Experimental studies of the influence of hydrodynamics on heat transfer at evaporation and boiling of a film irrigating a bundle of horizontal finned tubes

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Abstract. A feature of operation of some devices, such as heat pump steam generators, evaporators of refrigerating machine, steam generators of binary freon station, etc., is a low value of specific heat fluxes. Heat transfer regimes occur, when boiling in the film is just beginning and the heat transfer rate during evaporation is comparable to that during boiling. Tube finning is a well-proven way to intensify heat transfer, both during evaporation and boiling. However, there are insufficient data on the choice of finning parameters and methods for evaluating heat transfer during boiling of a film that irrigates a bundle of finned tubes. This paper presents an algorithm for calculating heat transfer during evaporation and boiling of a film that irrigates a bundle of finned tubes, and compares the calculated and experimental data. The calculated parameters turn out to coincide satisfactorily with the experimental data.

1. Introduction
The film heat exchangers are successfully used in many industries. They become especially attractive if taking into account that finning leads to essential stabilization of the film flow and multiple intensification of heat transfer. However, the number of publications in journals that study heat transfer at evaporation and boiling of a film that irrigates a bundle of horizontal finned tubes is extremely limited. The paper attempts to describe heat transfer at evaporation and boiling by equations obtained on the basis of physical studies of hydrodynamics of the film flows on the finned tubes. Heat transfer at film boiling is always accompanied by evaporation heat transfer, and wall cooling depends on both processes.

2. Features of hydrodynamics of the film flow irrigating a bundle of finned tubes
Figure 1 shows a physical model of liquid flow through a finned tube. When the finned tubes are irrigated by ethyl alcohol, the microbubbles make the trajectory of liquid flow along the lateral surface of the fin clearly visible. The diagram of this motion is shown in figure 1.

Our visual observations and videos show that liquid flows from the top of the fin down the valley under the influence of surface tension forces along the lateral surface of the fin in a path indicated in figure 1. A thin film on the lateral surface of the fin wets its surface and evaporates intensively. This means that when the finned tube is irrigated by a liquid film, the height of the fin becomes a characteristic linear dimension.
Moreover, a finned tube can be considered an isothermal wall, since the fin efficiency for a given tube, determined by dependence given in [2], is close to unity.

When $\alpha \leq 1.0$, the height of a liquid layer held in a valley coincides well with the height of the capillary rise of liquid in an infinite vertical gap with a width equal to the distance between the fins. For trapezoid ribs there is a more general relation where we use the empirical distance proposed in [3] as the linear size of the depression.

The experimental results on capillary retention on the finned cylinders with rectangular and trapezoidal fin profiles are satisfactorily described by a single relationship.

In general, the proportion of a flooded tube surface is a function of dimensionless complex $\alpha D$.

3. Features of heat transfer at evaporation of a film irrigating a bundle of horizontal finned tubes

Tube finning is a well-proven method of intensifying heat transfer at evaporation and boiling. However, there are very few experimental data on evaporation of a film irrigating a bundle of finned tubes [4;5].

Below there are the results of an experimental study of heat transfer at evaporation of R-21 Freon, irrigating a bundle of horizontal finned tubes. The experiments were performed on a setup with forced circulation of a substance. Schematic diagram of setup is given in [14]. Liquid freon is fed to the experimental volume using a glandless pump through the heater at $T_s = T_f$.

Freon irrigates a bundle of experimental tubes through the distribution gap. The experimental tubes are heated by water flowing inside the tubes from a constant header tank. Water is supplied through the collector to the experimental sections, equipped with rotameters.

The experimental evaporator is made of a stainless steel cylindrical shell. The cross-section of the evaporator is shown schematically in figure 2.

4. Measurement method

The temperature of water, heating the experimental sections, was measured by semiconductor temperature sensors. The sensors were calibrated with an error of not more than 0.02°C. The wall
temperature was calculated by the heat transfer coefficient of water inside the tube, which was calculated by the Petukhov formula cited in [15].

The heat flux density in the experimental sections was calculated by dependence

\[ q = \left( m \cdot C_p \cdot \Delta T \right) / F. \]  

Equation (1)

The flow rate of heating water was determined by the readings of rotameters or flow sensors of 0.5 class. The experiments were carried out at tube bundle irrigation with R-21 freon at \( T_s = 40^\circ C \). The Re number (\( Re = G/2 \mu \)) of the film irrigating the tube bundle was varied in the range of \( 380 \leq Re \leq 1500 \). The bundle consisted of 12 tubes arranged in a vertical row. The tubes located at the bottom of the bundle were the experimental ones. The smooth penultimate tube of the bundle (the second from the bottom) made of M-1 copper had technical frequency \( R_c = 2.5 \mu m \), wall thickness \( \delta_w = 2 \) mm, and \( D = 10 \) mm. The third tube from the bottom made of M-1 copper was finned. Fin thread had a pitch of 1.07 mm and a finning ratio of 1.46.

