START-UP DYNAMICS OF AN ARTERIAL HEAT PIPE FROM THE FROZEN OR CHILLED STATE

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The authors present results of an experimental investigation of start-up of an arterial heat pipe from the frozen or chilled state. The physical processes then occurring are described.

The development of systems to provide thermal conditions has generated interest in making a detailed investigation of unsteady processes of heat transfer and hydrodynamics in heat pipes. During operation of heat pipes one can vary the heat load and the heat removal conditions. A sharp fall of the temperature of the surrounding medium can lead to transition of the working liquid to the solid phase, and lower the heat-transfer efficiency. For practical use of devices based on heat pipes, it is important to define criteria for reliability of start-up and its duration for any initial state of the heat-transfer agent, and the behavior of these devices for variables levels of the external factors. The study of unsteady regimes, development of a mathematical model, and a technique for calculating the start-up are urgent problems whose solution will promote wider application of heat pipes to industry.

Previous investigations of the problem of start-up of heat pipes from a state with frozen heat-transfer agents have taken several directions. The unsteady regimes of high-temperature heat pipes have been investigated in most detail. It was shown in [1, 2] that the start-up characteristics of heat pipes depend on the structure, the heat sink temperature, and the limiting thermal resistance in the condenser. In the experiments of [3] a heat pipe with channels jacketed by a mesh was successfully brought up to steady operating conditions. The gradual movement of the temperature front to the end of the condenser was recorded. However, the start-up of a heat pipe with open channels proved to be a complex process, since it was difficult to return the liquid to the evaporator in a number of cases.

The importance of calculating the heat-transfer limit due to the vapor stream reaching the speed of sound at the evaporator exit was confirmed in tests with liquid metal heat pipes. Structures were investigated with a composite wick made of perforated screen [4] and with rectangular grooves [5], to which heat was supplied by an electric heater and from which it was removed by radiation. For rapid start-up they recommended preliminary warm-up of the body over its whole length, to promote development of continuous flow of vapor.

References [4, 6] examined the question of the role of noncondensible gas, especially for heat pipes with a long condenser ($L_c/L_{ev} > 5$). It was established that the presence of noncondensible gas makes start-up conditions easier and reduces the danger of drying out the wick due to freezing of the heat-transfer agent back to the condenser zone.

There are very few papers dealing with investigation of start-up characteristics of low-temperature heat pipes. Experiments have been conducted in heat pipes made of copper, and stainless steel, with water as the heat-transfer agent. Heat was supplied to the evaporation mainly by an electric heater. Here, the condenser was cooled by free convection of air or by circulating a liquid. It was established in [7, 8] that the time to bring the heat pipe up to steady conditions depends on the method of supplying heat and the heating rate, and also on the heat removal law.

Interesting results from the practical viewpoint were obtained in investigations of heat pipes with metal-fiber and mesh structures, designed for cooling of electronic equipment [9, 10]. From analysis of the experimental data it was concluded in principle, that traces of noncondensible gas considerably affect the dynamic characteristics. This is due to the low

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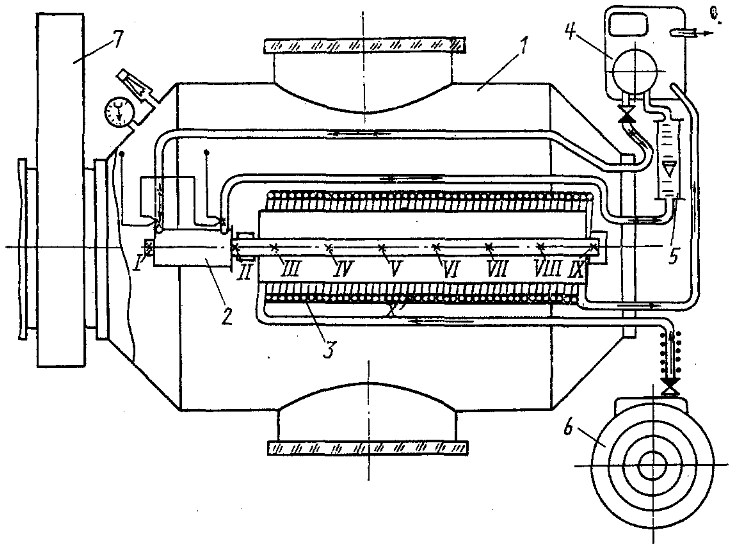

