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# Heat Transfer Enhancement of Natural Convection and Development of a High-Performance Heat Transfer Plate*

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A new enhancing technique was developed for natural convection adjacent to a vertical, heated plate. In order to enhance the heat transfer, V-shaped plates of which the edges faced upstream, were attached to a vertical plate in a staggered layout. These plates work not only as an extended surface but also as a heat transfer promoter. The overall heat transfer coefficients of the above enhanced surface were measured and compared with a nontreated, flat surface and a conventional finned surface. The results show that the highest heat transfer performance achieved is for the experimental surface and that the heat transfer coefficient is 40% higher than that of a conventional finned surface with the same total surface area and fin height.

***Key Words***: Heat Transfer, Heat Transfer Enhancement, Natural Convection, Extended Heat Transfer Plate, Heat Exchanger, Diverter Plate

## 1. Introduction

A number of techniques have been developed to improve the heat transfer performance of forced convection, and various devices such as roughness elements, turbulence promoters, twisted tapes, rifled tubes, interrupted fins, and so on, have been utilized to enhance forced convection heat transfer. These techniques and devices have been successfully applied to practical heat exchangers, and have contributed to reduce their size and the volume.

On the other hand, the only feasible and practical means to improve natural convection heat transfer has been by the use of a finned surface. For example, vertical plate fins, shown in Fig. 1, have been conventionally adopted to enhance the heat transfer from a vertical, heated plate. The main function of these fins is not to increase the heat transfer rate itself, but to enlarge the effective heat transfer area. Thus, in order to obtain a high level of heat transfer performance with these fins, the surface area of the fins should be far larger than that of the base plate. Such a large fin area can only be achieved by the use of high fins. This, however, prevents compactness of the heat transfer plate, in particular, in the normal direction to the base plate.

Furthermore, it should be noted that vertical fins are, basically, inapplicable to the heat transfer enhancement of a tall, vertical plate. This is because the boundary layer developed over the tall plate becomes

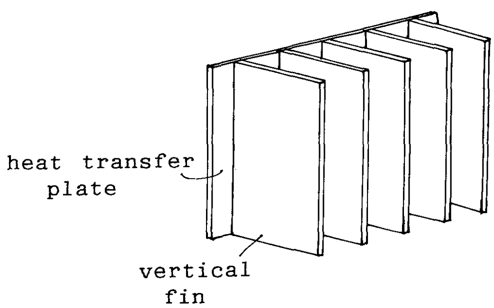


Fig. 1 Conventional heat transfer plate

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very thick, while the fins should be much higher than the boundary layer thickness in order to obtain an appreciable improvement of the heat transfer. However, such high fins are no longer practical. In order to dispose of these restrictions inherent in finned surfaces and to realize a compact but high-performance heat transfer plate, a technique that increases the heat transfer rate itself should be developed.

## 2. Enhancement of Natural Convection Heat Transfer and Aims of the Present Study

### 2.1 Previous studies

A number of previous workers have engaged in heat transfer enhancement of natural convection since the earliest era of natural convection study. Both active and passive enhancing techniques have been proposed and tested. Among these techniques, the simplest and the most practical is to install roughness elements on the heat transfer surface, because the technique has already been applied to forced convection and yielded successful results. Therefore, the main concerns of the previous workers have been directed to the heat transfer of rough surfaces. The problem was first dealt with by Prasolov[1], who measured the heat transfer coefficients from a horizontal cylinder to air. The surface of the cylinder was covered with roughness elements 0.08 to 0.36 mm high. He showed that the overall heat transfer coefficients were increased by 100 to 200% compared with the analytical results for a smooth cylinder.

His results attracted the attention of several workers. Heya et al.[2] reexamined Prasolov's experiment using almost the same experimental setup. They tested roughness elements 0.15 to 0.72 mm high. The overall heat transfer coefficients of the rough cylinder were measured with both water and air. However, no appreciable increase in the heat transfer coefficients was observed. The same result was reported by Fujii et al.[3]. They measured the local heat transfer coefficients around the roughness elements attached to a vertical, heated plate. Roughness elements 0.5 to 1.0 mm high were tested. However, very little variation of the heat transfer coefficients was shown. Based on the latter two results, it is now generally recognized that roughness elements whose height is far less than the boundary layer thickness will exert no appreciable influence on the heat transfer of natural convection.

