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Boiling Heat Transfer on Finned Tube Bundle with Lower Tubes Heated with Constant Heat Flux

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■ The boiling heat transfer on a finned tube bundle was experimentally studied in R11 at a pressure of 1 bar. The bundle is composed of 18 finned tubes in a 6×3 inline arrangement. The heat transfer coefficient of the bundle with a pitch-to-diameter ratio $s/d = 1.6$ or 1.15 and the time-averaged liquid velocity under the bundle were measured when the tubes of the lower rows were heated with high constant heat flux. The results indicate that the boiling heat transfer is strongly enhanced by the strong two-phase flow induced by the tubes of the lower rows, especially in the intermediate region, with the maximum enhancement of heat transfer relative to that with the lower rows not heated was as much as 150%. The greatest time-averaged liquid velocity under the bundle occurred with the increase of total heat flow. The mass flow rate through the bundle decreased with high heat flow. These results show us a way to enhance the boiling heat transfer in the intermediate region between natural convection and fully developed boiling and suggest that the bundle heat exchanger should be designed to work in the intermediate region.

Keywords: *boiling heat transfer; finned surfaces, heat transfer augmentation*

INTRODUCTION

Finned tubes are widely used in bundle heat exchangers, especially in the bundle evaporators of refrigeration machinery. The performance of fins of various shapes for boiling heat transfer has been investigated by many workers [1–5]. In recent years, numerous experimental studies on boiling heat transfer from finned tubes and finned tube bundles have been reported [6–11]. For example, the effects of pool geometry, fluid flow, and enhanced convection have been studied. This paper reports on an experimental investigation of the boiling heat transfer coefficients of a finned tube bundle when the tubes in the lower rows of the bundle were heated with high constant heat flux. The fluid velocities under the bundle were also measured to clarify whether the thermal drag phenomenon exists in pool boiling.

EXPERIMENTAL SETUP

The experiments were performed in R11 ($CFCl_3$) at a pressure of about 1 bar and a saturation temperature around 23.31° C. The test system is shown schematically in Fig. 1. The vessel of the evaporator was made of stainless steel and measured $680 \times 600 \times 370$ mm. Three sides of the vessel were equipped with glass windows to allow observation. The vapor produced in the evaporator was condensed in a controllable condenser, and the condensed liquid was fed back through a thermostatic preheater into the vessel. The entire apparatus was encased in a temperature-controlled compartment, its air temperature adjusted to the saturation temperature inside the vessel. This proved helpful for the control of pressure in the evaporator and for the reduction of heat losses from the vessel.

The finned heater tubes were made of copper with a finned length of about 270 mm. The cross sections and the parameters of the finned tube are given in Fig. 2. The bundle consists of 18 tubes with 26 fins per inch arranged vertically and horizontally in an inline configuration and with a pitch-to-diameter ratio $s/d = 1.15$ or 1.6 as shown in Fig. 3. Each tube could be heated separately with an internal electric heater (3 kW at 220 V). To prevent lateral flow and to simulate a fraction of a large bundle, the test bundle was placed between vertical glass plates, the upper ends of the plates extending out of the fluid by about 70 mm to prevent splashing.

