Heat Transfer Characteristics in a Channel Flow with a Rectangular Cylinder*

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An experimental study of heat transfer has been conducted in a channel flow with a rectangular cylinder having various width-to-height ratios, b/h , of 0.5, 1, 2 and 3. Time-averaged heat transfer coefficients and the fluctuations of wall heat flux have been measured. Heat transfer is augmented extensively at the downstream region of the cylinder. In order to examine the relationship between the heat transfer enhancement and the shedding vortices from the cylinder, flow visualization by a smoke-wire method using a high-speed video recording system has been carried out simultaneously with wall heat flux measurement. It is clarified that the heat transfer is augmented by the wall-ward flow induced by the clockwise and counterclockwise vortices shed alternately from the cylinder.

Key Words: Heat Transfer Enhancement, Vortex, Rectangular Cylinder, Flow Visualization, Wall Heat Flux Fluctuation, Channel Flow

1. Introduction

In a large number of engineering equipment, flows are turbulent and often separated from the solid surface. The separating and reattaching flows cause heat transfer to be augmented. Extensive investigations have been made so far in connection with the heat transfer enhancement.

K. Suzuki and H. Suzuki⁽¹⁾ studied both numerically and experimentally the mechanism of heat transfer enhancement in a laminar channel flow obstructed by a square cylinder. A high vorticity region emerges near the heated channel wall in phase with the shedding vortices from the cylinder. They observed that the induced vortex near the heat transfer surface entrains the cooler fluid from the downstream side and that this fluid motion contributes to the enhancement of the wall heat transfer.

In a turbulent channel flow obstructed by a square cylinder, Yao et al.⁽²⁾ measured the velocity and temperature fluctuations simultaneously with a combination of hot and cold wires. From this analysis, heat transfer enhancement was found to be related to the intensification of the cold and hot interactions toward and away from the wall, respectively. Sano⁽³⁾ also studied the heat transfer in a turbulent channel flow with a flat body. Heat transfer coefficient and velocity characteristics near the wall were measured. He found that the heat transfer coefficient is related to the increase of the turbulent intensity in the near wall region. Chung et al.⁽⁴⁾ investigated the heat transfer enhancement of a thermally developing region in which a square cylinder was inserted with various orientation angles. They showed that the heat transfer coefficient is maximum near the reattachment point of the separated shear layer to the channel wall.

In spite of these efforts, however, the mechanism of heat transfer enhancement in a turbulent channel flow with a bluff body is not clearly understood, because the unsteadiness and the complexity of the phenomena make the problem difficult.

On the other hand, the wake of a rectangular cylinder having various width-to-height ratios has been closely investigated in a uniform flow⁽⁵⁾⁻⁽⁷⁾. As the width-to-height ratio increases, the separated

* Received 27th October, 1998. Japanese original: Trans. Jpn. Soc. Mech. Eng., Vol. 64, No. 622, B (1998), p. 1817-1824 (Received 20th October 1997)

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shear layers at the leading edges reattach to the side walls of the cylinder. The reattachment of the shear layers causes a change of flow pattern in the near wake.

We have investigated the flow and thermal fields in a turbulent channel flow with a rectangular cylinder having various width-to-height ratios^{(8),(9)}. The heat transfer measurements indicated that the heat transfer coefficient increases extensively with an insertion of the rectangular cylinder at the downstream region of the cylinder. Spectral analyses⁽⁸⁾ of the velocity and wall heat flux fluctuations reveal that they have the same predominant frequency. This fact suggests a close relationship between the shedding vortices and the heat transfer enhancement at the channel wall. In addition, a change of the flow pattern depending on the width-to-height ratio was observed by flow visualization and confirmed by detailed flow measurements⁽⁹⁾ of the statistical and phase-averaged characteristics with laser Doppler velocimetry. When the width-to-height ratio of the rectangular cylinder $b/h \geq 2$, the separated shear layers reattach to the side walls of the cylinder, while they are entrained immediately behind the cylinder for $b/h \leq 1$. Particularly, when b/h is equal to 2, an intermittent reattachment of the flow causes two flow patterns which differ in wake width.

