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HEAT TRANSFER OF TUBES CLOSELY SPACED IN AN IN-LINE BANK

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Abstract – An experimental investigation of heat transfer around four cylinders closely spaced in a cross-flow of air has been conducted. The cylinders are settled in tandem with equal distances between centers. Their in-line pitch ratio is in the range of $1.15 \leqq c/d \leqq 3.4$ (c = distance between cylinders' centers, d = cylinder diameter); the Reynolds number ranges from 10^4 to 5×10^4. It is found that there exists a critical Reynolds number Red_c at which the heat transfer behavior changes drastically, and is correlated with the in-line pitch ratio by

$$Red_c = 1.14 \times 10^5\ (c/d)^{-5.84}.$$

Variations of characteristic features of the mean and local Nusselt numbers are discussed in relation to the length of the vortex formation region behind the cylinder.

NOMENCLATURE

- c, longitudinal spacing between cylinders' centers;
- C_p, pressure coefficient $= (P - P_\infty)/\frac{1}{2}\rho U_\infty^2$;
- d, cylinder diameter;
- h_θ, local heat transfer coefficient $= q/(T_w - T_\infty)$;
- Num, mean Nusselt number;
- Nu_θ, local Nusselt number $= h_\theta d/\lambda$;
- l_v, length of vortex formation region;
- P, static pressure;
- q, heat flux per unit area and unit time;
- Red, Reynolds number $= U_\infty d/\nu$;
- T, temperature;
- T_w, wall temperature;
- u', turbulent fluctuating velocity along flow direction;
- U_∞, velocity at upstream uniform flow;
- x, y, coordinates.

Greek symbols

- θ, circumferential angle from forward stagnation point;
- θ_a, θ at which Nu_θ attains a maximum;
- λ, ν, ρ, thermal conductivity, kinematic viscosity and density of air.

Subscripts

- c, critical;
- ∞, upstream infinity.

1. INTRODUCTION

PREDICTION of heat transfer from cylinders in tube banks is very important in relation to various engineering aspects and there has been much work [1–4]. In designing heat exchangers there have recently been trends that spacings between cylinders are reduced in order to make them as small as possible.

In his review paper [4], Žukauskas has reported that the mean Nusselt number of the cylinder in tube banks increases generally with decreasing the spacing between the cylinders in the flow direction since the turbulence intensity around the cylinder becomes large, and that such a flow situation is thought to be very similar to a single cylinder settled behind the turbulence grid where the turbulence intensity increases with decreasing the distance between them. On the other hand, several cases can be found in which the heat transfer rate decreases with decreasing the spacing at small Reynolds numbers [2].

Ishigai *et al.* [5] made a visualization study for the flow around cylinders in tube banks with the Schlieren method; the cylinder diameter tested was 31.7mm. They revealed that in the case of small in-line pitch ratio ($c/d \leqq 1.5$), the shear layer separated from the upstream cylinder and re-attached onto the downstream cylinder without forming a vortex. It is presumed from their results that in such a flow situation, the main flow may not be entrained into the neighborhood of cylinder surfaces and the flow between those cylinders is separated from the main flow by the separated shear layer and is very stagnant. Therefore the heat transfer from those cylinders is expected to level down.

The Reynolds number ranged from 1400 to 13 500 for the in-line arrangement in the work of Ishigai *et al.* [5] who found that the flow pattern changed little in this Reynolds number range. However Bloor [6] measured the length of the vortex formation region for a single cylinder and found that it decreased steeply when the Reynolds number exceeded 10^4 for a cylinder

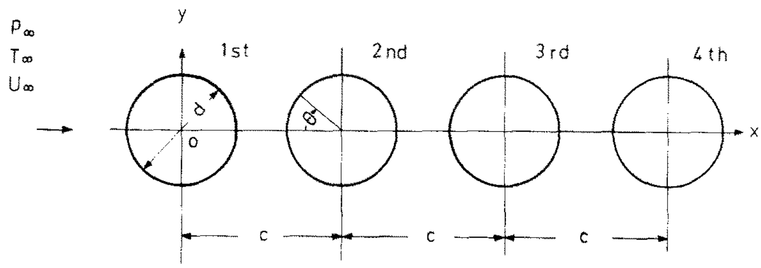

FIG. 1. Arrangement of cylinders and coordinate system.

of 1 in. diameter, which is not much different from that of Ishigai *et al.* It may be expected then that in the region of $Red > 10^4$, the flow pattern around the cylinders in tube banks varies with the Reynolds number even in the region of $c/d \leqq 1.5$. Kostić *et al.* [7] measured the heat transfer from two aligned cylinders in cross-flow, but unfortunately the in-line pitch ratio was studied only in the range of $c/d \geqq 1.6$.

