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THERMAL RESISTANCE OF THE VAPORIZATION ZONE FOR A HEAT TUBE WITH A THREADED CAPILLARY SYSTEM

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UDC 536.242

We present theoretical relationships with which to estimate the thermal resistance of the vaporization zone in a low-temperature heat tube with a capillary structure in the form of threading on the inside surface of the frame.

A threaded capillary structure (CS) allows us to achieve high heat-flux density in the vaporization zone, and it is simple from a technological standpoint. This type of CS has therefore gained widespread acceptance in arterial heat tubes (HT). A number of studies [1-4] has been devoted to the determination of thermal resistances in threaded CS. However, no consideration was given in these studies to the actual thread geometry nor to the influence of the heat flow on thermal resistance. In a number of cases, the assumptions based on the derivation of relationships for the calculation of thermal resistance failed to take into consideration contemporary concepts regarding meniscus vaporization. It is the goal of the present study to investigate the thermal resistance of the vaporization zone in a HT with a threaded CS, with consideration given to the above-indicated factors. The lateral cross sections of these variants of arterial HT can be seen in Fig. 1a, b, with the threading profile shown in Fig. 1c. The fluid at the edge of the meniscus can be divided into three characteristic regions [5]: the equilibrium film, a vaporization zone, and the meniscus itself. The adhesion forces in the equilibrium film, a consequence of the interaction between the molecules of the fluid and the wall, are so great that the fluid does not vaporize. In the vaporization film, whose thickness is greater than that of the equilibrium film, the adhesion forces diminish and vaporization becomes possible. The adhesion forces beneath the actual meniscus become negligibly small, and the flow of the fluid can be ascribed to the capillary pressure gradient generated by the change in the curvature of the meniscus. This change is slight in comparison to the change of curvature in the vaporization film, where it is altered by factors of ten. Let us determine that segment k–*a* of the actual meniscus and the corresponding angle $(\varphi_k - \varphi_a)$, i.e., see Fig. 1c, making the following assumptions: the flow is laminar, the radius of the meniscus within the limits of the angle $(\varphi_k - \varphi_a)$ changes only by 5%, and the change in the rate of flow and in the lateral cross section of the fluid flow between sections k–k_1 and a–o_2, owing to vaporization and the change in the radius of the meniscus, is small, while the density of the heat flow in the vaporization zone is constant. Under these assumptions, the pressure losses due to friction in the flow of the fluid along the segment x_{k_1}–x_{o2}:

$$\Delta p_{k-o_2} = B \int_{x_{k_1}}^{x_{o2}} \frac{dx}{y^3}, \tag{1}$$

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 60, No. 3, pp. 363-368, March, 1991. Original article submitted December 21, 1989.

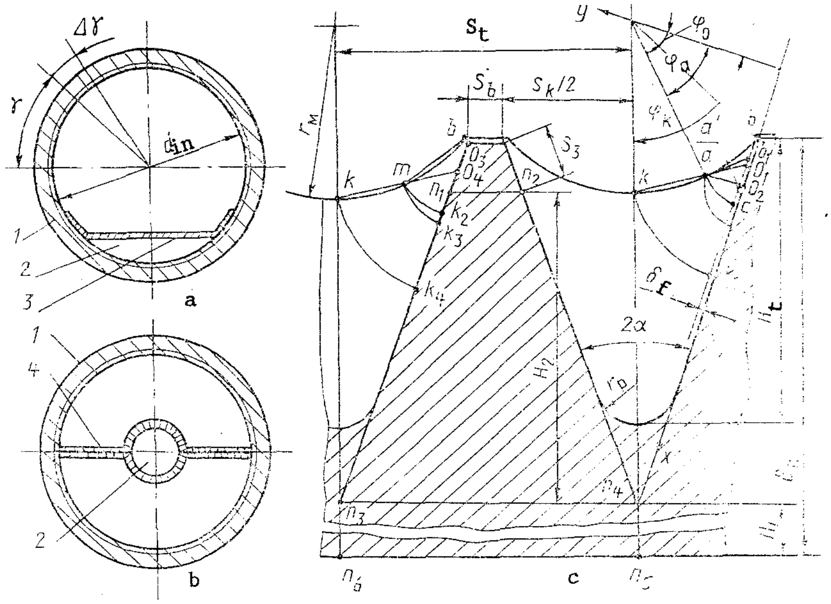

Fig. 1. Transverse sections of the heat-tube variance and the thread profile on the wall of the frame.

