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Turbulent heat transfer in a channel with two right-angle bends

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INTRODUCTION

IN THE PRESENT study a channel which has two right-angle sharp bends is considered. This configuration is a basic type of corrugated channel wall heat exchanger. The flow in such a channel shows quite complex flow patterns, including separation, reattachment, recirculation and flow deflection. Experimental studies on this flow configuration have been performed by Goldstein and Sparrow [1] and Izumi *et al.* [2]. Goldstein and Sparrow measured the local mass transfer rates in a corrugated wall channel by using the naphthalene sublimation technique. Izumi *et al.* measured heat transfer rates on a channel wall with two right-angle bends. A numerical study has been performed by Amano [3] for a laminar flow using the same channel configuration of Izumi *et al.* [2]. The agreement between the computation and experiment was quite reasonable. In the present paper this study is extended to the turbulent flow regime by modifying the most recent near-wall model of the author [4] for a channel with two bends.

MATHEMATICAL MODEL

The flow field in the geometry considered in this paper can be described by the steady, two-dimensional form of continuity, momentum and energy equations coupled with the standard k–ε turbulence model.

Since the turbulence model employed in the present computation is the high Reynolds number form of the turbulence model, it is necessary to introduce the 'wall-function' approach into the computations of wall adjacent numerical cells in order to account for the viscous effect from the wall. The near-wall model adopted for the evaluation of the k and ε equations in the wall-vicinity region is based on the three-layer model given in Amano [4].

RESULTS AND DISCUSSION

Flow field

In Fig. 1, the streamwise velocity profiles are shown in the channels of $H/W = 1$, 2, 3 and 4 for Reynolds numbers of $Re = 3000$ and 8000. In every channel, a fully-developed flow from upstream separates at the first bend corner B. This separated flow reattaches on the wall CD for the small step height ($H/W = 1$), while it reattaches on the wall BC for $H/W > 1$. When the step height is large, the flow redevelops in the channel between two bends BF–CG. This under-developed flow separates again at the corner G and deflects toward the wall CD, resulting in a high heat transfer rate on the CD wall. This aspect is shown in Figs. 2 and 3. A recirculating region is created along the wall GH just downstream of the corner G due to this separation. Then the flow again re-develops in the downstream region of the channel.

It is generally observed that the effect of Reynolds number on the velocity profile is minor; this is in contrast to the laminar flow case [3].

Local heat transfer

Local Nusselt numbers were computed along the channel walls to investigate local heat transfer characteristics affected

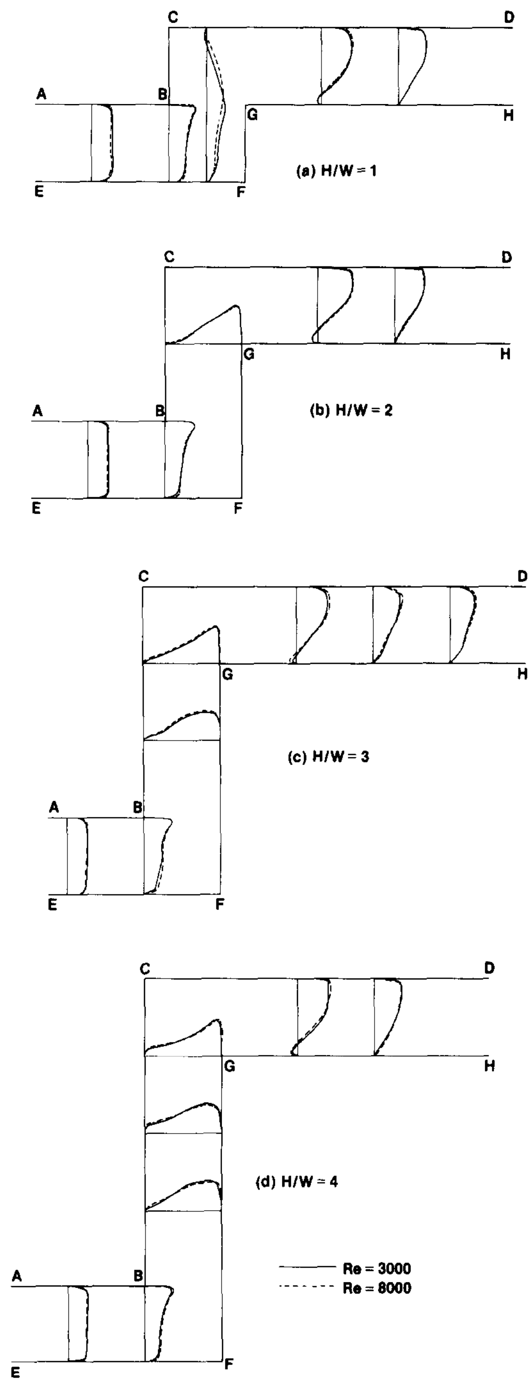

FIG. 1. Streamwise velocity profiles in the channel.

