Improving Centrifugal Compressor Performance By Optimizing Diffuser Surge Control (Variable Diffuser Geometry) and Flow Control (Inlet Guide Vane) Device Settings

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IMPROVING CENTRIFUGAL COMPRESSOR PERFORMANCE BY OPTIMIZING DIFFUSER SURGE CONTROL (VARIABLE DIFFUSER GEOMETRY) AND FLOW CONTROL (INLET GUIDE VANE) DEVICE SETTINGS.

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ABSTRACT

Surge control problems have been around as long as the centrifugal compressor itself. Many different approaches have been taken to improve operating range to surge (both in head and flow) depending on what type of surge mechanism is present in the compressor system. Compressor surge triggered by diffuser stall can be suppressed by variable diffuser geometry whereas surge from impeller stall can be eliminated by the use of variable-geometry inlet guide vanes. A given compressor duty in terms of flow and pressure ratio can be realized by an infinite number combinations of inlet guide vane / variable diffuser geometry settings. These various realizations of the same duty point have different compressor efficiencies. This paper describes a method that allows for optimal inlet guide vane/variable-geometry diffuser positioning (for a diffuser surge driven system) using two or three pressure measurements along the flow path, i.e. impeller inlet pressure, impeller exit/diffuser inlet pressure and diffuser exit pressure. Maximum obtainable diffuser pressure recovery can be used to determine the onset of surge. These maximum pressure recovery values are a function of variable-geometry diffuser setting only and are over most of the operating range independent of flow, head or inlet guide vane setting and can quickly be determined experimentally by pressure measurements. During operation, the known maximum pressure recovery value can be compared to the one determined from real time pressure measurements, and a determination as to the optimal setting of the diffuser can be made. Test data shown in this paper suggest that for the most efficient operation of a compressor, the diffuser should be positioned such that its pressure recovery value is close to its maximum. This in effect brings surge close to the operating point, but with careful control and safety factors it is surmised that stable operation is possible.

1. INTRODUCTION

Pushing efficiency numbers higher has long been the goal of centrifugal compressor designers. Of course there is also the desire for the design of stable, wide ranged compressor operation. In many instances, these desirable features are not mutually inclusive.

One method of stability control is through the use of variable diffuser geometry. In many cases the variable geometry configuration controls not only stability of the compressor system but the flow rate as well. In the case where another flow control device is used (i.e. Inlet Guide Vanes), there is the possible trade-off of performance versus efficiency for the different combinations.

The type of variable diffuser geometry used in the experiments discussed in this paper is the pipe diffuser. Salvage (1999) described in detail the performance, benefits and some geometric sensitivities of this type of diffuser. Also in his paper, a simple optimization scheme was detailed to determine the most efficient combination of diffuser/IGV settings using no measured information of the flow field or operating parameters except the actual diffuser/IGV orientation. Therefore, Salvage chose a one-to-one correspondence of IGV location to diffuser orientation based on criterion discussed in his paper. This had the effect of allowing the surge line of the compressor to be tailored to a desired characteristic, but also gave away efficient operation at lower IGV settings and pressure duty.

Brasz (2000) detailed the pressure recovery inside a variable geometry pipe diffuser. In his paper, data showing an increase in the overall pressure recovery coefficient with opening the diffuser throat is presented. This detailed description of the physics inside the variable diffuser passage supports the theory put forth later in the paper that the best operating condition is to open the diffuser as much as possible while avoiding surge.

The key to further optimization of the system is in understanding the basic flow phenomenon and using a flow measurement metric that can accurately, consistently and reliably determine which is the optimal positioning of the IGV and variable diffuser. This paper describes one such metric that shows the potential to determine the best positioning for efficient operation of a compressor at higher load points. Specifically, for the case of a compressor utilizing Inlet Guide Vanes and a pipe diffuser with variable throat geometry, a parameter describing the pressure ratio across the diffuser can be shown to give valuable insight in where surge will occur and allow for the maximum efficiency of operation. In essence this metric describes the most efficient operation of the diffuser with the objective of avoiding expensive mapping of all operating conditions apriori (flow, pressure rise for all IGV/Diffuser orientation combinations), taking highly accurate measurements installed in field applications and measuring or estimating compressor flow rate in the field.

2. EXPERIMENTAL SETUP

2.1 Description of compressors.

The compressor geometry used in this investigation is shown in Figure 1. The components of interest from inlet to exit are the IGV's, composed of a set of seven uncambered vanes, backswept 22 bladed compressor (11 main, 11 splitters), small vaneless space to a pipe diffuser, and a constant cross-sectional area collector. The impeller is 15.852 inches in diameter, with a blade exit height of 0.642 inches. The exit angle is approximately 50.0 degrees and the operational speed is 9200 RPM running at a wheel Mach number (U_{tip}/a_0) of about 1.3.

