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EFFECT OF DOUBLE ROW DIFFUSERS ON PERFORMANCE AND FLOW FIELD OF A CENTRIFUGAL COMPRESSOR

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ABSTRACT

The present paper reports measurements carried out in a centrifugal compressor equipped with double row of low solidity partial vane diffuser. Performance of the compressor is measured with three sets of double row diffusers with different radial spacing between the diffusers. From these measurements best configurations are identified and static pressures on the diffuser walls and probe measurements inside the diffuser passages are carried out and contours of diffuser wall static pressures and flow properties inside the passage are presented.

Keywords: Centrifugal compressor, double row partial vane low solidity diffusers, leading edge radius ratio, performance, flow field measurements.

1. INTRODUCTION

The fluid leaving the impeller of a centrifugal compressor possesses a large amount of kinetic energy. It is therefore necessary to convert this kinetic energy into pressure energy. This is usually done with the help of diffusers. The reduction in flow velocity and the accompanying rise in pressure are achieved by an increase in the area in the direction of flow.

Two different types of diffusers are used in centrifugal compressors, namely vaned diffuser and vaneless diffuser. Vaneless diffuser has the advantages of low cost, good operating range with large mass flow and good off design efficiency. But a large outer radius with attendant friction losses is required if a large pressure recovery is required. Vaned diffuser has the advantage of higher pressure recovery at a smaller radius ratio and good design efficiency. But off-design efficiency falls rapidly and flow is choked at a lower mass flow. Senoo [1] patented low solidity vaned diffuser (LSVD) in 1978. LSVD has the advantage of no throat; hence the flow will not be choked. Yoshinaga et al. [2] presented results on partial vane diffuser fixed to the shroud of the centrifugal compressor. Sitaram et al. [3] had combined both concepts and extensively tested partial vane low solidity diffuser fixed to both the hub and shroud at half the vane spacing. Senoo et al. [4] had further tested tandem vane diffuser in low solidity configuration. This concept had been further investigated by Hayami et al. [5] and Sakaguchi et al. [6]. Tandem vanes are extensively tested in axial turbomachines. Tandem vanes can be treated as a special of case of two rows of diffuser vanes. Many patents with different configurations of two rows of diffuser vanes are disclosed [7-11].

For present configuration two rows of low solidity partial vaned diffuser is used. Two rows have more flexibility to improve the performance of the compressor. Low solidity partial vaned diffuser (LSPVD) is a subclass of the low solidity vaned diffuser vaned diffuser. The main characteristics of LSPVD are low solidity without throat (l/s<1) and blade height less than the diffuser passage width.

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2. OBJECTIVE

The objective of the present investigation is to find the best diffuser configuration which includes optimum inlet radius ratio and optimum inlet angle of diffuser vane. With the optimum inlet angle the incidence loss can be reduced. To find out optimum blade angle, the vane angle of inner as well as outer rows of diffuser is varied $\pm 5^o$ from the nominal inlet angle so that no throat is formed for the both the rows. Performance test is carried on each configuration at different inlet radius ratio of outer row to find out optimum inlet radius ratio of outer row. The inner row is fixed at inlet radius ratio 1.10 (i.e. $R_3=r_3/r_2=1.1$) and outer row is changed according to the radius ratio of 1.25, 1.30 and 1.35 (i.e. $R_4=r_4/r_2=1.25$, 1.30 and 1.35). A total of twenty configurations are tested by variation of distance between the rows and variation in angle of the diffuser vanes.

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The vanes on the hub and shroud are staggered at half vane spacing. The second row is staggered at half vane spacing from the first row.

3. EXPERIMENTAL FACILITY AND INSTRUMENTATION

3.1 Experimental Facility

The present experimental investigation is carried out on a single stage centrifugal compressor available in Thermal Turbomachines Laboratory, Department of Mechanical Engineering, IIT Madras. The low specific speed impeller used for the present experimental investigation is without shroud and with the vanes being radial at exit. The impeller has an axial inducer integral at the inlet. The major design details of the impeller are given in Table 1. More details on the impeller and facility are given in [12].

