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Centrifugal Compressor Configuration, Selection and Arrangement: A User's Perspective

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ABSTRACT

The detailed sizing and selection of centrifugal compressors may be somewhat of a mystery to the typical user. Once the suction and discharge process conditions and required flow rates are established, they are submitted to an equipment supplier for sizing and selection. The resulting combination of impeller diameters, casing sizes and operating speed determined by the equipment supplier can be puzzling to the engineer charged with evaluating the selections.

This tutorial is intended to introduce the dimensional and dimensionless similarity parameters that can be utilized to perform an independent, equivalent selection for a compression application or provide a more thorough evaluation of the selections provided by an equipment supplier. These performance parameters will allow prediction of design point head and efficiency, approximate impeller diameters, and approximate compressor operating speeds. Furthermore, the inter-relationship between impeller diameter and rotational speed will be investigated to highlight the trade-offs in changing either of these design variables. The tutorial will also investigate useful mechanical parameters and guidelines that allow comparative preliminary assessment of proposed designs.

When significant head and/or flow requirements exist for a specific application, it becomes necessary to divide the compression service into multiple sections and in some situations multiple casings. Although a few different ways to configure multiple compressor sections exist, there are benefits and issues associated with each of these configurations. The potential benefits and issues of different arrangement options will also be addressed in this tutorial.

INTRODUCTION

Once required process conditions of suction pressure and temperature, discharge pressure, flow rate and gas composition are provided to a compressor supplier, the subsequent actions that are completed to arrive at a compressor selection are largely unknown to the purchaser and user. The resulting combination of impeller diameters and operating speeds are not readily predictable in many applications, and sometimes do not appear consistent with previous experience. The intent of this tutorial is to provide background information and present a relatively simple method to develop approximate, independent selections from the equipment suppliers. While these selections are not intended to replace the more accurate designs proposed by a supplier, they can provide reasonable, preliminary estimates prior to engagement of the supplier and a reference for more informed discussions with the equipment supplier.

A review of the most commonly utilized compressor configurations will first be covered. Configuration is defined as the grouping and orientation of impellers within a given compressor casing. Different configurations have been developed to address varying process requirements and applications. Benefits and limitations of each configuration will be presented, providing information behind why a specific configuration may be selected.

Next, a reasonably accurate method will be introduced that allows independent compressor selections to be developed. While these are not intended to replace selections provided by equipment suppliers, they should be comparable and provide insight into supplier selections and potential variations in selection parameters. The method will be developed from definition of required process parameters through calculation of relevant dimensional quantities needed, and then to the introduction and utilization of a selected number of dimensionless parameters and relationships that allow the more definitive design parameters of impeller diameters, rotational speeds and polytropic efficiencies to be predicted. Variations in the number of impellers, impeller diameters, and rotational speed may also be explored for their impact on predicted performance at design point conditions.

Finally, as pressure ratio and flow demands increase for particular applications, additional compressor sections and casings are necessary to meet these requirements. Various arrangements of compressor casings can be provided that address these increased demands. Driver selections may also influence both compressor selections and arrangements. Fundamental forms of these arrangements along with their benefits and negatives will also be explored.

CENTRIFUGAL COMPRESSOR CONFIGURATION

It is appropriate to begin an evaluation of centrifugal compressor design and selection by reviewing the different machinery configurations that are currently available to address the wide number of potential applications where they might be utilized. Accordingly, this discussion will begin with an introduction and review of various design configurations, however, it is first necessary to establish some descriptions of components and sub- components that are included in a typical centrifugal compressor. Some of this terminology is used interchangeably within the industry, so it is important to provide definitions that will be used for consistency in this document. These are:

- Stage: A stage is defined as a single inlet section, impeller and discharge section including diffuser, return bend, and return passage or, if applicable, discharge volute.
- Section: A section is defined as one or more stages arranged with a single inlet and discharge without any intermediate extraction for cooling or mass addition or extraction. An individual section is designated by a yellow trapezoid in the following figures.
- Casing: A casing is defined as a pressure containing structure that surrounds one or more compressor sections, supplied with two or more main process piping nozzles, and shaft connections for power input. Casings are designated by blue rectangles with dashed black boundaries in the following figures.
- Train: A collection of one or more drivers and compressor casings with connected inlet and discharge flows between individual casings. Multiple compression trains are normally supplied in parallel to achieve total flow and head requirements of a specific application.

While it is impractical to introduce all current and potentially known compressor configurations, the intent herein is to provide a description along with supporting information for the most commonly applied compressor designs. As potential new applications are introduced and designs evolved, it will be important for the user and purchaser to evaluate and understand the relative strengths and weaknesses of any given design.

Straight-Through Configuration

The most fundamental configuration of a centrifugal compressor is the straight-through design. This layout is composed of one or more impellers, aligned in the same direction, contained within a single casing fitted with a single inlet nozzle and a single discharge nozzle to accommodate the gas flow. A schematic diagram of the straight-through configuration is provided in Figure 1. While the number of stages contained within the section is a function of the produced head requirements, there are practical limitations to the number of stages that can be included.

One of these limitations is connected with the resulting discharge temperature that is a function of the overall pressure ratio, compression efficiency, and gas thermodynamic properties. This temperature limitation may be due to material temperature limits of components within the compressor or gas temperature limits imposed by other equipment or the process in which the compressor is operating.

An important limitation in the number of stages allowed in a single, straight-through configuration is associated with lateral rotordynamic considerations. Current beam-style designs limit the number of stages in a single casing to 10 or less, however, this may be subject to the magnitude of the flow coefficients of the stages. This is a subject that will be covered in more detail later. Beam-style

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compressors are configured with all of the impellers and a balance piston, if applicable, on the rotor located between two radial bearings. Lateral rotordynamic stability is one of the primary considerations in limiting the number of stages to 10 or less. Regardless of the number of stages provided in a design, it is good practice to provide a preliminary stability screening utilizing the evaluation method of the critical speed ratio (CSR) plot provided in API 684, “API Standard Paragraphs Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals, and Rotor Balancing.” This analysis examines both the rotor flexibility (in terms of the rotor operating speed to first rigid critical speed ratio) and the average density of the gas which impacts the magnitude of the aerodynamic excitation forces generated. Other similar figures that have been generated through the years have replaced average gas density with the product of discharge pressure and pressure differential across the compressor. The information required to evaluate rotordynamic stability using this plot is normally available on API 617, “Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services,” data sheets. Another useful parameter that can be calculated using data sheet information is the bearing span-to-impeller bore diameter ratio of the rotor. It is more likely that lateral stability is provided if this ratio is 10 or less. Values above 10 should be evaluated in more detail, and any value above 12 should be considered for an API 684 Level II stability analysis.

The pressure profile that is present around each impeller results in an unbalanced axial thrust force in the direction from the back disk towards the eye of the impeller. A summation of these unbalanced forces represented by each impeller are balanced by the combination of a balance piston at one end of the rotor and a thrust bearing. The balance piston is provided with a cross-sectional area where discharge pressure is imposed on one end of the piston and suction pressure on the other, partially negating the unbalanced axial force due to the impellers.

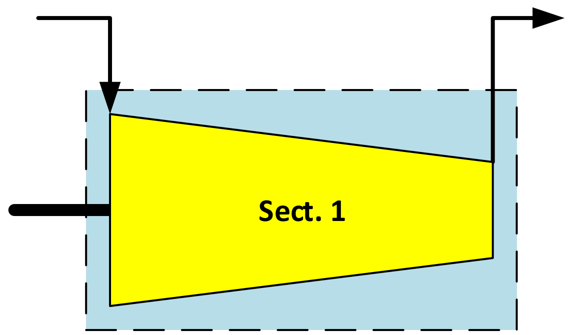

Figure 1: Straight-Through Design Configuration

Variants of the beam-style, single section, straight-through design exist that deserve identification. One of these is the overhung, single stage design that is commonly utilized in pipeline applications where a relatively low pressure ratio is required. The layout of this design includes a single impeller placed outside of the bearing span. The number of impellers included in this design is normally limited to a single impeller due to overhung moment influences on lateral rotordynamic behavior.

Another variant of the single section, straight-through configuration is the integrally geared compressor. In the simplest form of this design, a single impeller is directly connected to the end of a pinion which is driven by a bull gear to a specified speed. These impellers are often of an open design, which means that the vanes are connected to the back disk but there is no cover provided that encloses the flow path on top of the vanes. Although the lack of an impeller cover increases leakage, this is offset by the increased head generating capability due to higher allowable tip speeds when compared to a shrouded impeller. More complex versions of the integrally geared compressor exist where impellers are attached to both ends of the pinion and multiple pinions are associated with a single bull gear. The rotational speed of each of the pinions can be optimized to enhance the overall efficiency of the compressor.

Compound Configuration

There are a substantial number of applications where intercooling of the gas is required to maintain temperatures at acceptable levels. One solution is to provide multiple, single section casings with intercooling provided between the casings. An alternative is to provide multiple sections within a single physical casing. This is known as a compound section configuration. A benefit of the compound configuration is reduced footprint, since a single casing requires less space than two casings with double the amount of radial bearings and the space required by a coupling between two separate casings.

Additionally, sometimes process requirements dictate the need for a sidestream addition or extraction to be available at some intermediate pressure of the overall compression application. This is another important benefit of the compound compressor configuration. A schematic of the compound compressor design is provided in Figure 2.

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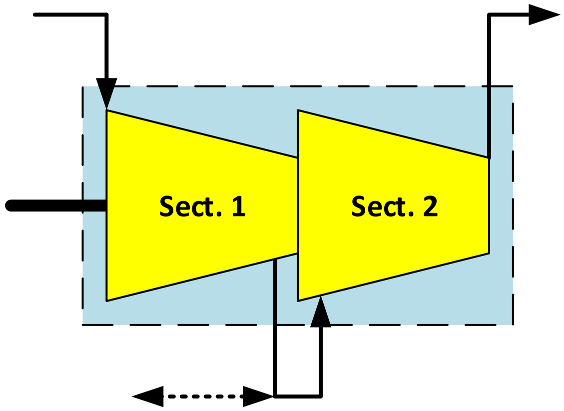

Figure 2: Compound Design Configuration

While there is significant advantage in applying the compound configuration when appropriate, there are some drawbacks to this design. One of these is the fact that inlet and discharge nozzles for each section are required to be supplied on the casing. Adequate spacing to accommodate these nozzles on the casing has to be considered when the equipment supplier lays out the casing internal and external design. Although there are no theoretical limitations on the number of sections contained within a single casing, limits do exist on the space available in a given casing to accommodate a large number of sections.

A pressure differential also exists between the discharge of a given section and the suction of the following section due to pressure losses that must exist in the piping and equipment between the two sections of compression. The differential pressure that exists within the compressor casing results in the potential for leakage from the lower pressure section discharge into the higher pressure section suction. This results in an overall loss of efficiency of the compressor. Some type of a controlled leakage internal seal must be provided to limit the amount of leakage and increase the compression efficiency. The addition of the seal requires some amount of space and a reasonable rule of thumb is that each seal displaces a potential stage of compression. Accordingly, a reasonable estimate of the maximum number of stages available in a two section compressor is nine. This reduces to a maximum number of eight for a three section machine. Notwithstanding the issue with space available for nozzles on the casing, it is fairly obvious that a compound compressor with more than three sections is only feasible for relatively low compression ratios per section.

One final characteristic of the compound design that should be identified is concerned with the transient operation of the machine. Upon shutdown, a compressor will settle-out to an equilibrium pressure somewhere between the suction and discharge pressures that is dependent upon a number of factors, primarily relative suction and discharge volumes associated with the compressor and its connected systems. This settle-out pressure calculation is straightforward for a single section casing. In the case of a casing with multiple sections, however, each section will settle-out to a different pressure level and then all sections will settle out to a common settle-out pressure due to leakage through the controlled leakage seals between each section. Associated piping, vessels, and other process equipment must be designed to accommodate this combined settle-out pressure.

Back-to-Back Design Configuration

A unique version of the compound design is the back-to-back configuration, where the stages of the first section are oriented within the casing opposite to the second section. Generally the eyes of the impellers of each section are oriented towards the shaft ends of the casings. A schematic of this type of compressor configuration is provided in Figure 3.

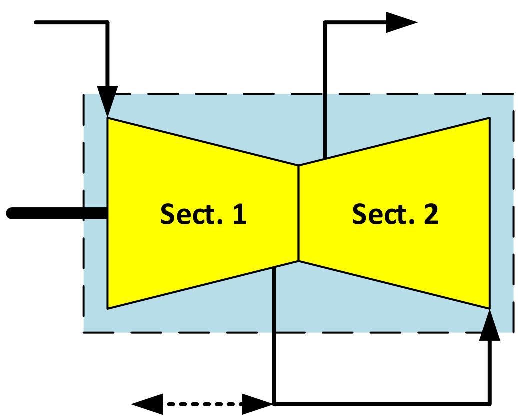

Figure 3: Back-to-Back Design Configuration

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The back-to-back configuration is similar to the compound design with the ability to provide intercooling and mass flow addition or extraction between the sections, although this is an option. A significant advantage of the back-to-back design is its inherent characteristic to reduce, and roughly balance the axial thrust force generated in the stages of each section. Since the two sections are oriented in an opposite direction, the unbalanced axial thrust forces of each section are acting in opposite directions. This is a very distinct benefit in high pressure, high density compression applications such as gas injection services where unbalanced thrust forces can be substantial. In such an application the duty and size of the thrust bearing could be prohibitive without the balancing feature.

A close clearance, controlled leakage seal must be provided between the two sections in a similar fashion to the compound machine design. Unlike the balance piston required in the other designs, this internal, controlled leakage seal (commonly known as the division wall seal) is only subject to approximately half of the overall differential pressure of the two section casing. While this reduced pressure difference has the potential to reduce internal leakage, this can be offset by a possible need for increased clearance due to the location of the seal near the center of the bearing span with the accompanying increased deflection of the shaft in this location. Additionally, the location of the division wall seal near the center of the rotor has previously been a challenge with the potential for aerodynamic excitation. Technology and design developments over the past several years have reduced this issue with the introduction of hole pattern seal designs and swirl brakes.

Final settle-out conditions of the back-to-back design are similar to the compound configuration, where each section settles out to a different pressure level followed by an equalization of pressure between the two sections to a common settle-out pressure. It is critical to design associated piping and equipment to this common settle-out pressure to prevent overpressure of equipment, particularly on the suction side of the lowest pressure section. An additional phenomenon that is observed with the back-to-back design concerns the leakage across the division wall seal and its impact on the flow leaving the first section and entering the second section. This flow will actually be the sum of the first section flow rate, any sidestream flow, and the division wall leakage which may impact inter-stage process piping, process equipment and intercooler duty and design.

Double Suction Configuration

Significant inlet volumetric flow rates can result in excessive magnitudes of impeller flow coefficients. Additional detail on the impacts of high flow coefficients will be explained in more detail, however, beyond a certain point it is beneficial to reduce the flow into an individual section. One way to accomplish this is to provide parallel sections in separate casings. An alternative is to select a compressor of the double suction design configuration. This design is geometrically similar to the back-to-back configuration, but both sections are of equal design. The inlet flow rate is split in half externally and introduced into the casing through two suction nozzles at each end of the casing. Upon passing through the two equal sections, the flow is combined into a single discharge nozzle. This configuration is shown schematically in Figure 4.

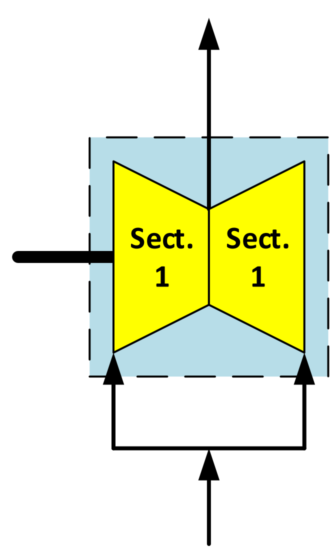

Figure 4: Double Suction Design Configuration

Given that the two sections of the compressor are oriented in separate directions and are aerodynamically similar, there is theoretically no net axial thrust produced. Since there can be differences in both internal and external suction losses and manufacturing tolerances between the two opposed sections, some small amount of axial thrust is anticipated and a thrust bearing is included, albeit limited in capacity.

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The double suction configuration is very useful for large volumetric flow rates that are often associated with low inlet pressures as stated previously. It should be assumed that the maximum number of stages per section is limited to no more than four due to considerations of the dual suction inlets and single, combined discharge. Although possible, this machine design is probably limited to a single section of compression with no more than two suction nozzles and a single discharge nozzle.

Sideload Design Configuration

A final centrifugal compressor configuration that warrants further explanation is the sideload design. This configuration is similar to the compound design in that multiple sections oriented in the same direction are contained in a single casing. The sideload design is substantially different because there are no intermediate intercooled flows that leave and re-enter the casing. One or more sidestream flows may be introduced or extracted from the casing, however, all or most of the flow from the preceding section does not leave the casing. Sidestream flows that are introduced into the casing mix with the discharge of the preceding section, and any cooling occurs through the mixing of the predominately lower temperature sidestream. A schematic of the sideload configuration is provided in Figure 5.

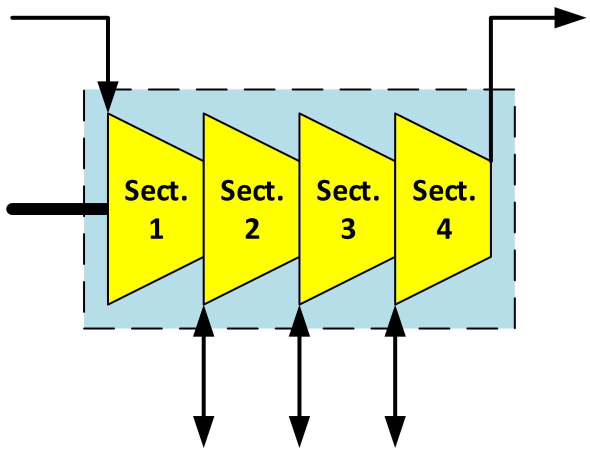

Figure 5: Sideload Design Configuration

The sideload compressor design is well-suited for refrigeration applications where the refrigerant is introduced at progressively lower temperatures to the lower pressure sections. These sidestreams originate from economizers that operate at intermediate pressure levels in order to increase the overall efficiency of the refrigeration process. Temperature increases of the gas in a given section are reduced by the mixing of the sidestream flow into the suction of the following section of the compressor, effectively maintaining the gas temperature through the machine at reasonable levels. It is possible to extract flow from one of these sidestreams, but this tends to be an exception.

Limitations on the potential number of sections and stages per section do exist for the sideload design. Assuming the rule of thumb that a limit of ten stages are allowed in a single casing and that any sidestream nozzles are roughly equivalent to a stage, it is evident that there are limitations to the number of sections that can be contained in such a design. Recent trends in some applications, particularly liquefied natural gas (LNG) processing, are pushing volumetric flows to ever-increasing levels. This results in increasing flow coefficients of individual stages with increasing axial lengths required for each stage of compression. Accordingly, this reduces the number of stages further. This will be discussed in more detail in the next section.

There are additional complications associated with the sideload compressor configuration. One of these is involved with the process control of the machines themselves. Generally each section of compression is protected from the potential to surge through a recycle line dedicated to that section which prevents the volumetric flow rate from falling below a prescribed level. This is complicated in the sideload configuration because recycle flow can only be obtained from the final discharge. Relative effects of flow capacities of each section and the preceding sections influence the selection of these minimum volumetric flows. It is also more difficult to monitor the a aero-thermodynamic performance of these machines since mixture temperatures of the internal gas flows are not measured, thereby making individual section performance only able to be estimated based upon predicted section efficiency.

While it is acknowledged that additional centrifugal compressor design configurations exist (e.g. isothermal designs), they tend to be very unique and limited in application. As noted previously, the configurations that have been described are those that are most commonly observed in industry and warrant the descriptions provided. It should also be noted that multiple configurations or combinations of configurations may be applied to any potential application. This flexibility provides the user and purchaser both opportunities and challenges to select the most optimum combination in many cases.

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CENTRIFUGAL COMPRESSOR SELECTION

The detailed sizing and selection of a compressor section for a specific application depends upon a minimal number of input parameters. These parameters are critical to the accurate sizing of the compressor and are typically available early in the design phase of a project. While there may be multiple sets of conditions that are imposed in a given application, there should be one chosen set of conditions that determine the design point and performance guarantee basis of a selection. The critical parameters are:

1. Gas composition or mixture molecular weight, MW
2. Suction pressure, P_s
3. Suction temperature, T_s
4. Discharge pressure, P_d
5. Gas volumetric flow rate, Qa
6. Required sidestream pressures and temperatures (for multiple section applications)
7. Sidestream flow additions or extractions (for multiple section applications)

Although these parameters are necessary inputs to sizing a compressor, they are not sufficient alone in their basic form to establish the design parameters of the machine.