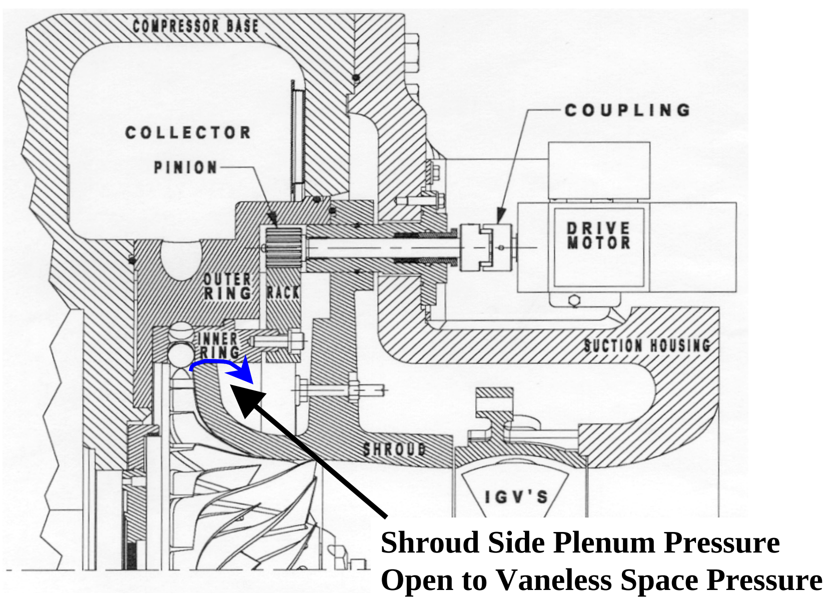
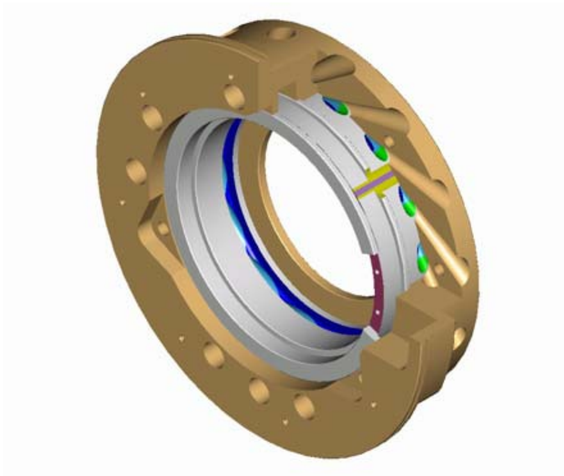


Figure 1 Compressor Geometry

This compressor is operated on a chiller system. The working gas (r134a) is pulled from an evaporator vessel, compressed, the discharged to a condenser vessel.

Pressure measurements were made in the evaporator, condenser and a plenum adjacent and connected to the vaneless space before the diffuser (see Figure 1). Pressure measurements inside this plenum can be used to get an approximation to the average pressure inside the vaneless space upstream of the diffuser inlet with minimal fluctuations and thus reduce more costly signal conditioning or expensive measurement devices.

The pipe diffuser geometry consists of 3 basic parts, a short constant area throat section (0.642 inches in diameter), a length of 4-degree divergence and then a divergence of 8 degrees. Shown in Figure 2 is a cross sectional view of the pipe diffuser geometry. The pipe diffuser is also used as a flow stability device. As show in Figure 2, a rotatable inner ring is present that adjusts the throat area of the diffuser depending on angular rotation. It is this rotation that is referenced throughout this paper as diffuser orientation.



