TESTS ON A PASSIVE-CONTROL HEAT PIPE

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Measurements have been made on the thermal stabilization provided by a gravitational thermal pipe containing a controlling piston.

Controlled heat pipes are widely used as thermostats and regulators. There are many types of these control pipes, which differ in control technique and design [1-4]. An effective control method is to vary the thermal resistance in the condensation zone by means of the area. There are several ways of doing this, e.g., by displacing the boundary between the vapor and the uncondensed gas in a gas-regulated tube [5] or by blocking part of the condenser with excess liquid [6], or else by blocking the condenser with a magnetizable ball controlled by an external magnetic field [7].

We have examined the stabilization provided by a passive pipe whose vapor channel contains a piston leaving a capillary gap from the wall. The control parameter is the temperature in the adiabatic zone, and the piston is the control element. The control is provided by varying the transmitted thermal power. The piston alters the condensation area, and thus the resistance. The amount of heat radiated by the pipe is

$$Q = kA(T - T_0)/\delta.$$

In the steady state, the radiated heat equals the input. If Q = const, the vapor temperature is related to the radiating area:

$$T = T_0 + \frac{Q\delta}{k}\frac{1}{A},$$

where $A = \pi DL$, so

$$T = T_0 + \frac{Q\delta}{\pi Dk}\frac{1}{L}.$$

This shows that one can vary the condenser length L to control the vapor temperature and thus the pressure.

In the steady state, the vapor pressure is proportional to the balancing piston mass. To a first approximation, the piston mass can be determined as follows. The weight is

$$G = mg.$$

The force exerted by the vapor on the piston is [8]

$$F = c_x S \rho_v \frac{v_v^2}{2}.$$

At equilibrium, G = F, so the mass is

$$m = \frac{c_x S}{2g}\rho_v v_v^2.$$

On increased input to the evaporation zone, the vapor pressure and temperature rise; the pressure lifts the piston, which increases the condensation area and disposes of the excess heat. As the piston's mass is constant, the vapor pressure balancing it will also be constant. The capillary gap between the piston and the wall prevents the vapor from penetrating the space above the piston. Also, the gap contains a condensate film, which reduces the friction and lag. To prevent the vapor after condensation from solidifying above the piston and preventing it from moving, it is necessary to maintain the condenser temperature above the melting point.

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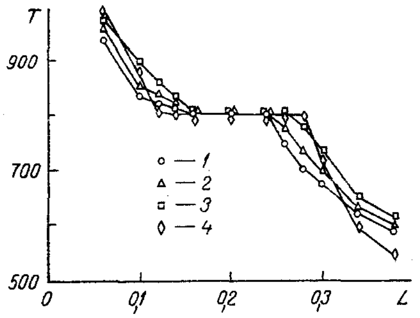

Fig. 1

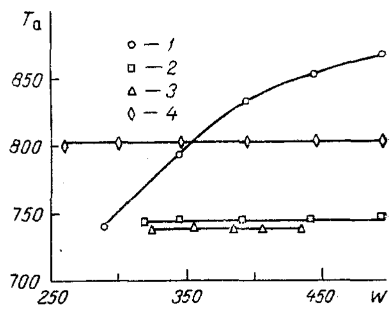

Fig. 2

Fig. 1. Temperature T (K) along the length L (m) of a tube with 35 g piston: 1) W = 260 W; 2) 300; 3) 345; 4) 395.

Fig. 2. Adiabatic-zone temperature T_a (K) as a function of power W (W): 1) unregulated tube; 2) gas-regulated tube (gas pressure 960 Pa); 3) tube with 20 g piston; 4) tube with 35 g piston.

We used a vertical stainless-steel Kh18N10T wickless tube 800 mm long, outside diameter 20 mm and wall thickness 1 mm. The heat was provided by an ohmic heater 100 mm long in the lower part. The heat was removed by a cooling liquid flowing on the outside of the condenser. The temperature distribution along the pipe was measured with Chromel–Alumel thermocouples and an R386 multimeter. The coolant was cadmium. We tested two pistons made of Kh18N10T steel 20 mm high. One had mass 20 g and the other 35 g. The gap between piston and wall was 0.1 mm. The motion was monitored from the displacement at the end of a rod joined to the piston, which was visible through a glass tube in the upper part of the condensation zone. The residual-gas pressure after outgassing was not more than 10^{-3} Pa.

