MAXIMIZING COMPRESSOR EFFICIENCY WHILE MAINTAINING RELIABILITY

by

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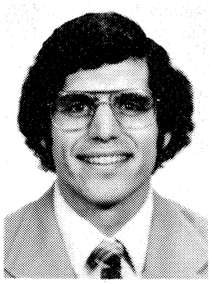

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ABSTRACT

The natural gas and chemical processing industries have historically and necessarily demanded high reliability from their centrifugal compressors, which has led to a significant emphasis on field experience of designs. This emphasis has sometimes resulted in new units which reflect design and manufacturing practices which can be improved upon. Users can, in many cases, significantly increase efficiencies by considering designs which use more recently developed technologies that create only refinements of machines historically used in their applications.

Accurately machined three-dimensional impellers are an example of an under-utilized available technology for multistage compressors. These designs used in several stages can provide significant efficiency gains, particularly at higher flows.

At lower flows, impeller efficiencies can be improved by a process called abrasive flow machining. This process can provide similar benefits in process compressors by improving surface finish in areas that cannot be reached with conventional metal finishing techniques.

Advancements in machine tool technology have also allowed changes in compressor casing designs. Numerical control (NC) machine tools can be used to machine inlets and variable area discharge volutes in the same axial casing space, thereby improving efficiencies through generous volute sizing without requiring additional diameter and bearing span.

Specific examples of uses of these design and manufacturing technologies and comparisons to alternative designs are detailed. The data presented show that these technologies can be used with confidence to provide high compressor efficiencies while maintaining reliability.

INTRODUCTION

Multistage centrifugal compressors are used in many applications in the petrochemical and energy industry. In many instances, whole processes or plants are dependent on the operation of a single compressor train. In virtually all cases, significant expense and lost income will occur if the compressor must be unexpectedly shut down, or does not perform as required by the field conditions. This situation has led to a significant emphasis on unit reliability, and the ability of the unit to perform as expected once it reaches the job site.

Historically, a common method used to ensure that this goal is achieved has been to insist on a design which has been used previously under similar conditions. With process compressors, this has been possible, especially when the process in which the compressor is to be used is well established. However, new processes are continually requiring different operating conditions which require at least modifications to existing compressor designs. Natural gas compressors are even more unique in their requirements due to the widely varying conditions at the many gas fields around the world. The design purchased for a specific application is, therefore, a new or modified compressor design in most instances.

In an attempt to require some experience base in a new application, users often require component experience in lieu of complete unit experience. This often takes the form of requiring existing casing designs, or existing impeller designs. In general, casings, or at least case designations, are flexible enough to fit staging for many different applications. Since stages themselves must be stacked together to form a compressor selection, vendors have found it possible to meet impeller experience requirements by designing families of standard stages, which cover a wide range of flow coefficients (NOMEMCLATURE), and can be expanded or contracted in size or stacked differently, to meet each application's requirements. Each vendor has thereby built up his experience base of existing gas path components and casing components, allowing him to claim at least some experience base for a given design.

The reliance on experience lists has provided users with many successful compressor trains throughout the world. It has also meant, however, that many of the compressors being bought at the present time consist primarily of components originally designed years ago. Although this is not significant for some components, those parts of the design which could have been affected by design and manufacturing technology improvements will be at a disadvantage compared to newer designs. Each vendor is therefore continually looking for opportunities to gain field experience on new components, thereby providing an experience base for future orders and keeping their designs competitive.

The following describes some technologies used sparingly in multistage process and natural gas compressors, advantages of these newer technologies to the user, and comparisons between designs which use the new technologies and those which do not. The advances described provide improved efficiencies, without any changes to the basic compressor design which would compromise reliability.

The first and primary technology discussed is that of three-dimensional (3D) impeller design and manufacture by using automated, precision machining techniques. These techniques result in stages which are predictable in performance, while improving efficiency of the compressor. The advances compared to traditional technologies consist of automated, computerized design procedures which accurately define an impeller geometry that performs as expected due to the consistency of the design procedure, and the ability to translate these designs into metal by multiaxis NC machine tools, which produce pieces that accurately reflect the designer's intended configuration.

The resulting impellers provide higher efficiencies and the ability to perform at higher Mach numbers than standard two-dimensional (2D) impellers. When compressors are selected to take advantage of the characteristics of 3D impellers, significant performance improvements can be made without fundamental changes in the overall design of the compressor train. The types of compressor trains that can best benefit from 3D impellers, and the benefits obtained, are discussed.

A second technology, known as abrasive flow machining, can enhance compressors which have flows well below values required for 3D impellers to be of benefit. In applications with large pressure ratios, large volume reductions, or limitations in driver capabilities or rotordynamics, impellers are generally used which have tip widths and blade shapes that preclude standard welding or metal finishing procedures in the flow channel of the impeller. Most manufacturers have developed some form of "slot" welding to connect the impeller disk and shroud, but less has been done historically to improve the surface finish of the inaccessible impeller flow path. Abrasive flow machining, or honing, can finish impellers to a high quality finish without the need for access to the flow path. Because the process can be used on low flow impellers which have a significant percentage of their losses caused by skin friction, the improved surface finish can provide significant efficiency improvements.

Abrasive flow machining has been used for several years on aircraft engine components. It is also used on small, intricate components for other industries. However, relatively complicated tooling is often required to direct the abrasive media through the areas requiring finishing. The bulk of the applications have, therefore, been those requiring finishing of multiples of the same part, where the tooling costs can be spread across many parts.

In process compressors, units are often sold one at a time, and the volume reduction through the compressor train means that often only one of each impeller design is used on a given unit. There is, therefore, hardly ever enough manufacturing volume to allow extensive tooling. However, the flow path of a shrouded impeller can provide its own "tooling", since it encloses and restricts any abrasive media passing through it. This makes it possible to finish individual impellers with a minimum of tooling. The effects of this type of finishing on several compression sections are discussed, and a means of estimating the potential effect on a given selection is provided.

A third design and manufacturing technology which can benefit the user both in efficiency and reduced compressor size and weight is the use of variable area discharge volutes machined into vertically split casings. This allows larger volutes to be used within the same diameter and casing length constraints. As is shown, inlet and discharge volutes can even occupy the same axial shaft span, providing the shortest possible bearing span.

Historically, vertically split compressors in back to back configurations have used either discharge volutes in the bundle or machined in the casing. When the volutes are in the bundle, either axial length is used or the diameter of the bundle is increased. When the volutes are in the casing, annular grooves are machined from the bore of the casing, requiring an increase in the casing diameter to contain the pressure. Because the thinnest casing wall occurs at only one location when a machined variable area volute is used, casing diameter can be reduced.