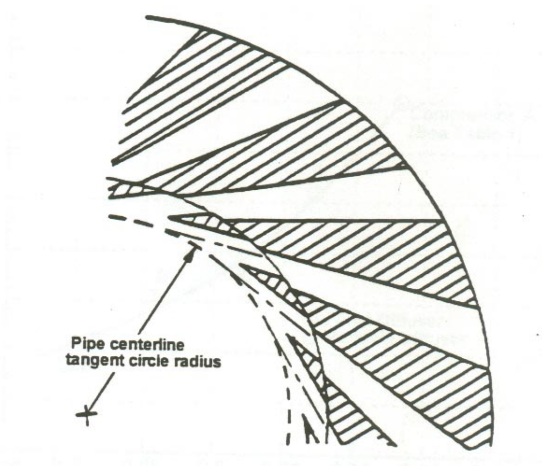


Figure 2 Variable (Pipe) Diffuser Geometry

3. RESULTS AND ANALYSIS

3.1 Overall Performance

To understand the effects of variable diffuser orientation on flow efficiency and stability, the surge line with a fully open diffuser using only IGV's as flow control is first determined (see blue diamond line Figure 3). In this figure Pevaporator is the evaporator static pressure and Pcondenser is the condenser static pressure. Also, the surge line for fully open IGV and only using the variable diffuser geometry orientation as flow control is denoted (see purple circled line Figure 3.). Between these two lines is the potentially unstable operating region of the compressor. Again, due to the fact that surge is initiated in the diffuser for this particular compressor system, sensitivity of the surge region was investigated for different diffuser/IGV orientations for the same overall pressure duty. To determine key physics and a metric to describe the optimal control of the variable diffuser/IGV setting, 10 measurement conditions are chosen. Nine measurement conditions designated by combinations of high, medium and low flow with high, medium, and low pressure operation, eight of which are inside the potentially unstable region are investigated. To compare to a non-surge flow point, one of the nine combinations (high flow, low pressure) is outside the surge region as well as another point at a much higher flow point with medium duty (well inside the stable operation region). At each of these operating conditions, different combinations of variable geometry orientation/IGV position are tested and a corresponding compressor performance points taken. This is shown in the cluster of points taken for each of the ten pressure rise/flow combinations (Figure 3).

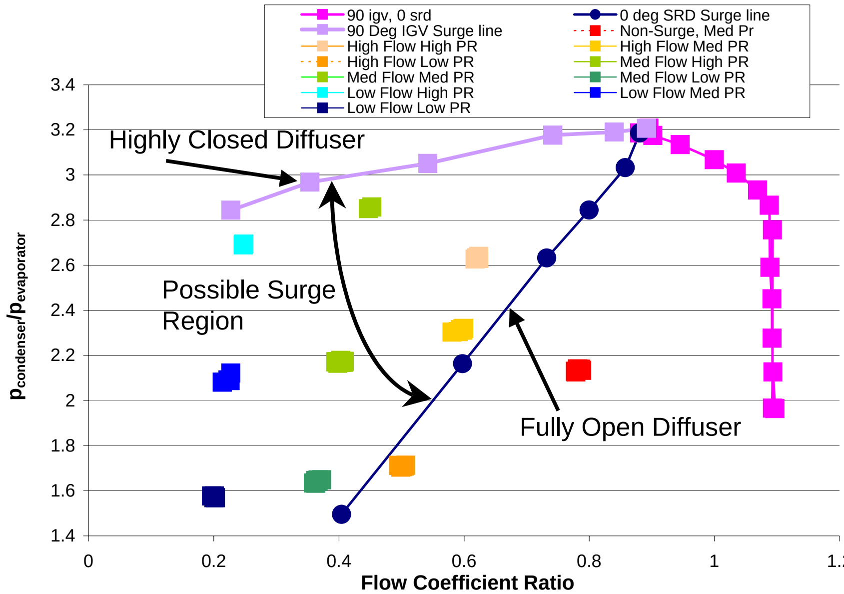


Figure 3 Performance Characteristics and Surge Zone

Shown in Figure 4 are the corresponding efficiency points. As a reminder, each of these points has a constant overall pressure ratio, but now the effect of diffuser geometry orientation can be evaluated. Each of the combinations box colors is associated with a diffuser geometry location, where more closed is toward the red and more open is towards the blue. From Figure 4 it is apparent that as the diffuser is opened, the efficiency is increased, up to the point of surge (or fully opened for the cases inside the stable envelope).

