INVESTIGATION OF GASDYNAMICS OF A VAPOR STREAM IN HEAT PIPES UNDER TURBULENT CONDITIONS

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Results are presented of an experimental investigation of the gasdynamics of a turbulent vapor stream in heat pipes, during simultaneous operation of the evaporation section, the adiabatic section, and the condensation section.

The evaporation and condensation sections of heat pipes can be regarded as channels with porous walls through which there is alternately blowing and suction of gas.

In [1, 2] results were presented of an experimental investigation of the gasdynamics of a turbulent stream with uniform blowing and suction in a circular porous tube modeling the isothermal conditions of heat pipe evaporation and condensation zones. The gasdynamic processes for each zone were examined separately, although they were in fact interconnected. A flow expansion zone preceded the stabilization section.

It is of interest to conduct comprehensive investigations of the gasdynamics of the heat-pipe vapor channel, consisting of evaporation, adiabatic (impermeable), and condensation sections. By doing this one can establish the interaction of these sections and can refine methods for hydraulic computation of the heat-pipe vapor channel.

The test object was a porous tube of internal diameter D = 26.5 mm and porosity $\varepsilon_f = 0.285$. The tube was made of a perforated strip of thickness $\delta = 0.5$ mm with apertures of diameter d = 1 mm, located in a checkerboard fashion. The maximum length of the porous section was L = 800 mm. The lengths of the blowing and suction sections were altered by means of pistons, inserted from dead-end channels. The geometric dimensions of the test models are given in Table 1.

The experimental investigations were conducted in the air test facility described in [2]. The facility consisted of a porous tube, a rotational flowmeter, a reservoir for smoothing out pressure fluctuations, a system of pipes, and a measuring device. In order to create uniform flow in blowing and suction, the porous channel was divided into separate sections of length 50 mm by means of thin rubber disks, and the air was supplied to each section (the blowing section) or withdrawn (the suction section) via throttle valves with large hydraulic resistance. There were no taps in the adiabatic section. Each model was tested at no less than three conditions in the Reynolds number range $Re_0 = (1.1\text{–}3.6) \cdot 10^4$. Figures 1 and 2 show some typical results of investigation of the static pressure variation along the porous channel. It should be noted that the coordinate x is calculated from the beginning of the flowing section from the blind end for all the sections.

Figures 1 and 2 show a comparison of the test and theoretical data. In the blowing section the longitudinal pressure gradient was calculated from the data of Olson and Ekkert [1], which can be approximated by the relation

$$\frac{\Delta p}{\left(\frac{u_0^2}{2}\rho\right)_c} = -2{,}19X_c^2 . \tag{1}$$

We note that Eq. (1) is in good agreement with the theoretical relation of Leont'ev et al. [3], obtained with critical blowing in the evaporation section.

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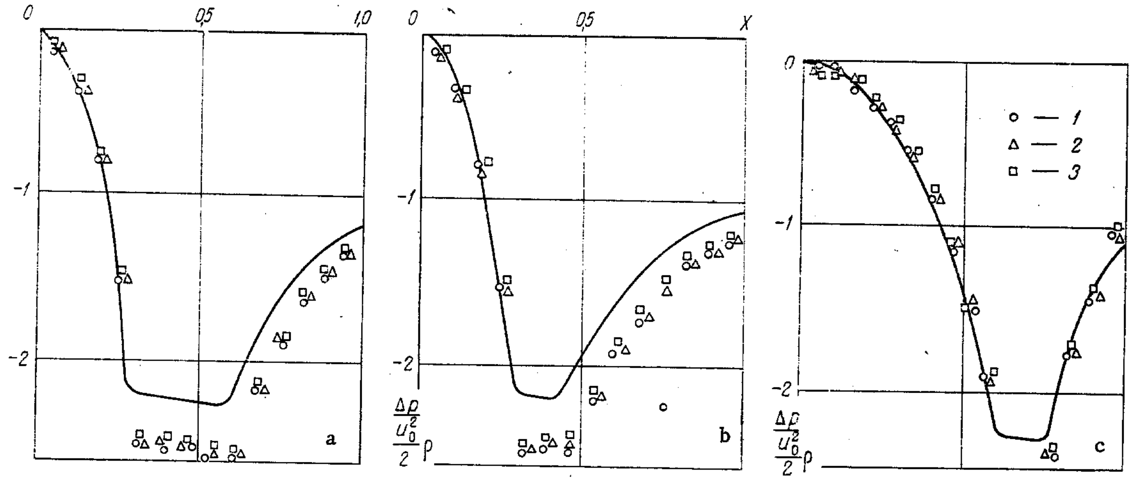

