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Aerodynamic Studies in the Static Components of a Centrifugal Compressor Stage

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ABSTRACT: Aerodynamic studies in the static components of a centrifugal compressor stage were conducted using the CFD solver FLUENT. A typical centrifugal compressor stage geometry having a flow coefficient of 0.053 was chosen for the simulation study. The study is confined to the static components of the centrifugal compressor stage like crossover bend (180° U-bend), a radial cascade of return channel vanes and exit ducting (90° L-turn). The aerodynamic performance is reported in terms of total pressure loss coefficient, static pressure recovery coefficient, return channel vane surface static pressure distribution and stage exit swirl angle distribution. The simulated flow through the static components covered five different operating conditions of the actual centrifugal compressor stage namely, design point (with 100% flow rate) and off-design operating conditions (with 70%, 80%, 110% and 120% flow rates). The standard k-ε model was used with standard wall functions to predict the turbulence. A minimum total pressure loss coefficient was observed near the 80% flow rate when the average flow angle at U-bend inlet was 24°. The static pressure recovery was seen to be better with 70%, 80% and 100% flow rates. The swirl angle distribution at the stage exit was observed to be satisfactory.

KEYWORDS: Return channel vanes, total pressure loss coefficient, static pressure recovery coefficient, vane surface static pressure coefficient, swirl angle

I. INTRODUCTION

The development of modern centrifugal compressors demands extensive optimization of all the flow path components. In a multistage centrifugal compressor the pressurized fluid from one stage is passed on to the subsequent stage through a 180° circumferential U-bend, a cascade of de-swirl vanes known as “return channel vanes” and L-turn ducting with a 90° bend for connection to the subsequent stage. The function of return channel vanes is to guide the swirling flow leaving the diffuser to the eye of the subsequent impeller stage with near zero inlet swirl. The flow through U-bend and its interaction with downstream row of de-swirl vanes presents a complex fluid dynamics problem. Author [1-10] presented some design guide lines for the design of return channel passages of a centrifugal compressor stage.

Author [11-20] reported the experimental and numerical studies in the crossover system of a typical centrifugal compressor stage with two different return channel vane configurations namely RCV1 and RCV2. In their study the static components of the centrifugal compressor stage like crossover bend, cascade of return channel vanes and the exit L-turn ducting together was referred as “crossover system”. They conducted experimental investigations in a static model of 1:1 size simulating the actual flow conditions at the design and off-design operating conditions. The numerical studies were conducted by simulating the same flow conditions for the purpose of comparison of the results. A good agreement between the experimental and numerical results was reported on the performance of the crossover system in terms of total pressure loss coefficient, static pressure recovery coefficient and vane surface pressure distribution. RCV1 configuration was observed to give better performance than RCV2. Author [21-35] conducted experimental investigations on a static test rig with three different geometries of return channel vanes. They reported about the nature of flow taking place through the return vane channel vanes and emphasized the need to describe the flow with the aid of simplified calculation models. In a similar fashion [36-45] conducted experimental investigations on static test rig and reported the presence of secondary flow in U-turn and exit L-turn sections. They concluded that two dimensional blades are adequate for the De-swirl vanes. They also concluded that the losses in deswirl vanes are mainly due to the separation on vane surface. Author [46-54] conducted CFD studies and showed that the deceleration of the flow introduces large separations and recirculation which decreases the efficiency. They also demonstrated about the qualitative agreement between the CFD results and experimental data. The inverse design and optimization of a multistage radial compressor stage consisting of a vaneless diffuser, cross over bend and return channel was presented

by [55-60]. They also studied the impact of vane lean on secondary flows and showed performance improvements with negative lean. Oh et. al (2005) solved the U-bend problem using FLUENT solver. It is shown that the wake/jet flow coming into the inlet of the U turn bend develops the secondary flow in the downstream side. An inverse method to design a circular cascade for a return channel of a centrifugal turbo machine, whose vane height varies in the radial direction, was developed by [61-64] using singularity method. They also developed a circular cascade model and tested its performance experimentally. They concluded that the flow pattern through the cascade is very complicated owing to the interference between the secondary flow on the end wall and the flow separation on the vane surface.

