CFD Analysis of Impeller-Diffuser Interaction in a Centrifugal Compressor with Twisted Vaned Diffuser

P. Venkateswara Rao, G.V. Ramana Murty and G. Venkata Rao

Abstract The effect of variation of diffuser radius ratio on the performance of a centrifugal compressor stage with vaneless diffuser (VLD), low solidity vaned diffuser (LSVD), and twisted vaned diffuser (TVD) was studied using Computational Fluid Dynamics (CFD). ANSYS CFX software was used in the present study. The diffuser vane was generated by modifying the trailing edge of a cambered aerofoil profile (NACA 2410). The stagger angle was varied from hub to shroud forming a twisted vane diffuser keeping the same leading edge. The present analysis was carried out at a tip Mach number of 0.35. The performance was assessed in terms of stage power coefficient, stage efficiency, total pressure loss coefficient and static pressure recovery coefficient for five different flow coefficients in the diffuser. From the present study, the optimum radius ratio is 1.10 for TVD with the chosen impeller diffuser configuration.

Keywords Centrifugal compressor · Diffuser · Twist · Performance · Interaction

Nomenclature

C_p	Static Pressure Recovery Coefficient, $(p_4\text{-}p_2)/(p_{02}\text{-}p_2)$
D	Diameter (m)
g	Acceleration due to gravity (m/s^2)
H	Total head (m), $\frac{\gamma}{\gamma-1}RT_{01}\left[\left(\frac{P_{04}}{P_{01}}\right)^{\gamma-1/\gamma}-1\right]\frac{1}{g}$
l	Chord length of the Vane (m)
LE	Leading Edge
LSVD	Low Solidity Vaned Diffuser
$\dot{m}$	Mass flow rate (kg/s)
n	Speed (rpm)
P	Pressure (Pa)

P.V. Rao (✉) · G.V.R. Murty · G.V. Rao
Department of Mechanical Engineering, Vasavi College of Engineering, Hyderabad, India
e-mail: pvrao_vce@yahoo.co.in

A.K. Saha et al. (eds.), *Fluid Mechanics and Fluid Power – Contemporary Research*, Lecture Notes in Mechanical Engineering,
DOI 10.1007/978-81-322-2743-4_86

r	Radius (m)
R	Gas constant
S	Pitch (m), $\pi D_3/Z$
TPLC	Total pressure loss coefficient, $(p_{02}-p_{04})/(p_{02}-p_2)$
TVD	Twisted Vaned Diffuser
u	Peripheral velocity (m/s), $\pi Dn/60$
VLD	Vaneless Diffuser
Z	Number of diffuser vanes
α	Flow angle (deg)
β	Blade angle (deg)
η	Total-to-static stage efficiency (%), $\left[\dfrac{\left(\frac{P_4}{P_{01}}\right)^{\gamma/\gamma-1}}{\left(\frac{T_{04}}{T_{01}}-1\right)} - 1\right] * 100$
λ	Power coefficient, $\phi\psi/\eta$
ω	Angular velocity, rad/s
ϕ	Flow coefficient, $m/(\rho u_2 \pi D_2^2/4)$
γ	Ratio of specific heats
ρ	Inlet density (kg/m^3)
σ	Solidity, l/s
ψ	Head coefficient, $H/(u_2^2/2\ g)$

Subscripts

0	Total quantities
1	Inlet of the Impeller
2	Exit of the Impeller
3	Inlet of the Diffuser
4	Exit of the Diffuser

1 Introduction

The industries such as fertilizers, refineries, steel mills deploy centrifugal compressors for achieving pressure rise in the process. Similarly aircraft industry uses centrifugal compressors for aviation applications. Rotating impeller and stationary diffuser are the primary components of a centrifugal compressor. Significant pressure rise is achieved by rotating the flow from axial to radial direction in the impeller. Flow in the impeller is three dimensional due to meridional bend and circumferential bend. Flow parameters at the diffuser and impeller interface are difficult to compute and understand due to the inaccessibility. At the exit of the impeller a non-uniform jet-wake flow pattern (non-uniform) is observed. Thus non uniform flow enters the diffuser circumferentially at different inlet angles and velocity fluctuations. Impeller flow is affected by geometry, flow conditions, and unsteady pressure disturbances due to diffuser vanes. The study is challenging as flow field characteristics are three dimensional, viscous, turbulent and unsteady.

