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PERFORMANCE OF LOW SOLIDITY AND CONVENTIONAL DIFFUSER SYSTEMS FOR CENTRIFUGAL COMPRESSORS

Naresh K Amineni and Abraham Engeda
Turbomachinery Lab, Mechanical Engineering Dept.
Michigan State University
East Lansing, Michigan

William C Hohlweg and Gregory L Direnzi
Product Development
Elliott Turbomachinery Company
Jeannette, Pennsylvania

ABSTRACT

Centrifugal compressors have the widest compressor application area, covering aircraft engines, small stationary gas turbines, process and refinery industries, the refrigeration industry, and turbochargers. Despite the vast literature coverage of diffuser systems for centrifugal compressors, there are not more than twenty publications in the open literature on the family of vaned diffusers known as Low Solidity Vaned Diffuser (LSVD). This is highly surprising, in light of the fact, that practically all process and refrigeration compressors manufacturers, at one time or another, have attempted to design and test LSVD. Therefore with the strong belief that any work on LSVD either theoretical or experimental will be welcomed, this paper presents the performance of two newly designed LSVD.

Comparative experimental studies on diffuser systems for centrifugal compressors, performed at the Michigan State University Turbomachinery Lab are presented. A vaneless, a conventional vaned and two low solidity vaned diffusers were tested. The results are compared for the effect of the diffuser systems on the stage performance, the maximum efficiency, and the operating range of the compressor. The effect of the vane number in low solidity vaned diffuser on the performance is also discussed.

NOMENCLATURE

a = sonic velocity
b = passage hub to shroud width
c_p = specific heat
Cp = pressure recovery, $2\Delta p/\rho U_2^2$
L = vane chord length
LSVD- Low Solidity Vane Diffuser
M_t = tip rotational Mach number, U_2/a_0
$\dot{m}$ = mass flow rate
%R = range, $100(\phi_{choke} - \phi_{surge})/\phi_{choke}$
S = vane leading edge pitch
r = radius
U_2 = tip speed
Z = number of diffuser vanes
β = vane angle with respect to tangent
ϕ = inlet flow coefficient; $\dot{m}/\rho U_2 \pi r_2^2$
ϕ_N = normalized flow coefficient
η_N = normalized relative efficiency
θ = turning angle, $\beta_4 - \beta_3$
σ = solidity, L/S
ψ = head coefficient; $2c_p(T_{06} - T_{01})/U_2^2$
ψ_N = normalized head coefficient

Subscripts

h = hub
s = shroud
0 = impeller inlet location, total condition
2 = impeller tip location
3 = diffuser vane inlet location
4 = diffuser vane exit location
5 = diffuser exit
6 = compressor exit

INTRODUCTION

It is common practice by the centrifugal compressor manufacturers to offer a standard line of compressors which would provide large flow range for a variety of inlet conditions and process gases. Thus, vaneless diffusers have been the obvious choice for manufacturers due to their ability to accept wide range of inlet variations without any significant loss in performance.

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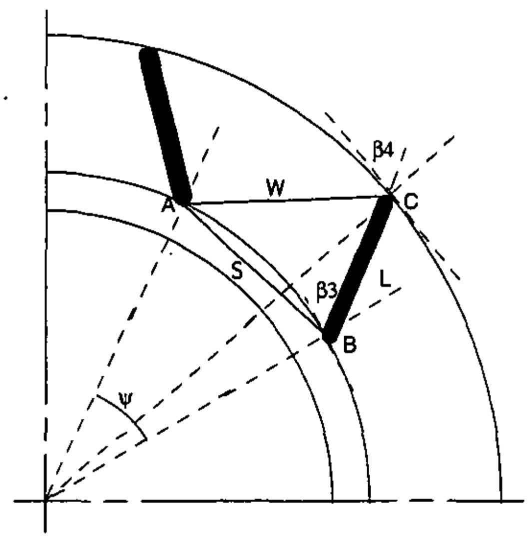

FIGURE. 1 GEOMETRY OF LSVD

Often, the requirement for higher efficiency forces the designer to replace the standard vaneless diffuser with a vaned diffuser. It has been seen from experience that even though the vaned diffusers increase the design point efficiency by improving diffuser pressure recovery, they reduce the operating range of the compressor by throat choking at high flows and vane stalling at low flows. Vaned diffusers find extensive application in gas turbines where flow range requirements are limited. However, they are notalways desired in the industrial compressors due to the limited flow range offered by them.

In an attempt to overcome the deficiencies of the vaned diffusers, Senoo et al. (1983, 1989) proposed low solidity vaned diffusers (LSVD) and showed that they offered improved efficiency and diffuser pressure recovery without any significant loss in operating range. Osborne and Sorokes (1988) conducted further experiments using design procedures derived from Senoo with simple flat plate vane construction and obtained similar results. They also demonstrated significant performance gains over a vaneless diffuser for designs covering a wide range of specific speeds.

