Heat transfer promotion with a cylinder array located near the wall

Y. Kawaguchi*, K. Suzuki* and T. Sato†

Heat transfer from a flat plate has been investigated when a cylinder array is located near the wall. Each cylinder in the cylinder array was positioned normal to the flow direction and parallel to the flat plate surface. Measurements of the heat transfer coefficient and the optimum value for the cylinder pitch and spacing between the cylinders and the flat plate surface were obtained. A comparison of the heat transfer mechanism in this flow system with that obtained previously for the case when a single cylinder is inserted in the boundary layer was made.

Keywords: *convective heat transfer, heat transfer promoter, complex turbulent flow, relaxing turbulent flow*

Introduction

There have been a number of investigations on the heat transfer from the roughened wall of a duct[1-3]. In these studies, the effects of rib height, rib pitch, Reynolds number, rib geometry etc on the overall heat transfer coefficient and pressure drop are examined experimentally. A slightly different situation has been treated in another paper[4]. In Ref 4, heat transfer from a flat plate when the turbulent boundary layer is disturbed by a single cylinder is discussed. The present paper is an extension of that work and deals with the cases when a number of cylinders are inserted in a line at various streamwise pitches in a region close to the plate. One of the purposes of the present study is to see if there exists an optimum value for the spatial pitch between cylinders to enhance the flat plate heat transfer. The mechanism of heat transfer enhancement is also discussed.

Experimental apparatus and procedure

The experimental apparatus used in the present study is basically the same as that used in Ref 4. Therefore, only important differences between the present experimental system and that of Ref 4 will be described.

Fig 1 shows a schematic view of the cylinder/flat-plate system used in the present study. The cylinders are made of synthetic resin. The cylinder diameter, d (8 mm), and the velocity of approaching flow, U_∞ (14 m/s), are the same as those values used in Ref 4, where a single cylinder was used as a turbulence generating body. It is, therefore, possible to compare directly the present data with the previous results. The cylinder Reynolds number $Re_d = dU_\infty/\nu$ is about 7700 and is in the subcritical range. The Karman-like vortices are thus found even when a single cylinder is located deep in the boundary layer but at reduced frequency[5].

The first cylinder is located at a streamwise position of 1400 mm downstream from the leading edge of the flat plate. This is the position where a single cylinder was inserted in previous studies[4-7]. The boundary layer thickness, δ, at the first cylinder position is about 28 mm when it is removed. The value of δ/d is thus about 3.5. Tests are made for four different pitches of cylinders: $p = 50$ mm, 100 mm, 200 mm and 400 mm. Experiments are also repeated for the case treated in Ref 4, ie when a single cylinder is inserted. The reproducibility of the results was confirmed to be good. These results are presented and identified as those with a pitch $p = \infty$.

The insertion of the cylinder array increases the momentum loss. No attempt has been made to keep the pressure field constant along the x direction when the cylinder array is inserted. Thus, the pressure field differs from one case to another. However, the cylinder diameter to tunnel height ratio is small in the present study. The difference in the free stream velocity between the upstream of the first cylinder and the downstream of the last cylinder was confirmed to be, at most, less than 4%. This rate of flow acceleration corresponds to the value 4×10^{-8} for the acceleration parameter $K = (\nu/U_\infty^2)(dU_\infty/dx)$. This is two orders of magnitude smaller than the critical value of K for flow relaminarization[8].

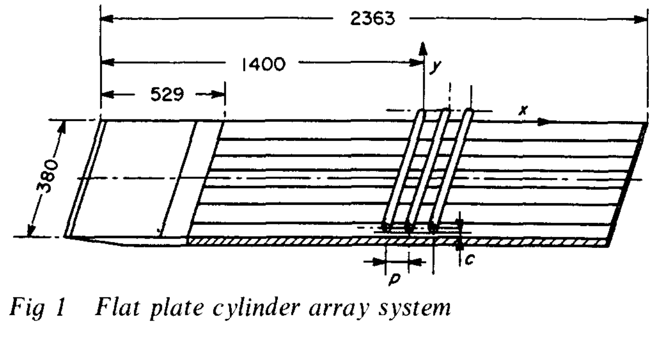

Fig 1 Flat plate cylinder array system

* Department of Mechanical Engineering, Kyoto University, Kyoto, Japan
† Department of Mechanical Engineering, Setsunan University, Osaka, Japan
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The space between the plate and the cylinder, c, is changed in three steps. It was suggested in Ref 4 that the mechanism of heat transfer augmentation may be different in the regimes of $c<4$ mm and of $c>4$ mm. A representative value of c in each regime, $c=2$ mm and 11 mm, was chosen in this study. For the case when a single cylinder is inserted, detailed information on flow and turbulence characteristics are available at these two values of c from Refs 5–7. In addition to these two cases, experiments were also performed when the cylinders were attached directly to the plate. This case corresponds to a roughened plate with ribs.