In the present study the aerodynamic performance of the static components like U-bend, cascade of return channel vanes and exit L-turn ducting with a modified return channel vane configuration namely RCV3 is presented. The aerodynamic performance of RCV3 is compared with that of RCV2 and RCV3 reported by [65-68]. The numerical solution with RCV3 configuration is obtained with the same operating conditions used in the study of RCV1 and RCV2. The computational details are described in the following section.

II. COMPUTATIONAL DETAILS

The numerical solution obtained by solving the Reynolds-Averaged Navier-Stokes (RANS) equations in relation to the present problem and solved by the code is given as follows.

$$1. \quad \text{Conservation of mass} \quad \frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0 \quad (1)$$

$$2. \quad \text{Conservation of momentum} \quad \frac{D u_i}{D t} = - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\mu \frac{\partial u_i}{\partial x_j} + \rho u_i u_j' \right) - \delta_{ij} \frac{\partial u_k}{\partial x_k} \quad (2)$$

Modeling Details

The model is created using GAMBIT preprocessor with three different blocks namely crossover bend, return channel vane and exit L-turn ducting. Three dimensional sector models are used as the flow passages are axi-symmetric. This procedure also minimized the computer memory requirement and allowed grid refinement in critical regions. The crossover bend and the L-turn ducting were meshed with structured hexahedral volumes. The return channel vane sector was meshed with unstructured hex/wedge volumes. Boundary layers were created on the vane surface to resolve the flow conditions at critical regions where flow separation is likely occur. The three blocks were coupled with “interface” feature available in the program. The meridional view of the flow path is shown Figure 1. The meshed geometry of the three blocks is shown in Figure 2. Grid independence studies were carried out to determine the

optimum grid size. The details of grid independence studies are shown in Table 1. The grid generated with an interval spacing of 0.002 m is used in all simulations as it was established as the optimum grid interval size. The design details of return channel vane configuration RCV3 is shown in Table 2. The flow path width at various locations is shown in Table 3.

Table 1: Grid independence studies

S No.	Grid interval size, metres	Total number of elements	Total pressure loss coefficient
1	0.003	72610	0.50185
2	0.0025	131950	0.45027
3	0.002	251460	0.4259
4	0.0015	497232	0.4230

Table 2: Flow Path Dimensions

Location	Flow path width
U-bend inlet	24.5 mm
U-bend exit	20.5 mm
L-turn exit	47.5 mm

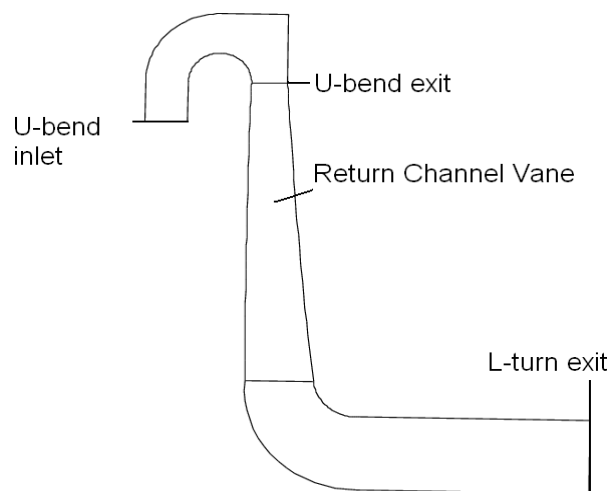
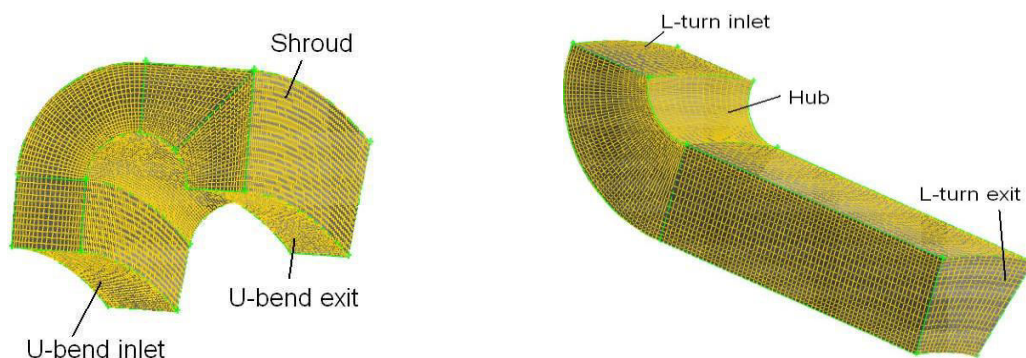


