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STUDY OF SWIRLING TURBULENT FLOW AND HEAT TRANSFER CHARACTERISTICS IN CONICAL DIFFUSERS

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ABSTRACT

This paper provides and validates a numerical procedure for calculation of turbulent swirling flow and heat transfer characteristics in conical diffusers. The numerical model is based on the fully-conserved control-volume represents fully elliptic Navier-Stokes and energy equations in cylindrical coordinates system. Turbulence is simulated via the two-equation k-ε model. The study is performed to flows with inlet swirl intensity ranged from 0 to 0.9 and the Reynolds number ranges from 10^4 to 1.53×10^5. An experimental study with cold airflow through conical diffusers is conducted to verify the numerical model. To vary the intensity of separation, diffusers of various divergence angles ranged from 4 to 40 degrees are studied. The axial length of the diffusers is taken constant and equal to 4.3 times the inlet diameter. A comparison with available experimental results shows that the numerical method predicts the essential features of various diffuser heat transfer and fluid flow observed in the experiments. The enhancement of heat transfer coefficient increases with the increase of both swirl intensity and Reynolds number. In addition, when the angle of divergence is increased, heat transfer decrease due to the presence of flow separation. At high Reynolds number and large swirl intensity the recirculation region is formed at the centerline of the wide-angle diffuser which causes a sharply increase of the average Nusselt number due to increasing of the turbulence intensity and the tangential velocity.

Keywords: Heat transfer, Swirl flow, Diffuser performance, Turbulence, Flow separation.

1- INTRODUCTION

Swirling flow in confined geometries is an important subject because of its wide industrial use. The theory and modeling of turbulent swirling flows has been reviewed extensively by Gupta et al. [1] and Sloan et al. [2]. Most attention has been given to strongly recirculating swirling flows in combustor geometries or free swirling jets. But swirling turbulent flow in diffusers also occurs in a number of commonly used fluid mechanical devices. For this reason many experiments have been performed to analyze which of the effects of swirl on the overall diffuser performance are important for efficient use [3 to 7], but measurements of the turbulence quantities have not been undertaken very often. The industrial application of such devices makes it essential that accurate and economical prediction can be obtained for both the mean velocity field and the turbulence quantities. In nonswirling diffuser flows, separation or near-separation is caused by the occurrence of a region of low axial momentum near the wall, because of the positive axial pressure gradient. It has been found that the inclusion of swirl upstream of the diffuser inlet can prevent separation occurring for diffuser angles and area ratios at which it would otherwise occur, [5]. Measurements of turbulence quantities for a 20 degrees swirling conical diffuser flow have been made in [6]. They found an optimum inlet sold-body rotation swirl level to give neither flow separation at the diffuser wall nor axial flow reversal at the centerline. A free-vortex inlet swirling flow through a 12 and 16 degrees conical diffusers are also considered, but only mean flow (axial and tangential velocities) experimental results [5] are available. With swirling flow, the pressure gradient along the axis was increased by a swirl, and the axial velocity near the center was reduced downstream. The stronger the swirl, the larger the low velocity core is near the center, and the pressure recovery coefficient is decreased by the core if the swirl is stronger than the optimum value. Abdalla [8] and Abdalla et al. [9 to 12] carried out studies on the turbulence and swirl effects on smooth and rough wall conical diffusers with unseparated and separated flow. Results showed that the greatest improvement in the diffuser performance occurred with an optimum inlet swirl intensity corresponding to the relative surface roughness and the divergence angle.

Swirling flow has been used as an alternative to enhance heat transfer in heat exchanger applications. This is because the swirling flow is usually accompanied with high tangential velocity and turbulence intensity, which provides an additional mechanism to increase the heat transfer. Different methods of generating the swirling flow may lead to different characteristics of flow structure and heat transfer, [1 to 12]. Experiments were performed to study the heat transfer process of swirling flow into a head conical diffuser with a half divergence angle of 7 degrees, [13]. The Reynolds number ranges from 10^4 to 2×10^5 and the swirl intensity from 0 to 0.65. The nature of these swirling flows and their enhancement on the heat transfer have been studied

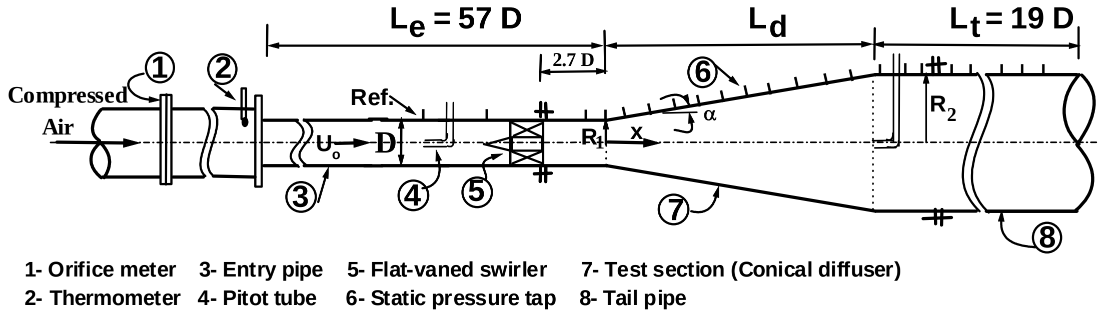


Fig. 1 Layout of experimental setup

extensively. Numerical studies on the variation of velocity field and turbulence intensity in a circular pipe have also been performed [14 to 17]. A comparison between the experimental data and the prediction was made to validate the numerical model. One may expect that the swirling flow may have a significant effect on the flow and heat transfer process in a gas turbine combustor. Where, the swirling flow is just issued from the swirler and has relatively high tangential velocity and turbulence intensity. The heat transfer data in the combustor are very important for designing the cooling process of a combustor. Therefore, the objective of the present work is to study the swirling flow heat transfer process in conical diffusers with different divergence angles. This kind of work has been rarely found in the literature.

