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An experimental investigation of a block moving back and forth on a heat plate under a slot jet

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Abstract

This experimental study aims to investigate heat transfer phenomena caused by a moving block under a jet flow. The experimental apparatus includes three main systems of a jet flow, periodical movement and heating control. The work fluid is air and the data runs are performed for jet Reynolds numbers and speeds of moving block. The comparison between experimental and numerical results show good consistence, and the destruction of boundary layers discovered by previous authors' numerical results is validated. The enhancement of heat transfer rate is generally accompanied with the increment of jet Reynolds number and speed of moving block.

Keywords: Moving boundary; Convection heat transfer; Experimental

1. Introduction

Up to now, numerous methods have been proposed to enhance heat transfer rate of a heat body. These methods mainly include the passive and the active methods. The examples of the former one are treated surfaces and swirl flow devices, while the examples of the active methods are surface vibration, fluid vibration, injection and suction which were summarized and reviewed in detail by Bergles [1,2]. A jet impingement which has high heat transfer rate is one of the above active methods and widely used in the cooling apparatus for high heat flux system such as electric cooling and turbine blade cooling. However, accompanying with the progress of semiconductor technology, the smaller and more compact device is produced indefatigably. The heat generated by the new device is always several times of the former one and becomes the main defect of the failure of device. As a result, how to increase the heat transfer rate of the jet impingement becomes a very important issue.

In the past, Mujumdar and Douglas [3], Martin [4], and Jambunathan et al. [5] reviewed the contemporary literature. Marple et al. [6] used a flow visualization technique to study a confined water jet impingement and observed the laminar flow for jet Reynolds number up to 2300. Chou and Hung [7] studied fluid flow and heat transfers of a confined slot jet numerically and found that the Nusselt number at the stagnation line was proportional to jet Reynolds number in a 0.5 power and the ratio of separation distance to jet width in a −0.17 power. Chakroun et al. [8] investigated heat transfer of a round air jet impinging normally on a heated rough surface and found that the local and average Nusselt numbers of a rough surface were larger than those of a smooth surface by 8.9% to 28%. Chung et al. [9] investigated heat transfer characteristics of an axisymmetric jet impinging on a rib-roughened convex surface. The average Nusselt numbers on the rib-roughened convex surface were more than those on the smooth surface by 14% to 34%. Besides, in several experimental studies, an extra mechanism in the jet impingement system was

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Nomenclature

A	area [m^2]	*Greek symbols*	
h	heat transfer coefficient [w m^{-2} K^{-1}]	θ	subtending angle [rad]
H	distance from wind tunnel exit to heat plate [m]	ν	kinematic viscosity [m^2 s^{-1}]
I	current [A]	w	angular velocity [rad s^{-1}]
k	thermal conductivity [w m^{-1} K^{-1}]		
$\overline{Nu}$	average Nusselt number	*Subscripts*	
Q	heat energy [w]	b	block
R	length of linkbar (Figs. 3 and 7) [m]	h	heat plate
T	dimensional temperature [K]	in	input
v	fluid velocity [m s^{-1}]	j	jet
V	voltage [V]	lose	lose
V_b	dimensionless velocity of moving block [$v_b \cdot v_j^{-1}$]	s	stagnation
w	width of wind tunnel exit [m]	t	theoretical value
Δy	thickness of basswood [m]		

installed to enhance the heat transfer rate of the jet impingement. Zumbrunnen and Aziz [10] investigated an intermittent water jet impinging on a constant heat flux surface experimentally, and found that the enhancement due to the intermittent flow only occurred as the frequency of the intermittence was sufficiently high enough. Haneda et al. [11] enhanced heat transfer rate of a jet impingement by inserting a suspended cylinder between the jet exit and the heat plate. The mechanically oscillatory motion of the cylinder vibrated the flow field, and the maximum Nusselt number around the stagnation point was augmented by about 20% compared to that without the insertion of a cylinder.

From the above literature, the increment of heat transfer rate of the jet impingement seems to have limitation in spite of installing any extra equipment. The reason could be the stillness of velocity and thermal boundary layers on the heat plate. Consequently, as the huge enhancement of heat transfer rate is expected, the boundary layers mentioned above are necessarily destroyed.

Based upon the results of Fu et al. [12] that a thin block moving back and forth along a heat plate under an impinging jet could improve the heat transfer rate of the heat plate remarkably. In this study the boundary layer on the heat plate was destroyed by the moving thin block and the new one reformed immediately after the moving thin block which was the main mechanism to enhance the heat transfer rate hugely. However, the study [12] was purely numerical analyses and extremely short of relating experimental studies to verify its accuracy and availability.

