Int. J. Heat Mass Transfer. Vol. 10, pp. 1349–1359. Pergamon Press Ltd. 1967. Printed in Great Britain

# STUDIES OF HEAT-TRANSFER RATE OF A SUBCOOLED BOILING LIQUID IN TUBES

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(*Received* 10 *January* 1966)

**Аннотация**—Излагаются резудьтаты экспериментального исследования интенсивности теплообмена при кипении недогретой до температуры насыщения жидкости в трубах с большим отношением длины обогреваемого участка к диаметру в условиях низких давлений. Опыты проведены на воде и нормальном пропиловом спирте.

## NOMENCLATURE

$L, L_v$, distances between points of pressure and voltage tapping in the test section, respectively;

$L_h$, heated section length [m, mm];

$x_h$, distance from the heated section inlet to the cross-section where the local heat-transfer coefficient or wall temperature is to be determined [m, mm];

$d$, internal diameter of the test tube [m, mm];

$f$, cross-sectional area of the tube [m$^2$];

$\rho, \rho''$, liquid and vapour density, respectively [kg/m$^3$];

$c_p$, specific heat at constant pressure [J/kg°K];

$\zeta$, vaporization heat [J/kg];

$t_{in}$, liquid temperature at the inlet to the heated section [°C];

$t_l$, liquid temperature in the cross-section at the distance $X_h$ from the inlet [°C];

$t_s, T_s$, saturation temperature [°C, °K];

$t_{bo}$, temperature of the onset of high-rate boiling of subcooled liquid [°C];

$t_w$, temperature of the tube internal surface [°C];

$\Delta t_{sub}$, $= t_s - t_l$, local subcooling in a certain cross-section of the tube [°C];

$P_{in}$, absolute pressure at the inlet to the heated section [N/m$^2$];

$G', G''$, mass-flow rate of liquid and vapour phases, respectively [kg/s];

$G$, total mass-flow rate of both phases [kg/s];

$W$, $= G/\rho f$, liquid velocity [m/s];

$W'_0$, $= G/\rho f$, $W''_0 = G''/\rho'' f$, reduced velocity of liquid and vapour phases, respectively [m/s];

$\beta$, $= W''_0/W'_0 + W''_0$, vapour flow fraction in the absence of the "slip" effect of vapour phase;

$\varphi$, actual vapour volume flow fraction;

$q$, specific heat flux [W/m$^2$];

$\alpha_0$, convective heat-transfer coefficient in single-phase flow [W/m$^2$];

$\alpha_\varphi$, heat-transfer coefficient in vapour–liquid flow with a certain value of $\varphi$ with no effect of vaporization on the heat-transfer rate [W/m$^2$°K];

$\alpha_b$, heat-transfer coefficient in high-rate boiling of sub-cooled liquid [W/m$^2$ °K];

$\Delta p$, total pressure drop due to hydraulic resistance and flow acceleration [N/m$^2$];

$\Delta P_{fr}$, pressure drop due to friction in a liquid phase flow with a total mass-flow rate [N/m$^2$];

$Q$, $= q\pi d X_h$, heat quantity supplied to flow along the tube $X_h$ [W];

$Q$, heat quantity spent for liquid vaporization over the section length $X_h$ [W].

Similarity numbers

$K_w$, $= q/\zeta\rho''W$;

$K_s$, $\zeta/C_pT_s$.

RECENT research work [1, 2] shows that the transfer mechanism of boiling varies fundamentally depending on the conditions of the experiments. At low values of heat flux, heat is carried away from the region adjacent to the wall mainly by the superheated liquid which is displaced by vapour bubbles during their growth and also at the moment of their breaking. As the heat flux is increased the contribution of the phase–conversion heat in the form of vaporization in the direct vicinity to the heat-transfer surface begins to play an important part. At the same time, in each case, a considerable amount of liquid is moved towards the surface from the flow core which fact gives rise to additional disturbance of the region adjacent to the wall.

In pool boiling, mass transfer due to vaporization ensures essential increase of the heat-transfer rate even though heat flux values are considerably small. Under conditions of forced liquid motion the turbulence of a homogeneous flow (with appropriate values of the Reynolds numbers) may be sufficiently high, so that vaporization affects heat transfer only at certain relations between the vaporization rate $q/\zeta\rho''$ and liquid velocity $W$. Under the conditions of forced motion of saturated boiling liquid the region of the effect of mass transfer due to vaporization may be defined by inequality of [3].

$$\frac{q}{\zeta\rho''W}\left(\frac{\rho''}{\rho}\right)^{1\cdot45}\left(\frac{\zeta}{C_pT_s}\right)^{0\cdot333} > 0\cdot4 \times 10^{-5}. \quad (1)$$

In the case of a boiling liquid the main mass of which is below its saturation temperature, the heat-transfer rate is essentially influenced by the temperature of the flow core. Measurement of temperature fields shows that due to turbulent fluctuations the subcooled liquid penetrates deeply into the two-phase layer adjacent to the wall [4, 5] which results in the disturbance of a normal process of formation and growth of vapour bubbles. For substantial subcooling even if inequality (1) is satisfied, conditions may exist under which the vaporization process will not intensify the heat-transfer rate. Thus, in boiling of a subcooled liquid the lower boundary of the region in which the heat-transfer rate is affected by vaporization may be displaced considerably toward the larger values of the group in equation (1).

Subcooling of the flow core should also be accounted for when the effect of velocity on the heat-transfer rate is estimated. Increase of the flow turbulence with velocity intensifies heat transfer; but this is accompanied by a deeper penetration of a larger mass of subcooled liquid into the region adjacent to the wall. This curtails vaporization and leads to the reduction of the thickness of the two-phase layer.