2- EXPERIMENTAL SETUP AND MEASUREMENTS

The layout of the experimental setup is shown in Fig. (1). The air is supplied to the test diffusers with the aid of two electrically driven compressors through two-interconnecting air reservoirs of 1.57 and 7.16 m^3 capacities. Each compressor draws a maximum flow rate of 880 m^3/h at maximum operating pressure of 8 bar. The air from reservoirs flows through the main pipeline, a main control valve and then to the test rig. With adjusting the main control valve, the required air flow rate could be supplied to the test diffusers. Three conical diffusers made of aluminum are tested. They have the same inlet diameter of 28 mm and divergence angles of 8, 14 and 20 degrees at area ratio of 3.8, 4.2 and 6.3, respectively. The axial length of the tested diffusers is 6.8 D_1 for 8 degrees diffuser and 4.3 D_1 for both 14 and 20 degrees diffusers. On leaving the test diffusers, the air enters a tail pipe and is finally discharged into the atmosphere. The length of the upstream pipe is 57 times the inlet hydraulic diameter of the diffuser and this is arranged to ensure a fully developed turbulent flow at the diffuser inlet. The swirl generator is located 2.7 times the inlet hydraulic diameter ahead of the diffuser inlet. The swirl flow was produced using flat-vaned swirlers. Four swirlers were fabricated to produce different swirl angles at the diffuser entry. The swirl vane made of a steel flat sheet of 2 mm thickness. The flat-vaned swirlers used have a pitch/chord ratio of approximate 0.4 in order to obtain a constant tangential velocity at the diffuser entrance. The vane angles of these swirlers are 5, 11, 15 and 20 degrees, respectively. A total of four different swirlers with an internal diameter of 28 mm and a hub diameter of 9 mm are made. The swirling flow is usually characterized by swirl intensity (S_o) which is a nondimensional parameter representing the axial flux of swirl momentum divided by the axial flux of the axial momentum times the swirler radius. It has been obtained, [18], that the swirl vane angle (β) and the swirl intensity (S_o) are related by,

$$S_o = \frac{2}{3}\left[\frac{1-(D_h/D_s)^3}{1-(D_h/D_s)^2}\right]\tan\beta \qquad (1)$$

Accordingly, the swirl intensity of the swirling flow produced by each of these swirlers is 0.06, 0.14, 0.19 and 0.26, respectively. To avoid flow separation and generation of shocks and turbulence before the swirlers entrance, the hub is streamlined in the front of the swirler. The tests with no-swirl are conducted without swirl generators. Static pressure taps are located along the upstream pipe, the diffuser wall and the tail pipe for measuring the static pressure distributions. The pressure taps having 1mm internal diameter are drilled normal to the wall. The wall static pressure taps were spaced at an equal intervals. The static pressure distributions were measured using a water multi-tube manometer. The total flow rate was measured by a calibrated orifice-meter with diameter ratio of 0.43 which gives a discharge coefficient of 0.62. The average inlet velocity (U_o) was calculated from the measured flow rate and checked by the measured axial velocity using total-static Pitot tube that was placed at the center of the entrance pipe 6D_1 ahead the diffuser inlet and parallel to the flow direction. The flow direction was determined by getting a maximum dynamic pressure reading. The pressure recovery coefficient (C_P), which is defined as the actual static-pressure rise of the diffuser to the inlet dynamic pressure, was measured. All three diffusers were tested for nonswirling and swirling flow conditions. The effect of swirl intensity on the performance of conical diffusers is studied. The experiments are conducted with Reynolds number ranging from 4.8×10^4 to 1.53×10^5.

The experimental uncertainties are determined according to the procedure proposed by Kline and McClintock, [19]. The maximum uncertainty of the pressure recovery coefficient is 1.5%, the Reynolds number is 2% and the swirl intensity is 0.6%.

3- NUMERICAL MODELLING

3-1 General Form of the Differential Equations

The problem under consideration is governed by the steady two-dimensional axisymmetric form of the continuity and the time-averaged Navier-Stokes equations. The cylindrical coordinate-system (x, r, θ) is used to describe the swirling flow in the axisymmetric conical diffuser, Fig.(2). For the present study, the steady state equations for incompressible, axisymmetric, turbulent swirling flow may be written as shown in [20 and 21]. The equations governing the continuity, momentum, stagnation enthalpy and turbulence model in generalized form can be written as follow,

$$\frac{1}{r}\left[\frac{\partial}{\partial x}(\rho u r \varphi)+\frac{\partial}{\partial r}(\rho v r \varphi)\right]-\frac{1}{r}\left[\frac{\partial}{\partial x}(r\Gamma_{\varphi}\frac{\partial \varphi}{\partial x})+\frac{\partial}{\partial r}(r\Gamma_{\varphi}\frac{\partial \varphi}{\partial r})\right]=S_{\phi} \qquad (2)$$

where, u, v and w are the axial, radial and tangential velocities, respectively. ϕ is the general dependent variable. x, r, and θ are the axial, radial and tangential coordinates, respectively. ρ and Γ_{ϕ} are the density and the effective diffusivity coefficients. S_{ϕ} is the source of ϕ . In the present calculations, equations were solved for continuity and with dependent variables, ϕ , corresponding to the axial, radial and tangential velocity components, and temperature. The effective diffusivity (Γ_{ϕ}) was calculated from the two-equations k- ε turbulence model.

