
Heat Transfer Enhancement in a Narrow Concentric Annulus in Decaying Swirl Flow

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The characteristics of decaying swirling flows and forced convective heat transfer on the conditions of both laminar and turbulent flow in a narrow concentric annulus were simulated. The governing equations are solved numerically via a finite volume method. A uniform wall temperature at the inner wall and adiabatic conditions at the outer wall are considered as thermal boundary conditions. Solutions for the axial and swirl velocity distributions and the Nusselt number are obtained for different values of the inlet swirl number and the Reynolds number. Simulations show that the inlet swirl number have great influences on the heat transfer characteristics. Under both developing laminar and developed turbulent flow conditions, the increases of the inlet swirl number will enhance the heat transfer. When the inlet swirl number increases it increases the axial velocity near the wall and reduces it at the mid-gap to achieve the conservation of mass due to the existence of secondary flows in the annulus due to centrifugal forces. The increase of the near-wall velocity, in turn, produces larger temperature gradients and a higher heat transfer rate. The swirl velocity profiles decay gradually downstream as a result of friction which leads to damping of the tangential velocity. The swirl has a pronounced effect on the turbulent kinetic energy which is increased evidently with the swirl number. Obviously, a higher turbulence level leads to a considerable improvement in the heat transfer rate. Turbulence level improvement can be attributed to the high velocity gradients. Numerical results show that the turbulent kinetic energy is lower in the mid-gap and higher in the near-wall regions. Moreover, the turbulent structures near the outer wall are more activated than those near the inner wall. The comparison between predicted and experimental data of average Nusselt numbers was found to be in good agreement.

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199 ISSN1064-2285 ©2011 Begell House, Inc.
1. INTRODUCTION

