Fluid Motion and Heat Transmission in a Horizontal Liquid Layer Heated Locally from Free Surface*

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Heat transfer from a heated wire and a heated vertical plate, located just below a liquid surface, was studied experimentally. The curve representing the heat transfer coefficient as a function of the temperature difference between the heaters and a cooled tray could be divided into four parts. The range of each part depended strongly upon the size of the heaters, the depth of the tray and the liquid properties. The mechanism of heat transfer from the heaters in each part was discussed. The following observations were made. In the first part, where the temperature difference was the smallest, convective heat transfer was obscured by conduction. The heat transfer was mainly due to natural convection in the second part, and was mainly due to Marangoni convection in the fourth part. The third part could be considered as a transition regime. Furthermore, it was found that the transition was suppressed by the meniscus of the liquid surface which was in contact with the heaters.

Key Words: Heat Transfer, Multiphase Flow, Surface Tension, Natural Convection, Marangoni Convection, Heated Wire, Horizontal Liquid Layer

1. Introduction

It is well known that not only buoyancy induced convection, but also convection induced by surface tension inhomogeneity can be observed. This convection, caused by surface tension gradients in a free surface, is called Marangoni convection. In a gravitational field, Marangoni convection is usually obscured by buoyancy-induced natural convection. However, Marangoni convection plays an important role in ignition of a pool of liquid fuel, spread of flame over the pool and crystal growth from melts[1],[2]. Many basic studies have been performed to obtain insight into these phenomena. Some of them focused on fluid motion in a liquid pool whose free surface was heated locally.

Murad et al.[1] observed fluid motion around a heated wire just wetted by the top of a liquid surface and claimed that the fluid motion was the net result of the surface tension and the buoyancy forces. However, Isoda[3] considered that the fluid motion was due to the surface tension gradients. Matsumoto and Saito[4] showed that the curve representing heat transferred from the wire as a function of temperature difference between wire and liquid could be divided into three parts. They considered that the fluid motion was caused by the surface tension gradients and regarded the fluid motion as laminar flow in the first part, where the temperature difference was smallest, and as turbulent flow in the third part. The experiments of Nishihara et al.[5] were apparently performed under the condition of the third part defined by Matsumoto and Saito[4] (turbulent Marangoni convection regime), because the slope of the transferred heat curve agreed with that in the third part. However, the temperature distribution in the liquid pool almost agreed with the results of laminar numerical analysis. Tanasawa and Maekawa[2] examined heat transfer to a cooled flat plate placed on a liquid surface. There was a critical temperature difference below which the heat transfer coefficient

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was almost constant, and the heat transfer coefficient increased monotonically with the temperature difference for larger temperature differences. They considered that the fluid motion was mainly due to the surface tension gradients in the small temperature difference range and that buoyancy forces also affected the fluid motion in the larger temperature difference range. Kashiwagi et al.[6] obtained a similar heat transfer coefficient curve from experiments using flat plate heaters. However, they considered that the heat was mainly transmitted by conduction in the small temperature difference range, and the surface-tension-driven fluid motion became effective in the larger temperature difference range.

As mentioned above, there have been many studies on fluid motion and heat transmission in a liquid pool whose free surface was heated locally. However, each study yielded different results and indicated a different heat transfer mechanism. Therefore, more detailed and systematic research is required. The purpose of this study is to obtain basic knowledge of fluid motion and heat transmission in a horizontal liquid layer heated locally from a free surface, which may be useful for obtaining insight into several phenomena which accompany surface-tension-induced fluid motion, and which would provide useful hints for future research on Marangoni convection in low gravity fields. First, fluid motion and temperature distribution in the liquid layer were analyzed numerically, and features of natural convection and Marangoni convection were examined. Based on the results, heat transfer from a heated wire and a heated vertical plate, which were located just below the liquid surface, was studied experimentally. Convection in a rectangular cavity with a free surface heated from the side was also observed, and the effect of the meniscus developed at the junctions of the liquid surface and side walls upon the flow pattern was examined. Finally, it was shown that heat transmission in the liquid layer was due to not only Marangoni convection but also natural convection and conduction, and the role of each factor depended strongly upon the shape of the liquid surface which was in contact with the heater as well as the sizes of heater and tray, the temperature difference and the liquid properties.

2. Features of Natural Convection and Marangoni Convection

Fluid motion and temperature distribution in a liquid pool whose free surface was heated locally were analyzed numerically to examine features of natural convection and Marangoni convection. The simplified model shown in Fig. 1 was employed to represent the somewhat complicated experimental model described later. An isothermally heated a semicircular heater was located at the center of a semicircular tray which was cooled isothermally. It was assumed that the free surface of the liquid pool remained flat and was in contact perpendicularly with the heater and the tray. Fluid motion was assumed to be two-dimensional and symmetrical with respect to the vertical center line. Properties of the liquid, except for density in buoyancy term and surface tension, were assumed to be constant (properties of n-propanol at 300 K[7] were employed). Solutions of the governing equations (equation of continuity, momentum equations and energy equation) were obtained numerically by employing the control volume and the SIMPLE algorithm[8].

