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Enhancement of Heat Transfer with Swirling Flows Issued into a Divergent Pipe

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Experiments have been performed to study the heat transfer process of swirling flow issued into a heated divergent pipe with a divergent angle of 7 deg with respect to the pipe axis. A flat vane swirler situated at the entrance of the pipe is used to generate the swirling flow. The heat transfer results along the pipe wall demonstrate that at low Reynolds numbers without the swirl, the separation point of the recirculation flow caused by the divergence of the pipe makes a minimum in the heat transfer. The presence of the swirler itself can have a significant effect on the heat transfer. The minimum disappears and the heat transfer results approach the case of a constant diameter pipe when the flow is swirled. The enhancement of the heat transfer is small and is mostly caused by the swirl velocity. At high Reynolds numbers, the large turbulence intensity generated by the impingement of the flow stream on the swirl vane can significantly enhance the heat transfer. A qualitative flow structure can be inferred from the heat transfer measurements. During the experiments, the Reynolds number ranges from 10^4 to $2 \; 3 \; 10^5$, and the swirl number from 0 to 0.65. Because the increase of the heat transfer with the swirl number is monotonic, correlations of the Nusselt number in terms of the swirl number and the Reynolds number are obtained.

Nomenclature

D = diameter of the pipe at the inlet
D_h = diameter of the swirler hub
D_s = i.d. of the swirler
h = local heat transfer coefficient
$\bar{h}$ = average heat transfer coefficient over the entire pipe length
k = thermal conductivity of air
L = total length of the pipe
Nu = local Nusselt number, hD/k
$\overline{Nu}$ = average Nusselt number, $\bar{h}D/k$
Q = total rate of heat transfer
q = heat flux
Re = Reynolds number, UD/ν
S = swirl number, Eq. (1)
T = temperature
U = mean velocity at the inlet of the pipe
x = axial distance from the inlet of the pipe
θ = vane angle
ν = kinematic viscosity of air

Introduction

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 swirl velocity and turbulence intensity, which provides an additional mechanism to increase the heat transfer. In the past, different methods of generating the swirling flow have been adopted, which include 1) imparting a tangential flow into a tube,[1–5] 2) flow passing a twisted tape,[6–8] and 3) flow passing a radial or an axial-flow type of swirler.[9–12] These different methods may lead to different characteristics of flow structure and heat transfer. The nature of these swirling flows and their enhancement on the heat transfer have been studied extensively. Numerical studies on the variation of velocity field and turbulence intensity in a circular pipe have also been performed.[13–15] A comparison between the data and the prediction was made to validate the numerical model.

In the current experiments, a single flat-vaned swirler is adopted to generate the swirling flow. This kind of swirler is frequently used in a gas turbine combustor. The inlet air is usually issued through a swirler to increase the mixing rate of fuel jet with air and, therefore, the combustion efficiency. One may expect that the swirling flow may have a significant effect on the flow and heat transfer process along the wall, particularly in the primary zone of the combustor where the swirling flow is just issued from the swirler and has relatively high swirl velocity and turbulence intensity. The heat transfer data in the primary zone are very important in designing the cooling process of a combustor. However, the swirling flow effect on the heat transfer process along the wall has been overlooked in the past and needs to be assessed. Because the shape of the combustor in the primary zone may be complicated, it makes the analysis of the experimental data difficult. Therefore, one may simplify it as a divergent pipe. Therefore, the objective of the present work is to study experimentally the swirling flow heat transfer process in a divergent pipe. This kind of work has not been found in the literature.

Experimental Apparatus

Experiments are performed in a divergent, circular pipe with a divergent angle of 7 deg with respect to the pipe axis. A schematic of the pipe and the experimental setup is shown in Fig. 1. The pipe is made of aluminum that has an i.d. of 12.4 cm at the entrance and a 1 cm wall thickness. Uniform airflow entering the pipe is supplied by an open-cycle wind tunnel, and the used air is discharged into the outside environment. The velocity of the uniform airflow was maintained at desirable values, and the maximum turbulence intensity is less than 0.7%. The pipe is heated with electric resistance wire wrapped uniformly outside to provide the desired uniform heat flux. To ensure uniform wiring, the resistance wire is inserted into a spiral V-shaped groove that was cut with a lathe outside the pipe wall like a screw thread. The voltage and current passing through the wire is controlled with a variable transformer. In

