Unsteady Flow and Heat Transfer in a Plate-Fin and Tube Heat Exchanger*

Hiroshi YAMASHITA**, Genichiro KUSHIDA**
and Ryotaro IZUMI**

Fluid flow and heat transfer in a plate-fin and tube heat exchanger were studied, using a fundamental model consisting of a pair of parallel plates and a square cylinder passing perpendicularly through the plates, which simulated plate-fins and a tube, respectively. The analytical results are reported about the unsteady flow and heat transfer at the Reynolds number $Re=1\,000$ and the Prandtl number $Pr=0.71$. The unsteady flow around a cylinder is shown in terms of time histories, velocity fluctuation intensity contours and the Strouhal numbers. Complicated aspects of the fluid flow are shown in terms of velocity vectors, streamlines, streak lines, pressure contours and wall shear stress contours. Furthermore, heat transfer characteristics under a constant wall heat flux condition are shown in terms of distributions of temperatures and local Nusselt numbers, and the mechanisms of fluid flow and heat transfer are systematically examined in detail.

Key Words: Convective Heat Transfer, Unsteady Flow, Numerical Analysis, FDM, Plate-Fin and Tube Heat Exchanger, Three-Dimensional Flow

1. Introduction

Plate-fin and tube heat exchangers, which consist of a set of parallel plates and an array of tubes passing perpendicularly through the plates, are typically compact, high-performance heat exchangers. They are widely used in engineering devices as air-conditioning machines. An extended surface on the air side of the tubes, that is, a plate-fin, is very useful when heat is transferred between the air and the liquid in the tubes, which have very different heat transfer coefficients. Such heat exchangers are the most important basic type of fin and tube heat exchangers, and have often been studied and developed. Owing to a complicated flow between fins, however, most of such studies are concerned with the performance test of actual heat exchangers. Some technical data is available, but few systematical examinations of the mechanisms of fluid flow and heat transfer between the plate-fins have been performed. In the present study,

the fluid flow and heat transfer in a plate-fin and tube heat exchanger are studied, using a fundamental model consisting of a pair of parallel plates and a rectangular cylinder passing perpendicularly through the plates, which simulate plate-fins and a tube, respectively.

In the previous paper, the authors studied the steady flow⁽¹⁾ and heat transfer⁽²⁾ at the Reynolds number $Re=200$ using a model in which the base of the cylinder is square and the channel width is equal to the side length of the cylinder, and pointed out in these papers that the fluid flow around a cylinder at $Re=200$ is steady and symmetric. Also, the authors experimentally studied the fluid flow of the similar model for the various Reynolds numbers⁽³⁾. It was proved by these experiments that a fluid flow around a cylinder became unsteady at a relatively large Reynolds number. The fluid flow in the neighborhood of a cylinder shows periodic fluctuations. The heat transfer characteristics also seem to be strongly affected by the unsteady flow. Therefore, the study on such a unsteady flow is necessary to examine the fluid flow and heat transfer. In the present paper, the unsteady flow and heat transfer at a relatively large Reynolds num-

* Received 17th July, 1987. Paper No. 87-4007A

** Faculty of Engineering, Nagoya University, Furocho, Chikusa-ku, Nagoya, 464, Japan

ber is theoretically analyzed using the similar model such as that in the previous paper. The analysis is made numerically, and the three-dimensional continuity equation, the Navier-Stokes equation and the energy equation are solved by a finite difference method for an incompressible viscous flow at the Reynolds number $Re=1000$ and the Prandtl number $Pr=0.71$. As for the finite difference method used here, the SIMPLE algorithm by Patankar⁽⁴⁾ is modified, and a central-difference scheme is used for the convection terms. The unsteady flow around a cylinder is shown in terms of time histories, velocity fluctuation intensity contours and the Strouhal numbers. Complicated aspects of the fluid flow are shown in terms of velocity vectors, streamlines, streak lines, pressure contours and wall shear stress contours. Also heat transfer characteristics under a constant wall heat flux condition are shown in terms of distributions of temperatures and local Nusselt numbers, and the mechanisms of fluid flow and heat transfer are systematically examined in detail.

