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Advances and Breakthroughs in Diffusers for Compressors and Pumps: A Short History of Diffusers for Centrifugal Machines

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The diffuser plays an indispensable role in setting the useable flow range and efficiency for centrifugal machinery. Key aspects of the history and design of such diffusers are overviewed followed by a look at current, novel ideas for better systems. Improved machinery can be expected in the coming years due to decades of development work in this field.

Keywords: diffusers, flow range, efficiency, centrifugal compressors, centrifugal pumps.

1. Role of diffuser in centrifugal stages

In centrifugal stages, at least eight (8) basic functions can be involved in the performance of a diffuser; sometimes, certain ones are chosen as design targets for optimization; in other instances, all functions may be involved. The eight are:

1. Kinetic energy recovery (except high-flow choke conditions) always losing total pressure,
2. Flow regulation (good impeller may operate at low flow by matching to a low-flow diffuser),
3. A potentially significant structural element,
4. A possible flow match to next element (set the velocity triangles),
5. Strong possible interaction (coupling) with the impeller flow field,
6. Tendency to "clean up" the flow field (steadier and more uniform),

7. Reduction of radial side loads, and
8. Potential reduction of noise and vibration (or make it worse).

Just how these attributes are utilized will have enormous impact on the final stage design and its performance. They are often the key to stability, and surely to stage efficiency. These points, and many more, are background for this review and are detailed in Japikse, 1984 [1] and 1996 [2].

2. History of diffuser research

The fluid device, called a diffuser, has been around since Roman times. Rouse and Ince [3], page 28, state: "Each consumer... did not pay for the amount of water he actually used, but a flat rate for the lease of a certain discharge. The standardized size of the distributing pipe leading from the receptacle was originally taken as the discharge measure. However, it was soon found by dishonest consumers... that the lead pipes [the standardized discharge element per the lease] could be easily pounded out to yield a greater cross section." By Bernoulli's principle, one can see how the pressure rise along the device must increase while always matching the downstream exit pressure, which would be the atmospheric pressure. Hence, the pressure at the inlet to the device would drop, and a greater pressure difference along the pipe system would exist, giving a higher flow rate.

The first research into the performance of diffusers, evidently, did not occur until about 1910 in a study by Gibson [4], and while this may be the first study to examine the actual fluid dynamics of diffusers, it barely scratched the surface of a deep understanding of diffuser performance. The first mapping of a family of diffusers was achieved by Reid [5] at Stanford University in 1953, as shown in Figure 1.

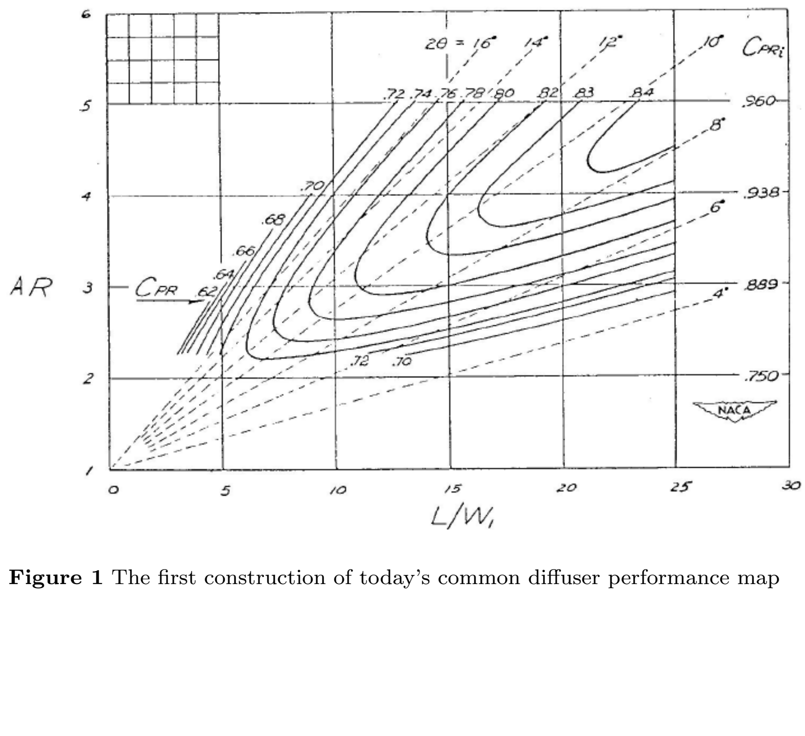


Figure 1 The first construction of today's common diffuser performance map

Reid's map presents the performance (Cp = static pressure rise/inlet dynamic head) of a set of different diffusers constructed at various lengths (L) for given inlet width (W) and an overall exit over inlet area ratio. In subsequent plots, other authors have used the symbols S or N for L, but the definitions are the same, and L is now the common symbol.