All the geometries tested in this study are tabulated in Table 1. For wider use of the present results, it is desirable to describe the geometrical parameters and the coordinates in dimensionless forms. However, there is a difficulty in doing this because there are three fundamental length scales in the present study: the cylinder diameter, d; and the outer and inner length scales of the turbulent boundary layer, δ and ν/u_τ (u_τ is the friction velocity and ν the kinematic viscosity). In this first attempt to study the effects of cylinder pitch and of the space between the cylinders and the flat plate, the last two length scales are left uncontrolled. For general use, additional experiments must be made in the future to see the effects of each length scale. In this study, the cylinder diameter, d, is used temporarily as a representative length scale in rendering dimensionless the geometrical parameters. This is preferable to compare the present results for $c=0$ and those of Refs 1–3. Dimensionless values of two parameters, c/d and p/d, are also shown in Table 1. However, the later discussion of the results will be made in terms of dimensional geometrical parameters and coordinates since it is still unknown which length scale is suitable for rendering them dimensionless when c is not zero.

Table 1 Geometry of cylinder array

c, mm	d, mm	c/d	p/d
0	50	0.0	6.25
0	100	0.0	12.5
0	200	0.0	25.0
0	400	0.0	50.0
0	∞	0.0	∞
2	50	0.25	6.25
2	100	0.25	12.5
2	200	0.25	25.0
2	400	0.25	50.0
2	∞	0.25	∞
11	50	1.375	6.25
11	100	1.375	12.5
11	200	1.375	25.0
11	400	1.375	50.0
11	∞	1.375	∞

The streamwise distance x is measured downstream from the centre of the first cylinder, and the normal distance y is measured outward from the plate. Seven strips of thin metal sheets each 50 μm thick are glued in parallel to the plywood flat plate. Care was taken to minimize the waviness of the metal sheets. Surface waviness or roughness is less than 0.3 mm and the wall can be regarded as almost hydrodynamically smooth. The space between the neighbouring sheets is 1 mm or less. These sheets are electrically connected in series and are heated by passing an alternating electric current through them. One hundred and twenty-six thermocouples are attached to the back surface of the centre metal sheet in the streamwise region of $-100\text{ mm}<x<963$ mm. These are used to measure the streamwise distribution of heating wall surface temperature. Twenty-four thermocouples are attached to the remaining six metal sheets to confirm the two-dimensionality of the thermal field. Except for the two sheets most remote from the centreline of the test plate, the spanwise non-uniformity of the measured heat transfer coefficient is within 2%. A number of thermocouples are also attached on the back surface of the plywood plate. The measured temperature difference between the surfaces of the plywood plate and the one-dimensional heat conduction equation are used to determine the conduction rate of heat loss through the plywood plate. Although this rate of conduction heat loss is about 5% of total heat flux, it is subtracted from the Joule heating flux of the metal sheets to yield a corrected local heat flux. The radiation heat exchange between the flat plate and the upper wall of the wind tunnel is about 1% of the electrical heating flux, thus it is ignored. The value of the heat transfer coefficient is defined as:

$$h_x=\frac{q_w}{T_w-T_\infty}$$

where q_w is the corrected wall heat flux, and T_w and T_∞ are the flat plate surface temperature and the free stream temperature, respectively. When the cylinder array is attached to the flat plate surface, heat conduction between the cylinders and heating surface can occur. However, both the thermal conductivity of the cylinder material and the contact area between the cylinder and the flat plate surface are small. Radiative heat loss to the cylinder is, at most, equal to the magnitude mentioned for the upper wall of the tunnel. Thus no attempt was made to subtract the magnitude of this heat loss due to the surface extension.

Notation

- c Clearance between cylinders and flat plate, mm
- d Diameter of cylinder, mm
- F_x Parameter of heat transfer augmentation
- h_x Local heat transfer coefficient, kW/m²K
- h_{x0} Local heat transfer coefficient for undisturbed boundary layer, kW/m²K
- T Mean temperature, K
- T_w Wall temperature, K
- T_∞ Free stream temperature, K
- U_∞ Free stream velocity, m/s
- u_τ Friction velocity, m/s
- p Pitch of the cylinder array, mm
- q_w Wall heat flux, kW/m²
- x Streamwise distance measured from the first cylinder, mm
- y Distance from the wall, mm
- δ Boundary layer thickness at $x=0$, mm
- ν Kinematic viscosity, m²/s

In a supplementary study, a fine thermocouple of 0.1 mm diameter was traversed in the y direction at nine streamwise locations: $x = 25, 50, 75, 125, 150, 175, 225, 250$ and 275 mm for two typical cases to be discussed later.