Nomenclature

- a : thermal diffusivity
 - C_p : pressure coefficient $= (p - p_s) / (\rho u_m^2 / 2)$
 - d : side length of rectangular cylinder
 - d_e : characteristic diameter $= 2H$
 - f : Fanning's friction coefficient
 - f' : frequency
 - H : channel width
 - h : heat transfer coefficient
 - I, J, K : grid numbers
 - Nu : Nusselt number
 - O : origin
 - p : pressure
 - Pr : Prandtl number
 - q : heat flux
 - Re : Reynolds number $= u_m 2H / \nu$
 - St : Strouhal number $= f' d / u_m$
 - t : time
 - u : velocity
 - x_1, x_2, x_3 : coordinates
 - θ : temperature
 - λ : thermal conductivity
 - ν : kinematic viscosity
 - ρ : density
 - τ : shear stress
 - φ : dependent variable
- Subscripts**
- b : bulk
 - c : center
 - in : inlet
 - m : mean
 - max : maximum

- n : number of iterations
- ref : reference
- s : starting point
- w : wall
- $*$: modified
- $'$: dimensionless or fluctuation
- $-$: integral mean

2. Calculation Procedure

The analytical model is shown in Fig. 1, and the coordinates and dimensions are shown in Fig. 2. We assume a square cylinder with side length d between parallel plates, and take the three-dimensional Cartesian coordinates (x_1, x_2, x_3) where the origin O is located at the center of the cylinder, x_1 is measured in the flow direction, x_2 in the direction perpendicular to the plates and x_3 in the spanwise direction. In this analysis, the channel width H is equal to d . The calculation domain is taken as wide as possible in the flow and spanwise directions, that is, $-6.4 \leq x_1/d \leq 16.2$ and $-6.4 \leq x_3/d \leq 6.4$. The characteristic diameter of the channel d_e and the Reynolds number Re are defined as follows.

$$d_e = 2H \quad (1)$$

$$Re = u_m d_e / \nu \quad (2)$$

Considering a steady incompressible viscous laminar flow in the channel and constant physical properties of the fluid, the basic equations are the continuity, the Navier-Stokes and the energy equations :

Fig. 1 Analytical model

Fig. 2 Coordinates and dimensions

$$\nabla \cdot \mathbf{u} = 0, \quad (3)$$

$$D\mathbf{u}/Dt = -\frac{1}{\rho}\nabla p + \nu\Delta\mathbf{u}, \quad (4)$$

$$D\theta/Dt = \alpha\Delta\theta. \quad (5)$$

The boundary conditions for velocity are as follows. At the walls and faces of the cylinder, velocities are zero. At the upstream boundary in the channel, the flow is assumed to be hydrodynamically fully developed, and the parabolic velocity profile is applied. At the downstream boundary, it is assumed that none of the velocity components change in the flow direction. At the boundary in the spanwise direction, the parabolic velocity profile is applied. Pressure corrections are zero at all boundaries. The boundary conditions for temperature are as follows. It is assumed that all the walls and faces of the cylinder are heated, such that the wall heat flux is constant. At the upstream boundary, we use a developed temperature profile which is calculated in the same way but for a flow without a cylinder between parallel plates. At the downstream boundary, the temperature gradient is kept constant in the flow direction. At the boundary in the spanwise direction, the temperature gradient in that direction is zero.

The calculation procedure is similar to that used in the previous papers^{(1),(2)}, and we carry out numerical calculations using the finite difference method. In this method, the SIMPLE algorithm by Patankar⁽⁴⁾ is modified. The differential grids of the channel are shown in Fig. 3. The grids number from $I=1$ to 46 in the x_1 -direction, $J=1$ to 11 in the x_2 -direction, and $K=1$ to 39 in the x_3 -direction. The grid sizes are not equal, and they are small in the neighborhood of the cylinder and the walls, as shown in the figure. The coordinates of the grids are shown in Table 1.

In this analysis, numerical calculations are carried out for the unsteady flow and heat transfer at Reynolds number $Re=1000$ and the Prandtl number $Pr=0.71$. A central-difference scheme⁽⁵⁾ is applied for

the convection terms to avoid the influence of false diffusion. A fully implicit method is used to advance the solution in time. We take 0.4 as the value of the relaxation parameter. We set $\max(|\varphi_{i,j,k}^n - \varphi_{i,j,k}^{n-1}|/\varphi_{\text{ref}}) < 10^{-5}$ for the convergence criterion, where φ is a variable, n is the number of iterations, and φ_{ref} is u_m for the velocity, $\rho u_m^2/2$ for the pressure correction and $q_w d_e/\lambda$ for the temperature. The number of iterations was about 30 per one time step, and the CPU time for one time step was 120 seconds.

