sion of practical importance has been obtained, allowing determination of flow conditions for both fresh and salt water at which internal icing of the line will be excluded.

NOTATION

P, T, V, S_w, pressure, temperature, velocity, and area of "live" flow section; Θ, salt concentration in flow; ρ_T, ρ_ℓ, density of ice and solution; C_ℓ, specific heat of the solution; λ_T, λ_ℓ, λ_{gr}, thermal conductivity coefficients of ice, solution, and ground; α, heat liberation coefficient from liquid to the tube wall; α^*, heat liberation coefficient from liquid to ground; ξ, hydraulic resistance coefficient; T_H, external temperature; L, heat of phase transsition; R_0, S_0, radius and area of tube cross section; G, thermal resistance of disturbed ground; δ_{in}, λ_{in} thickness and thermal conductivity coefficient of insulation.

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COMPREHENSIVE INVESTIGATION OF STARTUP REGIMES FOR A FROZEN HEAT PIPE

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UDC 536.248.2

The authors describe results of an experimental investigation of startup of an ammonia heat pipe made of aluminum with longitudinal channels and radiative heat rejection. They determine the influence of the startup dynamics of the location of the frozen heat transfer agent and the slope of the heat pipe to the horizontal. They propose a simplified method of determining the startup characteristics when the main mass of heat transfer agent is frozen in the heater zone.

The startup of low-temperature heat pipes is characterized by a number of special features which give rise to certain problems. These include the possible startup of a heat transfer agent from the frozen state with a given means of heat supply and removal, and the need to account for possible operation of the heat pipe at the hydrodynamic power limit. Startup of a heat transfer agent from the frozen state is a sequence of complex physical processes involving phase transitions of substances and unsteady effects.

The dynamic characteristics of low-temperature heat pipes have been investigated in a number of papers [1-3], where, in the main, startup from a state with a liquid heat transfer agent was investigated. One should note the theoretical paper [4], which formulated the problem of a frozen heat pipe with a homogeneous wick, and also papers [5, 6] where theoretical and experimental data for startup of a water heat pipe were examined.

The aim of the present paper is a comprehensive investigation of startup conditions of an ammonia heat pipe when heat is supplied to the evaporator by convection to the liquid and heat is removed by radiation.

The heat pipe had 45 longitudinal open grooves of depth $0.82 \cdot 10^{-3}$ m and width $0.5 \cdot 10^{-3}$ m. The material of the body was aluminum. The geometric dimensions of the pipe were: total length 1.72 m, length of the heater zone 0.2 m, length of the cooler zone 1.48 m,

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outer diameter 0.018 m, inner diameter 0.016 m, volume of charge 0.036 kg.

The experimental facility simulated the actual conditions of operation of the heat pipe as an element of a radiation panel [7]. The heat flux was supplied with boundary conditions of the third kind by means of a liquid heat exchanger. The condenser was equipped with a radiator of width 0.12 m. Heat rejection was accomplished by radiation to a nitrogen screen ($\varepsilon = 0.2$). The maximum absolute error of temperature measurement was ±1.5 K. An adiabatic state relative to the surrounding medium was achieved by locating the heat pipe in a vacuum volume. During the experiments we recorded the temperature field along the shell and the radiator. The sensors were welded copper-constantan thermocouples.

The experimental technique included preliminary defrosting of the object by two methods and subsequent startup. The first method achieved defrosting of the main mass of heat transfer agent in the condenser zone, and the second method did the same in the evaporator zone.

Startup of a long heat pipe under conditions of considerable heat rejection was accomplished by supplying heat only to the evaporator with no supplementary heating of the condenser. The temperature of the working liquid arriving at the heat exchanger was held constant at 313 K.

We studied the influence of location of the frozen heat transfer agent on the startup dynamics. In the case of startup of a heat pipe from the state with the heat transfer agent frozen in the condenser zone, most of the pipe could not be heated to the triple point over a considerable time period ($\tau = 18{,}900$ sec). This means that there is not enough heat transfer by axial heat conduction of the body and that there is no stable evaporation-condensation cycle.

