Critical heat flux and heat transfer in a thermosiphon with enhanced surfaces for boiling and condensation

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Abstract. The technology of solid-state light source is associated with the future of a number of sectors of economy. Duration of failure-free operation, optical radiation power and other output characteristics of the LEDs are closely linked with p-n junction temperature, which makes the development of the cooling systems an important step in creating LED systems. In this work, we have created a new type of thermosiphon for studying heat transfer from a local heat source. Boiling heat transfer on the local heaters with the diameter of 1 and 5 mm has been investigated. It is shown that on the finned surfaces overheating relative to the saturation temperature in comparison with a smooth surface decreases up to three times for the heater with the diameter of 5 mm. For finned surfaces on the heater with a diameter of 1 mm, surface overheating relative to the saturation temperature decreases four times. More than three times increase is observed for the heat transfer coefficient on finned surfaces as compared to the smooth ones. It is shown that the value of the critical heat flux (CHF) under the conditions of a large volume of liquid for the heaters with $D = 1.6$ and 5 mm is in good agreement with the known dependences. Radial finning of the heater has no effect on CHF. The value of CHF for $D = 1$ mm turns out to be higher than the calculated one. The greatest increase is achieved by using a head with finning.

1. Introduction

The future of a number of sectors of the economy is linked with the technology of solid-state light sources and one of the main characteristics of uptime is the removal of heat. Study of the evaporation mechanism on the surfaces of small size was started in [1]. The main objective was to study the temperature fields of the heating surface and liquid as well as the conditions for the formation and existence of a thin liquid film at the base of the vapour bubble. In that work, as in the follow-up, a single center of vaporization was studied [2]. The heat flux was limited by the value at which there is a steady generation of a single center of evaporation. A review of studies on heat transfer during vaporization at the surface of small size is contained in [3]. In our days the problem of creating an effective compact cooling system for powerful LEDs whose efficiency and durability are highly dependent on the efficiency of the power removal remains unsolved.

Heat transfer at boiling of dielectric liquids on the heaters of small size surface (SSS) in submerged vertical channels was studied in [4]. A review of works on heat transfer at evaporation on SSS may be found in [3]. In the literature, there are dependences for the critical heat flux density for each type of SSS, having the general form:
\[ \frac{q_{cr}}{q_{cr,\infty}} = k \left(\frac{d_s}{l_s}\right)^n, \]

but differing in correlation multipliers; and \(d_s\) is the diameter of the heating surface. Here

\[ q_{cr,\infty} = 0.13r_l\rho_f \left(\frac{\sigma (\rho - \rho_v) g}{\rho_v^2}\right)^{0.25}, \]

\(r_l\) is the latent heat of vaporization, \(\rho_v\) is the vapor density, \(\rho\) is the fluid density, \(\sigma\) is the surface tension, and \(g\) is the gravity acceleration.

For locally heated horizontal plates \(k = 3.4, n = -0.5\) [3], for horizontal cylinders \(k = 1.4\) [5], and for spheres \(k = 2.5\). The reason for such significant effect of the size of SSS on \(q_{cr}\) is the feature of hydrodynamic situation that occurs around the surface of small size (SSS).

One of the most effective ways of LEDs cooling is using thermosiphons. The expediency of using the contoured thermosiphons for LED lamp cooling to enhance their reliability and efficiency is shown in [6]. Disadvantage of such systems is a required careful evacuation of heat tubes before filling them with liquid. Nevertheless, during operation, the condensable impurities dissolved in the working fluid and located on internal surfaces accumulate in the upper part of the condenser, preventing efficient condensation of vapor. A new thermosiphon design was proposed and investigated in [7, 8]. In it, noncondensable gas was pushed into a special chamber under the action of a jet of newly formed vapor though failing to prevent vapor condensation. Many different types of heat pipes exist, so there is a wide range of possible solutions.

This paper proposes a new type of thermosiphon with enhanced surfaces for boiling and condensation. Vapor condensation in the finned tubes was studied theoretically in [9], taking into account the influence of capillary forces. The effectiveness of a new type of surface for the condensation process is grounded in [10]. This paper presents investigation results on heat transfer and heat transfer crisis at boiling on intensified surfaces with radial fins in a new thermosiphon.

2. Design of thermosiphon

To study the heat transfer processes that occur at boiling on the longitudinal finned surfaces, a new type of thermosiphon has been created. Views and scheme of the thermosiphon are shown in figure 1. Thermosiphon consists of heater 1, vapor channel 2 and radiator-condenser 3 with an intensified surface for condensation. Vapor channel 2 is made in the form of a cylinder, one end of which adjoins the bottom of the thermosiphon. Radiator 3 is made of a closed thin-walled corrugated sheet profile with oval corrugations in the form of a multipetal drum; it is installed coaxially with vapor channel 2.