The concentric annular pipe flow is of considerable industrial importance, as it is frequently encountered in heat exchangers, gas-cooled nuclear reactors, gas turbines, extruders, and oil/gas drilling wells. Forced convection in an annulus between two horizontal concentric cylinders has been investigated extensively in the literature (Shah and London, 1978; Kakac et al., 1987; Cengel, 2006) because of its wide variety of practical and technical applications such as heat exchangers. The production of small-size and inexpensive but effective heat exchangers to transfer more heat from the surfaces of heat exchangers is a very important subject in heat transfer. Swirl flows are found in nature, such as tornadoes, and are utilized in a very wide range of applications, such as cyclone separators, agricultural spraying machines, heat exchangers, gasoline engines, diesel engines, gas turbines, and many other practical heating devices (Gupta et al., 1984). Swirl flow devices are designed to impart a rotational motion about an axis parallel to the flow direction to the bulk flow and are one of the passive enhancement techniques used for increasing the rate of heat transfer (Kreith and Bohn, 1993). The use of decaying swirl flow is one of most promising techniques for enhancing the mass and heat transfer. The swirling effect to the fluid is given in the upstream section, and then it is allowed to decay along the length of a tube (Razgaitis and Holman, 1976). The presence of a swirl will increase the flow path, decrease the free area, and introduce an angular acceleration to the fluid flow (Kuheln and Goldstein, 1979). Swirling flows can be imparted to the flow by use of various swirl-generating methods (Jawarbeh et al., 2005, 2007; Yilmaz et al., 2002) where part of fluid enters axially while the remainder is injected tangentially using a vortex generator. Different swirl intensities can be achieved by varying the fluid flow rate at the axial and tangential inlets. Yowakim and King (1988) have investigated experimentally swirl flows in a cylindrical annulus using a hot-wire anemometer. Their experimental work is based on a relatively large gap of 73 mm and a very high Reynolds number of 300,000. Martemianov and Okulov (2004) proposed a theoretical model of heat transfer in axisymmetric swirl pipe flows. Their study shows that wake-like swirl flows will increase the heat transfer compared with the axial flow, while jet-like swirl flows can diminish the heat transfer. Zeng et al. (2007) investigated the characteristics of the flow and convective heat transfer on the conditions of both developed laminar and turbulent flows in bilaterally heated narrow annuli. The convective heat transfer depends on the effect combining the heat-flux ratio and the Reynolds number. Under laminar flow conditions, decreases of the gap size will lead to the heat transfer deterioration both on the inner and outer walls. However with respect to the case of turbulent flow conditions, it is quite different. The decrease of the gap size will yield heat transfer deterioration on the inner wall,
but it will enhance the heat transfer coefficient on the outer wall. Akpinar et al. (2004) investigated the effect of a swirl generator on heat transfer rates; swirl generators were placed in the entrance section of the inner pipe of a heat exchanger. Experiments were carried out for both parallel and counter flow models of the fluids at different Reynolds numbers. It was observed that the Nusselt number could increase up to 130% by giving rotation to the air with the help of the swirl elements. Several studies have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications (Berger et al., 1983; Janssen and Hoogendoorn, 1978; Prabhanjan et al., 2002). The centrifugal force due to the curvature of the tube results in the development of secondary flows which assist in mixing the fluid and enhance the heat transfer. These situations can arise in the food processing industry for the heating and cooling of either highly viscous liquid food, such as pastes or purées, or products that are sensitive to high shear stresses. Lu and Wang (2008) have investigated experimentally the convection heat transfer characteristics of water flow in a narrow annulus without swirl. Experimental results show that the heat transfer characteristics of a single-phase water flow in the narrow annulus are different from those in circular tubes. The transition from laminar to turbulent heat transfer in the narrow annulus is earlier than that in circular tubes. When the Reynolds number is lower than 150, heat transfer is deteriorated, and the axial heat conduction has an important influence on the overall heat transfer. The transition from laminar to turbulent heat transfer is in the Reynolds number range from 800 to 1200. In the turbulent flow area, a narrow annulus can achieve the enhancement of heat transfer. Dirker and Meyer (2002) conducted experiments of heat transfer coefficients at the inner wall of smooth concentric annuli for turbulent flow of water within a wide range of diameter ratios and the Wilson plot technique was used to develop a convection heat transfer correlation. It was found that the convection heat transfer correlation for an annulus was dependent on the annular diameter ratios. The deduced correlation predicted Nusselt numbers accurately within 3% of measured values for a Reynolds number range, based on the hydraulic diameter, from 4000 to 30,000. Ichimiya et al. (1984) carried out experiments of forced convection heat transfer in a narrow concentric annulus by turbulence promoters to examine the effects of promoters. A double-pipe helical heat exchanger was studied experimentally using two differently sized heat exchangers (Rennie and Raghavan, 2005). Overall heat transfer coefficients were calculated and heat transfer coefficients in the inner tube and the annulus were determined using the Wilson plots. The effect of swirl in an axial flow through a narrow gap has been studied by Mateescu and Paidoussis (1987). The characteristics of turbulent swirling flow in an axisymmetric annulus using the PIV technique have been studied by Chang (2004). A numerical study has been performed by De Parais et al. (1998) to estimate the hydrodynamics of an annular swirling decaying flow induced by means of a single tangential inlet in the laminar flow regime. The mean swirl intensity in the whole annular gap thickness has been found to decrease from the entrance section.
In a decaying vortex flow, where the fluid is enforced to flow downstream by imposing swirl at the entrance, the methodology of calculating the Nusselt number and velocity components in the annulus of the heat exchanger is quite limited in the open literature. Hence, this paper concerns a numerical study of laminar and turbulent flows of an incompressible viscous fluid confined between two stationary tubes with a small gap using a finite volume method. The Reynolds stress model (RSM) is implemented in the CFD code developed by Fluent 6.2 (Fluent Inc.), which is based on the finite volume method. Therefore, the main objective of this paper is to show the effect of the inlet swirl number on heat transfer in the form of the Nusselt number for laminar and turbulent flows within the annulus.