Fig. 1. Schematic of the experimental equipment: 1) chamber; 2) test heat pipe; 3) cryogenic circuit; 4) thermostat; 5) rotameter; 6) nitrogen tank; 7) gate valve; I-X) location of thermocouples; arrows) circulation of liquid nitrogen; arrows with crosses) circulation of ethanol.

pressure of the heat-transfer agent vapor during heating of the pipe up to the triple point temperature. The authors demonstrated that one can start up a low-temperature heat pipe without violating the temperature limits in the evaporator zone by reducing the cooling intensity and introducing a certain amount of noncondensible gas.

In contrast with these named references, reference [11] studied start-up from the state where all the working liquid was evaporated in the hot zone, and the vapor was condensed and frozen in the cold zone. The heat pipe had a staggered braided wick of quartz fiber. Heat was transmitted to the frozen liquid only by the conduction of the body and the wick. The start-up time was cut by one half when an auxiliary heat pipe warming up the condenser was used.

Thus, the investigations published so far in the low-temperature area are limited in nature. Their results cannot be extended to the conditions for start-up of heat pipes of complex construction with different boundary conditions in the evaporator and condenser. In practice in heat-exchange equipment one often meets arterial heat pipes with radiative cooling, and with power supplied by a liquid heat-transfer agent. Therefore, there is a need to conduct investigations in this direction.

A cylindrical heat pipe with arteries was examined in [12]. It had a grooved evaporator, and a condenser with an imbedded wick made of steel mesh. The geometric dimensions of the heat pipe, in m, were: total length 0.95, evaporator length 0.165, condenser length 0.76. The body material was aluminum alloy. The working liquid was chosen to be ammonia. It does not cause corrosion of the aluminum alloys, and can be used over quite a wide temperature range: from 405.6 (critical point) to 195.4°K (triple point). The high saturation vapor pressure of ammonia (10.6 atm) at room temperature allows the heat pipes to be kept for a long time, eliminating ingress of noncondensible gas. To study the problem of bringing an arterial heat pipe up to steady operating conditions, we investigated start-up from frozen (i.e., below the triple point) and chilled (i.e., around the triple point) conditions.

<u>Experimental Equipment and Experimental Technique</u>. An experimental facility was developed to take account of the special features of the problem. In contrast with [7, 10] the heat flux was supplied to the evaporator with boundary conditions of the third kind, and the heat was removed by radiation, we ensured that the heat pipe was adiabatic with respect to the surrounding medium, and we examined the start-up at a different level of heat removal at the initial time.

The experimental facility (Fig. 1) included a vacuum chamber 1, the heat pipe unit, systems for supplying and removing heat, the control and measurement apparatus, and a set of

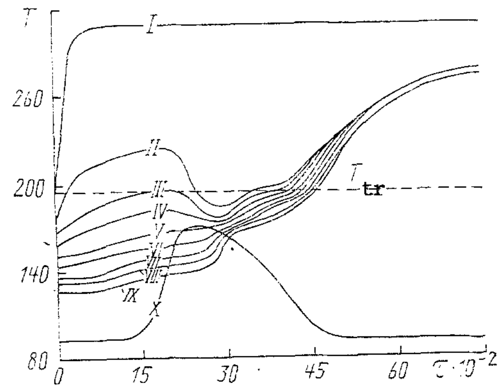

Fig. 2. Dynamics of temperature variation of the heat pipe surface when switched on from the frozen state: I — X, thermocouple number; T, °K; $\tau\cdot 10^2$, sec.

elements to provide the required pressure in the chamber. The arterial heat pipe 2 was located in the chamber. The heat flux was supplied to the heat pipe with the aid of a type U-10 thermostat 4. The heat-transfer agent (ethanol) was supplied to the liquid heat exchanger via hoses and vacuum lead-ins. To increase the heat removal the condenser section was equipped with a radiator. Heat was removed by radiation to the cryogenic circuit 3 through which nitrogen was circulated. The TRZhK-4M nitrogen tank 6 completed the heat-removal system. If need be nitrogen was also used to cool the heat-transfer agent in the thermostat. The adiabatic conditions during the tests were achieved by locating the heat pipe in an evacuated volume, the vacuum low-temperature chamber. The chamber was switched to a vacuum collector, which served as a pumping system with mechanical and diffusion pumps. The working vacuum was 10^{-2} mm Hg. In this way the influence of convection was negligible. The radiation from the chamber walls acted only on the evaporator, since the condenser was shielded by a radiation screen of aluminum foil. In order to reduce the influence of the gravity field, the heat pipe was located horizontally on thermally insulated supports of a clamp.