![Figure 1](image-url)

**Figure 1.** Design of thermosiphon. a – side view, b – top view, c – scheme. 1 – heater, 2 – vapor channel condenser, 3 – radiator-condenser, 4 – socket for liquid input.
Figure 2. Design of heaters. a – scheme of heater $D = 5$ mm, b – photo of heater $D = 5$ mm with the longitudinal finned surface. c – pictures of head on heater $D = 1$ mm. 1 – location of thermocouple.

This shape of radiator allows maximizing the heat exchange surface, which in turn provides for condensation of vapor with natural air circulation. The radiator petals have the rounded peaks, channels between them have the rounded bases, and the radii of petals are 2-3 times larger than the radii of the channel bases. All this ensures intensive condensation of vapor on the inner surface of petals near vertices.

The horizontal layer of liquid is formed in lower part of thermosiphon. The thickness varies depending on the operating parameters of the experiment. In the center of the layer from the side of the substrate there is local heating performed by a copper heater, figure 2. The power of the heating element is controlled by the power source. The temperatures of the heater, the liquid and the vapor are measured by a number of thermocouples. The data from thermocouples is collected using a control and measuring system, and the temperature measurement precision is 0.1°C.

3. Results and discussion

In this study, we carried out experiments for the conditions of a large volume of fluid ($H > 10$ mm) and maintained the fluid at saturation temperature. For these purposes, the work area was covered with additional heater and a protective insulation. Curves of boiling heat transfer enhancement for heaters with $D = 5$ and 1 mm on smooth and finned surfaces are presented in figure 3. The boiling curves for the finned surfaces are higher than for smooth ones. Heat transfer enhancement ratio $\alpha/\alpha_0$ is more than 1 ($\alpha_0$ – heat transfer coefficient for smooth surfaces, $\alpha$ – heat transfer coefficient for finned surfaces). Experiments on the heaters with $D = 5$ mm with smooth and finned surfaces show that overheating on the finned surfaces is three times less. And the heat transfer coefficient increases up to two times on the finned surface compared with the smooth one.

Figure 3. Dependence of heat transfer enhancement ratio $\alpha/\alpha_0$ on heat flux for the heater with a) $D = 5$ mm. 1 - large fins, 2 - small fins; b) $D = 1$ mm. 1 - head with large fins, 2 - head with small fins.
The crisis of heat transfer was registered on the heater with large fins and amounted to 256 W/cm². The studies of heat transfer during boiling on a heater with a diameter of 1 mm are shown in figure 3b. On the smooth surface overheating temperature reached about 200°C. The heat flux density before the onset of critical phenomena reached 900 W/cm². On the ribbed surfaces overheating relative to saturation temperature decreased up to four times. More than three times increase was specific for the heat transfer coefficient on the finned surface compared with the smooth one. It should be noted that the type of finning has no significant effect on the boiling curve. As a result of the experimental work the density of heat flux reached 1400 W/cm² for heater of D = 1 mm with finned head (figure 2c).

The obtained experimental data on CHF are compared in figure 4 with calculated dependence (1) from [3] and results of [4]. It can be seen from figure 4 that the value of the critical heat flux density (CHF) in a large volume of liquid for the heaters with D = 1.6 and 5 mm is in good agreement with dependence (1) and experimental data for the vertical heaters. The size of the heater has a significant effect on CHF. Radial finning of the heater does not affect CHF, and a decrease in the height of the layer and flow rate of supplied liquid leads to a significant decrease in CHF. The value of CHF for D=1 mm is higher than the calculated one. Moreover, the greatest increase is achieved by using a finned head, which is shown in figure 2c.

Experiments on the heaters of 5-mm diameter with the smooth and finned surfaces show that on the finned surfaces superheating relative to the saturation temperature decreases three times. When using the finned heads on the 1-mm diameter heater, superheating relative to the saturation temperature decreases more than four times (°C). Data on the influence of liquid temperature around the heat-exchange surface on removed heat flux density for the saturation conditions (points 2-4) and at the initial temperature of 25-30°C (points 1) are presented in figure 5. Such regimes are possible in thermosiphons and heat pipes, if the condensed liquid is supercooled. It is seen that in this case the temperature of the LED crystal model decreases substantially. It is shown that for the heat flux densities of up to 1000 W/cm², the temperature of the heat model of LED semiconductor crystal does not exceed 120°C.

**Figure 4.** Dependence of CHFD on the heater size. 1 – calculation by formula (1) from [3], 2 – data of [4], 3 – D = 1.6 and 5 mm, 4 – D = 1 mm, 5 – D = 1 mm with head.
Figure 5. Dependence of removed heat flux density on heater temperature, $D = 1$ mm. 1 – large finning under the conditions of saturation $T_0 = 25 – 30^\circ$C, 2 – large finning, 3 – small finning, 4 – smooth surface.

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