HEAT AND MASS TRANSFER IN HEAT PIPES WITH NONCONDENSING GAS

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An experimental study is described concerning the performance parameters of heat pipes with noncondensing gas. The test results are found to agree closely with calculations.

Heat pipes with noncondensing gas, just as conventional heat pipes, are now widely used in industry.

The performance characteristics of a heat pipe with noncondensing gas derive from the fact that the vapor drives the noncondensing gas toward the cold end of the pipe, where a so-called gas lock then forms as a result. Heat pipes with noncondensing gas can be used for thermostatting various heat emitting apparatus components.

The purpose of this study was to determine how the amount and the properties of the noncondensing gas affect the transport of energy and mass through heat pipes.

It is well known that, as vapor flows through a noncondensing gas, the latter can be trapped in either of three modes: turbulently at high velocities and high pressures, viscously at low velocities and high pressures, or diffusively at low velocities and low pressures [1].

When the heat pipe is started, liquid evaporates to form a vapor—gas mixture. From the instant on when the evaporation rate begins to exceed the rate of concentrative and thermal diffusion, the vapor—gas interface gradually shifts toward the condensation zone in the heat pipe. The molar current of vapor is in this case larger than the diffusion current. The condition for a shift of the vapor—gas interface is

$$Q/r \geqslant -\rho\left(D\nabla n_i + \frac{D_T}{T}\nabla T\right). \qquad (1)$$

The displacement of the vapor—gas interface is affected by two processes.

1. As the volumes of vapor and gas change under equal pressures in both, vapor begins to condense on the gasfree cold wick surface. Gas molecules are trapped during the molar flow of vapor molecules toward the condenser. In most cases the trapping at this stage may be regarded as diffusive. The starting stage of heat pipe operation ends when the trapping becomes viscous, i.e., the number of gas molecules

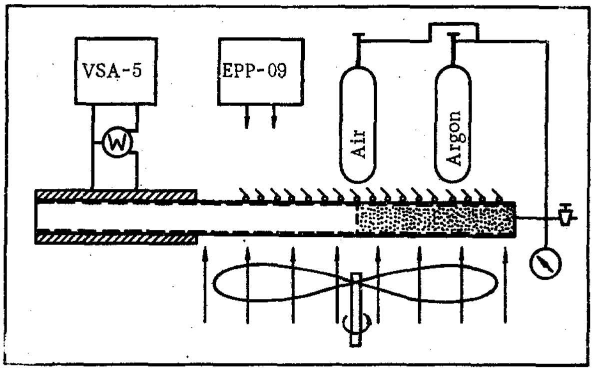

Fig. 1. Schematic diagram of the test apparatus.

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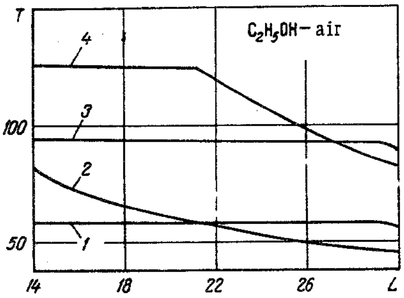

Fig. 2. Temperature profile (°C) along the condenser segment (cm), during heat dissipation by natural convection: 1, 3) without gas; 2, 4) with noncondensing gas; 1, 2) Q = 8 W; 3, 4) Q = 15 W.