3-2 Inlet and boundary conditions

The following boundary conditions are used at the inlet, outlet, the axis of symmetry and the solid wall. The inlet velocity profile in the axial direction corresponding to fully developed flow is considered at the inlet section. In the case of swirling flows, the tangential velocity profile of a uniform swirl profile was assumed at the inlet section. The inlet conditions required for the turbulence model are the turbulence intensity and the turbulence length scale. These values are set according to [23 and 24], while the turbulence intensity was taken as 3 % when experimental values were not available, and the characteristic length scale is 0.05 multiplied by the inlet diffuser diameter. The air temperature in the upstream of the swirler measured and taken as the inlet temperature of flow. The wall temperature is presumed constant at the diffuser wall. But with comparison of experimental work for the others, the local wall temperature is taken from the available experimental data. The zero-gradient conditions were applied to the exit and symmetry axis. The radial velocity component at the center of symmetry was set to zero. On the solid boundary, the no-slip velocity boundary conditions are adopted. The wall boundaries were treated using logarithmic law of the wall, [20].

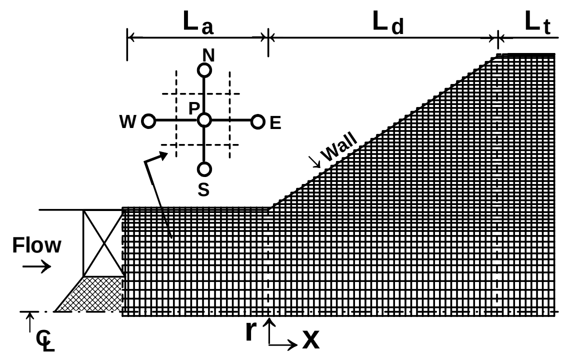


Fig. 2 Computational mesh (98×59 grids).

3-3 Solution Procedure

The system of equations and boundary conditions were solved using a control volume method, which is created by the cylinderical grid system of coordinates to provide the finite difference equations. The discretization scheme used is a hybrid system (an upwind-central difference scheme) explained in detail by Launder and Spalding [20]. The set of the resulting finite-difference equations of mass, momentum, turbulence model and the energy were solved numerically by an iterative, line-by-line procedure [21]. The geometry used for the calculations is shown in Fig. (2). It consists of a cylindrical pipe with a swirler and a diffuser placed 2.7 D downstream the swirler and it is followed by a tail pipe. A mesh consisting of 98×59 points is used in the calculation. In the axial direction, the mesh is uniform and fine over the upstream and along the diffuser and is then stretched in the tail pipe. In r-direction, the mesh is fine adjacent to the wall to capture all the flow features. The computations are started immediately downstream from the swirler to study the effect of flow blockage and wake caused by the swirler-hub on the flow characteristics at the diffuser entrance. The solution was considered to be converged when the maxima of the mass flux and momentum flux residuals summed at all nodes were less than 0.05 % of the inlet flux.

4 - RESULTS AND DISCUSSION

Computational fluid dynamics validation is an important aspect of code development program. Validation of the k-ε turbulence model for turbulent flow in conical diffusers will focus on nonswirling and swirling flow with and without heat transfer.

4-1 Validation of the Numerical Method

4-1-1 Flow in Conical Diffusers Without Heat Transfer

To verify the numerical model, predicted results for the pressure distributions in case of nonswirling flow through 8, 14 and 20 degrees conical diffusers are compared with the present experimental results. For nonswirling flow, the effect of the hub placed ahead the diffuser entrance is also theoretically examined. A comparison of the pressure distributions between the case with hub and the case without is shown in Fig. (3) and are presented by dashed-lines. It can be observed from this figure that, the pressure distributions are improved for all tested diffusers when the hub is placed upstream the diffusers. The rate of improvement increases with increasing the divergence angle of the diffuser. This means that the presence of the hub control the growth of the boundary layer and hence the separation in the wide-angle diffusers is suppressed. This is observed from Fig. (4) which presents the streamline contours of 20 degrees diffuser as a sample of the predicted results. In addition, the hub blockage can make the flow move along the divergence wall and increases the turbulence mixing due to the wake flow behind the hub. This will reduce the separation tendency or shrinks the separation bubble in wide-angle diffusers.

The comparison between the experimental and predicted results of C_P -distributions are conducted for nonswirling flow and swirling flow with different swirl intensities, namely S_o = 0, 0.06, 0.14, 0.19 and 0.26, as shown in Fig. (3). For 8 degrees diffuser, it is established from both experimental and computational results for nonswirling flow that, the wall pressure gradually increases along the diffuser wall up to the diffuser exit. This means that, there is a forward flow very close to the wall and that separation is not reached. The distributions of the pressure recovery coefficient along the 14 degrees diffuser indicate that a separation commences near the diffuser exit. The separation location is designated in the figure by a symbol (S). For 20 degrees diffuser, the position of separation is moved towards the diffuser entrance and started at about x/D = 1.2. The effect of inlet swirl intensity on the pressure distributions of tested diffusers is also shown in Fig. (3). The results indicate that the swirl can significantly enhances the pressure recovery coefficient of the tested diffusers. The experimental and predicted results for swirling flow indicate that, an optimal inlet swirl intensity is required for improving the diffuser performance as shown in Fig. (5-a). The optimal swirl intensities are found as 0.14 for 8 and 14 degrees diffusers and 0.19 for 20 degrees diffuser. A further point of interest is that swirling inlet flow caused larger performance, (ΔC_{P_d}), for wide-angled diffusers than that with

