Novel Flow-Wise Grooves in Radial Turbomachines

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Abstract

Flow-wise cover grooves have been introduced into centrifugal compressors and pumps, and other turbomachines, to utilize misoriented secondary flow in the control of impeller exit profiles and to improve diffuser entry conditions. The design background for these grooves is explained, as well as their computational foundation. Experimental results are presented and the favorable impact on performance is shown. Proper applications have improved surge margin without efficiency penalty and have worked with both vaneless and vaned diffusers.

Keywords

Grooved-Covers – Centrifugal Compressors – Stability

Nomenclature

C_θ	Absolute frame velocity, tangential component
C_m	Absolute frame velocity, meridional component
r_2	Radius to impeller exit
m, M	Meridional distance from impeller leading edge
L2F	Laser 2 Focus velocimetry
VNLS	Vaneless (Diffuser)
α	Absolute flow angle from meridional reference

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INTRODUCTION

The core inspiration for the flow-wise grooved-covers (or close-coupled flow-guides, CCFG) is embodied in the laser velocimetry done in earlier investigations, (Japikse and Karon, 1989) and is a response to the characteristics of conventional tests and CFD evaluations. The stage used for this study is based on a small turbocharger compressor wheel as documented by Japikse and Karon, ibid. Figures 1a-1c illustrate the great variations found in the flow field at the diffuser inlet. In fact, the flow field is essentially fighting itself in the tight confines of the impeller blade passage as the secondary flow regime develops and causes large blockage and then large mixing losses. Considerable energy is involved in this mixing, since the C_θ values vary by 70 m/s out of a base level of just 160 m/s for the core flow (see Figure 1b). Likewise, the absolute flow angle varies by 20° out of a base of 46° (see Figure 1c). Fortunately, this process is often located near the shroud surface, and therefore should be amenable to flow-wise cover grooves for open front-face impellers. The results of this study have revealed strong problems in circumferential distortion (both vaneless and vaned diffusers), plus some very strange wall pressure patterns, especially for the highly pinched vaneless diffuser (Japikse and Krivitzky, 2016). It is believed that the grooved-cover may lessen these effects. Figure 2, below, shows an embodiment of the CCFG, Close-Coupled Flow Guide, or flow-wise grooved-cover concept. The CCFG concept is much broader than the work of this present study and is presented in the patent, US 8,926,276 B2 (and other patents pending). In principle, there are a myriad of ways that a more desirable coupling between the diffuser and the impeller can be forced; the flow-wise cover groove is one part of this process.

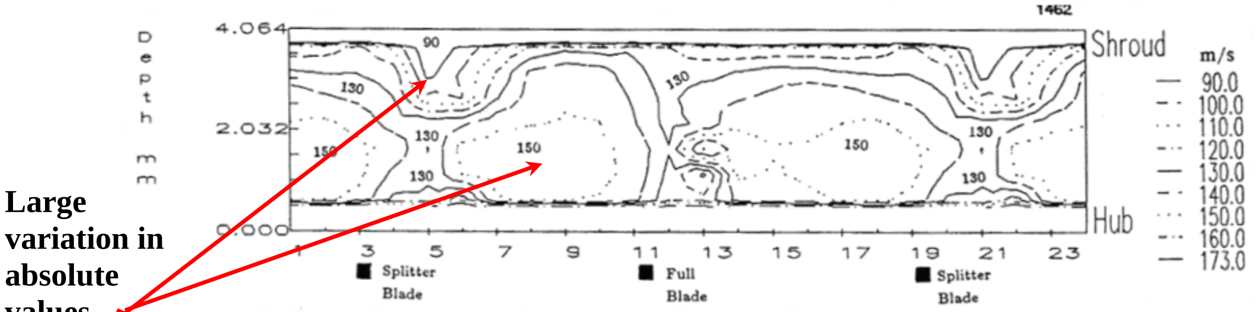


Figure 1a. L2F measurements: C_m; near diffuser inlet, $r = 1.1r_2$; 80 krpm, 0.303 lbm/sec

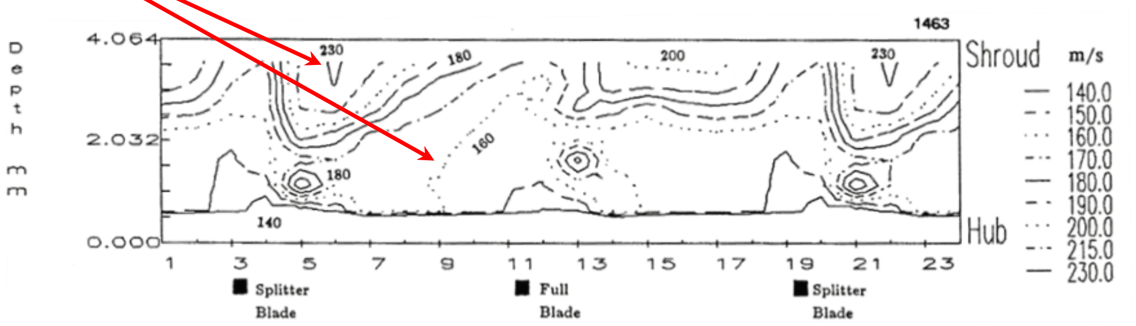


Figure 1b. L2F measurements: C_θ; near diffuser inlet, $r = 1.1r_2$; 80 krpm, 0.303 lbm/sec