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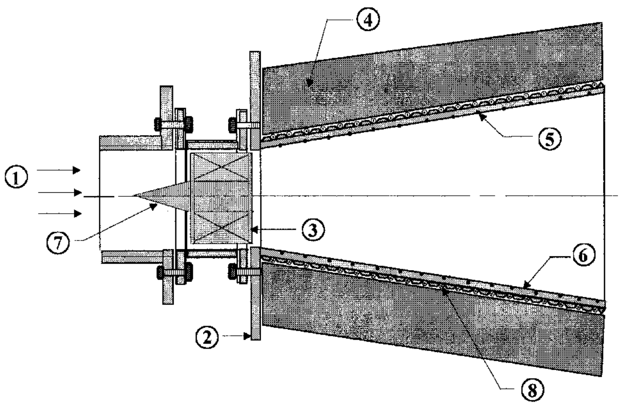

Fig. 1 Schematic of experimental setup: 1, uniform air; 2, holding plate; 3, swirler; 4, insulation; 5, thermocouples; 6, divergent pipe; 7, guide cone; and 8, resistance wire.

this way, uniform heating at a desired flux can be readily achieved. To reduce the axial conduction of heat in the pipe wall, the aluminum pipe is sliced into different sized rings, and a 1-mm-thick insulation plate is inserted between the rings. The surface of the wall is polished after all of the rings and the insulation plates are bolted together. The total length of the pipe is 34.9 cm. The entire pipe wall is well insulated outside. The heat loss rate to the environment is estimated to be less than 0.2% of the total thermal energy supplied by the electric wire.

The flat-vaned swirler used has both a vane aspect ratio and a space/chord ratio of approximate unity and is located at the entrance of the pipe to generate the swirling flow. The swirler is made of steel and has 10 pieces of 1-mm-thick flat vanes. A total of seven different swirlers with an i.d. of 12.4 cm and a hub diameter of 5 cm are made. The vane angles of these swirlers are 0, 7.7, 15, 22, 28, 34, and 39 deg, respectively. The swirling flow is usually characterized by the swirl number S, 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[10] that the swirl vane angle and the swirl number are related by

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

Accordingly, the swirl number of the swirling flow made by each of these swirlers is 0, 0.1, 0.2, 0.3, 0.4, 0.5, and 0.65, respectively. To avoid flow recirculation and the generation of turbulence before the air enters the swirler, a guide cone is placed in front of the hub. The swirler at the inlet of the pipe can be rotated at different angles and replaced for different sets of experiments.

The chromel–alumel thermocouples are used to measured all of the temperatures. Twenty-two thermocouples are used to measure the axial temperature distribution of the pipe walls, which are used to evaluate the local heat transfer coefficient. Each of the thermocouples was inserted radially from the outside into a 1.5-mm-diam hole until its junction close to the surface of the pipe wall. The holes were sealed after the installation. The nonuniformity of the circumferential temperature of each ring as a result of the swirling flow is checked with six equally spaced thermocouples. However, the nonuniformity of the temperature of each ring is found to be significantly small. The temperature of the entrance air is also measured. All of the thermocouples are calibrated in a constant temperature bath and the measurement error is found within +0.1°C. All of the temperature signals are acquired with a FLUCKE-2287A data logger and sent to a personal computer for managing and plotting. The temperature data are taken when the entire system reaches steady state, usually in 3–4 h.

The local heat transfer coefficient is evaluated with the following equation:

$$h = \frac{(Q - Q_{\mathrm{loss}})}{A_s(T_w - T_0)} \qquad (2)$$

where Q is the total thermal energy supplied by the electricity, which is equal to the product of the voltage and current passing through the resistance wire; A_s is the total surface area inside the pipe wall; and T_0 is the air temperature at the inlet of the pipe. The experimental uncertainties are determined according to the procedure proposed by Kline and McClintock.[16] The maximum uncertainty of the local Nusselt number is 5.6%, the Reynolds number is 6.2%, and the swirl number is 0.78%.