Senoo [1] was the first person to talk about the LSVDs. Geometrical throat is not formed in the LSVD passages and thus increases the operating range of the compressor compared to conventional diffusers. Krain [2] experimentally observed that the flow at the impeller exit is highly non-uniform. Further he stated that the radial gap between impeller and the LSVD plays a significant role.

Rodgers [3] and Clements et al. [4] confirmed the dependence of stage total-to-static pressure ratio on radial gap between impeller and LSVD. They showed that the flow characteristics at impeller-diffuser interface can lead to an enhanced stage performance. Computations with different types of diffuser vanes in centrifugal compressor stage are performed by Anish and Sitaram [5, 6]. They concluded that the performance of partial vaned diffuser is little affected by the interaction levels. Abdelwahab [7] proposed twist for the diffuser vane in the direction same as that of the impeller rotation direction using cambered NACA 65 at a speed of 36800 rpm. While twisting from hub to shroud, chord length of the profile is varied. Strong secondary flows are prevented by the three dimensional airfoil vanes near the shroud, along the suction surface of the diffuser. Uniform blade loading is the outcome of this change. Venkateswara Rao et al. [8] showed that a twist angle of 9° at a setting angle of 24° with respective to tangential direction provides optimum performance for chosen centrifugal compressor stage with NACA 2410 as diffuser vane profile.

The present work is primarily focused to study the effect of radial distance between the impeller and TVD on the stage performance and flow field of an industrial centrifugal compressor stage. Three different types of diffusers are chosen for this analysis and comparison: viz. VLD, LSVD, and TVD. Leading edge of the diffuser vane is located at three different radial locations. For CFD simulations, steady-state model is assumed. The diffuser vane is formed from a NACA 2410 aerofoil profile with minor modifications in the trailing edge region. The setting angle for the hub profile is 24° with respective to tangential direction. Shroud profile is twisted by 9° [8] in the opposite direction of the impeller rotation (z-axis) to obtain a three dimensional diffuser with leading edge as the reference. As a result of the profile rotation to form the twist, the chord length remains constant from hub to shroud. All CFD simulations were analyzed at 0.35 impeller tip Mach number as experimental data of VLD and LSVD is available for comparison.

2 Details of Centrifugal Compressor

A low speed centrifugal compressor stage is chosen for the current study. Table 1 shows the specifications of the compressor stage. Five flow rates, (80, 90, 100, 110 and 120 % of design mass flow rate) are considered for the analysis. These

Table 1 Details of the centrifugal compressor stage

Impeller inlet blade angle, β_1	27°	Mass flow rate, $\dot{m}$	1.3 kg/s
Impeller exit blade angle, β_2	45°	Number of impeller blades	17
Diffuser blade chord length at hub, l	90 mm	Number of diffuser blades, Z	14
Impeller radius at inlet (r_1)	150 mm	Speed, n	4500 rpm
Impeller outlet radius (r_2)	250 mm	Diffuser passage width, b	24.5 mm

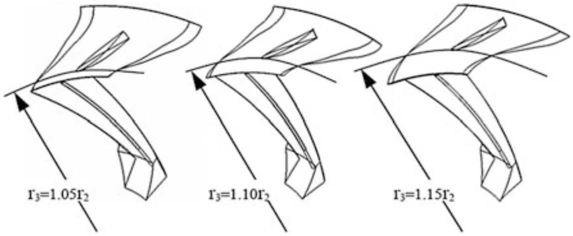


Fig. 1 Leading Edge position in the diffuser for TVD

represent the design and off design cases for all the three types of diffusers i.e. VLD, LSVD and TVD.