Figure 1: Meridional View of the Flow Path



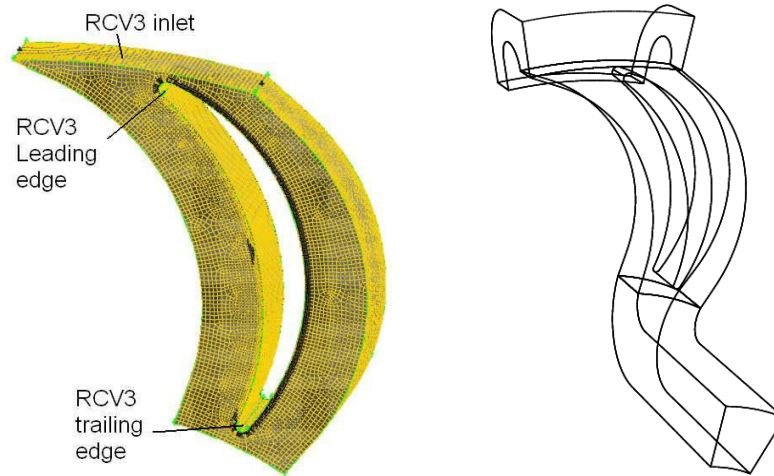


Figure 2: Meshed geometrical components and the combined model

Table 3: Vane Parameters

Parameter	Value
Inlet flow angle	25°
Exit flow angle	82°
Chord length	178 mm

Solver Settings

Pressure based solver with implicit formulation under 3-D steady flow conditions with absolute velocity formulation is chosen for the present study. Author [69] conducted numerical studies of U-bend in return channel systems of multistage centrifugal compressors. They used Reynolds stress turbulence model and also two equation $k - \epsilon$ turbulence model. In their study $k - \epsilon$ model predicted the turbulence closer to the experimental observations. In the present study the $k - \epsilon$ model is used to predict the turbulence. Standard wall functions were used to capture the flow phenomenon near the walls. At the inlet section “Total Pressure” was specified along with flow component directions. Also the density of air is specified. At the exit section of 90° bend the “static pressure” with “radial equilibrium pressure distribution” option with target mass flow rate is used as outlet boundary condition. The solution is assumed to converge when the maximum residual values are equal to 1e-06. The operating conditions (the average flow angle at U-bend inlet and the corresponding mass flow rate) at which the simulations were carried out were taken based on the experimental investigations reported by [70-74].

III. RESULTS AND DISCUSSION

Figure 3(a) shows the variation of meridional velocity from hub to shroud at U-bend exit for RCV3 configuration. Almost uniform velocities are observed from hub to shroud at all operating conditions. Higher magnitudes of meridional velocities are observed at this location than at U-bend inlet at the given operating conditions. The increase in meridional velocity from inlet to outlet of U-bend may be attributed to the narrowing of flow path width at the exit of U-bend. The distribution of circumferentially averaged flow angle distribution from hub to shroud for RCV3 configuration is shown in Figure 3(b). A very uniform distribution in flow angle is observed at all the studied operating conditions. The magnitude of average flow angle is seen to be higher at U-bend exit than at U-bend inlet at the given operating conditions. The increase in flow angle may also be attributed to the decreased flow path width at U-bend exit. The decreased flow path width causes the meridional component to increase and hence the increase in flow angle.