In recent years, Bhavnani and Bergles[4] examined the influence of higher roughness elements on heat transfer. They investigated the heat transfer around two-dimensional, rectangular ribs attached horizontally on a vertical plate. The height of ribs ranged from 3.6 to 6.35 mm, and was of an order comparative to the boundary layer thickness. The result showed that the local heat transfer coefficients just upstream and downstream of the ribs decreased to less than that of the smooth plate owing to stagnation of the flow. Thus, the overall heat transfer coefficients of the ribbed surfaces became 7 to 40% smaller than those for a smooth plate of equal projected area.

The above result suggests that roughness elements placed in a natural convection boundary layer work as a flow retarder rather than a heat transfer promoter. This result is explained by the Reynolds numbers of the flow around the roughness elements. Because the flow velocities induced by the buoyancy force are, in general, very small, the Reynolds numbers remain so small that flow separation may hardly occur.

### 2.2 Previous works by the present authors

Based on the above discussion, it is worthwhile investigating the cases in which the roughness elements are much higher than the boundary layer thickness, because the Reynolds numbers of the flow around these roughness elements become higher, and flow separation will occur behind them. In this respect, the present authors[5] carried out heat transfer experiments with higher roughness elements whose configuration is shown in Fig. 2(a). The two-dimensional partition plate was adopted as a roughness element and installed horizontally on a vertical, heated plate. The height of the partition plate was varied from 10 to 70 mm. The local heat transfer coefficients of the vertical plate to water were measured at the location around the partition plate.

The result showed that the heat transfer in the downstream region of the partition plate is markedly enhanced when the plate height exceeds certain critical values, which are much higher than the boundary layer thickness adjacent to the vertical plate. The flow field around the partition plate during the above enhancement was also made visible with dye. The result is schematically illustrated in Fig. 2(a). The fluid upstream, vertical plate turns at the tip of the partition plate and separates from the upper surface after flowing some distance from the tip. Replacing this separated flow, the fluid above the partition plate penetrates the separation region and reaches the vicinity of the downstream plate. Here, the temperatures of the fluids ascending the upstream plate and above the partition plate are considered high and low, respectively. Thus, it is revealed that the high heat transfer coefficients of the downstream plate are caused by the inflow of the low-temperature fluid into the separation region. This result provides a key to the heat transfer enhancement of natural convection.

As was mentioned in the above, the high partition plate can promote heat transfer. However, in order to



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obtain a marked increase in the heat transfer coefficient, the height of the partition plate should be several to ten times higher than the boundary layer thickness developed over the base plate. However, the adoption of such high fins will be impractical, and also impedes the compactness of the heat transfer plate. Therefore, roughness elements other than the partition plate should be exploited.

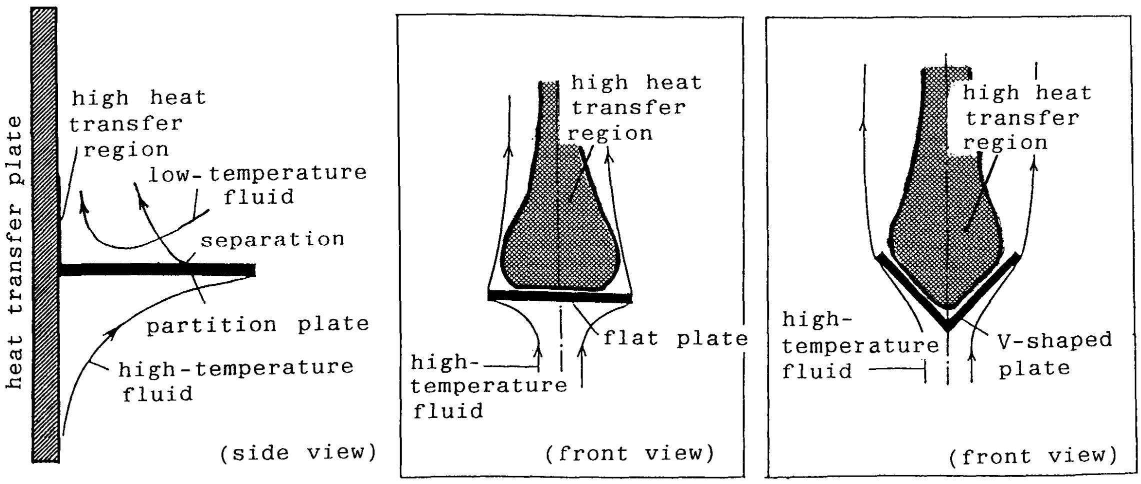
The two-dimensional partition plate redirected all of the high-temperature fluids toward the outer region of the boundary layer. On the other hand, there is another way to redirect the high-temperature fluid. Let us consider a case in which the spanwise length of the partition plate is finite, as is shown in Fig. 2(b). Then, the high-temperature fluids near the both sides of the partition plate will go around the plate. Provided that the span of the plate is short enough, the plate will redirect all of the high-temparature fluids toward both sides. Thus, the low-temperature fluids will enter the region behind the plate instead, and will enhance the heat transfer.