In evaporators with long tubes subcooling decreases along the flow not only because heat is supplied to the liquid, but also because the saturation temperature decreases, which fact provides additional difficulties in the analysis and correlation of experimental data. The effect of the decrease of the saturation temperature is especially important when the pressure drop in the heated section is commensurable with the absolute pressure in the system. This case is very important in practice and is of great theoretical interest. In order to study the phenomena under these conditions, the present experiments have been carried out at low pressures, using the tube with a large ratio of the heated length to the diameter, $L_h/d$.

## EXPERIMENTAL UNIT AND EXPERIMENTAL PROCEDURE

The study of the heat-transfer rate of boiling water and normal propyl alcohol was carried out using an apparatus consisting of a closed loop with forced circulation provided by a glandless pump. The test section (Fig. 1) was made of a horizontal brass tube with an internal diameter of 8·97 mm and wall thickness of 0·5 mm. The tube was heated by alternating electric current of low voltage. The length of heated section of tube No. 1 was 522 mm, that of tube No. 2 was 896 or 1207 mm.

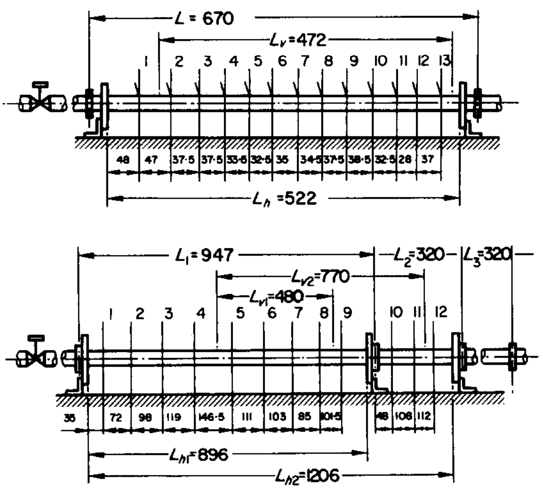


FIG. 1. Schematic drawing of the experimental set. Nos. 1–13 are thermocouples.

The temperature of the heat-transfer surface was measured with copper–constantan thermocouples, 0·2 mm in diameter, which were tin soldered to the external tube surface.

Butt-soldered thermocouples were mounted exactly perpendicularly to the external surface of the tube. The solder spot was filed to the thickness of the thermocouple junction and then the wires insulated from the tube with a thin layer of mica were wound one turn round the tube. The e.m.f. of the thermocouple was recorded by a potentiometer with a sensitive galvanometer. The thermocouples were calibrated against a thermometer with divisions of 0·1 degC. The schematic diagram of the location of the thermocouples on the heated sections is shown in Fig. 1.

For measurements of static pressure 0·9 mm holes were drilled in the tube. In each cross-section the holes were linked by a circular chamber. Measured were the absolute pressure at the inlet and the pressure drops along the tube. However, the data on hydraulic resistance were not analysed in the present work. Measurements of the pressure drop allowed systematic control of the surface condition of the test tubes. All runs showed that the tubes may be considered absolutely smooth. Deviations of experimental friction resistance coefficients from those predicted by the Blasius formula over the whole range of the Reynolds number did not exceed $\pm 4$ per cent.

To maintain pulsation-free conditions, a butterfly valve was mounted upstream of the test tube. The distance between the butterfly valve and the inlet cross-section of the heated portion for tube No. 1 was about 100*d*, and 60*d* for tube No. 2.

The temperature of the working fluid at the inlet to the heated section was measured by a laboratory thermometer and a thermocouple soldered to the tube wall in front of a current-supplying flange. This thermocouple fixed at the unheated section of the tube provided very exact readings of the liquid temperature. The cooler followed by the pump and electric heater mounted before the test tube (the butterfly valve) allowed smooth adjustment of the liquid temperature at the test section inlet. In all runs this temperature was below the saturation temperature corresponding to the inlet pressure. The rate of the circulating liquid was measured by a double membrane connected to a differential manometer filled with dichlorethane.* In the experiments with water the pressure in the system was created by vapour generated in a special tank. The tank was connected to a separator of the main loop by a tube of a small diameter. In experiments with alcohol, pressure in the system was created by nitrogen which was supplied to the separator from a special vessel. The very small solubility of nitrogen in

* In experiments with alcohol, the usual inverted U-tube manometer was used, the rate being measured with a circulating liquid column.

normal propyl alcohol was taken into account but the authors considered that it could not have any noticeable effect on the heat-transfer coefficient.

The experimental study consisted of a series of runs. In each series the heat flux, liquid velocity and pressure at the inlet to the heated section were constant. The temperature at the inlet to the heated section was variable. Thermocouple readings fixing the temperature of the heating surface were recorded under steady-state conditions. If the thermocouple readings showed changes of temperature exceeding 0·2 degC, the run was considered inadequate.

Before each series of runs water was degassed by long-time boiling and blowing of the unit.