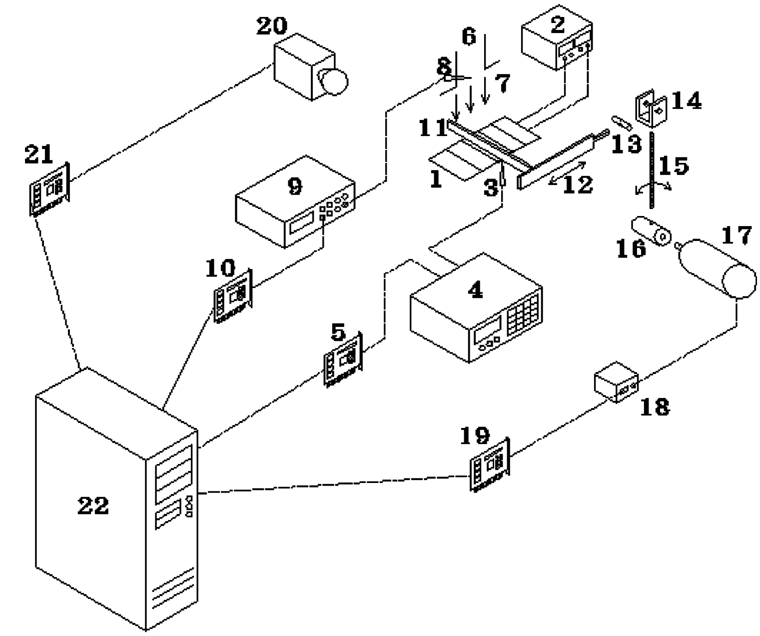
Therefore, a further investigation of this issue will be carried out by an experimental method. The apparatus used in the experimental method includes three main systems of a jet flow, periodical movement and heating control. The jet flow system is made of a fan and a wind tunnel and provides necessary velocity distribution. The periodical movement system maintains periodical motion of a thin block moving back and forth on the heat plate which plays a leading role. Due to the limitation of the thin block speed, a jet Reynolds number is smaller than 1500. The heating control system is combined with heat plates, thermalcouples and power supply, and is to maintain constant temperature condition on the heat plates which is consistent with the thermal condition used in the previous numerical study [12]. Since a stepping motor is used to drive a moving bar periodically, the acceleration and deceleration motions of the moving bar during the initial and final stages of periodical movement are unavoidable. The average velocity during the periodical movement is then utilized. Comparing both the numerical and experimental results, the consistence gives good agreement.

2. Physical model and experimental procedure

In order to simulate the physical model adopted in the pervious study [12] as consistently as possible, the relating experimental apparatus conducted in this study is shown in Fig. 1, and the test section (1, 3 and 7) is equivalent to the physical model of the previous study [12].

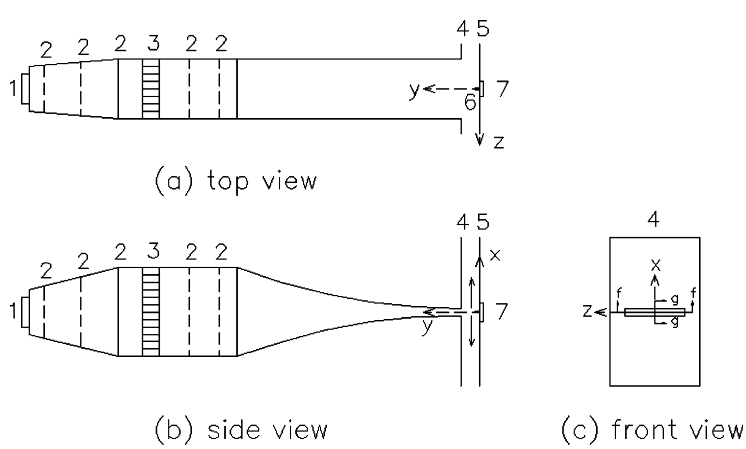
Air streams are sucked by a fan (without being indicated in Fig. 1) and rectified by a wind tunnel (6). Then the air streams could be regarded as a jet flow (7) and blow uniformly and stably at the exit of the wind tunnel. For rectifying the air streams, the honeycomb (3) and screens (2) are then set at appropriate positions in the wind tunnel shown in Fig. 2.

The detailed figure of the periodical movement system (11–17) shown in Fig. 1 is renewebly indicated in Fig. 3. A linkbar (7) of which one side is fixed on a stepping motor (8) and the other side is connected with a connector (6). Another linkbar (5) is installed perpendicularly to the linkbar (7) and adjoins the connector (6), and a thin moving block (2) is perpendicularly inserted in the linkbar (5). For keeping the movement of the moving block (2) limited in the heating plates (1) stably and periodically, the barriers (3) and bearings (4) are used. As a result, when the stepping