Results and discussion

Before entering the discussion on the distribution of the local heat transfer coefficient, h_x, for each case studied, an understanding of the overall influence of the cylinder array will be developed by analysing the experimental data from a macroscopic viewpoint. For this purpose, the following parameter used in the previous study is used:

$$F_x = \frac{\int_{x0}^{x} h_x \, dx}{\int_{x0}^{x} h_{x0} \, dx}$$

where h_{x0} is the local heat transfer coefficient obtained for the case where no cylinder is inserted in the boundary layer, and -100 mm ($x_0/d = -12.5$) was adopted for the value of x_0 appearing in the lower integration limit. The parameter F_x is the ratio of the mean heat transfer coefficients between the case of interest and the one when no cylinder is inserted. It describes a mean effectiveness of enhancement of the flat plate heat transfer averaged over the distance $(x - x_0)$. The streamwise variation of F_x was computed for every case studied by numerically integrating the measured values of h_x. All the results obtained are compared in Figs 2 to 4. First, it is seen that locating cylinders at such a remote position as $c = 11$ mm ($c/d = 1.38$) is not efficient in enhancing the heat transfer from the flat plate. When $c = 0$ or 2 mm ($c/d = 0$ or 0.25), much better enhancement of heat transfer is attained if the pitch of cylinders is appropriately chosen. Appropriate values for the pitch are 100 mm or 200 mm ($p/d = 1.25$ or 25) when the cylinders are attached to the plate, ie when $c = 0$. The values 100 mm or 200 mm ($p/d = 12.5$ or 25) are also appropriate for the case when a small space is present between the flat plate and the cylinder, ie when $c = 2$ mm ($c/d = 0.25$). A comparison of these two cases, $c = 0$ and

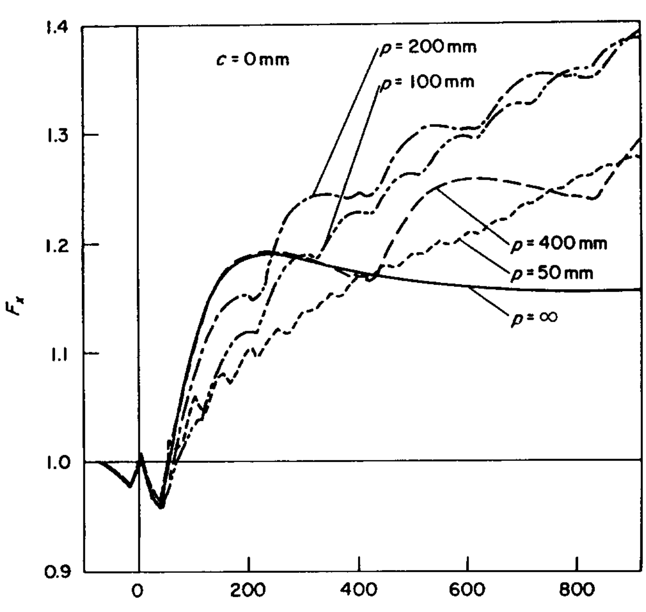

Fig 2 Overall heat transfer augmentation performance of cylinder array (c = *0 mm*)

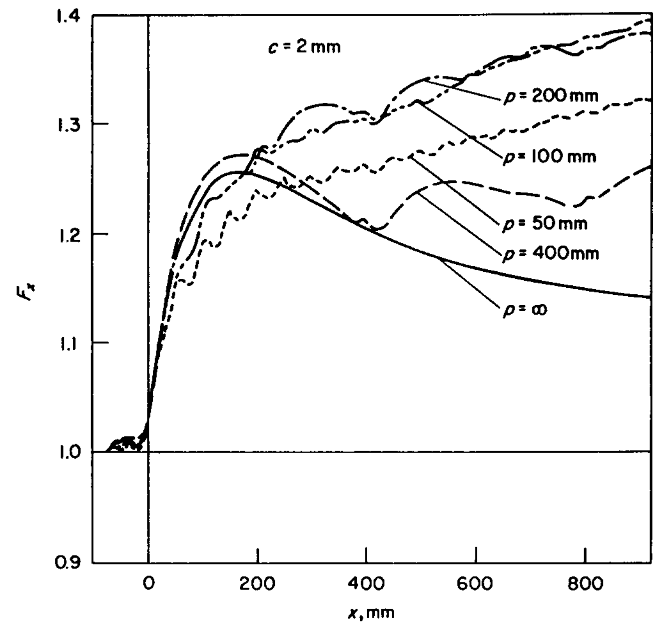

Fig 3 Overall heat transfer augmentation performance of cylinder array (c = *2 mm*)

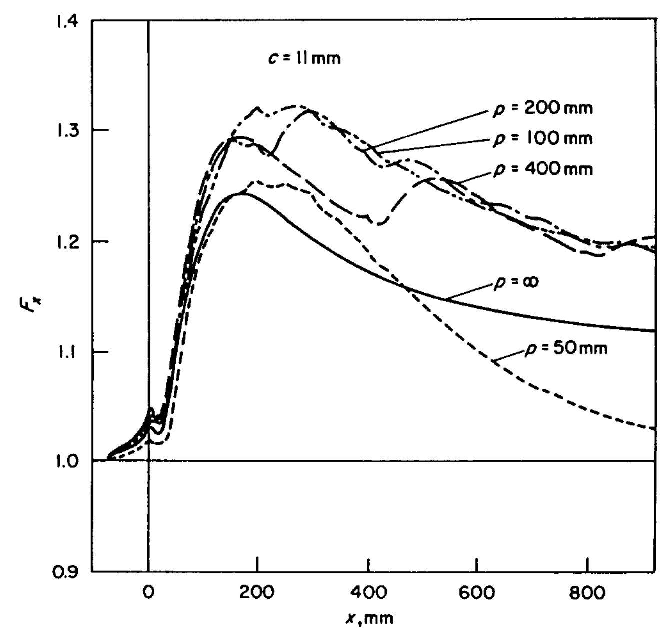

Fig 4 Overall heat transfer augmentation performance of cylinder array (c = *11 mm*)