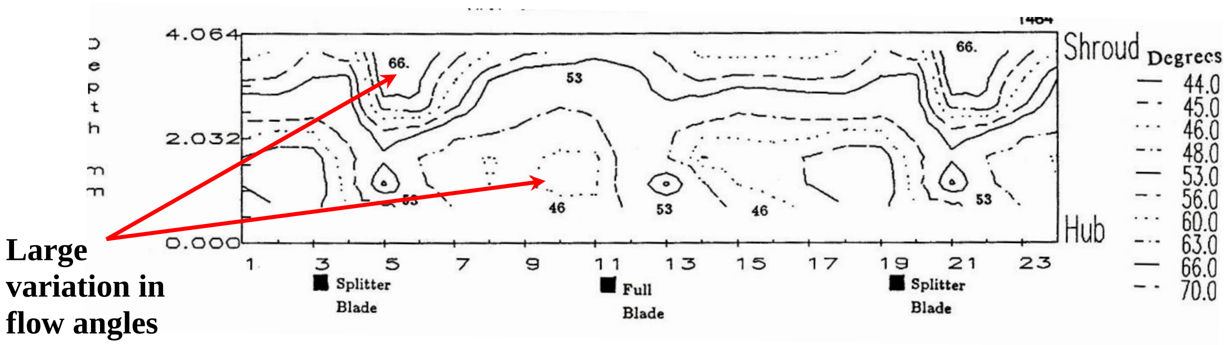


Figure 1c. L2F measurements: α; near diffuser inlet, $r = 1.1r_2$; 80 krpm, 0.303 lbm/sec

It was a bit difficult to initiate a cover design of this type, since it had never been done before. Hence, there are no useful guidelines from past experience! So the study was begun by examining CFD results for operation of the test impeller at various operating points.

1. DESIGN STUDIES FOR THE CCFG WITH VANELESS DIFFUSERS

Figure 3 shows the pitch-wise averaged values of absolute frame flow angles at various spanwise locations from a proper CFD analysis of the design case performance. The evidence of secondary flow evolution near the shroud surface is strong. The core flow settles into a pattern of nearly constant flow angle (a log spiral) of about 60° from 70% meridional distance and outward. The other (pseudo) streamtubes bend over much further towards tangential (the absolute flow angle approaches 75°) and will lead to stall conditions in the diffuser at lower flow rates. These are the streamtubes that should be trapped in the cover grooves, hence forcing them to take on a flow angle nearly the same as the core value of 60°. These errant streamtubes seem to incorporate at least 10% and probably more of the passage height based on Figure 3.

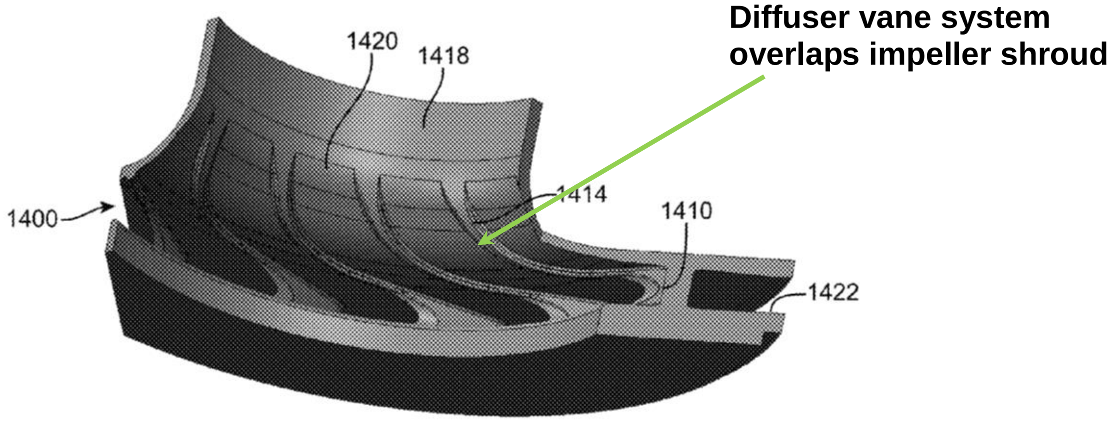


Figure 2. Embodiment of the flow-wise grooved-cover CCFG (Close-Coupled Flow-Guide); Patent 8,926,276