The impeller and diffuser interaction intensity in vane type diffuser configurations is changed by keeping the diffuser vane leading edge at three different locations(radial) as shown in Fig. 1. Locations considered are

$$R_3 = r_3/r_2 = 1.05\,(\text{Interaction is strong})$$
$$R_3 = r_3/r_2 = 1.10\,(\text{Interaction is medium})$$
$$R_3 = r_3/r_2 = 1.15\,(\text{Interaction is weak})$$

3 Geometry and Mesh

The rotating component in the compressor is impeller while the stationary component is diffuser. Three dimensional flow domain is considered for the CFD analysis. AutoCAD is used to model impeller and Vaned diffuser geometry. GAMBIT software was used for modelling and meshing of vaneless diffuser. Flow in each passage is considered periodic. Figure 2a–c shows the generated mesh for the analysis.

ANSYS Turbo Grid is used to generate structured hexahedral grids. O—grid and C—grid topology are selected to improve grid quality around the blade edges (leading edge and trailing edge). At design flow condition, Grid independence studies are conducted for the chosen configurations. With increasing count of elements, four grids are generated. Optimum grid size is selected based upon the convergence of one of the performance parameter namely static pressure recovery coefficient (C_p). This grid is used for all analyses. A grid minimum y-plus and

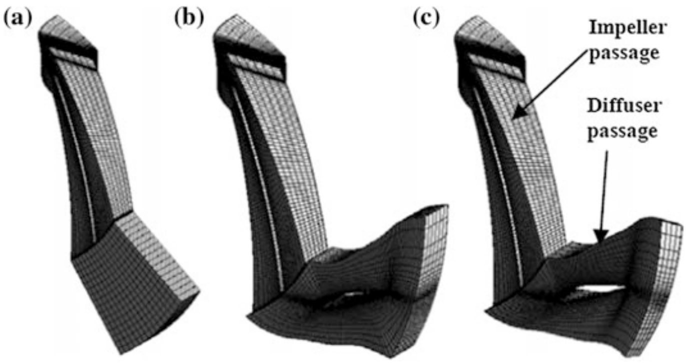


Fig. 2 Mesh details of compressor stage, impeller with **a** VLD **b** LSVD NACA 2410 **c** TVD NACA 2410

maximum y-plus value of 4 and 9 respectively used for the analysis which is in acceptable limits [9]. Three dimensional mass, momentum and energy equations are solved in the simulation using ANSYS CFX 14.0. In impeller, the equations are solved using a rotating frame. The diffuser variables are solved using the stationary frame. The transfer of information between the stationary and rotating domain is carried out using stage interface mixing plane model.

3.1 Boundary Conditions and Numerical Methodology

The computational domain at the inlet is kept in front of the eye of the impeller in order to ensure that the inlet boundary conditions are not affected by the back pressure of the impeller blade. Uniform total pressure is chosen as the boundary condition at the inlet for the stationary frame. Flow at inlet to impeller is assumed to be uniform with no swirl. Smooth wall with no-slip boundary condition was chosen for all surfaces viz. hub, shroud and the blade surfaces. Air is assumed as ideal gas and it enters axially at the inlet of the compressor. The reference absolute pressure used for simulation is 95000 Pa. Total energy model is specified for the temperature calculations. The specified total temperature at the inlet boundary is 310 K. Standard k-ε turbulence model is used for calculations. Turbulence intensity is selected as 0.05. Eddy length scale is 0.03. This is in line with the computations carried out by Meakhail and Park [10] on a centrifugal fan. For domain side walls (impeller and diffuser) boundary condition selected as rotational periodic. Boundary condition selected at outlet is mass flow rate. High resolution advection scheme is opted to descretiaze the governing partial differential equations. The fluid time step is specified as 0.1/ω. All the governing equations used for analysis are solved with residual error less than 1e-05. Post processing of results was done using ANSYS CFX Post.