In the process of understanding the behavior of low solidity vaned diffusers Sorokes and Welch (1991, 1992) developed a rotatable low solidity vaned diffuser and obtained comprehensive data showing the effects of various setting angles on both diffuser performance and overall stage performance. Hohlweg et al. (1993) performed experiments on two different types of compressors and compared the performance of the vaneless, vaned and low solidity vaned diffusers for each compressor. Their results showed the importance of the effect of incidence on the performance of the compressor.

Based on study of literature and analysis of past experience, two new LSVD were designed. A well performing vaneless and a conventional vaned diffuser were tested along with the two new LSVD downstream of the same impeller for comparative study. The overall stage performance with the three diffuser systems are compared. The effect of the diffuser system on the maximum efficiency of the stage and the operating range is discussed along with the effect of LSVD vane number on the performance of the LSVD. The pressure recovery of the vaneless and a LSVD is also discussed in order to understand the diffusion process in the LSVD.

LSVD DESIGN

The design geometrical parameters needed for a simple geometry LSVD are shown in Fig. 1. The design of the present LSVDs was based on the experience drawn from the previously tested and reported [Hohlweg et al. (1993)] LSVDs. It was found that negative incidence at design point enhanced the flow range of the stage when compared to the positive incidence. Thus, for the present designs, the design incidence was set at -2°. The choice of -2° was based on the theoretical calculations of the flow angles coming out of the impeller for good matching of the components.

Further at the initial stage of this current work, the LSVDs reported by Hohlweg et al. (1993) were numerically analyzed for flow behavior inside the LSVD. The numerical analysis was done using the BTOB3D code. The details of this analysis is reported by Amineni et al. (1995). Numerical experimentation with the number of vanes (Z) in the diffuser showed that the flow at surge could be improved by increasing the number of vanes (Fig. 2). Since the blade turning angle (θ) which determines the amount of blade loading in the diffuser is a function of the number of vanes, when the solidity and the vane inlet angle of straight flat plate LSVD are fixed. By increasing the number of vanes the blade loading is decreased causing the flow to be more stable. Thus in order to study the effect of vane number on the performance, one of the present LSVD was designed with 16 vanes and the other one with 14 vanes.

The solidity of the two new designs was kept at 0.7. The choice of 0.7 solidity was made so as to compare the results of present studies with the other published reports. The vane inlet angle was obtained from the design flow angle and the design incidence. The vane inlet radius for both designs was maintained same, since earlier reports by Sorokes et al. (1988) have shown that inlet radius ratio does not have a significant effect on the LSVD performance. Since the LSVD are competing with the vaneless diffusers, the shape of the vanes were straight flat plate type so as to keep the fabrication cost to minimum possible.

Once the solidity (σ), the diffuser inlet radius (r_3) and the inlet vane angle (β_3) is determined, the number of vanes (Z) in the diffuser is needed to determine the vane exit radius (r_4) and the exit vane angle (β_4) from the LSVD geometrical relationships:

$$[\sigma]^2 = \frac{\left(\frac{r_4}{r_3}\right)^2 + 1 - 2\left(\frac{r_4}{r_3}\right)\cos\theta}{2 - 2\cos\left(\frac{2\pi}{Z}\right)} \qquad (1)$$