TABLE 1: Design Details of the Impeller

Parameter	Value
Pressure ratio, PR	1.08
Mass flow rate, m	0.84 kg/s
Speed, N	4500 rpm
Shape number, $N_{SH}=N\sqrt{V}/(60W^{3/4})$	0.076
Inducer hub diameter, D_{1h}	110 mm
Inducer tip diameter, D_{1t}	225mm
Impeller exit diameter, D_2	393 mm
Number of blades, N_B	20
Blade angle at inducer hub, β_{1h}	45 Deg.
Blade angle at inducer tip, β_{1t}	29 Deg.
Blade angle at exit, β_2	90 Deg.
Exit diameter of vaneless diffuser, D_5	600 mm

All angles are measured w.r.t. tangential direction

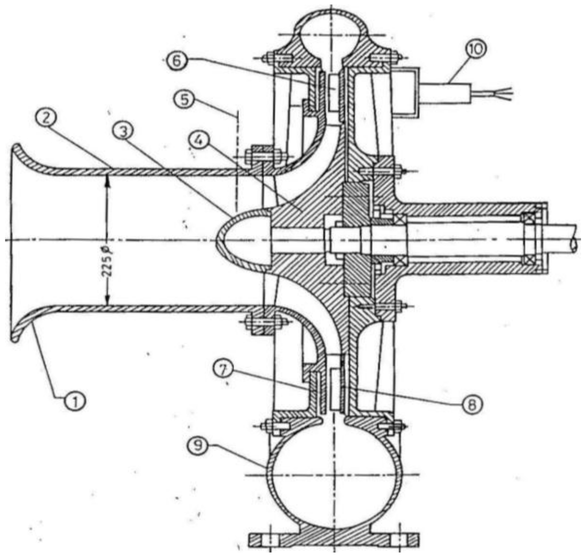
A schematic of the impeller is shown in Fig. 1.

The details of diffuser vanes used for the inner and outer rows are given in Table 2.

Table 2: Design Details of the Diffusers

Parameter	Value (Range)	
	Inner row	**Outer row**
Leading Edge Radius Ratio	R_3=1.10	R_4=1.25, 1.30 & 1.35
Inlet Angle	75^o	65^o
Exit Angle	65^o	DR1.25=58^o DR1.30=56^o DR1.35=54^o
Vane Thickness	3.2 mm	4 mm
Vane Height	6 mm	
Solidity	0.7	
Number of Blades	21 for both rows	

All angles are measured w.r.t. tangential direction



1 Inlet nozzle 2 Inlet duct 3 Hub 4 Impeller 5 Inlet traverses station 6 Diffuser 7 Diffuser shroud
8 Diffuser hub 9 Scroll casing 10 Traverse mechanism

Fig.1. Meridional view of the compressor showing impeller and hub wall vane diffuser

3.2 Instrumentation

The instrumentation used in the present investigation is primarily consisting of scanning box, micromanometer, tachometer and three hole cobra probe. A brief description of this instrumentation is given below.

For the measurement of pressure a micro manometer manufactured by M/s Furness Control Ltd., Bexhill, U.K and a scanning box (FC0 91-3) is used. Scanning box has twenty channels. These are numbered sequentially. The pressures to be measured are connected to the numbered inputs. Output of the scanning box is connected to the micro manometer, where the pressure is displayed. The micro manometer, differential pressure measuring unit, is capable of reading air pressure in the range of $\pm$ 1999 mm WG with accuracy of 0.1 mm. The flow in the passages of inner and outer rows of the is measured with a pre-calibrated three hole probe. The probe is used in non-nulling mode. Total and static pressures and flow angle are obtained from the probe measurements and the calibration curves. The calibration curves of the probe are shown in Fig. 2. The probe tip geometry is shown as an inset of the figure. The calibration is done at two velocities, representative of the flow field to be measured. The calibration curves show that the effect of Reynolds number on the calibration coefficients is negligible. The impeller inlet flow is one-dimensional, while the impeller exit flow is mildly three-dimensional with low axial velocities. Hence the use of a two-dimensional probe, such as a three-hole probe is justified. A three-dimensional probe such as a five hole probe causes large blockage, causing relatively large errors in the flow parameters to be measured.