4 CFD Model Validation

It is assumed for the simulations that the flow within a diffuser passage does not vary with the relative position of impeller and diffuser. CFD model was validated with the available experimental data of Siva Reddy et al. [11].

Figures 3 and 4 show the variation of head coefficient and stage power input with the flow coefficient in a normalized form for the chosen cases of VLD and

Fig. 3 Validation of CFD results (Head coefficient)

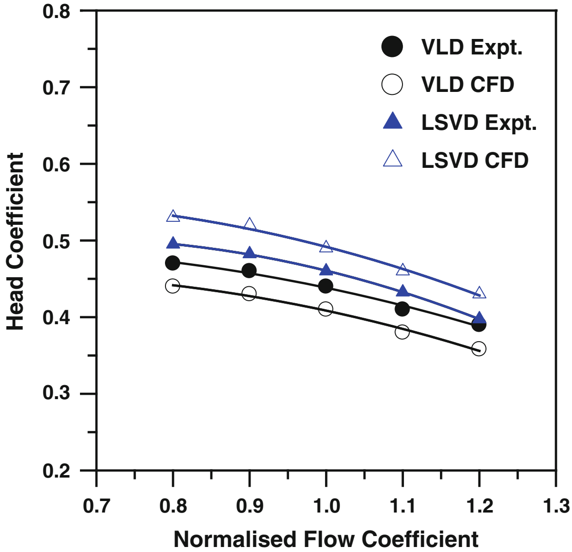
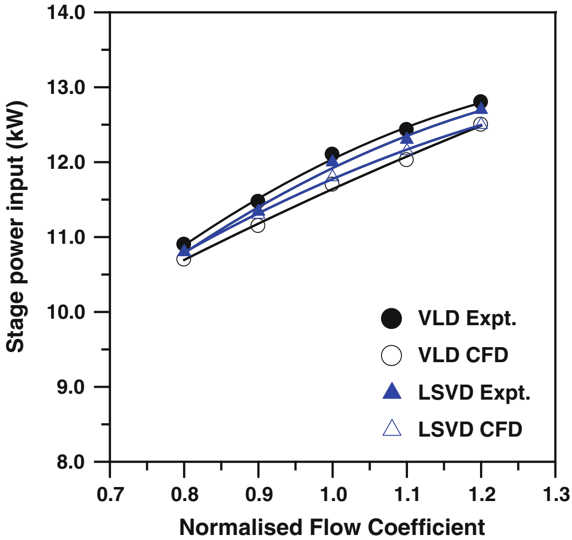


Fig. 4 Validation of CFD results (Stage power input)



LSVD configurations for the purpose of comparison with experimental data. The nature of variations is as expected for a centrifugal compressor stage. It is observed from these figures that the CFD studies predict high performance. The interaction between impeller and diffuser is unsteady which is represented in experimental results where as steady state simulations are considered in the CFD studies.

5 Results and Discussion

The performance with various diffuser combinations viz., VLD, LSVD NACA 2410, and TVD NACA 2410 blades with different radius ratio is discussed in the following sections.

5.1 Effect of Diffuser Radial Distance on Stage Performance

Results are presented in the form of normalized parameters. Vaneless diffuser data at the design point is used for normalization. The performance of the stage is explained in terms of stage efficiency, stage power coefficient and diffuser static pressure recovery coefficient (SPR).

5.1.1 Stage Efficiency

Total-to-static stage efficiency is used to evaluate the stage performance of a centrifugal compressor. The total-to-static stage efficiency is computed based on the conditions at the inlet of impeller domain and at outlet of diffuser domain. This way, the combined performance of the impeller and diffuser is accounted. Figure 5 shows variation of total-to-static stage efficiency for different diffusers with flow coefficients at radial distance R_3 = 1.05, 1.10 and 1.15 respectively. Stage efficiency is observed to be maximum at design flow coefficient for all types of diffusers considered at R_3 = 1.05. The efficiency of the TVD is more compared to LSVD because TVD offers better flow guidance. TVD decelerates the flow more efficiently than LSVD for the flow coefficients considered. Peak efficiency is observed for TVD configuration at design flow coefficient for R_3 = 1.1. It is 8.8 % higher compared to VLD and 1.72 % higher compared to LSVD. At all the radial distances considered, TVD shows improved performance in comparison to LSVD for all the flow coefficients. Maximum efficiency is also observed at far away distance from the impeller position (R_3 = 1.15) at design flow coefficient for all types of diffusers considered.