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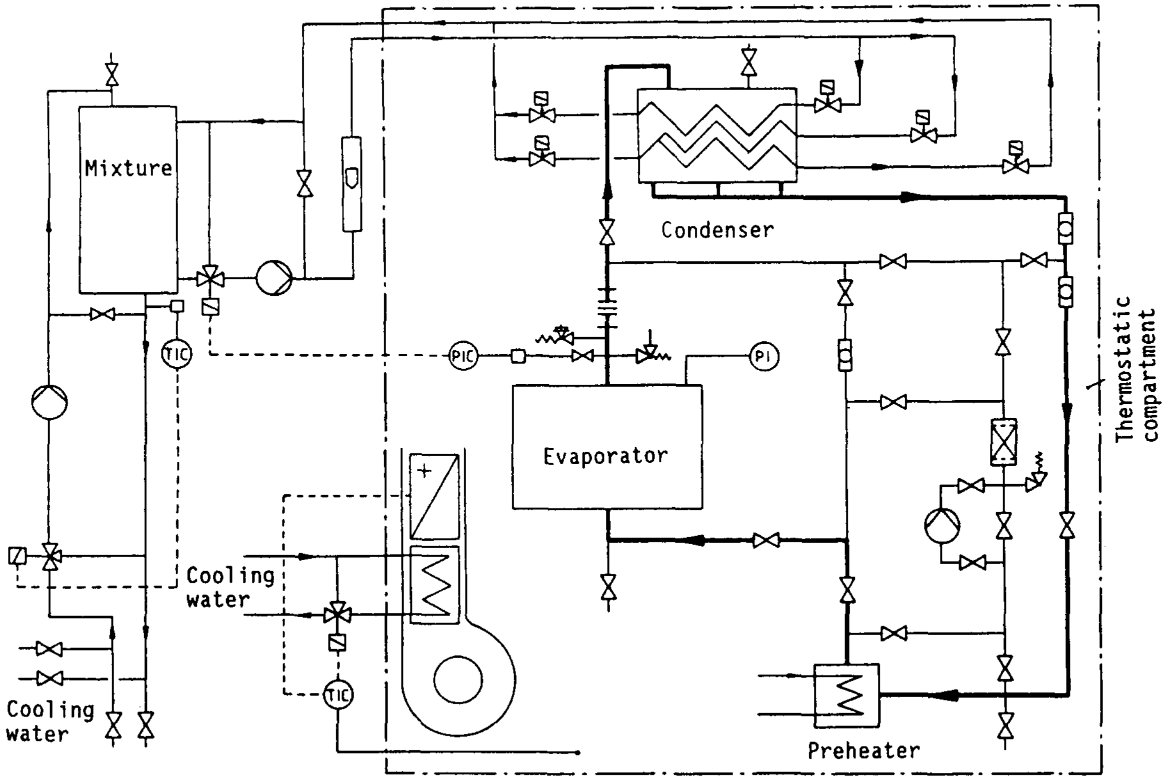


Figure 1. Schematic drawing of test system.

MEASUREMENTS

Temperature

Temperatures were measured with Phillips NiCr-Ni-coated thermocouples of 0.5 mm O. D. The liquid and vapor temperatures were measured with six thermocouples at the locations shown in Fig. 3. The inside wall temperature of the finned tube, ϑ_{wi}, was measured with six thermocouples that were fixed at intervals along the tube and around its inner circumference as shown in Fig. 2. The six tubes in the middle column were equipped with interior thermocouples. The outside wall temperature of the finned tube is calculated with the equation

$$\vartheta_{wo} = \vartheta_{wi} - \frac{Q \ln(d_R/d_i)}{2\pi k L}, \tag{1}$$

where ϑ_{wi} is the arithmetic mean of six inside wall temperatures measured with six thermocouples for each tube, Q is the electric heating power for each heater tube, d_R and d_i are tube diameters described in Fig. 2, k is the thermal conductivity of copper and L is the finned tube length of the heating section. The temperature difference between the outside wall temperature ϑ_{wo} and the temperature of the saturated vapor above the liquid ϑ_v is used to calculate the heat transfer coefficient.

Pressure

The saturation pressure of the vapor was measured at the vapor outlet of the vessel by a class 0.1 pressure gauge within the range 0–1.5 bar.

Heating Power

Each of the 18 finned tubes could be heated separately. The heating power was measured individually with a class 0.1 power meter. At the beginning and end of a full measurement, the heating power of each tube was measured twice. The mean value was used to calculate the outside wall temperature and the heat transfer coefficient to minimize the measurement error resulting from power supply fluctuation.

