GUIDELINES FOR SPECIFYING AND EVALUATING NEW AND RERATED MULTISTAGE CENTRIFUGAL COMPRESSORS

by

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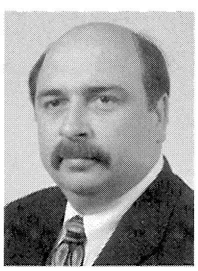

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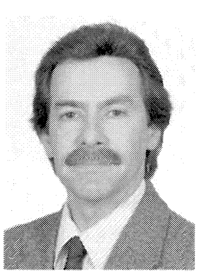

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ABSTRACT

Refineries, petrochemical companies, and gas processors are continually trying to increase plant production while budgets are getting tighter. Rerating process centrifugal compressors has become an ever-increasing trend as a solution to this problem. Existing hardware, however, has its limitations. The limiting factors can be different for different applications of the same

equipment. When rerating existing compressors cannot adequately meet increased compression production demands, new equipment must be added to an existing plant or a new plant must be built. Both of these scenarios require the purchase of new compressors.

Realizing that eventually a new plant may expand or increase production to respond to increased demand, and that the differential cost for a new compressor is high, it makes sense to consider the future uprateability of new compressors when they are initially purchased and built.

Discussions in this paper will give the rotating equipment engineer guidance when evaluating the design of a centrifugal compressor—–either new or rerated, including a list of items that should be reviewed; and guidelines that should be followed. A case study and examples will be used to emphasize and help explain the various limiting factors for both new and rerated multistage centrifugal compressors. Cost implications and tradeoffs along with the impact on reliability will also be analyzed. With this information, the rotating equipment engineer will be able to work with process engineers to effectively optimize compressor hardware selections for the required process duty.

WHERE DO I START?

It is important for the user to begin planning projects early enough to allow adequate time for the analysis and evaluation of many items associated with the plant design. Often, the basis for the compressor design is the result of preliminary discussions between the user and the compressor supplier. Once an initial set of conditions for a particular service is available, these conditions can be sent to compressor suppliers for preliminary performance data and for budget pricing. At this time, the supplier can offer suggestions as to how the process conditions could be revised to better suit the compressor design. It is important to note that the analysis of a rerated compressor is often more complex than for a brand new “clean sheet of paper” compressor, since the hardware that can be applied to the rerate is limited by the configuration of the existing compressor. This imposes several additional constraints on rerate selections that would not apply to a new compressor, and quite possibly may necessitate adjustments to process conditions.

User Specifications

From the equipment supplier’s point of view, the best way to begin working on any new request for quotation (RFQ), is to take some time to review the information provided in the RFQ package. The documents in the package should define whether the project is dealing with new or existing equipment, or a combination of both. A quick review of the process data sheets for each service will provide important information, such as flow rates, pressure levels, temperatures, gas composition, and type of driver. Oftentimes, there are special notes included with data sheets that define special performance requirements such as tolerance bands for intermediate pressures (sideloads, extractions, and pressure levels at coolers), discharge temperature limits, special material requirements, etc. Other special requirements may be defined in the requisition, general equipment specification, and/or compressor specification, so the application engineer should also review these documents prior to beginning any equipment selection. Taking time up front in a project to become familiar with the various project requirements can save hours of time downstream.

Whether the end user or a contractor is assembling the RFQ package, the responsible engineer should take care in providing a complete and legible copy of all pertinent documents to the bidders. Process data sheets should show all of the anticipated operating points for each different service, including any startup cases. The guarantee case should be clearly identified. The API 617 (API, 1995) data sheets should also indicate if the compressor will be used in an air dryout mode. The gas analysis for each compression section should be complete. If possible, the molecular weight of each gas constituent should be given along with the molecular weight of the gas mixture. The sum of all the gas fractions should add up to 100 percent. It is not uncommon to find that the gas analysis given on page two of the data sheets does not match the molecular weight shown on page one of the data sheets. Taking care to make sure all of the process data are consistent can eliminate time delays and the need for numerous questions between the user and the supplier. It is important to note that a variety of gas properties programs are utilized by users and suppliers worldwide. As a result, it is not uncommon for “k” (ratio of specific heats) and “z” (compressibility) values to vary, if different gas properties programs are used. The supplier should note the calculated “k” and “z” values on the performance data sheets.

The project specifications include many important requirements that must be considered. As an example, some users specify impeller stress limitations that are more stringent than the API 617 requirements. This is an important consideration that must be accounted for in initial selections. This would tell the equipment engineer that he needs to be more conservative in selecting impellers for a particular duty and that he must keep the speeds for any specified operating points below the maximum capability of the worst case (generally the highest flow) impellers. The user should keep in mind that such a requirement may increase the compressor cost by requiring the use of additional stages or special high strength materials. For instance, a duty that can be handled in four compression stages may have to be performed by five stages to meet this directive.

Time spent reviewing user specifications prior to selecting compressor hardware is always time well-spent. This effort often uncovers user preferences and firm requirements that can be taken into consideration for initial equipment selections.

Experience

Experience clauses are frequently included in project specifications. Some users require that the equipment or components offered be in service for at least two or more years. Such a requirement can limit the selection of components that may be applied to a particular service. This type of requirement could also preclude the use of “latest technology” components that may have been proven by single stage testing or other similar means in a development setting.

Temperatures and Materials

Sometimes special materials are required, due to low process temperatures, low ambient temperatures, gas mixtures being at the saturation point, or corrosive/erosive components in the gas stream. In general, standard compressor materials are acceptable for temperatures at or above −20°F (−29°C). It is not unusual for operating conditions to require materials acceptable for temperatures as low as −150°F (−101°C) in ethylene refrigeration service, or even −320°F (−195°C) for boil-off compressors. As the minimum design temperature for a compressor decreases below −75°F (−60°C), leadtimes for materials tend to increase due to materials availability problems. Some users have preferences for certain materials in certain services, such as requiring stainless steel shaft sleeves in wet gas service. In cracked gas service, many projects still require that the maximum yield strength be limited to 90,000 psi (6210 bar), even though the latest edition of API 617, by reference to NACE MR0175-93, defines the need for limited yield materials as a function of the concentration of H_2S present in the process gas stream and the total pressure.

Discharge temperature limits can be an important consideration. These are often imposed in cracked gas, HCL, and chlorine applications where there is a potential for fouling, corrosion, or reactive chemical problems. A discharge temperature limit usually dictates that “highest efficiency” aerodynamic components be applied. This requirement may also increase the number of intercooling points that are needed.

Intermediate Pressure Levels

For multisection applications, especially for refrigeration services, limits may be imposed on intermediate pressure levels due to process considerations. It is important to know by how much these pressures are allowed to vary, so that the hardware can be selected to meet the required process conditions. A reasonable range for intermediate levels is a variation of $+/-2$ percent in head for variable speed compressors, with a wider range for fixed speed applications. For sideload and extraction machines where no pressure ranges are given and the pressures specified cannot be met exactly, it is usually better to be a little below the specified pressure for sideloads and a little higher than the specified pressure for extractions. At the expense of additional power, this will ensure that the expected refrigeration loads can be met.

Performance Considerations

Some end users and contractors require that certain performance parameters are guaranteed along with the usual guarantees for flow, head, and horsepower. These are often in the form of head or pressure rise to surge guarantees, guarantees for intermediate pressure levels, surge flow guarantees, and/or guarantees for overload flow capability. The end user should recognize, however, that such guarantees often exact additional price or performance compromises. For instance, assume that a compressor manufacturer must guarantee that the surge flow for a multistage-stage centrifugal compressor will not be greater than 70 percent of the rated flow at a fixed speed. The compressor engineer would tend to select hardware that is moderately loaded. This has several effects:

- The hardware may not be selected at optimum efficiency at the normal point, since often, peak efficiency occurs near mid-flow for most centrifugal compressor stages.
- There will be less overload capability than if the staging was selected near the middle of the flow range for that particular stage.
- Additional costs for testing may be required to prove that such a requirement is met.

The bottom line for any special guarantees is that they should be tailored to the particular duty under consideration, and they should make sense for that duty. It does not make a whole lot of sense, for instance, to impose a requirement that the head rise to surge must be greater than six percent for a hydrogen compressor that is a fixed speed application. With the low molecular weight of the gas, it would be very difficult to meet such a requirement. This is due to the fact that for a given impeller design with a given blade configuration, an impeller compressing a high molecular weight gas will produce more head than the same impeller at the same speed and flow compressing a lighter molecular weight gas (Hallock, 1968). Although controllability is usually the issue when head rise to surge guarantees are required, the user must also recognize that machines with low head rise can be controlled using flow control instead of pressure control for antisurge systems. Flow control has been successfully applied to many compressors.

The potential for future uprateability is a very important consideration in the selection of centrifugal compressors. Keeping in mind that many compressors are uprated some time during their "life," it makes sense ideally to select equipment that can accommodate capacities on the order of 20 to 25 percent higher than the capacities specified in the initial RFQ. This way, it is less likely that the relatively expensive compressor becomes the plant bottleneck, obstructing a future plant capacity upgrade. Purchasing equipment that is very close to its maximum flow capability may save the end user some money up front, but it will end up costing *much* more money in the future, when whole casings must be replaced and existing foundations and piping must be modified.

Communication

Good communications among all parties involved in a project is essential to selecting and purchasing the right equipment. Although reliability and safety are normally the most important considerations overall, the supplier has to know what the end user expects of the equipment and which performance parameters are most important. Is efficiency most important? Is increased capacity most important? Is turndown capacity most important? Is a steep curve required to improve controllability? Is a flat curve required so that there is not much change in performance as the flow changes? Also, many times, there are areas within the RFQ that are not clear. This is particularly true for instrumentation, and also applies to equipment options such as single flow versus double flow for cracked gas service. Open communication among all parties involved—users, contractors, and suppliers—is vital in identifying different options. Open communication also provides opportunities to clarify ambiguous areas, so that effort is not wasted due to misinterpretation of the project specifications.

THE INITIAL SELECTION

After reviewing the RFQ package, the compressor application engineer is ready to begin an initial selection of the equipment that will be required. The project data sheets (API 617 format) provide information about the different process points at which the compressor will operate, along with identification of the type of driver that will be used. A good place to start with any selection is with the operating condition designated as the guarantee point. As defined by API 617, at the guarantee point, head and capacity are to have zero negative tolerance and the horsepower at this point cannot exceed 104 percent of the specified value (API, 1995). It is essential to note where the alternate operating points are flow and head-wise in relation to the guarantee point. This tells if the compressor staging (at the guarantee point) should be selected toward the underloaded portion of the curve, toward the overloaded portion of the curve, or in the middle of the curve.

