CENTRIFUGAL COMPRESSOR PERFORMANCE ENHANCEMENT THROUGH THE USE OF A SINGLE STAGE DEVELOPMENT RIG

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ABSTRACT

Despite recent advances in computation fluid dynamics, the complex flows through centrifugal compressor components are still not totally understood. These uncertainties detract from the designer's ability to accurately predict component performance. Consequently, design verification remains largely dependent on performance testing. Some data can be gathered during testing of production units. However, this information can be somewhat limited, i.e., only overall or stage data, no component data. A more direct approach is to gather the required data in a single stage research vehicle. Recent experiences in the use of such a rig are presented.

The advantages of rig production testing are described. Also included is a description of the test rig facilities used: the rig itself; the instrumentation; and the data acquisition system.

A recent performance comparison of two three-dimensional impellers is discussed. A stage containing the first wheel exhibited an undesirable droop in pressure rise toward surge. Therefore, a new arbitrary bladed, inducer-style impeller was developed and validated in the rig. Test results for both wheels are presented.

Tests being conducted using a variable geometry low solidity vaned diffuser (LSO) are also cited. Included are comments on the advantages/disadvantages of LSOs, prior experience, and the factors which prompted the test program. Recent test results are submitted which show the effect of LSD setting angle on stage performance.

The authors conclude that single stage rig testing is essential to: acquire a better understanding of existing stages; verify new stage or component designs; validate design procedures; and develop the technology necessary to achieve enhanced compressor performance.

INTRODUCTION

A significant number of recent technical papers have addressed the advances made in the field of computational fluid dynamics. Many suggest that, within a few years, researchers and designers will have the ability to accurately model the complex flowfields through centrifugal compressor components such as impellers, vaned or vaneless diffusers, return channels, volutes, etc. And, while it cannot be argued that these new analytical tools will improve the understanding of the flow physics involved, it is doubtful that they will ever totally replace the "real world' information obtained through production or prototype testing.

Most researcher and designer engineers agree; in the development or confirmation of new concepts or analysis/prediction techniques, there is no substitute for good test data. The various means are addressed that centrifugal compressor designers use to obtain data and comments on how this information is utilized to enhance compressor performance. The advantages/disadvantages of each option are discussed. Emphasis is placed on the need for a single stage research vehicle in the development and validation of new design procedures. Recent performance test results are cited which demonstrate the effectiveness of a rig in deriving and confirming new design and prediction techniques.

PERFORMANCE TESTING

As stated, centrifugal designers rely heavily on performance test data to:

- validate new designs.
- confirm performance maps.

- calibrate prediction techniques.
- develop new design, analysis, or prediction techniques.
- demonstrate experience to customers.

As such, manufacturers of centrifugal compressor must have the ability to collect good (accurate) data.

Production Testing—Flange To Flange Results

As opportunities are abundant, many turbomachinery vendors try to collect the information described above during production testing. However, in most cases, such testing gathers only flange to flange performance. That is, a compressor (or compressor section) may contain five centrifugal stages (Figure 1). Flange to flange data will reflect the combined performance of all five stages operating in unison. While adequate for confirming overall compressor performance, flange data is of marginal assistance when problems result or when the analyst is interested in the performance of one particular stage or stage component.

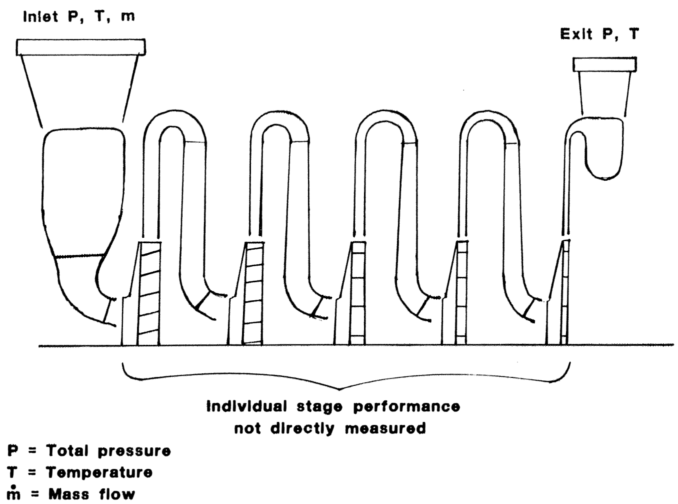


Figure 1. Flange to Flange Performance Data.

Obviously, the data are not totally without value because: if the data matches predictions, then one can assume all components are acting as expected; and overall data trends can indicate the effectiveness of any modifications applied to a machine. As an example of the latter, the overall performance for a three stage compressor section is shown in Figure 2. Changes were required to improve the first stage impeller's capacity. Despite only having sectional performance, the effectiveness of the modifications are obvious as overload capacity improved. In short, flange to flange data were sufficient in this situation.

Production Testing—Stage Results

On some production centrifugals (typically low pressure units), it is possible to collect individual stage data; i.e., measure the performance of one impeller along with all of its associated stationary hardware (guidevane, diffuser, return channel, or volute, etc.). While still not providing discrete component performance, stage data does provide more insight than flange to flange information; direct evaluation of the performance of individual stages. Some typical locations for stage instrumentation are illustrated in Figure 3.

Under most circumstances, stage data can be sufficient to demonstrate the effectiveness of a new design. As with overall data, one can compare the test results to predicted stage curves and

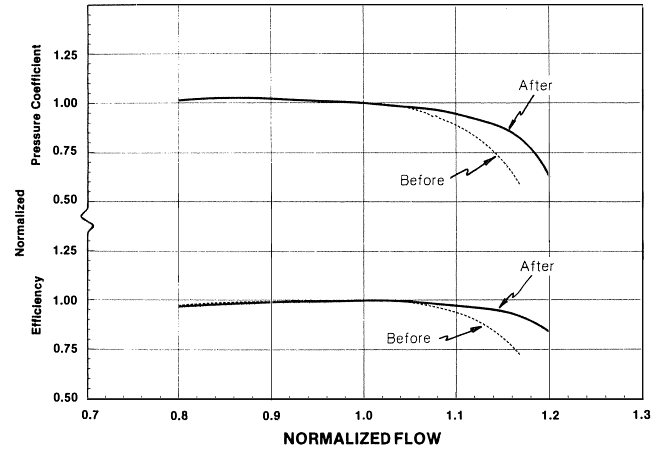


Figure 2. Sample Flange to Flange Data Before and After Modifications.

determine that the components, acting together, conform to expectations. However, should deviations exist, the analyst must somehow judge which of several components (impeller, diffuser, guidevane, etc.) is causing the problem. Or worse, the deficiency may result from the combination of components (i.e., impeller-guidevane, impeller-return channel, diffuser return channel) rather than any individual element acting alone.

One method used to derive component performance from stage data involves the use of performance prediction or 1-D analysis software. An analysis engineer performs an iterative parametric study in an attempt to identify the source of a stage deficiency. The losses associated with the various stage components are adjusted until a stage model is obtained which matches the measured data. Possible adjustments might include: increased return channel

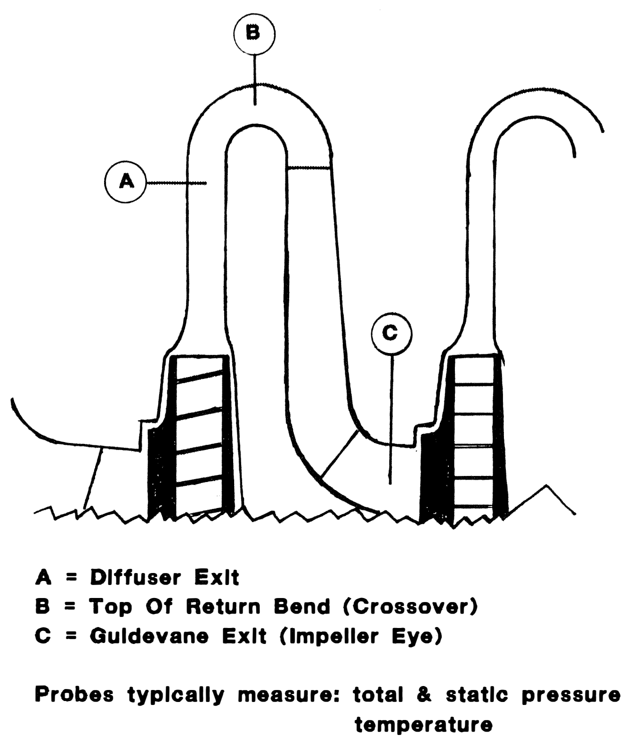


Figure 3. Typical Probe Locations for Measuring Stage Performance.

losses; introduction of impeller inlet preswirl; deration of impeller passage area or increase in boundary layer blockage, etc. Once an acceptable model is achieved, the engineer then tests various corrective measures until a satisfactory solution is found.