Liquid Velocity

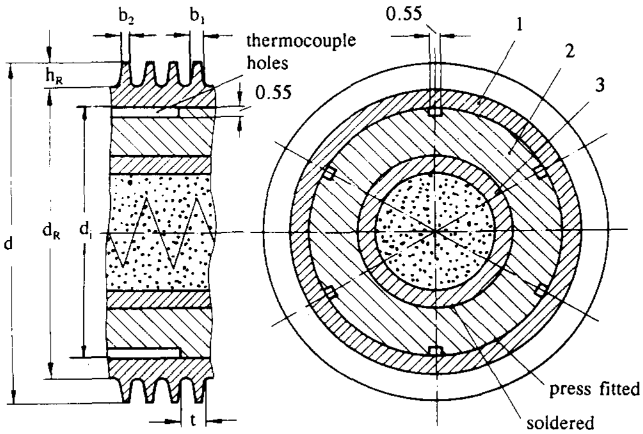
The liquid velocity under the bundle was measured by using a small velocimeter placed about 70 mm under the bundle as shown in Fig. 3. The liquid velocity was unsteady due to the action of the two-phase flow. Therefore, the time-averaged velocity was calculated from the velocity data (over 1500) collected over a period of about 5 min.

EXPERIMENTAL PROCEDURE

The heat transfer coefficient for the finned tubes is obtained from the equation

$$\alpha = Q/[A(\vartheta_{wo} - \vartheta_v)], \tag{2}$$

where Q is the measured heating power for each tube, A is the total surface area of the finned tube, ϑ_{wo} is the outside temperature of the tube, and ϑ_v is the vapor temperature above the liquid in the vessel. The mean heat transfer coefficient of the bundle is the arithmetic mean of the heat transfer coefficients of each row of the bundle.



fins per inch	26
finned tube diameter d [mm]	18.89
root diameter d_R [mm]	15.85
inside diameter d_i [mm]	13.90
fin height h_R [mm]	1.52
fin spacing t [mm]	1.03
fin thickness (base) b_1 [mm]	0.44
fin thickness (tip) b_2 [mm]	0.22
total surface area A [m^2/m]	0.1952
area ratio A/A_R	3.920

Figure 2. Finned heater tube geometry. *1*, Finned tube; *2*, filler tube; *3*, heating element.

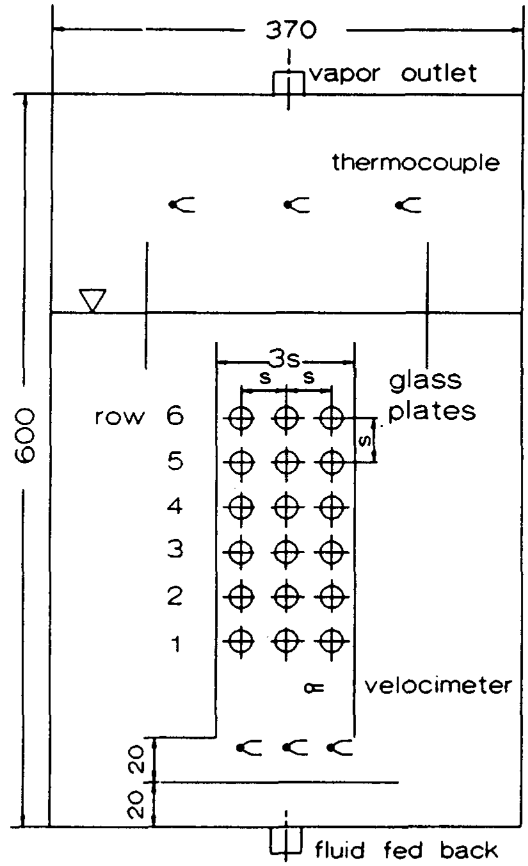


Figure 3. Bundle arrangement.

Before the experiments were performed, the test tubes had to be heated with a high heat flux for at least 50 h to minimize the starting effects and to obtain reproducible results. The temperatures of the inside wall of the finned tubes were measured 10 times in about 12 min. All experiments were performed with heat flow being decreased in steps from large to small values. During the experiments, the vapor pressure was exactly controlled to be 1 ± 0.005 bar.

