K and k, coefficients; L, length of the heater interval, m; N, number of bubbles; P, pressure, N/m^2; q, specific heat flux, W/m^2; R, heater radius, m; $\tilde{R}$, dimensionless radius; T, time, sec; $\eta = a/g$, acceleration load; ϑ, liquid subcooling at heater level, K; λ, wavelength, m; Λ, dimensionless wavelength; ρ, density, kg/m^3; σ, surface tension, N/m; ω, angular frequency, 1/sec. Subscripts: d, dominant wave; ', liquid; ", vapor; F, a plane heater; cr1, transition from nucleate to film boiling; cr2, transition from film to nucleate boiling; A, buoyancy force; σ, surface tension force; and 0, moment of loss of stability.

LITERATURE CITED

1. R. J. Simoneau and K. J. Baumeister, Adv. Cryog. Eng., 16, 416-425 (1971).
2. J. H. Lienhard and K. Sun, Trans. Am. Soc. Mech. Eng. Ser. C, J. Heat Transfer, 92, No. 2 (1970).
3. M. Pomerantz, Trans. Am. Soc. Mech. Eng. Ser. C, J. Heat Transfer, 86, No. 2 (1964).
4. N. M. Levchenko, "Investigation of cryogenic liquid heat transfer in a centrifugal force field: Visualization. Onset of boiling. Boiling crises," Preprint, Physicotechnical Insitute of Low Temperatures, Academy of Sciences of the Ukrainian SSR, Kharkov (1986).
5. S. I. Sergeev, O. M. Popov, and I. P. Vishnev, Inzh.-Fiz. Zh., 46, No. 1, 100-107 (1984).
6. J. H. Lienhard and P. T. Y. Wong, Trans. Am. Soc. Mech. Eng. Ser. C, J. Heat Transfer, 86, No. 2 (1964).
7. R. Bellman and R. Pennington, Q. Appl. Math., 12, 151 (1954).
8. Yu. A. Kirichenko, Inzh.-Fiz. Zh., 25, No. 1, 5-13 (1973).

OPERATION OF A U-SHAPED THERMOSIPHON AT SMALL ANGLES OF INCLINATION

V. A. Krivonos

UDC 536.248.2

The effect of the volume of heat-transfer agent and the angles of inclination of the branches of a thermosiphon on its internal thermal resistance is determined.

Two-phase thermosiphons are being increasingly used in various branches of technology and this has found expression in the volume of experimental research [1-9].

In a number of cases it is necessary to use thermosiphons of complex configuration. Thus, in [8] a U-shaped thermosiphon with two straight tubes (branches), parallel to the horizontal and to each other, was considered. The charge of heat-transfer agent and the spatial arrangement of the branches were not varied during the experiments.

Our object was to determine experimentally the optimum charge of heat-transfer agent for a U-shaped thermosiphon and the effect on its operation of the angles of inclination of the thermosiphon as a whole and its individual branches.

The thermosiphon (Fig. 1) had the following geometric characteristics: outside diameter 16 mm, wall thickness 1.5 mm, total length 1250 mm, lengths of heating and cooling zones 195 and 600 mm, respectively; the thermosiphon material was copper and the heat-transfer agent was distilled water.

On the inner surface of the heating zone we formed a metric thread with a 1-mm pitch. The heating and cooling zones make up the two branches, which are arranged one above the other at an angle of up to 10°. This type of thermosiphon is intended for use in the heat-transfer devices of the cooling systems of radioelectronic assemblies. The thermometric measurements were made with Chromel–Copel thermocouples. The thermocouple emf was recorded by means of a KSP-4 potentiometer (accuracy class 0.25). The temperature of the heating zone was measured with four thermocouples. In the adiabatic part on the surface of the thermosiphon beneath the insulation in the steady-state operating regime the surface temperature was equal to the vapor temperature. This temperature was determined by the thermocouple at the point T_v. Along the length of the cooling zone the temperature was measured at seven uniformly spaced points. The heating zone was wrapped with one layer of mica paper, on which nichrome wire 1.0 mm in

A. V. Lykov Institute of Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR, Minsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 53, No. 1, pp. 20-26, July, 1987. Original article submitted April 10, 1986.