2. ANALYSIS

2.1 Governing Equations

The flow under study is steady, incompressible, and axisymmetric. The outer and inner radii of the concentric annulus are \( r_o \) and \( r_i \), respectively. Figure 1 shows the schematic of the annulus and coordinate system. The fluid enters the gap between the cylinders uniformly with two velocity components \((V, U)\) in \( z, \theta \) directions, where \( V \) and \( U \) denote the axial and tangential inlet velocity components, respectively. The thermophysical properties of fluid are assumed constant except density for which the Boussinesq approximation is employed such that the variation of density with temperature has been neglected. The basic equations governing the fluid flow and heat transfer in the annular geometry are the laws of conservation of mass, momentum, energy, turbulent kinetic energy, and its dissipation rate. The equations will be solved assuming that the gap between the two cylinders is small, and the radial velocity is also assumed to be zero since it doesn’t have the space and time to develop. The governing equations in the annulus are written as:

**Continuity:**

\[
\frac{\partial}{\partial x_i} (\rho u_i) = 0
\]  

\[ (1) \]

![Diagram of swirling flow between two concentric cylinders.](image)

Fig. 1. Swirling flow between two concentric cylinders.
Momentum:

\[
\frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} [\mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_l}{\partial x_l} \right)] + \frac{\partial}{\partial x_j} (-\rho \overline{u_i u_j'}) \tag{2}
\]

Energy:

\[
\frac{\partial}{\partial x_j}[u_i(\rho E + p)] = \frac{\partial}{\partial x_j} \left\{ \left( k + \frac{c_{\mu} \mu}{\sigma_k} \right) \frac{\partial T}{\partial x_j} + u_i(\tau_{ij})_{\text{eff}} \right\}, \tag{3}
\]

where \( E \) is the total energy and \((\tau_{ij})_{\text{eff}}\) represents the viscous heating, defined as

\[
(\tau_{ij})_{\text{eff}} = \mu_{\text{eff}} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \mu_{\text{eff}} \frac{\partial u_l}{\partial x_l} \right). \tag{4}
\]

In CFD analysis, the time-averaged governing equations were solved numerically, which were closed by the Reynolds stress turbulence model (RSM). Since RSM takes into consideration the effects of severe streamline bending due to swirl in a more proper way than the one- and two-equations models, it is best suited for the present study. Moreover, in swirling flows, due to anisotropy of the strain and Reynolds stresses tensors, eddy-viscosity models fail to capture the turbulence effects well. The transport equations for the transport of the Reynolds stresses, \(-\rho u_i u_j\), can be written as follows:

\[
\frac{\partial}{\partial x_k} \left( \rho u_i u_j \right) = \frac{\partial}{\partial x_k} \left( \frac{\mu}{\sigma_k} \frac{\partial u_i u_j}{\partial x_k} \right) - \rho \left( \frac{u_j}{\sigma_k} \frac{\partial u_i}{\partial x_k} + \frac{u_i}{\sigma_k} \frac{\partial u_j}{\partial x_k} \right) + \Phi_{ij} - \frac{2}{3} \delta_{ij} \rho \varepsilon. \tag{5}
\]

The term on the left-hand side of this equation represents convection, the terms on the right-hand side represent turbulent diffusion as proposed by Lien and Leschziner (1994), molecular diffusion, stress production, pressure strain, and dissipation, respectively. The pressure strain term, \(\Phi_{ij}\), is simplified according to Gibson and Launder (1978). The dissipation rate, \(\varepsilon\), is computed with a model transport equation as

\[
\frac{\partial}{\partial x_j} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} [(\mu + \frac{\mu}{\sigma_{\varepsilon}}) \frac{\partial \varepsilon}{\partial x_j}] = \frac{1}{2} \frac{P_{\mu}}{k} \frac{\varepsilon^2}{k} - C_{e2} \frac{\rho \varepsilon^2}{k}, \tag{6}
\]

where \(\sigma_k = 1.0\), \(C_{e1} = 1.44\), and \(C_{e2} = 1.92\) are constants taken from Launder and Spalding (1974).

2.2 Materials and Boundary Conditions

Calculations have been performed for different Reynolds numbers and different inlet swirl numbers. In the present simulations, the Reynolds number range, based on
the annular hydraulic diameter \( D_h = 2(r_o - r_i) \) of 50–10,000 is covered and the inlet swirl number range of 0–3 is covered. The Reynolds number \( Re \) and the inlet swirl number \( S \), which measures the ratio of the rate of the injected tangential momentum flux to the rate of axial momentum flux, are defined as

\[
Re = \frac{\rho V D_h}{\mu},
\]

\[
S = \frac{U}{V}.
\]

We define the nondimensional quantities as follows:

\[
\bar{V}_z = \frac{V_z}{V}, \quad \bar{V}_\theta = \frac{V_\theta}{U}, \quad \bar{Z} = \frac{z}{L}, \quad \bar{r} = \frac{r}{r_o}, \quad \alpha = \frac{r_i}{r_o},
\]

where \( \bar{V}_z, \bar{V}_\theta, \bar{Z}, \bar{r}, \) and \( \alpha \) are the dimensionless axial velocity, dimensionless swirl velocity, dimensionless axial coordinate, dimensionless radial coordinate, and radius ratio, respectively.