The earliest analytical modeling of the common diffuser has some surprising turns. The first published attempts may have been by Stodola [6] in about 1945, Figure 2. This figure covers a set of converging-diverging nozzles, significant in its own right, but in the subsonic, downstream portion, it is simply a conical diffuser.

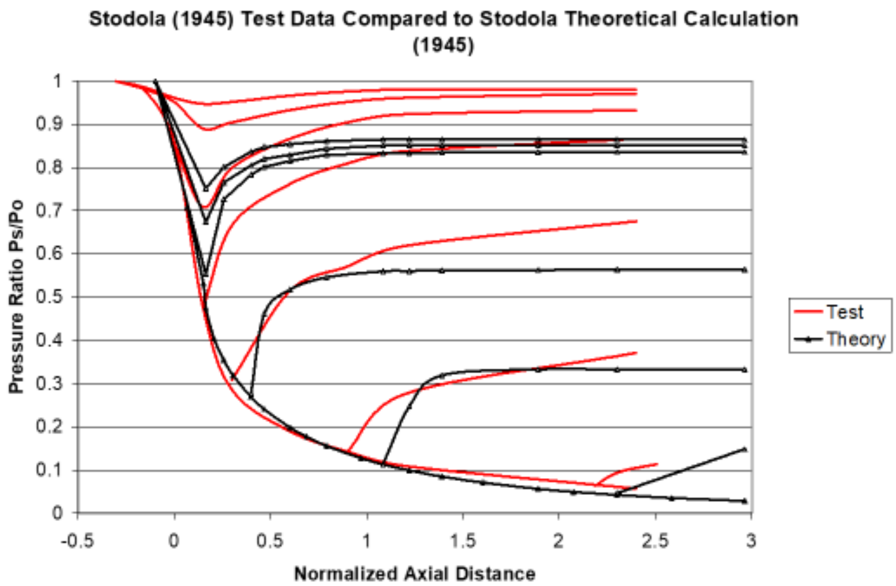


Figure 2 Performance of converging-diverging nozzles from Stodola, 1945. The subsonic portions on top are clear measurements of conical diffuser performance

The results in Figure 2 show a generally useful comparison of the pressure coefficient along the length of the nozzle subject to various backpressures. The measurements and calculations were reported separately in Stodola's reference, and it is not evident if they were meant for a direct comparison, but such a comparison has been rendered and shown here in Figure 2. One might conclude that if Stodola had made a precise case-by-case comparison at the specific back pressures, he would have done rather well with this very simple one-dimensional (1D) calculation process. Globally, he clearly did so.

The next worthy study of diffusers was conducted at Stanford University and partially illustrated in Figures 3–5. Figure 3, from Bardina, et al., 1981 [7], show a very good match between data and model for a nominally unstalled case

(Fig. 3-left) and for a moderately well-stalled case (Fig. 3-right). Figure 4 show the continuous trend from unstalled to highly stalled flow. The studies were all based on a very well-constructed integral boundary layer solution to the equations of motion using advanced understanding of the law of the wall and wake plus entrainment and stability models. Similar comparisons were then made for an entire family of diffusers with ever increasing divergence angles, well into a heavily stalled regime, as shown above in Figure 4 from Childs, et al., 1981 [8]. Finally, mapping calculations covering several flow regimes, are shown in Figure 5, from Bardina op.cit. In the first case, a fixed length diffuser is given at various divergence angles, with good modeling all the way out to deep stall. In the second case, lines of first and appreciable stall are both experimentally and analytically defined.

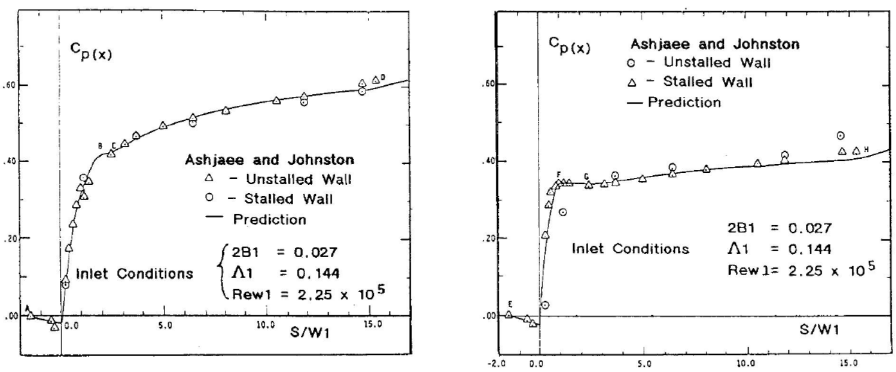


Figure 3 Unstalled diffuser modeling, Bardina et al. (left), Stalled diffuser modeling, Bardina et al. (1981) (right)