0022-0841/87/5301-0753$12.50

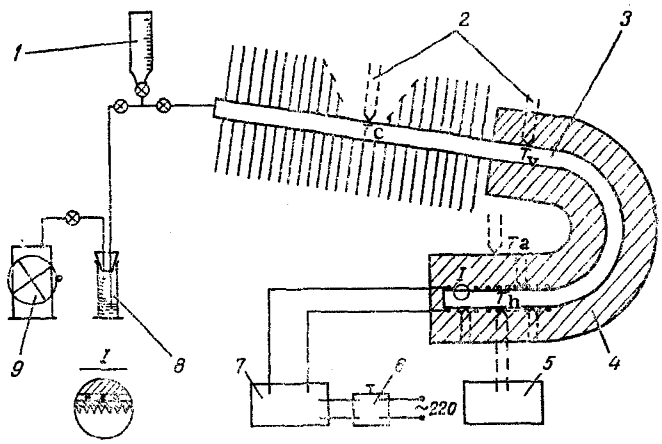

Fig. 1. Diagram of experimental apparatus for determining the heat-transfer characteristics of a U-shaped thermosiphon: 1) reservoir; 2) location of thermocouples; 3) thermosiphon; 4) insulation; 5) potentiometer; 6) regulating transformer; 7) wattmeter; 8) measuring vessel; 9) Komovskii pump.

diameter was wound, and then insulated with two layers of glass cloth. The whole was covered with a layer of asbestos cord 40 mm thick whose outer surface was wrapped witb copper foil. To the foil we soldered a thermocouple at the point T_a in order to estimate the heat losses from the heater to the ambient space. Cooling was accomplished by natural convection of the ambient air at room temperature. In order to reduce the external thermal resistance, the cooling zone was constructed in the form of rectangular aluminum fins measuring 180 × 100 × 1.5 mm and spaced 7 mm apart. Power was supplied to the heating zone through a regulating transformer (LATR) and registered by a wattmeter. The power was varied from 60 to 250 W. As shown by systematic measurements, the heat losses from the heater to the ambient space were not more than 5%.

Before loading with the heat-transfer agent, we first degreased the thermosiphon with acetone and washed it with distilled water. In [8] a similar system was filled by the "evaporation" method, which is the simplest and quite reliable. In our case, in view of the developed heat-shedding surface in the cooling zone, it is difficult to use this method, especially when the volume of heat-transfer agent is a variable parameter of the investigation, since it is necessary to create special thermal insulation for the finned cooling zone. We therefore used a method which consisted of filling the entire volume of the thermosiphon, which was known beforehand, with heat-transfer agent and then withdrawing it in batches into a preevacuated measuring vessel. Thus, during the experiment the entire mass of heat transfer agent was constantly present in the thermosiphon–measuring vessel system. From the amount of heat-transfer agent in the measuring vessel, in both liquid and gaseous phases, it is possible at any time to determine quite accurately the amount of heat-transfer agent in the U-shaped thermosiphon.

It is clear from Fig. 2 that the volume of heat-transfer agent has a very strong influence on the efficiency of the cooling zone by reducing the heat-removal surface. This finally leads to an increase in the average temperature in the heating zone (Fig. 2b), i.e., in the cooled object. The observed scatter of the temperature values (Fig. 2b) is constant in all the experiments and is associated both with the difference in the position of the thermocouples around the circumference of the heating zone (see Fig. 1) and the care in attaching them. These graphs give only a qualitative picture of the effect of the charge level on the thermal resistance of U-shaped thermosiphons, without supplying a quantitative answer to the question: what value of V minimizes the thermal resistance, i.e., makes the cooling zone isothermal? In order to determine the optimum value of V, from the standpoint of obtaining minimum temperatures in the heating zone, we plotted a graph of the dependence of the length of the isothermal cooling zone on the volume of heat transfer agent. The data were taken from Fig. 2a. In Fig. 3 for a series of values of V we have noted the coordinates of two adjacent measurements along the length of the cooling zone beweeen which isothermicity clearly broke down. Through the three points we drew straight lines and extrapolated them to intersection with the 600 mm ordinate (end of cooling zone). It was found