where

$$B = \frac{4 v_{\ell} Q S_{p}}{\pi d_{in} l_{su} L}; \tag{2}$$

$$y = r_{m} (\cos \varphi_0 - \cos \varphi_x) + \delta_{f}. \tag{3}$$

The capillary head offsetting the pressure losses is given by

$$\Delta p_{CAP} = 0{,}05 \frac{\sigma}{r_{m}}. \tag{4}$$

Equating the right-hand sides of (1) and (4) and carrying out the integration, we derive an expression from which we can determine the angle

$$0{,}05 \frac{\sigma r_{m}}{B} = \frac{2}{(1+\cos\varphi_0)^3 (1-\cos\varphi_0)} \left\{ \frac{\cos\varphi_0}{1+\cos\varphi_0} \left[\frac{\mathrm{tg}\frac{\varphi_a}{2}}{\left(\mathrm{tg}^2\frac{\varphi_a}{2} - b^2\right)^2} - \frac{\mathrm{tg}\frac{\varphi_k}{2}}{\left(\mathrm{tg}^2\frac{\varphi_k}{2} - b^2\right)^2} \right] + \frac{3(\cos\varphi_0 + 2)}{2(1-\cos\varphi_0)} \left[\frac{\mathrm{tg}\frac{\varphi_a}{2}}{\mathrm{tg}\frac{\varphi_a}{2} - b} - \frac{\mathrm{tg}\frac{\varphi_k}{2}}{\mathrm{tg}\frac{\varphi_k}{2} - b} \right] + \frac{4-\cos\varphi_0}{4b(1-\cos\varphi_0)} \left[\ln \frac{\mathrm{tg}\frac{\varphi_a}{2} + b}{\mathrm{tg}\frac{\varphi_a}{2} - b} - \ln \frac{\mathrm{tg}\frac{\varphi_k}{2} + b}{\mathrm{tg}\frac{\varphi_k}{2} - b} \right] \right\}, \tag{5}$$

where

$$b = \left(\frac{1-\cos\varphi_0}{1+\cos\varphi_0} \right)^{0,5}; \tag{6}$$

$$\varphi_k = \frac{\pi}{2} - \alpha; \quad \varphi_0 = \varphi_k - \arcsin \frac{r_{t} \cos\alpha + [H_{t} - r_{t}(1-\sin\alpha)]\,\mathrm{tg}\,\alpha}{r_{m}}. \tag{7}$$

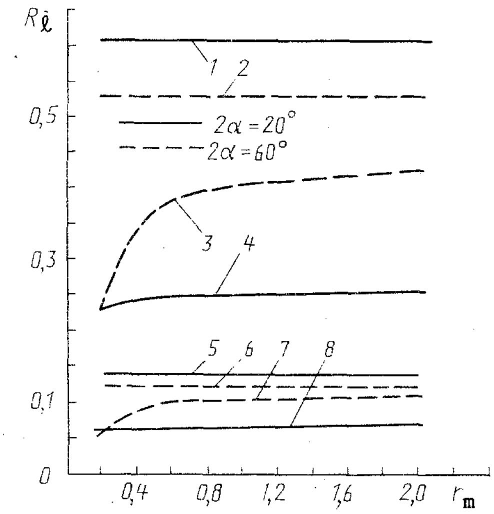

Fig. 2. Linear fluid resistance in one thread groove as a function of the meniscus radius: 1, 2) X22, R_{max}; 3, 4) X22, R_{min}; 5, 6) NH_3, R_{max}; 7, 8) NH_3, R_{min}·R_l, m·K/W; r_m, mm.

Formula (5) includes the meniscus radius r_m which corresponds to the given magnitude of the heat flow Q to the vaporization zone of the HT. Thus, in order to determine the angle φ_a, it is necessary that for the given HT that we first calculate the relationship $r_m = f(Q, \gamma)$. Thus a calculation can be accomplished by using the method presented in [6]. The calculations accomplished in accordance with this method for a large number of HT of various length and with various thread dimensions demonstrated that in the case of heat flows no greater than 85% of the maximum, the meniscus remains at the starting point of the threading groove (it does not penetrate into the groove), and that the relationship $r_m = f(\gamma)$ is nearly linear (the difference from the linear is no greater than 5%). This makes it possible to calculate the angle φ_a averaged over the length of the thread groove from the average magnitude for the radius r_m for the given heat flow.

In order to determine the thermal resistance to the transfer of heat through the fluid, we must know the configuration of the a—b_1 film vaporization segment which exhibits a pronounced change in curvature. The method for the calculation of this segment has been treated in [5]. However, in order to carry out the calculations by means of this method, we require experimental data on the fluid—metal pairs characteristic of low-temperature HT, and such data are presently not available. Therefore, instead of determining the exact magnitude of the thermal fluid resistance we propose estimating the boundaries within which these fall.