Of course, the stage model and, therefore, the proposed solutions, can only be as good as the assumptions made regarding the source of the performance problem. And, these assumptions will be based on the analyst's prior experience with similar performance patterns. Still, even the most experienced engineer can be misled by data trends. For example, consider the stage performance map displayed in Figure 4. There are at least two components which could cause the performance trend shown. First, the impeller may be undersized, leading to a premature choke. Second, some downstream component might be restricting the overload capacity (i.e., insufficient throat area or adverse incidence on a vaned diffuser or return channel). In the use of a 1-D analysis code, one would be able to duplicate the trend shown either by decreasing the impeller overload capacity or by reducing the return channel passage area. Therefore, the final decision must often be based on either intuition or some prior knowledge regarding the particular components involved. For those curious, the source of the performance problem in this case was an undersized return channel.

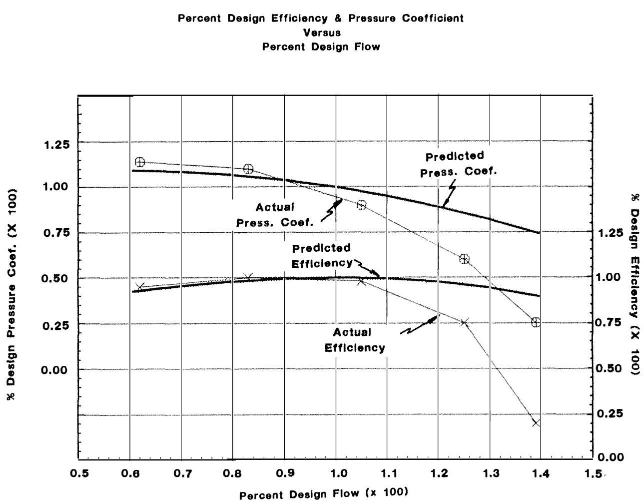
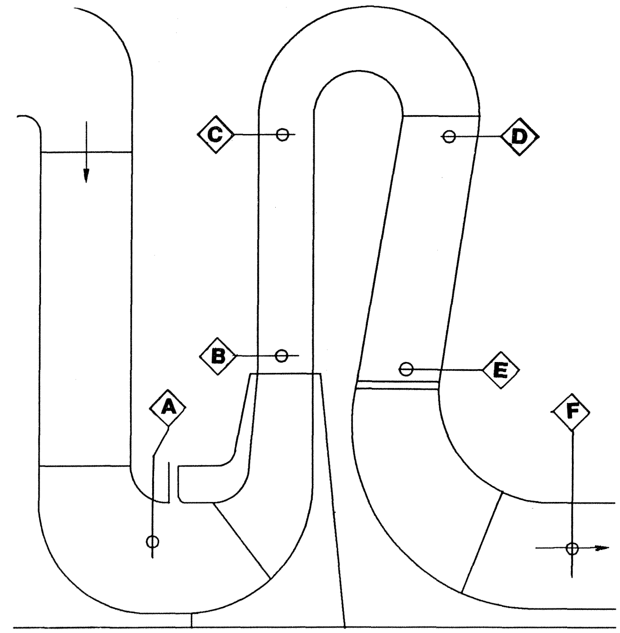


Figure 4. Sample Stage Data.

Had component data been available in the above case, the engineer could have quickly identified the problem source. Performance levels for each component would have been compared to expectation and the return channel restriction would have been obvious. However, the installation of component instrumentation is difficult (if not impossible) on a multi-stage centrifugal. To illustrate, the minimum recommended instrumentation shown in Figure 5 must be installed to obtain accurate and reliable component results. One can imagine the difficulties associated with locating all of the required probes. And, even if it were feasible to physically position the probes, it may be impossible to get the associated tubing and/or wiring out through the case. In short, gathering component data in multistage compressors is not practical nor cost effective.

The Single Stage Test Rig

A far more practical means of collecting component performance information is a single stage test rig (SSTR). The purpose of a rig is to duplicate a centrifugal stage in both geometry and flow conditions, but do so in a manner that allows easy access to each



Location	Static Pressure Probes	Total Pressure Probes	Temperature Probes
A	3	3	3
B	3	3	
C	3	3	
D	3	3	
E	3		
F	3	3	3

Note: Probes located 120 degrees apart

Figure 5. Minimum Recommended Probes for Collecting Component Performance.

stage component. The following section describes the rig that was used to gather the data described herein.

Test Rig Configuration And Associated Hardware

The single stage rig is a permanent closed loop system dedicated to measuring aerodynamic performance. Located away from the production test facility, the rig has become an independent test cell.

An Overview Of The Test Facility

An overall view of the test facility (Figure 6) best describes the rig configuration. A 1500 hp variable speed motor and speed increasing gear drive the compressor with allowable speeds in excess of 12000 rpm. The test loop consists of a 40 ft straight run from the discharge flange to the process cooler. Process temperature control is implemented by opening or closing a bypass line. From the cooler process, gas proceed through a 30 foot flow straightening and orifice run, leading to the test rig inlet.

SSTR Internal Components

The single stage compressor is designed for cost effectiveness and ease of assembly. Concentric rings, fastened by tiebolts, allow assemblers to unstack internals without entirely disassembling the rig. That is, an impeller or diffuser changeout may be completed without disturbing the remainder of the rig assembly (Figure 7). Further, the concentric rings are oversized to allow for entire compressor reconfiguration without manufacturing and assembling a complete case.

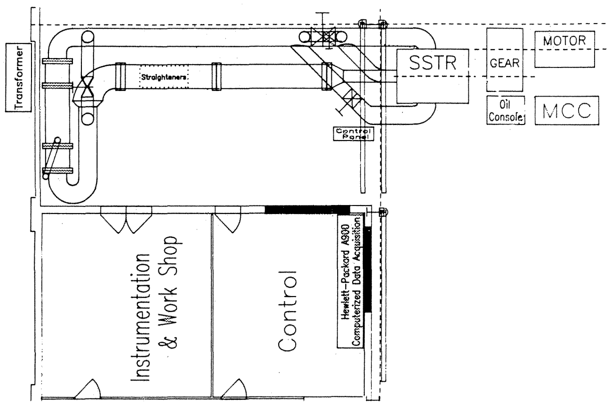


Figure 6. Overview of Single Stage Rig Test Facility.

The compressor inlet (also shown in Figure 8) consists of an inlet plenum followed by a false or pseudo-return channel to simulate the inlet conditions seen by a multistage centrifugal impeller. One other configuration, an axial inlet, has been designed and tested within the rig. The axial inlet will be used extensively during future pipeline booster development tests.

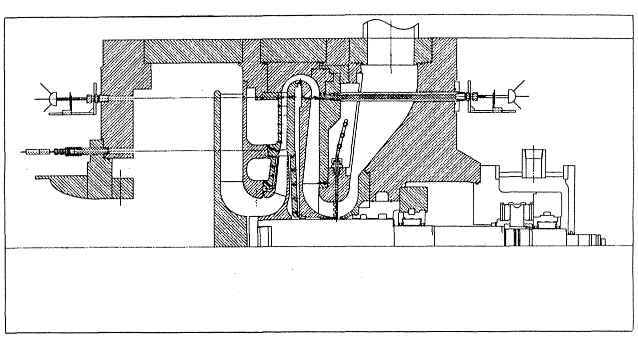
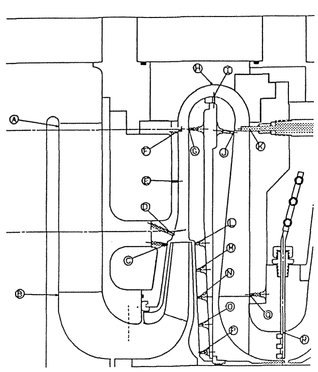


Figure 7. The Single Stage Test Rig.