$c = 2$ mm ($c/d = 0$ and 0.25), indicates that the latter case is slightly more effective in enhancing the flat plate heat transfer. Based upon these observations, the following discussion on the local heat transfer coefficient are mainly concentrated on the cases when $c = 0$ and 2 mm ($c/d = 0$ and 0.25).

All the distributions of the local heat transfer coefficient, h_x, obtained are shown in Figs 5 to 7 in a normalized form of h_x/h_{x0}. Each figure includes five diagrams (a) to (e). Diagram (a) presents the results obtained when a single cylinder is inserted, ie $p = \infty$. The other diagrams (b) to (e) are the results obtained for other cylinder pitches: $p = 400, 200, 100$ and 50 mm ($p/d = 50, 25, 12.5$ and 6.25).

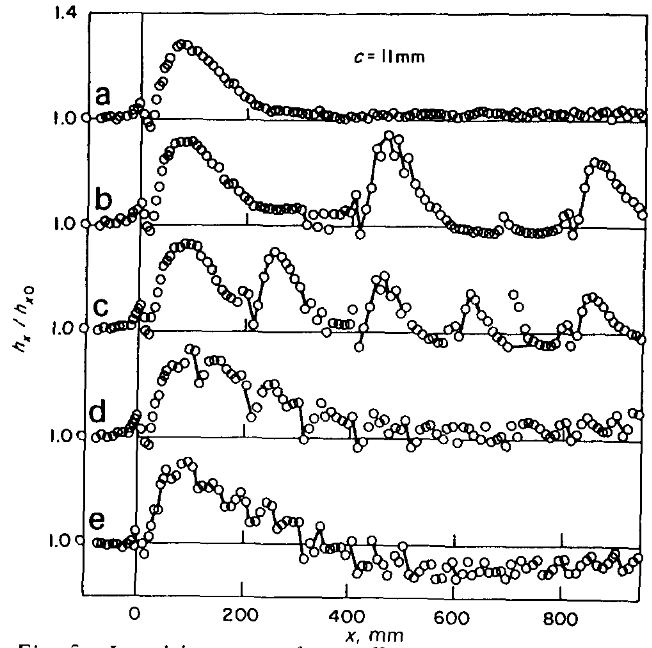

Fig. 5 *Local heat transfer coefficient for* c = *11 mm*: (*a*) p = ∞; (*b*) p = *400 mm*; (*c*) p = *200 mm*; (*d*) p = *100 mm*; (*e*) p = *50 mm*

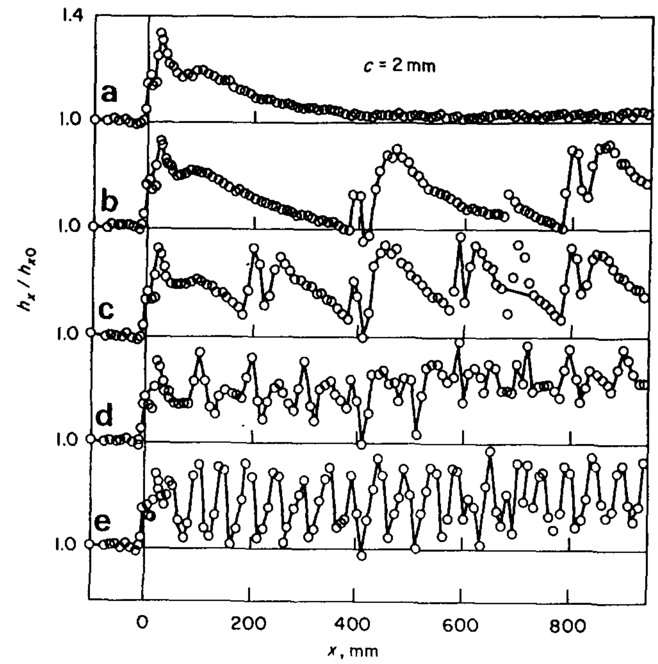

Fig 7 *Local heat transfer coefficient for* c = *2 mm*: (*a*) p = ∞; (*b*) p = *400 mm*; (*c*) p = *200 mm*; (*d*) p = *100 mm*; (*e*) p = *50 mm*