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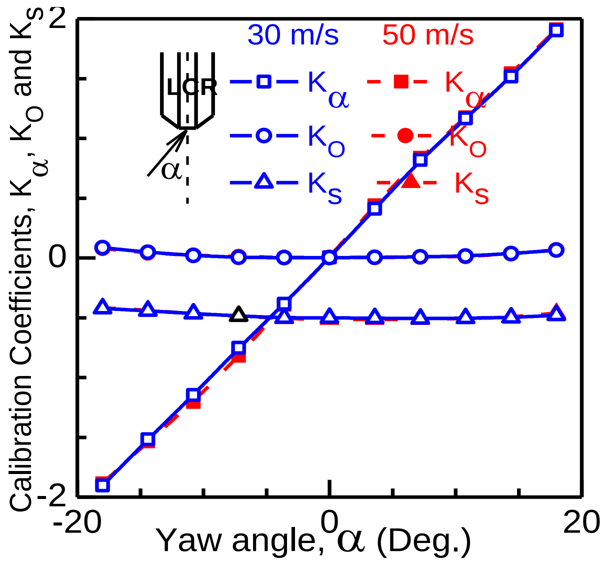

Fig.2. Non-nulling calibration curves of three hole probe

4. RESULTS AND DISCUSSION

4.1. Performance Characteristics

Performance tests are conducted on all configurations at a constant speed of 3,000 rpm as the effect of Reynolds number is found to be negligible beyond an impeller speed of 3,000 rpm. The configurations are listed in Table 3.

Table 3: Double Row Diffusers Tested

DR1.25 0H0S	DR1.30 0H0S	DR1.35 0H0S
DR1.25 0H5S	DR1.30 0H5S	DR1.35 0H5S
DR1.25 0H-5S	DR1.30 0H-5S	DR1.35 0H-5S
DR1.25 5H0S	DR1.30 5H0S	DR1.35 5H0S
DR1.25 5H5S	DR1.30 5H5S	DR1.35 5H5S
DR1.25 5H-5S	DR1.30 5H-5S	DR1.35 5H-5S
DR1.25 -5H5S	DR1.30 -5H5S	DR1.35 -5H5S

The performance of centrifugal compressor with all the twenty configurations is compared and from this comparison, the best configuration for each outer row position is presented in Table 4 and plotted in Fig. 3.

From Table 4 and performance curves, it can be concluded that the double row diffuser with outlet radius ratio 1.35 with 5H5S angle change gives slightly better performance in terms of peak energy coefficient and operating range compared to all other configurations. Hence further experiments are conducted on this diffuser only.

For this configuration, static pressure on the diffuser walls at 357 holes, placed at 17 radial locations is measured at four different flow coefficients, viz., ϕ=0.23 (below design flow coefficient), 0.34 (design flow coefficient), 0.45 and 0.60 (above design flow coefficient). A precalibrated three hole probe is used to measure flow parameters inside the diffuser passages at twelve axial locations, six radial locations and at seven circumferential locations at the above flow coefficients. At each axial plane, total pressure, static pressure, flow angle and velocity are determined. The non-dimensionalised flow parameters are axially averaged and are presented. The radial distribution of mass averaged non-dimensional properties is also presented.

4.2 Static Pressure Contours on the Diffuser Walls

Contours of static pressure coefficient on the diffuser walls at the four flow coefficients are shown in Fig. 4. As the static pressures are nearly identical on the hub and shroud walls, on the static pressures on the shroud wall are shown.

