T is the transport section;
C is the condenser section;
V is the vapor-flow passage;
HP is the heat pipe;
R is the reservoir;
e is the end of sylphon bellows;
av is the average (for siphon bellows);
0 are the initial parameters;
c is the conduction;
r is the radiation.

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MAXIMUM HEAT-TRANSFER CAPACITY OF A VERTICAL TWO-PHASE THERMAL SIPHON

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A survey of the experimental data on the maximum heat-transfer capacity of a two-phase thermal siphon is presented; a physical model that describes many of the experimental data on the heat-transfer limits for thermal siphons is proposed.

A two-phase thermal siphon works with an evaporation–condensation cycle and represents an efficient heat-transfer device that can often compete successfully with other heat exchangers.

The limiting heat flux carried by such a siphon is a major working characteristic; however, at present there is no agreed view on the limit to the heat transfer through a vertical two-phase siphon. This limit may be called the critical heat transfer. Various types of crisis should be distinguished [1] in terms of the physical principles. There is a deterioration in the heat transfer if the layer of liquid at the wall is disrupted by the interaction between the phases (type I crisis). The film of liquid evaporates on account of inadequate supply in a type II crisis. Here we consider the crisis arising from interaction between the phases, which disturbs the countercurrent flow in the two-phase boundary layer. Many of the experimental results are qualitative rather than quantitative.

For example, the drying occurring at the heating surface has been discussed [2, 3] in terms of interaction between the countercurrents of vapor and liquid. A qualitative description of this phenomenon has been given [3], while the relationship given in [2] applies only for the conditions considered in that paper, and it cannot be used, for example, to explain the heat-transfer limit due to instability in the liquid film [4]. In [5],

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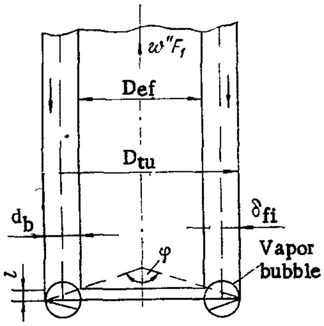

Fig. 1

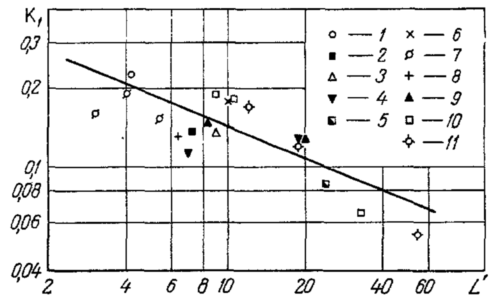

Fig. 2

Fig. 1. Scheme for determining the resistance coefficient.

Fig. 2. Stability criterion K_1 as a function of L': 1) water [2]; 2) n-hexane [2]; 3) CCl_4 [2]; 4) ethane [2]; 5) Freon 11 [5]; 6) water [5]; 7) water [4]; 8) Freon 11 [4]; 9) water [6]; 10) ethanol [6]; 11) Freon 12 [6].

the initial force-balance equation was so extensively simplified that the picture was not that of the real physical situation at all.

Considerable interest attaches to [6], in which the crisis in heat (mass) transfer was represented as due to hydrodynamic instability in the two-phase flow. This corresponds to the actual situation in the process, since the hydrodynamic instability is fundamental. Unfortunately, the main conclusions of [6] are incorrect. For example, it was asserted that the heat-transfer crisis is independent of the geometrical dimensions of the thermal siphon, which is in conflit with the author's own data as well as with the results of [5]. There is no justification for distinguishing two modes of crisis, since the graph (Fig. 4 on p. 596 of [6]) for the two conditions can be described by a single equation with an error of ±35%. There is no basis for relating the maximum heat transfer and the change in flow conditions to the physically unimportant thickness of the liquid film.

Here we describe the available data on the heat-transfer crisis in a thermal siphon from a unified viewpoint and derive a relationship for the onset of the limiting heat flux.

The heat transfer in a siphon occurs by evaporation, mass transfer, and condensation; the mass transfer is disrupted primarily on account of radical change in the hydrodynamic and thermal parameters. We now describe the hydrodynamic flow pattern preceding the crisis. The description is based on our own visual observations.

The direction of the heat flux has a marked effect on the motion of the vapor in such a siphon (heat input from the side or end). The heat transfer with input from the end is the more similar of the two to boiling in a large volume, so we consider it as a particular case of the latter later on.

We first consider the more complicated and more common case of lateral input; here there are distinct cases in which the working liquid fills either a small (1-5% of the cavity) or a large (5-33%) proportion. In the first case, the heat transfer occurs in the flowing film of condensate. The vapor is generated in the film and initially moves radially, but then moves axially in response to the overall pressure difference along the siphon. The change in direction involves some loss of pressure arising from eddies, which is the greater, the larger the radial component of the velocity. If the heat loading is small ($q < 1$ W/cm^2 for water), evaporation occurs directly from the film, whereas at higher heat loads, up to the critical value, one gets boiling in the film, with the formation of a large number of bubbles. When these break, they eject jets of vapor in a radial direction (the momentum of which is the larger, the greater the heat input).