The type of driver to be used for the compressor can often have an effect on how the compressor is selected. For instance, many gas turbines have fixed speed ranges. If the compressor selected cannot run within this speed range, a gear is required. For fixed speed motor drive applications, the compressor must be selected to meet the highest flow, highest head case, with no negative tolerance on head. Operating conditions at lower flows may require suction or discharge throttling or flow recycling. Suction throttling requires that a throttle valve be located in the main suction piping to the compressor. The performance at lower flow conditions is achieved by partially closing the throttle valve, thereby lowering the suction pressure and raising the inlet volume flow to meet the required discharge pressure. Care must be taken in applying suction throttling to refrigeration systems. If the suction pressure can go to subatmospheric levels, there could be an undesirable leakage of air into the refrigeration loop. In discharge throttling, the gas is compressed to the pressure level developed by the compressor at the particular suction pressure, flow, and speed. This pressure level is higher than the required discharge pressure. A throttle valve in the discharge piping is then used to reduce the pressure to the required level. The drawback to discharge throttling is that compressing the gas to a discharge pressure higher than what is needed consumes more power. With flow recycling, the compressor antisurge system is used to take flow from the discharge end of the compressor and return this flow to the compressor inlet (or an intermediate nozzle) after it is cooled. This recirculated flow increases the total flow moving through the compressor, raising the operating point to a higher flow value along the curve to where the pressure ratio meets the required level. The drawback to flow recycling is that additional power is consumed in order to compress the gas that is recycled. Of these three methods for fixed speed applications, suction throttling usually consumes less power. Steam turbine drive applications are generally the most flexible, since there is usually a turbine that can operate at the conditions required by the compressor and the speed can easily be changed to meet the various specified operating conditions. Considering the future uprateability of drivers is also an important issue, but it is not within the scope of this paper.

It is essential that the contractor/end user understand that compressor manufacturers, for the most part, build standard frame sizes that cover different volume flow ranges. Each compressor frame size has its own set of inlet, discharge, and intermediate nozzles that are normally used, along with a standard set of choices for bearing and seal sizes and main shaft diameters.

AERODYNAMIC CONSIDERATIONS

Frame Selection

The combination of volume flow and molecular weight defines the frame size of the compressor. While looking at the volume flow alone will generally indicate the frame size that should be selected, the molecular weight of the gas being compressed must also be considered. High molecular weight will reduce the allowable flow limit for the inlet nozzle and also raise the mach number (Equation (3)) at the first stage impeller. If the flows for the various operating conditions are in the high end of the flow range for a particular compressor frame size, it may be prudent to choose the next larger frame size. This will provide margin for future uprating and may result in higher efficiencies since more optimum flow coefficient stages can be used (Equation (5)).

Nozzle Velocities

A velocity of 150 ft/sec (46 m/sec) is generally regarded as the upper limit for inlet nozzles operating on air or gases with acoustic velocities similar to that of air. In practice, it is not desirable to design or purchase a new compressor that has a nozzle velocity this high, since:

- At higher flows along the compressor curve velocities will be even higher,
- A velocity this high leaves reduced potential for future uprateability, and
- High gas velocity through process piping is a major source of noise.

Acceptable limits for nozzle velocities may be lower than 150 ft/sec (46 m/sec) depending on the gas conditions. One method uses acoustic velocity, which for ideal gases is a function of the molecular weight, "k" value (ratio of specific heats C_p/C_v), "Z" value (compressibility), and gas temperature, to define the maximum nozzle flow. For this method, one would compute the $\sqrt{\theta}$, which is defined by:

$$\sqrt{\theta} = \sqrt{\frac{(26.2)\,(MW)}{(k)\,(Z)\,(T)}} \tag{1}$$

Where:
MW = Molecular weight of the gas being compressed
k = Ratio of specific heats (C_p/C_v)
Z = Compressibility factor
T = Temperature entering the nozzle (°R)

Using the value for $\sqrt{\theta}$ computed above, the maximum volume flow for a particular nozzle size can be found by using the charts labeled as Figures 1 and 2. Note that the flow limits shown on Figures 1 and 2 for $\sqrt{\theta} = 1.0$ are based on a velocity limit of 150 ft/sec (46 m/sec) for air at 80°F (27°C). As seen in Equation (1), higher molecular weight and/or lower temperature will increase the value for $\sqrt{\theta}$, which in turn will reduce the allowable flow for the nozzle.

Higher nozzle velocities translate into higher pressure drops through the nozzles, resulting in lower overall compressor efficiency and higher power requirements. Pressure drop through a nozzle can be defined by the equation:

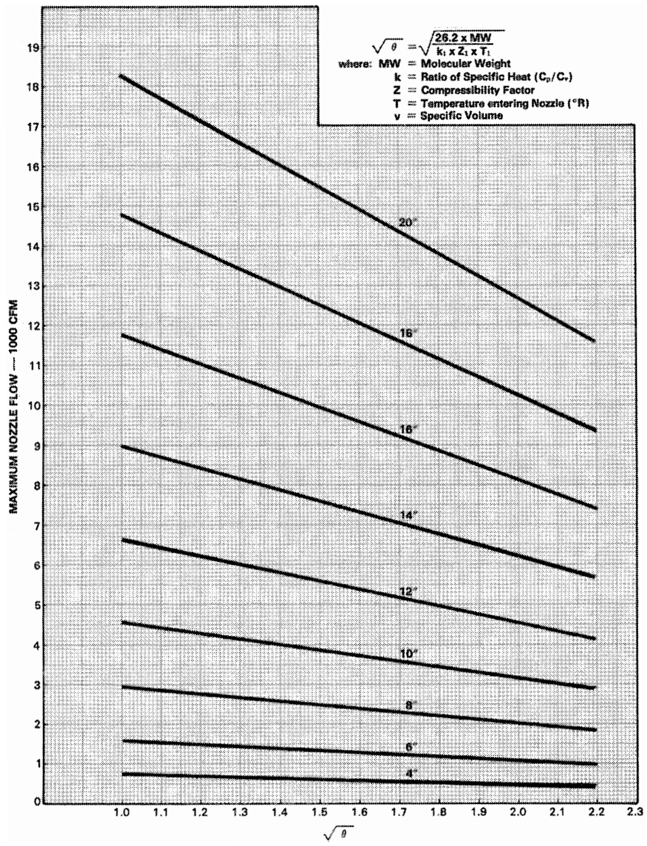

Figure 1. Nozzle Flow Limits (Four Inch through 20 Inch Nozzles).

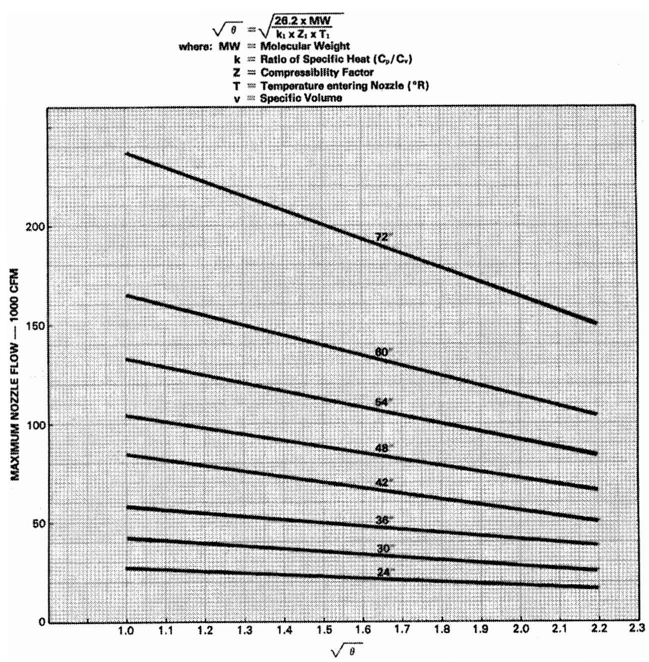

Figure 2. Nozzle Flow Limits (24 Inch through 72 Inch Nozzles).

$$\Delta P = \frac{c\,V^2}{2g_c(144)(\nu)} \tag{2}$$

Where:
ΔP = Pressure drop (psid)

c = Loss coefficient constant
V = Gas velocity (ft/sec)
g_c = 32.2 (gravitational constant (lb_m-ft/lb_f-sec^2)
v = Specific volume of the gas (ft^3/lb_m)

The loss coefficient, "c," can have different values depending on the type of nozzle under consideration and the design of the nozzle. Values for "c" are determined empirically. As a guide, "c" generally has a value of approximately 1.5 to 1.8 for main inlet nozzles and internal volute intercooler nozzles; sideloads with blank stage diaphragms have loss coefficients around 1.8 to 2.2; external volute intercooler nozzles have loss coefficients of approximately 2.3 to 2.7; sideloads and extractions with the flow passing through the diaphragm core have loss coefficients on the order of eight to nine.

Figure 3 shows an internal volute intercooler section design. Figure 4 shows an external volute intercooler section. For an internal volute, the volute section is located inside the casing, and requires more axial space than an external volute. For an external volute design, the volute section is located outside the aerodynamic flowpath. The external volute is positioned on the periphery of the compressor casing.

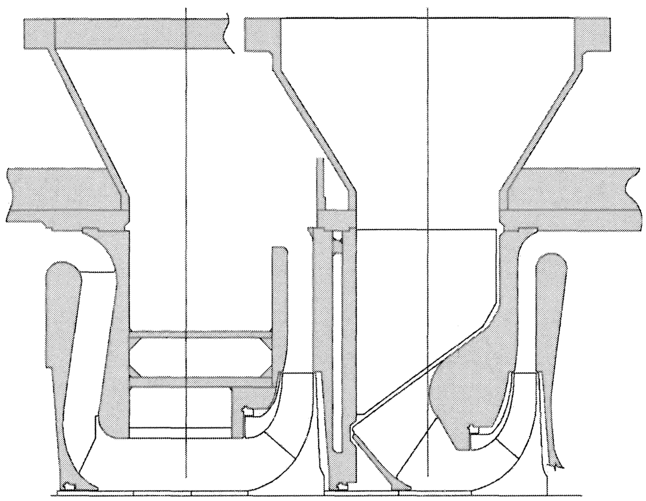

Figure 3. Internal Volute Intercooler Section.

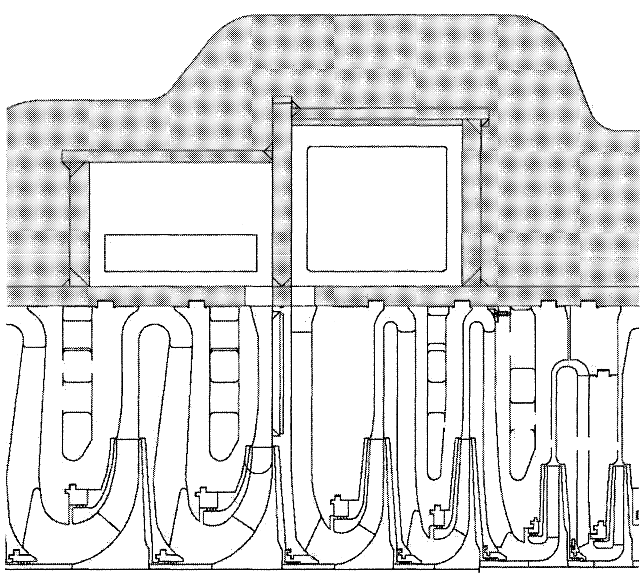

Figure 4. External Volute Intercooler Section.

Likewise, blank stage sideloads (Figure 5) require more axial space than lightener core sideloads (Figure 6). For a lightener core sideload, the flow is introduced through the open area in the diaphragm. Lightener core sideloads can only be used for relatively low flows.