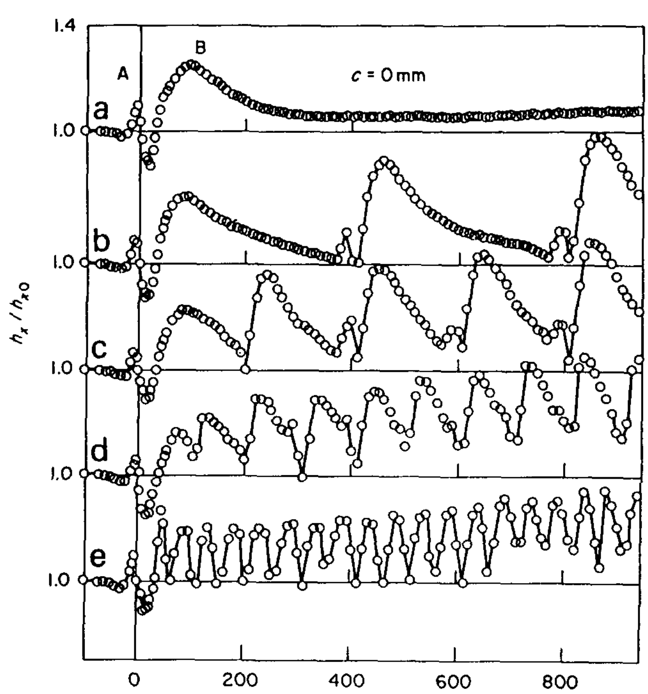

Fig 6 *Local heat transfer coefficient for* c = *0 mm*: (*a*) p = ∞; (*b*) p = *400 mm*; (*c*) p = *200 mm*; (*d*) p = *100 mm*; (*e*) p = *50 mm*

The results obtained for the largest space between the plate and the cylinder, ie when $c = 11$ mm ($c/d = 1.38$), are shown in Fig 5. For the pitch $p = 400$ mm ($p/d = 50$), Fig 5(b), a periodical streamwise change in h_x is observed. Similar periodical changes in h_x are also shown in Fig 5(c) when the pitch is reduced to $p = 200$ mm ($p/d = 25$). In this case, and in the other two cases with smaller pitches, $p = 100$ and 50 mm ($p/d = 12.5$ and 6.25), while a periodical change in h_x is observable to some extent, the mean value of h_x averaged over one spatial period clearly decreases in the downstream direction. A single cylinder located at this height causes a disturbance in the flow and temperature fields thus enhancing the flat plate heat transfer, as shown in Fig 5(a). In a similar manner, enhancement of heat transfer is brought about by the first cylinder in Figs 5(b) to (e). However, the disturbance created by the first cylinder affects the enhancement created by cylinders located downstream. The heat transfer enhancement obtained from the second cylinder is less than that attained if the second cylinder is located far from the first one. Similar reduction in the level of heat transfer enhancement continues to occur for other cylinders located further downstream. Supplementary experiments to investigate the reason for this will be described later.

Incidentally, in Figs 5(b) and 5(c) an odd fluctuation of h_x appears around $x = 700$ mm. This oddity also appears in two of the following figures. This is an error caused by the detaching of thermocouples from the metal sheets as a result of repeated heating. Thus the solid curves connecting experimental data are drawn smoothly there, disregarding the odd data.

The h_x distributions obtained for the case of zero space between the cylinder and the flat plate ($c/d = 0$) are shown in Fig 6(a) to (e). In all of Figs 6(b) to (e), periodical variation of h_x is observed. Contrary to the previous case, $c = 11$ mm ($c/d = 1.38$), the value of h_x averaged over one spatial period increases as one moves downstream. The results shown in Fig 6(b), for example, show that the h_x variation for the second cylinder is quite similar to that found around the first one. However, the magnitude of the h_x peak is higher for the second pattern than for the first. Disturbances caused by the first cylinder, in this case, magnify the enhancing performance of the second cylinder. As discussed in Ref 4, the turbulence generated in the shear layer existing at the edge of the recirculating bubble behind the cylinder plays an important role in heat transfer enhancement. This causes the peak B of h_x, specified in Fig 6(a), where the intense turbulence from the shear layer reaches the wall region. This peak also appears clearly in Figs 6(b) and 6(c) in every cyclic pattern of h_x variation. The turbulence generated in the shear layer formed in the wake of the first cylinder appears to intensify the turbulence in the shear layer formed in the wake of the second cylinder. Thus a little more augmented heat transfer may result, as indicated by the increase in the magnitude of h_x in the vicinity of the second cylinder. This reasoning can explain another difference between the first and the second patterns of h_x cyclic variation. In Fig 6(a) the peak B of h_x is located about 95 mm downstream from the centre of the cylinder. The same location is found for the first peak of h_x in both Fig 6(b) and 6(c). However, the corresponding distance for the second peak is reduced to 60 mm in Fig 6(b) and to 45 mm in Fig 6(c). Turbulence existing in the flow sweeping the surface of the second cylinder may give an effect similar to that of free stream turbulence on the flow around a single cylinder. Turbulence acts to raise the momentum diffusivity. This leads to a spatial delay in flow separation from the cylinder surface. When this is efficient enough, flow separation occurs on the back of the cylinder. Thus, a separated shear layer starts from a lower normal position, or from a position closer to the flat plate. This allows the shear layer to reach the wall region in a shorter distance. Moreover, the increased eddy diffusivity affects the flow pattern downstream of the cylinder and enables the shear layer turbulence to reach the wall region faster. Thus the noted decrease in the distance between the cylinder and the peak B for the second h_x pattern can be explained. A similar phenomenon may continue to occur when the cylinder pitch is reduced further, but in a somewhat different manner. This is because, when the cylinder pitch is too small, the main part of the h_x variation observed for a single cylinder cannot be completed in the spatial distance between two neighbouring cylinders. In fact, the shear layer from a cylinder has to reach the wall at a distance somewhat shorter than the pitch of cylinders, since another recirculating bubble can be formed in front of the succeeding cylinder. This results in a change in the patterns of the h_x variation certainly for the second and other successive cylinders, and sometimes even after the first cylinder. The latter can be found in Fig 6(e), where the first pattern of periodical h_x variation is quite different from the counterparts found in the other figures. In this case, the distance between two neighbouring cylinders is too short for the mechanism causing the peak B in Fig 6(a) to work effectively.