Polytropic Head and Efficiency

A most fundamental parameter used to describe and design a compressor section for an application is related to the specific work imparted to the gas, namely the required head produced by the machine. The resulting value of head may be described by a few different thermodynamic models, but the most prevalent and flexible in process applications is the polytropic model. An isentropic model also exists, but is more limited in its accuracy and applicability under changing suction conditions. Two different forms of the polytropic head equation are presented below:

$$Hp = C_1 \cdot \frac{n}{n-1} \cdot \left[P_d \cdot v_d - P_s \cdot v_s \right] = C_2 \cdot \frac{Z_s \cdot T_s}{MW} \cdot \frac{n}{n-1} \cdot \left[\left(\frac{P_d}{P_s} \right)^{n-1/n} - 1 \right] \qquad \text{(Eqn. 1)}$$

$$where \ : n = \frac{\ln\left(\frac{P_d}{P_s}\right)}{\ln\left(\frac{v_s}{v_d}\right)}$$

Polytropic head as stated above is the specific work or energy imparted to the gas to raise it from given suction conditions to the discharge pressure, however, this is not the total energy transferred through the compressor to the gas. The sum of the head and this additional energy which is lost work is equal to the total enthalpy rise across the machine. The polytropic efficiency is defined as the ratio of the polytropic work to the total enthalpy increase observed in the gas, or:

$$\eta_p = \frac{Hp}{h_d - h_s} \qquad \text{(Eqn. 2)}$$

It is evident upon closer evaluation of the above equations that either a value of discharge temperature or polytropic efficiency is required to accurately solve these relations. In the equation for polytropic head, the discharge specific volume, v_d, and polytropic exponent, n, both require a knowledge of the discharge temperature. Similarly, the equation for polytropic efficiency needs the value of the discharge enthalpy, h_d, which can either be obtained from an estimate of the discharge temperature or the polytropic efficiency.

A general rule of thumb is that the head generated in a single stage of a closed impeller design, centrifugal compressor should fall between approximately 6500 and 14,000 ft-lbf/lbm (19,429 and 41,847 J/kg). This upper limit is increased for open impeller designs. Average generated head per stage was evaluated for more than 60 actual compressor sections from multiple equipment suppliers within the author’s personal experience. This is presented in Figure 6, where average head per stage is plotted against section inlet flow coefficient. The data was obtained from information available on API 617 data sheets. Flow coefficient is a dimensionless parameter that will be more fully defined in a later section. The plot also contains two limits identified as minimum and maximum tip speed limits that will be defined in the next section. The important observation to make in the following figure is the distribution of average head per stage across the entire actual experience data included. Although there are examples that fall above and below the typical limits, the majority of the data falls within the prescribed limits.

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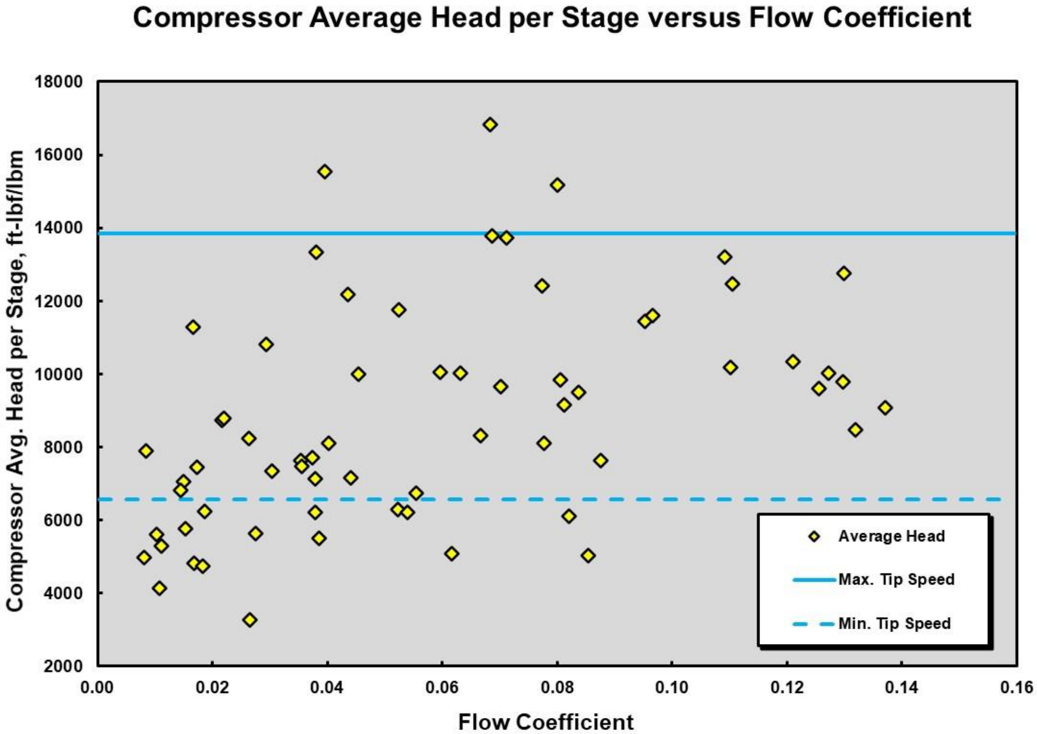

Figure 6: Average Head per Stage versus Flow Coefficient for Closed Impellers

There is an additional limitation that is reflected on the average head per stage figure above that is associated with some of the lower flow coefficient data. In high pressure, injection applications, a maximum differential pressure limit across a single stage may be imposed. As the suction pressure is increased, a constant pressure ratio results in increasing differential pressure ratio. These increasing differential pressures impact the deflections of stationary components and the amount of thrust imposed across a single impeller. This added criteria may result in reduced head rise allowed across a stage when applied. A number of these reduced head examples are represented in this data.

After the values of the polytropic head and efficiency are obtained, the power absorbed by the compression process can be calculated as the product of the mass flow rate and the enthalpy rise (or polytropic head divided by the polytropic efficiency). These relations form the basis of most compressor selections provided by equipment suppliers, where the polytropic efficiency is estimated and provided by the supplier.

$$PWR = C_3 \cdot \frac{\dot{m} \cdot Hp}{\eta_p} \quad \text{(Eqn. 3)}$$

Acceptance testing and operational performance monitoring utilize actual measurements of discharge temperature that allows the polytropic head and efficiency to be calculated directly. These are then compared to the values predicted by the equipment supplier to assess ultimate compressor performance acceptability and ongoing operational health. Such an ongoing performance monitoring effort, when combined with regular vibration monitoring and other mechanical parameter evaluation, such as lube oil and bearing temperature monitoring, form the basis of a robust machinery condition monitoring program and evolution to a more condition based maintenance philosophy.

Impeller Tip Speed

Another parameter associated with centrifugal compressor design that is often overlooked is the tip speed of impellers composing a single section. The tip speed equation that relates to each individual impeller is:

$$U_{tip} = C_4 \cdot D \cdot N \quad \text{(Eqn. 4)}$$

A general rule of thumb is that impeller tip speed should normally range between 650 and 900 ft/sec (198 and 274 m/sec) for fully enclosed impeller designs.

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Impeller tip speed is related to both mechanical and aerodynamic limitations associated with compressor design. Material stress levels within the impeller are directly proportional to the square of the tip speed. Accordingly, there is a maximum value of tip speed that is associated with the allowable stress for a given material of construction. This material strength limitation is most often approached in applications involving very low molecular weight gases where a high value of head is required for a given compression ratio. The maximum tip speed limitation is increased for open impeller designs due to the reduced centrifugal forces generated with the absence of the mass of the cover.

Heavy molecular weight gases often impose a different, lower tip speed limit on impeller designs associated with gas acoustic velocities. The machine Mach number is defined as the ratio of the impeller tip speed and inlet acoustic velocity.

$$Mn = {U_{tip}}/{c} \qquad \text{(Eqn. 5)}$$

Gas acoustic velocity is directly related to specific heat ratio and temperature and inversely related to gas molecular weight. Heavier molecular weight gases normally are associated with lower acoustic velocities and lower allowable tip speeds. The machine Mach number does not represent an actual physical quantity since the flow field in the tip region of the impeller is multi-dimensional and the local acoustic velocity is different from the inlet, so choking of the flow is not necessarily related to a machine Mach number of unity. In fact, there are several examples where impeller machine Mach numbers greater than unity have been manufactured and are operating without issue. A more important quantity is the inlet relative Mach number, Mir, which is the Mach number distribution at the eye of the impeller. Here localized choking regions can exist that impact impeller flow fields and efficiency. Unfortunately, the inlet relative Mach number is not readily derivable from data sheet information and must be supplied separately by the equipment supplier. Casey, et al. (2010) demonstrated that the inlet relative Mach number is proportional to the machine Mach number and the flow coefficient. Accordingly, consideration of this parameter is only critical in applications that are pushing established limits of this quantity.

Once again referencing the database of over 60 compressor sections, a plot of impeller tip speeds versus inlet flow coefficient is provided in Figure 7. Here the previously identified rule of thumb limits are included along with two different measures of the impeller tip speed for separate compressor sections. One of these tip speed measures is based upon the maximum impeller diameter included in a section. The other is based upon a weighted average of the impeller diameters based upon the square of the tip speeds. The importance of this will be discussed in a later section.

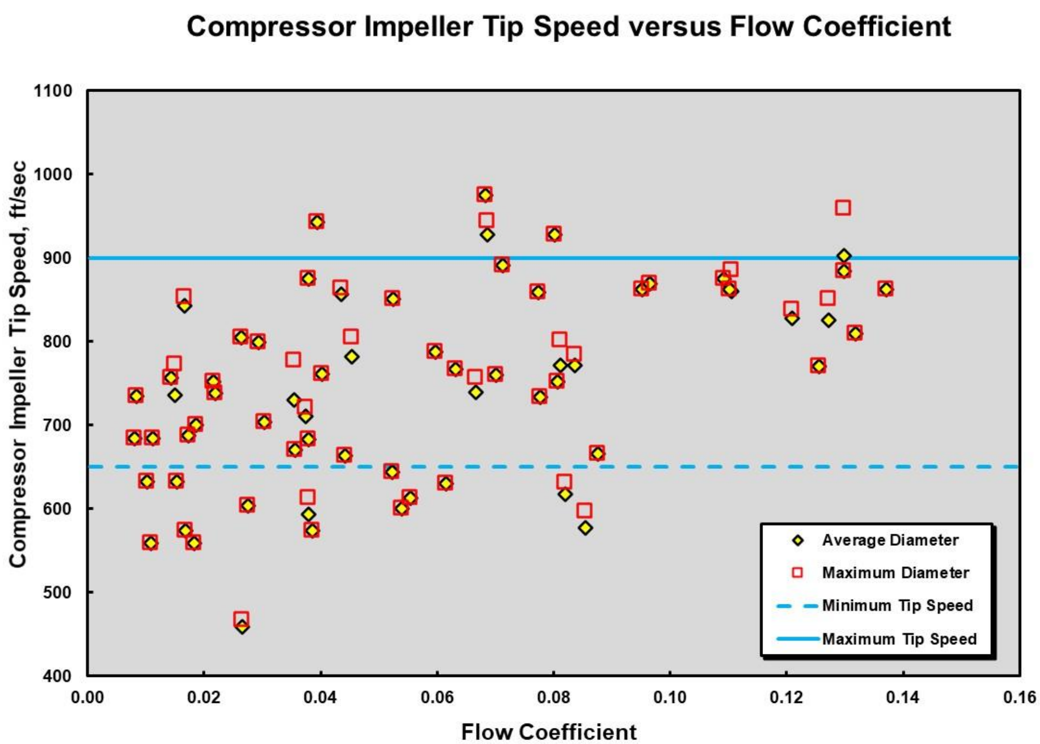

Figure 7: Impeller Tip Speed for Closed Impellers

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In those cases where the yellow diamonds representing the weighted average diameter fall within the red boxes of the maximum diameters, this signifies that all of the impellers within the compressor section are of the same diameter. This is not the case when the yellow diamonds fall outside their related red boxes, where the relative spacing between the two data points represents a larger difference of one or more impellers from the average. Although there are a limited number of cases where the tip speeds exceed the rule of thumb values given, the vast majority of examples fall within or below the limit. This serves to further validate that the stated limits are reasonable assumptions to make in a given compressor design.

Flow Coefficient

While the parameters addressed to this point are certainly important to compressor design and assessment, they are not sufficient to allow independent selections or detailed evaluations to be completed on supplier designs. The remaining parameters to be discussed will be based upon the laws of similitude between varying process, physical and geometric properties that can be involved. A dimensionless quantity that often is used as a primary independent variable used in the selection procedure is the flow coefficient. It has already been utilized above as the independent variable in the comparison of impeller tip speeds and average head per stage among several actual selections provided by a number of equipment suppliers. Specifically, the flow coefficient is defined as the ratio between the inlet volumetric flow rate and the product of the tip speed and some characteristic area. The flow coefficient is given by:

$$\varphi \approx \frac{Qa}{U_{tip} \cdot A_{char}} = C_5 \cdot \frac{Qa}{N \cdot D^3} \qquad \text{(Eqn. 6)}$$

This dimensionless parameter not only allows comparisons between compressors operating at different pressure and density levels, variations in gas molecular weight, differences in rotational speed, and different impeller diameters, but it also provides insight into the geometric relationships that exist between the designs of different impellers. The flow coefficient is directly related to the relative amount of volumetric flow that the impeller must accept. Very low flow coefficients are characterized by a flow path through the impeller that incorporates an almost right angle turn downstream of the impeller eye. Additionally, the ratio between the diameters of the eye of the impeller at the shroud to the outer diameter is relatively small. As the flow coefficient increases, this eye-to-outer diameter ratio approaches unity as the eye diameter increases to accommodate larger volumetric flows. At higher flow coefficients, the flow path through the impeller transitions from a radial exit to one at a lower angle relative to the axis of the shaft. These are commonly known as mixed flow impeller designs. Ultimately, the flow coefficient can rise to levels where the impeller changes from a radial to an axial design. This geometric relationship is illustrated in the following photograph of low and high flow coefficient radial impellers to the left and right, respectively.

Photo Courtesy of Dresser-Rand: A Siemens Business

Figure 8: Impeller Flow Coefficient Comparison

The magnitude of the flow coefficient also has an impact on other relative geometric factors between impellers. Aungier (1995, 2000) proposed an axial length-to-diameter ratio as a function of the impeller flow coefficient in an initial paper covering preliminary compressor design. He later offered a modification to this relationship in a text that included another correction for eye hub-to-diameter ratio. The following figure is based upon Aungier's original formulation.

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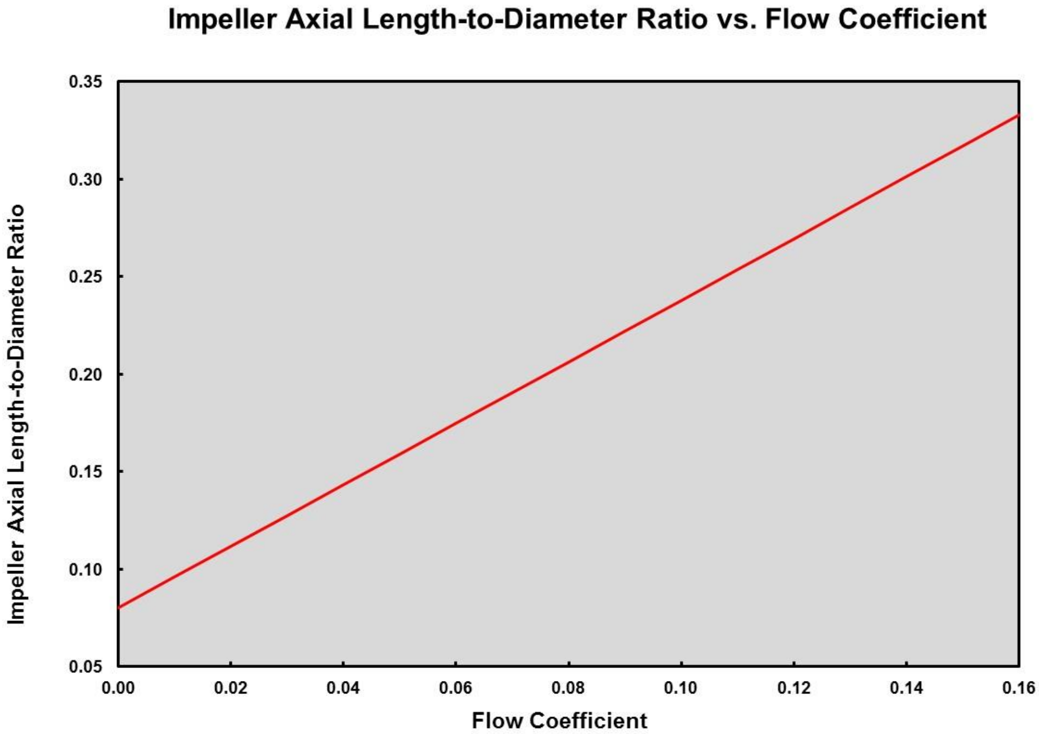

Figure 9: Aungier Impeller Axial Length Ratio

The value of the results of this relationship is its potential impact on lateral rotordynamics. Essentially the implication is that a fewer number of high flow coefficient impellers may be accommodated in a given compressor design than more moderate flow coefficients due to the increased axial length of the high flow coefficient impellers and the impact on bearing span. This is best illustrated in high flow, propane refrigeration sideload compressors where typical configurations are limited to five impellers or less. It is also apparent in the previous photograph (Figure 8), where the impellers are roughly the same diameter but considerably different in axial length.

Fundamental Dimensionless Parameters

Additional similarity parameters exist that allow further comparisons between machines operating under widely varying conditions. It has been previously demonstrated that the required conditions of flow rate, gas molecular weight, pressures, and temperatures can be converted into volumetric flow, head, efficiency and power for a given application. These can be further reduced using similarity principles to allow specific impeller and other component sizes to be applied to a range of very different process conditions. The first of these similarity parameters is the head coefficient. As noted previously, the polytropic model is only one of the thermodynamic models that can be applied to describe compressor performance, although it is the most universally used in process applications. The polytropic head coefficient for a single stage is defined as:

$$\mu_p = C_6 \cdot \frac{Hp}{U_{tip}^{\ 2}} \qquad \text{(Eqn. 7)}$$

This dimensionless parameter demonstrates that the head produced by a single impeller is directly proportional to the square of the operating tip speed of that impeller. It can also be shown that, for a given staging configuration, the polytropic head coefficient varies across the flow range from surge point to overload comparable to the way the efficiency varies from surge to overload. The head coefficient may also be applied to an entire compressor section with any number of stages (a total number j as shown below). The above equation transforms to:

$$\mu_p = C_6 \cdot \frac{Hp}{\sum_{i=1}^{j} U_{tip,i}^{\ 2}} \qquad \text{(Eqn. 8)}$$

This also leads to an equation that can be utilized to calculate an equivalent, average diameter of the same number of stages:

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$$D_{avg} = \sqrt{\frac{\sum_{i=1}^{j} D_i^2}{j}} \qquad \text{(Eqn. 9)}$$

The polytropic efficiency has already been presented and exists in a non-dimensional form as the ratio between the work transferred to the gas being compressed and that absorbed by the compressor with the difference being lost work. An additional dimensionless parameter can be introduced that corresponds to the total work input, and is commonly referenced as the work input coefficient.

$$\tau = \frac{\mu_p}{\eta_p} \approx \left(h_d - h_s\right) \qquad \text{(Eqn. 10)}$$

Aungier (1995, 2000) has developed relationships for polytropic head coefficient, work input coefficient and polytropic efficiency as a function of the design inlet flow coefficient. These are available in both his referenced technical paper and book and are provided for both open and covered impeller stage designs. According to Aungier (1995), "…they reflect a combination of good efficiency and stable operating range that are considered to be readily achievable with conventional aerodynamic design technology." It is imperative to understand that these relations do not represent the performance of a single stage across its flow range, but rather the characteristics of a family of stages of different flow coefficients at their design or best efficiency points. Accordingly, they can be utilized to predict the performance characteristics of a stage at its design point but offer no insight into the shape of the performance curve, flow range, or head and efficiency variation across the individual stage. Nevertheless, their ability to predict design point performance characteristics allows their use in providing preliminary sizing and design of a compressor stage or section. Figure 10 provides a plot of the predicted values of polytropic head coefficient as provided by the Aungier (1995, 2000) relations for both vaneless and vaned stage diffuser designs.

