may be found in the custom-built European designs of axial compressors.

#### WHEN TO USE AXIAL-CENTRIFUGAL COMPRESSORS

A combination of axial/centrifugal staging is used:

- a. When for large inlet flows in excess of 50,000 ACFM, higher compression discharge pressures above 150 psig are required.
- b. The centrifugal impellers placed at the tail end of axial compression stages optimize the low flow-high head capability of the centrifugal stage, thereby reducing the overall length of the rotor.

#### FUTURE TRENDS

Compression machinery in any plant plays a key role. Careful aerodynamic, thermodynamic and design considerations are necessary as the reliable and efficient operation of a plant is dependent on how the compressors are selected and operated. In recent years much attention has been directed to improve the compressor reliability through surveillance and efficiency through improved techniques in the fabrication. Our knowledge of the limiting phenomena such as surge, stonewall and rotor dynamics needs further advances such that the control systems and process operations can be adopted to fully utilize the complete range of performance without any precipitating damage. The materials of construction is another area that needs our attention. Materials need to be developed which could provide more confidence in predicting their stress behaviour under corrosion, and cyclic fatigue. Improved machining, assembling and surface finishing techniques for the aerodynamic flow paths in the compression machinery has been recently reported to have improved the compression efficiency to a limited extent, Ref 5. Extensive equipment surveillance with data logging and trending of rotor vibrations and bearing temperatures is rightfully becoming popular in improving the availability of the turbomachinery in critical plant services.

The advent of high pressure reinjection of natural gases for gas lift in oil/gas industry has brought to the fore the requirements of correctly predicting physical properties of gas mixtures throughout compression cycle, improvements in compressor seal designs in containing high pressures and improving knowledge of rotor response.

#### ACKNOWLEDGEMENTS

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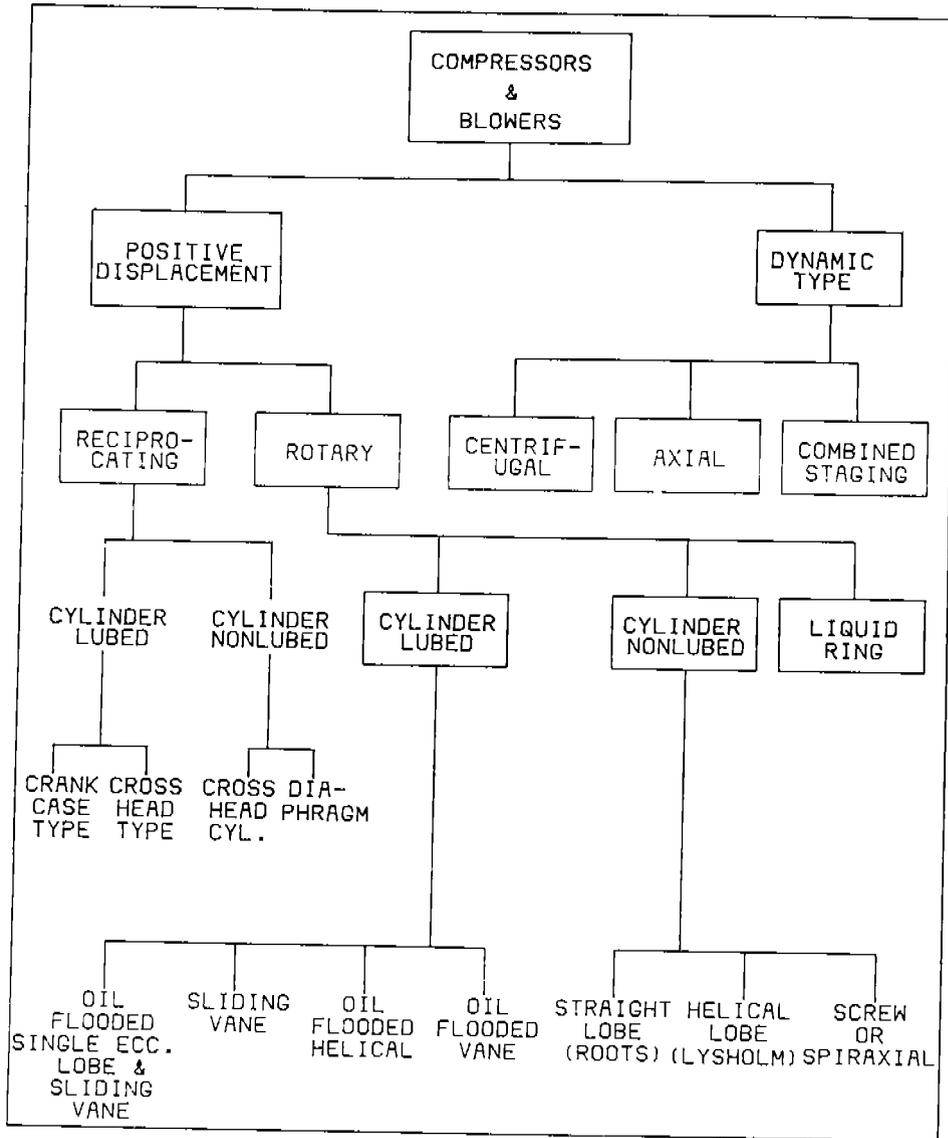