Moreover, it is worthwhile considering a case in which the above plate is folded in half to make a V-shape as shown in Fig. 2(c), which will redirect the high-temperature fluid more easily than the horizontal plate. The stagnation of high-temperature fluids in front of the plate will also be diminished by use of the V-shaped plate. Thus, the reduction of the heat transfer in the upstream region will be minimized.

Based on the above concept, the present authors[6] conducted the experiments on the heat transfer and the fluid flow around a V-shaped plate. The result was satisfactory. The heat transfer was markedly enhanced in the downstream region of the V-shaped plate even with a plate 2.5 mm high, which was smaller than the boundary layer thickness. The high heat transfer region expanded in the downstream direction with increasing height and inclination angle of the plate. The heat transfer coefficients just behind the V-shaped plate increased to be almost the same as those at the leading edge of the test plate. The result suggests that a new boundary layer develops behind the plate. On the other hand, the local heat transfer coefficients immediately outside and upstream of the promoter became somewhat smaller than those without the plate due to the inflow and the stagnation of the high-temperature fluids, respectively. However, these reduced heat transfer regions covered only a small portion of the total heat transfer area. Therefore, the overall heat transfer coefficients around the V-shaped plate became much higher than those without the plate.

### 2.3 Aims of present study

The above experiments were carried out with a single promoter made of a low-conductivity acrylic resin plate. We consider here a case in which a number of V-shaped promoters are installed on a vertical, heated plate in a staggered layout as shown in Fig. 3. Each promoter will redirect the high-temperature fluid toward both sides, and will introduce the low-temperature ambient fluid into the downstream region of the promoter. Therefore, it is expected that multiple promoters would yield a marked increase in the overall heat transfer coefficients of the base plate. Moreover, they would work not only as heat transfer promoters but also as an extended heat transfer surface if they were made of high-conductivity materials such as copper or aluminum. This would result in a further increase in the overall heat transfer coefficients.




(a) Two-dimensional partition plate (b) Horizontal plate of finite length (c) V-shaped plate

Fig. 2 Flow and heat transfer around plates



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In light of the above, heat transfer plates with staggered arrays of V-shaped promoters were fabricated and tested in this study. In order to ascertain that the experimental plates show higher heat transfer performance than conventional heat transfer plates, plates with vertical fins that have the same fin height and total surface area as the experimental plate were also fabricated. The overall and local heat transfer characteristics of these plates are investigated experimentally. Furthermore, some discussions are presented on the future development of the plates and also on the applicability of the plates as the practical heat exchangers.

## 3. Experimental Apparatus and Measurements

Three test plates, shown in Figs. 4(a),(b) and (c), were examined in the experiments. These are a smooth plate, a plate with V-shaped promoters, and a plate with vertical fins, respectively. The smooth plate was made of a 2 mm-thick, 280 mm- wide and 240 mm -high copper plate. The promoters and the vertical fins were soldered on copper base plates with the same dimensions as the smooth plate. Stainless-steel foil heaters 30 μm thick were glued to the back of the test plate. The double-adhesive vinyl tapes electrically insulated the copper plate from the heaters. The heaters were divided into four heating sections, and each section was heated by an independent a.c. power supply. By adjustment of the electrical input to each heating section, the surface temperatures of the plates were maintained almost uniform.

