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DETERMINING THE INFLUENCE OF GEOMETRIC FACTORS AND THE TYPE OF HEAT CARRIER ON THE THERMAL RESISTANCE OF MINIATURE TWO-PHASE THERMOSYPHONS

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This paper reports experimental data on the total thermal resistance of copper two-phase thermosiphons with internal diameters of 3 mm, 5 mm, and 9 mm, 700 mm long. Water, ethanol, methanol, and freon-113 were used as heat carriers. During the study, thermosiphons were located vertically. The length of the heating zone varied from 45 mm to 200 mm while the length of the condensation zone was constant and equaled 200 mm. The filling coefficient of thermosiphons varied from 0.3 to 2.0. Two series of experiments were conducted. The first series was distinguished by the fact that the filling coefficient of three thermosiphons with an internal diameter of 9 mm varied from 0.3 to 0.8 with the same length of the heating zone of 200 mm. The second series of experiments was carried out on thermosiphons with internal diameters of 3 mm and 5 mm. With the same amount of heat carrier, the length of the heating zone changed from 45 mm to 200 mm. As a result of research, it was determined that the total thermal resistance of thermosiphons is influenced by both their geometric factors (internal diameter and filling coefficient) and the type of heat carrier. The main factor that influenced the value of thermal resistance was also the transmitted heat flux. An increase in heat flow led to a significant decrease in thermal resistance. The maximum heat flux was determined with minimal thermal resistance. To calculate the value of the thermal resistance of thermosiphons, two dimensionless dependences were derived, which hold for two ranges of Reynolds numbers. For small Reynolds numbers (until 2000), which characterize the beginning of the action of vaporization centers and their gradual increase, the degree indicator was –0.8, and for larger Reynolds numbers, up to critical phenomena, the degree indicator was at the level of –0.3

Keywords: miniature thermosiphon, heat transfer intensity, thermal resistance, heat flux, heat carrier, filling coefficient

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1. Introduction

The current trend of reducing the mass and size characteristics of electronic equipment while increasing its functionality predetermines an urgent task of designing effective thermal stabilization systems for such miniature devices. The use of devices for heat dissipation, which employ the evaporation-condensation cycle for this purpose, makes it possible to some extent to solve this problem. A thermosiphon can be used as such a device. It is a two-phase closed heat transfer device containing a certain amount of liquid that uses the latent heat of evaporation and condensation to transfer heat between the heat source and the drain without any external devices. The equivalent thermal conductivity of such devices is several orders of magnitude higher than the thermal conductivity of naturally known metals (copper, silver, and others) [1, 2]. Thermosiphon requires only a temperature difference to transfer a large amount of heat, so that such reliable heat transfer devices can be used in cooling and thermal stabilization systems of various objects, such as small-sized electronic devices, and large power equipment.

The phrase «miniature thermosiphons» refers to thermosiphons, the geometric characteristics of which (primarily the inner diameter) are commensurate with the scale of the physical processes occurring inside them. Since the vaporization process takes place to transfer thermal energy in such devices, the dimensions of the internal space affect their heat transfer characteristics. First of all, it affects the process of occurrence, growth, and separation of steam bubbles. In [1], it is noted that the complex that determines the ratio of gravity forces and surface tension forces is the Bond *Bo* number:

$$Bo=\frac{d_{in}}{\sqrt{\frac{\sigma}{g(\rho'-\rho'')}}}, \quad (1)$$

where d_{in} is the inner diameter of the thermosiphon, m; σ – surface tension coefficient, N/m; g – acceleration of gravity, m/s^2; ρ', ρ'' – densities of liquid and vapor, kg/m^3.

The influence of such forces begins to manifest itself at the numbers $Bo<4$ [1]. It should also be noted that, depending on the selected heat carrier, the limiting diameters of thermosiphons, at which changes in heat transfer characteristics are already beginning to appear, may differ from each other. Thus, for water at atmospheric pressure, thermosyphons with an internal diameter of less than $10\cdot10^{-3}$ m can be classified as miniature. For methanol, thermosyphons with an internal diameter of less than $7\cdot10^{-3}$ m can already be considered miniature. Since limited conditions begin to affect the processes of vaporization and heat transfer, the study of such an effect can be useful in the design of miniature cooling systems based on two-phase thermosiphons for electronic and computer equipment.

For the successful and effective implementation of such elements, it is necessary to thoroughly investigate the processes occurring in them, and on this basis to derive dependences that will make it possible to calculate the optimal operating modes and parameters of cooling systems. Therefore, scientific research in this area is important and necessary for practical application.

2. Literature review and problem statement

The main characteristics of thermosiphons are the minimum thermal resistance $R_{\min}$ at the maximum power transmitted, $Q_{\max}$. The value of the total thermal resistance depends on the intensity of heat transfer in the zones of evaporation (heating) and condensation. In works [3, 4], a study of the thermal resistance of a copper two-phase thermosiphon with a length of 0.2 m with an internal diameter $d_{in}=6\cdot10^{-3}$ was carried out. The tests were performed when the filling coefficient F_r changed from 30 to 100 %, and distilled water and organo-fluorine dielectric liquids (FC-84, FC-77, and FC-3283) were used as a heat carrier. The cited paper notes that dielectric liquids have a low thermal resistance *R* with a small heat flow compared to water. With an increase in the heat flow, the total thermal resistance *R* of the thermosiphon with water is much lower than that of dielectric liquids. It was also shown that the values of *R* in the evaporation and condensation zone are practically the same. However, in [1], it is noted that the main contribution to the total *R* is made by the heating zone, where the bubble boiling process occurs.