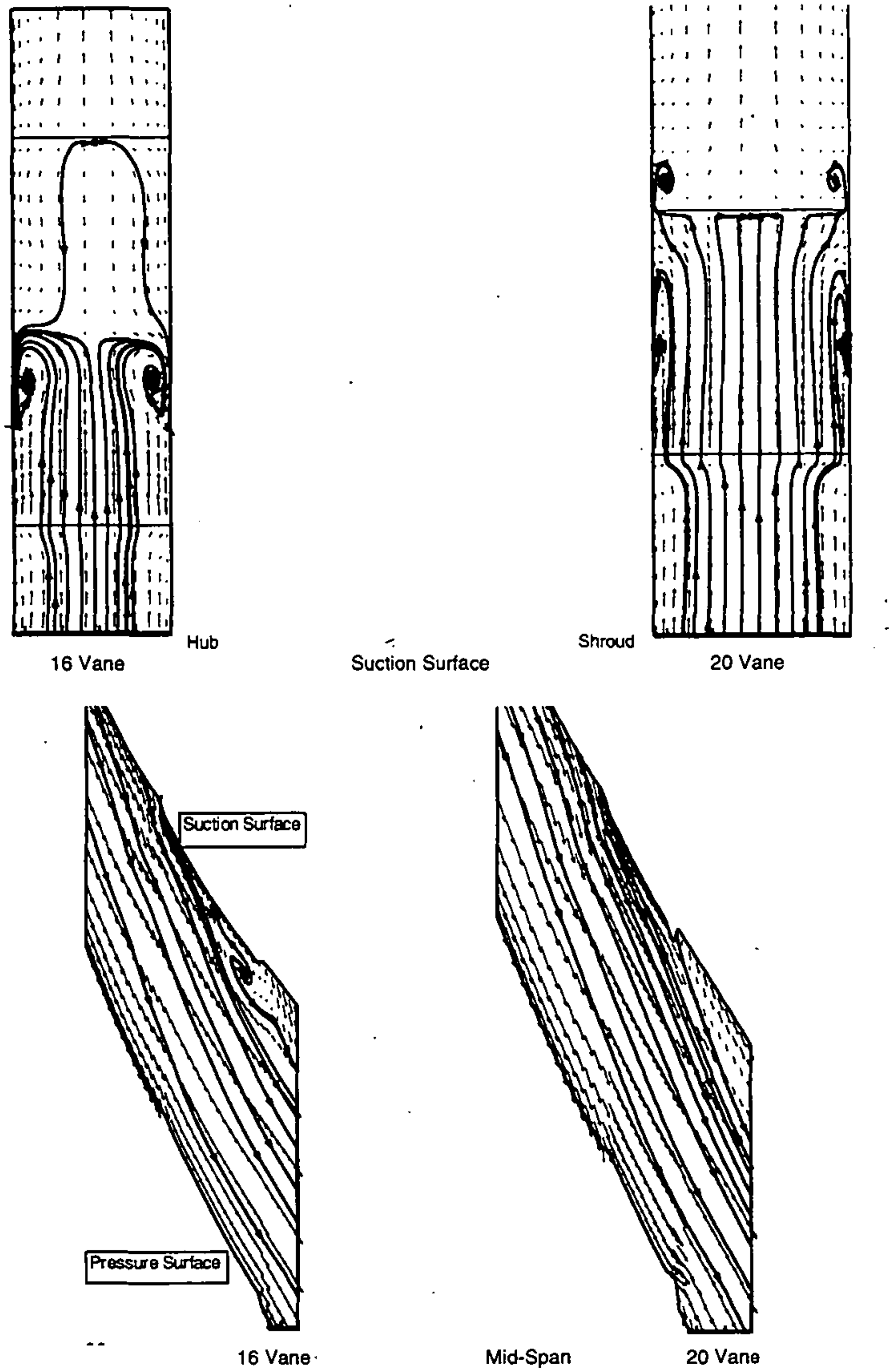

FIGURE. 2 FLOW IN 16 VANE AND 20 VANE LSVD AT SURGE.

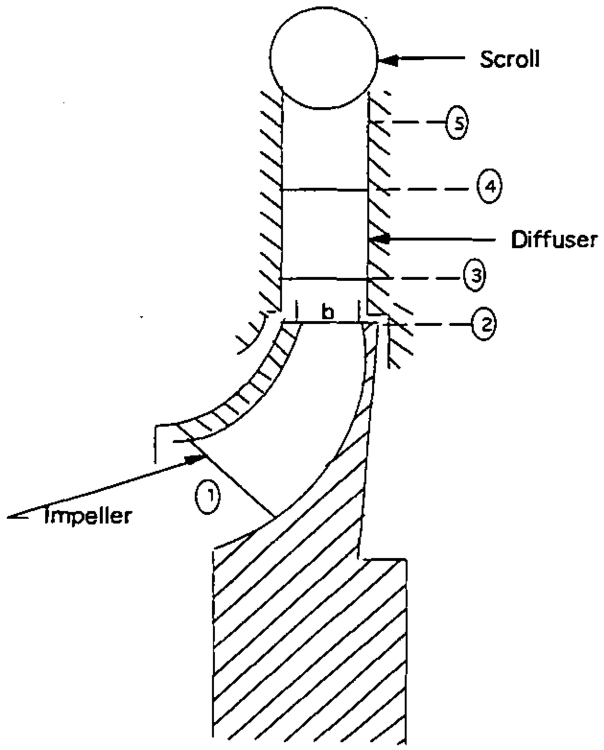

FIGURE. 3 VARIOUS DIFFUSER STATIC TAPS LOCATIONS AND ALSO USED IN TABLE. 1

where

$$\sigma = \frac{L}{S} \qquad (2)$$

and for straight vanes

$$r_4 = r_3 \frac{\cos\beta_3}{\cos\beta_4} \qquad (3)$$

THE TEST FACILITY

The comparative performance tests of the vaneless, conventional vaned and two LSVD were performed at the Michigan State University Turbomachinery Lab. The tests were performed on Test Rig I of the lab's two major centrifugal compressor rigs, each of which has variable speed, 225 kW drive and fully automated data acquisition and monitoring systems. Both test rigs have been designed to accommodate a wide configuration of impellers and diffuser systems.

The total pressures and the total temperatures at the suction and discharge sections, were measured using 1/16 and 1/8 inch diameter Kiel type probes and 1/8 inch diameter total temperature probes consisting of copper-constantan thermocouples. These probes were located at 4 circumferential locations 90 degrees apart. The static pressures at the corresponding locations were also recorded using 1/16 inch holes. In addition to these, static pressure in the diffusers was measured at four different radial positions. Fig. 3 shows the radial location of the static taps.

The compressor casing was covered with insulating materials to eliminate the heat loss.