For comparison we examined startup under analogous conditions from a state with the heat transfer agent frozen in the evaporator. Startup was accomplished with slope angles of 0; +2.5; +5; +10°. As an example Fig. 1 shows characteristic temperature distributions along the body for $\phi = +10°$. The dynamics of the temperature field variations with time show that after heat is supplied and the ice melts in the evaporator the pipe subsequently warms up along its length. The startup occurs in a front manner as the channel fills with condensible vapors. The temperature front, moving along the pipe, switches on different sections of the condenser to the heat and mass transfer process. Even in the horizontal location the pipe came up to steady conditions in 1 hour.

One can obtain a clear idea of the influence of pipe orientation relative to the action of gravity forces on startup dynamics from the time at which the temperature front τ_{tp} passes the section where the thermocouples are welded, for different positions of the pipe (Fig. 2). The maximum absolute error of determining this time was ±1 sec. With increased slope the heating time up to the triple point, and therefore the startup duration was reduced. This is due primarily to the fact that the limiting power of this configuration, which determines the heating of the cold zone (see Table 1), depends on the slope angle. At small slope angles the process of heat and mass transfer is accomplished in a stable manner not only by the capillary forces, but also by gravitational forces. The approximate dependence of startup duration on pipe orientation was verified. The best agreement with the experimental data is obtained with a fourth degree polynomial:

$$\tau_s = 3640 - 1{,}841\varphi + 321\varphi^2 - 38{,}1\varphi^3 + 1{,}52\varphi^4.$$

Thus, as the orientation varies there is a sharp change in the dynamic characteristics of the heat pipe and of the startup time from the state with the heat transfer agent frozen in the evaporator. Also, the position of the frozen heat transfer agent clearly affects the startup time appreciably. When it is concentrated in the heating zone one obtains reliable and nonprotracted startup of a long heat pipe when heat is supplied only to the evaporator.

We attempted to develop a mathematical model of a grooved heat pipe with heat transfer agent frozen in the heater zone. The analysis was based on the following assumptions: the parameters are lumped, the heat transfer agent is uniformly frozen in the evaporator, sublimation and desublimation can be neglected, and the condenser is heated by condensation of vapors at a rate determined by the limiting heat transfer for the given structure.

We arbitrarily divide the startup process into four periods. We separately solved the problems of determining the heating duration of the evaporator and the condenser to T_{tp},

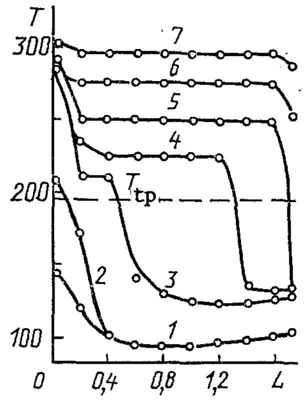

Fig. 1

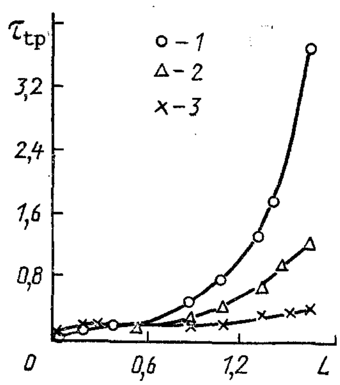

Fig. 2

Fig. 1. Temperature distribution along the body during startup of a heat pipe with heat transfer agent frozen in the evaporator, for various times: 1) $\tau = 0$; 2) 96; 3) 192; 4) 384; 5) 528; 6) 672; 7) 1488 sec; $T_\ell = 313$ K, $\phi = 10°$. T, K; L, m.

Fig. 2. Dependence of body heating time to the triple point on the heat pipe slope angle: 1) $\phi = 0$; 2) 2.5; 3) 10°; $T_\ell = 313$ K. $\tau_{tp} \cdot 10^{-3}$, sec.

melting of the heat transfer agent in the heater zone, and heating of the structure to steady temperatures [8, 9].

From an analytical solution of the heat balance equation, allowing for loss of heat along the body for the evaporator zone

$$\alpha \pi d_e L_e (T_\ell - T_e) = \frac{dT_e}{d\tau} C_{HP} + \frac{T_e - T_C}{R^*}; \tag{1}$$

$$\tau = 0,\ T_e = T_{0,e} \text{ and } \tau = \tau_1,\ T_e = T_{tp}$$

we found the variation of the evaporator temperature during the first period

$$T_e = \left(T_{0,e} - \frac{A}{B}\right) \exp\left(-\frac{\tau B}{R^* C_{HP}}\right) + \frac{A}{B} \tag{2}$$

and the heating time to T_{tp}

$$\tau_1 = \frac{R^* C_{HP}}{B} \ln \frac{T_{0,e} B - A}{T_{tp} B - A}, \tag{3}$$

where

$$C_{HP} = C_e^f + \eta C_c;\ C_i = \sum_{K=1}^{m} V_{ic} \rho_{ic} C_{ic},\ i = e, cK;$$

$$A = \alpha \pi d_e L_e R^* T_\ell + T_C;\ B = \alpha \pi d_e L_e R^* + 1.$$

The coefficient η takes into account the temperature difference between the condensation zone and the evaporator.