The technologies discussed can be seen to provide substantial opportunity for increased operating efficiency without creating major changes in designs or design philosophy when compared to traditional process compressors.

DESIGN AND CONSTRUCTION OF 3D IMPELLERS

Differences Between 3D and 2D Impellers

Typical 3D impeller geometries are shown in Figures 1 and 2, while a 2D impeller is depicted in Figure 3. A full 3D impeller is shown in Figure 1. It is characterized by an axial inlet, relatively long axial length and a three dimensional blade shape. A semi 3D impeller is depicted in Figure 2. It's geometric characteristics are similar to the full 3D impeller except that it has a shortened axial length and an inlet that is between axial and radial. The 2D impeller in Figure 3 has the shortest axial length, a radial inlet and a two dimensional blade shape. A cross section of a compressor using all three types of impellers is shown in Figure 4.

In a 3D impeller the blade angles along the leading edge may be varied from disk to shroud to match the incoming flow field and obtain optimum incidence angles. This is especially important for higher flow coefficient designs, where the blade passage is wider and has more variation in the flow field from disk to shroud. A 2D impeller has a basically constant blade angle from

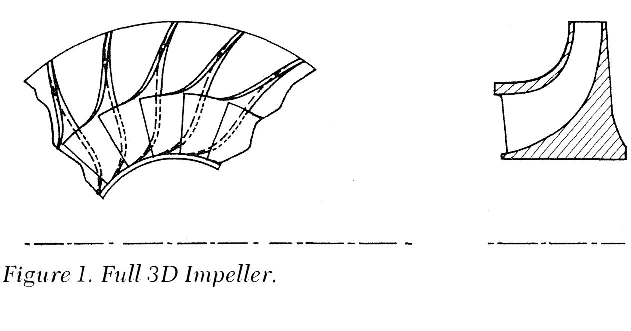

Figure 1. Full 3D Impeller.

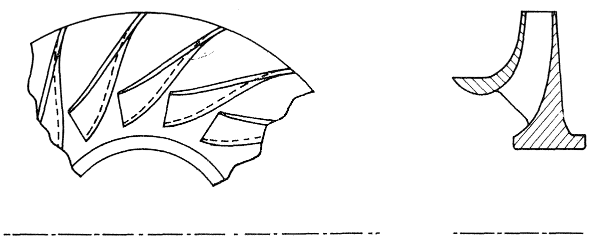

Figure 2. Semi 3D Impeller.

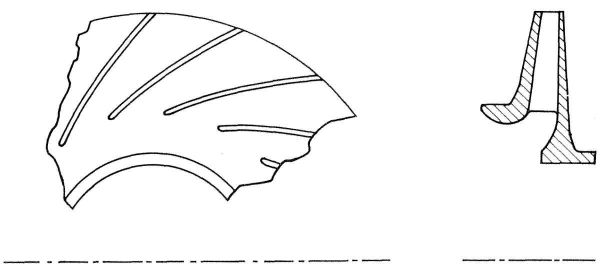

Figure 3. 2D Impeller.

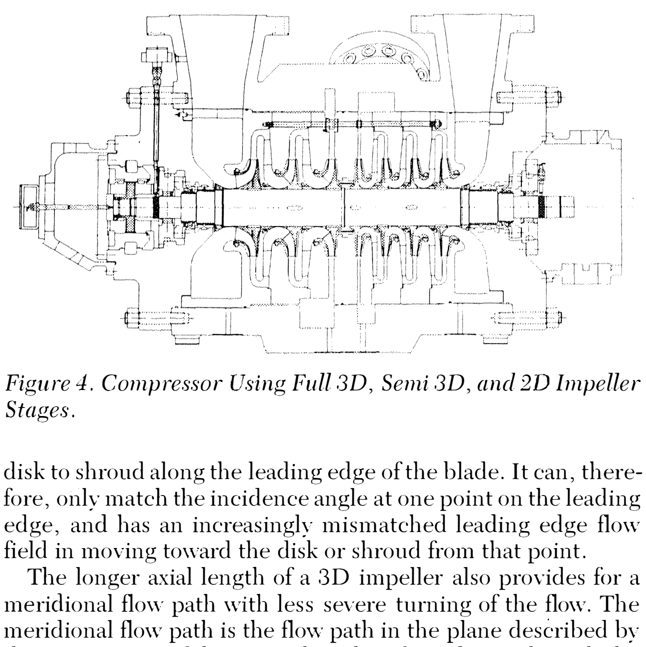

Figure 4. Compressor Using Full 3D, Semi 3D, and 2D Impeller Stages.

disk to shroud along the leading edge of the blade. It can, therefore, only match the incidence angle at one point on the leading edge, and has an increasingly mismatched leading edge flow field in moving toward the disk or shroud from that point.

The longer axial length of a 3D impeller also provides for a meridional flow path with less severe turning of the flow. The meridional flow path is the flow path in the plane described by the intersection of the gas path with a plane drawn through the axis of rotation, or more simply, the flow path one sees when viewing a cross section of the compressor. The reduced severity of turning helps control flow separation off the shroud much more effectively than 2D designs at higher flow coefficients. In addition, the longer 3D blades offer the flow more guidance, and are better able to control flow separation off the blade suction surface, thus keeping blade loading losses to a minimum.

Full 3D impellers are, therefore, best suited for high design flow coefficients. They are also better suited than 2D impellers at higher tip speed Mach numbers (NOMEMCLATURE). The inlet is at a lower radius and hence lower blade speed than the equivalent 2D design. The inlet also produces less of a variation in meridional velocity from disk to shroud. This gives lower inlet relative Mach numbers for the same tip speed Mach numbers.

Due to their relatively long axial length, full 3D impellers are generally used only as the first stage in a section for inline process compressors. Multiple full 3D stages would increase the shaft length significantly relative to 2D impellers, making a typical multistage compressor not practical due to rotordynamic considerations. One exception is natural gas pipeline compressors, which have relatively low head requirements, and sometimes use two or three full 3D impellers in succession.

Semi 3D impellers are suitable for both first stage and intermediate stage applications. They are more suited to the lower flow coefficients than full 3D impellers. At lower flow coefficients skin friction losses are more dominant than blade loading and meridional flow path turning losses. The shorter semi 3D blades have a smaller surface area, thus minimizing skin friction losses while still maintaining control over blade loading and flow path turning losses. Semi 3D impellers are also an improvement over 2D impellers at higher tip speed Mach numbers, but are not quite as efficient as full 3D impellers in this regard. They provide these advantages while minimizing any extra axial span relative to 2D impeller designs.