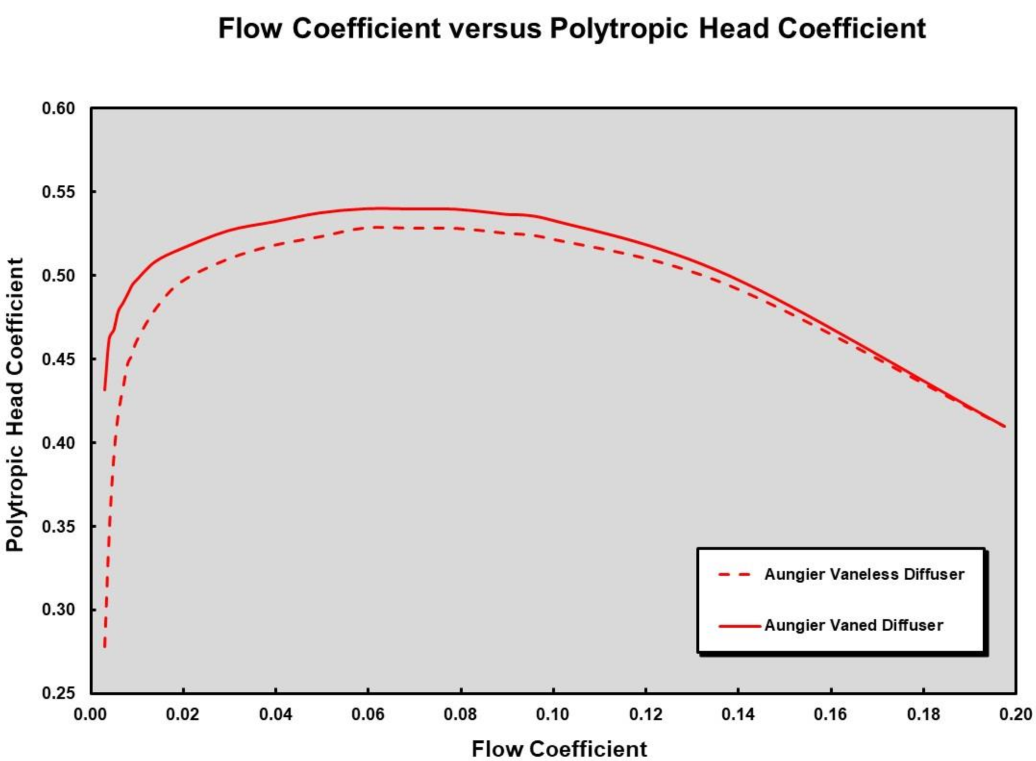

Figure 10: Aungier Polytropic Head Coefficient Characteristics

Figure 11 provides a similar set of characteristics for the polytropic efficiency across a family of stages of different design flow coefficients as predicted by Aungier (1995, 2000).

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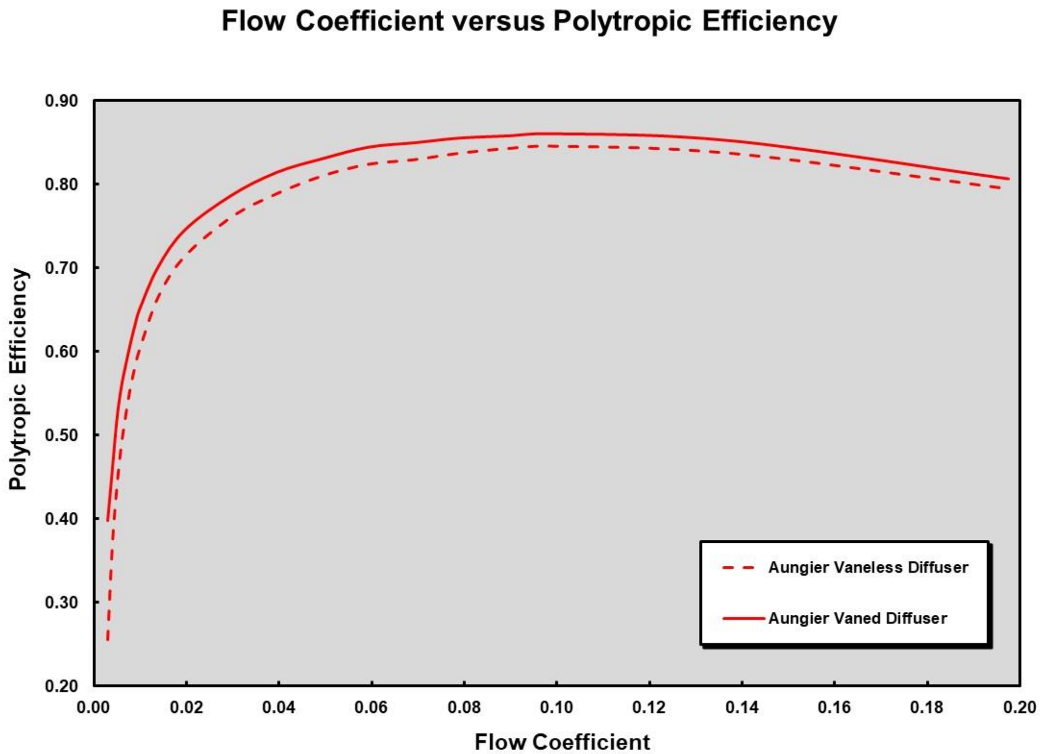

Figure 11: Aungier Polytropic Efficiency Characteristics

Although the data presented above is for a single stage, it can be extended to an entire section. An examination of both the head coefficient and the efficiency shows a near constant value for both the head coefficient and efficiency across the range of flow coefficients between a value of 0.05 and 0.11. As long as it is assumed that subsequent stages after the first stage are designed with flow coefficients near the best efficiency points, these characteristics can also be applied across a section with reasonable accuracy. Flow coefficients outside of this range may still be applied, but with an expected reduction in performance. A selection of the flow coefficient allows prediction of the polytropic head coefficient and efficiency. When these are combined with an assumed impeller tip speed to satisfy the required head for an application (or combination of tip speeds for a multi-stage section), a wide range of impeller diameters and rotational speeds are derived. Ultimately, an approximate compressor selection can be derived.

While the methodology presented to this point is able to provide reasonable compressor selections, both the ease of making a selection and compatibility with actual selections provided by equipment suppliers can be improved with the inclusion of some additional similarity parameters. These will be introduced and further developed below.

Additional Dimensionless Parameters

The use of two additional similarity parameters allow for more straightforward and accurate compressor selections. Although well established, they are not commonly utilized. These two parameters are the specific speed, Ns, and the lesser known specific diameter, Ds. The dimensional forms of these two similarity parameters are:

$$Ns = \frac{N \cdot \sqrt{Qa}}{Had^{0.75}} \qquad Ds = \frac{D \cdot Had^{0.25}}{\sqrt{Qa}} \qquad \text{(Eqn. 11)}$$

There are several different forms of both these parameters that include variables with different dimensional units. Care should be exercised in using and comparing these parameters without first ensuring consistent units are being applied. It should also be noted that the head included in these relations is the reversible, adiabatic, or isentropic, head per stage. Replacement with the polytropic head per stage will change the results to some degree and has been used throughout the following analysis. There are also some references that suggest replacing the inlet volumetric flow with average or discharge volumetric flow rate in calculating the flow coefficient in multi-stage sections. Inlet volumetric flow has also been consistently applied here.

Specific speed and specific diameter may also be expressed in dimensionless forms that allow more generalized application and can also be related to the dimensionless flow coefficient and polytropic head coefficient. A number of the references included herein

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provide more detailed development, but the final forms of the equations are provided below. First, the specific speed:

$$ns = C_7 \cdot \frac{N \cdot \sqrt{Qa}}{\left({}^{Hp}\!\!\diagup_{nstg} \right)^{0.75}} = C_8 \cdot \frac{\sqrt{\varphi}}{{\mu_p}^{0.75}} \qquad \text{(Eqn. 12)}$$

And, the specific diameter:

$$ds = C_9 \cdot \frac{D \cdot \left({}^{Hp}\!\!\diagup_{nstg} \right)^{0.25}}{\sqrt{Qa}} = C_{10} \cdot \frac{{\mu_p}^{0.25}}{\sqrt{\varphi}} \qquad \text{(Eqn. 13)}$$

Finally, the product of these two parameters can be expressed as:

$$nsds = \frac{C_{11}}{\sqrt{\mu_p}} \qquad \text{(Eqn. 14)}$$

The importance of the inclusion of these two variables is evident in the fact that the rotational speed and impeller diameter have been separated. Flow coefficient and polytropic head coefficient are based upon a combination of these two parameters, either separately or in the form of the tip speed. Application of the specific speed and specific diameter thus allows the approximate magnitude of these two variables to be identified independently.

Balje (1962, 1981) provided one of the most illustrative comparisons between the specific speed, specific diameter, head coefficient and efficiency for compressors. This is provided in Figure 12. Although this figure is limited to compressors, Balje's extensive work not only included dynamic and positive displacement compressors, but pumps, turbines and other types of expanders as well.

Although there is a substantial amount of information included in this diagram, the most applicable to centrifugal compressors are isolated to values of specific speed between 0.1 and 3, and values of specific diameter between 1 and 20. The solid curves are characteristics of constant efficiency. Superimposed lines of constant head coefficient are dashed. A characteristic passing through the maximum value of efficiency for a given value of specific speed provides an optimum efficiency combination of specific speed and specific diameter. The assumption is that an equipment selection will be made along this optimum efficiency characteristic, if possible. Selections that are not possible along this optimum may also be evaluated for expected performance in terms of efficiency and head making capability. Generalized areas defining the machine design between purely radial, mixed flow, and axial construction are also identified.

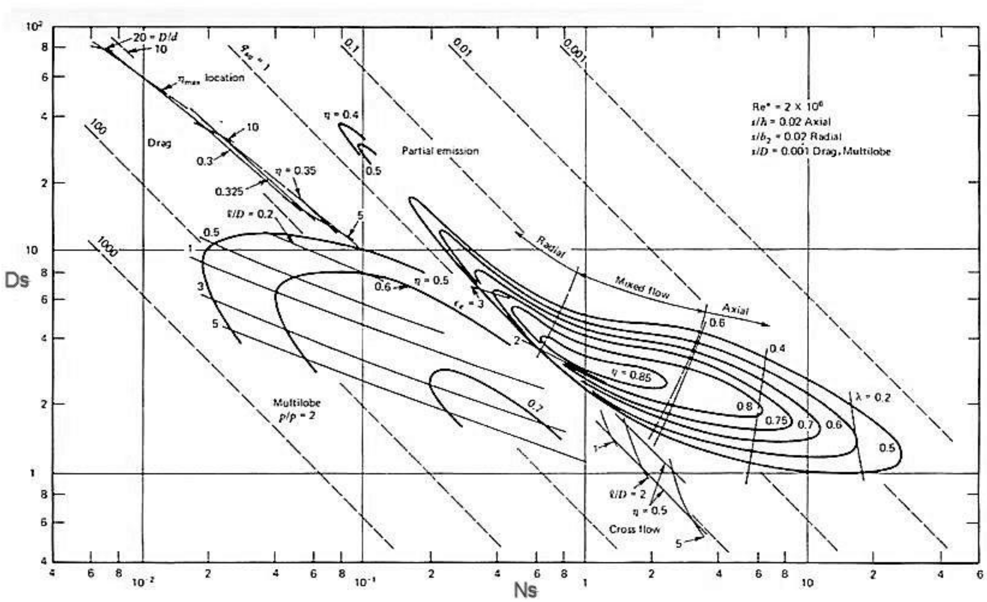

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Figure 12: Balje NsDs Diagram

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Prior to Balje's (1962, 1981) published work, Cordier (1955) provided a specific speed versus specific diameter correlation based upon a comparison of a significant number of machines that he considered to be of superior design. This work, known as the Cordier diagram, provided a more distinct optimum relationship between specific speed and specific diameter than the figure developed by Balje. The Cordier diagram is reproduced from Lewis' (1996) text in Figure 13 below.

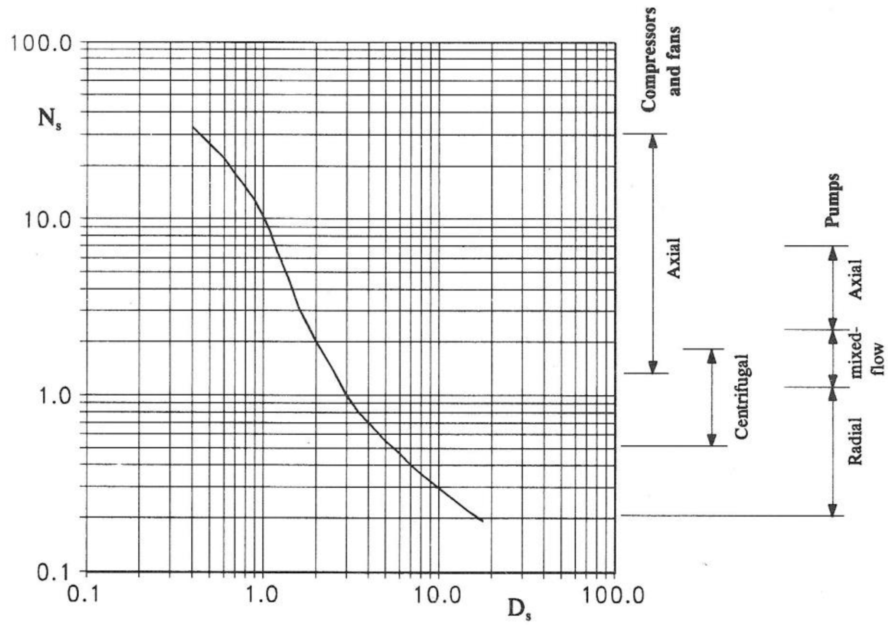

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Figure 13: Cordier Diagram

More recent work by Casey, et al. (2010) has re-examined the Cordier characteristic, particularly in the mixed flow impeller range, and has provided not only a review of existing correlations for the characteristic, but also an alternate equation to predict the values along the Cordier characteristic. Figure 14 below, reproduced from their paper, superimposes the Cordier characteristic on the original Balje diagram and also includes some data from actual machines evaluated by Cordier (1955) and some representative cross sectional drawings that further illustrate the geometric variations between impellers of increasing flow coefficient from left to right.

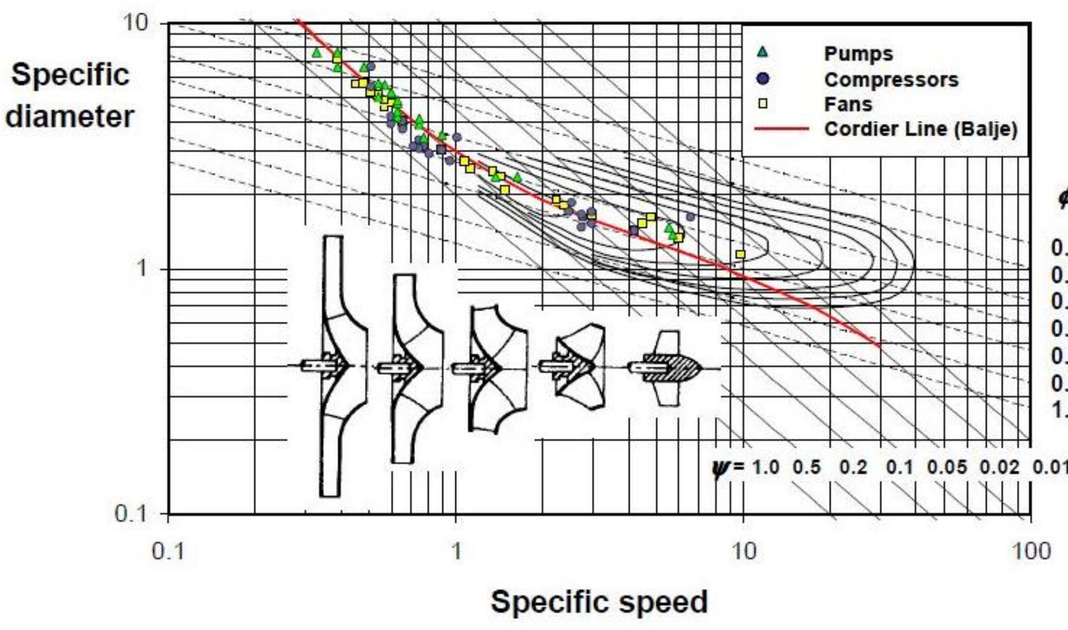

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Figure 14: Casey, et al. Balje NsDs Diagram

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These additional performance parameters, specific speed and specific diameter, are able to provide an increased number of variable combinations that might be utilized to predict compressor performance level and design point parameters. A selected few of these combinations appear to offer superior results and will be presented in order to establish the selection methodology that will be described in more detail in the following section.

The first of these correlations will be the dimensionless specific diameter as a function of the inlet flow coefficient. This is provided in Figure 15 which includes plots derived from the relation presented by Cordier (1955), correlations proposed by Casey, et al. (2010), and also developed from the Aungier (1995, 2000) equations for both vaneless and vaned diffusers. Actual equipment supplier selection data is also superimposed that demonstrates the utility of the correlation. It should be reiterated that this actual selection data includes over sixty separate compressor sections that span a variety of applications including gas gathering and processing, high pressure gas injection, large volume flow process refrigeration and light molecular weight recycle services. The similarity between the Cordier and the Casey, et al. curves which were derived from specific speed versus specific diameter correlations, and the Aungier curves which were derived from flow coefficient versus polytropic head coefficients is significant. It should also be noted that relationships were based upon both polytropic and isentropic relations. The close agreement of the actual selection case data with the predicted curves is also remarkable along the wide range of flow coefficients represented by the case data.

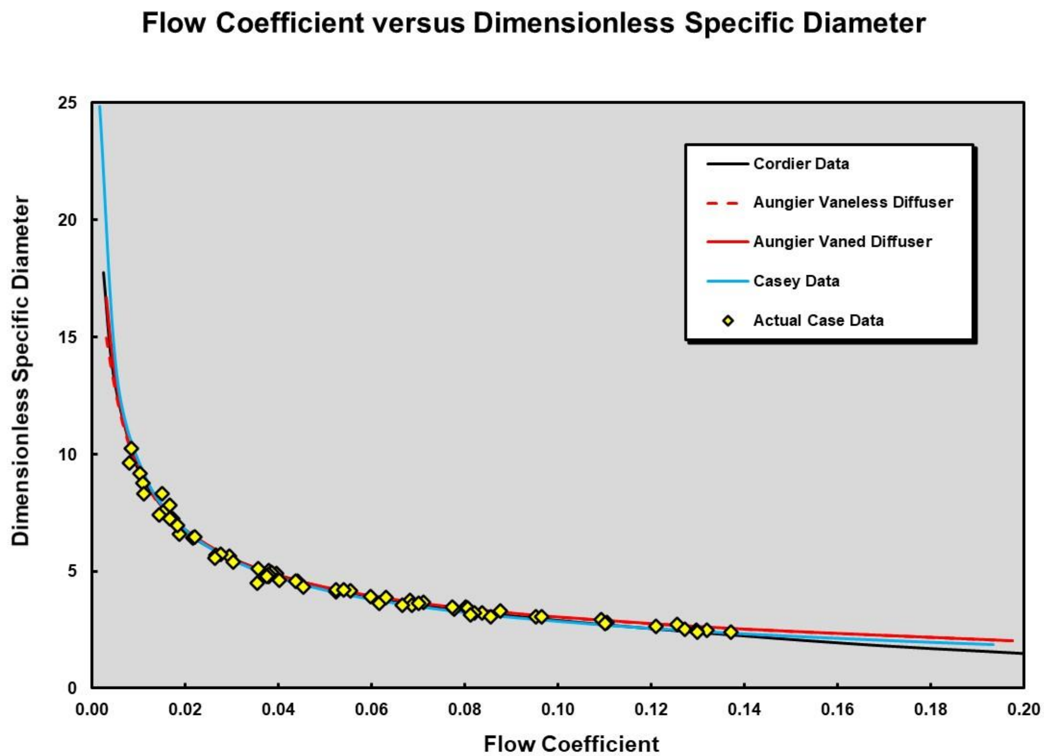

Figure 15: Dimensionless Specific Diameter versus Flow Coefficient

The significance of this information is based upon the ability to accurately predict the specific diameter with a given or assumed value of the flow coefficient. With reference to the equation for specific diameter (13), this allows a prediction of the average impeller diameter to be provided for given values of the average polytropic head per stage and the inlet volumetric flow rate. These two quantities are set by the specific process conditions necessarily established for a specific application.

A second critical relationship developed from the additional parameters of specific speed and specific diameter is the Cordier characteristic itself. Figure 16 provides the Cordier characteristic with separate relations developed from the data provided by Cordier (1955) and the equations proposed by Aungier (1995, 2000) and Casey, et al. (2010), similar to that provided in Figure 15. Actual case data from equipment supplier selections is again included to further illustrate the accuracy of the correlation between the parameters.