The value of thermal resistance and the maximum heat flux $Q_{\max}$ are also influenced by the amount of filled heat carrier and the length of the heat exchange zones. Thus, in [5], a study of the heat transfer characteristics of thermosiphons with a length and internal diameters of $6.7\cdot10^{-3}$ m; $9.5\cdot10^{-3}$ m; and $12.0\cdot10^{-3}$ m was reported. Water, methanol, ethanol, and acetone were used as heat carriers. It was shown that $Q_{\max}$ almost does not depend on the amount of heat carrier filling. Dependence was observed only on the internal diameter and saturation temperature (on pressure). Nevertheless, the unambiguous effect of the degree of filling on the intensity of heat transfer and the value of thermal resistance was not found.

The values of the heat transfer coefficients in the evaporation zone of a miniature thermosiphon differ significantly from the calculations according to the well-known Imura formula [6], which does not take into consideration the change in geometric characteristics and as claimed, does not depend on the amount of refueling with the heat carrier. With an increase in the internal diameter beyond the limit of $Bo=4$, the intensity of heat transfer in the evaporation zone can be calculated according to the dependence from [6], but with some error. This is given in work [7] where the maximum heat flux and heat transfer coefficients in the evaporation zone of the thermosiphon with an internal diameter of $25\cdot10^{-3}$ m with a change in the length of the heater were investigated. As it was shown, there is an effect of the amount of refueling on the value of the heat flow *Q*, and the intensity of heat transfer in the heating zone can be calculated according to the dependence from [6]. However, the effect of the amount of refueling at different lengths of heating zones is very difficult to detect.

Computer simulation of phase change processes of evaporation and condensation inside thermosiphon systems was carried out in [8]. The simulation results were compared with the experimental study of a thermosiphon with a length of 0.4 m and an internal diameter of $20.2\cdot10^{-3}$ m. It was shown that the minimum value of thermal resistance is observed with the vertical arrangement of the thermosiphon and the degree of filling with a heat carrier of 65 % of the volume of the heating zone. However, it is not entirely clear how this parameter is affected by a change in the inner diameter and length of the thermosiphon.

Reducing the internal diameter of the steam space of the thermosiphon also affects its main heat transfer characteristics. This is primarily due to the vaporization process in the heating zone. The conditions for the occurrence of steam bubbles differ from boiling on the surface in a large volume. The thickness of the thermal boundary layer in a limited space can be commensurate with the internal diameter of the heating zone and the heat carrier will be in a metastable (overheated) state almost along the entire height. The superheated liquid evaporates into a steam bubble, moving due to lift. The amount of steam increases to the size of the inner diameter of the thermosiphon heating zone and pushes part of the liquid to the condensation zone. This phenomenon is termed «geyser boiling» [9]. Such periodic emissions of heat carrier from the heating zone to the condensation zone lead to a change in heat transfer coefficients over time, which in turn will affect the thermal resistance value [10]. The influence of the angle of inclination and the amount of refueling on the occurrence of pulsation phenomena was studied in [11]. It was shown that temperature pulsations depend on these factors, but the pattern of influence is difficult to identify since the data are given during transient processes. And what is the effect of the amount of refueling and the amplitude of temperature pulsations on thermal resistance is still difficult to say. In [12], the effect of reducing the size of heat exchange surfaces while limiting the steam space on the intensity of heat transfer during boiling of heat carriers is shown. The intensity of heat transfer decreases under cramped conditions. However, the mechanism of this effect for different heat carriers has not been identified. Mechanical fluctuations in the movement of the heat carrier and the associated temperature pulsations of the surface [13] lead to a change in the intensity of heat transfer and, accordingly, a change in thermal resistance. On such a complex process of heat transfer in thermosiphons, the miniaturization factor is also superimposed. Therefore, the heat transfer characteristics of miniature thermosiphons will differ from the same evaporative-condensing devices with greater geometric characteristics. All this suggests that it is advisable to conduct a study aimed at determining the decisive factors that affect the transfer of heat in miniature two-phase thermosiphons. Such knowledge will give direction for the design of effective heat dissipation systems from small-sized devices.

3. The aim and objectives of the study

Our studies aimed to determine the features of heat transfer processes in miniature evaporative-condensing devices (thermosiphons) with low-temperature heat carriers and changes in geometric parameters to search for their optimal designs.

To achieve this goal, the following tasks were solved:

– to establish the influence of the internal diameter of thermosiphons on their thermal resistance;

– to determine the effect of the filling coefficient F_r and the thermophysical properties of heat carriers on the total thermal resistance of miniature thermosiphons;

– to derive a dependence for calculating the thermal resistance of miniature thermosiphons in the range of selected parameters of experimental samples.

4. The study materials and methods

Copper miniature thermosiphons with various heat carriers (distilled water, methanol, ethanol, and freon R113) were used as the object of our study. The volume of heat carrier was controlled by the weight method and was determined by the filling coefficient. It was the ratio of the volume of liquid V_l to the internal volume of the heating zone V_{ev} of the thermosiphon ($F_r=V_l/V_{ev}$). With the same internal diameter d_{in}, the filling coefficient was defined as the ratio of the filling height of the heat carrier L_l to the length of the heating zone L_{ev} ($F_r=L_l/L_{ev}$).