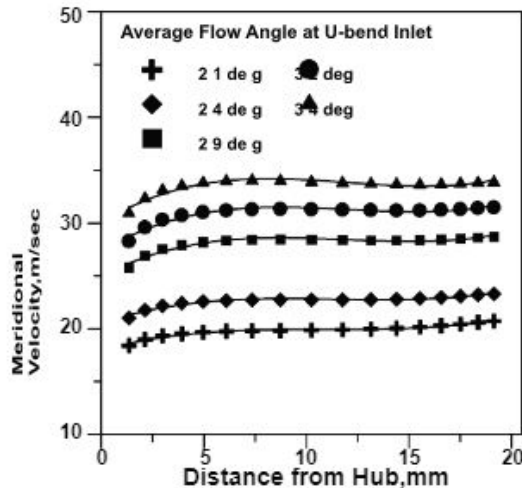


Figure3 (a): Meridional Velocity Distribution at U-bend exit

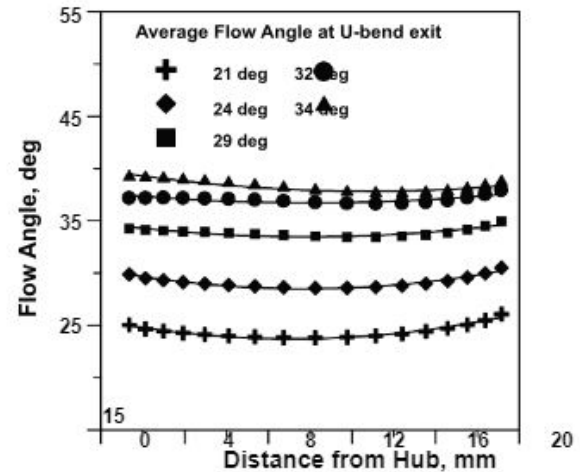


Figure3 (b): Flow Angle Distribution at U-bend exit

The vane surface pressure coefficient plotted against the %chord length from the leading edge is shown in Figure 4. The vane surface static pressure coefficient is obtained by dividing the static pressure difference between the local static pressure on vane surface and the average static pressure at U-bend inlet with average dynamic pressure at U-bend inlet. The vane is observed to be uniformly loaded at the design point (average flow angle at U-bend inlet = 29°). With increase in negative incidence on the leading edge of the RCV3 configuration, acceleration of the fluid is observed up to 30% chord length from the leading edge on suction side and there after deceleration of flow resulting in the recovery of static pressure towards the trailing edge. On the pressure side immediately after the L-turn exit a sudden drop in static pressure is noted at about 5% chord length and there after changing to positive static pressure values. This indicates the possible flow separation at this zone and there after reattachment to vane pressure surface particularly at higher negative incidence angles.

The distribution of circumferentially averaged meridional velocity from hub to shroud at the L-turn exit is shown in Figure 5. The meridional velocity is seen to be more concentrated towards the hub and is decreasing in magnitude towards the shroud. Due to the increased mass flow rate with increase in average flow angle at U-bend inlet increasing meridional velocities are observed. The L-turn is causing the flow to be concentrated towards the hub.