The specific heat \( C_p \), density \( \rho \), dynamic viscosity \( \mu \), and Prandtl number \( Pr \) of the water as the working fluid were constant values of 4178 J kg\(^{-1}\) K\(^{-1}\), 996 kg m\(^{-3}\), 0.798 \( \times 10^{-3} \) kg m\(^{-1}\) s\(^{-1}\), and 5.42, respectively. The outer radius of the annulus was \( r_o = 20.05 \) mm, while the inner radius was \( r_i = 15.93 \) mm, thereby forming an annular gap of 4.12 mm and a radius ratio of \( \alpha = r_i/r_o = 0.8 \). The length of the annulus was equal to \( L = 1500 \) mm. The boundary conditions are the no slip at the solid walls, uniform velocity and temperature profiles at the inlet and developed flow at the outlet. At the outlet boundary there is no information about the variables and some assumptions have to be made. The diffusion fluxes in the direction normal to the exit plane are assumed to be zero. The pressure at the outlet boundary is calculated from the assumption that the radial velocity at the exit is neglected, so that the pressure gradient from \( r \)-momentum is given by

\[
\frac{\partial p}{\partial r} = \frac{\rho V^2_\theta}{r}.
\]

Due to memory limitations, axis-symmetric solutions were obtained. The centerline boundary was considered to be the axis of symmetry. A uniform wall temperature \( (T_w = \text{const}) \) as a thermal boundary condition is assumed for the inner wall. The outer wall is assumed to be perfectly insulated \( (q_w = 0) \). For the uniform wall temperature boundary conditions, \( \varphi \) given by:

\[
\varphi = \frac{T - T_w}{T_{in} - T_w},
\]

where \( T_w \) is the inner wall temperature and \( T_{in} \) is the inlet fluid temperature. For the case of a uniform inner wall temperature, the Nusselt number is given by:

\[
Nu = \frac{hD_h}{k} = \left. \frac{2(1 - \alpha)}{\varphi_b} \frac{\partial \varphi}{\partial \bar{r}} \right|_{\bar{r}=\alpha},
\]
where $h$ is the heat transfer coefficient and

$$\varphi_b = \frac{T_b - T_w}{T_{in} - T_w}$$

(13)

is the dimensionless bulk temperature. The bulk temperature, $T_b$, is given by:

$$T_b = \frac{\int_{r_i}^{r_o} uTrdr}{\int_{r_i}^{r_o} urdr}$$

(14)

Inlet boundaries for $k$ and $\varepsilon$: The turbulence intensity, $I$, can be estimated from the following formula derived from an empirical correlation for pipe flows:

$$I = \frac{u'}{V} = 0.16(Re)^{-1/8}.$$ 

(15)

An approximate relationship between the turbulence length scale, $L_o$, and the physical size of the annulus hydraulic diameter, $D_h$, is

$$L_o = 0.07D_h.$$ 

(16)

The relationship between the turbulent kinetic energy, $k$, and turbulence intensity, $I$, is

$$k = \frac{3}{2} (VI)^2.$$ 

(17)

The turbulence dissipation rate $\varepsilon$ can be determined as

$$\varepsilon = C_{\mu} \frac{k^{3/2}}{L_o}.$$ 

(18)

In this study, the nonequilibrium wall functions were used near the wall as proposed by Kim and Choudhury (1995).

