CONTROL OF JET IMPINGEMENT HEAT TRANSFER BY A WAKE FLOW BEHIND AN ARRAY OF CIRCULAR CYLINDERS

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Local controllability of jet impingement heat transfer was investigated by immersing an array of circular cylinders within a slot jet. Heat transfer on the central stagnation line was augmented most effectively due to the acceleration and surface-renewal effects when the cylinder array was placed from four to six cylinder diameters upstream of the heat transfer surface in a 'passing' arrangement. The maximal point of heat transfer was shifted, without attenuation, to half the cylinder pitch off the jet axis on both sides by a 'blocking' arrangement whereas the central stagnation-line heat transfer was suppressed very much due to the blocking effect. The mechanism for augmentation of the stagnation-line heat transfer was explained successfully by the surface-renewal model proposed.

Introduction

During the last decade, interest has grown in thermal control of electronic components and equipment from the viewpoint of the operating speed and frequency and the power density in the circuit.

To elucidate the mechanism for enhancement of heat transfer in impinging jets, large-scale organized structures of turbulent jets have been investigated by many researchers.[1,6] Kataoka *et al.*[2-4] have pointed out that the remarkable enhancement of heat transfer at the stagnation point is due to the surface renewal effect of large-scale turbulent eddies impinging on heat transfer surfaces.

It would be very advantageous to further increase the heat transfer artificially in the region of the stagnation point. From the viewpoint of surface renewal, the wake effect of cylindrical obstacles on jet cooling was studied with a view to artificial augmentation (Kataoka *et al.*[4,5]). This is the second-phase work, which makes use of an array of circular cylinders immersed in a slot jet stream for local control of jet impingement heat transfer.

1. Experimental

As shown in **Fig. 1**, the experimental setup was almost the same as the previous one.[3-5] A convergent slot nozzle of PVC plastic resin was used as the jet exit: the shorter side length (characteristic length) $D = 40$ mm, the longer side length $= 150$ mm, and the contraction ratio $= 1/10$. A water jet was discharged horizontally from the convergent slot nozzle into a large volume of stagnant water. The initial turbulence was kept sufficiently low (of the order of 1% or less). The submerged water jet struck normally against a large flat plate (termed 'impingement plate'). The temperature of the water jet was kept equal to that of the receiving stagnant water. The jet Reynolds number was set at 4,500.

The test array consisted of six circular cylinders (diameter $d = 8$ mm, length $= 150$ mm) arrayed in

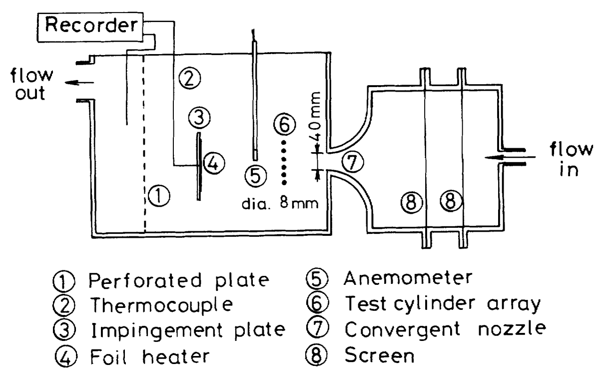

Fig. 1. Experimental setup

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parallel with a pitch of 20 mm and fixed at both ends. The cylinder diameter of the test array was chosen to be of the same order or less than the size of developing vortices in the jet shear layer. The cylinder pitch was also chosen to be of the same order as the jet half-radius. The narrowest gap between two adjacent cylinders was 12 mm. The test array was placed normal to the slot jet stream according to the following two rules of arrangement. As shown in **Fig. 2**, the first arrangement (termed 'passing') was to put the centerline of the central gap of the test array perpendicularly to the flow direction on the jet axis plane, so that the jet stream would 'pass' through the central gap.

The second arrangement (termed 'blocking') was to put the axis of the third cylinder of the test array on the jet axis, so that the jet stream would be blocked at the jet axis by the third cylinder. The impinging jet experiment was also conducted in a 'test array-free' arrangement to obtain reference data for comparison under the same jet-flow condition.