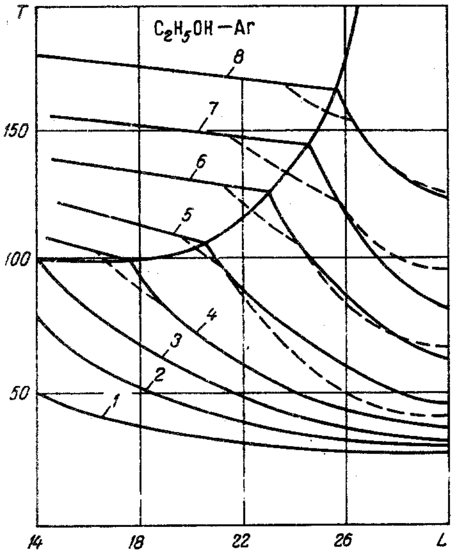

Fig. 3. Temperature profile (°C) along the condenser segment (cm), during heat dissipation by forced convection, with argon as the noncondensing gas: 1) Q = 17 W; 2) Q = 28 W; 3) Q = 36 W; 4) Q = 50 W; 5) Q = 70 W; 6) Q = 84 W; 7) Q = 100 W; 8) Q = 130 W.

in the vapor zone becomes insignificant so that the condensation ratio increases fast and thus also the rate of vapor transport.

The preceding analysis is correct when applied to laminar vapor flow, i.e., when Re < 1000 (which is typical of heat pipes with nonmetallic heat carriers). It is to be noted that the mode of vapor flow through a heat pipe depends on the kind of heat carrier, on the level of transmitted power, and on the boundary conditions.

In most common cases the vapor temperature is determined from boundary conditions of the third kind in the cooler and in the heater, namely:

$$T_\mathrm{v} = \frac{K_\mathrm{e} l_\mathrm{e} T_\mathrm{e} + K_\mathrm{c} l_\mathrm{c} T_x}{K_\mathrm{e} l_\mathrm{e} + K_\mathrm{e} l_\mathrm{e}}, \tag{2}$$

where

$$K_\mathrm{e} = \frac{1}{\dfrac{1}{\alpha_1 d_2} + \dfrac{1}{2\lambda} \ln \dfrac{d_2}{d_1} + \dfrac{1}{\alpha_2 d_1}},$$

$$K_\mathrm{c} = \frac{1}{\dfrac{1}{\alpha_2 d_1} + \dfrac{1}{2\lambda} \ln \dfrac{d_2}{d_1} + \dfrac{1}{\alpha_1 d_2}}.$$

2. What further influences the displacement of the vapor—gas interface is that, on the one hand, the vapor pressure under operating conditions is a logarithmic function of the temperature along the saturation line but, on the other hand, the gas volume is a linear function of the pressure.

For most liquids the P = f(T) relation follows the empirical equation according to Antoine [1]:

$$\lg P = A - \frac{B}{T_\mathrm{v} + C}. \tag{3}$$

Under steady operating conditions in a heat pipe with noncondensing gas, the latter is completely trapped and driven into the condensation zone or the displacement volume.

Inserting then the Antoine equation (3) into the equation of state for an ideal gas will yield the volume of the gas lock as a function of the temperature:

$$V_\mathrm{G} = \frac{mR \cdot T_\mathrm{G}}{M \cdot 10^{A - \frac{B}{T_\mathrm{V} + C}}}. \tag{4}$$

For a heat pipe of circular cross section, the length of the gas lock is

$$L_\mathrm{G} = 4 \frac{mRT_\mathrm{G}}{M\pi d_\mathrm{V}^2} 10^{\frac{B}{T + C} - A}. \tag{5}$$

In this case the area of the condensation zone, assuming that $L_a = 0$, is equal to the difference between the total condenser area and the area of the condenser portion filled with noncondensing gas:

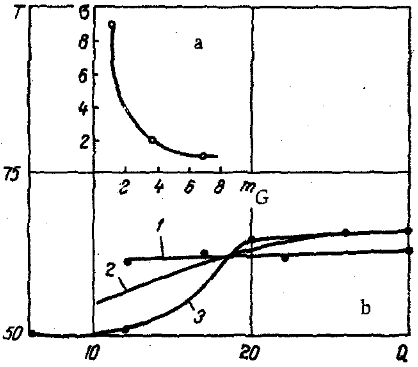

Fig. 4. Temperature sensitivity σ (W/°C) as a function of the gas mass m_G (10^{-5} kg) (a) and operating temperature T (°K) as a function of the transmitted thermal flux (b): 1) $m_1 = 1.13 \cdot 10^{-5}$ kg; 2) $m_2 = 3.63 \cdot 10^{-5}$ kg; 3) $m_3 = 6.85 \cdot 10^{-5}$ kg.