EXPERIMENTAL RESULTS AND DISCUSSION

Boiling Heat Transfer of Tube Bundle

The heat transfer coefficient of the tube bundle were measured with the tubes of the lower two rows heated with high constant heat flux. In the experiments, the tubes of the two bottom rows were heated separately from those of the other rows, with a constant heat flux of 30,000, 20,000, or 10,000 W/m^2. The results are shown in Fig. 4 as mean heat transfer coefficients of the four upper rows versus the heat flux of these tubes. It is known that in pool boiling from a finned tube bundle, a natural convection region, an intermediate region, and a fully developed boiling region can be distinguished according to the heat transfer performance. It can be seen from Fig. 4 that the heat transfer from the four upper rows is strongly enhanced by the strong two-phase flow induced from the tubes heated with high constant heat flux. The maximum enhancement is found in the intermediate region around $q = 1000$ W/m^2. In the fully developed region, with a heat flux higher than 20,000 W/m^2, the heat transfer enhancement due to the strong two-phase flow is very small. The heat transfer in this region is dominated by nucleate boiling, and the two-phase flow induced by boiling from the tubes of the two lowest rows has very little effect on the heat transfer of the four uppermost rows.

In the natural convection region around $q = 100$ W/m^2, the heat transfer is also strongly enhanced. The increase in the heat transfer coefficient compared to that when the lower two rows were not heated is much smaller than in the intermediate heat flux region. In the natural convection region, the tubes were heated with a low heat flux and only a few bubbles were created, and the increase in the heat transfer coefficient was due only to convective enhancement. But in the intermediate region, a lot of bubbles were created at the tubes of the four upper rows and were intensely washed away by the action of the two-phase flow induced from the lower two rows. Thus, boiling from the tubes of the upper rows was strongly enhanced. Heat transfer enhancement in the intermediate heat flux region consisted of convective enhancement and boiling enhancement. For the case of a constant heat flux of 30,000 W/m^2, the maximum increase in the heat transfer coefficient relative to that of the upper rows with the two lower rows not heated is about 150%. These results suggest a way for enhancing heat transfer within

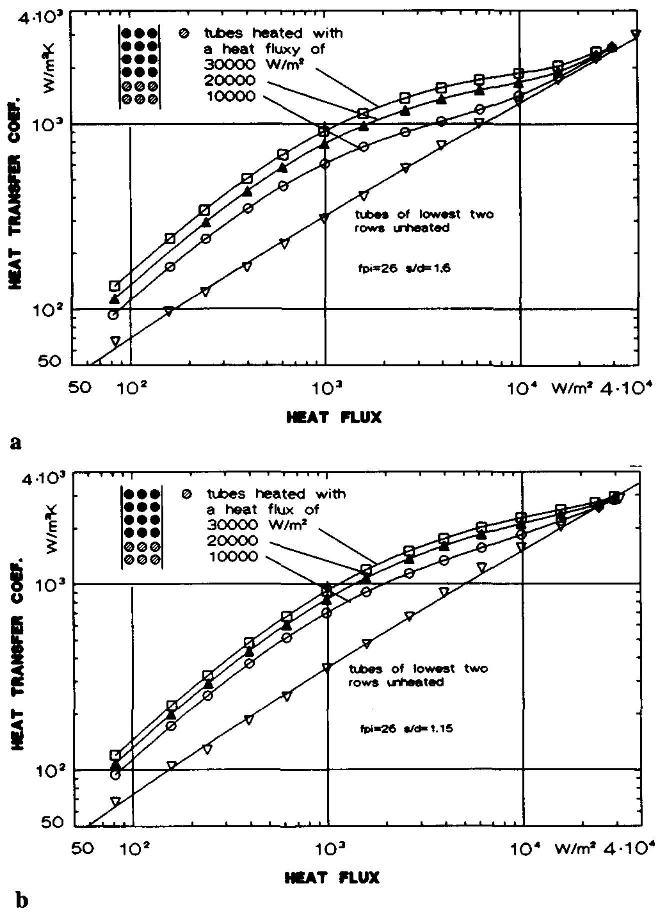


Figure 4. Enhancement of heat transfer of the upper four rows due to extra heating.