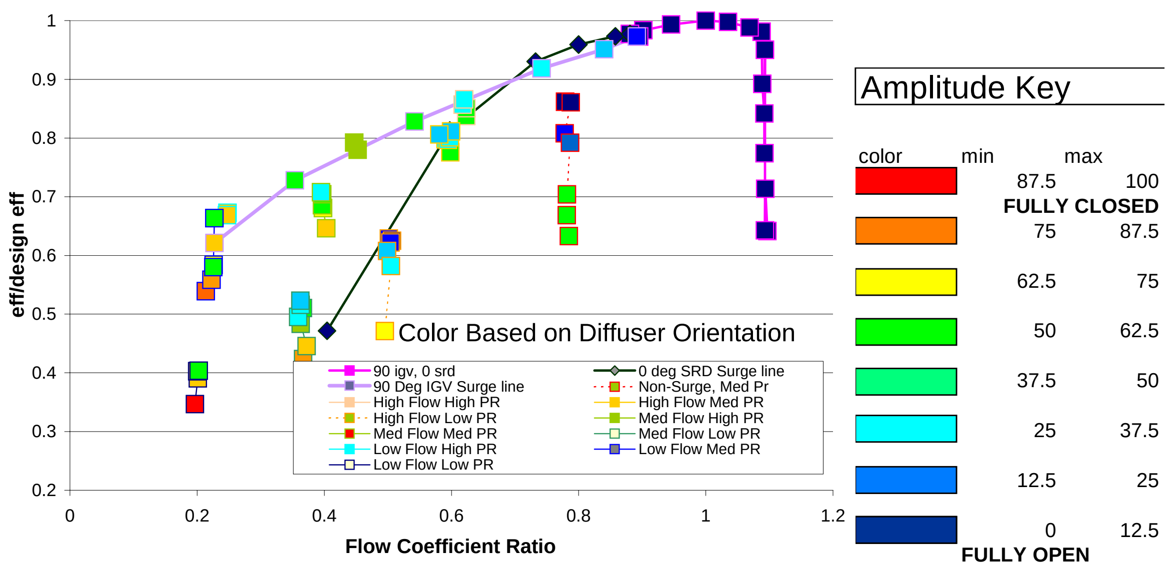


Figure 4 Efficiency of Compressor System at Zone Points, Fully open Variable Diffuser Surge, and Maximum Surge line Using Variable Diffuser

The real issue is to determine what metric will give the correct information of when maximum efficiency (nearest to surge) has occurred while constrained by the following needs: (1) a simplistic model to reduce pre-installation testing (i.e. complete compressor map for every build, diffuser orientation), (2) the fact that that the compressor performance changes over time, (3) maintaining extremely accurate instrumentation through the units life cycle, (4) the unit cannot be allowed to surge once and (5) no flow measurement is to be made.

Because the diffuser drives this system stability, one metric investigated represented the pressure ratio across the diffuser. To make this measurement, pressures were taken before and after the diffuser as described in section 2. As a reminder, for ease of measurement and to get lower fluctuating pressure measurements for a more stable average, the pressure before the diffuser was taken in a plenum chamber adjacent to the vaneless space. Although this plenum pressure measurement does describe the pressure in the vaneless space, it is an estimation of the actual vaneless diffuser space pressure and not precisely accurate. A plot of this ratio (Pcondenser/Pplenum) versus flow is shown in Figure 6.

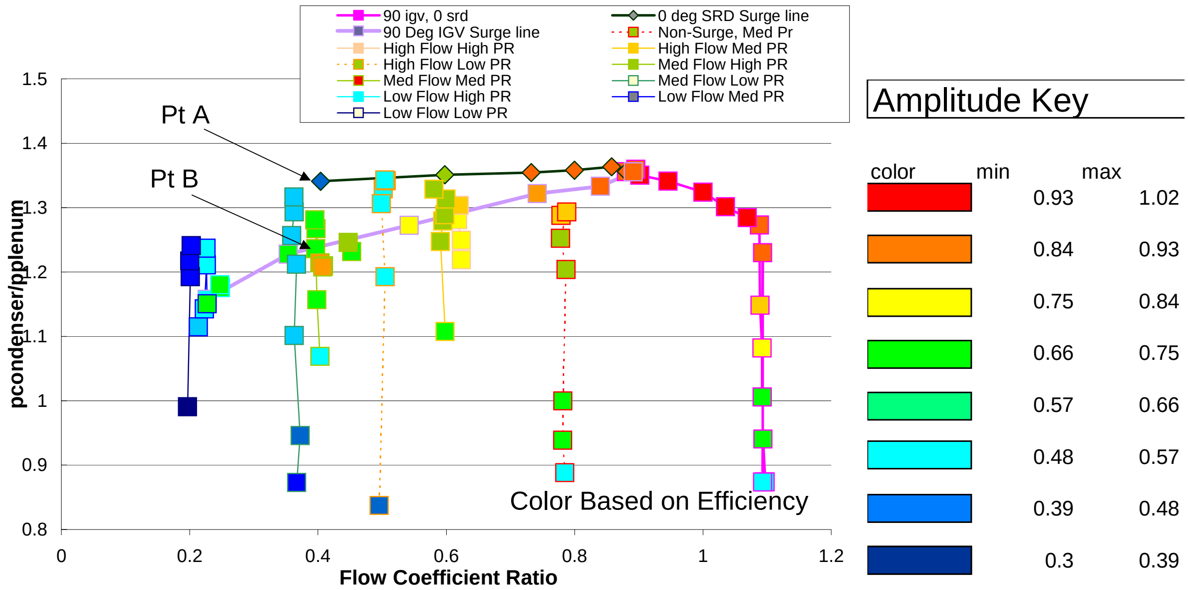


Figure 5 Diffuser Pressure Recovery Parameter Correlation to Efficiency

The remarkable aspect of the Pcondenser/Pplenum metric, is now a narrowly defined region where surge (maximum efficiency) is defined. For example, at 40% the design flow rate there is only a 7% difference between Pcondenser/Pplenum at fully opened diffuser (1.34 at Pt A) and Pcondenser/Pplenum at the closed diffuser position (1.2 at Pt B). As expected, the more open the diffuser throat, the more diffusion and the higher the efficiency (figure 6).