The purpose of this work is to clarify the relationship between the shedding vortices from the cylinder and the heat transfer enhancement. Our attention is focused on the unsteady characteristics of heat transfer, and the flow visualization with a high-speed video system has been synchronized with the measurement of heat flux fluctuation at the wall.

Nomenclature

- b : width of a rectangular cylinder
 C : constant in the empirical Eq. (3)
 d : hydraulic diameter of the channel
 h : height of a rectangular cylinder
 H : channel height
 Nu : local Nusselt number ($= aH/\lambda$)
 Nu_{av} : averaged Nusselt number based on the hydraulic diameter
 Nu_{∞} : averaged Nusselt number at the fully developed region
 Nu_{max} : second maximum value of local Nusselt number
 Pr : Prandtl number
 q_w : wall heat flux
 Re : Reynolds number ($= U_m H/\nu$)
 Re_d : Reynolds number based on the hydraulic diameter
 T_w, T_f : temperatures of wall and fluid

- U_m : sectional mean velocity
 x : streamwise coordinate
 y : coordinate normal to the channel wall
 α : heat transfer coefficient ($= q_w/(T_w - T_f)$)
 λ : thermal conductivity of fluid
 ν : kinematic viscosity

2. Experimental Apparatus and Procedures

Figure 1 shows the experimental apparatus schematically. Uniform flow of air, supplied as the working fluid by a blower, entered the test section through a diffuser, a straightening section and a contraction. Flow rate was adjusted with valves and was measured with an orifice flowmeter.

A schematic diagram of the test section is shown in Fig. 2. The channel height, H , was 50 mm and the spanwise dimension was 350 mm. Streamwise and normal coordinates, x and y , have their origin on the symmetry axis at the inlet of the test section. The height of the rectangular cylinder, h , was 10 mm and its width, b , was varied from 5 to 30 mm so that the width-to-height ratios, b/h , of the rectangular cylinder were 0.5, 1.2 and 3. The cylinder was placed symmetrically in the channel and the streamwise position of its front surface was $0.6H$ from the inlet of the test section.

Fig. 1 Experimental apparatus

Fig. 2 Schematic diagram of test section

Heat transfer coefficients were measured with a test plate made of luan. A stainless steel foil of 20 μm thickness was glued to the entire surface of the plate and heated electrically so that the wall heat flux would be constant. In order to minimize heat loss from the back surface, glass wool was used for thermal insulation. Fifty thermocouples were equipped to measure the temperature of the stainless steel foil and the back of the test plate. Conductive heat loss from the back surface was taken into account for obtaining the wall heat flux and was estimated by assuming one-dimensional heat conduction.

Local heat transfer coefficient is expressed as

$$\alpha = q_w / (T_w - T_f) \quad (1)$$

where q_w is the wall heat flux compensated for heat loss. Local Nusselt number is defined as follows:

$$Nu = \alpha H / \lambda \quad (2)$$

where λ is the thermal conductivity of air at the temperature $(T_w + T_f)/2$.

Fluctuations of the wall heat flux were measured with thin-film heat flux sensors^{(8),(10)}. Figure 3 shows the thin-film heat flux sensor and the test plate for the measurement of heat flux fluctuations. The heat flux sensor was made of nickel film of 3 μm thickness and processed by photo-etching into the shape shown in Fig. 3(b). Light pulse with a frequency range of 5 Hz to 250 Hz was radiated to the sensor in order to calibrate the dynamic response, and it was confirmed⁽⁸⁾ that the sensor responded sufficiently to the fluctuations brought about by the shedding vortices.

The sensor was operated in the constant temperature mode in the same way as a hot wire anemometer. The heat transfer surface therefore consisted of a copper plate and copper blocks equipped with thermocouples. Each component was heated separately by electric heaters. The detector of the sensor was attached to the heat transfer surface of a copper block with a double-sided tape which also served as an electric insulator. The reference of the sensor was attached to the side wall of the copper block and was put between them to detect the temperature of the test plate. The copper blocks were capable of being arranged in an arbitrary order so that the measurement position could be altered.