The performance of the compressor stage was evaluated on the total to total basis. The flow rate through the compressor was controlled with a valve at the discharge piping and the mass flow rate was calculated based on ASME orifice plate differential pressure measurements in the suction pipe.

TEST COMPONENTS

The tests were performed at three speeds corresponding to tip Mach numbers of M_t= 0.69, 0.88, and 1.02. The impeller had the following configuration:

- tip diameter, d_2 = 244 mm
- exit blade width, b_2 = 12.85 mm
- inlet shroud diameter, d_{s1} = 147.67 mm
- inlet hub diameter, d_{h1} = 47.09 mm
- exit blade angle, b_2 = 70.70 deg.
- inlet blade angle, b_{s1} = 28.60 deg.
- blade number, Z = 19

One vaneless diffuser, one conventional vaned diffuser and two low solidity vaned diffusers were tested. The vaneless diffuser was pinched whereas the vaned and the low solidity vaned diffusers had the same width as the impeller exit blade width. Simple flat plate vanes were used for low solidity vaned diffuser. Even though a solidity of up to 0.9 is considered as low solidity, 0.7 solidity was chosen for the present LSVD designs as in most of the published literature. Since the solidity was fixed for the two LSVDs, the number of vanes determined the amount of blade loading in the diffusers. The principle specifications along with the designation of the diffuser are given in Table. 1. Fig. 3 shows the notation used to define the various locations in the diffuser.

Table. 1 Diffuser Design Parameters

Diff. Name	b_3/b_2	b_4/b_2	r_3/r_2	r_4/r_3	r_5/r_3	Z	θ	β_3	σ	Diff. Type
VNL	0.838	0.838	1.089	-	1.406	-	-		-	Vaneless
VND	1.00	1.00	1.09	-	1.406	15		15		Vaned
LSVD1	1.00	1.00	1.09	1.10	1.406	16	12.9	17	0.7	LSVD
LSVD2	1.00	1.00	1.09	1.12	1.406	14	14.6	17	0.7	LSVD

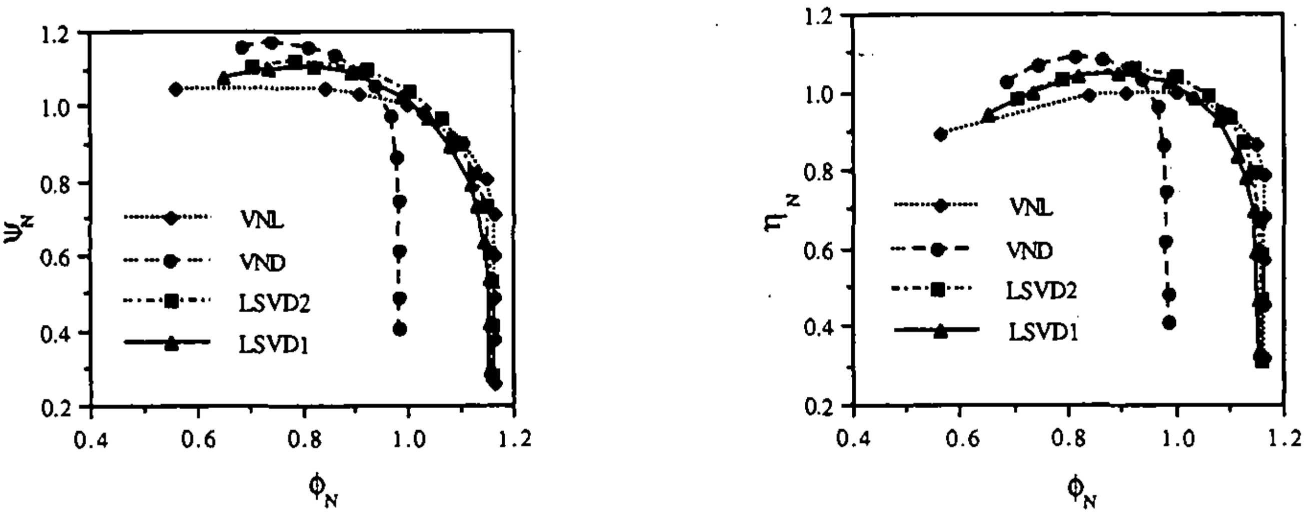

FIGURE. 4 PERFORMANCE CHARACTERISTICS AT M_t = 1.02.