It may be concluded from these results that the heat transfer characteristics of tubes in in-line arrangement are still not clear for the cases of very small pitch ratio, which are very important in relation to designing heat exchangers. From this standpoint, the objective of the present study is to investigate the heat transfer characteristics of four aligned tubes in a cross-flow of air especially considering cases of small in-line pitch ratio.

2. EXPERIMENTAL APPARATUS AND TECHNIQUE

The wind tunnel used in the present study is the same as that employed in previous studies [8, 9], and the test section is a rectangle 325 mm high and 225 mm wide. The turbulence intensity of the upstream uniform flow $(\overline{u'^2})^{1/2}/U_\infty$ was about 0.5–0.7% through the experiments. Four cylinders are settled at equal intervals at the center of the test section as shown in Fig. 1 which includes the coordinate system employed. All the cylinders are 26 mm in diameter and the in-line pitch ratio c/d was varied from 1.15–3.4 and the Reynolds number Red based on the cylinder diameter from 10^4–5×10^4. Cases with small values of c/d were especially investigated in detail.

For the heat transfer measurements, a stainless steel ribbon wound helically around the cylinder of hard vinyl chloride tube was electrically heated and the inside of the tube was filled with a rigid urethane foam to minimize heat loss. All the four cylinders were heated under the condition of the same heat flux. The wall temperature was measured at intervals of 10° around the cylinder surfaces with 0.065 mm copper–constantan thermocouples stuck to the back of the stainless steel ribbon. The heat flux to the four cylinders was ranged from 1.02 to 2.73 kW/m².

Measurements of the wall static pressure were made using four other cylinders which had pressure holes of 0.5 mm diameter without heating all of them. Temperature in the flow field was measured with a temperature probe of a 0.065 mm copper–constantan thermocouple.

In the present results, no corrections were made for the tunnel wall effects upon the heat transfer and flow characteristics. The symmetry of the flow around four cylinders with respect to the horizontal axis was confirmed from the results of the local heat transfer coefficient and the pressure distribution and in addition, the two-dimensionality of the flow field was found to be consistent with the visualization of surface flow pattern.

3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1. *Mean Nusselt number*

Variations of the mean Nusselt number of four cylinders with the in-line pitch ratio are typically shown at $Red = 4.1 \times 10^4$ in Fig. 2 which includes, for comparison, the results for the second cylinder of three cylinders in a tandem arrangement [8]. The present data for the second cylinder has, in general, a similar trend to the previous data, but at the small value of c/d, the former is lower than the latter because all four cylinders are heated in the present study (only the second cylinder was heated in [8]).

The result for the first cylinder has qualitatively the same trend as that for the second one as clearly shown in Fig. 2 and shows a quite complicated variation with the in-line pitch because of the presence of the downstream cylinders, though it is generally accepted that the heat transfer of the tube in the first row of the

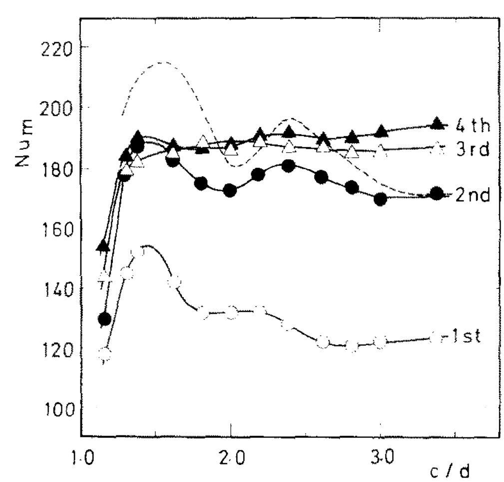

FIG. 2. Variation of mean Nusselt number with in-line pitch ratio at $Red = 4.1 \times 10^4$. ---, Aiba *et al.* [8], $Red = 4.0 \times 10^4$.

bank is similar to that of a single cylinder. The first and second cylinders have maxima at about $c/d = 1.5$, while there is no distinct peak there for the third and fourth cylinders and their heat transfer rate changes little in the range of $c/d > 1.3$. It is very clear in Fig. 2 that the heat transfer rate decreases steeply in the region of $c/d < 1.3$ for all the cylinders, though Žukauskas [4] reported that the heat transfer rate generally increased on decreasing the in-line pitch.

When c/d increases above 3.4 shown in Fig. 2, the well-known jumping phenomena [10] occur for all the cylinders at about $c/d = 3.8$. This is confirmed by the following facts in the additional experiments. That is, in the region of $c/d > 3.8$, *Num* for the first cylinder becomes nearly equal to that for the single cylinder. On the other hand, the values of *Num* for other cylinders approach very gradually to that for the single cylinder. For example, at $c/d = 5.0$ the former is 20–35 % larger than the latter. Similar results are also obtained by Kostić *et al.* [7] for the second cylinder of two aligned cylinders. These results say that the wake turbulence originating from the first cylinder exerts its effects upon the heat transfer of the downstream cylinders, far downstream.