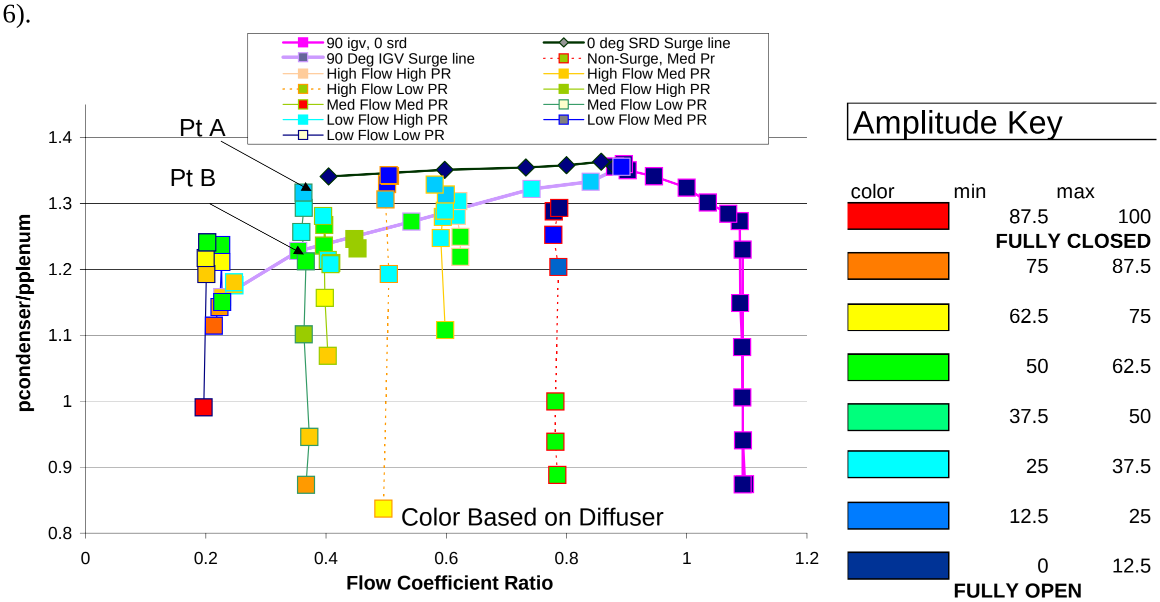


Figure 6 Diffuser Pressure Recovery Parameter Correlation to Flow Rate and Variable Diffuser Orientation

At this point, a curve fit describing the bottom surge line (purple figure 6) could be determined and used as an upper limit to the diffuser parameter during operation. This would in effect be a conservative control. To further increase system efficiency, some more information is needed.

Because there is still not a total collapse of the Pcondenser/Pplenum metric at surge (Figure 6), not all the physics of the problem has been accounted for. The correct orientation of the diffuser geometry is a prime candidate to incorporate into the analysis. To do this Pcondenser/Pplenum is plotted against the diffuser orientation (Figure 7).