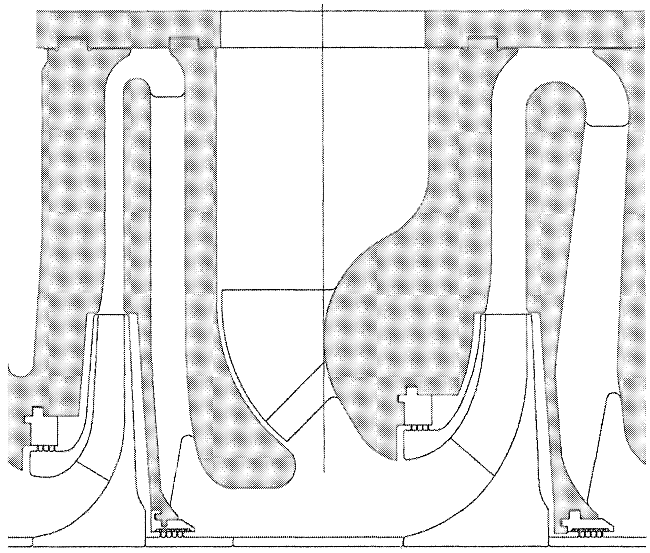

Figure 5. Blank Stage Sideload Design.

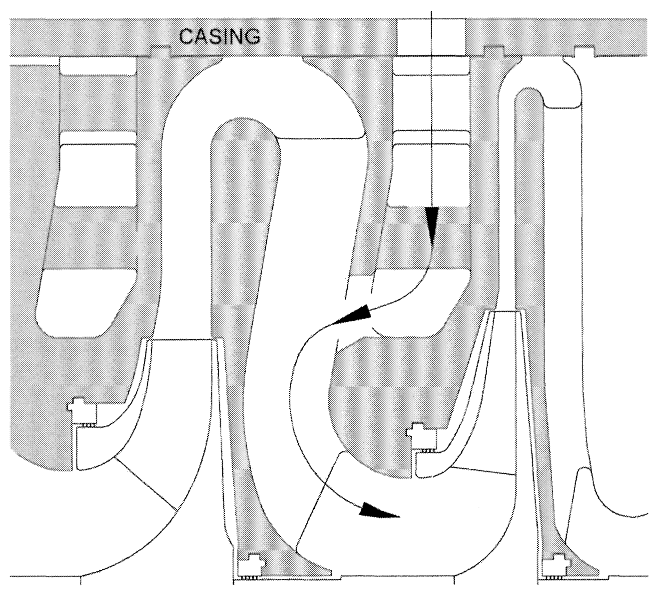

Figure 6. Lightener Core Sideload Design.

The nozzle flow limits given in Figures 1 and 2, and the 150 ft/sec (46 m/sec) velocity limit are good practice to follow for any new compressors. For rerate applications, where an existing compressor casing and existing nozzle configuration would be reused at higher flow conditions, these limits may be exceeded at the price of higher pressure drops through the nozzle and higher power consumption.

A different method can be used to check the suitability of existing nozzles at new conditions. This method uses a comparison of the nozzle mach number at the anticipated conditions to the nozzle mach number for air at 80°F (27°C). The nozzle mach number is determined by comparing the inlet velocity of the gas at the nozzle with the acoustic velocity.

$$\text{Nozzle Mach No.} = \frac{\text{Inlet Velocity}}{\text{Acoustic Velocity}} \qquad (3)$$

Where:

Inlet Velocity $= \frac{Q}{60A}$

Q = Inlet volume flow (acfm)

A = Nozzle area (ft^2)

$$\text{Acoustic Velocity} = a = \sqrt{g_c\, kZRT} \qquad (4)$$

Where:

g_c = Gravitational constant = 32.2 ($lb_m - ft/lb_f - sec^2$)

k = Ratio of specific heats, C_p/C_v

Z = Compressibility factor

R = Specific gas constant = 1545/molecular weight

T = Temperature (°R)

The acoustic velocity of air at 80°F is approximately 1139 ft/sec (347 m/sec). At a velocity of 150 ft/sec (46 m/sec), the nozzle mach number is 150/1139 or 0.132. This is used as a basis for comparison to other gases at other gas conditions.

For a rerated compressor, for nozzles with relatively high nozzle velocities (over 125 ft/sec or 38 m/sec), the nozzle mach number should be calculated using Equation (3). This value should then be compared with the standard value of 0.132. For mach number ratios (actual/0.132) up to 1.25, a loss coefficient on the order of 2.0 (versus 1.5 to 1.8 for a new properly designed nozzle) would be used to account for the increased velocity. This factor would increase to around 2.5 for mach number ratios of 1.25 to 1.35. Nozzle mach number ratios in excess of 1.35 are not recommended.

We have given guidelines to follow for inlet nozzles; but what about discharge nozzles? In general, compressor manufacturers are not very concerned about discharge velocities, since a high discharge velocity does not affect the compressor itself. The user, however, needs to consider these, since high velocities may affect the process downstream and increase noise levels. Oftentimes, process piping size increases downstream of the compressor body to reduce gas velocities and minimize pressure drops through process piping.

With respect to the compressor, although discharge velocity is not a major worry, it is imperative that the discharge volute be properly sized to accommodate all of the specified process conditions. This applies to all discharge volutes—main discharge, double flow discharge, intercooler discharges, and volute type extractions. An undersized discharge volute results in higher pressure drops. In the worst case, this could mean that the compressor cannot meet the expected or guaranteed level of performance, especially in fixed speed applications. Higher pressure drops will translate into higher speed and higher power requirements.

Discharge Volutes

For new equipment, discharge volutes (Figure 7) should ideally be sized to allow uprateability of the compressor on the order of at least 20 percent to 25 percent. For rerated equipment, often there is no choice of selecting different size volutes. Many older compressors are cast construction with external or cast-in volutes. For these machines, the area of the existing volute is fixed and cannot be easily changed, other than by grinding to increase the volute area. For an existing volute, the only way to improve the volute loading is to lower the volume flow into the volute or increase the head developed by the stage immediately preceding the volute. The volume flow can be reduced artificially by increasing the pressure ratio for the compression section located just before the volute. The head can be increased by selecting larger diameter impellers, using impellers with low backward lean, or by scaling an impeller to a larger diameter.

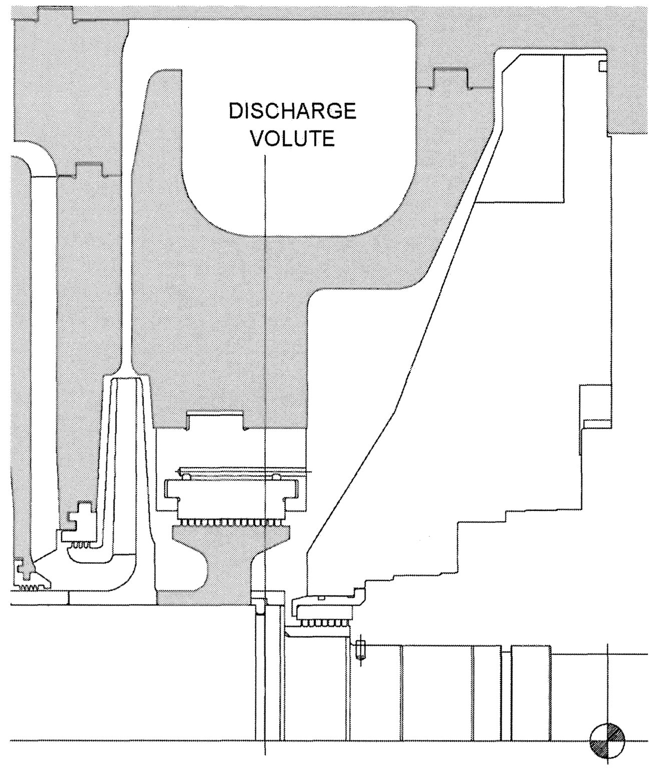

Figure 7. Location of Balance Piston and Discharge Volute for an Inline Compressor.

Flow Coefficient

As time goes on and the compressor industry improves upon and extends the limits of proven designs, maximum acceptable flow coefficients for impellers have also crept upward. Whereas impeller design coefficients of approximately 0.15 were generally looked upon as the upper limit a few years ago, design flow coefficients on the order of 0.18 are now quite common. Flow coefficient can be defined as:

$$\phi = \frac{700\, Q}{N\, d^3} \qquad (5)$$

Where:

Q = Impeller inlet volume flow (ft^3/min)

N = Rotational speed (rpm)

d = Impeller outside diameter (in)

The higher flow coefficient stages available today should not be of great concern to the end user, since they are a natural extrapolation of proven high flow levels. Rigorous single stage testing by suppliers is more sophisticated than in the past, and design and simulation programs are also more sophisticated. These improved tools and techniques raise the level of confidence in actual versus predicted performance for a compressor stage.

Although flow coefficients themselves should not be cause for concern, the combination of the inlet nozzle and the inlet impeller should be designed properly to minimize pressure losses and ensure even flow distribution at the eye of the impeller. This is very important since any performance deficiencies at the first stage are carried through the machine in the form of reduced head and possibly higher operating temperatures.

Relative Inlet Mach Number

Along with flow coefficient, the relative inlet mach number of an impeller is an important consideration. Higher mach numbers generally mean reduced operating range for a particular stage, and stage efficiency is usually lower at higher mach numbers. Some user specifications limit the maximum mach number to values of 0.8 or 0.9. This is not a problem for machines compressing low molecular weight gases, but it does affect high molecular weight and/or low temperature applications such as propylene, propane and butane compressors, to name a few. As a general guideline, caution should be exercised when the relative inlet mach number exceeds a value of 0.9. When this value is exceeded, the supplier should provide supporting data and experience references.

The mach number is a direct function of the molecular weight of the gas being compressed and the geometry of the impeller at its inlet. The relative inlet mach number is defined as:

$$M_{w1} = \frac{W_1}{a} \qquad (6)$$

Where:

M_{W1} = Relative inlet mach number
W_1 = Relative velocity at impeller inlet at static gas conditions
a = Acoustic velocity of the gas at static conditions (Equation (4))

and

$$W_1 = \sqrt{(u_1^2 + V^2)} \qquad (7)$$

Where:

u_1 = Impeller rotational velocity (ft/sec)
u_1 = Nd/229
N = Rotational speed (rpm)
d = Root mean square diameter at the impeller inlet (in)
V = Absolute inlet velocity at static conditions (ft/sec)

The relative velocity at the inlet to the impeller depends upon the inlet volume flow to the impeller along with the area at the eye of the impeller and the rotational velocity of the impeller. The rotational velocity is a function of the speed and impeller blade diameter at the inlet. Therefore:

- Higher flow stages have higher mach numbers than lower flow stages at the same gas conditions.
- At the same gas conditions, a full or partial inducer type impeller (Figure 8) has a lower inlet mach number than would a noninducer stage (Figure 9), since the blade diameter at the inlet to the impeller is lower.

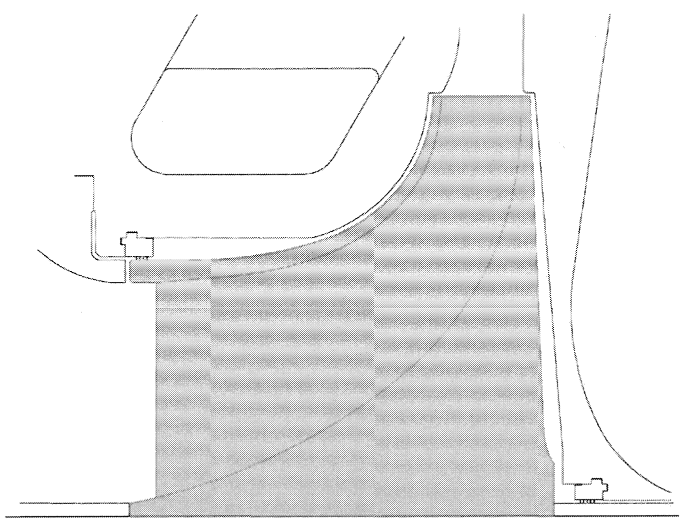

Figure 8. Inducer Type Impeller.