Design Techniques

Aerodynamic impeller design is based on both past experience and analytical techniques. Because of the complex nature of the flow field, it is imperative that the designer have a good test data base, proven design guidelines, and the proper analytical tools before designing a 3D impeller.

An excellent method that has been used to develop a test data base is described by Benvenuti [1]. It involves the use of specially instrumented single stage test rigs where component as well as overall stage performance can be measured. This type of data, when supplemented by correlation of production test data on complete machines, provide an excellent database for the design process.

Design guidelines tend to be evolutionary in nature. To begin, the designer must make some assumptions based on the physics of the problem. As more test data become available, the design rules evolve to produce more optimum and predictable designs. The design rules for 3D impellers of the type described here have had many years of evolution, and are well defined at this point in time. Some factors which enter into a set of design rules are: limits on overall impeller diffusion and blade loading, optimum blade loading distributions, optimum incidence angles on impellers and return channel vanes, limits on inlet relative Mach numbers, expected efficiency levels based on past designs and test data, expected surge margins based on past designs and test data, and guidelines as to which shapes of flow path components produce the best results.

Current production aerodynamic blade design programs are quasi-three dimensional in nature. They generally use an axisymmetric streamline curvature method in the meridional plane, and a separate blade to blade loading calculation. They also generally include CAD/CAM features which assist in the design iteration process, and can be used to generate the definition of the final geometry. A program of this type is described in the literature [2, 3]. A plot of the streamlines in the meridional plane of a high flow coefficient impeller is presented in Figure 5. The corresponding blade loading diagram is shown in Figure 6. The calculation procedures of Jansen, et al. [2], have been used to generate the diagrams of Figures 5 and 6. This procedure can be followed for any given impeller design. The basic streamline curvature methods have been in use for many years, and are described by Novak [4].

Performance prediction techniques are needed to convert blade designs into stage performance parameters. Some current performance prediction programs, Jansen, et al. [5], are empirical flow models which have been calibrated to extensive test data. They can be used to supplement actual test data to allow the designer to accurately determine the performance of new intermediate designs.

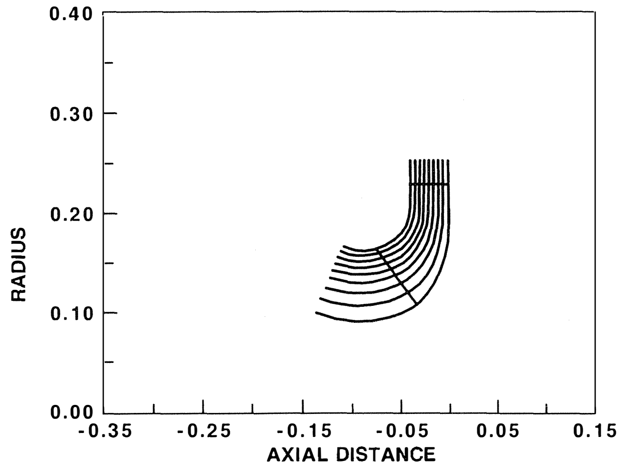

Figure 5. Impeller Meridional Contour and Streamlines.

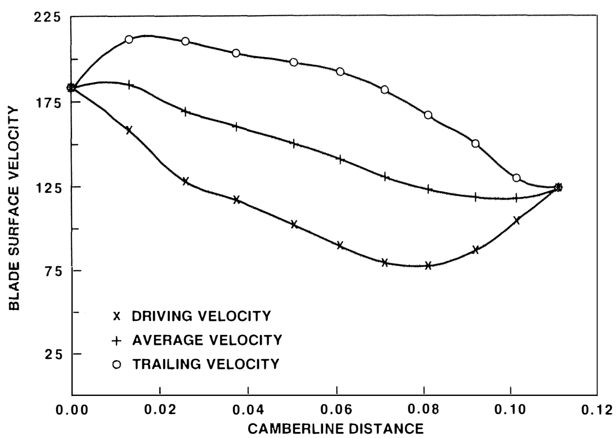

Figure 6. Impeller Blade Loading Diagram.

Diaphragms for use with 3D impellers can be designed in the same manner as those for 2D impellers, and often can be the same components. The designs discussed here use simple constant width vaneless diffusers and two dimensional deswirl vanes. The return channel geometry and width are adjusted to give optimum deswirl vane incidence angles. These designs, when combined with the 3D impellers designed with the above procedures, result in conservative but efficient stages.

Mechanical design of 3D impellers has historically had to accommodate applications which have taken advantage of the high Mach number capabilities. They have often been used in high tip speed, high head designs not typical to process compressors. These high speed impellers are usually designed without shrouds, and are found in applications such as small gas turbines and integrally geared compressors. Many design details, from materials used to shaft connection design, are unique to a given design or vendor.

However, when tip speed and head requirements are kept within typical limits, the mechanical design of 3D impellers in multistage compressors can be essentially the same as that for 2D impellers. In most cases, covered or shrouded impellers are used, with the shroud welded to the blades which are machined from the disk. Stress criteria, shroud and disk thicknesses, labyrinth seal interface surfaces, and the impeller to shaft attachment design can all be identical. For given tip speeds, 3D impellers often have lower shroud stresses because a longer blade is connecting it to the disk. Weld joint design and blade thickness can also be identical to similarly sized 2D impellers. When 3D impellers are used in this manner, any mechanical design or reliability issues due to differences from 2D designs are avoided, while retaining the performance advantages.

3D Impeller Manufacturing Processes

Historically, designs for 3D impellers in process compressors have usually incorporated separate disks, shrouds and blades. These designs evolved from older 2D impeller designs, incorporating separate blades primarily for manufacturing reasons. Forgings to be used for the disk did not have to incorporate the axial length of the blades, and could be purchased in pre-formed dished shapes which minimize the amount of machining to be done. Shrouds could be similarly formed. Blades of the 2D designs were simple shapes which could be bent or rolled to their final shape and welded to the disk and shroud.

When this design and manufacturing method was adapted to 3D designs, disks and shrouds could be designed as before, but the blade shape was now a complex three dimensional shape which could not be easily generated with simple metal forming methods. Manufacturers have tried using dies to stamp blades to the proper shape, but the die cost can be very expensive for small numbers of impellers. Others have tried to bend blades to shape with crude bending and stamping procedures. This reduces costs considerably, but can result in inconsistent, poorly shaped blades. A third method that has been used is casting either blades or whole impellers with precision investment casting methods, but pattern costs can be prohibitive for small numbers of impellers.

