

SELECTION OF TURBOMACHINERY—CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

One of the primary functions of a the rotating machinery engineer is the application and selection of turbomachinery. During the early stages of a project, the process engineer needs to know the performance of turbomachinery, the horsepower, etc., in order to complete the arrangement of equipment in the process. As the project progresses, selection of the turbomachinery becomes focused on materials of construction, rotor design, bearings, and seals.

The configuration of a centrifugal compressor employing radial flow type impellers in a typical refinery process is presented, along with the methods used to select, compute estimated performance and mechanical design parameters, including materials of construction. A discussion is included on the non-dimensional quantities of aerodynamic (rotating) elements; specific speed, suction specific speed, and flow coefficient.

The compressor selection discussions are directed primarily to radial flow type compressors that are commonly used in hydrogen rich services for a refinery unit.

Here, "selection of turbomachinery" is considered to be computation (estimation) of thermodynamic performance, metallurgy and rotor arrangement (shaft, bearings, seals, etc.). The selection process occurs before the machinery is purchased from the OEM. On many projects however, that are so called front end engineering only some of the details of hardware design are not considered.

INTRODUCTION

Selection of a centrifugal compressor starts with performance calculations. After basic machine performance is determined, the mechanical construction is addressed. The primary areas of concern are metallurgy, shaft sealing and rotordynamics.

Rotordynamics analysis (RDA) of turbomachinery designs should be made during selection. A lateral critical speed study includes undamped critical speed analysis, plot of the undamped critical speeds as a function of stiffness, synchronous unbalance response analysis, and stability analysis. The stability analysis is concerned with all calculated subharmonic, self-excited vibrations of the rotor. Oil whirl is one such common example of subharmonic instability of concern in design of the rotor bearing support of turbomachinery. Other instabilities result from disturbing/destabilizing forces from aerodynamic sources or shaft seal design.

Rotordynamics of a centrifugal compressor with oil film seals are of particular interest because of the possibility of the seals to unload the journal bearings, and allow the destabilizing forces to dominate a rotor's behavior. Logarithmic decrement and its relationship to rotor stability is also discussed.

SELECTING TURBOMACHINERY

Define process configuration

A complete definition of the conditions of service is required. Many times the alternate operating conditions are not known upfront and when finally identified can drastically effect the selection arrangement, materials, etc. Runin is another "operating case" that can have significantly different conditions than the so called normal operation. For example if the normal condition of service has a suction pressure of 800 psia and gas molecular weight of seven the head required could be 50,000 ft. Startup on nitrogen, which is commonly performed, would require the compressor to pump a gas at a suction pressure of 100 psia and with a molecular weight of 28. The difference is would result a much different head, power requirement, discharge temperature, etc.

Studies and Optimizations

The rotating equipment engineer is involved in calculating turbomachinery horsepowers and other performance parameters during the process design work. Questions are also addressed as to the suitability of proposed arrangements, operating conditions, temperatures, etc.

Tradeoff studies are performed on drivers for the compressor; motor vs steam turbine or backpressure turbine vs condensing turbine depending on customer utility costs. Condensing turbines can exhaust to an air cooled condenser or water cooled (surface) condenser. A study is performed for cost and steam rate of air cooling vs water cooling. Accurate steam rates of the steam turbine driver are required to perform the study.

The following calculation procedures can be used to estimate turbomachinery performance.

Centrifugal Compressor Selection Calculations

A radial split barrel type centrifugal compressor is commonly used to compress hydrogen rich/hydrocarbon gas mixtures in refinery services. Normally the compressor is arranged with five to seven impellers (stages) in a straight through design. The radial split design is selected because of the high pressure and low molecular weight of the gas mixture compressed (approximately five to six).

The shaft seals of the centrifugal compressor are of particular concern. Most manufacturers use an oil film type of shaft seal for containing the gas in the compressor casing. The seals can be a proprietary design type or so-called bushing type. The number of seal rings with the bushing type is determined by the operating pressures, higher pressures requiring more sealing rings.

These type seals are analyzed carefully for rotordynamic stability. There have been installations with oil film seals used where rotor instability problems developed. To avoid this problem, consideration must be given early in the project before the rotor and seal design is final.

Dry gas seals are becoming popular with some users of turbomachinery for both new installations and retrofits. The popularity is due to several factors. For one, the seal oil system with its pumps, filters, coolers, etc., can be not required. For another, the operating cost, steam usage, etc., of the seal oil pumps is saved. Space is saved, seal oil leakage is eliminated, and finally, the disposal of contaminated seal oil is also eliminated. However, the rotordynamic analysis and stability analysis of the turbomachine must take into consideration the effects of the dry gas seal.

The rotor stability analysis assumes that the oil film seals are "floating" or locked up. The effects on stability is then neglected when the seals are so-called centered and floating. At this point, they provide friction damping and contribute significantly to shaft stability, since they are located inboard of the bearings. When gas seals replace the oil film seals, this source of damping is removed. Another source of damping needs to be added to replace this friction damping. If not, the shaft will tend to be unstable. Dry gas seals do not by themselves provide very much damping. These factors need to be considered in the rotordynamic analysis of the compressor and in the decision of the engineer or user to utilize an oil film seal or dry gas seal.

Gas Analysis Calculations

The compressor selection methodology is a "general polytropic head method." This method is a simplification, but was selected because it is accurate and can be applied with hand calculations (calculator), or can be programmed with a digital computer.

A more accurate method would be to compute gas properties for each impeller or stage and then compute the performance for each individual stage or impeller.

The mollier method is another common compressor method of calculation, but it requires the use of a mollier diagram that is specific for the gas mixture. If the gas that is to be compressed is a single component gas, the mollier diagram is probably readily available. However, for mixtures, which is usually the case, the mollier diagram must be constructed using a digital computer program.

Process engineering provides the gas mixture, basic properties and preliminary operating conditions early in the project. It is then necessary to compute the remaining fluid properties and performance including head, horsepower, and speed from a gas analysis, which is usually given as a listing of components and a percentage of each.

The given gas composition is used to calculate the mixture molecular weight, the ratio of specific heats, k , pseudocritical temperature, and pseudocritical pressure.

$$MW_{(mixture)} = \sum (\text{mol } \% \times MW) \quad (1)$$

$$C_{P(mixture)} = \sum (\text{mol } \% \times C_p) \quad (2)$$

$$T_{C(mixture)} = \sum (\text{mol } \% \times T_c) \quad (3)$$

$$P_{C(mixture)} = \sum (\text{mol } \% \times P_c) \quad (4)$$

The procedure is demonstrated in Table 1.

Using the value of specific heat, C_p calculated above, the ratio of specific heats, K , must be calculated by using Equation (5).

$$K = \frac{C_p}{C_v} = \frac{MC_p}{MC_p - 1.986} \quad (5)$$

The ratio of specific heats, K is computed at inlet and discharge conditions. For estimating performance, however, an average value is computed at the average compressor operating temperature.