Equivalent Tip Speed

Another way of looking at the effect of mach numbers is to consider the equivalent tip speed of an impeller. Equivalent tip

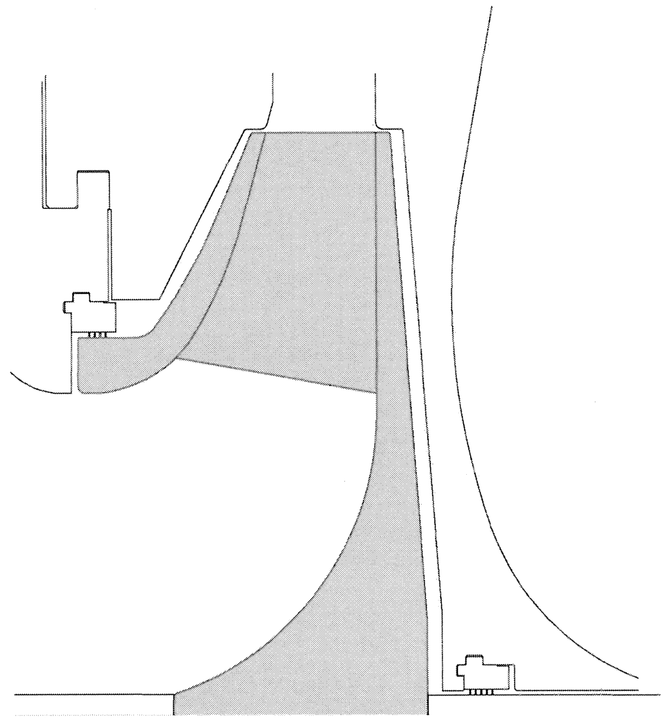

Figure 9. Noninducer Type Impeller.

speed describes aerodynamically what the performance of a particular impeller would be if operating on air rather than the actual gas being compressed. Normally, equivalent tip speeds are at a value of 1200 ft/sec (366 m/sec) or lower. However, with newer impeller designs and better matching with stationary components, it is not uncommon to see equivalent tip speeds as high as 1400 ft/sec (427 m/sec). An example of a typical individual stage performance curve operating at various equivalent tip speeds is given as Figure 10.

In looking at a particular impeller for an experience comparison, at the same flow coefficient and same equivalent tip speed, the mach number will be the same. The advantage of using equivalent tip speed is that it is easier to calculate than mach number. Equivalent tip speed is defined by:

$$u_{2e} = \sqrt{\theta\, u_2} \qquad (8)$$

Where:

u_{2e} = Equivalent tip speed (ft/sec)
$\sqrt{\theta} = \sqrt{\frac{(26.2)\,(MW)}{(k)\,(z)\,(T)}}$ as shown in Equation (1) above
$u_2 = \frac{Nd_2}{229}$
N = Rotational speed (rpm)
d_2 = Impeller outside diameter (in)

Discharge Temperature

It is important to consider discharge temperature limits when selecting a compressor. The most common applications for which temperature limits are imposed are cracked gas, wet gas, HCL, and chlorine compressors. For such applications, corrosion, chemical reactivity, and fouling can be problems, especially for stationary aerodynamic components such as diaphragms and guide vanes. In cracked gas and wet gas services, the rate of polymerization dramatically increases at temperatures in the range of 195°F to 210°F (90°C to 99°C), depending on the feed stock. For chlorine compressors, the discharge temperature for all specified operating points should be limited to a maximum of 250°F (121°C). For

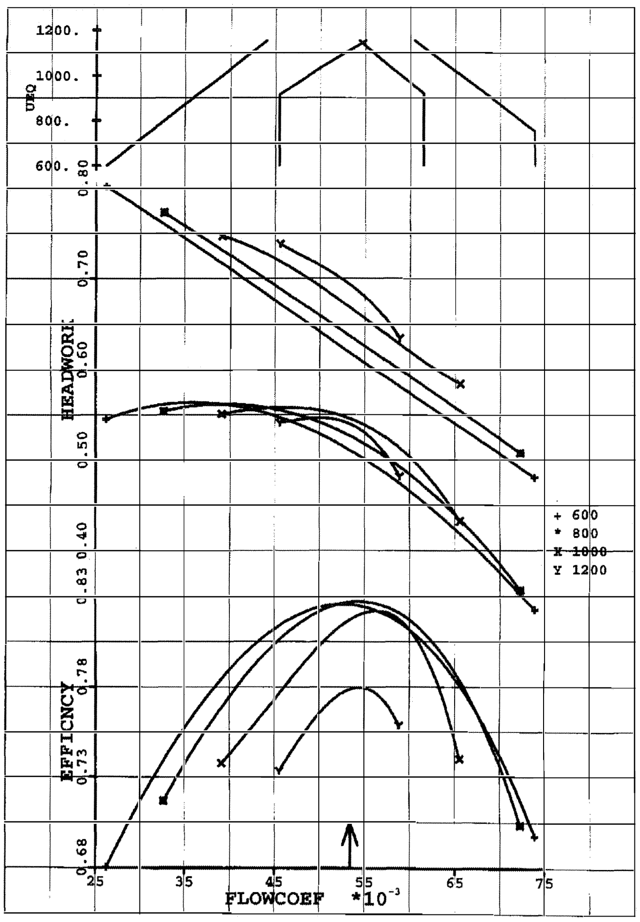

Figure 10. Impeller Curve at Various Equivalent Tip Speeds.

HCL applications, if the operating temperature is held below 350°F (177°C), corrosion should not occur. With fouling, polymers build up, the flow passages narrow, and efficiency falls. This translates into reduced capacity and higher power requirements. Corrosion and chemical reactivity can lead to component failure. To limit such effects, high efficiency staging should be used in conjunction with multiple cooling points. However, high efficiency staging often requires more axial length for an optimum flowpath. Adding cooling points also adds to the axial space required and may even require more compression sections or an additional compressor body. Increased axial spacing lowers the critical speeds.

In the absence of any temperature limits required due to the type of gas being compressed, most compressors can withstand discharge temperatures in the 350°F (177°C) to 400°F (204°C) range without taking any special design measures. These limits are set by elastomers (normally Viton) and balance piston seal materials (lead or fluorosint).

MECHANICAL CONSIDERATIONS

Theoretically, it would always be possible to select a compressor with an overall efficiency at or above 85 percent, if mechanical limitations were ignored. However, every final selection has to be a compromise between aerodynamic and mechanical considerations. Generally, the compressor application engineer works up a preliminary compressor selection that has a high efficiency level. Using this as a base selection, some key mechanical parameters are checked. These parameters include: bearing span, critical speeds, impeller speed limits, and shaft end sizes. An acceptable final equipment selection is a mix of the best aerodynamics, with all components operating within their respective mechanical limits.

Bearing Span/Critical Speeds

If the bearing span for the initial selection is too long, aero components that require shorter axial space must be selected. This often results in lower efficiency levels and increased power requirements. There are two ways to determine if a bearing span is too long. The first way is to look at historical data for the particular frame size of compressor with respect to bearing span versus maximum continuous speed. For such an investigation, you also need to consider whether the compressor in question and the reference compressors are "drive-through" bodies, i.e., compressor bodies situated between two other bodies in the equipment string. If this information proves inconclusive or is too close to call, a preliminary unbalance response analysis should be run to determine the expected values for first and second critical speed. For long rotors and rotors with high gas densities, a preliminary log decrement/stability analysis should also be run.

Impeller Limits

Hand-in-hand with critical speed considerations, the mechanical limits for each selected impeller must be checked. To confirm that a particular impeller is within its design limits, the engineer compares the required spin speeds for the particular application with the maximum allowable spin speed for the impellers, based on the specific material required for the duty. Each compressor supplier has mechanical limits for various materials for each impeller. These limits are established by actual test data or by analytical means. API 617 requires that impellers be overspeed tested to at least 115 percent of maximum continuous speed. Some user specifications have more stringent requirements. It is important to note that overspeed limitations may preclude the use of certain impellers for a particular application. Again, this may necessitate a compromise in which less efficient staging with higher mechanical limits is selected.

Shaft Ends

Shaft end sizes need to be verified as part of the evaluation procedure for new and rerated compressors. In extreme conditions where power requirements are very high, the size of the shaft end(s) could dictate that a larger frame size compressor be used. The maximum size for the main drive shaft end is limited to the diameter of the journal bearing. For compressor bodies located between two other bodies in the same string, the "drive-through" shaft end size is limited by the maximum bore for the thrust bearing. The required shaft end diameter can be defined by:

$$\text{S.E.} = \sqrt[3]{\frac{321000\ (\text{hp})\ (\text{DSF})}{(\text{S})\ (\text{N})}} \tag{9}$$

Where:
S.E. = Shaft end diameter (in)
hp = Horsepower
DSF = Driver sizing factor
S = Torsional shear stress of the shaft end material (psi)
N = Speed (rpm)

In sizing a shaft end, the combination of horsepower and speed that yields the highest ratio (highest torque) should be used. A value of 1.1 is commonly used for the driver sizing factor for steam turbine applications, since the turbine is normally sized for 110 percent of the worst case conditions of speed and power for the compressor. For this equation, the torsional shear stress varies directly with the shaft material, and is found by multiplying the ultimate tensile strength of the given material by 0.10. An exception to this is for compressors driven by synchronous motors, for which the permissible shear stress is obtained by multiplying the ultimate tensile strength of the given material by 0.075. This adjustment accounts for the transient torque requirements.

Although compressor shaft ends are normally sized for the maximum conditions the compressor is exposed to, the shaft ends can also be sized to accommodate the maximum power developed by the driver. This is often the case for fixed speed motor applications when the motor is not grossly oversized, since it is quite likely that, at some point in time, the plant will increase capacity and will use the maximum power that the motor will produce. For gas turbine applications, it may *not* be feasible to size the compressor shaft end for the maximum power that the gas turbine can develop since the maximum gas turbine power at the minimum specified site ambient conditions may be significantly higher than the power required by the compressor. Oversizing the shaft ends will affect coupling selection, possibly requiring heavier couplings. Oversizing shaft ends can also increase compressor overhang lengths. Increased overhang lengths and heavier couplings lower critical speeds, so one needs to be careful when specifying that shaft end sizes be oversized.

Bearings

Thrust and journal bearing loads should be checked for all compressors. Journal bearings generally have ultimate load limits as high as 600 psi (41 bar), but should be limited to 275 to 300 psi (19 to 21 bar) static loading for continuous long term operation. Journal bearing load is calculated by:

$$P = \frac{W}{(L)\,(D)} \tag{10}$$

Where:
P = Bearing load (psi)
W = Static bearing load (lb)
L = Bearing pad axial length (in)
D = Bearing nominal diameter (in)

Accurate journal bearing loads are best obtained from rotordynamic runs, which provide a calculated rotor weight. In the absence of actual data, the static load for each journal bearing can be approximated by using one-half of the estimated rotor weight.