Typical examples of numerical solutions are shown in Fig. 2. Figure 2(a) shows net results of surface tension gradients (Marangoni convection). Figure 2(c) shows net results of buoyancy forces

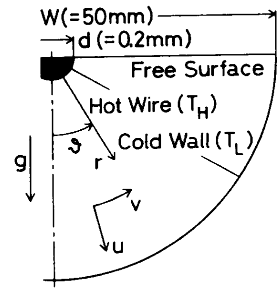

Fig. 1 Numerical model

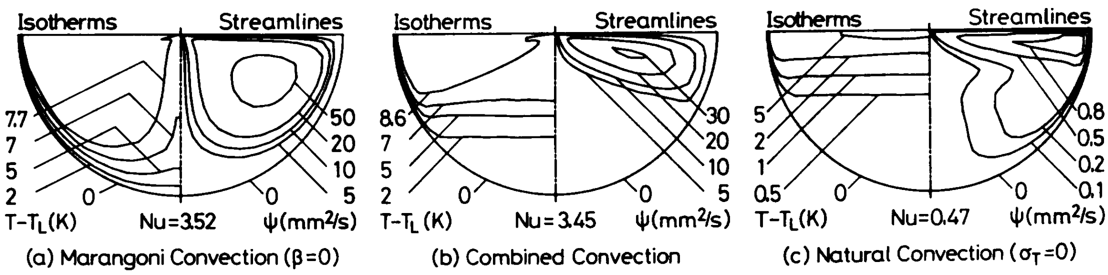

Fig. 2 Representative numerical results. (T_H-T_L=50 K, d=0.2 mm, W=50 mm, liquid : n-propanol (300 K))

(natural convection). Figure 2(b) shows results of both surface tension gradients and buoyancy forces (mixed convection). The left-hand side of each figure shows isotherms, and the right-hand side shows streamlines. Nu is the Nusselt number defined as $Nu = Q/[\pi\lambda(T_H - T_L)]$, where Q is heat input from the heater per unit length, λ is thermal conductivity, T_H is temperature of the heater and T_L is temperature of the cooled tray. As shown in Fig. 2(a), strong circulating flow is induced and low-temperature liquid is led to the heater in the case of Marangoni convection. In the case of natural convection, heated liquid slowly moves away from the heater along the free surface, and the temperature distribution is almost stratified. The value of Nu for natural convection is very small compared with that for Marangoni convection. Maximum velocity of fluid motion for mixed convection is comparable to that for Marangoni convection. However, the influence of buoyancy forces is clearly seen in Fig. 2(b), for example, the area of the circulating flow is limited to the upper half of the tray and the temperature distribution is stratified in the lower half.

3. Wire Heating

Next we described detailed experimental investigation of steady-state fluid motion and heat transmission in a horizontal liquid layer whose free surface is heated locally by a wire.

3.1 Experimental apparatus

Figure 3 shows a schematic diagram of the experimental apparatus, which consisted of a small rectangular tray. Two sides of the tray were double-glazed windows to facilitate observation. Other sides of the tray were thermally insulated. The tray was 100 mm thick in the direction of observation, 73 mm wide, and 50 mm high. Depth, H, of the test liquid was 15 mm. Nickel wire for electrical heating, spanning the thickness of the tray, was placed just below the liquid surface so that the top of the wire was in contact with the liquid surface. The bottom of the tray consisted of a brass block, with a water channel to permit dissipation of the heat transferred from the wire. Temperature, T_H, of the wire was estimated from the electric resistance curve as a function of temperature which was obtained in advance. Temperature, T_L, of the bottom of the tray was measured using a thermocouple. Heat transfer data were presented in terms of Nusselt number which was defined as $Nu = Q/(\pi\lambda\Delta T)$ using temperature difference $\Delta T = T_H - T_L$ to avoid uncertainty due to error in bulk temperature measurement, where Q was the power dissipated in the wire per unit length. Diameter, d, of the wire was varied from 0.04 mm to 0.8 mm. Lower alcohols were employed as test liquids. Water was not employed, because water surface was easily spoiled and free surface motion might be suppressed[9]. For this experiment, Q was considered to be accurate to within $\pm 0.5\%$, and the error of ΔT was smaller than $\Delta T \times 1.5\% + 0.3$ K.