Experimental samples of copper miniature thermosiphons, which were investigated, are described in Table 1.

The study of thermal resistance was carried out at an experimental installation, which is shown in Fig. 1. Heat was supplied to the heating zone of the thermosiphon by an electric heater, which was wound on the body of the thermosiphon on top of heat-resistant and dielectric films with a thickness of $0.1 \cdot 10^{-3}$ m.

Heat removal from the condensation zone was carried out by running water through a pipe-in-pipe condenser and was controlled according to the RM 025Zh (8) rotameter readings. The temperature in the main zones of thermosiphons was determined using copper-constantan thermocouples, with a diameter of electrodes of $0.16 \cdot 10^{-3}$ m. Hot adhesions of thermocouples were soldered to the body of thermosiphons. The signal from thermocouples through an analog-to-digital converter was transmitted to a personal computer. Absolute error from the thermocouples did not exceed 0.1 °C.

The orientation in the space of the thermosiphon was vertical. The change in the heat flux of the heating zone of the thermosiphon was carried out using a laboratory autotransformer (5) and was controlled by the readings of the wattmeter (4).

To reduce heat loss to the environment, the thermosiphon was completely insulated with basalt fiber.

Experiments were carried out with a stepped supply of heat. The transition to the next mode was carried out only after the establishment of a stable mode, when the temperatures of all heat exchange zones did not change over time. The experiment ended when the temperature in the heating zone began to rise sharply uncontrollably as a result of the occurrence of a film boiling mode.

Table 1

Geometric characteristics of thermosiphons

Inner diameter, d_{in}, mm	Overall length, l_S, mm	Heating zone length, l_{ev}, mm	Condensation zone length, l_c, mm	d_{in}/l_{ev}	Filling factor, $F_r=L_l/L_{ev}$	Heat carrier
3.0	700	80	200	0.0375	2.14	Water
3.0	700	125	200	0.024	1.37	Water
3.0	700	200	200	0.015	0.83	Water
5.0	700	45	200	0.111	1.93	Water, methanol, freon R113
5.0	700	90	200	0.056	0.96	Water, methanol, freon R113
5.0	700	200	200	0.025	0.44	Water, methanol, freon R113
9.0	700	200	200	0.045	0.3	Water
9.0	700	200	200	0.045	0.57	Water
9.0	700	200	200	0.045	0.8	Water, Ethanol

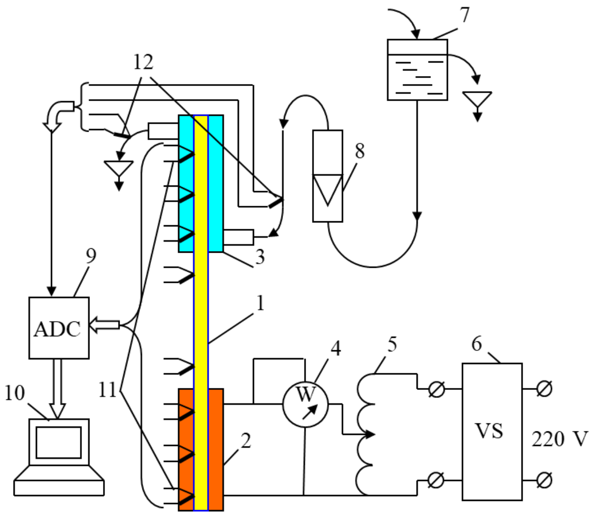

Fig. 1. Scheme of the experimental installation for the study of heat transfer characteristics of thermosiphons: 1 – two-phase closed thermosiphon; 2 – ohmic heater of the evaporation zone; 3 – condenser of the «pipe in pipe» type; 4 – wattmeter; 5 – laboratory autotransformer; 6 – voltage stabilizer; 7 – pressure tank; 8 – rotameter; 9 – analog-to-digital converter; 10 – personal computer; 11 – copper-constantan thermocouples (8 pcs); 12 – copper-constantan thermocouples for controlling the temperature of cooling water input-output (2 pcs)

The thermal resistance of thermosiphons was defined as the ratio of the difference between the external average temperatures of the heating zones T_{ev} and condensation T_c to the heat flux transmitted by this device Q_{out}:

$$R=\frac{\bar{T}_{ev}-\bar{T}_{c}}{Q_{out}}. \qquad (2)$$

The heat flux diverted from the thermosiphon was calculated according to the dependence:

$$Q_{out}=G\cdot C_p\left(T_{out}-T_{in}\right), \qquad (3)$$

where G *is* the flow rate of water cooling the condensation zone [kg/s]; C_p – specific heat capacity [J/kg·K]; T_{out}, T_{in} – water temperature after the condenser and before entering the capacitor [°C].

The error of determining Q_{out} did not exceed 5 %, and the thermal resistance R did not exceed 7 %.