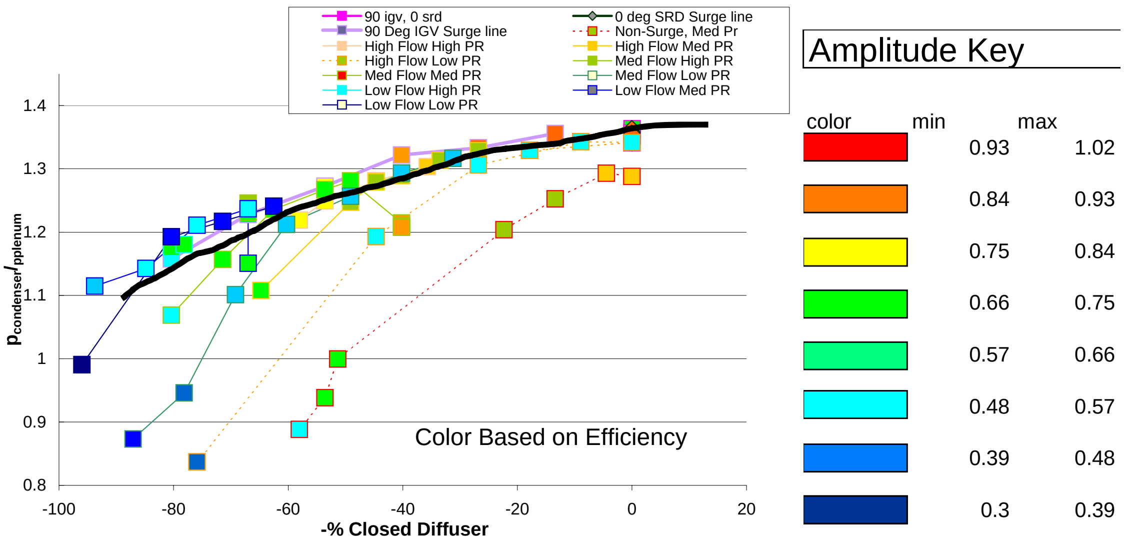


Figure 7 Diffuser Pressure Recovery Parameter Correlation to Variable Diffuser Orientation

Surge can be seen to fall along a single line (denoted black line on Figure 7). There is a define curve of maximum attainable diffuser pressure rise that is possible for any given diffuser orientation. To demonstrate the collapse further, only the points from figure 7 of maximum efficiency at the 8 test points in the surge zone are plotted along with the two surge lines (Figure 3). This, in essence, is a subset of the data shown in Figure 7 and defines the upper limit of the pressure recovery of the diffuser (Black line Figure 7).

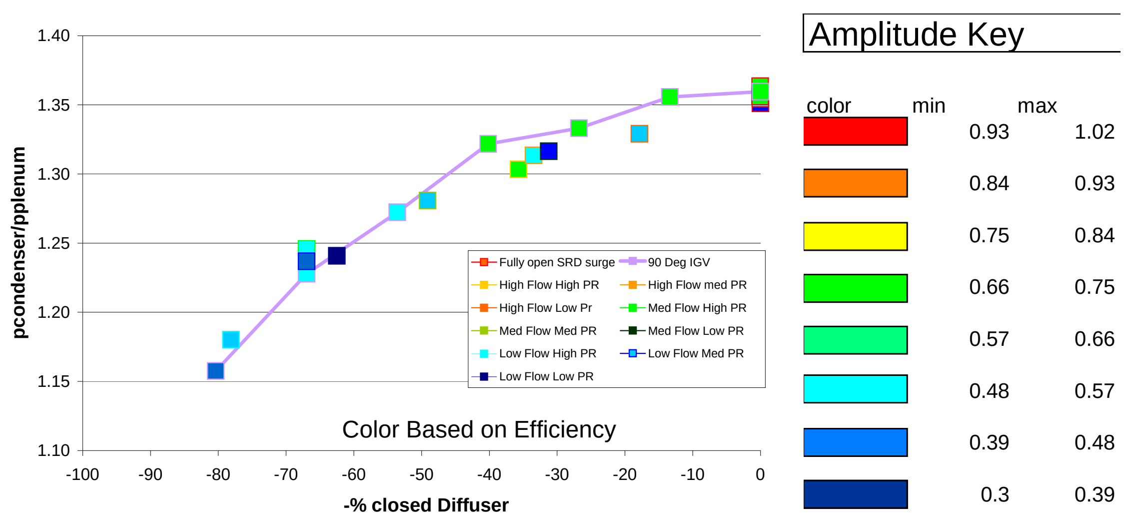


Figure 8 Correlation of Diffuser Pressure Recovery Parameter vs. Diffuser Orientation

Now it is clearly defined when maximum efficiency (or surge) will occur and a control scheme based on the current diffuser orientation can easily be devised to utilize this curve to control for maximum efficiency. As the current value of Pcondenser/Pplenum approaches the maximum value of Pcondenser/Pplenum (with an added factor of safety) for a given diffuser orientation, the system can now be stopped short of surge for maximum efficiency. The control curve can be determined by a minimal amount of test points (4-8) along any surge line. Also, a minimal amount of measurements are necessary (namely shroud plenum pressure, condenser pressure and diffuser orientation) to optimize the system.

It is also important to note that in no way is the IGV orientation expressly used to define this curve, and the surge criterion is determined mainly by the diffuser orientation. The weak function of Pcondenser/Pplenum on IGV location is shown in the next figure. This chart is a contour chart of the data presented in Figure 8 with the third dimension being the IGV position. The vertical contours in Figure 8 show that the value of Pcondenser/Pplenum is relatively constant at surge for diffuser position, irrespective and independent of IGV location.

