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CENTIFUGAL COMPRESSOR EVOLUTION

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Jim is a member of ASME and the ASME Turbomachinery Committee. He has authored or co-authored over fifty technical papers and has instructed seminars and tutorials at Texas A&M and Dresser-Rand. He currently holds four U.S. patents and has several other patents pending. He was elected an ASME Fellow in 2008 and a Dresser-Rand Fellow in 2015.

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ABSTRACT

This paper addresses the advancements that have been made during the past 70 years in the design, analysis, and manufacturing methods for centrifugal compressors. The paper provides a historical perspective on these disciplines, citing how they and other technological innovations have contributed to significant improvements in the aerodynamic and mechanical performance of modern turbomachines.

INTRODUCTION

The first industrial application of centrifugal or radial compressors was in conjunction with early gas turbines work done around the turn of the 20th century. Some of the earliest work was done by Elling, who developed the first gas turbine to generate positive power in 1903 (Bolland & Veer, 2003). Such compressors have also been used by the process industry since the early 1900s. One of the earliest applications was as blast furnaces blowers for steel mills. For example, this Original Equipment Manufacturer's (OEM's) first record of a centrifugal compressor was serial number 7, sold to Scullin Steel Company of St. Louis, MO, USA in 1912. These blowers were quite often very large machines even by today's standards. While they included fundamentally the same components (bearings, seals, impellers, diffusers, etc.), the machine components of that era differed significantly from the complex internals in compressors offered by OEMs today.

This paper traces the evolution of centrifugal compressors and their internals from the relatively primitive machines of the early 20th century to the sophisticated turbomachines of the 21st century. The paper will cite the role of advanced manufacturing methods as well as the importance of advanced analytical methods in the development of today's complicated equipment.

Historical Perspective

Improved manufacturing methods have been a key factor in the development of today's high-performance centrifugal turbomachines. It would make little sense to apply modern sophisticated analytical and design techniques if manufacturing methods were not available to precisely build the complex shapes required to achieve the high performance. Today's manufacturing methods are critical to the current high efficiency levels. Such has not always been the case.

In the early days of process centrifugal compressor development, design choices were restricted in large part by the manufacturing methods available at that time. OEMs had to create designs that could be fabricated with the limited number of methods available. These included machining (i.e., turning, 3-axis milling), joining (i.e., welding, riveting) and castings.

Machining techniques were limited to turning and 3-axis milling. Such methods are capable of creating fairly simple 2-D shapes. While these are adequate for a wide range of compressor applications, they proved to be inadequate for high flow and/or high Mach number machines (more on this later). To create the more sophisticated shapes for higher flow applications, OEMs were forced to use welded fabrications with die pressed or custom rolled/hammered blades, or castings. In fact, the welded impeller did not become commonplace in process machinery until the late 1950s or early 1960s. Therefore, prior equipment included impellers that were either cast or fabricated via riveting. Some of the earliest riveted impellers date back to the early 1920s.

Likewise, stationary components were fabricated via welded assemblies or casting. Castings were the method of choice for most OEMs because of (a) the cost advantages attained when multiple copies of the same component were required; and (b) optimized performance was not a critical consideration at this time. The use of castings for compressor casings and other stationary components was prevalent through the mid-1950s. The most common were sand castings; wherein the mold was formed from compact sand. Castings formed using this approach were typically had very rough surfaces, which compromised the achievable aerodynamic performance. Still, it was not unusual for the flow path of a process centrifugal to be composed almost entirely of cast components. In most machines from that era, parts that were not cast were either welded, bolted, or riveted fabrications.

In these early machines, the primary driver (as defined by the end users) was that the unit compresses gas. Energy consumed was not a major consideration. However, as energy costs increased and as competition escalated between OEMs, it became necessary to develop compressors capable of delivering higher performance levels.

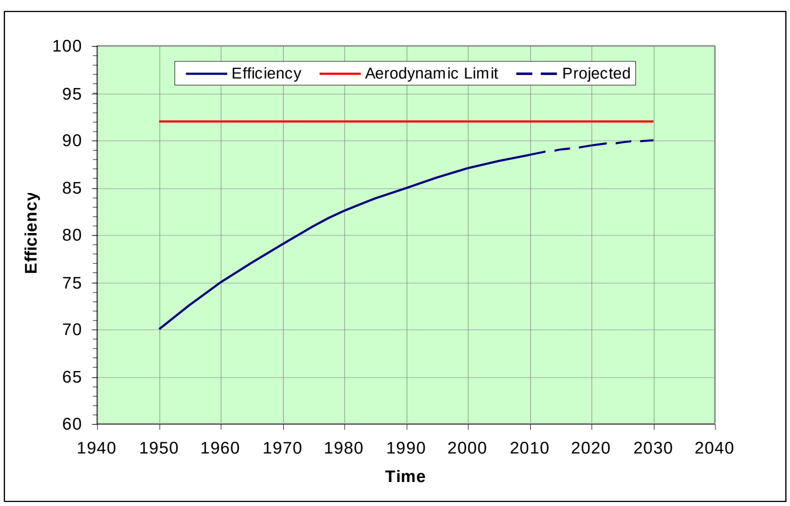


Figure 1 – Efficiency Trends

The general trend of peak compressor efficiency over the past 60 years is shown in Figure 1. Note that this curve represents centrifugal compressor stages with a flow coefficient, ϕ, greater than 0.080. Stages with lower flow coefficients have lower attainable peak efficiencies due to higher frictional losses, etc. As can be seen, in the 1950s, efficiencies were typically in the range of 70 percent to 75 percent. Energy was abundant, so this relatively low centrifugal compressor performance was not a concern. However, the energy crisis of the mid-to-late 1970s caused compressor users and OEMs to give greater attention to reduced power consumption. Significant strides were made in driver and compressor performance during those years as efficiency levels reached 80 percent to 85 percent. Further improvements continued through the 1990s and into the new century and optimal efficiency levels have now reached the upper 80s and are pushing toward 90 percent. However, it is commonly held that the multi-stage centrifugal compressor industry is approaching an asymptotic efficiency limit. The ultimate limit is the subject of much conjecture but most place the limit between 90 percent and 92 percent polytropic efficiency. That is, it might not be practical or even possible to design a multi-stage process centrifugal compressor that will achieve greater than 92 percent efficiency. Obviously, Newton's Laws and the Laws of Thermodynamics are sacrosanct and dictate the impossibility of reaching 100 percent efficiency. In addition, there are some fundamental losses (i.e., secondary flows, boundary layer effects, leakage effects, windage, bearing friction, shear forces, etc.) that cannot be avoided in centrifugal stages. Again, current thinking is that these losses will limit the industry to approximately 90 to 92 percent for multi-stage machines (Shepherd, 1956; Japikse & Baines, 1994; Sorokes, 1995; Aungier, 2000).

Note the reduced slope of the efficiency increase over the past decade as compared to prior decades. Clearly, there is diminishing return for the investments being made to achieve higher efficiency simply because there is less efficiency improvement to be gained. Further improvements are possible by: (a) addressing what had previously been considered secondary or tertiary performance factors such as leakage paths, external surface finishes on impellers, etc.; (b) developing ever more sophisticated aerodynamic components; or (c) merging of axial and radial centrifugal technologies. These efforts might result in higher stage or compressor efficiency but they might do so at the expense of overall flow range (Sorokes, 2003). While it is possible that the theoretical ceiling will be broken, incremental changes in efficiency over the next decade will certainly be measured in quarter, half and single points rather than the five to ten point improvements achieved in earlier decades.

The subject of increased performance will be addressed further in the Looking Forward section later in the paper. However, before considering the future, the discussion will turn to the factors that contributed to the performance improvement from the 1950s through 2018. These will be broken down in two major categories: aerodynamics and mechanical. In each category, the key components will be presented along with the improvements possible because of the advances in manufacturing and analytical techniques.

AERODYNAMICS

The key aerodynamic components in the centrifugal compressor are the inlet, inlet guide vanes (IGV), impeller, diffuser, return channel, volute, and sidestream (or side entry / exit). All have benefited greatly from improved manufacturing and analysis methods. Several of these components will now be discussed in decreasing order of importance.

Impellers

Clearly, attaining high performance from a centrifugal compressor demands superior aerodynamic designs and there is no more critical component in achieving such than the centrifugal impeller. Impellers are responsible for all of the work added to the gas stream, so it is not possible to achieve high efficiency in an overall compressor or in a compressor stage without a very well-designed impeller. No amount of tuning in the stationary components can make up for a poorly-designed impeller. A very large portion of the efficiency gain over the past few decades has been due to the more sophisticated impeller geometries made possible by the progress in manufacturing and analytical practices.

As mentioned in the introduction, early impellers were typically fabricated via welding, brazing, riveting or castings. Each method imposed limits on the attainable impeller geometry and, consequently, on the achievable performance.

In the 1950s or 1960s, OEMs began offering welded impellers. These fell into two categories: two-piece or three-piece welded. In the two-piece configuration, the impeller blades were 3-axis milled onto the cover (shroud) or disk (hub) and were then fillet-welded to the disk (hub) or cover (shroud). Because of the 3-axis milling, the blade shapes were typically 2-D in nature; i.e., sections of circles, ellipses or some other 2-D geometry shape. Such shapes severely limit the aerodynamics of the design but this was all that 3-axis milling allowed. Further, to do the fillet welding, the impeller passage opening had to be large enough to accommodate the weld fixtures (typically 0.6 inch (15.25mm) or larger). Therefore, welded impellers for very low-flow coefficient impellers, with their inherently narrow flow passage, were not possible. Such narrow impellers were built via "through the blade" riveting (see Figure 2) and/or casting.

Figure 2 – Riveted impellers

To achieve reasonable performance in higher flow coefficient stages; i.e., $\phi > 0.040$, it is imperative that the impeller inlet blade angles match the non-uniform inlet flow angles. An explanation is in order.