Thrust bearing loads are calculated in a manner similar to that for journal bearings. For thrust bearing load, simply divide the net aerodynamic thrust load by the area of the thrust bearing. The aerodynamic thrust load is calculated by adding the thrust loads for the individual impellers, plus the thrust loads due to steps in the shaft or pressure differences at the two ends of the rotor, and then subtracting out any compensating load due to a balancing piston (for inline machines) or center seal (for back-to-back units) (Elliott Co., 1975). Ultimate load limits for thrust bearings are a function of the bearing size and speed. For smaller size thrust bearings (approximately 12 in^2 to 25 in^2) the ultimate load limits are in the 525 to 650 psi (36 to 45 bar) range. For larger thrust bearings, the ultimate load limits are in the 700 to 850 psi (48 to 59 bar) range. Maximum thrust bearing loads should be limited to one-half of the ultimate design load as required by API 617 (API, 1995).

In lieu of bearing load limits, some user specifications define bearing metal temperature limits. Bearing metal temperatures are based on empirical or actual test data and are dependent upon the bearing load and rubbing velocity. Bearing metal temperature limits are usually not difficult to meet for thrust bearings, but are sometimes difficult to meet for journal bearings. This is due to the fact that journal bearings are often more highly loaded than thrust bearings. To reduce thrust bearing loads, the balancing piston or center seal diameter can be adjusted; but there is no such adjustment for journal bearing loads. For each compressor frame size, there are often three or four choices for standard thrust bearing sizes; whereas there may be only one or two choices for journal bearing size. Journal bearing loads can only be decreased by reducing the weight of the rotor or increasing the journal bearing diameter or length.

AUXILIARIES

As part of any rerate study, auxiliary equipment such as couplings, oil systems, and seal buffer systems must be considered. When compressor or driver shaft end diameters need to be increased, it is unlikely that an existing coupling can be reused. When torque requirements increase, it is best to have the coupling vendor review the new conditions to determine if the existing coupling may be reused. Sometimes the existing coupling can be used for the new duty, if the user will accept a reduced service factor.

Lube oil systems should be reviewed to ensure that they have adequate capacity and heat removal capabilities. Bearing oil requirements and heat loads increase with increasing speed. For gears, oil flows, and heat loads increase as the power increases.

OTHER CONSIDERATIONS

Standard Designs

Compressor suppliers normally have standard compressor frame sizes. Each frame size is designed to handle a particular range of volume flows, and there is usually some flow overlap from one frame to another. The volume flow that one supplier's compressor frames can accommodate is different from the flow range for other suppliers' equipment. For a particular application, the required service may be optimum for one supplier's compressor, while it may be less than optimum for another supplier. Casing design pressures also vary from supplier to supplier. Each compressor frame size has standard nozzle sizes, but there is some flexibility to install larger or smaller nozzles. Larger nozzles require more axial space, which lengthens the bearing span and lowers critical speeds.

Diffusers

Compressor diaphragms can be supplied with vaned (Figure 11) or vaneless (Figure 12) diffusers. Typical performance curves for a vaned diffuser stage and a vaneless stage are given as Figures 13 and 14. When to apply vaned diffusers is the topic of some debate. Vaned diffusers can boost stage efficiency along with increasing the amount of head (pressure ratio) produced by the stage. This translates into lower power requirements and could even reduce the number of stages required for a particular service. The drawbacks of vaned diffusers can be: reduced operating flow range, increased noise (if located immediately prior to a discharge nozzle), and more surface area for polymers to accumulate. However, some studies have shown that by applying low solidity vaned diffusers to a particular stage, the loss of flow range can be minimized by optimally matching aerodynamic components (Hohlweg, et al., 1993). For vaned diffuser applications, the diffuser vanes are positioned to achieve optimum incidence at the stage design flowrate. With conventional vaned diffuser designs, if the flow increases enough, the minimum area in the passage will cause the diffuser to choke, and thereby limit the capacity of the stage. As flow is reduced, the incidence on the leading edge of the vane changes, and the flow eventually separates from the vanes. When this separation becomes large enough, stall occurs. It may not be advisable to use vaned diffusers in dirty gas services and in applications that have a broad range of flows.

Main Casing Seals

The type of shaft end seals to use for a particular application must also be considered when assembling specifications for a compressor inquiry. The main types of seals available are: labyrinth, carbon ring or circumferential, mechanical contact, bushing, and gas face seals. Labyrinth seals are best suited for low pressure, nontoxic, or air applications. It is important to note that labyrinth seals are clearance seals. That is, they cannot hold pressure in the compressor casing; and some process gas or buffer gas or both, will leak to atmosphere. The clearance between the labyrinth and the shaft sleeve at the seal will allow process gas to escape. Labyrinth seals are used where relatively high leakage flows are tolerable. If no process gas leakage to atmosphere is

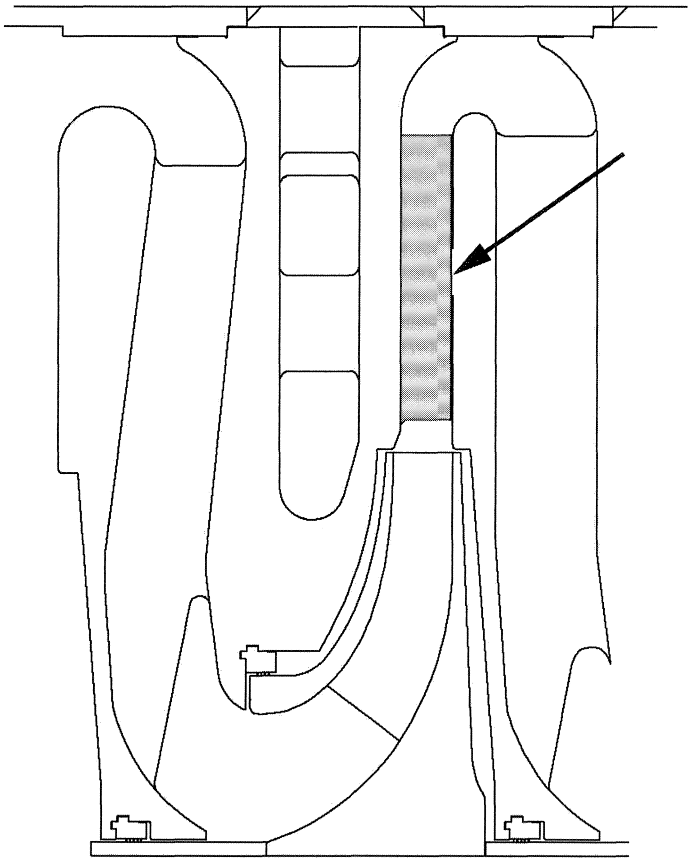

Figure 11. Vaned Diffuser Stage.

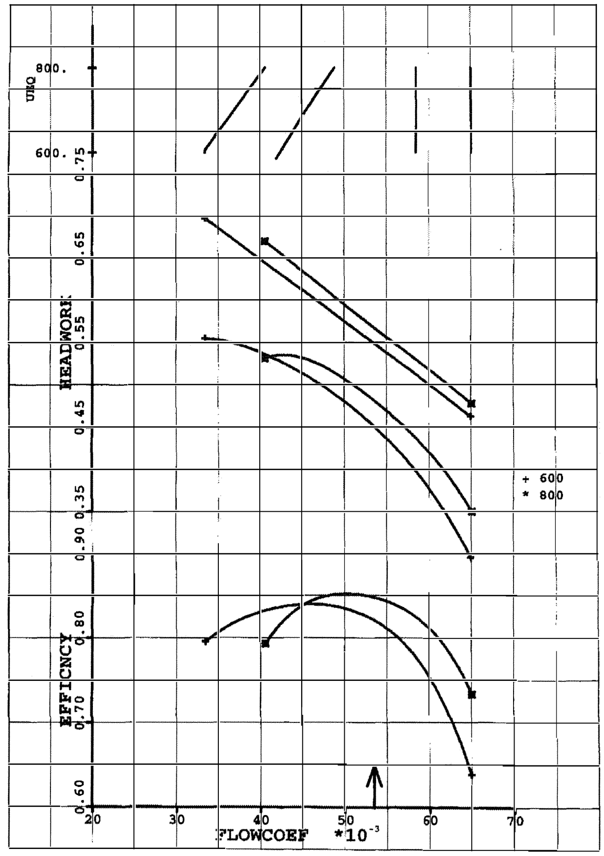

Figure 13. Performance Curve for Vaned Diffuser Stage.

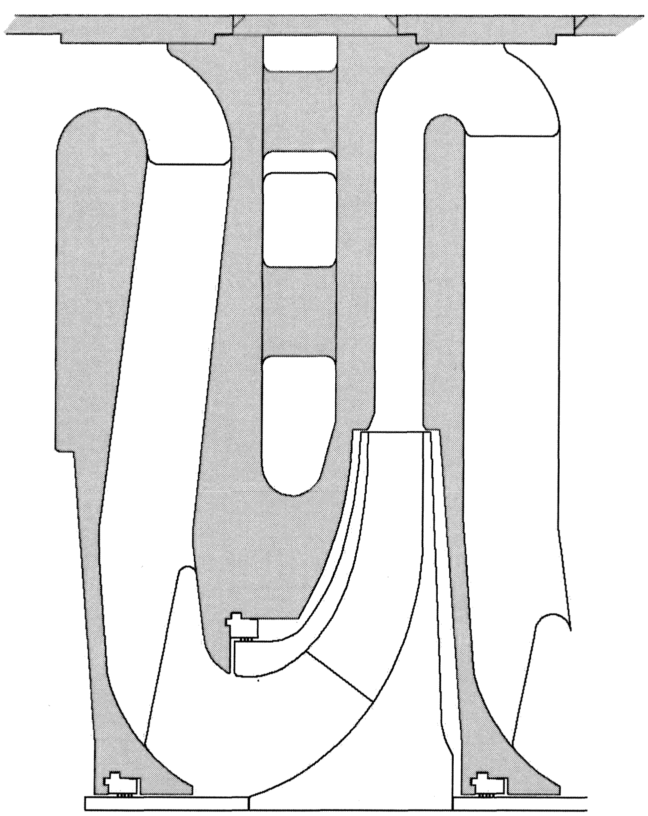

Figure 12. Vaneless Diffuser Stage.