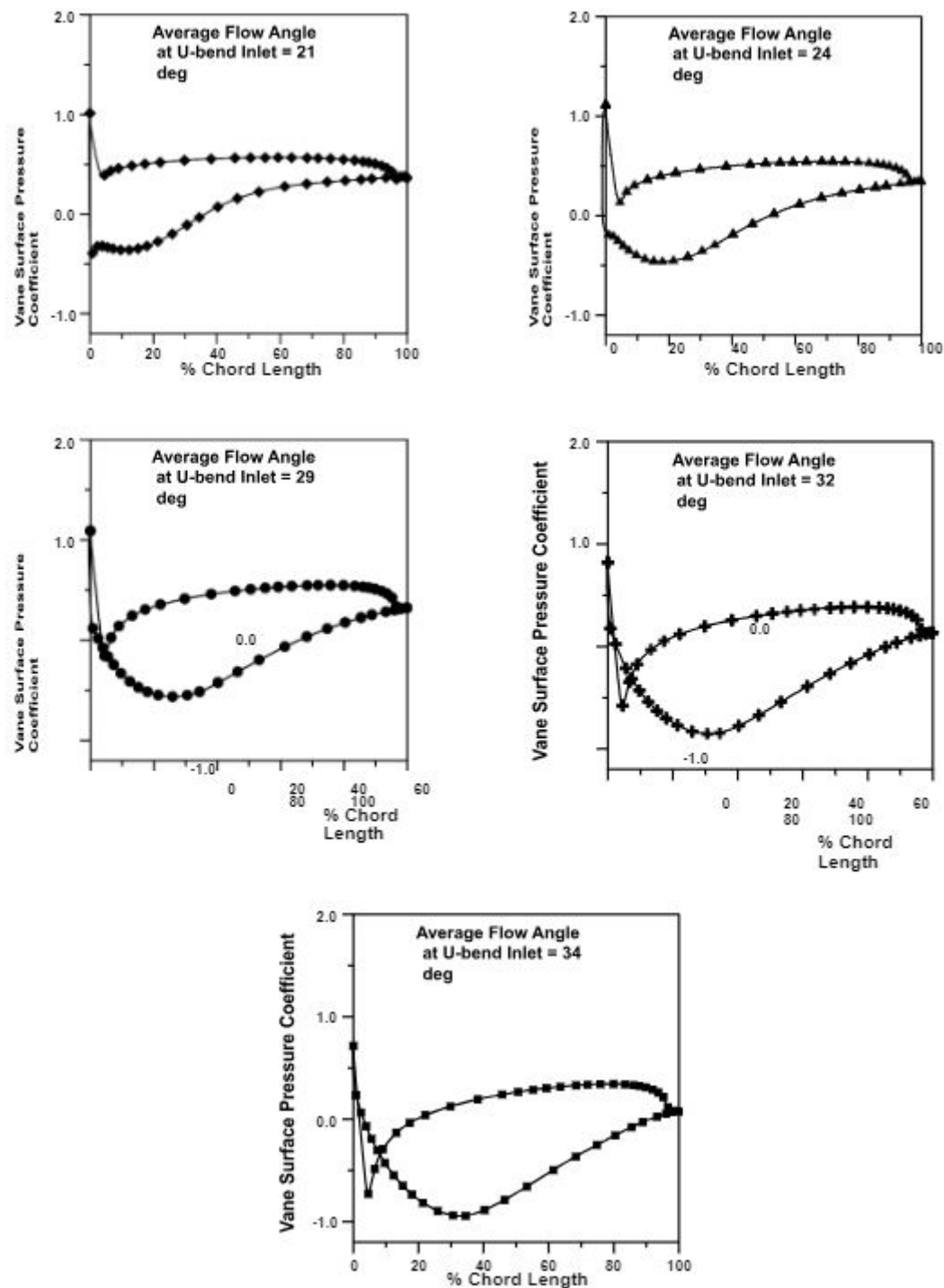


Figure 4: Vane Surface Static Pressure Distribution for RCV3 Configuration

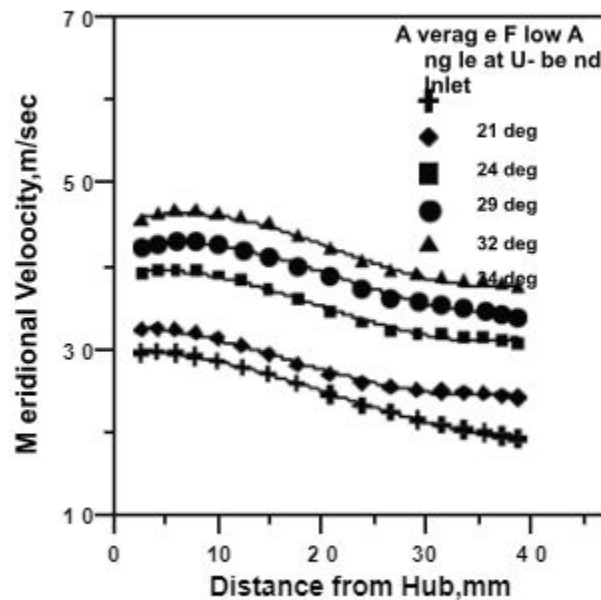


Figure 5: Meridional Velocity Distribution at L-turn exit

Figure 6 shows the variation meridional velocity along the circumference at the L-turn exit section for the design point operating condition. A very uniform meridional velocity distribution is observed near the hub i.e., at a distance of 4.9mm from the hub. With the increase in distance from the hub, the meridional velocity is seen to be decreasing in magnitude and also to vary in a wavy fashion. The wavy nature observed to subside towards the hub. The L-turn bend appears to be causing the flow to migrate towards the hub. The vane trailing edge appears to have an effect on the meridional velocity distribution in causing the wavy nature of variation.

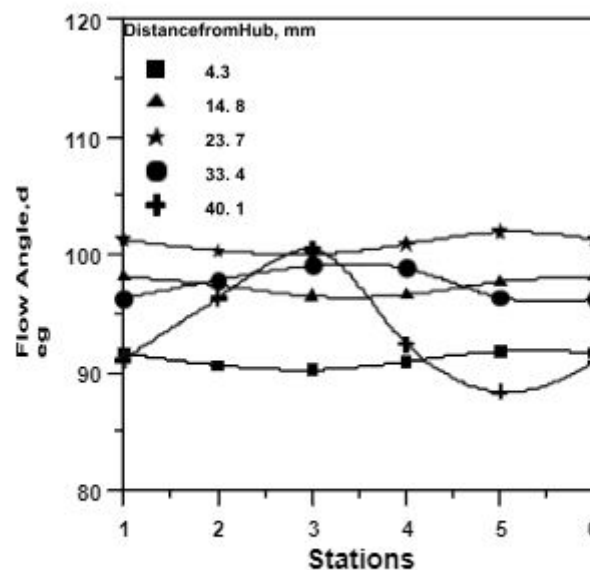


Figure 6: Meridional Velocity Distribution at L-turn exit (Design Point)

Figure 7 shows the variation of flow angle (swirl angle) at L-turn exit section along circumference at the design point. The flow angles are measured with respect to tangential direction. Negative swirl (swirl angle greater than 90°) is