In high-flow coefficient stages, there is a large variation in approach angle of the gas at the impeller inlet from the shroud to the hub. This is due to the variation in the peripheral and meridional velocity at the impeller inlet (Figure 3). The peripheral velocity, U1x, is a function of the diameter at the various locations so U1s is much higher than U1h in high flow coefficient stages where there is inherently a large change in diameter from shroud to hub.

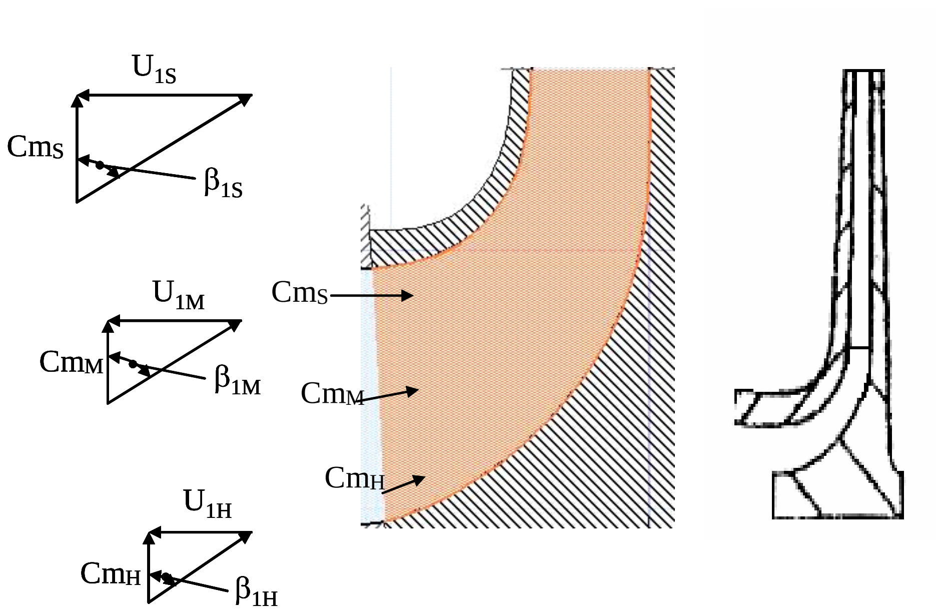


Figure 3 – Impeller Inlet Velocity Distribution

In addition, there is also a change in local curvature along the leading edge between the shroud and hub. The meridional velocity, Cxm, is impacted by this local curvature such that C1s is higher than either C1m or C1h. For further discussion on this subject, see Sorokes and Kopko (2007). Because of the variation in U1x and C1x, there is a significant difference in the gas approach angle between the shroud and hub. The only way to match these angles is to use a 3-D blade shape that has a varying inlet angle across the leading edge.

As end users required higher and higher flow rates, many OEMs did apply 2-D blading in high-flow coefficient stages but the performance of these machines was compromised because of the less-than-optimal flow incidence caused by the 2-D shapes.

As the demand for higher performance increased, OEMs were forced to develop methods to manufacture 3-D blading. Early approaches included castings or 3-piece fabrications. In the former, the 3-D blade shapes was created via casting of complex patterns, providing the leading edge angles necessary to achieve reasonable incidence angles. In the 3-piece fabrication, the “three pieces” were the shroud, the hub, and X number of blades. In the most primitive 3-D designs, the blade shapes were sections of a cone, cylinder or torus. These could be readily formed via rolling or stamping. However, these shapes, while providing improved incidence, did not provide adequate control over the area distribution through the impeller passage. More complex shapes were required… the so-called arbitrary blade shapes. The name arbitrary is derived from the fact that there is no geometric shape that can replicate the blade. The shape is defined by line elements in space or a mesh of points. The need to define such shapes gave rise to a number of “geometry generator” software packages, the descendants of which are still in use today (more on this later).

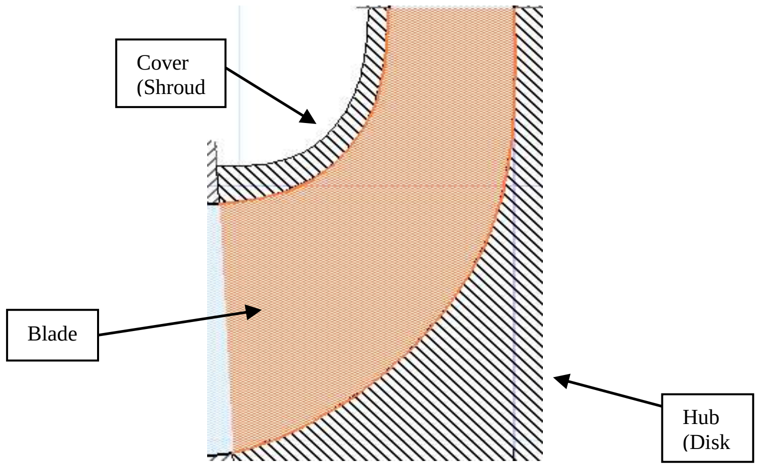


Figure 4 – Definition of Impeller Components

The blades (geometric or arbitrary) were formed via die-pressing or other methods of forming. The formed blades were often compared against a check block to insure the proper shape. The blades often deviated from the desired shape because of “spring-back” (caused by the elasticity of the material itself or due to shape change as the blade cooled). The blades were then welded or riveted to both the cover and disk to create the impeller (see Figure 4). Blade locating fixtures, ranging from the simple to the elaborate, were developed to minimize the placement variation blade-to-blade. These variations were typically small in the riveted impellers but could be significant in welded designs due to distortions caused by the welding and heat treatment (i.e., thermal distortion due to the high level of heat input to the part). Further, to permit surfaces to be inspected with die-penetrant or magnetic particle non-destructive test methods, the welded impellers were often grit-blasted to remove any weld residue (or “splatter”) or the oxide scale formed during the heat treat process. This grit-blasting caused a rough surface finish so impellers often had to be ground, polished, or honed to achieve the desired finish.

All of the concerns regarding blade accuracy and most of the surface finish issues were resolved with the introduction of 5-axis milling. Flank or point milling eliminated the need for die-pressing or forming as the blades could be milled from a disk forging. This reduced the amount of welding because one end of the blade was integral with the disk (or cover). Of course, the positioning of the milled blades was also far more precise than was achievable with the 3-piece fabrications. The more uniform blade-to-blade spacing also contributed to higher aerodynamic performance.

In recent years, OEMs have begun single-piece machining covered impellers from a single forging, thus totally eliminating the need to weld or join the cover (or disk) to the blading. The individual impeller passages are machined by plunge milling from the inside and outside diameter of the impeller. The inner and outer cuts are then tied together near the center of the individual flow passage. The “integral joint” in such a design is stronger than any welded or brazed joint, providing a more robust impeller in applications requiring additional joint strength. However, there are limits to the single-piece milling approach due in large part to limits in cutter technology. In the plunge milling process, it is impossible to reach some of the passage surfaces with conventional cutters. Therefore, special “lollipop” cutters were developed to reach these locations (see Figure 5). However, there is a limit in the overall length to diameter ratio of these cutters, thereby limiting the depth that can be cut in narrower flow passage. As such, single-piece milling is typically limited to larger diameter, higher flow coefficient impeller designs (i.e., $\phi > 0.040$, $D_2 > 15$ inch (381mm)).

One method to overcome the tooling and “reach” issues associated with single-piece milling is electro-discharge machining (or EDM). In this method, specially shaped electrodes are used to reach or “machine” areas that cannot be reached via conventional milling. The EDM method is often used in combination with milling to constructed high flow coefficient, single-piece impellers that could not have been machined 10 years ago.

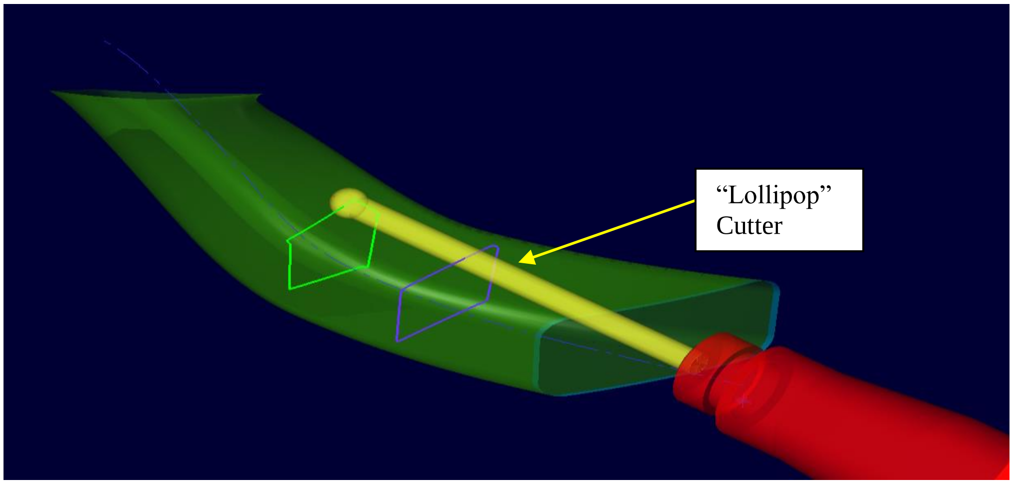


Figure 5 – Special "Lollipop" Cutters for Single-Piece Milling

The fabrication of low-flow coefficient stages provided another challenge to OEMs. Such stages are very important for high pressure applications such as re-injection, syn-gas applications, and the like. Given the very low flows required, it became necessary to develop methods to accurately fabricate impellers with very narrow flow passages. As noted previously, fillet welding required a minimum opening of 0.6 inch (15mm). Therefore, fillet welding was not an option for the small passages required.