This method is simplified for illustrative purposes. A more accurate calculation of specific heats and compressibility can be obtained by using various equations of state such as the Benedict Webb Rubin (BWR) Equation of State [1] or the Redlich Kwong Equation of State [2]. The API Technical Data Book [3] also provides sophisticated computer procedures such as the Lee Kessler Equation of State [4] for determining the specific heats and compressibility of gas mixtures that employ the Pitzer acentric factor.

Each equation of state is valid only for a specific range of temperature and pressure. The BWR equation of state [1] has been applied for hydrogen rich gases at pressures less than 3000 psi and temperatures less than 300°F. The calculation procedure uses eight empirical constants that are determined experimentally for each pure gas. Correlation of the constants for an equation of state takes considerable computation time. It should, therefore, be solved with a digital computer.

If the gas mixture is given in "weight percent" then the constituents need to be converted to a molar percentage basis. The procedure for the correlation of gas data starting with the weight flow composition is demonstrated in Table 2.

The gas mixture compressibility can also be obtained from generalized compressibility charts. For these charts, the reduced pressure, P_r , and reduced temperature, T_r , must be calculated. Equations 6 and 7 are used to compute the reduced temperature and pressure of the gas mixture.

$$T_r = \frac{T, \text{ }^\circ\text{R}}{T_c, \text{ }^\circ\text{R}} \quad (6)$$

$$P_r = \frac{P, \text{ psia}}{P_c, \text{ psia}} \quad (7)$$

The compressibility, Z , is dimensionless value, represents the deviation of the gas from an "ideal gas," and is calculated at both the suction and discharge conditions. The value of this dimensionless factor can be close to unity at low pressures, but can be much higher than unity for higher pressures (pressures above

Table 1. Gas Analysis Calculations Procedure.

Gas Constituent	Molar % Ea Constituent	Molecular Weight, MW	[Mol % / 100] x MW	Specific Heat C _p @ 150°F Btu/Lb-Mol°F	[Mol % / 100] x C _p @ 150°F Btu / Lb-Mol°F	Critical Pressure P _c , psia	[Mol % / 100] x Critical Pressure, P _c , psia	Critical Temperature T _c , °R	[Mol % / 100] x Critical Temperature, T _c , °R
Gas #1	Mol % _{GAS 1}	MW _{GAS 1}	Mol % x MW _{GAS 1}	C _p GAS 1	Mol % x C _p GAS 1	P _c GAS 1	Mol % x P _c GAS 1	T _c GAS 1	Mol % x T _c GAS 1
Gas #2	Mol % _{GAS 2}	MW _{GAS 2}	Mol % x MW _{GAS 2}	C _p GAS 2	Mol % x C _p GAS 2	P _c GAS 2	Mol % x P _c GAS 2	T _c GAS 2	Mol % x T _c GAS 2
Gas #3	Mol % _{GAS 3}	MW _{GAS 3}	Mol % x MW _{GAS 3}	C _p GAS 3	Mol % x C _p GAS 3	P _c GAS 3	Mol % x P _c GAS 3	T _c GAS 3	Mol % x T _c GAS 3
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.
.
TOTAL			MW		C _p		P _c		T _c

The specific heat of the gas mixture is computed @ 150°F, approximate average operating temperature of the recycle gas compressor.

Table 2. Mass Analysis Calculation Procedure.

Gas Constituent	(A) Weight %	(B) Weight Flow Lb / hr	(C) Molecular Weight, MW	(D) Lb - Moles / Hr	(E) Mol %
Gas #1	Wt % (GAS 1)	(A) ₁ x Weight	MW (GAS 1)	(B) / (C) (GAS 1)	(D) / total moles/hr
Gas #2	Wt % (GAS 2)	(A) ₂ x Weight	MW (GAS 2)	(B) / (C) (GAS 2)	(D) / total moles/hr
Gas #3	Wt % (GAS 3)	(A) ₃ x Weight	MW (GAS 3)	(B) / (C) (GAS 3)	(D) / total moles/hr
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.
TOTAL	Weight		MW	Total moles / hr	100.00

approximately 500 psi), and it does not vary linearly. For this reason, it is necessary to consider an accurate estimate of compressibility.

Inlet Volume Flow, Standard Volume Flow, and Mass Flow (Capacity)

The compressor capacity is usually given in one or more different ways:

- mass flow, lb/hr or lb/min
- volumetric flow, at the standard conditions of temperature and pressure, 60°F and 14.697 psia. The normal standard volume flow is in mmscfd.

If the weight flow is known, then volumetric flow can be calculated as follows:

$$m = (n) (MW_{mixture}) \tag{8}$$

$$Q \text{ (SCFM)} = 379.46 (n) \text{ scfm} \tag{9}$$

The flow at inlet conditions of temperature and pressure must be calculated in order to correctly select a centrifugal compressor. The flow at inlet conditions is calculated as follows:

$$v_s = z_s \left[\frac{1545}{(MW_{mixture})} \right] \left[\frac{T_s, \text{ }^\circ\text{R}}{(144) (P_s)} \right] \tag{10}$$

$$ICFM = Q_s = (m) (v_s) \tag{11}$$

The inlet flow can also be calculated directly from standard flow rate by Equations (12) or (13).

$$Q_s = ICFM = \left[\frac{MMSCFD \times 10^6}{60 \times 24} \right] \left[\frac{14.7}{P_s} \right] \left[\frac{T_s}{520} \right] \left[\frac{z_s}{1.0} \right] \tag{12}$$

$$Q_s = ICFM = SCFM \left[\frac{14.7}{P_s} \right] \left[\frac{T_s}{520^\circ\text{F}} \right] z_s \tag{13}$$

Compressor Head

Compressor head can be expressed as adiabatic or polytropic. Herein, the “compressor process” is considered polytropic rather than adiabatic. An adiabatic process by definition is one in which the heat transfer is zero.

The equation for polytropic head is given by:

$$H_{poly} = \left[\frac{(Z_s + Z_d)}{2} \right] \left[\frac{1545}{MW} \right] (T_s) \left[\frac{R_c^{[(n-1)/n]} - 1}{(n-1) / n} \right] \tag{14}$$

and:

$$\left[\frac{(n - 1)}{n} \right] = \left[\frac{(k - 1)}{(k) (E_{poly})} \right] \tag{15}$$

The overall compressor polytropic efficiency is required to compute the head and gas horsepower. This efficiency includes all losses except bearing and seal losses (the mechanical losses). The losses that determine the (thermodynamic) efficiency in a centrifugal compressor include (disk) friction, balance piston leakage, slip, diffuser, and recirculation. Mechanical losses are not included. Since compressor efficiency is a combination of impeller and diffuser efficiency, the diffuser is considered because of the influence it has on the performance of the impeller.

Polytropic head divided by the head per stage will determine the number of stages (impellers) that the compressor will have. As a general rule of thumb most compressor manufacturers limit the head per stage to no more than 10,000 to 12,000 feet. A conservative value would be 9,000 ft per stage. Since the rotating speed is determined by the head per stage, a lower head per stage will give a slower rotating speed, keeping the tip speed below the acoustic velocity of the gas. Tip speeds for a compressor like this are limited to 700 to 850 fps.