Results and Discussion

Because of the lack of divergent pipe data, the heat transfer predictions in the entrance region reported in Ref. 17 for the constant diameter pipe with Reynolds numbers ranging from 10^4 to 2×10^5 are adopted and correlated (Fig. 2). All of the data collapse into a single line when the Nusselt numbers are divided by the Reynolds number to the 0.67 power. The correlation is well performed and has a standard deviation of 0.37. The correlated results are then compared with the data for the divergent pipe, as shown in Fig. 3a for the Reynolds numbers from 10^4 to 4×10^4, and in Fig. 3b for Reynolds numbers from 6×10^4 to 2×10^5. At a low Reynolds number, the effect of divergence can be clearly seen from the deviation of the data for the divergent pipe from the correlation results for the constant diameter pipe. For $Re = 10^4$, the agreement of the data between the divergent pipe and the constant diameter pipe is very good for $x/L \leq 0.26$. For $x/L \geq 0.26$, however, the heat transfer results for the divergent pipe are significantly higher. This apparently is a result of the occurrence of the separation and recirculation of the flow that drives the cold flow from the outside near the exit and moves upstream along the heated wall. This flow makes the heat transfer in the downstream region higher than in the upstream. It is expected that the separation point is close to the location for the minimum in the heat transfer. The recirculation flow that occurred along the divergent pipe wall is observed by using cigarette smoke. The smoke at the exit near the wall tends to move into the pipe along the divergent wall and then exit with the core flow.

As the Reynolds number increases, the minimum in the heat transfer associated with the point of separation moves upstream. The locations for the minimum in the heat transfer for Re from 2×10^4 to 4×10^4 are very close to $x/L = 0.18$ (Fig. 3a). The upstream motion of the separation point tends to enlarge the recirculation region, which leads to intense mixing between the recirculation flow and the mainstream, and eventually results in a reduction of the cold reversed flow from the outside and of the recirculation region. This action leads to a reduction in the heat transfer in the downstream region. In

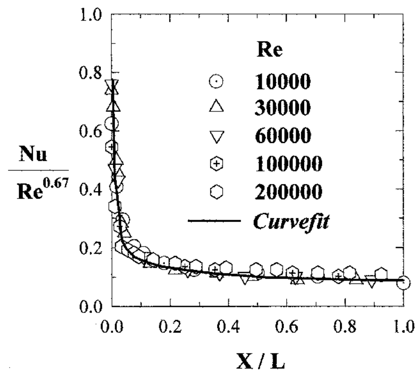

Fig. 2 Correlation of predicted normalized Nusselt numbers for constant diameter pipe (data are taken from Kays and Perkins[17]).

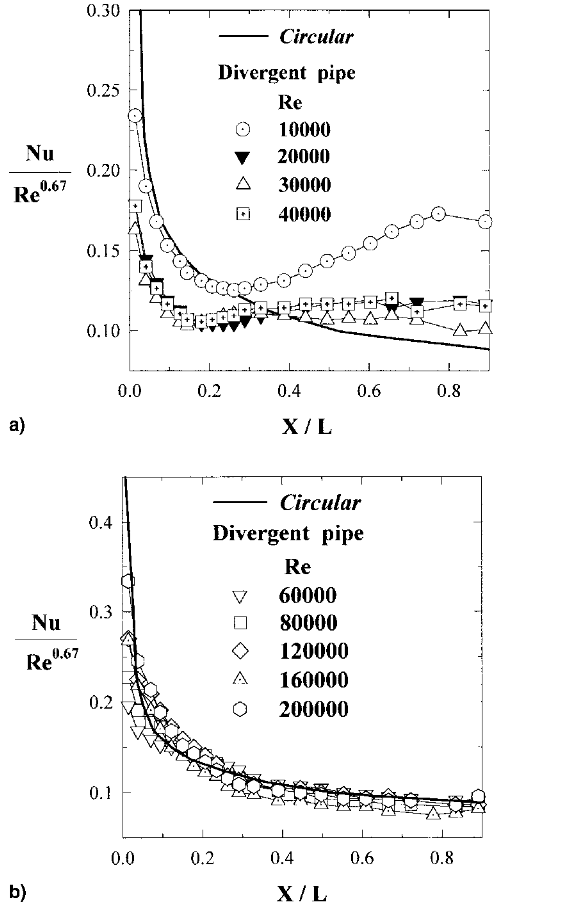

Fig. 3 Normalized Nusselt number distributions in the divergent pipe for a Reynolds number from a) 10^4 to 4 × 10^4 and b) 6 3 10^4 to 2 3 10^5.