3. Calculation Results and Examinations

3.1 Time histories

The present analysis examines the periodic fluctuations of a fluid flow in a quasi-steady state. We used the calculated result obtained after a sufficiently long period of time starting from a certain state. The present results correspond to 800 steps ($\Delta t'=0.05$), where dimensionless time is defined as $t'=tu_m/H$. The time histories of the velocities u_1/u_m , u_2/u_m and u_3/u_m , and the dimensionless temperature θ' are shown in Fig. 4.

These values correspond to those at time $t'=0$ to 40 (800 time steps) that the initial state does not affect. The symbols A, B and C in the figure correspond to the typical locations around a cylinder as shown in Fig. 6(a). None of these oscillations are perfectly regular, but they show periodic fluctuations. The frequencies of these fluctuations are analyzed by a maximum entropy method⁽⁶⁾. Using these frequencies f' , the Strouhal numbers $St(\equiv f'd/u_m)$ for u_3/u_m are shown in Table 2. There is a fluctuation of $St=0.20$ at the location C downstream of the cylinder, which seems to be the Karman vortex streets. At the location A immediately upstream of the cylinder, there is a weak fluctuation. At the shear layer in the neighborhood of the side face of the cylinder, there is a fluctuation of $St=1.23$ which is larger by one order of magnitude than that of the Karman vortex streets.

Fig. 3 Differential grids

Table 1 Coordinates of grids

J	6	5	4	3	2	1
	7	8	9	10	11	
$ x_2/d $	0	0.14	0.26	0.36	0.44	0.50

I, K	20	19	18	17	16	15	14	13	12	11
	21	22	23	24	25	26	27	28	29	
$ x_1/d $	0	0.14	0.26	0.36	0.44	0.50	0.56	0.64	0.74	0.86
$ x_3/d $										

I, K	10	9	8	7	6	5	4	3	2	1
	30	31	32	33	34	35	36	37	38	39
$ x_1/d $	1.0	1.2	1.5	1.9	2.4	3.0	3.7	4.5	5.4	6.4
$ x_3/d $										

I, K	40	41	42	43	44	45	46
x_1/d	7.5	8.7	10.0	11.4	12.9	14.5	16.2

This fluctuation is generated by Kelvin-Helmholtz shear-layer instability. The other velocity components and the temperature also have the three kinds of fluctuations at the three locations, respectively.

The streak line at $t'=30.8$ is shown in Fig. 5, where each point indicates the location at one time step ($\Delta t'=0.05$). The streak line shows the complicated aspects in the neighborhood of the side face and behind the cylinder, where vortices are formed.

The aspects of the flow field which is averaged over the whole range of time histories ($t'=0$ to 40) shown in Fig. 4 are shown below.

3.2 Velocity vectors

The resultant vectors of the velocity components at various channel sections are shown in Figs. 6 to 8. Figures 6(a) and (b) correspond to the sections of $J=6$ and 2, respectively. At the central section of channel width ($J=6$), the upstream flow impinges on the cylinder, and is divided at the impinging point. In

the neighborhood of the front corner of the cylinder, the velocity is very large and the fluid flow is accelerated. On the other hand, there is a dead water zone behind the cylinder. At the section ($J=2$) near the wall, a backflow zone exists in front of the cylinder, and the velocity near the front corner of the cylinder is very large. Figures 7(a) and (b) correspond to the sections of $K=20$ and 14. At the central section ($K=20$) in the spanwise direction, the flow forms a vortex in front of the cylinder as a result of impinging against the cylinder and the walls. There is a backflow dead water zone behind the cylinder. At the section ($K=$

Fig. 5 Streak line ($t'=30.8$)

Table 2 Strouhal numbers

Location	$(x_1/d, x_2/d, x_3/d)$	St
A	$(-0.56, 0, 0)$	0.38
B	$(-0.26, 0, -0.64)$	1.23
C	$(3.00, 0, 0)$	0.20

Fig. 4 Time histories of velocity and temperature

(a) $J=6$

(b) $J=2$

Fig. 6 Velocity vectors ($J=6$ and 2)

(a) $K=20$

(b) $K=14$

Fig. 7 Velocity vectors ($K=20$ and 14)

14) near the side face of the cylinder, there are two vortices which have contrary rotations in the neighborhood of the cylinder. Figures 8(a), (b) and (c) correspond to the sections of $I=14$, 20 and 26 perpendicular to the flow direction. At the section ($I=14$) immediately upstream of the cylinder, the velocity vectors extend radially about the impinging point of the front face of the cylinder because the upstream flow impinges on the cylinder. The flow which turns to the walls reverses, and the flow which turns to the side goes downstream. At the section ($I=20$), four pairs of vortices which have contrary rotations are seen in the neighborhood of the cylinder. The vortices which are far from the cylinder correspond to the horseshoe vortices, as will be shown later. At the section ($I=26$) immediately downstream of the cylinder, the velocity is very small although there are some vortices present.

