HEAT TRANSFER AT BOILING OF LIQUID FILM IRRIGATING A HORIZONTAL BUNDLE OF ROUGH TUBES

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Abstract. According to experiments with film irrigation of packing of tubes with artificial roughness, boiling on them begins at very low-temperature heads. Heat transfer at boiling on these tubes is noticeably more intense than that on a smooth tube.

Introduction

Effect of wall roughness on boiling heat transfer under free convection is known long ago [1 -3]. In [4] a criterial dependence describing satisfactorily experimental results at boiling on walls with various roughness, is presented.

There are very few experimental data on heat transfer at boiling of liquid film irrigating walls with different roughness. It is known that boiling heat transfer is always accompanied by evaporation heat transfer. In the transition area between evaporation and boiling their contribution to the total heat removal is commensurate.

The below-described experiments were carried out in a closed circulation experimental test-bench. Copper and duralumin tubes both plain and rough in the bottom part of a horizontal tube bundle were used as the experimental tubes. The one-row tube bundle consisted of 20 tubes. The 2nd from the bottom copper tube with the plain surface has the surface finish $R_z \approx 2.5$ microns, the wall thickness $\delta_w = 2$ mm, and $D = 10$ mm. The 3rd from the bottom copper tubes has roughness $R_z \approx 20 - 30$ microns because of special treatment with corundum particles. The 4th from the bottom duralumin tube has roughness $R_z \approx 20 - 30$ microns because of special treatment with corundum particles in the air stream.

The bank was irrigated with Freon R21 at $T \approx 40^\circ$C. The tubes were heated by water flowing from a constant header tank inside the tubes. Measurements were carried out simultaneously for all experimental tubes at the same heating water flow rate.

Measurement technique

The temperature of the water heating experimental section is measured by semiconductor temperature sensors. The sensors are calibrated with the accuracy of less than 0.02°C. The sensors calibration was carried out after their placement into the test volume by means of thermostats. For the plain tubes, the wall temperature was determined as an arithmetic mean value of measurement of 5 thermocouples sealed around perimeters of tubes from 0 to 180°. For the rough tubes, the wall temperature was calculated using the heat transfer coefficient.

A Freon flow rate irrigating the tube bundle was measured with a turbine flow meter.

The heat flux density on the experimental sections is calculated according to the dependence:

$$q = (m \cdot C_p \cdot \Delta T)/F$$  \hspace{0.5cm} (1)

The heating water flow rate is determined on the readings of rotameters as well as flowmeters with the accuracy rating 0.5. The rotameters installed in every section are calibrated individually.

The measured parameters were the following: temperature of heating water at the inlet and outlet of the experimental section, temperature drop between the inlet and outlet from the section, temperature of
Freon liquid and vapor in the experimental volume, pressure in the evaporator and a condenser, flow rate of liquid Freon irrigating the tube bundle, the power of electric heaters of water and Freon, geometry of the tube bundle.

Experimental results on heat transfer at boiling on the plain and rough tubes are shown in Fig. 1. It is seen that boiling began on the copper plain tube at $\Delta T = 4.5 - 5^\circ$C ($q = 10^4$ W/m$^2$), on the copper rough tube at $\Delta T = 2.5 - 2.8^\circ$C ($q = 6 \times 10^3$ W/m$^2$), and on the duralumin rough tube at $\Delta T = 3.5^\circ$C ($q = 7 \times 10^3$ W/m$^2$). The effect of heat release of wall material is quite clear as well as at pool boiling [8, 9]. Heat transfer on the copper tube is considerably higher than on the duralumin tube with the same geometry characteristics. Reduction of the temperature drop at which the boiling begins because of roughness increase of a heat release wall is extremely important for heat exchangers with the limited value of available temperature drop. Boiling heat transfer on a copper rough tube at $q = \text{idem}$ increases one and a half times in comparison with a smooth wall. The use of rough tubes is a simple and reliable way to enhance heat transfer of the film irrigating a tube bundle.

![Figure 1. Heat transfer at film evaporation and boiling in the tube bundle.](image)

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**Figure 1.** Heat transfer at film evaporation and boiling in the tube bundle.

R21, $T_s = 40^\circ$C, $D = 10$ mm, $S/D = 1.2$, $\delta_w = 2$ mm;  
1 – copper, $R_z=2.5$ microns, 2 – copper, $R_z= 20 - 30$ microns, 3 – duralumin D16T, $R_z= 20 - 30$ microns; $Re = 300 - 600$.