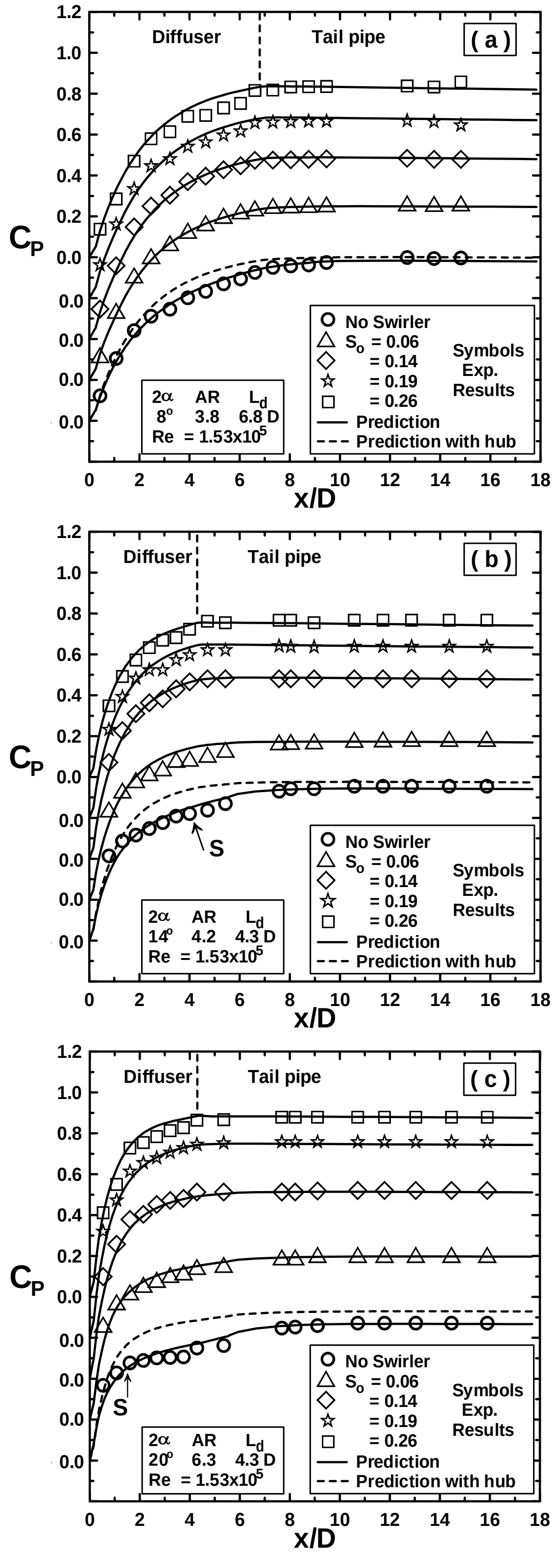
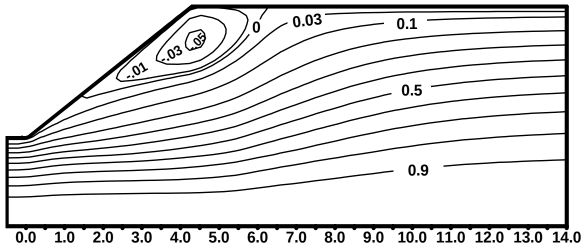
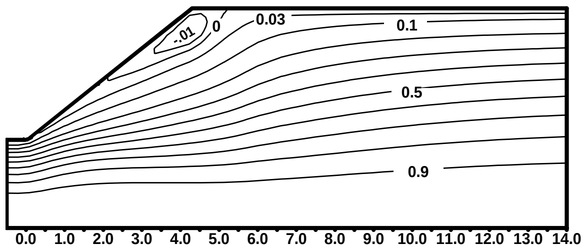


Fig. 3 Distribution of wall static pressure recovery coefficient C_P for tested diffusers.



Without Hub



With Hub

Fig. 4 Predicted streamline contours with and without hub without swirling flow for 20 degrees diffuser (Re = 1.53×10^5).

small angle diffuser, as shown in Fig. (5-a). This can be explained as follows. The swirl intensity in the no-separation diffuser is decayed faster than that of separation flow case, as shown in Fig. (5-b). On the other hand, when the mainstream is swirled in wide-angled diffusers, the tangential velocity makes the flow move along the divergent wall. This means that, the boundary layer thickness is decreased with increasing the inlet swirl intensity and, correspondingly, the separation zone is diminished. In addition, a complex wake flow behaviour downstream the hub of the swirler is accompanied with high turbulence intensity. Therefore, increasing the tangential velocity and the turbulence intensity significantly improve the overall performance of the wide-angled diffusers. With strong swirling flow, the degraded performance may be due to the formation of recirculation at the diffuser centerline which causes larger total losses and a reduction of the flow effective area. To summarize, the existence of swirl is effective to suppress remarkably the development of the boundary layer and, hence, to control the flow separation in wide-angled diffusers [12]. This explains the reason why the performances of conical diffusers are improved with swirling flow. Good agreement is obtained for tested diffusers.

4-1-2 Flow in Conical Diffusers With Heat Transfer

Gau and Chen [13] conducted an experimental study of the swirling flow with heat transfer in a divergence pipe with a half divergence angle of 7 degrees and an area ratio of 2.86.

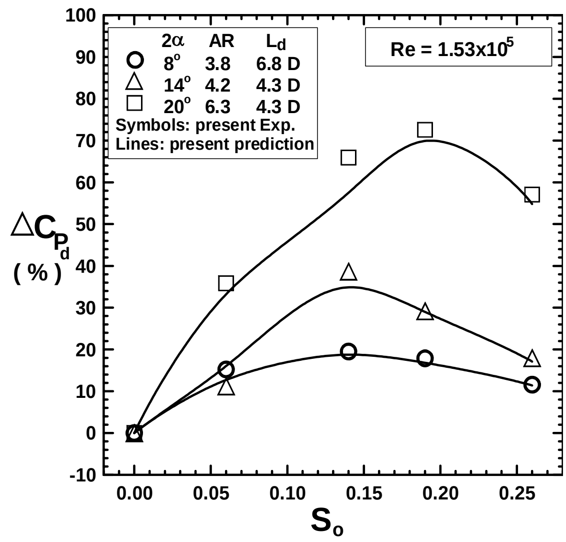


Fig. 5-a The increasing of the overall pressure recovery coefficient C$_P$ for tested diffusers for swirling flow.

