Effect of swirl flow on characteristics of the annular diffuser

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Abstract. The current work deals with the axial flow annular diffuser's performance characteristics with divergent casing angle 9° with an area ratio of 3. We performed the numerical simulation to examine the inlet swirl's effect on the annular diffusers. The velocity profiles were measured along the span of diffusers at a number of locations. Analysis of the results shows that the inlet swirl improves the stability at outlet flow, gains the recovery of pressure, and delays the flow separation on the casing. It is found that static pressure recovery increases by 5% with swirl flow. The results prove that the best performance is observed in the unequal hub and diverging casing diffuser with an equivalent cone angle of 15° and the inlet swirl angle of 12°.

Keywords: annular diffuser; swirl angle; longitudinal velocity profile; static pressure recovery coefficient

1. Introduction

The diffuser is the expanding cross-section, which recovers the fluid's highest possible static pressure by reducing its flow velocity. The flow gets decelerated when it passes across the diffuser, as part of the fluid's kinetic energy is transformed into potential energy of pressure. The effective diffuser is that which changes maximum pressure energy from kinetic energy. In the gas turbine, the central core shaft is required, and the necessity is fulfilled with the annular diffuser. Many researchers have been investigated swirl flow through the diffuser and examine the flow behaviour inside the diffusers [1-2]. Stevens and Williams [3] carried out the work on annular diffusers having uniform diameter centre bodies and diverging casing walls. It was found that increasing the turbulence level at the inlet improvement in the pressure recovery and stable flow at the outlet. Adkins [4] studied the optimum design of annular diffuser geometries, but those methods do not play a vital role in the designing of diffusers because of the complex geometry and flow field. Annular diffusers with swirl flow at inlet have very less literature available, but an extensive range of literature is available for conical diffusers of different geometries. Arora et al. [5] have performed the numerical study of the swirling flow on the annular diffuser by applying the RNG k-ε turbulence model. They have found that flow separation is completely removed from the casing wall by using swirling flow. The paper's primary focus is to analyze the diffuser's performance with fully developed turbulent swirl and non-swirl flow.

1.1 Geometric model

The annular diffuser with casing wall angle 9° and hub wall 5° of area ratio (AR) 3 have been selected for present investigations. The fully developed turbulent flow with swirl angle 0°- 25° is to be chosen to generate swirl velocity type distribution at the diffuser's inlet. Based on the existing literature optimized geometric values of the diffusers are used to carry out the numerical analysis [6, 7, 8]. The annular diffuser's geometric value has been optimized for maximum static pressure recovery and minimum total pressure loss coefficient. Figure 1 and Table 1 show the geometrical configuration details of the unequal hub and diverging casing axial annular diffuser parameters.

Table 1. Geometrical dimension of Unequal Hub and Diverging Casing (UHDC) annular diffuser.

| Geometrical parameters | Dimensions |
|------------------------|------------|
| Area ratio (AR)        | 3          |
| Diffusing axial length (L), cm | 37.55      |
Equivalent cone angle (2$\Theta$)  
Hub inlet diameter, cm 7.6  
Casing inlet diameter, cm 15.5  
Hub wall angle ($\Theta_h$) 5°  
Casing wall angle ($\Theta_c$) 9°

Figure 1. Diffuser geometrical parameter

2. Turbulence model

Equations (1) and (2) are the governing equations for $k$ and $\varepsilon$, and equations (3) to (4) give the auxiliary relations for the RNG $k$-$\varepsilon$ turbulence model. The transport equations are given below:

$$\rho u_i \frac{\partial k}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \alpha_k \mu_{eff} \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon + S_k$$  
(1)

$$\rho u_i \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \alpha_\varepsilon \mu_{eff} \frac{\partial \varepsilon}{\partial x_j} \right] + C_{\mu} \frac{\varepsilon}{k} \frac{G_k}{k} - C_{\varepsilon} \frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon$$  
(2)

$$\mu_{eff} = \rho C_\mu \frac{k^2}{\varepsilon}$$  
(3)

$$R_\varepsilon = \frac{C_\rho \rho \eta^3 \left(1 - \frac{\eta}{\eta_0}\right) \varepsilon^3}{1 + \beta \eta^2}$$  
(4)

Here $\alpha_k$ represents the counter-effective Prandtl number for $k$ and $\alpha_\varepsilon$ represents the counter effective Prandtl numbers for $\varepsilon$. $S_k$ and $S_\varepsilon$ are the source terms defined by the user. The term $\eta = \frac{sk}{\varepsilon}$ and value of the constants are considered in the equation as follows. $C_{\mu} = 1.42; C_{\varepsilon} = 1.68; C_\rho = 0.0845; \eta_0 = 4.38$; and $\beta = 0.012$.

Previous studies conducted on the annular diffusers by (Singh et al., [9], Mohan et al., [7], Arora, [10]) have considered different two-equation turbulence models, such as standard $k$-$\varepsilon$ (Sadasivan et al.,[11]), and RNG $k$-$\varepsilon$ (Arora, [10]) to analyze the flow field variables. Here the RNG $k$-$\varepsilon$ turbulence model has been chosen due to its better performance in predicting the velocity profile against the experimental results of Arora and Pathak [12].
3. Grid independence test

The grid study has been conducted for the unequal hub and diverging casing annular diffuser. The grid's size depends upon the diffuser's geometrical parameters, but different diffusers will have varied grid sizes. Four grids cell counts between 0.9 to 1.9 million are created (coarse grid=0.9×10^5, medium grid=1.2×10^5, fine grid=1.6×10^5, and finer grid =1.9×10^5). The numerical analysis has been performed with swirl flow using the RNG k-ε turbulence model at the Reynolds number 2.5x10^5. It is observed that static pressure coefficient profiles for the fine grid and finer grid match with each other, as shown in Figure 2. Thus fine grid size (1.6×10^5 cells) has been selected to conduct the numerical study.