addition, the recirculation flow region acts as a blockage, which can reduce the flow velocity in the upstream, and thus, the heat transfer. For Reynolds numbers greater than 6×10^4, the normalized Nusselt numbers $Nu/Re^{0.67}$ for the divergent pipe approach that for the constant diameter pipe (Fig. 3b). This fact suggests that the mixing between the recirculation flow and the mainstream is so intense that the recirculation flow is completely washed away by the mainstream.

Smoke visualization is also made at these high Reynolds numbers. However, the smoke at the exit near the wall does not move into the pipe, but move out into the ambient. Therefore, the mainstream near the wall for the divergent pipe is the same as the one for a constant diameter pipe. In this way, to satisfy the mass conservation, the recirculation flow is expected to occur in the central region of the divergent pipe. Note that this recirculation flow is also expected to occur when the mainstream is swirled.

When the swirler with a guide cone in the front and its swirl number $S = 0$ is placed at the entrance of the divergent pipe, the normalized Nusselt number $Nu/Re^{0.67}$ increases significantly (Fig. 4), particularly at low Reynolds numbers. The monotonic decrease of the normalized Nusselt number toward downstream for Re from 10^4 to 4×10^4 suggests that the separation and the recirculation flow along the pipe wall does not occur. It appears that the swirl hub tends to guide the mainstream to move along the divergent wall of the pipe and washes away the recirculation flow. A wake flow behind the swirl hub is expected to be generated which, because of shearing with the mainstream, can generate a flow with high turbulence intensity. This causes significant enhancement of heat transfer, as shown in Fig. 4 for $Re = 10^4$ or 2×10^4. However, the normalized Nusselt number decreases with an increasing Reynolds number. This decrease occurs because as the Reynolds number increases, the mainstream tends to move along the divergent wall, and shearing of the mainstream with the wake flow in the central region becomes relatively weak, which can generate a flow with relatively low turbulence intensity. For Reynolds numbers greater than 4×10^4, the normalized Nusselt numbers for the divergent pipe with insertion of the swirler at the entrance approach (Fig. 5) the one with absence of the swirler and the one for a constant diameter pipe.[17] It appears that at high Reynolds numbers, the presence of the swirler has no effect on the flow and heat transfer.

The effect of the guide cone placed in the front of the swirler is also examined. Comparison of the normalized Nusselt numbers between the case with the guide cone and the case without it are shown in Fig. 6a. For the case without the cone, the flow in front of the swirl hub may have intense circulation and shearing with the flow passing through the swirler. This can generate additional turbulence. Therefore, one may expect that the normalized Nusselt numbers for the case without the cone are higher than with the cone. However, the opposite results are obtained. This can be explained by the fact that the swirl hub without the cone can create a greater blockage for the mainstream than with the cone. The greater blockage can make the mainstream move along the divergent wall much earlier; thus the shearing between the mainstream and the wake flow behind the swirl hub is weaker. The normalized Nusselt numbers also decrease with an increase of the Reynolds numbers, and approach the case for a constant diameter pipe. At high

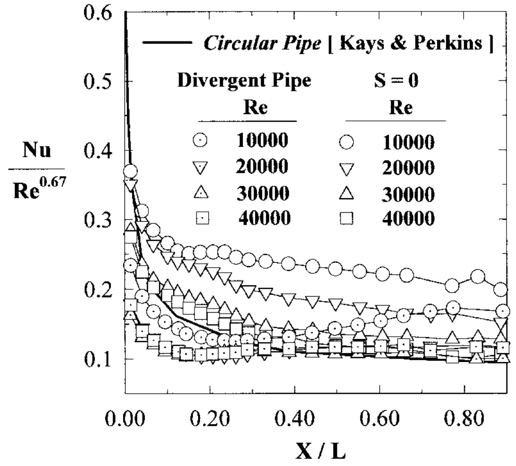

Fig. 4 Comparison of the normalized Nusselt number distributions for a Reynolds number from 10^4 to 4 3 10^4 between the case without swirler and the case with the swirler for $S = 0$.