Figure 3 shows variations of the mean Nusselt number with the Reynolds number at $c/d = 1.3$. It is very interesting to notice that all the results for the four cylinders vary discontinuously at about $Red = 2.1 \times 10^4$. *Num* for all the cylinders is lower than that for the single cylinder in the region of $Red < 2.1 \times 10^4$. It increases suddenly at $Red = 2.1 \times 10^4$ (its rate of increase is largest for the second cylinder) and the values for the second, third and fourth cylinders then become larger than that for the single cylinder by about 10–35 %. Furthermore differences in *Num* between them in the region of $Red > 2.1 \times 10^4$ are very small compared with those in the $Red < 2.1 \times 10^4$ region. All the results obtained for the first cylinder are smaller than that for the single cylinder in the Reynolds number range studied.

The results shown in Fig. 3 suggest that the heat

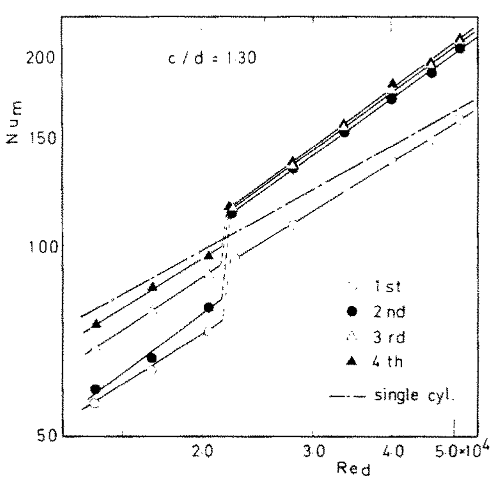

FIG. 3. Variation of mean Nusselt number with Reynolds number.

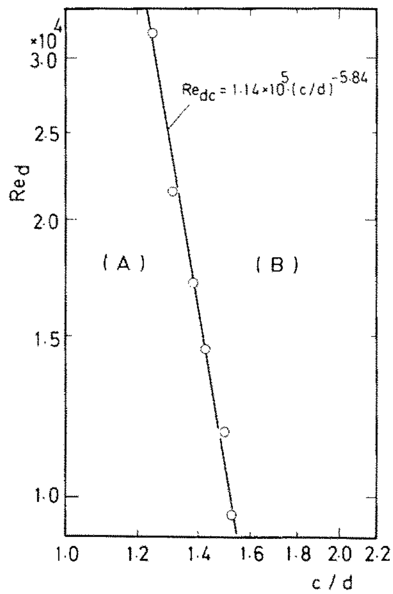

FIG. 4. Critical Reynolds number in-line pitch ratio.

transfer and flow characteristics change drastically at $Red = 2.1 \times 10^4$ for the case of $c/d = 1.3$. Such a Reynolds number is hereafter called the critical Reynolds number Red_c. As shown in Fig. 4 it decreases with increasing c/d, and is a function of the in-line pitch ratio c/d; it is expressed by

$$Red_c = 1.14 \times 10^5 \ (c/d)^{-5.84}. \tag{1}$$

Note that equation (1) is obtained in the range examined in the present study; that is, $1.15 \leqq c/d \leqq 3.4$ and $10^4 < Red < 5 \times 10^4$. It can be considered that equation (1) expresses reversely the critical value of in-line pitch ratio $(c/d)_c$ as a function of *Red*. It shows that $(c/d)_c$ increases with decreasing *Red*. However as described later, the length of the vortex formation

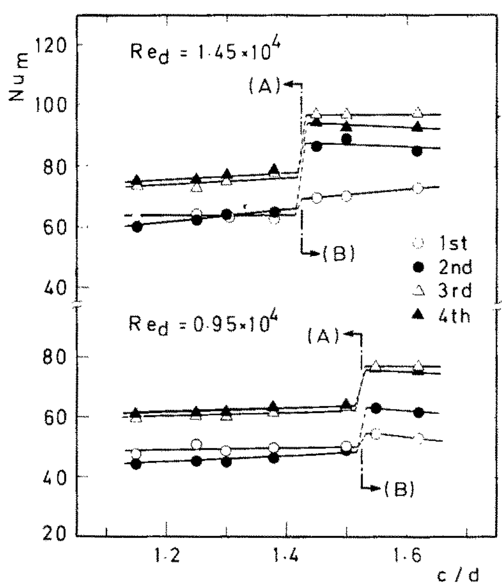

FIG. 5. Variation of mean Nusselt number with in-line pitch ratio.