5. The results of studies of the influence of determining factors on the heat-transmitting ability of miniature thermosiphons

5. 1. The effect of the inner diameter of thermosiphons on their thermal resistance

One of the main characteristics that determine the heat transfer capacity of a heat pipe or thermosiphon is thermal resistance. It should be said that the total thermal resistance of a thermosiphon is defined as:

$$R=R_{ev}^{w}+R_{ev}+R_{vap}+R_{c}+R_{c}^{w}, \qquad (4)$$

where R_{ev}^{w}, R_{c}^{w} is the thermal resistance of the thermosiphon wall in the heating and cooling (condensation) zones, R_{ev}, R_c – thermal heat transfer resistances in the heating and cooling zones, R_{vap} – thermal resistance of the steam space.

Given that the body of the thermosiphon is usually made of metal and has a small thickness, the thermal resistances of the wall in the heating and cooling zone can be neglected since they are several orders of magnitude smaller than other components.

Thermal resistance in steam space is determined by the hydrodynamics of the movement of the vapor-liquid mixture of the heat carrier and, above all, the speed of steam.

According to [2, 14, 15], this thermal resistance can be compared with R_{ev} and R_c at vapor velocities above 0.2M (M is the Mach number). This situation for conventional heat carriers occurs at pressures much lower than atmospheric ($P\ll 10^5$ Pa) and for liquid metals. For the temperatures and heat carriers that were studied, the value of the steam velocity is much smaller than the specified value, so the R_{vap} component can also be neglected.

As a result, we obtain that the thermal resistance of the thermosiphon is the sum of the thermal resistances of the heating zone R_{ev} and condensation R_c, that is, it is determined by the intensity of heat transfer processes in these zones.

Studies have shown that the thermal resistance of a thermosiphon is significantly affected by its diameter, moreover, this effect is different in the heating and cooling zone (Fig. 2).

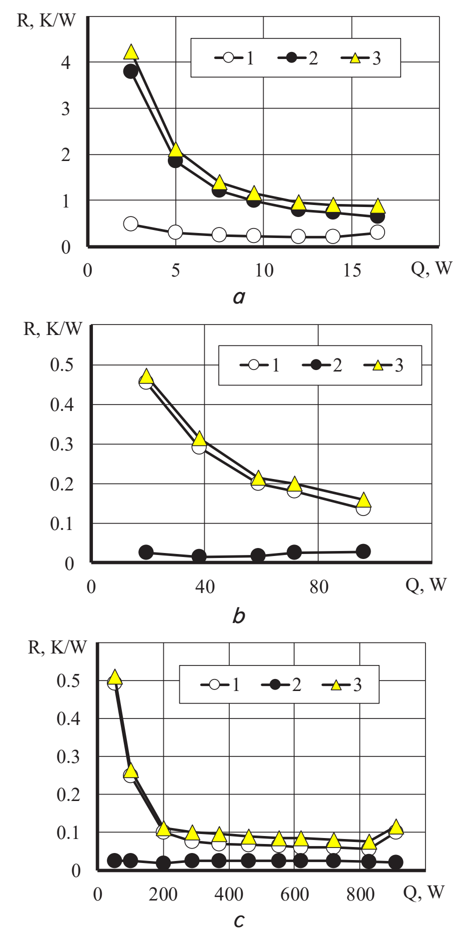

Fig. 2. The effect of the transmitted heat flux and the inner diameter on the thermal resistance of the thermosiphon with water as a heat carrier: *a* – d_{in}=3·10^{-3} m; *b* – d_{in}=5·10^{-3} m; *c* – d_{in}=9·10^{-3} m; 1 – R_{ev}; 2 – R_c; 3 – R_Σ

Thus, for thermosiphons with an internal diameter of 5·10^{-3} m and 9·10^{-3} m, R_{ev} significantly higher than R_c, and for a diameter of $3{\cdot}10^{-3}$ m, the thermal resistance R_c significantly exceeds R_{ev} at approximately the same lengths of the heating and condensation zone.

Such a discrepancy between thermal resistances in heat exchange zones relative to d_{in} can be explained by the fact that when boiling in the heating zone, the tear-off diameters of the steam bubbles become commensurate with the internal diameter of the thermosiphon. Then the capillary forces begin to exceed the forces of gravity and conditions are created for the appearance of a liquid plug in the condensation zone, which prevents the condensate from returning to the heating zone and the movement of steam in the other direction. For thermosyphons with an internal diameter of $3{\cdot}10^{-3}$ m, the probability of a liquid plug will be three times higher than for thermosyphons with an internal diameter of $9{\cdot}10^{-3}$ m. At the same time, in [16] it was also shown that in the condensation zone the thermal resistance was significantly higher than in the heating zone, despite the fact that the internal diameter of the thermosiphon was $8{\cdot}10^{-3}$ m.

With small dimensions of the steam space (Bo<4) [1], the transfer of heat in the thermosiphon mainly takes place during the projectile flow mode of the vapor-liquid mixture from the heating zone to the condensation zone. In the opposite direction, the condensate moves under the action of gravitational forces, which must exceed capillary forces. If inertial forces are not taken into consideration, then due to capillary forces, a condensate plug l_{pl} is formed, the length of which can reach significant proportions, but not more than the amount of heat carrier in the thermosiphon. Such a situation may arise when the action of these forces will be the same and the condensate plug will be stationary.

Capillary pressure P_{cap}, which acts on the condensate plug, can be calculated from the dependence [2]:

$$P_{cap}=\frac{4\sigma}{d_{in}}\cos\theta, \tag{5}$$

where θ is the angle of wetting the heat exchange surface with liquid.