Three parallel strips of SUS foil heater were attached to the impingement plate as the heat transfer surface. The width, length, and thickness of each foil heater were 20 mm, 160 mm, and 50 μm respectively. Local coefficients of heat transfer under the uniform wall heat-flux condition were determined from the temperature difference measured between the hot junction of a C–C thermocouple (wire diameter = 100 μm) attached to the rear surface of the central foil heater and the cold junction immersed in the bulk water. Flow measurements were made by moving a hot-wire I-probe (20 μm-diameter and 5 mm-long Pt wire) along the jet axis in the presence of the impingement plate. Analog signals of the fluctuating velocity detected on the jet axis were sampled at 500 Hz for about 40 s for statistical analysis by a digital computer.

2. Results and Discussion

To compare heat transfer augmentation among the three arrangements on a common basis, as was done in previous papers,[4,5] the local Froessling number was defined using the nozzle exit velocity and the characteristic dimension of the nozzle exit: $Fs_o = Nu/Reo^{1/2}Pr^{1/3}$.

As reported in a previous paper,[2] it was confirmed in advance that in the case of test array-free arrangement the Froessling number Fs_{os} at the stagnation line becomes maximal when the heat transfer surface is positioned at $Z = 6D$ downstream of the jet exit, i.e. when $H/D = 6$.

Figure 3 compares the lateral distributions of local Froessling number between the passing and blocking arrangements. It can be seen that the heat transfer at the central stagnation line, i.e. at $y/D = 0$ is augmented very much in the passing arrangement, especially when $H/D = 2$–6. The maximal Fs_o is approximately 1.8 times the maximal Fs_o in the test array-free arrangement when the test array is placed in an axial position from four to six cylinder diameters upstream of the heat transfer surface, i.e. when $Xc/d = 4$–6.