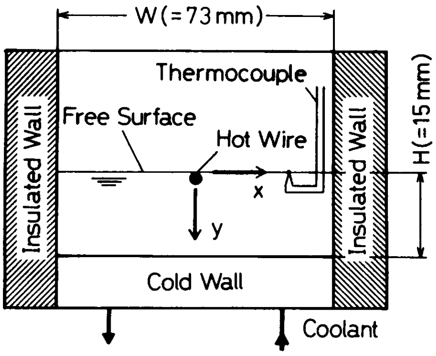

Fig. 3 Experimental apparatus

3.2 Heat transfer from wire

Figure 4 shows the variations of Nu with increase of ΔT for various values of d. As shown in the figure, the larger d is, the larger is the value of Nu. The slopes of Nu curves depend upon the range of ΔT, and every Nu curve can be divided into four parts except for the curve for $d = 0.8$ mm (ability of the power supply was not sufficient). Taking the case of $d = 0.2$ mm as an example, Nu does not change much with increase in ΔT in the range of $\Delta T < 5$ K, but Nu increases slightly with ΔT in the range of 5 K $< \Delta T <$ 25 K, increases rapidly with ΔT in the range of 25 K $< \Delta T <$ 35 K, and increases gradually with ΔT in the range of 35 K $< \Delta T$. In this study, these four parts are called regimes (I), (II), (III) and (IV), respectively. Regimes (II) and (IV) should correspond respectively to the laminar convection regime and to the turbulent convection regime of Matsumoto and Saito[4].

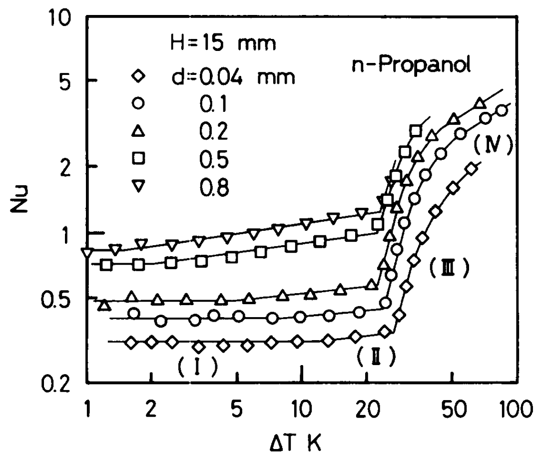

Fig. 4 Variations of Nusselt number of wire with temperature difference (effect of wire diameter)

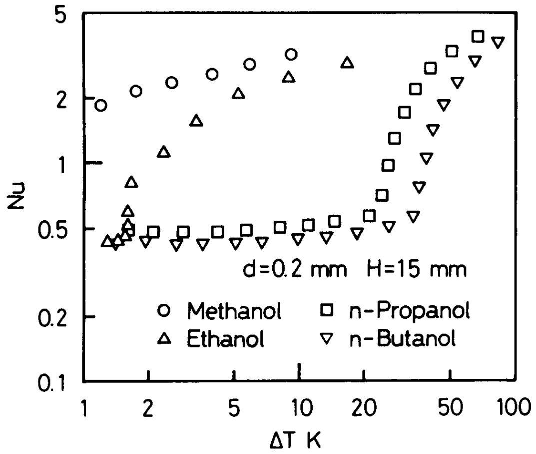

Fig. 5 Variations of Nusselt number of wire with temperature difference (effect of test liquid)

Figure 5 shows the variations of Nu with increase of ΔT for various test liquids. The shape of the Nu curve of n-butanol is similar to that of n-propanol, which was described in the previous paragraph, and the curve could be divided into four parts. However, regimes (I) and (II) are not so obvious in the Nu curve of ethanol, and regimes (I) and (II) cannot be found in the Nu curve of methanol. As shown in Figs. 4 and 5, the range of ΔT of each regime depended strongly upon the liquid properties as well as the size of the heater. This is probably one of the main reasons why previous studies yielded different heat transfer characteristics.

3.3 Fluid motion and temperature distribution

(I), (II), (III) and (IV) of Fig. 6 show representative photographs of fluid motion in each regime, which were visualized using aluminum tracer particles. τ indicates the exposure time of photography. As shown in Fig. 6, heated liquid moves very slowly away from the heater along the free surface within regime (I). Within regime (II), the velocity of the fluid motion increases. A pair of vortices appears below the heater within regime (III). The size of the vortices becomes larger with increase in ΔT as shown in Fig. 6(IV).

(I), (II), (III) and (IV) of Fig. 7 show representative temperature distributions in each regime, which were measured using a thermocouple (diameter of the thermocouple was 0.25 mm). As shown in Fig. 7, isotherms near the heater are roughly semicircular within regime (I). Temperature distribution is almost stratified within regime (II). The shape of isotherms within regime (III) is similar to that within regime (II). However, low-temperature liquid flows towards the heater within regime (IV). This may be due to the strong vortex flow shown in Fig. 6(IV).