Subject to perfect wetting, cosθ=1, and the dependence is simplified:

$$P_{cap}=\frac{4\sigma}{d_{in}}. \tag{6}$$

Capillary pressure does not depend on the angle of inclination of the thermosiphon and acts along the axis in the opposite direction from the movement of the condensate plug.

The pressure due to the gravitational forces acting in the direction of the heating zone depends on the angle of inclination and is calculated from the known dependence:

$$P_{g\varphi}=\rho' g h\sin\varphi, \tag{7}$$

where $h=l_{pl}\sin\varphi$ is the length of the condensate plug; φ – the angle of inclination of the thermosiphon to the horizon.

Then the pressure of the forces of gravity will be equal to:

$$P_{g\varphi}=\rho' g l_{pl}\sin^2\varphi. \tag{8}$$

Provided that the capillary and gravitational pressures are equal, the condensate plug becomes stationary, and it is possible to determine its length at different angles of inclination of thermosiphons.

Taking into consideration equations (5) and (8), the length of the fixed plug of condensate l_{pl} in the general form will be equal to:

$$l_{pl}=\frac{4\sigma\cos\theta}{d_{in}\rho' g\sin^2\varphi}. \tag{9}$$

There are limiting lengths of condensate plugs for different angles of inclination, heat carriers, and internal diameters of the steam space of thermosiphons. With a vertical arrangement (φ=90°) of a thermosiphon with $d_{in}=3{\cdot}10^{-3}$ m for water, the condensate plug l_{pl} reaches $9{\cdot}10^{-3}$ m. And for a thermosiphon with $d_{in}=9{\cdot}10^{-3}$ m, it does not exceed $3{\cdot}10^{-3}$ m. When the gravitational forces exceed capillary ($P_{g\varphi}>P_{sar}$), then the plug moves towards the heating zone. An increase in the length of the heat carrier plug leads to a change in forces in the opposite direction ($P_{g\varphi}<P_{sar}$) and the plug is either stationary or moving towards the condensation zone.

At the same time, in [3, 4], it was shown that in a thermosiphon with an internal diameter of $6{\cdot}10^{-3}$ m, the thermal resistances of the heating and condensation zone had approximately the same value. But the length of the thermosyphon was 0.2 m. This indicates that, in addition to the internal diameter, other factors also affect the thermal resistance (the length of the heating and condensation zones, the total length of the thermosyphon, the amount of heat carrier, etc.).

The tendency to increase the value of thermal resistance with a decrease in the internal diameter of thermosiphons characterizes the processes of heat transfer in such evaporative condensation systems. At the same time, the maximum value of the heat flux $Q_{\max}$ decreases while simultaneously increasing $R_{\min}$. The value of thermal resistance at the filling coefficient $F_r\approx0.93$ for water obeys the dependence of $R\sim Q^{-0.8}$ (Fig. 3).

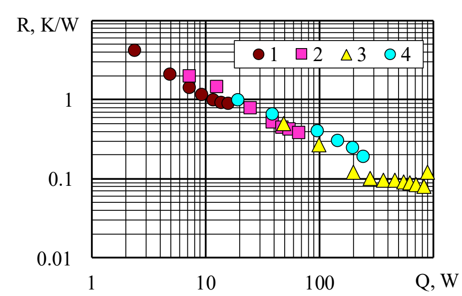

Fig. 3. The dependence of thermal resistance on the heat flux at different d_{in} (heat carrier – water; T_{in}=20 °C): 1 – $d_{in}=3.0{\cdot}10^{-3}$ m, F_r=0.83; 2 – $d_{in}=5.0{\cdot}10^{-3}$ m, F_r=0.96; 3 – $d_{in}=9.0{\cdot}10^{-3}$ m, F_r=0.8; 4 – data from [3], $d_{in}=6.0{\cdot}10^{-3}$ m, the total length of the thermosiphon is 0.2 m, F_r=1.06

An increase in $Q_{\max}$ and a decrease in $R_{\min}$ can be ensured by increasing the internal diameter of thermosiphons. Thus, for water in thermosiphons with $d_{in}=3{\cdot}10^{-3}$ m, $Q_{\max}$ does not exceed 15 W, and $R_{\min}$ reaches 1 K/W. At the same time, in thermosiphons with $d_{in}=9{\cdot}10^{-3}$ m $Q_{\max}$ reaches 900 W while $R_{\min}$ does not exceed 0.1 K/W.

5. 2. Influence of filling coefficients and thermophysical properties of heat carriers on the thermal resistance of miniature thermosiphons

To determine the effect of filling coefficients at a constant length of the heat carrier (h_l=const), experiments were carried out on thermosiphons with an internal diameter of $3\cdot10^{-3}$ m, $5\cdot10^{-3}$ m. In these thermosiphons, with the same filling of the heat carrier, the length of the heating zone changed (Fig. 4).