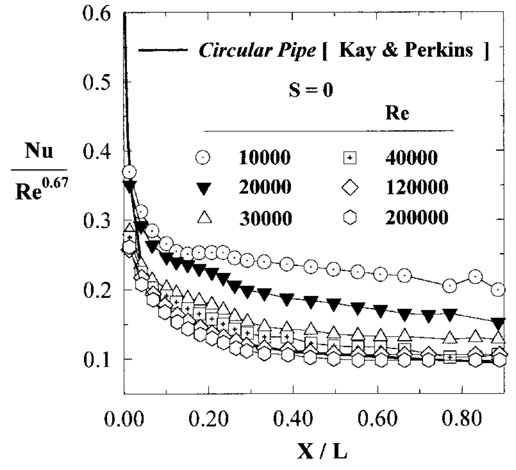

Fig. 5 Normalized Nusselt number distributions for the case with swirler ($S = 0$) and a Reynolds number from 10^4 to 2 3 10^5.

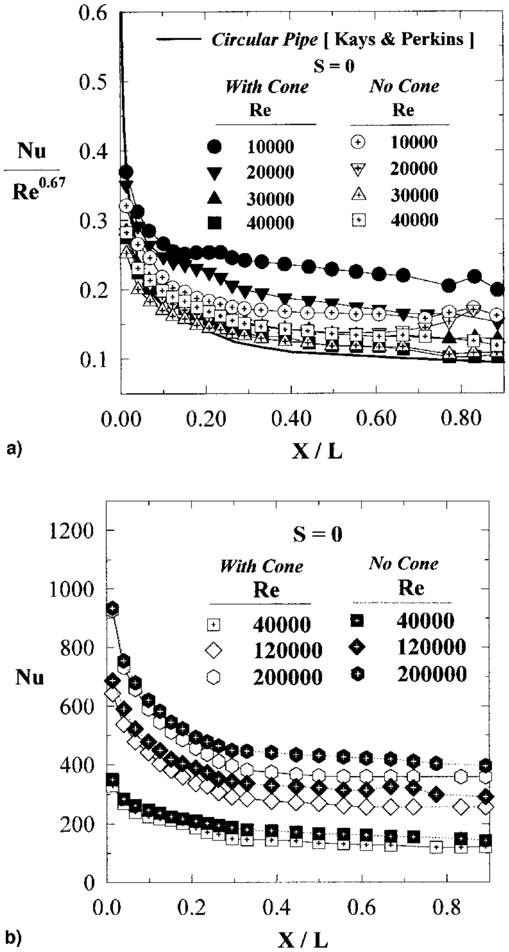

Fig. 6 Comparison of the normalized Nusselt number distributions for a Reynolds number from a) 10^4 to 4 3 10^4 and b) 4 3 10^4 to 2 3 10^5, and S = 0 between the case with the cone placed in the front and the case without the cone.