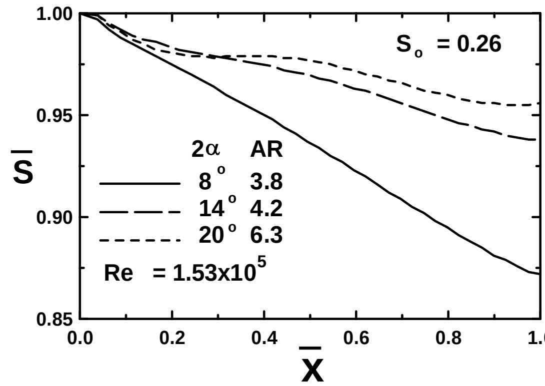


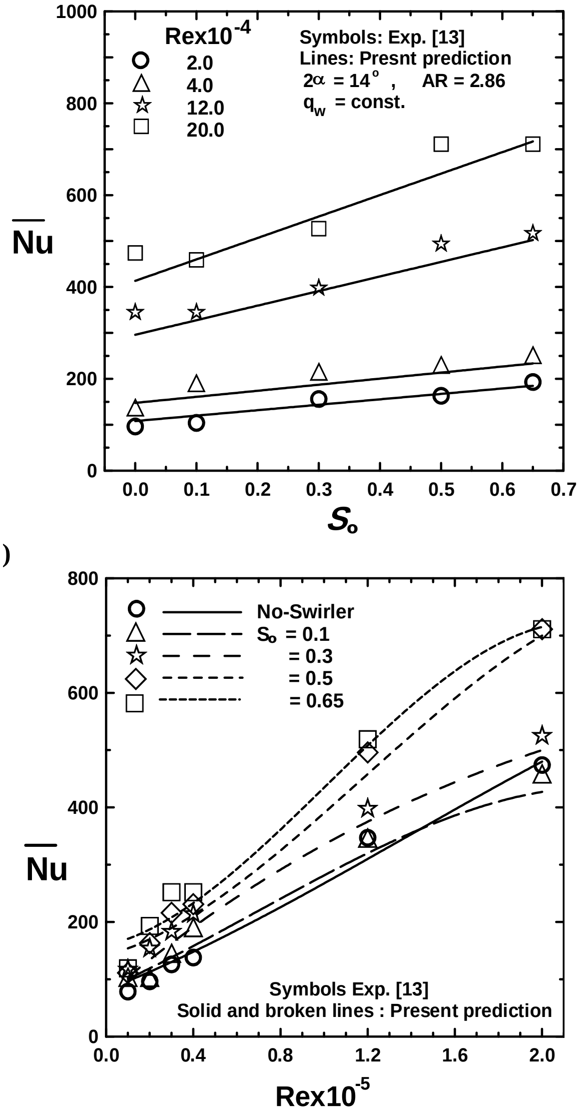
Fig. 5–b Predicted swirl decay in tested diffusers.

During the experiments, the Reynolds number ranges from 10^4 to 2×10^5 and the swirl intensity from 0 to 0.65. Measurements were made of the local heat transfer coefficient for airflow with a constant heat flux condition. The local heat transfer coefficient evaluated with the following equation :

$$h = \frac{(Q - Q_{Loss})}{A_s(T_w - T_{in})} \tag{3}$$

where Q is the total thermal energy; A_s is the total surface area inside the pipe wall; and T_{in} is the air temperature at the inlet of the pipe.

Calculations of swirling and nonswirling flow with heat transfer are performed for the same configuration given in [13] at the same inlet conditions. A comparison between the calculated average Nusselt numbers ($\overline{Nu}$) and the measurements is given in Fig. (6). The results show that $\overline{Nu}$ increases with increasing the inlet swirl intensity as well as with increasing the inlet Reynolds number (Re). The theoretical predictions are in good agreement with the experimental data.



(a)

(b)

Fig. 6 Comparison between predicted average Nusselt numbers ($\overline{Nu}$) and experimental data [13].

5- PRESENTATION OF PREDICTED RESULTS

Computations are performed for swirling flow heat transfer in conical diffusers with different total divergence angles which are ranged from 4 to 40 degrees and area ratios from 1.69 to 16.97 while the dimensionless axial length is kept constant at $L_d / D = 4.3$. All tested diffusers have the same inlet diameter (D). The inlet conditions are taken at 2.7 times the inlet diameter ahead the diffuser entrance. During the computations, the Reynolds number ranged from 4.8×10^4 to 1.53×10^5 and swirl intensities ranged from 0 to 0.9. the wall temperature is kept constant at 3.7 times the inlet air temperature.

Figure (7) shows the influence of the divergence angle of the conical diffusers on the distributions of local Nusselt number along the tested diffusers in case of nonswirling and swirling flows. For nonswirling flow the local Nusselt number distributions are decreased with increasing the divergence angle due to the occurrence of flow separation. The maximum Nusselt numbers occur at positions near to the diffuser inlet decrease in magnitude with increasing the total divergence angle of the diffuser. At small divergence angles, where