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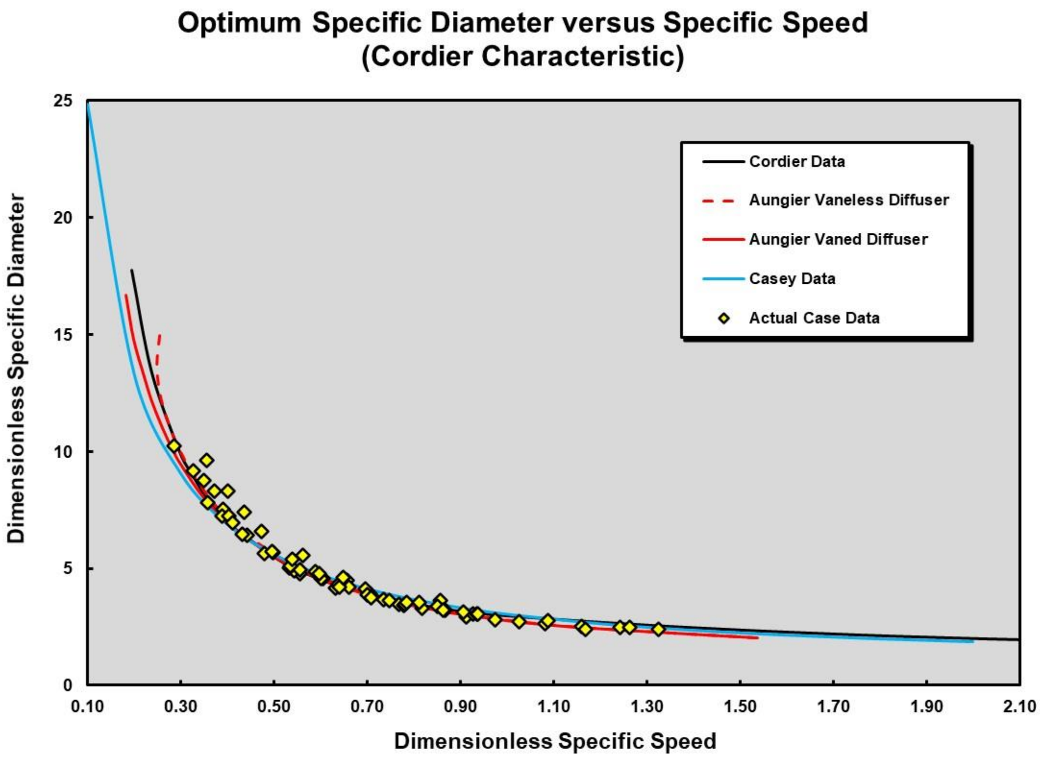

Figure 16: Dimensionless Specific Diameter versus Specific Speed

This relationship allows the dimensionless specific speed to be determined for a given value of specific diameter. Determination of the specific speed from the specific diameter then allows the design speed to be established for given values of the average polytropic head per stage and the inlet volumetric flow rate from Equation 12. Accordingly, these two correlations (Equations 12 and 13) allow the prediction of impeller diameter and rotational speed from an established value of the flow coefficient. This leads to the derivation of the average impeller tip speed and then the polytropic head coefficient to be determined.

Once these parameters have been established, the only remaining parameter that is needed to provide a preliminary selection is a value of the polytropic efficiency. This can be determined from the relations proposed by Aungier (1995, 2000) and is provided in Figure 17.

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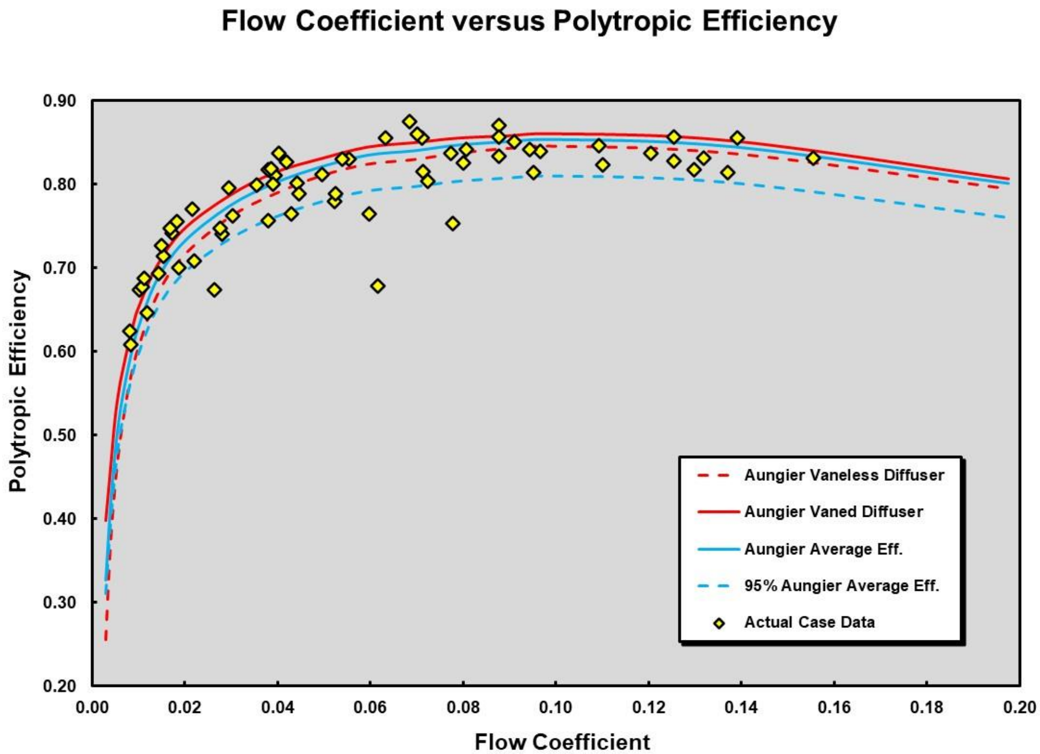

Figure 17: Polytropic Efficiency

There is obviously more scatter in the actual case data provided for polytropic efficiency than is evident with the other parameters. While this may appear to potentially impact the accuracy of the results, it does not affect the impeller sizing or speed significantly due to its limited influence on the value of the polytropic head. Its primary effect is on the required power necessary to provide the compression application and the resulting gas discharge temperature. In fact, the impact on the power may be one explanation for the variation. Since the equipment suppliers must meet some form of a performance guarantee which most often includes some limit on required power, there may be some tolerance applied to the predicted efficiency to ensure that any such performance guarantee is satisfied. Another potential reason for the variation noted in the actual case data efficiencies could be due to less than optimum matching provided in all stages of a multi-stage section, thereby reducing the overall efficiency level of the section. Regardless of the cause, a method is proposed to evaluate the impact of any variation in efficiency on the preliminary selection. First, an average efficiency between the Aungier vaneless and vaned diffuser is calculated and used as the baseline value. This can then be factored to increase or reduce the value to evaluate any impact. A 95% factor has also been plotted and it is clear that this covers the vast majority of the case data.

Selection Methodology

The information developed in this section allows for preliminary compressor selections to be completed independently from the equipment supplier. They will probably be slightly different from that ultimately provided by the equipment supplier, but they do offer the user and purchaser the ability to evaluate one or more potential configurations prior to engaging a supplier. This methodology will also provide additional tools and criteria for evaluation of equipment supplier selections. The following three methods of selection and evaluation are based upon five dimensionless parameters; namely the flow coefficient, polytropic head coefficient, polytropic efficiency, dimensionless specific speed, and dimensionless specific diameter. Depending upon the actual application, one or more of these evaluation procedures may be utilized to produce a preliminary equipment selection or evaluation of a proposed selection for one or more compression sections.

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Flow Coefficient

A primary and flexible approach to making a selection is through an assumption of the flow coefficient. If reference is given to the Aungier plots of polytropic head coefficient and efficiency, it is apparent that the highest efficiency and value of head coefficient occur across a flow coefficient range from a value of approximately 0.05 to 0.11. Optimum selections should, if possible, lie within this range of flow coefficients. Ideally, the selected section inlet flow coefficient should lie near the top of this range since successive stages in a section will likely have progressively smaller flow coefficients. Additionally, if there are one or more sections connected on the same shaft (such as multiple sections within a casing) or the same shaft system, higher pressure, subsequent sections may also result in lower values of flow coefficients. Of course, sidestream flows, molecular weight changes and other factors can influence this impact and must be considered, however, an attempt to maintain all sections within the optimum flow coefficient range is desirable. A suggested approach should include selections at the extents of this range of 0.05 to 0.11 and potentially one or more within this range. The solution steps are:

1. Calculate required polytropic head and inlet volumetric flow rate from provided process data [Eqns. 1 and 2]. Determine if multiple stages are required for a given section and calculate the average head per stage. Provide an initial estimate of the polytropic efficiency (reference to the Aungier plot of polytropic efficiency is a good start).
2. Select an assumed value or range of values of the flow coefficient (Note: optimum values normally lie between values of 0.05 to 0.11).
3. Determine the value(s) of the specific diameter from the flow coefficient versus specific diameter relationship [Figure 15]. Calculate the average impeller diameter(s) from the specific diameter equation [Eqn. 13].
4. Determine the value(s) of the specific speed from the specific speed versus specific diameter relationship [Figure 16]. Calculate the design operating speed(s) from the specific speed equation [Eqn. 12].
5. Determine the polytropic head coefficient(s) using the product of specific speed and specific diameter and the related equation [Eqn. 14].
6. Determine the range of polytropic efficiencies for each flow coefficient using the modified Aungier relationship (100% and 95% of predicted average efficiencies) [Figure 17].
7. Iterate on these steps with resulting efficiencies until efficiency value convergence if improved accuracy is desired.

Situations exist where the maximum diameter derived at the minimum flow coefficient of the range is smaller than that available from a given manufacturer's design range. It should be noted that impeller sizes will be larger at the lower flow coefficients and rotational speed higher at the higher flow coefficients. In the case where the resulting impeller diameter is smaller than is actually available, the selected flow coefficient will need to be lowered to derive larger impeller diameter solutions. Alternatively, the fixed diameter solution method presented below may be utilized with a given minimum impeller diameter provided by an equipment supplier.

Conversely, the resulting rotational speed at the highest flow coefficient in the optimum range may fall below that needed for a specific driver or application. In this case, a higher flow coefficient will need to be evaluated, or the fixed rotational speed method may be utilized if the required speed is known. Speed increasing or reducing gears may also be included before or between casings, but mechanical efficiency losses attributed to gears and added train complexity should be weighed against compressor efficiency reductions.

Fixed Impeller Diameter

In the case where average impeller diameter is known or selectable from a choice of discrete sizes, the fixed impeller diameter methodology provides a direct sizing approach that could only be achieved through iterations with the flow coefficient method. Here the average impeller diameter becomes the independent parameter and all other performance parameters, including the inlet flow coefficient, are obtained from the calculation procedure. The solution steps are:

1. Calculate required polytropic head and inlet volumetric flow rate from provided process data [Eqns. 1 and 2]. Determine if multiple stages are required for a given section and calculate the average head per stage. Provide an initial estimate of the polytropic efficiency (reference to the Aungier plot of polytropic efficiency is a good start).
2. Select one or more assumed value(s) of the average impeller diameter. Actual equipment supplier information may be useful.
3. Determine the value(s) of the dimensionless specific diameter(s) from the specific diameter equation [Eqn. 13].
4. Determine the value(s) of the specific speed from the specific speed versus specific diameter relationship [Figure 16]. Calculate the design operating speed(s) from the specific speed equation [Eqn. 12].
5. Calculate the resulting flow coefficient(s) from the resulting value(s) of average impeller diameter and rotational speed utilizing the equation for flow coefficient [Eqn. 6].
6. Determine the polytropic head coefficient(s) using the product of specific speed and specific diameter and the related equation [Eqn. 14].

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7. Determine the range of polytropic efficiencies for each flow coefficient using the modified Aungier relationship (100% and 95% of predicted average efficiencies) [Figure 17].
8. Iterate on these steps with resulting efficiencies until efficiency value convergence if improved accuracy is desired.

Should one or more impellers in a multi-stage section be of different diameters, use of a weighted average diameter is recommended although it is recognized that this could impact the calculated value of the flow coefficient. As long as the difference between the average and first stage diameters are reasonably close, this will not impact the overall results significantly and should be within the expected accuracy of the overall methodology.

Fixed Rotational Speed

The final method to be presented is based upon the assumption of a constant rotational speed where it becomes the independent variable. All other design parameters are derived from the various relations. This method is most useful in applications such as direct drives utilizing fixed speed electric motors or limited speed range drives such as single shaft gas turbines.

1. Calculate required polytropic head and inlet volumetric flow rate from provided process data [Eqns. 1 and 2]. Determine if multiple stages are required for a given section and calculate the average head per stage. Provide an initial estimate of the polytropic efficiency (reference to the Aungier plot of polytropic efficiency is a good start).
2. Select one or more assumed value(s) of the rotational speed. This may be dictated by the driver involved.
3. Determine the value(s) of the dimensionless specific speed(s) from the specific speed equation [Eqn. 12].
4. Determine the value(s) of the specific diameter from the specific speed versus specific diameter relationship [Figure 16]. Calculate the impeller average diameter(s) from the specific diameter equation [Eqn. 13].
5. Calculate the resulting flow coefficient(s) from the resulting value(s) of average impeller diameter and rotational speed(s) utilizing the equation for flow coefficient [Eqn. 6].
6. Determine the polytropic head coefficient(s) using the product of specific speed and specific diameter and the related equation [Eqn. 14].
7. Determine the range of polytropic efficiencies for each flow coefficient using the modified Aungier relationship (100% and 95% of predicted average efficiencies) [Figure 17].
8. Iterate on these steps with resulting efficiencies until efficiency value convergence if improved accuracy is desired.

Utilization of one or more of the aforementioned methods can result in reasonably accurate compressor selections. These may be used in applications that include one or more compressor sections in one or more individual casings. It also allows evaluation of variations in configuration and process conditions. While these selections provide the user and purchaser with an independent design for a given application, it must be acknowledged that selections provided by the equipment supplier must ultimately be used. The expectation is that the equipment supplier's design must govern and it is assumed that the supplier will support their selection with some form of performance guarantee. Accordingly, as already stated, the selections derived from these methods will provide reasonably accurate estimates of potential selections provided or allow additional evaluation of supplier proposed selections and configurations.

Six actual case selections will be examined with the methods presented and compared against the actual supplier selections. These are provided in the Appendix for different applications to illustrate how the methods may be applied and demonstrate their accuracy of prediction.

CENTRIFUGAL COMPRESSOR ARRANGEMENT

As head and flow demands increase for a particular application, single casings may no longer be capable of satisfying the requirements. When multiple casings are required, they may be arranged in different ways that can possess both benefits and issues that warrant further explanation. The drivers utilized for a compression application can also influence how the machines are arranged. The vast majority of centrifugal compressor applications employ electric motors, gas turbines or steam turbines as drivers. A brief review of the characteristics of each of these types of drivers is introduced below, prior to compressor arrangement evaluation.

Centrifugal Compressor Driver Characteristics

Electric Motors

Electric motors are available in a wide range of output power capabilities. Although smaller sized units are normally only offered in discrete power capacities, larger rated power motors can be designed and built with more flexibility. Constant speed motors are only available in a limited number of fixed operating speeds that are a function of the number of poles associated with the motor design. Synchronous motors operate at defined fixed speeds, whereas induction motors operate with some amount of slip (reduction in speed) that is related to the applied torque from the driven equipment.

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Most electric motors can be modified to become variable speed drivers through the placement of variable frequency drives between the main power supply and the motor. These drives modify the frequency of the alternating current power supply source. High speed motor designs exist but rotordynamic considerations seriously limit the power rating as the design speeds increase significantly. Variable frequency drives can become quite complex in larger power capacities and can have impacts on the power supply system and equipment train rotordynamics that normally have to be addressed through detailed design modifications. They can also become physically very large and become prohibitive in applications where footprint size is important such as on offshore platforms.

Fixed speed electric motor installations have process control and starting implications that are more restrictive than variable speed designs. Operating at only a single speed essentially limits operation to only a single point on the curve at a fixed amount of head rise. The impact on process control requires that any capacities below the design point either be accommodated by suction throttling and/or partial recycle. Fixed speed operation also does not tolerate changes in gas molecular weight as readily as variable speed operation. Finally, there are starting implications with fixed speed electric motor drives. The rapid rise in speed on start-up from settle-out conditions can overload the motor, creating the need for partial or total blowdown of the compressor loop prior to starting. Additionally, the in-rush current required on starting can have very negative impacts on the overall power supply system if the motor represents a substantial portion of the overall power grid system capacity. This is most notable on “islanded” power systems, such as would exist on an offshore platform. These influences are reduced or eliminated with variable frequency drives where large starting torques are generated at relatively low power draw levels upon starting.

Gas Turbines

Gas turbines exist in both variable speed and essentially constant speed designs with very little speed margin. Unfortunately, gas turbines are only available in discrete blocks of power and with output powers that are sensitive to ambient temperature variations. Generally they are selected based upon the maximum ambient temperature expected at the installation. Performance is also expected to degrade between major overhauls, so some tolerance must also be included for this degradation.

Two major design concepts exist for gas turbines, the industrial design and the aero-derivative design. The aero-derivative engine has a relative higher power density, meaning that the produced power to weight ratio is higher than the industrial design. This is accomplished through the application of lighter weight high-alloy casings, higher compression ratios and higher firing temperatures. In comparison, industrial designs are defined by heavier casings, lower compression ratios and lower firing temperatures. Light industrial designs also exist that possess aspects of both industrial and aero-derivative designs.

The multi-shaft, variable speed versions of gas turbines are able to operate over fairly wide output speed ranges (generally more significant than that actually required by the driven equipment) with power turbine efficiencies relatively constant over most of the speed range. Variable speed designs are not sensitive to starting load limitations and are typically able to start compressors from full settle-out conditions since the hot gas generator portion of the turbine is not mechanically linked to the power turbine.

Constant speed, single shaft gas turbines are subject to many of the same weaknesses as fixed speed electric motors. More specifically, with their limited allowable speed range, process control of the connected compressors can be more complex and suction throttling and/or partial recycle may be required beyond that necessary for anti-surge control. Available starting torque limitations generally require partial or full blowdown of the compressor loop to allow start-up. Some form of starting means is also required and the standard size supplied for power generation, which is the most common application, must be modified to provide greater starting torque.

Another aspect of gas turbines that deserves coverage is their need for periodic maintenance which requires some period of idling the machine. There are normally one or more interim inspections that are relatively short in duration that ultimately lead to some form of a major inspection. A major inspection is usually required after some number of equivalent years of operation and can be influenced by the number of starts and the firing level of the engine (duty cycle). The major inspection for aero-derivative or light industrial designs requires either removal of the engine for replacement with another engine or temporary usage of a rental unit. These generally entail a few days to complete. Larger, lower power density industrial engines may require major maintenance activities to be accomplished in place which may require a few weeks of down time. These maintenance requirements must be considered when selecting gas turbines as drivers.

Steam Turbines

Steam turbines represent a driver that may be designed flexibly for varying rotational speed and rated power levels. The most critical issue that exists for the application of steam turbines is the necessity and complexities associated with the supply of a steam generating system. Steam systems represent added complexity of operation and additional equipment, along with increased maintenance requirements. Due to the large plot space requirements of a steam system they are generally limited to onshore application with few exceptions. If a steam system is already required for other services (such as process heating), however, a steam turbine may be the most logical choice for a compressor driver.

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Steam turbines should be selected and sized with some consideration given to fouling and its effect on power output during operation. Periodic maintenance of steam turbines will also need to be considered, although typically not as significant as that required for gas turbines.

Increased Head Requirements

When the required compression head needed to satisfy a set of process conditions extend beyond that capable of being supplied by a single casing, additional casings and additional sections must be provided. There are two different fundamental types of arrangements that will be described. The first is the series arrangement and is shown schematically in Figure 18.

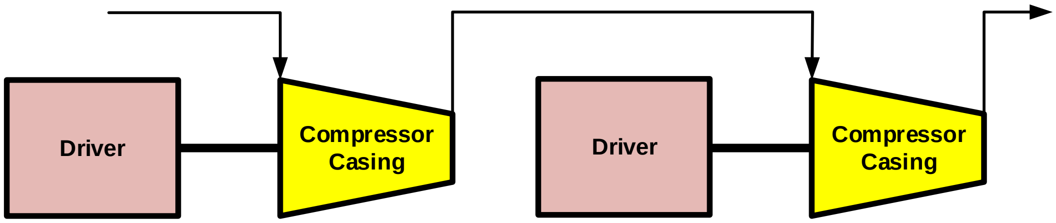

Figure 18: Series Compression Arrangement

The series arrangement is characterized by each individual compressor casing being driven by a dedicated driver. Although only two separate casings are shown in the schematic, this should not be considered as a limit. A major benefit of this type of arrangement is the ability to drive each casing at a different rotational speed, allowing each casing selection to be optimized to the degree possible. Of course, each casing may be composed of one or more sections, so a significant compression ratio can be provided. Sidestream additions or extractions can also be accommodated between casings or within a casing.

There are some deficiencies that exist with the series arrangement. Multiple, smaller capacity drivers are required that will likely cost more than a single, larger power driver. These multiple driver-compressor arrangements will also occupy a larger footprint that will be less desirable where space is limited or comes at increased cost. The greater number of drivers involved likely results in decreased overall reliability since a failure of a single driver results in the shutdown of the entire compression train. Depending upon the type of driver involved, the fit of available driver compared against the required compression power must also enter into the consideration of potential benefit of this type of arrangement. Finally, the process control of multiple, variable speed capable drivers in a series arrangement can become quite complex or require some amount of compromise in control flexibility.

The alternative to the series arrangement is the tandem arrangement, depicted in the schematic shown in Figure 19. In contrast, the tandem arrangement uses only a single driver for multiple compressor casings. Two separate casings are designated in the following figure but, as was the case with the series arrangement, this should not be considered as a limit. Tandem arrangements with three or four compressor casings commonly exist and have been in operation for some time.