3.3 Streamlines

The complicated three-dimensional flow in the channel is shown in terms of streamlines. The streamlines shown here express the flow field which is averaged. Streamlines are distinguished by the values of J or K at the section ($I=6$). Figures 9 to 11 show the three-dimensional oblique projections (a) and the orthogonal projections (b) of streamlines. As shown in Fig. 9, the streamlines at the central section ($J=6$) go along this section because the flow field is averaged. In the neighborhood of the front corner of the

Fig. 8 Velocity vectors ($I=14, 20$ and 26)

Fig. 9 Streamlines ($J=6$)

cylinder, the stream lines are curved, showing the shear layer. Figure 10 corresponds to those at the section ($K=19.95$) near the central section in the spanwise direction. The streamlines form a vortex and curl towards a vortex center in front of the cylinder. Then they turn aside in the spanwise direction near the vortex center, and go downstream curling spirally. This is a horseshoe vortex. Figure 11 corresponds to those at the section ($J=1.5$) near the wall. These streamlines go downstream after turning to the side direction fairly upstream of the cylinder. They show the aspects of a horseshoe flow in front of the cylinder.

3.4 Velocity fluctuation intensity contours

The intensity of velocity fluctuations is defined as $\sqrt{u'^2}/u_m$ using a root mean square velocity $\sqrt{u'^2}$. The velocity fluctuation intensity contours for all the velocity components at the central section ($J=6$) are shown in Figs. 12(a), (b) and (c), respectively. $\sqrt{u_1'^2}/u_m$ is very large at the shear layers in the neighborhood of the front corners of the cylinder, and is maximum at $x_1/d=2.5$ and $x_3/d=\pm 0.5$. $\sqrt{u_2'^2}/u_m$ is

(a) Oblique projection

(b) Orthogonal projection

Fig. 10 Streamlines ($K=19.95$)

(a) Oblique projection

(b) Orthogonal projection

Fig. 11 Streamlines ($J=1.5$)

maximum in the neighborhood of the front corners of the cylinder. The location of the maximum downstream of the cylinder is at $x_1/d=1.5$, which is more downstream than $\sqrt{u_1'^2}/u_m$. As for $\sqrt{u_3'^2}/u_m$, its maximum downstream of the cylinder, at $x_3/d=0$ and $x_1/d=2$, which is different from those of the other components, is larger than that in the neighborhood of the front corners of the cylinder. $\sqrt{u_1'^2}/u_m$ is largest of all the components, and $\sqrt{u_3'^2}/u_m$ is also relatively large, but $\sqrt{u_2'^2}/u_m$ is quite small.

Figures 13 (a) and (b) correspond to the contours of the velocity fluctuation intensity $\sqrt{u_1'^2 + u_2'^2 + u_3'^2}/u_m$ at the central section ($J=6$) and the section ($J=2$) near the wall, respectively. The location of a maximum at the central section is similar to that of $\sqrt{u_1'^2}/u_m$ shown in Fig. 12(a). The velocity fluctuation intensity near the wall is much smaller than that at the central section.

3.5 Pressure contours

(a) $\sqrt{u_1'^2}/u_m$

(b) $\sqrt{u_2'^2}/u_m$

(c) $\sqrt{u_3'^2}/u_m$

Fig. 12 Velocity fluctuation intensity contours

The pressure contours at various sections in the channel are shown in Fig. 14. Figure 14(a) corresponds to the central section ($J=6$), Fig. 14(b) to the section ($K=15$), and Figs. 14(c) and (d) to the

(a) $J=6$

(b) $J=2$

Fig. 13 Velocity fluctuation intensity contours ($\sqrt{u_1'^2 + u_2'^2 + u_3'^2}/u_m$)