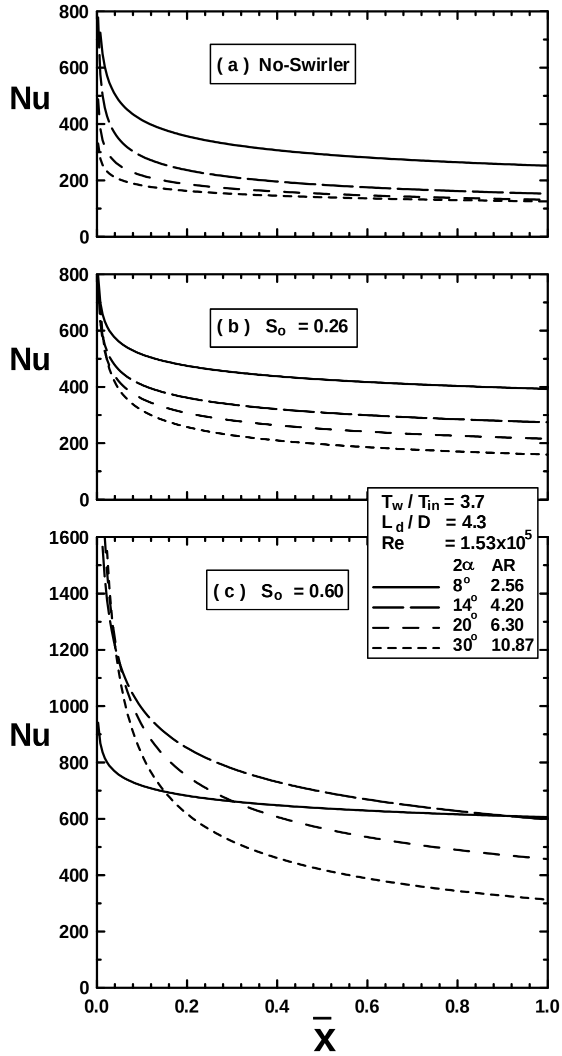
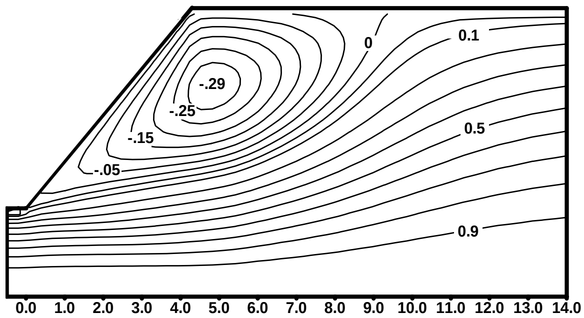
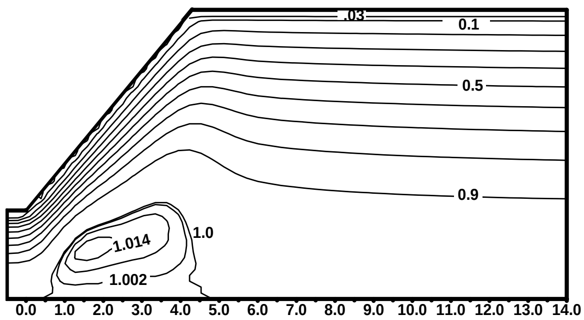


Fig. 7 Predicted results of local Nusselt number for different S_o and 2∝.

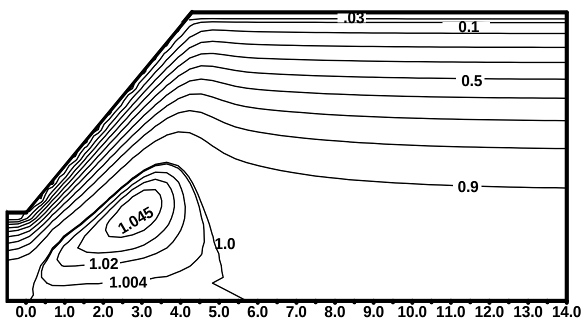
no-separation is present, the local Nusselt number distributions decrease with downstream distance because the boundary layer grows along the diffuser wall and hence the heat transfer is reduced. For large angles, the separation flow region acts as a blockage which can reduce the flow velocity and thus the heat transfer, depending upon the strength of separation development. The computed axial distribution of the local Nusselt number for nonswirling and swirling flows is presented in Fig. (7). The results indicate that increasing the swirl intensity from 0 to 0.26 does not influence qualitative features of the local Nusselt number along all the diffusers. However, the magnitude of Nusselt number is greatly influenced by the swirl intensity. A further increase of swirl intensity ($S_o = 0.6$) strongly affects the magnitude of the local Nusselt number. Increasing the divergence angle in the presence of swirl flow causes an increase of the local Nusselt number due to the suppression of the wall separation and the formation of the central recirculation regions. However, the maximum Nusselt numbers increase in magnitude and move downstream the diffuser inlet with increasing the swirl intensity. The distribution of the local Nusselt number of 14 degrees diffuser with a higher swirl intensity ($S_o = 0.6$) shows the highest magnitude of the local Nusselt number distribution. This enhancement of heat transfer occurs due to the disappearance of flow separation and the formation of the recirculation region. A further increase of divergence angle to 30 degrees enlarged the recirculation region and a significant increasing of Nusselt number is noticed, as shown in Fig. (8). At low swirl intensity, the enhancement of Nusselt number is mainly due to the tangential velocity. However, the Nusselt number increases monotonically with increasing swirl intensity because of an increase in the tangential velocity and turbulence intensity in the mainstream.



(a) No-Swirler



(b) S_o = 0.26



(c) S_o = 0.60

Fig. 8 Predicted streamline contours with nonswirling and swirling flows for 30 degrees diffuser ($Re = 1.53\times10^5$).

The effect of Reynolds number on the average Nusselt number with swirling flow through conical diffusers is examined. A sample of predicted results is presented in Fig. (9-a) for 20 degrees diffuser. The heat transfer predictions in 20 degrees diffuser with nonswirling flow are compared with the constant diameter pipe for Reynolds number ranged from 10^4 to 1.53×10^5. The correlated results of the average Nusselt number appearing in the figure is developed in [25] for fully developed heat transfer in constant diameter pipe. The effect of swirl can be seen from the deviation of the predicted data for 20 degrees diffuser from the correlation results for the constant diameter pipe. The average Nusselt number results for 20 degrees diffuser are lower when the mainstream has no-swirl. This is a result of the occurrence of the flow separation. When the flow is swirled, the heat transfer results in 20 degrees diffuser are significantly higher. The results indicated that, when the Reynolds number is relatively low, the enhancement of the heat transfer by the swirling flow is not large. This may be because the enhancement of heat transfer at a low Reynolds number is mainly a result of the tangential velocity. As the Reynolds number increases, the high speed flow tends to move along the divergent wall and makes heat transfer approach the wall of the diffuser. In addition, the swirling flow enhances the heat transfer due to the relatively high turbulence intensity. When the swirl intensity becomes more than 0.4, a sharp increase behaviour on the average Nusselt number is observed at low Reynolds numbers.