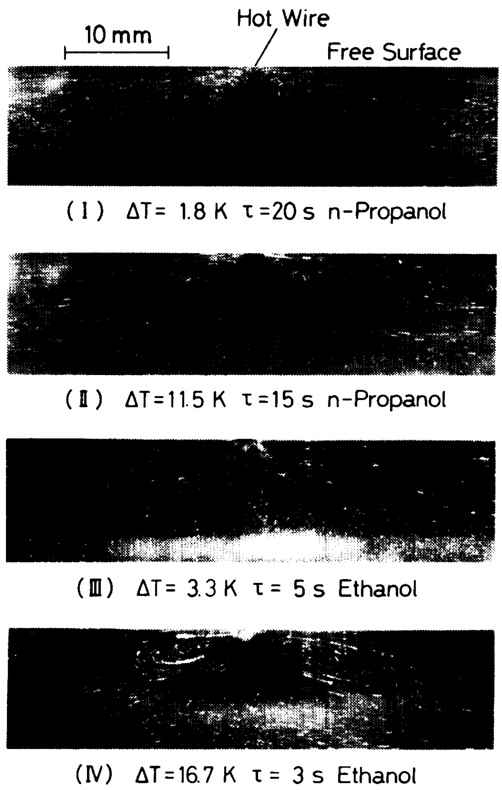

Fig. 6 Representative flow patterns around wire (d=0.2 mm, H=15 mm)

Fluctuation of the flow pattern and corresponding wire temperature fluctuation were also observed in the case of relatively large ΔT within regime (IV). Matsumoto and Saito[4] considered that the fluid flow transited to a turbulent one, based on the temperature fluctuation. However, we consider that the fluid flow did not transited to a turbulent one within this experimental range, because the temperature fluctuation was roughly periodic (the period was almost 100 s) and amplitude of the fluctuation was very small (below 1% of ΔT).

3.4 Discussion on mechanism of heat transfer

Nu within regimes (I) and (II) was obtained as a function of Rayleigh number, Ra, considering that fluid motion and temperature distribution within regime (II) shown in Fig. 6(II) and Fig. 7(II) had many similar features to those of the numerical solution for natural convection shown in Fig. 2(c). The result is shown in Fig. 8. Ra is defined as $Ra = g\beta\Delta T d^3/(\alpha\nu)$, where g is gravitational acceleration, β is thermal expansion coefficient, α is thermal diffusivity and ν is kinematic viscosity. Nusselt number of natural convection heat transfer around a wire in unbounded fluid[10] is also shown in Fig. 8 by a solid line. For all test liquids and values of d, Nu approaches the broken line as Ra increases, though data within regime (I) are just above the line. The

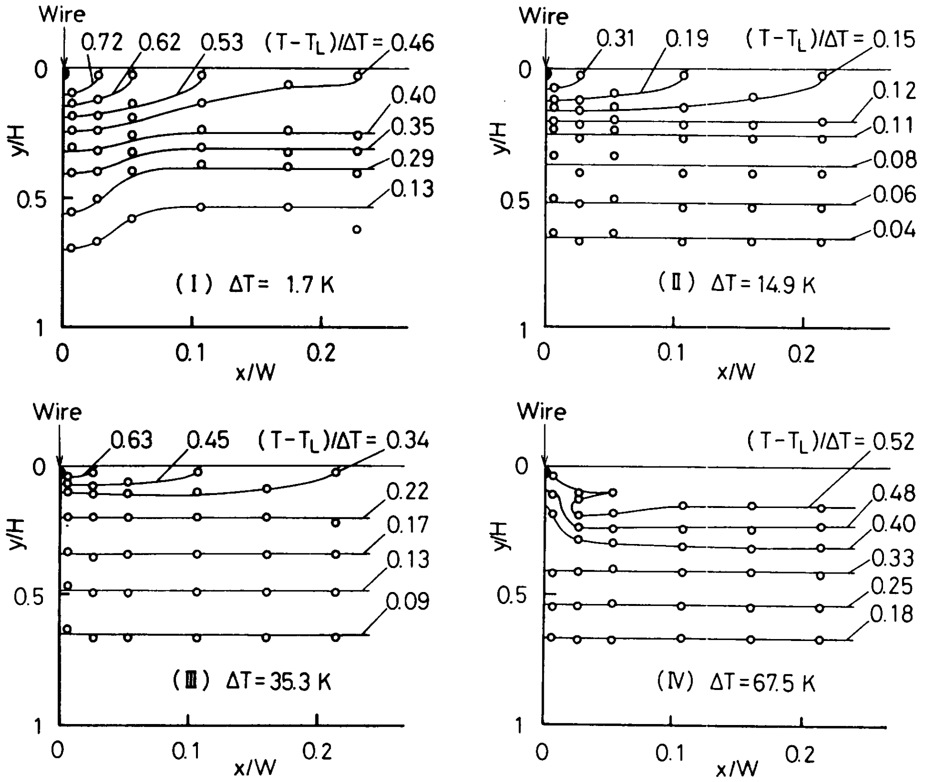

Fig. 7 Representative temperature distributions around wire (d=0.2 mm, H=15 mm, liquid : n-propanol)