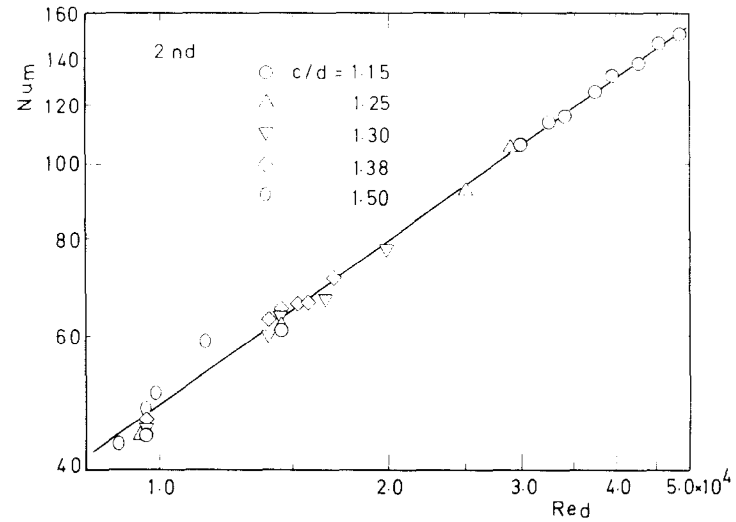

FIG. 6. Variation of mean Nusselt number with Reynolds number of the second cylinder in the range of $Red < Red_c$.

region for the single cylinder of 1 in. diameter measured by Bloor is roughly equal to 2.5 d in the region of $Red = 10^3$–10^4 but it decreases on decreasing Red lower than 10^3 (see Fig. 12). It is conjectured from this fact that $(c/d)_c$ may not increase beyond 2.5.

At a given value of c/d, the heat transfer rate is lower than that of the single cylinder in the Reynolds number range included in the region A, and it increases drastically at the critical Reynolds number. Subsequently in the region B, *Num* is much larger than that in A. It is very important to note in designing a heat exchanger that there exists such a critical state in the heat transfer characteristics of the tube bank.

The variation of *Num* with c/d is typically shown in Fig. 5 at $Red = 1.45 \times 10^4$ and 0.95×10^4 for which the critical values of in-line pitch ratio are 1.43 and 1.52 respectively. It is clear that *Num* for all the cylinders shows little change with c/d in the region A; that is, $c/d < (c/d)_c$. The heat transfer coefficient increases drastically when c/d increases over $(c/d)_c$ at a constant Reynolds number.

Figure 6 shows a typical example of variation of *Num* with *Red* for the second cylinder. All the data presented are included in the region A defined in Fig. 4. It is quite interesting that they are brought into a single correlation independently of the in-line pitch. Such a result is easily inferred from the data shown in Fig. 5. In Fig. 7 the results obtained in the region A for the four cylinders are summarized without plotting the data for clearness. The scatter of the data is almost the same as that in Fig. 6. The results are expressed independently of the in-line pitch, within the error of $\pm 7\%$, as follows:

$$Num = 1.64 \times 10^{-1} \cdot Red^{0.618} \quad \text{for the first cylinder,} \tag{2}$$

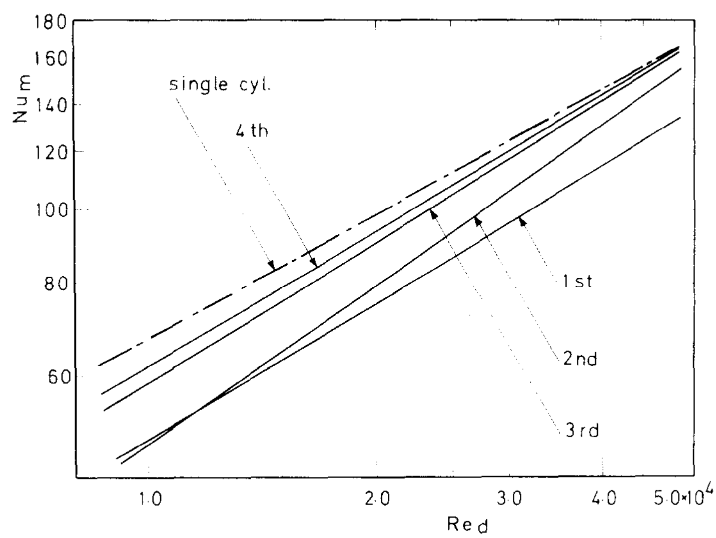

FIG. 7. Variation of mean Nusselt number with Reynolds number of four cylinders in the range of $Red < Red_c$.

$Num = 0.695 \times 10^{-1} \cdot Red^{0.711}$ for the second cylinder, (3)

$Num = 1.84 \times 10^{-1} \cdot Red^{0.626}$ for the third cylinder, (4)

$Num = 2.50 \times 10^{-1} \cdot Red^{0.599}$ for the fourth cylinder. (5)

Note that they are valid in the Reynolds number range of $Red < Red_c$. By the way, the results shown in Fig. 5 may give an impression that the data included in the region B are summarized independently of the in-line pitch ratio as in the region A. However the range of c/d belonging to the region B is much wider than that shown in Fig. 5 and *Num* shows large variations with c/d as shown in Fig. 2. Therefore, in the region B, the results for *Num* can not be expressed independently of c/d.