These cost and manufacturing issues have kept use of 3D impellers in process compressors below the level which can be justified by their efficiency benefits. The use of automated procedures in design and manufacture of machined 3D impellers avoids many of these problems.

The design techniques described earlier utilize a straight line element blade construction to facilitate flank milling. A blade shape constructed with straight line elements using the methods described by Jansen, et al. [2], is shown in Figure 7. Once these methods have defined the geometry of a given impeller, an NC machining tape can be generated to machine the blades. This automatically produces an inherently accurate impeller with evenly spaced, identical blades. Tooling costs are also minimized because the machining can start with a simple pancake forging. This makes the process practical for small numbers of impellers.

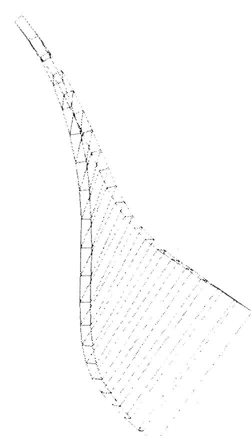

Figure 7. 3D Impeller Blade Constructed with Straight Line Elements.

The blades are machined from the disk forging, which allows sufficient room for the machine tool head. Shrouds are machined in the same manner as required by 2D impellers, prior to welding to the blades. From a manufacturing standpoint, the whole process is identical to what is done for a 2D impeller, except for the 5-axis machine tool used for the blade machining operation. Forging materials, NDT requirements, welding procedures and heat treating procedures are all interchangeable between 2D and 3D designs. Final machining, balancing, overspeed testing and rotor assembly procedures also duplicate those used for 2D impellers. The similarity of operations ensures component reliability, by minimizing any lack of familiarity with the impeller requirements at all steps of the manufacturing process.

WHY, WHEN, AND HOW TO USE 3D IMPELLERS

Why

The improved control of the flow field possible with 3D impellers means that higher efficiencies are usually obtained. A comparison is given (Figure 8) of relative stage design point efficiency levels over a range of flow coefficients, which is typical for 3D and 2D impeller designs. There is obviously some scatter about these lines, but for the purpose of discussion, they will suffice as presented. In the medium to high flow coefficient range, it can be seen that 3D impellers offer a distinct efficiency advantage over 2D designs. This advantage increases with increased flow coefficient. At lower flow coefficients, this advantage is diminished.

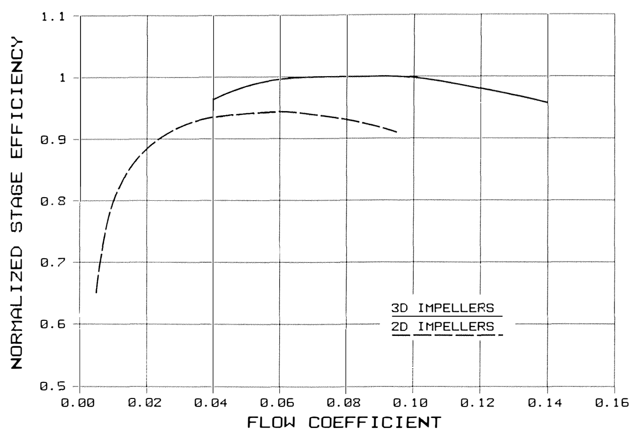

Figure 8. Comparison of 3D vs *2D Impeller Stage Design Point Efficiency* vs *Flow Coefficient.*

A specific example of 3D *vs* 2D impeller stage performance, for two impellers of approximately the same flow coefficient, is shown in Figure 9. The 3D impeller stage design point efficiency is approximately ten percent better than the 2D impeller stage. The corresponding 3D head coefficient (NOMEMCLATURE) is also approximately ten percent higher, indicating that the work coefficients (NOMEMCLATURE) are approximately equal, and that the higher head coefficient is due to the higher efficiency. The 3D data is taken from a closed loop shop performance test of a single stage compression section, using carbon dioxide as the test gas. A data point taken in the field, also shown on Figure 9, corroborates the shop test data. The 2D impeller was tested in an open loop air test single stage development test rig.

The 3D impeller was tested with an inlet volute, a 1.55 diameter ratio radial vaneless diffuser, and a variable area discharge

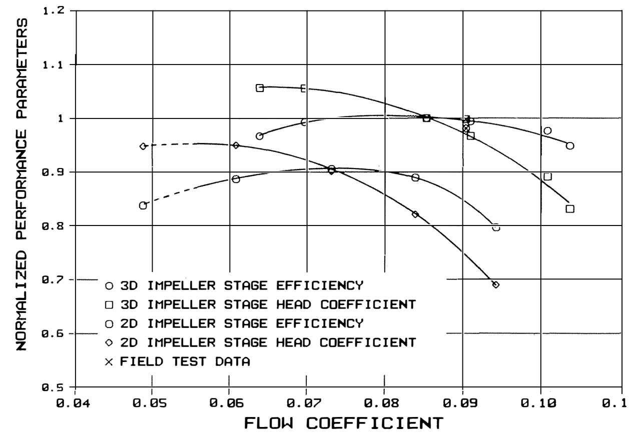

Figure 9. 3D vs *2D Impeller Stage Performance Test Data.*

volute. The impeller blade diameter was 728.6 mm, trimmed from an original 755 mm, leaving the disk and shroud at the original diameter. The 2D impeller was tested in a stage rig with a vaneless diffuser having a 1.6 diameter ratio, and a return channel. Its diameter was 450 mm. Because of these factors, this comparison is not a one to one correspondence. The 3D impeller is favored due to its larger size and hence smoother relative surface roughness, and due to the fact that it was tested at a higher Reynolds number. It is estimated that these factors account for on the order of three in efficiency, reducing the difference from ten percent to seven percent, which is in line with Figure 8.

The 3D and 2D impeller stage efficiency curves of Figure 8 are superimposed over compressor section test data in Figure 10. The section test data are taken from ASME Class III shop performance tests. The data was taken on a variety of impeller sizes and at various test Reynolds numbers. No corrections have been applied to the data presented here to account for these variations. The shop test data on the figure are coded by symbol as to which sections consist of all 2D impellers, all 3D impellers or a mixture of 2D and 3D impellers. Also noted are those sections in which the impellers were abrasive flow machined. In general, it is seen that the shop test data follows the trends of the 2D and 3D impeller stage curves. The abrasive flow machined compressor sections are high in efficiency as expected. This will be discussed in greater detail later in this paper.