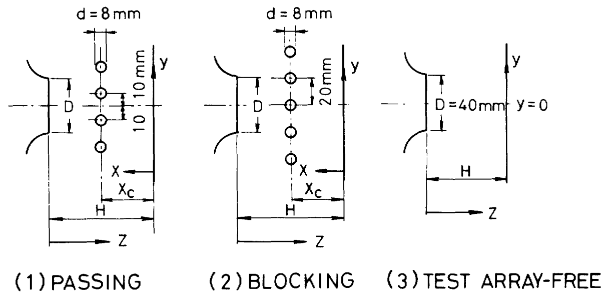

Fig. 2. Arrangement of test array and the coordinates used

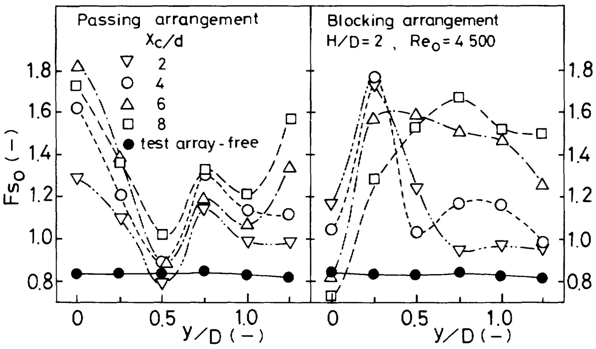

Fig. 3a. Lateral distributions of local Froessling number

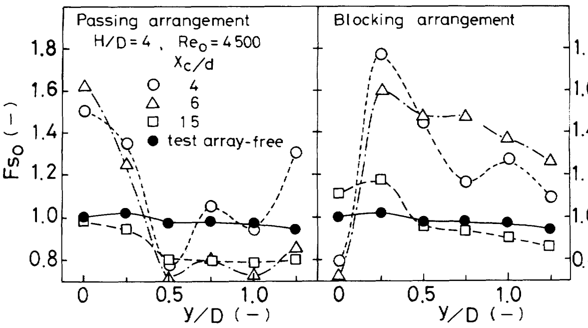

Fig. 3b. Lateral distributions of local Froessling number

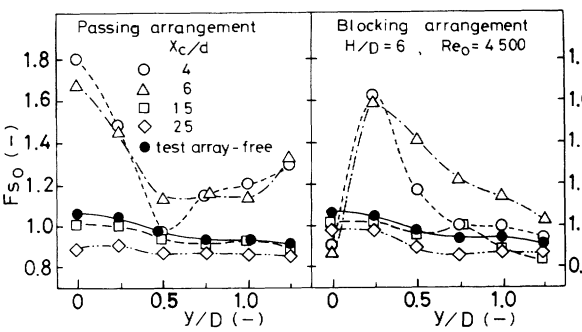

Fig. 3c. Lateral distributions of local Froessling number

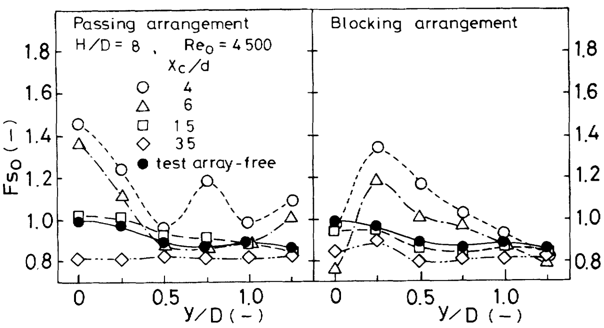

Fig. 3d. Lateral distributions of local Froessling number

However, the central stagnation-line heat transfer is suppressed considerably in the blocking arrangement. The local Froessling number in this arrangement becomes maximal at $y/D = \pm 0.25$ when $Xc/d =$ 4–6. The maximal value is almost as high as that in the passing arrangement. This implies that the maximal heat transfer at the central stagnation line in the passing arrangement can be shifted without attenuation to half the cylinder pitch off the jet axis on both sides by the blocking arrangement.

The great augmentation caused at $y/D = 0$ in the passing arrangement and at $y = \pm 0.25$ in the blocking arrangement must be due firstly to the acceleration effect of the corresponding gaps in the test array. If the test array is placed more than $15d$ from the heat-transfer surface, i.e. if $Xc/d > 15$, the lateral distribution curves of local heat transfer in both the passing and blocking arrangements become very similar to, and even slightly lower than, that in the test array-free arrangement. This is due to downstream decay of the accelerated flow and its turbulence. The effective augmentation attained when $H/D \leqq 6$ is also limited in the central impingement region. In the passing arrangement the cylinder gaps located at $y/D = \pm 0.5$ do not provide very much augmentation effect, due to the crossflow effect. It can be considered that the local controllability of heat transfer distribution depends on the nozzle width, the nozzle-to-plate distance, the cylinder pitch and diameter, and the test array position.

Figure 4 shows the variation of stagnation line heat transfer with distance of the test array from the heat-transfer surface in the passing arrangement. It is seen from the figure that stagnation-line heat transfer is augmented effectively if the test array is positioned between $Xc/d = 4$–6, regardless of where the heat-transfer surface is placed within the range $Z/D = 2$–6. The effect of wake flow is amplified between the test array and the heat-transfer surface when Xc/d is small, but is attenuated with increasing Xc/d.

Figure 5 shows the axial variation of the time-averaged velocities measured on the jet axis in the presence of a flat plate placed at $Z/D = 6$. The solid

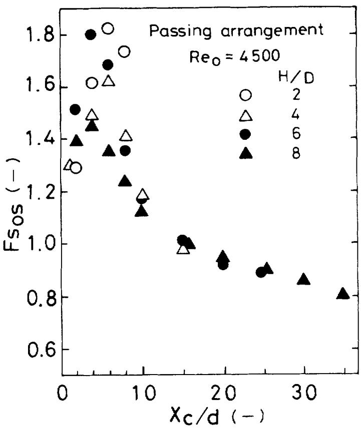

Fig. 4. Variation of central stagnation-line heat transfer with axial distance of test array from heat-transfer surface for the 'passing' arrangement

