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To Study And Analyze the Variation of Pressure Recovery Coefficient at the Walls of Axial Annular Diffuser at Area Ratio 2.5 and Casing Diverging Angle of 6° using CFD

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ABSTRACT

The present work relates to the analysis of flow through the parallel hub and diverging casing annular diffuser. In this analysis variation of pressure recovery coefficients have been visualized at the diffuser walls with and without swirl. The casing angle and area ratio have been taken as 6^0 and 2.5 .The swirl angle was varied from $0^0$to 25^0 at the inlet of diffuser. The analysis was carried out for flow regime with various experimentally obtained inlet velocity profiles with or without swirl [1]. Pressure coefficients have been calculated along the casing and hub walls of axial annular diffuser. In the present work CFD approach was used to find the results by using $RNG\ \ k-\varepsilon$ turbulence model.

Keywords: *Annular diffuser; Pressure Recovery Coefficient; Area Ratio; Casing Diverging Angle and CFD.*

1.0 Introduction

A diffuser is a device which decelerates the flow and to regain total pressure or device which increases the pressure of a fluid at the expense of its kinetic energy. Diffuser is also used to maintain the uniform flow at the exit. For obtaining the uniform flow at the exit the magnitude of the secondary components should be less than 10%.

Annular diffuser is generally used in turbo machines where fluid may have to flow through a hub or a central shaft. It was well demonstrated that the diffusers of annular type are complex in nature due to the presence of inner wall which makes the flow complex through annular diffuser.

In the annular diffuser there are many unknown parameters which are interconnected, by changing the one parameter whole setup has to be changed. So it is very difficult to perform experimental works on annular diffuser. To overcome this problem Numerical methods are generally used which are less costly and need less time. So Computational Fluid Dynamic approach is generally used to analyze the flow through annular diffuser.

Performance of diffuser depends upon geometrical as well as dynamical parameters. Geometrical parameters are inlet length, size of duct, area ratio of the diffuser, divergence angle, diffusion length of the diffuser, shape of the exit duct with free or submerged discharge conditions. Dynamic parameters are inlet velocity profile, boundary layers parameters, Reynolds Number and Mach Number etc. In annular diffuser maximum pressure recovery is achieved within shortest possible length due to presence of hub and internal surface to guide the flow outwards.

Sovran & Klomp (1), Shrinath (2), Hoadley (3), Colodipietro et al (4), Shaalan & Shabaka (5), Kumar (6), Lohmann et al (7), Sapre et al (8), Agrawal et al (9), Singh et al (10), Kochevsky (11), Mohan et al (12), Japikse (13) and Yeung & Parkinson (14) showed that diffuser performance depends on swirl velocity and increases with introduction of swirling flow. Pressure recovery of diffuser increases up to a certain point after that it deteriorated with swirling flows.

The effectiveness and pressure recovery coefficient of annular axial diffusers decreases as separation occurs on the walls of the diffuser i.e. at the casing and on the hub surface. The separation of the flow can be shifted or delayed from one position to another by introducing the swirling flow. The efforts have been made by Howard (15), Stevens (16), Singh et al (17) developed an annular diffuser without flow separation, however they have successful up to some extent. Manoj Kumar, B.B.Arora [18] analyzed the flow in annular diffuser