During the tests we recorded the temperature field along the heat pipe body. The sensors were Chromel — Alumel thermocouples glued to the outer surface. The thermocouple information was recorded on type ÉPP-9 chart recorded potentiometer, so that it could be analyzed in real time and could actively influence the investigations. The heat flux supplied to the evaporator was determined from the change of enthalpy of the heat-transfer agent in the time for it to pass through the heat exchanger. The flow rate of the heat-transfer agent was measured with the type RS-7 rotameter 5. The temperature of the cryogenic circuit and of the heat-transfer agent were held constant or varied, depending on the experimental conditions.

The experimental technique provides for preliminary freezing (or chilling) of the heat pipe. To do this the working chamber was evacuated, the pumping system was switched on, and the pressure was brought down to 10^{-2} mm Hg. The pressure was regulated with pointer-type and ionization thermocouple vacuum gauges. Then we began to cool the cryogenic circuit with nitrogen issuing from the tank. The experiments showed that the process of freezing the heat pipe by radiative heat transfer alone was very lengthy (4-5 h). To shorten the experiment and economize on liquid nitrogen, after chilling the circuit we isolated the chamber from the vacuum collector, and supplied a certain amount of helium to the chamber (up to a pressure of 0.1 atm). Due to convection the process of freezing the heat pipe was accelerated, and took approximately 1 h. By reconnecting to the pumping system the helium was then removed, and the vacuum in the chamber was brought down to 10^{-2} mm Hg. Then the heat pipe start-up process was investigated. We began to pump ethanol through its heat exchanger, and held its temperature constant with the thermostat.

To elucidate the question of the influence of the state of the working liquid on the start-up characteristics of the heat pipe we conducted two series of experiments. We examined the start-up from the state determined by an initial heat pipe temperature lying in the range: a) T_0 = 140-180°K, the frozen state; and b) T_0 = 190-203°K, the chilled state.

Start-Up from the Frozen State. The heat pipe was frozen down to a temperature of 140°K. The cryogenic circuit temperature was 103°K, and the temperature of the ethanol in the liquid heat exchanger was 283 or 313°K. Heating of the pipe was regulated from the thermocouple readings. Initially a large temperature gradient between the evaporator and

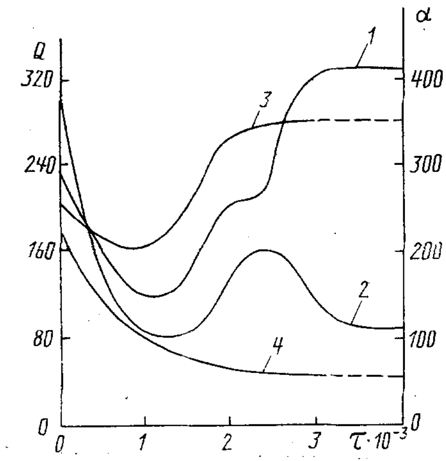

Fig. 3

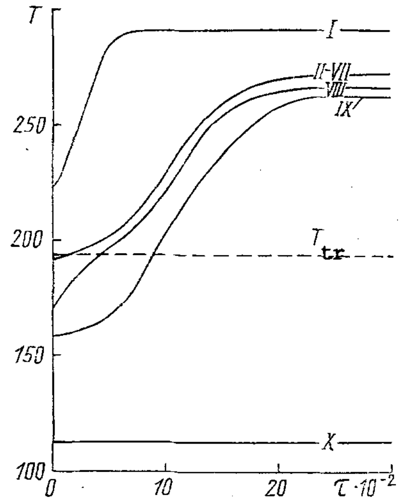

Fig. 4

Fig. 3. The effective heat removal coefficient α, W/(m²·°K), (1, 3) and the heat flux Q, W, (2, 4) as a function of time, τ, sec: 1, 2) start-up from the frozen state; 3, 4) from the chilled state.