The static pressure distribution is nearly uniform in the circumferential direction for design flow coefficient. For all flow coefficients the flow becomes more uniform after the vane trailing edge.

Static pressure distribution at ϕ=0.23 is somewhat uniform. This is due to that at lower flow coefficient incident to the leading edge is lower. This reduces flow separation and results in uniformity at the leading edge.

Table 4: Comparison of Performance of Different Diffuser Configurations

Configuration	VLD	DR1.25 5H-5S	DR1.30 5H0S	DR1.35 5H5S
ψ_{max}	1.265	1.360	1.360	1.370
ϕ_{max}	1.036	0.990	1.050	1.030
ψ_d: Energy coefficient at design flow coefficient	1.275	1.340	1.340	1.350
$\Delta\psi_{max}$: $\left(\psi_{max} - \psi_{max\,vld}\right)/\psi_{max\,vld}$	0.000	0.075	0.075	0.083
ϕ at ψ_{max}: Flow coefficient at which peak energy coefficient occurs	0.323	0.400	0.430	0.430
ψ at ϕ_{max}: Energy coefficient at maximum flow coefficient	0.426	0.780	0.840	0.810
$\Delta\phi_{max}$: $\left(\phi_{max\,vld} - \phi_{max}\right)/\phi_{max\,vld}$	0.000	0.046	-0.014	0.004
ϕ_{op}: Operating range	0.713	0.590	0.620	0.600

H: Hub S: Shroud DR1.25 5H-5S: Second row L.E. at R_4=1.25, hub vanes set at 5^o and shroud vanes set at -5^o from original angle

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At flow coefficient ϕ=0.45 static pressure is non-uniform because of high incident angle. In present investigation of DR1.35 5H5S, as the vane spacing is large, it is not possible for the flow to mix completely on its way to the vane exit, because of the insufficient flow guidance. As a consequence, non-uniformity of static pressure still prevails. In this type of blades static pressure drop is very small, even though the incident angle is high. At ϕ=0.45, the static pressure rise is higher than all other flow coefficient.

At higher flow coefficient of ϕ=0.60, the pressure distribution is highly non-uniform. This is due to the fact that at higher flow coefficient incident to the leading edge is higher. This causes flow separation and non-uniformity at the leading edge. The static pressure at the exit of vane is very low.

4.2 Total Pressure Contours in the Diffuser Passage

Contours of total pressure coefficient in the diffuser passage at the four flow coefficients are shown in Fig. 5. These contours are averaged values at the twelve axial stations. All other contours at the four flow coefficients are available in [3].

Total pressure coefficient decreases as radius ratio increases for both blades. The decrease in total pressure occurs due to losses occurring in the flow passage. The total pressure distribution is different at different axial planes in the passage. The total pressure distribution near the diffuser shroud and hub are low compared to that at the centre. Not only is that, at the centre the distribution uniform in the circumferential direction. Axially averaged total pressure distribution for four flow coefficient is presented. Axially averaged total pressure distribution is uniform in the circumferential direction for ϕ=0.23 and ϕ= 0.34. At higher flow coefficient, i.e. at ϕ=0.45 at ϕ=0.60 constant total pressure lines are radially outwards which indicates that total pressure variation in circumferential direction is very high. This may be due large velocity variation in this region.

4.4 Variation of Flow Parameters with Radius

The flow parameters measured in the diffuser passages are mass averaged and plotted against the radius. The mass averaged values are shown in Fig. 6.

The mass averaged total pressure decreases as radius increases. The decrease in total pressure indicates the losses occurred in the flow passage. But in case of DR1.35 5S5H, it has no throat and it provides a large frontal area for flow adjustment in the passage, so as compared to other type of diffuser, DR1.35 5S5H gives lower loss.

Static pressure increases as the radius increases. At ϕ=0.45 is having a higher value of static pressure coefficient than other flow coefficients. The static pressure rise for ϕ=0.60 is very low.