![Figure 2: Grid independence test](image)

4. Parameters of performance

The most important parameters are evaluated and are given below:

**Static pressure recovery (Cp):**
The amount of kinetic energy transformed into pressure energy in terms of static pressure rise from exit to the inlet section of the diffuser due to diffusing action [13].

\[ C_p = \frac{P_{ex} - P_{in}}{0.5 \rho u_{avg}^2} \]

Here \( P_{ex} \) represents the mass average static pressure at the exit, and \( P_{in} \) represents the mass average static pressure at the inlet section of the diffuser while \( u \) represents the average velocity of flowing fluid.

**Diffuser effectiveness (η):**
It is the ratio of actual recover of the pressure of the diffuser to the ideal recover of the pressure of the diffuser for the similar area ratio.

\[ \eta = \frac{C_p}{C_{pi}} \]

\[ C_{pi} = 1 - \frac{1}{AR^2} \]

Where AR represents the area ratio.
Total pressure loss coefficient ($C_{p_l}$):
The amount of total pressure loss due to turbulent intermixing of fluid due to viscous forces to the inlet
dynamic head.

$$C_{p_l} = \frac{P_{in} - P_{ex}}{0.5 \rho u_{avg}^2}$$

Where $P_{in}$ represents the mass average total pressure at the inlet, and $P_{ex}$ represents the mass average
total pressure at the exit section of the diffuser.

5. Analysis of velocity distribution

In the current work, the effect of swirling and non-swirling flow is evaluated. Moreover, the impact of
the recirculation zone (RZ) on the performance is analyzed. The detailed analysis of streamline
contours of UHDC diffuser at swirl angle 0°-25° is presented in Figure 3. At swirl angles 0°-12° of
UHDC diffuser, velocity contours show symmetry of flow between the walls. As observed from the
figure, there is no flow separation and reversal of flow from the walls. The primary intention behind
the correct selection is the geometrical parameters and the inlet swirl angles.

Figure 3. Streamline contours of UHDC diffuser at inlet swirl angles (i) 0°, (ii) 7.5°, (iii) 12°, (iv) 17°,
and (v) 25°.

The longitudinal velocity distribution of UHDC annular diffuser at swirl angles 0°, 12°, and 25° with
area ratio 3 is presented in Figure 4. The diffusion takes place continuously along the diffuser length.
For the 0° inlet swirl angle, most of the flow close to the hub wall and separation occurs at a swirl
angle 25°, as seen in Figure 4. The reverse flow is occurred on the inner wall at the inlet swirl angle
25° for x/L=0.3-0.9, respectively. The swirl velocity distribution is shown in Figure 5 for the inlet
swirl angles 12°- 25°. These plots indicate that swirl velocity increases toward the casing wall of the
diffuser. Due to the forced vortex nature, swirl velocity near the outer wall is higher in the case of a
25° inlet swirl. The high swirl velocity near the casing indicates that the flow separations from the
hub. The above phenomena indicate that swirl decay as the flow moves in the downward direction of
the diffuser.
The flow distribution of velocity profiles very well matched with the Caladipietro et al. [14] and Hoadley [15] with swirl flow, who attributed this behavior to centrifugal effect force. The centrifugal force pushes the flow toward the diffusers' casing wall, delayed the flow separation, stabilizes the flow, and escalates the flow separation from the hub wall. The wall angle increases, then velocity
profiles are distorted, as seen in the velocity profiles. The performance of the diffuser is susceptible to the wall angle as well as the swirl angle.

6. Pressure coefficient at casing

The mass average coefficient of pressure recovery has been determined in the annular diffuser with inlet swirl angles 0°-25° at the casing wall of the diffuser. It is found that there is an increase in pressure recovery using swirl flow in comparison to non-swirl flow in the diffuser. The pressure recovery coefficient increases from 0° to 12° swirl angle in the UHDC diffuser, whereas the further increase of swirl angle results in a decrement of the pressure recovery, as shown in Figure 6. The improvement of pressure recovery up to a 12° swirl angle is very distinct. For 17° to 25° swirl angle, drop in Cp due to RZ exist at the hub wall. A sudden drop in pressure recovery is observed for the 25° swirl angle after x/L=0.18 due to the sharp change in the velocity profile.

![Figure 6. Pressure recovery coefficient at the casing wall of UHDC diffuser](image)

7. Conclusion

- By introducing swirl, the flow shifts from the hub wall to the casing wall due to which no flow separation occurs on the casing wall.
- Due to the boundary layer development velocity profiles have different shapes along with the downstream of the diffuser passage.
- The UHDC diffuser's performance is degraded due to the presence of a recirculation zone (RZ) on the hub wall at the inlet swirl angles 17° and 25°.
- The best performance is observed on the optimized swirl angle 12° for the UHDC diffuser.

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