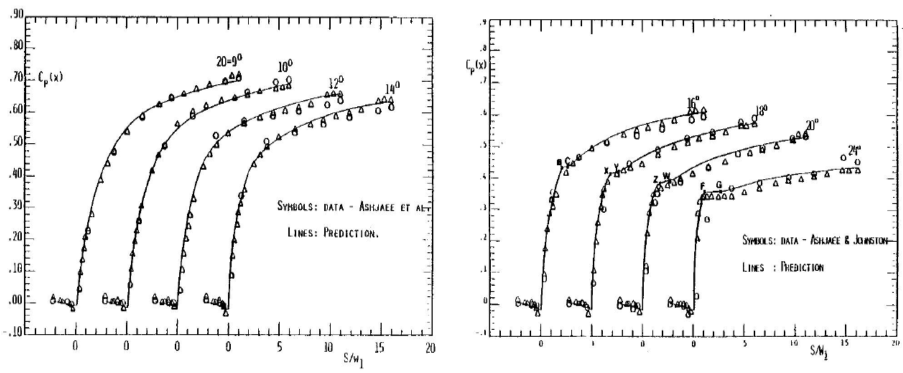


Figure 4 Diffuser modeling, low divergence, (left),Diffuser modeling, high divergence (right)

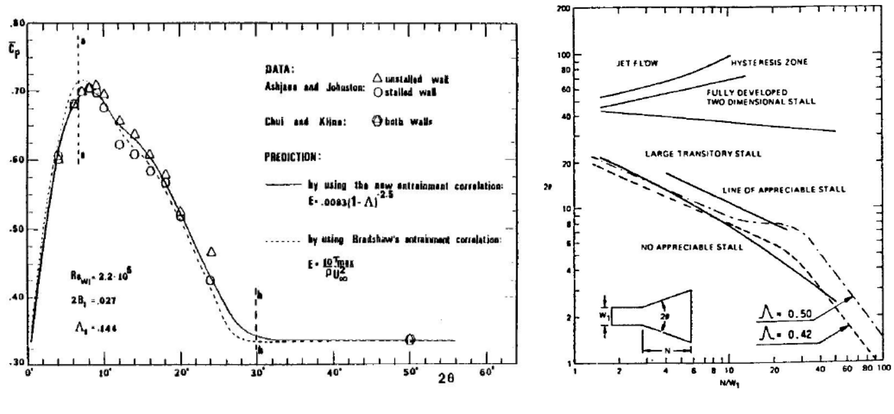


Figure 5 Modeling through multiple flow regimes (left), Modeling stall levels (right)

The next step in modeling is clearly for CFD, which is commonly applied to a full machine problem, in this case the diffusers for radial flow compressors and pumps. Unfortunately, while it should be quite practical with today's tools and a very diligent investigator, we seem to be lacking any study that has significantly surpassed the much simpler boundary layer analysis given above when it comes to modeling a full diffuser map! Nonetheless, recent studies in the laboratory and with CFD have shown great promise for stage design even if not for the simpler diffusers given above (which elements are used in complete stage layouts). Ohta et al. (2010) [9] have carefully studied the role of the diffuser leading edge vortex; Robinson et al. (2012) [10] have closely examined the impeller-diffuser interaction with powerful unsteady CFD calculations and rendered some convincing evidence of the voracity of time accurate modeling; Borm and Kau (2012) [11] tested several CFD codes and turbulence models against impeller-diffuser test data and gave insights to application, while Everitt et al. (2016) [12] gave a broad study of impeller outlet conditions and their view of how these conditions impact the diffuser performance.

3. Classes of diffusers

Diffusers of many types have been used in a great variety of pump and compressor stages. Table 1, below, is a matrix summary of the studies conducted within the Concepts NREC (*CN*) High-Performance Diffuser Consortium. There are seven generic types of diffusers, with many variations possible.

Some of these could be combined into a common genre, but the nearly independent treatment given to each in the technical literature conspires for this listing. Variations around these types are noted by degree of passage pinch (width reduction), use of flow control grooves, and so forth.