The results obtained when $c=2$ mm ($c/d=0.25$) are plotted in Fig 7(a) to (e). On the whole, the variation of h_x distribution with this cylinder pitch looks to be similar to that observed in Fig 6. However, there are some differences. The results for the case when a single cylinder is inserted in the flat plate boundary layer are plotted in Fig 7(a), and indicate that there are two narrow peaks of h_x and one mild one, located respectively at $x \sim 0$, $x=30$ mm and $x=80$ mm. The second and the last peaks are caused again by the turbulence coming from the shear layers formed in the wake of the preceding cylinder. When a cylinder is located detached from the flat plate, two shear layers appear, one originating from the separation point on the side of the cylinder facing the plate (lower shear layer) and another from that on the outer side of cylinder (upper shear layer). The upper shear layer causes the mild peak in h_x in Fig 7(a) while the lower shear layer causes the second narrow peak. The first narrow peak is caused by the flow acceleration beneath the cylinder. Details of these points are discussed in Ref 4. The same reasoning given in the discussion of the results in Fig 6 may be used to explain the mild peak which appears at a shorter distance after the second cylinder than for the first cylinder. However, when the separation points on the cylinder surface move downstream due to the effect of turbulence caused by the preceding cylinder, the lower shear layer starts from a higher normal position. Thus, it takes more distance for its effect to reach the wall region. This leads to an absorption of the second narrow peak by the mild one caused by the turbulence from the upper shear layer. This may be the reason why better enhancement is attained when $c=2$ mm ($c/d=0.25$) than when $c=0$ mm ($c/d=0$). This may also explain why only two h_x peaks appear in the pattern of its periodic variation for the second and other successive cylinders in Figs 7(b) and 7(c). However, in the case of the smallest pitch $p=50$ mm ($p/d=6.25$), there is no chance for the upper shear layer to reach the wall because of the short distance between the neighbouring cylinders. In this case, only one peak appears in the pattern of h_x variation for the second and other successive cylinders. This peak may be generated as the combination of the effects of the turbulence from the lower shear layer and the flow acceleration beneath the cylinder. In the case when $p=100$ mm ($p/d=12.5$), the distance between the neighbouring cylinders is too short for the two-peak-type heat transfer to occur constantly and too long for the one-peak heat transfer to occur in a regular fashion. Thus, the pattern may fluctuate with respect to time and space. Therefore, between one pair of cylinders, the former type of heat transfer can occur while, between another pair of cylinders, the latter may replace it. This may be the reason for the incomplete periodicity of the spatial variation in h_x found in this case.

It was pointed out previously that the flat plate heat transfer is not effectively enhanced when the cylinders are positioned at $c=11$ mm ($c/d=1.38$). When the cylinder pitch is less than 200 mm ($p/d<25$), the heat transfer enhancement attained by the second and the following cylinders is reduced to a lower level than that found for the case when only a single cylinder is used. While this is not a case of particular interest it may be worthy of additional studies to obtain a more general understanding of controlling turbulent heat transfer. Therefore, supplemental experiments to obtain temperature profiles were made for two cases, $c=2$ mm and 11 mm ($c/d=0.25$ and 1.38). The cylinder pitch, $p=100$ mm ($p/d=12.5$), was the same for these two experiments. A set of mean temperature profiles obtained at nine streamwise positions for each case is shown in Fig 8 for $c=2$ mm ($c/d=0.25$), and Fig 9 for $c=11$ mm ($c/d=1.38$). Hollow circles in these figures represent the wall temperature which was used previously in the calculation of local heat transfer coefficient h_x. All the plotted data are dimensional temperatures measured from free stream temperature T_∞.