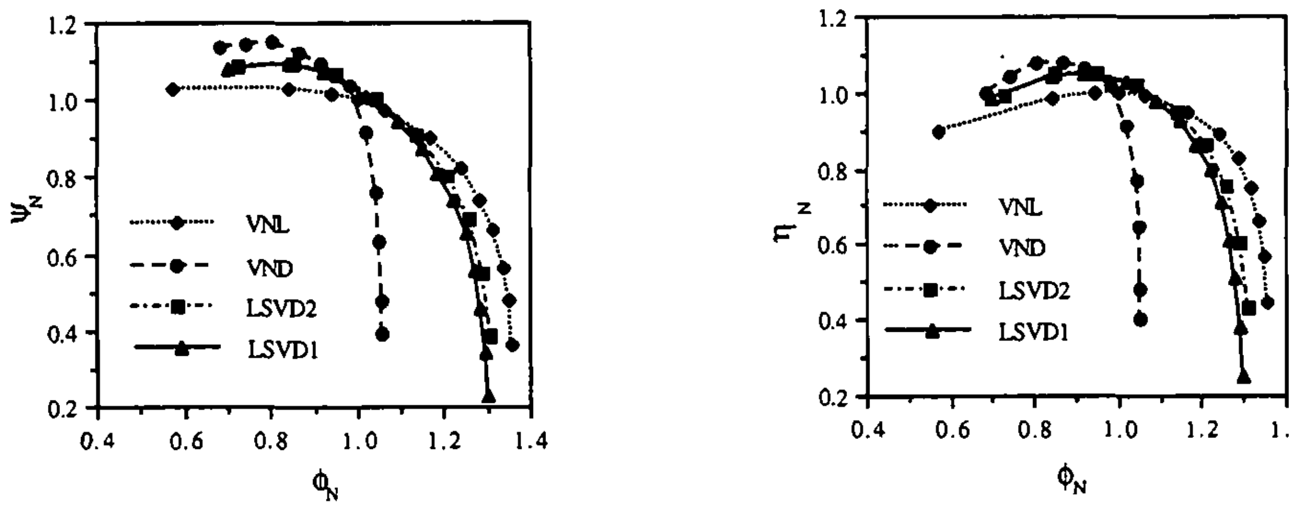

FIGURE. 5 PERFORMANCE CHARACTERISTICS AT M_t = 0.88.

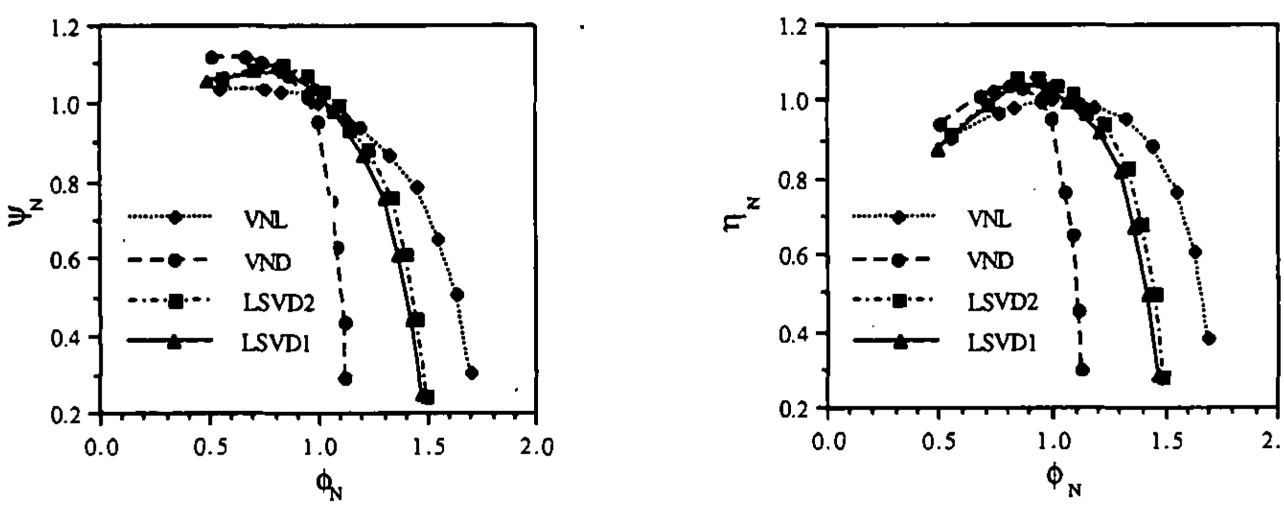

FIGURE. 6 PERFORMANCE CHARACTERISTICS AT M_t = 0.69.

PERFORMANCE CHARACTERISTICS

The characteristic curves of the compressor with different diffusers at three impeller tip rotational Mach numbers (M_t) (0.69, 0.88 and 1.02) are presented in Fig. 4 through Fig. 6. Each of these figures shows the variation of efficiency (η) and head coefficient (ψ) with the flow coefficient (ϕ) at a particular tip Mach number. The headcoefficient and efficiency were calculated based on stage total to total conditions. The work input of the impeller was estimated from the stagnation temperature difference between the inlet and exit of the compressor. All the performance curves are normalized using the rated design point values of the vaneless diffuser stage as the reference.