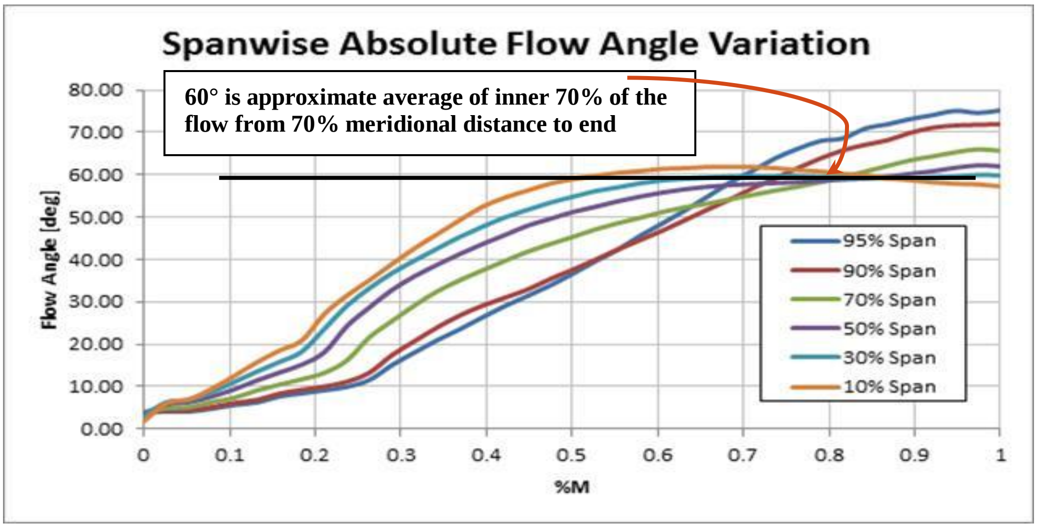


Figure 3. Blade-to-blade averaged flow angles at various % spanwise locations showing 1) crossovers at 70% - 80% meridional distance, 2) a group of angle traces that settle out to a level of about 60°, and 3) two traces that rise to higher angles; impeller at 4° incidence

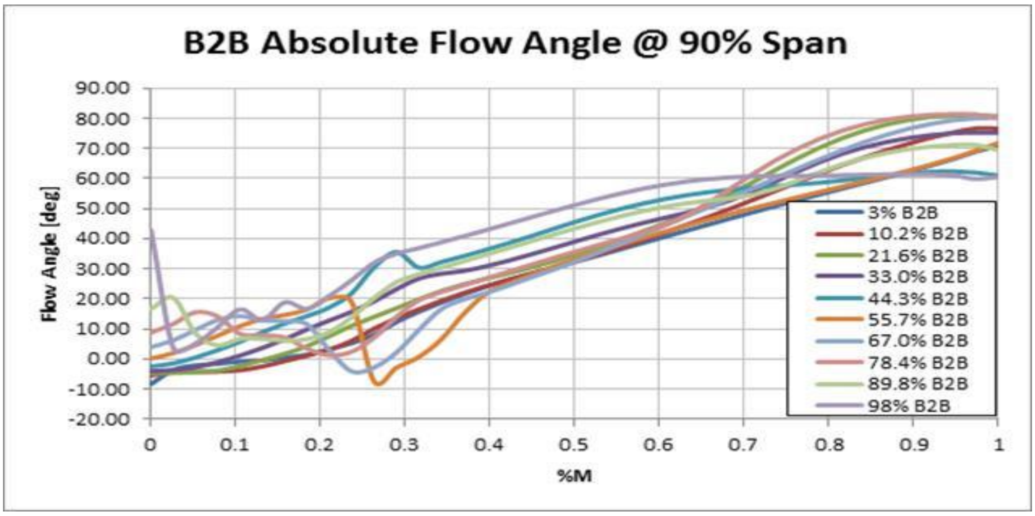


Figure 4. Blade-to-blade flow angle variations taken at the 90% spanwise location; note that these angles cross over each other at 70% distance; at this point, the secondary flow has formed into a tight passage vortex and develops a pattern distinct from the primary flow; impeller at 4° incidence

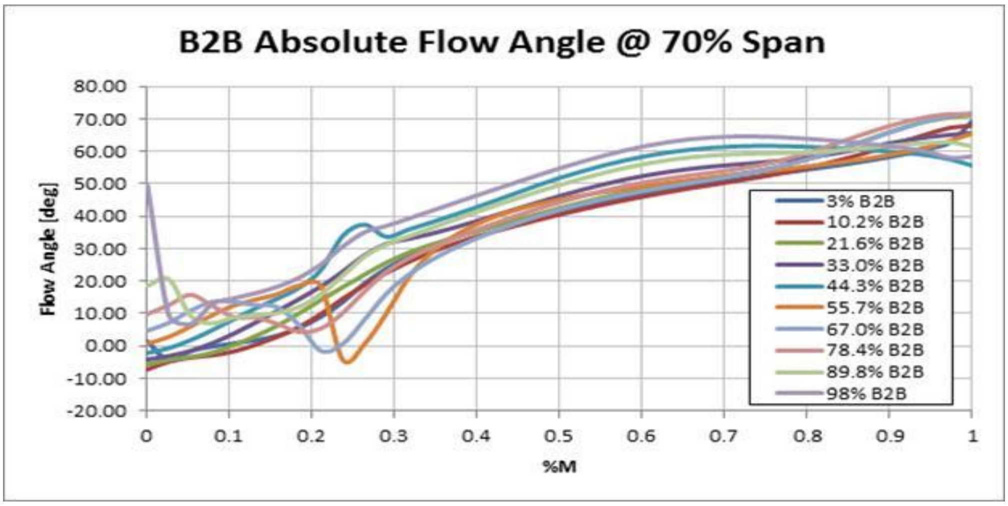


Figure 5. Blade-to-blade flow angle variations at the 70% spanwise location; same trends as Figure 4; impeller at 4° incidence

The blade-to-blade variations at two spanwise locations are shown in Figures 4 and 5. A 60° groove angle, with respect to the meridional, seems quite reasonable, as these blade-to-blade variations seem to cluster around this level. It is also important to consider what might occur at other operating points along the same speed line. For this reason, the same characteristics were considered at a point operating with 8° of incidence, i.e., a lower flow point. The results at 8° incidence showed increased angles of Figures 3 and 4 of about 5°, but the results of Figure 5 shifted up a bit less, perhaps about 3° especially in the exit region, which is of greatest interest. Examples of grooved-covers, based on this design criterion, are shown in Figures 6a and 6b, below.