The smoke wire method was used for flow visualization. The nichrome wire used was 0.2 mm in diameter and it was stretched vertically just ahead of the cylinder. The wire was heated electrically by a smoke generator to produce paraffin oil vapor. A high-speed video recording system (Kodak 4540; 2250 frames per second) and a halogen lamp were employed to record the flow pattern. The high-speed video recording system and an A/D converter for the heat flux mea-

Fig. 3 Schematic diagram of test plate for heat flux measurement: (a) test plate and (b) heat flux sensor

surement were synchronized with a triggering signal from the smoke generator.

3. Results and Discussion

3.1 Time-averaged heat transfer characteristics

The streamwise distribution of the local Nusselt number for $Re=15\,000$ is shown in Fig. 4. In this figure, the results for the case without a cylinder are also plotted. Heat transfer is seen to be enhanced extensively by insertion of the rectangular cylinder, and to depend on the width-to-height ratio b/h of the cylinder.

The local Nusselt number decreases rapidly near

Fig. 4 Local Nusselt number distribution for $Re=15\,000$ Fig. 5 Local Nusselt number distribution for $b/h=1$

the inlet of the test section, since the flow and thermal fields are in the inlet region. The local Nusselt number takes a maximum both at the inserted position of the cylinder around $x/H=0.6$ and at a farther downstream position. The former results from flow acceleration due to the blockage effect of the cylinder, while the latter is caused by the periodically shedding vortices from the cylinder⁽⁸⁾. We refer to the latter as the second maximum of local Nusselt number, which is discussed below.

The local Nusselt number distributions for Reynolds numbers ranging from 7 500 to 22 500 are shown for the case of $b/h=1$ in Fig. 5. The distributions are similar to each other regardless of the Reynolds number. The same results are obtained for the other cases of b/h . This fact is in accord with the results of Yao et al.⁽¹¹⁾ and Sano⁽³⁾, who reported that the fundamental structures of the flow and thermal fields are identical and independent of the Reynolds number in a turbulent channel flow.

The second maximum of the local Nusselt number is plotted against the Reynolds number in Fig. 6. The maximum value of the local Nusselt number

Fig. 6 Maximum Nusselt number versus Reynolds number

Fig. 7 Streamwise position of maximum Nusselt number versus Reynolds number

increases as the Reynolds number, Re , increases, and its dependence on Re is almost the same for all the cylinders.

Figure 7 shows the relationship between the streamwise position of the second maximum and the Reynolds number. In the case of $b/h=0.5$, the position is farthest and independent of the Reynolds number. The second maximum of the local Nusselt number appears at a farther upstream position in the cases of $b/h=1, 2$ and 3 with an increase in the Reynolds number.

Senda et al.⁽¹²⁾ also showed in their study of an axisymmetric confined jet with a cylindrical ring that the position of the maximum heat transfer coefficient moves upstream with the Reynolds number. They pointed out that the recirculation region behind the ring spreads wider in the radial direction when the Reynolds number increases, and that the spatial scale of the shedding vortices (vortex rings) increases. The present result may have been due to the same reason.

The averaged Nusselt number, Nu_{av} , over the heat transfer surface ($x/H=0-10$) is shown in Fig. 8. The abscissa and the ordinate of the figure are the Reynolds number and the averaged Nusselt number

Fig. 8 Averaged Nusselt number versus Reynolds number

based on the hydraulic diameter, respectively. In this figure, the following empirical equation⁽¹³⁾ at the inlet region of an internal flow is given as reference.

$$Nu_{av} = Nu_{\infty} [1 + C/(x/d)] \quad (3)$$

$$Nu_{\infty} = 0.019 Re_d^{0.8} Pr^{0.5} \quad (4)$$

where Nu_{∞} is the averaged Nusselt number for a fully developed flow and C is a constant depending on the shape of the duct entrance. We chose the value of C to be unity in this study, because C is between 0.7 and 1.2 when the duct entrance has the shape of a bell-mouth.

The averaged Nusselt number is the largest for $b/h=0.5$ and smallest for $b/h=2$. The Reynolds number (Re_d) dependence of the averaged Nusselt number is approximately equal to that of Eq. (3) for all the cases, and agrees with the result of Sano⁽³⁾.