Comparing these two figures, it is found that the temperature boundary layer is thinner when $c=11$ mm ($c/d=1.38$) than when $c=2$ mm ($c/d=0.25$). This suggests that the thermal diffusion is not effectively enhanced by the insertion of the cylinders in the fully turbulent region when $c=11$ mm ($c/d=1.38$). This must lead to a conclusion that the heat transferred from the flat plate is stored in a narrow region along the flat plate. Thus the mean fluid temperature in that region must be higher at $c = 11$ mm ($c/d = 1.38$) than at $c = 2$ mm ($c/d = 0.25$). This is actually observed over the entire streamwise region of the boundary layer, as can be seen from a comparison of Figs 8 and 9. The higher fluid temperature near the wall should be the main cause of the ineffective enhancement of heat transfer found in the case when $c = 11$ mm ($c/d = 1.38$). Since it is outside the scope of the present study, no attempt was made to explain the physical background of this phenomenon. However, it may be interesting to make detailed studies in the future of why the disturbance caused by each cylinder and the turbulence generated in the wake of each cylinder do not increase the thermal diffusion effectively when the cylinder is positioned at a large distance from the plate.

Conclusion

When a cylinder array is located very close to, or attached to, a flat plate, enhancement of heat transfer from the flat plate is attained when the pitch of cylinders is appropriately chosen. In these cases, the turbulence generated in the shear layer behind a cylinder moves toward the wall region and plays an important role in enhancement of heat transfer. The above two cases can be compared with each other, and indicate that a little better enhancement is obtained when a small space is present between the flat plate and the cylinder. This is because the turbulence coming from the lower shear layer can agitate the flow near the wall in addition to a similar effect from the turbulence generated in the upper shear layer. When the cylinders are located at positions rather remote from the flat plate, the disturbance generated by a cylinder works to suppress the enhancing effect of the successive cylinders. The heat transfer resistance in the turbulent region does not seem to be reduced effectively by the