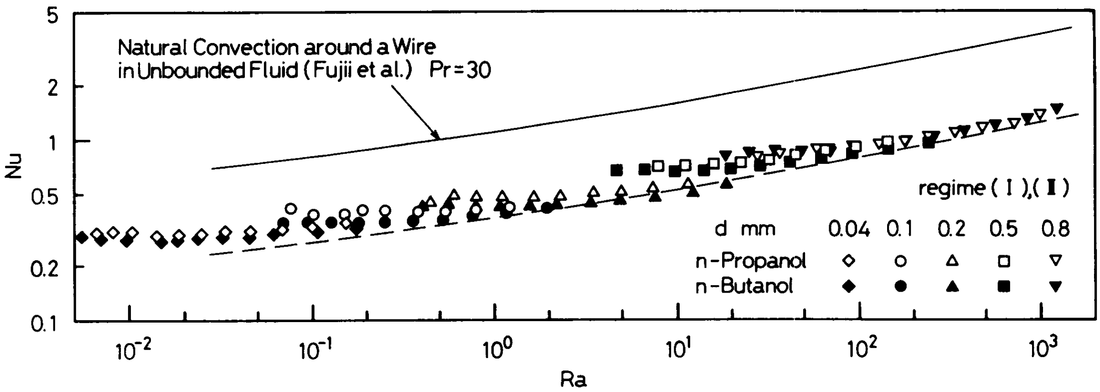

Fig. 8 Relationship between Rayleigh number and Nusselt number of wire

slope of the broken line is similar to that of the solid line. Based on these results, the heat transfer within regime (II) can be considered as natural convection heat transfer, and fluid motion should be caused mainly by the buoyancy forces.

Nu within regimes (III) and (IV) was obtained as a function of Marangoni number, Ma, considering that fluid motion and temperature distribution within regime (IV) shown in Fig. 6(IV) and Fig. 7(IV) had many similar features to those of the numerical solution for Marangoni convection shown in Fig. 2(a). The result is shown in Fig. 9. Ma is defined as $Ma=|\sigma_T|\Delta Td/(a\mu)$ using d as the characteristic length, where σ_T is temperature coefficient of surface tension and μ is dynamic viscosity. For all test liquids and values of d, Nu approaches the solid line indicating $Nu \propto Ma^{0.25}$ as Ma increases, though data within regime (III) are below the line (the exponent is similar to that reported by Tanasawa et al.[2] and Kashiwagi et al.[6] in the larger temperature difference range). Based on these results, the heat transfer within regime (IV) can be regarded as Marangoni convection heat transfer, and the fluid motion around the heater should be due to the surface tension gradients. However, fluid motion far from the heater might be affected by buoyancy forces, because the

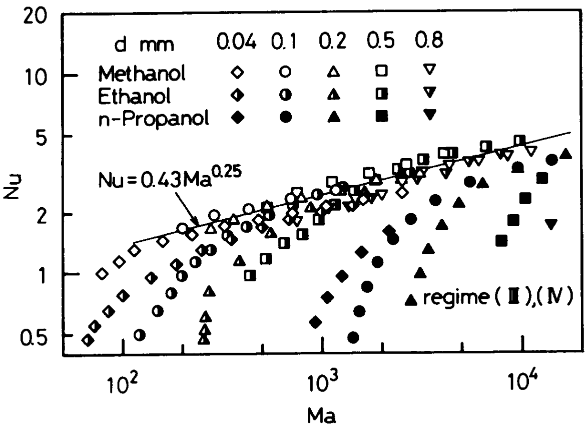

Fig. 9 Relationship between Marangoni number and Nusselt number of wire

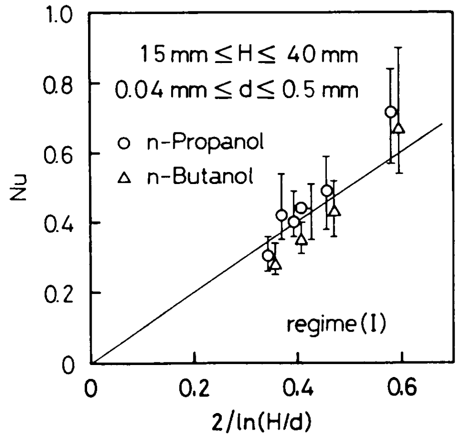

Fig. 10 Relationship between $2/\ln(H/d)$ and Nusselt number of wire

temperature distribution shown in Fig. 7(IV) resembles Fig. 2(b) rather than Fig. 2(a).