observed in the mid- region of the L-turn section that is at distances of 14.8, 23.7, and 33.4 mm from the hub with in the range of 96° to 102° . Near the hub a very uniform and near zero swirl angle distribution is observed. However at distance of 40.1mm from hub the swirl angle is observed to fluctuate between 88° and -101° in a wavy fashion. The RCV3 vane trailing edge appears to have an effect on the nature of swirl angle distribution.

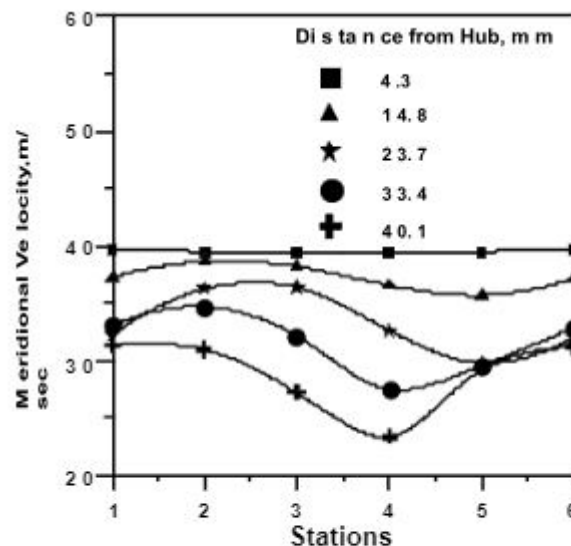


Figure 7: Swirl Angle Distribution at L-turn exit at the Design Point

Figure 8 shows the variation of total pressure loss coefficient for RCV3 which is also compared with that of RCV1 and RCV2 reported. The total pressure loss coefficient is obtained by dividing the average total pressure difference between L-turn exit and U-bend inlet with the average dynamic pressure at U-bend inlet. The total pressure loss coefficient is observed to be minimum for an average flow angle of 24° at U-bend inlet. The total pressure loss is seen to increase with increase in flow angle at U-bend inlet. The total pressure loss occurs due to the skin friction between the walls and the fluid and also due to the internal friction arising due to secondary flows.