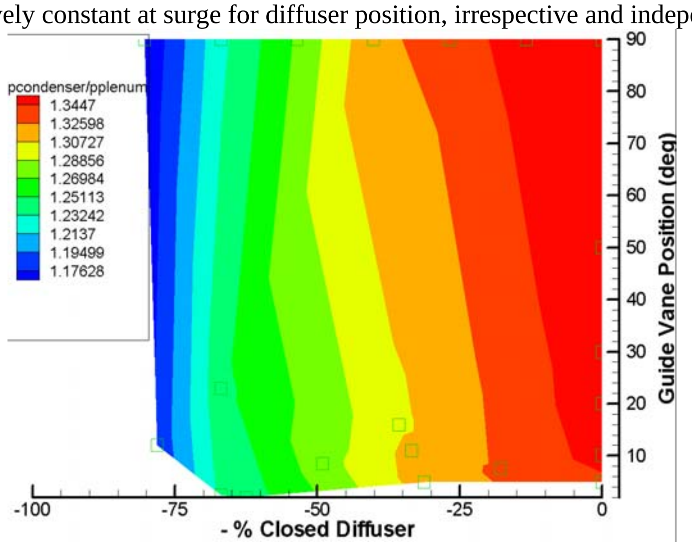


Figure 9 Effects of IGV and Diffuser Orientation on Diffuser Pressure Recovery Parameter

Although there seems to be great promise in using this metric to avoid surge and maximize efficiency, one unknown issue still must be investigated. One of the characteristics of the Pcondenser/Pplenum metric as it nears the maximum attainable value is its close approach (along somewhat parallel lines) to the maximum envelope (black line Figure 7). If there is a measurement inaccuracy or a quick transient the system could be sent into surge. If a factor of safety is added (black line of Figure 7 moved down), then a significant available diffuser opening would be ignored, and thus some efficiency improvement lost. Subsequent testing of this parameter in an actual control scheme is necessary to quantify the utility of using Pcondenser/Pplenum as a control parameter.

3.2 Physical Description

The previous data analysis showed the utility of using the Pcondenser/Pplenum metric with diffuser orientation to determine the optimal operational combination of diffuser and IGV settings in the possible surge region (shown in figure 3) for the highest efficiency. To describe the physical processes and why this metric works, the following pictorial description of the pressure rise through the compressor/diffuser system will be used (Figure 10).

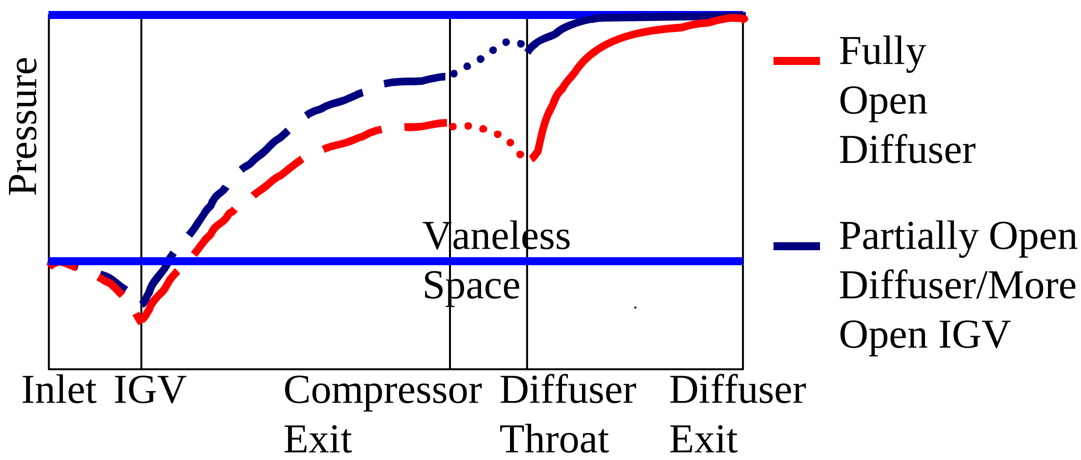


Figure 10 Compressor Component Pressure Rise Example for Two Different IGV/Diffuser settings at same overall load - Compressor/Diffuser Comparison.

Two cases are described making the same pressure duty, one with a fully open diffuser, the other with the diffuser at some arbitrary closed position. As shown in Brasz (2001), the fully open diffuser has the largest static recovery coefficient. This is depicted by the larger increase in diffuser pressure recovery for the fully open diffuser case (Figures 10 and 11). Therefore, in order to make the same pressure duty, the compressor for the fully opened diffuser case must be operating at a lower pressure rise (Figure 11), i.e. more closed IGV positioning. This means that more pre-swirl is present for the fully open diffuser case than any closed case and will adversely effect the operational efficiency.