Test Rig Control Facility and Instrumentation

Due to the permanence of the test facility, the control center has become dedicated to the SSTR (and other adjacent test rigs). Test rig startup and shutdown along with prompting data acquisition and actuation of the pneumatic discharge valve are regulated from the center.



STATION	Static Press. Probes	Total Press. Probes	Temp. Probes	Total Press. Rakes	Temp. Rakes	Press. Trav. Probes
A	3			3	3	
B	3			4	3	
C	3					
D	3	3	3			1
E	3	3				1
F	3	3	3			1
G	3					
H	3					
I	3					
J	3					
K	3					1
L	3					
M	3					
N	3					
O	3					
P	3					
Q	3					
R	3		3	3		1

DATA ACQUISITION STATIONS

INLET	-	A-B
IMPELLER	-	B-D
DIFFUSER	-	D-F
RETURN BEND	-	F-K
RETURN CHANNEL	-	K-R
OVERALL STAGE	-	B-R

Figure 8. SSTR Instrumentation List.

Test instrumentation includes 122 pressure, 44 temperature, and six traverse probes together with probes for monitoring mechanical concerns. To facilitate the acquisition of data from each probe, a multiple scanivalve system is employed. All pressure instrumentation within the loop passes through the scanivalve en route to an HP A-900 computerized data acquisition center. Thermocouple data are converted to mV and read directly by the A-900 system.

The computer system has an online real time monitoring program which allows the test engineer to check compressor performance while adjusting the operating conditions (inlet pressure, temperature, flow, or compressor speed). This is accomplished by reading a sampling of key pressure and temperature probes every 10 seconds. Using this data, flow, efficiency, and work coefficient are calculated and displayed along with inlet temperature and pressure and compressor skin temperatures. By monitoring these values on the CRT, the test engineer can verify that a performance point has settled prior to data acquisition.

To read a full performance point, the operator accesses a program which actuates the scanivalve system. Four scans are read and sent to a disk file. After all readings for a given flow condition are stored, the data is scanned for faulty information (i.e., failed probes, transient response in the scanivalve system, etc.). After all erroneous data has been filtered out, the data is averaged and run through a performance routine. Final summary sheets compile the various aerodynamic parameters which are then reviewed by a test engineer. Once the point is approved, the operator will move on to the next flow point. Upon completion of an entire speed line, the computer generates the required compressor maps (flow *vs* efficiency, flow vs diffuser pressure recovery, etc.).

By studying Figure 8, the reader will understand how overall stage and component performance is determined. Pressure and temperature probes are located throughout the flow passages of the test stage. Overall performance is calculated using data from measuring stations "B" and "R." Individual component performance can be determined as described on Figure 8.

Section Summary

In summation, the single stage test rig was designed for ease of component changeout and to allow maximum access for the installation of component instrumentation. The amount of probes installed far exceeds the minimum recommendations specified on Figure 5. The data acquisition system and test center; including the scanivalves, control panels, computer equipment, software, etc., were designed to give the operator total control of all rig functions. In short, the facility provides aerodynamic researchers with an invaluable tool for gathering component as well as stage data.

The discussion will now proceed with presentations on how the SSTR has been employed to validate new designs and to assist in the development of new prediction techniques. The first example deals with the solution of an impeller problem. The second treats the acquisition of data that will be used to develop better prediction techniques for low solidity vaned diffusers (LSDs).

THE SSTR FOR DESIGN VALIDATION

The Improved Impeller

A problem associated with some centrifugal stages is a failure to generate continuous rise to surge. This phenomenon may take on a variety of appearances as shown in Figure 9. The stage to be addressed in this discussion exhibited curve 3; often called a "camel back" characteristic. Obviously, this trend can cause serious problems to an engineer attempting to design a surge control system. And, while operation in the 'drooped' area is not detrimental to the compressor, system designers or users often must treat point S' on curve 3 as surge when setting their control system.

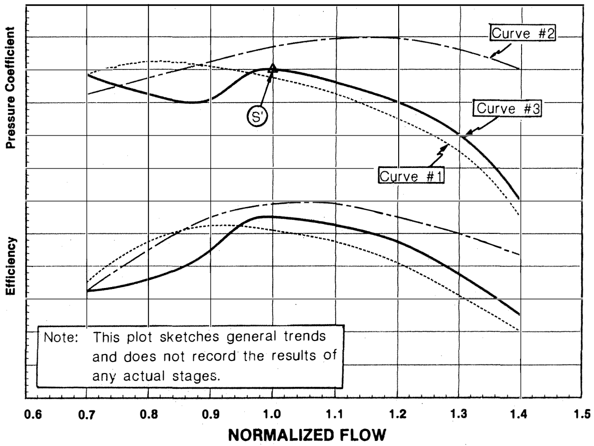


Figure 9. 'Drooping' Characteristics Seen in Centrifugal Stage Test Curves.

For the record, the internals for the stage in question were designed over 20 years ago. A sketch of the impeller involved is shown in Figure 10. The blading is three-dimensional in nature; and was defined by taking a torus section. The impeller was preceded by a standard radial flow (no prewhirl) inlet guidevane and followed by a vaneless diffuser.

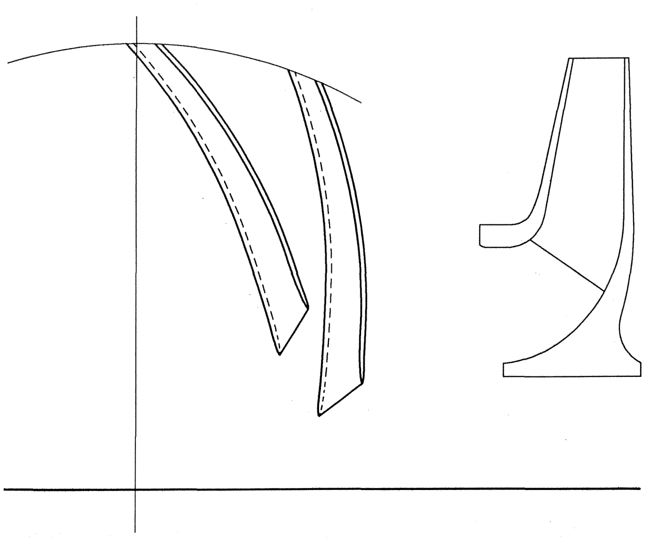


Figure 10. Axial and Meridional Views of the Original Impeller.

The "camel back" tendency evidenced itself in stage data taken during production testing. Several corrective measures were applied while attempting to correct the problem, but with limited success. For example, experience showed that installation of prewhirl guidevanes upstream of the impeller would alleviate the problem; either by improving rise to surge or by shifting the 'droop' out of the required operating range. Still, as the guidevanes did not actually eliminate the problem, a detailed analysis was performed in an attempt to isolate the root cause.

1-D And 2-D Computer Analyses

In performing a 1-D analysis on the overall stage, no unusual aerodynamic parameters were discovered. All impeller mach numbers, velocity ratios, flow angles, and incidence levels gave no indications of a problem (Table 1). Diffuser flow angles and return

Table 1. Results of 1-D Analysis on the Original Impeller.

Parameter	Value
Impeller Incidence - Shroud	**+1.0 Deg.**
Mean	**+2.0 Deg.**
Hub	**+5.0 Deg.**
Impeller Inlet Relative Mach Number Range	**0.42 to 0.75 ***
Impeller Relative Velocity Ratio Range, Ws1/W2	**1.51 to 1.86 ***
Impeller Exit Flow Angle Range	**61 to 70 Deg. ***
Impeller Exit Mach Number Range	**.30 to .77 ***
Diffuser Flow Angle Range	**57 to 61 Deg. ****
Return Channel Incidence Range	**3 to 15 Deg. ****

*** - For allowable U2/Ao range**
**** - Geometries varied for different applications**

channel incidence angles were all nominal. In short, the 1-D analysis yielded no clues as to the cause of the phenomenon.