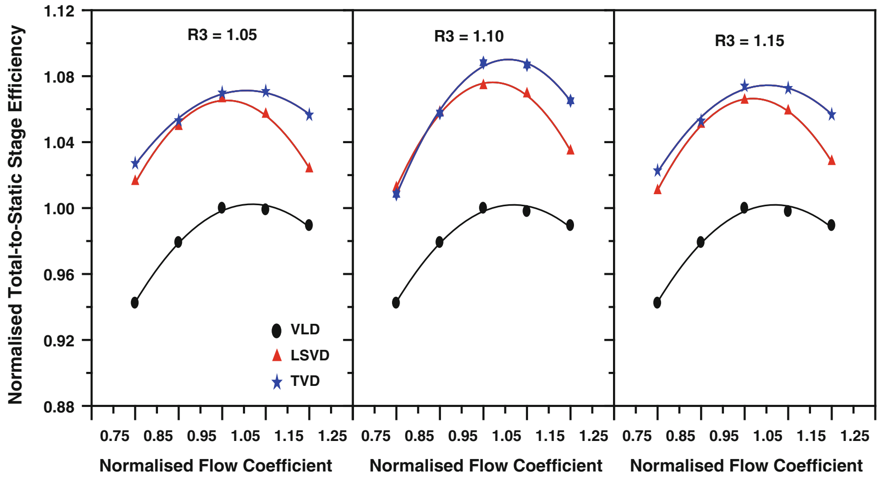


Fig. 5 Effect of diffuser radius ratio on total-to-static stage efficiency

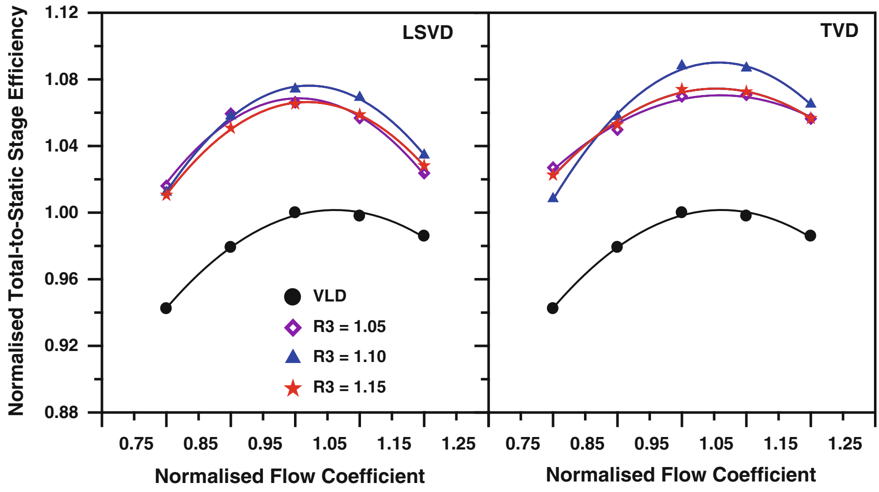


Fig. 6 Effect of vane shape on total-to-static stage efficiency at various radial positions

Normalized stage efficiency at different diffuser vane leading edge position is shown in Fig. 6 and compared with the results of VLD. The peak value of efficiency is observed at design flow coefficient for all the different leading vane positions for both LSVD and TVD.