The variation of static pressure recovery coefficient with the average flow angles at U-bend inlet is shown in Figure 9. The static pressure recovery coefficient is obtained by dividing the average static pressure static pressure difference between the L-turn exit and U-bend inlet with the average dynamic pressure at U-bend inlet. The static pressure is expected to recover well at lower incidence angles. With the increase in incidence angle on the negative side (increase in flow angle at U-bend inlet) there is a decrease in static pressure recovery. At higher negative incidence angles the static pressure recovery coefficient is observed to be negative. The performance of the stage in terms of total pressure loss coefficient and the static pressure recovery for RCV3 is observed to be superior to RCV1 and RCV2.

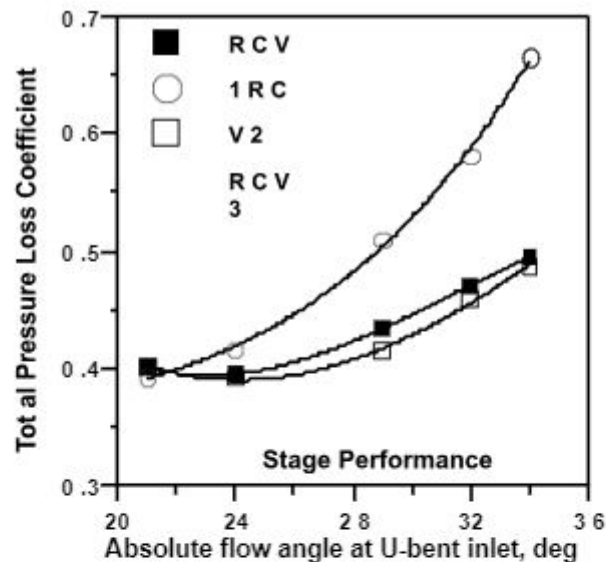


Figure 8: Variation of total pressure loss coefficient

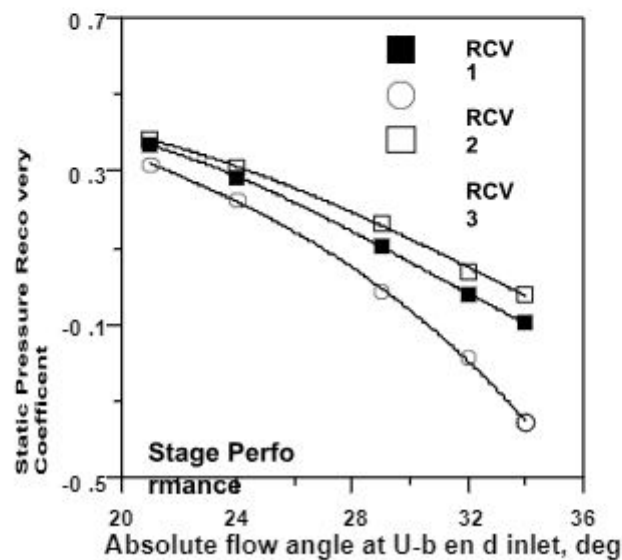


Figure 9: Variation of static pressure recovery coefficient

Qualitative plots

The flow separation phenomenon observed in the case of RCV1 and RCV2 studies [75] occurring on the suction side of the return channel vane trailing edge is almost eliminated in the case of RCV3. Figure shows the qualitative velocity vector plots depicting the flow phenomenon on the suction side of the trailing edge at the design point. Figure 10(a) shows the velocity vector plots on a plane passing the mid span of the vane for the three vane configurations. Figure 10(b) shows the magnified view of

the flow phenomena on the suction side of the trailing edge. The flow separation phenomena is observed to be eliminated in the case RCV3 configuration as can be seen in the magnified vector plots.