Total compressor head is divided by 9,000 ft of head per impeller to obtain the number of impellers the compressor will

have. That number is then rounded off (up or down) to get an integer number of impellers or stages for the centrifugal compressor. Limiting the head and rotating speed will result in an impeller tip speed below the acoustic velocity of the gas. A "straight through" compressor arrangement is limited to eight or nine impellers maximum because of rotordynamics. More than nine impellers increases shaft length (bearing span) and decreases responsive criticals which could encroach on the API separation margin [6].

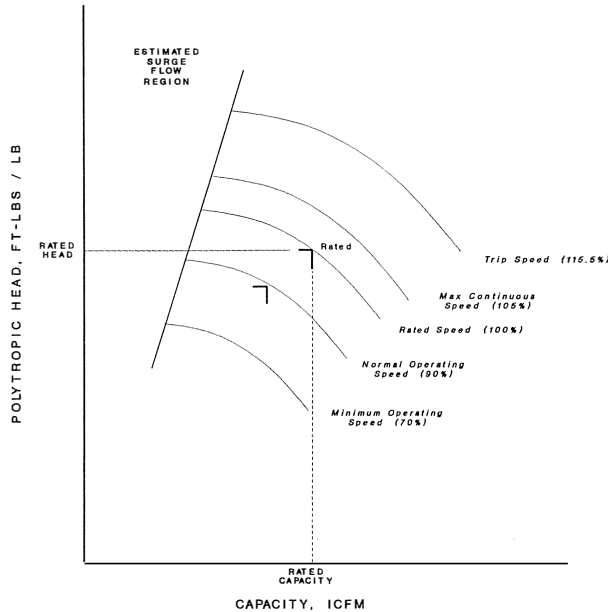


Figure 1. Predicted Performance Curve.

Compressor Horsepower

The polytropic head is used to calculate gas horsepower and then bearing and seal losses to obtain the total brake horsepower of the compressor.

$$\text{GHP} = \frac{(m) \times (H_{\text{poly}})}{33,000 \times (E_{\text{poly}})} \quad (16)$$

$$\text{Compressor BHP} = \text{compressor GHP} + \text{bearing losses} + \text{seal losses} \quad (17)$$

Bearing and seal losses can be estimated separately from Equations (18) and (19) [5].

$$\text{Bearing losses} = B_L N / 1000)^2 \quad (18)$$

$$\text{Seal losses} = S_L (N / 1000)^2 \quad (19)$$

Bearing and seal loss calculations relate compressor frame size (which is a function of capacity in actual cfm) and speed to power. Mechanical losses are not a function or percentage of gas horsepower and must be calculated. Many bearing manufacturers publish data, curves and the like relating mechanical losses to speed, clearance, and oil film thickness for various types of journal and thrust bearings. If such design data is available, the engineer would need to estimate the shaft size to utilize the information.

Table 3. Estimated Bearing and Seal Loss Coefficients.

Capacity, ICFM	Loss Coefficients	
	Bearings, B_L	Seals, S_L
1250 - 4225	0.174	0.093
2100 - 6500	0.321	0.174
2750 - 8500	0.603	0.321
4250 - 12,700	1.110	0.603
6350 - 21,000	2.077	1.110
8500 - 31,000	3.887	2.077
12,700 - 42,300	7.238	3.887
17,000 - 53,000	13.401	7.238

Source of data: Borsig Technical Book, Berlin, 6th Edition. (Ref. # 5)
Data was originally computed for kW loss, correlated for hp loss.
 R_S loss coefficient is applicable for oil film type seals only.

In addition, mechanical losses will vary with the type of journal bearing that is selected. Plain sleeve bearings and tilting pad journal bearings will have different mechanical losses.

Compressor Discharge Temperature

The polytropic discharge temperature, T_d , °R, is calculated as follows:

$$T_{d(\text{poly})} = [T_s, \text{°R}] \left[R_c^{(n-1)/n} \right] \text{°R} \quad (20)$$

The maximum allowable discharge temperature for a recycle gas compressor is generally limited to 350°F. The discharge temperature limitation is dictated by small internal clearances and the materials of construction. The compressor discharge temperature limit is an important value for a centrifugal compressor. Startup procedures may require circulating nitrogen at a suction pressure of approximately 100 psig. During this operation, the pressure ratio will be significantly higher than on hydrogen rich gas and, consequently, so will the discharge temperature. Without proper consideration for discharge temperature the startup operation could be limited to a lower speed and flow.

Impeller Diameter and Tip Speed

The impeller tip speed, U , is calculated from Equation (21):

$$U = 2\pi \left[\frac{D}{2} \right] \left[\frac{N}{60 \times 12} \right] \text{fps} \quad (21)$$

An alternate method to determine tip speed, indirectly, relates the impeller tip speed to head coefficient.

$$U = \left[\frac{(H_{\text{poly}}) (g)}{(N_{st}) (\mu)} \right]^{1/2} \text{fps} \quad (22)$$

The impeller tip speed is then compared to the acoustic velocity of the gas and then checked for stress limitations. Specifications should limit stresses to no more than 60 percent of impeller material yield at the maximum continuous speed (MCS). If the stresses are in fact higher than allowable, then the selection should be modified.

Acoustic Velocity and Mach Number

The acoustic velocity, V_a , is the speed of sound in the gas mixture and is calculated at the suction temperature of the mixture using Equation (23).

$$\text{Acoustic velocity @ inlet } V_a, \text{ ft/sec} = \left[k g R T_s Z_s \right]^{1/2} \quad (23)$$

Mach number, MN, relates the tip speed of the impeller at maximum continuous speed to the acoustic velocity.

$$MN = \frac{U}{V_a} \quad (24)$$

The Mach number for a hydrogen rich compressor is very low, due to the high acoustic velocity of the gas mixture. An acoustic velocity of 3000 fps (which is approximately the value of the acoustic velocity for a three or four mole weight hydrogen rich gas mixture), would result in a Mach number of approximately 0.30, using an impeller tip speed of 900 fps.

Nondimensional Analysis, Flow Coefficient, Head Coefficient

Centrifugal compressor performance can also be expressed in non-dimensional form. Specific speed has been described as a correlation of head, speed, and flow to a classification of applicable impeller geometry or types. The correlation is applicable to all types of turbomachines. Balje [7] relates specific speed, N_s , to specific diameter, D_s (Figure 2).