The influences of the divergence angle and the swirl intensity on the average Nusselt number are shown in Fig. (9-b). The divergence angle is varied from 4 to 40 degrees and the swirl intensity ranges from 0 to 0.9. The results are plotted at constant Reynolds number of 1.53×10^5 and dimensionless wall temperature of 3.7. The predicted results of the heat transfer with swirling flow in tested diffusers are compared

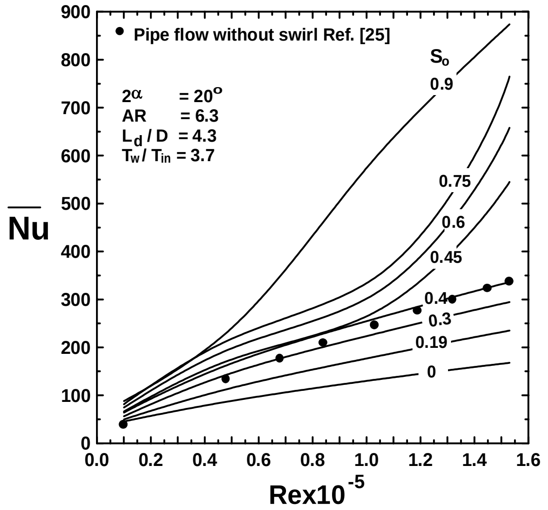


Fig. 9-a Effect of Re on $\overline{Nu}$ for different S_o for 20 degrees diffuser.

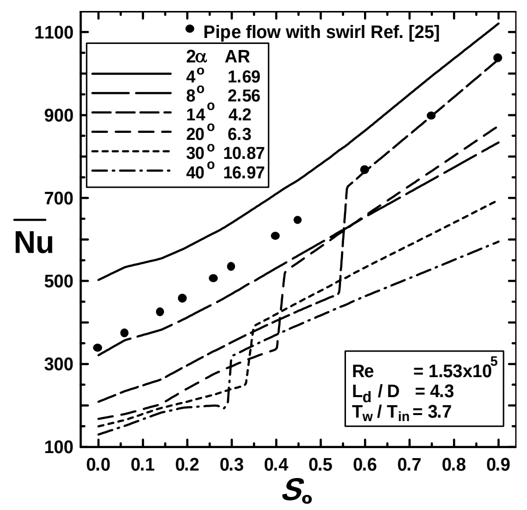


Fig. 9-b Effect of S_o on $\overline{Nu}$ for different diffusers.

with the heat transfer correlation for free swirling flow in a constant diameter pipe developed in [25]. In general, increasing the divergence angle of the diffuser lowering the average Nusselt number than that in fully developed swirling flow in pipes, due to the presence of pressure gradient. For small angle diffusers ($2\propto$ = 4 and 8 degrees) the average Nusselt number increases gradually with an increase of swirl intensity. An increase in tangential velocity is generally accompanied by a significant increase of turbulence intensity in the flow. This results in an increase of heat transfer with the swirl intensity. For wide-angle diffusers, the figure indicates that increasing the swirl intensity from 0 to 0.29 increases the average Nusselt number gradually due to the suppression of the separation zone. Afterwards, the average Nusselt number shows a sharply peaked behaviour with increasing the swirl intensity due to the appearance of the recirculation at the centerline. As the divergence angle increases from 14 to 40 degrees, the swirl intensity required for this behaviour is decreased with increasing either the divergence angle or the Reynolds number. A further increase of the swirl intensity causes a monotonic increase in the average Nusselt number due to increased turbulence kinetic energy. This is observed in Fig. (10) which presents contours of turbulence kinetic energy and flow temperature for S_o = 0.19 and 0.33 as a sample of results for 30 degrees diffuser. Correspondingly, the maximum of average Nusselt number strongly depends on the swirling flow regime and the turbulence intensity in conical diffusers.

6- CONCLUSIONS

The objective of the present work is to study the swirling flow heat transfer process in conical diffuser with different divergence angles. The study is performed to flows with inlet swirl intensity ranged from 0 to 0.9 and the Reynolds number from 10^4 to 1.53×10^5. The main conclusions obtained are:

1- The numerical method predicts the essential features of various diffuser heat transfer and fluid flow observed in the experiments.

1- Swirling inlet flow causes larger improvement of the performance of wide-angled diffusers than that for conical diffusers having small divergence angles.

3- The optimum swirl intensity strongly depends on the divergence angle and the flow regime in conical diffuser.

4- Increasing the divergence angle of conical diffuser lowering the average Nusselt number with no-swirl than that of a constant diameter pipe. The results of the average Nusselt number approach the case of a constant diameter pipe when the flow is swirled.

5- The swirling turbulent flow in non-separated conical diffusers achieves the highest average Nusselt number than that the separated diffusers.

6- The average Nusselt number of conical diffusers increases with increasing both the swirl intensity and Reynolds number. The intensification of the average Nusselt number in wide-angle diffusers at high Reynolds number and with strong swirling flow is due to the formation of recirculation regions, which results in increasing of the turbulence intensity and the tangential velocity.