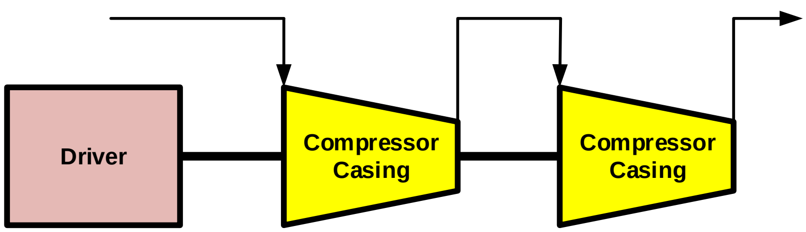

Figure 19: Tandem Compression Arrangement

Benefits of the tandem arrangement are associated with the use of a single driver. Although the power rating of the driver must be larger than for equivalent series arrangement, there should be an economy of scale which would lead to an overall lower capital cost. Secondly, the footprint of the tandem arrangement should be smaller as well. While these are certainly important considerations, the fact that all casings are connected to the same shaft system results that all casings must operate at a common rotational speed. This could lead to overall reduced efficiency since one or more casings may have to operate away from its optimum speed. The need to operate at a common speed can be offset through the inclusion of a gear between casings, but the mechanical losses associated with the gear must be taken into account. Tandem arrangements are also more complicated with respect to torsional critical speed considerations, particularly with the addition of one or more gears. The mechanical design of the shafts of inboard compressor casings must also be able to handle the drive-through torque requirements of the more outboard casings. Additionally, design considerations associated with material handling must be provided to maintain inboard casings, particularly if they are of radially split, barrel type designs.

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While the series and tandem descriptions are the most fundamental descriptions of multi-section, multi-casing compressor arrangements, actual installations may also be combinations of these arrangements due to a number of reasons. Obviously driver selection and sizing may influence an arrangement. Retro-fitting or expansion of existing facilities could also lead to a mixture of compressor arrangements. Regardless of actual installation complexity, though, any overall arrangement should be able to be divided into combinations of these two fundamental arrangements.

Increased Flow Requirements

As the required flow rate to handle a specific application increases, the ability to handle the entire duty in a single train of compression equipment is not possible. Multiple, parallel trains of equipment must be supplied to meet the flow demand. In critical applications, spare installed trains of equipment may be desired, however, the cost and justification of an installed spare train of major equipment should be weighed against other ways to address the issue such as additional capacity being designed into individual trains or a robust capital sparing program.

Similar to single trains discussed above, there are two fundamental parallel arrangements that deserve some discussion, series-parallel and tandem-parallel. In the series-parallel arrangement, separate drivers are provided for each compressor casing as shown schematically in Figure 20. A benefit of this arrangement, as with the single train, is the ability to design rotational speeds of individual casings more optimally. Process control of the series-parallel arrangement becomes even more difficult than the alternative and can become a significant challenge with more than two casings in series.

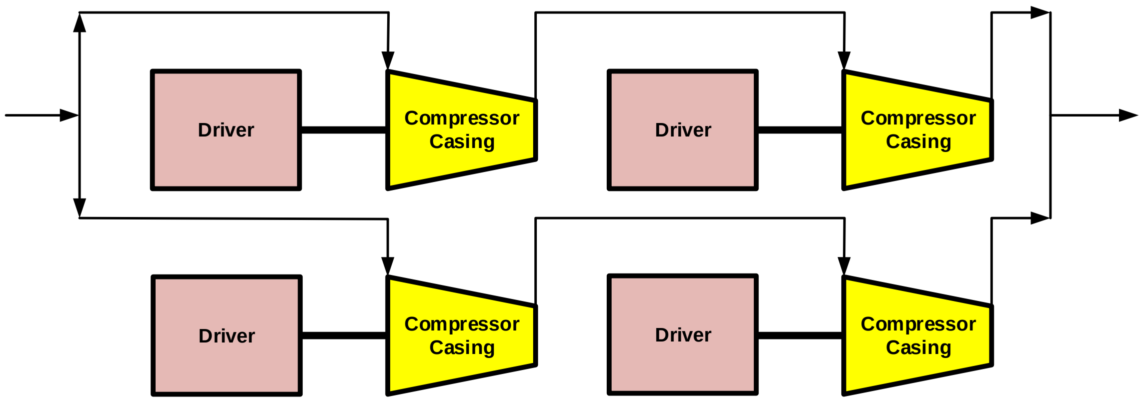

Figure 20: Series-Parallel Arrangement

The tandem-parallel arrangement is shown in Figure 21. A positive aspect of the tandem-parallel arrangement with common driver sizes of a comparable series-parallel arrangement is that the loss of a single driver during operation has less of an impact on capacity than the equivalent series-parallel arrangement. For example, with a common driver size, the capacity of the tandem-parallel arrangement shown in Figure 21 requires only a single series arrangement. The loss of a single driver in the series arrangement results in a complete loss of capacity, whereas the loss of a single driver in the tandem-parallel arrangement results in approximately only 50% of the total capacity. Similar reasoning can be applied against more numerous parallel trains relative to the series-parallel arrangements.

While the tandem-parallel arrangement appears to be more beneficial from a reliability perspective, this does come at a cost. A comparison of equivalent flow capacity series-parallel and tandem-parallel arrangements results in more compressor casings needed in the tandem-parallel arrangement. This increase in casing count also results in an increase in associated piping and equipment such as scrubber vessels, coolers, valving, and instrumentation. The overall impact of this difference is likely higher initial capital cost and larger required footprint for the tandem-parallel arrangement.

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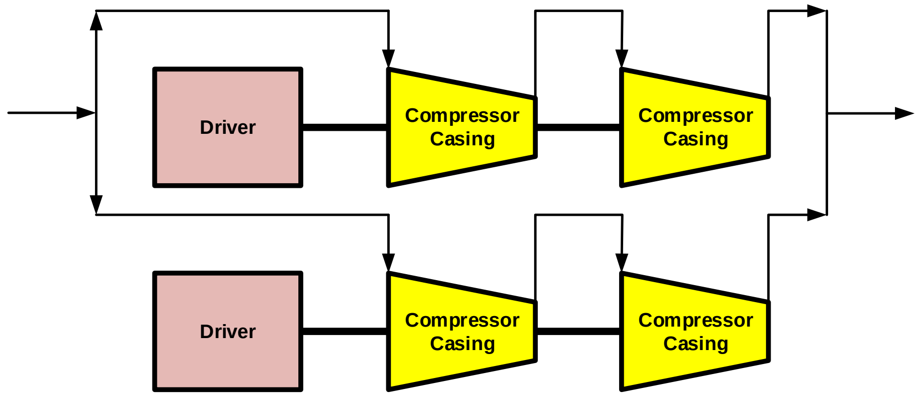

Figure 21: Tandem-Parallel Arrangement

Another issue can be associated with the tandem-parallel arrangement when the volumetric flow rates fall to small magnitudes. In this case, the small flow coefficients involved result in low expected polytropic efficiencies due to relatively excessive internal leakage rates, and may ultimately result in operational problems if internal seal clearances increase. The series-parallel arrangement may be the best alternative in such cases.

As noted previously, the schematics provided here for both parallel arrangements are limited to two casings, however, this does not limit more casings from being provided in series for either arrangement. The tandem-parallel arrangement represents a more simple control arrangement, since there is only a single shaft speed that can be controlled in each train. This simpler control problem may be offset to some degree with lower overall efficiencies. The tandem arrangement also represents a more challenging arrangement with respect to torsional critical speeds, especially if gears are included to vary relative speeds between individual casings to improve efficiency. Finally, as the number of tandem casings in a train increase, considerations must be given to the layout of lube oil drain piping since the elevation required for the gravity return drain lines may become significant.

Reliability, availability and maintainability (RAM) analyses have become popular over the past few years to evaluate differences in designs and machinery arrangements on reliability and production rates, often referred to as production efficiency. When applied to the tandem-parallel arrangement, the analysis is relatively straightforward. A loss of a driver or compressor casing in a train will result in the loss of that train and its corresponding flow capacity. This may not be the case with the series-parallel arrangement. Generally, the loss of a driver or compressor in a train is assumed to result in the loss of that portion of the total power produced by all drivers. The loss in capacity is then assumed to be equal to the percentage loss in driver power.

In order to demonstrate that the loss of a single driver or compressor in a series-parallel arrangement may not result in an equal percentage of flow, the following examples will be used. It is assumed that there are only two drivers, each with a single compressor casing in a single train. Each train is supplied with common headers connecting individual casing suction and discharge piping that allow sharing of flow between the parallel trains. The loss of one low pressure compressor results in a loss of low pressure capacity in the remaining parallel operating low pressure compressors. Since the low pressure and high pressure mass flow rates must be equal, the flow rates in the running low pressure machines must increase. This is illustrated in the left plot of Figure 22, where the operating point moves to a higher flow rate in the direction of the arrow. The opposite is true in the running high pressure compressors. Again referring to Figure 22, the high pressure compressor operating point is denoted in the right hand plot. Here the total mass flow must be reduced to balance with the operating low pressure compressors. The operating points on all high pressure machines move to lower flow, higher head operating points. As long as the individual compressor curves possess adequate range, from initial operating point to overload in the low pressure compressors and from initial operating point to surge control line in the high pressure compressors, then the changes in flow in proportion to the lost driver/compressor are consistent. If, however, adequate range does not exist, the flow capacity will be reduced further either through limitation of flow rate in the low pressure compressors due to overload or recycle due to the operating point reaching the surge control line in the high pressure machines.

A good rule of thumb to follow is that installations with fewer than four series-parallel trains may be subject to flow reductions beyond the proportional loss in driver power. This is based on the assumption that flow range varies by approximately 50% from surge to overload for each compressor. Of course, this also assumes that the initial operating points on both stages is in the middle of the range of each compressor. This obviously is not necessarily the case in each application or in actual operation.

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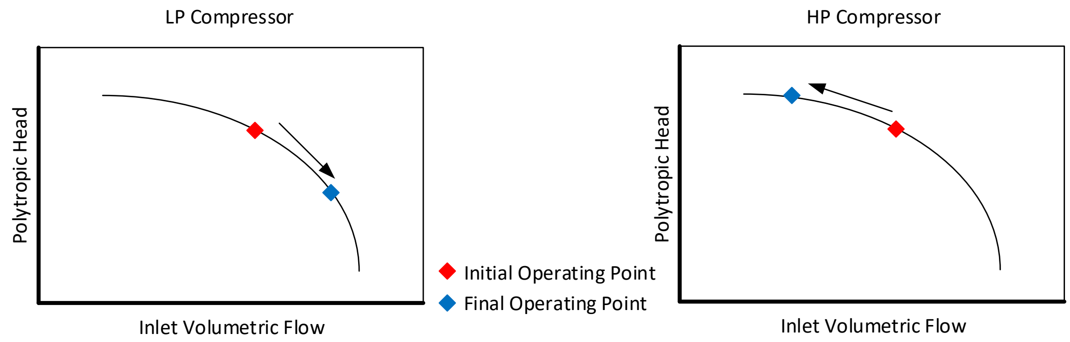

Figure 22: Loss of Low Pressure Compressor

A similar analysis can be performed on the loss of a high pressure compressor. This is demonstrated in Figure 23. With the loss of a high pressure compressor, the flow will have to increase in the remaining high pressure compressors and reduce in all of the low pressure machines. This is illustrated in the figure with low pressure compressor flow moving from an initial operating point to a lower flow towards the surge control line. The flow in the operating high pressure compressors move to a higher rate to accommodate the loss of capacity.

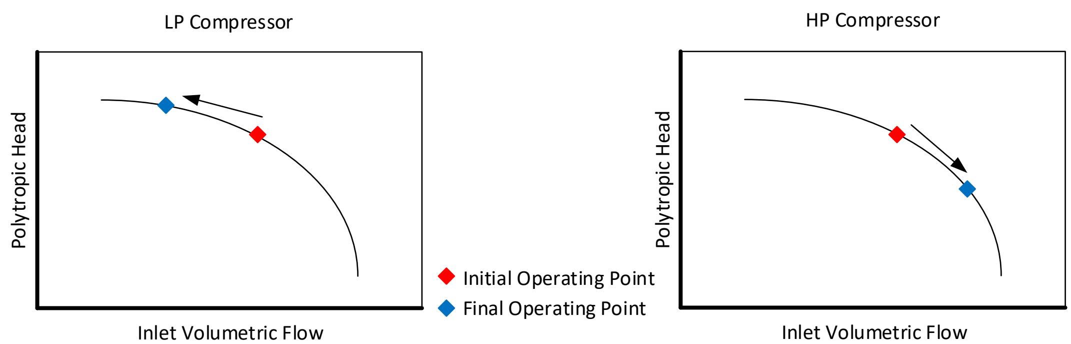

Figure 23: Loss of High Pressure Compressor

These examples demonstrate the general impact of the loss of a compressor in a series-parallel configuration. Such impacts are more pronounced in installations with constant speed or limited variable speed drives. Variable speed drives will respond to a lesser extent. In cases where the flow will tend to increase, the speed will increase to the degree possible to move to a higher flow rate. There may still be some movement to a lower generated head, but this is dependent upon the specific conditions. Conversely, the speed will tend to decrease in those situations where the operating point needs to move to a lower flow rate.

Another impact that is observed in the loss of a compressor in a series-parallel arrangement is associated with the variation in inter-stage pressure. This pressure reduces in the case of loss of a low pressure machine and increases upon the loss of a high stage compressor. The magnitude of this pressure variation is dependent upon the specific compressor selection, but should be evaluated to determine any impacts on piping and process equipment that is located between the two compressor sections.

While this qualitative assessment provides some general insight into the impacts of losing one or more compressors in parallel trains of equipment, simulation of these potential actions should be completed using predicted or as tested compressor curves to more accurately represent the behavior. Parallel compressor trains composed of more than two sections or casings also further complicate the prediction of behavior and must be simulated to gain reasonable insight.

CONCLUSIONS

The information and methodologies presented in this tutorial are intended to provide the purchaser and user the ability to develop independent, preliminary compressor selections and evaluate supplier proposed equipment with additional tools and criteria. This must be combined with actual selections along with detailed discussions with equipment suppliers to develop one or more compressor designs to satisfy the design requirements for any given application.

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Specifically, this tutorial has been created to:

- Introduce and evaluate different compressor configurations that are commonly used and are likely included in an equipment supplier's proposal. An understanding of the benefits of one type of configuration in comparison to another for a specific application should result in more optimum selections.

- Provide a methodology that allows independent selections to be developed from those provided by one or more equipment suppliers. Although they should not be considered of the same accuracy as the selections provided by equipment suppliers for their specific machine designs, they can provide a reasonable approximation of supplier selections and potential alternatives. It should be understood that the resulting selections are obtained for the design point, however, and are not intended to provide any information on compressor performance map curve shape or other parameters. These can only be generated from detailed stage selections based upon unique supplier stage designs.

- Define fundamental compressor arrangements and demonstrate how they can be utilized for applications as pressure ratio and flow rates increase. This will allow comparison and contrast of the benefits each possess to accommodate increasingly complex design arrangements.

While the material presented herein could be considered as a way to circumvent the engagement of the equipment supplier in making selections for an application, this is not the intent. It is believed that more informed purchasers and users are better customers with enhanced understanding and ability to appreciate benefits of any unique design. This is also not intended to be a complete coverage of all potential compressor design alternatives with full acknowledgement given to evolving technological advancements that are consistently being introduced. Ultimately, the equipment supplier's selection and predicted performance must become the basis of any design with the expectation that the supplier will support their design with some level of performance guarantee.

NOMENCLATURE

P_s, P_d = Compressor section suction and discharge pressure, respectively
v_s, v_d = Compressor section suction and discharge specific volume, respectively
T_s = Compressor section suction temperature
Z_s = Compressor section suction compressibility factor
n = Polytropic exponent
Hp = Polytropic head (work)
η_p = Polytropic efficiency
h_s, h_d = Compressor section suction and discharge enthalpy, respectively
$\dot{m}$ = Mass flow rate
PWR = Compressor section absorbed gas power
D = Impeller diameter
D_{avg} = Impeller weighted average diameter
N = Rotational speed
U_{tip} = Impeller tip speed
c = Compressor section inlet acoustic velocity
Mn = Compressor machine Mach number
Qa = Compressor section inlet volumetric flow rate
φ = Dimensionless flow coefficient
μ_p = Polytropic head coefficient
τ = Compressor work input coefficient
Had = Adiabatic (isentropic) head
Ns = Specific speed, dimensional form
Ds = Specific diameter, dimensional form
ns = Specific speed, dimensionless form
ds = Specific diameter, dimensionless form
$nstg$ = Number of stages (impellers) in a compressor section
$nsds$ = Product of dimensionless specific speed and specific diameter
C_x = Equation unit conversion constants (x is number) as defined below

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Equation Unit Conversion Constants

Equation Unit Conversion Constants	U.S. Customary Units	SI Units
Equation Parameter Units	P_s, P_d in psia v_s, v_d in ft^3/lbm T_s in R *Hp*, *Had*, h_s, h_d in ft·lbf/lbm $\dot{m}$ in lbm/min *PWR* in hp U_{tip}, *c* in ft/sec *N* in rev/min *D* in inches *Qa* in ft^3/min	P_s, P_d in bara v_s, v_d in m^3/kg T_s in K *Hp*, *Had*, h_s, h_d in J/kg $\dot{m}$ in kg/hr *PWR* in kW U_{tip}, *c* in m/sec *N* in rev/min *D* in mm *Qa* in m^3/hr
C_1	144.00	100,000.00
C_2	1545.349	8314.460
C_3	3.0303×10^{-5}	2.7778×10^{-7}
C_4	4.3633×10^{-3}	5.2360×10^{-5}
C_5	700.333	6.7548×10^{6}
C_6	32.174	1.000
C_7	1.00074×10^{-3}	1.74533×10^{-3}
C_8	1.7724	1.7724
C_9	1.5373	0.0600
C_{10}	1.1284	1.1284
C_{11}	2.0000	2.0000

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APPENDIX

This appendix provides selection results of six different applications that were derived from the over sixty examples that were utilized in demonstrating the accuracy of the methodology developed earlier. These examples were chosen in an effort to illustrate the wide utility of the methodology proposed and show its accuracy in providing compressor selections as compared to those that may be proposed by equipment suppliers. They were selected to cover the range of the methods proposed and extremes of potential applications that challenge actual machinery selections.

Thermodynamic properties were obtained through the use of the National Institute of Standards and Technology (NIST) REFPROP software application (Lemmon, et al., 2018). This was chosen due to its superior accuracy in property prediction over a broad range of process conditions, ease of use, and extensive number of available gas components. While the use of alternative equations of state may affect the results, they are not expected to be significant in many of the examples. Utilization of the most accurate thermodynamic properties is nevertheless always a good practice to follow.

Case Study 1: Gas Gathering Application

This first case study is a relatively common application where associated natural gas is produced with the primary product, crude oil. As the produced fluid is progressively reduced in pressure from the wellhead conditions, additional gas that is soluble in the oil is flashed. In this specific application, the variable speed, gas turbine driven compression duty is divided into two sections in a back-to-back configuration. Design process conditions are provided in Table A1.1 below.

Section	**1**	**2**
Volumetric Flow, Qa (acfm)	6928	2407
Gas Molecular Weight, MW	21.07	21.07
Suction Pressure, P_s (psig)	50.3	165.5
Suction Temperature, T_s (F)	120	110
Discharge Pressure, P_d (psig)	183.0	425.3

Table A1.1: Case Study 1 Design Process Conditions

The first section required polytropic head was estimated to be 53,326 ft-lbf/lbm assuming a polytropic efficiency of 80%. Polytropic head of the second section was approximately 75% of this value and is provided for information only since this analysis will be focused on the first section only. The approximate total head to be generated in these two sections is approximately 93,550 ft-lbf/lbm. Assuming a maximum value of head per stage at 14,000 ft-lbf/lbm, a minimum number of stages to satisfy this requirement is seven, four in the first section and three in the second. The addition of another stage in both sections brings the total impeller count to nine which is considered near the recommended limit. Figure A1.1 provides the results for required impeller diameter and rotational speed derived from using the variable flow coefficient methodology presented previously across the optimum range of flow coefficients.

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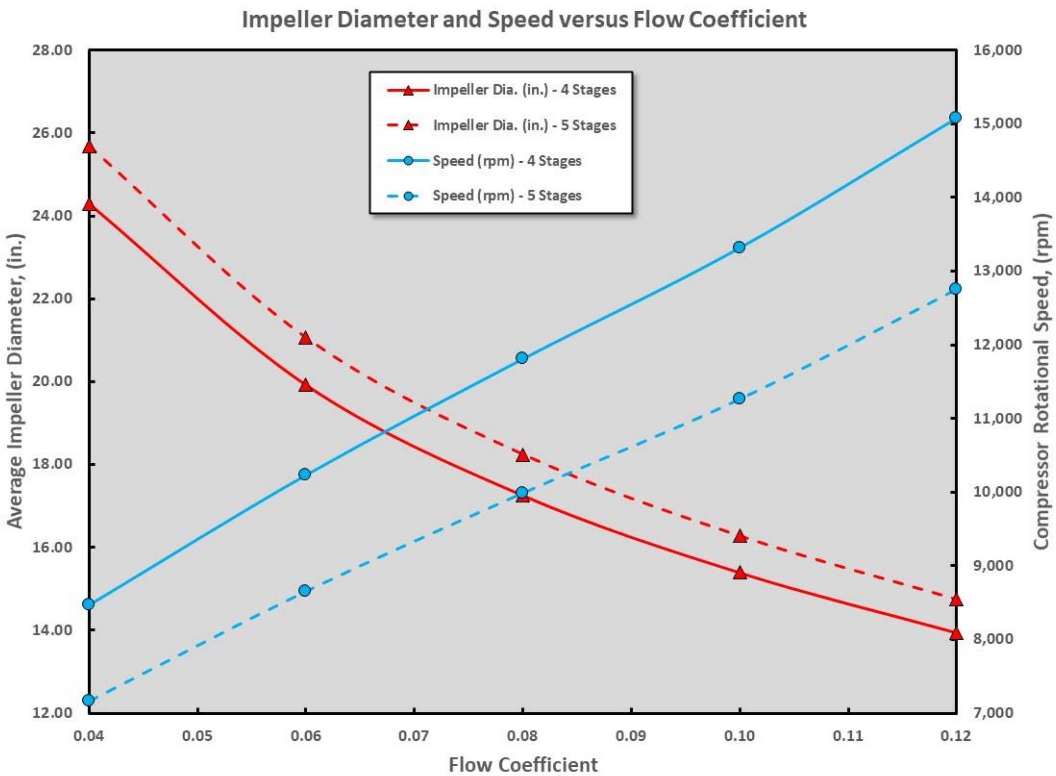

Figure A1.1: Case Study 1 Predicted Impeller Diameter and Speed

It is interesting to note that an increase in the impeller count actually results in larger required impeller diameters which is offset by a more significant reduction in rotational speed. Another trend to note is that rotational speed tends to increase while impeller diameter tends to decrease as flow coefficients become larger. In order to maximize the efficiency across both sections, it is desirable to maintain the inlet flow coefficients within the optimum range. Since the volumetric flow decreases by a factor of nearly three, the first section flow coefficient should be selected on the high side of the range.