Table 1 A ranking of diffuser types and levels of research conducted at <u>*CN*</u> through consortium programs

Impeller Type	Diffuser Class Testing and Type (y = yes; n = no; p = planned; n/a = not applic.)												Gr'ved Covers	
	Vls-f	Vls-r	Vls-lt	Vls-b	Vls-pa	Ch-t	Ch-a	Ch-d	Con	LSA	HSA	Tnd	Flat	Vls	Vaned
Ns[1] = 110, pr = 3.5	y	y	n	y	n	y	n	y	n	y	n	y	y	y	y
Ns[2] = 110, pr = 3.5	n	n	n	n	n	n	n	y	n	y	n	n	y	n	p
Ns[3] = 110, pr = 3.5	n	n	n	n	n	n	n	p	n	p	n	n	p	n	p
Ns = 85, pr = 4.5	y	n	n	n	n	y	n	n	y/n	y	n	n/a	n	n	n
Ns = 55, pr = 1.8	y	n	y	n	n	n/a	n/a	n/a	n/a	y	n/a	n/a	n	n	n

Vls-f = Vaneless-front pinch	Ch-t = Channel-tangential divergence	LSA = Low Solidity Airfoil
Vls-r = Vaneless-rear pinch	Ch-a = Channel-axial divergence	HSA = High Solidity Airfoil
Vls-lt = Vaneless-linear taper	Ch-d = Channel-double divergence	Tnd = Tandem Airfoils
Vls-b = Vaneless-both sides pinched	Areas of special interest	Flat = Flat Plate LSA Equivalent
Vlspa = Vaneless-partial height vanes	Con = Circular X-section	Ns[1]: r_2=1.35" Im-1; Ns[2]: r_2=1.35" Im-2; Ns[3]: r_2=2.70" Im-3

The seven generic types are:

1. Vaneless (VLS, the most common of all, by far)
2. The low solidity airfoil (LSA)
3. The high solidity airfoil (HSA)
4. The channel diffuser (Ch)
5. The conical diffuser (Con)
6. The tandem diffuser (Tnd)
7. The flat-plate diffuser (Flat)

In developing a design, there are many common considerations, including space available, performance required, cost of product, operating range needed, and so forth. Specialized research has found that best performance requires care with the impeller exit /diffuser inlet and may be the most important area for diffuser design, with all other design parameters coming second. There is a strong probability of using Flow-wise Grooved Covers[1] as part of future designs, whether vaneless or vaned (patents involved). There is always a reasonable chance of a standardized design working well; perhaps a flat-plate variant will meet many needs. It must be said, however, that much more work is needed to learn the complete physics of impeller exit distortion.

[1] Japikse, D., "Flow control structures for turbomachines and methods of designing the same", United States Patent No. 9,970,456 B2, May 15, 2018.

4. Reliability of design

There is no doubt that CFD will play the dominant role in understanding diffuser performance in general, and specifically for centrifugal stage design. Accumulative work leads in one direction: one must be able to model unsteady, transitory behavior with transitional shear layers, and diverse inlet distortions covering velocity, pressure, and flow angle profiles, as well as turbulence and vorticity variations. Until recently, this was a very tall order to fill, virtually unobtainable as it would seem. However, today it is nearly all within one's reach: commercial CFD offers competitive options to cover all the issues listed above, and the use of Cloud computing offers essentially unlimited computational resources that readily translate into speed as well.

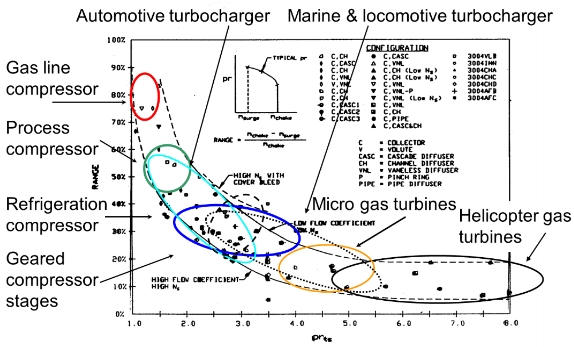


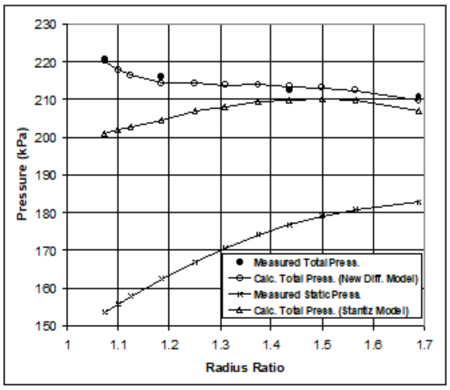
Figure 6 Range versus single-stage pressure ratio for centrifugal compressors. Diversity of diffusers is used across the range of compressor applications

However, the issue of turbulence modeling remains significant. Current models trace their roots back to studies of flat-plate shear layers with only mild adverse pressure gradients, and many turbomachinery flow fields have strong swirl, strong adverse pressure gradients, and complex turbulence and vorticity. While this will take quite a while to sort out, it is reassuring to note the early 1980s studies cited above showing remarkable agreement just using boundary layer theory to achieve broad and useful agreement. We await further work and understanding in the near future.