Such was not the case with the 2-D (or quasi-3-D) results. When the impeller was analyzed using a streamline curvature program, several unattractive trends became apparent. However, even these were not obvious at first. The relative mach number distributions along the shroud and hub surfaces are shown in Figure 11. When the wheel was designed, these distributions would have been acceptable and no further analyses would have been required. However, when applying today's criteria, the apparent rapid deceleration along the driving surface is unacceptable. Such a rapid decrease in relative velocity is a strong indication that a separation

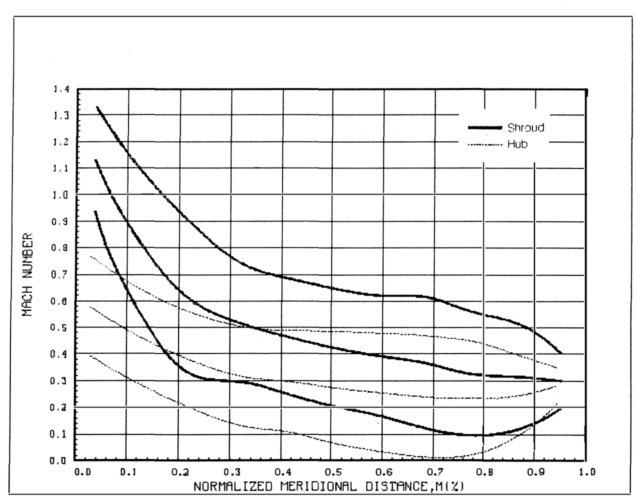


Figure 11. Mach Number Distribution from 2-D Analysis on Original Impeller.

or stall cell will form in the impeller passage. Further, the aerodynamic loading on the old impeller exceeds allowable limits as shown in Figure 12.

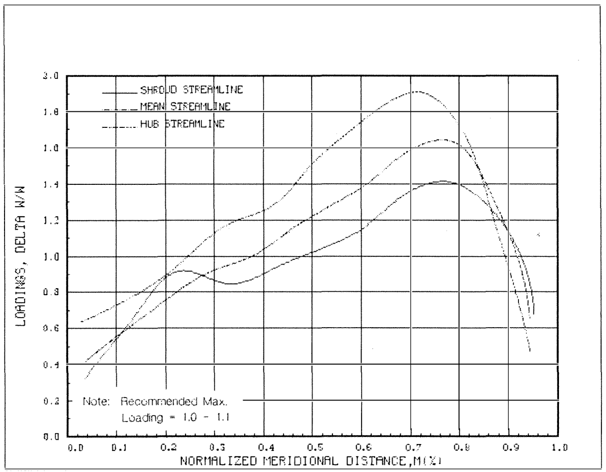


Figure 12. Aerodynamic Loading Distribution from 2-D Analysis on Original Impeller.

Since 2-D analyses of the stationary components showed no adverse patterns, an impeller stall or separation was deemed to be the most likely cause of the "camel back" trend.

Based on the preceding information, a new impeller design was completed; the intent being to eliminate the adverse velocity and loading trends exhibited by the original wheel. After several iterations through geometry generators and flowfield codes, an acceptable design was derived. The reader should note that while the older design was developed from a torus section, the new impeller contains a totally arbitrary blade shape. That is, the blading is defined by straight lines in space and cannot be duplicated by taking a section of any geometric figure (cone, cylinder, torus, etc.). A sketch of the new geometry is given in Figure 13.

The velocity and loading distributions for the new wheel are shown in Figures 14 and 15, respectively. One can see the improvements by comparing the new distributions with those of the earlier impeller (Figures 11 and 12). The area of rapid deceleration has been eliminated and the loading diagram has improved significantly.

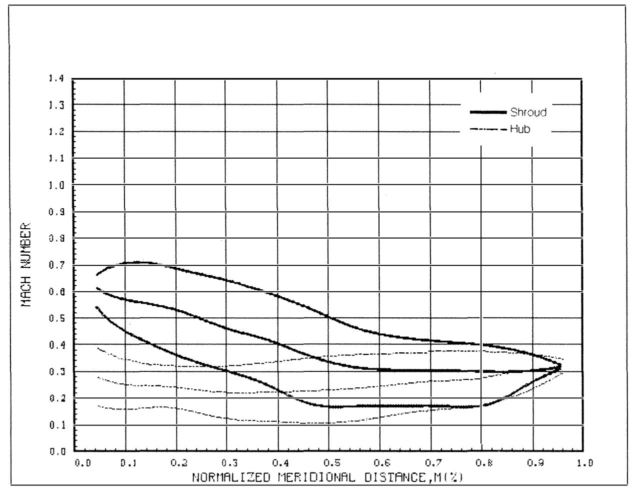


Figure 14. Mach Number Distribution from 2-D Analysis on New Impeller Design.

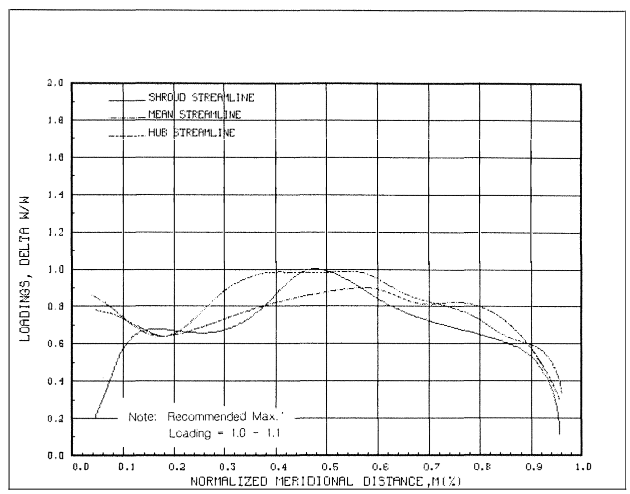


Figure 15. Aerodynamic Loading Distribution from 2-D Analysis on New Impeller Design.

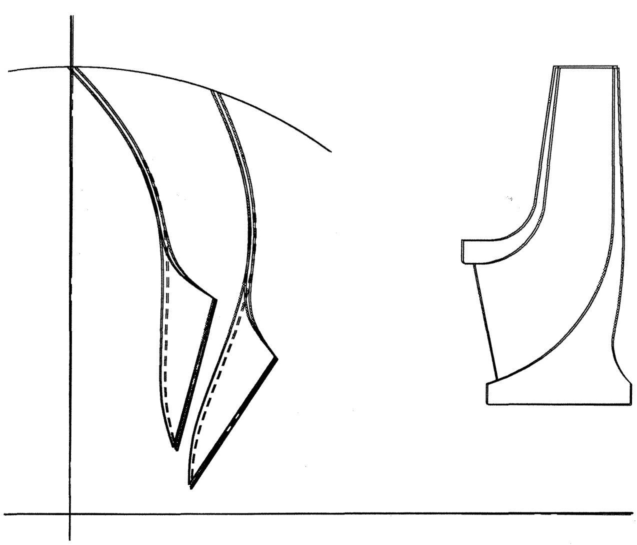


Figure 13. Axial and Meridional Views of the New Impeller Design.

Performance Testing

Having produced an improved impeller on paper, the next step was to validate the design through performance testing. A test program was developed to compare the old versus new wheels. Component data were mandatory and the wheels had to be subjected to a variety of operating conditions to ensure that the "camel back" trend had been eliminated. Therefore, development rig testing was the logical choice.

The original impeller was tested first. Four speed lines were taken (U_2/A_o = 0.60, 0.78, 0.96, and 1.06). To further define the "camel back" region, very closely spaced flow points were taken at U_2/A_o = 0.78 as shown in Figure 16. For this specific run, conditions were painstakingly monitored using the real time display to ensure that flow, pressure, temperature and operating speed

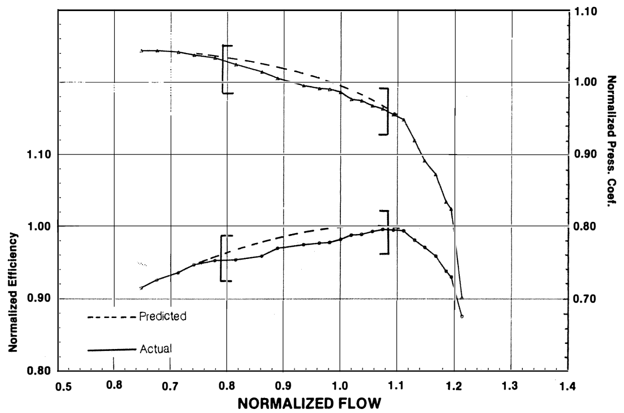


Figure 16. Test Results on Original Impeller Showing Camel-Back Characteristic.

were held constant. If adjustments were required, no data were taken until engineers were satisfied that any transients resulting from these adjustments had "settled out."

The "camel back" is easily identified but, for convenience, is enclosed in brackets. Of interest, the general location and shape of the phenomenon matched prior production stand testing. Also, sensors read no increase in shaft vibration, nor were there any significant pressure pulsations, while operating in the "camel back" region.

