NUMERICAL AND EXPERIMENTAL INVESTIGATION OF THE FLOW IN ANNULEAR DIFFUSER

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ABSTRACT
Numerical calculation of turbulent separated flow characteristics in an axial steam turbine engine exhaust diffuser having 22 cylindrical struts in its passage is presented. The used numerical code is based on the resolution of the averaged Navier-Stokes equations and a finite volume formulation. Turbulence is simulated by the k-ε model and the Reynolds stress model.

Computations are performed for turbulent flow in a sector of 1/8 of exhaust diffuser (2 struts / 45° of the total geometry). In this simplified geometry, the presence of 6 struts is neglected. The comparison between the numerical results with the experimental data reveals an important difference for the static pressure recovery coefficient. A small part of this difference is attributed to numerical reasons: turbulence model, discretization scheme, wall function, swirl treatment in the model, etc. These numerical factors are already observed in previous work concerning the different types of diffusers. A large part of this difference is attributed to effects of the number of struts neglected in these calculations.

KEYWORDS
Annular Diffuser, Turbulent Flow, k-ε model, RSM model.

NOMENCLATURE
\( \bar{C}_p \) static pressure recovery coefficient
\( \bar{p} \) mean static pressure
\( \bar{v}_t \) mean total velocity
\( \bar{\alpha} \) swirl angle
\( \rho \) density
Subscripts
1 inlet
2 outlet

INTRODUCTION
Annular exhaust diffusers are used in turbomachinery applications; such as axial flow compressors and turbines; to increase the static pressure and reduce the velocity of the discharge flow. These diffusers of central hub deal with flows of varying degrees of swirl. The hub is sometimes supported by struts. The wake of struts may deteriorate the pressure recovery of the diffuser and therefore, special care should be taken for the design of these struts.

In the studies of exhaust diffusers, several interesting features can be observed such as the presence of the adverse pressure gradient, regions of recirculating flow as well as the wakes of the struts. The existing swirl in the flow stabilizes the flow on the outer wall and prevents separation there. However, excessive swirls may create recirculating zone near the hub.

Annular diffusers situated downstream of a turbomachine have inlet conditions different from ducted flow inlet conditions. These last conditions concern the annular diffusers situated downstream of an annular duct or upstream of a ventilator. In turbomachinery inlet conditions, the flow presents a swirl component, a high level of turbulence and a periodical component induced by the presence of the upstream turbomachine. A higher pressure recovery coefficient \( \bar{C}_p \) is obtained in the diffusers situated downstream an axial turbomachine due to the increased turbulent mixing which results in a later onset of separation.

A considerable amount of experimental investigations was carried out on annular diffuser flows with ducted flow inlet conditions (e.g., Sovran...
and Klomp, 1967; Lohmann et al., 1979; and Kumar and Kumar, 1980). On the other hand, little contributions concerning annular diffusers downstream a turbine (e.g., Kruse et al., 1983; and Quest, 1990), or a compressor (e.g., Adenubi, 1976; Pfeil and Göing, 1987; and Zierer, 1995) can be found in the literature survey.

In the domain of the numerical studies of annular diffuser turbulent flows, few numerical researches on the swirling flow and the boundary-layer development through annular diffusers of different shapes have been tested. Most of these studies are focused on annular diffusers with ducted flow inlet conditions and without strut (k-ε model/RSM model, e.g., Jones and Manners, 1989; Djebedjian et al., 1995; and Shuja and Habib, 1996). Published numerical contributions concerning annular diffusers with turbomachinery inlet conditions are rarely available. Using the algebraic eddy-viscosity model of Baldwin and Lomax and neglecting the presence of exhaust struts, Baskharone (1991) obtained displaced curves for the computed and measured pressure recovery coefficients. From these studies and others in the domain of two-dimensional and conical diffusers, it is observed that the numerically predicted pressure recovery coefficient is superior to the experimental one.

The experimental researches studying the influence of wake of struts on the pressure recovery indicate the importance of strut geometry, stagger angle and swirl angle (Senoo et al., 1981, and Kruse et al., 1983), upstream and downstream distances from the edges of strut to the inlet and outlet of diffuser, (Gogolev et al., 1974), and the interaction of strut’s with guide vanes wakes (Desideri and Manfrida, 1995).

In the present study, a numerical study has been conducted to make a comparison of the predicted results for the pressure recovery coefficient for an exhaust annular diffuser with experimental data for swirling and non-swirling turbulent flows.