5. Overview of applications, current and future

An overview of the centrifugal compressor application field is suggested in Figure 6 from Japikse 1996 [13]. Stable operating range is plotted against the pressure ratio of a single-stage compressor, and while efficiency is likewise very important, it is usually range considerations that force some compromises in the efficiency goals. The figure below reveals why one set of rules does not cover the entire industry: each segment operates in a specific Mach number range and with specific market needs. Hence the technology must cover the widest set of possibilities. Today, for example, one could operate on the left-hand side of the chart only with a vaneless diffuser and on the right-hand side only with a channel or conical diffuser, and market needs would never be met if one tried to reverse these!

The details of diffuser design for all markets are vast and cannot be covered here, but a starting point can be found in Chapter 3 of the reference by Japikse (1996), *ibid.* To open the matter for discussion, the illustrations given below do not represent current design, but do point toward future advances that may change the field considerably.



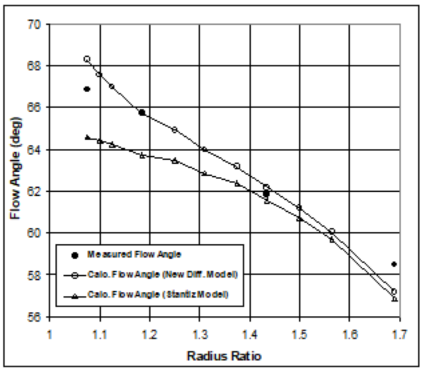


Figure 7 Comparison of modeling for a vaneless diffuser using a mixing plane method (Stanitz model) versus a coupled rotor-stator calculation using progressive mixing of the impeller exit flow entering the diffuser over considerable radial distance

5.1. *Lessons from vaneless diffuser data matching*

The ubiquitous vaneless diffuser can illustrate important issues. Figure 7 from Dubitsky and Japikse (2008) [14] shows a detailed modeling study compared to detailed data, including full flow field traverse data giving mass-averaged total pressure and yaw angle, as well as wall static pressures for a vaneless diffuser. The modeling is a time-cyclic model of the two-zone impeller model, but without sudden impeller exit mixing and rather progressive mixing into the vaneless diffuser. This modeling was carried out using the one-dimensional differential equations, common to the art, for both radial momentum and tangential momentum and various mixing and mass transport relationships for the flow after it leaves the impeller. Hence, it is a two-dimensional model, with time variation, and with first-order viscous effects. Current mixing plane CFD calculations are inferior, as they suppress all the tangential variations into a mixed-out average state. Key observations can be made: two comparisons on total pressure are given, and one of them closely matches the measured values, while the other fails even the proper trend (while each is forced to match the measured static pressures). The correct trend could only be achieved by suppressing sudden expansion mixing and using the progressive mixing with the radius. This agreement is also true for the flow angles given in the figures. This study fundamentally shows that a mixing plane solution for CFD modeling of an impeller and diffuser is highly suspect at best!

5.2. *Grooved covers*

The most complex part of the impeller exit / diffuser inlet flow problem is caused by the impact of the secondary flow leaving the impeller and entering the diffuser with a highly distorted set of velocity/pressure profiles. The so-called secondary flow leaving the impeller is weak in radial momentum, and hence, is tangential in direction when leaving the impeller, in the absolute frame of reference. Nonetheless, this flow element has considerable velocity and total pressure that currently is largely wasted. One test of this issue is to use flow-wise cover grooves near the impeller exit to re-guide some of this flow in a better direction (this technology is thoroughly patented in many countries). Figure 8 shows an example.