**Data representation in dimensionless coordinates**

As shown in [15], the total heat transfer at film evaporation and boiling can be calculated according to the dependence:

$$\bar{\alpha} = \alpha_0 \left(1 - \frac{L_d}{L}\right) + \alpha_d \frac{L_d}{L} - \alpha_e \left(1 - \frac{L_d}{L}\right)$$  \hspace{1cm} (2)

Here $L_d$ is a part of tube perimeter where convective heat transfer on the initial part of the thermal boundary layer, $L = \pi \cdot R$ is a tube semi-perimeter, $\alpha_d$ is convective heat transfer on the initial part of the thermal boundary layer, $\alpha_0$ – boiling heat transfer, $\alpha_e$ – evaporation heat transfer.

Temperature measurements along the perimeter of the plain tube wall showed that local $\Delta T = T_w - T_s$ is independent of the thermocouple location on the tube perimeter. This dense tube bank should be considered as a vertical wall because there is no initial part of the thermal boundary layer on the bank.
tubes. Here a drop dimension exceeds the distance between bank tubes, so the drop flows smoothly from an upper to the lower tube. The drop temperature is practically constant.

Without the initial part of the thermal boundary layer the dependence (2) is the following:

\[ \alpha = \alpha_0 + \alpha_e \]  

(3)

A calculation algorithm for heat transfer at film evaporation and boiling at the wave and turbulent film flow is stated in [15]. The presented comparison between experimental data for a smooth tube bundle and calculated values is quite satisfactory.

As follows from Fig. 2 for rough tubes, heat transfer is changed only at film boiling. The boiling begins at considerably lower temperature difference between wall and liquid in comparison with smooth tubes.

The presented comparison between experimental data for a smooth tube bank and calculated values is quite satisfactory.

Heat transfer calculation at boiling must be performed according to the dependence:

\[ Nu^* = 0.01Re^{0.8}Pr^{1/3}bKt^{0.4}\left(\frac{\rho_C}{\rho_{wall}}\right)^{-0.2}\frac{\bar{R}_e^{0.2}}{Re_c} \]  

(4)

Here \( \left(\frac{\rho_C}{\rho_{wall}}\right)^{-0.2} \) is the ratio of physical properties of heat carrier to physical properties of the cooled wall. This parameter takes into account the effect of the wall properties on boiling heat transfer rate. \( \bar{R}_e = \frac{R_e}{(\nu^2/\theta)^{1/3}} \) is a parameter taking into account the effect of wall roughness on boiling heat transfer.

As shown in [4], at \( \bar{R}_e < 0.9 \) heat transfer is enhanced as wall roughness grows. At \( \bar{R}_e > 0.9 \) change of wall roughness has no effect on heat transfer.

It should be noted that film evaporation heat transfer is independent of tube roughness and agree satisfactorily with a calculation based on the three-layer model of turbulent flow stated in [16] for \( Re > Re_{cr} \). \( Re_{cr} \) is determined by [9].

A heat transfer calculation algorithm at film evaporation and boiling for wave and turbulent flow is stated in [15]. The presented comparison of experimental data with calculated dependencies is quite satisfactory [9 – 14].

At wave film, flow heat transfer is practically independent of irrigation density and determined by the thermal resistance of a “residual” film thickness [15].

At the beginning of the film boiling, it is turbulized by growing vapor bubbles, and it’s surface area increases considerably. In this case, heat transfer determination for a single-phase flow should be considered as estimating and understated. As follows from Fig. 2, at maximum (in our experiments) heat fluxes the deviation of the experimental data from the calculated achieves 10 – 20%.
Currently, it is not possible to determine heat transfer during evaporation of the boiling film more accurately.

At $T_s = \text{const}$, the dependence (4) takes the form of

$$Nu^* = C Re_*^{0.8}$$

(5)

Here $C = 0.01Pr^{1/3}bK_t^{0.4} \left( \frac{i_{wall}c_p}{i_{wall}c_p} \right)^{-0.2} \bar{R}_x^{0.2}$ at $\bar{R}_x \leq 0.9$

or $C = 0.01Pr^{1/3}bK_t^{0.4} \left( \frac{i_{wall}c_p}{i_{wall}c_p} \right)^{-0.2} \bar{R}_x > 0.9$

In Fig. 2 the comparison of experimental data at boiling on rough tubes is presented in the form of $\frac{\alpha}{\bar{g}} = f(Re_*)$ according to the calculation on the dependence (3). Experimental data on tubes with increased roughness agree with the calculated data satisfactorily.

The maximum deviation of the experimental data from the calculated values of heat transfer coefficient is observed during comparison with experiments on the plain tube. In our opinion, it is due to the fact that at lower heat fluxes boiling did not take place around the whole tube perimeter. The similar phenomenon was observed in the experimental works [17, 18].
Conclusions

1. The use of tubes with increased roughness is a simple and reliable way to enhance boiling heat transfer. Decrease of $\Delta T$, which corresponds boiling origin, is especially actual for devices with the limited temperature drops.

2. The proposed calculation of heat transfer at film evaporation and boiling for rough tubes agrees satisfactorily with measurement results.

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