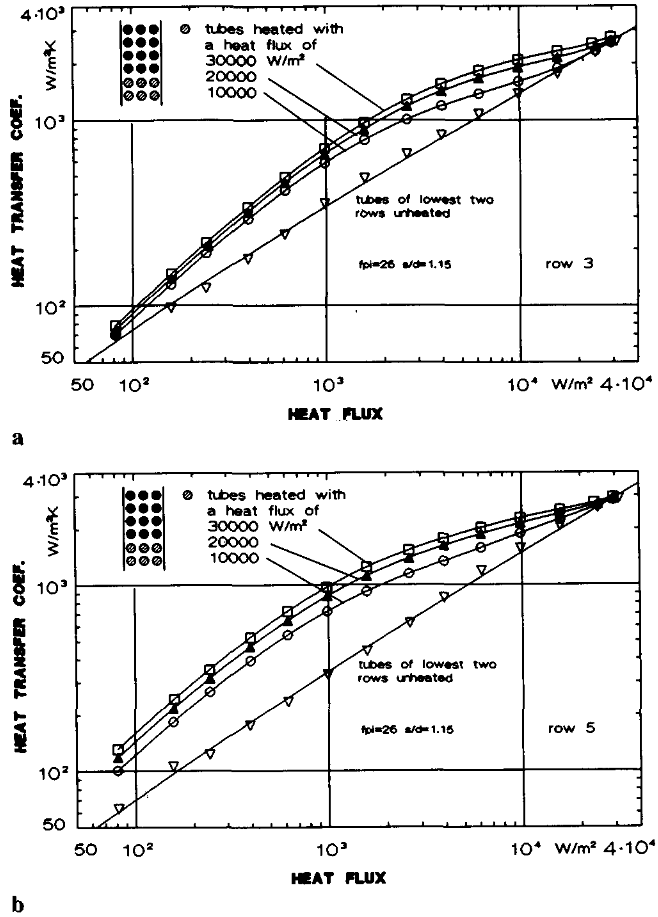


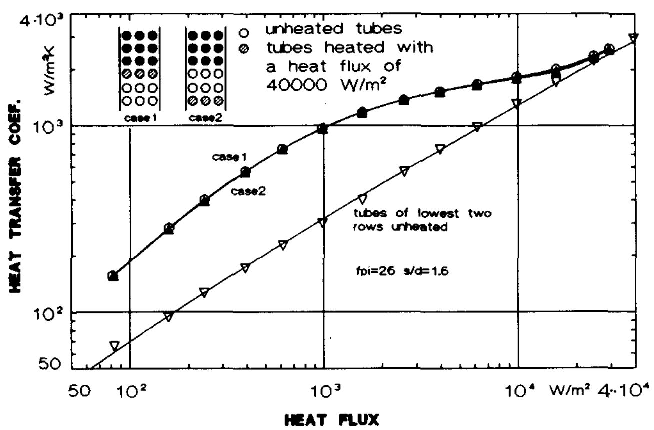
Figure 5. Heat transfer coefficients of rows 3 and 5 with extra heating.

the intermediate heat flux region between natural convection and fully developed boiling.

The heat transfer coefficients for each of the four upper rows were also measured. The heat transfer coefficients of rows 3 and 5 versus the heat flux for these tubes are plotted in Fig. 5. It can be seen that the enhancement of heat transfer for row 3 is much smaller than that for row 5 due to the strong two-phase flow induced by heating the tubes of the bottom two rows with a high constant heat flux. The difference in the heat transfer enhancement of different rows must be due to differences in the two-phase flow at the different positions. When the 12 tubes of the top four rows were heated with the same heat flux and the tubes of the lower two rows were unheated, the experimental results show that the heat transfer coefficient of row 3 was smaller than those of the other rows, and the heat transfer coefficient of row 6 was the largest. For example, when the tubes were heated with a heat flux of 9850 W/m^2, a tube pitch-to-diameter ratio $s/d = 1.15$, the heat transfer coefficients for rows 3–6 were 1382, 1454, 1567, and 1878 $W/(m^2K)$, respectively. Above the tube bundle, the cross section for the two-phase flow is much larger than that in the bundle. The bubbles created by the tubes of row 6, therefore, are much easier to move away. This is why the heat transfer coefficients of the row 6 tubes are much larger than those of the other tubes. As analyzed above, the tubes are intensely washed by the two-phase flow induced by heating the tubes of the lower two rows with a high heat flux, which enhances the boiling heat transfer of the bundle. It can be concluded that the enhancement of heat transfer of row 6 is smaller than that of row 5 because the washing effect of the two-phase flow is relatively smaller.