Considering the circumstances mentioned above, we conclude that the fundamental characteristics of heat transfer are not affected by the Reynolds number in the range of 7 500 to 22 500. The following experiments were, therefore, conducted at $Re=15\,000$.

3.2 Simultaneous measurement of flow visualization and wall heat flux fluctuation

The images of the visualized flow and the output signals of the heat flux sensor are shown in Fig. 9 for $b/h=0.5$. Figure 9(a) shows the temporal change of the output signals from the three heat flux sensors. The positions of the heat flux sensors are at $x/H=2.16, 2.64$ and 3.00 , and are indicated by black rectangles below the images in Fig. 9(b). The ordinate of Fig. 9(a) is the output voltage of the heat flux sensor and the wall heat flux is proportional to the square of this output voltage. Numbered arrows in Fig. 9(a) indicate the time when the corresponding images shown in

Fig. 9 (a) Heat flux fluctuation and (b) flow visualization for $b/h=0.5$; ■, position of heat flux sensor

Fig. 9(b) were recorded. From Fig. 9(a), it is seen that the wall heat flux fluctuates periodically. The frequency of its fluctuation is confirmed to be identical to that of vortex shedding⁽⁸⁾.

The vortex shed from the upper side of the cylinder is located at the left end (The fluid flows from left to right.) of image ① in Fig. 9(b). This vortex is referred to hereafter as the negative vortex, because it rotates clockwise and has negative vorticity. On the other hand, we refer to the vortex shed from the lower side as the positive vortex. There exists a positive vortex right above the heat flux sensor at $x/H=2.16$.

These negative and positive vortices move downstream with time. The positive vortex located above the sensor of $x/H=2.16$ in image ① arrives at the downstream side of the sensor in image ②. There appears a subsequent negative vortex at the upstream side of the sensor ($x/H=2.16$). A pair of these counter-rotating vortices induces the downward flow between them from the channel core to the bottom wall. This wall-ward flow is responsible for the temporal maximum of the wall heat flux fluctuation at this instant at $x/H=2.16$, which is seen clearly from Fig. 9(a).

The vortex pair travels downstream and brings about a maximum wall heat flux at $x/H=2.64$ and at the instant of image ④. In image ④, there are a negative vortex and a positive vortex at the downstream and upstream sides of the heat flux sensor of $x/H=2.16$, respectively. At this instant an upward flow is induced between the vortices. Contrary to the above-mentioned mechanism of the heat transfer enhancement, the fluid flows away from the channel wall and the heat flux fluctuation takes a local minimum there, as seen from the wall heat flux fluctuation at $x/H=2.16$.

Subsequently the wall heat flux fluctuation at $x/H=3$ takes a temporal maximum when the pair of these vortices reaches the heat flux sensor, as shown in image ⑥ of Fig. 9(b). In this way, the heat transfer is enhanced extensively at the downstream region of the cylinder as the vortices move through the channel.

K. Suzuki and H. Suzuki⁽¹⁾ studied numerically the flow and the heat transfer in a laminar channel flow obstructed by a square cylinder. They pointed out that a vortex is generated near the channel wall in phase with the shedding vortices from the cylinder and this vortex causes the wall-ward flow near the wall from the downstream side. This wall-ward flow supplies cooler fluid to the heated channel wall and augments heat transfer. The heat transfer enhancement caused by the wall-ward flow in the present study is considered to be based on the same mechanism as that reported by them.

The images of the flow visualization and the output signals of heat flux sensor are shown in Fig. 10 for $b/h=1$. It was found⁽⁹⁾ that the wall heat flux fluctuates at the frequency of vortex shedding. There is a negative vortex at the left end of image ① in Fig.

Fig. 10 (a) Heat flux fluctuation and (b) flow visualization for $b/h=1$; ■, position of heat flux sensor

10(b). This negative vortex moves to only the upstream side of $x/H=2.16$ and induces the wall-ward flow in cooperation with a positive vortex located at its downstream side in image ②. The heat flux

fluctuation, therefore, takes a temporal maximum at this instant. The vortex pair moves farther downstream and brings about a local maximum in the wall heat flux at $x/H=2.64$ and $x/H=3.00$ at the instant shown in images ④ and ⑥, respectively, which is seen clearly in Fig. 10(a).