Riveting was used for early applications but was unacceptable as the required operating speeds and associated stresses increased. The next alternatives developed were brazing and slot-welding. These are still in use today but both have a few disadvantages. The brazed joint lacks the strength of most welded joints, especially if the braze material is too thick. This can result if the gap between blades and adjacent wall is too large. A quality braze requires a very small gap between the blades and adjacent wall; i.e., on the order of 0.001 inch – 0.003 inch (0.025 – 0.075mm). Slot-welding provides a stronger joint than brazing. However, in slot-welding, the base metal of the disk (or cover) and the base metal of the blading are melted, leading to inaccuracies in the impeller passage height. As a result, the distortion in the flow passages caused by the welding can impact the capacity of the impeller, causing it to be smaller than expected. The blade material is not melted during the brazing process, leaving an inherent "mechanical lock" that keeps this "shrinkage" from happening but, again, the braze joint is weaker.

To address these shortcomings, OEMs turned to an electron beam welding process, a method originally developed by the aircraft industry. In electron beam welding, only the center portion of the base metal of the blading is melted. The outer portion that is not melted provides a "mechanical lock" similar to brazing, ensuring passage height accuracy. In the early 1990s, a variation to the electron beam process was developed, called EBraze™ welding (Miller, 1996). Under this new method, the cover (or disk) is joined to the blading with a combination of electron-beam welding and brazing. This greatly improves the fatigue strength in the joint by eliminating the stress riser associated with the un-fused portion of the traditional electron beam weld. Therefore, accurate, robust low flow coefficient impellers can be fabricated via the various types of electron beam welding.

Other alternatives for low flow coefficient designs include the afore-mentioned electro-discharge machining (EDM) as well as electro-chemical machining (ECM). It is also possible to use laser sintering with powdered metals.

More recently, significant advances have been made in constructing impellers via additive manufacturing or 3-D printing. Much has been written of late on how 3-D printed parts are being used in the turbomachinery industry. Many of the shortcomings of the process are rapidly being eliminated; i.e., voids in printed parts, inconsistent mechanical properties, and surface roughness; and many OEM are beginning to use parts built via additive manufacturing. One limitation facing the industry is the size of the part that can be built via these processes. However, larger "printers" are being developed to increase printable part sizes. Of course, a significant advantage of additive manufacturing is that virtually any shape can be formed, from the simplest to the most complex blade shape. Once sufficiently large "printers" are available, any impeller of any flow coefficient will be printable. Granted, issues related to printing "unsupported" surfaces must still be addressed but this is another issue that will be overcome as the technology advances. See Figure 5a which illustrates the concept of an unsupported surface.

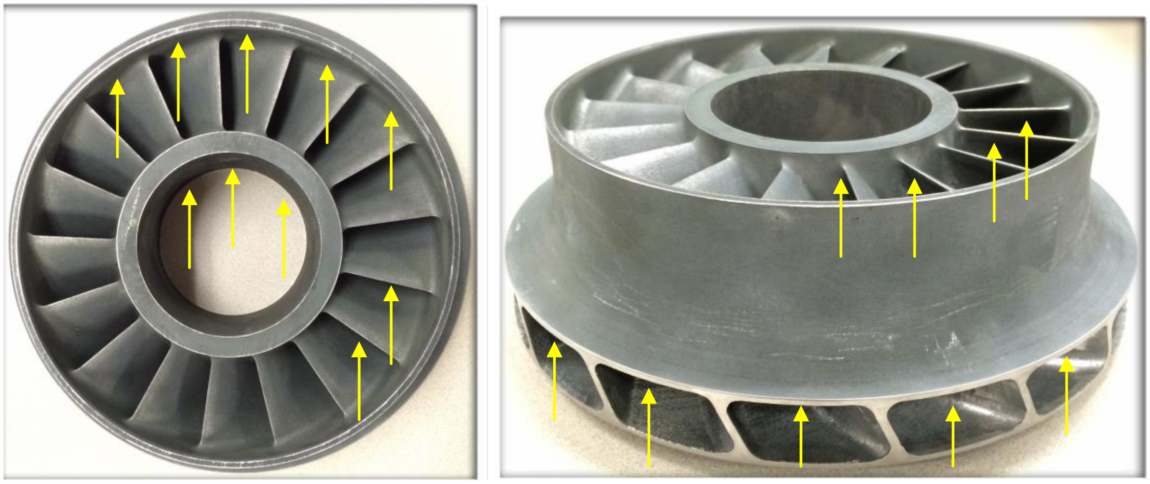


Figure 5a – Impeller built with additive manufacturing – select unsupported surfaced indicated by yellow arrows – the unsupported surfaces depend on the orientation of the part during printing

One final point about low-flow coefficient impellers: 2-D blade shapes are very appropriate given the narrow flow passages common in such designs because there is little or no variation in the flow angles from hub-to-shroud. Therefore, it is commonplace to find 2-D blades in low-flow coefficient impellers.

Diffusers

Diffusers are the second most important aerodynamic component in a centrifugal stage so it comes as no surprise that they have evolved substantially from the early days of centrifugal compressors. The diffuser converts a portion of the remaining kinetic energy in the gas stream (velocity pressure) into static pressure (potential energy), further reducing the volumetric flow.

Centrifugal compressor diffusers fall in two broad categories: vaneless and vaned. As indicated by their name, vaneless diffusers contain no vanes. Conversely, vaned diffusers contain one or more rows of vanes. In general, vaneless diffusers offer the widest flow range because there are no vanes to interfere with the gas as it passes through the diffuser. However, the static pressure recovery in vaneless diffusers is not as high as in their vaned counterparts. Therefore, the peak attainable efficiency for stages with vaneless diffusers is not as high (Figure 6).

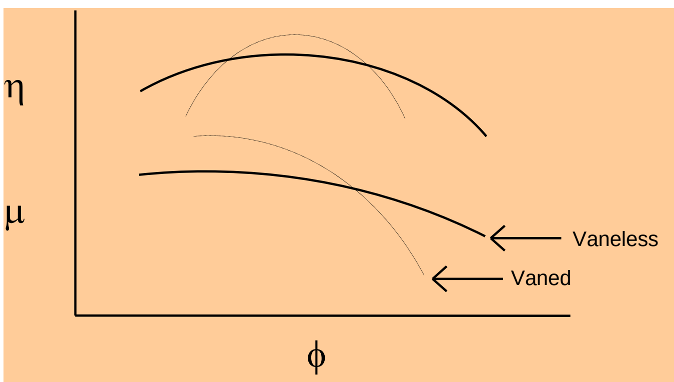


Figure 6 – Performance Characteristic – Channel v. Vaneless Diffuser

Vaneless diffusers were the dominant style in early centrifugal compressors because of their simplistic design. The parallel or tapered walls were easy to machine via turning. Therefore, it was possible to achieve some very good surface finishes, a necessity for high performance in vaneless diffuser. However, the limited peak static pressure recovery (typically less than 50 percent) restricted the peak efficiency achievable with vaneless diffusers.

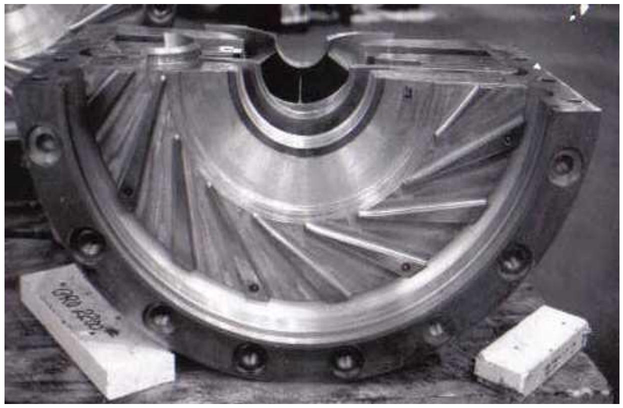


Figure 7 – Channel Diffuser

Some process compressor OEMs attempted to apply channel diffusers. The name comes from the fact that two adjacent diffuser vanes form a passage or channel (see Figure 7). These diffusers do provide superior static pressure recovery, C_P, with peak C_P levels reaching 75 percent - 80 percent. However, channel diffusers also cause a substantial reduction in flow range, making them undesirable for compressors that must operate over a range of flow conditions. As a result, channel diffusers are rare in process compressors but somewhat popular in air machines, gas turbine gas generator compressors, or turbochargers, which do not require wide flow range.

Figure 8 – Low Solidity Vaned Diffuser (LSD)

Vaneless diffusers were the most widely-used style in industrial centrifugals until the late 1980s when some OEMs began applying a style known as the low solidity vaned diffuser. Unlike the channel diffuser, the vanes in a low solidity vaned diffuser (LSD) do not form a channel and also have no true geometric throat (Figure 8). Numerous publications in the late 1980’s and early 1990s touted the advantages of the LSD style including Senoo et al (1983), Osborne and Sorokes (1988), Sorokes and Welch (1992), and Amineni and Engeda et al (1995, 1996). The most important benefit is that LSDs provide nearly the same operating range as a vaneless diffuser yet provide greater pressure recovery and, therefore, higher stage efficiency. The introduction of the LSD provided a step change in stage efficiency without significantly reducing the flow range. However, the efficiency enhancement seemed to be limited to medium- to low-flow coefficient stages; i.e., $\phi < 0.080$ and smaller; with the greatest benefit being in flow coefficients, ϕ, less or equal 0.030.

More recently, select OEMs have re-introduced the rib diffuser, a special class of LSD with vanes that do not cross the entire flow passage (Figure 9). These diffusers were first suggested in the mid-to-late 1970s but did not gain broad acceptance until much later. Again, this style of diffuser provides an efficiency boost in some flow coefficient ranges but is ineffective in others. Those seeking more details on this style are directed to the work of Sorokes and Kopko (2001).