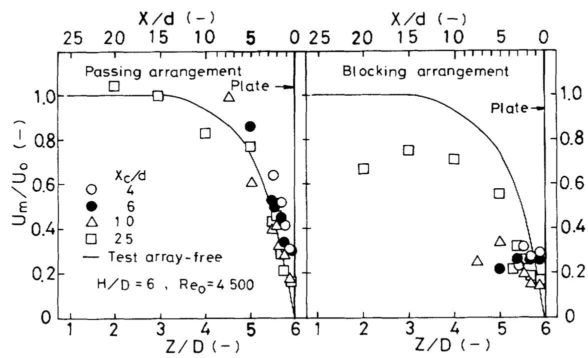

Fig. 5. Axial variation of time-averaged centerline velocity

lines given in the figure indicate the experimental curve in the test array-free arrangement. The arrival velocity as the characteristic velocity scale was defined at an axial station 20 mm upstream of the heat-transfer surface on the central jet axis.

As mentioned above, it can be seen from the figure that the passing arrangement causes an accelerating effect owing to the sudden contraction of the jet flow passage between the two central cylinders, so that the arrival velocity becomes much larger than in the other two arrangements. The blocking arrangement has arrival velocities lower than those of the test array-free arrangement. The augmentation of the impingement heat transfer may be due secondly to the effect of intermittent surface renewal action on the boundary layer developing on the heat transfer surface.

The **'surface renewal' model** proposed in previous papers[2,3] was also tested for this flow system. The 'surface renewal parameter' is defined as the product of intensity and frequency:

$$SR = \frac{\sqrt{\bar{u}^2}}{Um} \cdot \frac{fD^2}{\nu} \tag{1}$$

The intensity of surface renewal action can be assumed to be the turbulence intensity, $\sqrt{\overline{u^2}}$, of the arrival velocity. The frequency of the surface renewal event can be determined by making a statistical analysis[2,3,5] of the fluctuations of the arrival velocity as follows.

Figure 6 shows time-traces of the instantaneous arrival velocities measured at $X = 20$ mm on the centerline of the approaching stream. The determination of the threshold value, shown by the horizontal lines in the figure, may be arbitrary. The value was taken to be 90% of the RMS value of the arrival velocity fluctuations when $H/D = 6$ in the test array-free arrangement. It can be assumed that the boundary layer in the central impingement region undergoes surface renewal action once if u goes one or more times beyond either of the two threshold lines during a time period for successively crossing the abscissa twice, i.e. $u = 0$. The former condition (to check whether u goes beyond one of the two threshold lines) can detect large-scale motions with large kinetic energy. The latter condition (to check whether u crosses the abscissa twice) is necessary as a backup condition so as not to count small-scale structures with high-frequency components. It can be considered that the surface renewal action occurs by a sweep-like motion when u goes beyond the upper threshold line and by an ejection-like motion when u goes beyond the lower one. That is, a sweep-like motion with a rapidly approaching fluid washes away the boundary layer along the heat transfer surface whereas an ejection-like motion with a slowly approaching fluid strips the boundary layer from the heat-transfer surface. The surface renewal frequency was calculated by counting how many times u crosses the threshold lines under these conditions.

It has been confirmed that the frequencies as well as the turbulence intensities become higher in the passing arrangement than in the other two arrangements.[5] It is seen from **Fig. 7** that the surface renewal parameter remains very high even near the flat plate in both the test array arrangements, especially when $Xc/d = 4$–6. This may suggest that large-scale structures are produced between the test array and the heat-transfer surface without decay, rather than with amplification of turbulence. This effect can also be considered to occur at $y/D = \pm 0.25$ in the blocking arrangement. It has been observed by a hydrogen bubble method that some large-scale circulating vortex flow occurs intermittently between the test array and the impingement plate when $Xc/d = 4$–6. It can be concluded that the augmentation of impingement heat transfer due to the turbulence effect is directly related to the surface renewal parameter evaluated near the heat-transfer surface.