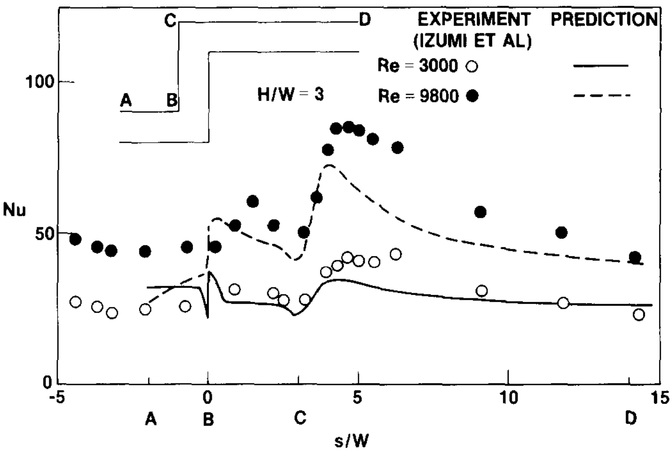

FIG. 2. Nusselt number distribution along the ABCD wall.

by the channel bends. In Fig. 2 the computed results are compared with the experimental data obtained by Izumi *et al.* [2] for a step ratio of $H/W = 3$ and for $Re = 3000$ and 9800. These data are measured on the ABCD wall. Generally, agreement between experimental data and computed results is fairly reasonable showing similar trends, although there is a maximum discrepancy of about 25%. In both experiment and computation the maximum heat transfer rate occurs about one to two channel widths downstream of the concave corner C.

In Fig. 3 the computed local Nusselt number distributions are shown. In each H/W case similar features are observed in the distribution of local Nusselt numbers. Along the ABCD wall, the Nusselt number shows the first peak at the convex corner B due to the flow separation. Then it slowly decreases along the BC wall and becomes minimal at the concave corner C; this is attributed to the flow recirculation in this region. The heat transfer rate then increases rapidly from the corner C to the second peak at the location two channel widths ($2W$) downstream therefrom, and then slowly decreases toward the outflow. This second peak is mainly due to the deflected flow which separates at the convex corner G on the opposite side of the wall. The flow is also highly accelerated at the location of the corner G when it separates as is represented by the velocity profile at this point in Fig. 1, and it results in maximum heat transfer rates on the CD wall.

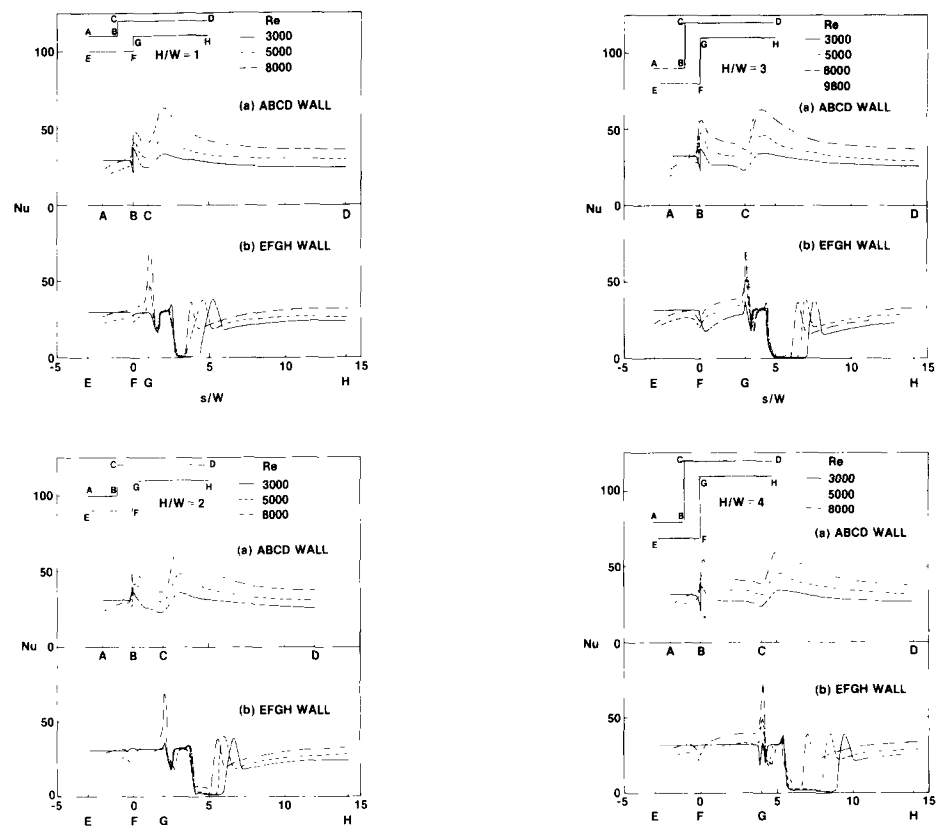

FIG. 3. Nusselt number distribution along the channel wall.