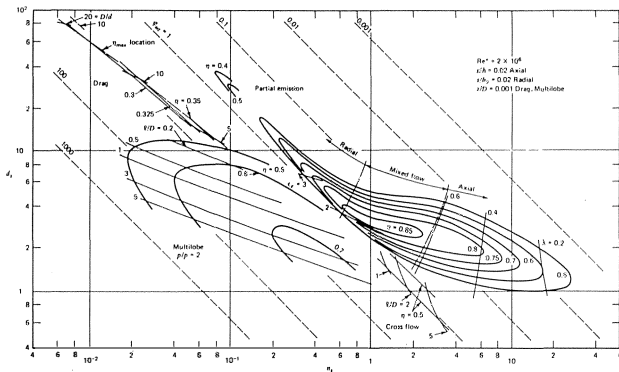


Figure 2. $N_s D_s$ Diagram for Single Stage Compressors.

Specific speed, N_s , and specific diameter, D_s , are calculated by Equations (25) and (26).

$$N_s = \frac{\left[N \right] \left[Q \right]^{1/2}}{\left[H \right]^{(0.75)}} \quad (25)$$

Capacity, Q, is in cubic feet per second, cfs.

$$D_s = \frac{\left[D \right] \left[H \right]^{1/4}}{\left[Q \right]^{1/2}} \quad (26)$$

The D_s vs N_s diagram relates various turbomachinery impeller geometry configuration to stage efficiency. A recycle gas

compressor will have a low flow coefficient with backward leaning impeller blades.

Head coefficient and flow coefficient are also dimensionless values that are calculated by Equations (27) and (28).

$$\text{Head Coefficient, } \mu = \frac{(H_{sig}) (g)}{U^2} \quad (27)$$

The head coefficient, μ , relates the specific work of compression to the specific kinetic energy of the gas. It is therefore the dimensionless specific work of compression for a stage of compression.

$$\text{Flow coefficient, } \Phi = \frac{\left[700.3 \right] \left[Q_s \right]}{\left[N \right] \left[D \right]^3} \quad (28)$$

The dimensionless flow coefficient, Φ , relates the velocity of the gas to the speed of the impeller. This “gas velocity” is determined by means of the volume flow entering the impeller surface. The flow coefficient is, therefore, considered as a dimensionless suction volume flow.

Similarly, the head coefficient relates head to the tip speed of the impeller.

Efficiency is also a function of the inlet flow coefficient. As the flow coefficient decreases, there is a decrease in impeller efficiency. Similarly, as the flow coefficient is increased, the compressor operating point moves to the right toward choke and impeller efficiency also decreases.

Relationships between efficiency and flow coefficient, as well as between the head coefficient and flow coefficient for a given stage (and the compressor) can be plotted as dimensionless performance maps. A plot of μ and E_{poly} vs Φ is provided in Figure 3. It is helpful to apply curve fitting principles and model these curves for use in predicting a compressor’s performance.

Impeller geometry and flow coefficient is related in Figure 3. The figures from Balje [7] are shown to identify impeller geometry as a function of specific speed, N_s , while Figure 3 is used to relate efficiency and impeller geometry against flow coefficient. It is a good check for the type of selection utilized for the application. The lower flow coefficients correspond to shrouded, two dimensional impeller geometry with backward leaning blades. This geometry is the type utilized for the low flow requirements of high pressure, hydrogen rich centrifugal compressors. Conversely, this figure shows that an axial flow design would be utilized for an application with a high flow coefficient.

Dimensionless curves are useful for predicting the performance at alternate operating conditions. For example, the user may wish to evaluate performance if the unit had to process an alternate feed. The dimensionless performance curves will be useful to estimate a new discharge pressure or capacity based on the new or alternate inlet conditions. Once nondimensionalized, the curve will depict a compressor’s performance without regard to a particular set of operating conditions.

Mechanical Design

Casing design

A high pressure, hydrogen rich centrifugal compressor in refinery service requires a double case with vertically split outer casing and cast inner casing. Centrifugal compressors with a very high discharge pressure can be fabricated with a seal ring on the outer head which provides greater strength.

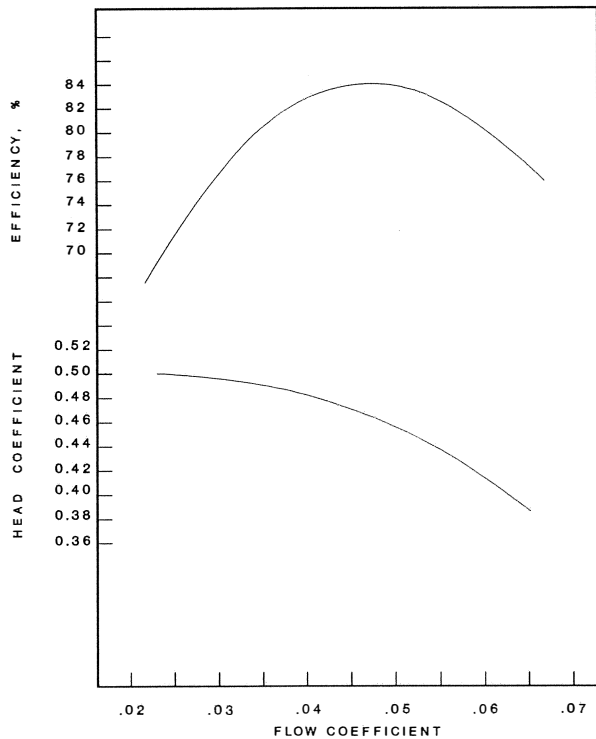


Figure 3. Dimensionless Performance Curve—Centrifugal Compressor.

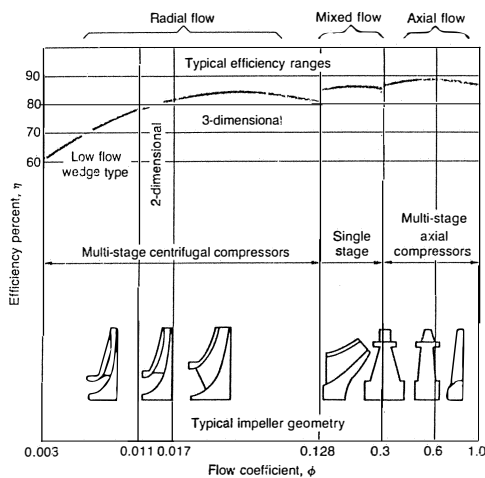


Figure 4. Impeller Geometry Vs Flow Coefficient.

Metallurgy

Selection of materials of construction is crucial in refinery applications due to the common presence of sulfur compounds, and hydrogensulfide. NACE standard MR-01-90, "Sulfide Stress Cracking Resistant Metallic Materials for Oil Field Equipment" [8] is applied for that reason. Materials selected have a yield stress limit of 90,000 psi for carbon steel. Stainless steels such as precipitation hardened martensenic stainless is limited to a lower yield stress.

Typical materials of construction for major components of a centrifugal compressor in refinery (hydrogen rich/hydrocarbon) service are tabulated in Table 4. Parts such as the impeller, shaft seal parts are critical, since they are exposed to the harsh process

Table 4. Typical Materials of Construction – Centrifugal Compressor.