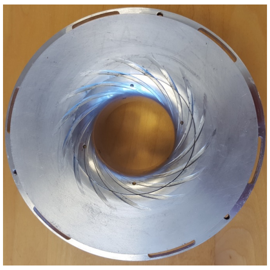


Figure 8 Illustration of a grooved cover

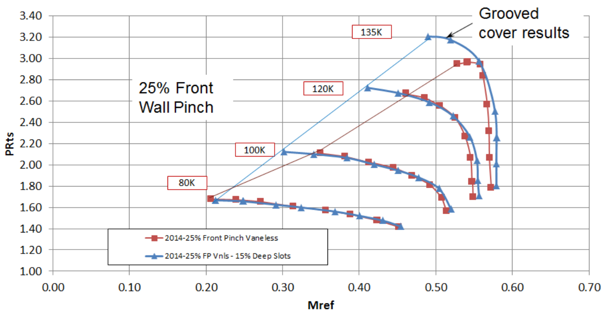


Figure 9 Characteristic improvements in stability using flow-wise cover grooves

Measured results for the grooved cover are given below, in Figure 9, for one instance, with many more already reported. The remarkable improvement in range is noteworthy.

5.3. *Interdigitated vanes*

Figure 10 shows the use of partial height vanes on one surface and another set on the opposite surface, with or without the possible use of a few full height vanes. The case in point here uses a set of partial height vanes on the shroud side and a different set of partial height vanes on the opposite surface forming an interdigitated set of opposing and offset partial height vane groupings. The results were excellent for range extension on both sides of the map, with little or no compromise in efficiency. Indeed, the new map, in Figure 11 is broader and is an easier map to work with in design. Best methods for design optimization are being worked out at this time. The final illustration, Figure 12 is a flow visualization picture showing the 'phantom effect' of the missing vane sections.

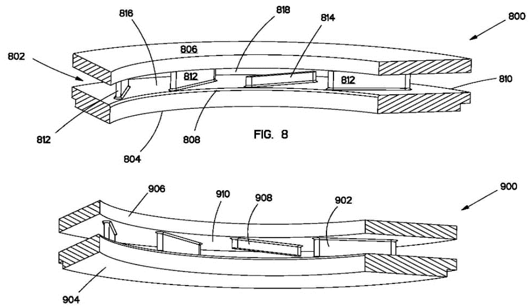


Figure 10 Interdigitated partial height vane, notional diagram

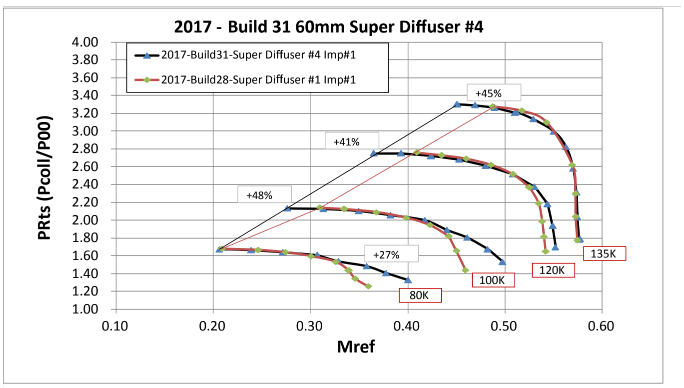


Figure 11 Performance of the interdigitated vane set

This is a picture of an interdigitated LSDA diffuser set. Where the word 'vane' appears in the picture, a partial height vane is actually present and can be seen. Where the words 'no vane' appear, there is no vane close to the surface, and the partial height vane is opposed and attached to the other surface with a considerable gap with respect to the illustrated surface. Nonetheless, there does appear to be evidence of a vane actually being there! This is a phantom image created by the pressure field of the vane on the opposite surface; there is a good flow-guiding effect, even though the vane is well cut back. This method has proven useful in extending the range of the stage with little or no adverse impact on efficiency.

Figure 12 Illustration of the 'phantom' diffuser vane effect showing clear flow guidance

6. Future of advanced diffusers

Is it possible to achieve 80% diffuser recovery in industrial design? This is a very difficult question; it appears as though no one has ever broken this threshold in a centrifugal machine, but the question is greatly complicated by the difficulty of measurement. For more than five decades, this author and colleagues have been measuring p_2 and then modeling p_{02} with conservation equations, within sensible uncertainty bounds. Recent research has shown that precise measurement of p_2 is more complex than previously thought. By increasing the circumferential array of taps, the distributions of pressure measured are illustrated in Figures 13 and 14. Figure 13 is for the 100% speed line for a 15-vane LSA diffuser operating at an angle of attack of 0°. Figure 14 is for a similar 15-vane flat-plate diffuser operating near design speed, but at a -4° angle of attack. In the first case, the harmonics and phase angle shifts are nominal and easily understood: the vane count of 15 is evident, and so is the first harmonic, indicating some circumferential distortion. For the second illustration, many harmonics are contributing to variations in amplitude and to phase angle adjustments. These may all be completely correct, or some of them could be false interpretations of the data due to aliasing errors while attempting to pursue the Fourier analysis.