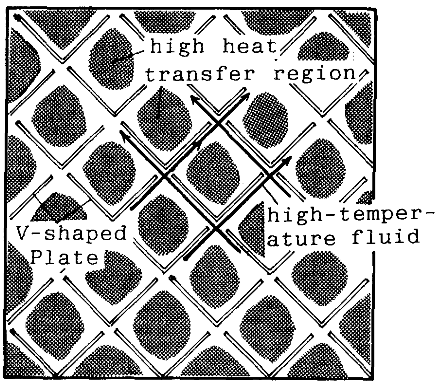
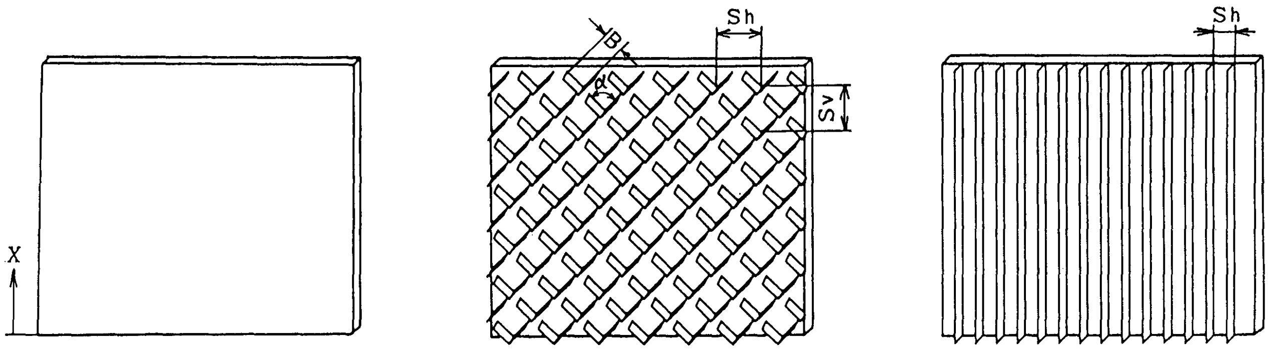



Fig. 3 Plate with V-shaped promoters

An acrylic resin plate 10 mm thick was installed on the back of the heaters to support the test plate. Styrofoam thermal insulation 20 mm thick was also glued to the back of the acrylic plate to prevent conduction heat loss through the acrylic plate to the ambient fluid. For the purpose of the local temperature measurements of the test plate, forty-six Chromel-Alumel thermocouples of 100 μm diameter were spot welded on the back of the copper plate. They were distributed densely near the promoters or the fins to confirm the uniformity of the surface temperatures. The test plates were placed vertically in the large water tank of 600×600 mm² cross-sectional area and 900 mm depth. Water at room temperature was used as the test fluid.

The V-shaped promoters utilized in the experiments were made of a 1 mm-thick copper plate and their dimensions were as follows: length of the plate $B=20$ mm, opening angle of the plate $\alpha=90$ deg., horizontal pitch $S_h=40$ mm, and vertical pitch $S_v=40$ mm. (See also Fig. 4(b) for reference.) Meanwhile, the height of the promoters was varied at 5, 10 and 20 mm. These promoters were attached onto the base plate in a staggered layout. The vertical fins shown in Fig. 4(c) were also made of 1 mm-thick copper plate. In order to compare the heat transfer performance, the height and the total surface area of the vertical fins were the same as those of the V-shaped promoters. Thus, the height $H$ and the horizontal pitch $S_h$ of the vertical fins were determined as $H=10$, 20 mm and $S_h=18.6$ mm, respectively.

The effective heat transfer area of the three plates is tabulated in Table 1. The experiments were performed under temperature differences between the




(a) Smooth plate

(b) Experimental plate with V-shaped plates

(c) Conventional plate with vertical fins

Fig. 4 Test plates



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plate and the ambient water of $\Delta T=2.5$, 5.0 and 7.5 K. The maximum temperature deviation against the mean temperature of the test plates was kept to less than 0.4 K even for the case of $\Delta T=7.5$ K. This confirms the temperature uniformity of the test plate.