Having acquired a good baseline test, the new impeller was installed in the rig. Inlet conditions (flow, pressure, temperature, gas) again were monitored closely to ensure that they matched those of the prior testing. As with the earlier testing, four U_2/A_o lines were run. The performance data shown in Figure 17 was obtained at 0.78 U_2/A_o plotted against the data gathered on the original impeller. Obviously, the "camel back" trend has been eliminated as the curve now exhibits continuous rise to surge. Note also, as compared to the old design, the new impeller is somewhat higher in efficiency, head, and overload capacity. These increases are attributed to the arbitrary blading as well as the improved hub and shroud contours.

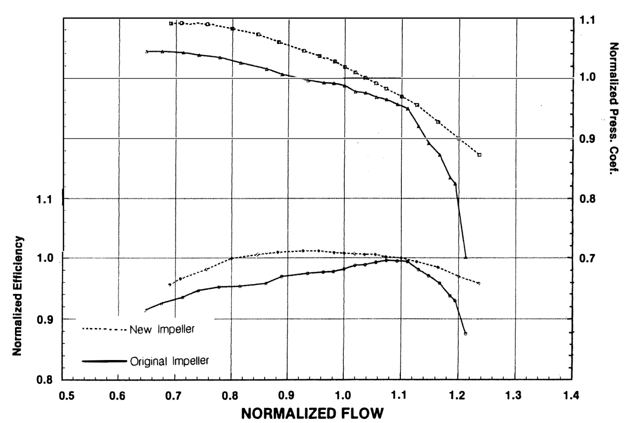


Figure 17. Test Results on New Impeller Plotted Against Results from Original Design.

Section Summary

The single stage rig was effectively used to test verify a new impeller design. The successful results confirmed the hypothesis that an adverse impeller velocity distribution was indeed causing the "camel back" problem. Further, the analyst was able to validate new design criteria, ensuring that the "camel back" phenomenon will be avoided in future impellers.

THE SSTR AS A RESEARCH TEST VEHICLE

The Adjustable LSD

The next issue to be addressed is the use of the SSTR to gather the data required to develop an improved prediction technique for low solidity vaned diffusers (LSDs). However, before proceeding, it is important that the reader be briefly introduced to the LSD and its performance advantages.

Background

Centrifugal compressor designers are continually searching for ways to improve stage performance. One common practice has been to replace the standard vaneless with a vaned (i.e., channel) diffuser. But experience has shown that the channel diffuser, though increasing design point efficiency via improved diffuser pressure recovery, does reduce the operating range (surge to choke). Full vaned diffusers are applied extensively in gas turbines where flow range requirements are limited. However, since most industrial centrifugal users require good performance over a fairly wide capacity envelope, vaned diffusers are not always desirable.

In the early 1980s, Senoo [1, 2, 3) suggested that the low solidity vaned diffuser (LSD) could be used to achieve improved efficiency and diffuser pressure recovery without any significant loss in operating range. Additionally, he suggested that LSDs could be most effective at lower specific speeds, based primarily on the more tangential impeller discharge flow angles and the typically narrow vaneless diffusers required at low N_s.

The diffuser Senoo proposed was very simple in nature as shown in Figure 18. The vanes are quite short and clearly form no geometric throat to restrict capacity. Further, the vaned region constitutes only a small percentage of the overall diffuser length, the remainder being vaneless.

To test the new concept, two LSDs were designed following guidelines derived from Senoo's works (also shown on Figure 18). The diffusers were installed (with approval of the customer) in a production unit. Minimal component instrumentation was also inserted to measure diffuser pressure recovery. (Note: the LSDs and extra diffuser probes were removed from the unit before shipment.) The results were very encouraging as both efficiency and pressure coefficient increased significantly. The vaneless *vs* LSD stage performance is compared in Figure 19. As further details regarding these results are reported [4], no additional comments will be included here.

Given the success of the testing, the LSD concept was adopted as a viable alternative to vaneless diffusers. However, questions soon arose concerning specific design parameters and application limits. Some of the unresolved issues were as follows:

- the effects of geometric considerations, such as stagger (or setting) angle, chord length, and leading and trailing edge radius ratios, on stage performance.
- the effective (performance enhancing) application range of LSDs, i.e., specific speed limits, impeller exit flow angle limits, etc.
- the effect of the LSD exit flow on downstream components such as return channels or volutes.
- how to accurately predict the effects that LSDs have on stage characteristics, i.e., under varying operating conditions, with different impeller types, etc.

Attempts were made, with some limited success, to glean information from the numerous LSDs tested in production units.

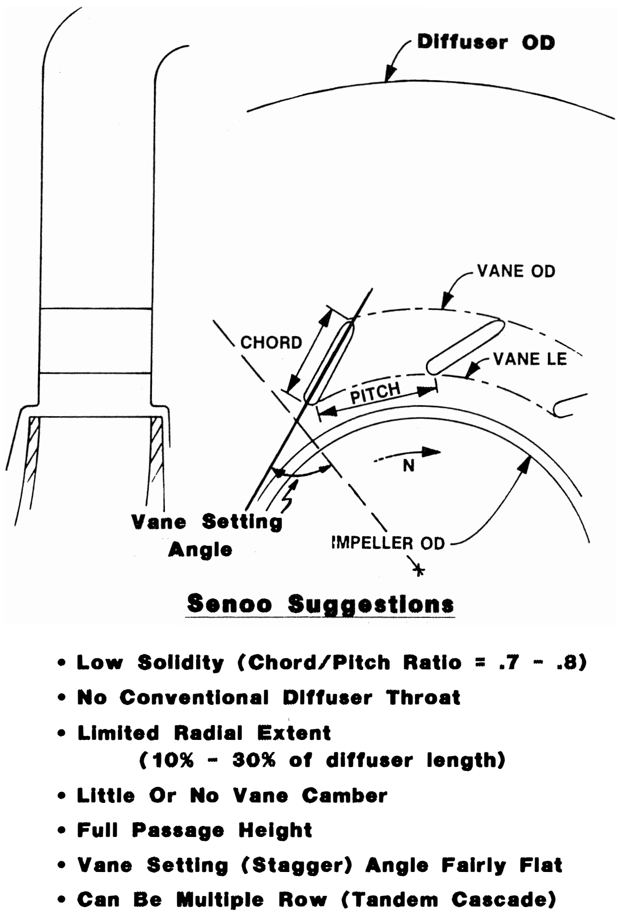


Figure 18. The Low Solidity Diffuse As Suggested by Senoo.

For example, production testing demonstrated that LSDs can be improve stage performance at high specific speeds. (Recall, Senoo had suggested low N, application.) But, the data also suggested that the magnitude of the improvement is far more sensitive to geometric considerations (setting angle, vane shape, chord length, etc.) at high N_s than at low N_s.

Other production data suggested that performance characteristics such as rise to surge, surge margin, and even choke margin, can be strongly affected by LSD geometry regardless of N_s. Such data brought into question Senoo's claim regarding the flow range of an LSD vs a vaneless diffuser. Still, it was difficult to confirm or quantify any of these effects since the production diffusers, though very similar, were sufficiently different to preclude any definite conclusions.

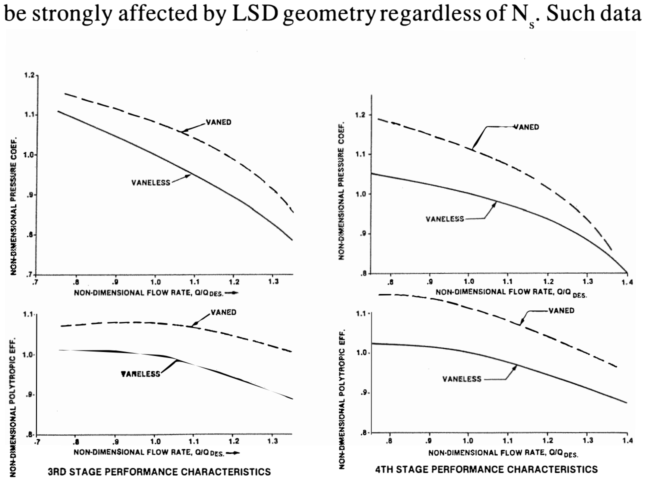


Figure 19. Test Data from Initial LSD Designs (Two Stages in Multi stage Centrifugal).

