Heat Transfer around Tubes in Staggered Tube Banks*

by

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An experimental study was conducted to investigate the heat transfer and flow around tubes in staggered tube banks in a cross flow of air. The cylinder spacings examined $C_y/d \times C_x/d$ are 1.6×1.6 and 1.2×1.2 where C_x and C_y denote the cylinder spacings along and normal to the upstream uniform flow direction, and d the cylinder diameter, respectively. Main results obtained are as follows.

In case of 1.2×1.2 , the mean Nusselt numbers for the second and subsequent cylinders are almost equal to each other and much higher than that for the first cylinder. In case of 1.6×1.6 , the third cylinder exhibits the maximum heat transfer rate. This may be due to the high velocity of an oncoming flow to that along with its high turbulence intensity.

Key Words: Convective Heat Transfer, Experimental Study, Tube Bank, Heat Exchanger, Turbulence

1. Introduction

Prediction of heat transfer and flow around cylinders in tube banks is very important in relation to various engineering aspects. Recent energy crisis has been promoting developments of high performance heat exchangers for the effective use of energy. For that purpose, it is, first of all, necessary to clarify the detailed heat transfer characteristics of cylinders in tube banks.

Many of the previous works are mainly concerned with the heat transfer and the pressure loss through the bank⁽¹⁾⁻⁽⁵⁾. There has been little information as to the heat transfer process in connection with the fluid flow behaviors. Furthermore there have not been enough data on the tube banks of narrow spacing, which may be very important in relation to the design of compact heat exchangers.

From this standpoint, the present authors investigated, in detail, the heat transfer and flow around a group of three or four cylinders in an in-line or a staggered arrangement in a cross flow of air⁽⁶⁾⁻⁽¹¹⁾. The cylinder spacing was widely varied in order to obtain the basic data on the heat transfer in tube banks.

In case of the staggered arrangement, the heat transfer features for a narrow

spacing resemble those of a cylinder in a two-dimensional impinging jet. On the other hand, for a wide spacing, they approach those of a single cylinder in a uniform flow⁽¹⁰⁾⁽¹¹⁾. However effects of neighboring cylinders on the heat transfer characteristics may be serious in a staggered tube bank constructed of a large number of cylinders, especially in the narrow spacing.

Colburn⁽¹²⁾, Tucker⁽¹³⁾ and Kays et al.⁽¹⁴⁾ have represented the heat transfer data on which the effects of the cylinder spacing are quite small. On the other hand, Grimson⁽¹⁾ and Fishenden⁽²⁾ have demonstrated quantitatively the effects of the cylinder spacing based on the data by Pierson⁽²⁰⁾. It may be very important to make clear their discrepancy in connection with the design of heat exchangers. Furthermore, the local heat transfer behaviors of cylinders in tube banks may be necessary to estimate the thermal stress of the tubes heated under the condition of constant heat flux.

The purpose of the present study was to clarify the heat transfer characteristics around tubes in staggered tube banks, as shown in Fig. 1. Two cases are examined. One of them is a very narrow cylinder spacing, for which there has been little information on the heat transfer features, and the other is a moderate cylinder spacing which is widely used in industries. Heat transfer rate around each cylinder was measured along with the distributions of static pressures, mean velocities and turbulence intensities. Their mutual correlations are discussed in the present paper.

Nomenclature

C_D : pressure drag coefficient $= \int_0^\pi C_p \cos \theta d\theta$

C_p : pressure coefficient $= (P - P_f) / \frac{1}{2} \rho U_\infty^2$

C_x, C_y : cylinder spacing along and normal to upstream uniform flow direction

d : cylinder diameter

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h : heat transfer coefficient
 N : row number
 Nu : Nusselt number
 Nu_{mt} : mean Nusselt number expect the first and second rows
 Q : heat flow rate
 Re : Reynolds number = $U_t d / \nu$
 S : area of heated surface
 T : temperature
 U : mean flow velocity
 U_t : mean velocity through minimum clearance = $U_\infty C_y / (C_y - d)$
 $\sqrt{u'^2}$: r.m.s. of streamwise turbulent fluctuating velocity
 x, y : coordinates
 θ : angle from forward stagnation point
 Subscripts
 ave : mean value Nu_N of Nu_m from the first cylinder to the Nth one
 f : forward stagnation point
 m : mean
 max : maximum
 w : wall
 ∞ : upstream infinity

