FORCED CONVECTION HEAT TRANSFER FROM A RECTANGULAR CYLINDER: EFFECT OF ASPECT RATIO

Mohammad Rahnama*, Seyed-Majid Hashemian*, Mousa Farhadi**

*Mechanical Engineering Department, Kerman University, Kerman, Iran,
**Mechanical Engineering Department, Mazandaran University, Babol, Iran.

Corresponding author: rahnama@mail.uk.ac.ir, Tel. & Fax: (+98) 341 3220064

Abstract

Convective heat transfer from a rectangular cylinder placed in the middle of a channel was investigated numerically. The unsteady laminar flow equations were discretised using finite volume method for the range of Reynolds numbers between 50 and 200. Computations were performed for cylinder aspect ratios of 0.5, 1 and 2. Results of flow and thermal fields were obtained for both the instantaneous and mean flow. Nusselt number distribution along each side of the cylinder showed that front side has maximum heat transfer rate compared with other sides. Results of mean total Nusselt number variation with Reynolds number for different aspect ratios showed that increasing aspect ratio decreases total Nusselt number for the Reynolds numbers considered in this study.

1 Introduction

Heat transfer from a rectangular cylinder placed in the middle of a channel occurs in many engineering applications such as cooling of electronic components and heat transfer in heat exchangers. Fluid flow in such geometries is laminar at low Reynolds numbers; a steady type of flow exists at Reynolds numbers less than about 50 while unsteady laminar flow appears for higher Reynolds numbers. Bruer et. al. [1] showed that two-dimensional computation of flow could be done with good accuracy for Re<300 and three-dimensional effects are negligible in this range of Reynolds number.

Unsteady flow computations for a square cylinder placed in a channel was done by various authors [2,3,4] while heat transfer predictions has not been done in detail to the same extent. Forced convection simulation of steady and unsteady two-dimensional flow around a square cylinder was conducted by Kelkar and Patankar [5]. Their computations showed that the temperature fields in the wake for steady and unsteady flow are quite different while the overall heat transfer from the square cylinder in unsteady flow was almost found the same as that in steady one. Some authors investigated the effect of vortex shedding on heat transfer from a square cylinder [6,7] and showed that downstream temperature fields were affected by this phenomena considerably while small variation in overall Nusselt number was observed.

Recently Turki et. al [8] studied unsteady flow field and heat transfer characteristics in a channel with a built-in heated square cylinder. Two important issues in their study were to investigate the effects of blockage ratio and mixed convection in heat transfer in such geometry. They showed that pure forced convection could be occured for the value of Richardson numbers less than 0.05. Sharma and Eswaran [9] studied the effect of channel confinement on the two-dimensional flow and heat transfer across a square cylinder. To the best of authors’ knowledge, no published paper was found to consider the effect of aspect ratio on heat transfer from a rectangular cylinder. The focus of the present work is to reveal the effect
of aspect ratio on total heat transfer rate from a rectangular cylinder.

2 Governing equations

The flow is assumed to be unsteady, two-dimensional and laminar, for which the governing conservation equations of mass, momentum and energy can be written in the following forms:

\[
\frac{\partial U_i}{\partial x_i} = 0 \tag{1}
\]

\[
\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = \frac{\partial P}{\partial x_i} + \frac{1}{Re} \frac{\partial^2 U_i}{\partial x_i \partial x_j} \tag{2}
\]

\[
\frac{\partial T}{\partial t} + U_j \frac{\partial T}{\partial x_j} = \frac{1}{Re \cdot Pr} \frac{\partial^2 T}{\partial x_i \partial x_j} \tag{3}
\]

In the above equations, \( U, T, P, Re \) and \( Pr \) are dimensionless fluid velocity, temperature, pressure, Reynolds number and Prandtl number respectively. The maximum inlet velocity and block height, ‘d’ (see Fig. 1), were used in nondimensionalizing the above mentioned parameters. ‘T’ is the ratio of local temperature difference to block temperature with inlet fluid temperature. The geometry and the relevant dimensions are shown schematically in Fig. 1, in which a fixed two-dimensional rectangular cylinder place in the middle of the channel exposed to a fully developed laminar flow.

The boundary conditions used for the flow of fluid composed of a parabolic velocity profile at the inlet and convective boundary condition at the outlet. No-slip boundary condition was used for fluid in contact with solid walls of channel and rectangular surfaces. Both the inlet fluid and the rectangular block surfaces were at different constant temperature. The walls of the channel were assumed to be adiabatic.

3 Computational details

The computational domain is shown in Fig. 1. The upstream and downstream distances, ‘L_u’ and ‘L_d’, were selected as 10d and 15d respectively. The channel height was selected as 4d (or a=2d in Fig. 1) which corresponds to the blockage ratio of 25%.