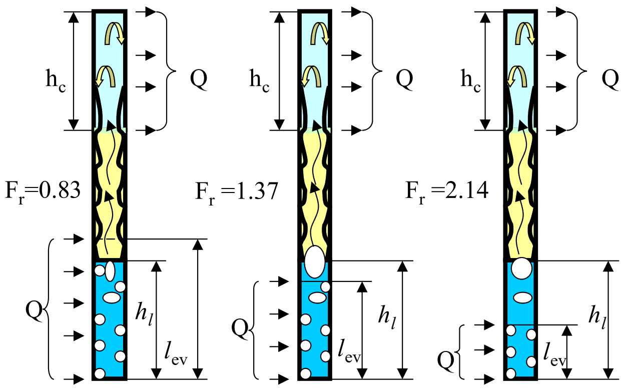

Fig. 4. Scheme of operation of a thermosiphon with an internal diameter of $3\cdot10^{-3}$ m with different F_r (h_l=const)

In a thermosiphon with an internal diameter of $9\cdot10^{-3}$ m with the same length of the heating zone (l_{ev}=const), the amount of the filled heat carrier changed (Fig. 5).

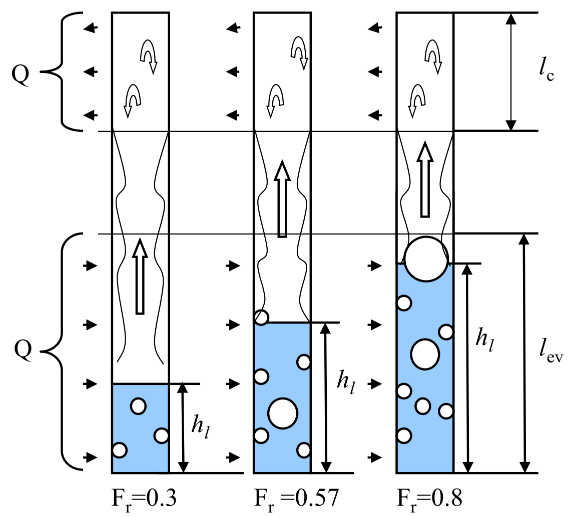

Fig. 5. Scheme of operation of a thermosiphon with an internal diameter of $9\cdot10^{-3}$ m with different F_r (l_{ev}=const)

The study of the effect of the filling coefficient F_r on thermal resistance at the same length of the heating zone (l_{ev}=const) showed that with an increase in the amount of heat carrier, the heat transfer characteristics of thermosiphons improve. This applies to increasing the maximum heat flux (Fig. 6). However, the minimum thermal resistance value for all filling coefficients was about the same up to Q_{max}.

A significant effect on the thermal resistance of the thermosiphon is exerted by the thermophysical properties of the heat carrier. Thus, Fig. 6 shows that the thermal resistance of a thermosiphon with an internal diameter of $d_{in}=9\cdot10^{-3}$ m with ethanol is significantly higher than the thermal resistance of a thermosiphon with water under other identical conditions. An obvious explanation for this is the lower thermophysical properties of ethanol compared to water. The coefficient of thermal conductivity λ is about 4 times and 3 times the heat of vaporization r.

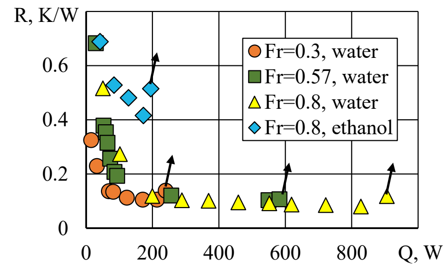

Fig. 6. Thermal resistance of a thermosiphon with a diameter of $9\cdot10^{-3}$ m for different filling coefficients F_r and heat carriers (l_{ev}=const)

The arrows indicate the mode of uncontrollable growth of thermal resistance caused by the phenomena of the heat exchange crisis in the heating zone. That is, the film boiling mode, at which the temperature of the thermosiphon wall in the heating zone rises sharply and the amount of steam decreases. Steam practically does not enter the condensation zone and the temperature difference between the heat exchange zones is growing at a rapid pace.

If we consider the change in the value of thermal resistance with an increase in the heat flux discharged, then the existence of two characteristic modes of heat transfer is visible. The first is observed at low thermal loads. During this period, the formation of thermal and hydrodynamic layers occurs. The temperature in the heating zone rises until the first vaporization center is activated. When this occurs, the bubble grows to the size of the inner diameter of the thermosiphon, and it pushes part of the heat carrier into the condensation zone. This process is repeated periodically. The average value of thermal resistance during this period is high. An increase in heat flux leads to the activation of the next vaporization center and the thermal resistance decreases. This nature of the decrease in thermal resistance is observed until the maximum number of vaporization centers occurs on the heating surface and the thermal resistance stabilizes at a certain minimum value. A further increase in the heat flux does not lead to a significant decrease in thermal resistance. Because the number of vaporization centers is maximum, and the intensity of heat transfer increases only by increasing the frequency of separation of steam bubbles.

The growth of Q_{max} with an increase in F_r is due to the fact that heat transfer in the heating zone occurs only in the area where the heat carrier is located. The larger the area of the heat exchange surface, the more centers of vaporization and the higher the intensity of heat transfer. Therefore, Q_{max} also increases.

A similar picture of the influence of thermophysical characteristics of heat carriers is observed for a thermosiphon with a diameter of $5\cdot10^{-3}$ m with different heat carriers with approximately the same filling coefficient F_r (Fig. 7).