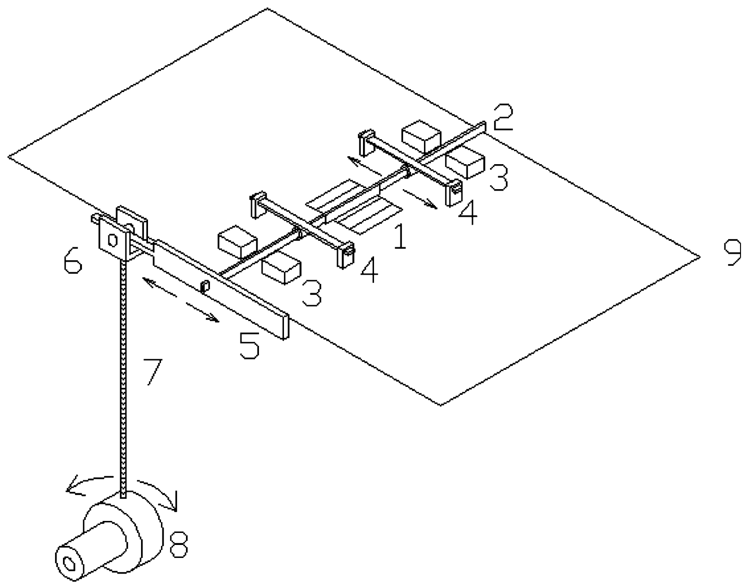
(2) power supply
(3) thermalcouples
(4) temperature indicator
(5) data acquisition card
(6) wind tunnel
(7) jet flow
(8) hot wire
(9) hot wire animometer
(10) data acquisition card
(12) ~(16) periodical movement system
(17) stepping motor
(18) motor driver
(19) interface card
(20) camera
(21) data acquisition card
(22) personal computer

Fig. 1. Experimental apparatus.



(1) fan
(2) screen
(3) honey comb
(4) confined plate
(5) impinging plate
(6) moving block
(7) heat plates region

Fig. 2. Wind tunnel.



(1) heat plates region
(2) moving block
(3) barrier
(4) bearing
(5) linkbar
(6) connector
(7) linkbar
(8) stepping motor
(9) impinging plate

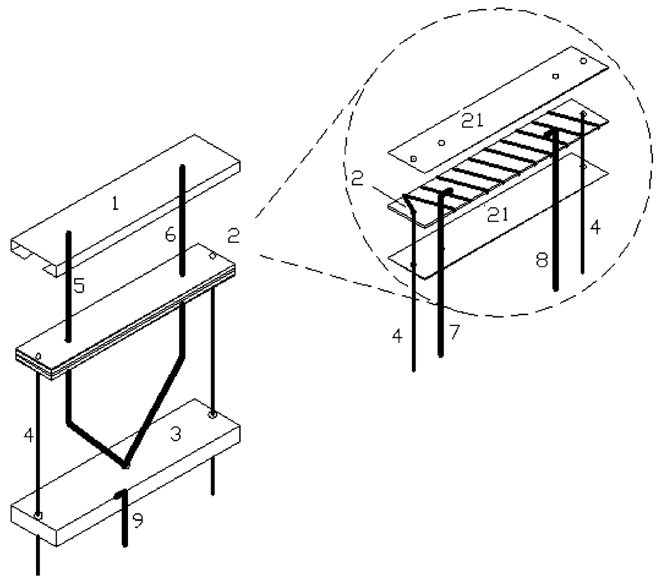
Fig. 3. Periodical movement system.

motor (8) revolves a small angle back and forth periodically, by way of the linkbar (7), connector (6) and linkbar (5) , the moving block (2) could move short but approximate horizontal distance back and forth on the heat plates (1). Since the moving block and heat plates are solid materials. For avoiding damage caused by the direct contact between the moving block and heat plates, the contacting side of the moving block is made of soft material.

The heating control system is combined with heat plates, thermalcouples and power supplies. The heat plates have five pieces which are separately controlled by an individual power supply in order to maintain surface temperature of each heat plate at a constant temperature condition. The heat plates set on the left and right most sides are used as the guard plates which protect the heat energy of the other three heat plates set in the central region to dissipate from the both most sides. As a result, the data reduction of the heat energy of the central heat plate carried away by the jet flow mentioned above becomes simple and accurate.

Shown in Fig. 4, a fine Ni–Cr line (4) 0.1 mm in diameter is used as a heater during electric power applying on both ends of the line winds uniformly and tightly around an insulator of mica (2) of which the length, width and thickness are 40, 8 and 0.4 mm, respectively. Two thermalcouples (7 and 8) are installed on the upper side of the mica and prohibited to touch the Ni–Cr line. Afterward, two pieces of thin mica (21) used as an electric insulator 0.1 mm in thickness cover both sides of the mica winded by the Ni–Cr line to prevent the direct contact between the Ni–Cr line and thin copper plate (1). Since the jet flow directly impinges on the flat heat plates assumed in the pervious study. A piece of thin copper plate (1) then covers all materials mentioned above and the upper side of copper plate impinged by the jet flow should be adjusted as smoothly as possible. Two pairs of thermalcouples (5 and 6) are stuck on the inner side of the upper side of copper plate to indicate the surface temperature of heat plate.