Fig. 1. The quantity $\Delta p/[(u_0^2)\rho]=f(X)$ as a function of Re_0 in the evaporation section, the adiabatic section, and the condensation section: a) L_e = 200 mm; L_a = 200 mm; L_c = 300 mm [1) Re_0 = 15,360, 2) 21,260, 3) 25,430]; b) L_e = 200 mm; L_a = 100 mm; L_c = 400 mm [1) Re_0 = 1750, 2) 22,920, 3) 27,400]; c) L_e = 500 mm; L_a = 100 mm; L_c = 200 mm. [1) Re_0 = 17,260, 2) 21,130, 3) 23,370].

TABLE 1. Basic Structural Dimensions of the Models

Model No.	Blowing-section length L_e, mm	Impermeable-section length L_a, mm	Expansion-section length L_c, mm	Model No.	Blowing-section length L_e, mm	Impermeable section length L_a, mm	Expansion-section length L_c, mm
1	200	0	150	14	200	200	200
2			200	15			300
3			400	16			400
4			500	17	300	0	200
5			600	18		0	400
6	200	50	200	19		50	200
7			300	20		200	200
8			400	21	400	0	200
9			500	22		0	300
10	200	100	200	23		0	400
11			300	24		50	200
12			400	25		100	200
13			300	26		200	200
				27	500	50	200
				28		100	200

In the impermeable section the relative pressure drop was determined from the well-known formula

$$\frac{\Delta p}{\left(\frac{u_0^2}{2}\rho\right)_a} = -\xi_0\left(\frac{L}{D}\right)_a X_a. \qquad (2)$$

From the test data of [2], the pressure variation in a section with uniform suction was determined from the relation

$$\frac{\Delta p}{\left(\frac{u_0^2}{2}\rho\right)_c} = 2[1-(1-X_c)^2] - 1.545\left(\frac{L_c}{D}\right)^{-0.27}[1-(1-X_c)^{1.73}] - \qquad (3)$$
$$-\frac{\xi_0}{3}\left(\frac{L_c}{D}\right)[1-(1-X_c)^3] - m\left(\frac{L_c}{D}\right)[0.2-(1-X_c)^4(0.2+0.8X_c)],$$

where m is an experimental factor, depending on the intensity of suction at the beginning of the porous section $K_{\perp 0}$, and determined from the formula:

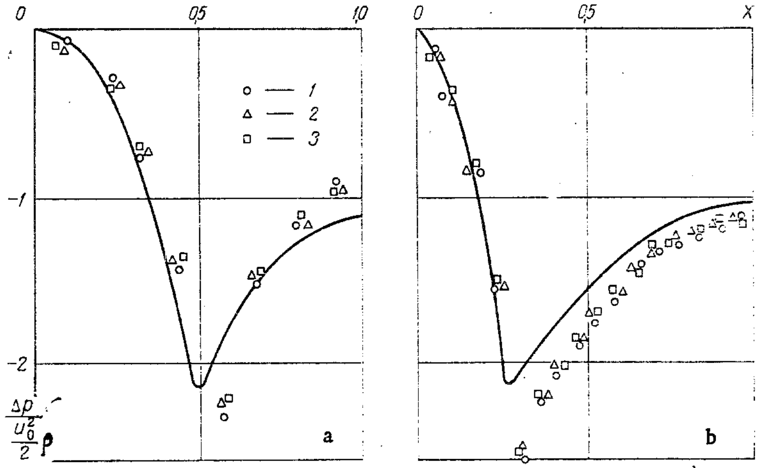

Fig. 2. The quantity $\Delta p/[(u_0^2/2)\rho] = f(X)$ as a function of Re_0 as a function of a) L_e = 200 mm; L_a = 0; L_c = 200 mm [1) Re_0 = 13,420, 2) 17,720, 3) 20,670]; b) L_e = 200 mm; L_a = 0; L_c = 600 mm [1) Re_0 = 18,220, 2) 24,440, 3) 28,930].