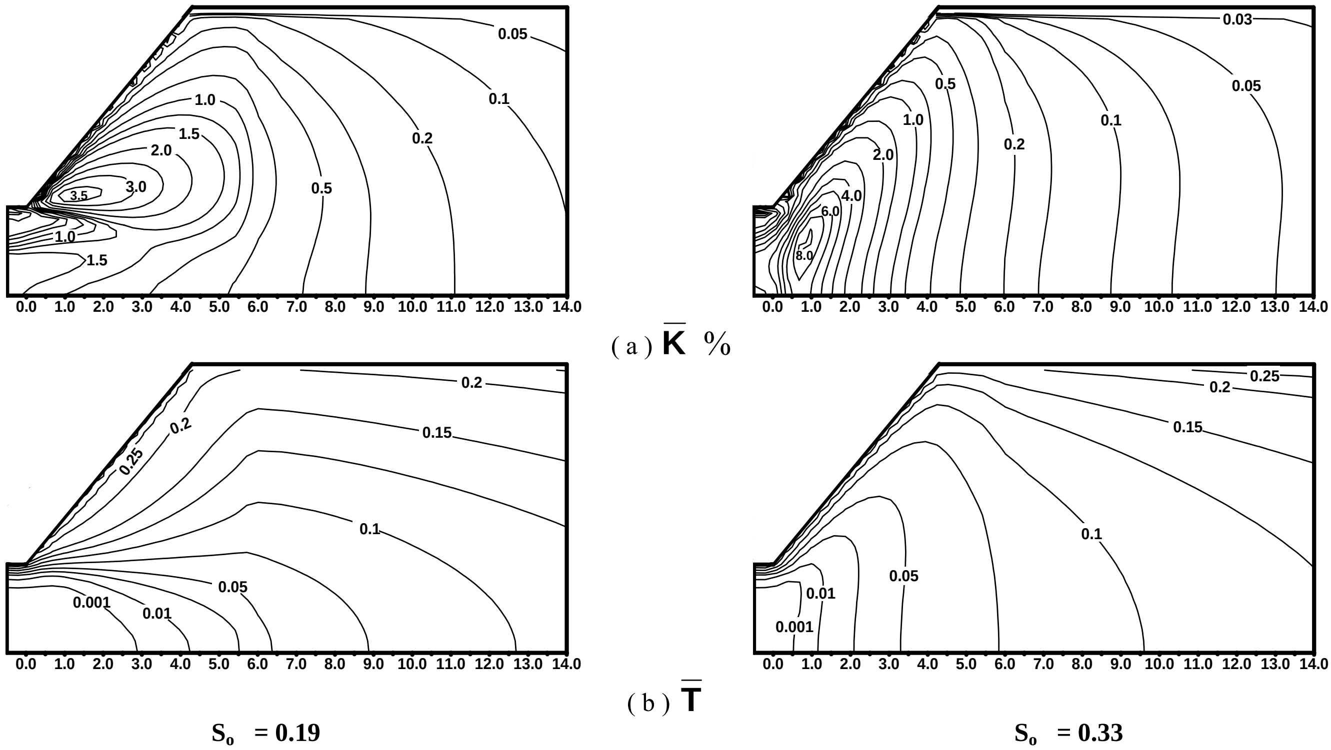


Fig. 10 Predicted streamline contours of turbulence kinetic energy ($\overline{K}$ %) and temperature ($\overline{T}$) for 30 degrees diffuser.

NOMENCLATURE

A cross-sectional area, (m^2)

AR diffuser area ratio, (A_2 / A_1)

C_p local pressure recovery coefficient,

$C_p = (p_x - p_1) / (\frac{1}{2} \rho U_o^2)$

C_{p_d} overall pressure recovery coefficient,

$C_{p_d} = (p_2 - p_1) / (\frac{1}{2} \rho U_o^2)$

D diffuser inlet diameter, (m)

h local heat transfer coefficient, $h = q_w / (T_w - T_b)$, ($W/m^2 . K$)

$\bar{h}$ average heat transfer coefficient, $\bar{h} = (\int_0^{L_d} h\, dx) / L_d$

κ thermal conductivity of air, (W/m. K)

k turbulent kinetic energy, (m^2 / sec^2)

$\overline{K}$ dimensionless value of turbulent kinetic energy, $\overline{K} = k / U_o^2$

L_a entry pipe length between swirler and diffuser, (m)

L_d diffuser axial length, (m)

L_e entry pipe length, (m)

L_t tail pipe length, (m)

Nu local Nusselt number, $Nu = hD / \kappa$

$\overline{Nu}$ average Nusselt number, $\overline{Nu} = \bar{h} D / \kappa$

p static pressure, (N / m^2)

q heat flux, (W/m^2)

Re Reynolds number, $\rho U_o D / \mu$

S_o swirl intensity, $S_o = (\int_0^R u w r^2\, dr) / (R_1 \int_0^R u^2 r\, dr)$

T temperature, (K)

$\overline{T}$ dimensionless temperature, $\overline{T} = (T_r - T_{in}) / (T_w - T_{in})$

T_b the bulk temperature, $T_b = \int_0^R \rho u T\, dr / \int_0^R \rho u\, dr$

U_o mean-bulk longitudinal velocity, (m/sec)

u, v, w local mean velocity in axial, radial and tangential coordinates,(m/sec)

x, r, θ axial , radial and tangential coordinates, (m)

$\overline{X}$ dimensionless axial distance, (x / L_d)

ΔC_{P_d} the increasing of C_{P_d}, $\Delta C_{P_d} = \frac{(C_{P_d})_{S_o} - (C_{P_d})_{S_o=0}}{(C_{P_d})_{S_o=0}}$

Greek Symbols

α diffuser half-cone angle, (degrees)

β van angle of the swirler, (degrees)

ε dissipation rate of turbulence kinetic energy, (m^2 / sec^3)

ρ density, (k_g / m^3)

μ effective viscosity, ($N.sec / m^2$)

ν kinematic viscosity, (m^2 / sec)

ψ stream function, $\psi = \int_0^R u\, dr$

Subscripts

1 diffuser inlet condition
2 diffuser exit condition
in inlet condition at swirler
c centerline condition
b bulk
h hub of the swirler
r local radial position
s outer surface of the swirler
w wall condition

Superscripts

— dimensionless and average conditions

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