There are two reasons that result in the two-phase flow becoming stronger toward the top of the bundle. First, the bubbles become larger as the static pressure decreases, which makes the velocity of the two-phase flow increase. Second, the collisions among bubbles become more intense due to the insertion of the bubbles created by the tubes of rows 3–5, which results in stronger turbulence. According to this analysis, the enhancement of the heat transfer of row 3 due to the extra heating is smaller than that of the other three rows.

To study the effect of extra heating position on boiling heat transfer of the tube bundle, two experiments were performed. In case 1, the tubes of row 3 were heated with a constant heat flux of 40,000 W/m^2 and tubes of rows 1 and 2 were unheated; in case 2, the tubes of rows 2 and 3 were unheated and the tubes of row 1 were heated at a constant heat flux of 40,000 W/m^2. The experimental results are shown in Fig. 6. For the case of $s/d = 1.6$, the position of the extra heating does not affect the heat transfer enhancement of the tube bundle. For both cases 1 and 2, the mean heat transfer coefficients of the upper



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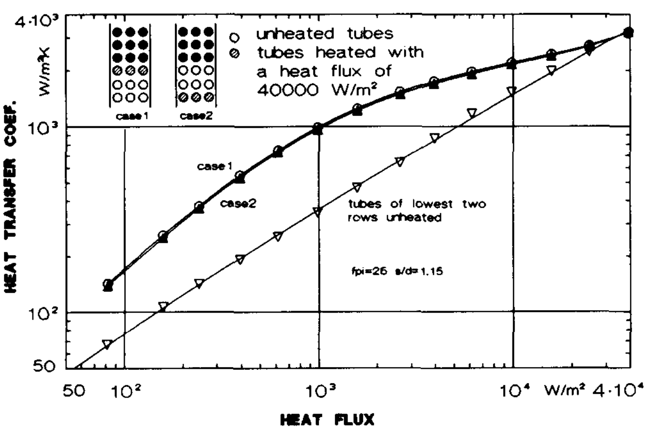

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Figure 6. Effect of the position of extra heating on heat transfer.

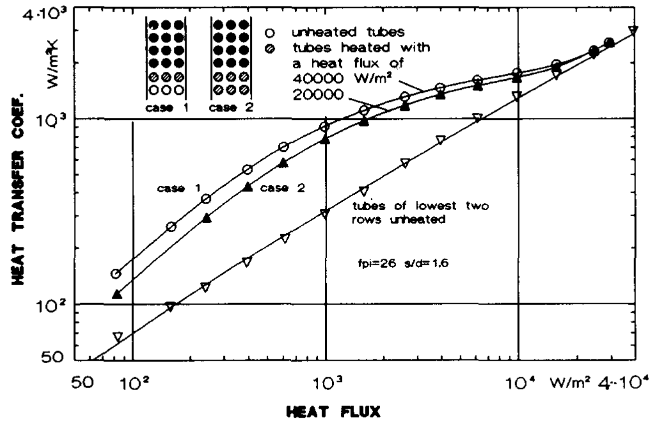

Figure 7. Effect of heat flux of extra heating on heat transfer.

three rows are almost the same. For the case of $s/d = 1.15$, the difference between the mean heat transfer coefficients of the three top rows obtained for cases 1 and 2 is very small. In other words, the two-phase flow induced by the extra heating does not depend on the position of the extra heating if the total heat flow and heat flux are unchanged. In contrast, the increases in heat transfer coefficients of the tube bundle were very different when the heat flux of the extra heating was different (total heat flow unchanged). Figure 7 shows the results for two cases, case 1, with tubes of row 1 unheated, and tubes of row 2 heated with a heat flux of 40,000 W/m^2, gives greater heat transfer enhancement for the top four rows than case 2, in which tubes of rows 1 and 2 were heated with a heat flux of 20,000 W/m^2. It is clear that the tubes heated with the larger heat flux (total extra heat flow unchanged) gives greater enhancement of heat transfer for the tubes of the top four rows. In experiments on boiling heat transfer it is very difficult to simulate a larger tube bundle, especially in the region of high heat flux because doing so requires a large power supply, a high-capacity cooling system, etc. Before we carried out our experiment it was suggested that tubes of one or two rows heated with high heat flux could be used to simulate more rows heated with a lower heat flux. Experimental results shown in Fig. 7 show that this idea does not work. The reason is that the two-phase flow induced by extra heating depends not only on the total heat flow but also on the heat flux for each tube.

