Heat Transfer and Friction Characteristics in Turbulent Rhombic Duct Flows[*]

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1   Introduction

Turbulent heat transfer is of great importance in a wide range of engineering applications. In many turbomachines and heat exchangers, conduits of various cross-sectional shapes are being used to improve the heat transfer and pressure loss.   They may be parallel, triangular, rectangular or hexagonal partitions.   In present work, we focus our attention on a square and its derivatives, i.e., rhombuses.

In non-circular ducts, anisotropic and inhomogeneous Reynolds stresses would cause secondary flows of Prandtl's second kind.   Their effect on momentum and heat transfer is significant.   However, studies on turbulent flow in non-circular ducts are relatively few.   Only two DNS data sets [1, 2] and some experimental results [e.g., 3] in square ducts were reported.   The understanding of the effect due to the presence of two intersecting walls on turbulent transport mechanism is neither complete nor satisfactory.

From the above background, we study the fully developed turbulent velocity and temperature fields in square and rhombic ducts.

2   Numerical Procedure

The flow geometry and the coordinate system are shown in Fig. 1.   DNS is carried out in five different rhombic ducts.   The acute angle θ is changed from 90 degrees of a square duct to 75, 60, 45 and 30 degrees. The Reynolds number, Re_b, based on the bulk mean streamwise velocity, U_b, and the hydrodynamic diameter, L_h, is set to be about 4470.   No-slip condition is imposed on the walls, while the periodic condition is applied at the streamwise boundaries.   The Prandtl number is 0.71.   As for the thermal boundary conditions, the cooling rate is constant along the duct axis with the wall peripheral temperature being constant [4].   Details are available in Fukushima & Kasagi [5].

3   Results and Discussion

In Fig. 2, three ratios of friction factor, f_0/f_p, j-factor, j_0/j_p and Reynolds analogy factor, $(j_0/f_0)/(j_p/f_p)$ are shown against θ.   The mean values, f_0 and j_0, have been ob-

tained by averaging local f and j over the side length. Here, the Petsukov empirical equations, f_p and j_p, are used as references.   Comparing f_0/f_p and j_0/j_p in the five different rhombic ducts, it is clear that both values decrease appreciably as the acute apex angle θ is de-
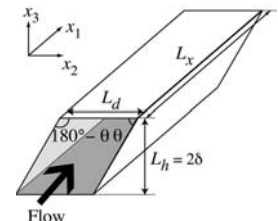

Fig. 1 Flow geometry.

creased.   The rate of decrease of f_0/f_p is smaller than that of j_0/j_p, therefore $(j_0/f_0)/(j_p/f_p)$ also decreases with θ. The square duct should give the best performance among the present five rhombic ducts in terms of friction and heat transfer properties.

Figures 3(a)-(e) illustrate the variation of the mean secondary velocity vectors at different acute angles, θ. Near the corners of acute angle, a pair of distorted counter-rotating vortices appears with their centers located further away from the corners as θ is decreased. On the other hand, near the corners of obtuse angle, there appear a pair of somewhat smaller, but more circular counter-rotating vortices.   The maximum secondary velocity near the acute angle corners is larger than that near the obtuse angle corners.   However, the vortices near the acute corners cannot spread deep into the corner. These facts imply that the enhancement of turbulent heat and momentum transfer by the secondary flow is stronger near the obtuse corners than near the acute ones.

The effects of these secondary flows on the mean streamwise velocity, U/U_b, and the mean local temperature difference, $(T-T_w)/(T_b-T_w)$, fields are shown in Figs.
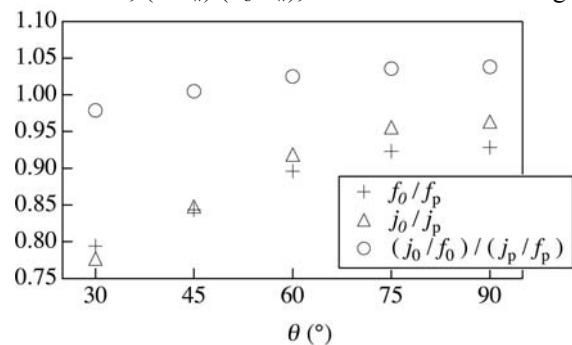

Fig. 2 Variation of f_0/f_p, j_0/j_p and $(j_0/f_0)/(j_p/f_p)$ with θ.

4(a)-(e). The subscript w denotes the wall value. Since the induced secondary flow transports efficiently high-momentum and high-temperature fluid from the center to the corner of the duct, the contours of both U/U_b and $(T-T_w)/(T_b-T_w)$ are distorted accordingly. As the acute intersecting angle becomes smaller, the deformation of $(T-T_w)/(T_b-T_w)$ appears smaller than that of the U/U_b.