### 2.3 Solution Procedure

The governing differential equations for mass, momentum, and energy, the transport equations for the transport of the Reynolds stresses and its dissipation rate were solved using the control-volume-based finite difference method described by Patankar (1980). The convective and diffusive terms were discretized using the power law scheme and the SIMPLER algorithm (Patankar, 1980) was used to resolve the pressure-velocity coupling. For all the simulations performed in this study, converged solutions were usually achieved with residuals as low as $10^{-6}$ for all governing equations. The grid points are not distributed uniformly over the computational domain. They have a greater density in the radial direction and have a lower density in the axial direction. The 2D axisymmetric case with 312,000 quadrilateral grid cells is chosen and the structured grid was used for the present simulation. To give grid in-
dependent solutions, the \((r \times z)\) grid node numbers of \(52 \times 3000\), \(104 \times 3000\), and \(208 \times 3000\) are chosen, respectively. A mesh refinement study showed a grid of \(312,000\) nodes to be fine enough to capture all the flow features. The nearest nodes to the wall are approximately within \(y^+ \approx 1–2\). The mesh is sufficiently refined in order to resolve the expected large flow parameter gradients. The under-relaxation parameters on the velocities were selected 0.3–0.5 for the radial and axial, and 0.9 for the swirl velocity components. Segregated, implicit solver has been applied through an annulus. When using the present turbulent model it is necessary to run the simulation for a significant number of iterations, beyond normal convergence criteria. Experience has shown that typically 9000 iterations needed before the peak tangential velocity in the simulation stabilizes.

3. RESULTS AND DISCUSSION

Before proceeding further, it is important to validate the present simulations. We should refer here, due to the lack of experimental/numerical data from the literature that uses the decaying swirl flow technology in heat exchangers, the experimental work of Nouri et al. (1993) who performed laser Doppler velocimetry (LDV) measurements in concentric annuli without swirl at a radius ratio \(\alpha = 0.5\) and the numerical work of Chung et al. (2002) who performed a direct numerical simulation (DNS) of concentric annular pipe flows without swirl at Re of 8900 at radius ratios \(\alpha = 0.1, 0.5\). These works have been adopted as a benchmark for the present simulation because simply nothing better is available. The present predicted results of the average axial velocity profiles were compared against the published study by the above two references as shown in Fig. 2, and they were found to be in good agreement.

![Fig. 2. Mean axial velocity distribution: a) \(\alpha = 0.5\); b) \(\alpha = 0.1\).](image-url)
The problem of forced convection between two concentric horizontal cylinders is studied experimentally by Lu and Wang (2008) for different Reynolds numbers ranging from laminar to turbulent heat transfer. In order to gain confidence and understand the modeling methodology that is required to adequately simulate the heat transfer in the annulus, the experimental work of Lu and Wang (2008) is used as a study to validate the modeling approach presented in this paper. Computations were performed for this geometry without swirl to validate the accuracy of the numerical results by comparing with the experimental results. Table 1 compares the predicted and the measured average Nusselt numbers for different Reynolds numbers, the mean Nusselt number is obtained by integrating the local Nusselt number over the inner cylinder. The results presented in Table 1 were obtained using the laminar model for the Reynolds number ranging from 50 to 1000, while the RSM model was used for Reynolds numbers of 5000 and 10,000. The results shown in Table 1 represent good agreement between the present computations and experiments of Lu and Wang (2008). The predicted values of Nusselt numbers compare well with the measured values with maximum percentage error less than 5%.

The characteristics of the swirling flow and forced convective heat transfer on the conditions of both developing laminar and developed turbulent flow in narrow annuli are now presented and discussed. The solutions were carried out for laminar and turbulent flows of different inlet swirl numbers S and Reynolds numbers Re. Under laminar flow conditions, flow in the entrance region is a hydrodynamically developing flow since this is the region where the axial velocity profile develops as shown at axial locations, \( Z \), of 0.1, 0.15, and 0.2 (Fig. 2) for Re = 500 and S = 1. The hydrodynamically fully developed region beyond the entrance region in which the axial velocity profile is fully developed and remains unchanged at specified swirl number is shown in Fig. 3 for axial location \( Z \geq 0.3 \). The axial velocity distribution in the fully developed region without swirl flow (S = 0) exhibits a Poiseuille-like shape profile in laminar flow. Under turbulent flow conditions, the entrance length effect can be neglected; the flow is hydrodynamically fully developed and somewhat flatter in turbu-
lent flow as expected. The strength of vortex flow has a great influence on the axial velocity profile. Figure 4 shows the developed axial velocity profiles in the annulus in laminar flow, $Re = 500$, at $\bar{Z} = 0.5$ for inlet swirl number $S = 0$ and 2, while Fig. 5 shows the developed axial velocity profiles in the annulus in turbulent flow, $Re = 5000$, for inlet swirl number of $S = 0$, 2, and 3. It can be seen from Figs. 4 and 5 that when the inlet swirl number increases it increases the axial velocity near the wall and reduces it at the mid-gap to achieve the conservation of mass due to the existence of the recirculation zone and centrifugal forces in laminar flow in addition to eddy motion and more strong mixing in the radial direction in turbulent flow. In a decaying swirl flow, the velocity characteristics have a large effect on the heat trans-