For $b/h=2$ and $b/h=3$, images of the visualized flow and the output signals of heat flux sensor are shown in Figs. 11 and 12, respectively. When $b/h \geq 2$, the separated shear layers at the leading edges of the cylinder become reattached to the side walls of the cylinder. In the case of $b/h=2$, two flow patterns that differ in the width of the wake are observed⁽⁸⁾ to appear intermittently in the cylinder wake. Consequently, the periodicity of velocity fluctuation weakens near the wall and the wall heat flux fluctuation does not show any predominant frequency⁽⁸⁾.

The wall heat flux of $b/h=2$ is seen to fluctuate less regularly compared to the other cases because of the intermittent change of the flow pattern in the near wake. Figure 11 corresponds to the case of the wide wake. It is inferred that the heat transfer is very likely augmented by the wall-ward flow as in the cases of $b/h=0.5$ and 1. At the instant shown in image ① in Fig. 11(b), a positive vortex locates on the upstream side of the heat flux sensor of $x/H=2.16$. When this vortex moves downstream, the wall heat flux takes a maximum value at $x/H=2.64$ as seen in images ③ and ④, and at $x/H=3.0$ in images ⑤ and ⑥.

The fluctuation of wall heat flux again becomes rather periodic for the case of $b/h=3$. At the instant shown in image ③, a positive vortex located at $x/H=2.64$ brings about a temporal maximum in the wall heat flux. As the vortex moves downstream, the heat transfer enhancement is found to occur at $x/H=3.0$, as seen from images ⑤ and ⑥.

The spatial relationship between the vortices and the position of the maximum wall heat flux differs somewhat, depending on b/h . When $b/h \leq 1$, the wall heat flux fluctuation shows a maximum between the positive vortex and the successive negative vortex located upstream. When $b/h \geq 2$, on the other hand, it appears that the wall heat flux has a maximum at the downstream side of the positive vortex. This difference is caused by the spatial arrangement of the shedding vortices. The length scales of the vortex of $b/h=0.5$ and 1 are larger than those of $b/h=2$ and 3⁽¹⁴⁾ and are comparable with the channel height. The vortices are aligned along the center line of the channel and induce wall-ward flow which is perpendicular to the channel wall. In contrast to the cases of $b/h=0.5$ and 1, the positive vortices of $b/h=2$ and 3 are shed towards the lower wall of the channel and the vortex centers lie below the channel centerline, and the nega-

Fig. 11 (a) Heat flux fluctuation and (b) flow visualization for $b/h=2$; ■, position of heat flux sensor

tive vortices are the opposite. These vortices are located in a staggered array and the wall-ward flow induces flow somewhat upstream when observed from the frame moving with the vortices.

Fig. 12 (a) Heat flux fluctuation and (b) flow visualization for $b/h=3$; ■, position of heat flux sensor

4. Conclusion

We have measured the time-averaged heat transfer coefficients and the wall heat flux fluctuations in a

channel flow with a rectangular cylinder. Flow visualization using a high-speed video recording system was synchronized with the measurements of wall heat flux fluctuations. The present investigation leads to the following conclusions.

(1) Heat transfer on a channel wall is enhanced extensively by the insertion of a rectangular cylinder. The distribution of the time-averaged local Nusselt number has a maximum at the downstream region of the cylinder.

(2) The vortices shed periodically from the cylinder induce the wall-ward flow from the core region of the channel. Wall heat flux fluctuation reveals a spatial and temporal maximum accompanied by the wall-ward flow.

(3) The wall heat flux fluctuates in phase with the shedding vortices from the cylinder. The position of the maximum wall heat flux moves downstream as the shedding vortices travel through the channel, which results in extensive heat transfer enhancement.

Acknowledgments

We are grateful to Professors M. Kumada and J. Mimatsu of Gifu University for their kind advice on measurements using the dynamic heat flux sensor. Thanks are also offered to Professors H. Fujimoto and J. Senda of Doshisha University for the provision of the high-speed video system. This work was partially supported by Doshisha University's Research Promotion Fund.

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