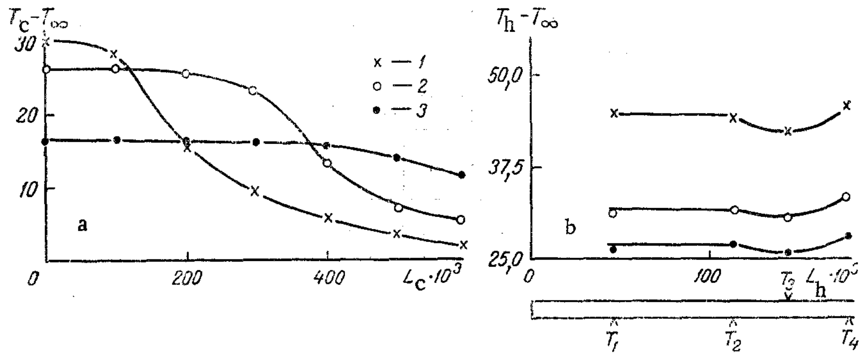

Fig. 2. Temperature distribution over the length of the individual zones of the thermosiphon as a function of the volume of heat-transfer agent [$Q = 120$ W, $\varphi_h = 0°$, $\varphi_c = 8°$; 1) $V = 75$ cm^3 (45%); 2) 55 (33%); 3) 39 (23%)]: a) cooling zone; b) heating zone. T, °C, K; $L \cdot 10^3$, m.

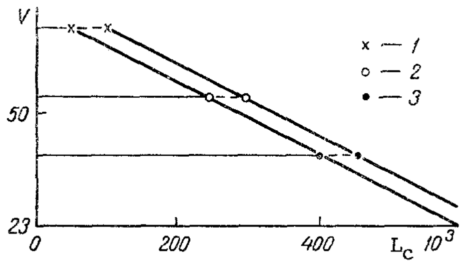

Fig. 3. Determination of the optimum volume of heat transfer agent: 1) $V = 75$ cm^3 (45%); 2) 55 (33%); 3) 39 (23%). $L \cdot 10^3$, m.

that at $V < 23$ cm^3 the effect of the volume of heat-transfer agent on the operation of the cooling zone is eliminated, as was also confirmed experimentally. A further decrease in V leads to a fall in the critical power. Accordingly, allowing for the experimental error, as the nominal volume we chose $V = 20$ cm^3. This is approximately 12% of the internal volume of the thermosiphon and 75% of the volume of the heating zone (volume of heating zone 16%). In this case the cooling zone will possess low thermal resistance, and the heating zone will have minimum temperatures.

It should be noted that, in all the graphs, as the temperature values we have taken the difference between the corresponding temperature and the temperature of the ambient air. For example, in Fig. 2a at the beginning of the cooling zone ($L_c = 0$) for $V = 75$ cm^3 the temperature was 55°C at an air temperature of +24.8°C. Correspondingly, the values of the temperature difference for $V = 75$ cm^3 (curve 1) in Fig. 2b must be increased by 24.8°C in order to obtain the absolute values of the temperatures in the heating zone. However, it is not necessary to know the air temperature to obtain the generalized characteristics.

The quantities α_h and R_{ts} can be determined from Fig. 4. The quantity α_c was not determined experimentally since that would have involved certain difficulties relating to the correct measurement of the temperature difference $\Delta T_c = T_c - T_y$, which is very small because of the high efficiency of the condensation process, the considerable length of the cooling zone, and the relatively low power values.

We also investigated various positions of the U-shaped thermosiphon as a whole in the vertical plane in order to determine the angular limits of operation, i.e., the point at which the temperature of the heating zone begins to rise continuously at a constant value of the power supply. The results of the investigation are presented in Table 1 for the case in which the angle between the branches of the thermosiphon $\varphi_{ts} = 3.5°$. The values of the angles φ_c and φ_h are assumed to be positive if the resulting inclination of the heating and cooling zones does not lead to a deterioration in their operation. If the operation of the zones deteriorates when their angular position is changed, the values of φ_c and φ_h are considered to be negative.