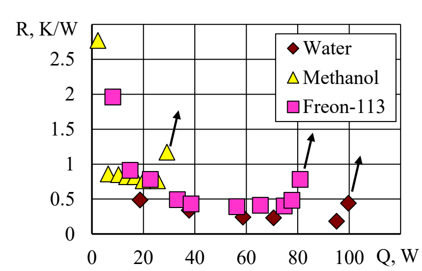

Fig. 7. The effect of the type of heat carrier on the thermal resistance of a thermosiphon with a diameter of $5\cdot10^{-3}$ m with a filling coefficient $F_r\approx0.48$

The effect of the filling coefficient at the same height of filling with a heat carrier (h_l=const) is shown in Fig. 8. The change in the filling coefficient F_r occurred at different lengths of the heating zone.

Fig. 8 demonstrates that at the same height of filling with heat carrier (h_l=const), the reverse pattern is observed compared to l_{ev}=const. With an increase in F_r, the maximum heat flux $Q_{\max}$ decreases, and the value of thermal resistance increases. Such an ambiguous influence of the filling factor F_r on the main heat transfer characteristics of thermosyphons $Q_{\max}$ and $R_{\min}$ is connected with the change of the effective length of the thermosyphon. The effective length of the thermosiphon is defined as the distance between the middles of the heating and condensation zones. It varied from 0.5 m at F_r=0.44 to 0.58 m at F_r=1.93. Changing the distance between the heat exchange zones when l_{ev}=const leads to a different effect of the filling factor on the thermal resistance. Increasing the effective length of the thermosiphon reduces its heat transfer capacity and, accordingly, reduces $Q_{\max}$ while increasing $R_{\min}$.

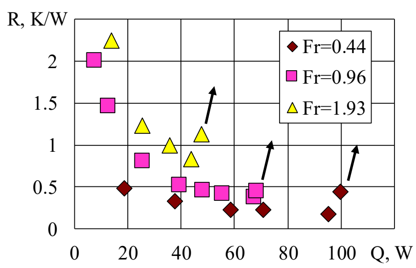

Fig. 8. Influence of the filling coefficient F_r on the thermal resistance of a thermosiphon with a diameter of $5\cdot10^{-3}$ m (heat carrier – water) (h_l=const)

5. 3. Generalization of research results on the influence of determining factors on the heat transfer capacity of miniature thermosiphons

As a result of the analysis of the study data, it was shown that the value of thermal resistance depends mainly on the filling coefficient F_r and the Reynolds number Re. With an increase in F_r, the thermal resistance increases to a degree of 0.5. At the same time, an increase in the Reynolds number leads to a decrease in thermal resistance. Also, the coefficient of thermal conductivity of the heat carrier λ and the inner diameter d_{in} of the thermosiphons are also inversely proportional. Experimental data on determining a thermal resistance are summarized thru the following dependences:

– at numbers Re<2,000,

$$R_\Sigma = 0.21\frac{1}{\lambda d_{in}}F_r^{0.5}\,\mathrm{Re}^{-0.8}; \qquad (10)$$

– at Re>2,000,

$$R_\Sigma = 6.2\cdot10^{-3}\,\frac{1}{\lambda d_{in}}F_r^{0.5}\,\mathrm{Re}^{-0.3}. \qquad (11)$$

The number Re was determined as follows:

$$\mathrm{Re} = \frac{W_s\cdot d_{in}}{\nu},$$

where ν is the kinematic viscosity of steam, m^2/s; W_s is the speed of steam:

$$W_s = \frac{4Q_{out}}{r\rho''\pi d_{in}^2},$$

where r is the heat of vaporization of the heat carrier, J/kg.

A generalization of experimental data for water, methanol, ethanol, and freon 113 is shown in Fig. 9. Dependences are built in the range of changes in the filling coefficient F_r=0.44...1.93 (Fig. 9).

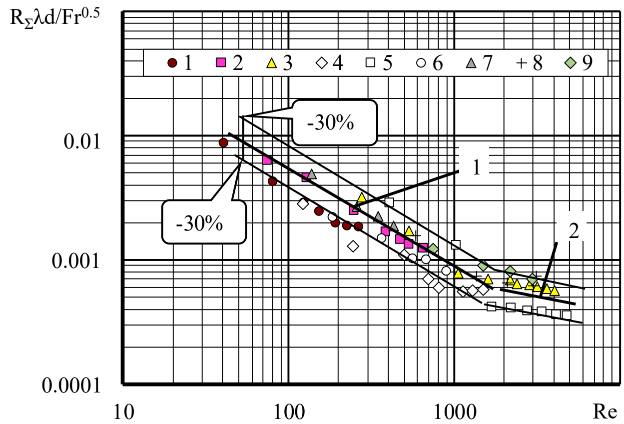

Fig. 9. Generalization of experimental data: line 1 – dependence (10), line 2 – dependence (11), 1 – d_{in}=$3.0\cdot10^{-3}$ m, F_r=0.83; 2 – d_{in}=$5.0\cdot10^{-3}$ m, F_r=0.96; 3, 8, 9 – d_{in}=$9.0\cdot10^{-3}$ m, F_r=0.8; 4 – d_{in}=$5.0\cdot10^{-3}$ m, F_r=0.49; 5 – d_{in}=$5.0\cdot10^{-3}$ m, F_r=0.5; 6 – d_{in}=$5.0\cdot10^{-3}$ m, F_r=0.44; 7 – d_{in}=$5.0\cdot10^{-3}$ m, F_r=1.93. 1–3, 6, 7 – heat carrier – water; 4 – methanol; 5 – freon 113; 8, 9 – ethanol

Approximation was performed by the least squares method. The dependences summarize 85 % of the experimental data for water, methanol, ethanol, and freon 113 with a spread of ±30 %. They are valid in the range of internal diameters d_{in} from $3.0\cdot10^{-3}$ m to $9.0\cdot10^{-3}$ m and the lengths of thermosiphons of 0.7 m. Characteristic change in the power index of the generalizing dependence at Re≈2,000 is associated with the maximum number of vaporization centers in the heating zone. This, in turn, leads to a change in the flow regimes of the vapor-liquid mixture in the thermosyphon.