The process of melting of the heat transfer agent in the evaporator along a radius is described by the system of differential equations

$$\alpha \pi d_e L_e (T_\ell - T_e) = C_{HP} \frac{dT_e}{d\tau};\ \lambda_{ef}^{\ell} = \frac{T_e - T_{tp}}{x_{me}} = r^* \rho_\ell \varepsilon' \frac{dx_{me}}{d\tau}; \tag{4}$$

$$\tau = 0,\ x_{me} = 0,\ T_e = T_{tp};\ \tau = \tau_2,\ x_{me} = h + \delta_f\ T_e = T_{e,2}.$$

We estimated the melting time by solving the transcendental equation obtained by solving system (4):

$$\exp\left(-\frac{\alpha\pi d_e L_e \tau_2}{C_{HP}}\right) = \frac{\alpha\pi d_e L_e \tau_2}{C_{HP}} + 1 + \frac{(h+\delta_f)^2 r^* \rho_\ell \varepsilon' \alpha\pi d_e L_e}{2\lambda_{ef}(T_\ell - T_{tp}) C_{HP}}.$$

The evaporator body tèmperature at the end of melting is

$$T_{e,2} = T_\ell + (T_{tp} - T_\ell)\exp\left(-\frac{\alpha\pi d_e L_e \tau_2}{C_{HP}}\right).$$

After the heat supply zone thaws out we have the processes of evaporation and subsequent condensation of vapor on the cold section of the pipe. Vapor condensation leads to filling of the grooves and simultaneously they are heated.

During warmup of the condenser at each time instant the condition must hold that the power supplied does not exceed the heat transfer limit. As a first approximation we consider that the two values are equal, i.e., the heat balance equation for the third period was solved assuming $Q_3 = Q_{max}$:

$$Q_{max} = C_c \frac{dT_c}{d\tau} + \sigma\varepsilon A_{rad}(T_c^4 - T_c^4),\ \ \tau = 0,\ T_c = T_{c,0};\ \ \tau = \tau_3,\ T_c = T_{tp}. \tag{5}$$

The heat transfer limit for a given heat pipe is due to the capillary power constraints, and also to vapor-liquid interaction. The value of Q_{max} was computed by the method of [10], developed for a pipe with open channels:

$$Q_{max} = 16N \frac{r}{\nu_\ell} \frac{W^3 D^3}{(D+W)^2} \frac{\rho_\ell g \sin\varphi + \dfrac{\sigma\cos\theta_0}{WL_0} - \left|\dfrac{dP_v}{dz}\right|_a}{(f\,\mathrm{Re})_\ell}, \tag{6}$$

$$L_0 = L_a + (L_e + L_c)(1 - \sin\varphi),\ \ 90° > \varphi \geqslant 0°,$$

whence the heating time of the condensed zone to T_{tp} is

$$\tau_s = \frac{C_c}{2T_i^3\sigma\varepsilon A_{rad}}\left\{\mathrm{arctg}\frac{T_{tp}}{T_i} - \mathrm{arctg}\frac{T_{c,0}}{T_i} + \frac{1}{2}\ln\left|\frac{(T_i + T_{tp})(T_i - T_{c,0})}{(T_i - T_{tp})(T_c + T_{c,0})}\right|\right\},\ T_i^4 = \frac{Q_{max} + \sigma\varepsilon A_{rad} T_c^4}{\varepsilon\sigma A_{rad}}. \tag{7}$$

Analogously to what was done for the third period we also found the time for the condenser temperature to rise to a steady value. We considered the 80% and 90% approximation to this value. We postulated that during the fourth period the power supplied to the cold zone is close to the steady value, i.e., $Q_4 = Q_{st}$. Following the melting the heat transmission capability of the heat pipe sets in, i.e., its effective heat conduction becomes quite high (at the level 10^4 W/(m·K)). Therefore for operation under steady conditions the heat pipe is considered as practically isothermal, at a temperature which one finds from the balance between the power supplied and removed:

TABLE 1. Dependence of Q_{max} on the Slope Angle of a Heat Pipe of Length 1.72 m with T_e = 313 K.