At off design condition incidence is high. This may lead to flow separation. Static pressures always increase with radius ratio for any flow coefficient.

Flow angle decreases as radius increase. DR1.35 5S5H shows a large variation at flow angle at the trailing edge because of flow separation. Out of four flow coefficient flow angle at ϕ=0.45 shows somewhat less variation inflow angle compared to the other flow coefficients. Higher value of flow angle is seen at ϕ=0.60 throughout the flow passage.

As radius increase total velocity decreases continuously for all flow coefficient. The rate of decrease is high for ϕ=0.45 and ϕ=0.60. The energy transformation mainly occurs within the diffuser vane passage. The kinetic energy of the fluid is converted into static pressure. It can be seen that at any radial location the velocity of flow decreases as flow coefficient increases. Hence among the four flow coefficients maximum velocity values are occurred for ϕ=0.60. That means conversion of kinetic energy into static pressure rise is less at higher flow coefficients. Also losses are more at higher flow coefficients. These losses are occurred mainly due to flow separation. At off design conditions flow separates from the vane.

Meridional velocity decreases as radius increase to maintain the continuity of the flow. The variation of meridional velocity is almost linear from inlet to exit of the passage, which we can predominantly observed for the flow coefficient ϕ=0.45. Highest meridional velocity is observed at flow coefficient of 0.60, but on average ϕ=0.45 shows higher meridional velocity.

Reduction in tangential velocity shows that kinetic energy is converted to static pressure. Higher tangential velocity is observed at ϕ=0.45 compared to that at other flow coefficients.

5. CONCLUSIONS

1 Performance tests show that the DR1.35 5S5H shows the higher energy coefficient and efficiency than the other tested configurations.
2 At the leading edge of DR1.35 5S5H provides large area for flow adjustments and weaker flow separation zones, due to lower number of vanes, which results in small pressure drop near the leading edge. At flow coefficient ϕ=0.45 static pressure is non-uniform because of high incident angle. In present investigation of DR1.35+5SH, as the vane spacing is large, it is not possible for the flow to mix completely on its way to the vane exit, because of the insufficient flow guidance. As a consequence, non-uniformity of static pressure prevails. In this type of blades static pressure drop is very small, even though the incident angle is high

3 The mass averaged total pressure decreases as radius increases. The decrease in total pressure indicates the losses occurred in the flow passage.

But in case of DR1.35 5S5H, it has no throat and it provides a large frontal area for flow adjustment in the passage, so as compared to other type of diffuser, DR1.35 5S5H gives lower loss.