Within regime (I), fluid motion was very slow, Nu did not change much with increase in $\varDelta T$, and the value of Nu depended strongly upon d and H. These features were similar to those of conduction. Therefore, the relationship between Nu within regime (I) and $2/\ln(H/d)$, which should correspond to the Nusselt number of conduction if the heater and tray are semi-circular, was examined. The result is shown in Fig. 10. Nu is almost proportional to $2/\ln(H/d)$. Based on this result, it can be considered that convective heat transfer was obscured by conduction within regime (I).

4. Vertical Plate Heating

Heat transfer from a heated vertical plate which was placed just below the liquid surface was examined experimentally to confirm the hypotheses in the previous section. The tray and test liquid were similar to those in the previous section, but depth, H, of liquid was 40 mm. A nickel plate 0.02 mm thick, spanning the thickness of the tray, was placed vertically just below the liquid surface so that the top of the plate was in contact with the liquid surface, and the plate was heated uniformly by electric power dissipation. Height, b, of the plate was varied from 1 mm to 5 mm. Power dissipation, Q, and mean temperature, $\overline{T_H}$, of the plate were measured by the same methods as the previous section.

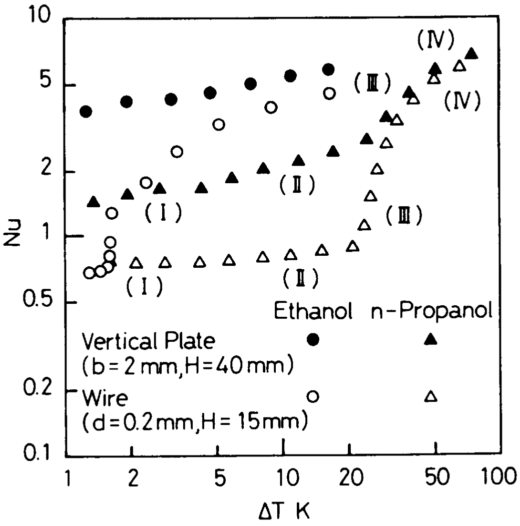

Fig. 11 Variations of Nusselt number of vertical plate with temperature difference

Typical examples of the variation of Nusselt number, Nu, with increase of $\varDelta T$ are shown in Fig. 11. Nu is defined as $Nu=Q/(2\lambda\varDelta T)$, where $\varDelta T=\overline{T_H}-T_L$. Heat transfer data for wire heating are converted and shown again in the figure as references. Taking the case of n-propanol as an example, the Nu curve for vertical plate heating can be divided into four parts in the same way as that for wire heating, though regime (I) is not so obvious and Nu within regime (III) does not increase so rapidly.

Figure 12 shows the relationship between Nu within regimes (I) and (II) and modified Rayleigh number, Ra^*, defined as $Ra^*=g\beta Qb^3/(2a\nu\lambda)$. Nusselt number of natural convection in unbounded fluid[10] is also shown in the figure by a solid line. For all test liquids and values of b, Nu within regime (II) almost agrees with that of natural convection in unbounded fluid, though data within regime (I) are just above the solid line. Figure 13 shows the relationship between Nu within regimes (III) and (IV) and modified Marangoni number, Ma^*, defined as $Ma^*=|\sigma_T|Qb/(2a\lambda\mu)$. For all test liquids and values of b, Nu approaches the solid line, indicating $Nu\propto Ma^{*0.13}$, as Ma^* increases, though data within regime (III) are below the line. These results confirm us the hypoth-

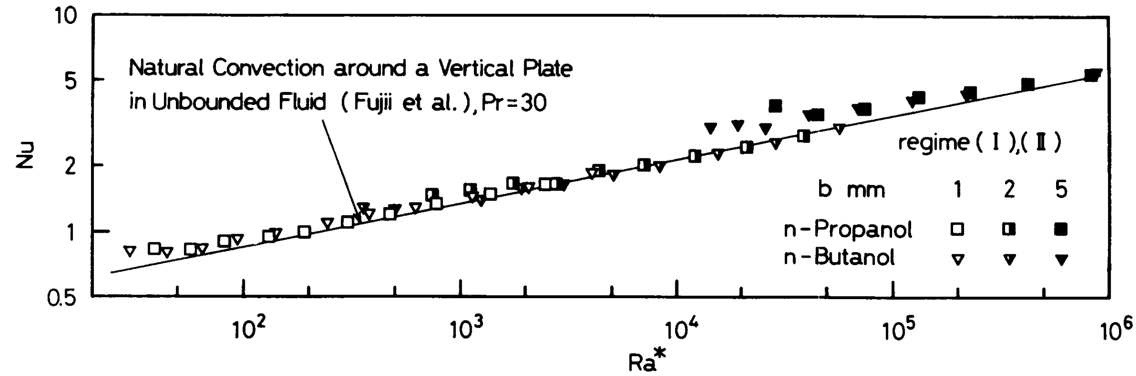

Fig. 12 Relationship between modified Rayleigh number and Nusselt number of vertical plate