In order to determine the upper bound of the thermal resistance of the fluid, we will replace the actual vaporization surface k—b_1 with two straight-line segments: a—a' and k—a. The segment a—a' lies on the straight line drawn from the point a, perpendicularly to the wall of the threading groove. Its height

$$H_{a-a'} = r_m (\cos\varphi_0 - \cos\varphi_a). \tag{8}$$

The segment k—a lies on the straight line drawn from the point k through point a. In order to simplify the calculation formulas, we assumed that the streamline in the transfer of heat to the segment a—a' forms a circle with its center at the point o_2, while in the case of heat transfer to the segment k—a it forms a circle whose center lies at the point o_1. We have neglected the transfer of heat through the fluid for the segment situated below the arc k—k_1. When we take into consideration the above assumptions, the linear thermal resistance of the fluid (per unit length of threading groove) for a single threading groove is given by

$$R_{\ell\max} = 0{,}25\,\lambda_\ell^{-1}\left(\frac{2}{\pi}\ln\frac{y_a}{\delta_f} + \frac{1}{\beta}\ln\frac{l_{o_1k_1} + \dfrac{\delta_f}{\beta}}{l_{o_1c_1} + \dfrac{\delta_f}{\beta}}\right)^{-1}, \tag{9}$$

where

$$\beta = \frac{\pi}{2} - \frac{\varphi_k - \varphi_a}{2} + \alpha; \tag{10}$$

$$l_{o_1c_1} = y_a(\operatorname{ctg}\beta + 1); \tag{11}$$

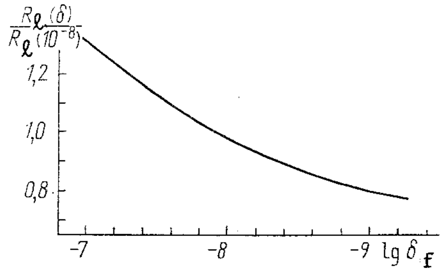

Fig. 3. Thermal fluid resistance as a function of the thickness of the unvaporized film.

$$l_{o_1k_1} = r_m\left[2\sin\frac{\varphi_k-\varphi_0}{2}+\frac{\cos\varphi_0-\cos\varphi_a}{\cos\left(\frac{\varphi_k-\varphi_a}{2}+\alpha\right)}\right]. \tag{12}$$

The lower bound of the thermal resistance to the transfer of heat through the fluid can be determined if we assume that the meniscus over the entire extent from the point k to the point b exhibits a constant diameter. For the sake of simplification, if we replace the arc k—b by the broken line kmb (km = mb), under assumptions analogous to those which were adopted in the determination of $R_{l\,max}$, we obtain

$$R_{\ell\,\mathrm{min}} = 0{,}25\,\lambda_\ell^{-1}\left(\frac{1}{\beta_1}\ln\frac{\delta_f+S_1\beta_1}{\delta_f}+\frac{1}{\beta_2}\ln\frac{\delta_f+S_1\beta_1+S_2\beta_2}{\delta_f+S_2\beta_2}\right)^{-1}, \tag{13}$$

where

$$\beta_1=\varphi_0+\frac{\varphi_k-\varphi_0}{4};\quad \beta_2=\pi-\varphi_0+\frac{3}{4}(\varphi_k-\varphi_0);$$

$$S_1=2r_m\sin\frac{\varphi_k-\varphi_0}{4};\quad S_2=S_1\frac{\sin\beta_1}{\sin\beta_2}. \tag{14}$$

For purposes of calculating $R_{l\,max}$ and $R_{l\,min}$ we need data on the thickness δ_f of the vaporizing film, and this thickness depends on the properties of the liquid and the surface which it wets, as well as on the presence on that surface of oxidizing films, contamination, etc. From the data cited in [7, 8] we might assume that δ_f ranges from 10^{-9} to $15 \cdot 10^{-8}$ m. For purposes of comparison, let us note that the thicknesses of the monomolecular liquid layers of low-temperature and cryogenic heat carriers (Freon, ammonia, nitrogen, oxygen, etc.) range from $2 \cdot 10^{-10}$ to $4.4 \cdot 10^{-10}$ m, while the height of the thread surface irregularities ranges from $0.5 \cdot 10^{8}$ to 10^{-8} m. The level of cleansing for the technical thread surfaces in the HT is lower than in the case of laboratory specimens which provided the data for [7, 8]. Contamination leads to impairment of wetting, which bears out the fact that the adhesion forces become weaker. It can therefore be assumed that δ_f in the threading does not exceed 10^{-8} m.