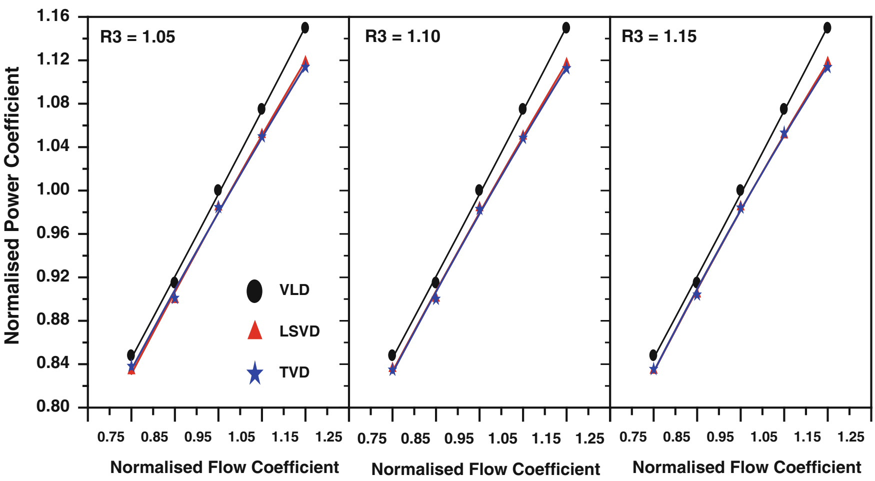


Fig. 7 Normalized power coefficient variation

5.1.2 Stage Power Coefficient

The energy required by the fluid as it passes through the impeller and diffuser to the exit is indicated by a performance parameter called stage power coefficient. As the normalized flow coefficient increases from 0.8 to 1.2, the power coefficient increases uniformly for all diffusers at any position of the diffuser vane leading edge. The variation in power coefficient follows the trend of a backward curved blade impeller. The power coefficient of VLD is more than TVDs. Due to the increase in efficiency at high flow coefficients for all configurations of vaned diffusers, as observed from Fig. 5 the power coefficient reduces Fig. 7. The effect of diffuser leading edge position on stage power coefficient is insignificant.

5.1.3 Diffuser Performance

Diffuser performance can be assessed based on the properties of flow at the diffuser inlet and outlet. The diffuser performance is estimated in terms of total pressure loss coefficient and static pressure recovery coefficient (C_p).

(a) Static Pressure Recovery Coefficient: C_p illustrates the amount of static pressure recovered from the existing dynamic head at the diffuser inlet. The variation in SPR coefficient against the normalized flow coefficient at different leading edge positions is shown in Fig. 8. The static pressure coefficient values for LSVD and TVD are improved than VLD at design flow coefficient. The uppermost pressure recovery coefficient of LSVD is lower than that of TVD. TVD offers better flow assistance and thereby decelerate the flow more efficiently than LSVD or VLD; hence, it shows a better value of C_p. At off-design flow conditions the C_p