## GENERAL RESULTS

Tables 2 and 3 and Figs. 4 and 5 show the temperature data in a heated section for certain sets of runs. In two sets of runs 1–8 and 31–39, the value of the group in the left-hand side of inequality (1) is less than $0{\cdot}4 \times 10^{-5}$. The heat-transfer rate in these experiments is independent of the process of vapour bubble generation at the heating surface. It should be noted that independence of the heat-transfer rate of vaporization process is also found in runs 9–14 although in this case the value of the above group exceeds somewhat its boundary value of $0{\cdot}4 \times 10^{-5}$. In this connexion the heat-transfer coefficients in a given cross-section of the tube are estimated from the ratio of the heat flux to the difference of the wall temperature and the liquid temperature determined from the heat-balance equation:

$$q \, . \, \pi d x_h = \frac{\pi d^2}{4} W \, . \, \rho \, . \, C_p (t_l - t_{in}). \tag{2}$$

In Fig. 2(a) the distribution of the dimensionless heat-transfer factor over the length of the heated section is plotted from the data of runs 9–14. As a scale for $\alpha_\varphi$, the value of the heat-transfer coefficient in convective heat transfer in a single phase flow $\alpha_0$ was taken which was determined according to the formula of Mikheev [6]. The values of $\alpha_0$ were calculated also from formula of Hausen [7]. These points for run No. 9 are designated by empty circles. Figure 2(a) shows that the present data for this run are 4–5 per cent higher than $\alpha_0$ determined by

*Table 1. Comparison of actual local vapour volume flow fractions $\varphi$ predicted by formulae (3) and (4)*

| Run No. | | Dimensionless distance from the heated section inlet $x_h/d$ | | | |
|---|---|---|---|---|---|
| | | 20 | 30 | 40 | 50 |
| 11 | $\varphi$ from [3] | — | — | — | 0·1 |
| | $\varphi$ from [4] | — | — | — | 0·133 |
| 12 | $\varphi$ from [3] | — | 0·184 | 0·323 | 0·474 |
| | $\varphi$ from [4] | — | 0·165 | 0·342 | — |
| 13 | $\varphi$ from [3] | 0·124 | 0·310 | 0·458 | 0·562 |
| | $\varphi$ from [4] | 0·180 | 0·320 | — | — |
| 14 | $\varphi$ from [3] | 0·210 | 0·390 | 0·517 | 0·610 |
| | $\varphi$ from [4] | 0·216 | 0·370 | — | — |

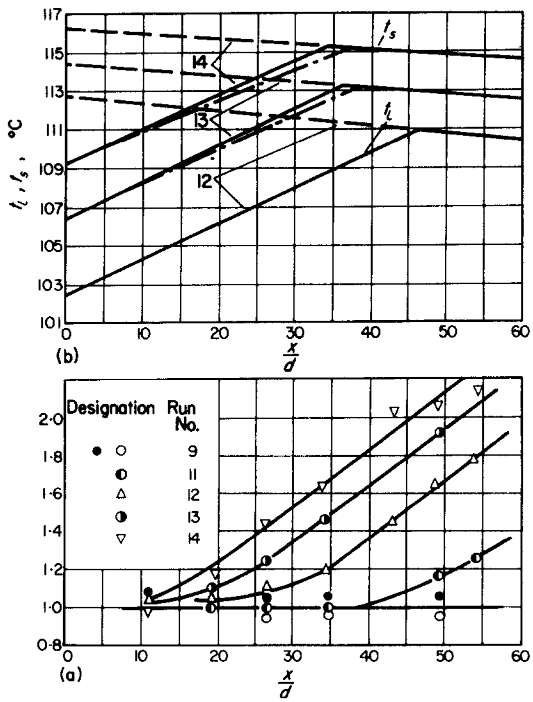


FIG. 2. Heat-transfer coefficient (a) liquid temperature and saturation temperature (b) along the heated section.