Since all of the preceding findings led to unacceptable levels of uncertainty in specifying geometry and predicting LSD effectiveness, a research project was initiated to address these concerns. The objectives of the project are to: determine the sensitivity of LSD performance to various geometries; firmly establish application guidelines (N_s, impeller exit flow angle); and generate a database that will serve as a baseline for an enhanced LSD prediction technique.

Obviously, the research must involve testing impellers of varying specific speeds, coupled with assorted LSD geometries. And, since determining detailed component performance was the justification for the research, the only practical alternative was rig testing.

The Adjustable LSD Apparatus

A review was conducted on all prior test data (and available literature) to identify those geometric parameters likely to have the greatest effect on LSD performance. Selected were stagger angle (beta 3), inlet radius ratio (r_3/r_2), and chord length. Regarding the latter, having decided to conform to the "solidity" (defined here as chord/pitch) range suggested by Senoo (0.71, 0.78), a chord length change forces a change in pitch, i.e., increase or decrease in number of vanes. Therefore, chord length and vane number had to be treated as dependent variables.

Given the preceding considerations, attempts were made to design an apparatus which would allow variation of all selected parameter. However, it soon became apparent that changing all variables within a single device was impractical (if not impossible). Therefore, a design was chosen which facilitated the adjustment of stagger angle. Any change of chord length and/or radius ratio (r_3/r_2) requires a disassembly and rebuild.

A sketch depicting the LSD vanes at various setting angles is given in Figure 20. However, no additional details of the device design will be included herein, as they are not pertinent. All the reader need recognize is that the vane setting angle can be adjusted externally; the test rig may be in operation or at rest.

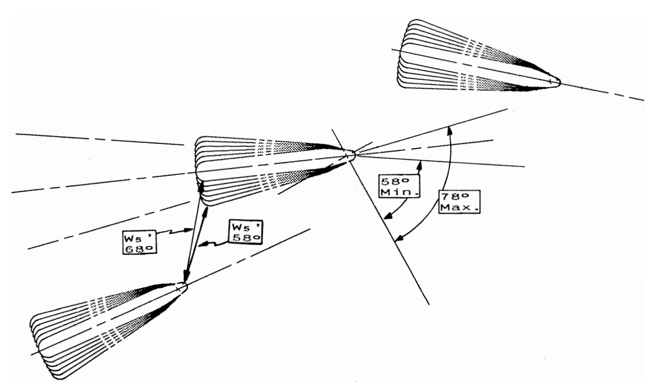


Figure 20. Schematic Showing Adjustable LSD Setting Angles.

Two important points must be made before proceeding further. First, the LSD vanes, actuating devices, and specifics regarding performance levels achieved are considered proprietary and, as such, cannot be discussed in great detail. Second, at the publication deadline, only one impeller, one diffuser vane geometry, and one diffuser inlet radius ratio had been tested. Therefore, additional details regarding LSD effectiveness at various specific speeds, radius ratios, and chord lengths will have to be addressed in future publications.

Test Hardware, Procedures, and Conditions

The impeller design used to test the adjustable LSD is also considered company sensitive, but sufficient information is contained in Table 2 to give the reader insight into the tested geometry.

Table 2. Test Rig Impeller Design Parameters and Operating Conditions.

U2/Ao	Inlet Rel. Mach Number	Ws1/W2	Abs. Exit Flow Angle	Inlet Pressure	Inlet Temperature	Test Gas
0.60	0.43	1.56	65.5 Deg.	30 psia	100 Deg. F	Nitrogen
0.76	0.56	1.69	66.0 Deg.	30 psia	100 Deg. F	Nitrogen
0.96	0.70	1.60	69.6 Deg.	20 psia	100 Deg. F	Carbon Dioxide
1.06	0.76	1.93	71.5 Deg.	20 psia	100 Deg. F	Carbon Dioxide

Tests were conducted at several speeds using different gases to obtain varying impeller tip mach numbers; and, therefore, varying impeller exit flow angles. The operating conditions and the resulting U_2/A_os and exit flow angles are also reflected in Table 2. Please note the increase in exit flow angle with increased speed; the reason will become apparent.

Information about the LSD geometry used in this test phase is exhibited in Table 3. Note that the available stagger angle range encompasses all anticipated impeller exit flow angles.

Table 3. Low Solidity Diffuser Geometry.

Parameter	Value (Range)
Leading Edge Radius Ratio	1.08
Trailing Edge Radius Ratio	1.16 to 1.24
Inlet Setting Angle	58 to 78 deg.
Exit Angle	48 to 65 deg.
W5'	1.3" to 2.0"
Maximum Vane Thickness	.32"
Chord	2.5"
Pitch	3.4"

Testing began with the LSD vanes set at nominal incidence for the impeller best efficiency point (BEP) exit flow conditions. A full line (choke to surge flow at constant speed, typically five to seven points) was taken at this setting angle. To ensure redundancy, four full scans of data were taken at each flow point. Upon finishing tests at the initial stagger angle, the vanes were rotated and another full line of data were read; beginning again at maximum attainable flow and progressing to surge. To establish trends, a minimum of five setting angles were tested at each speed.

Test Results

The results displayed in Figures 21, 22, 23, 24, 25, 26, 27, 28, 29, 30, 31, and 32 were obtained for the four U_2/A_o lines (0.61, 0.78,

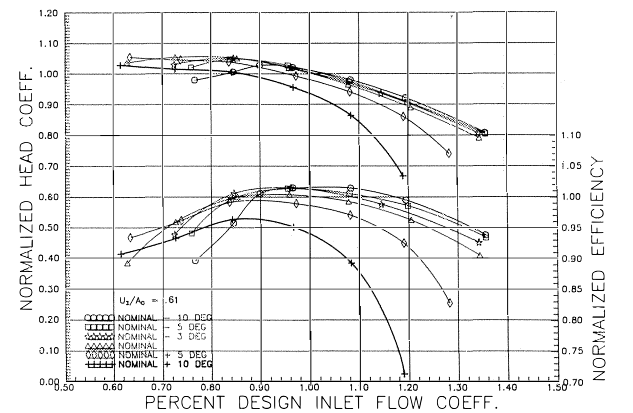


Figure 21. Normalized Efficiency and Head Coefficient Vs Normalized Flow (U_2/A_o = 0.61).

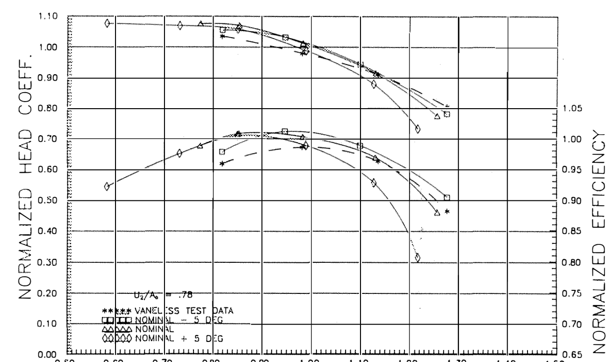


Figure 22. Normalized Efficiency and Head Coefficient Vs Normalized Flow (U_2/A_o = 0.78).

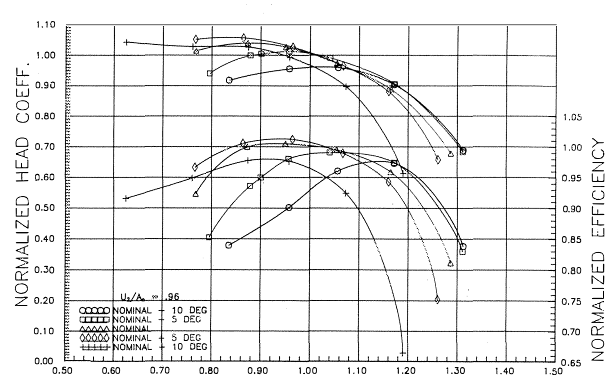


Figure 23. Normalized Efficiency and Head Coefficient Vs Normalized Flow (U_2/A_o = 0.96).