EXPERIMENTAL FACILITY

Fig. 1 shows a cross-section of the test rig. Compressed air passes through a single stage turbine and the axial exhaust. Behind the exhaust, the flow expands into a condenser followed by a perforated plate. Based on the mean axial velocity and the hydraulic diameter at the diffuser inlet, the average inlet Reynolds number was greater than 5.3*10^5 for all experiments. Measurements were carried out with 5 hole probes, static pressure taps, and hot film probes. The details of the test rig and the measurements are mentioned in Djebedjian (1997).

Model Description

The model exhaust diffuser, Fig. 2, consists of three annular diffusers in series with a casing of expanding octagonal shape for the second and third one. There are 22 cylindrical struts (20 inclined and 2 vertical struts) in the exhaust diffuser passage for the rigidity of the outer casing.

The annular diffuser’s inlet casing and hub radii are 0.069 m and 0.14 m, respectively. At the outlet, the radius of the hub is 0.031 m. The length of the diffuser is 0.321 m. The area ratio of the exhaust is 3.47 and the ratio of length-to-inlet height is 4.5.

Diffuser Performance

The evaluation of the diffuser performance can be done by many coefficients such as the static pressure recovery coefficient. It is defined as:
where \( \bar{p}_1 \) and \( \bar{p}_2 \) are the mean static pressures at the inlet and the outlet of the diffuser, respectively. \( \rho \) is the density and \( \bar{V}_{t1} \) is the mean total velocity at the inlet.

**NUMERICAL APPROACH**

The used numerical code was based on the resolution of the time-averaged equations of conservation of mass and momentum. These equations were discretized along with the suitable transport equations for the turbulence model by a finite-volume method using a non-staggered grid arrangement. In this study, turbulence was simulated by the standard \( k-c \) model, (Launder and Spalding, 1974), and the Reynolds stress model (RSM), (Launder et al., 1975). The face values of the unknowns were evaluated by the Quadratic Upstream Interpolation for Convective Kinematics (QUICK), (Leonard, 1979). The SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm (Patankar, 1980) was used for the pressure-velocity coupling and the line-by-line solution method for solving the linearized equations with an additive correction multigrid for the pressure-correction equations.

**Boundary Conditions**

**Inlet.** At the inlet of the diffuser, total pressure, flow angles and turbulence intensity, deduced from measured data, were specified. The inlet turbulent energy dissipation was calculated from:

\[
\varepsilon = C_{\mu}^{0.75} k^{1.5} / l_m
\]

where the length scale, \( l_m \), was 5.5 percent of the annular gap.

**Outlet.** Constant static pressure was specified at the diffuser outlet. This specified pressure was determined from measured data.

**Walls.** The no-slip condition was used for all velocities. The flow close to the wall was handled by the wall function; (Launder and Spalding, 1974).

**Geometry Sector**

For economical reasons of the calculation time, three-dimensional studies were undertaken on a sector of 1/8 of annular diffuser (2 struts / 45° of the total geometry), Fig. 3. It does not represent exactly the real unsymmetrical exhaust and neglects the presence of 6 struts. Consequently, the resulted sector exhibits geometric periodicity and gives rise to flow characteristics that are spatially periodic. In general, the periodicity can be of two types, transitional and rotational. In this configuration, rotational periodic conditions were specified on the surfaces \( I = 1 \) and \( I = I_{max} \), Fig. 4.

**Grid Generation**

A generalized body-fitted coordinate system that conforms to the physical boundaries of the diffuser was used. The resulted algebraic grid was smoothed by the Poisson equations solver.

**RESULTS AND DISCUSSION**

**Grid Density Study**

For this study, three grid sizes \((l_{max} x J_{max} x K_{max})\), namely, \(11 x 23 x 50\) grid, \(21 x 23 x 50\) grid and \(25 x 27 x 90\) grid, Fig. 4, were used. For the swirl angle \( \alpha = 20^\circ \), the coefficient \( C_p \), predicted by the
RSM model is plotted in Fig. 5 as a function of the inverse of the mesh size. Two important effects interpose:
- The first is the pressure loss effect due to the recirculation region at the outlet of the exhaust. The increment of the mesh size is accompanied by an increment of $C_p$ for the studied axisymmetrical diffuser (Djebedjian, 1997).
- The second effect is the pressure loss due to the presence of struts in the axial exhaust. The refinement of grid around the struts was not sufficient to capture this loss as the $C_p$ shows an important dependence on the mesh size.