The experimental results shown in Figs. 4–7 exhibit almost no scatter in heat transfer coefficients. It can be assumed, therefore, that the extra heating of the tubes of the lower rows has a stabilizing effect upon the heat transfer of the upper tubes.

Fluid Velocity under the Bundle and the Thermal Drag Phenomenon

To investigate the effect of extra heating on the heat transfer of the tube bundle and to clarify whether the thermal drag phenomenon exists in pool boiling, the fluid velocities under the bundle with and without extra heating were measured for the case of a tube pitch-to-diameter ratio $s/d = 1.15$. In our experiments, to simulate a fraction of a large bundle, the finned tube bundle was channeled by vertical glass plates. The fluid velocity under the tube bundle was measured by using a small velocimeter. Because the fluid velocity is unsteady under the action of two-phase flow, the instantaneous velocity is difficult to measure. Figure 8 shows the experimental results for time-averaged velocity. During a period of about 5 min, over 1500 velocity data were collected. The time-averaged velocity is the arithmetic mean of these data.

The fluid velocity is strongly dependent upon the heat flux of the tubes of the four top rows if the tubes of the

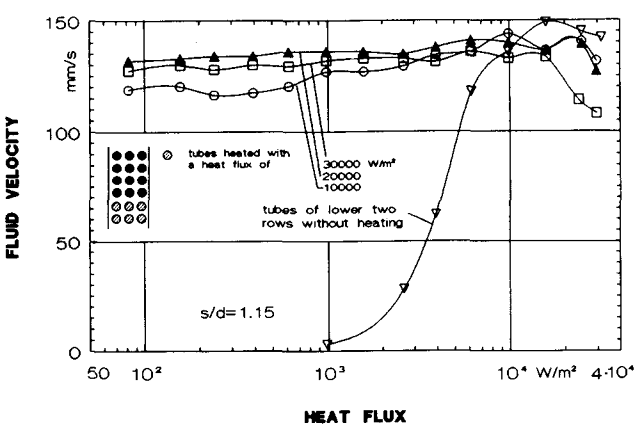

Figure 8. Fluid velocity under the bundle.

two bottom rows are unheated. When the heat flux is less than 1000 W/m^2, the fluid velocity is too low to measure with the velocimeter. When the heat flux is greater than 1000 W/m^2, the fluid velocity increases sharply with increases in the heat flux. When the heat flux is higher than 20,000 W/m^2, that is, in the fully developed boiling region, the fluid velocity decreases as the heat flux increases. A small fluid velocity means a small mass flow rate through the channel. Thus, this result indicates that the boiling heat transfer coefficient of the bundle would be smaller for very high heat flux.

The fluid velocity under the bundle with extra heating is very different from that without extra heating. Under the action of the strong two-phase flow induced by the extra heating, the fluid velocity under the bundle seems to be independent of the heat flux of the tubes of the four top rows. Only in the high heat flux region does the fluid velocity vary with the heat flux. An interesting result is that the fluid velocity first increases then decreases with increases in the heat flux of the tubes of the two lower rows. Among the three cases shown in Fig. 8, the intermediate amount of extra heating (20,000 W/m^2) gives the greatest fluid velocity in the region of $q < 10{,}000$ W/m^2. These results show us that a maximum fluid velocity exists. It is also seen that in the fully developed boiling region, a larger heat flux of extra heating corresponds to a smaller fluid velocity. This indicates that an increase in total heat flow would result in a decrease in the mass flow rate. This phenomenon is similar to that found in narrow-channel natural convective boiling [12]. The reason is that there are two forces acting on the fluid flow. Under our experimental conditions, the driving force of the fluid flow is buoyancy induced by heating and increases with heat flow. On the other hand, thermal drag resulting from the volume expansion due to heating also increases with increases in heat flow. Combining the actions of buoyancy and thermal drag, it can be concluded that a maximum fluid velocity must exist. These results indicate that a bundle heat exchanger must be designed to work in the region in which fluid velocity increases as heat flux increases.