At $M_t = 1.02$, the characteristic curves of the VNL diffuser and the LSVDs are vertical at the right end, and the flow rate at this speed is constant for these three diffusers. On the other hand the VND diffuser maximum flow was about 16% less than the other three. This clearly indicates that the maximum flow through the compressor was controlled by the vaned diffuser throat choke in case of the VND and the inducer choke in case of the VNL and LSVDs. Similar results were presented by Hohlweg et al. (1993) and Hayami et al. (1990). At lower speeds ($M_t = 0.88$ and 0.69) it can be noticed that even though the LSVDs lack a true diffuser throat, their overload capacity decreases with the speed. The loss of overload capacity by LSVDs over VNL is about 7% at $M_t = 0.88$ and 11% at $M_t = 0.69$. Thus at low speeds the high negative incidence causes pressure surface stall of the LSVD vanes and therefore it can be said that LSVD tend to behave more like vaned diffuser than the vaneless diffuser. Sorokes and Welch (1992) showed that LSVD displayed flow separation on the pressure surface of the vanes at negative incidence, thus indicating LSVD are sensitive to the incidence as seen in the present study.

The left end of the curves show that the minimum flow attainable by the different diffusers is different. At $M_t = 1.02$ the surge flow coefficient of LSVD1 was smaller than the VND by 7% and greater than the VNL by 15%, where as LSVD2 and VND reached almost the same surge flow. These observations are different from those of Hayami et al. (1990) ($\theta = 10°$) and the reason for this could be that the LSVD vanes in the present study ($\theta_{LSVD1} = 12.9°$, $\theta_{LSVD2} = 14.6°$) have higher loading and are made of thin flat plate which caused them to stall. This situation might be improved by using complex airfoil vanes. VND, LSVD1 and LSVD2 all attained about the same surge flow, which was greater than the VNL at $M_t = 0.88$, but at $M_t = 0.69$ the VND, LSVD2 and VNL reached surge at the approximately same flow and LSVD1 surged at 12% lower flow than the others. Thus indicating the LSVD1 vanes acted as flow stabilizers for possible stall in the diffuser at low speed.

The efficiency of the compressor with LSVDs was better than the VNL diffuser for a wide range of flow at all tip speeds. However, the LSVDs fell short of the VND diffuser at $M_t = 1.02$ and 0.88. At $M_t = 0.69$ the LSVD2 was able to out perform even the VND diffuser over a wide flow range.

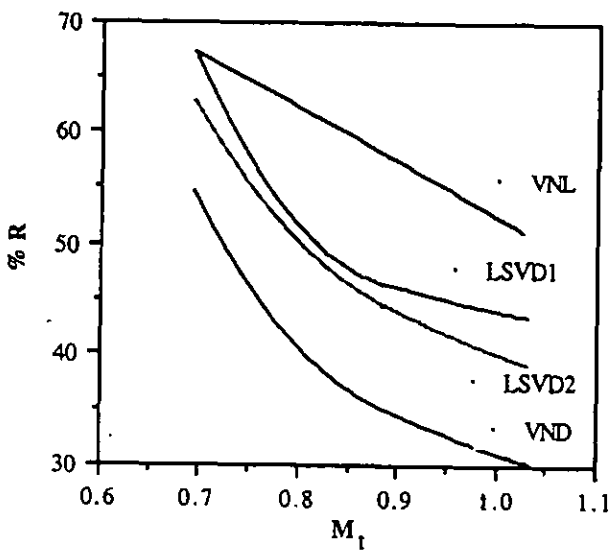

FIGURE. 7 VARIATION OF %RANGE WITH M_t

In general at all speeds the efficiency and the pressure ratio of the compressor with LSVDs were superior to the VNL diffuser over a wide range and the flow range attained by LSVD is comparable to VNL diffuser. Thus LSVD gives flexibility in design as they can be adjusted to peak in efficiency over a wide range of impeller flow conditions.

COMPARATIVE STUDY

Operating Range (%R)

The percentage range (choke to surge) or the total operating range of the compressor as function of impeller tip Mach number is presented in Fig. 7. The VND diffuser has the lowest and the VNL diffuser has the highest operating range as expected. The LSVDs seem to have an operating range in between the VND and VNL diffuser showing clear advantage of the LSVD over the vaned diffuser. As the vanes of both VND and LSVDs stall by pressure side separation due to negative incidence at high mass flow rate and by suction surface separation at low mass flow rates the difference in the operating range of these diffusers with VNL diffuser is maximum at $M_t = 0.88$. However, at low speeds the difference between the VNL and LSVDs seem to be very little as the small vanes of the LSVD seem to prevent the diffuser stall at low flow rates and the gain in surge flow seems to compensate for loss in flow range at the right end of the characteristic curve. An interesting observation which can be seen in Fig. 7 is that the VND and the LSVDs show similar trend in decrease of operating range and VNL diffuser has a linear decrease as the M_t increases, this clearly indicates that the LSVDs behave more like VND diffusers.