TABLE 1. Effect of Angle of Inclination of U-Shaped Thermosiphon on Its Operation

φ_h, deg	φ_c, deg	T_c, °C	T_v, °C	Heater, °C				T_∞, °C	Steady state
				T_1	T_2	T_3	T_4		
0	3,5	38,0	41,5	46,0	46,0	46,5	50,5	24,0	Yes
+2,0	1,5	37,3	40,5	43,5	43,5	45,0	49,5	23,0	»
+2,5	1,0	36,2	40,0	43,5	43,5	44,5	49,5	23,0	»
+3,0	0,5	36,0	39,3	>150	87	62	51	23,0	No
—1,5	5,0	37,8	41,2	45,5	46,0	47,0	50,5	23,5	Yes
—4,5	8,0	37.0	42	>150	120	63	51	23,5	No

<u>Remark.</u> $Q = 100$ W; $\varphi_{ts} = \varphi_c + \varphi_v = 3.5°$; $V = 20$ cm³.

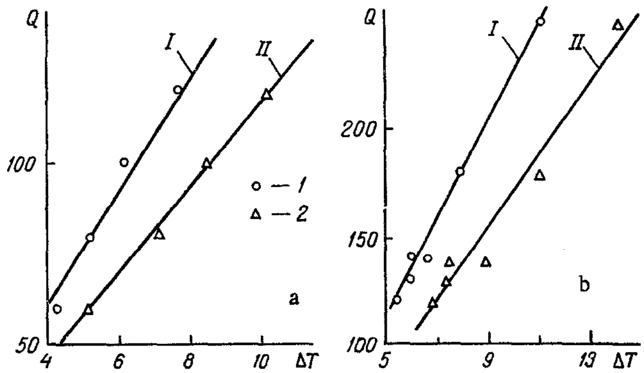

Fig. 4. Temperature difference as a function of the power supply [$V = 20$ cm³ (12%), $\varphi_h = 0°$; 1) $\Delta T = T_h - T_v$, 2; $\Delta T = T_h - T_c$]: a) $\varphi_c = 3.5°$; I) $\alpha_h = 1553$ W/(m²·K); II) $R_{ts} = 0.087$ K/W; b) $\varphi_c = 8°$; I) $\alpha_h = 2278$ W/(m²·K); II) $R_{ts} = 0.058$ K/W. Q, W.

In Fig. 2 it is possible to distinguish a tendency for the temperature drops along the length of the U-shaped thermosiphon to decrease with decrease in the volume of heat transfer agent. The presence of noncondensing gases in the tubes leads to a similar temperature distribution along the length of the cooling zone. In the case in question the main cause is the accumulation of liquid heat transfer agent in the cooling zone. In all the experiments, with increase in power the reduced-temperature zone increased. In the case of a noncondensing gas the opposite situation should be observed.

It was also noted that on the range of angles φ_c from 3.5° to 8° the mode of formation of the liquid plug remained almost unchanged.

The complete cycle of investigations showed that for a U-shaped thermosiphon with the optimum amount of heat transfer agent (20 cm³) at an angle of inclination of the cooling zone $\varphi_c = 3.5°$ the critical power is equal to 140 W. When $\varphi_c = 8°$, crisis conditions were not reached even at $Q = 250$ W. Higher powers were not investigated, since at $Q = 250$ W the temperature of the heating zone had already reached 70°C (the temperature limit of normal operation of most semiconductor devices). One particular point is worth noting. In the experiments the power was usually raised in steps of 10-20 W following the establishment of steady-state conditions in each previous regime. However, if the power was raised abruptly from 0 to 250 W, then after a certain time crisis conditions developed in the thermosiphon. For this method of power supply the maximum power was 180 W.

As the angle φ_c was varied from 3,5° to 8°, the heat-transfer coefficient in the heating zone α_h increased from 1553 to 2278 W/(m²·K). Usually, a change in the angular position of the cooling zone of straight thermosiphons has little effect on the value of α_h [9]. In a U-shaped system during the operating period most of the liquid phase of the heat-transfer agent is in the lower branch of the thermosiphon. It is constantly in a state of pulsation, moving back and forth between the end of the heating zone and the beginning of the bend. Thus, the thermosiphon operates in a regime similar to the "choking" regime with constant coverage of its internal cross section at different points along the length of the lower branch. As the power increases, the pulsation zone gradually shortens as a result of its being forced away from the end of the heating zone owing to an increase in the partial pressure of the vapor phase of the heat transfer agent. Consequently, the liquid phase penetrates increasingly rarely into the end of the heating zone, which leads to overheating and the onset of crisis conditions.