![Fig. 5 Grid density effect on the pressure-recovery coefficient $C_p$ for the swirl angle $\alpha = 20^\circ$](image)

**Diffuser Performance**

The computed and measured pressure recovery coefficients $C_p$ are presented in Fig. 6. The numerical results were obtained with the $25 \times 27 \times 90$ grid and the RSM model for the configurations without and with 16 struts. It can be observed that the evolution of the curves representing the numerical results is in good agreement with the experimental one and that the pressure loss due to the struts is evident. From previous numerical simulations of turbulent flows in diffusers without strut, an over-estimation of the coefficient $C_p$ is usually observed. Beside this fact, the difference between the predicted and experimental $C_p$ can be attributed to the following reasons:
- The neglected struts proposed to simplify the numerical prediction.
- The insufficient number of grids around the struts which can evaluate correctly their influences on the pressure loss.
- The unsteadiness of the flow (The pressure loss due to the existence of the struts is under estimated).

![Fig. 6 Pressure recovery coefficient as a function of the swirl angle for the numerical predictions without strut and 16 struts](image)

**Turbulence Models Study**

The experimental and the predicted coefficients $C_p$ from the standard $k-\varepsilon$ model and the RSM model using the coarsest grid $11 \times 23 \times 50$ are plotted for different swirl angles, Fig. 7. The over-estimation of the RSM model is less than that of the $k-\varepsilon$ model and it can be concluded the superiority of the RSM for the numerical simulation of the treated case.

![Fig. 7 Pressure recovery coefficients predicted by the $k-\varepsilon$ and the RSM models](image)
Number of Struts Effect on The Pressure Recovery Coefficient

The utilization of a 2 struts / 45° sector of the axial exhaust neglects the presence of 6 struts and increases the evaluated pressure recovery coefficient. A simplified approach, which applies a blockage factor and presents a mean to know the neglected head loss in the numerical calculation, is studied.

The head loss depends on the configuration and the section of strut. In this study, there are 20 inclined cylindrical struts, i.e. elliptical sections. For a swirling flow with an angle $\alpha$ and a strut of length $L$ and elliptical section with maximum lengths $D$ and $l_t$ ($= D \cos \beta$ where $\beta$ is the angle of inclination of strut), Fig. 8, the projected surface of the strut $A_p$ is given by: $A_p = l_t L$ where $l_w$ is the width perpendicular to the velocity $V'$. It is calculated from: $l_w = D/\left(\sin^2 \alpha \cos^2 \beta + \cos^2 \alpha\right)^{0.5}$. The struts blockage factor is defined as the report $\sum_{i=1}^{n} l_w l_t / A_w$ for $n$ struts ($A_w$ is the cross-sectional area of the exhaust at the distance between the two struts).

Fig. 8 Velocity component $V'$ of the swirling flow and strut width $l_w$ perpendicular to $V'$

Fig. 9 shows the variation of $C_p$ with the struts blockage factor. The evaluated numerical results were obtained using the RSM model and the 25 x 27 x 90 grid and the experimental data corresponds to the actual exhaust with 22 struts. The $C_p$ coefficients for the configuration of 8 struts were obtained by keeping only the first strut of the 45° sector, Fig. 3, and applying the same boundary conditions. The variation of $C_p$ for a blockage factor equal to zero represents the influence of swirl on the pressure recovery in the exhaust without strut. That recovery is maximum when $\alpha = 20^\circ$. The flow with $\alpha = 0^\circ$ is very influenced by the interaction of the wakes of the two struts, while for the other swirling flows, these tendencies are slightly less important. Taking into account the curves slopes of the numerical results, good agreement with the measured $C_p$ will be resulted from applying the total geometry of exhaust with the 22 struts.

CONCLUSIONS

The pressure recovery coefficient of an axial exhaust diffuser installed downstream a single-stage turbine was measured for different swirl angles. The simplified rotationally periodic sector of 45° of the geometry with 0, 1 and 2 struts was used in the numerical computations. From this variation of the total number of struts, the following major conclusions emerge:

- The used assemble of struts have an important role in decreasing the pressure recovery in the exhaust.
- The numerical evolution of the coefficient $C_p$ with the swirl angle is similar to that measured experimentally.
- Depending on the swirl angle, the pressure recovery is more or less influenced by the interaction between the two struts' wakes.
- Using the struts blockage factor approach, the application of the total geometry of the exhaust is expected to give good agreement with the measured $C_p$.

REFERENCES


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