Only part of the problem of obtaining good diffuser entry static pressure is illustrated above. Additionally, it has been learned that the distortion along the front face of the compressor is often different along the rear face. Hence, the problem of obtaining precise measurements is truly complex, and the ability to evaluate the inlet conditions to a compressor or pump diffuser is not fully in hand at this time. Inadequate methodology still dominates the issue of establishing with confidence the true values needed to assess the level of Cp to be found in any compressor or pump stage. We could easily be in error by multiple points of recovery working with the best measurements available, even now.

So, are we close to achieving 80% recovery in an applied stage diffuser? Maybe, but refinements in both design and measurements are needed before the claim can be firmly established.

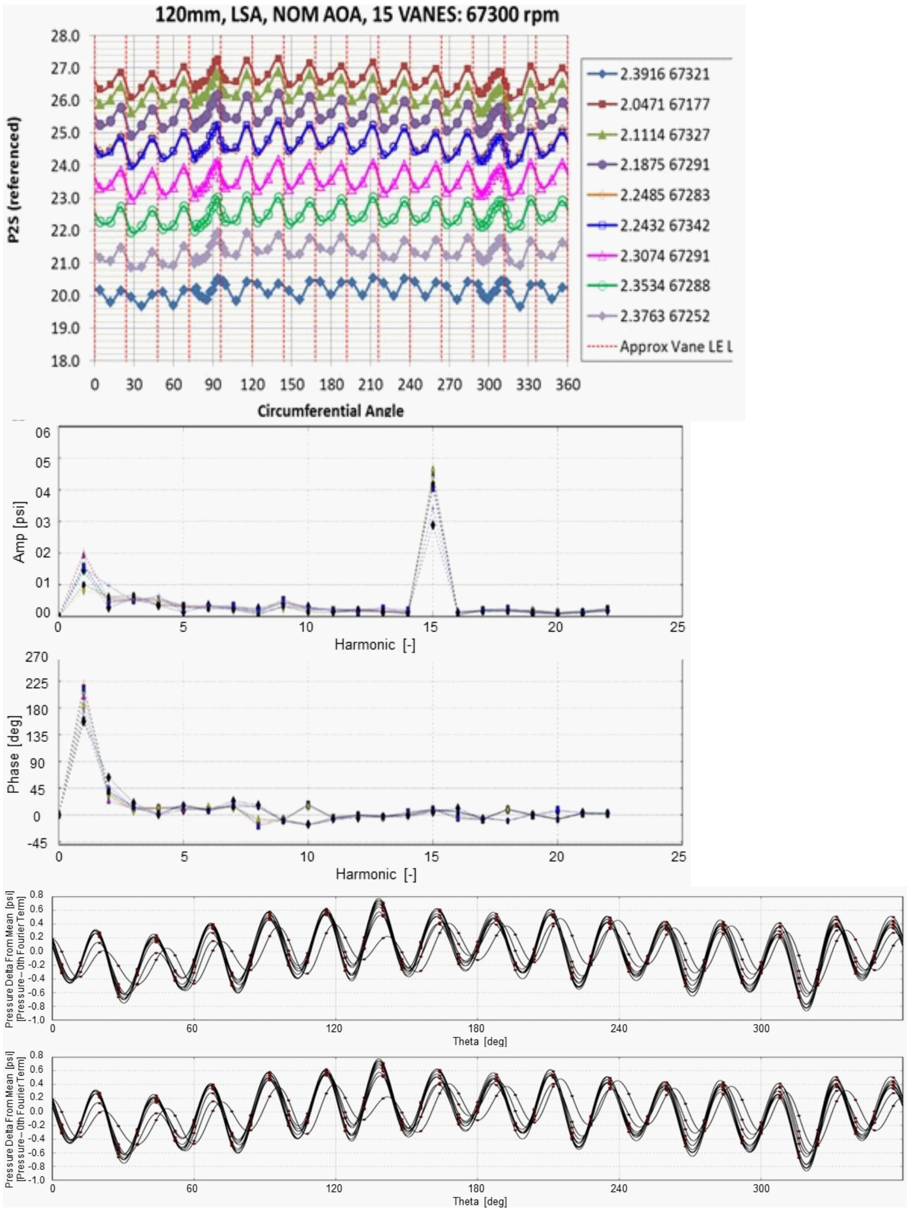


Figure 13 Measured p_2 values vs. circumferential position and Fourier amplitudes and phase angles at different harmonics. <u>*CN*</u> 120 mm rig data, LSA diffuser at AOA 0°, at 100% speed