The heat transfer rates of three cylinders other than the first one approach that of the single cylinder with increasing Reynolds number. On the other hand, the result for the first cylinder is somewhat smaller than that for the single cylinder and its approach to the latter is very slow. The reason for such a result will be clear in what follows. It is interesting to notice that the mean Nusselt number of the first cylinder becomes nearly equal to that of the second one at small Reynolds number around $Red = 10^4$.

3.2. *Local heat transfer*

Shown in Fig. 8 are representative examples of the local Nusselt number distribution at $c/d = 1.3$ for the cases of $Red = 2.04 \times 10^4$ in the region A and $Red = 2.16 \times 10^4$ in the region B. It is very clear that their characteristic behaviors are quite different from each other though the difference in the Reynolds numbers between them is very small.

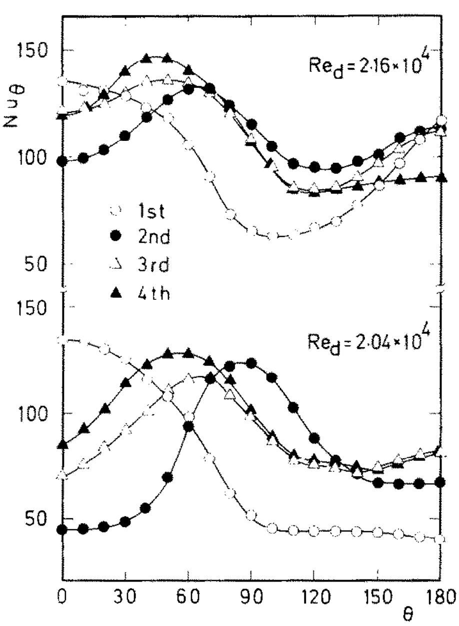

FIG. 8. Local Nusselt number distribution in the vicinity of Red_c for $c/d = 1.3$.

As far as the first cylinder is concerned, the Nusselt number distributions show qualitatively the same trend for the front face in both cases, but indicate very different behavior for the rear face. That is, Nu_θ shows little change in the region from about $\theta = 100$–$180°$ and its value is very small for the case of $Red = 2.04 \times 10^4$. On the other hand, Nu_θ has a minimum around $\theta = 100°$ and after that increases monotonically with the circumferential angle for the case of $Red = 2.16 \times 10^4$. This is very similar to findings for the single cylinder at the same order of Reynolds number as in the present study.

As to the second cylinder, Nu_θ for the case of $Red = 2.04 \times 10^4$ has a maximum around $\theta = 85°$ but Nu_θ on the front face is as small as that on the rear face of the first cylinder. It may be inferred from these results that the flow between the first and second cylinders is very stagnant at $Red = 2.04 \times 10^4$. For the case of $Red = 2.16 \times 10^4$ Nu_θ shows a maximum around $\theta = 65°$ which locates upstream by about 20° compared with that for the case of $Red = 2.04 \times 10^4$ and Nu_θ on the rear and front faces is 1.7–2 times larger than that for the latter case. Similar behavior can be found for the third and fourth cylinders and the circumferential angle θ_a at which Nu_θ exhibits a maximum is 50° and 45° for the third and fourth cylinders respectively at $Red = 2.16 \times 10^4$, while $\theta_a = 65°$ and 55° for the third and fourth cylinders respectively at $Red = 2.04 \times 10^4$.

It may be concluded from these results that the flow between cylinders shows a drastic change at about $Red = 2.1 \times 10^4$ for the case of $c/d = 1.3$. In the case of $c/d = 1.3$ discussed here, the local Nusselt number distributions show little change with the Reynolds number in the regions of $Red < 2.1 \times 10^4$ and $Red > 2.1 \times 10^4$.

As previously shown in Fig. 2, *Num* at $Red = 4.1 \times 10^4$ decreases steeply at very narrow spacing such as $c/d = 1.15$. The local Nusselt number distribution is very similar to that for the case of $c/d = 1.3$ and $Red = 2.04 \times 10^4$ which belongs to the region A. In the case of $c/d = 1.15$, all the Reynolds numbers tested in the experiments are lower than Red_c, and correspondingly the heat transfer characteristics belonging to the region B could not be reached.

It is already shown in Fig. 8 that the circumferential angle θ_a at which Nu_θ attains a maximum changes drastically at the critical Reynolds number in a similar way to the mean Nusselt number. Therefore it is possible to find out the critical Reynolds number from the characteristic behavior of θ_a with *Red* which is shown in Fig. 9. In the region of $Red < Red_c$, θ_a is almost independent of c/d and $\theta_a = 80$–$90°$, 60–65°, and 50–60° for the second, third and fourth cylinders respectively; that is, θ_a decreases on shifting downstream. It decreases suddenly at the critical Reynolds number. Subsequently in the region of $Red > Red_c$, the value of θ_a is smaller than that in the region of $Red < Red_c$.