(a) $J=1$

(b) $K=15$

(c) $I=15$

(d) $I=25$

Fig. 14 Pressure contours

sections ($I=15$ and $I=25$) perpendicular to the flow direction, respectively. The numerals written alongside the curved lines in the figures indicate the pressure coefficients $C_p = (p - p_s) / (\rho u_m^2 / 2)$, where p_s is the standard pressure at the center of the upstream boundary. As shown in Fig. 14(a), the pressure contours are complicated in the neighborhood of the cylinder. The pressure rises abruptly near the front face of the cylinder and has a maximum at the impinging point. On the other hand, it drops abruptly in the neighborhood of the front corner of the cylinder. Figures 14(b), (c) and (d) correspond to the sections which include the side face, the front face and the back face of the cylinder, respectively. At the side face, the pressure drops abruptly in the neighborhood of the front corner. At the front face of the cylinder, the pressure is quite large around the maximum of pressure at the impinging point. On the other hand, the pressure at the back face of the cylinder is uniformly low.

3.6 Wall shear stress contours

Fanning's friction coefficient is defined as $f = |\tau| / (\rho u_m^2 / 2)$. The wall shear stress contours expressed by the product fRe of Fanning's friction coefficient f and the Reynolds number Re at the wall are shown in Fig. 15. As shown in the figure, the wall shear stress contours are very complicated in the neighborhood of the cylinder. As the upstream flow impinges on the front face of the cylinder, the flow is accelerated, turning to the side direction. Thus, the wall shear stress has a very large maximum value in the neighborhood of the front corners of the cylinder. It has another maximum upstream of the cylinder where the reverse flow again impinges on the wall and is accelerated. On the other hand, it has a minimum at the location where the flow encounters that from the upstream.

3.7 Instantaneous streamlines and wall shear stress contours

The instantaneous streamlines at $t' = 7.55$ are shown in Fig. 16. They indicate complicated aspects downstream of the cylinder.

Fig. 15 Wall shear stress contours

The instantaneous wall shear stress contours at $t' = 7.55$ are shown in Fig. 17. The asymmetrical flow pattern is shown downstream of the cylinder, which seems to be the Karman vortex streets. As shown in Figs. 16 and 17, at relatively large Reynolds numbers, much disturbance is produced by the unsteady flow downstream of the cylinder.

3.8 Temperature distributions

The temperature contours at various sections of the channel are shown in Fig. 18. Figures 18(a) to (d) correspond to the sections of $J=1$, $K=15$, $I=15$ and $I=25$, respectively. The numerals written alongside the curved lines in the figures indicate the dimensionless temperatures $\theta' = (\theta - \theta_s) / (q_w d_e / \lambda)$, where θ_s is the standard temperature at the center of the section of the upstream boundary. In the case without a cylinder, the temperature in a channel should increase uniformly in the flow direction because of a constant wall heat flux condition. In the case with a cylinder, the aspects of temperature contours become very complicated as is evident in Fig. 18. At the wall, the flow from upstream separates after impinging on the front face of the cylinder, and so the temperature is very high in the backflow zone downstream of the cylinder. In a horseshoe region around the cylinder,

(a) Oblique projection

(b) Orthogonal projection

Fig. 16 Instantaneous streamlines ($t' = 7.55$)

Fig. 17 Instantaneous wall shear stress contours ($t' = 7.55$)

Fig. 18 Temperature contours

however, the temperature is low. The temperature downstream of the dead water zone behind the cylinder is also relatively low. It seems that the temperature field is disturbed by the unsteady flow downstream of the cylinder. Figures 18(b) to (d) show the temperature contours at the sections which involve the cylinder faces. At the side face, the temperature is low upstream of the cylinder face and abruptly increases downstream of it. At the front face of the cylinder, the temperature is uniformly low. At the back face, the temperature is very high as a whole. As shown in the figures, the temperature is high near the walls and low near the center as far as the channel width direction is concerned, the exception being in the neighborhood of the cylinder, where it is similar to the case without a cylinder.

3.9 Local Nusselt number distributions

The local Nusselt number Nu is generally defined as follows.

$$Nu = \frac{hd_e}{\lambda} = \frac{(q_w/\lambda)d_e}{\theta_w - \theta_b} \quad (6)$$

In the above equation, θ_b denotes the bulk temperature, which corresponds to the mixing mean temperature at the section perpendicular to the flow direction for the flow between parallel plates. As pointed out in the previous paper⁽⁷⁾, the wall temperature sometimes becomes lower than this bulk temperature in the three-dimensional flow field. Therefore, it is necessary to modify the definition of the local Nusselt number.