## 4. Results and Discussion

The overall heat transfer coefficients for the three types of test plates are demonstrated in Fig. 5. The coefficients $h_m$ are calculated from the following equation, and are plotted against the temperature difference $\Delta T$.

$$h_m=Q/A_b(T_w-T_f) \qquad (1)$$

where, $Q$ : total heat input to heaters, $A_b$ : surface area of the base plate, $T_w$ : temperature of the plate, and $T_f$ : ambient temperature of the fluid. The Rayleigh numbers based on the height of the test plate are shown in the figure. The analytical solution[7] for a laminar natural convection adjacent to a vertical, flat plate is represented by the solid line in the figure and is also expressed as :

$$Nu_m\,(=h_m l/\lambda)=0.60Ra^{1/4} \qquad (2)$$

As is obvious from the figure, the result for the smooth plate coincides well with the above equation. This confirms that the measurements were carried out satisfactorily.

Table 1 Effective surface areas of test plates

| | Surface Area of Plates(cm²) | | | | | |
|---|---|---|---|---|---|---|
| | Smooth | Vertical Plate Fin | | V-shaped Plate | | |
| Height | - | 10mm | 20mm | 5mm | 10mm | 20mm |
| Base Plate | 672 | 672 | 672 | 672 | 672 | 672 |
| Fin or V-shaped Plate | 0 | 672 | 1344 | 336 | 672 | 1344 |
| Total | 672 | 1344 | 2016 | 1008 | 1344 | 2016 |

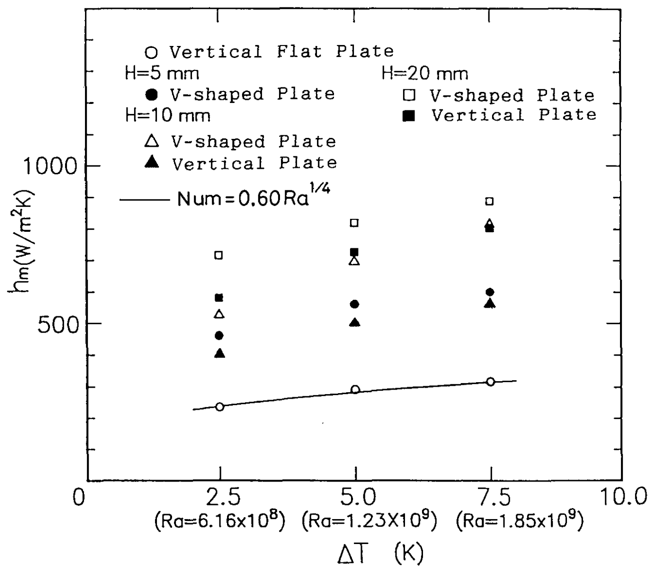


Fig. 5 Overall heat transfer coefficients of test plates

Meanwhile, the overall heat transfer coefficients for the plate with conventional fins and also for the plate with V-shaped promoters become much higher than those for the smooth plate. The ratios of the heat transfer enhancement for the former two plates to the smooth plate are as follows: 1.7 and 2.3 for the plate with 10 mm- and 20 mm-high vertical fins, respectively, and 1.9, 2.4 and 2.9 for the plates with 5 mm-, 10 mm- and 20 mm-high promoters, respectively. On the other hand, the ratios of the surface area enlargement to the smooth plate are calculated from Table 1 as, 2.0 and 3.0 for the plate with 10 mm- and 20 mm-high fins, and 1.5, 2.0 and 3.0 for the plates with 5 mm-, 10 mm- and 20 mm-high promoters, respectively. It is obvious from these results that the ratios of the heat transfer enhancement exceed the ratios of the surface enlargement, in particular, for the test plates with 5 mm- and 10 mm-high promoters. This result indicates that the promoters substantially increase the heat transfer coefficient.

Although the effective surface areas of the test plates are identical between the vertical fins and the promoters, the overall heat transfer coefficients of the plates show 40% and 28% higher values than those of the conventional plates with 10 mm- and 20 mm-high fins, respectively. With consideration of the fact that there exists no practical heat transfer plate that has a comparable performance to the conventional plate, the result is noteworthy. Moreover, it should be recalled that even the test plate with 5 mm-high promoters yielded higher heat transfer coefficients than the conventional plate with 10 mm-high vertical fins. This result will encourage further development of the compact and material-saving heat transfer plate.