φ, deg	0	1	2,5	5	10	15
Q_{max}, W	42,7	53	71,11	91,6	146,2	200,95

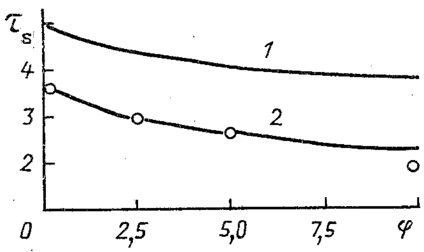

Fig. 3. Dependence of the heat pipe startup time, from a state with the heat transfer agent frozen in the evaporator, on the slope angle: 1) theory, 90% approximation to the steady-state value; 2) theory, 80% approximation to the steady state value; the points are experimental data. $\tau_s \cdot 10^{-3}$, sec; φ is in deg.

$$Q_{st} = \alpha \pi d_e L_e (T_\ell - T_{st}) = \sigma\varepsilon A_{rad} (T_{st}^4 - T_C^4).$$

The heat pipe startup characteristics in this period were described by the equations

$$Q_{st} = C_c \frac{dT_c}{d\tau} + \sigma\varepsilon A_{rad} (T_c^4 - T_C^4), \quad \tau = 0, \; T_c = T_{tp}; \; \tau = \tau_4, \; T_c = T, \tag{9}$$

$$\tau_4 = \frac{C_c}{2T_{st}^3 \varepsilon\sigma A_{rad}} \left\{ \operatorname{arctg} \frac{T}{T_{st}} - \operatorname{arctg} \frac{T_{tp}}{T_{st}} + \right. \left. + \frac{1}{2} \ln \left| \frac{(T_{st} + T)(T_{st} - T_{tp})}{(T_{st} - T)(T_{st} + T_{tp})} \right| \right\} \tag{10}$$

For the heat pipe investigated (ϕ = 10°) the time to reach the steady regime from a state with the heat transfer agent frozen predominantly in the evaporator was (at the 80% approximation)

$$\tau = \tau_1 + \tau_2 + \tau_3 + \tau_4 = 2{,}3 + 37{,}9 + 561{,}2 + 1729{,}7 = 2327{,}2 \text{ (sec)}.$$

Figure 3 shows the theoretical computations and the experimental data on startup for various orientations of a grooved heat pipe in a gravity force field. We obtained favorable agreement in spite of a number of simplifications in comparing with startup of the actual structure.

Thus, the method suggested can be used to predict the startup characteristics for small slope angles of a heat pipe with a heat transfer agent frozen in the evaporator. With this method one can estimate the maximum possible time to thaw out a low-temperature heat pipe for which hydrodynamic power constraints play the main role in the startup.

NOTATION

A, B) coefficients; A_{rad}) area of radiating surface; C) heat capacity; d) diameter; h) height of grooves; L) length; Q_{max}) maximum thermal power transmitted by the heat pipe; R*) thermal resistance in heat transmission in the axial direction; r*) heat of melting; T) temperature; T_C) temperature of the cooler (the cryogenic circuit); T_i) intermediate temperature; V) volume; x_{me}) position of the liquid-ice interface (at the heat pipe wall x_{me} = 0); α) heat transfer coefficient; δ) thickness of the heat transfer agent, frozen in the evaporator above the grooves; ε') porosity; ε) emissivity of the radiator surface; η) coefficient; λ) thermal conductivity; ρ) density; σ) Stefan-Boltzmann constant; τ) time; ϕ) slope angle; N) number of grooves; r) specific heat of vaporization; ν_ℓ) kinematic viscosity of the liquid; W) half-width of a groove; D) groove depth; Re) Reynolds number; θ) wetting angle; g) acceleration due to gravity; z) axial coordinate. Subscripts: e) evaporator; c) condenser; a) adiabatic zone; ℓ) liquid; f) frozen state; st) steady state; HP) heat pipe; tp) triple point; ef) effective; v) vapor; 0) initial state; 1, 2, 3, 4) period number.

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