6. Discussion of results of studying the influence of geometric factors and thermophysical properties of heat carriers on the thermal resistance of miniature thermosiphons

Features of heat transfer in miniature thermosiphons are mainly due to the influence of their geometric characteristics. With Bond numbers Bo<4 (compressed conditions), the formation of a thermal boundary layer depends on the inner diameter of the thermosiphon. With small diameters, there is a possibility that all the liquid will be superheated with respect to the saturation temperature. Then the appearance of the first steam bubble will lead to an increase in its size up to the inner diameter of the thermosiphon. Due to the increase in pressure in the heating zone and the action of inertial forces, this steam formation will push part of the heat carrier into the condensation zone. The temperature in the heating zone decreases, and in the condensation zone increases. This leads to the destruction of the boundary layer, and for the appearance of the next steam bubble, it is necessary to overheat the liquid again so that the vaporization center is activated again. And such a periodic phenomenon is observed at the initial heat fluxes [9–11]. At the same time, thermal resistance is of great importance, which is observed in Fig. 2. An increase in heat flow leads to the activation of new vaporization centers and a decrease in the impact of pulsation phenomena. And subsequently, the thermal resistance gradually decreases to a minimum value with a maximum thermal flow. It should be noted that an increase in the internal diameter reduces the minimum thermal resistance (Fig. 2). This is consistent with works [3, 17] but the authors did not consider the effect of the inner diameter on the value of the minimum thermal resistance. The data of our study on the influence of geometric factors and thermophysical properties of heat carriers on the thermal resistance of miniature thermosiphons suggest the following:

– the main difference in heat transfer in miniature thermosiphons is associated with a significant influence of their internal diameter;

– heat transfer characteristics also depend on the type of heat carrier, the number of refueling, and the lengths of the heat exchange zones (heating and condensation zones);

– the thermal resistance of miniature thermosiphons in the range of selected designs can be calculated from the resulting dependences (10), (11).

From a practical point of view, dependences (10), (11) can be used in the calculation of small-sized cooling systems for electronic equipment.

The application of dependences (10), (11) is limited to the use of heat carriers (water, methanol, ethanol, and freon 113), the range of internal diameters from 3.0 mm to 9.0 mm, and the total length of thermosiphons of 0.7 m. Employing the above dependences for conditions that differ from those studied requires additional experiments.

An increase in the heat transfer characteristics of miniature thermosiphons can be carried out by performing grooves on its inner surface [18], which will provide high intensity in the heat exchange zones and a low value of thermal resistance.

In the future, it is necessary to identify the influence of the effective length of miniature thermosiphons on their heat-transmitting characteristics and to search for the optimal amount of heat carrier refueling for different thermal loads.

7. Conclusions

1. It was found that an increase in heat flow leads to a decrease in thermal resistance in all cases until the specified increase leads to a crisis in the heating zone and the thermosiphon ceases to function. Reducing the internal diameter of the thermosiphon leads to an increase in its thermal resistance and a decrease in the maximum heat flow.

Thus, with a decrease in the internal diameter d_{in} from 9.0 mm to 3.0 mm, an increase in the minimum total thermal resistance R_{Σ} of the thermosiphon was recorded, from 0.075 K/W to 0.88 K/W. Simultaneously, a decrease in the maximum heat flux $Q_{\max}$ is observed (from 827 W to 16.5 W when changing the internal diameter d_{in} from 9.0 mm to 3.0 mm). A dependence was also derived for calculating the condensate plug, which is formed inside the thermosiphon due to the interaction of gravitational and capillary forces.

2. It is shown that there is an effect of the filling coefficient on the heat transfer characteristics of miniature thermosiphons depending on the method for determining F_r. At l_{ev}=const, the minimum thermal resistance is practically independent of F_r, and $Q_{\max}$ increases with an increase in F_r. With h_l=const, the increase in F_r leads to a decrease in $Q_{\max}$ while $R_{\min}$ increases. At the same time, there is an effect of the effective length of the thermosiphon. An increase in the effective length leads to a decrease in $Q_{\max}$, and an increase in $R_{\min}$. Data have also been obtained indicating that the thermophysical properties of heat carriers significantly affect the basic heat-transmitting characteristics of miniature thermosiphons $Q_{\max}$ and $R_{\min}$.

3. Two generalizing dependences were derived to calculate the total thermal resistance of miniature thermosiphons. Thermal resistance is mainly influenced by the thermal conductivity coefficient of the heat carrier λ, the inner diameter d_{in}, the filling coefficient F_r, and the Reynolds number Re. The use of the above dependences is limited to the list of working fluids (water, methanol, ethanol, and freon 113), the range of internal diameters d_{in} from 3.0 mm to 9.0 mm, and the total length of thermosiphons of 0.7 m. It is shown that there are two modes of heat transfer, which are characterized by changes in the movement of steam and steam liquid mixture.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

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