Table A1.2 provides the results for a four stage section with variation in flow coefficient across a range including that of optimum efficiency and head coefficient. This was completed utilizing the average Aungier predicted efficiency relationship and included iterations on polytropic head resulting from predicted efficiency. This is most evident in the variation of calculated head across the flow coefficients. Although the values of head do vary, this variation is relatively small and comparison to the non-iterated value of head also demonstrates a small deviation.

Volumetric Flow, Qa (acfm)	6928	6928	6928	6928	6928
Polytropic Head, Hp (ft·lbf/lbm)	53305.0	53051.3	52960.8	52915.1	52924.2
Number of Impellers in Section	4	4	4	4	4
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	24.2968	19.9220	17.2554	15.3914	13.9409
Rotational Speed, N (rpm)	8466.5	10230.1	11806.7	13314.2	15075.8
Impeller Tip Speed, U_{tip} (ft/sec)	897.56	889.26	888.93	894.15	917.04

Table A1.2: Flow Coefficient Analysis Results for 4 Stage Section

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A similar analysis for a five stage section is provided in Table A1.3 again using the average calculated value of polytropic efficiency from the Aungier relations.

Volumetric Flow, Qa (acfm)	6928	6928	6928	6928	6928
Polytropic Head, Hp (ft·lbf/lbm)	53305.0	53051.3	52960.8	52915.1	52924.2
Number of Impellers in Section	5	5	5	5	5
Flow Coefficient, $Qa/N{\cdot}D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	25.6907	21.0650	18.2454	16.2745	14.7407
Rotational Speed, N (rpm)	7161.8	8653.6	9987.2	11262.4	12752.6
Impeller Tip Speed, U_{tip} (ft/sec)	802.81	795.38	795.08	799.75	820.22

Table A1.3: Flow Coefficient Analysis Results for 5 Stage Section

The actual equipment supplier involved with this application offered "standardized" impeller diameters of 14.9, 17.3 and 20.0 inches within the calculated range. Evaluation of the calculated data shows that the 14.9 inch selection with four stages in the section provides a solution with the best match of flow coefficient near the maximum end of the optimum range between 0.05 and 0.11. This in turn will allow a near optimum selection of flow coefficient for the second section, therefore is the best solution for the specific application. The selection of the smaller diameter impeller also results in a shorter bearing span with fewer impellers and a smaller required casing size, although this impact is not considered that significant.

A further evaluation was completed on the basis of a four stage section with an assumed lower polytropic efficiency based upon 95% of the Aungier average predicted value. The results of this analysis are provided in Table A1.4.

Volumetric Flow, Qa (acfm)	6928	6928	6928	6928	6928
Polytropic Head, Hp (ft·lbf/lbm)	53647.5	53380.7	53285.4	53237.4	53246.9
Number of Impellers in Section	4	4	4	4	4
Flow Coefficient, $Qa/N{\cdot}D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.7624	0.7933	0.8050	0.8110	0.8098
Work Input Coefficient, τ	0.6981	0.6802	0.6697	0.6565	0.6251
Impeller Diameter, D_{avg} (in.)	24.2579	19.8912	17.2291	15.3681	13.9198
Rotational Speed, N (rpm)	8507.2	10277.7	11860.9	13374.9	15144.7
Impeller Tip Speed, U_{tip} (ft/sec)	900.44	892.01	891.65	896.86	919.83

Table A1.4: Flow Coefficient Analysis Results for 4 Stage Section with 95% Efficiency

When a comparison is made between the results provided in Tables A1.2 and A1.4, it is evident there are relatively small differences in the calculated values of impeller diameter, rotational speed and impeller tip speed. This should provide assurance that these calculated values are not very sensitive to variations between predicted and actual levels of polytropic efficiency. The major influences of polytropic efficiency can be found in the absorbed power and discharge temperature of the specific application.

Finally, a comparison is made between the actual equipment supplier selection and calculated values derived from assuming supplier values of flow coefficient, impeller diameter and rotational speed at the 100% Aungier average predicted efficiency. The assumed

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value calculations utilize the three different methods presented earlier, namely sizing based upon an assumed value of flow coefficient, impeller diameter, or rotational speed, respectively. The comparison is provided in Table A1.5 below.

Case	Supplier Selection	Equal Flow Coefficient	Equal Impeller Diameter	Equal Rotational Speed
Volumetric Flow, Qa (acfm)	6928	6928	6928	6928
Polytropic Head, Hp (ft·lbf/lbm)	52808.0	52910.6	52937.6	52937.6
Number of Impellers in Section	4	4	4	4
Flow Coefficient, $Qa/N \cdot D^3$	0.1092	0.1092	0.1061	0.1023
Specific Speed, ns	0.9120	0.9536	0.9373	0.9103
Specific Diameter, ds	2.9457	2.9079	2.9475	3.0122
Specific Speed · Diameter, nsds	2.6865	2.7730	2.7626	2.7421
Polytropic Head Coefficient, μ_p	0.5542	0.5202	0.5241	0.5320
Polytropic Efficiency, η_p	0.8470	0.8543	0.8543	0.8540
Work Input Coefficient, τ	0.6543	0.6089	0.6135	0.6229
Impeller Diameter, D_{avg} (in.)	14.8790	14.6812	14.8790	15.2055
Rotational Speed, N (rpm)	13485.0	14120.7	13884.0	13485.0
Impeller Tip Speed, U_{tip} (ft/sec)	875.47	904.55	901.37	894.68

Table A1.5: Selection Criteria Comparison

This analysis shows close agreement between the actual compressor selection provided by the equipment supplier and preliminary selections based upon the methods developed in this tutorial. There is an approximate difference in the calculated polytropic head between the different cases of less than 0.3% which may be likely attributable to differences in calculated thermodynamic properties, solution methods and polytropic efficiency. All other parameters demonstrate reasonably close agreement between the actual data and the different methods. Most importantly, the variations in impeller diameter and rotational speed are minimal. The deviation in selected impeller diameter is with+/- 2% and the deviation in estimated speed is within 5%.

In summary, this example shows that the methods introduced in this work have the ability to predict a compressor selection that closely approximates an actual selection provided by an equipment supplier. While there is some flexibility in the number of stages that are possible, additional considerations as presented can provide further criteria to impact the selection.

Case Study 2: Gas Gathering Application

This example is also a relatively common application where associated natural gas is produced with the primary product, crude oil. Unlike Case Study 1, in this application, the variable speed, gas turbine driven compression duty is provided by a single section in a straight-through configuration. Design process conditions are provided in Table A2.1 below.

Volumetric Flow, Qa (acfm)	17,316
Gas Molecular Weight, MW	21.14
Suction Pressure, P_s (psig)	50.3
Suction Temperature, T_s (F)	120
Discharge Pressure, P_d (psig)	350.3

Table A2.1: Case Study 2 Design Process Conditions

The required polytropic head was estimated to be 88,137 ft-lbf/lbm assuming a polytropic efficiency of 80%. Assuming a maximum value of head per stage at 14,000 ft-lbf/lbm, a minimum number of stages to satisfy this requirement is seven impellers. The addition of one or two additional stages will be examined. Figure A2.1 provides the results for required impeller diameter and rotational speed derived from using the variable flow coefficient methodology presented previously across the optimum range of flow coefficients.

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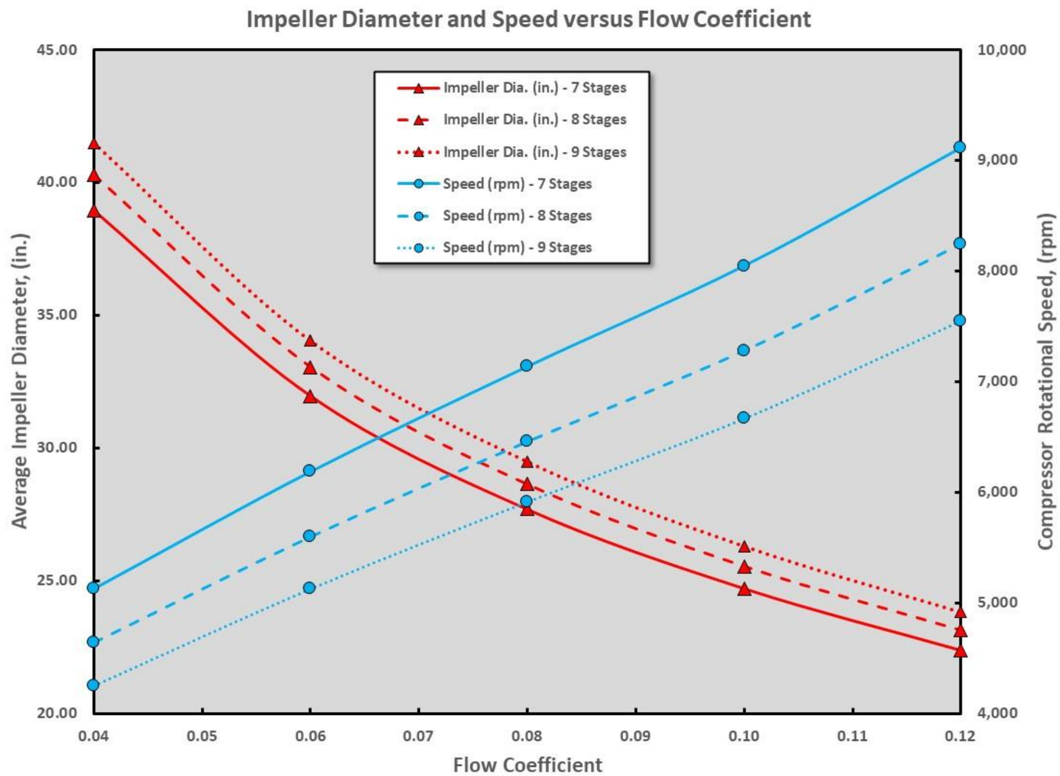

Figure A2.1: Case Study 2 Predicted Impeller Diameter and Speed

Similar to what was demonstrated in Case Study 1, an increase in the impeller count results in larger required impeller diameters which is offset by a more significant reduction in rotational speed. Another trend to note is that rotational speed tends to increase while impeller diameter tends to decrease as flow coefficients become larger. In order to maximize the efficiency across all stages within the section, it is desirable to maintain the inlet flow coefficient near the top of the optimum range.

Table A2.2 provides the results for a seven stage section with variation in flow coefficient across a range including that of optimum efficiency and head coefficient. This was completed utilizing the average Aungier predicted efficiency relationship and included iterations on polytropic head resulting from predicted efficiency. This is most evident in the variation of calculated head across the flow coefficients. Although the values of head do vary, this variation is relatively small and comparison to the non-iterated value of head also demonstrates a small deviation.

Volumetric Flow, Qa (acfm)	17316	17316	17316	17316	17316
Polytropic Head, Hp (ft·lbf/lbm)	88087.3	87475.3	87256.7	87146.3	87168.2
Number of Impellers in Section	7	7	7	7	7
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	38.9665	31.9680	27.6945	24.7054	22.3766
Rotational Speed, N (rpm)	5129.9	6188.3	7137.8	8046.7	9111.9
Impeller Tip Speed, U_{tip} (ft/sec)	872.21	863.18	862.52	867.41	889.65

Table A2.2: Flow Coefficient Analysis Results for 7 Stage Section

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A similar analysis for an eight stage section is provided in Table A1.3 again using the average calculated value of polytropic efficiency from the Aungier relations.

Volumetric Flow, Qa (acfm)	17316	17316	17316	17316	17316
Polytropic Head, Hp (ft·lbf/lbm)	88087.3	87475.3	87256.7	87146.3	87168.2
Number of Impellers in Section	8	8	8	8	8
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	40.2893	33.0532	28.6347	25.5440	23.1362
Rotational Speed, N (rpm)	4641.1	5598.6	6457.6	7279.9	8243.6
Impeller Tip Speed, U_{tip} (ft/sec)	815.87	807.43	806.82	811.39	832.19

Table A2.3: Flow Coefficient Analysis Results for 8 Stage Section

Finally, the same analysis was completed for a nine stage section is given in Table A2.4.

Volumetric Flow, Qa (acfm)	17316	17316	17316	17316	17316
Polytropic Head, Hp (ft·lbf/lbm)	88087.3	87475.3	87256.7	87146.3	87168.2
Number of Impellers in Section	9	9	9	9	9
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	41.4933	34.0409	29.4904	26.3074	23.8276
Rotational Speed, N (rpm)	4248.7	5125.2	5911.6	6664.4	7546.6
Impeller Tip Speed, U_{tip} (ft/sec)	769.21	761.26	760.67	764.98	784.60

Table A2.4: Flow Coefficient Analysis Results for 9 Stage Section

The actual equipment supplier involved with this application offered “standardized” impeller diameters of 23.3 and 27.0 inches near the top of the calculated range. Evaluation of the calculated data shows that the 23.3 inch selection with seven stages in the section provides a solution with the best match of flow coefficient near the maximum end of the optimum range between 0.05 and 0.11. The selection of the smaller diameter impeller also results in a shorter bearing span with fewer impellers and a smaller required casing size, although casing size impact is not considered that significant.

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A further evaluation was completed on the basis of a seven stage section with an assumed lower polytropic efficiency based upon 95% of the Aungier average predicted value. The results of this analysis are provided in Table A2.5.

Volumetric Flow, Qa (acfm)	17316	17316	17316	17316	17316
Polytropic Head, Hp (ft·lbf/lbm)	88912.6	88269.7	88040.1	87924.2	87947.2
Number of Impellers in Section	7	7	7	7	7
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.7624	0.7933	0.8050	0.8110	0.8098
Work Input Coefficient, τ	0.6981	0.6802	0.6697	0.6565	0.6251
Impeller Diameter, D_{avg} (in.)	38.8758	31.8958	27.6327	24.6506	22.3269
Rotational Speed, N (rpm)	5166.0	6230.4	7185.8	8100.5	9172.9
Impeller Tip Speed, U_{tip} (ft/sec)	876.28	867.09	866.39	871.27	893.62

Table A2.5: Flow Coefficient Analysis Results for 7 Stage Section with 95% Efficiency

When a comparison is made between the results provided in Tables A2.2 and A2.5, it is once again evident there are relatively small differences in the calculated values of impeller diameter, rotational speed and impeller tip speed. This should provide assurance that these calculated values are not very sensitive to variations between predicted and actual levels of polytropic efficiency. The major influences of polytropic efficiency can be found in the absorbed power and discharge temperature of the specific application.

Finally, a comparison is made between the actual equipment supplier selection and calculated values derived from assuming supplier values of flow coefficient, impeller diameter and rotational speed at the 100% Aungier average predicted efficiency. Another difference is evident when examining the value of the supplier selection impeller diameter, which is different between the supplier "standardized" size provided earlier and the average value found in Table A2.6. A few of the impeller diameters were trimmed to a smaller value to increase the required rotational speed to accommodate a direct drive from the selected gas turbine driver. As before, the assumed value calculations utilize the three different methods presented earlier, namely sizing based upon a supplier selected value of flow coefficient, impeller diameter, and rotational speed, respectively. The comparison is provided in Table A2.6 below.

Case	Supplier Selection	Equal Flow Coefficient	Equal Impeller Diameter	Equal Rotational Speed
Volumetric Flow, Qa (acfm)	17316	17316	17316	17316
Polytropic Head, Hp (ft·lbf/lbm)	87305.0	87137.0	87361.5	87361.5
Number of Impellers in Section	7	7	7	7
Flow Coefficient, $Qa/N \cdot D^3$	0.1105	0.1105	0.1165	0.1144
Specific Speed, ns	0.9730	0.9611	1.0045	0.9725
Specific Diameter, ds	2.7909	2.8902	2.7913	2.8394
Specific Speed · Diameter, nsds	2.7155	2.7779	2.8040	2.7613
Polytropic Head Coefficient, μ_p	0.5424	0.5184	0.5087	0.5246
Polytropic Efficiency, η_p	0.8375	0.8542	0.8533	0.8537
Work Input Coefficient, τ	0.6477	0.6068	0.5962	0.6145
Impeller Diameter, D_{avg} (in.)	22.6058	23.4213	22.6058	22.9949
Rotational Speed, N (rpm)	8720.0	8601.5	9007.2	8720.0
Impeller Tip Speed, U_{tip} (ft/sec)	860.11	879.02	888.43	874.91

Table A2.6: Selection Criteria Comparison

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The analysis shows close agreement between the actual compressor selection provided by the equipment supplier and preliminary selections based upon the methods developed in this tutorial. There is an approximate difference in the calculated polytropic head between the different cases of less than 2.5% which may be likely attributable to differences in calculated thermodynamic properties, solution methods and polytropic efficiency. All other parameters demonstrate reasonably close agreement between the actual data and the different methods. Most importantly, the variations in impeller diameter and rotational speed are minimal. The deviation in selected impeller diameter is with 4% and the deviation in estimated speed is within +/- 3%.

This example also shows that the methods introduced in this work have the ability to predict a compressor selection that closely approximates an actual selection provided by an equipment supplier. While there is some flexibility in the number of stages that are possible, additional considerations, such as output speed range of the driver, can provide further criteria to impact the selection.

Case Study 3: Gas Injection Application

Gas injection can represent a fairly unique compression application. The combination of high operating pressures and gas density, along with the potential for relatively small volumetric flow rates result in challenging selections. Design process conditions for the injection example are provided in Table A3.1 below. This is actually the high pressure section of a two section machine, the low pressure section conditions have been included, as well, in the second column.

Section	**HP**	**LP**
Volumetric Flow, Qa (acfm)	257.7	386.5
Gas Molecular Weight, MW	19.11	19.11
Suction Pressure, P_s (psig)	4180.3	2470.3
Suction Temperature, T_s (F)	122	122
Discharge Pressure, P_d (psig)	5935.3	4195.3

Table A3.1: Case Study 3 Design Process Conditions

The required polytropic head was estimated to be 16,587 ft-lbf/lbm assuming a polytropic efficiency of 80%. Assuming a maximum value of head per stage at 14,000 ft-lbf/lbm, a minimum number of stages to satisfy this requirement is two impellers, however, in this case the maximum head per stage is not the governing parameter. The total differential pressure for this compressor section is 1755 psi, which would result in a stage differential of nearly 900 psi for two stages. The equipment supplier actually selected three stages which reduced this differential to approximately 600 psi per stage.

Table A3.2 provides the results for a three stage, high pressure section with variation in flow coefficient across a range including that of optimum efficiency and head coefficient. This was completed utilizing the average Aungier predicted efficiency relationship and included iterations on polytropic head resulting from predicted efficiency. This is most evident in the variation of calculated head across the flow coefficients. Although the values of head do vary, this variation is relatively small and comparison to the non-iterated value of head also demonstrates a small deviation.

Volumetric Flow, Qa (acfm)	257.7	257.7	257.7	257.7	257.7
Polytropic Head, Hp (ft·lbf/lbm)	16584.8	16561.9	16553.7	16549.6	16550.5
Number of Impellers in Section	3	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	5.8389	4.7835	4.1420	3.6940	3.3460
Rotational Speed, N (rpm)	22691.3	27487.8	31753.0	35823.7	40560.1
Impeller Tip Speed, U_{tip} (ft/sec)	578.10	573.72	573.86	577.41	592.15

Table A3.2: Flow Coefficient Analysis Results for 3 Stage Section

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It is important to note from the data presented in this table the combinations of impeller diameter and rotational speed. All impeller diameters are below 6 inches and the design operating speeds are all above 22,000 rpm. The impeller diameters required all fall below the actual equipment supplier's range for the proposed machines. In order to raise potential impeller diameters into a range provided by the supplier, the design flow coefficient will have to be extended to values below the optimum range. Table A3.3 gives the results for calculations performed in this lower range.