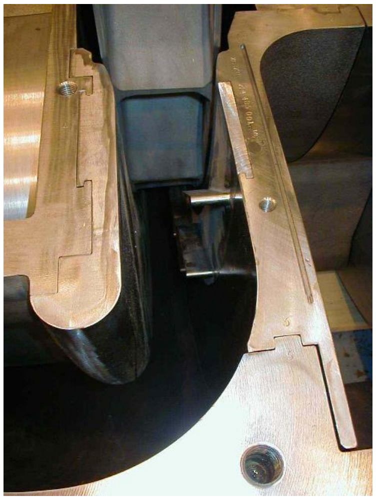


Figure 9 – Rib Diffuser

One other class of vaned diffusers deserves mentioning… the so-called tandem or multi-row diffuser. Rather than a single row of vanes, these diffusers have multiple rows (two or more) of vanes with each row contributing to the static pressure recovery process. It is commonly known that if one attempts to achieve too much turning in diffuser vane, the flow will separate from the vane surface, leading to excess losses or possible premature stall. Therefore, if high levels of turning are desired, the turning is better achieved with two or three rows of vanes rather than one. The multiple rows of vanes also help to control boundary layer growth and reduce losses in the diffuser passage. In recent years, tandem rib-LSD or LSD-LSD diffusers have proven to provide performance benefits when compared to single row cascades (see Figure 9a).

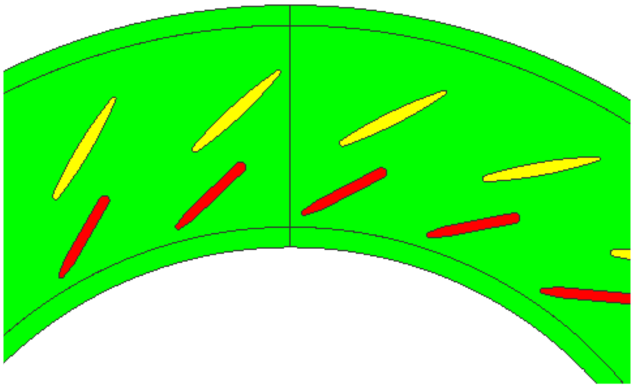


Figure 9a Tandem (2-row) Diffuser

Other Components: Inlet Guides, Return Channels, Volutes, Inlets, Sidestreams & Casings
The stationary components in most early centrifugals were constructed via casting. Typical issues associated with the casting techniques included core shifts, varying vane thicknesses, and rough surface finishes. All contributed to increased losses and greater uncertainty in performance predictions. Simply put, if the geometric dimensions of the parts do not match the dimensions used to predict the compressor performance, one cannot expect to have great accuracy in the performance predictions.

The desire to minimize the number of casing patterns also led to some interesting issues when the return bends or cross-overs were cast into the case. This meant that the diffusers and return channel passages had to align with the fixed return bend location and size. As the return channels were also cast with integral bulbs, the bulb size (or inner portion of the return bend) could not be custom-sized for each application, leading to arrangements such as that shown in Figure 10. These step changes in the flow path were not conducive to high efficiency and also were a prime location for rotating stall cells to form (Marshall, Sorokes 2000).

Figure 10 – Diffuser Discharging into Cast Return Bend

To address this and other similar issues, OEMs began machining the stationary flow path components. Again, early machine tool technology limited this to the return channels because the vanes could easily be machined on a 3-axis mill. However, as larger 5-axis mills became available, it was possible to machine complex components such as volutes and inlets. It was also possible to mill prewhirl inlet guide vanes via 5-axis machining. By the year 2000, OEMs were building compressors with nearly 100 percent fabricated / machined internal components as compared to the 1950s when nearly 100 percent of the components were cast.

When fabricating components, attention must be given to the methods to be used to assemble the various parts. Advances in joining technologies contributed to improved quality, reliability and safety. In addition to better welding techniques, innovations in bolting technology also play an important role in modern machines. The advent of hydraulic tensioning, super-nuts, and the like facilitated assembly and disassembly of parts and increased product flexibility.

It must be noted that castings remain a viable option for component fabrication especially for very large parts where material removal would be costly or when many copies of the same or very nearly the same component are required. Steps must be taken to ensure the accuracy of the flow path geometry and a high quality surface finish. It is also advantageous to use steel castings rather than cast iron as it is much easier to repair defects in steel via welding or other processes.

Analytical Techniques

The evolution in analytical methods has also been a major contributor in the advancement of centrifugal compressor aerodynamic technology. Much of this has been the direct result of the advances in computer technology. As computer technology improved, it was possible to perform more complex mathematic computations in less time. The result was more sophisticated modeling of individual components or entire compressors.

1-D methods

The most common approach used in 1-D models is the so-called “velocity triangle” methodology. Formulations based on the Euler turbomachinery equation, the Bernoulli equation, conservation of mass, conservation of angular momentum, and other empirical performance models are used to solve for the meridional, tangential, and relative velocities and flow angles at various key locations within a centrifugal stage. Such codes focus primarily on the inlet and exit region of each component and “know” little about the geometry between those locations. As such, they have limited value today other than doing some basic sizing calculations. However, this was the only approach available to turbomachinery designers in the early days.

Prior to the late 1950s, all designs were completed using this approach… and development testing. The design system typically included a slide rule, a pencil, a ruler, a compass, a protractor, a drafting board, lots of paper, and human creativity / intellect. Despite the lack of computers and analysis procedures, some very sophisticated designs were created in the 1930 and 1940s, many as the result of military aircraft engine developments that later found their way into industry. Others were the result of the time-consuming “cut and try” testing where researchers tried a variety of design geometries until a configuration was discovered that met the performance objectives. Clearly, this was not a cost effective way of developing compressor products but it was the only technique available at the time.

2-D methods
Introduced commercially in the late 1950s, two-dimensional or 2-D methods provided another level of insight for designers in the development and analysis of aerodynamic components. Unlike 1-D codes, 2-D codes require definition of the entire flow path including hub and shroud profiles as well as blade or vane angle definition and thickness.

The most common 2-D method is the streamline curvature approach. For those unfamiliar with such, streamline curvature codes divide the flow passage into "streamtubes" of constant mass flow, such as show in Figure 11. Velocities are calculated based on the local curvatures in the meridional (or hub-shroud) profiles and the mass flow through the streamtube area. Some of these codes are also sensitive to curvature in the blade-to-blade direction.

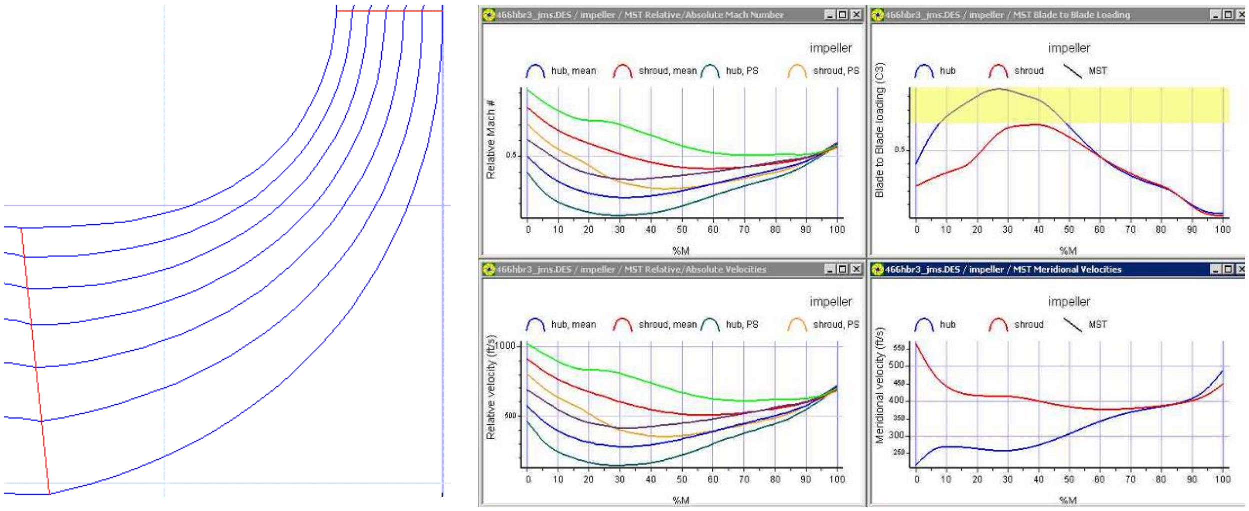


Figure 11 – 2-D Streamtube Analysis

There are also modeling parameters that can be adjusted when running the various 2-D streamline curvature codes. Some of these include: the number of calculation stations (quasi-orthogonals) from inlet to exit; the number of streamtubes dividing the flow passage; loss distributions; curve fits on the geometry; and convergence tolerances.
3-D methods
Three-dimensional Computational Fluid Dynamics (CFD) codes are the most rigorous analytical techniques used to calculate the flow through aerodynamic components. First widely available to the industrial compressor industry in the late 1980s, such codes provided a major step forward in the ability to understand the flow physics inside the rotating impeller and stationary components as well as the interactions between such.

CFD analyses or simulations are performed using computational grids that break the flow passage up into small polyhedrons (e.g. hexahedrons or tetrahedrons), essentially the aerodynamic equivalent of finite element analysis. Consequently, such codes can account for all facets of the aerodynamic component geometry and provide a far more comprehensive approximation of the flow physics than any of the less sophisticated flow codes. As a result, using such codes can lead to superior aerodynamic designs and, therefore, superior performance because untoward flowfield characteristic can be identified and eliminated or minimized.

Highly computationally intensive, use of CFD analyses has become more commonplace as high-end computer workstations and workstation clusters have become more readily available. Putting this into perspective, the analysis of a single impeller passage operating at a single flow condition and single speed required 18 hours on the most advanced workstation in 1990. The same analysis can be run in approximately five minutes today on the average laptop computer.

Because of the computation times involved, initial CFD applications tended to focus on isolated components; in particular, the impeller. However, by the mid to late 1990s, further advances in analytical codes and computers made it possible to include multiple components in the computational domain (see examples in Figure 11a). Even further advances have allowed analysts to conduct unsteady analyses, allowing an assessment of how transient or time-dependent flow, pressure, or temperature fluctuations impact component or stage performance. Such analyses represent the closest method yet to approximating the real world flow physics inside a compressor.