2. Experimental apparatus and technique

The wind tunnel used in the present study has a rectangular test section of 120 mm high and 225 mm wide. Two staggered tube banks examined are schematically shown in Fig. 1(a) and (b), which include the coordinate systems employed for each case. All the cylinder diameters are 25 mm. Fig. 1(a) exhibits a staggered tube bank of four transversal and seven longitudinal rows having $C_x/d = 1.2$ and $C_y/d = 1.2$, where C_x and C_y are the cylinder spacings along and normal to the upstream uniform flow direction, as included in the figure. Fig. 1(b) shows a case of three transversal and seven longitudinal rows having $C_x/d = 1.6$ and $C_y/d = 1.6$.

The case of Fig. 1(a) may correspond to a tube bank of very narrow cylinder spacing for which there has been little information on the heat transfer features, and the case of Fig. 1(b) may be a tube bank, which is widely used in engineering. The turbulence intensity of the upstream uniform flow $\sqrt{u'^2}/U_\infty$ was at most 0.06% for the case of Fig. 1(a) and about 0.02% for that of Fig. 1(b) under the condition of setting cylinders, respectively.

Experimental procedures were almost the same as in the earlier works (6)-(9). Heat transfer measurements were conducted by heating only the measured cylinder under the condition of constant heat flux (15)(16). Heat transfer coefficient and Nusselt number are defined as

$$h_0 = Q/S(T_w - T_\infty) \quad \text{and} \quad Nu_0 = h_0 d / \lambda$$

The mean velocity and the streamwise turbulence intensity at the center sections of

Fig. 1 Flow configuration and coordinate

neighboring rows were measured with a constant temperature hot-wire anemometer with a linearizing circuit. The hot-wire used was a 0.005 mm tungsten wire of effective length 1 mm.

Static pressure and velocity measurements were conducted under noheating of all the cylinders. The Reynolds number ranged from 8600 to 36000. Full care was taken in setting the cylinder since the flow deflected for specific values of C_x/d (17). The symmetry of the flow was found to be satisfactory through the distributions of the heat transfer coefficients and the static pressures over the whole circumference.

3. Experimental results and discussion

3.1 Heat transfer coefficient and pressure distribution

Represented in Fig. 2(a) and (b) are the distributions of static pressures and Nusselt numbers around each cylinder in case of $C_x/d = C_y/d = 1.6$ and $Re = 36000$. The results for the first cylinder show similar trends to those of a single cylinder (16)(18) though the flow separates about 10° downstream of the separation point for the latter, resulting in a downstream shift of minimum Nusselt number position. In cases of the second and third cylinders, the pressure decreases steeply from the forward stagnation point $\theta = 0^\circ$ to about $\theta = 30^\circ$ and the base pressure is the lowest for the second cylinder. The pressure distribution shows no material change for the fourth and subsequent cylinders. This suggests that the flows around them are nearly equal to each other. Furthermore their profile resembles that for a single cylinder in a high turbulent flow or at a Reynolds number higher than the critical one (19), and the flow separates at about $\theta = 120^\circ$.

As far as the heat transfer coefficient is concerned, that for the second cylinder is much higher than that for the first cylinder, suggesting that the oncoming flow to the former possesses a high velocity (see Fig. 9). It is clear that the heat transfer coefficient for the second cylinder is the highest near the rear stagnation

Fig. 2 (a) Local pressure distribution

Fig. 2 (b) Local Nusselt number distribution

point $\theta = 180^\circ$. This may be due to the lowest pressure there as described in Fig. 2 (a), resulting in a high velocity in the near wake of the second cylinder. The Nusselt number distribution exhibits little variation for the third and subsequent cylinders, the second peak existing around $\theta = 120^\circ$. Such profile is frequently found for the single cylinder in a high turbulent flow. Based on these descriptions along with the pressure distribution, in case of a relatively wide cylinder spacing such as 1.6×1.6 , it may not be unreasonable to say that the flow and heat transfer characteristics around the third and subsequent cylinders are nearly equal to those for the single cylinder in a high turbulent flow.