As the angle φ_c increases (increase in radius of bend), the liquid phase is distributed more and more uniformly over the internal volume of the two branches and does not prevent to the same degree as before the escape of vapor phase from the heating zone, which also leads to an increase in α_h and the critical power. The thermosiphon begins to operate in a regime more characteristic of phase transitions in liquid films.

In the course of the experimental investigations we repeatedly noted periodically recurring hammer, accompanied by a characteristic "metallic" sound, in the region of the bend. As the radius of the bend increased, the intensity and force of these shocks noticeably decreased.

An analysis of the data in Table 1 shows that if some change in the position of the thermosiphon in the vertical plane does not result in a deterioration of the operation of the condensation zone, then it will have a negative effect on the operation of the heating zone and vice versa. In fact, whereas the cooling zone operates successfully, i.e., normally supplies the heating zone with the necessary amount of condensed heat transfer agent, at angles $\varphi_c > 0.5°$, in the heating zone even at $\varphi_c = 8°$ crisis conditions may develop as a result of the impossibility of heat transfer agent entering the end of the zone because of its elevation ($\varphi_h = -4.5°$). Wheras in our case the heating zone works well at angles $\varphi_h > -4.5°$, at $\varphi_h = +3.0°$ the cooling zone is practically perpendicular to the gravity field vector and does not transport the necessary amount of condensed heat transfer agent into the heating zone. Hence, it follows that the values of φ_c and φ_h should lie on the following intervals: $8° > \varphi_c > 0.5°$, $3.0° > \varphi_h > -4.5°$, if the angle between the branches $\varphi_{ts} = 3.5°$.

Further investigation showed that in the case of U-shaped thermosiphons with angles between the branches of up to 10°, for which the ratio of the heating and cooling zone lengths is 1:3, it is possible to recommend the following branch angle working intervals: $\varphi_{ts} - 0.5° > \varphi_h > -4.5°$; $\varphi_{ts} + 4.5° > \varphi_c > 0.5°$.

It should be noted that the occurrence of crisis conditions in the operation of the U-shaped thermosiphon was always first recorded in relation to the temperature T_1 (end of heating zone) with the subsequent successive response of the thermocouples T_2, T_3, and T_4. In individual cases, for example, at $\varphi_c = 8°$, $\varphi_h = -4.5°$, crisis effects were never observed at the point T_4.

NOTATION

T, temperature; Q, power; φ, angular position of the thermosiphon zones; L, length; V, volume of heat-transfer agent; α, heat-transfer coefficient; and R, thermal resistance. Subscripts: c, cooling zone; h, heating zone; ∞, ambient space; ts, thermosiphon; and v, vapor.

LITERATURE CITED

1. A. N. Alabovskii, A. I. Beloivan, N. Yu. Koloskova, et al., in: Heat Conduction and Convective Heat Transfer [in Russian], Kiev (1980), pp. 33-35.
2. M. G. Semena, Inzh.-Fiz. Zh., 35, No. 3, 397-403 (1978).
3. S. N. Fainzil'berg, N. Yu. Koloskova, and M. G. Semena, Izv. Vyssh. Uchebn. Zaved., Energet., No. 6, 86-88 (1978).
4. E. Hahne and U. Gross, Proc. 4th Int. Heat Pipe Conf., London (1982), pp. 125-136.
5. H. Nguen Chi, M. Groll, and Thong Dang-Van, AIAA Paper No. 1106 (1979), pp. 1-7.
6. I. G. Kiselev, A. I. Isakeev, V. V. Filatov, and N. I. Istomin, in: Boiling and Condensation [in Russian], Riga (1979), No. 3, pp. 74-83.
7. D. Chisholm, I. D. R. Grant, and P. J. Duncan, 6th Int. Heat Transfer Conf., Toronto (1978), Vol. 3, pp. 117-122.
8. K. T. Feldman and S. J. Munje, J. Energy, 3, No. 4, 211-216 (1978).
9. M. G. Semena and Yu. F. Kiselev, Inzh.-Fiz. Zh., 35, No. 4, 600-605 (1978).