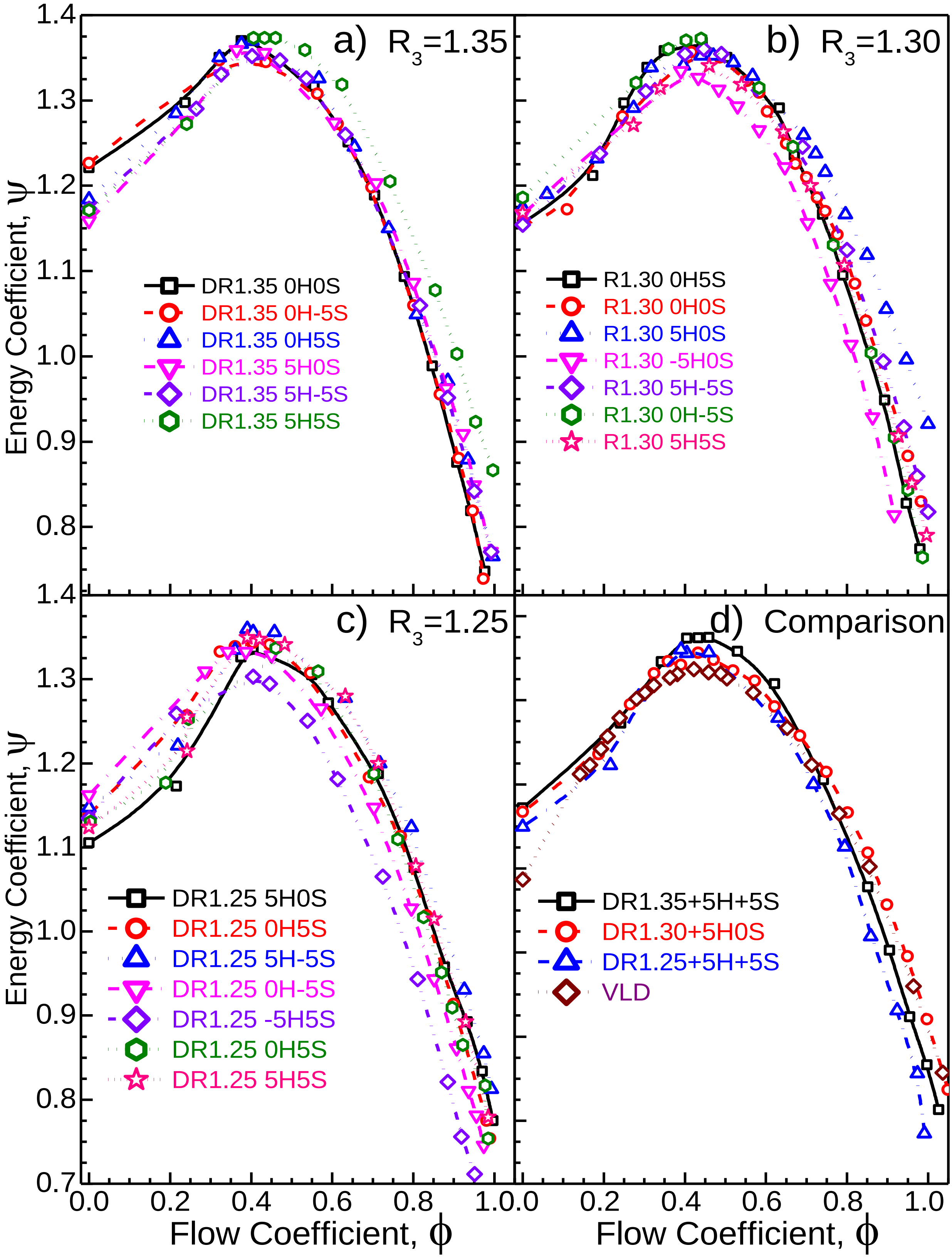

Fig. 3. Comparison of performance characteristics of a centrifugal compressor with double row diffuser with different radius ratio

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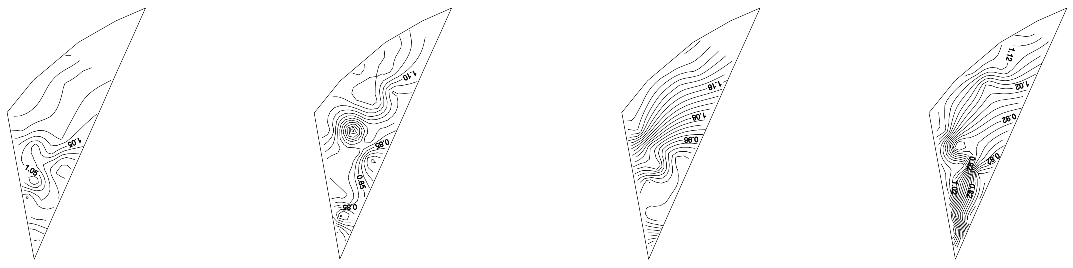
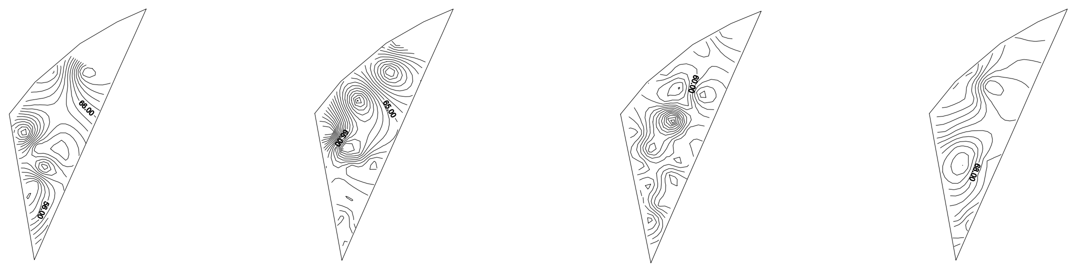