MEASUREMENT ERROR

The uncertainty of the evaluated heat transfer coefficient can be estimated from the equation

$$\left|\frac{\Delta\alpha}{\alpha}\right| = \left|\frac{\Delta Q}{Q}\right| + \left|\frac{\Delta A}{A}\right| + \left|\frac{\Delta(\Delta\vartheta)}{\Delta\vartheta}\right|, \tag{3}$$

where $\Delta Q/Q$ and $\Delta A/A$ are about $\pm 0.6\%$ and $\pm 3.0\%$, respectively. The main error in the heat transfer coefficient results from the temperature measurements. All the thermocouples used in the experiments were calibrated. The absolute error of the difference between the outside wall temperature of the finned tube and the vapor temperature is about $\pm 0.21°$ C, which is estimated from the measurement error of the thermocouples and the error of the digital multimeter. In our experiments, the temperature differences used to calculate the heat transfer coefficient are very different for different values of heat flux. The minimum temperature difference is about 1.25° C in the natural convection region, and the maximum temperature difference as much as 20° C in the fully developed boiling region. Therefore, the relative error of the temperature difference ranges from $\pm 1.0\%$ to $\pm 16.6\%$, and the maximum uncertainty of the heat transfer coefficient is about $\pm 4.6\%$ in the fully developed boiling region and $\pm 20\%$ in the natural convection region. The uncertainty of time-averaged liquid velocity is about $\pm 5.0\%$, which is dependent only on the velocimeter; the uncertainty in the data processing can be neglected.

PRACTICAL SIGNIFICANCE

This study presents a method to enhance the heat transfer from a finned tube bundle. If there is strong two-phase flow induced by extra heating or by some other method, then the heat transfer in the intermediate region between natural convection and fully developed boiling will be strongly enhanced. The experimental results show that there is maximum fluid velocity as the heat flux increases. This indicates that the finned tube bundle should be controlled to work in the intermediate region where the fluid velocity under the bundle is below maximum.

CONCLUDING REMARKS

1. Remarkable enhancement of heat transfer of the top four rows in a six-row tube bundle was observed when the tubes of the two lower rows were heated with high constant heat flux. The greatest enhancement in the intermediate heat flux region was as much as 150% compared to heat transfer without extra heating. This indicates a way to enhance heat transfer in this region.
2. The position of extra heating has little effect on boiling heat transfer, and one row heated with high heat flux cannot simulate several rows heated with low heat flux even if the total heat flow is the same.
3. A maximum fluid velocity occurs as the heat flux increases. The fluid velocity under the bundle with extra heating is almost independent of the heat flux of the tubes of the upper rows for a heat flux less than 10,000 W/m^2. In the fully developed boiling region, the fluid velocity decreases as heat flux increases for both cases of the bundle with and without extra heating.
4. Pool boiling from a bundle channeled with two glass plates is much like natural convective boiling in a narrow vertical channel. The thermal drag resulting from volume expansion due to heating does affect the mass flow through the channel. Therefore, a bundle heat exchanger should be designed to work in the intermediate region between natural convection and fully developed boiling.

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NOMENCLATURE

A surface area, m^2
b fin thickness, mm
d diameter, mm

h fin height, mm
k thermal conductivity, W/(m K)
L tube length of the heated section, mm
Q heat flow rate, W
q heat flux, W/m^2
s tube pitch, mm
t fin spacing, mm

Greek Symbols

α heat transfer coefficient W/(m^2 K)
ϑ temperature, K

Subscripts

H heater
c channel
R root
wi inside wall
wo outside wall
1 tube base
2 tube tip

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