Refinery Service, (Hydrogen / Hydrocarbon Gas Mixture)		
Component	Material (s)	Comment(s)
Casing	A266, forged steel, Class 2	
Diaphragms	ASTM A48 cast iron	
Shaft	AISI 4140 forged steel	90,000 psi material yield limit
Impellers	Carbon steel	90,000 psi material yield limit
Labyrinth seals	Aluminum	
Shaft sleeves	AISI 410 ss	

environment. For carbon steels, NACE limits the material yield to 90,000 psi and hardness to R_c 22 maximum.

Journal Bearings

Turbomachinery bearing designs are determined on load carrying ability and also on rotordynamic considerations. A high speed centrifugal gas compressor is commonly manufactured with tilting pad bearings.

Tilting pad bearings are chosen because they provide approximately twice the damping characteristics as compared to conventional hydrodynamic sleeve bearings, which is effective in reducing or eliminating oil whirl effects [9]. Stiffness in all radial planes, controlled clearance, inherent self-alignment capability and characteristics are easily modified.

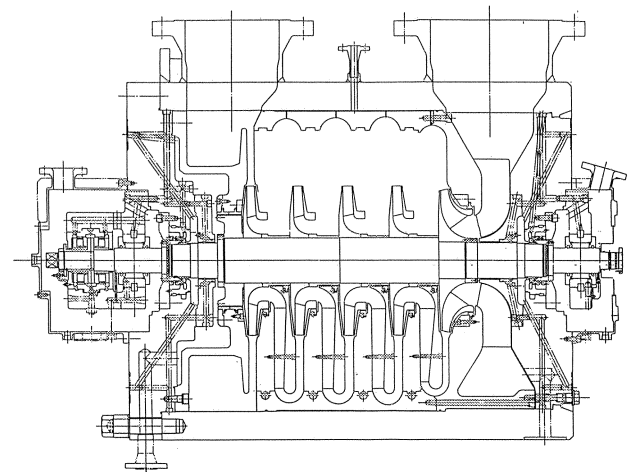


Figure 5. Cross Section of Centrifugal Compressor—Vertical Split Design.

Seals

The centrifugal gas compressor uses mechanical or oil film type seals for containing the process gas in the compressor casing. (Gas seals are another choice that is growing in popularity recently.) The common seal selection in refinery processes however is the oil film seal. There are a variety of designs offered by the compressor manufacturers today. Most use a version of the bushing design where high pressure oil fills a very small clearance between the shaft sleeve and the seal ring or rings, as the case may be. Other proprietary designs use a similar principle.

Rotordynamics, General

During turbomachinery selection, it is common to conduct a preliminary rotordynamics study. In order to avoid field prob-

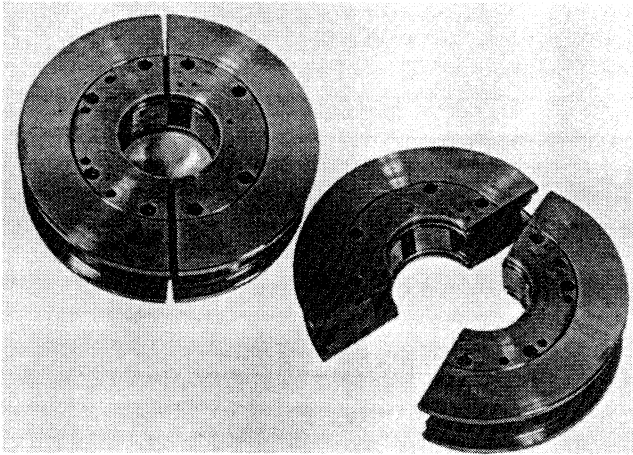


Figure 6. Tilting Pad Journal Bearing.

lems, rotor instability should be addressed in the turbomachinery selection process early in the project.

Critical speed studies include the following elements.

- undamped critical speed analysis including the critical speed map.
- a plot of rotor mode shape for each resonant speed.
- calculated unbalance rotor response analysis.
- a rotor dynamic/bearing stability analysis.
- plot of log decrement vs rotor speed.
- calculation of the amplification factor.

Undamped critical speed analysis—The undamped critical speed analysis involves calculation of the undamped natural frequencies. Programs typically use the transfer matrix analytical techniques developed by Prohl [10]. Definition of the rotor configuration is required, including shaft dimension, location of bearings and seal parts, and all loads such as impellers, couplings, balance drum and the thrust collar.

During equipment selection, final (detailed) rotor data and geometry is not always available, so it is estimated by the manufacturer. The critical speeds calculated are “placed” outside the operating speed range. If a suitable margin between a critical and the speed range cannot be achieved, shaft geometry (or journal bearing selection) can be modified to change the responsive criticals. This issue is crucial, since the buyer should be fully aware of the sensitivity of his turbomachine before he buys it.

A *critical speed map* is generated for frequencies in the range from zero to at least 25 percent above the maximum operating speed of the turbomachine. This process will identify the machine’s natural frequencies and the corresponding mode shapes.

The *critical speed map* referred to is a plot of critical speeds vs support stiffness. The undamped bearing support stiffness vs speed is plotted on this map from minimum to maximum bearing clearance. Intersections of the undamped bearing support stiffness curves and undamped critical speed curves are a prediction of the system’s critical speeds. The *actual* system critical speeds are determined by a synchronous response analysis. An example of a critical speed map is shown in Figure 7. This map was generated for a high pressure recycle gas compressor.

Unbalance Rotor Response Analysis (Synchronous Response Analysis)

Bearing and rotor system reaction to unbalance is of interest to the compressor designer. The responsive, synchronous modes

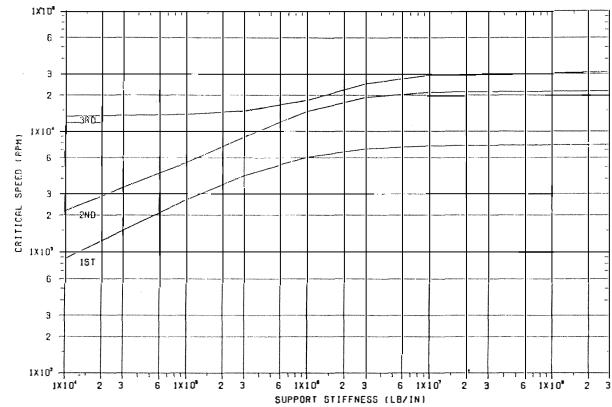


Figure 7. Undamped Critical Speed Map.

will be the critical speeds to be avoided in any operating speed range.

To calculate the synchronous response, the engineer must know rotor bearing system geometry, bearing dynamic coefficients as a function of speed, support stiffness and damping, plus the location and value of rotor unbalance.

A synchronous response analysis, if conducted, will provide the unbalance response amplitude vs speed for specific locations of interest, major axis amplitude and phase angle vs rotor speed, and the bearing force vs speed for bearings.