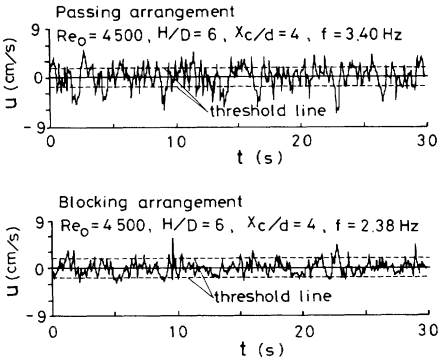

Fig. 6. Time-traces of instantaneous arrival velocities

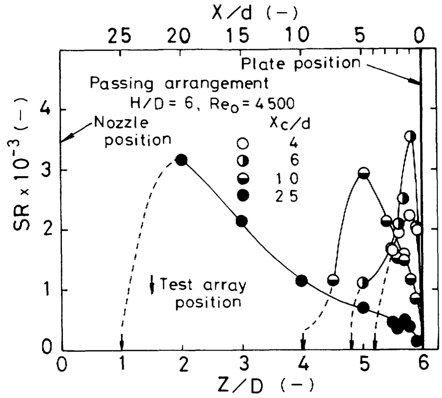

Fig. 7a. Axial variation of surface renewal parameter in the presence of flat plate when $H/D = 6$

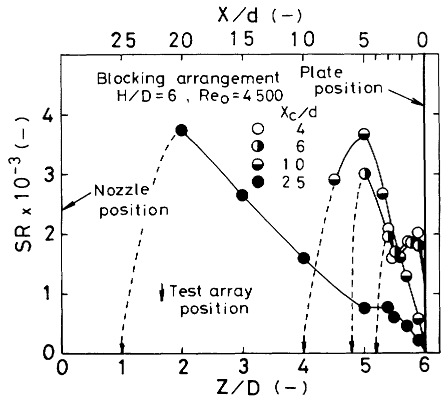

Fig. 7b. Axial variation of surface renewal parameter in the presence of flat plate when $H/D = 6$

Conclusion

(1) When the test array is placed in the region $4 \leqq Xc/d \leqq 6$, the passing arrangement can augment the heat transfer around $y/D = 0$ by about 1.8 times the maximal heat transfer obtained in the test array-free arrangement.
(2) The blocking arrangement in the region $4 \leqq Xc/d \leqq 6$ shifts the position of maximal heat transfer to approximately half the cylinder pitch off the jet axis without attenuation of the maximum value.
(3) Artificial augmentation of jet impingement heat

transfer depends increasing the surface renewal parameter as well as accelerating the arrival velocity.

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Nomenclature

D	= Shorter-side length of slot nozzle exit	[m]
d	= Cylinder diameter	[m]
Fs_o	= Local Froessling number $= Nu/Reo^{1/2}Pr^{1/3}$	[—]
Fs_{os}	= Froessling number at central stagnation line	[—]
f	= Frequency of surface renewal	[Hz]
H	= Spacing between jet exit and heat-transfer surface	[m]
h	= Local coefficient of heat transfer	[W/m²K]
Nu	= Local Nusselt number $= hD/\kappa$	[—]
Pr	= Prandtl number	[—]
Reo	= Jet Reynolds number $= UoD/\nu$	[—]
SR	= Surface-renewal parameter defined by Eq. (1)	[—]
Tuo	= Turbulence intensity $= \sqrt{\bar{u}^2}/Uo$	[—]
t	= Time	[s]
Um	= Jet centerline velocity	[m/s]
Uo	= Jet exit velocity	[m/s]
u	= Fluctuation of arrival velocity	[m/s]
X	= Axial distance measured in the upstream direction from heat-transfer surface	[m]
Xc	= Distance between test array and heat-transfer surface	[m]
y	= Lateral distance along heat-transfer surface from geometrically central stagnation line	[m]
Z	= Axial distance measured from jet exit	[m]
κ	= Thermal conductivity of fluid	[W/mK]
ν	= Kinematic viscosity of fluid	[m²/s]

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