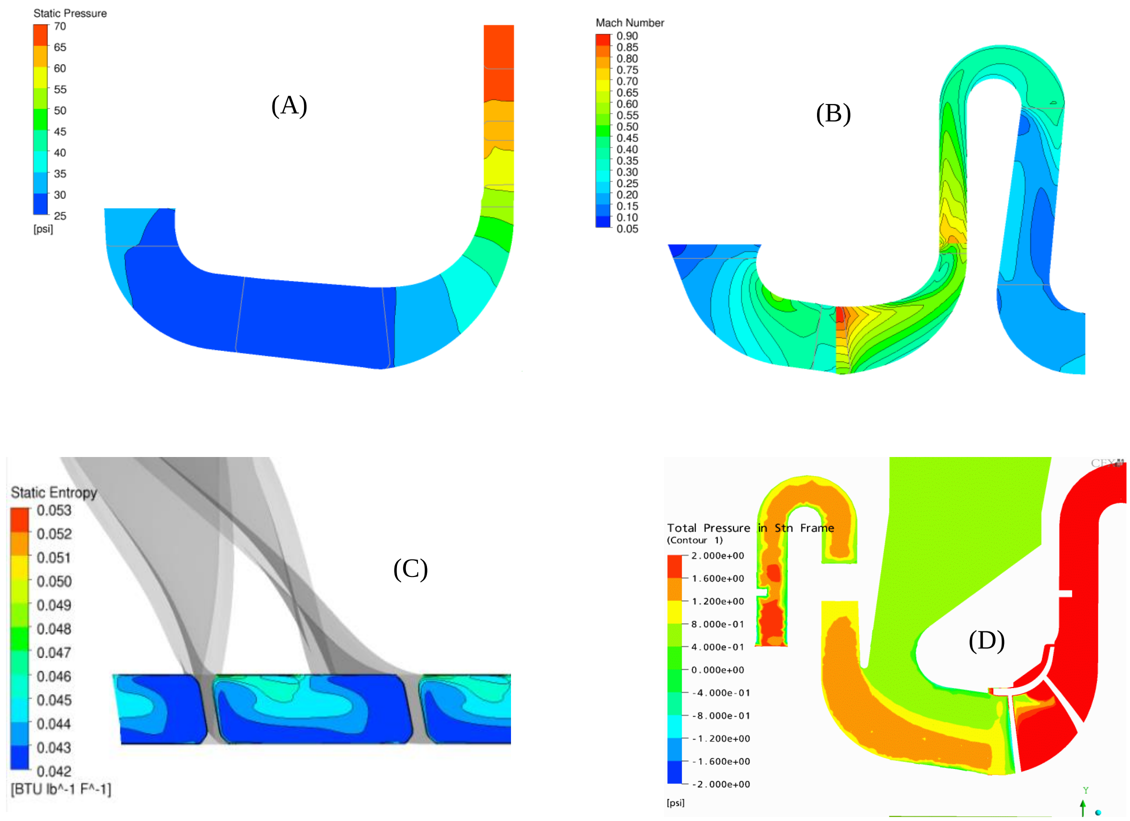


Figure 11a. Examples of contour plots: (A) – Static pressure distribution through inlet guide and impeller; (B) Mach number distribution through a centrifugal stage; (C) Entropy distribution at impeller exit plane; (D) Pressure distribution in mixing section of sidestream.

In recent years, the most sophisticated simulations have begun to address and blend the aerodynamic and mechanical disciplines. Code developers are striving to simulate Fluid Structure Interaction (FSI); i.e., determining how the fluid forces, thermal gradients or the like impact the mechanical structure and/or determining how material deflection, vibrations or the like influence the behavior of the flow field in a component. By linking the aerodynamic and mechanical simulations, designers can gain critical insight into the multi-disciplinary nature of their design.

One further advance in the field of computational methods must be mentioned; that being design optimization. There are many well-documented approaches that can be applied but the basic concept behind all of them is to allow the computer to create and assess various component geometries (in many cases, completely randomly) until it finds a configuration that meets the user's design objectives. In most cases, the user will apply limits on the geometry generator to keep it from developing untenable designs or to force the solution to fit in a given space or meet other operating constraints; i.e., operating speed, impeller diameter, bore diameter, etc. This helps reduce the computation time when compared to allowing the geometry generator to develop designs complete at random. Regardless, high-end optimization studies require extensive amounts of time on large computational clusters or super-computers. As such, these studies are not yet part of the day-to-day design process. However, with further evolution of computers and parallel processing, the day will come when design optimization will be as commonplace as use of the slide rule was 60 years ago.

In summary, the advances in computer technology have also contributed to the ability to design more efficient centrifugal compressor components.

ROTORDYNAMICS

Undamped Critical Speed Analysis

In the mid-1940's Myklestad developed a new method of calculating modes of uncoupled bending vibration of an airplane wing and other types of beams. (Myklestad, 1944). One year later Prohl developed a general method for calculating critical speeds of flexible rotors (Prohl, 1945). These two references form the basis of the Myklested-Phohl method, which is a transfer matrix solution technique at the heart of the undamped critical speed map commonly used today. Just as an aerodynamicist would use a 1D analysis as a starting point for a new impeller design, a rotordynamic analyst would use, and still does today, an undamped critical speed map to analytically determine the location of the rotor's natural frequencies as a function of the bearing support system. The undamped critical speed program generates undamped circular synchronous critical speeds from rotor geometry input as a function of symmetrical bearing stiffness coefficients (Figure 12). In the late 1940's though the 1960's the calculation of the first critical speed (NC_1) was conducted by hand with the objective of ensuring the intended running speed range of the compressor did not coincide with NC_1. As computer technology advanced and journal bearing coefficients became common place, the bearing coefficients would be plotted on the undamped critical speed map to "zero-in" on the location of the critical speeds.

Often, on the OEM's test stand or in the field, a coin (in the United States, often a U.S. nickel) would be placed on its edge on a flat spot on the bearing cap (housing). If the nickel did not vibrate and fall, the machine would be classified as 'running smooth'. This vibration monitoring technique has now since been replaced with eddy current proximity probes and advanced data acquisition systems. Nevertheless a nickel is still used, in jest, as a sign of a smooth running machine by designers.

Synchronous Unbalance Response

In May of 1965, J. W. Lund released Part V of a report prepared for the US Air Force Aero Propulsions Laboratory (Lund, 1965). This landmark publication contained a computer program to determine the unbalance response analysis of a rotor on fluid film bearings. In addition, the report contained the analytical foundation on which the programs are based. Lund describes the basic theory of the rotor response program in his 1967 paper. These tools, along with bearing and seal programs which are used to determine oil film stiffness and damping (more on this later), enabled more advanced analysis on rotors. The basic kernel of these codes is still used today but is now being superseded by finite element analysis approaches.

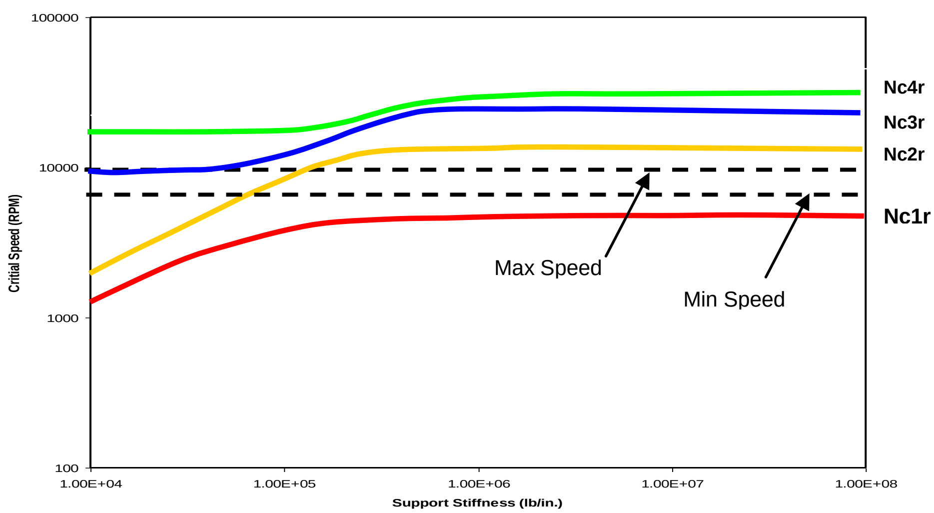


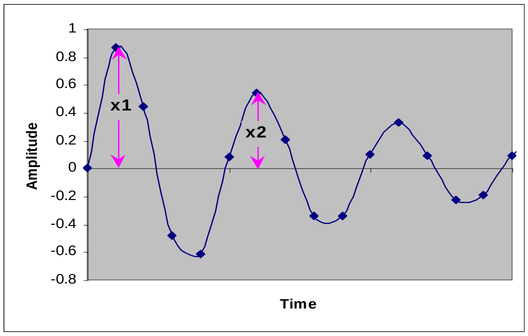
Figure 12 – Critical Speed Map

The forced response tool allowed the designer to apply unbalance to the rotor to calculate the rotor's response to unbalance, the location of NC_1 and the amplification factor (API 617, 2002) and (Childs, 1993). In addition the sensitivity to unbalance could now be calculated and compared to the OEM's test and field experiences.

Rotor Stabililty Analysis

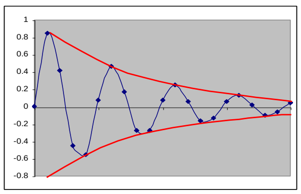
In the early 1970s a series of stability problems were seen in high-pressure gas injection and synthesis gas compressors. See Fowlie (1975) and Smith (1975) for discussions of the Kaybob compressor, see Wachel (1975) for a discussion of the Kaybob and Ekofisk compressors, and Booth (1975) for the Ekofisk compressor. In 1974 Lund published a break-through paper on the analysis of rotor stability (Lund, 1974). A program was written based on the theory in this paper (Smalley, 1974). The lateral stability program of Lund was used in the analysis of instability problems at the first natural frequency and in the design of centrifugal compressors to resist subsynchronous vibration. It was common to refer to this as doing a Lund analysis.