(a) ϕ=0.23 (b) ϕ=0.34 (c) ϕ=0.45 (d) ϕ=0.60

Fig.4. Contours of static pressure coefficients on the diffuser shroud wall of DR1.35 5H5S for different flow coefficients



(a) ϕ=0.23 (b) ϕ=0.34 (c) ϕ=0.45 (d) ϕ=0.60

Fig. 5 Contours of axially averaged total pressure coefficients in the diffuser passage of DR1.35 5H5S for different flow coefficients

4 Static pressure increases as the radius increases due to flow diffusion. Static pressure at ϕ=0.45 is higher than that at other flow coefficients. The static pressure rise for ϕ=0.60 is very low. At off design condition incidence is high. This may lead to flow separation. Static pressures always increase with radius ratio for any flow coefficient. As radius increase total velocity decreases continuously for all flow coefficient. The rate of decrease is high for ϕ=0.45 and ϕ=0.60.

5 The energy transformation mainly occurs within the diffuser vane passage. The kinetic energy of the fluid is converted into static pressure. It can be seen that at any radial location the velocity of flow decreases as flow coefficient increases. Velocity component decreases as radius increase to maintain the continuity of the flow. The variation of velocity component are almost linear from inlet to exit of the passage, which is predominantly observed for the flow coefficient ϕ=0.45.

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NOMENCLATURE

C	Velocity (m/s)
C_d	Coefficient of discharge of the inlet nozzle
C_m	Meridional velocity (m/s)
C_u	Tangential velocity (m/s)
C_o	Velocity in the suction duct at nozzle throat (m/s)
D	Diameter (m)
DRLSPVD	Double row low solidity partial vaned diffuser
DR1.25 5H5S	Double row diffuser with radius ratio 1.25 and 5 degree increase in inlet angle of outer row of diffuser vanes on the hub and shroud
H	Hub
	Height difference between suction and delivery pressure taps (m)
LSPVD	Low solidity partial vaned diffuser
N	Speed in rpm
P_a	Ambient pressure (N/m^2)
P_d	Average static pressure at delivery side (N/m^2)
P_o	Total pressure (N/m^2)
P_s	Average static pressure at suction side (N/m^2)
P_{ws}	Wall static pressure (N/m^2)
R	Radius ratio, r/r_2
r	Radius (m)
S	Shroud
V	Volume flow (m^3/s)
VLD	Vaneless diffuser
W	Specific work (m^2/s^2) = $\frac{(P_d - P_s)}{\rho} + \left(\frac{C_d^2 - C_s^2}{2}\right) + gH$
α	Flow angle (degree)
ϕ	Flow coefficient = $\frac{V}{\pi D_2 b_2 u_2}$
ρ	Density of air (kg/m^3)
σ	Solidity=chord/spacing
ψ	Energy coefficient = $\frac{2W}{u_2^2}$

Subscripts

1	Impeller (Inducer) inlet
2	Impeller exit
3	Leading edge of first row diffuser vanes
4	Leading edge of second row diffuser vanes
5	Exit of vaneless diffuser
h	Hub
max	Maximum value
maxvld	Maximum value of compressor with vaneless diffuser
op	Operating range
t	Tip