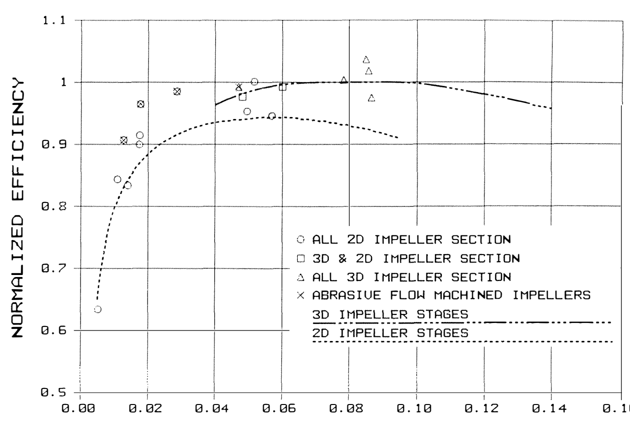

Figure 10. Compressor Section Performance Test Data.

When and How

It should be obvious from the data presented that 3D impellers should be used whenever practical. A prudent strategy, therefore, is to identify and avoid those situations which keep them from being advantageous on a particular compressor.

The performance advantages of 3D designs have been shown to be best utilized at higher flow coefficients. Therefore, the compressor designer should try to use selections at higher flow coefficients. This means in general higher speed, smaller diameter compressors for a given application. Low speeds and large compressor sizes result in low flow coefficients which result in lower efficiencies for a given application.

To achieve the desired high speed with various drivers, a gearbox is sometimes required. This is normal and expected with motor drivers. Often, however, compressors are force-fit to gas turbine speeds, and can also be affected by speed limitations when driven directly by steam turbines. Performance advantages could be obtained in both these cases, even after accounting for gear losses. An example of this is shown in Figure 11, where high speed and low speed selections for two sections of compression are compared. As shown by Figure 12, both the high speed and low speed selections perform the same task, but the high speed selection can provide better performance even after gear losses are added, since the efficiency difference is between four and five percent. This is true even though the speed was changed in this case by a factor of only 1.39.

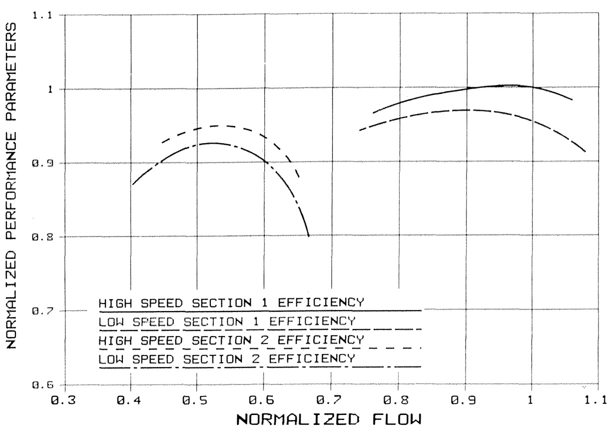

Figure 11. Efficiency Comparison for High and Low Speed Selections.

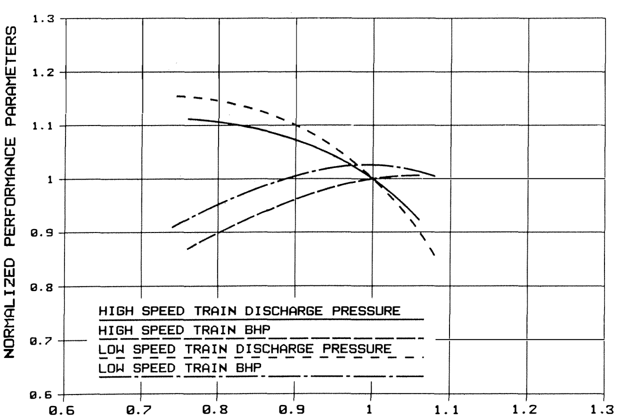

Figure 12. High Speed vs Low Speed Selection—Overall Train Performance.

In general, the user has to weigh the performance and size benefits against any additional train complexity when looking at the potential addition of a gear to a compressor train. The normal rotordynamic considerations also have to be taken into account.

Even when higher flow coefficient designs are not practical, 3D impellers can provide performance benefits at flow coefficients as low as approximately 0.040 or even below. This occurs often in typical process compressors, providing a means of improving performance with virtually no change to existing train configurations. At times when even lower flow coefficients are required, traditional 2D impellers are sufficient. In this case, other technologies can be used to improve performance.

ABRASIVE FLOW MACHINING

The Process

Abrasive flow machining is a process which forces an extremely viscous media through a work piece, thereby improving its surface finish. The media is composed of an organic polymer base and abrasive particles. As the media is forced through work piece restrictions at pressures of typically several hundred psi, the abrasives hone or polish the surfaces comprising the restriction. Selection of media viscosity, abrasive size, media pressure and flow rate, and the number of cycles all affect the result, which can vary from significant metal removal to a simple surface polish. Vendors are available to provide machines and media, or to do the finishing operation itself on a job shop basis.

Abrasive flow machining is used extensively in aircraft engines [6]. It also is used on other products where production runs can justify the tooling costs and schedules. An example of typical tooling for an open impeller is depicted in Figure 13. It is discussed in more detail by Perry and Stackhouse [6]. On closed impellers, the impeller flow channel itself comprises the restriction, and on a single part basis abrasive flow machining can be done with no other tooling than a cylinder mating to the impeller eye and a blank to cover the impeller bore. Media forced through the impeller from eye to exit can be simply dumped to the worktable for later recirculation. There are obviously intermediate approaches between full tooling and this minimal tooling which can be used depending on the economics of a given situation.

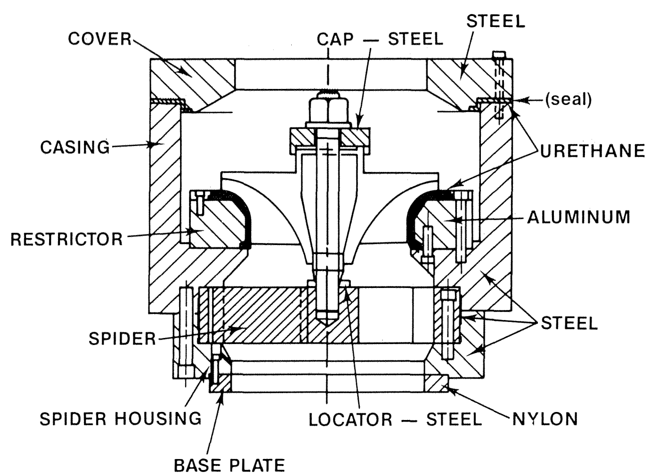

Figure 13. Abrasive Flow Machining Tooling.