The rotordynamics stability code provided the added insight of logarithmic decrement or log dec. The log dec of a system can be characterized in the time domain as the amplitude of successive aptitude peaks. If the amplitudes are decaying over time the log dec is positive, if growing the log dec is negative (Wachel, 1975), (Kuzdzal, 1994), (Ramesh, 2004) (Figure 13). Certainly, a negative log dec is undesirable. The rotor stability program can also be used to analyze oil-whir problems in hydrodynamic oil film bearings (DeSantiago, 2007).

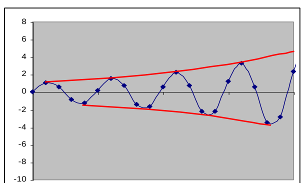


LogDec = ln(x1/x2)

Stable : if Logdec > 0
Unstable: if Logdec <0



Stable system



Unstable system

Figure 13 – Definition of Log Decrement

Hydrodynamic Oil Film Bearing

In the 1970’s and 1980’s the sources of instability were sometimes difficult to quantify. Certainly, it was understood that if the rotor showed an instability at its first forward whirling mode while operating at full speed it was likely a pure rotor instability. While if the subsynchronous vibration tracked speed or was not at the first critical it may be more related to a forced excitation, an aerodynamics stall or bearing whirl/whip. (Marshall et al, 2000). Hydrodynamic oil film bearings have been used in centrifugal compressors for many decades, but before the introduction of proximity probes, data acquisition systems and rotor stability methods a designer could only characterize a machine as running ‘smooth’ or ‘rough’. If a machine was running ‘rough’ one really has no practical means to know why. Could it be the first critical speed, bad bearings, or an excessive unbalance?

As hydrodynamic bearing technology advanced, the designer recognized the need to optimize the plain sleeve bearing. In general, a plain sleeve bearing has a comparatively large load carrying capacity, but performed poorly in rotor dynamics stability, often being the source of the issue. Oil whirl (Pinkus, 1956), (Newkirk, 1956) was a common term used in the 1960’s and 1970’s to characterize the maximum speed the bearing could operate at before it generated undesirable instability forces. Resonant whip was used to describe when the unstable frequency locked on to the first natural frequency. Designers in that era worked to modify the inner geometry (profile) of the journal bearing to control the oil whirl and whip characteristics with the intent of increasing the bearing instability threshold speed (Figure 14). Various geometric details such as three and four axial groove bearings, elliptical or lemon bore bearings, offset half bearings and pressure dam bearings were all designed and employed to improve the rotor stability threshold speed. To improve this parameter, generally load carrying capacity or synchronous unbalance response ability was compromised (Figure 15). The advent of four lobe bearings further improved the situation. In this design the curvature of the fixed profile (non movable pads) were cut at a larger diameter to form what is today know as bearing preload. (Nicholas, 1980, 1981, 1985).

Although tilt pad thrust bearings were invented by Mitchell in 1905 (Australian – English patent) and Albert Kingsbury (US) in 1907, tilt pad journal bearings did not start to become popular until the late 1960’s. This OEM’s first known use of a tilt pad journal bearing was a 2.5-inch (63.5-mm) diameter Waukesha bearing used in 1964. Tilt pad bearings offered a distinct advantage over a fixed profile bearing, in that the bearing had movable pads that significantly reduced the oil film cross-coupling stiffness, thereby increasing the rotor stability.

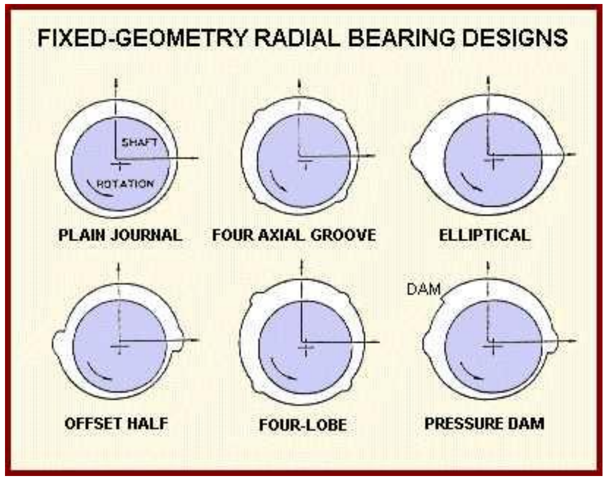


Figure 14 – Fixed Geometry Radial Bearing Design

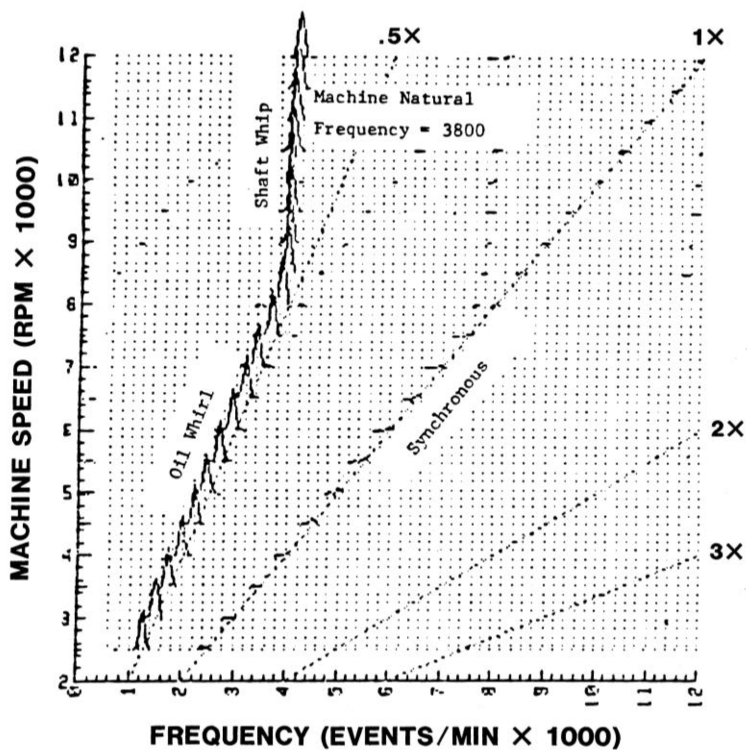


Figure 15 – Effect of Oil Whip

The first usable program to analyze tilt pad bearings was based on the ground-breaking paper by (Lund 1964). The program used the pad assembly method to assemble the synchronous stiffness and damping coefficients from those of the individual pads. Now it is more common to use the program developed by Nicholas (Nicholas et al, 1979) to analyze tilt pad bearings.To this day, tilt pad bearings are the workhorse hydrodynamic bearing for centrifugal compressors used in the oil and gas industry. Today's experience has shown successful operating of tilt pad journal bearings with light load at speeds as a high as 570 ft/sec (174 M/sec) and with unit loads as high as 775 PSI (5434 KPa) at moderate surface speeds. Nevertheless, the world continually strives for new solutions and the bearing arena is no exception.

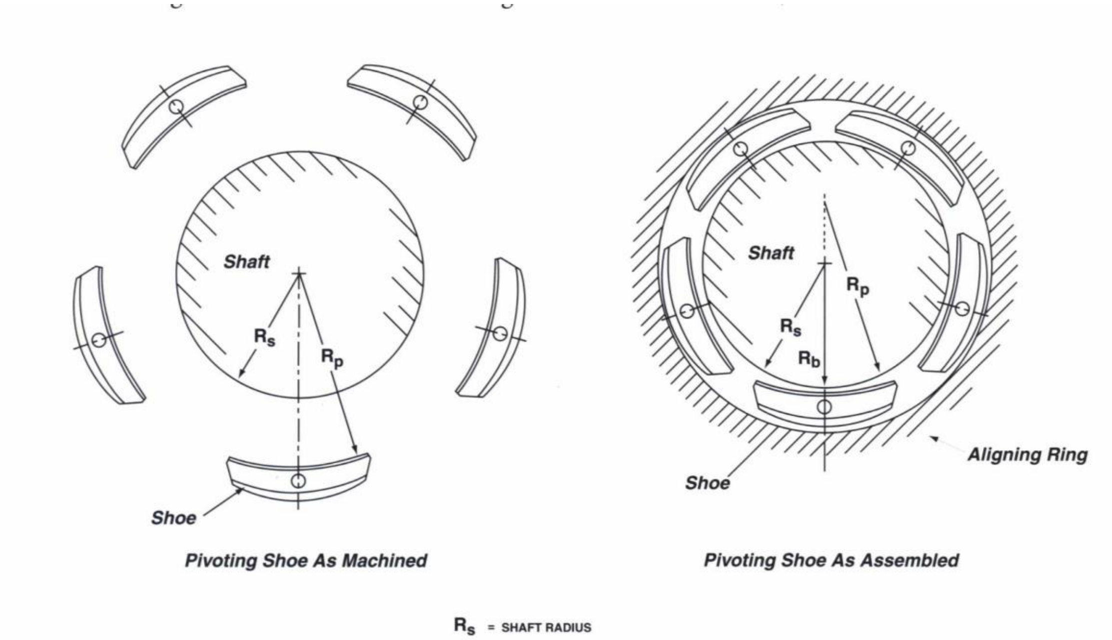


Figure 16 – Tilt Pad Bearing

Morton effect

For a modest to highly loaded bearing, the minimum oil film thickness is in the bottom half of the bearing, generally near the 6 o'clock position and the bottom-half of the bearing will be hotter than the top-half of the bearing. The shaft rotates through the hot spot and becomes circumferentially heated. The shaft heating is evenly distributed circumferentially.