Fig. 3. The development of the axial velocity profiles in the annulus in laminar flow.

Fig. 4. The developed axial velocity profiles in the annulus in laminar flow at different swirl numbers.
fer and momentum characteristics. Figure 6 shows the development of the nondimensional swirl velocity in the annulus in laminar flow at different axial positions $\tilde{Z} = 0.1, 0.3, \text{ and } 0.6$. Generally, at the specified inlet swirl number and Reynolds number, as an example $S = 2$ and $Re = 500$, the swirl velocity exhibits parabolic profiles and these profiles decay downstream as a result of wall friction which leads to damping of the tangential velocity. The maximum tangential velocity, as shown by the previous figure, occurs at the mid-gap. Figure 7 shows the development of the nondimensional swirl velocity in the annulus in turbulent flow at different axial positions $\tilde{Z} = 0.07, 0.13, 0.33, 0.5, \text{ and } 0.6$. In general, at the specified inlet swirl number and Reynolds number, as an example $S = 2$ and $Re = 5000$, the swirl velocity profiles also decay along the annulus due to viscous effects at the wall which leads to damp-
ing of the tangential velocity. The maximum tangential velocity does not occur at the mid-gap as the case of the laminar flow but the behavior of semi-free and forced-vortex modes is captured. A closer consideration, however, reveals that in addition to the radial-axial plane flow there is also a substantial centrifugal force, which decays with the length, thus shaping the development of the overall flow field. The local Nusselt numbers at the lower wall for the horizontal flow in the annulus for laminar and turbulent flows are simulated and plotted for different inlet swirl numbers in Figs. 8 and 9. Simulations show that the inlet swirl numbers have great influences on the heat transfer characteristics. The variation of the local Nusselt number along the annulus in laminar flow, thermally developing flow, for a uniform surface temperature is given in Fig. 8, the local Nusselt number was plotted as a function of the dimensionless axial location $\bar{Z}$ for $Re = 500$ and for the range of swirl numbers of $S = 0$, $S = 2$. 

![Swirl velocity profiles in the annulus in turbulent flow.](image)

**Fig. 7.** Swirl velocity profiles in the annulus in turbulent flow.

![Variation of the local Nusselt number along the annulus in laminar flow for a uniform surface temperature.](image)