$$m = 0.0256BK_{\perp_0}^{0.435}. \tag{4}$$

Here

$$K_{\perp_0} = \frac{v_w}{u_0} = \frac{1}{4\frac{L_c}{D}};$$

and B is a coefficient depending on the porosity and determined from the transcendental equation

$$\sin(2\varepsilon_f + 3B) = 4.5\varepsilon_f - 3B. \tag{5}$$

Equation (3) can be used to determine the longitudinal pressure gradient in porous channels with uniform suction in the porosity range $0.115 \le \varepsilon_f \le 0.6$ and relative length $20 \le L/D \le 72$. For this case the friction factor was determined from the following empirical relation:

$$\xi = \xi_0 + 15.6K_{\perp}^{1.27} + \frac{m}{K_{\perp}}\left(1 - \frac{K_{\perp_0}}{K_{\perp}}\right). \tag{6}$$

From the graphs presented it can be seen that the pressure variation along the porous section, as calculated from Eqs. (1)-(5), coincides with the experimental data to within 20%. The largest deviation between the calculated and test data occurs at the interface of the blowing and impermeable sections, and the least discrepancy is at the interface between the blowing and the suction sections. The reason apparently is that the flow velocity profile becomes redistributed beyond the blowing section, which leads to an additional pressure drop. Some disagreement between the test data and calculated values obtained from Eq. (3) for the suction section may arise from the effect of the unsteady velocity profile at the entrance to this section. Equation (3) generalizes the experimental results for the hydrodynamics of channels with flow expansion through a porous wall, for a universal velocity profile in hydraulically smooth tubes at the entrance to the porous section. Evidently, it would be more reasonable to compute according to Eqs. (1)-(5) from the beginning of the suction zone, to reduce the error in calculation.

The results of the experimental investigations have shown that the flow gasdynamics in the blowing and suction sections depend on the relative length of these sections, the radial Reynolds number ($Re_w = v_w D/\nu$), and the channel porosity. A calculation using Eq. (1) does not show the influence of these parameters, and at the end of the blowing section it gives a value for the relative pressure change of $\Delta p/[(u_0^2/2)\rho] = -2.19$.

NOTATION

p	is the static pressure;
ρ	is the density;
u	is the mean velocity over the channel section;
v_w	is the radial velocity at the wall;
L	is the length of the channel section;
D	is the channel diameter;
d	is the diameter of apertures in the porous wall;
ε_f	is the channel porosity;
x	is the coordinate;
$X = x/L$	is the relative coordinate along the channel;
ξ_0	is the friction factor for flow in a channel with solid walls;
Re	is the Reynolds number.

Indices

e	is the evaporation section;
a	is the adiabatic section;
c	is the condensation section;
0	are the values of the parameters at the entrance to the condensation section.

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HEAT TRANSFER WITH BOILING OF A THIN WATER FILM ON A TEFLON SURFACE

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An experimental investigation has been made of heat transfer with boiling of a thin water film on a horizontal heat-generating surface, made of Teflon-4.

Contemporary literature has practically no information on heat transfer with boiling and break-up of thin liquid films on poorly wetted heater surfaces, which might serve as a guideline for the present investigation.

Tests were made on a circular horizontal surface of diameter 28 mm (the end of a copper thermal wedge) to which (with type PU-2 OFTU-G0029004 adhesive) a thin Teflon film (Teflon -4) was attached. The total thickness of the film and the polymerized substrate of adhesive, measured by means of a digital indicator with resolution of 0.01 mm, was 0.09 mm. This liquid-layer thickness above the heat-generating surface was maintained by influx from the periphery of liquid heated to the saturation temperature and was monitored by a needle contact method to within ±0.01 mm. The heat-flux density and the temperature of the heat-generating surface, allowing for the temperature drop in the Teflon layer, were determined from the temperature gradient along the wedge body. The tests were carried out in distilled water at atmospheric pressure in the heat-flux range 20-300 kW/m^2. The Teflon surface was prepared using fine emery paper.

When water boils in a large vessel (liquid-layer thickness not less than 100 mm), the heat-emission coefficients are practically the same as the heat-emission coefficients on the metallic surfaces of a heater. One should note especially that, in a large vessel, boiling is observed at individual centers for very low heat-flux values (on the order of 3 kW/m^2), and correspondingly, at low excess temperatures of the heat-generating surface of 1.5-2°C.

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