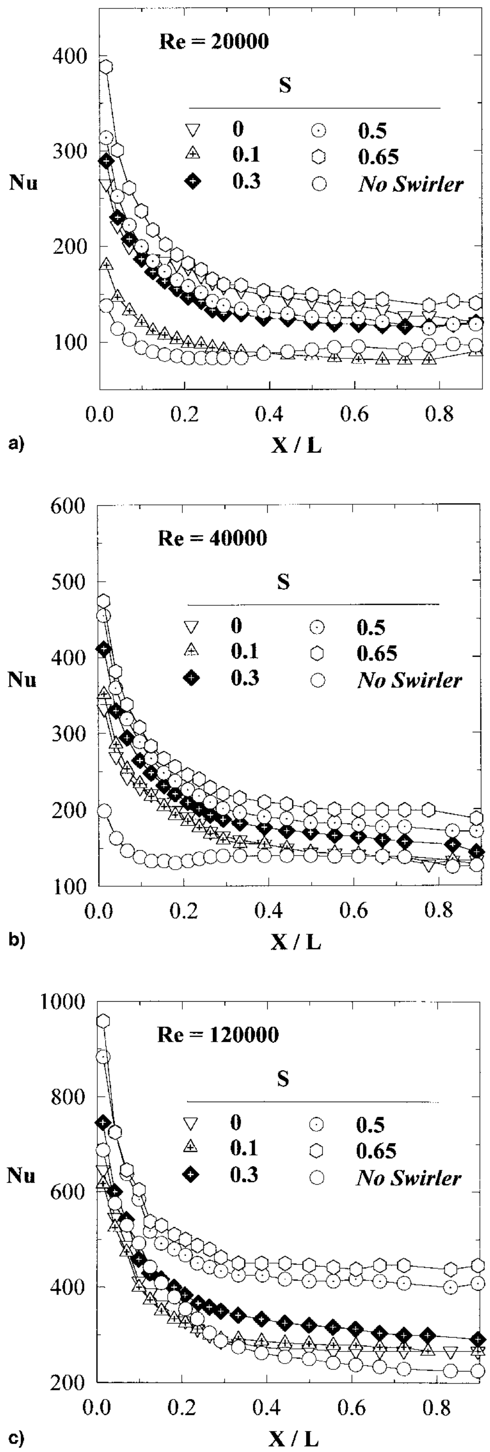

Fig. 7 Swirling flow effect on the Nusselt number distributions for *Re* = a) 2 3 10^4, b) 4 × 10^4, and c) 1.2 3 10^5.

Reynolds numbers, however, the Nusselt numbers for the case without the cone approach the case with the cone (Fig. 6b). It appears that at high Reynolds numbers, this blockage effect on the mainstream caused by the absence of the guide cone is negligible.

When the swirling flow is generated inside the channel, the heat transfer can change significantly with the swirl number of the flow (Figs. 7 and 8). Because the swirl flow generated by the flat-vaned swirler can produce a flow with high turbulence intensity,[11] it is expected that the swirl velocity associated with high turbulence intensity can significantly enhance the heat transfer.[9] When the Reynolds number is low, however, as shown in Fig. 7a for S = 0.1, the slight increase of the swirl does not increase, but decreases the heat transfer. This is apparently because the swirl velocity tends to make the flow move along the divergent wall of the pipe. Thus, the shearing of the mainstream with the wake flow behind the swirl hub becomes weak, which can significantly reduce the heat transfer. Despite the case of S = 0, however, the Nusselt number indeed increases monotonically with an increasing swirl number, because of an increase in the swirl velocity and turbulence intensity in the mainstream. However, the Nusselt number for the extreme case of S = 0.65 is only slightly higher than the case of S = 0. Therefore, an increase of the swirl number at a low Reynolds number does not benefit the heat transfer.

However, the situation improves at high Reynolds numbers (Figs. 7b and 7c). The swirl can significantly enhance the heat transfer, particularly at high Reynolds numbers (Fig. 7c). For $Re = 4 \times 10^4$, the Nusselt number at S = 0.65 is approximately 1.5 times higher than the Nusselt number at S = 0. For Re = 12×10^4, however, the Nusselt number at S = 0.65 is approximately 1.7 times higher than the Nusselt number at S = 0. However, the flow with low swirl number, e.g., S = 0.1, has only a slight effect on the heat transfer.

The explanation that a swirling flow at high Reynolds numbers gives a better heat transfer result is given here. It is noted that the swirling flow generated by the flat-vaned swirler has the nature that when the swirl number increases, both the swirl velocity and the accompanying turbulence intensity increase as a result of the impingement of the flow stream on the swirl vane. These can significantly enhance the heat transfer. At low Reynolds numbers, i.e., the flow velocity is relatively low, the impingement of the flow stream on the swirl vane is not so

Fig. 8 Reynolds number effect on the normalized Nusselt number distributions for S = a) 0.1, b) 0.3, and c) 0.65.

severe, and the turbulence intensity generated by the impingement is not so large. The enhancement of the heat transfer is mostly caused by the swirl velocity. However, the swirl velocity has an opposite effect to reduce the shearing of the mainstream with the wake flow behind the swirl hub and to reduce the heat transfer. Therefore, an increase of the swirl number of the flow at low Reynolds numbers does not significantly enhance the heat transfer. At high Reynolds numbers, however, the situation is somewhat different. An increase in swirl velocity is generally accompanied by a significant increase of turbulence intensity in the flow because of the impingement of the flow stream on the swirl vane. This results in an increase of the heat transfer with the swirl number that is much greater in the high Reynolds number region.