FIG 1: CLASSIFICATION OF COMPRESSORS/BLOWERS

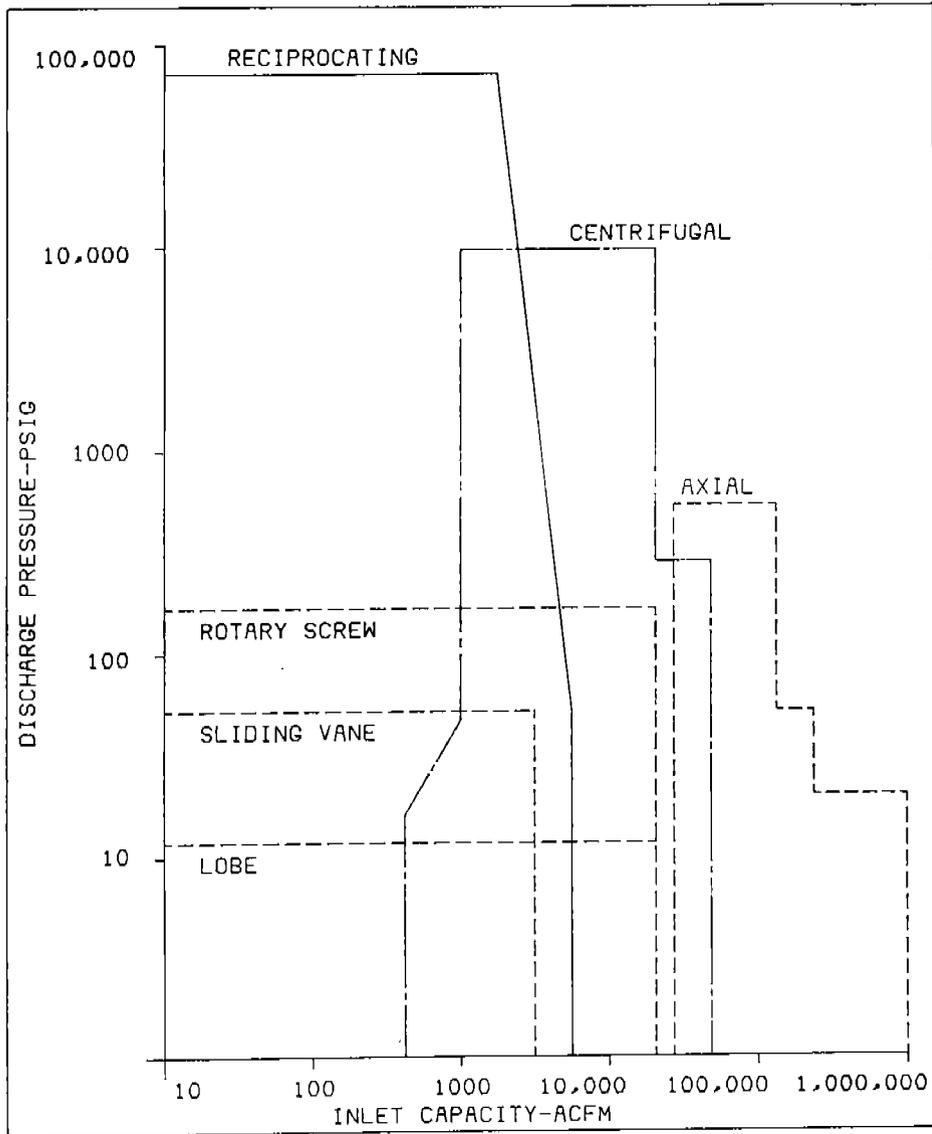


FIG 2: APPROXIMATE INLET CAPACITY V/S DISCHARGE PRESSURE RANGE OF COMPRESSORS

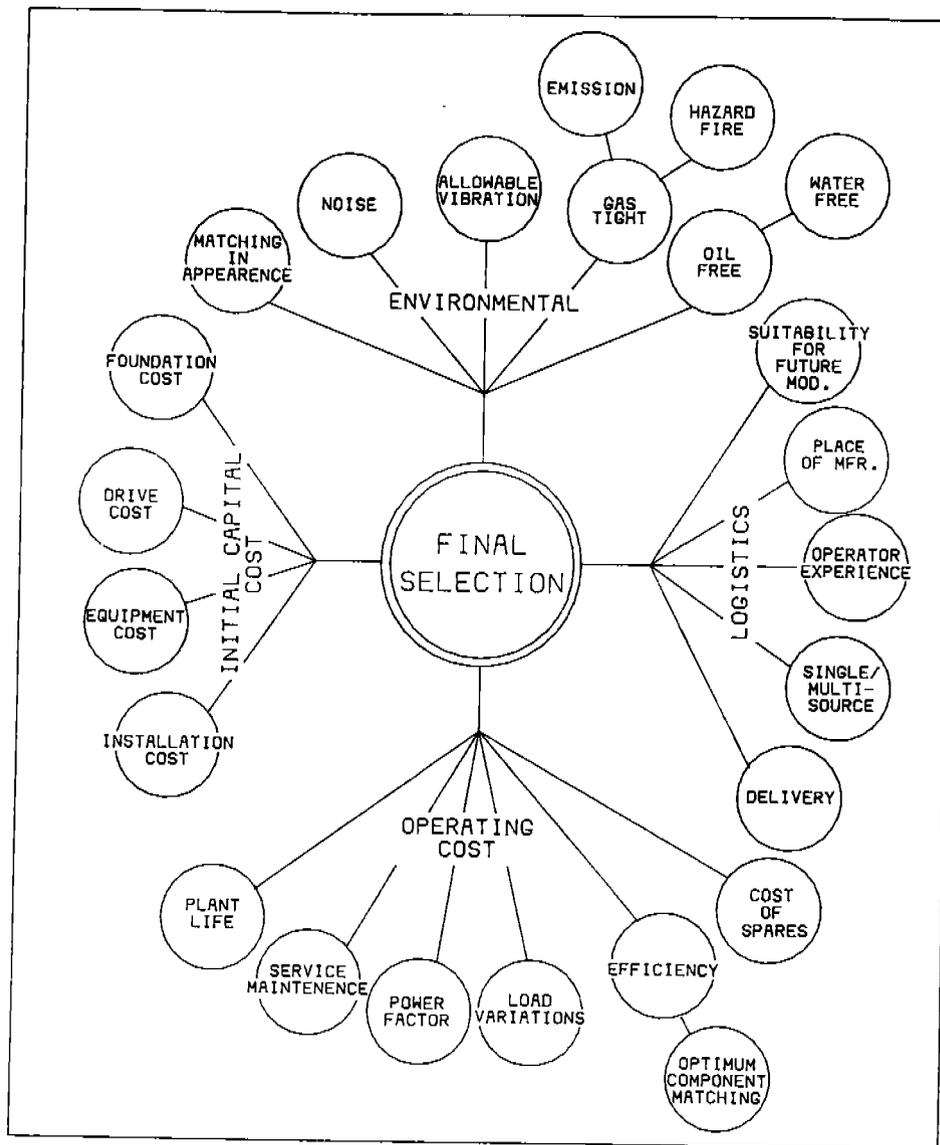