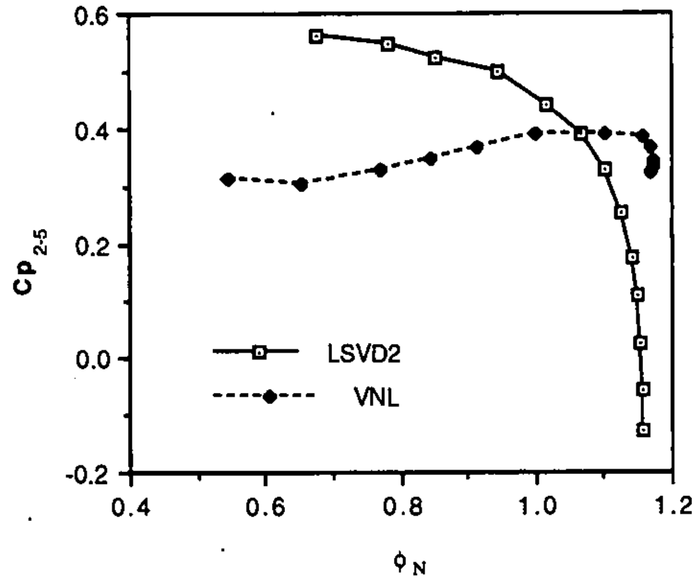

FIGURE. 8 TYPICAL PRESSURE RECOVERY IN VNL AND LSVD.

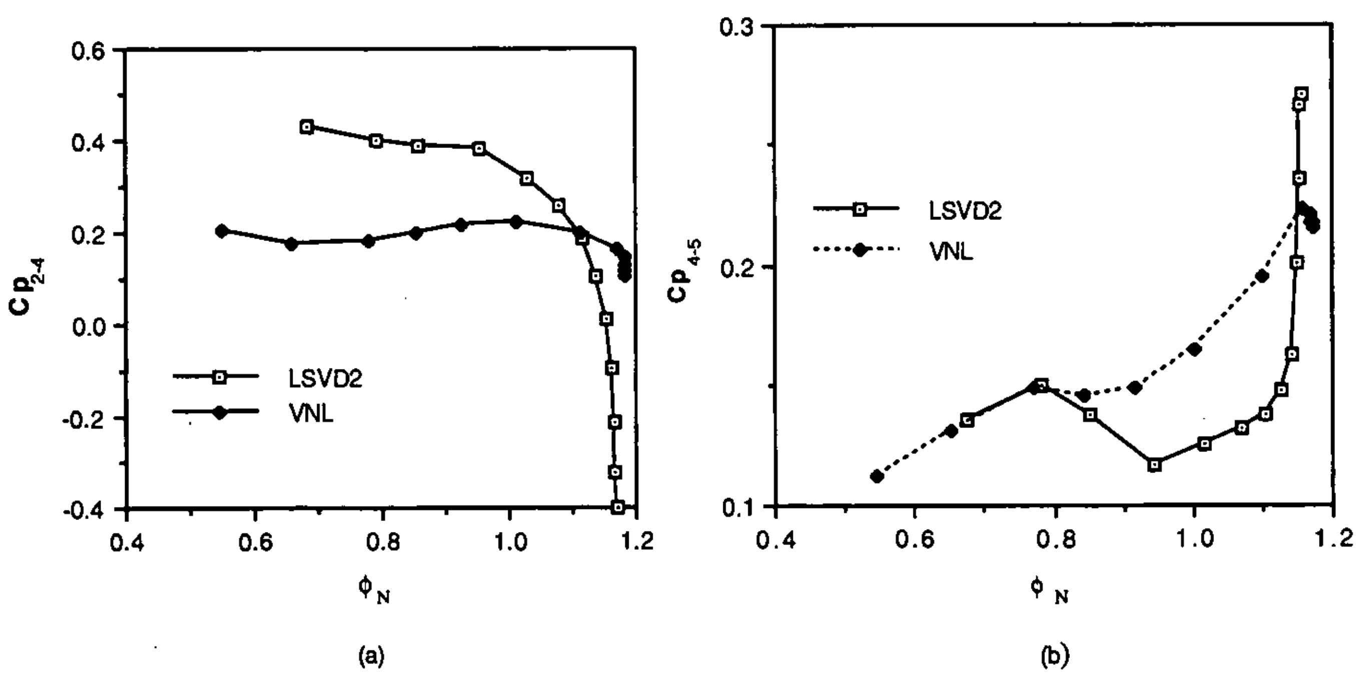

FIGURE. 9 PRESSURE RECOVERY IN DIFFERENT REGIONS OF LSVD.

Blade Number (Z) -

As noted in Table. 1, LSVD1 had 16 vanes and LSVD2 had 14 vanes and both diffusers had same solidity of 0.7. Since solidity and vane number are related, decrease in the vane number causes an increase in the blade turning angle (θ) or the blade loading, when the vane inlet angle is fixed and the vanes are made of thin straight flat plate. From the characteristic curves and Fig. 7 it is clearly evident that LSVD2 did not have the operating range as good as the LSVD1 and this can be attributed to the highly loaded vanes which stalled at low mass flow rates. The difference in operating rage between LSVD1 and LSVD2 was about 5% at M_t = 1.02 indicating that vane loading has a great effect on operating range achievable by the LSVD at high speeds. The peak efficiency of LSVD2 was greater than that of LSVD1 by 1 percentage point at M_t = 1.02 and they both had almost same peak efficiency at M_t = 0.88. Therefore it seems for a given speed there is an optimum blade turning which can provide good efficiency and operating range.