Figure 8a shows that for $S = 0.1$, an increase in the Reynolds number decreases the normalized Nusselt number and makes the heat transfer results approach the case of a constant diameter pipe without swirl. In this case, the swirl does not increase, but decreases the heat transfer. This is attributed to the fact that the swirl velocity can make the flow move along the divergent wall like a constant diameter pipe. As the swirl number increases, the high swirl velocity associated with high turbulence intensity can significantly increase the heat transfer. The normalized Nusselt number is significantly higher than the case of a constant diameter pipe without swirl (Figs. 8b and 8c). In addition, at high swirl, the normalized Nusselt numbers for Re from 10^4 to 2×10^5 tend to collapse into a single line. This suggests that at high swirl, the normalized Nusselt numbers $Nu/Re^{0.65}$ in the divergent pipe are the same as ones in a constant diameter pipe. The effect of divergence on the heat transfer at high swirl number is relatively small.

The average heat transfer coefficients over the entire heated pipe are calculated by dividing the average heat flux by the mean temperature difference between the heated wall and the entrance air. The average Nusselt numbers, which are defined as $\bar{h}D^h/k_f$, are presented in Fig. 9. 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 the heat transfer at a low Reynolds number is mainly a result of the swirl velocity. The turbulence intensity generated by the impingement of the flow stream on the swirl vane is not large, which does not play an important role in affecting the heat transfer. In addition, the divergence of the pipe that causes separation and recirculataion of the flow can enhance the heat transfer when the mainstream has no swirl. When the mainstream is swirled, even with a low swirl number, the swirl velocity makes the flow move along the divergent wall and washes away the recirculation flow. This makes the heat transfer in a divergent pipe approach the case of a constant diameter pipe. As the Reynolds number increases, the high-speed flow tends to move along the divergent wall and makes the heat transfer approach the wall of a constant diameter pipe. In addition, the swirling flow can enhance the heat transfer caused by the relatively high turbulence intensity generated by the impingement of the flow stream on the swirl vane. This can be argued from the small increase of the Nusselt number at low swirl numbers from $S = 0$ to 0.1, and at high swirl numbers from $S = 0.5$ to 0.6. At low swirl numbers, the increase of the swirl number or the vane angle does not significantly enhance the impingement of the flow stream on the swirl vane. At high swirl numbers, because the turbulence intensity in the main-

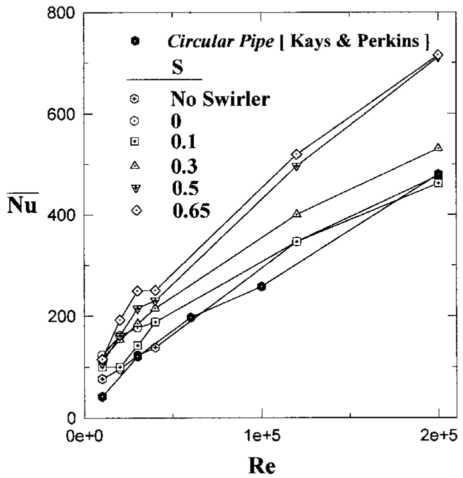

Fig. 9 Comparison of the average Nusselt number distributions for the case of the constant diameter pipe, the divergent pipe, and the case with the swirler placed at the entrance of the pipe.

stream has already reached a high level, an additional increase of the vane angle could not cause a significant increase in the turbulence intensity.

Because the Nusselt number increases monotonically with increasing both the Reynolds number and the swirl number in the high Reynolds number region, and the heat transfer results are very close to those of a constant diameter pipe, the divergence of the pipe, even with a 7 deg angle, has no appreciable effect on the heat transfer. Therefore, an attempt was made to correlate the Nusselt number in terms of the Reynolds and swirl numbers. To include the data for $S = 0$ in the correlation, instead of using S, one uses the parameter $S + 1$. For $4 \times 10^4 \leq Re \leq 2 \times 10^5$ and $0 \leq S \leq 0.65$, the following equation is obtained:

$$Nu = 0.165 Re^{0.62}(S + 1)^{0.89}(x/L)^{-0.22} \tag{3}$$

The preceding correlation has a standard deviation of 6.87%. Note that the power of the Reynolds number is very close to 0.67, which is for the case when a uniform flow moves into a constant diameter pipe. This further suggests that with or without the presence of the swirling flow, the divergence of the pipe has no appreciable effect on the local heat transfer when the Reynolds number is higher. Therefore, an attempt is made to correlate the Nusselt number in terms of $(S + 1)$, x/L, and $Re^{0.67}$; this results in the following equation:

$$Nu = 0.097 Re^{0.67}(S + 1)^{0.89}(x/L)^{-0.22} \tag{4}$$

The preceding correlation has a standard deviation of 1.5%, which is an improvement over that of Eq. (3). The least-square fit of the data is presented in Fig. 10a. At a low Reynolds number and low swirl number region, however, the recirculation of the flow caused by the divergence of the pipe complicates the heat transfer process and makes the correlation difficult. Therefore, an attempt is made to correlate the Nusselt number in the high swirl number region. For $10^4 \leq Re \leq 4 \times 10^4$ and $0.3 \leq S \leq 0.65$, one obtains:

$$Nu = 0.79 Re^{0.48}(S + 1)^{0.67}(x/L)^{-0.23} \tag{5}$$

The preceding correlation has a standard deviation of 8.6%. The least-square fit of the data is presented in Fig. 10b.

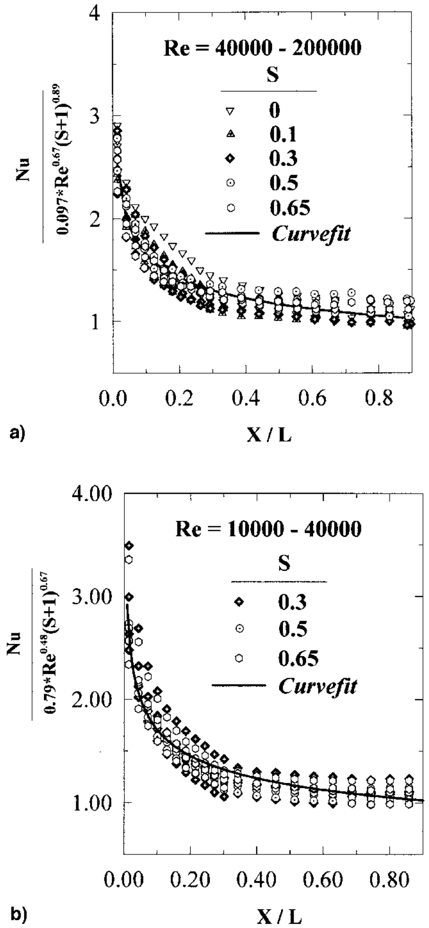

Fig. 10 Correlations of the Nusselt number distributions for a) *Re* from 4×10^4 to 2×10^5 and *S* from 0 to 0.65, and b) *Re* from 10^4 to 4×10^4 and *S* from 0.3 to 0.65.

Conclusions

The experiments considered the case with the absence of the swirler and the cases with the presence of the swirlers for S from 0 to 0.65. All of the current data were compared with the published results for the case of a constant diameter pipe. For the case at low Reynolds numbers without the swirler, the divergence of the pipe, which causes recirculation of the flow, can enchance the heat transfer, particularly in the downstream region of the pipe. In addition, the presence of the swirler itself or a guide cone in the front can also enhance the heat transfer as a result of the occurrence of a wake flow behind the swirl hub, which increases the turbulent mixing of the flow. At high Reynolds numbers, the flow tends to move along the divergent wall and washes away the recirculation flow. The divergence of the pipe with even a 7 deg of angle has no appreciable effect on the heat transfer. The heat transfer results approach the case of a constant diameter pipe.

When the swirling flow is generated at low Reynolds numbers, the enhancement of the heat transfer is relatively small and is mostly a result of the effect of the swirl velocity. As the Reynolds number increases, the turbulent intensity generated by the impingement of the flow stream on the swirl vane becomes dominant, which can significantly enhance the heat transfer. Because the Nusselt number is found to increase monotonically with both the Reynolds and swirl numbers, correlations of the Nusselt number in terms of these parameters are successfully obtained. These correlations further suggest that with or without the presence of the swirling flow, the divergence of the pipe has no appreciable effect on the heat transfer when the Reynolds number is higher.

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