A block of basswood (3) with 40, 10 and 3 mm as length, width and thickness respectively is used as a thermal insulator to prevent the dissipation of heat energy from the bottom side of heat plate. Its upper side is stuck tightly on the bottom side of heat plate mentioned above and a pair of thermalcouples (9) are installed on the bottom side of basswood. Utilizing the temperatures measured from the mica (7 and 8) and the bottom side of basswood (9), the quantity of heat energy of the heat plate transferring to the basswood by heat conduction could be calculated which means the heat loss of heat plate from the bottom side could be obtained. For flow visualization, a smoke-wire techniques is used to observed variations of flow field which compare with the results of numerical study [12] qualitatively.



(1) thin copper plate	(4) Ni-Cr line
(2) mica	(5)~(6) thermalcouples on the bottom side of thin copper plate
(21) mica	(7)~(8) thermalcouples in mica
(3) basswood	(9) thermalcouples on the bottom side of basswood

Fig. 4. Heat plate.

Each experimental data run includes three measurements and a brief outline is given as follows:

(1) The measurement of jet flow. Start the fan and use a hot wire anemometer to measure a velocity profile at the exit of wind tunnel. The distance from the exit of wind tunnel to the heat plate is 50 mm. The velocity profile of f–f section shown in Fig. 2 measured at the exit is indicated in Fig. 5. Except the regions near both walls, the velocity distribution on most central region in which the moving block moves back and forth is flat, therefore, the assumption of two dimensional flow made in the previous study [12] is approximately consistent with this experimental study. The jet Reynolds number Re_j of jet flow is defined as follows:

$$Re_j = \frac{v_j w}{\nu} \tag{1}$$

v_j is the fluid velocity measured at the center of wind tunnel exit, w is the width of the wind tunnel exit and ν is the kinematic viscosity of the working fluid.

Adjust the revolution of fan and a designed fluid velocity could be obtained.

(2) The measurement of periodical movement. A stepping motor having resolution of 36000 steps per revolution is used. The linkbar (7) shown in Fig. 3 has a length of 200 mm and the width of the exit of wind tunnel w is 20 mm which is the one-way moving distance of the moving block. The theoretical velocity of moving block v_{bt} could then be calculated by the following equation:

$$v_{bt} = R\omega\cos\theta \tag{2}$$

R is the length of linkbar (7) (=200 mm), ω is the angular velocity of stepping motor and θ is the subtending angle between circumferential velocity of linkbar and velocity of moving block.

As a result, the maximum clearance between the moving block and heat plates due to revolution of the linkbar is about 0.27 mm. For filling the clearance, one kind of soft material which contacts with the heat plates directly has the length to be slightly longer than 0.27 mm and is installed on the bottom side of moving block.

Realistically, the moving distance 20 mm including acceleration and decelerating regions converted into the steps of stepping motor is about 600 steps or 0.11 rad in minute circumferential angle. Therefore, the velocity of moving block driven by the stepping motor maintained at a constant magnitude is impossible. The distribution of the velocity of moving block on the heat plates region of 20 mm is shown in Fig. 6 for a certain case. Conveniently, the average velocity of moving block v_b used in the data reduction is obtained by that the distance which is summation of the moving block moving back and forth in 20 mm length 24000 times divides the waste time. Shown in Fig. 6,

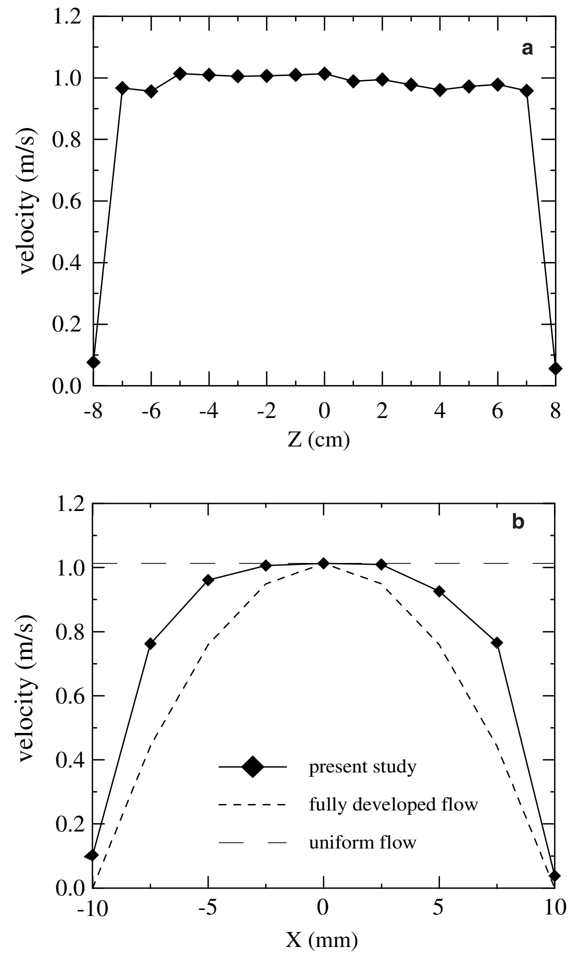


Fig. 5. Velocity profiles on the exit of jet flow for $Re_j = 1386$.