0.96, and 1.06). Efficiency and head coefficient data are shown in Figures 21, 22, 23, and 24 plotted against flow. Diffuser pressure recovery trends are shown in Figures 25, 26, 27, and 28 while figures 21, 22, 23, and 24 plot flow diffuser loss coefficient pre plotted in Figures 29, 30, 31, and 32. Note, on Figures 21, 22, 23, and 24, the performance at BEP flow for the nominal setting was used to normalize the data on each plot. The levels of pressure recovery and loss coefficient are not normalized. To clarify, diffuser pressure recovery is defined as the percentage of impeller exit dynamic (or velocity) pressure which is converted to static pressure by the diffuser. The diffuser loss coefficient specifies the amount of impeller exit dynamic pressure lost in the diffuser through friction, vortex, or other parasitic losses.

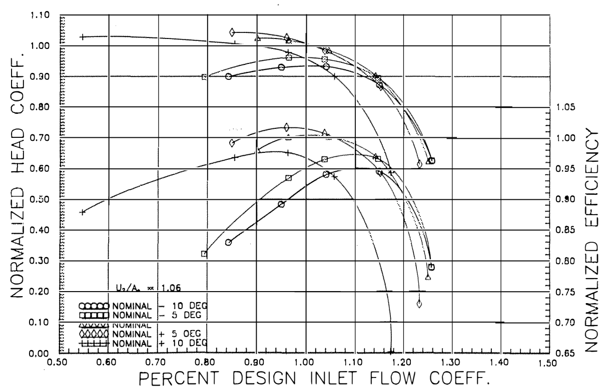


Figure 24. Normalized Efficiency and Head Coefficient Vs Normalized Flow (U_2/A_o = 1.06).

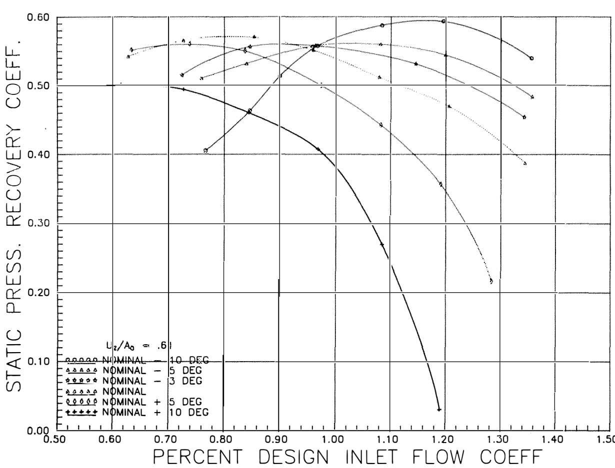


Figure 25. Diffuser Pressure Recovery Vs Normalized Flow (U_2/A_o = 0.61).

As can be seen on Figures 21, 22, 23, and 24, the variation in curve shape with setting angle is quite dramatic. BEP flow, rise to surge, stability, and choke margin all exhibit strong sensitivity tu stagger angle.

Other observations about the results include:

- As setting angle was decreased from nominal, it is very apparent that the LSD vanes were stalling. This is evidenced by the droop to surge in both the pressure coefficient and efficiency curves. Note that the surge margin is reduced even for a small decrease (two to three degrees) from nominal incidence.

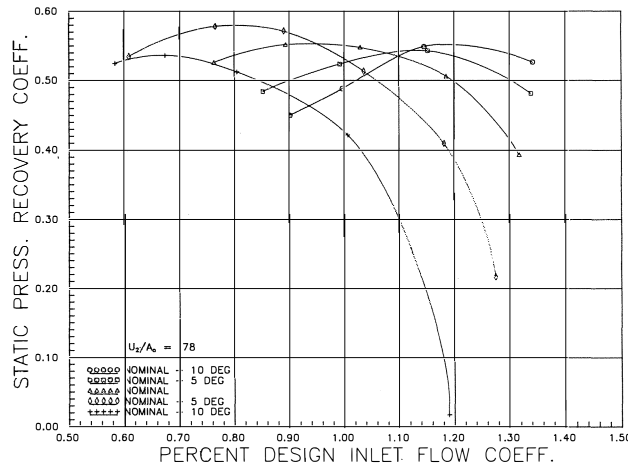


Figure 26. Diffuser Pressure Recovery Vs Normalized Flow (U_2/A_o = 0.78).

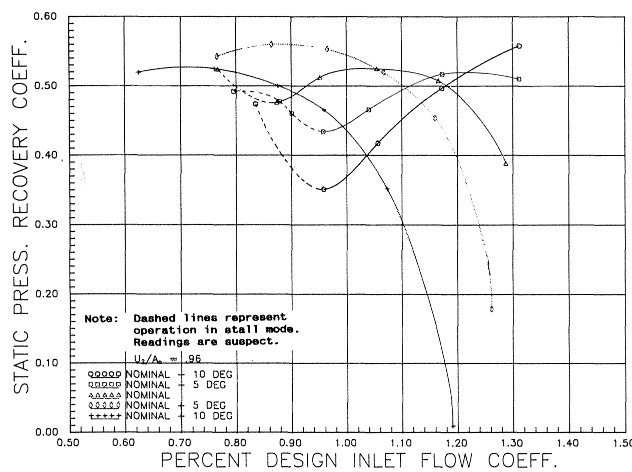


Figure 27. Diffuser Pressure Recovery Vs Normalized Flow (U_2/A_o = 0.96).

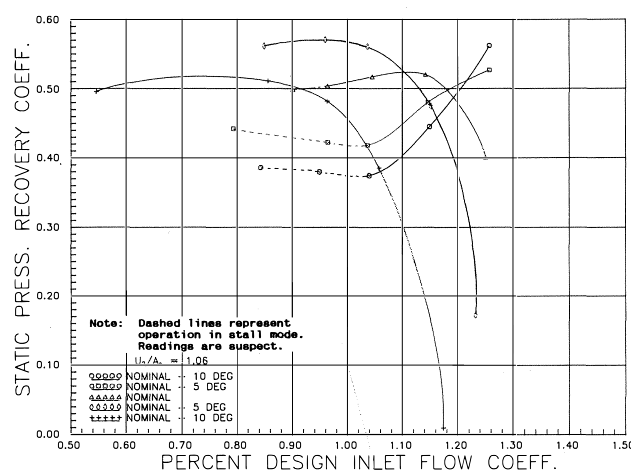


Figure 28. Diffuser Pressure Recovery Vs Normalized Flow (U_2/A_o = 1.06).

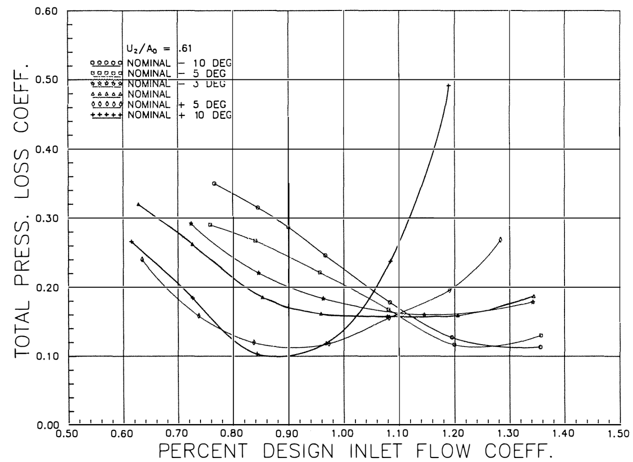


Figure 29. Diffuser Loss Coefficient Vs Normalized Flow (U_2/A_o = 0.61).

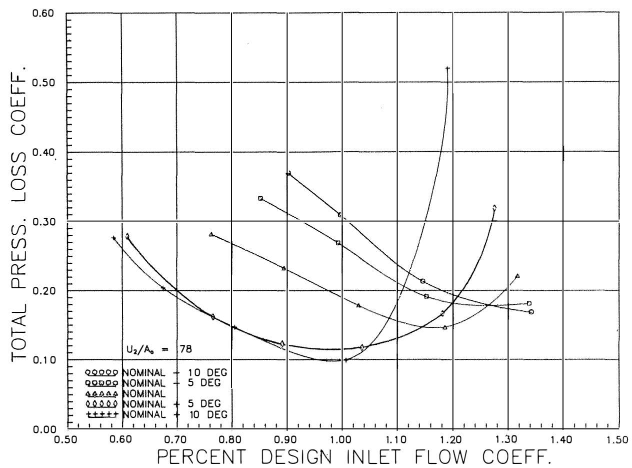


Figure 30. Diffuser Loss Coefficient Vs Normalized Flow (U_2/A_o = 0.78).