Figure 1 gives temperature curves showing that there is a stable isothermal zone; this migrates because the piston is displaced when the power increases. We examined the position of the piston in the condensation zone as a function of the heat flux. A linear relation was found between the transfer area and the heat input. Varying the power from 260 to 500 W caused fluctuations in the adiabatic-zone temperature of not more than ±2 K for the 20 g piston or ±2.5 K for 35 g. The control also involved some lag. The response time to input power change did not exceed 30 sec. The temperature drift in the adiabatic zone in that time did not exceed the above values.

The larger the piston mass, the greater the vapor pressure required, and consequently the higher the stabilized temperature. Figure 2 shows that increasing the mass from 20 to 35 g increased the stabilized temperature from 742 to 803 K. For comparison, we show the adiabatic temperature as a function of power for an unregulated tube and for a gas-regulated one with helium as the gas. The stabilization in our tube was comparable with that in the gas one in the power range used.

If the tube deviates from vertical by 10°, there is no effect on the operation. The piston and condenser walls were examined after the experiment and showed less than 1 g of cadmium there. During the experiments lasting about 200 h, the tube worked in a stable fashion and retained its stabilization behavior completely.

NOTATION

Q, amount of heat; k, thermal conductivity of tube wall material; A, heat-transfer surface; δ, wall thickness; T, adiabatic-zone temperature; T_0, radiating-surface temperature; G, piston weight; g, acceleration due to gravity; m, piston mass; F, vapor pressure force; c_x, drag coefficient; S, piston area; D, tube diameter; L, length; ρ_v, vapor density; v_v, vapor speed.

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ENERGY TRANSPORT BY RADIATION IN A COMPOSITE CHANNEL

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An approximate solution is proposed for the heat flux and temperature distribution in a channel comprised of cylindrical and conical sections.

The simplest composite channels (Fig. 1) are windows in chambers with high temperature or pores of a continuous body. They can be made complicated by the quantity of sections and by their shape. There is no transverse energy transport. We also neglect longitudinal transport in the walls. But the walls reflect and reradiate the energy absorbed, and the medium dissipates it. Sources and sinks are arranged only on the endfaces. Consequently, the power of the resultant flux is constant, Q = const, W, while the channel is called conservative or adiabatic. The geometric and optical characteristics of the bodies and the endface temperature are given. The magnitude of Q and the temperature distribution over the channel length are determined.

The solution of the general problem is given in [1] on the basis of differential equations but it turns out to be quite complex even for gray bodies. An approximate, very simplified solution is proposed in this paper.

Represented in Fig. 2 are systems of coaxial cylinders or concentric spheres and channel section specimens. The analytic solutions for the system 2a are in handbooks [2, 3] but with the flaw that they do not take account of the jump in potential in a layer of the medium because of expansion of the flux of the radiant energy transport vector. Given below is a correct solution.

The systems of bodies in Fig. 2 are simulated by an electrical loop. We consider $\Theta \equiv n^2\sigma T^4$ the transport potential. The quantity $\Theta_1 - \Theta_2$ is the analog of the electromotive force and is distributed over the external and internal sections of the closed loop. Three outer sections are shown in the diagram. The jumps in the potential are written by analogy to the Ohm's law for the loop sections:

$$\Delta\Theta_1' = \frac{Q}{F_1}\frac{R_1}{A_1},\ \Delta\Theta_2' = \frac{Q}{F_2}\frac{R_2}{A_2},\ \Delta\Theta' = \frac{Q}{F_1}r_{12}.$$

The first two are lumped in points on the surfaces while the third is distributed in the medium. The internal reduction in the potential is also comprised of three parts with jumps due to $\Delta\Theta_1''$ — the action of sources in body 1, $\Delta\Theta_2''$ and $\Delta\Theta''$ — the action of sinks in body 2, where $\Delta\Theta''$ appears during "leakage" of the sinks over the surface F_2 when the density of the resultant flux is reduced. Similarly to electrodes of the current source, surfaces 1 and 2 comprise a unit. The channel section is a heat machine in which the heater cannot act without a refrigerator according to the second law of thermodynamics. But the surfaces are separated more greatly as contrasted to the electrodes, the jumps $\Delta\Theta_1''$ and $\Delta\Theta_2''$ are at their external sides while $\Delta\Theta''$ is between them. In fact, $\Delta\Theta_2''$ and $\Delta\Theta''$ are not related to the resistance to the flux. Nevertheless, we simulate all the jumps by sections of the loop. The complete jumps in the potential on and between the surfaces equal

$$\Delta\Theta_1 = \Delta\Theta_1' + \Delta\Theta_1'',\ \Delta\Theta_2 = \Delta\Theta_2' + \Delta\Theta_2'',\ \Delta\Theta = \Delta\Theta' + \Delta\Theta''.$$

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