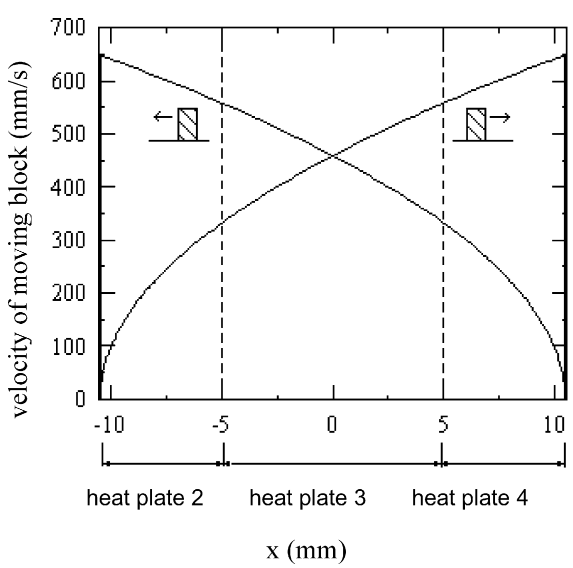


Fig. 6. Distribution of velocity of moving block versus location of heat plates.

the moving block running through the heat plate 3 being used to calculate heat transfer rate from a heat plate almost keeps constant speed which is approximately equal to the average velocity mentioned above. Then the average velocity could be approximately regarded as a constant velocity which is assumed in the previous study [12].

(3) The measurement of heat transfer rate of a heat plate. The heat plates mentioned above are wound tightly and uniformly by a fine Ni–Cr line and the heat plates arranged on both most sides are used as guard heaters, then as electric power is applied on heat plates, the uniform temperature distribution on the central heat plates could be obtained. The difference of two pairs of thermalcouples shown in Fig. 4 installed on the central three heat plates are not larger than 0.1 °C. As a result, the condition of the constant temperature on heat plate which is assumed in the previous study [12] could almost be satisfied.

The heat transfer rate of the heat plate 3 (middle heat plate) could be calculated by the following equations:

(a) Total input heat energy

$$Q_{in} = I \times V \tag{3}$$

Q_{in} is the total input heat energy (w), I, V is the current [I] and voltage [V] of electric power, respectively.

(b) Heat energy dissipation Q_{lose} from the basswood installed the bottom side of heat plate.

$$Q_{lose} = K_b \times A_b \times \Delta T/\Delta y \tag{4}$$

K_b is the thermal conductivity of basswood ($\doteq$0.055 $\mathrm{w\,m^{-1}\,K^{-1}}$), A_b is the area of basswood ($0.04\,\mathrm{m} \times 0.01\,\mathrm{m} = 4 \times 10^{-4}\,\mathrm{m}^2$), ΔT is the temperature difference between the upper and bottom sides of basswood (K) and Δy is the thickness of basswood (=0.003 m).

(c) Calculation of average Nusselt number $\overline{Nu}$ of heat plate 3.

$$\overline{Nu} = \frac{hw}{k} = \frac{Q_{con}}{A_h \Delta T} \cdot \frac{w}{k} \tag{5}$$

h is the convection heat transfer coefficient [$\mathrm{w\,m^{-2}\,K^{-1}}$], w is the width of jet flow exit [=0.02 m], k is the thermal

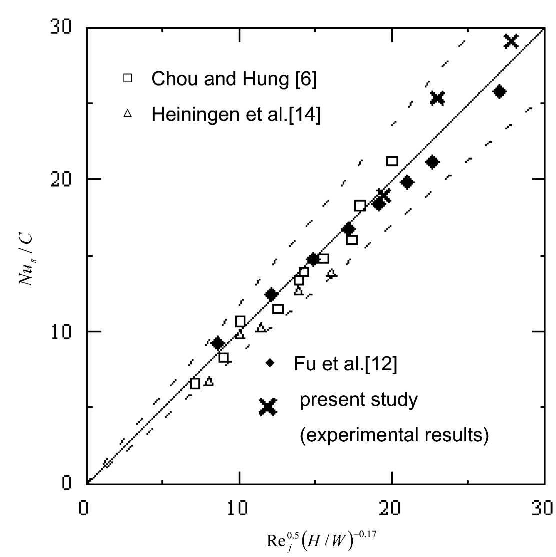


Fig. 7. Comparison of the results of present study with previous studies.

conductivity of air [=0.025 w m^{-1} K^{-1}], Q_{con} is the convection heat transfer energy ($= Q_{in} - Q_{lose}$), A_h is the area of heat plate 3 [=10^{-4} m^2] and ΔT is the temperature difference between surface of heat plate 3 and inlet air [K].

Total pairs of thermalcouples used in the experimental system are 26, the scanning speed of temperature indicator for scanning the transient variations of total thermalcouples can not catch with the speed of moving block. For convenience, enough time is taken to measure the total thermalcouples and an average temperature of each thermalcouples is used for calculating the heat transfer rate of each experimental run.