Rotordynamic/Bearing Stability Analysis

The rotor bearing stability analysis portion of the rotordynamics study is conducted to ascertain the stability of the rotor system—the resistance of the system to destabilizing forces. Rotor instabilities identified in various literature are synchronous (bearing related), self-excited, and forced, nonsynchronous. The destabilizing sources commonly encountered in high speed turbomachinery are:

- Hydrodynamic bearings (oil whirl)
- Fluid ring seals (oil seals)
- Labyrinth seals
- Turbomachinery aerodynamic cross coupling
- Tip clearance excitation
- Impeller-diffuser interaction forces
- friction rubbing of internal parts

Log Decrement

Logarithmic decrement of the response gives an indication of rotor system stability, and its ability to suppress destabilizing forces. Lund [12] states that a log decrement of 1.0 or higher indicates a well damped mode. The higher the value of the logarithmic decrement, the more stable the rotor system. A low or negative value indicates an unstable rotor. The transition between a stable rotor condition and an unstable condition occurs between -0.1 and +0.3 [13].

Log decrement can be calculated by Equation (29)

$$\delta = \frac{-2\pi\Omega}{N_d} \quad (29)$$

where Ω is the damping exponent, sometimes called the growth factor (sec^{-1}), and is the real part of the eigenvalue analysis resultant. N_d is the damped natural frequency.

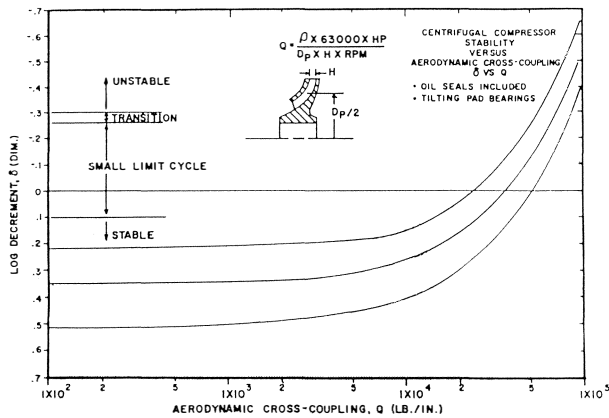


Figure 8. Unbalance Response.

In the event of a log value of δ for a particular mode, one of the methods used to increase the stability of a rotor is the use of tilting pad journal bearings. The damping forces of tilting pad bearings are significantly higher than for plain sleeve bearings. The user or compressor manufacturer will often select tilt pad journal bearings, even though they generally cost more than sleeve bearings, in order to avoid any rotor instability especially for a higher speed turbomachine.

The preliminary rotordynamics reports should include the above analyses.

Rotordynamics—Hydrodynamic Effects of Oil Seals

It was mentioned earlier that oil film shaft seals need to be analyzed for rotordynamics effect. Analysis of rotor vibration in fluid film bearings has become more and more difficult as operating speeds of turbomachinery increase. One of the machinery components that has contributed to this condition is the shaft seal.

High pressure centrifugal compressor seals are usually a close clearance (oil film) bushing design. The seals, located just inboard of the journal bearings, can have two or more seal rings, depending on the suction pressure of the compressor. When the oil bushings act as a bearing, the journal bearings are unloaded and the rotor system can become unstable. The seal is reacting to a combination of hydrodynamic squeeze film and sliding friction behavior. However, the friction (sliding) force and rotor weight eventually exceed the lifting hydrodynamic force developed by the lockup of the seal and seal rings [14].

High pressure seals can contribute or cause instability. Seal instability is but one of many causes of nonsynchronous, self-excited rotor instability. Instability is nonsynchronous, self-excited and potentially damaging to a turbomachine train. Self-excited vibrations appear spontaneously and increase in amplitude until limited by some barrier. The undamped critical speed analysis and unbalance response analysis referred to above is concerned with the *forced response* of the rotor system, instability is concerned with *self-excited* forces. Reducing these destabilizing forces is necessary to obtain a smooth running turbomachine.

The compressor designer must also be concerned with the range of compressor suction pressures. During startup, the suction pressure will be approximately 100 psig for run-in on nitrogen. After run-in cycle, hydrogen is added to the loop until the nitrogen is replaced. The pressure and temperature is increased to normal suction pressure. This wide range of operating pressures complicates the design of this seal. Lower suction pressure operation may require larger clearance to cool seal

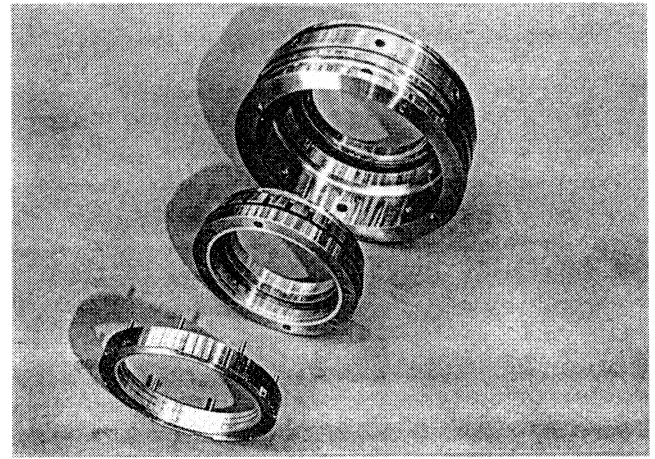
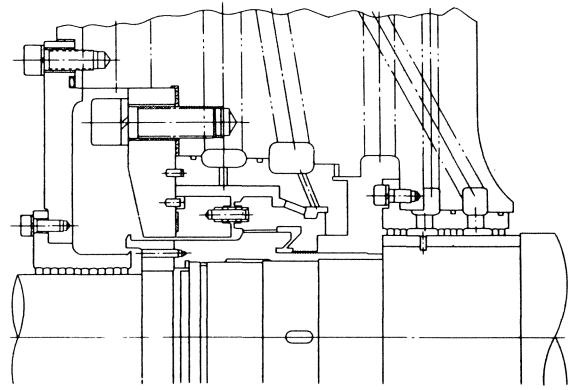


Figure 9. High Pressure Oil Film Seal.

parts while at higher suction pressure flows are adequate at small clearances.

CONCLUSION

Selection of turbomachinery (centrifugal compressors) involves more than just computing the head and horsepower. Proper selection procedure takes the problem from analyzing the process application to determining the gas properties, polytropic head, discharge temperature, horsepower, staging, tip speed, rotating speed, and impeller diameter. The mechanical or physical arrangement and materials of construction are determined in conjunction with standards and experience.

After the mechanical configuration (number of stages, casings, speed, etc.) is determined, it is advisable to address the machine's rotordynamics and to compute the lateral critical speeds and a stability analysis. The engineer is then able to proceed with procurement of the machine, complete the process study, etc., and know that the turbomachinery selection is correct.

APPENDIX 1 - CALCULATIONS FOR A CENTRIFUGAL COMPRESSOR

Sample calculations:

The following example illustrates the selection of a high pressure, hydrogen rich centrifugal compressor, using the polytropic head method presented previously. The operating conditions shown are the actual operating conditions for a hydrotreater recycle gas compressor for a recent project that was actually built and is in operation.