Fig. 4. Dynamics of the temperature variation of the heat pipe surface during start-up from the frozen state: I-X) thermocouple number.

the condenser (80-85°K) was observed. The heat pipe warmed up slowly, but did not start up. To create the necessary conditions for start-up and to reduce the heat leakage from the pipe to a minimum, the cryogenic circuit temperature was raised from the time $\tau = 1200$ sec onwards. When it reached a value on the order of 173°K the temperature field of the heat pipe changed qualitatively and quantitatively. There was a sudden fall of the temperature at the place where the evaporator and condenser zones join, and the temperature at the end of the condenser increased, i.e., the temperature gradient along the pipe decreased sharply (40-45°K). The heat transmitting function of the pipe was restored. Subsequently, the pipe operation was not inhibited, in spite of the cryogenic circuit temperature being lowered to the original value. The temperature drop between the evaporator and the condenser was reduced to 10°K.

Figure 2 shows the start-up characteristics of the frozen heat pipe. It can be seen from the dynamics of its variation that the start-up process can be divided conventionally into three periods. Within the time interval $\tau < 1500$ sec (the first period) the evaporator receives preliminary heating. The result is melting of the frozen liquid located in the evaporator and also in the periphery of the wick. Leakage of heat due to radiation is prevented by heating of the condenser. The intense evaporation is accompanied by heating of the vapor in this zone. An insignificant amount of noncondensible gas is displaced to the end of the pipe, and does not show any influence on the process. Drying out of the evaporator is the reason for breakdown of the evaporation–condensation cycle. The pipe then warmed up like a hollow rod.

In the second period of start-up ($3500 > \tau > 1500$ sec) the formation of the temperature field is determined mainly by the reduced intensity of radiation and the axial heat conduction of the body. This creates favorable conditions for increased heat flux to the arteries and thawing of the ice in the part of the pipe adjacent to the heating zone. The working liquid at temperature T_{tr} reaches the mesh adjacent to the heated wall and evaporates strongly. For this reason the body is cooled (second and third thermocouples). As the solid ammonia phase melts in the pipe the evaporation–condensation cycle is restored, and the axial temperature gradient is reduced.

A special feature of the third period is heating of the entire pipe above T_{tr}. The heating process occurs simultaneously with redistribution of the heat-transfer agent mass between the zones. Warm-up of the condenser causes an increase of pressure in the channel, and therefore, reduces the overheat of the wall above the saturation temperature of the working liquid, leading to filling of the grooves and arteries of the evaporator. Closed circulation of liquid is seen over the entire length. The heat pipe gradually settles to a steady operating regime.

From analysis of the experimental data we have been able to establish the variation with time of the power supplied to the evaporator, and the effective heat-removal coefficient in the liquid heat exchanger (Fig. 3). In the unsteady regime these magnitudes depend not only on the flow rate, and the thermophysical properties of the working liquid, but also on the rate of change of the boundary conditions, and the temperature of the evaporator wall. The nature of the variation of the heat flux and the heat-removal coefficient with time was determined by a number of physical processes occurring during start-up of the heat pipe. For example, the heat flux and the heat-removal coefficient in the first period of start-up were reduced simultaneously because of unsteady warming of the body and drying out of the evaporator. The beginning of warm-up of the condenser and of the evaporation–condensation cycle caused these to increase in the second period. With the onset of the third period we observed a secondary drop of heat flux, that coincided in time with the end of the melting process and an increased rate of heating of the body. This quantity stabilized in a balance between the heat fluxes supplied and removed. It should be noted that at the secondary drop of heat flux there is a bend in the curve showing the heat-removal coefficient as a function of time. The heat-removal coefficient tends to a steady value which is determined by the possible heat rejection in the condenser and the geometry of the heat exchanger.