Based upon the measurements mentioned earlier, experimental procedures are briefly indicated as follows:

(a) Start the fan to provide a designed jet velocity v_j and calculate the corresponding jet Reynolds number Re_j.
(b) Start individual power supply to apply approximate power Q_{in} composing of current and voltage to each heat plate.
(c) Adjust each power supply carefully to make the temperature of each heat plate to be the same magnitude and the temperature difference among the heat plates is not larger than 0.1 °C.
(d) Take the necessary data to calculate the heat transfer rate of heat plate 3 under a pure jet flow condition.
(e) Start the movement system to obtain a designed velocity of the moving block v_b, the experimental condition becomes transient. For economizing run-

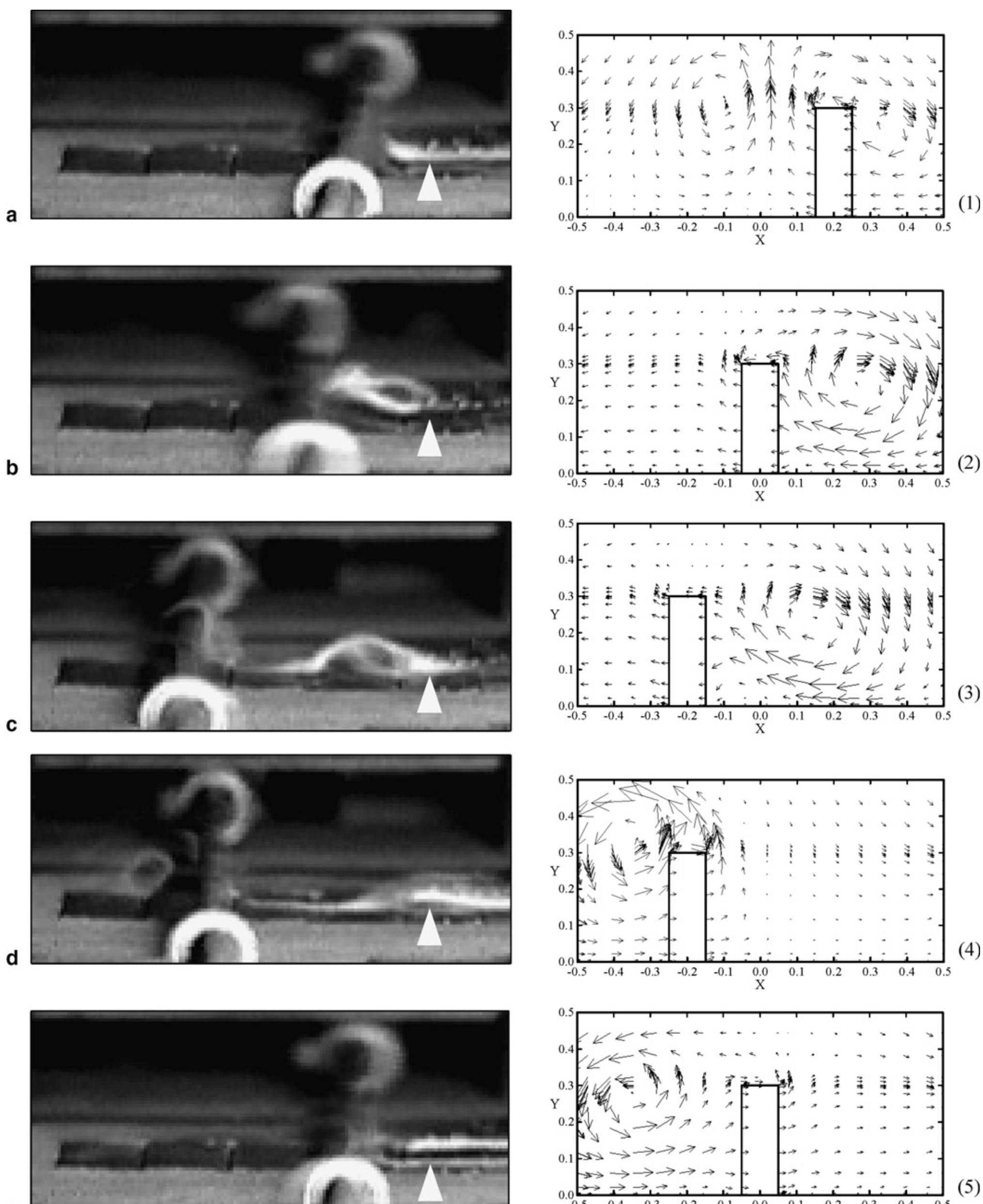


Fig. 8. Comparison of the flow visualization with numerical results for $Re_j = 1054$, $V_b = 0.30$.

ning time, increase an approximate quantity of power of individual heat plate according to the numerical results of the previous study [12] first. After the temperature variation of each heat plate becomes slight, the fine adjustment of power supply is executed sequentially until the temperation difference condition is satisfied.

(f) Take the necessary data to calculate the heat transfer rate of heat plate 3 of a moving block under a jet flow condition.