Specific volume:

$$v_s = 1.026 \left[\frac{1545}{3.73} \right] \left[\frac{580.67^\circ\text{R}}{(144)(537 \text{ psia})} \right]$$

$$v_s = 3.191 \text{ ft}^3/\text{lb}$$

Inlet flow:

$$Q_s = (239.8 \text{ lbs/min} \times 3.191 \text{ ft}^3/\text{lb})$$

$$Q_s = 765.2 \text{ ICFM}$$

Ratio of Compression:

$$R_c = \text{ratio of compression} = \frac{800 \text{ psia}}{537 \text{ psia}}$$

$$R_c = 1.489$$

Polytropic exponent:

$$\left[\frac{n-1}{n} \right] = \left[\frac{(1.375-1)}{(1.375)(0.785)} \right] = 0.347$$

Compressor head:

$$H_{\text{poly}} = \left[\frac{1.026+1.037}{2} \right] \left[\frac{1545}{3.73} \right] (580.67^\circ\text{R}) \left[\frac{1.489^{(0.347)} - 1}{0.347} \right]$$

$$H_{\text{poly}} = 106,069 \text{ ft-lbs}_f / \text{lb}_m$$

Number of stages:

$$\text{stages} = \frac{106,069 \text{ ft}}{9000 \text{ ft/stage}} = 11.8 \text{ rounded to } 12$$

Since the compressor manufacturer selected a compressor with 15 impellers, the remaining calculations will be made with 15 impellers. Fifteen impellers were selected in order to reduce the polytropic head developed per impeller, the corresponding rotating speed, and thus keep the tip speed of each impeller low enough to meet the ratio of actual stress to yield stress. The result was lower stresses in the impeller. The impeller stress limit was 60 percent of material yield @ maximum continuous speed. Consequently, the impellers selected produce less than 9000 ft per impeller guideline. The actual value was 106,069/15 = 7071 ft/impeller.

Table 5. Gas Analysis Calculation Results – Project Example.

Gas Constituent	Molar %	Molecular Weight, MW	Specific Heat, $M C_p @ 150^\circ\text{F}$ (Btu/Lb-mole ^o F)	[Mol % / 100] x $M C_p @ 150^\circ\text{F}$ (Btu/Lb-mole ^o F)	Critical Pressure, P_c , PSIA	[Reduced Pressure] [Mol % / 100] x Critical Pressure, PSIA	Critical Temperature, T_c , °R	[Reduced Temperature] [Mol % / 100] x Critical Temperature, °R	
Water Vapor	00.53	18.016	0594	8.07	0.028	3188.80	10.58	1188.83	3.84
Hydrogen sulfide	00.88	34.076	2898	8.27	0.070	1500.00	11.05	872.12	5.71
Hydrogen	92.81	2.016	1.8730	8.94	6.647	188.11	174.77	59.85	95.80
Methane	3.41	16.042	5470	8.95	0.305	698.40	22.72	343.00	11.70
Ethane	1.95	30.068	4960	13.78	0.213	708.50	10.88	549.89	8.82
Propane	00.82	44.096	2723	19.82	0.121	616.00	3.82	685.75	4.13
i-Butane	00.10	58.120	2681	25.77	0.025	557.80	0.83	734.13	0.73
n-Butane	00.06	58.120	2523	25.81	0.023	550.80	0.80	728.29	0.69
i-Pentane	00.03	72.148	2218	31.86	0.009	490.40	0.18	828.77	0.25
n-Pentane	00.03	72.148	2218	31.86	0.009	488.80	0.18	845.47	0.25
Hexane +	00.08	86.172	2069	37.63	0.020	458.90	0.38	813.27	0.73
Total	100.000		3.7308	7.278		238.54			92.15

For generalized compressibility charts use $T_c = 819^\circ\text{R}$ & $P_c = 327 \text{ psia}$ for hydrogen.
Using these values for hydrogen, the critical properties of the gas mixture are $P_c = 364.68 \text{ psia}$ and $T_c = 113.89^\circ\text{R}$

Source of data, API Technical Data Book, Volume I, Fourth Edition, 1982

From equation (9) K is calculated for the actual gas mixture;

$$K = \frac{M C_p}{M C_p - 1.986}$$

$$K = \frac{7.278}{7.278 - 1.986}$$

$$K = 1.375$$

A two casing compressor train design arrangement will accommodate the 15 impellers selected. Single casings are limited to nine stages for nonintercooled straight through designs. The two casings are designated “LP casing” and “HP casing” for low pressure casing and high pressure casing respectively. The LP casing has seven impellers and the HP casing has eight impellers. The arrangement described is shown in Figure A-1.

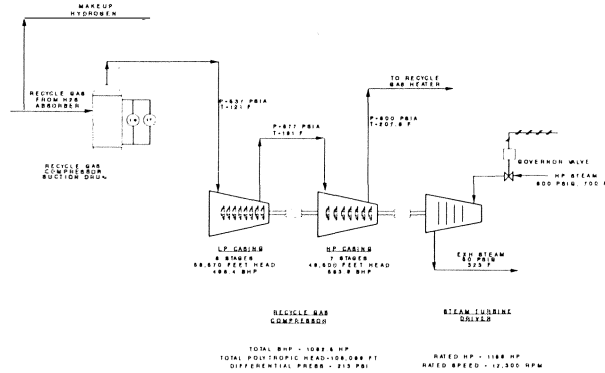


Figure A-1. Schematic Diagram, Hydrogen Gas Compressor—Project Example.

$$G_{\text{hp}} \text{ (LP casing)} = \frac{(239.8 \text{ lbs/min}) \times (56570 \text{ ft})}{33,000 \times (0.785)}$$

$$G_{\text{hp}} \text{ (LP casing)} = 458.2 \text{ hp}$$

$$B_L \text{ (LP casing)} = (0.174 (12,300/1000))^2$$

$$B_L \text{ (LP casing)} = 26.3 \text{ hp}$$

$$S_L \text{ (LP casing)} = (0.093) (12,300/1000)^2$$

$$S_L \text{ (LP casing)} = 14 \text{ hp}$$

$$B_{\text{hp}} \text{ (LP casing)} = 498.5 \text{ hp}$$

$$G_{\text{hp}} \text{ (HP casing)} = 523.6 \text{ hp}$$

$$B_L \text{ (HP casing)} = 26.3 \text{ hp}$$

$$S_L \text{ (HP casing)} = 14 \text{ hp}$$

$$B_{\text{hp}} \text{ (HP casing)} = 563.9 \text{ hp}$$

$$\text{Total } B_{\text{hp}} \text{ (LP casing + HP casing)} = 1062.50 \text{ hp}$$

Polytropic discharge temperature:

$$T_{\text{d(poly)}} = 121 + 459.67^\circ\text{R} \times 1.489^{0.347}$$

$$T_{\text{d(poly)}} = 666.9^\circ\text{R} = 207.2^\circ\text{F}$$

The impeller diameter is selected as a value consistent with the casing size, ICFM required, and rotating speed. For this application, a 12 to 13 in impeller diameter would be appropriate while limiting the rotating speed to 10 to 11,000 rpm. The compressor manufacturer selected a 12.6 in diameter for all 15 stages and a speed of 12,300 rpm. Therefore, the remaining calculations are based on these values.