Previous works⁽¹⁵⁾⁽¹⁶⁾ show the highest mean Nusselt number for the third cylinder,

as in the present study. This may be due to the oncoming flow of high velocity and also of high turbulence intensity, as shown later in Fig. 9.

The local Nusselt number distribution makes no essential variation with the Reynolds number for both cases of 1.2×1.2 and 1.6×1.6 though the results are not shown in this paper.

Fig. 3 describes the variation of the mean Nusselt number Nu_m for each cylinder at various Reynolds numbers. It is clear that the average heat transfer rate is the lowest for the first cylinder and the highest for the third one as noted previously.

Shown in Fig. 4 is the mean value Nu_N of Nu_m from the first cylinder to the Nth

Fig. 3 Streamwise variation of mean Nusselt number

Fig. 4 Comparison of Nu_N with previous results

one, in which Nu_{ave} is Nu_7 . Previous results by Snyder⁽¹⁵⁾ and Pierson⁽²⁰⁾ are for tube banks of ten rows. In their results, consequently Nu_{ave} corresponds to Nu_{10} . The present data are in very good agreement with those by Pierson and are a little higher than those by Snyder.

The results for the case of smaller cylinder spacing 1.2×1.2 are shown in the following. Represented in Fig. 5(a) and 5(b) are the distributions of the static pressures and the heat transfer coefficients for each cylinder at $Re \approx 30000$. It is very clear that they are quite different from those for the case of 1.6×1.6 shown in Figs. 2(a) and 2(b).

The first cylinder shows the heat transfer and flow features, which are similar to

Fig. 5 (a) Local pressure distribution

Fig. 5 (b) Local Nusselt number distribution

those for the single cylinder in a high blockage condition⁽¹⁸⁾ because of the very narrow cylinder spacing in the direction normal to the upstream uniform flow. That is, the local Nusselt number Nu_θ reaches maximum at about $\theta = 70^\circ$ and it may be considered from the pressure distribution that the flow separates at about $\theta = 95^\circ$.

The pressure decreases steeply from $\theta = 0^\circ$ to 30° on the second cylinder. This may suggest the high velocity of the flow oncoming to that. It results in a heat transfer behavior similar to that for the third and subsequent cylinders, as described in Fig. 5(b). This is different from the case of 1.6×1.6 . The minimum and maximum pressure coefficients locate at about $\theta = 40^\circ$ and 60° respectively. Flow visualization study was conducted using a water channel with floating aluminium powder. It is revealed that the flow near the wall is stagnant in the neighborhood of $\theta = 40^\circ$ to 60° . Similar flow feature was also detected with further downstream cylinders, and this may have some effect upon the heat transfer and static pressure distributions around those cylinders.

The static pressure around a single cylinder in a two-dimensional jet⁽²¹⁾⁽²²⁾ makes a large decrease on the upstream face similar to that in the region from $\theta = 0^\circ$ to 30° on the second and subsequent cylinders of the tube bank. Its decreasing rate, however, becomes small downstream. On the other hand, the pressure in the region from $\theta = 40^\circ$ to 60° decreases more steeply downstream. These behaviors of static pressure may produce the heat transfer characteristics that Nu_θ decreases to downstream in the region from $\theta = 0^\circ$ to 30° and increases in the same direction in the region from $\theta = 30^\circ$ to 70° .