(g) Change the parameters of v_j, v_b and Q_{in}, and repeat the above procedures.

As for the uncertainty analysis, the relative uncertainty proposed by Kline [13] being used to analyze the results and the uncertainties for Nusselt and Reynolds numbers are about 7% and 5%, respectively.

3. Results and discussion

In the situation of confined impinging system, the relationship between the Nusselt number Nu_s at stagnation point and jet Reynolds number Re_j could be expressed as the following equation:

$$Nu_s = cRe_j^{0.5}(H/w)^{-0.17} \tag{6}$$

$$c = 0.574$$

The solid line shown in Fig. 7 is obtained from Eq. (6), both dashed lines are 15% deviation from the solid line. The previous studies of Chou and Hung [7] Heiningen [14], and both numerical [12] and experimental results of the authors' studies are consistent well with the results of Eq. (6).

A flow visualization is shown qualitatively in Fig. 8. A fine metal line used as a heater and applied by paraffin is set at a top angle of a white triangle shown in Fig. 8. The fine metal line can generate white smoke within a short time when the electric power is on. The figures of (a)–(e) are experimental results and the figures of (1)–(5) are obtained by the numerical method. Shown in (a) as the moving block moves toward the left, the fluid on the right side of moving block is sucked by the moving block, the fluid then flows toward the left which causes white smoke also flowing toward the left. The direction of vectors of fluid velocities close to the wall shown in (1) is to the left which is consistent with the result of (a). When the moving block moves toward the left further shown in (b), the white smoke forms a circulation zone on the right of the moving block, similarly the vectors of fluid velocities in (2) indicate a circulation region on the right of the moving block. In (c), the moving block moves to the left most side, the distance between the moving block and fine metal line becomes large, and a part of white smoke close to the wall continuously moves toward the left and a part of white smoke forms a larger circulation zone. These results are similar to the numerical results shown in (3). In (d), the moving block starts to move toward the right and the fluids are pushed by the moving block. As a result, the white smoke flows to the right side of fine metal line, and the remnant white smoke from the above motion (c) appears on the left side of the moving block. In the meanwhile, the direction of the corresponding fluid velocities is also toward the right and indicated in (4). The moving block continuously moves to the right which results in both white smoke and fluid velocities flowing to the right and being shown in (e) and (5), respectively.

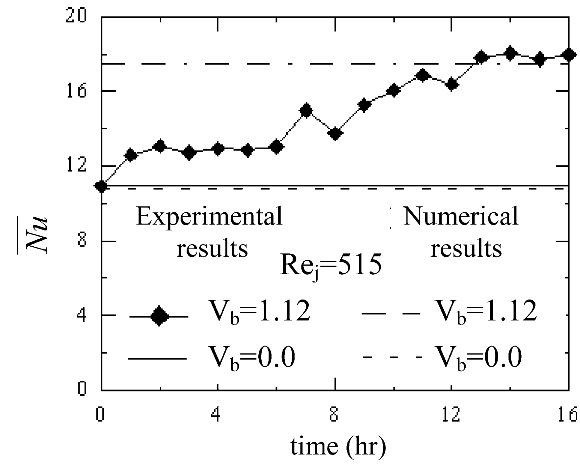


Fig. 9. Variations of average Nusselt numbers with time for $Re_j = 515$, $V_b = 0.0$ and 1.12 conditions.

Variations of the experimental and numerical results obtained from [12] with time for $Re_j = 515$, $V_b = 1.12$ and 0.0 are shown in Fig. 9. During the experimental run, a minute increment of electric power is necessary, and the experimental run consumes much time to reach the stable state which means the variation of the heat surface temperature is smaller than 0.1 °C. During the experimental run, the electric power is sometimes increased excessively which causes the monotous increment of the variations of $\overline{Nu}$ to be difficult. Due to the existence of

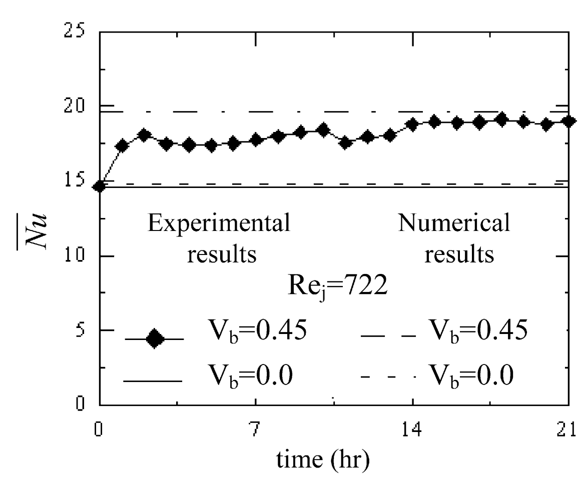


Fig. 10. Variations of average Nusselt numbers with time for $Re_j = 722$, $V_b = 0.45$ and 0.0 conditions.

destruction of thermal and velocity boundary layers by a moving block, from the running of moving block the $\overline{Nu}$ is always larger than that of the situation without moving block ($V_b = 0$). Finally both experimental and numerical results become consistent, and the enhancement of heat transfer rate is about 60%.