*Table 2. Experimental data on heat transfer to water. Tube No. 1*

| Run No. | $T$ $t_w$ (°C) | | | | | | $t_{in}$ (°C) | $P_{in} \times 10^{-4}$ (N/m²) | $q \times 10^{-3}$ (W/m²) |
|---|---|---|---|---|---|---|---|---|---|
| | 2 | 4 | 6 | 8 | 10 | 12 | | | |
| | | | | $W$ = 1·2 m/s | | | | | |
| 1 | 115·1* | 115·9 | 116·6 | 117·2 | 117·6 | 118·6 | 95·7 | 14·81 | 149·7 |
| 2 | 117·0* | 117·6 | 118·2 | 118·9 | — | 120·3 | 97·3 | 14·91 | 149·0 |
| 3 | 118·0* | 118·7 | 119·4 | 120·1 | 120·5 | 121·1 | 98·6 | 14·91 | 152·0 |
| 4 | 119·7* | 120·7 | 121·3 | 121·6 | 121·6 | 122·3 | 100·7 | 14·91 | 152·0 |
| 5 | 122·6* | 122·8 | 122·6 | 121·7 | 121·1 | 121·1 | 105·4 | 15·10 | 152·2 |
| 6 | 124·4* | 124·2 | 122·8 | 122·6 | 121·7 | 121·1 | 108·1 | 15·50 | 153·3 |
| 7 | 125·0* | 125·0 | 124·4 | 123·4 | 122·2 | 121·9 | 109·6 | 15·60 | 154·5 |
| 8 | 126·2* | 126·2 | 124·4 | 122·6 | 120·9 | 120·0 | 110·3 | 16·00 | 154·5 |
| 9 | 105·1 | 106·6 | 108·9 | 110·4 | — | 113·3 | 77·2 | 14·64 | 245·5 |
| 10 | 119·8 | 120·6 | 121·4 | 123·1 | — | 126·0 | 93·0 | 14·64 | 237·0 |
| 11 | 123·1 | 124·6 | 125·2 | 126·6 | 125·6 | 126·2 | 96·4 | 14·86 | 236·0 |
| 12 | 127·8 | 128·3 | 125·5 | 128·3 | 126·8 | 126·0 | 102·0 | 15·73 | 233·8 |
| 13 | 132·4 | 131·6 | 130·0 | 129·0 | 127·3 | 126·1 | 106·4 | 16·73 | 233·8 |
| 14 | 134·7 | 132·6 | 130·1 | 129·7 | 127·2 | 126·9 | 108·9 | 17·69 | 233·8 |
| 15 | 127·6 | 128·3 | 128·9 | 129·7 | 130·3 | 130·1 | 80·5 | 14·42 | 412·0 |
| 16 | 128·7 | 129·8 | 130·4 | 129·8 | 128·6 | 127·0 | 85·4 | 14·91 | 425·0 |
| 17 | — | 129·3 | 129·5 | 128·9 | 128·1 | 127·7 | 89·8 | 14·91 | 429·0 |
| 18 | 134·2 | 130·4 | 130·6 | 129·2 | — | 128·6 | 90·8 | 15·20 | 430·0 |
| 19 | 135·2 | 132·3 | 131·0 | 129·8 | — | 128·5 | 93·8 | 15·70 | 430·0 |
| 20 | 135·2 | 132·3 | 129·6* | 127·1 | — | 127·1 | 95·3 | 15·50 | 431·0 |
| 21 | 133·7 | 134·9 | — | 137·1 | 137·4 | 137·1 | 109·7 | 24·70 | 226·0 |
| 22 | 136·9 | 137·7 | 138·4 | 139·8 | 138·8 | 137·6 | 112·5 | 24·70 | 223·0 |
| 23 | 142·5 | — | 142·5* | 141·1* | 139·4* | 137·3* | 117·0 | 24·97 | 226·0 |
| 24 | 143·5* | 143·5* | 141·8* | — | 139·0* | 135·5* | 119·3 | 25·48 | 228·0 |
| 25 | 133·1 | 135·4 | 138·2 | 139·2 | 140·0 | 142·1 | 90·0 | 24·70 | 388·0 |
| 26 | 137·8 | 138·8 | 141·1 | 142·3 | 142·8 | — | 95·1 | 24·70 | 383·0 |
| 27 | 143·4 | 143·8 | 145·8 | 146·4 | 145·5 | — | 101·0 | 24·60 | 388·0 |
| 28 | 145·0 | 145·3 | 146·9 | 145·7 | 144·0 | 143·1 | 103·2 | 24·69 | 389·0 |
| 29 | 144·5 | 145·9 | 148·6 | 144·3 | — | — | 104·0 | 24·72 | 388·0 |
| 30 | 145·5 | 146·8 | 147·2 | 145·0 | 143·0 | — | 105·4 | 24·83 | 399·0 |
| | | | | $W$ = 1·5 m/s | | | | | |
| 31 | 112·9 | 113·1 | 114·2 | 114·7 | 114·9 | 115·8 | 99·2 | 14·89 | 142·8 |
| 32 | 113·9 | 114·2 | 115·3 | 115·8 | 116·1 | 117·2 | 100·0 | 15·00 | 141·5 |
| 33 | 114·6 | 114·4 | 115·6 | 115·7 | 115·8 | 116·8 | 100·9 | 15·00 | 141·5 |
| 34 | 117·0 | 116·7 | 117·8 | 118·6 | 118·8 | 119·2 | 104·3 | 14·88 | 144·0 |
| 35 | 118·8 | 118·4 | 119·7 | 120·2 | 119·9 | 119·7 | 106·0 | 15·20 | 144·0 |
| 36 | 120·5 | 120·4 | 120·8 | 121·4 | 120·7 | 120·9 | 108·6 | 15·42 | 144·0 |
| 37 | 124·0 | 122·8 | 122·8 | 122·1 | 121·3 | 120·3 | 110·5 | 16·75 | 145·3 |
| 38 | 123·6* | — | 121·2 | — | 119·5 | 118·3 | 113·2 | 16·50 | 145·3 |
| 39 | 122·4* | — | 120·5* | — | 118·9 | 117·6 | 113·6 | 16·64 | 145·3 |
| 40 | 125·3 | 127·7 | 130·1 | 130·9 | 131·8 | — | 50·6 | 14·47 | 721·0 |
| 41 | — | 129·4 | 130·4 | 131·2 | 132·9 | 133·9 | 55·5 | 14·47 | 702·0 |
| 42 | — | 133·8 | 135·2 | 136·4 | 136·4 | 136·5 | 63·0 | 14·49 | 725·0 |
| 43 | — | 137·1 | 137·6 | 138·4 | 137·1 | — | 71·0 | 14·38 | 725·0 |
| 44 | 139·3 | 138·3 | 137·5 | 138·8 | 136·4 | 135·5 | 73·7 | 14·58 | 746·0 |
| 45 | 140·5 | 138·8 | 138·0 | 139·2 | 136·3 | 134·9 | 75·1 | 14·70 | 752·0 |
| 46 | 139·6 | 137·9 | 137·4 | 137·8 | 135·5 | 134·0 | 76·0 | 14·59 | 722·0 |
| 47 | 133·1 | 133·5 | — | 135·9 | 136·3 | 136·8 | 46·6 | 14·51 | 856·0 |
| 48 | — | 136·2 | 137·1 | 137·5 | — | 137·7 | 58·0 | 14·51 | 879·0 |