Start-Up from the Chilled State. The heat pipe was cooled to a temperature close to the ammonia triple point. Here the surface temperature of the adiabatic and the condensation zones were from 185 to 193°K, and the end of the evaporator was 210°K. The experiments were carried out at heat-transfer agent temperature in the liquid heat exchanger of 253, 273, and 294°K. The cryogenic circuit temperature was 123°K, i.e., the heat rejection was quite strong. Figure 4 shows typical start-up characteristics of the chilled heat pipe, obtained for $T_\ell = 273$°K. The readings of thermocouples II-VII practically coincide. The delay of the thermocouples VIII and IX is due to the presence of traces of noncondensible gas, which influences the warm-up at the beginning. Start-up was observed without stable breakdown of circulation of the working liquid. Probably because of the low effective heat conduction of the mesh, the ammonia was chilled only in the peripheral wick, and was in the liquid phase in the arteries. After the heat was brought on the evaporator and the section of condenser adjacent to it warmed up instantly. The vapor–gas front moved into the condensation zone. The intense evaporation in the channels led to their local short-duration drying out. The vapor condensed in the peripheral wick, heating it. From time $\tau = 750$ sec there was almost isothermal heating of the condenser, with subsequent melting of the heat-transfer agent in the mesh. A part of the thawed heat-transfer agent was returned through the arteries to the grooves. As early as $\tau = 2500$ sec the temperature field of the heat pipe stabilized. However, the supply of power was continued up to time $\tau = 3000$ sec. The laws of variation of the heat flux and the effective heat removal coefficient during the start-up process are analogous to those obtained for the frozen pipe. The nature of the dependences is different, as can be seen from Fig. 3. The chilled heat pipe goes quickly to a steady regime of operation, and therefore its characteristic $Q = f(\tau)$ decreased abruptly. In the initial start-up period, when heat capacity played a very large part, the heat-removal coefficient decreased due to the sharp fall of power expended in heating. After the evaporation–condensation cycle came into operation the curve of heat flux variation took on a gentle slope. Heating of the evaporator wall relative to the reference decreased, and therefore the heat-removal coefficient increased to its constant value.

It should be noted that the chilled heat pipe was started up much faster than for the case of the completely frozen heat-transfer agent. This seemed to stem from the absence of processes heating the heat-transfer agent at the end of the heat pipe and of drying out of the evaporator. In addition, the time to heat the object to the triple point temperature of the working liquid was substantially reduced.

Thus, the tests conducted have shown that the initial state of the heat-transfer agent considerably influences the dynamics of start-up of the arterial heat pipe. From the chilled state (temperature of the heat-transfer agent in the arteries close to the triple point) the start-up proceeded independently of the intensity of heat rejection. From the frozen state (temperature of the heat-transfer agent in the arteries below the triple point) the start-up proceeded normally if the intensity of heat rejection at the initial time was close to zero.

NOTATION

L_{ev}, length of the evaporator, m; L_c, length of the condenser, m; Q, heat flux, W; T_{tr}, triple point temperature, °K; α, heat-removal coefficient, W/(m²·°K); τ, time, sec.

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THERMAL STATE OF A POROUS PLATE UNDER COOLANT FILTRATION CONDITIONS

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The steady-state temperature field is considered in a finite porous plate during filtration of a coolant.

One of the important problems in the design of porous cooling systems is study of the temperature fields within porous bodies in the presence of filtration processes. In particular, [1, 2] were dedicated to this problem. Because of their complexity, problems of this type are solved in two stages: initially the dynamic problem is solved, i.e., the pressure (velocity) field in the porous body is found, after which the temperature field is constructed from the solution so obtained. Use of this approach is based on the assumption of the dominant effect of the velocity field on the temperature field in the porous body with only a weak effect in the opposite direction.

The goal of the present study is to obtain an analytical solution for the temperature field in a two-dimensional porous body when an incompressible liquid is used as the coolant.

We will consider the thermal state of a finite porous plate ABCDEK (Fig. 1a [3]), to which coolant is supplied through the face AC under a pressure P_1. A pressure P_2 exists on the surfaces AK and CD. In addition, we will assume the face KD to be impermeable and thermally nonconductive. Moreover, we assume that the temperature at the coolant input to the plate is equal to T_1, with temperature at the exit T_2, where T_2 is a known function of the coordinates.

The flow within the porous medium obeys the law

$$\frac{\alpha}{\mu(T)} \operatorname{grad} P = -\frac{f(v)}{v} \mathbf{v}, \qquad (1)$$

with consideration of which, heat and mass transport with a powerlike resistance law can be described by the equations

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