In order to assess the performance of the heat transfer plates, the local as well as the overall heat transfer characteristics should be investigated. In the experiments, the test plate was divided into four heating sections, thus, the local heat transfer coefficient for each section can be measured from the electrical input to the heater as follows :

$$h_i=Q_i/A_i(T_w-T_f) \qquad (3)$$

where, $h_i$, $Q_i$ and $A_i$ stand for the local heat transfer coefficient, the transferred heat and the surface area of the $i$th heating section, respectively.

The typical results of the local heat transfer coefficients thus obtained are shown in Fig. 6 for the case of $\Delta T=5$ K. Here, the height of the fins and the promoters is kept equal at $H=10$ mm, and, thus, the effective surface areas are identical for both plates. Figure 6(a) demonstrates that the heat transfer



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coefficient of the smooth plate is the smallest among the three plates at every heating section.

Meanwhile, the local heat transfer coefficients at the bottom heating section from $X=0$ to 60 mm are almost the same between the vertical fins and the promoters as shown in Figs. 6(b) and (c). However, the variation of these coefficients with the vertical distance differs somewhat between the two. The result for the experimental plate demonstrates that the local heat transfer coefficients far downstream of the plate are maintained almost as high as that at the bottom of the plate. On the other hand, the result for the conventional plate shows a marked decrease in the local heat transfer coefficients in the vertical direction. Such reduction is caused by the development of the boundary layers over the base plate and the vertical fins, and is inevitable in the conventional plate. Meanwhile, the promoters renew the boundary layers developed over the base plate and realize high heat transfer regions behind them. In light of these facts, the advantages of the plate are obvious. The relative performance of the experimental plate to the conventional plate will be improved further if the test plate becomes higher.

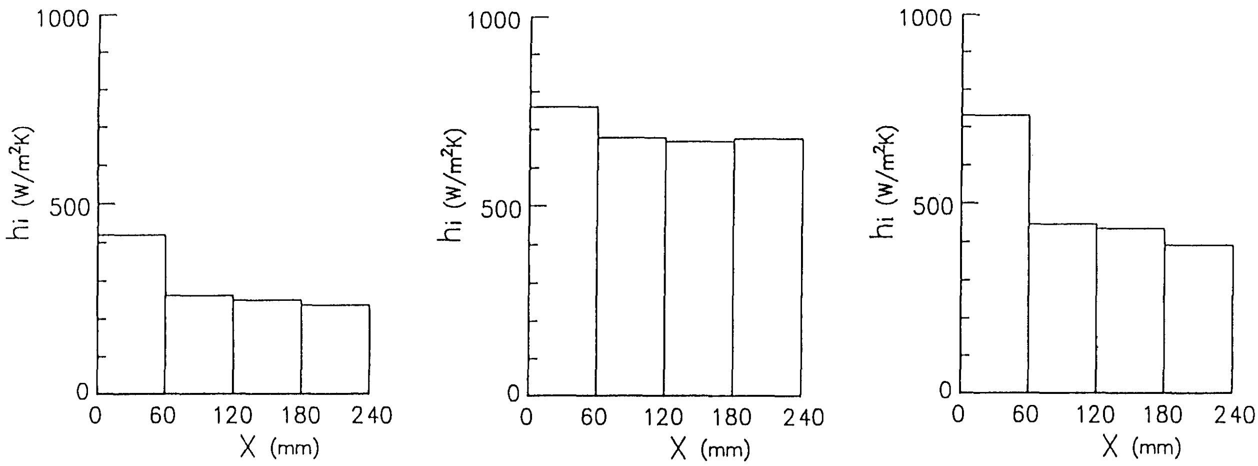
## 5. Discussion on the Future Development of the Enhanced Plate

As was shown in the previous section, the V-shaped promoters proposed in this paper have several advantages in the development of a compact as well as high-performance heat transfer plate. However, in order to apply these promoters to practical heat exchangers, the problems inherent in the promoters should be outlined and discussed.

The first problem is the fabrication of the V-shaped promoters. The experimental V-shaped promoters are difficult to install when compared with conventional vertical fins. However, this difficulty will be eased by the fact that even the small promoters less than several to ten millimeters high are effective in the heat transfer enhancement, and such small promoters can be fabricated by a press without difficulty.