Nu_θ in the region from $\theta = 90^\circ$ to the rear stagnation point $\theta = 180^\circ$ is nearly equal for the third to the sixth cylinders. A deflection of the wake downstream of the seventh cylinder⁽¹⁷⁾ may produce a lower value of Nu_θ around it. It is not reasonable to determine the separation point on the second and subsequent cylinders only with the results shown in Fig. 5(a) and 5(b). Flow pattern visualized with the water channel shows that their separation point locates at about $\theta = 110^\circ$, though the Reynolds number is an order of 1000 and is quite small as compared with 30000 in the figures.

Fig. 6 represents the variation of mean Nusselt number Nu_m with the Reynolds number for all the cylinders. The heat transfer rate of the first cylinder is lower by about 45 to 50% than that for others and also lower by about 20% than that for the single cylinder⁽²³⁾. The value of Nu_m for the present case of 1.2×1.2 is lower than that for the case of 1.6×1.6 shown in Fig. 4. Furthermore Nu_m of the third cylinder is not necessarily the largest in the present narrow cylinder spacing of 1.2×1.2 and it is different from the results for 1.6×1.6 and the previous data⁽¹⁵⁾⁽¹⁶⁾. It is clear that Nu_m for the second through the seventh cylinders is nearly equal to each other. A dash-dot-dot-line included in the figure plots the average value of Nu_m for all the cylinders. The present result is in excellent agreement with that by Pierson⁽²⁰⁾ but

Fig. 6 Variation of mean Nusselt number with Reynolds number. ---, mean value of seven cylinders; —, Num by Pierson⁽²⁰⁾; —, Num by Žukauskas⁽¹⁸⁾; ---, Num for single cylinder⁽²³⁾; Φ , Δ , Kostić et al.⁽¹⁶⁾

is a little higher than that by Žukauskas⁽¹⁸⁾. Data by Kostić et al.⁽¹⁶⁾ for a bank of three transversal and nine longitudinal rows of 1.5×1.5 are shown in Fig. 6 for comparison. The symbol Φ stands for the third cylinder and Δ for the sixth one.

Tube banks used in industries are usually composed of a large number of cylinders. In order to estimate their mean heat transfer coefficient, it may be reasonable to apply the present mean Nusselt numbers except those for the first and second cylinders, since their values are quite different from those for the third and subsequent cylinders. The results obtained are given in Fig. 7. That is, Num_t denotes the mean Nusselt number from the third to the seventh cylinders. It is very clear that the results for both the cases of 1.2×1.2 and 1.6×1.6 are in very good agreement with each other, though the local heat transfer behaviors are quite different. An average line is expressed by

$$Num_t = 0.243 Re^{0.64}$$

as included in the figure. These results may be favorable to the design of compact heat exchangers, though the pressure loss naturally increases as the cylinder spacing decreases.

Demonstrated in Fig. 8 is the pressure drag coefficient for each cylinder. The results for the case of 1.2×1.2 show that C_D is the largest for the first cylinder, the minimum for the second one, and is nearly equal for others. There is no such large variation for the case of 1.6×1.6 , and the value for the latter is much lower than that for the former. However under the condition of an equal Reynolds number, the mass flow rate decreases with a decrease of the cylinder spacing and it results in a decrease of the power driving the fluid flow around the cylinders in spite of a large pressure loss.

Fig. 7 Variation of Num_t with Reynolds number

Fig. 8 Streamwise variation of pressure drag coefficient. $\circ$, $C_y/d = C_x/d = 1.2$, $Re = 30000$; Δ , $C_y/d = C_x/d = 1.6$, $Re = 36000$

The mean heat transfer coefficient decreases with a decrease of the cylinder spacing in case of in-line tube banks⁽²⁴⁾ and, on the other hand, it makes no material change with the cylinder spacing in case of staggered tube banks as shown in Fig. 7. It may be concluded from these descriptions that the staggered arrangement is more favorable than the in-line arrangement for the design of compact heat exchangers.