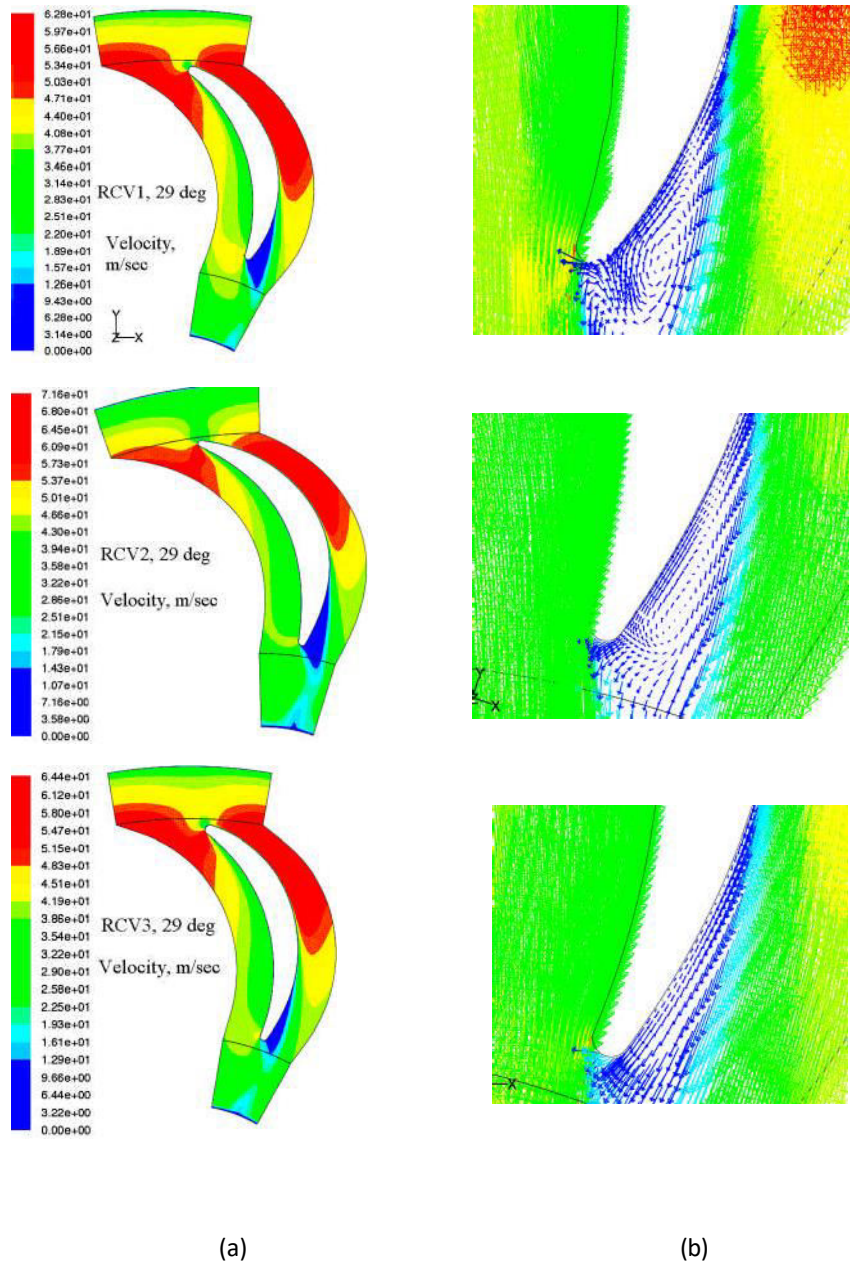


Figure 10: Velocity Vector plots on a plane passing through the mid span of the vane

IV. CONCLUSIONS

A numerical study was conducted to study the aerodynamic performance of the static components of centrifugal compressor stage with a vane configuration namely RCV3. A very uniform velocity and flow angle distributions were observed at the U-bend exit. The flow separations phenomenon that was seen in RCV1 and RCV2 configurations is eliminated with RCV3 configuration. The meridional velocity and swirl angle distributions are observed to be influenced by the return channel vane trailing edge. The stage exit swirl angle distribution is observed vary between 80° and 102° . The present study is found to be helpful in understanding the nature of flow through return channel passages of a centrifugal compressor stage.

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