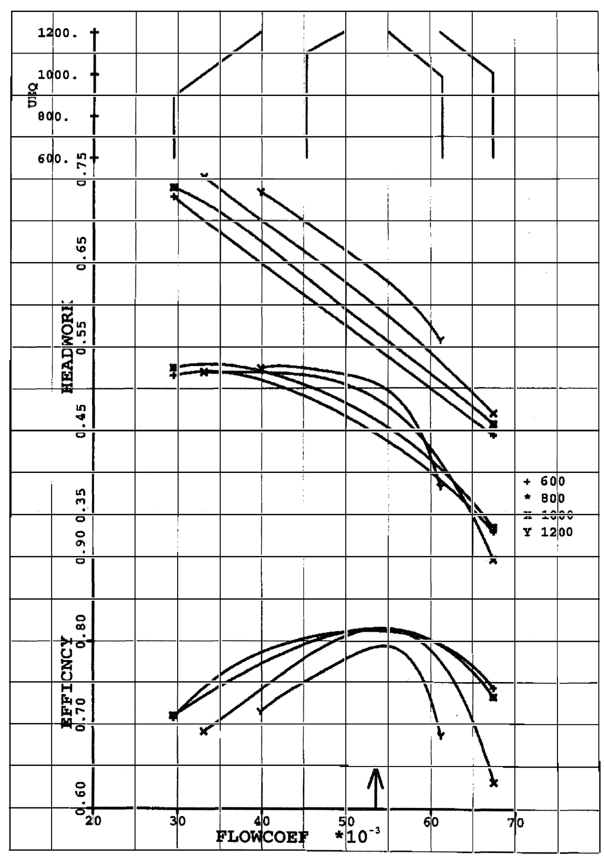

Figure 14. Performance Curve for Vaneless Diffuser Stage.

permitted, a neutral buffer gas must be injected along the seal path. This neutral buffer then mixes with the process gas and also exhausts to the atmosphere or to flare. The limiting factor for the application of labyrinth seals is the ability to get buffer gas in and out of the seal and to avoid pressurizing the bearing housing. Labyrinth seals can have stationary or rotating teeth, and the labyrinths can be straight, stepped, or interlocking. The labyrinth teeth can be made of aluminum, carbon steel, stainless steel, or composite materials, depending on the particular service. The mating component is normally steel or an abradable (such as lead or fluorosint) material.

Carbon ring seals are frequently applied to chlorine compressors and to lower pressure applications. Carbon ring seals, like labyrinth seals, are clearance seals, and usually consist of multiple carbon rings located by steel spacers, with or without springs. The carbon rings float on the shaft and move with the shaft, automatically compensating for shaft displacement. This arrangement inherently has smaller clearances than a labyrinth seal and, thus, lower leakages. Closer clearances are possible since carbon is "self-lubricating." However, because of the smaller clearances, carbon rings wear more rapidly than do labyrinths and generally must be replaced more often than labyrinth seals. The disadvantage of carbon ring seals is the potential for ring breakage caused by the rings hanging up on the packing box wall. This situation can cause excessive rubs on the shaft and may eventually score the shaft. As a rub occurs, the shaft heats up and expands, compounding the rub, and resulting in enlarged clearances or broken rings.

Mechanical contact seals are used in almost any compression service for operating pressures up to 750 psig (52 barg) and shutdown pressures up to 900 psig (62 barg). It is an effective seal that can be used in applications where little or no process gas leakage can be tolerated. The basic sealing device in the mechanical contact seal consists of a stationary contact sleeve and a rotating seal ring with a carbon ring sandwiched in between. The sealing faces are ground to a high degree of flatness to provide maximum contact. The seal faces are held together by the force of springs located around the periphery of the seal. Oil is supplied to the seal at some pressure differential above the gas pressure for lubrication, cooling, and positive sealing action. This results in a slight flow of seal oil across the seal faces and prevents the outward flow of process gas.

Bushing seals are suitable for operating pressures up to 4000 psig (275 barg). For this type of seal, seal oil is injected between two sleeves or bushings that run at close clearance to the shaft. The seal oil, in addition to acting as a sealant, also provides cooling and lubrication for both the rotating and stationary seal sleeves.

Oil free dry gas seals have been gaining in popularity in recent years and have been successfully applied in various compressors and applications. Dry gas seals consist of a stationary spring loaded ring and a rotating mating ring that can be either grooved or notched. As the shaft rotates, the mating ring design forces gas inward toward the ungrooved part of the face, which is in close tolerance with the stationary sealing face. The small gap between the two rings restricts gas leakage to a very small amount. There are several gas seal configurations available—single seals with one sealing face, double seals with two sealing faces on opposite sides of a common mating ring, and tandem or triple seals, which are multiple versions of the single seal. The major benefits of gas seals are the elimination of sometimes complicated seal oil systems and lower fugitive emissions. This can translate into lower utility costs, fewer shutdowns, and lower gas vent flows. Gas seals also have very low mechanical losses when compared with mechanical contact or bushing seals. However, gas face seals require a clean and dry buffer source. Buffer systems for dry gas seals must include filters, and usually include stainless steel piping that adds to the cost of the overall system. Buffer systems can range from simple designs to elaborate systems with a plethora of instrumentation.

Existing compressors can usually be retrofit with a seal type different from the original seals, if it is economically feasible to do so. Mechanical seals and bushing seals are often replaced with dry gas seals to save on utility costs. In many cases, this is easily done with minor modifications to the existing seal housings and to the shaft sleeves. The supplier must make sure that there are adequate connections on the seal housings and endwalls to accommodate the buffer and leakoff porting required for the gas face seal. Rotordynamics must also be evaluated by the compressor supplier any time that oil lubricated seals are replaced with dry gas seals, since the amount of damping at the seals is different for the dry gas seals. If a seal retrofit is contemplated, it may make sense to make the seal change at the same time that a unit is rerated. This would reduce the amount of downtime and eliminate the need for an extra shutdown to make the seal change.

Impeller Eye and Interstage Shaft Seals

Improvements in stage efficiencies over the last 20 years are attributable in part to the application of abradable type seals at impeller eyes and at interstage shaft seals (Figure 15). In the past, aluminum was widely used as the labyrinth material for these seals, and the labyrinth teeth were stationary (Figure 16). Aluminum seals have conservative clearances that result in higher recirculation flows around the impeller, thereby lowering the individual stage efficiency and increasing the power requirement. Developments have been made in the application of thermoplastic polymers to labyrinth type seals. Polymer seals offer improved corrosion resistance (versus aluminum) in such services as wet gas or wet hydrogen sulfide. Clearances for polymer seals are normally slightly smaller than those used for aluminum seals, so they provide a little advantage performance-wise, but not on the order seen for abradable seals. Abradable seals (Figure 17) normally consist of a set of rotating steel labyrinth teeth and a stationary component made of mica filled tetrafluorethylene (TFE), silicone rubber, nickel-graphite, or aluminum silicon polyester, to name a few. The use of abradable seals allows tighter clearances, while limiting the risk of damage to the rotating and stationary seal components if a rub does occur. This is especially important for compressors that operate above the first lateral critical speed and must pass through the first critical during startup and shutdown. The rotating labyrinth teeth provide an additional positive feature in that they are self-cleaning through centrifugal force. In the event that a rub does occur, the steel teeth will cut into the stationary abradable material and remove material. The effective clearance between the seal teeth and stationary seal remains unchanged and leakage through the seal will not increase significantly (Dowson, et al., 1991). The tighter clearances used for abradable seals can cut recirculation flows by as much as 50 percent when compared with conventional aluminum seals. This increases the flow potential for a particular stage, while improving the overall stage efficiency. Abradable materials may also be applied to balance piston seals for inline compressors and center seals for back-to-back compressors. Again, for these seals, the advantage is in decreased leakage that reduces the amount of gas that must be compressed. For inline machines with balance piston seals, reduced balance piston leakage also reduces the temperature increase at the point to which the balance piston line reenters the compressor. This combined effect improves the overall efficiency of the compressor and reduces the power requirement.

COMPRESSOR RERATES

Existing compressors can usually be rerated when operating conditions change enough to warrant such an expenditure. Oftentimes, technological advances are made in processes such that plant output can be significantly increased by changing or upgrading process components. Other times process demand may decrease or a new application may be identified for an idle existing machine. Improved efficiency may also be a driving force to justify rerating an existing unit.

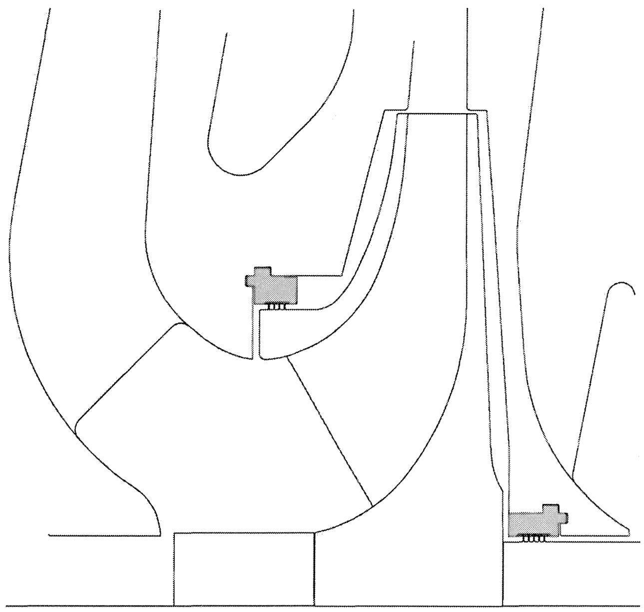

Figure 15. Impeller Eye and Interstage Shaft Seals.

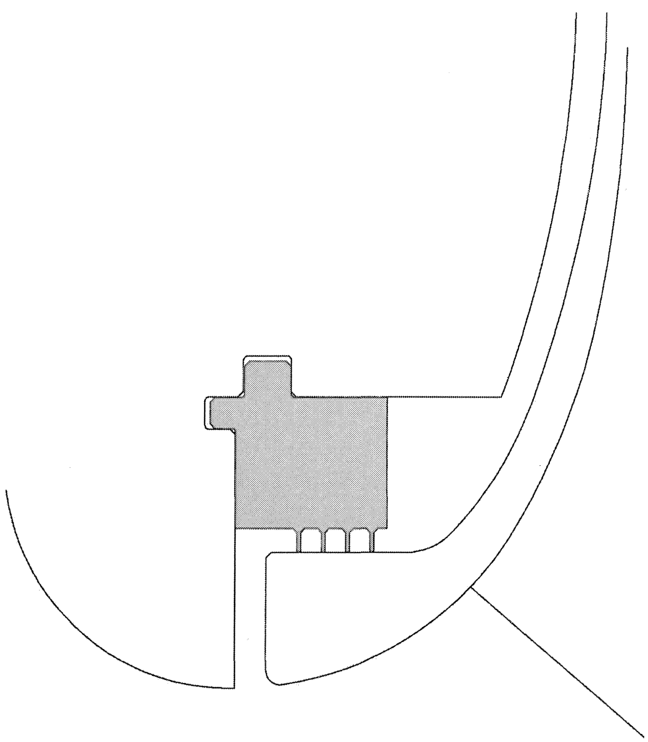

Figure 16. Impeller Eye Seal with Stationary Labyrinth Teeth.