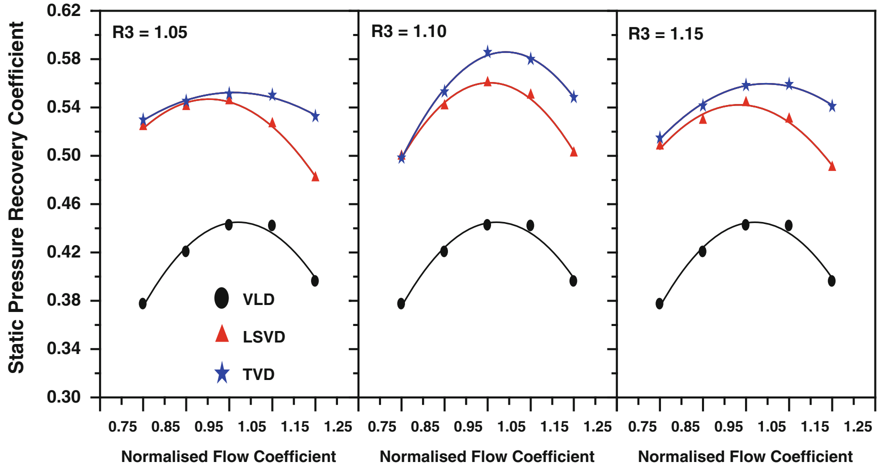


Fig. 8 Static pressure recovery coefficient variation

value is found to be decreasing for the vane-type diffuser configurations. When the diffuser vane leading edge is moved to R_3 = 1.10, C_p varies similarly with R_3 = 1.05, but its magnitude is higher except at low flow coefficient. The C_p values turn out to be even lesser when the leading edge is shifted to R_3 = 1.15.

Static pressure in the diffuser is non-dimensionalized using diffuser inlet dynamic pressure. Figure 9 shows static pressure variation at 80, 100 and 120 % mass flow rate from the diffuser inlet to outlet in the streamwise direction. These values are taken at mid-span location. At all radius ratio considered, the static

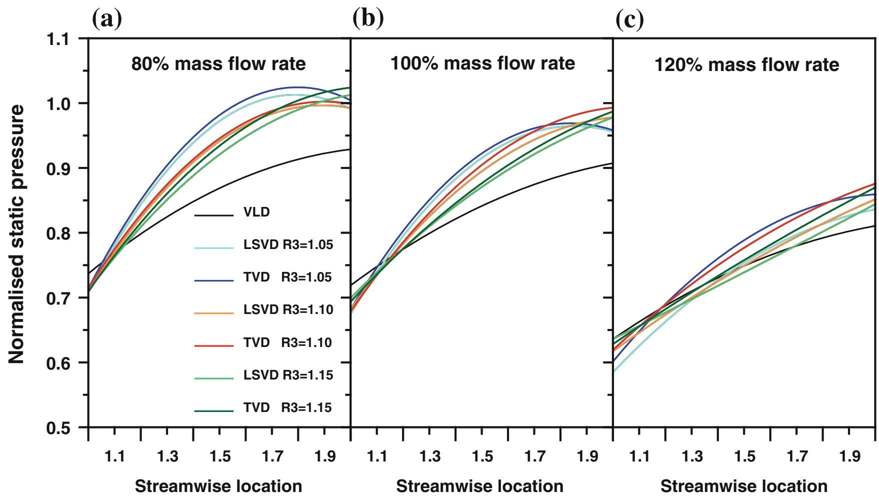


Fig. 9 Variation of normalized static pressure in the diffuser

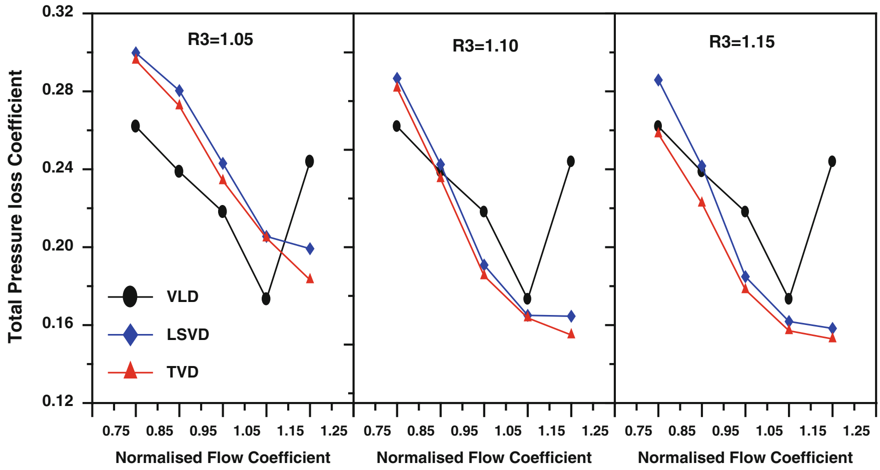


Fig. 10 Comparison of total pressure losses

pressure rise of TVD is more compared to LSVD. Peak pressure rise is observed for TVD at design mass flow rate for radius ratio 1.10. At $R_3 = 1.10$ the static pressure values of TVD are much higher as compared to $R_3 = 1.05$ and 1.15. This shows that there will be an optimum radius ratio for static pressure rise.