As an example, Fig. 2 shows results from calculation of R_{lmax} and R_{lmin} as functions of the meniscus radius. The calculations were carried out for threading having the following parameters: $H_t = 0.185$ mm, $r_t = 0.05$ mm, $S_t = 0.35$ mm. In the case of "Khladon-22" the thermophysical properties corresponded to a temperature of 213 K, while in the case of ammonia they corresponded to 293 K. For HT with the given dimensions the radius of the meniscus in the vaporization zone changes as a function of the heat-flow magnitude. Thus, the data shown in Fig. 2 indicate that $R_{l\,max}$ is independent of the heat flow, and noticeable changes in $R_{l\,min}$ are observed only for small r_m, corresponding to larger flows of heat.

Figure 3 shows the influence exerted by the thickness of the unvaporized film on the thermal resistance of the liquid. As we can see from the figure, when δ_f changes within the limits indicated in [7, 8], the thermal resistance of the liquid changes approximately by 20%. It should be noted that in this case no consideration was given to the effect of sorbed gases, metal oxides, the chemical-reaction products for oxides with the coolant, etc. This effect must be accounted for in special studies.

Calculation of the components in the thermal resistance of the liquid, accomplished with utilization of the relationships obtained in the derivation of formulas (9) and (13), demonstrated that no less than 75% of the heat transferred through the liquid is transmitted through that segment of its side surface S_3 adjacent to the edge of the thread protrusion, that side surface extended over 0.05 mm. When we take this into consideration, the linear thermal resistance to the transfer of heat through that segment of the frame which includes a single threading protrusion can be presented in simplified form as the sum of the resistances of the trapezoid $n_1n_2n_3n_4$ and the cylindrical segment $n_3n_4n_5n_6$ (see Fig. 1):

$$R_k = 0{,}5\,\lambda_k^{-1}\left(\frac{H_3}{S_t - l_{n_1n_2}}\ \ln\frac{S_tH_3}{l_{n_1n_2}H_3} + \frac{H_1}{S_t}\right), \tag{15}$$

where

$$l_{n_1n_2} = S_t - S_k + 2S_3\sin\alpha; \tag{16}$$

$$H_1 = \delta_k - \operatorname{ctg}\alpha\,\frac{S_k}{2}\ ;\ H_3 = \operatorname{ctg}\alpha\,\frac{S_k}{2} - S_3\cos\alpha. \tag{17}$$

The total thermal resistance of the vaporization zone can be calculated by means of the formula

$$R_{\text{sur}} = \frac{(R_\ell + R_k)\,S_t}{l_{\text{sur}}\pi d_{\text{in}}K}, \tag{18}$$

where K is a coefficient which takes into consideration that fraction of the perimeter of the lateral cross section to which the heat is transferred. For example, if the heat is directed to half of the perimeter, then K = 1.

Calculations of the thermal resistance of the vaporization zone in a heat tube with a housing-frame thickness of $\delta_h = 0.5$ mm made of 12Cr18Ni10Ti steel, M1 copper, and an AD1 aluminum alloy, with the same thread parameters for which the data presented in Fig. 2 were derived, demonstrated that:

a) for coolants with low coefficients of thermal conductivity (Freon, propylene, nitrogen, oxygen) the contribution of the liquid to the overall thermal resistance of the heat-transfer zone of the HT with a housing frame made out of 12Cr18Ni10Ti steel is not less than 50%; in the case of ammonia, this contribution is no less than 30%;

b) despite the fact that in the case of the AD1 alloy the thermal conductivity is higher than in the case of the 12Cr18Ni10Ti steel, by a factor of approximately 14, replacement of the steel housing frame by one made of the AD1 alloy leads to a reduction in the thermal resistance for the heat-transfer zone by no more than 70% for low-temperature coolants and by a factor of 2.5-3 in the case of ammonia; substituting copper for the AD1 alloy makes it possible to reduce the thermal resistance of the transfer zone in the case of HT with low-temperature coolant only by 1-3.5%.

NOTATION

S_t and H_t, threading pitch and height; 2α, threading groove profile angle; r_m, radius of meniscus; l_{sur}, length of vaporization zone; d_{in}, inside diameter of HT housing frame; Q, heat flow; ν_l, kinematic coefficient of viscosity; σ, coefficient of surface tension; δ_f, film thickness; H, height; β, γ, φ, angles (see Fig. 1); L, heat of vapor formation.

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