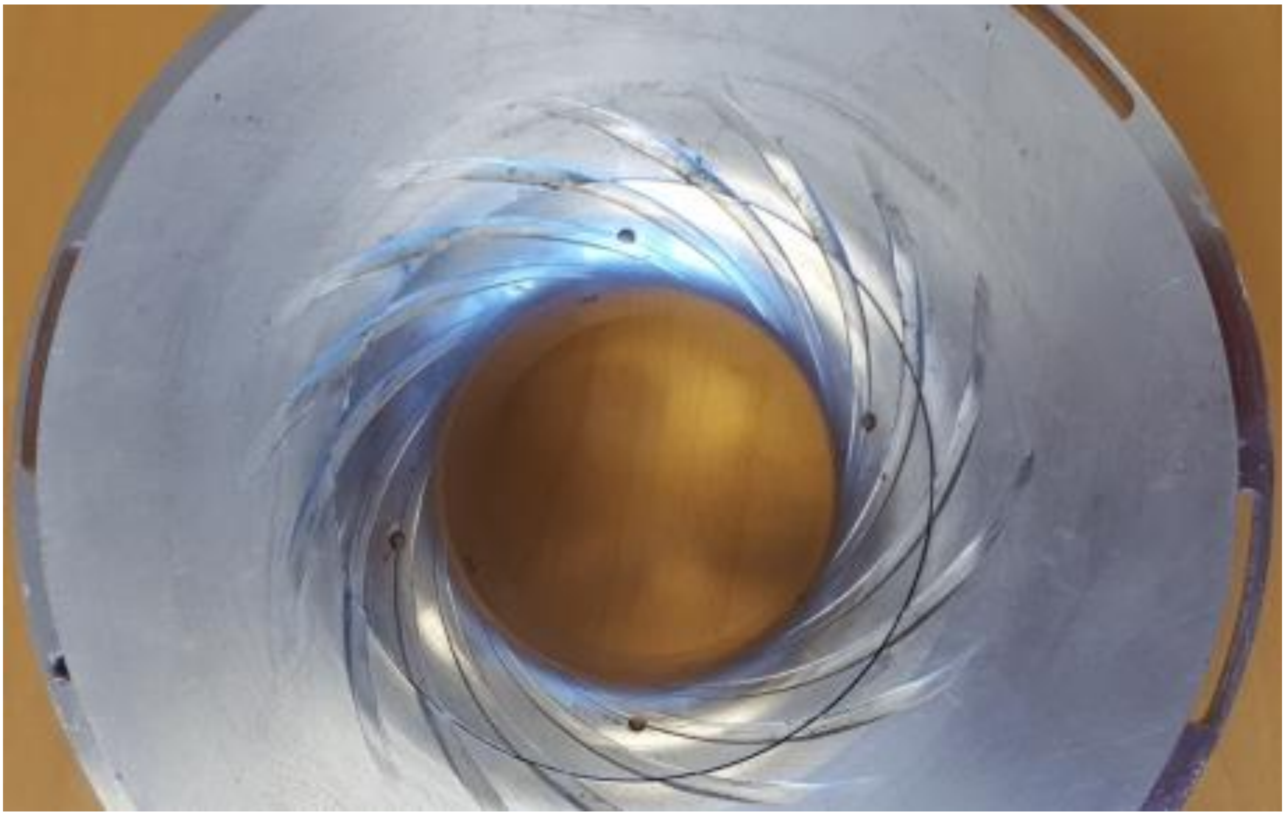


Figure 6a. The flow-wise grooved-cover (aka CCFG) showing the impeller cover below the reference circle and outside the circle is the diffuser front plate (25% front pinch configuration); grooves to 30% meridional distance past impeller

Figure 6b. The flow-wise grooved-cover (aka CCFG) showing the impeller cover below the reference circle and outside the circle is the diffuser front plate (25% front pinch configuration); grooves to end of the diffuser

2. EXPERIMENTAL EVALUATION OF FLOW-WISE GROOVED-COVER

The experimental evaluation of the flow-wise grooves was carried out as part of the International Diffuser Consortium.

The first set of tests has been conducted with the so-called 60 mm rig using a workhorse compressor impeller from the turbocharger industry.

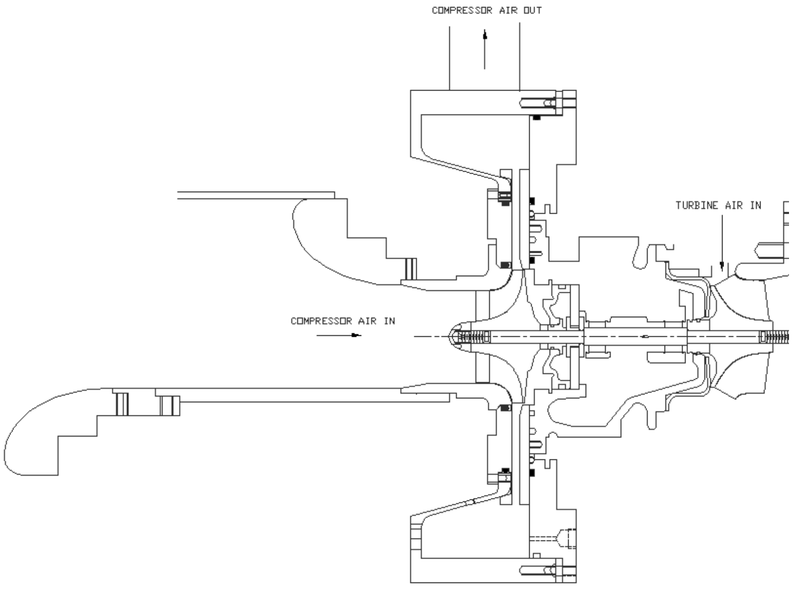


Figure 7. 60 mm test rig as used for consortium work; both the short and long inlets are illustrated in this diagram. Four exit ports from the collector are used.