Pressure Recovery (Cp)

The pressure recovery of the three diffuser systems was measured using the static pressure taps located on the hub, at four different radial positions. The positioning of these static taps was such that, one set of taps was at the exit of the impeller, one set at the vane inlet radius of VND and LSVD, one set at the vane exit radius of LSVD, the last set was located at the exit of the diffuser. Thus, it was not only possible to measure Cp of the whole diffuser, it was also possible to measure the Cp of different regions of the LSVD.

Fig. 8 shows the typical Cp measured of the VNL and LSVD at M_t = 1.02. It can be clearly seen that the pressure recovery of the LSVD2 is much better than the VNL. The Cp of LSVD2 at high mass flow rates suffers from the high negative incidence, since the LSVDs were designed with a design incidence of -2°.

In order to understand the pressure recovery mechanism in the LSVD. The Cp of the diffuser is broken down into two parts 1) Cp from diffuser inlet (station 2) to vane exit (station 4); 2) from vane exit (station 4) to diffuser exit (station 5). Fig. 9 compares the Cp of these two regions for LSVD2 with the corresponding regions of the VNL at M_t = 1.02. It can be observed that nearly 75% of the total pressure recovery of LSVD is attained between diffuser inlet (station 2) and vane exit (station 4). The $Cp_{2\text{-}4}$ of LSVD2 is simillar to the pressure recovery between diffuser inlet and the throat of a vaned diffuser as reported by Baghdadi (1977), Kano et. al. (1982) and Stein and Rautenberg (1988). Since the $Cp_{2\text{-}4}$ of LSVD2 is higher than that of VNL over a wide range of flow rate, the hub and shroud boundary layers at the vane exit of LSVD2 are thick due to which the $Cp_{4\text{-}5}$ of LSVD2 is lower than that of VNL (Fig.9(b)). The thick boundary layers at the vane exit cause higher blockage and increased losses in the downstream vaneless space of the LSVD2.

CONCLUSIONS

To complement and supplement the few reported cases of LSVD performance and design, the performances of a vaneless, a CVD and two new designs of LSVD were compared. One objective was to maintain simple vane geometry and a consequent objective was to identify the influence of geometrical vane parameters on the LSVD performance.

It has been observed that the general pressure recovery mechanism in the LSVD is similar to that of the CVD inlet to throat region. However, the magnitude of Cp associated with LSVD tends to be higher. Moreover it was found that, for a given design speed there is an optimum blade turning which can provide good efficiency as well as operating range.

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REFERENCES

[1] Baghdadi, S., 1977, "The Effect of Rotor Blade Wakes on Centrifugal Compressor Diffuser Performance -- A Comparative Experiment", Journal of Fluids Engineering, Trans. ASME, pp. 45-52.

[2] Kano, F., Tazawa, N. and Fukao, Y., 1982, "Aerodynamic Performance of Large Centrifugal Compressors", ASME Paper No. 82-GT-17.

[3] Senoo, Y., Hayami, H. and Ueki, H., 1983, "Low Solidity Tandem Cascade Diffusers for Wide Flow Range Centrifugal Blowers", ASME Paper No. 83-GT-3.

[4] Stein, W. and Rautenberg, M., 1988, "Analysis of Measurements in Vaned Diffusers of Centrifugal Compressors", Journal of Turbomachinery, Vol. 110, pp. 115-121.

[5] Osborne, C., and Sorokes, J. M., 1988, "The Application of Low Solidity Diffusers in Centrifugal Compressors", Flows in Non-Rotating Turbomachinery Components, ASME FED-Vol.69.

[6] Hayami, H., Senoo, Y., and Utsunomiya, K., 1989, "Application of Low-Solidity Cascade Diffuser to Transonic Centrifugal Compressor", ASME Paper No. 89-GT-66.

[7] Sorokes, J. M., and Welch, J. P., 1991, "Centrifugal Compressor Performance Enhancement Through the Use of a Single Stage Development Rig", Turbomanchinery Symposium Proceedings, Texas A&M, pp. 101-112.

[8] Sorokes, J. M., and Welch, J. P., 1992, "Experimental Results on a Rotatable Low Solidity Vaned Diffuser," ASME Paper No. 92-GT-19.

[9] Hohlweg, W. C., Direnzi, G. L. , and Aungier, R. H., 1993, "Comparison of Conventional and Low Solidity Vaned Diffusers", ASME Paper No. 93-GT-98.

[10] Amineni, N. K., Engeda, A., Hohlweg, W. C., and Boal, C. F., 1995, "Flow Phenomenon in Low Solidity Vaned Diffusers of an Air Packaging Compressor", ASME Paper No. 95-WA/PID-1.