**Fig. 8.** Variation of the local Nusselt number along the annulus in laminar flow for a uniform surface temperature.
2, and 3. The Nusselt number keeps decreasing downstream at the specified inlet swirl number because the flow is a thermally developing flow. The variation of the local Nusselt number along the annulus in turbulent flow, thermally developed flow, for a uniform surface temperature is given in Fig. 9 for the range of swirl numbers of $S = 0, 2,$ and $3$. The Nusselt number decreased approximately up to ten annulus hydraulic diameters, and then it remained nearly constant at the specified inlet swirl number because of the thermally developed flow. The last two figures indicated that the Nusselt number of both laminar and turbulent flows in the annulus with swirl is higher than that without swirl for the same Reynolds numbers. The enhancement in the local Nusselt numbers compared to values obtained in the axial flow was higher at a higher swirl number because of the higher swirl intensity. The centrifugal force results in the development of secondary flows which help in mixing the fluid and enhance the heat transfer. In addition, the Nusselt numbers and thus the convection heat transfer coefficients are much higher in the entrance region. It can be also observed that as the swirl number increases, the Nusselt number increases. This behavior is related to the wall velocity gradient, which increase with the widening of the axial velocity around the mid-gap. In the early work by Razgaitis and Holman (1976), the existence of the recirculation zone has been mentioned as a possible mechanism of heat transfer enhancement in the swirl flows. Indeed, the recirculation zone improves the convective heat transfer because it increases the axial velocity near the wall. The increase of the near-wall velocity, in turn, produces larger temperature gradients and a higher heat transfer rate. It was shown experimentally (Chang and Dhir, 1995) that one of the major mechanisms of heat transfer enhancement is a high axial velocity near the wall. For swirling flow, the local Nusselt number decayed along the annulus in the axial direction as the axial distance increased, but the effect of swirl was still
considerably evident at the end of the annulus. As the fluid flows along the annulus, the angular momentum and tangential velocity decay due to friction losses at the wall. Some researchers, including Baker and Sayre (1974), Kreith and Sonju (1965), and Wolf et al. (1969), observed that the decay process showed an approximately exponential behavior. It was also observed that the decay of swirl decreases with increasing axial Reynolds number. Figure 10 shows clearly that the swirl has a pronounced effect on the turbulent kinetic energy which is increased evidently with the swirl number. Obviously, a higher turbulence level leads to a considerable improvement in the heat transfer rate. Turbulence level improvement can be attributed to the high velocity gradients. It can be seen that the turbulent kinetic energy is lower in the mid-gap and higher in the near-wall regions. Moreover, the turbulent structures near the outer wall are more activated than those near the inner wall and the last conclusion also has been proven by Chung et al. (2002).

4. CONCLUSIONS

The characteristics of the swirling flow and convective heat transfer under the conditions of both laminar and turbulent flows in a narrow annulus were solved by means of a finite volume technique. The predicted average Nusselt numbers were found to be in good agreement with experimental data. Simulations in the present studies revealed that the inlet swirl number has great influences, in both developing laminar and developed turbulent flows, on the heat transfer characteristics. Nusselt numbers were seen to increase with imposing swirl to the entrance of the annulus. When the inlet swirl number increases it increases the axial velocity near the wall and reduces it at the mid-gap due to centrifugal forces. The increase of the near-wall velocity produces larger temperature gradients and a higher heat transfer rate. The
swirl velocity profiles decay along the annulus as a result of wall friction which leads to damping of the swirl velocity. The swirl has a pronounced effect on the turbulent kinetic energy which is increased evidently with the swirl number. Obviously, a higher turbulence level leads to considerable improvement in the heat transfer rate.

**NOMENCLATURE**

\( D_h \) hydraulic diameter  
\( I \) turbulence intensity  
\( k_{\text{eff}} \) effective thermal conductivity  
\( L \) length of the annulus  
\( L_o \) turbulence length scale  
\( \text{Nu} \) Nusselt number  
\( \text{Nu}_{\text{in}} \) average Nusselt number  
\( p \) pressure  
\( \text{Pr} \) Prandtl number  
\( q_w \) inner wall heat flux  
\( \bar{r} \) dimensionless radial coordinate, \( r/r_o \)  
\( r_i \) inner radius  
\( r_o \) outer radius  
\( r, \theta, z \) components of the location in the cylindrical coordinates  
\( \text{Re} \) Reynolds number  
\( S \) inlet swirl number  
\( T \) temperature  
\( T_b \) bulk temperature  
\( T_{\text{in}} \) inlet temperature  
\( T_w \) inner wall temperature  
\( u \) velocity  
\( V, U \) axial and tangential inlet velocity components  
\( V_z, V_\theta \) axial and tangential velocity components  
\( \bar{V}_z, \bar{V}_\theta \) dimensionless axial and tangential velocity components  
\( Z \) dimensionless axial position  

**Greek symbols**

\( \alpha \) radius ratio, \( r_i/r_o \)  
\( \varepsilon \) turbulent dissipation rate  
\( k \) turbulent kinetic energy  
\( \mu \) dynamic viscosity  
\( \mu_t \) turbulent viscosity  
\( \nu \) kinematic viscosity  
\( \rho \) density  
\( \varphi \) dimensionless temperature  
\( \varphi_b \) dimensionless bulk temperature  
\( \Psi \) swirl constant.
REFERENCES