**Figure 3.** Scheme of test section.

Figure 3 shows the dimensions of the finned tube.

5. Measurement results

On a technically smooth copper tube, boiling began at \( \Delta T = 4.5 - 5^\circ C \) (\( q = 10^4 \) W/m\(^2\)).

In these experiments, boiling on a finned tube was not within the entire studied range of heat fluxes. This is especially evident in figure 4, showing that for a finned tube \( \alpha \approx \text{Const} \) in the entire studied range of parameters. In figure 4, specific heat flux is calculated for the full surface of the finned tube. It follows from this figure that evaporation heat transfer on a finned tube is noticeably more intense than on a smooth one, even if calculation of specific heat flux is related to the full surface of the finned tube.

**Figure 4.** Heat transfer coefficient vs. heat flux at evaporation of freon R-21 on the finned (2) and smooth tubes (1).

The calculation of heat transfer at evaporation of a film irrigating a bundle of finned tubes is complicated, first of all, by very complex hydrodynamics of irrigation of each fin. The lateral surface of the fin in its upper part is irrigated by a thin laminar film of liquid flowing down from the fin end. The lower part of the fin, located in the depth of the valley, is irrigated by the main liquid flow, which is transported along the valley between the fins, and it is turbulent.

It is practically impossible to determine correctly the interface between the laminar and turbulent flows. It can only be estimated in the first approximation. Heat transfer at film evaporation from different surfaces of the fin can be calculated only approximately under the following assumptions:
a) surface tension forces are significantly higher than gravity. The upper part of the lateral fin surface is irrigated with liquid that flows from the end surface of the fin into a valley along the trajectory indicated in figure 1.

b) regularities of heat transfer on the end part of the fin and in the valley between the fins similar to a smooth tube were determined using a three-layer turbulence model, as it was done in [6].

c) the mass flow of liquid irrigating the end face of the fin and its lateral surface is determined from the balance relationships.

d) a fin is in the initial region of the thermal boundary layer with a characteristic linear size (the height of the fin).

e) the thickness of the turbulent film in-between the fins is calculated according to the Duckler-Bergelin dependence [7] taking into account the geometry of finning.

\[ \bar{\delta} = \text{Re}^{0.2} \left( \frac{3\nu^2}{g} \right)^{\frac{1}{13}} \text{Re}^{8/15} \]  

(2)

The critical Reynolds number of the film here is determined by Brauer [8].

\[ \text{Re}_{cr} = 35K a^{1/10} \]  

(3)

The film thickness calculated by (2) almost coincides with the film thickness determined by the dependence given in [9] and results of experimental studies given in [10]. Here

\[ K_a = \left( \frac{l_w}{l_v} \right)^6 = \frac{\sigma}{\nu^3 \rho g} \]  

Kapitza number;

Heat transfer in the initial section of the thermal boundary layer (the upper part of the lateral surface of the fin) is determined by dependence:

\[ Nu_{+2} = 0.6 \text{Pr}^{1/3} \left( \frac{\text{Re}_x}{\bar{h}} \right)^{1/9} \]  

\[ \bar{h} = h/l_w \]  

(4)

Dependence (4) is empirical. It is written by analogy with the analytical formula given in [11]. To determine the heat transfer coefficient on the end part of the fin and in the valley under the turbulent regime of the film flow, we can use the three-layer model described in [12], where the turbulent viscosity is piecewise approximated by the corresponding dependences. A table of results of numerical calculation of the Nusselt number vs. the film Reynolds number for some integer values of the Prandtl number is presented in [12]. Calculating the film thickness according to (2) and taking into account that the valley has the shape of a trapezoid, we can approximately determine the areas of the fin sides, washed by the turbulent or laminar liquid flows. Using the experimental data on \( \Delta T = T_w - T_s \), then the specific heat fluxes on each part of the finned tube may be calculated. The power removed from the tube will consist of the following components:

\[ \Sigma Q = (Q_l + Q_w + 2Q_t + 2Q_v)k \]  

(5)

Here \( Q_l \) [W] is the power removed by the fin end; \( Q_w \) [W] is the power removed by the horizontal part of the valley between the fins; \( Q_t \) [W] is the power removed by the lateral side of the fin, washed by the laminar liquid flow; \( Q_v \) [W] is the power removed by the lateral side of the fin, washed by the turbulent liquid flow; and \( k \) is the number of fins on the test tube.