It has been shown [7] that the pressure loss in natural convection arising from sharp changes in direction makes a substantial contribution to the overall pressure loss in the thermal siphon.

If the filling factor is small, the surface drying is local and causes an increase in the local resistance; the pressure loss arising from the local resistance at loads close to the critical value tends to retard the film of condensate and break it up. The disruption by the bubbles also tends to produce uncovered areas, since drops of liquid are ejected [8]. The pressure loss due to eddies is substantially dependent on the size of the siphon. If the diameter D_{tu} is small (6-8 mm), the film is disrupted and retarded, with the result that the cooling zone tends to become choked with the condensate [4, 9].

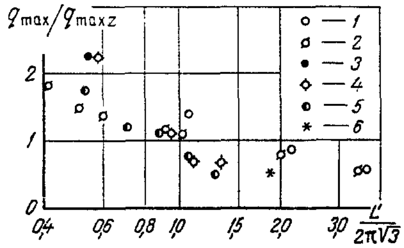

Fig. 3. Comparison of the present results with those of [11]: 1) ethanol; 2) water; 3) ethanol [11]; 4) isopropanol [11]; 5) methanol [11]; 6) water [11].

A somewhat cifferent picture precedes the onset of crisis when the filling factor is large; the transitional state of boiling shows a large number of partially fused bubbles at the evaporation surface. The size of the evaporation patches is small by comparison with the total heat-transfer surface, and the escape of vapor into the vapor space is hindered, which results in an increase in the vapor content within the liquid. The vapor content increases rapidly as the critical load is approached, and there is a rise in the overall pressure difference in the system, so some of the vapor–liquid mixture is displaced to the upper part of the volume, and a region with an elevated vapor content is formed within the core of the working liquid, while a film moves along the wall, as in the first case. Uncovered surface areas then arise as described above. This process is accompanied by the ejection of much of the liquid to the condensation end in a small-diameter siphon, which resembles the process described above.

Therefore, the main cause of a limit to the mass transfer is the increase in the local resistance arising from the sharp change in vapor flow direction, which results in a fall in the pressure at any local resistance.

A key aspect of this is to determine the resistance coefficient for a two-phase siphon; Fig. 1 shows an approximation for our model, where the entry of the vapor into the cavity is taken as resembling the entry of liquid into a collector with a straight generator (conical collector without an end wall) [10]. It that case the resistance coefficient is

$$\zeta = \zeta_1 \left(1 - \frac{F_1}{F_0}\right), \tag{1}$$

where ζ is dependent on the relative length l/D_{ef} and on the convergence angle φ of the flow, F_1 while is the effective cross section of the channel, which is defined as the difference between the cross section F_0 of the tube and that of the annular region formed by the equivalent diameter of the bubble. The convergence angle and the relative length are somewhat arbitrary, since they are dependent on the angle subtended by the bubble at the instant of bursting (vapor ejection). Experimental data on this aspect are lacking, so we are forced to estimate φ and l/D_{ef}.

The qualitative picture of [8] and electrical simulation of bubble bursting [4] indicate that the angle of escape of the vapor from a bubble varies within fairly narrow limits (about 30-60°). The angle tends to be less if a flow of liquid strikes the top of the bubble; correspondingly, φ varies from 120 to 150°. Geometrical considerations indicate that l can vary from $0.5r_b$ to $0.85r_b$ within this angular range. Then ζ may be determined with an error not more than ±6% within these parameter ranges. Here we assumed $\varphi = 140°$, $l = 0.7r_b$, and $D_{ef} = D_{tu} - 2d_b$.

We have seen above that ζ is dependent on the tube diameter; we take the capillary constant d_b as our length scale to relate ζ to D_{Tu}/d_b; the $\zeta = f(D_{tu}/d_b)$ function constructed from the data [2, 4-6] shows that the resistance increases considerably in tubes of small cross section, while there is a monotonic fall as the diameter increases.

The resistance coefficient then allows us to discuss the stability of the countercurrent system. The stability of a two-phase flow is dependent on Kutateladze's stability criterion [8]:

$$K = \omega''_{cr} \sqrt{\rho''} / \sqrt[4]{\sigma g (\rho' - \rho'')}. \tag{2}$$

In our model, this takes the form

$$K_1 = \zeta\omega''_{cr}\sqrt{\rho''}/\sqrt[4]{\sigma g(\rho'-\rho'')}. \tag{3}$$

The published measurements and our own studies indicate that K_1 is dependent on the linear dimension of the system $L' = D_{tu}/d_b$, i.e., the correlation equation $K_1 = f(L')$ applies. Figure 2 shows K_1 as a function of L'. This approximate relation for K_1 applies for a wide range of pressures (0.02-20 bar) and for wide ranges in the geometrical parameters (D_{tu} = 6-50 mm) for various liquids (water, ethanol, Freons, n-hexane, and CCl_4) and is

$$K_1 = 0.36\,(L')^{-0.4} \tag{4}$$

with an error of ±35%.