Impeller Tip speed:

$$U = 2\pi \left[\frac{12.6}{2} \right] \left[\frac{12,300 \text{ RPM}}{60 \times 12} \right]$$

U - 676 fps

Acoustic Velocity

$$V_a = \left[(1.375) (1545/3.73) (580.67^\circ R) (32.16 \text{fps}^2) (1.026) \right]^{1/2}$$

$V_a = 3303 \text{ fps}$

The Mach number, U / V_a , will be approximately 0.2 using an acoustic velocity of 3300 fps. (The speed of sound in air is approximately 1200 fps.)

Specific speed:

$$N_s = \frac{\left[12,300 \text{ RPM} \right] \left[765.2/60 \text{ cfs} \right]^{1/2}}{\left[7071 \text{ ft} \right]^{(0.75)}}$$

$N_s = 56.963$

Specific diameter:

$$D_s = \frac{\left[12.6/12 \text{ ft} \right] \left[7071 \text{ ft} \right]^{1/4}}{\left[765.2/60 \text{ cfs} \right]^{1/2}}$$

$D_s = 2.696$

Flow coefficient:

$$\Phi = \frac{\left[700.3 \right] \left[765/2 \text{ ICFM} \right]}{\left[12,300 \text{ RPM} \right] \left[12.6 \right]^3}$$

$\Phi = 0.02177$

Head coefficient:

$$\mu = \frac{\left[7071 \text{ ft} \right] \left[32.16 \text{fps}^2 \right]}{\left[676 \text{ fps} \right]^2}$$

$\mu = 0.4976$

The calculations for the project specific example are summarized in Table 6 below.

NOMENCLATURE

- MW Molecular weight
- C_p Constant pressure specific heat
- C_v Constant volume specific heat
- T_c Critical temperature
- P_c Critical pressure
- k Ratio of specific heats
- T_r Reduced temperature
- P_r Reduced pressure
- m weight flow, lb/min
- n number of moles per minute
- Q capacity, cfm (usually inlet cfm, icfm)
- v specific volume, ft³/lb

Table 6. Hydrogen Gas Compressor Selection Calculations Summary.

OPERATING CONDITIONS	Equation (s) used	Given or Calculated value
Capacity, MMSCFD	Process Data	36.9
Capacity, m. Lb / min	Process Data	239.832
Suction pressure, P_s , psia	Process Data	537
Suction temperature, T_s , °F	Process Data	121
Suction temperature, T_s , °R	Process Data	580.67
Discharge pressure, P_d , psia	Process Data	800
GAS DATA		
Capacity, SCFM	Equation (9)	25,623
Molecular weight, MW	Table 1	3.7308
Gas constant, R	$R = 1545 / \text{MW GAS MIXTURE}$	414.12
Critical pressure, P_c , psia	Table 1	235.54
Critical temperature, T_c , R	Table 1	92.15
Ratio of specific heats, k, @ 150 F	Equation (5)	1.375
Reduced pressure, P_r (1)	Equation (7)	2.28
Reduced temperature, T_r (1)	Equation (6)	6.30
Compressibility @ suction, Z_s	Generalized Compressibility Charts	1.026
Compressibility @ discharge, Z_d	Generalized Compressibility Charts	1.037
INLET VOLUME FLOW		
Specific volume, v_s , ft ³ /lb	Equation (10)	3.1905
Weight flow, m. lbs/min	Given	239.832
Polytropic efficiency, E_{poly} , %	Input by R. E. engineer	78.5%
Inlet capacity, Q_1 , ICFM	Equation (11) or (12)	765.2
COMPRESSOR POLYTROPIC HEAD		
Polytropic head, H_{poly} , ft-lbs/lb _m	Equation (14)	106,069
Number of stages	Head / 10,000	8 (LP CASING) + 7 (HP CASING)
COMPRESSOR HORSEPOWER		
Gas horsepower, Ghp	Equation (16)	458.2 + 523.6 = 981.8 GHP
Bearing losses, B_L , hp	Table 3 + Equation (18)	26.3 + 26.3 = 52.6 HP
Seal losses, S_L , hp	Table 3 + Equation (19)	14 + 14 = 28 HP
Compressor brake horsepower, Bhp	Equation (17)	498.5 + 563.9 = 1062.5 BHP
DISCHARGE TEMPERATURE		
Discharge temperature, T_d , °R	Equation (20)	666.9 °R
Discharge temperature, T_d , °F		207.2 °F
IMPELLER DIAMETER & TIP SPEED		
Impeller diameter, D_2 , inches		12.6 inches
Impeller tip speed, U, fps	Equation (21) or (22)	676 FPS
ACOUSTIC VELOCITY		
Acoustic velocity, V_a , fps	Equation (23)	3303 FPS
DIMENSIONLESS VALUES		
Specific speed, N_s	Equation (25)	56.9 Dimensionless
Specific diameter, N_d	Equation (26)	2.7 Dimensionless
Flow coefficient, ϕ	Equation (28)	0.02177 Dimensionless
Head coefficient, μ	Equation (27)	0.4976 Dimensionless
Compressor Speed, RPM	$N = 229 U / D$	12,300 RPM

(1) For generalized compressibility charts, $P_r = 1.47$, $T_r = 5.10$ for the gas mixture specified in this project example

- Z Compressibility, dimensionless
- T Temperature, °R
- P Pressure, psia
- icfm Inlet cubic feet per minute
- scfm Cubic feet per minute, at 60°F, 14.7 psia
- mmscfd scfm × 10⁶
- H Specific head, ft-lb_f/lb_m
- R Ratio (of compression)
- R Gas constant, dimensionless
- E Efficiency, percent
- GHP Gas horsepower, hp
- BHP Brake horsepower, hp
- B_L Bearing loss coefficient, dimensionless
- S_L Seal loss coefficient, dimensionless
- D Diameter (impeller diameter), inches
- N Rotating speed, rpm and natural frequency in log decrement analysis
- g acceleration constant, 32.16 fps²
- N_{st} Number of stages
- V_{st} velocity, fps
- U Tip speed (impeller tip speed), fps
- MN Mach number, dimensionless
- N_s Specific speed, dimensionless

D_s Specific diameter, dimensionless

Greek letters

δ log decrement

Ω damping exponent

Φ Flow coefficient, dimensionless

μ Compressor head coefficient, dimensionless

Subscripts

s property at suction conditions

d property at discharge conditions

c compression

c a critical property

r reduced property (pressure, temperature, and compressibility)

poly polytropic value

a acoustic

1 property at inlet condition(s)

2 property at outlet condition(s)

L Loss (used for bearing and seal loss computation)

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