*Table* 2 (*continued*)

| Run No. | $T$ $t_w$ (°C) | | | | | | $t_{in}$ (°C) | $P_{in} \times 10^{-4}$ (N/m$^2$) | $q \times 10^{-3}$ (W/m$^2$) |
|---|---|---|---|---|---|---|---|---|---|
| | 2 | 4 | 6 | 8 | 10 | 12 | | | |
| 49 | 139·2 | 138·7 | 139·7 | 139·6 | 139·8$_*$ | 139·5 | 61·9 | 14·51 | 892·0 |
| 50 | 140·4$_*$ | 137·9 | 136·9 | — | 135·5 | 134·4 | 71·1 | 14·30 | 879·0 |
| 51 | 140·4$_*$ | 137·1 | 137·3 | 136·7 | 134·9 | 133·3 | 76·2 | 15·60 | 856·0 |
| 52 | 150·4 | 149·9 | 150·1 | 149·9 | 149·3 | — | 75·6 | 24·59 | 872·0 |
| 53 | 151·5 | 150·5 | 149·9 | 149·1 | 148·3 | — | 82·8 | 24·71 | 872·0 |
| 54 | 151·4 | 149·7 | 149·5 | 148·5 | 147·9 | — | 85·5 | 24·82 | 872·0 |
| 55 | 151·5 | 150·1 | 149·7 | 148·1 | 147·3 | 147·5 | 88·0 | 25·06 | 872·0 |
| 56 | 150·4 | 149·7 | 148·9 | 147·7 | 147·7 | 146·8 | 90·4 | 25·30 | 877·0 |
| 57 | 150·8 | 149·5 | 149·3 | 147·5 | 146·5 | 146·5 | 92·0 | 25·44 | 877·0 |
| | | | | $W$ = 2·0 m/s | | | | | |
| 58 | 121·4 | 122·4 | 123·0 | 124·1 | 124·7 | 125·8 | 88·8 | 14·81 | 418·0 |
| 59 | 123·4 | 124·1 | 125·1 | 126·5 | 127·9* | 128·4 | 91·6 | 14·85 | 423·0 |
| 60 | 125·9 | 126·9 | 128·2 | 129·3 | — | 128·3 | 96·2 | 15·30 | 412·0 |
| 61 | 129·7 | 130·3 | — | 130·5 | — | 129·5 | 99·7 | 16·30 | 417·5 |
| 62 | 129·9 | — | 133·8 | 135·7 | 136·3 | 136·5 | 79·0 | 14·61 | 737·5 |
| 63 | 134·0 | 136·3 | — | 139·1 | 138·9 | — | 85·5 | 15·40 | 741·0 |
| 64 | — | 136·8 | 138·6 | 139·7 | 139·4 | — | 87·0 | 15·60 | 746·0 |
| 65 | — | 137·8 | 140·1 | 140·0 | — | — | 87·6 | 15·90 | 749·0 |
| 66 | 128·8 | 130·6 | 132·9 | 134·7 | 135·5 | 134·9 | 55·3 | 14·00 | 921·0 |
| 67 | 131·3 | 132·7 | 135·1 | 136·6 | 137·4 | 136·6 | 59·2 | 14·52 | 950·0 |
| 68 | 133·3 | 134·6 | 136·6 | 138·6 | 138·6 | 138·4 | 61·6 | 14·52 | 932·0 |
| 69 | 135·4* | 136·3 | 138·6 | 139·9 | 139·5 | 139·2 | 67·1 | 14·52 | 923·0 |
| 70 | 138·6* | 138·8 | 140·2 | 138·8 | 138·6 | — | 74·9 | 14·77 | 923·0 |
| 71 | 138·3* | 138·8 | 140·5 | 139·2 | 138·8 | — | 76·0 | 15·05 | 923·0 |
| 72 | 139·4* | 139·9 | 140·3 | 139·7 | 139·6 | 138·8 | 78·0 | 15·20 | 928·0 |

* Figures 2, 4, 6, 8, 10 and 12 designate the thermocouple numbers according to their location along the heated section (see Fig. 1). The asterisk above the figure is a reading of the thermocouple intermediately following that of interest down the flow. The asterisk below the figure is the reading of the preceding thermocouple.

formula of [6], and 6–7 per cent lower than the values of $\alpha_0$ determined by formula of reference [7]. Similar relations for convective heat-transfer regimes free of vaporization are also obtained for other runs. In the calculation of the local heat-transfer coefficient $\alpha_0$ from the formula of reference [7], the correction for the effect of the tube length is taken in the form $[1 + \frac{1}{3}(d/x_h)^{\frac{2}{3}}]$.

Figure 2(a) shows also that with a certain subcooling of the liquid at the tube inlet, the ratio $\alpha_\varphi/\alpha_0$ becomes larger than unity. The less subcooling, the higher the rate of this increase. With the reverse sign of inequality (1), the excess of the heat-transfer coefficient $\alpha_\varphi$ over its value in the convective heat transfer in a single-phase medium is possible only due to vapour volume fraction $\varphi$ increasing down the flow which causes an increase of the true velocity of the liquid. Since in a turbulent flow the value of $\alpha_\varphi$ is proportional to the velocity to the power 0·8, the value of $\varphi$ in any section of the tube is readily obtained from the known distribution $\alpha_\varphi/\alpha_0 = f(x_h/d)$, [8]. For low pressures, when the ratio $\rho''/\rho \gg 1$, the relation between $\varphi$ and $\alpha_\varphi/\alpha_0$ is of a very simple form:

$$\varphi = 1 - \left(\frac{\alpha_0}{\alpha_\varphi}\right)^{1\cdot25}. \tag{3}$$

In Table 1 a comparison of the local values of $\varphi$ determined by formula (3) with those predicted according to formula (4) is given:

*Table 3. Experimental data on heat transfer to normal propyl alcohol. Tube No. 2, $L_h$ = 1206 mm*

| Run No. | $T$ $t_w$ (°C) | | | | | | $t_{in}$ (°C) | $P_{in} \times 10^{-4}$ (N/m²) | $q \times 10^{-3}$ (W/m²) |
|---|---|---|---|---|---|---|---|---|---|
| | 2 | 4 | 6 | 8 | 10 | 12 | | | |
| | | | | $W$ = 2·0 m/s | | | | | |
| 73 | 125·4 | 126·9 | 127·1 | 127·3 | 130·8 | 130·6 | 87·8 | 23·65 | 150·0 |
| 74 | 129·0 | 134·3 | — | 133·1 | 136·1 | 135·3 | 98·1 | 23·82 | 142·0 |
| 75 | 135·4 | — | 134·7 | 134·7 | 136·3 | 135·7 | 101·1 | 24·10 | 142·0 |
| 76 | 134·7 | — | — | 132·4 | 131·8 | — | 106·9 | 22·65 | 142·0 |
| 77 | 126·5 | 128·3 | 130·6 | 129·0 | — | 130·2 | 68·3 | 24·62 | 229·0 |
| 78 | — | 129·6* | 133·2 | — | 134·3* | — | 78·3 | 24·62 | 230·0 |
| 79 | 129·8 | 132·0 | — | 134·1 | 133·9 | 134·7 | 82·9 | 24·62 | 230·0 |
| 80 | 134·9 | 136·5 | 138·5 | 137·9 | 137·5 | 137·1 | 94·0 | 24·62 | 230·0 |
| 81 | — | 137·3 | 139·1 | 138·3 | 137·7 | 136·7 | 95·8 | 24·62 | 229·0 |
| 82 | 137·3 | 138·7 | 139·5 | 139·1 | 137·6 | 138·9 | 97·5 | 24·62 | 227·0 |
| | | | | $W$ = 1·95 m/s | | | | | |
| 83 | 136·1 | 137·1 | 139·5 | 139·7 | 139·3 | 139·4 | 77·0 | 24·42 | 392·0 |
| 84 | 136·3 | 137·2 | — | 139·5 | 139·3 | 139·5 | 78·3 | 24·42 | 392·0 |
| 85 | 137·3 | 138·5 | — | 140·3 | 139·1 | 138·1 | 80·8 | 24·62 | 388·0 |
| 86 | 137·9 | — | 140·9 | — | 139·1 | 138·0 | 81·6 | 24·81 | 385·0 |

$$\varphi = 1 - 12\left(\frac{q \cdot \Delta t_{\text{sub}}}{W}\right)^{0.25} \cdot \frac{1}{p^{0.07}}. \quad (4)$$

Formula (4) is obtained in [9] from the experimental measurements of $\varphi$ by the tracer method. In the calculation of $\varphi$ from formula (4) the local subcooling in a certain cross-section is determined allowing for the change of the saturation temperature along the tube. The variation of the saturation and liquid temperature along the tube for runs 12–14 is shown in Fig. 2(b). As is shown in Table 1, the values of $\varphi$ determined from different formulas are in fair agreement.

In the calculation of local values of the heat-transfer coefficient in boiling subcooled liquid, some workers determine the flow temperature in the intermediate cross-section of the tube by linear interpolation between the measured temperatures of liquid at the inlet and outlet of the heated section. In this case at the outlet from the heated section the thermocouple is usually placed in a mixing chamber to provide complete condensation of vapour. The present experiments have demonstrated that a thermocouple placed in the mixing chamber may read an entirely uncertain temperature since in a large number of cases there is no complete condensation either in the tube or in the mixing chamber. In the case when the liquid temperature becomes the same as the saturation temperature in the middle sections of the tube [runs 12–14, Fig. 2(b)], the interpolation method is quite inadequate. It is also impossible to determine the saturation temperature distribution over the heated section from the reading of pressure at the inlet and the readings of the thermocouple placed in the mixing chamber, since the prediction of the law of pressure change along the tube between the outlet of the heated section and the mixing chamber is impossible. The present measurements of the pressure drop over the unheated section which is directly preceded by the heated one, have shown that with certain combination of parameters ($P_{in}$, $t_{in}$, $w$, $q$), the pressure at the outlet section does not fall, but in fact increases. At low pressures the reversible component of the total pressure drop due to acceleration of the liquid and vapour phases may be considerably higher than the irreversible component, therefore when vapour is condensed, pressure recovery occurs in the unheated section. In Fig. 3 a typical curve is presented of the pressure drop

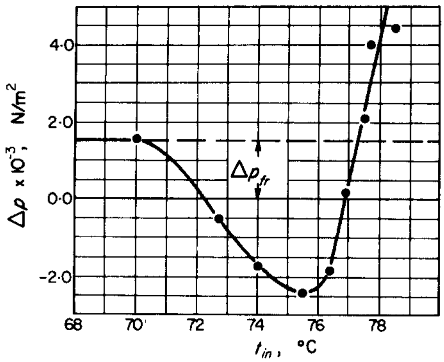


FIG. 3. Pressure drop over the non-heated section (which directly follows the heated one) vs. the liquid temperature at the inlet to the heated tube ($l_h$ = 896 mm). $P_{in}$ = 14·71 N/m²; $W$ = 2 m/s; $q$ = 663000 W/m².

in the unheated outlet section of a length of 320 mm vs. the liquid temperature at the inlet to the heated section ($l_h$ = 896 mm, tube No. 2). This set of runs was carried out with the following parameters at the heated section: $P_{in} = 14{\cdot}72 \,.\, 10^4$ N/m²; $W = 2{\cdot}0$ m/s; $q = 663000$ W/m².

In the cases when one portion of heat supplied to the surface of the vapour generating tube is spent on vaporization and the other for increasing the liquid enthalpy, a question always arises about the correspondence of the liquid temperature determined from the heat-balance equation to its true temperature at the same tube section. At low pressures vapour mass flow fractions at the end of the heated section are very low in boiling of subcooled liquid, therefore the portion of heat spent on vapour generation is considerably smaller than that spent on the increase of the liquid enthalpy. Therefore the actual integral mean temperature of the liquid will approach the temperature determined by equation (2).