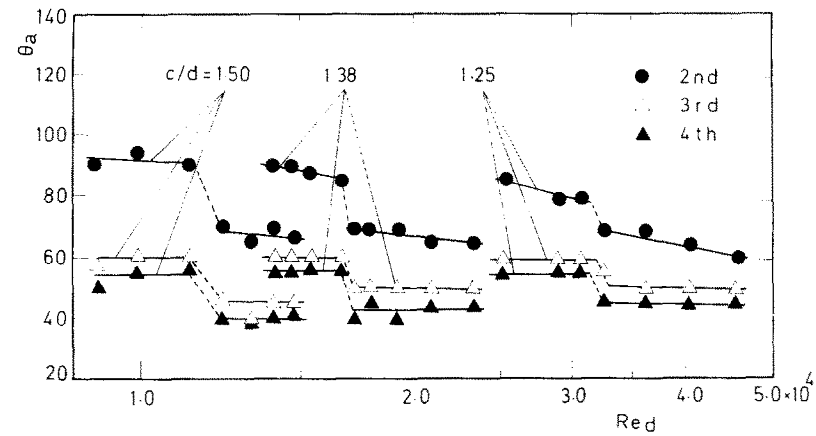

FIG. 9. Variation of maximum Nusselt number position θ_a with Reynolds number.

3.3. *Temperature and pressure distributions*

Figure 10 shows a typical example of temperature distribution in the flow for the case of $c/d = 1.3$ and $Red = 1.92 \times 10^4$ which belongs to the region A. The temperature is highest in the separated flow region between the first and second cylinders compared to that in the other regions, and it exactly corresponds to the fact that the heat transfer decreases on the rear face of the first cylinder and on the front face of the second one, as shown in Fig. 8. The temperature inside the separated flow region between the two cylinders decreases to the downstream.

The pressure distributions along the surfaces of the four cylinders are shown in Fig. 11. The case of $Red = 1.90 \times 10^4$ belongs to region A and that of $Red = 2.14 \times 10^4$ to the region B. Present data for $Red = 1.90 \times 10^4$ show qualitatively the same trend as those obtained by Ishigai *et al.* [5]. The pressure on the front face of the second cylinder is almost equal to the base pressure on the first cylinder, showing that the flow between those cylinders is very stagnant. Subsequently it results in the decrease of the heat transfer rate there. The pressure coefficient for the third and fourth cylinders reaches the peak values at which the shear layer, separated from the upstream cylinder, impinges onto the downstream one, while the pressure coefficient for the second cylinder has no peak.

The pressure distribution experiences quite large variations at the Reynolds number exceeding the critical one, similar to the local Nusselt number distribution; that is, the results obtained at $Red = 2.14 \times 10^4$ show that the angular position of the maximum value of C_p shifts upstream for the third and fourth cylinders and, in addition, a peak appears around $\theta = 60°$ for the second cylinder, and furthermore the variation of C_p along the cylinder surfaces is greater compared with that at the Reynolds number of $Red < Red_c$ such as $Red = 1.90 \times 10^4$.

3.4. *Discussion*

It is obvious that the shear layer separated from the upstream cylinder effects greatly the flow around the downstream one, and it is decisively important for the heat transfer whether that layer is or is not rolling up and forming the vortex upstream of the downstream cylinder. Ishigai *et al.* [5] described from their flow visualization study that in the Reynolds number range 1400–13 500, the vortex was not found at the in-line pitch ratio of $c/d \leq 1.5$ except behind the most downstream cylinder because of the insufficient spacing to form it.

On the other hand, Bloor [6] measured, using a wind tunnel of very low free-stream turbulence (lower than 0.03 %), the length of the vortex formation region

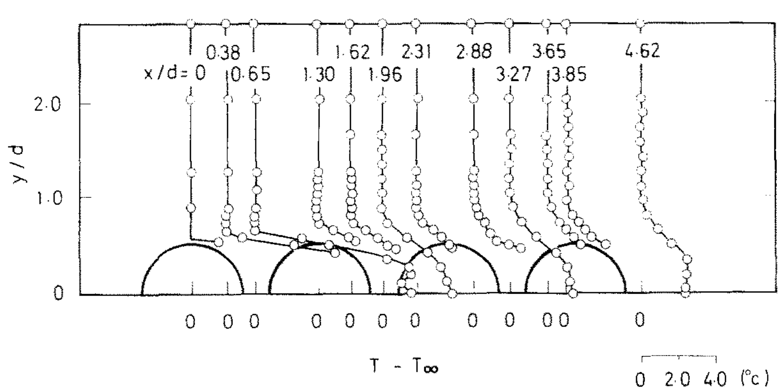

FIG. 10. Temperature distribution in the flow field for $c/d = 1.3$, $Red = 1.92 \times 10^4$ and $q = 1.02\,kW/m^2$.