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by using CFD technique and they calculated the effect of swirl on the flow behavior inside annular diffuser and found that the flow was moved towards the casing with the introduction of swirling flow. They also calculated the effect of swirl on pressure recovery coefficient. Ozturk Takar, Ali Pinarbasi [19] analyzed the flow in centrifugal compressor vaneless diffuser. They used the computational Fluid Dynamics approach to analyze the flow and calculated the different parameters such as velocity, Pressure and turbulent kinetic energy at different hub sections. CFD work was performed by using Fluent and Gambit. Rita J Schnipke et al [20] analyzed a vane diffuser by applying Finite element analysis. In this method a model geometry and meshing was prepared in the Gambit, then this mesh model is exposed to Finite Element Analysis. Here continuity and momentum equations are discretized by using Galerkins method. Finally the pressure contours and velocity contours are plotted. Majid Nabavi [21] analyzed the 3-D asymmetric flow through a planar diffuser. Here the flow was incompressible and diffuser with gradual expansion was used. The numerical approach was used to find the results and finally compared with experimental work. In this paper the effect of divergence angle, Reynolds number and aspect ratio on the flow asymmetry were calculated. Manoj Kumar, B. B. Arora [22] analyzed the flow in vaneless diffuser by using CFD approach. They found the effect of inlet swirl and area ratio on the performance of annular diffuser. They presented the results in terms of non-dimensional longitudinal velocity, swirl velocities, static pressure and total pressure. Dr. Basarat salim [23] analyzed the wide angle diffusers experimentally. The effect of area ratio and diffuser divergence angle were checked on the performance of asymmetric rectangular wide angle diffusers. Sparrow et al [24] investigated fluid flows in a conical diffuser with the help of 3-D numerical model. They found that symmetric flow separation occurred for the diffuser angle of 5^0 and Reynolds number less than 2000.Results for the 10^0 and 30^0simulation showed symmetric separation at all investigated Reynolds Numbers (5000-33000). R.Keerthana et al [25] analyzed a series of annular diffusers of divergence angle 9^0, 15^0, 21^0 and 27^0 by using CFD approatch.PRO-E and ANSYS FLUENT were used. Here results showed that pressure recovery increases by increasing the diffuser angle. Stefano Ubertini [26] analyzed experimentally the annular diffuser of an industrial gas turbine, measurements were performed on a scale model of 35% with and without the struts. The results were presented in terms of flow angle, Static pressure, total pressure and wall static pressure. Manoj Kumar gopaliya [27] presented the effect of offset on S shaped diffuser with $90^0/90^0$ turn. The diffuser has rectangular inlet and semicircular outlet with aspect ratio 2.For the analyses a software code on Finite volume Technique using k- ε turbulent model was employed and to modify the flow .The finally results show that outlet pressure recovery decreases with increase in non-uniformity at the exit due to offset.. Manoj Kumar gopaliya [28] presented the effect of horizontal and vertical offset on a Y shaped duct. Here the Y shaped duct has rectangular inlet and circular outlet with area ratio=2.The settling length was 1.5D at the $Re=2.74x10^5$.In this paper a computer based program on finite volume technique using k- ε turbulence model was employed to analysis the problem. The results obtained from this study indicate reduced outlet pressure recovery accompanied with increase in non-uniformity in the flow at the exit contributed by the offset effect Ali pinarbasi[29] measured the level of turbulence experimentally in the different plane of a centrifugal compressor vaneless diffuser .In this paper detailed flow measurements at the inlet of a centrifugal compressor vaneless diffuser are presented .Alysson et al[30] investigated local flow turbulence and velocity profiles by using four turbulent models for a radial air diffuser. For all the turbulence models ANSYS-CFX software was used. It was shown that for the turbulent flow through diffuser shear stress model (SST) is a good choice. D. P. Agrawal et al [31] analyzed the flow in a vaned radial diffuser and calculated the velocity distribution in the blade to blade plane. In the present paper both the experimental and numerical methods were performed to calculate velocity variation. Finally results show the good agreement between the Experimental and Numerical methods**.** F Frust [32] calculated the flow of fluid at the exit of planer diffuser with 3:1 suddenly expansion ratio in a duct by using Laser Doppler Anemometry experimentally. They measured the symmetric velocity profiles at Reynolds number of 56, 114 and 252.

Chithambaran et al (1984) and Buice et al (1997) showed that experimental analysis of annular diffuser is costly because it involves precision instrumentation and takes lot of time. Due to this Computational Fluid Dynamics is more economical tool to analyze the flow through annular diffuser accurately.

In the present investigation the computational Fluid dynamics (CFD) approach by using Fluent has been used for the detailed flow analysis in axial annular diffuser with parallel hub and diverging casing. Casing angle and area ratio have been taken as 6° and 2.5. The swirl angle has been varied from 0 to 25^0 at the inlet of diffuser.

In this paper we have calculated the variation of longitudinal velocity and swirl velocity in non-dimensional form at different diffuser passage heights at x0.1, x0.3, x0.5, x0.7 and x0.9 along the axial direction .Then development of flow, flow reversals and flow separation through the diffuser passage heights was analyze. Here RNG k- ε turbulent model have been used to solve the problem numerically due to the closeness of experimental results [1].

2.0 CFD Modelling

Annular diffuser geometry was drawn with proper meshing scheme with the help of ANSYS-15.0 module. Here *k*- ε , (RNG) and realizable turbulent model was used. The boundary conditions at the inlet is the same velocity profile as experimentally obtained with turbulence intensity of 3% and hydraulic diameter 7.76cm.

The outlet boundary condition is zero gauge pressure normal to the outlet boundary with turbulence intensity of 3%. Here the second order up winding scheme is used for momentum, swirl velocity, turbulence kinetic energy and turbulence dissipation rate.