Any increase in the diameter or in the heat input from the end should result in a closer analogy with boiling in a large volume for heaters of finite length, so we perform a comparison with that case. A thermal siphon or a heater in a large volume will show free convection and the same type of heat-transfer crisis. The difference lies in the mode of vapor generation. Lateral heat input causes additional pressure loss due to the eddies, which results in an earlier crisis at the heating surface.

We use the data of [11], in which the onset of crisis was examined with plates of finite dimensions, whereas in other studies boiling was examined in large tanks. The data [11] were processed in ($q_{max}/q_{max\,z}$, $L2\pi\sqrt{3}/\lambda_d$) coordinates, which requires a conversion for our purpose. The ratio of q_{max} to $q_{max\,z}$ represents the Kutateladze–Zuber equation, as has been demonstrated [12]; $q_{max\,z}$ was taken as characteristic of the boiling liquid to which q_{max} should be referred. The quantity $L' = L2\pi\sqrt{3}/\lambda_d$ is a dimensionless linear characteristic of the heater, which is expressed in terms of Taylor-instability wavelengths for the vapor escaping from the vapor–liquid surface, and $\lambda_d = 2\pi\sqrt{3\sigma/g(\rho'-\rho'')}$ is the most sensitive wavelength of the Taylor instability for a horizontal interface. A correlation relationship was formulated:

$$q_{max}/q_{maxz} = f(L'), \tag{5}$$

which applies also in our cases. We compare our conclusions with (6) by expressing our relative dimension $D_{tu}/\sqrt{\sigma/g(\rho'-\rho'')}$ in terms of wavelengths by dividing by $2\pi\sqrt{3}$.

Figure 3 compares the data examined here with those of [11]; the comparison is made for a single jet. In [11], Zuber's model was used with various numbers of vapor jets at the heater, including a single jet bounded by vertical walls, which is similar to the situation in a thermal siphon. The comparison was performed for ethanol and water. Although the range of our data for ethanol is narrower than that in [11], the results are in good agreement for $L' \gtrsim 1$; the agreement should also be satisfactory for $L' < 1$, because the relationship is of the form reported for ethanol [11] in the case of water, where the range of conditions was wider.

Borishanskii [16] has shown that q_{max} is effected by the viscosity, which can be characterized by $N = \rho\sigma/\mu^2$; it has also been pointed out [13] that induced convection has a considerable effect for $L' < 1$, i.e., viscosity affects q_{max}. If N is introduced into $I = \sqrt{NL'}$, which represents the ratio of the inertial and surface-tension forces to the viscous forces, we obtain [13] the following correlation:

$$q_{max}/q_{maxz} = f(L', I). \tag{6}$$

Figure 4 shows $q_{max}/q_{max\,z}$ as a function of I for the range L' = 0.4–4; clearly, no viscosity effect appears for the various liquids for I > 800, as has been concluded elsewhere [13]. Since we did not examine any values I < 800, no comparison is made for this range.

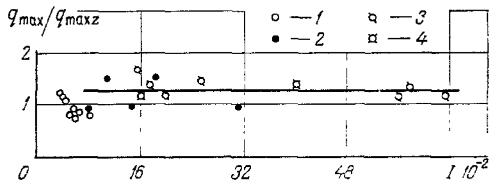

Fig. 4. Maximum heat flux as a function of I for L' = 0.4-4.0: 1) ethanol [11]; 2) ethanol [6]; 3) water [4]; 4) Freon 12 [6].

NOTATION

ζ	is the local resistance coefficient;
w''_{cr}	is the reduced critical rate of vapor formation;
ρ', ρ''	are the densities of liquid and vapor respectively;
φ	is the angle of flow constriction;
l	is the half-width of bubble core;
D_{tu}	is the internal tube diameter;
D_{ef}	is the effective diameter of vapor flow;
$r_b = \sqrt{\sigma/g(\rho' - \rho'')}/2$	is the equivalent bubble radius;
d_{eq}	is the equivalent bubble diameter;
σ	is the surface tension;
g	is the acceleration of gravity;
K	is the Kutateladze's stability criterion;
K_1	is the criterion for two-phase flow stability in siphon;
$L' = D_{tu}/\sqrt{\sigma/g(\rho' - \rho'')}$	is the linear dimension;
q	is the heat-flux density;
q_{max}	is the experimental maximum heat flux density;
$q_{max\,z}$	is the maximum heat-flux density given by Kutateladze – Zuber equation;
λ_d	is the wavelength of Taylor instability;
I	is the parameter;
N	is the Borishanskii number.

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