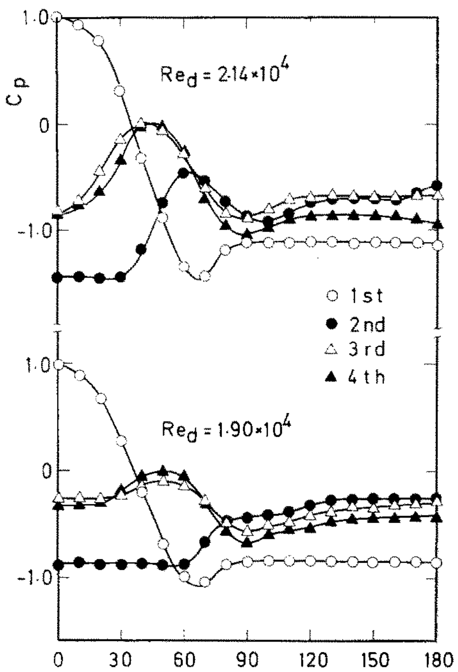

FIG. 11. Pressure distribution in the vicinity of Red_c for $c/d = 1.3$.

l_v downstream of the single cylinder. The results are shown in Fig. 12 and it is clear that the value of l_v/d is quite different depending on the cylinder diameter. She has described how this dependency of l_v/d upon d may originate from the fact that the base pressure coefficient varies with d under the very low free-stream turbulence.

The results for the cylinder of 1 in. diameter, which is nearly equal to that of the present cylinder, show no essential change in the region from $Red = 10^3$–10^4. It however decreases quite rapidly when the Reynolds number increases beyond 10^4; that is, the longitudinal distance from the cylinder for the initial formation of the vortex becomes shorter with increasing Red. It may therefore be presumed from this fact that as far as the tube bank in in-line arrangement is concerned, the shear layer separated from the first cylinder has a possibility of rolling up and forming the vortex upstream of the second cylinder even at the in-line pitch ratio of $c/d < 1.5$ in the region of $Red > 10^4$, though it may be considered that the flow around the first cylinder is different from that around the single cylinder because of the presence of downstream cylinders.

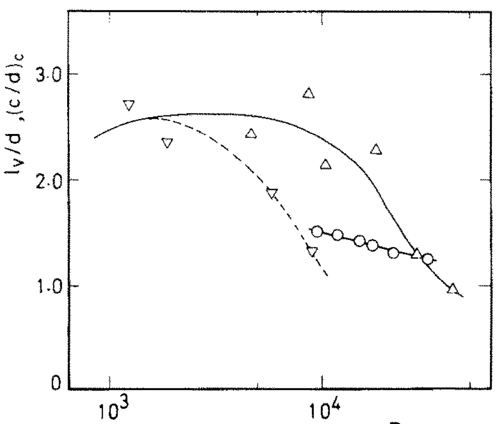

FIG. 12. Comparison of present critical in-line pitch ratio with length of vortex formation region by Bloor. Present data: ○. Bloor's data: △, $d = 1$ in.; ▽, $d = 1/4$ in.

In a situation where the vortex is not formed downstream of the first cylinder, the very stagnant flow region appears between the first and second cylinders and is separated from the main flow. Consequently the heat transfer rate levels down on the rear face of the first cylinder and on the front face of the second one. On the other hand, when the vortex is formed, the entrainment of the main flow around the faces mentioned before increases [11]. Subsequently the heat transfer rate increases drastically in such a situation. The heat transfer of the third and fourth cylinders also changes drastically at the critical Reynolds number as shown in Fig. 3. It may originate from the sudden change of the flow pattern around the first cylinder, and its effects continue but decrease to the downstream. Therefore the variation of the heat transfer rate at the critical Reynolds number becomes small for the third and fourth cylinders compared with that for the second one.

The critical in-line pitch ratio $(c/d)_c$ as a function of Red obtained in the present study is compared with the length of the vortex formation region l_v of Bloor in Fig. 12. The Reynolds number range examined in the present study corresponds to the region where l_v for the cylinder of 1 in. diameter decreases steeply with increasing Reynolds number, and the present data show a trend which is a little different from that of Bloor. As previously described, Bloor's data were obtained under a situation of very low free-stream turbulence. On the other hand, Gerrard [11] has described that on increasing the free-stream turbulence, the value of Red, at which l_v/d begins to decrease, decreases and approaches that for the cylinder of smaller diameter. As to the present data obtained using a wind tunnel of free-stream turbulence of 0.5–0.7%, it may be inferred for such free-stream turbulence that the present results for $d = 26$ mm, which is nearly equal to $d = 1$ in. in Bloor's study, approach her data for $d = 1/4$ in. around $Red = 10^4$. Furthermore in the case of the tube bank, the turbulence of oncoming flow may not be so low as that in Bloor's study, and the interference effects among cylinders are very large and $(c/d)_c$ never decreases below 1.0 because of the geometrical limitation of the tube arrangement. These facts may result in the deviation of the present data of $(c/d)_c$ from the data of l_v/d for $d = 1$ in. measured by Bloor.