Using the values of $\varphi$ from Table 1, the authors have plotted the true distribution of the liquid along the tube in runs 13 and 14. This distribution is shown in Fig. 2(b) by the dot–dash line. When plotting this figure two assumptions were made: (1) The absence of the "slip" effect of the vapour phase, i.e. for any cross-section of the tube $\varphi = \beta$ (2) the circulation velocity $W = G/\rho f$ is equal to the reduced velocity of the liquid phase $W'_0 = G/\rho f$. With these assumptions made and taking that $\beta = W''_0/(W'_0 + W''_0)$, the heat quantity spent in vapour generation is calculated:

$$Q'' = G''\,.\,\zeta = \rho'' W \frac{\varphi}{1-\varphi} f\zeta. \qquad (5)$$

The difference between the values of $Q = q\,.\,\pi d x_r$ and $Q''$ is spent on the increase of the liquid enthalpy. If the slip effect is taken into account, then in expression (5) the quantity $\varphi$ must be replaced by $\beta$. Since in the presence of the slip effect, $\beta > \varphi$, for the same values of $\varphi$ $Q''$ will be larger and the curve of the true temperature distribution will be lower. In this case for the determination of a new value of $\varphi$ one more correction for the new distribution $\alpha_\varphi/\alpha_0$ is necessary. The corrections act in different directions. The calculations show that under these conditions allowance for vapour slip has practically no effect on the distribution of the actual liquid temperature along the tube.

Thus, if the value of the group in the left-hand side of inequality (1) is less than $0{\cdot}4 \times 10^{-5}$, then the heat-transfer coefficient along the whole tube may be calculated by the formula for convective heat transfer in a single-phase medium, but true liquid velocity should be taken. In other words, in this case a reliable formula is necessary for the determination of the actual vapour flow fraction.

Figure 4 represents the temperature distribution at the wall along the heated portion for a set of runs in which the value of the group of unequality (1) exceeds considerably $0{\cdot}4 \times 10^{-5}$. Under these conditions the heat-transfer rate may change within a very wide range depending on the subcooling. With high subcoolings (run 1), the heat-transfer coefficient over the whole tube length is the same as the convective heat-transfer coefficient in a single phase liquid. As the subcooling decreases, the wall temperature increases, that favours nucleation and growth of vapour bubbles at the heating surface, increases the number of operating nucleation centres and the thickness of the two-phase layer adjacent to the wall. All this increases the heat-transfer rate due to both mass transfer involved by vaporization process and basic turbulent transfer, since when subcooling of the liquid is relatively large, the velocity effect still remains strong. The typical wall temperature distribution for these conditions is shown in Fig. 4 for runs 2–5.

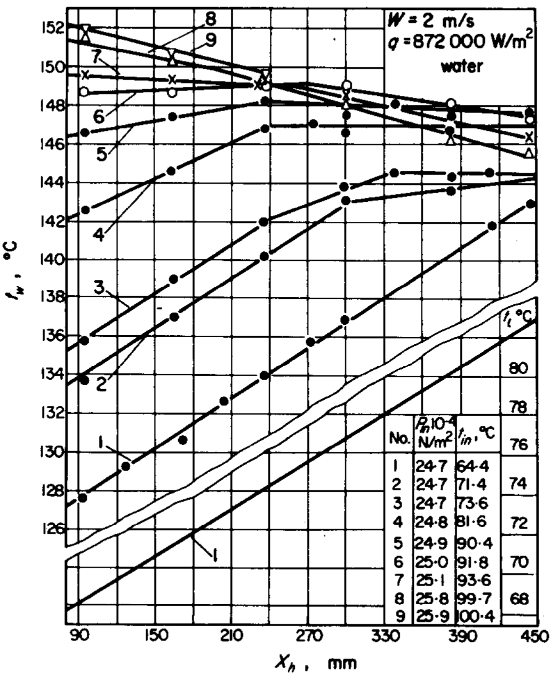


FIG. 4. Change of wall and liquid temperature along the heated tube.

When subcooling reaches a certain value depending on the parameters $q$, $P_{in}$, $W$, the heat-transfer rate does not depend any longer on the temperature of the core of the flow and becomes the same as in boiling of saturated liquid (runs 8 and 9). For this region the heat-transfer coefficient is better determined using the saturation temperature.

The flow temperature at which the heat-transfer coefficient of subcooled boiling liquid becomes the same as the heat-transfer coefficient of saturated boiling liquid is referred to as the temperature of intensive boiling onset and is designated by $t_{bo}$. Usually this temperature is experimentally determined on the basis of measured wall temperatures along the vapour generating tube. The flow temperature becomes equal to the temperature $t_{bo}$ in the tube cross-section where the wall temperature stops growing down the flow. However, such a method is quite inadequate in low-pressure boiling when low vapour mass flow fraction gives rise to very high vapour volume flow fraction. Under these conditions, the constant wall temperature at the end of the heated section (runs 2–4) or even its decrease down the flow (runs 5 and 6) may be attributed to the increase of the flow velocity rather than to the onset of high-rate boiling. To confirm this a typical set of runs was carried out, represented in Fig. 5 which shows the wall-temperature distribution (Fig. 5(a), the heat-transfer coefficient Fig. 5(b) and pressure Fig. 5(c) along the heated section obtained on tube No. 2 ($L_h$ = 896 mm). At the beginning of the tube, the wall temperature increases, the heat-transfer coefficients and hydraulic resistance have the same values as in a single-phase turbulent flow. In the second half of the tube an essential decrease of the wall temperature is observed, caused by the increase of the heat-transfer rate. This is accompanied by a rapid increase of resistance. In this case even at the point of maximum, the wall temperature does not attain the value which would occur in boiling of a saturated liquid, with other things being equal. Consequently, the growth of the hydraulic resistance and heat-transfer rate in these experiments may be mainly attributed to the increase of the flow velocity.