If the bearing is lightly loaded and the speeds are high, the shaft will want to run at a low eccentricity ratio or close to the bearing center. The Morton effect is a thermal instability where the circumferential temperature differential in the shaft causes a bent-shaft excitation. Childs (2013) describes that the temperature differential is created by the rotor's synchronous orbit within a hydrodynamic bearing. In a lightly loaded bearing with high synchronous orbits, the minimum oil film thickness will rotate with the shaft. For instance, this will occur when the bearing vibration (peak-peak) orbit is 1.5 mils, and the eccentricity is less than 0.75 mils. Hence the shaft segment on the outside of the orbit always faces the minimum oil film clearance. Fluid shearing and heat production are at a maximum at the minimum clearance and a minimum at the maximum clearance. This creates a temperature gradient circumferentially in the shaft. The temperature gradient bends the shaft, creating more unbalance and yet higher synchronous vibrations, compounding the matter.

Most OEM's will combat this effect with: ensuring the rotor natural frequencies are not in close proximity to the design speed, hence keeping sensitivity to unbalance and synchronous vibration low, creating asymmetry in the bearing forcing the shaft to a higher eccentricity or by utilizing a thermal barrier coating on the shaft.

Magnetic bearings

Magnetic bearings were introduced into the oil and gas market segment in the late 1970's and early 1980's (Hustak, 1986, 1987) when the gas pipeline industry was moving to an oil-free turbomachinery solution. Today most compressor OEM's offer a line of magnetically supported, hermetically sealed centrifugal compressors for operation on un-manned platforms and on the seabed. In the late 1990's there was a resurgence as magnetic bearings were seen as an enabler for hermetically sealed oil-free compressors. Finally, researchers developed a method to utilized magnetic bearings to purposely add know quantities of destabilizing forces to a rotor to measure log dec (more on this later).

Main compressor seals

Main compressor seals have also made significant advances other the years. In the 1910's centrifugal compressors were generally compressing air, which was used to fire blast furnaces in the steel industry. The main seals of these air compressors were aluminum labyrinth (laby) type seals that generally leaked to the atmosphere. But fuel was cheap and the leakage of compressed air was of little concern. In the later decades as centrifugal compressors were being used to compress methane (CH_4) and other combustible gases, leakage to the atmosphere could not be tolerated. (Kirk 1986)

Oil seal bushings were utilized for main compressor seals for higher-pressure natural gas applications. Records show oil film seals being used in the 1950's (Figure 17). These seals used oil supplied at a pressure higher than the compressor suction pressure to ensure the volatile gas did not leak to the atmosphere. The introduction of oil film seals brought at least two issues along with it. First, seal oil that came in contact with the process gas could become contaminated (soured) if the gas being compressed was sour; sour gas being any gas mixture containing acidic elements such as hydrogen sulfide (H_2S) or the like. This generated the need to properly dispose of the contaminated oil. Second, as the pressure increased, the need to understand the stiffness and damping coefficients of the oil film seals also increased. Not until the late 1970's and early 1980's is there evidence in the open literature of OEMs' working to understand these forces and how they impact machinery vibration characteristics (stability) of the rotor during high-pressure operation (Kirk, 1977). The author's company developed and started supplying a tilting pad oil-film seal in the early 1970s. The tilting pad seal had a significant positive effect on the stability of the compressor (Memmott, 1990, 1992). For a recent paper where a tilting pad seal was used to eliminate an instability seen with an oil-film ring seal see Memmott (2004a).

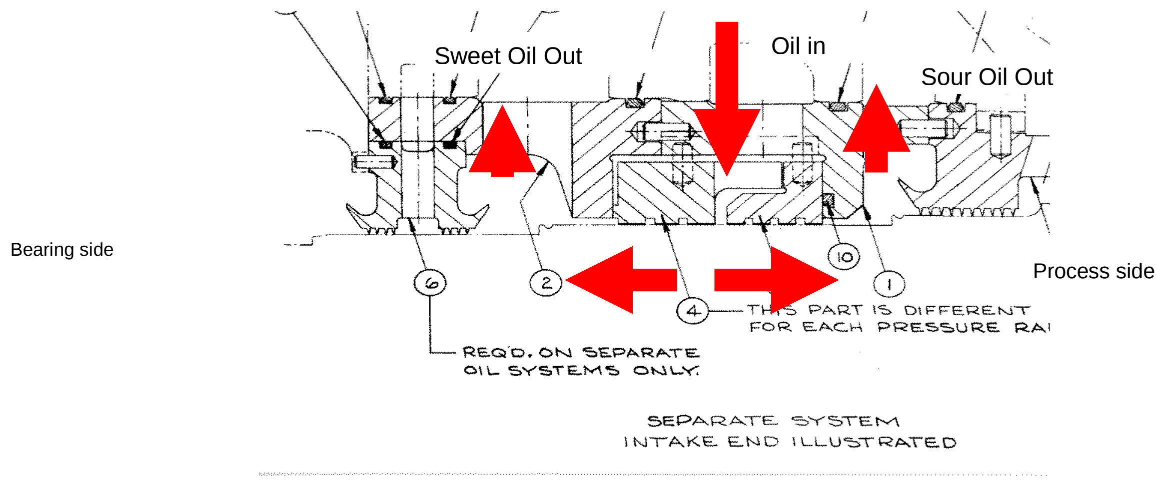


Figure 17 – Oil Film Seal

Today the use of oil film seals is generally reserved for revamp and repair activities. Nearly 100 percent of all new compressors sold in the oil and gas industry feature dry gas seals (DGS). The first known application of a DGS in the author's company was in a 30 PSIG (210KPa) single-stage overhung compressor in 1962. Since that time, the DGS industry has worked hard to gain market acceptance. Still today, the industry standard is a tandem DGS where the full seal delta pressure is let down across one seal face while the secondary seal face does little or no appreciable work (Stahley, 2005). Today as the technology and materials research advances DGS sealing pressure capability continues to rise to meet today's demanding high-pressure applications. Current seal delta pressure limits are near 6500 PSI (450 bar). From a rotor dynamics standpoint one negative attribute that a DGS introduces is an added rotating mass on the rotor. Interestingly enough, the sealing face is orthogonal to the shaft. As a result, the seal stiffness and damping parameters in the radial direction are ignored in a rotor dynamics analysis.

Internal Seals

As previously discussed, as speed increases the tangential velocity of the gas (or oil) surrounding the rotor also increases. These tangential forces, which are characterized as cross-coupled stiffness coefficients, can become large enough to drive the first forward natural frequency of the rotor unstable. When the destabilizing tangential force exceeds the stabilizing external damping force, any disturbance will cause the shaft to non-synchronously whirl. To that end, in high gas density machinery, much care is taken to control gas (or oil) tangential forces in secondary flow passages. See Ehrich, 1993.

As speed, power and gas density increase-managing forces inside a centrifugal compressor can mean the difference between a rotor that becomes unstable with increased gas density and a rotor that becomes more stable.

Although tooth labyrinth seals have generally been used to seal impeller stage rise from 'bleeding' back to low pressure areas in the machine for decades, a recent advancement to the eye seal and balance piston/division wall seal has been the swirl brake (Figure 18). When properly designed, these stationary vane-like devices can substantially reduce cross-coupling stiffness generated inside a tooth laby by controlling the gas tangential velocity (Moore, 2000). These swirl breaks can be used on tooth labyrinths as well as damper seals.

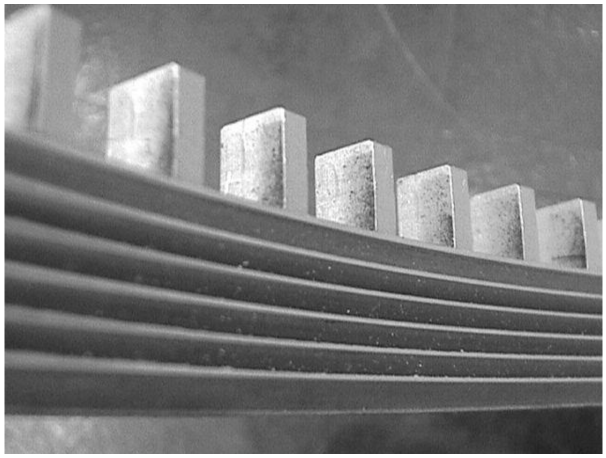


Figure 18 – Swirl Brakes

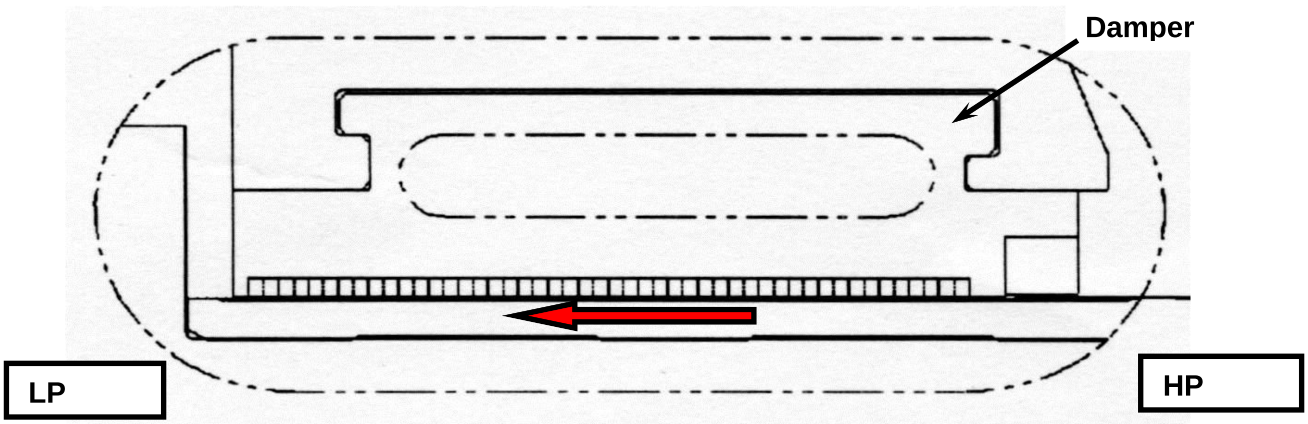


Figure 19 – Hole Pattern Seal

Damper seals generally take two possible forms: a honeycomb or hole pattern seal (Figure 19). In either form the intention of the seal is to introduce more desirable direct damping than undesirable cross-coupling stiffness, thereby improving the stability of the rotordynamic system. Researchers have completed much laboratory work (Kleyhans, 1996) (Childs, 2003) to ascertain accurate leakage and stiffness and damping parameters.