Using normal techniques, closed impellers can be very difficult to finish to a smooth gas path finish, especially when they are low flow designs. The various slot welding techniques that manufacturers have developed address construction problems but not surface finish. Gas paths are usually finished with hand grinding prior to welding, with blasting (and grinding where possible) being the primary finishing operation after welding.

Abrasive slurry vibratory finishing is sometimes used as well, but most covered impellers have only a blasted gas path surface, with finishes between 63 and 125 root mean square (rms), tending toward the higher number.

As noted above, the abrasive flow machining process can be controlled to remove a small amount of metal, without significant modification to the basic structure and design of the impeller. There are, therefore, no reliability issues relative to designs without this finishing operation. Gas path finishes can consistently be reduced to 32 rms or better when starting with an impeller finished in a conventional manner.

Performance Effects

Pressure losses due to fluid friction are a significant portion of the losses in a centrifugal compressor stage. The flow path surface finish directly affects these pressure losses. As noted above, abrasive flow machining can be used to improve flow path surface finish and hence improve efficiency.

The pressure losses due to fluid friction in a centrifugal compressor, like those in a pipe, are a function of the relative surface roughness, that is the surface finish divided by some characteristic dimension. In a pipe this can be the pipe diameter, or in the case of a compressor, the impeller tip width. This means that impellers with small tip widths (low flow coefficient and/or small diameter impellers) have an inherently larger relative surface roughness. It is these type stages that can benefit most from abrasive flow machining.

A method has been devised to estimate the effects of improving surface finish on stage efficiency. It involves using friction factors as presented by Strub, et al. [7], and assumes like those authors, that fluid friction accounts for 70 percent of the pressure losses. The performance of a stage with improved surface finish can then be estimated from its known performance at its normal surface finish.

This method has been used to estimate the effect on stage efficiency of using abrasive flow machining to improve the impeller surface finish from a normal 125 root mean square to 32 root mean square on 450 mm diameter impellers over a range of flow coefficients. This was done for numerous stages typical of the process industry, including both 2D and 3D impeller designs.

The ratio of stage efficiency is shown in Figure 14, with improved surface finish to stage efficiency with the normal surface finish *vs* flow coefficient. Although it is presented as a single line, actual calculated results are a band on the order of plus and minus 0.5 percent. The line was drawn favoring the low side at high flow coefficients to compensate for the assumption noted above that friction losses are a constant 70 percent of the total

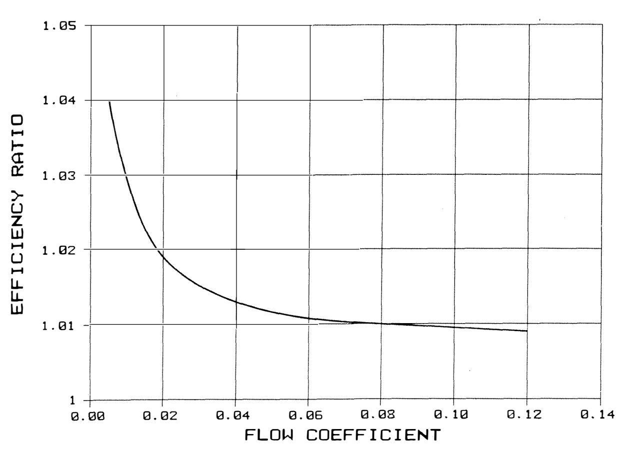

Figure 14. Effect of Abrasive Flow Machining on Stage Efficiency.

losses. At high flow coefficients friction losses are in actuality a lower percentage of the total losses than at lower flow coefficients.

The performance test data presented in Figure 10 for abrasive flow machined impellers are in general agreement with this simple estimating approach. The abrasive flow machined impeller sections show higher efficiencies than the corresponding sections with no special surface finish treatment, and the improvement is seen to be larger at the lower flow coefficients. The actual improvements are, if anything, larger than predicted by the estimate.

MACHINED DISCHARGE VOLUTES

Design Background

Many manufacturers use back to back compressor designs when an intercooled design is required to reduce power consumption or limit discharge temperatures. This type of design puts the discharges of both compression sections side by side in the center of the compressor (Figure 4). It is sometimes limited to horizontally split designs and lower pressure vertically split designs because of rotor stability and thrust balancing considerations, but is a very common type of design in the industry.

In horizontally split cases, the side by side discharge volutes are cast as part of the casing in cast designs, or formed from plate in fabricated designs. This allows reasonable freedom in designing the volutes, and presents no special problems. In vertically split designs, castings are sometimes used, but it is more common to see forged or rolled plate casings, especially in higher pressure ratings. In this type of design, compromises are usually made in the shape and size of the discharge volutes.

Two basic approaches have been used to design the discharge volutes in this type of vertically split forged or rolled plate compressor casing. One has been to put discharge volutes in the bundle, with the casing having only a radial opening to the discharge nozzle. This design minimizes case machining, but requires increases in casing bore diameter, rotor length, or both (Figure 15). The other approach has been to machine an annular groove in the casing, which either is used as machined (constant area), or has a variable area volute shaped by welded-in segments (Figure 16). This approach increases the casing outside diameter because the casing outside the annular groove must contain the casing discharge pressure. Both of these approaches put pressure on the compressor designer to minimize discharge volute size, thereby reducing casing size or length issues.

Design Capabilities

Large numerically controlled multiaxis machine tools can and have been used to machine variable area volutes in forged vertically split compressor casings. The volute can then be varied in both axial length and radial depth by continuous automatic adjustment of the cutting tool relative to the casing.

This capability allows the designer to incorporate generously sized discharge volutes while minimizing casing diameter and length. An example of a volute designed in each of three ways, is shown in Figure 17, illustrating this effect with three designs for the same pressure rating and discharge flow. Note that the variable area machined volute is as large as the constant annulus machined volute only at one point, and the asymmetry of the design requires a finite element analysis to determine minimum wall thickness criteria (Figure 18).