As part of the strategy for how the compressor supplier handles a request for a potential rerate, the user should specify what is to be accomplished by rerating the compressor in question. The user should indicate whether the existing driver will also be reused or if a new driver will be provided. The user should also state if the rerate is to maximize the use of the existing components or if the machine should use all new components to obtain optimum performance at the given operating conditions. Sometimes the conditions change so much that the existing parts cannot be reused. The condition of the existing parts also plays a part in the decision as to whether they can be reused while rerating the compressor. For parts to be reused, they must be thoroughly inspected using

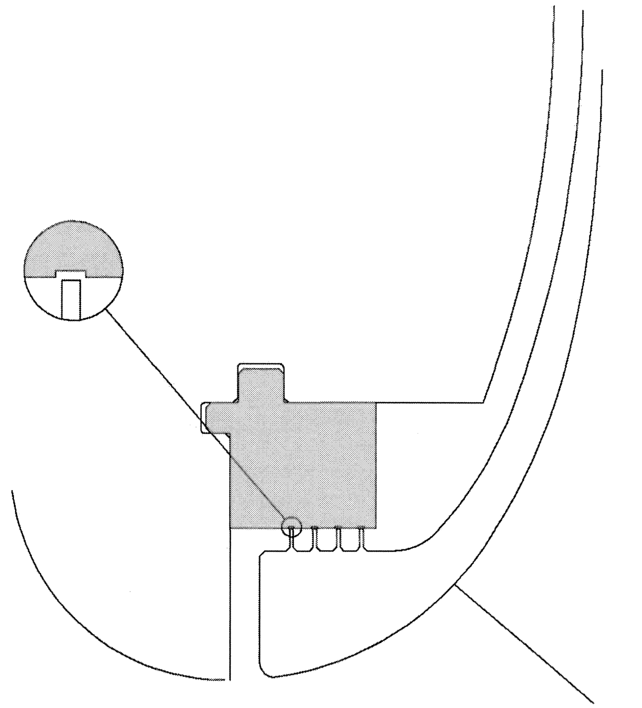

Figure 17. Abradable Type Impeller Eye Seal with Rotating Teeth.

nondestructive testing methods (i.e., magnetic particle inspection, liquid dye penetrant, ultrasound, etc.) and verified to meet the minimum standards for the original material. Damaged parts need to be either repaired (if possible) or replaced. Reusing existing parts must be weighed against expected efficiency improvements that could be realized if all of the compressor internals are changed out.

Reusing Parts

This brings up another important consideration for rerating a compressor. To reduce costs, it may be desirable to reuse the existing shaft(s) and as many of the existing impellers as possible. However, reusing existing components may increase the downtime required for the turnaround or may leave the user exposed for a time without a spare rotor. The problem here is that if components are to be reused, they have to be available to the compressor supplier so that they can be installed. If all new components are ordered, it may be more costly, but the supplier can manufacture the required parts without affecting the user's plant production schedule. If parts are to be reused from an existing spare rotor, the spare rotor must be shipped to the supplier so that the reused components can be removed from the existing rotor and installed on the rerated rotor. To minimize the effect on the leadtime for the rerated rotor, the spare rotor may have to be shipped to the supplier before the planned shutdown of the plant. This will leave the user exposed without a spare rotor should some problems occur. If there is no spare rotor, the rotor being run in the compressor would have to be shipped to the supplier. In this case, the rotor would not be available until the plant shutdown and the supplier cannot complete the rerated rotor assembly prior to the planned shutdown. The rotor (or components to be reused) would have to be removed from the casing, cleaned, boxed, and shipped to the supplier. The supplier would then receive the rotor, remove the components to be reused, and inspect these components. Assuming that the components are in reusable condition, they would then be installed on the new rotor assembly. If the components to be reused are not in acceptable condition, they must be repaired or replaced with new components, which would also add to the leadtime required for the rerate rotor assembly.

Compressor History

From the supplier side, a good background check of the existing equipment is essential to performing a thorough rerate study. What is the current machine configuration? What type of driver is used? What is the operating history of the machine? Has it been performing well or poorly? Have there been any mechanical problems? Has the compressor been rerated previously? Such questions need to be considered.

Current Conditions

It is extremely important that user process simulations, including the equipment performance model, be checked and verified by comparing with current plant operating data. Small errors in measuring the current machine operating pressures, temperatures, and flows can have significant impact on the performance calculations. Therefore, multiple measurements of these parameters are preferred.

Measurements should be made using calibrated instrumentation. The location and the quality of the sensing points are also quite important. Users must understand that sometimes small changes in rerate process conditions can increase the likelihood that a compressor can be reapplied. For example, when the inlet mass flow (lb_m/min) must be increased, raising the suction pressure will help to keep the actual inlet volumetric flow (ft^3/min) down. For compressors with sideloads or extractions, such as refrigeration compressors, allowing some variation in the intermediate pressures will increase the probability of rerating the compressor. Any allowable tolerances for sideload and extraction pressures need to be communicated between the user and the supplier.

Compressor Summary

Working on a rerate is much different from working on new equipment, since the size of the existing casing imposes additional constraints with respect to the aerodynamic hardware that can be used. Making a summary of the compressor in its current configuration is a good way for the application engineer to familiarize himself with the compressor under consideration. The following information can be tabulated:

- Identification number
- Frame size
- Casing design pressure
- Hydrotest pressure
- Maximum continuous speed
- Impeller spin speed
- Nozzle sizes
- Stage lineup
- Impeller diameters
- Impeller materials
- Shaft diameter at impellers
- Nominal shaft end diameter
- Journal bearing size and type
- Thrust bearing size and type
- Main casing seals size and type
- Balance piston/center seal diameter

This tabulation serves as a quick reference as the study progresses.

Rotor Sketch

After the summary is completed, a rough sketch of the compressor rotor should be made to define the axial space available for each stage and to show the casing nozzle and bearing locations. This information is most readily obtained from the compressor instruction book or from the original supplier's files. The space available in each compression section will be limited by the location of any intercooler, sideload, or extraction nozzles, since the existing nozzles will be reused and the location of holes in the casing for these nozzles cannot be changed.

Once the current configuration of the existing machine is established and the available space is defined, it is time to perform an initial hardware selection. The goal here is to optimize the hardware selection within the boundaries of the existing casing. It will not always be possible to use the highest efficiency staging since the available space is limited. Usually, a compromise has to be made between best efficiency and what will fit in the casing.

Rotordynamics

Rotordynamics can be simpler for rerated versus new equipment, especially if it is known that the compressor in its existing configuration is operating satisfactorily. For rerates, the bearing span is fixed and the shaft diameters do not normally change. In such instances, only the impeller and sleeve weights would change. Also, if the required shaft end diameter need not be changed, there is a high probability that the existing coupling may be reused (although the service factor may be reduced). Given the operating history and knowing that only the rotor weight will change, gives a fairly good indication that the compressor will operate somewhat the same as current after it is rerated.

Other Considerations

Many of the areas that are checked for a new machine should also be checked for a rerated compressor. These include impeller stresses at the proposed operating conditions, nozzle velocities, casing design pressure limits, casing temperature limits, shaft end sizes, couplings, and volute sizing. When existing impellers are reused in the rerate, it must be verified that the existing impellers' materials are suitable for the new conditions. This can especially be a concern for low temperature compressors where the impeller materials are changed throughout the machine, based on the expected operating temperatures. For instance, the first stage in an ethylene compressor often has to be designed for a minimum operating temperature of −150°F (−101°C). If an existing stage is moved to the first stage position to accommodate a new set of operating conditions, it must be verified that the material for the existing impeller is acceptable for a minimum temperature of −150°F (−101°C). If it is not, a new impeller must be fabricated using an acceptable material.

Nozzle Velocities

Although nozzle velocities for rerate conditions should be checked, the velocities cannot be reduced unless the conditions are changed, since the nozzles themselves are fixed and cannot be replaced. Using the guidelines for Equations (3) and (4), the engineer needs to determine the pressure drops associated with higher velocities and account for them in the compressor selection. If nozzle velocities exceed the maximum limits, the user should be notified.

Shaft End Sizing

The required nominal shaft end size(s) for the required duty must be calculated per Equation (9) or by using a similar method. If the required shaft end size exceeds the existing shaft end size, the existing shaft cannot be reused as-is. There are three options to choose from with respect to the shaft:

(1) A new shaft with a larger shaft end diameter can be supplied,

(2) The shaft end diameter for the existing shaft can be increased by weld overlay, or

(3) A new shaft made of higher strength material could possibly be provided if there is an acceptable high strength material.

Options (1) and (2) would require a new coupling, whereas for option (3), it may be possible to reuse the existing coupling, if it is suitable for the higher torque requirements. The shaft end maximum size is limited, based on bearing sizes as noted above. Changing the shaft end size will also affect the coupling. The coupling manufacturer must be consulted to determine if the existing coupling or some parts of the existing coupling can be reused for the new operating conditions.

Volute Sizing

It is extremely important to check volute sizing at the rerate conditions. An undersized volute can have a very detrimental effect on compressor performance. The higher losses associated with an undersized volute will also affect the performance of downstream stages and downstream compressor bodies.

CONSIDERATIONS FOR THE FUTURE

When specifications are written for a new compressor, the user should consider whether there is potential for increasing the capacity in the future. Too often, shortsighted economics dictate that lowest cost drive a project without accounting for future changes. Purchasing a compressor frame that is near its maximum flow may seem to be a good idea now when it saves a significant amount of money, but what happens five or 10 years into the future? Maybe at that time, a 30 percent increase in capacity can generate multiples of the savings initially realized; but the increased capacity requires a new compressor, a new driver, and a new foundation, in addition to larger process vessels and extensive piping changes. Designing *now* for the future can potentially save a lot of money and a lot of headaches later. How can a compressor be designed today to meet future needs? A few areas to consider are: casing design pressure, impeller overspeed, shaft end sizing, nozzle sizing, volute sizing, and stage spacing.

Casing Design Pressure

With respect to the casing design pressure, the user should be careful to make sure that the casing they are purchasing can cover any future operating conditions. API 617 contains guidelines for casing design pressures that should be followed whenever possible. It may be advantageous for users to specify some additional margin for casing pressure beyond the API 617 requirements, but this must be reviewed on a case-by-case basis, and is subject to the actual casing design limitations of the suppliers' casings. Requiring additional pressure margin for casing ratings could potentially mean that a particular casing will not have to be retested, if the discharge pressure increases over the life of the casing or if the casing is reapplied in the future to some service that has a higher pressure requirement. Having to remove a casing from its foundation and rehydrotest can be a very costly and time consuming project.

Impeller Overspeed

Industry standards dictate that individual impellers be overspeed tested to 115 percent of the maximum continuous speed for the particular service being considered. Some user specifications are more restrictive. Raising the impeller spin requirement to something like 120 percent or 125 percent of maximum continuous speed is too restrictive for most applications. Such a requirement may dictate that lower efficiency impellers be used for a particular application. However, it may make sense to go a little beyond the 115 percent requirement to a value such as 118 percent for impeller overspeed *if such a requirement does not compromise the compressor selection.* Adding to the overspeed margin works to:

- Preclude the need to respin impellers to meet industry standards when shop or field tests show that a speed increase is required to meet the specified operating conditions and
- Provide additional margin should the impeller be considered for use in some future uprate conditions.

An increase in speed of three percent is fairly significant, since such an increase will produce as much as nine percent additional head. *But this only makes sense when it does not compromise the compressor selection.* Having to change from a high efficiency impeller to one with lower efficiency only to meet a 118 percent overspeed requirement may not be in the user's best interest. In such instances, the best approach would be to overspeed test to 118 percent of the maximum continuous speed all of the impellers that can meet such a requirement, and test the other impellers at their design limit (between 115 percent and 118 percent of the maximum continuous speed).

Shaft Ends

Shaft ends can be sized using the procedure noted in Equation (9). Although there is a margin of safety built into this calculation procedure, future uprate conditions often require larger shaft end sizes due to higher torque requirements. When a larger shaft end size is needed, an existing shaft cannot normally be reused. But specifying larger-than-required shaft ends can detrimentally affect critical speed margins, especially on drive-through machines, by increasing overhang lengths and overhung weights. Larger shaft ends can also increase coupling sizes and costs. If initial costs are of primary concern for a project and it is doubtful that a particular compressor will ever be uprated, shaft ends should be sized in the normal manner. However, if there is a high probability that capacity will be increased in the future, it may make sense to size the shaft ends to the next larger increment above the current requirements *where possible.* If critical speeds are adversely affected such that the compressor no longer meets required critical speed margins with the larger shaft ends, the shaft ends should not be oversized. If, on a drive-through machine, you are using the maximum shaft end size that the standard size thrust bearing allows, the shaft ends should not be oversized. It would not make sense to have to use a larger frame size compressor just to meet a criterion that a shaft end size be larger than is required.