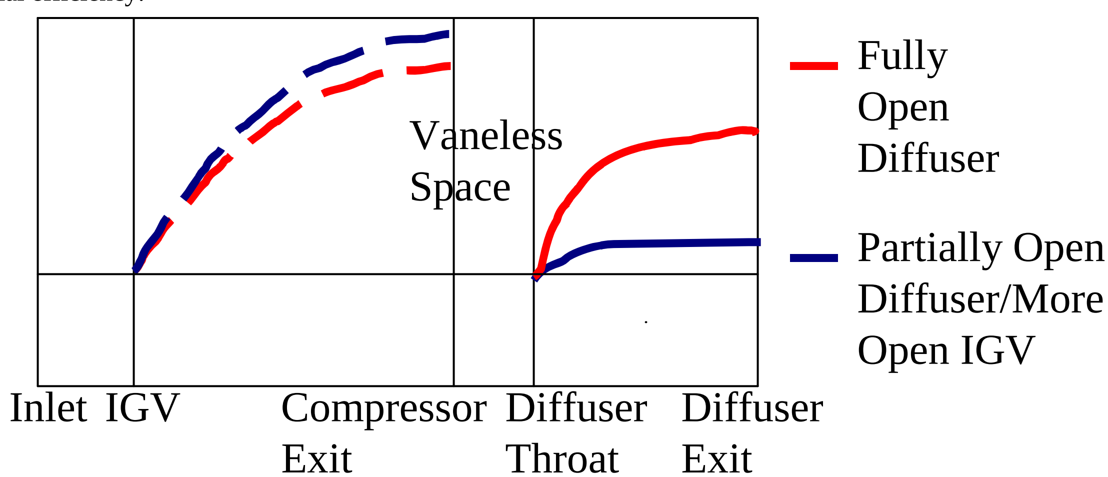


Figure 11 Compressor Component Pressure Rise Example for Two Different IGV/Diffuser settings at same overall load - Compressor/Diffuser Comparison.

Because losses of this system are dominated in the diffuser region when the diffuser is significantly closed, where pressure recovery coefficients for closed diffuser cases can be less than half that of the fully open case, the previously described small losses in efficiency in the compressor region due to more pre-swirl are more than offset by the increased losses in the diffuser.

The upshot of this is that the more opened the diffuser is, the more efficient the system becomes. Also, because stability (surge) characteristics of the system is dominated by the flow in the diffuser, both maximum efficiency and surge occur at very nearly the same point. Therefore, it is no surprise that the metric that describes the diffuser performance ($P_{condenser}/P_{plenum}$) is a good gauge of both stability and system efficiency.

4. 0 Conclusions

This paper has detailed a methodology and measurement standards that can be used to optimize a centrifugal compressor system that has inlet flow control with a variable diffuser geometry and where system stability is driven by the diffuser. The measurement metrics are the pressure ratio across the diffuser and diffuser orientation. For any given diffuser orientation, there is a maximum attainable pressure recovery value for stable operation. This is completely analogous to a maximum pressure recovery coefficient before separation in a classic parallel walled diffuser. In a centrifugal compressor system, this separation feeds into the system flow field and generates an unsteady and unstable flow.

Given that the diffuser efficiency increases as the diffuser is opened, and the most open a diffuser can be is determined by the diffusion stability (stall and surge), it is not surprising that a pressure recovery value would be a predictor of both surge and maximum efficiency.

The data in this paper suggests that a control scheme is possible that utilizes a measured pressure ratio across the diffuser to bound the operating conditions. The pressure measured before and after the diffuser are taken in plenum conditions, namely, inside an adjacent chamber to the vaneless diffuser for the upstream value and inside the condenser for the downstream value. This is done to reduce the effects of transients on the measured pressure.

For any given diffuser orientation there is a maximum attainable pressure recovery value irregardless of the inlet guide vane setting. The control scheme can be set up to insure that the diffuser operates as open as possible (maximum efficiency) but never above the maximum pressure recovery value (stall and surge).

One issue that has not been resolved in this paper is that given the close approach of the pressure recovery parameter as the diffuser positioning is being optimized to the maximum attainable value, the control scheme may be limited to lower than maximum efficiency gains given a reasonable factor of safety to avoid surge.

5.0 Authors

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Salvage, J.W., 1999, Development of a Centrifugal Compressor with a Variable Geometry Split-Ring Pipe Diffuser., *ASME Journal of Turbomachinery*, Vol. 121, pp. 295-304.