In some experiments of the present authors, in the starting portion of the tube or closer to its middle cross-section, the heat-transfer rate was the same as in boiling of saturated liquid. The results of these experiments agree well with the predicted relation of [3] for each case of saturated liquid boiling in tubes (Fig. 6). In Fig. 6 for each run are given the thermocouple numbers designating the cross-section for which

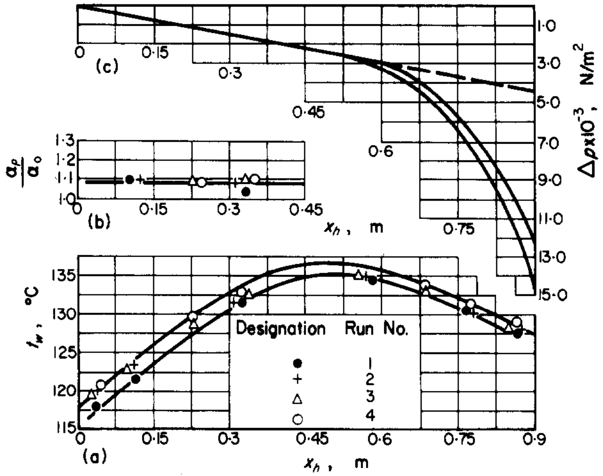


FIG. 5. Change of the wall temperature (a), heat-transfer coefficient (b) and pressure (c) along the heated section. Tube No. 2. $L_h$ = 896 mm; $W$ = 2 m/s; $q$ = 663000 W/m².
1. $P_{in} = 14{\cdot}75 \times 10^4$ N/m²; $t_{in} = 76{\cdot}7$°C.
2. $P_{in} = 15{\cdot}10 \times 10^4$ N/m²; $t_{in} = 78{\cdot}3$°C.
3. $P_{in} = 15{\cdot}15 \times 10^4$ N/m²; $t_{in} = 78{\cdot}7$°C.
4. $P_{in} = 15{\cdot}23 \times 10^4$ N/m²; $t_{in} = 79{\cdot}2$°C.

the heat-transfer coefficient is to be determined. In the cross-sections which are placed at a larger distance down the flow, the heat-transfer rate is higher. Growth of $\alpha$ in these experiments is caused by the effect of increasing vapour content. In the same figure the experimental data of work [10] are shown which are obtained in the case of boiling water in tubes with a very small ratio $L/d$. The heat-transfer process of boiling in short tubes cannot involve the effect of vapour content and therefore the treatment of the data of [10] on the basis of the liquid velocity at the tube inlet shows good agreement with the predicted relation of [3].

Thus when inequality (1) is satisfied and the flow temperature becomes higher than $t_{bo}$, the heat-transfer coefficient may be determined by Sterman's formula [3], the true liquid–phase velocity being known. If inequality (1) is satisfied, but the flow temperature is less than $t_{bo}$, the heat-transfer rate may change within a wide range depending on the subcooling of the

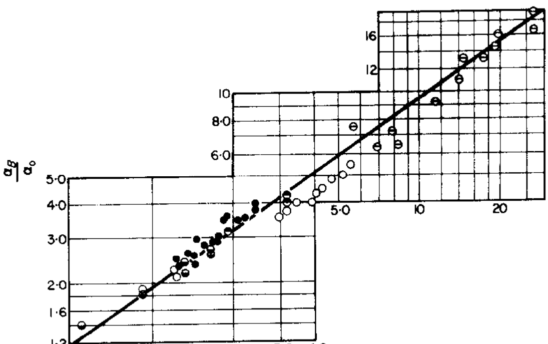


FIG. 6. Comparison of the present authors' data with those of [10] and the predicted relation of [3] with high-rate boiling of subcooled liquid.

The authors' data: ◒ = water. Runs No. 24 (thermocouple 2), No. 28 and 29 (thermocouples 4 and 6), Nos. 54–57 (thermocouple 2), Nos. 8 and 9 in Fig. 4 (thermocouple 2). ◓ = normal propyl alcohol; runs Nos. 75 and 76 (thermocouples 1, 2, 3) No. 82 (thermocouple 2), Nos. 84, 85, 86 (thermocouple 6 and 8).

Data of [3]: ● = $P_{in} = 11{\cdot}77 \times 10^4$ N/m²; ○ = $P_{in}$ = $58{\cdot}86 \times 10^4$ N/m²; ⊖ = $P_{in} = 206{\cdot}0 \times 10^4$ N/m² (water); — = predicted curve [3].

liquid. It would be too soon to propose any formulae for the determination of the heat-transfer coefficient within this temperature range.

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**Abstract**—The paper discusses the results of an experimental study of heat-transfer rate of a subcooled boiling liquid in tubes with a large heated length-to-diameter ratio under low pressure. In the experiments water and normal propyl alcohol were used.

**Résumé**—On discute les résultats d'une étude expérimentale du transport de chaleur par ébullition d'un liquide sous-refroidi dans des tubes avec un grand rapport de la longueur chauffée au diamètre et une faible pression. On a employé dans les expériences de l'eau et de l'alcool propylique normal.

**Zusammenfassung**—Die Arbeit behandelt die Ergebnisse einer experimentellen Untersuchung des Wärmeübergangs beim Sieden in unterkühlter Flüssigkeit in einem Rohr mit grossem Verhältnis von beheizter Länge zu Durchmesser und bei geringem Druck. Es wurde Wasser und Normalpropylalkohol verwendet.