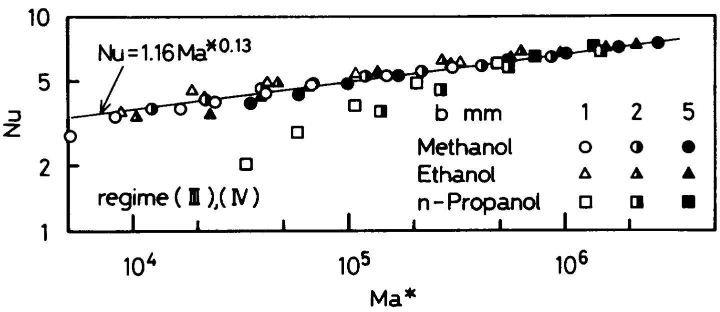

Fig. 13 Relationship between modified Marangoni number and Nusselt number of vertical plate

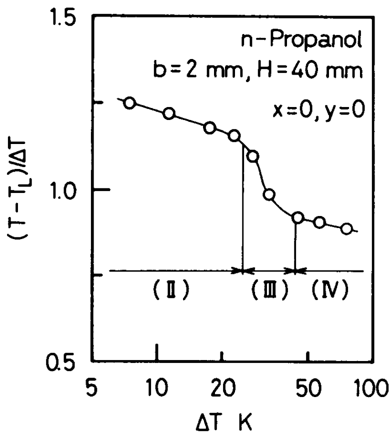

Fig. 14 Temperature of liquid surface in contact with top of plate

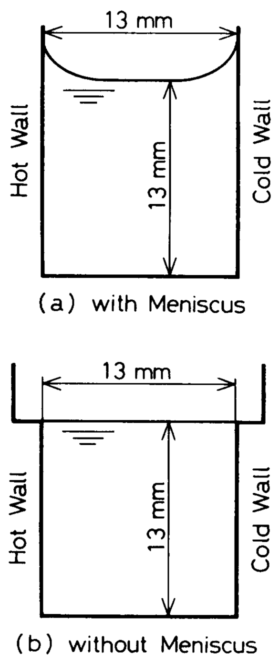

Fig. 15 Cross section of rectangular cavities

eses in the previous section that the heat transfer can be considered as natural convection heat transfer in regime (II) and as Marangoni convection heat transfer in regime (IV).

We now discuss the mechanism of heat transfer in regime (III) which was not discussed in the previous section. Figure 14 shows the variation of $(T-T_L)/\Delta T$ with increase in ΔT, where T is the temperature of the liquid surface which is in contact with the top of the heated plat (T can be regarded as the temperature of the top of the plate). Within regime (II), $T-T_L$ is larger than ΔT. This is because the local heat transfer coefficient of natural convection is largest at the lower edge of the plate. Within regime (IV), $T-T_L$ is smaller than ΔT. This is because the upper part of the plate is exposed to surface-tension-induced high-speed flow. $(T-T_L)/\Delta T$ decreases rapidly with increase in ΔT within regime (III). Accordingly, regime (III) can be regarded as a transition regime from natural convection to Marangoni convection, and the Marangoni forces become more effective with increase of ΔT in this regime.

5. Convection in Rectangular Cavity with Free Surface

As shown in Fig. 11, the Nusselt number curve of vertical plate heating also could be divided into four parts. However, the slope of the Nusselt number curve within regime (III) was smaller than that of wire heating. This difference might be due to the difference in shape of the heaters, because there are wedge-shaped slits between the upper side of the wire and the liquid surface in the case of wire heating, and fluid motion was considered to be hindered in the slits. The convection within a rectangular cavity in which meniscuses were developed at the junctions of the free surface and side walls was observed, and compared with the convection within a cavity in which development of meniscuses was prevented, because the meniscuses could be considered to play a similar role to the wedge-shaped slits.

Cross sections of two cavities, employed in the experiment, are shown in Fig. 15. Each cavity was composed of two copper blocks with water channels for isothermal heating/cooling as side walls, two double-glazed windows as front and back walls and an insulated bottom. Each cavity was 13 mm wide and 185 mm thick in the direction of observation. Depth of the test liquid was 13 mm. The side walls of the cavity of Fig. 15(a) had sharp edges at the same height as the liquid surface to prevent development of meniscuses, but the side walls of the cavity of Fig. 15(b) were plane so that meniscuses were developed (the liquid surface rose about 2 mm up the walls).