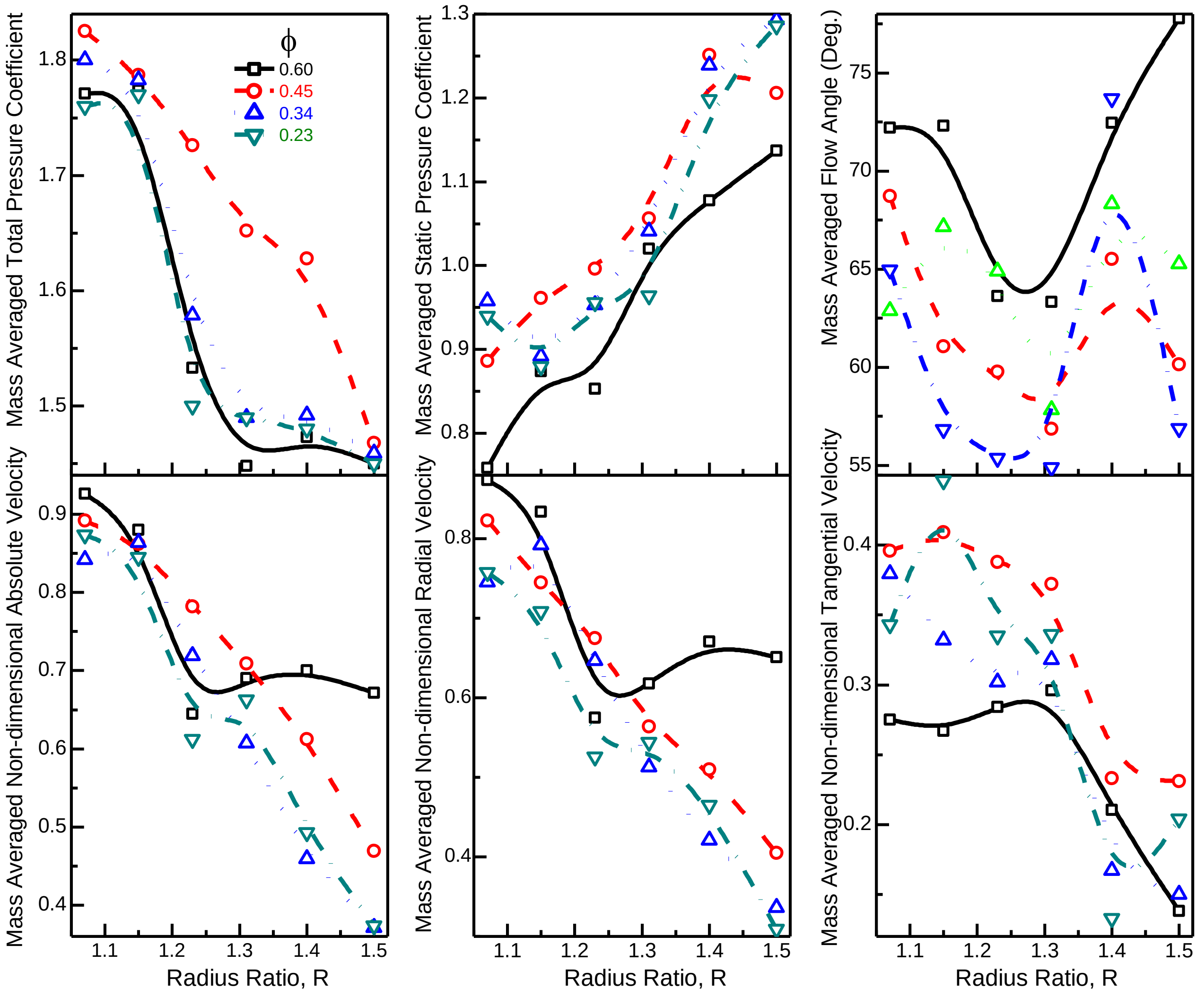

Fig. 6 Variation of mass averaged total and static pressure coefficients, absolute, meridional and tangential velocities and flow angle with radius ratio for DR1.35 5SH at different flow coefficients

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