It may be concluded from the comparison shown in Fig. 12 and the above discussions that the present critical Reynolds number explains the critical flow state at which the shear layer, separated from the first cylinder, rolls up and forms initially the vortex upstream of the second cylinder. The present data for Red_c shown in Fig. 4 can be considered through the descriptions noted previously not to exhibit a strong dependency upon the cylinder diameter. It is, however, necessary to conduct a detailed study to verify the above presumption.

4. FINAL REMARKS

Heat transfer measurements have been made for the in-line tube bank constituted by four cylinders along with the temperature and static pressure measurements. The main results obtained are summarized as follows:

There exists the critical Reynolds number Red_c at which the heat transfer and flow around the four cylinders (especially the second one) changes drastically, and that is expressed as a function of the in-line pitch ratio.

In the region of $Red < Red_c$, the heat transfer on the rear face of the first cylinder and on the front face of the second one decreases. The mean Nusselt numbers for all the four cylinders are smaller than that for a single cylinder, and they collapse into four correlations independent of the in-line pitch.

Beyond the critical Reynolds number, the heat transfer rate increases suddenly on the surfaces facing the flow between each two cylinders and the mean Nusselt number in the region of $Red > Red_c$ is much larger than that in the region of $Red < Red_c$.

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TRANSFERT THERMIQUE DE TUBES SERRES DANS UN RESEAU EN LIGNE

Résumé – On étudie expérimentalement le transfert de chaleur autour de quatre cylindres étroitement espacés dans une écoulement d'air transversal. Les cylindres sont placés en tandem à égale distance entre axes. Le rapport du pas longitudinal c au diamètre d est $1,15 \leqslant c/d \leqslant 3,4$; le nombre de Reynolds varie de 10^4 à 5×10^4. Il existe un nombre de Reynolds critique Red_c pour lequel le transfert thermique change fortement et il est lié au rapport précédent par

$$Red_c = 1,14 \times 10^5 \, (c/d)^{-5.84}.$$

On discute les variations des nombres de Nusselt moyens et locaux en relation avec la longueur de la région de formation des tourbillons derrière le cylindre.

WÄRMEÜBERGANG IN EINEM FLUCHTENDEN ROHRBÜNDEL

Zusammenfassung – Der Wärmeübergang an vier Zylindern, die in dichter Anordnung einem Querstrom von Luft augesetzt sind, wird experimentell untersucht. Die Zylinder sind paarig mit jeweils gleichem Abstand angeordnet. Ihre Längsteilung wird variiert zwischen $1,15 \leq c/d \leq 3,4$ (c = Mittenabstand der Zylinder, d = Zylinderdurchmesser); die Reynolds-Zahl liegt zwischen 10^4 und 5×10^4. Es zeigt sich, daß eine kritische Reynolds-Zahl Red_c existiert, bei der sich das Wärmeübertragungsverhalten drastisch ändert; sie ist mit der Längsteilung durch folgende Beziehung korreliert:

$$Red_c = 1,14 \times 10^5 (c/d)^{-5.84}.$$

Änderungen charakteristischer Werte der mittleren und der örtlichen Nusselt-Zahl werden im Hindblick auf ihren Zusammenhang mit der Länge der Wirbelschleppe hinter dem Zylinder untersucht.

ТЕПЛОПЕРЕНОС ОТ ТРУБ В КОРИДОРНОМ ЛУЧКЕ

Аннотация — Проведено экспериментальное исследование переноса тепла в поперечном потоке воздуха от четырёх близко отстоящих друг от друга цилиндров, расположенных последовательно один за другим с равными расстояниями между центрами в пределах $1{,}15 \leqq c/d \leqq 3{,}4$, где c — расстояние между центрами цилиндров, d — диаметр цилиндра. Значения числа Рейнольдса изменялись в диапазоне от 10^4 до 5×10^4. Найдено, что существует критическое значение числа Рейнольдса Red_c, при котором существенно изменяется картина переноса тепла и которое связано с шагом цилиндров следующим соотношением

$$Red_c = 1{,}14 \times 10^5 (c/d)^{-5{,}84}.$$

Рассмотрено изменение средних и локальных значений числа Нуссельта в зависимости от длины зоны формирования вихря за цилиндром.