Volumetric Flow, Qa (acfm)	257.7	257.7	257.7	257.7	257.7
Polytropic Head, Hp (ft·lbf/lbm)	17001.0	16749.1	16641.7	16605.8	16584.8
Number of Impellers in Section	3	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0050	0.0100	0.0200	0.0300	0.0400
Specific Speed, ns	0.2206	0.2992	0.4115	0.4960	0.5686
Specific Diameter, ds	13.2605	9.4765	6.7622	5.5557	4.8215
Specific Speed · Diameter, nsds	2.9253	2.8352	2.7827	2.7556	2.7414
Polytropic Head Coefficient, μ_p	0.4674	0.4976	0.5165	0.5268	0.5322
Polytropic Efficiency, η_p	0.4757	0.6290	0.7321	0.7749	0.8025
Work Input Coefficient, τ	0.9826	0.7911	0.7056	0.6798	0.6632
Impeller Diameter, D_{avg} (in.)	15.9595	11.4480	8.1821	6.7259	5.8389
Rotational Speed, N (rpm)	8968.9	12028.4	16465.0	19812.9	22691.3
Impeller Tip Speed, U_{tip} (ft/sec)	624.56	600.83	587.82	581.45	578.10

Table A3.3: Flow Coefficient Analysis Results for 3 Stage Section Below Optimum Range

Figure A3.1 provides the results for required impeller diameter and rotational speed derived from using the variable flow coefficient methodology presented previously across both the optimum range and lower extended range of flow coefficients. It is evident from this figure that the impeller diameter increases significantly in this lower flow coefficient range while the rotational speed is lowered. Investigation of Table A3.3 shows that this is only achieved at the cost of lower expected polytropic efficiency.

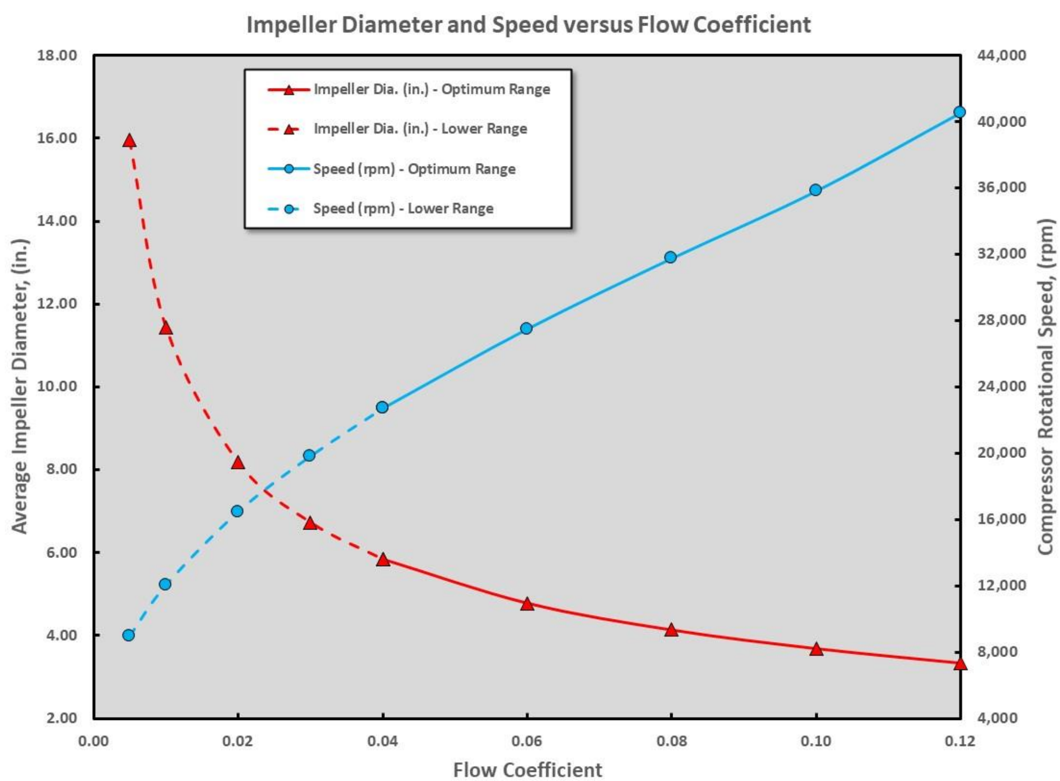

Figure A3.1: Case Study 3 Expanded Range Predicted Impeller Diameter and Speed

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A further evaluation was completed on the basis of a three stage section with design flow coefficients in this lower range with an assumed lower polytropic efficiency based upon 95% of the Aungier average predicted value. The results of this analysis are provided in Table A3.4.

Volumetric Flow, Qa (acfm)	257.7	257.7	257.7	257.7	257.7
Polytropic Head, Hp (ft·lbf/lbm)	17056.4	16789.7	16676.0	16638.1	16615.8
Number of Impellers in Section	3	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0050	0.0100	0.0200	0.0300	0.0400
Specific Speed, ns	0.2206	0.2992	0.4115	0.4960	0.5686
Specific Diameter, ds	13.2605	9.4765	6.7622	5.5557	4.8215
Specific Speed · Diameter, nsds	2.9253	2.8352	2.7827	2.7556	2.7414
Polytropic Head Coefficient, μ_p	0.4674	0.4976	0.5165	0.5268	0.5322
Polytropic Efficiency, η_p	0.4519	0.5975	0.6955	0.7361	0.7624
Work Input Coefficient, τ	1.0343	0.8328	0.7427	0.7156	0.6981
Impeller Diameter, D_{avg} (in.)	15.9465	11.4411	8.1779	6.7226	5.8362
Rotational Speed, N (rpm)	8990.9	12050.2	16490.5	19841.8	22723.1
Impeller Tip Speed, U_{tip} (ft/sec)	625.58	601.56	588.42	582.02	578.64

Table A3.4: Low Flow Coefficient Analysis Results for 3 Stage Section with 95% Efficiency

When a comparison is made between the results provided in Tables A3.3 and A3.4, it is once again evident there are relatively small differences in the calculated values of impeller diameter, rotational speed and impeller tip speed. Another item to note is the work input coefficient for the lowest flow coefficient in the first column. The fact that this value is above unity demonstrates that it is not a viable solution as this would require a greater enthalpy rise than available. In this case, the value of the estimated polytropic efficiency can influence the validity of the selection.

The minimum impeller diameter available from the actual equipment supplier was 11.0 inches for the compressor design utilized. A comparison is made between this actual equipment supplier selection and calculated values derived from assuming supplier values of flow coefficient, impeller diameter and rotational speed at the 100% Aungier average predicted efficiency. The comparison is provided in Table A3.5 below.

Case	Supplier Selection	Equal Flow Coefficient	Equal Impeller Diameter	Equal Rotational Speed
Volumetric Flow, Qa (acfm)	257.7	257.7	257.7	257.7
Polytropic Head, Hp (ft·lbf/lbm)	16818.0	16744.7	16676.4	16676.4
Number of Impellers in Section	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0102	0.0102	0.0108	0.0122
Specific Speed, ns	0.3256	0.3019	0.3097	0.3277
Specific Diameter, ds	9.1523	9.3853	9.1330	8.6002
Specific Speed · Diameter, nsds	2.9800	2.8335	2.8284	2.8180
Polytropic Head Coefficient, μ_p	0.4504	0.4982	0.5000	0.5037
Polytropic Efficiency, η_p	0.6738	0.6326	0.6427	0.6638
Work Input Coefficient, τ	0.6684	0.7876	0.7779	0.7588
Impeller Diameter, D_{avg} (in.)	11.0450	11.3385	11.0450	10.4007
Rotational Speed, N (rpm)	13131.0	12135.8	12410.6	13131.0
Impeller Tip Speed, U_{tip} (ft/sec)	632.82	600.40	598.10	595.90

Table A3.5: Selection Criteria Comparison

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The analysis shows reasonable agreement between the actual compressor selection provided by the equipment supplier and preliminary selections based upon the methods developed in this tutorial. Differences may be partially due to the more extreme process conditions and real gas behavior represented by this application. There is an approximate difference in the calculated polytropic head between the different cases of less than 0.9% which may be likely attributable to differences in calculated thermodynamic properties, solution methods and polytropic efficiency. All other parameters demonstrate reasonably close agreement between the actual data and the different methods. The deviation in selected impeller diameter is approximately 6% and the deviation in estimated speed is within approximately 8%.

Case Study 4: Process Refrigeration Application

Process refrigeration compression can also represent a very unique application. The high molecular weight of many refrigerants results in relatively low acoustic velocities and resulting high machine Mach numbers. Many refrigeration applications use either ambient air or cooling water as a heat sink to absorb the heat of compression and latent heat of the refrigerant, effectively setting the suction and discharge pressures for a given refrigerant. This allows only changes in refrigerant flow rates to adjust a specific refrigeration duty, which has resulted in significant volumetric flow rate requirements in such applications as large capacity petrochemical and LNG plants. Design process conditions for the refrigeration example are provided in Table A4.1 below. This is actually data for one section of a multi-section, sideload compressor design.

Volumetric Flow, Qa (acfm)	116,753.4
Gas Molecular Weight, MW	44.09
Suction Pressure, P_s (psig)	51.44
Suction Temperature, T_s (F)	45.6
Discharge Pressure, P_d (psig)	100.32

Table A4.1: Case Study 4 Design Process Conditions

The required polytropic head was initially estimated to be 9096 ft-lbf/lbm given a polytropic efficiency of 80%. Assuming a maximum value of head per stage at 14,000 ft-lbf/lbm, a minimum number of stages to satisfy this requirement is a single impeller. One additional constraint is that for this particular application, a large, single-shaft gas turbine with a very shallow speed range is used as the compressor driver. This shallow speed range effectively renders this as a constant speed drive application for compressor selection purposes.

Table A4.2 provides the results for a single stage section with variation in flow coefficient across a range including that of optimum efficiency and head coefficient. This was completed utilizing the average Aungier predicted efficiency relationship and included iterations on polytropic head resulting from predicted efficiency. This is most evident in the variation of calculated head across the flow coefficients. Although the values of head do vary, this variation is relatively small and comparison to the non-iterated value of head listed above also demonstrates a small deviation.

Volumetric Flow, Qa (acfm)	116753.4	116753.4	116753.4	116753.4	116753.4
Polytropic Head, Hp (ft·lbf/lbm)	9094.3	9077.4	9071.4	9068.4	9069.0
Number of Impellers in Section	1	1	1	1	1
Flow Coefficient, $Qa/N{\cdot}D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	109.7399	89.9150	77.8595	69.4397	62.8972
Rotational Speed, N (rpm)	1548.5	1875.2	2165.9	2443.4	2766.5
Impeller Tip Speed, U_{tip} (ft/sec)	741.47	735.68	735.80	740.31	759.23

Table A4.2: Flow Coefficient Analysis Results for Single Stage Section

It is important to note from the data presented in this table the combinations of impeller diameter and rotational speed. All impeller diameters are above 62 inches and the design operating speeds are all below 2800 rpm. As noted previously, the output speed of the

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driver is essentially fixed at a speed of 3600 rpm. In order to raise potential design speeds of the selection, calculations will need to be completed for flow coefficients above the optimum range. Table A4.3 gives the results for calculations performed in this higher range.

Volumetric Flow, Qa (acfm)	116753.4	116753.4	116753.4	116753.4	116753.4
Polytropic Head, Hp (ft·lbf/lbm)	9069.0	9070.8	9076.2	9083.4	9091.8
Number of Impellers in Section	1	1	1	1	1
Flow Coefficient, $Qa/N{\cdot}D^3$	0.1200	0.1300	0.1500	0.1700	0.1900
Specific Speed, ns	1.0179	1.0714	1.1544	1.2512	1.4297
Specific Diameter, ds	2.7615	2.6410	2.4379	2.2664	2.0991
Specific Speed · Diameter, nsds	2.8110	2.8297	2.8144	2.8357	3.0010
Polytropic Head Coefficient, μ_p	0.5062	0.4995	0.5050	0.4974	0.4441
Polytropic Efficiency, η_p	0.8524	0.8487	0.8375	0.8232	0.8071
Work Input Coefficient, τ	0.5939	0.5886	0.6030	0.6042	0.5503
Impeller Diameter, D_{avg} (in.)	62.8972	60.1506	55.5166	51.6000	47.7795
Rotational Speed, N (rpm)	2766.5	2912.3	3139.3	3404.5	3892.9
Impeller Tip Speed, U_{tip} (ft/sec)	759.23	764.36	760.46	766.52	811.57

Table A4.3: Flow Coefficient Analysis Results for Single Stage Section Above Optimum Range

Figure A4.1 provides the results for required impeller diameter and rotational speed derived from using the variable flow coefficient methodology presented previously across both the optimum range and upper extended range of flow coefficients. It is evident from this figure that the impeller diameter decreases in this upper flow coefficient range while the rotational speed is increased.

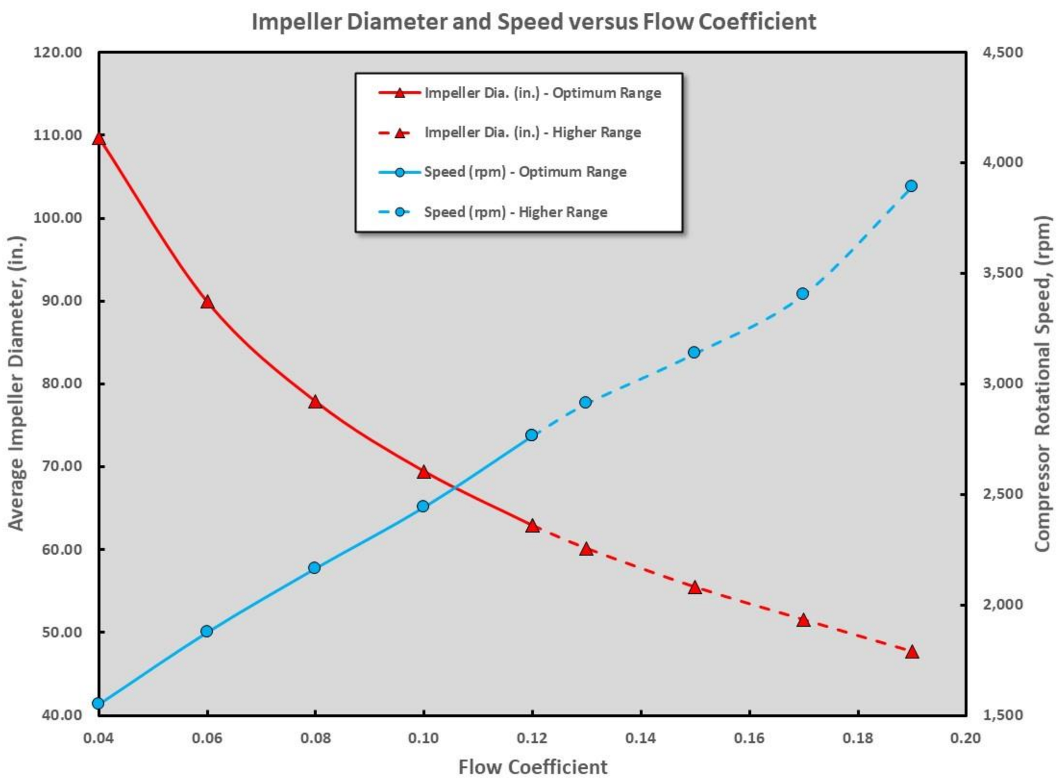

Figure A4.1: Case Study 4 Expanded Range Predicted Impeller Diameter and Speed

A further evaluation was completed on the basis of a single stage section with design flow coefficients in this upper range with an assumed lower polytropic efficiency based upon 95% of the Aungier average predicted value. The results of this analysis are provided in Table A4.4.

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Volumetric Flow, Qa (acfm)	116753.4	116753.4	116753.4	116753.4	116753.4
Polytropic Head, Hp (ft·lbf/lbm)	9090.4	9092.3	9098.0	9105.6	9114.4
Number of Impellers in Section	1	1	1	1	1
Flow Coefficient, $Qa/N{\cdot}D^3$	0.1200	0.1300	0.1500	0.1700	0.1900
Specific Speed, ns	1.0179	1.0714	1.1544	1.2512	1.4297
Specific Diameter, ds	2.7615	2.6410	2.4379	2.2664	2.0991
Specific Speed · Diameter, nsds	2.8110	2.8297	2.8144	2.8357	3.0010
Polytropic Head Coefficient, μ_p	0.5062	0.4995	0.5050	0.4974	0.4441
Polytropic Efficiency, η_p	0.8098	0.8062	0.7956	0.7820	0.7668
Work Input Coefficient, τ	0.6251	0.6196	0.6347	0.6360	0.5792
Impeller Diameter, D_{avg} (in.)	62.8601	60.1150	55.4834	51.5687	47.7498
Rotational Speed, N (rpm)	2771.4	2917.5	3145.0	3410.8	3900.1
Impeller Tip Speed, U_{tip} (ft/sec)	760.12	765.26	761.37	767.46	812.58

Table A4.4: Low Flow Coefficient Analysis Results for Single Stage Section with 95% Efficiency

When a comparison is made between the results provided in Tables A4.3 and A4.4, it is once again evident there are relatively small differences in the calculated values of impeller diameter, rotational speed and impeller tip speed. This should provide assurance that these calculated values are not very sensitive to variations between predicted and actual levels of polytropic efficiency. The major influences of polytropic efficiency can be found in the absorbed power and discharge temperature of the specific application.

The design rotational speed of 3600 rpm was utilized by the actual equipment supplier in making their selection. A comparison is made between this actual equipment supplier selection and calculated values derived from assuming supplier values of flow coefficient, impeller diameter and rotational speed at the 100% Aungier average predicted efficiency. The comparison is provided in Table A4.5 below.

Case	Supplier Selection	Equal Flow Coefficient	Equal Impeller Diameter	Equal Rotational Speed
Volumetric Flow, Qa (acfm)	116753.4	116753.4	116753.4	116753.4
Polytropic Head, Hp (ft·lbf/lbm)	9070.0	9072.5	9089.4	9089.4
Number of Impellers in Section	1	1	1	1
Flow Coefficient, $Qa/N{\cdot}D^3$	0.1371	0.1371	0.1556	0.1709
Specific Speed, ns	1.3245	1.1040	1.1655	1.3224
Specific Diameter, ds	2.4114	2.5636	2.4127	2.2420
Specific Speed · Diameter, nsds	3.1939	2.8303	2.8120	2.9648
Polytropic Head Coefficient, μ_p	0.3921	0.4993	0.5059	0.4551
Polytropic Efficiency, η_p	0.8140	0.8451	0.8338	0.8225
Work Input Coefficient, τ	0.4817	0.5908	0.6067	0.5533
Impeller Diameter, D_{avg} (in.)	54.9210	58.3847	54.9210	51.0360
Rotational Speed, N (rpm)	3600.0	3001.4	3173.0	3600.0
Impeller Tip Speed, U_{tip} (ft/sec)	862.70	764.60	760.36	801.67

Table A4.5: Selection Criteria Comparison

The analysis shows reasonable agreement between the actual compressor selection provided by the equipment supplier and preliminary selections based upon the methods developed in this tutorial. There is an approximate difference in the calculated

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polytropic head between the different cases of approximately 0.5% which may be attributable to differences in calculated thermodynamic properties, solution methods and polytropic efficiency. All other parameters demonstrate acceptable agreement between the actual data and the different methods. The deviation in selected impeller diameter is approximately 14% and the deviation in estimated speed is within 20%. One explanation for this greater deviation than found in the other examples may be due to the fixed speed application. It is common practice to provide additional impeller diameter or assume a reduced design speed in fixed speed applications to ensure that produced head performance guarantees can be achieved. This is one reason why many fixed speed compressor designs are found to produce excessive head during factory acceptance testing.

Case Study 5: Carbon Dioxide Injection Application

The compression of carbon dioxide represents a considerable range of application conditions. The high molecular weight of carbon dioxide, like many industrial and commercial refrigerants, results in relatively low acoustic velocities and high machine Mach numbers. Carbon capture and sequestration conceptual designs include both the collection of carbon dioxide at low pressures and relatively high volumetric flow rates coupled with high pressure, low volumetric flow injection duties. This results in potential compressor designs at the extents of possible flow coefficients. Some existing applications involve the use of integrally geared, multi-stage unshrouded impeller designs which are not specifically addressed herein. Although required carbon dioxide injection pressures are not as high as hydrocarbon-based injection applications, the higher molecular weight results in comparable or greater fluid densities and equivalent low volumetric flow rates. Design process conditions for a carbon dioxide injection example are provided in Table A5.1 below. This is actually data for the fourth section of a multi-section, injection compressor design with a constant speed driver connected through a speed increasing gear.

Volumetric Flow, Qa (acfm)	480.3
Gas Molecular Weight, MW	43.90
Suction Pressure, P_s (psig)	796.93
Suction Temperature, T_s (F)	111.2
Discharge Pressure, P_d (psig)	2654.00

Table A5.1: Case Study 5 Design Process Conditions

The required polytropic head was initially estimated to be 20,980 ft-lbf/lbm given a polytropic efficiency of 80%. Assuming a maximum value of head per stage at 14,000 ft-lbf/lbm, a minimum number of stages to satisfy this requirement is two impellers, however, this would result in a stage differential pressure of over 1000 psi. Three stages would reduce this level down to approximately 700 psi and four reduced further down to the 500 psi range. Actual machine selection was based upon three stages which will be analyzed here.