$$S_c = \pi d_V (L_r - L_{vap}) - 4 \frac{mRT_G}{Md_V} 10^{\frac{B}{T_V + C} - A} . \tag{6}$$

A simultaneous solution of Eqs. (2) and (6) yields the relations $T_V = f(Q)$ and $L_G = f(Q)$. When the temperature of the heat source in a heat pipe must be held constant, one usually adds a gas reservoir [2] or a displacement volume [3].

Expression (5) for the length of the heat pipe segment filled with noncondensing gas is

$$L_G' = 4 \frac{mR \cdot T_G}{M\pi d_V^2} 10^{\frac{B}{T_V + C} - A} - 4 \frac{V_r}{\pi d_V^2} . \tag{7}$$

If one considers a heat pipe with noncondensing gas and with an external reservoir filled with noncondensing gas, then the length of the cylindrical displacement volume in such a pipe is

$$L_G' = 4 \frac{mR \cdot T_G}{M\pi (d_V^2 - d_g^2)} 10^{\frac{B}{T_V + C} - A} - \frac{d_V^2 L_r}{d_V^2 - d_g^2} . \tag{8}$$

It follows from Eqs. (7) and (8) that, in order to improve the temperature stability, it is necessary either to increase the length or to decrease the diameter of a conventional heat pipe and, in the case of a heat pipe with a displacement volume, to reduce the size of the orifice between that displacement volume and the wick.

Analogous calculations can be made for any boundary conditions; when the gas is far from ideal, then Eq. (4) must be derived on the basis of the van der Waals relation.

The test apparatus is shown schematically in Fig. 1. It included a heat pipe made of copper, 12 mm (diameter) × 1 mm (wall thickness) and L = 300 mm long. The wick consisted of three layers of brass mesh with a volume V = 8.47 cm^3 and a 70% porosity. The active liquid was 96% ethyl alcohol. As the noncondensing gas we used air (M = 29, m = 0.019 g) in the first and the third test series, and argon (M = 39.9, m = 0.027 g) in the second test series. The thermal flux was generated by means of an electric heater L_e = 7 cm long. In order to ensure accurate power measurements, a standby heater had been placed above the main one. The heat pipe consisted of an evaporator and a condenser, without an adiabatic segment. The heat was dissipated by natural convection in the first test series but by forced convection in the second and the third test series, in the latter case with an air stream at a velocity V ≈ 10 m/sec. Along the condenser we had 17 copper–constantan thermocouples calked into the body of the heat pipe.

The purpose of the first test series was to compare the performance of a heat pipe without and with noncondensing gas (air) inside its condenser segment. The test results are shown in Fig. 2. Curves 1 and 3 represent the temperature profile along a heat pipe operating in the conventional mode. Curves 2 and 4 represent the temperature profile along a heat pipe operating in the controlled mode (i.e., with noncondensing gas present).

According to Fig. 2, the thermal resistance of the condensation zone is much higher in heat pipes with noncondensing gas than in heat pipes without it. The thermal resistance varies with the level of transmitted power.

In the second test series we evaluated the effect of a noncondensing gas – argon in this case – on the thermal resistance of a heat pipe. Argon was selected so as to avoid oxidation and electrochemical corrosion, since the heat pipe was to operate at elevated temperatures. The results of this test series are shown in Fig. 3.

TABLE 1. Results of the Experimental Study of a Heat Pipe with Noncondensing Gas, and Comparison with Calculated Values

Q, W	T_r °C	v_{test}, cm^3	L_{test}, cm	L_{calc}, cm	η, %
130	165	2,81	4,47	4,3	3,8
100	145	3,38	6,72	5,3	31,1
84	125	4,07	8,10	7,0	13,6
70	107	4,89	9,73	9,3	4,4
50	100	5,24	10,42	12,0	15,2

An analysis of the test results (Fig. 2) indicates that the distinct vapor—gas interface appears only at some threshold power within the $8 < Q_{thr} < 15$ W range, which agrees with Eq. (1).