One other advantage provided by the machined variable area volute is useful when there is only one impeller in a compression section, such as in an ammonia syngas train recycle impeller. As illustrated in Figure 19, if the bundle to casing seal is designed to follow the varying axial location of the edge of the machined volute, it is possible to locate the section inlet such that it over-

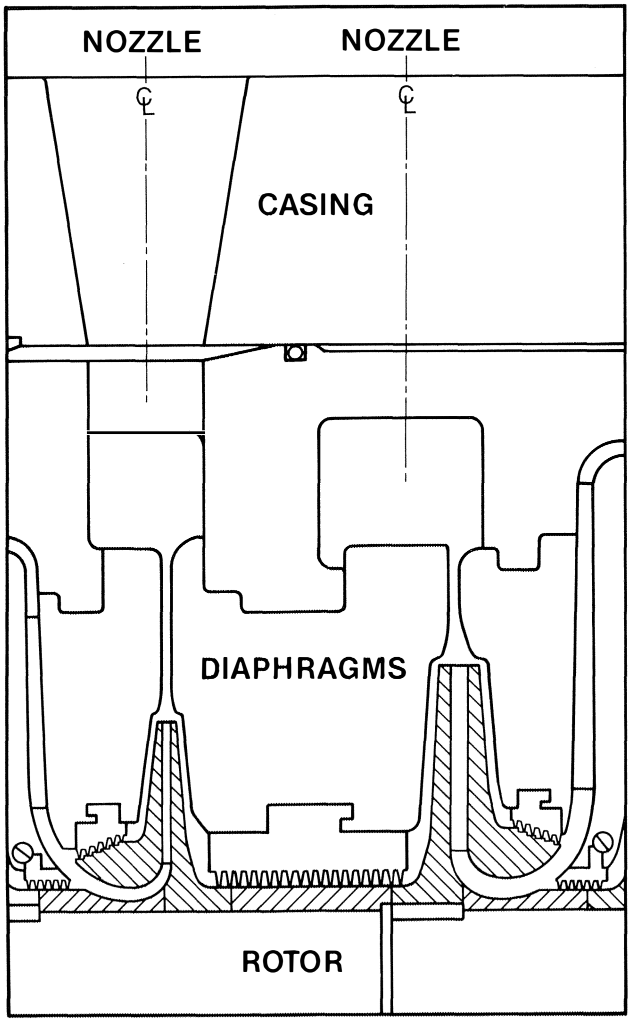

Figure 15. Bundle Volute Design.

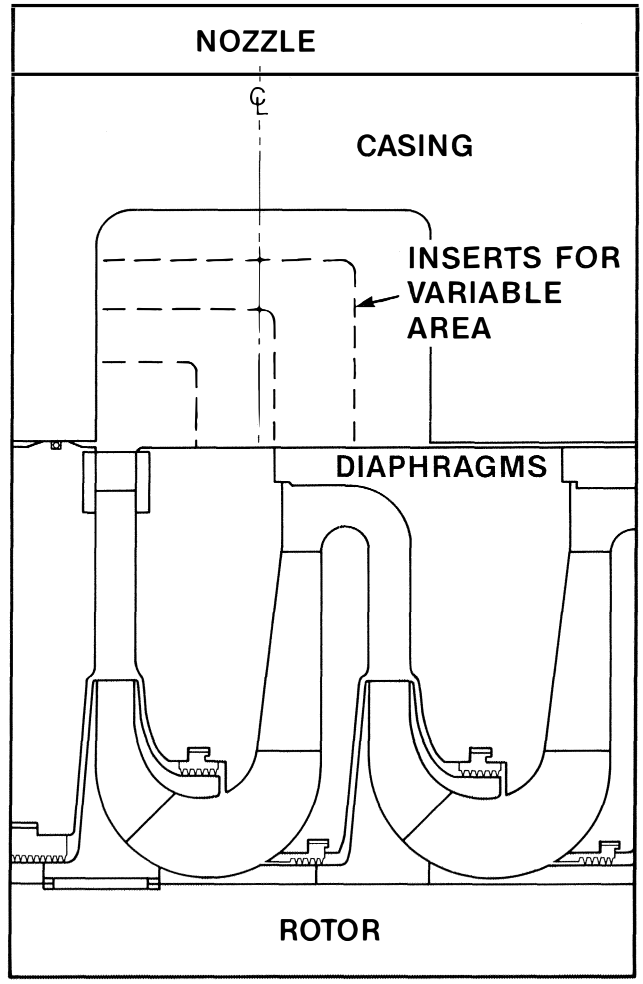

Figure 16. Casing Groove Volute Design.

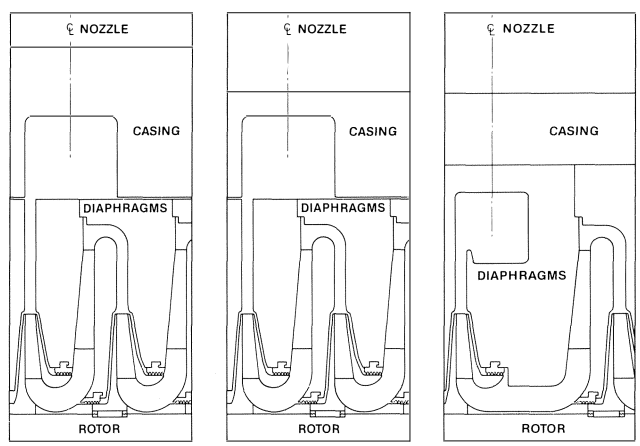

Figure 17. Comparison of Volute Designs.

laps the discharge opening. This can be done by locating the inlet circumferentially where the discharge volute is narrow relative to its width at the nozzle. This design also minimizes span while allowing the desired volute size.

Performance Effects

The discharge volute takes the flow from the exit of the last stage's diffuser. The fluid velocity at this point is in general higher than that in the system piping, so the flow must be diffused further in the volute. The geometry of the volute affects the efficiency of the diffusion process. An optimally sized variable area volute attempts to efficiently convert fluid velocity to pressure by providing a flow path with a gradually increasing area which simultaneously collects additional flow coming off the diffuser and diffuses the gas already in the volute. A constant area volute is basically a dump diffuser which loses most of the pressure associated with the velocity head.

Based on analytical techniques and available test data, engineers can estimate the stagnation pressure loss coefficient (NOMEMCLATURE) for a typical variable area volute in a vertically split casing to be approximately 0.55 while a corresponding constant area volute would have a stagnation pressure loss coefficient of approximately 0.89. A plot of the ratio of compressor section efficiency, with a variable area volute to the efficiency with a constant area volute *vs* the number of stages in the compressor is shown in Figure 20. The discharge volute affects only the performance of the last stage in the section, since the exit geometry of the previous stages is unchanged. The use of variable area volutes is thus most important on compressor sections with a small number of stages, and has a reduced effect on sections with large numbers of stages, as would be found on straight through, nonintercooled designs.