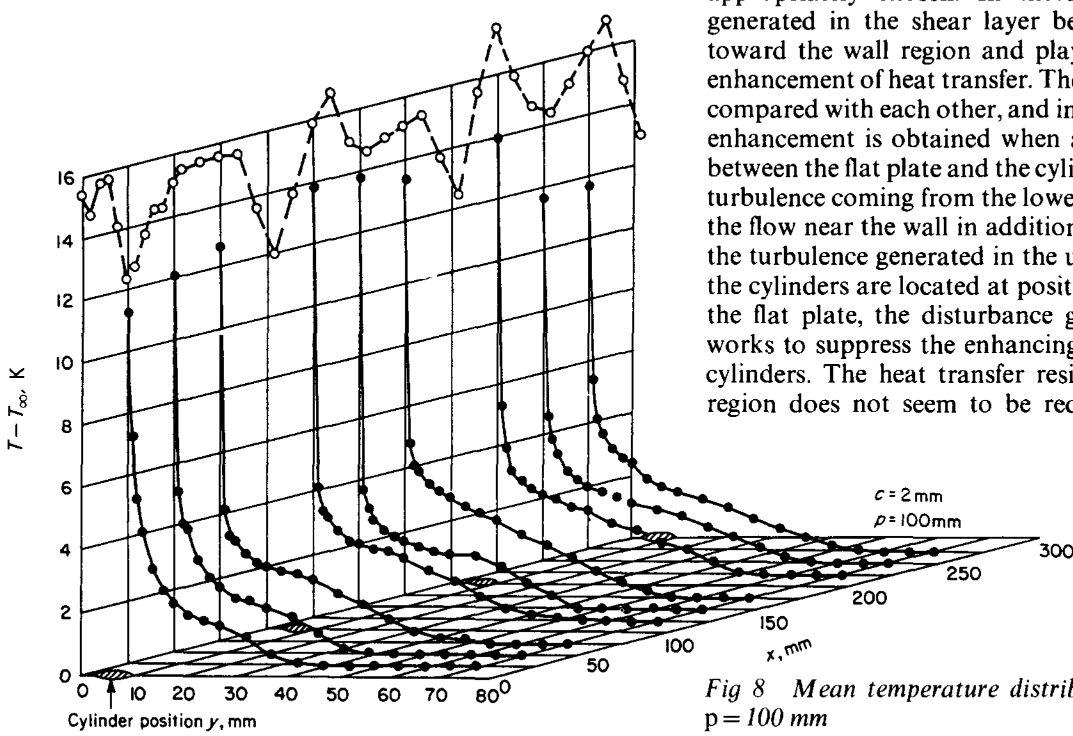

Fig 8 Mean temperature distribution for c = *2 mm and* p = *100 mm*

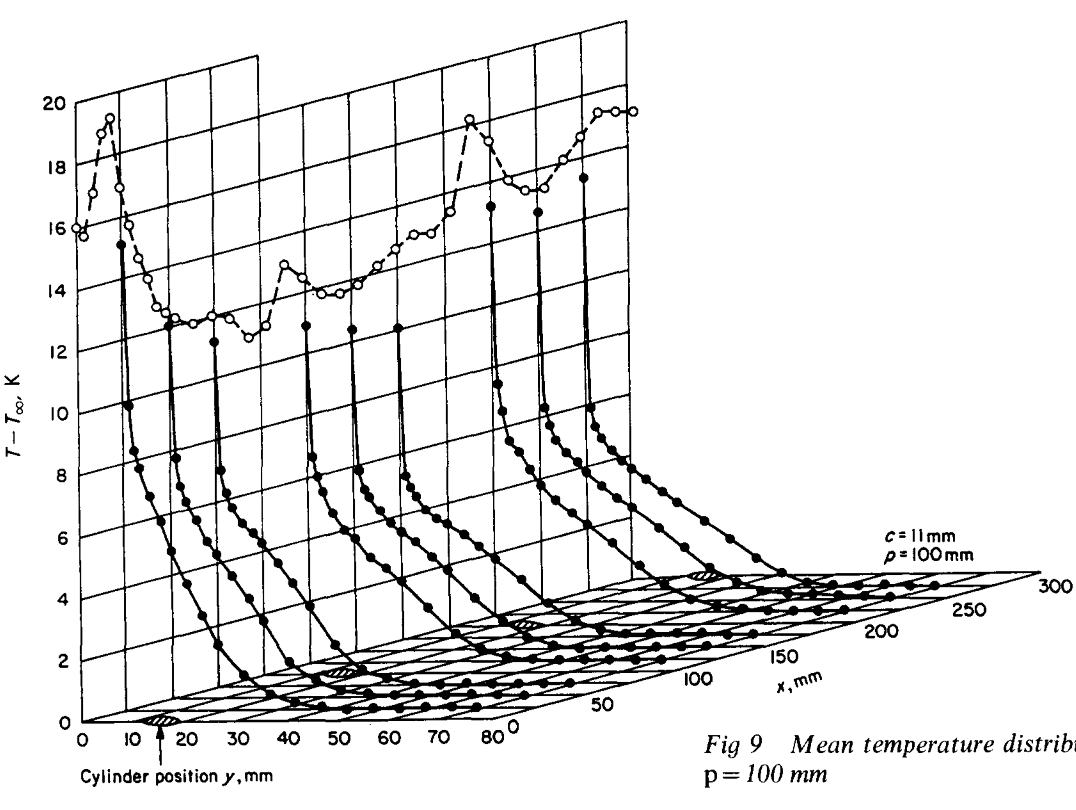

Fig 9 Mean temperature distribution for c = *11 mm and* p = *100 mm*

disturbance. The mechanism for this case is not important from the practical viewpoint of enhancing the flat plate heat transfer. However, to obtain a more general understanding of the control of turbulent heat transfer from the plate, a more detailed experimental study of the flow and turbulence fields for this case is desired.

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BOOK REVIEW

Thermal Sciences 16, Volumes 1 and 2 Proceedings of the 16th Southeastern Seminar

Ed. T. N. Veziroglu

Since its inception in 1965, the Southeastern Seminar on Thermal Sciences has experienced a steady increase in growth and popularity. Attesting to its current national and international reputation, the 16th seminar was attended by researchers from 36 nations, with more than one half of the participants coming from countries outside of the United States. The seminar was held in April 1982, and 71 papers were included in the proceedings.

Following a keynote paper which addresses opportunities in applied heat transfer research, the papers are divided into 17 subject areas which encompass a broad range of research activities. In Volume 1 specific areas include thermophysical properties (4 papers), measurement techniques (6 papers), turbulence (4 papers), forced convection (8 papers), natural convection (4 papers), radiation/convection (4 papers), heat transfer analysis (2 papers), and melting and solidification (4 papers). With the exception of the first two areas and the last area, most of the papers deal with theoretical (numerical) studies of extensions to classical problems, such as flows involving flat plates, spheres, concentric cylinders, parallel plates, and porous media. The section on measurement techniques includes several interesting papers on specialized topics such as temperature measurement during the casting of molten metals and flowrate measurements in multiphase systems.

The largest topical concentration appears in the first two sections of Volume 2, which deal with two-phase flows and heat transfer (6 papers) and two-phase instabilities (5 papers). There is a good blend of experimental and theoretical work, and several of the papers are of high quality. Sections on mass transport (2 papers) and combustion (6 papers) cover a broad range of highly diverse topics. The remaining sections deal with energy systems and include solar energy collection and storage (6 papers), solar energy applications (3 papers), cooling and dehumidifying (3 papers), thermal curing (2 papers), and hybrid energy systems (2 papers). Most of these papers involve models of overall system behavior and generally involve subject matter which is well covered in the existing literature.

The principal deficiency of the volumes relates to their lack of a central focus. Many different topics are covered and, even with the same section, there is often little cohesion between the included papers. In this age of specialization, it would therefore be difficult to recommend purchase by an individual. However, the volumes have a significant number of good papers, which should be accessible to individuals through institutional libraries.

F. P. Incropera
School of Mechanical Engineering,
Purdue University,
West Lafayette
IN, USA

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