The convergence criteria for residuals was 10^{-6} for various parameters involved in the present study such as continuity, axial velocity, radial velocity, swirl velocity, turbulent kinetic energy and dissipation rates.

2.1 Governing equations

The governing equations for 2D axisymmetric geometries with swirl are given below from the reference [1].

Conservation or Continuity equation may be written as follows :

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho \vec{v}) = S_m \qquad (1)$$

This is the general equation of continuity and it is applicable for compressible as well as incompressible fluid. The source S_m is the mass added to the continuous phase from the dispersed second phase (due to vaporization of liquid droplets) and any user defined sources.

For 2-D ax symmetric Geometries the mass balance equation is as follows [1]

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_x)}{\partial x} + \frac{\partial(\rho v_y)}{\partial x} = S_m \qquad \dots (2)$$

Here x and r are axial and radial directions, v_x is the axial velocity and v_y is the velocity in the radial direction.

Momentum Equation:

Momentum equation in the general form can be written as follows:

$$\frac{\partial(\rho \vec{v})}{\partial x} + \nabla.(\rho \vec{v}\vec{v}) = -\nabla p + \nabla.(\bar{\bar{\tau}}) + \rho.\vec{g} + \vec{F} \qquad \dots (3)$$

Here p represents the static pressure, τ shows the stress tensor, $\rho.\vec{g}$ shows the gravitational Force and $\vec{F}$ shows the body force. The stress tensor is given by the equation:

$$\bar{\bar{\tau}} = \mu[(\nabla\vec{v} + \nabla\vec{v}^r - \frac{2}{3}\nabla.\vec{v}I)] \qquad \dots (4)$$

Here μ shows the viscosity, I is the unit tensor and the second term on the right hand side is the effect of volume dilation.

For 2D ax symmetric geometries the axial and radial momentum equations can be written as

$$\frac{\partial(\rho v_x)}{\partial x} + \frac{1}{r}\frac{\partial(r\rho v_x v_x)}{\partial x} + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r v_x) = -\frac{\partial p}{\partial x} + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left\{2\frac{\partial v_x}{\partial x} - \frac{2}{3}(\nabla.\vec{v})\right\}\right] + \frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(\frac{\partial v_x}{\partial r} + \frac{\partial v_r}{\partial x}\right)\right] + F_x \qquad \dots (5)$$

$$\frac{\partial}{\partial t}(\rho v_r) + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_r) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r v_r) = -\frac{\partial p}{\partial r} + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(\frac{\partial v_r}{\partial x} + \frac{\partial v_x}{\partial r}\right)\right] + \frac{1}{r}\frac{\partial}{\partial r}[r\mu\left(2\frac{\partial v_r}{\partial r} - \frac{2}{3}(\nabla.\vec{\mathrm{v}})\right] - 2\mu\frac{v_r}{r^2} + \frac{2}{3}\frac{\mu}{r}(\nabla.\vec{v}) + \rho\frac{{v_z}^2}{r} + F_r \qquad \dots (6)$$

Where $\nabla.\vec{v} = \frac{\partial v_x}{\partial x} + \frac{\partial v_r}{\partial r} + \frac{v_z}{r}$

Where v_z is the swirl velocity

The tangential momentum equation for 2D swirling flows may be written as

$$\frac{\partial}{\partial t}(\rho v_{z)} + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_z) + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_z \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\frac{\partial v_z}{\partial x}\right] + \frac{1}{r^2}\frac{\partial}{\partial r}\left[r^3\mu\frac{\partial}{\partial r}\left(\frac{v_z}{r}\right)\right] \qquad (7)$$

3.0 Turbulence Modelling

Turbulent flows are described by fluctuating velocity components. These components mix with transported quantities such as momentum, energy and species concentration and cause the transported quantities to fluctuate as well. Since these fluctuations can be of small scale and high frequency, they are too computationally costly to simulate directly in practical engineering calculations. The instantaneous (exact) governing equation can be time averaged, ensemble averaged or otherwise manipulated to remove the small scales, resulting in a modified set of equations that are computationally less expensive to solve. However, the modified equations contains additional unknown variables and turbulence models are needed to determine these variables in terms of known variables.