Figure 16 shows representative flow patterns which were visualized using aluminum tracer. ΔT is the temperature difference between the side walls. The four photographs in the upper row show the flow patterns in the case with meniscus. The liquid very slowly rises along the heated wall (left-hand side of the photograph) and falls along the cooled wall (right-hand side) within a relatively small temperature difference range, as shown in Fig. 16(a-1). This convection is probably mainly due to buoyancy forces, because the flow patterns have similar features to that within regime (II) of the wire heating. As ΔT becomes large, a vortex flow, which was also observed within regime (III) of the wire heating (see Fig. 6(III)), tends to appear near the junction of liquid surface and heated wall. The vortex becomes larger with increase of ΔT, and the area of vortex flow covers the upper half of the cavity at very large ΔT, as shown in Fig. 16(a-4). Surface tension gradients should play an important role in the vortex flow. Similar flow patterns are observed in the case without meniscus, as shown in the lower row of Fig. 16. However, each flow pattern is observed at smaller ΔT than in the case with meniscus. That is, growth of the vortex flow with ΔT is suppressed in the case with meniscus. It can be considered that the meniscus acts to suppress the Marangoni effects upon the fluid motion. Moreover, in the case of wire heating, it can be considered that the transition from regime (II) is suppressed until the temperature difference becomes sufficiently large and then transition to regime (IV) occurs rapidly at very large temperature difference, because the wedge-shaped slits play a similar role to the meniscus.

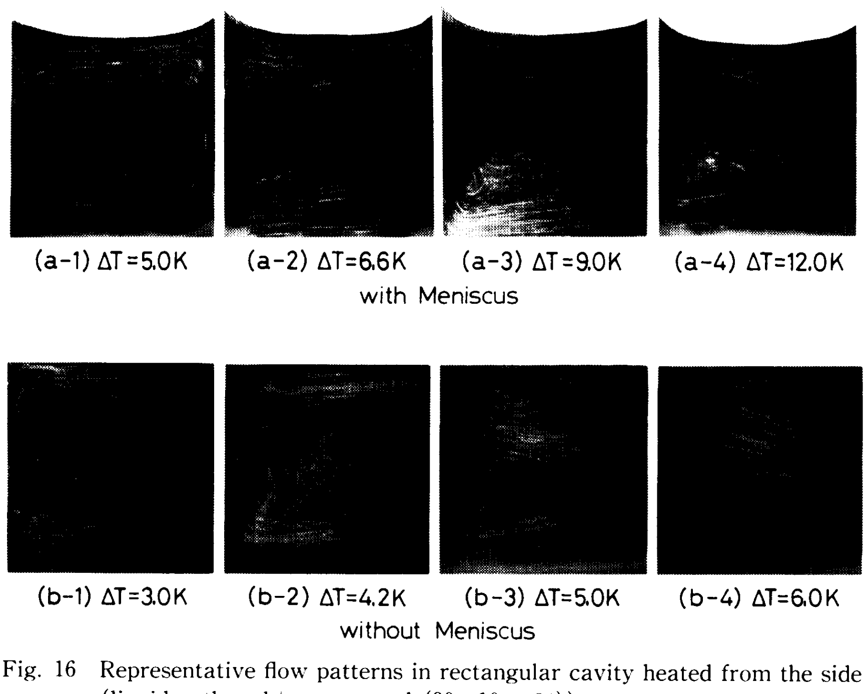

Fig. 16 Representative flow patterns in rectangular cavity heated from the side (liquid : ethanol + *n*-propanol (90 : 10 wt%))

As mentioned above, the fluid motion and the heat transmission in the liquid layer heated locally from a free surface is due to not only Marangoni convection but also natural convection and conduction, and the role of each factor depends strongly upon the shape of the liquid surface which is in contact with the heater as well as the sizes of heater and tray, the temperature difference and the liquid properties. Therefore, attentions should be paid to the shape of the liquid surface near the heater in the case that Marangoni forces might be significant.

6. Conclusions

Heat transfer from a heated wire and a heated vertical plate placed just below a liquid surface was studied experimentally, based on the features of Marangoni convection and natural convection obtained by numerical analysis. The following were deduced:

(1) The curve representing the heat transfer coefficient as a function of the temperature difference between heaters and a cooled tray could be divided into four parts. The range of each part depended strongly upon the size of heaters, the depth of the tray and the liquid properties.

(2) It was considered that convective heat transfer was obscured by conduction in the first part where the temperature difference was the smallest, and that the heat transfer was mainly due to natural convection in the second part and mainly due to Marangoni convection in the fourth part. The third part was regarded as a transition regime from the second part to the fourth part.

Furthermore, the experimental observation of the convection in rectangular cavity with a free surface heated from the side revealed that:

(3) The influence of Marangoni forces upon fluid motion was suppressed more in the case that meniscuses were developed at the junctions of liquid surface and walls than in the case that development of meniscuses was prevented.

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