Discharge Volutes and Nozzle Velocities

For any new machine selection, intercooler discharge volutes and final discharge volutes should ideally be sized to accommodate at least 20 to 25 percent additional flow. This will ensure that the volutes are adequately sized for the current operating conditions and will provide a reasonable amount of margin for any future conditions. For new compressors, nozzle velocities should be limited to 110 to 120 ft/sec (34 to 37 m/sec) or less (depending on the molecular weight) for any specified operating conditions.

Axial Space

Higher flow stages generally require longer axial space. The fact that axial space is fixed for an existing compressor can severely limit the choices for aerodynamic components when a unit is rerated. To improve this situation, it is possible to build extra axial space into a new compressor *if critical speed margins will allow.* One method of accounting for this is to have the compressor supplier verify that the next larger impeller (as compared to the impellers selected for the current conditions) could be installed in the casing under consideration. Although such a requirement would not cover all future requirements for all services, it is a reasonable way to design for future needs.

HOW MUCH IS TOO MUCH?

Purchasing a compressor that is designed and built to meet only the initial operating conditions may be desirable for some users, and such a practice will certainly limit initial plant costs. The end user needs to consider the future, since decisions made today can have a big impact on future equipment expenditures. It also needs to be recognized that guidelines are just that—guidelines. Sometimes it makes sense to not follow them if there is a good reason. Imposing more stringent requirements always adds associated costs in the form of higher cost materials, larger more expensive components, and sometimes even lower efficiencies to meet strict mechanical specifications. It is also difficult to make decisions when many projects are current-cost, bottom-line driven. Common sense, along with business decisions based on economics, has to play a part in answering the question "How much is too much?"

CASE STUDY

As part of a comprehensive uprate to increase the output of a hydrocarbon plant, the new operating conditions for an existing propylene compressor with two sideloads required an increase in the inlet volume flow and the sideload flows. Suction pressure and discharge pressure both increased for the uprate condition. A summary of the performance changes is shown in Table 1.

Table 1. Summary of Performance Changes.

Parameter	Change (Uprate vs. Original)
Process Gas	No Change
Inlet Volume Flow	+15.8%
Mass Flow	+31.1%
Molecular Weight	No Change
Inlet Pressure	+14.1%
Inlet Temperature	+5.5°F
1st Sideload Pressure	−6.6%
1st Sideload Temperature	−5°F
1st Sideload Volume Flow	+47.5%
1st Sideload Mass Flow	+39.0%
2nd Sideload Pressure	+0.5%
2nd Sideload Temperature	No Change
2nd Sideload Volume Flow	+28.1%
2nd Sideload Mass Flow	+29.6%
Discharge Pressure	+8.2%
Discharge Temperature	+10°F
Shaft Horsepower	+41.1%
Speed	−3.7%

Note that for the rerate condition, the sideload pressures specified by the user were at specific pressure levels. A tolerance band was allowed for these pressures. In the final rerate configuration, the expected pressures at the sideload nozzles calculated by the compressor supplier were different from those specified. The pressure calculated at the first sideload was 0.5 psia higher than specified, while the pressure calculated at the second sideload was 1.6 psia higher than specified. Using a performance simulation program, the user determined that the sideload pressures calculated by the compressor supplier were acceptable.

The first step in considering any compressor rerate, is to gather information. For this particular compressor, the original design specifications and compressor layout were available for reference. As a first step, a compressor summary was made, as shown in Table 2.

Table 2. Compressor Summary.

Frame Size	70M8-6
Casing Design Pressure	250 psig
Hydrotest Pressure	375 psig
Maximum Continuous Speed	3435 rpm
Impeller Spin Speed	4053 rpm (all impellers)
Nozzle Sizes	48 inch inlet
	20 inch first sideload
	16 inch second sideload
	30 inch discharge
Impeller Lineup	whl -whl - sl - whl - whl - sl - whl - whl
Impeller Diameters	46 inch (full diameter), 44.9 inch (full)
	46.8 inch (full), 45.6 inch (cut)
	46.8 inch (full), 44 inch (cut)
Impeller Materials	low temperature carbon steel (! 50EF) for the first impeller; all other impellers carbon steel with a minimum allowable temperature of ! 20EF
Shaft Diameter at Impellers	14.25 inches
Nominal Shaft End Diameter	6.0 inches
Journal Bearings	8 inch diameter, spherically seated
Thrust Bearing	91.1 square inches
Main Casing Seals	Tandem dry gas face seals
Balance Piston Diameter	27.75 inches

The next step was to make a rough sketch of the bearing span to show the location of all impellers and nozzles. The sketch for this particular compressor is shown in Figure 18. The sketch shows the axial space that is available at each stage, which aids in determining which stages will fit, with respect to a reselection to meet the new higher flow condition. The top part of the sketch shows the existing internals; the bottom part of the sketch is used to show the rerate selection of stages.

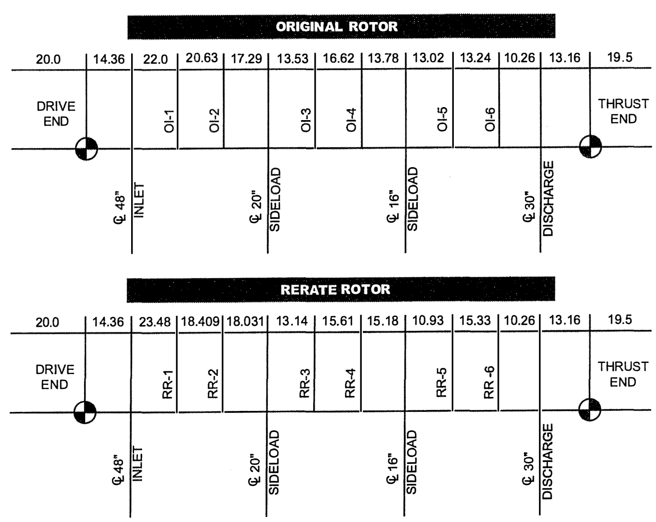

Figure 18. Rotor Sketch for Compressor Rerate.

After making the compressor tabulation and rotor sketch, a hardware selection was made. The intent here was to meet the specified conditions as closely as possible while selecting stages with axial space requirements similar to those of the existing stages. Once the preliminary selection was made, the axial space requirements were checked to make sure that the chosen aerodynamic components would fit in the available space. As part of the selection review, sectional maps were run to determine the maximum pressure that could potentially be developed by the last compression section. For the rerate conditions, the maximum pressure developed by the third section, based on surge at maximum continuous speed, is 228 psia, which is below the casing rating of 250 psig, so the existing casing design pressure is acceptable for the new conditions.

Based on the new selection, a horsepower of 28,735 at a speed of 3146 was required to meet the new operating condition. The shaft end size was then checked to determine the fate of the existing shaft. From Equation (9), we have:

$$\text{S.E.} = \sqrt[3]{\frac{321000\ (\text{hp})\ (\text{DSF})}{(\text{S})\ (\text{N})}} \tag{11}$$

The torsional shear stress for the existing shaft material is 12,500 psi. Therefore, we have:

$$\text{S.E.} = \sqrt[3]{\frac{321000\ (28735)\ (1.1)}{(12500)\ (3146)}} \tag{12}$$

$$\text{S.E.} = 6.37$$

The new conditions require a minimum shaft diameter of 6.37 inches. Since the existing shaft end size is 6.0 inches, a new shaft and a new coupling were required for the rerate. The new operating speed is 3146 rpm, which requires a maximum continuous speed of:

$$1.05 \times 3146 = 3303 \text{ rpm} \tag{13}$$

Since the original maximum continuous speed was 3405 rpm, this would not pose any problems for any of the existing impellers should they be reused as part of the rerate. However, as it turns out for this particular compressor, the flows changed enough so that none of the existing impellers could be used for the rerate.

For the original selection, the required area for the discharge volute based on the guarantee point operating conditions is 131 in^2. For the rerate conditions, the required volute area is 164 in^2. The particular volute used in this compressor has an area of 320 in^2, so the existing volute is acceptable for the new flow condition.

The bearing span for this compressor is not changing. The maximum continuous speed is not increasing. Critical speed margins for the original configuration are approximately 42 percent for first critical and 60 percent for second critical. With the new impellers required for the rerate, the rotor weight is increasing by approximately 10 percent, a larger shaft end is needed, and a new coupling is required. Although these changes would lower the critical speeds, the machine in its current configuration is operating satisfactorily, and the unbalance response report for the original machine and mechanical test data show that even with the increase in rotor weight, there should be no problem in meeting the required critical speed margins.

A comparison of the nozzle velocities for the original and rerate conditions is shown in Table 3.

Table 3. Nozzle Velocities Comparison.

Nozzle	Original	Rerate
48" Main Inlet	80 ft/sec	93 ft/sec
20" First Sideload	76 ft/sec	112 ft/sec
16" Second Sideload	75 ft/sec	96 ft/sec
30" Discharge	46 ft/sec	57 ft/sec

All of these nozzle velocities are well below 150 ft/sec. Using the $\sqrt{\theta}$ method to check the nozzles, Table 4 shows the flow limits for the nozzles.

Table 4. Nozzle Flow Limits.

Nozzle	$\sqrt{\theta}$	Flow Limit	Actual Flow
48" Main Inlet	1.51	88000 cfm	65909 cfm
20" First Sideload	1.48	15600 cfm	13591 cfm
16" Second Sideload	1.43	10200 cfm	7328 cfm

All of the flows for the rerate conditions are well below the flow limits given in Figure 1.

As an exercise, let us also look at the ratio of actual velocity to acoustic velocity (Table 5).

Table 5. Ratio of Actual Velocity to Acoustic Velocity.

Nozzle	Acoustic Velocity	Velocity	Ratio
48" Main Inlet	757.7 ft/sec	93 ft/sec	0.122
20" First Sideload	773.1 ft/sec	112 ft/sec	0.144
16" Second Sideload	800.8 ft/sec	96 ft/sec	0.120

Only the ratio of inlet velocity to acoustic velocity for the first sideload is over 0.132. However, since it is only nine percent above 0.132, no additional pressure drop must be accounted for at the first sideload nozzle.

CONCLUSION

When purchasing a new compressor or rerating an existing one, it is important to carefully consider safety, reliability, maintainability, and environmental soundness because of the tremendous financial losses that can result from unscheduled plant outages. The time and effort spent considering future uprateability of new compressors is time well spent. This paper gives the rotating equipment engineer guidance when evaluating the design of a new or rerated compressor. To minimize the amount of plant downtime, the rerate should be scheduled to coincide with a planned plant shutdown. Spare equipment (rotors, casings, etc.) should be considered for retrofit, and the necessary work should be performed in a shop rather than in the field whenever possible. With the proper planning, teamwork, and coordination among users contractors, and suppliers, compressor rerates can be a cost effective, reliable means of increasing plant production, while minimizing disruption to plant production.

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