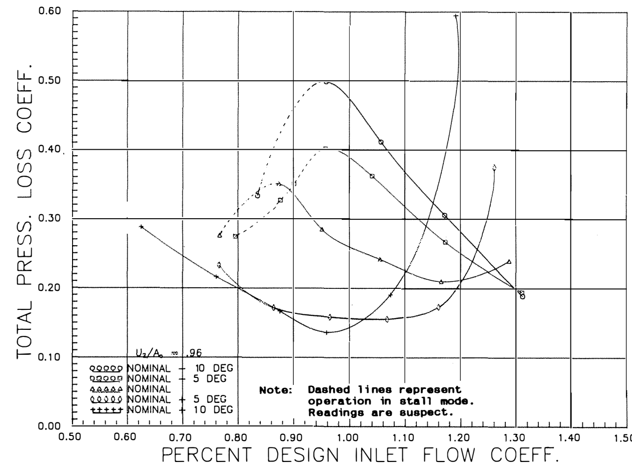


Figure 31. Diffuser Loss Coefficient Vs Normalized Flow (U_2/A_o = 0.96).

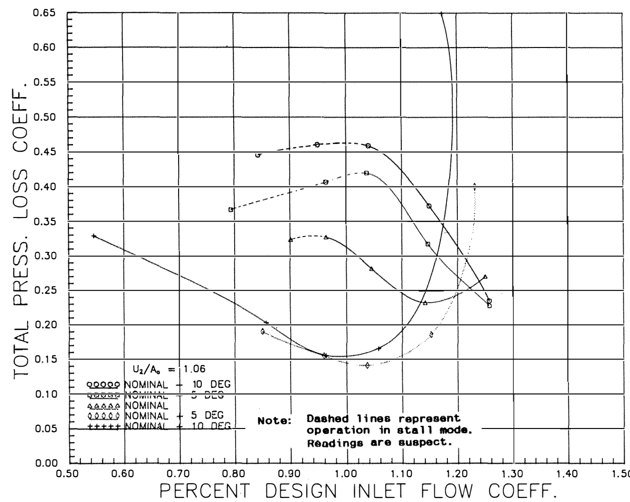


Figure 32. Diffuser Loss Coefficient Vs Normalized Flow (U_2/A_o = 1.06).

• Despite the stated absence of a true diffuser throat in an LSD, overload capacity displays a marked decrease as the stagger angle is increased from nominal. Two factors contributed to this effect. First, traverse data indicated that flow was separating from the suction surface of the vanes. The separation cells blocked a significant portion of the diffuser passage, thus reducing the effective area and limiting capacity. Second, a review of Figure 20 shows that though the LSD vane do not overlap (i.e., form a true diffuser throat), the dimension w_5' (the vane separation from trailing edge of one vane to leading edge of the adjacent vane) does decrease significantly as the setting angle becomes more tangential.

• Of particular interest is the improvement in rise to surge and stability as the setting angle is increased. A study of the pressure recovery and loss coefficient plots (Figures 25, 26, 27, and 28) shows that the peak diffuser pressure recovery and minimum loss have been shifted to lower flows as with increased stagger angle. The shift causes the stage to exhibit improved rise to surge; the diffuser performance near surge is higher than at the impeller BEP flow.

• As a final observation, Figure 33 shows the performance of the LSD at three setting angles plotted against the performance obtained during baseline vaneless diffuser testing. (As with Figures 21, 22, 23, and 24, the performance at BEP flow for the nominal LSD setting was used to normalize all data on Figure 33.) It is important to note that one can achieve the same flow range (surge to overload) with the LSD as one can with the vaneless diffuser. Further, with appropriate LSD setting angles, it is possible to achieve better rise to surge and surge margin with limited loss of efficiency in the overload region.

Section Summary

All of the above observations and others not included herein will prove vital in the enhancement of LSD design and prediction. Correlations of incidence angle to curve shape will be drawn and assimilated into various 1-D prediction codes, allowing the designer to tune the stagger angle for a particular application. Further, the testing has identified the need to consider the w_5' dimension when specifying vane geometry. Lastly, the data obtained may also be used as a basis for installing the rotatable LSD in production units (single or multistage).

As stated earlier, much testing remains to be completed and other knowledge regarding chord length and radius ratio will be gained. However, the lessons learned to date concerning LSD

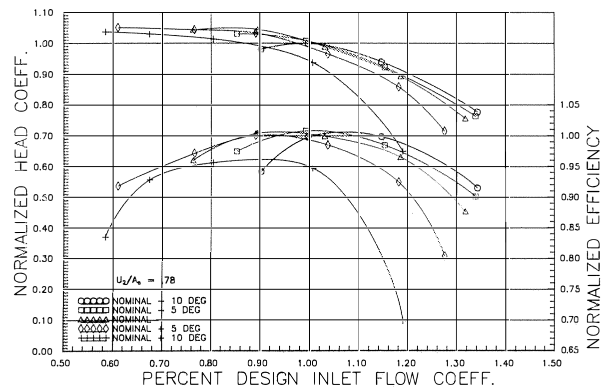


Figure 33. Vaneless Diffuser Performance Vs LSD Performance Using Various Setting Angles.

sensitivity to stagger angle mark an important step forward in the design, application, and performance prediction of low solidity vaned diffusers.

THE DANGERS OF RIG TESTING

The preceding sections have demonstrated the usefulness of a single stage test rig in: a) the confirmation of new designs and b) the development of new concepts and prediction techniques. However, rig testing must not be treated as infallible source of knowledge by researchers, designers, or, for that matter, purchasers of turbomachinery. Though the information collected is undeniably valuable and extremely beneficial, one must recognize that rigs typically operate under near ideal conditions. Such is not the case in the average industrial compressor installation.

For example, during rig testing, gas compositions are well known and samples are taken to ensure that correct properties are used. Test gases are also kept very clean (freed of particulates) by filtering, etc. Inlet conditions are monitored very closely and an optimized inlet geometry is typically employed, guaranteeing uniform impeller inlet conditions. Additionally, all stage elements (impellers, diffusers, etc.) are commonly aerodynamically matched to ensure that each operates at peak performance. Also, component surface finishes are kept very smooth and seal clearances are maintained at optimum levels. Further, tighter tolerances are frequently applied during the manufacturing of rig components. In short, every step is taken to ensure optimized performance and, as one should expect, the efficiency levels achieved are typically very high.

Conversely, while every attempt is also made to gain optimized performance in production units, practical limitations and real world operating concerns often hinder these attempts. For example, process gas mixtures often cannot be held constant and frequently contain foreign matter (dirt, rust, oil, or other process residues) which deposit on or otherwise foul flow passages. Further, normal (or transient) shaft vibrations or deflections will cause wear on seals (especially in a multistage compressor), making it difficult to hold optimum clearances; excess leakage results and performance deteriorates. Also, impeller inlet conditions will vary significantly dependent on the application. That is, in one situation, the impeller may follow "X" return channel; in another, it follows "Y." In yet another, the impeller may follow a sidestream inlet and be subjected to the associated pressure and temperature stratification; far from the idealized inlet installed in a rig. Lastly, in production compressors, it is frequently not practical nor cost effective to aerodynamically match all centrifugal stage elements. Doing so would force the compressor vendor to maintain an unmanageable inventory of drawings, tooling, routings, patterns, etc. Still, very attractive efficiency levels can be attained even without custom matching. However, one must accept that, in general, these performance levels will never quite match those obtained in an idealized test rig.

In summary, a single stage test rig can be an invaluable tool in the testing of new or existing components. However, one must keep the resulting performance levels in perspective. In some situations, it may be unrealistic to expect the same levels in production units.

CONCLUSIONS

The importance of test data in the design, performance prediction, evaluation, and validation of turbomachinery components are discussed. Comments were included regarding the different methods employed in data gathering and their limitations. Strong support was given to the use of a single stage test rig for the collection of component performance information.

Descriptions of and results from two rig test programs were presented. The first dealt with the validation of a new impeller design. The second addressed the acquisition of data necessary to improve low solidity vaned diffuser design and prediction methods.

Last, the reader was cautioned regarding the idealized nature of rig testing. While a valuable source of knowledge, users of such rigs must acknowledge that such idealized conditions will not always exist in production units.

In closing, the single stage test rig has been demonstrated to be an extremely valuable tool for gathering the data required to develop improved design, analysis, and prediction methods. The low solidity diffuser project, and other similar test rig programs, will continue to supply the knowledge and technology necessary to promote advanced centrifugal compressor performance.

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