The second problem is concerned with the layout of the promoters. The design procedures for the conventional vertical fins have already been established by Aihara[8] and others. Based on these previous studies, the vertical fins can be installed at a horizontal pitch of several to ten millimeters, which is smaller than that adopted in the present experiment. Such dense fins will result in a higher heat transfer performance than that obtained in the present experiment. Then, the problem arises as to whether the present promoters can be installed at smaller pitches or not. Although further experiments will be needed to answer this problem, we will briefly discuss the layout of the promoters in the following.

Firstly, we assume that the base plate is square as shown in Fig. 7, and also assume that the V-shaped promoters are installed at the same horizontal and vertical pitches as the vertical fins. Then, the layouts of the vertical fins and the promoters are given as in Figs. 7(a) and (b), respectively. It is obvious from these figures that the total surface area of the promoters becomes much larger than that of the vertical fins at the same height of the promoters and the fins. The result favors the experimental plate in view of the heat transfer enhancement. There will be, of course, some possibility that the promoters should be installed at larger pitches than the vertical fins. However, the surface area of the promoters will remain large unless the pitch of the promoters is about three times larger than that of the vertical fins. Figure 7(c) demonstrates the layout of the promoters whose




(a) Vertical flat plate $T=5.0$ K

(b) V-shaped plate $H=10$ mm, $\Delta T=5.0$ K

(c) Vertical plate fin $H=10$ mm, $\Delta T=5.0$ K

Fig. 6 Local heat transfer coefficients of test plates



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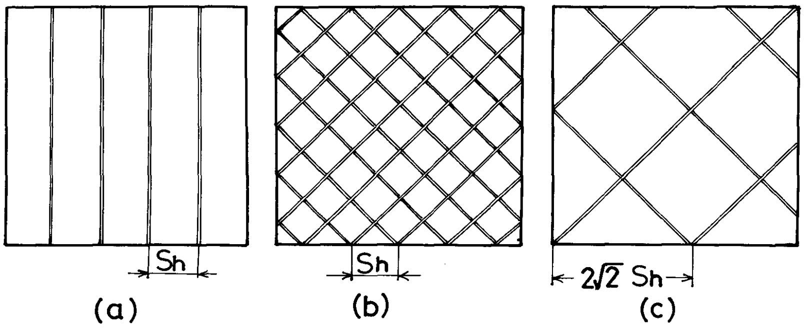


Fig. 7 Layout of vertical fins and promoters

surface area is equal to the vertical fins shown in Fig. 7(a). Here, the pitch of the promoters is $2\sqrt{2}$ times larger than that of the vertical fins.

In addition to these problems, we will encounter many other problems in the course of future applications of the experimental plate. One of such problem is the applicability of the experimental plate to air. Since air is often used as the working fluid in many practical situations, we are planning to perform experiments with air as the next stage.

## 6. Conclusions

This paper has developed a new enhancing technique for natural convection heat transfer from a vertical, heated plate. The technique is based on the idea that the heat transfer of natural convection can be promoted by redirecting the high-temperature fluids from the vicinity of the heated plate and by introducing low-temperature, ambient fluids into the near-wall region. V-shaped plates attached on the vertical plate were proposed as heat transfer promoters that could enable such fluid motion.

Heat transfer plates with staggered arrays of V-shaped promoters were fabricated, and their heat transfer characteristics were investigated. The overall as well as the local heat transfer coefficients of the plates were measured and compared with those of a conventional finned plate and also of a smooth plate. The results revealed that the following advantages are inherent in the present plate when compared with the conventional finned plate: 1) higher heat transfer performance is achieved even with smaller promoters than the vertical fins, and 2) heat transfer coefficients, even in the downstream region of the vertical plate, are increased with the promoters.

The conventional vertical fins have been installed so as not to retard the buoyant flow adjacent to the base plate. On the other hand, the present promoters positively inhibit the buoyant flow and renew the boundary layers over the base plate. The above concept for heat transfer enhancement will be applicable to not only natural convection over a vertical plate but also to the natural convection around vertical cylinders and over horizontal plates. Based on this concept, a compact as well as high-performance heat transfer plate could be developed.

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