The values of power calculated by (5) should be compared with the experimental value. The calculation data with a spread of ± 15% coincided with the experimental power value in the entire range of parameters of experimental research.

It should be noted that for the film flow parameters studied in this work, evaporation heat transfer occurred intensively and film boiling was not observed. The wall-liquid temperature head was always \( \Delta T < 5^\circ C \), that is, it met the condition when boiling started on a smooth tube.

6. Heat transfer at evaporation and boiling of a liquid film on a bundle of finned tubes
In a review of Fujita Y. and Tsatsui M. [4], the results of an experimental study of heat transfer at evaporation and boiling of a R11 freon film irrigating a bundle of finned and smooth tubes were presented. The bundle consisted of 5 tubes with only two tubes heated.

The paper [4] presented the geometrical parameters of the fins on a copper tube with an outer diameter $D = 25$ mm, $\delta_w = 7.5$ mm.

It follows from figure 5 that heat transfer at evaporation and boiling of a film irrigating a bundle of finned tubes is many times more intense than that at evaporation and boiling on a bundle of smooth tubes at $G=\text{idem}$. As it follows from the experiment, the greatest heat transfer intensification was achieved on the tube with the fins in the form of an acute triangle.

The authors of [4] do not indicate the reasons for such a significant intensification of heat transfer at boiling, shown in figure 5, although heat transfer at evaporation on a smooth tube is analyzed in detail.

The features of heat transfer on a finned tube are related to the fact that hydrodynamics of the film flow substantially depends on the surface tension, in addition to gravity and viscosity.

It has been already noted that calculation of heat transfer at evaporation of the film irrigating a bundle of finned tubes is complicated, first of all, by the complex hydrodynamics of the film irrigating each fin. The lateral surface of the fin in the upper part is irrigated by a thin laminar film flowing down from the end of the fin. The lower part of the fin deep in the valley is irrigated by the main liquid flow and it is turbulent. The interface between the turbulent and laminar flows can only be estimated using the assumptions listed above. Heat transfer in the initial section of the thermal boundary layer is calculated using dependence (4). Heat transfer at evaporation in a turbulent film flow is calculated with the three-layer model described in [12]. Boiling heat transfer can be calculated according to the dependence given in [13, 14].

The determination of heat transfer according to (6) suggests that the mechanism of heat transfer at film boiling does not differ from the mechanism of heat transfer at pool boiling.

Figure 5 shows the conditional value of specific heat flux calculated by the outer diameter of a smooth tube $D = 25$ mm. The true value of the heat flux is: $q_0 = q/n$, where $n$ is the finning coefficient.

When calculating boiling heat transfer by (6), the Re$_v$ number should be determined by the true value of the heat flux $Re_v = \frac{q \cdot l_c}{\rho \cdot \nu}$. Comparison of experimental data shown in figure 5 with the calculated results should be calculated using dependence:
Here $\alpha_d$ is the coefficient of evaporation heat transfer calculated by dependence (4) and $\alpha_b$ is the coefficient of boiling heat transfer calculated by dependence (6).

The lines in figure 5 show the value of heat transfer coefficient determined by dependence (7), and experimental data are indicated by the icons. It is seen that the calculation results are in satisfactory agreement with the experimental data.

The disadvantage of [4] is a lack of information about the roughness of the smooth and finned tubes. Comparison of experimental results with calculation can be performed only under the following assumptions. The surface of a smooth tube is close to the polished one, and its roughness $R_z \sim 0.5 \mu m$ or less. This assumption is confirmed by the very high temperature head, when boiling starts, $\Delta T_{bs} \sim 13^\circ C$. Most likely, the finned tubes have an increased roughness, since boiling on them starts at $\Delta T_{bs} \sim 3^\circ C$, which suggests that $R_z \geq 0.9$.

Under the accepted assumptions about tube roughness, the calculation data coincide satisfactorily with the experimental results.

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7. Conclusion

1. When irrigating a bundle of finned tubes, surface tension forces exceed gravity significantly and determine hydrodynamics of irrigating liquid. The characteristic linear dimension is the fin height, and the fin itself is in the initial region of the thermal boundary layer.

2. To date, it has been proven that boiling heat transfer depends on the physical properties of the cooled wall and its roughness.

3. The work presents an algorithm for calculating heat transfer at evaporation and boiling of a film that irrigates a bundle of finned tubes.

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