Rotor stability measurement while testing
As rotordynamic engineers advanced their understanding of aerodynamic and rotordynamic destabilizing forces, attention turned to measuring the log decrement on the OEM's test stand, generally during API 617 – type 1 tests. The full-load-full-pressure (FLFP) test offered an opportunity to assess rotordynamic stability. The outcome of the test was generally binary, either rotordynamically stable or unstable; characterized with a sub-synchronous vibration. But 'what if' the rotor could be excited with a non-synchronous asymmetric forcing function of increasing amplitude during the test, could one ascertain the amount of destabilizing force required to drive the rotor system unstable or determine the log decrement of the rotor?

As magnetic bearing technology transitioned from analog to digital controllers and mechanical engineers gained a stronger understanding of said technology, the magnetic bearing took on a second use: a test tool to purposely add a non-synchronous asymmetric forcing function of known/increasing amplitude during the test. As with any good test, it is generally designed such that the results are not skewed with the introduction of the measurement experiment. Most OEM's would mount the magnetic bearing on the rotor non-drive end without disturbing the oil film bearings, tooth labyrinths and damper seals. Care is taken to ensure the magnetic bearing rotor is small and light enough not to affect the rotor system, but large enough to impart a meaningful destabilizing force, see Moore (2002). As time has ticked by, designers has found a means to control destabilizing forces and introduce enough system damping where rotor bearing systems become more stable as speed, power and gas density increase.

In summary, OEMs have utilized full-load full-pressure testing along with magnetic bearings to inject know amounts of non-synchronous forces to demonstrate that with proper maintenance of forces a rotor can become more stable as speed, power and gas density increase (Moore, 2000). The industry certainly has come a long way in understanding the forces exerted on a rotor since the famous Ekofisk field rotor instability in the North Sea in 1974.

Operational modal analysis (OMA) was later used to assess rotordynamic stability without utilization of a magnetic bearing to provide the excitation forces. OMA can identify the modal parameters of a system over its entire operational range from measurement of the response due to (unknown) excitation. In the past, OMA has proven successful on non-rotating structures, but prior to 2013, has seldom been applied to rotating machinery. See Carden (2014) who presented three case studies demonstrating the use of OMA in identifying stability of lateral rotors modes based on measurements from existing radial proximity probes during normal production undertaken as part of commissioning campaigns.

Impeller analysis
More recent advances in structural dynamics have also occurred, one such area that has greatly improved machine reliability is impeller dynamics. In the early years a designer may run a simple hand calculation to ensure that impeller stresses were such that the impeller did not yield or slip on the shaft. This was sufficient when the impeller tips speeds may have been 50 percent or today's current state-of-the-art. As Finite Element Analysis (FEA) and computer technology advanced, more complicated FEA models were able to better assess the stress exerted on an impeller due to rotation. Nevertheless, most compressor users endured the occasional impeller incident. In many of the instances the failure may have been a result of a high cycle fatigue crack that resulted from an impeller running in a resonance condition. Today, impellers are analyzed using modal and forced response analysis in much the same way as rotors have been for four decades.

Understanding impellers natural frequencies and the aerodynamic forces exerted on these impellers from IGVs and LSDs have greatly reduced the number of impeller-related incidents in recent years (Schiffer, 2006).

LOOKING FORWARD
As noted in the Introduction, centrifugal compressor technology is rapidly approaching a limit in peak aerodynamic efficiency. However, there are further improvements that can be made to achieve higher efficiency and to broaden the flow range over which that peak efficiency occurs. So, in looking forward at what is to come, one can expect to see more sophisticated impeller blade shapes; more exotic diffusers; movable geometry in guide vanes, diffusers, and return channels; improved seal technologies; and further enhancements in other stationary components. All of these will depend heavily on further advancement in the aerodynamic and mechanical analysis tools providing more accurate simulations of the real world in the computational domain.

More sophistical impeller blade shapes will include blades that are more customized to address shortcomings discovered using the analytical techniques discussed in the following section. Further, as end users push to handle large flows in small equipment, OEMs will need to develop mixed flow style impellers. Such wheels have been used in single-stage pipeline boosters for many years but are not routine in multi-stage equipment. Yet this style of impeller represents the most suitable option for flow coefficients above 0.17 – 0.18 in centrifugal compressors.

More exotic diffusers will include more sophisticated vane shapes as well as tandem diffusers; i.e., multiple rows of LSDs or combinations of rib diffusers and LSDs. These will increase the static pressure recovery in the diffuser, leading to an increase in overall stage and compressor efficiency. However, these efficiency increases will have to be weighed against the probable decreases in operating range that will result.

Improved seal technology will be very important to low-flow coefficient stages. The impeller eye seal leakage represents a significant loss in such stages so anything that can be done to reduce that leakage will improve the efficiency of the stage. Improved technologies such as brush seals, abradable materials, spring-loaded seals and "directed leakage" will play a greater role in low flow stages.

Regarding external seals, the next logical step in compressor evolution will be the seal-less or hermetically-sealed compressor in which the there are no rotating shaft seals exposed to the atmosphere. Such machines are, in fact, very similar to the compressors found in common home refrigerators / freezers. Nearly all major compressor OEMs have introduced this new type of product configuration to the marketplace. With increased concerns over emissions, this new style of compressor will become more widely accepted in the oil and gas industry.

Stationary components such as inlet guide vanes and return channels will benefit from more advanced vane shapes (i.e., 3-D vaned diffusers or return channel vanes) or multi-blade row configurations designed via the advanced analytical and design tools available today.

Finally, the use of movable geometry in guide vanes, diffusers, and other stationary components will provide OEMs and end users with the opportunity to tune centrifugal stages to the application requirements. Consequently, while it might not be possible to increase the overall peak efficiency of the compressor, it will be feasible to increase the efficiency at off-design operating conditions. This can provide significant savings for the end user who wishes to operate their equipment over a very wide flow range. Movable geometry has been used for years in integrally-geared centrifugal compressors and axial compressors because of the ease of access to the inlet guide vanes and diffusers (Dresser-Roots Co., 2006). For similar reasons, movable geometry in the first stage inlet guide vanes and vaned diffuser of multistage beam-style centrifugal compressors dates back to the 1950s (Ferrara et al, 2005), (Sorokes and Welch, 1992). However, only recently has movable geometry been considered for all stages of a beam-style machine, as seen in Figure 20 (Sorokes et al, 2009). The complexities of implementing the movable geometry system in the limited stage spacing of a multi-stage machine make it impractical, yet the performance benefits of doing so might make it too attractive to ignore.

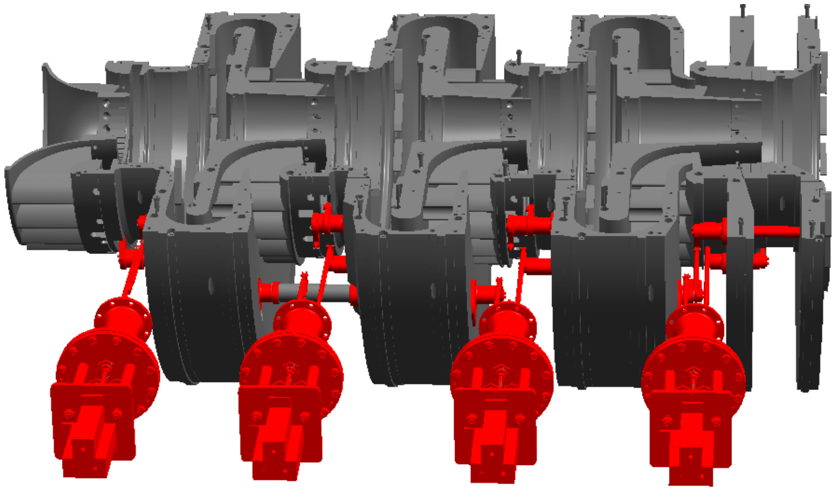


Figure 20 – Movable Geometry in Multi-Stage Centrifugal Compressor

CLOSING REMARKS

Tremendous advances have been made in the design and manufacturing of industrial centrifugal compressors. Many of these can be traced to the evolution of the analytical tools used in the design process. However, the sophisticated analytical methods would be a waste of effort if it was not possible to build the complex designs conceived by today's engineers. Therefore, one must recognize the critical role that improved manufacturing methods played in the advancement of centrifugal compressor performance. There is little value in designing a new component to the third decimal place if the manufacturing method cannot even assure accuracy to the first decimal place. Consequently, replacing 3-piece welding with 5-axis machining, switching from castings to machined fabrications, and the like have clearly contributed to higher quality components that, in turn, provide higher performance.

In closing, the industry has not reached the end of the evolutionary process for centrifugal turbomachinery. Further improvements can be achieved and with the ever-increasing demand for more energy efficiency equipment, OEMs will continue to strive for still higher performance levels and/or broader performance maps. It will be indeed be fascinating to watch the evolution of the next generation centrifugal compressor. To be continued…

NOMENCLATURE

ϕ = flow coefficient = $700.16\dfrac{Q}{ND_2^{\ 3}}$

D_2 = impeller exit diameter in inches
N = speed in rotations per minute
Q = flow in cubic feet per minute

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