Different turbulence models are given below:

3.1 Turbulence models

1. Spalart Allmaras model
2. k-ε models
   (a). Standard k-ε models
   (b). RNG .k-ε model
   (c). Relizable k-ε models
3. k- ω model
   (a) Standard k- ω model
   (b) Shear stress transport (SST) k- ω models
4. Reynolds stress model
5. Large eddy simulation (LES) model

It is very difficult to choose one model to a particular class of problems. The choice of turbulent model will depend on accuracy required, availability of computational resources and the amount of time available for the simulation work. It is very difficult to state the model which is best suited for a specific problem. Here we used RNG k-ε turbulence model to analyze the flow because it gives the results which were very close to the experimental results [1].

4.0 Results

Figures 1 shows the variations of pressure coefficients for the area ratios of 2.5 along the casing of axial annular diffuser. These pressure coefficients variations are shown for inlet swirl angles of 0°, 7.5°, 12°, 17° and 25°.

Figure 1 shows the following Results are given below:

(i) The maximum value of pressure coefficient is 0.651155 at 0^0 swirl angle and increases as the length increases.

(ii) The maximum value of pressure coefficient is 0,646819 at 7.5^0 swirl angle and increases as the length increases.

(iii) The maximum value of pressure coefficient is 0.664129 at 12^0 swirl angle and increases as the length increases.

(iv). The maximum value of pressure coefficient is 0.538173 at 17^0 swirl angle and increases as the length increases.

(iv) The maximum value of pressure coefficient is 0.336835 at 25^0 swirl angle and increases as the length increases initially then decreases.Figures 2 shows the variations of pressure coefficients for the area ratios of 2.5 along the hub of axial annular diffuser. These pressure coefficients variations are shown for inlet swirl angles of 0°, 7.5°, 12°, 17° and 25°. Figure2 shows the following Results are given below:

(i) The maximum value of pressure coefficient is 0.650812 at 0^0 swirl angle and increases as the length increases.

(ii) The maximum value of pressure coefficient is 0.64384 at 7.5^0 swirl angle and increases as the length increases.

(iii) The maximum value of pressure coefficient is 0.657989 at 12^0 swirl angle and increases as the length increases.

(iv). The maximum value of pressure coefficient is 0.521246 at 17^0 swirl angle and increases as the length increases.

(iv) The maximum value of pressure coefficient is 0.287723 at 25^0 swirl angle and increases as the length increases up to some length then constant and at last further increases.

5.0 Conclusions

Following conclusions are drawn for area ratio of 2.5 at various inlet swirl angles:

- As the flow occurs towards downstream, the pressure recovery coefficient increases at a given swirl flow.
- As the swirl flow increases the pressure recovery coefficient decreases at the casing and hub walls.

- At 25^0 swirl angle the pressure recovery coefficient initially increases up to some length then it decreases for the casing of annular diffuser.
- At 25^0 swirl angle the pressure recovery coefficient initially increases at the faster rate up to some length then remain constant and at last it further increases at the slower rate for the hub walls of annular diffuser.
- The maximum value of pressure recovery coefficients occurs at $12^{\ 0}$ swirl angle for both casing and hub walls of the diffuser.

Fig-1: Variation of Pressure Recovery Coefficients at Different Swirl Angles at the Casing Wall of Axial Annular

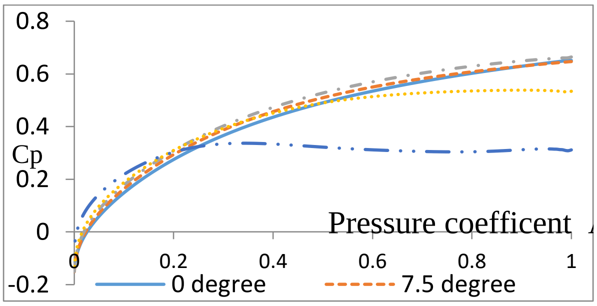
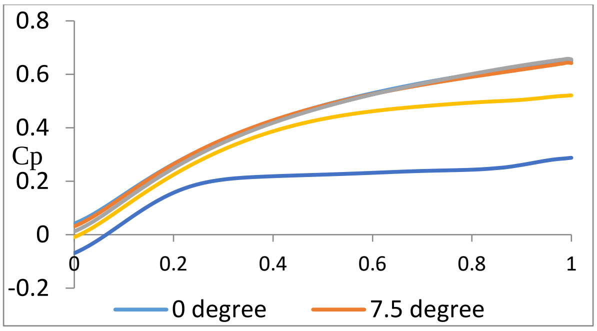


Fig 2: Variation of Pressure Recovery Coefficients at Different Swirl Angles at the Hub Wall of Axial Annular Diffuser.



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