Table A5.2 provides the results for a section with variation in flow coefficient across a range including that of optimum efficiency and head coefficient. This was completed utilizing the average Aungier predicted efficiency relationship and included iterations on polytropic head resulting from predicted efficiency. This is most evident in the variation of calculated head across the flow coefficients. Although the values of head do vary, this variation is relatively small and comparison to the non-iterated value of head listed above also demonstrates a small deviation.

Volumetric Flow, Qa (acfm)	480.3	480.3	480.3	480.3	480.3
Polytropic Head, Hp (ft·lbf/lbm)	20969.0	20835.3	20787.9	20764.0	20768.8
Number of Impellers in Section	3	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	7.5173	6.1663	5.3417	4.7650	4.3159
Rotational Speed, N (rpm)	19818.1	23917.1	27591.2	31107.4	35224.9
Impeller Tip Speed, U_{tip} (ft/sec)	650.04	643.50	643.08	646.76	663.34

Table A5.2: Flow Coefficient Analysis Results for Single Stage Section

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It is important to note from the data presented in this table the combinations of impeller diameter and rotational speed. All impeller diameters are less than 8 inches and the design operating speeds are all above 19,000 rpm. In order to lower potential design speeds of the selection, calculations will need to completed for flow coefficients below the optimum range. Table A5.3 gives the results for calculations performed in this lower range.

Volumetric Flow, Qa (acfm)	480.3	480.3	480.3	480.3	480.3
Polytropic Head, Hp (ft·lbf/lbm)	23501.6	21948.6	21304.4	21092.6	20969.0
Number of Impellers in Section	3	3	3	3	3
Flow Coefficient, $Qa/N{\cdot}D^3$	0.0050	0.0100	0.0200	0.0300	0.0400
Specific Speed, ns	0.2206	0.2992	0.4115	0.4960	0.5686
Specific Diameter, ds	13.2605	9.4765	6.7622	5.5557	4.8215
Specific Speed · Diameter, nsds	2.9253	2.8352	2.7827	2.7556	2.7414
Polytropic Head Coefficient, μ_p	0.4674	0.4976	0.5165	0.5268	0.5322
Polytropic Efficiency, η_p	0.4757	0.6290	0.7321	0.7749	0.8025
Work Input Coefficient, τ	0.9826	0.7911	0.7056	0.6798	0.6632
Impeller Diameter, D_{avg} (in.)	20.0939	14.6075	10.5014	8.6493	7.5173
Rotational Speed, N (rpm)	8375.5	10791.2	14515.0	17364.0	19818.1
Impeller Tip Speed, U_{tip} (ft/sec)	734.33	687.80	665.09	655.31	650.04

Table A5.3: Flow Coefficient Analysis Results for Single Stage Section Below Optimum Range

Figure A5.1 provides the results for required impeller diameter and rotational speed derived from using the variable flow coefficient methodology presented previously across both the optimum range and lower extended range of flow coefficients. It is evident from this figure that the impeller diameter increases in this lower flow coefficient range while the rotational speed is decreased.

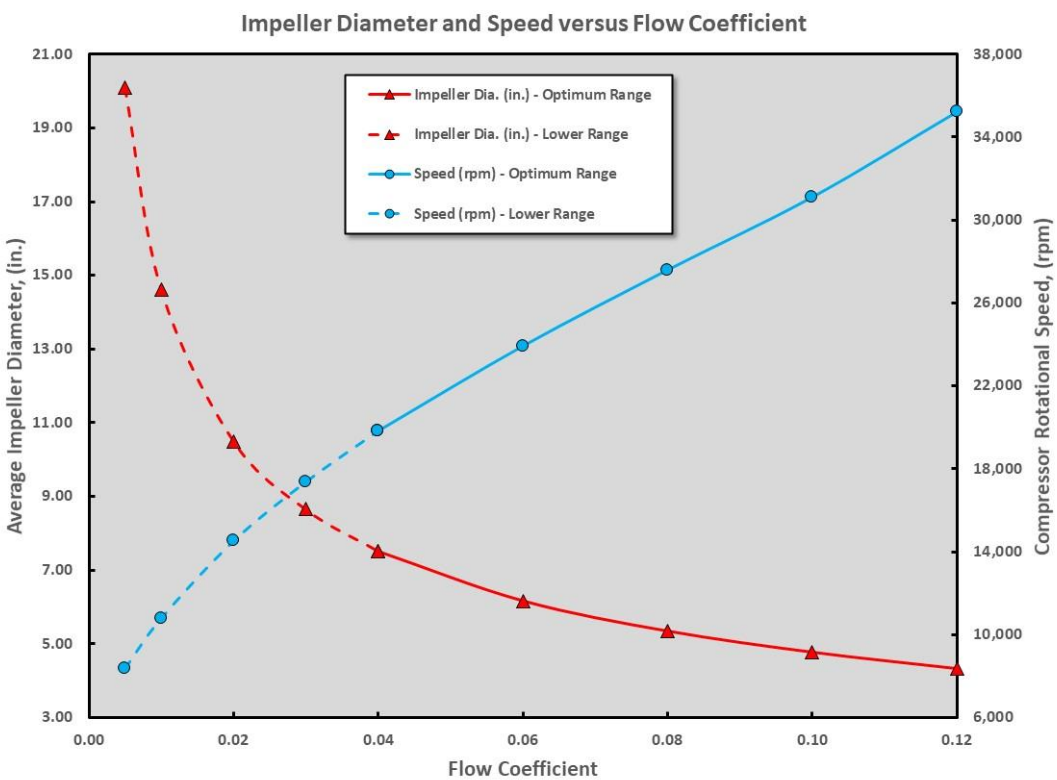

Figure A5.1: Case Study 5 Expanded Range Predicted Impeller Diameter and Speed

A further evaluation was completed on the basis of a single stage section with design flow coefficients in this lower range with an assumed lower polytropic efficiency based upon 95% of the Aungier average predicted value. The results of this analysis are provided in Table A5.4.

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Volumetric Flow, Qa (acfm)	480.3	480.3	480.3	480.3	480.3
Polytropic Head, Hp (ft·lbf/lbm)	23850.0	22195.1	21508.7	21283.1	21151.5
Number of Impellers in Section	3	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0050	0.0100	0.0200	0.0300	0.0400
Specific Speed, ns	0.2206	0.2992	0.4115	0.4960	0.5686
Specific Diameter, ds	13.2605	9.4765	6.7622	5.5557	4.8215
Specific Speed · Diameter, nsds	2.9253	2.8352	2.7827	2.7556	2.7414
Polytropic Head Coefficient, μ_p	0.4674	0.4976	0.5165	0.5268	0.5322
Polytropic Efficiency, η_p	0.4519	0.5975	0.6955	0.7361	0.7624
Work Input Coefficient, τ	1.0343	0.8328	0.7427	0.7156	0.6981
Impeller Diameter, D_{avg} (in.)	20.0201	14.5668	10.4764	8.6299	7.5011
Rotational Speed, N (rpm)	8468.4	10881.9	14619.3	17481.5	19947.3
Impeller Tip Speed, U_{tip} (ft/sec)	739.75	691.65	668.27	658.26	652.86

Table A5.4: Low Flow Coefficient Analysis Results for Single Stage Section with 95% Efficiency

When a comparison is made between the results provided in Tables A5.3 and A5.4, it is once again evident there are relatively small differences in the calculated values of impeller diameter, rotational speed and impeller tip speed. Another item to note is the work input coefficient for the lowest flow coefficient in the first column. The fact that this value is above unity demonstrates that it is not a viable solution as this would require a greater enthalpy rise than available. The major influences of polytropic efficiency can be found in the absorbed power and discharge temperature of the specific application.

A comparison is made between the actual equipment supplier selection and calculated values derived from assuming supplier values of flow coefficient, impeller diameter and rotational speed at the 100% Aungier average predicted efficiency. The comparison is provided in Table A5.5 below.

Case	Supplier Selection	Equal Flow Coefficient	Equal Impeller Diameter	Equal Rotational Speed
Volumetric Flow, Qa (acfm)	480.3	480.3	480.3	480.3
Polytropic Head, Hp (ft·lbf/lbm)	21158.0	21516.4	21634.1	21634.1
Number of Impellers in Section	3	3	3	3
Flow Coefficient, $Qa/N \cdot D^3$	0.0149	0.0149	0.0129	0.0155
Specific Speed, ns	0.3716	0.3579	0.3361	0.3655
Specific Diameter, ds	8.3207	7.8288	8.3671	7.6542
Specific Speed · Diameter, nsds	3.0923	2.8018	2.8118	2.7975
Polytropic Head Coefficient, μ_p	0.4183	0.5095	0.5059	0.5111
Polytropic Efficiency, η_p	0.6460	0.6942	0.6730	0.7001
Work Input Coefficient, τ	0.6475	0.7340	0.7518	0.7301
Impeller Diameter, D_{avg} (in.)	12.9440	12.1277	12.9440	11.8410
Rotational Speed, N (rpm)	13041.0	12717.6	11990.7	13041.0
Impeller Tip Speed, U_{tip} (ft/sec)	736.54	672.98	677.22	673.78

Table A5.5: Selection Criteria Comparison

The analysis shows reasonable agreement between the actual compressor selection provided by the equipment supplier and preliminary selections based upon the methods developed in this tutorial. There is an approximate difference in the calculated polytropic head between the different cases of approximately 2.5% which may be attributable to differences in calculated thermodynamic properties, solution methods and polytropic efficiency. All other parameters demonstrate acceptable agreement

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between the actual data and the different methods. The deviation in selected impeller diameter is approximately 10% and the deviation in estimated speed is within 9%. One explanation for this greater deviation than found in the other examples may be due to the fixed speed application. It is common practice to provide additional impeller diameter or assume a reduced design speed in fixed speed applications to ensure that produced head performance guarantees can be achieved. This is one reason why many fixed speed compressor designs are found to produce excessive head during factory acceptance testing.

Case Study 6: Hydrogen Recycle Application

The topic of hydrogen compression is currently of considerable interest although the scope and definition of potential applications remains largely ill-defined. Utilization of hydrogen as a pure fuel or a blending component of conventional hydrocarbon-based fuels is being evaluated along with the processes capable of producing adequate quantities to support the associated demand. Compression of hydrogen represents a substantial set of different design criteria when compared to the high molecular weight applications of industrial refrigeration and carbon dioxide compression. This much lower molecular weight gas results in higher acoustic velocities and lower pressure ratio for a given amount of produced head, effectively requiring higher impeller tip speeds. Stated differently, the low molecular weight gas corresponds to a mechanical limitation of stress due to high tip speeds versus a fluid mechanic limitation (choking) that is created with the high molecular weight gases. In this specific application example, a variable speed drive was applied to a more conventional hydrogen recycle application in a refinery catalytic reforming operation which is a mixture of primarily hydrogen and lesser amounts of hydrocarbons. Design process conditions are provided in Table A6.1 below.

Volumetric Flow, Qa (acfm)	52,329
Gas Molecular Weight, MW	7.27
Suction Pressure, P_s (psig)	34.60
Suction Temperature, T_s (F)	112.0
Discharge Pressure, P_d (psig)	101.80

Table A6.1: Case Study 6 Design Process Conditions

The required polytropic head was estimated to be 118,681 ft-lbf/lbm assuming a polytropic efficiency of 80%. Assuming a maximum value of head per stage at 14,000 ft-lbf/lbm, a minimum number of stages to satisfy this requirement is nine. The actual equipment supplier selection was based upon seven stages with impeller tip speeds above the normal range. An evaluation was provided with the selected seven stages along with an alternate selection of eight stages. Figure A6.1 provides the results for required impeller diameter and rotational speed derived from using the variable flow coefficient methodology presented previously across the optimum range of flow coefficients.

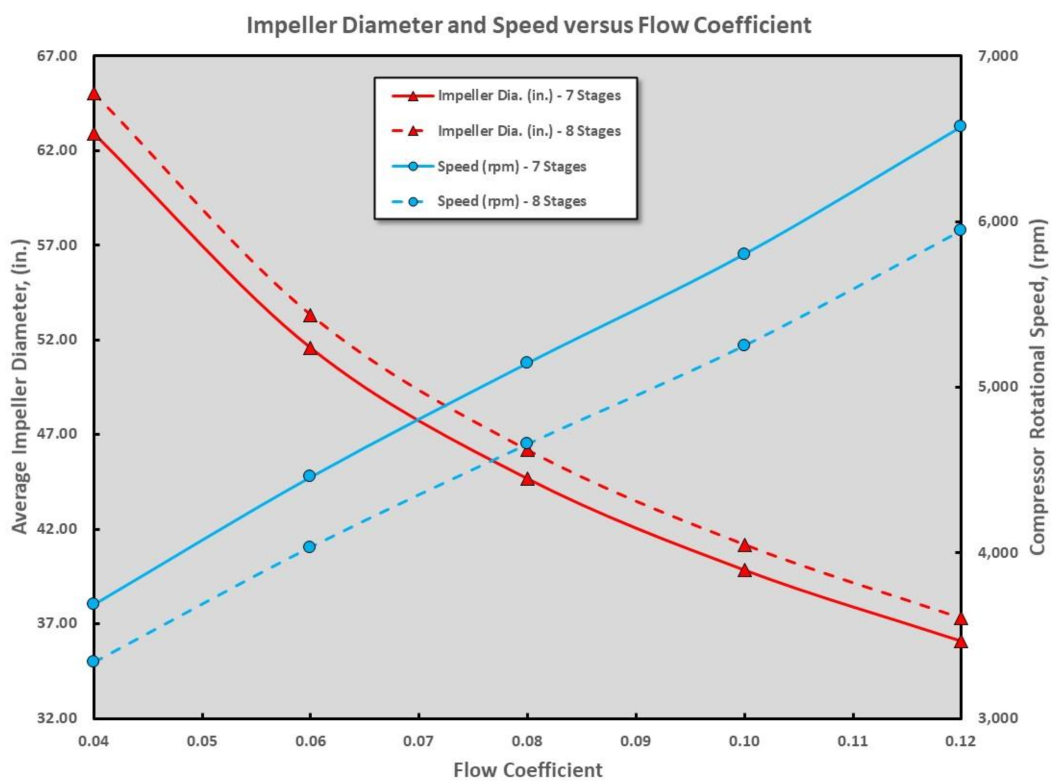

Figure A6.1: Case Study 6 Predicted Impeller Diameter and Speed

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A consistent observation is that an increase in the impeller count actually results in larger required impeller diameters which is offset by a more significant reduction in rotational speed. Another trend to note is that rotational speed tends to increase while impeller diameter tends to decrease as flow coefficients become larger. In order to maximize the efficiency across all stages within the section, it is desirable to maintain the inlet flow coefficient near the top of the optimum range, although the overall volume reduction is less than the previous hydrocarbon seven stage example.

Table A6.2 provides the results for a seven stage section with variation in flow coefficient across a range including that of optimum efficiency and head coefficient. This was completed utilizing the average Aungier predicted efficiency relationship and included iterations on polytropic head resulting from predicted efficiency. This is most evident in the variation of calculated head across the flow coefficients. Although the values of head do vary, this variation is relatively small and comparison to the non-iterated value of head also demonstrates a small deviation.

Volumetric Flow, Qa (acfm)	52329	52329	52329	52329	52329
Polytropic Head, Hp (ft·lbf/lbm)	118634.7	118072.2	117872.1	117771.1	117791.2
Number of Impellers in Section	7	7	7	7	7
Flow Coefficient, $Qa/N{\cdot}D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	62.8802	51.5581	44.6569	39.8329	36.0789
Rotational Speed, N (rpm)	3689.3	4457.8	5144.9	5801.8	6569.5
Impeller Tip Speed, U_{tip} (ft/sec)	1012.21	1002.85	1002.48	1008.37	1034.18

Table A6.2: Flow Coefficient Analysis Results for 7 Stage Section

A similar analysis for an eight stage section is provided in Table A6.3 again using the average calculated value of polytropic efficiency from the Aungier relations.

Volumetric Flow, Qa (acfm)	52329	52329	52329	52329	52329
Polytropic Head, Hp (ft·lbf/lbm)	118634.7	118072.2	117872.1	117771.1	117791.2
Number of Impellers in Section	8	8	8	8	8
Flow Coefficient, $Qa/N{\cdot}D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.8025	0.8351	0.8473	0.8537	0.8524
Work Input Coefficient, τ	0.6632	0.6462	0.6362	0.6236	0.5939
Impeller Diameter, D_{avg} (in.)	65.0148	53.3083	46.1728	41.1850	37.3037
Rotational Speed, N (rpm)	3337.7	4033.0	4654.6	5248.9	5943.4
Impeller Tip Speed, U_{tip} (ft/sec)	946.83	938.08	937.74	943.24	967.39

Table A6.3: Flow Coefficient Analysis Results for 8 Stage Section

The equipment supplier involved with this application offered constant impeller diameters of 48.97 inches. Evaluation of the calculated data shows that the selection with seven stages in the section provides a solution with the best match of flow coefficient near the middle of the optimum range between 0.05 and 0.11.

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A further evaluation was completed on the basis of a seven stage section with an assumed lower polytropic efficiency based upon 95% of the Aungier average predicted value. The results of this analysis are provided in Table A6.4.

Volumetric Flow, Qa (acfm)	52329	52329	52329	52329	52329
Polytropic Head, Hp (ft·lbf/lbm)	119397.6	118802.9	118591.2	118484.5	118505.6
Number of Impellers in Section	7	7	7	7	7
Flow Coefficient, $Qa/N \cdot D^3$	0.0400	0.0600	0.0800	0.1000	0.1200
Specific Speed, ns	0.5686	0.6895	0.7968	0.8991	1.0179
Specific Diameter, ds	4.8215	3.9486	3.4186	3.0487	2.7615
Specific Speed · Diameter, nsds	2.7414	2.7225	2.7239	2.7410	2.8110
Polytropic Head Coefficient, μ_p	0.5322	0.5396	0.5391	0.5324	0.5062
Polytropic Efficiency, η_p	0.7624	0.7933	0.8050	0.8110	0.8098
Work Input Coefficient, τ	0.6981	0.6802	0.6697	0.6565	0.6251
Impeller Diameter, D_{avg} (in.)	62.7796	51.4787	44.5890	39.7728	36.0244
Rotational Speed, N (rpm)	3707.0	4478.5	5168.4	5828.1	6599.3
Impeller Tip Speed, U_{tip} (ft/sec)	1015.45	1005.94	1005.54	1011.42	1037.32

Table A6.4: Flow Coefficient Analysis Results for 7 Stage Section with 95% Efficiency

When a comparison is made between the results provided in Tables A6.2 and A6.4, it is evident there are relatively small differences in the calculated values of impeller diameter, rotational speed and impeller tip speed. This should provide assurance that these calculated values are not very sensitive to variations between predicted and actual levels of polytropic efficiency. The major influences of polytropic efficiency can be found in the absorbed power and discharge temperature of the specific application.

Finally, a comparison is made between the actual equipment supplier selection and calculated values derived from assuming supplier values of flow coefficient, impeller diameter and rotational speed at the 100% Aungier average predicted efficiency. The assumed value calculations utilize the three different methods presented earlier, namely sizing based upon an assumed value of flow coefficient, impeller diameter, or rotational speed, respectively. The comparison is provided in Table A6.5 below.

Case	Supplier Selection	Equal Flow Coefficient	Equal Impeller Diameter	Equal Rotational Speed
Volumetric Flow, Qa (acfm)	52329	52329	52329	52329
Polytropic Head, Hp (ft·lbf/lbm)	112224.0	117993.2	117382.1	117382.1
Number of Impellers in Section	7	7	7	7
Flow Coefficient, $Qa/N \cdot D^3$	0.0695	0.0695	0.0666	0.0613
Specific Speed, ns	0.7208	0.7417	0.7267	0.6969
Specific Diameter, ds	3.7051	3.6710	3.7469	3.9064
Specific Speed · Diameter, nsds	2.6706	2.7226	2.7228	2.7224
Polytropic Head Coefficient, μ_p	0.5608	0.5396	0.5396	0.5397
Polytropic Efficiency, η_p	0.8744	0.8399	0.8385	0.8360
Work Input Coefficient, τ	0.6414	0.6425	0.6435	0.6456
Impeller Diameter, D_{avg} (in.)	48.9960	47.9408	48.9960	51.0808
Rotational Speed, N (rpm)	4486.0	4792.7	4677.6	4486.0
Impeller Tip Speed, U_{tip} (ft/sec)	959.04	1002.54	1000.00	999.84

Table A6.5: Selection Criteria Comparison

This analysis once again shows reasonable agreement between the actual compressor selection provided by the equipment supplier and preliminary selections based upon the methods developed in this tutorial. The approximate difference in the calculated polytropic head between the different cases of less than 5% which may be likely attributable to differences in calculated thermodynamic

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properties, solution methods and polytropic efficiency. All other parameters demonstrate reasonably close agreement between the actual data and the different methods. Most importantly, the variations in impeller diameter and rotational speed are minimal. The deviation in selected impeller diameter is within 7% and the deviation in estimated speed is also within 7%.

In summary, these examples show that the methods introduced in this work have the ability to predict a compressor selection that closely approximates an actual selection provided by an equipment supplier.

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