Figure 7 is a diagram of the Concepts NREC (*CN*) 60 mm facility. On the drive turbine side, a set of facility air compressors supplies compressed air which passes through a heater. This air is then expanded through the mixed-flow turbine. On the load, or test side, fresh ambient air is drawn through a remote clean air inlet. After collecting in an inlet plenum chamber, the air passes through a filter and a set of inlet screens (to provide a nominal level of turbulence), and then to a first bellmouth, and finally, through a 6-inch (152 mm) delivery pipe to the test compressor. At the inlet of the test compressor, a second bellmouth, shown above in Figure 7, is used to smoothly accelerate the flow towards the impeller. After passing through the test rig, the flow enters a collector with four large exit pipes, which minimizes circumferential variations in velocity and pressure. In turn, the flow passes through a flowmeter and then through the exit throttle.

The collector and exhaust duct system are thermally insulated to ensure an accurate assessment of compressor work input. The insulation keeps the compressor-generated heat contained, providing a truly adiabatic condition. In addition, heat is provided as needed to the turbine inlet air, in order to ensure that the compressor exit and turbine inlet temperatures are within 5 to 10°C of each other, in order to further control heat transfer to or from the compressor.

3. TEST RIG INSTRUMENTATION AND ACCURACY

Instrumentation is incorporated into the test rig based on the particular test configuration, in order to accurately assess the performance of the stage and to assess impeller and diffuser component performance. The test procedures are firmly established in ISO documents and use the best practices of the past 40 years.

Compressor exit flow rate is measured by an ASME sharp-edged orifice meter with flange pressure taps. The orifice size is chosen to keep the pressure differential across the orifice between 0.5 and 5.0 psi, thus providing indicated flow rates with overall uncertainties of about 2% or less.

The rotational speed of the compressor is measured with a one-pulse-per-revolution signal, as sensed by a magnetic probe that looks at a notch in the shaft. The absolute uncertainty of the speed measurement system is typically less than 0.01% at the common frequency range.

Temperatures at compressor inlet and exit are measured using type E (chromel-constantan) half-shielded thermocouples. These thermocouples are calibrated carefully in situ, as a system, including the extension cables and specific channels of the data acquisition system, with which they are used for tests. Calibration is made against a NIST traceable calibrated platinum resistance temperature device (RTD). Absolute accuracy of the calibration is typically better than ±0.1 R (±0.05°C). When recovery factor correction and flow field variations are included, the absolute temperature measurement accuracy can be held to better than ±0.5°F (±0.28°C) on a two sigma confidence level.

Steady pressure measurements include both total and static pressures at various locations throughout the rig. When possible, several circumferential locations are used at each measurement location and averaged after data are taken. This provides redundancy if a measurement station fails during the testing, and also allows for observing circumferential variations. Total pressures at stage inlet are measured using Kiel probes. Static pressures are measured with a small static pressure wall tap, which is a 0.5 mm (0.02 in.) diameter drilled hole or smaller, passing through the surface of the wall of a flow passage. Each static tap has a clean sharp edge. The outside connections are made to nylon “spaghetti tubing", which is used to route pressure signals to the transducers in the data acquisition system. Calibration of the pressure measurement instruments is made with a NIST traceable precision deadweight tester which provides for measurement accuracies of better than ±0.1 psi (±700Pa). Zero-offset drift is taken into account by subtracting the raw zero pressure readings from the pressure data.

All of the Kiel probes, wall statics, and thermocouples, as well as the flow rate and speed counter data, are automatically monitored and recorded by the *CN* Data Acquisition System (DAS). This system has the capability of measuring up to 90 pressures, 40 temperatures, and 2 speed counter inputs. Within the DAS, each pressure is measured by its own dedicated transducer which is chosen to match the pressure range expected at that rig location. This allows for quick data acquisition. Once the test point is stabilized (based on the compressor exit temperature), the data point is then recorded almost instantaneously for all measurements. At the beginning of the testing, the DAS transducers are calibrated to an accuracy of better than 0.05 psi (±350Pa).

Rub strips are used to accurately set the impeller hot running clearances. The rub strips are blue drafting pencil leads fixed in place in the impeller cover just inboard of the impeller inlet and exit. During shakedown runs, the rub strips are set to protrude into the flow path and be worn down by the action of the impeller blades. The impeller is run to a low speed (30-50 Hz); then the compressor is shut down, and the distance the rub strips protrude from the surface of the cover is measured. The cover is then centered about the impeller based on the rub strip results. One step at a time, the impeller speed is increased and the rub strips measured. Finally, the impeller is run at close to the design speed, the rub strips measured and the cover reset to produce the desired tip clearance and to center the cover around the rotor as close as possible when it is running at high speed. The rub strips are measured to an accuracy of about ±0.0005 inches (0.013 mm). These rub tests are repeated up to a maximum speed of about 120,000 rpm.