FIG 3: GROUP OF FACTORS INFLUENCING COMPRESSOR SELECTION

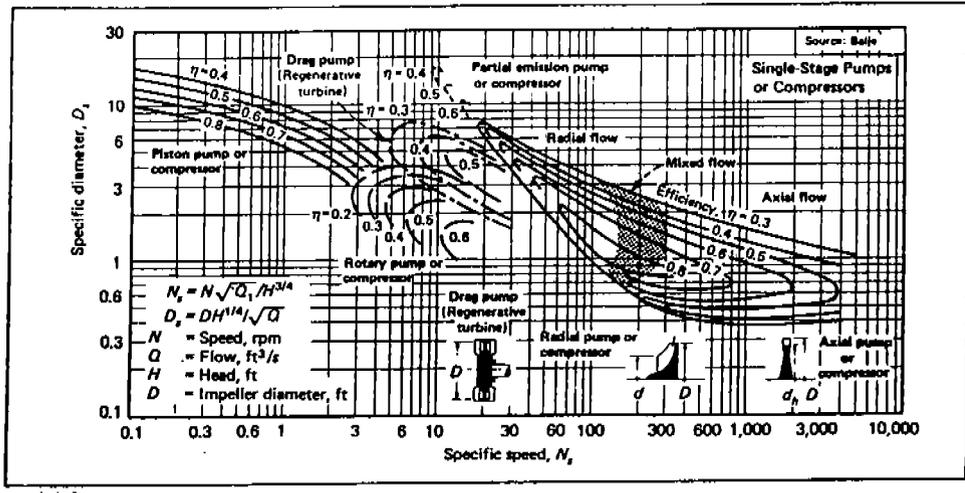


Fig (4) Specific Speed v/s Specific Diameter Range for Reciprocating, Rotary, Centrifugal and Axial Compressors.