The distributions of local friction factor and j-factor are plotted against the distance from the acute angle corner, x_2/δ in Fig. 5(a). The considerable decrease of f/f_p and j/j_p near the acute angle corner with the decrease of θ is observed. The dissimilarity between local momentum and heat transfer is illustrated in Fig. 5(b). Although the value near the obtuse angle corner approaches a constant value asymptotically, it decreases markedly near the acute angle corner. From a viewpoint of heat transfer equipment design, it is generally concluded that ducts which have only obtuse angle corners should be more efficient than those with acute angle corners.

Finally, the effect of the thermal boundary condition is considered. In the present study, we assumed a constant heat transfer rate along the axis with the peripheral wall temperature kept constant. With this boundary condition, the mean wall temperature gradient term, $-u\partial T_w / \partial x$, acts as the driving force in the equation of the local temperature difference, $(T-T_w)$. The distribution of this term in the cross-stream plane is equal to that of the streamwise velocity because T_w decreases linearly in the x direction [4]. Note that this velocity is driven by the pressure gradient which is almost uniform in the cross-stream plane. These facts result in deterioration of heat transfer properties near acute angle corners, where the streamwise velocity is very much decelerated.
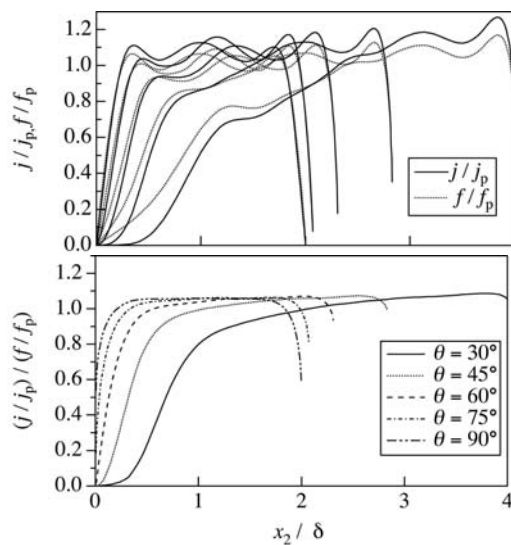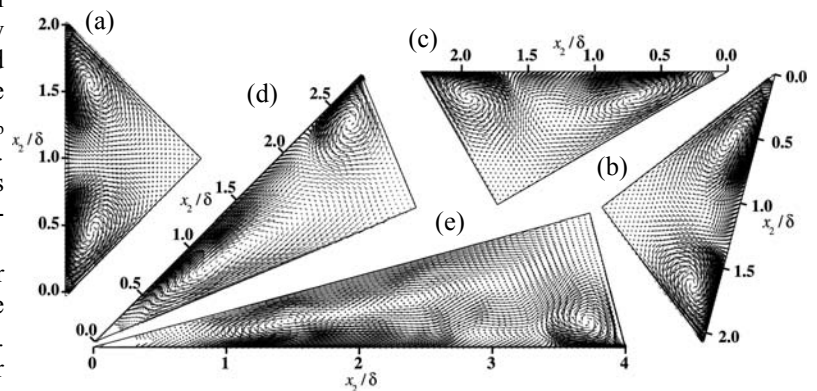

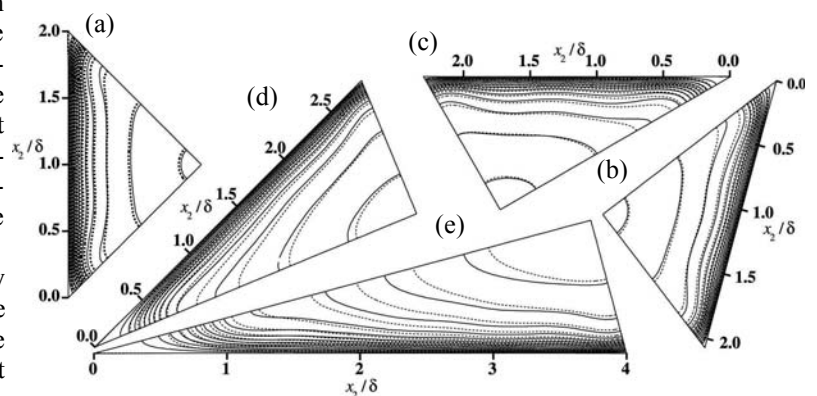Fig. 3 Mean secondary velocity vectors.

Fig. 4 Contours of mean streamwise velocity, U/U_b (solid line), and mean temperature difference, $(T-T_w)/(T_b-T_w)$ (dotted line). The increments are 0.1. (a) $\theta=90°$, (b) $\theta=75°$, (c) $\theta=60°$, (d) $\theta=45°$, (e) $\theta=30°$.

4 Conclusions

The square duct should give the best performance among five rhombic ducts tested in terms of friction and heat transfer properties. The heat transfer properties near the acute corners are deteriorated because of the dissimilarity inherent in the velocity and thermal boundary conditions. For designing thermal equipment, the duct with only obtuse corners would be more efficient.

Fig. 5 (a) Distribution of local friction factor, f/f_p, and local j-factor, j/j_p; (b) the ratio between f/f_p and j/j_p.

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