Fig. 10 indicates the time development of both experimental and numerical results for $Re_j = 722$, $V_b = 0.45$ and 0.0 situations. Similarly, the experimental run takes up much time and the experimental and numerical results are gradually close to the same value. The enhancement of heat transfer rate is about 45%.

Fig. 11 indicates the time development of both experimental and numerical results for $Re_j = 1054$ and $V_b = 0.31$ and 0.0 situations. Accompanying with the decrement of V_b, the enhancement generally decreases and is about 23%.

Shown in Figs. 12 and 13, there are the variations of $\overline{Nu}$ of experimental results with time for $Re_j = 1262$, $V_b = 0.26$ and 0.0 and for $Re_j = 1471$, $V_b = 0.22$ and 0.0, respectively.

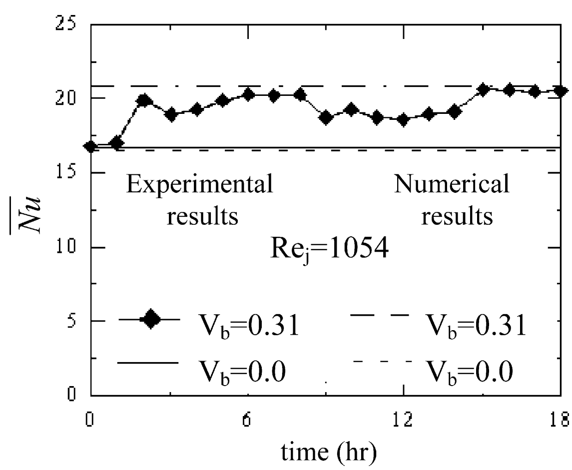


Fig. 11. Variations of average Nusselt numbers with time for $Re_j = 1054$, $V_b = 0.31$ and 0.0 conditions.

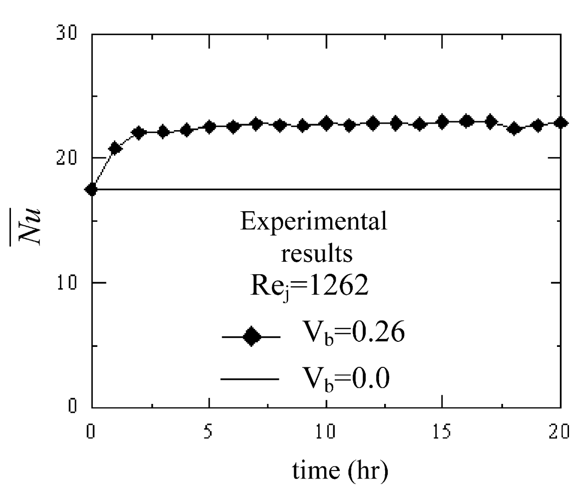


Fig. 12. Variations of average Nusselt numbers with time for $Re_j = 1262$, $V_b = 0.26$ and 0.0 conditions.

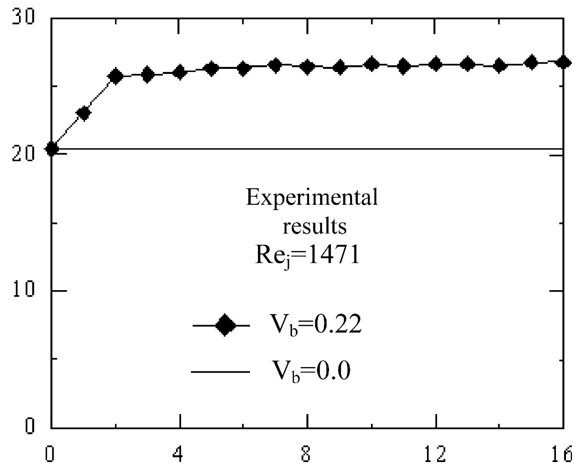


Fig. 13. Variations of average Nusselt numbers with time for $Re_j = 1471$, $V_b = 0.22$ and 0.0 conditions.

The jet Reynolds numbers are larger in these situations, the criteria of convergence of numerical calculation is hardly satisfied which causes the numerical results to be absent. As the moving block starts to move, the heat transfer rate is accompanied with increment in both situations. The enhancements of heat transfer rate are about 30%.

4. Conclusions

This experimental study employs a moving block to enhance heat transfer rate of a heat plate on which the moving block moves back and forth. Different combinations of jet Reynolds numbers and moving block velocities are taken into consideration and the conclusions are summarized as follows:

(1) The enhancement of heat transfer rate caused by the destruction of velocity and thermal boundary layers is validated experimentally.
(2) The larger moving block velocity is, the more remarkable enhancement of heat transfer rate becomes.
(3) Due to direct contact between the moving block and heat plate, time consuming is unavoidable.
(4) Good agreement is found between the experimental and numerical results.

Acknowledgement

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