4. GENERAL TEST PROCEDURES

After shakedown, actual performance tests are run. For the first tests in a new rig configuration, tests generally proceed from low to high speed. At each speed, the rig is run with the airflow valve fully open and the choke data point recorded. The flow is then reduced until stall is located (usually detected by an audible surge). The valve is opened slightly, and a test point is recorded just out of stall. Then the speed line is filled in with about 5 points between surge and choke, as uniformly spaced as possible. For modeling purposes, several points near and into choke are also added to the set. At each test point, time is taken to stabilize the system, in terms of mass flow, rotational speed, and collector temperatures, before recording the test point. This includes setting the speed precisely.

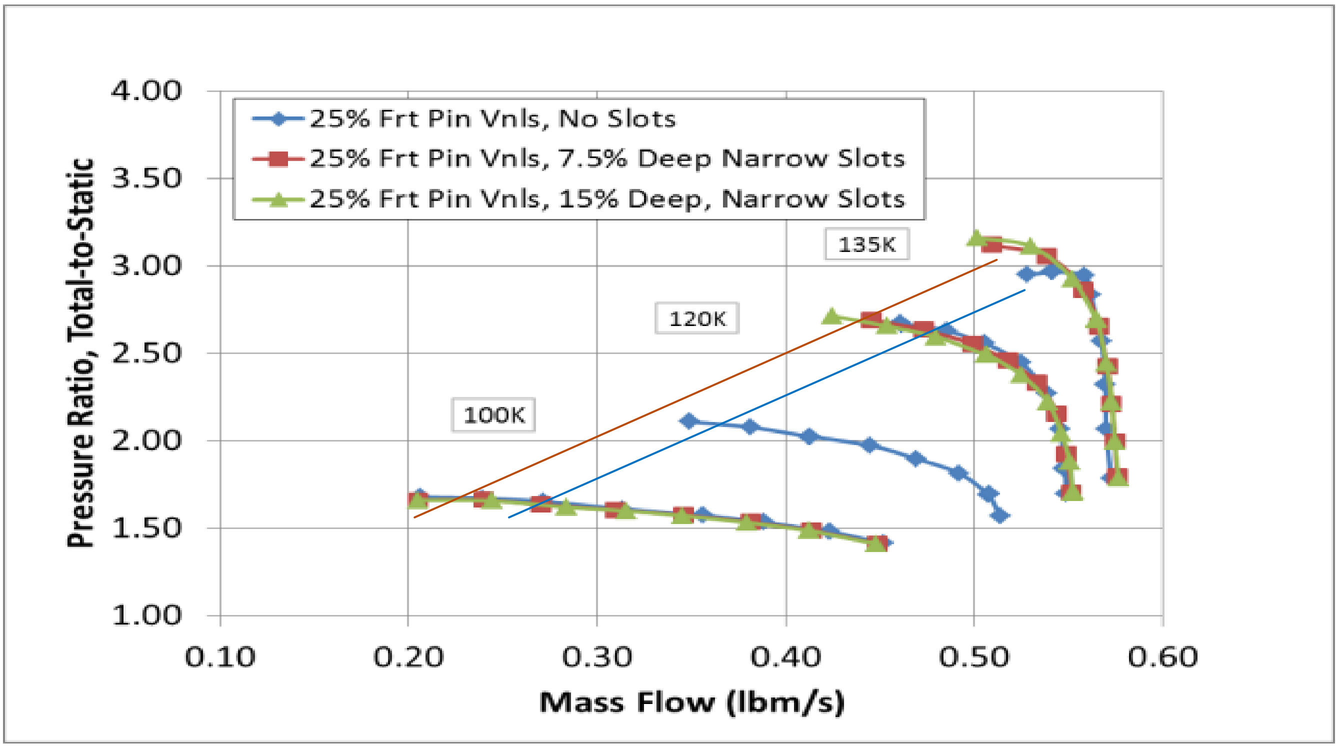


Figure 8a. Pressure Rise for the 7.5% and 15% deep grooves; Figure 6a cover

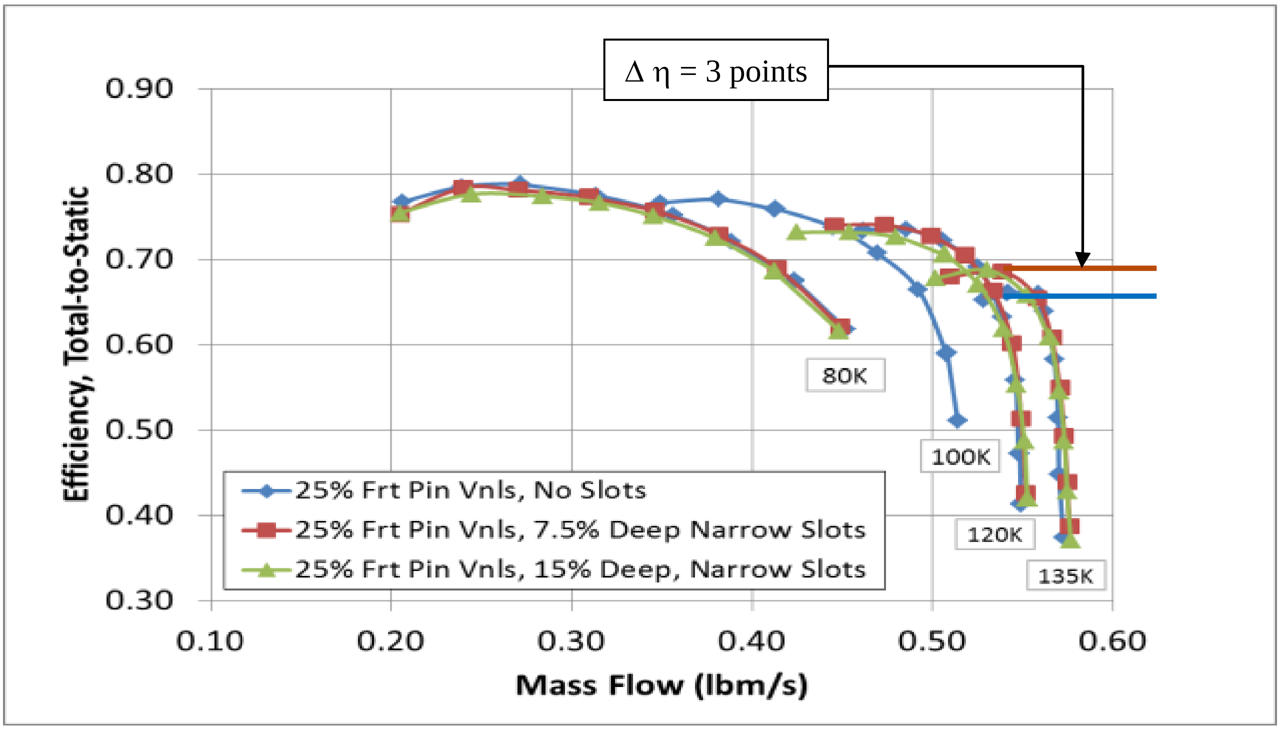


Figure 8b. Efficiency for the 7.5% and 15% deep grooves; Figure 6a cover

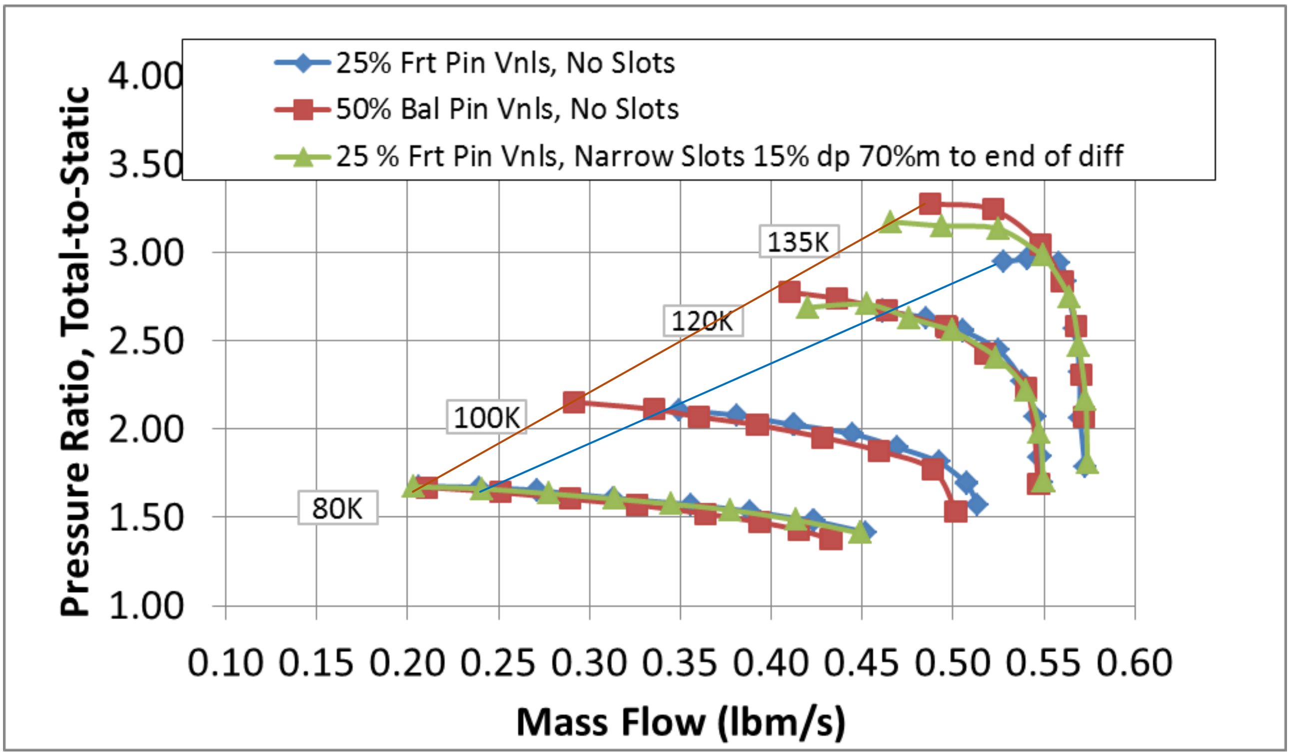
5. SELECT TEST RESULTS

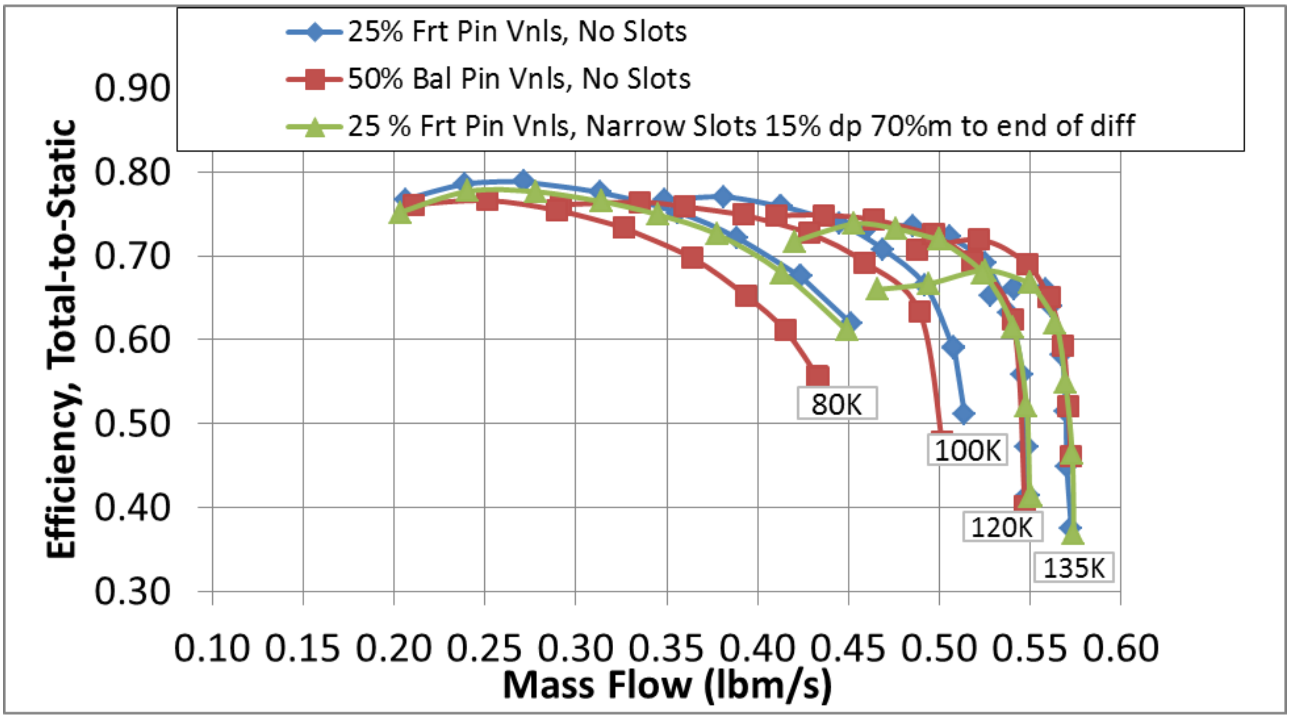
The use of narrow grooves and 7.5% (of b_2) depth worked well, as shown in Figures 8a and 8b as the reddish brown symbols. There is no real drop in pressure or efficiency compared to the ungrooved-cover, and range, efficiency, and pressure rise are all improved on the highest speed line. Comparison to the ungrooved case shows marked improvement, especially on the highest speed lines where efficiency has improved. These results are for the 25% front pinch vaneless diffuser, as used in many commercial machines.