Fig. 19 Modified local Nusselt number contours

Consequently we take the following modified local Nusselt number as proposed in the previous paper⁽⁷⁾.

$$Nu^* = Nu_c \times \left(\frac{Nu_m}{Nu_{cm}} \right)_{in} = \frac{(q_w/\lambda)d_e}{\theta_w - \theta_c} \left(\frac{\theta_w - \theta_c}{\theta_w - \theta_b} \right)_{in} \quad (7)$$

In Eq. (7), θ_c is the temperature which agrees with the value at the center of the upstream boundary and increases with the same gradient as θ_b in the flow direction. Nu_c is the Nusselt number which is shown by using θ_c instead of θ_b in Eq. (6). Nu_m and Nu_{cm} are the mean Nusselt numbers, which correspond to Nu and Nu_c , respectively. The subscript "in" indicates that the value is one at the upstream boundary.

The contours of the modified local Nusselt number Nu^* as defined in Eq. (7) at the wall are shown in Fig. 19. Nu^* is high in the horseshoe region around the cylinder. It is especially high in the neighborhood of the front corner of the cylinder. Such aspects are similar to the velocity fluctuation contours and the wall shear stress contours. Nu^* is low in the dead water zone immediately downstream of the cylinder, but it is relatively high downstream of it. This is due to the fact the velocity fluctuation intensity is large in this region, as shown in Figs. 12 and 13. These aspects are generated by the unsteady flow and are characteristic at large Reynolds numbers, and were thus not observed in steady flow investigations of the previous papers^{(1),(2)}. Consequently the heat transfer is augmented very much by the unsteady flow when the Reynolds number is relatively large.

4. Conclusions

The fluid flow and heat transfer around a square cylinder situated between parallel plates were considered as a model of a plate-fin and tube heat exchanger. The unsteady flow and heat transfer under constant wall heat flux conditions at Reynolds number $Re = 1000$ and Prandtl number $Pr = 0.71$ were calculated numerically by a finite difference method for the case where the channel width is equal to the side length of a square cylinder. The results are summarized as

follows :

(1) Numerical calculations were made using a finite difference method, and the unsteady flow and heat transfer around the cylinder were elucidated.

(2) The aspects of unsteady flow around the cylinder were shown in terms of time histories, velocity fluctuation intensity contours and Strouhal numbers, and the unsteady flow mechanism was elucidated.

(3) Velocity vectors, streamlines, pressure contours, wall shear stress contours, temperature contours and modified local Nusselt number contours were shown for the averaged flow and temperature fields around the cylinder.

(4) The modified local Nusselt number contours were similar to the velocity fluctuation intensity contours and the wall shear stress contours. When the Reynolds number was relatively large, the heat transfer was augmented by the unsteady flow around the cylinder.

Acknowledgements

The authors were permitted the use of the computer FACOM M-382 at the Nagoya University Computer Center to calculate the numerical values in this research, and would therefore like to express their sincere gratitude to the staff of the Computer Center.

They are also grateful to Mr. M. Hayakawa for his help with the figures.

References

- (1) Yamashita, H., Kushida, G. and Izumi, R., Fluid Flow and Heat Transfer in a Plate-Fin and Tube Heat Exchanger, Bull. JSME, Vol. 29, No. 254 (1986), p. 2562.
- (2) Kushida, G., Yamashita, H. and Izumi, R., Fluid Flow and Heat Transfer in a Plate-Fin and Tube Heat Exchanger, Bull. JSME, Vol. 29, No. 258 (1986), p. 4185.
- (3) Yamashita, H., Kushida, G., Izumi, R., Kato, K. and Kometani, H., Fluid Flow and Heat Transfer in a Plate-Fin and Tube Heat Exchanger, 23rd Nat. Heat Transf. Symp. Jpn., (in Japanese), (1986), p. 337.
- (4) Launder, B.E., Studies in Convection, Vol. 1 (1975), p. 1, Academic Press.
- (5) Yamashita, H., Izumi, R. and Kushida, G., Fluid Flow and Heat Transfer in a Two-Dimensional Miter-Bend, JSME Int. J., Vol. 30, No. 259 (1987), p. 93.
- (6) Hino, M., Spectral Analysis (in Japanese), (1977), p. 210, Asakura Shoten.
- (7) Kushida, G., Yamashita, H. and Izumi, R., Study on Three-Dimensional Flow and Heat Transfer in Miter-Bend, Bull. JSME, Vol. 28, No. 243 (1985), p. 1992.