For an analysis of the second test series, we have calculated the temperature profiles for the zone with noncondensing gas. Assuming that the heat is transmitted along the phase lock by conduction only, we can describe the temperature profile by the expression

$$t = t_0 \frac{ch a(x-l)}{ch al}, \tag{9}$$

where

$$a = \sqrt{\frac{4\alpha_1 d_2}{\lambda(d_2^2 - d_1^2)}}.$$

In order to verify the suitability of formulas (4)-(8), we have calculated the volume of gas and the length of the gas lock.

Equations (4) and (5) for ethyl alcohol and with a gas lock at the temperature t = 20°C are, in the International System of units,

$$P_G = 0.077 \cdot 10^{8.42 - \frac{1700}{T_V + 230}}, \tag{10}$$

$$V_G = 21.37 \cdot 10^{\frac{1700}{T_V + 230} - 8.42}. \tag{11}$$

Both theoretical and test data are listed in Table 1. It is evident here that the error does not exceed 21.1%, which is entirely permissible under the assumptions made here.

There is some divergence between theoretical and test values. This is because neither condensation of the diffusive vapor stream through the gas lock, nor heat conduction through the wick and heat transmission through the gas, nor the effect of the boundary conditions at the pipe end have been taken into account.

The downward trend of the test curve segments is explained by diffusion of gas into the vapor, while the higher level of the test curves in the gas lock range (Fig. 2) is explained by diffusion of vapor into the gas. It is to be noted that the slope of the curves along the active zone depends largely on the ratio

$$K = \frac{M_V}{M_G}. \tag{12}$$

When $K \gg 1$, there occurs thermal diffusion of gas into the vapor and this results in larger temperature drops across the active zone. In order to reduce the temperature gradient dT_V/dL, therefore, it is necessary to select a gas whose molecular weight is high.

For estimating the thermal stabilization in controlled heat pipes, we need the so-called coefficient of temperature sensitivity:

$$\sigma = \frac{dQ}{dT_V}. \tag{13}$$

The purpose of the third test series was to evaluate the effect of the gas mass in the reservoir on the temperature sensitivity of a controlled heat pipe. Here air served as the noncondensing gas. The gas mass, reduced to standard atmosphere, was $m_1 = 1.13 \cdot 10^{-5}$ kg, $m_2 = 3.63 \cdot 10^{-5}$ kg, and $m_3 = 6.85 \cdot 10^{-5}$ kg respectively. The temperature sensitivity σ as a function of the gas mass is shown in Fig. 4b. The relation $T = f(Q)$ is shown in Fig. 4a. According to the graph, the temperature sensitivity is an inverse function of the gas mass ($\sigma_{m_1} = 9$ W/°C, $\sigma_{m_2} = 2$ W/°C, and $\sigma_{m_3} = 1$ W/°C respectively).

Thus, our experimental study has shown that heat pipes with noncondensing gas can be successfully used as thermal regulators and thermostatting devices.

NOTATION

Q	is the thermal power;
r	is the latent heat of evaporation;
ρ	is the density;
D	is the concentrative diffusivity;
D_T	is the thermal diffusivity;
T, t	is the temperature;
α	is the heat transfer coefficient;
d	is the diameter;
P	is the pressure;
V	is the volume;
m	is the mass;
R	is the universal gas constant;
M	is the molecular weight;
S	is the surface area;
v	is the velocity;
λ	is the thermal conductivity.

Subscripts

e	refers to evaporator;
c	refers to condenser;
V	refers to vapor;
G	refers to gas;
g	refers to displacement volume;
r	refers to reservoir with gas;
p	refers to pipe.

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