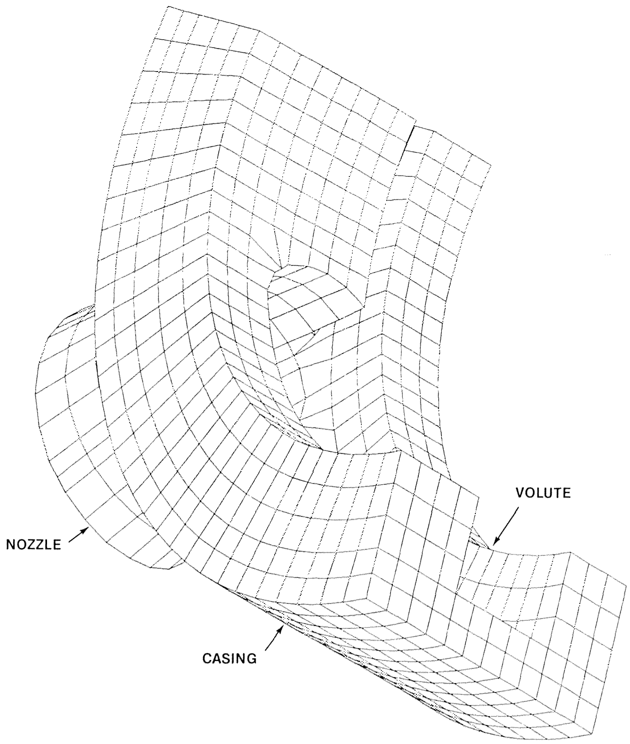

Figure 18. Finite Element Analysis Model.

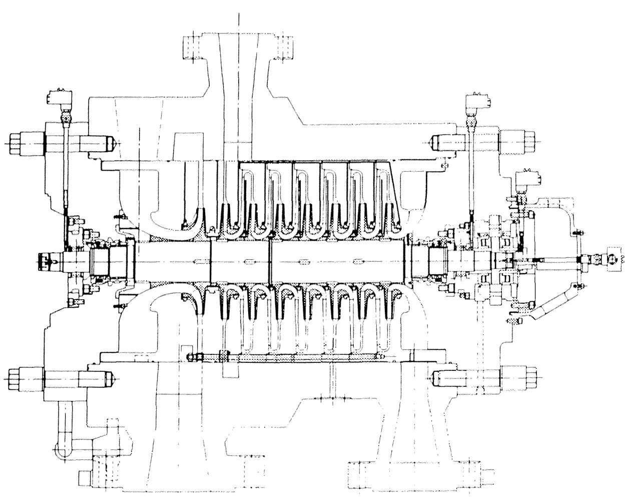

Figure 19. Compressor Using Overlapping Inlet and Discharge, with Variable Area Machined Volutes, 3D Impeller, and Abrasive Flow Machined Impellers.

When volutes are constrained to smaller than optimum areas, the fluid velocities increase, causing an increase in stagnation pressure loss due to skin friction. On the other hand, a variable area volute that is oversized can cause increased flow separation, which also increases the stagnation pressure loss. By allowing the volute to be sized properly, machined variable area volutes can minimize the stagnation pressure losses.

CONCLUSION

Three examples of available design and manufacturing technologies that can be used to benefit the compressor operator have been discussed. In each case, designs using these

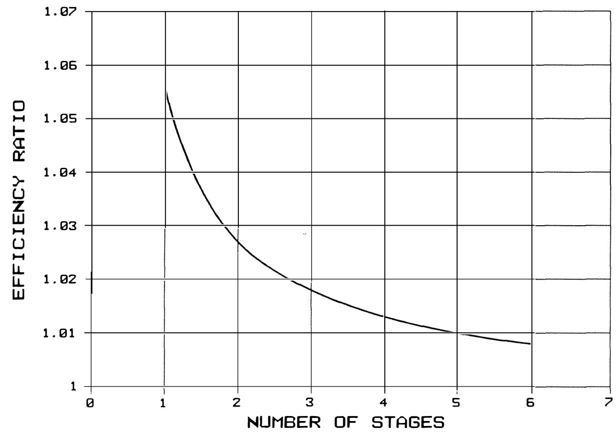

Figure 20. Effect of Variable vs *Constant Area Discharge Volutes on Section Performance.*

technologies can be applied within the realm of traditional or historical types of compressors, without significant departure from designs which have provided a long history of reliable service in process and natural gas applications.

The first and most significant technology discussed is the use of machined 3D impellers in multistage compressors. A summary of the advantages and how to apply them follows:

- Predictable improvements in excess of five percent in stage efficiency are possible, depending on the flow coefficient range.
- Improved operation and increased efficiency on high molecular weight gases, which cause high Mach numbers, are possible.
- The improvements can be maximized with high flow coefficients, but still exist at typical lower flow coefficients.

Another technology discussed is abrasive flow machining, which can be used to provide advantages as follows:

- Stage efficiency improvements on the order of two percent, depending on flow coefficient, are possible.
- No change in existing stage geometry is required to obtain the improvements.

The third technology involves machining capabilities of discharge volutes, which provide benefits as follows:

- High efficiencies on back to back vertically split compressors can be maintained through optimally sized and configured discharge volutes.
- Shaft spans can be shortened by overlapping the inlet and discharge axial locations while maintaining optimum volute sizes.

In some instances, all three of these technologies can be used in one compressor. An example of such a design is shown in Figure 19; a high pressure syngas makeup/recycle compressor with a 3D recycle impeller, abrasive flow machined impellers throughout (especially important on the low flow makeup impellers), and variable area machined volutes to minimize span and weight while increasing efficiency.

Whether a particular application can use any or all of these technologies to advantage should be considered when evaluating new equipment requirements. In most instances, it will be found that at least one will be applicable, and the user can improve operating efficiency, while maintaining the reliability to which he has become accustomed in his application.

ACKNOWLEDGMENT

The authors would like to thank T. Blehar for his extensive efforts in the original compilation, reduction and documenting of the performance data used in this paper, and R. Ide for his help in gathering and organizing the data.

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NOMENCLATURE

φ = Flow Coefficient (dimensionless) = $4Q/(\pi u_2 D_2^2)$
Q = Inlet Volume Flow
u_2 = Impeller Blade Tip Speed
D_2 = Impeller Diameter
M_u = Tip Speed Mach Number = u_2/a_{01}
a_{01} = Speed of Sound at Inlet Stagnation Conditions
$\tau\eta_p$ = Head Coefficient (dimensionless) = $gH_p/(\Sigma u_2^2)$
H_p = Polytropic Head
τ = Work Coefficient (dimensionless) = $gH/(\Sigma u_2^2)$
H = Enthalpy Rise
ω = Stagnation Pressure Loss Coefficient = $\Delta P_t/(P_t - P_s)$
P_t = Stagnation Pressure
ΔP_t = Stagnation Pressure Drop
P_s = Static Pressure
η_p = Polytropic Efficiency = H_p/H