The governing differential equations of (1) to (3) were discretised using finite volume method. A staggered nonuniform grid distribution was used for the present computations in which the velocity grid points are displaced compared to the pressure and temperature nodes. The convective and diffusive fluxes in the momentum equations were treated explicitly. A third-order Runge-Kutta algorithm was used for the time integration in conjunction with the classical correction method at each sub-step. The continuity equation (Eq. 1) and the pressure gradient term in the momentum equation (Eq. 2) were treated implicitly, while the convective and diffusive terms are treated explicitly. This method, which is called semi-implicit fractional step method, provides an approach that does not use pressure in the predictor step as in the pressure corrector method (such as the well-known SIMPLE family of algorithms). The linear system of pressure is solved by an efficient conjugate gradient method with preconditioning. The number of grid points used in the present computations was selected as 172×144 with the minimum grid spacing at the corners of the cylinder of 0.008d.

4 Results

The flow configuration is observed in Fig. 1. As the flow field is unsteady with vortex shedding for Re>50, a plot of vorticity field was shown in Fig. 2 to represent the relatively complex flow appeared in this geometry. In order to show the accuracy of the flow computations, variation of Strouhal number with Reynolds number is shown in Fig. 3 along with those obtained by Breuer et. al. [1]. As is observed in this figure, the differences are small which means that the present computations are of reasonable accuray. Heat transfer computations were presented in the form of local Nusselt number distribution along the sides of the cylinder and its mean value for the cylinder. Nusselt number could be obtained from the following equation:
where $q''$ is the local heat flux along the surface and $\Delta T$ is the temperature difference between the inlet fluid and cylinder surface. The local heat flux can be obtained from temperature gradient at the surface, which is written for the horizontal surfaces of the cylinder as:

$$q'' = k \left( \frac{T_w - T_{i,j,z}}{y_{j,z} - y_j} \right)$$

and for vertical surfaces of the cylinder as:

$$q'' = k \left( \frac{T_w - T_{x,j,z}}{x_{j,z} - x_j} \right)$$

As the mean flow is of more practical importance in actual flow situations, all of Nusselt number computations were done for mean flow.

Computations were done for aspect ratios of 0.25, 0.5, 1, 2 and 4 and Reynolds numbers of 100, 150 and 200. As was mentioned in the ‘Introduction’ part of the paper, the reason for selecting Reynolds number values less than 300 was to be sure of the accuracy of two-dimensional computations. As Turki et. al. [8] computed the total mean Nusselt number for the mean flow over a square cylinder, computations was done by the authors for the same geometry to check the accuracy of the results. Figure 4 shows variation of total mean Nusselt number with Reynolds number for a square cylinder. As is observed in his figure, good correspondence was achieved between the present computations and those of Turki et. al. [8].

Figures 5 and 6 show mean local Nusselt number distribution along the surfaces of the cylinder for aspect ratios of 0.5 and 2 respectively. As is observed in Fig. 5, the variation of Nusselt number along the top and bottom surfaces of the rectangular cylinder is the same. Increasing Reynolds number increases Nusselt number as is observed in many flow situations. The variation of Nusselt number along these surfaces is such that it starts from a maximum value at the beginning of the surface, followed by a steep decrease for $0<x/d<0.1$, after which a relatively constant value exists until near the end. At the end of these surfaces, an increase in the Nusselt number values is observed. The reason for such behavior is that the recirculating flow at the back of the cylinder extends to the end of the upper and lower surfaces. As this flow has a lower temperature compared to the flow near the end of the top or bottom surface coming from the upstream side, an increase in the heat transfer rate and subsequently in Nusselt number is observed.

Figures 5(c) and 5(d) show the variation of mean Nusselt number along the surfaces of the left and right (upstream and downstream) of the cylinder. As is observed in these figures, there is a minimum value for Nusselt number for the middle of the left surface compared to the its beginning and end. This behavior is contrary to what observed for the right surface. In the downstream of the right surface, there are two symmetric recirculation zones which make a symmetric distribution for Nusselt number, while a stagnation point exists in the left surface causes a minimum in Nusselt number distribution. Nearly the same behavior is observed in Fig. 6, which is for a cylinder with aspect ratio of 2. Figure 7 shows variation of total mean Nusselt number with aspect ratio for Reynolds numbers of 100, 150 and 200. It is observed that increasing aspect ratio decreases Nusselt number for all Reynolds numbers. Also there is an increase in Nusselt number with increasing Reynolds number.

5 Conclusions

Heat transfer from a rectangular cylinder in unsteady laminar flow showed that increasing aspect ratio in the range of 0.25 to 4 decreases mean total Nusselt number, while increasing Reynolds number in the range of 100 to 200 increases Nusselt number.
2.4 Diagrams and Figures

Fig. 1. The geometry of the flow configuration

Fig. 2. Vorticity contours at Re=200

Fig. 3. Variation of Strouhal number with Reynolds number for a square cylinder.

Fig. 4. Total mean Nusselt number distribution for a square cylinder.

Fig. 5. Mean Nusselt number distribution along (a) top, (b) bottom, (c) left and (d) right faces of cylinder for aspect ratio of 0.5
FORCED CONVECTION HEAT TRANSFER FROM A RECTANGULAR CYLINDER: EFFECT OF ASPECT RATIO

Fig. 6. Mean Nusselt number distribution along (a) top, (b) bottom, (c) left and (d) right faces of cylinder for aspect ratio of 2.

Fig. 7.

References