3.2 Velocity and turbulence intensity

There has been little information as to the velocity and turbulence intensity in the flow through the tube bank. It may not be easy to measure the velocity and turbulence in such a complicated and high turbulent flow. However their data may be very useful to infer the heat transfer process of the tube bank. Therefore the present authors measured the mean velocity and turbulence intensity with a hot-wire anemometer⁽⁶⁾⁻⁽¹¹⁾ for the tube banks consisting of a small number of cylinders.

Fig. 9(a) and 9(b) demonstrated the measured results for the case of 1.6×1.6 at $Re \approx 36000$, and they correspond to those at $x/d = 0.8$ and 2.4 respectively. Data are shown in the region of $y > 0$ because of their symmetry on each side of x -axis. The shear layer separated from the first cylinder possesses a very large velocity gradient and $(U/U_\infty)_{max}$ attains about 2.2. The velocity decrease in the vicinity of $y/d = 0$ and 1.6 is due to the existence of the second

row cylinders. Furthermore the turbulence intensity therein is about equal to that of an upstream uniform flow.

On the other hand, the results shown in Fig. 9(b) reveal that $(\sqrt{u^2}/U_\infty)_{\min}$ increases up to about 25% between the second and third cylinders and $(U/U_\infty)_{\max}$ decreases to about 1.8. The velocity gradient of the shear layer is much smaller than that from the first cylinder. The data measured at other cross sections show a similar trend to that found in Fig. 9(b), though the results are not given in this paper.

Figs. 10(a) and 10(b) describe the results obtained for the case of 1.2×1.2 at $Re \approx 30000$. The data at $x/d = 0.6$ are shown in Fig. 10(a) and those at $x/d = 5.4$ in Fig. 10(b), respectively. The shear layer separated from the first cylinder has a large velocity gradient and $(U/U_\infty)_{\max}$ reaches about 4.1, but $(\sqrt{u^2}/U_\infty)_{\min}$ is relatively small in the order of that of the upstream flow. A velocity profile of w-shape found in the vicinity of $y/d = 1.2$ suggests the existence of a reverse flow behind the first cylinder, forming a vortical flow region. The maximum velocity decreases drastically downstream of the second cylinder. However the velocity profile shows no large variation toward the downstream though the velocity fluctuation becomes violent, the fluctuation range being shown by $\bar{Q}$ in Figs. 9 and 10. On the other hand, a distance between the points of $(U/U_\infty)_{\max}$ increases in the downstream direction. This may correspond to an increase of the through flow width because of the turbulent diffusion. The pressure and heat transfer behaviors from $\theta = 0^\circ$ to 30° around the second and

Fig. 9 Streamwise velocity (O) and turbulence intensity (●) distributions for $C_y/d = C_x/d = 1.6$, $Re \approx 36000$.

(a) $x/d = 0.6$ between first row and second one

(b) $x/d = 5.4$ between fifth row and sixth one

Fig. 10 Streamwise velocity (O) and turbulence intensity (●) distributions for $C_y/d = C_x/d = 1.2$, $Re \approx 30000$.

subsequent cylinders resemble those of a cylinder impinged by a two-dimensional jet²²⁾

4. Concluding remarks

The heat transfer and fluid flow measurements were conducted for two staggered tube banks of $C_x/d = C_y/d = 1.2$ and 1.6 in the Reynolds number range from 8600 to 36000. The main results obtained are summarized as follows.

In case of 1.6×1.6 , the heat transfer rate for the third cylinder is the maximum independently of the Reynolds number as found in the previous works. This may be due to the high velocity of the flow oncoming to the third cylinder along with its high turbulence intensity.

In case of 1.2×1.2 , the heat transfer rate of the first cylinder is extremely low. The mean Nusselt number for the second and subsequent cylinders makes no material change. A very large velocity of the flow oncoming to the second cylinder produces a high heat transfer rate as compared with that in case of 1.6×1.6 .

In spite of a large difference of the local heat transfer behaviors, the mean heat transfer coefficient of the downstream cylinders is almost equal for both the cases of 1.2×1.2 and 1.6×1.6 at the same Reynolds number.

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