Along the EFGH wall, the distribution of Nusselt numbers is entirely different from that on the ABCD wall. The Nusselt number displays an increase on the FG wall having minimum values at the concave corner F. It then reaches its maximum value at the convex corner G because of the highly accelerated and deflected flow at this location, as explained above. Beyond the corner G the heat transfer rate undertakes rather complex features showing a rapid drop at about two channel widths downstream from the corner G followed by a slow recovery toward the downstream of the channel. As is shown in Fig. 1, the velocity profiles near the corner G on the GH wall are negative which indicates a flow recirculation in this region. In the region about two to three channel widths (2–3W) downstream from the corner G, the velocity gradient $\partial U/\partial y$ is very small and thus results in low levels of heat transfer rates attributed to low shear stresses.

With regard to the effect of Reynolds number, it is seen that the pattern of the local Nusselt number does not vary with Reynolds number on the ABCD wall, while a slight dependency of the behavior of the Nusselt number on Re is detected in the downstream region of corner G on the EFGH wall. This latter feature may be attributed to the difference in the flow pattern caused by changing Reynolds numbers in the redeveloping region; that is, the recirculating region in this part seems to be contracting as the Reynolds number increases.

Maximum and average heat transfer

In Fig. 4 the maximum Nusselt number on each wall is plotted as a function of Reynolds number for the step ratio $H/W = 3$. The maximum value of the Nusselt number on the ABCD wall usually occurs at $2W$ downstream from the concave corner C where the flow is accelerated due to the deflection caused by the separated flow at the convex corner G. Meanwhile, the maximum point on the EFGH wall occurs at the convex corner G where the strong convection resulted due to the deflection from upstream. Although a visible discrepancy is depicted between the experimental data of Izumi *et al.* [2] and the present computation, agreement of the slope is fairly reasonable (within 15%). In both experiment and computation, the dependency of Nu_{max} on Re is approx. $Re^{0.6}$.

In Fig. 5 the average Nusselt number is plotted as a function of Re for the step ratios $H/W = 1$–4. Again, the agreement between the measured values and the computed results is fairly reasonable (within 20%). In the turbulent flow regime the dependency of Nu on Re shows approx. $Re^{0.5}$ which is in accordance with the experimental data. It is also noticed that the average Nusselt number does not depend on the step ratio H/W of the channel.

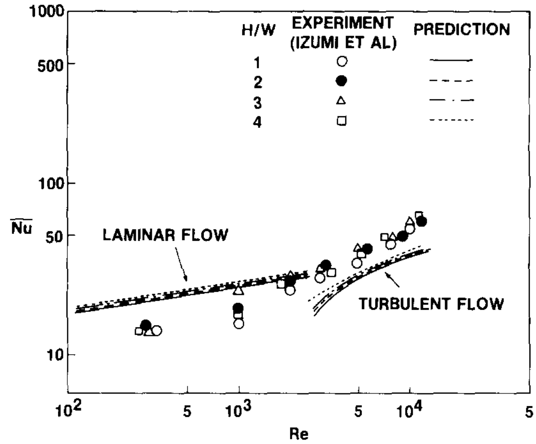

FIG. 5. Average Nusselt number as a function of Reynolds number.

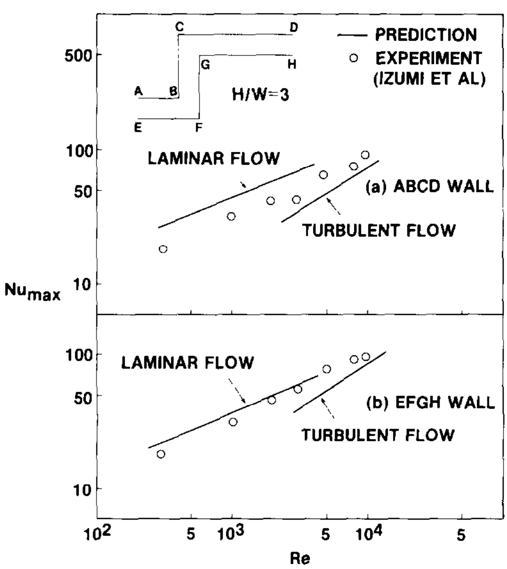

FIG. 4. Maximum Nusselt number as a function of Reynolds number.

In both Figs. 4 and 5 Nusselt numbers for the laminar flow regime are also displayed. The computed results for this regime were taken from the earlier work of Amano [3].

CONCLUSIONS

A numerical analysis was performed for the flow in a channel with two right-angle bends. It was found that such a channel created a complex flow pattern involving separation, impingement, reattachment, recirculation and flow deflection. The main conclusions to emerge from this study are summarized as follows:

1. The effect of Reynolds number on the local heat transfer rate on the channel wall is negligibly small for turbulent flow, which is in contrast to that for laminar flow.
2. The effect of step ratio on the heat transfer rate is small.
3. The dependency of the average Nusselt number on the Reynolds number is approx. 0.5 power.

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