(b) Total Pressure Loss Coefficient: Figure 10 shows total pressure loss coefficient (TPLC) variation with flow coefficient. It is observed that the total pressure loss coefficient increases with decrease in normalized flow coefficient for all the three cases of diffusers studied. It is observed that as the mass flow rate increases, flow angle increases, TPLC decreases. At all interaction levels minimum TPLC occurs for TVD compared to LSVD. This shows the superiority of TVD over LSVD. As the radial gap increases the flow is stabilized and TPLC decreases.

6 Conclusions

CFD investigations have been carried out to study the effect of radial distance on the stage performance of a centrifugal compressor for varying mass flow rates. Three types of diffusers are used for this study at five different flow coefficients. The following conclusions are made from the computational analysis for the chosen configurations. At design mass flow rate, maximum efficiency is observed for $R_3 = 1.10$ among the three different radial positions investigated for all vane type diffuser configurations. Radial gap has negligible effect on the power coefficient. When the diffuser vanes are positioned at $R_3 = 1.15$, the effect of interaction on the stage efficiency is insignificant. However, the stage efficiency observed to be affected as the diffuser vanes are brought closer to the impeller tip. For all the diffusers TPLC is higher for low design flow coefficient at all radius ratios. It is observed that as the mass flow rate increases, flow angle increases, TPLC decreases. At all interaction levels minimum TPLC occurs for TVD compared to LSVD. This shows the superiority of TVD over LSVD.

Acknowledgments The authors thank the Principal and Management of Vasavi College of Engineering for their cooperation in carrying out this work.

References

1. Senoo, Y.: Japanese patent application disclosure 119411/78, Oct (1978)
2. Krain, H.: Swirling impeller flow. ASME J. Turbomach. **110**, 122–128 (1988)
3. Rodgers, C.: The performance of centrifugal compressor channel diffusers, ASME Paper 82-GT-10 (1982)
4. Clements, W., Artt, D.: The influence of diffuser vane leading edge geometry on the performance of a centrifugal compressor, ASME Paper 89-GT-163 (1989)
5. Anish, S, Sitaram, N: Computational investigation of impeller-diffuser interaction in a centrifugal compressor with different types of diffusers. In: Proceedings of the IMechE vol 223 Part A: Power and Energy, pp. 167–178 (2008)
6. Anish, S., Sitaram, N., Kim, H.D.: A numerical study of the unsteady interaction effects on diffuser performance in a centrifugal compressor. J. Turbomach. **136**(1), 011012 (2013). doi:10.1115/1.4023471
7. Abdelwahab, A.: On the use of three-dimensional airfoil-shaped vaned diffusers with industrial centrifugal compressors. In: 4th AIAA Theoretical Fluid Mechanics Meeting 6–9 June 2005, Toronto, Ontario Canada
8. Venkateswara Rao, P., Ramana Murty, G.V., Venkata Rao, G.: Performance comparison of twisted low solidity vaned diffusers with different shapes. In: Proceedings of ASME Turbo Expo 2014: Turbine Technical Conference and Exposition, GT2014, June 16–20, Düsseldorf, Germany (2014)
9. Ansys CFX—Solver Modeling Guide—Domain interface Modeling
10. Meakhail, T., Park, S.O.: A study of impeller-diffuser-volute interaction in a centrifugal fan, ASME Paper No.GT-2004-53068 (2004)
11. Siva Reddy, T.Ch., Ramana Murty, G.V., Mukkavilli, P., Reddy, D.N.: Effect of the setting angle of a low-solidity vaned diffuser on the performance of a centrifugal compressor stage. Proc. Inst. Mech. Eng., Part A: J. Power Energy **218**, 637–646 (2004)