TABLE 1 - SUMMARY OF COMPRESSOR LIMITS

COMPRESSOR TYPE	Aerodynamic Parameters <sup>2</sup>			PERFORMANCE LIMITS <sup>1</sup>							
	Spec. Speed N <sub>s</sub>	Range Of Sp. Dia D <sub>s</sub>	Flow Coeff Ø	Min. Flow ACFM	Max. Flow ACFM	Approx. Surge Limit % Rated	Approx. Range Of Pressure Ratio/ Casing	Max. Disch. Temp/ Casing	Typical Adiabatic Efficiency	BHP Max	RPM Range
(A)	0.15-1.5	3-20	N.A. <sup>8</sup>	10	3,000	None	As Req'd <sup>5</sup>	400	0.75-0.85	As Req'd	300-1000
(B)	0.15-1.5	3-20	N.A.	10	3,000	None	As Req'd <sup>5</sup>	350	0.75-0.85	As Req'd	300-1000
(C)	3.0-30	0.3-10	N.A.	10	20,000	None	1-2.0	Varies	.70-.75	500	900-1800
(D)	3.0-30	0.3-10	N.A.	50	20,000	None	2-5.0	Varies	.75-.80	6,000	1500-1500
(E)	3.0-30	0.3-10	N.A.	100	3,000	None	2-5.0	180	.70-.75	6,000	1500-1500
(F)	3.0-30	0.3-10	N.A.	50	2,000	None	2-5.0	350	.70-.75	300	400-1200
(G)	3.0-30	0.3-10	N.A.	50	10,000	None	2-7.0	180	.55-.65	500	1800-3600
(H)	30-3000	0.3-6	0.01-0.1	350	200,000	65%	1.2-20	450	0.70-.75	5,000	3000-11000
(I)	30-3000	0.3-6	0.01-0.1	1,200	16,000	65%	1.2-20	450	0.70-.80	2,500 <sup>11</sup>	3000-40000
(J)	30-300	0.3-6	0.01-0.1	500	190,000	65%	3-7.0 <sup>6</sup>	450	0.77-.80	35,000	3000-20000
(K)	30-300	0.3-6	0.01-0.1	1,200	130,000	60%	3-7.0	450	0.77-.80	35,000	3000-20000
(L)	1000-10,000	0.3-2	0.01-0.1	50,000	1,000,000	60%	1.2-7	700 <sup>7</sup>	0.75-0.85	As Req'd <sup>9,11</sup>	3000-6000

COMPRESSOR TYPE :

Reciprocating Piston  
 (A) Non-Lubricated  
 (B) Lubricated

(C) Lobe - Roots Type  
 (D) Screw - Dry Helical  
 (E) Screw - Flooded  
 (F) Sliding Vane  
 (G) Liquid Ring

Centrifugal  
 (H) Single Stage Overhung<sup>3</sup>  
 (I) Multi-Poster API-672 Type<sup>4</sup>  
 (J) Multistage Horiz. Split  
 (K) Multistage Vertical Split

Notes:

(L) Axial

- (1) These limits are typical only. Significant variations may be observed between various vendors.
- (2) Refer to text of the paper for definitions.
- (3) Low pressure air service considered. Vertical split fabricated custom built design are also available.
- (4) Specially modified multi-stage multi-poster design for process service exists.
- (5) Special application in hyper class have reached 40-50,000 psig.
- (6) Discharge pressures in excess of 10,000 psig have been attained in gas lift compression service.
- (7) With dry labyrinth and buffer gas.
- (8) N.A. - Not applicable. This parameter exclusively refers to dynamic compressors.
- (9) There exists a capacity overlap between centrifugals and axials. Smaller axials handling 30,000-50,000 ACFM may be found in industry.
- (10) In cryogenic service inlet temperature may be as low as -250°F. API 618 limits discharge temperature to 350°F if pressures exceed 300 psig.
- (11) Larger special applications exceeding 50,000 BHP per train exist in centrifugal/axial configurations.