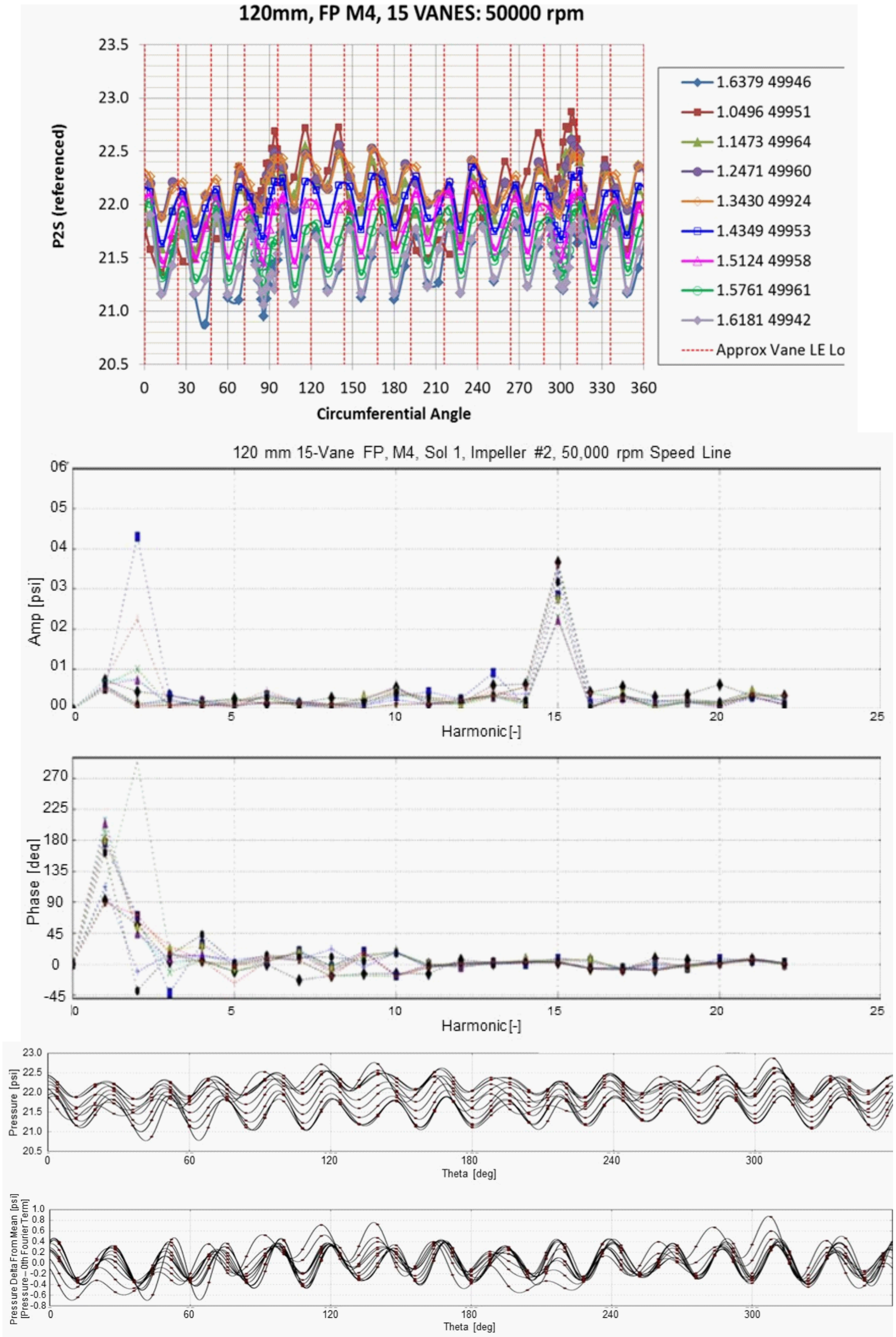


Figure 14 Measured p_2 values vs. circumferential position and Fourier amplitudes and phase angles at different harmonics. <u>*CN*</u> 120 mm rig data, flat-plate diffuser at AOA -4°, at 75% speed

7. Conclusion

The diffuser is an important part of all centrifugal compressor stages and many pump stages. It has been employed for a very long time, but significant issues still exist for its optimum application in truly advanced designs. This paper points to progress and issues in this process.

8. Acknowledgements

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