The effect of slot ending location is found with Figures 9a and 9b, below. The results, shown with the green symbols, show a very good effect when compared against the equivalent 25% front pinched vaneless diffuser with no grooves. For comparison, the 50% pinched vaneless shows better performance at the highest speed without any grooves, but shows efficiency penalties of about 3 points at the lower speeds. The grooved-cover gives a good balance between the two ungrooved cases.

The 50% pinch case would not be a good choice for all compressors needing low-speed performance such as turbochargers and gas-line compressors, but it would be a good choice for some barrel process compressors. More importantly, the data reveal a provocative conundrum: high performance is achieved with this balanced 50% pinch diffuser at high speeds and almost as good as a vaned diffuser. Other data, not included, show considerable diffuser inlet distortion in this inlet region and somehow this pinch schedule utilizes some of the available energy. The continuous groove for 25% diffuser pinch captures essentially all of the range and most of the efficiency of the 50% pinch case *without* losing low-speed performance, but some is yet to be captured at high speed.

Initial tests with vaned diffusers have been conducted and results lead to expecting very good performance in subsequent tests.





Figures 9a and 9b. Mixed diffuser pinch levels without grooves against the best grooves (to the diffuser exit); Figure 6b cover

CONCLUSIONS

Flow-wise grooved-covers have been shown to be effective for diffuser performance, by rerouting misdirected high-energy impeller secondary flow into a useful flow angle orientation, in the absolute frame of reference, yielding an improvement in pressure rise and stable operating range and high speed efficiency. There is little or no efficiency penalty across the entire map when done correctly. Classical CFD can be used to guide the design layout.

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REFERENCES

Japikse, D., Karon, D. M., (1989). *Laser Transit Anemometry Investigation of a High Speed Centrifugal Compressor*. Proceedings of ASME Gas Turbine and Aeroengine Congress and Exposition, Toronto, Ontario, Canada

Japikse, D., Wight, S. E., (2015). Concepts NREC Technical Memorandum No. 1817, Rev 4, *High-performance diffusers with strong impeller-diffuser coupling, Phase VI Final Report,* Concepts NREC, Wilder, VT, USA

Japikse, D., Krivitzky, E. M., (2016). *Radial Stages with Non-uniform Pressures at Diffuser Inlet*. Proceedings of ASME Turbo Expo 2016: Turbomachinery Technical Conference and Exposition, Seoul, South Korea

Japikse, D., (2015). Structures and methods for forcing coupling of flow fields of adjacent bladed elements of turbomachines, and turbomachines incorporating the same. US Patent 8,926,276