THE MECHANISM OF VAPOR CONDENSATION FROM HUMID AIR IN NARROW CHANNELS AND THE HYDRODYNAMICS OF A TWO-PHASE FLOW DURING DROPLET CONDENSATION

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A study has been made concerning the effect of the condensation mode in a forced stream of humid air on the processes of heat and mass transfer in narrow channels.

Depending on the condition of the condensation surface (wettable or unwettable), during the condensation of vapor on a cooled surface, the liquid phase will form either as a film or as discrete droplets.

During the condensation of vapor which is "pure" or has small admixtures of an inert gas, the mode of condensation determines the rate of heat transfer. In the case of "pure" vapor, as has been found by many researchers [1], the heat transfer coefficient is by one order of magnitude higher in the droplet mode than in the film mode of condensation.

So far, however, the mode of vapor condensation from a stream of humid air in narrow channels and its effect on the processes of heat and mass transfer have not yet been explained.

Results of studies concerning the processes of heat and mass transfer during droplet condensation of vapor from humid air are shown in Fig. 1, for the case of a slot channel with a rectangular cross section and a distance between plates h = 1.5, 3.0, or 8.0 mm. In order to ensure droplet condensation, the surfaces of copper plates were first treated mechanically (polished to a ∇7 finish) and then exposed to contact with water for 3-4 days. The copper plates thus became covered with an oxide film, which made them unwettable. In all subsequent tests, droplet condensation occurred on these plates (Fig. 2). The condensation trends were observed and the condensation surfaces were photographed through a lateral wall of the channel made of acrylic glass.

The channel was illuminated through the opposite wall with a collimated light beam.

The test procedure and the test conditions have been described earlier in [3, 4].

Film condensation was seen only on nonoxidized and thoroughly degreased surfaces. A very thin liquid film was seen on the upper and on the lower plate, moving in the direction of the stream. According to the data, the mass transfer rate is lower during film condensation than during droplet condensation.

Test values of the vapor flow rate are shown in Fig. 1 as a function of the difference between partial vapor pressure in air and saturation pressure at the plate surface temperature. The curves here approximate the test points obtained for droplet condensation from humid air.

According to the diagram, the vapor flow rate is lower (by up to 15%) during film condensation than during droplet condensation. The rate of convection is insignificantly lower during film condensation than during droplet condensation.

The lower rates of heat transfer and mass transfer during film condensation are explained by the fact that the liquid film of condensate on the plate surfaces produces a thermal resistance to both heat and mass transfer.

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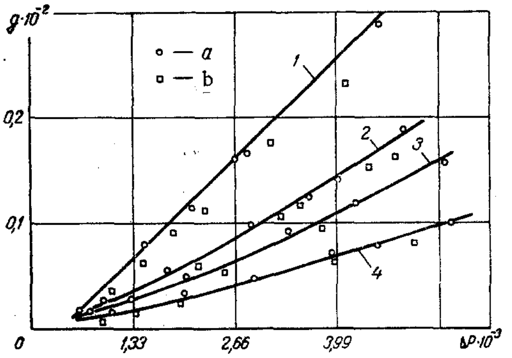

Fig. 1

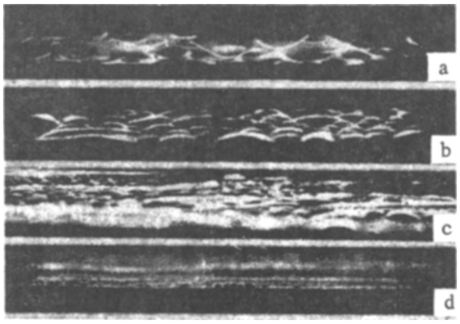

Fig. 2

Fig. 1. Vapor flow rate as a function of the partial-pressure difference ($P_V - P_S$), for (a) droplet condensation and (b) film condensation of vapor from a humid air: Re = 2010 (1), 1060 (2), 790 (3), 480 (4).

Fig. 2. Photographs of plate surfaces with droplet condensation of vapor in the channel: width h = 5 mm and $\rho_{mix} w_{mix}^2/2$ = 5 (a), 25 (b), 105 (c), 180 (d).

Therefore, a hydrophobic surface is most favorable to a maximum rate of heat transfer during the condensation of vapor from a forced stream of vapor-air mixture.

Droplet condensation was also seen on a rough oxidized duralumin surface, but with a wetting angle somewhat smaller than in the case of a polished copper plate.

During the tests we tracked the trend of droplet formation on, buildup on, and drain from the plate surface, at various orientations of the heat exchanger in the gravitational field.

It was noted, thus, that a partial-pressure difference ($P_V - P_S$) < 400 N/m^2 there occurred no vapor condensation. It was sufficient to produce on the plate surfaces large droplets first, however, to have vapor condensation occur then at the same partial-pressure difference.

This could be explained as follows. The condensation process was seen to be triggered by the formation of very small droplets on dry surfaces of the plate. Since the pressure above a droplet was higher than the pressure above a flat surface, hence the difference between partial vapor pressure in the stream and saturation vapor pressure above a forming droplet ($P_V - P_r$) was the massmotive potential. The value of P_r follows from the equation

$$P_r - P_f = \frac{2\sigma\rho}{R(\rho_1 - \rho)}. \tag{1}$$

If $P_V \approx P_r$, therefore, then condensation will not occur at all — as has also been noted in our tests.

The magnitude of the partial-pressure difference ($P_V - P_S$) at which condensation will not occur on a hydrophobic surface depends on the surface condition, on the kind of liquid, etc.

The observed trend of the condensation process at various velocities of the humid air stream through the h = 5 mm wide channel indicates that, as the velocity head of the air increases, the condensate droplets on the plate surface decrease in size before they are carried away by the stream.

Photographs of the horizontal surfaces are shown in Fig. 2 for various values of the velocity head: the upper plate (Fig. 2a) and the lower plate (Fig. 2b, c, d) of a heat exchanger. Approximate measurements of the droplet dimensions have shown that, as the velocity head decreases from 160 to 1 N/m^2, the maximum droplet dimension (at the contact between a droplet and the solid surface) increases from 1.5 to 8.0 mm. The condensate droplets are drained from the lower surface while the velocity head exceeds 5 N/m^2 at the upper surface of the horizontal heat exchanger. Only when the velocity head is less than 5 N/m^2 do some larger droplets fall from the upper surface, due to their weight, and are carried away by the stream along the lower surface.

A series of tests performed with vertical plates has shown that the rate of vapor condensation from an air stream is somewhat higher than in the case of horizontal plates. A vertical arrangement of plates may be considered preferable to a horizontal arrangement for attaining maximum coefficients of heat and mass transfer in narrow channels.

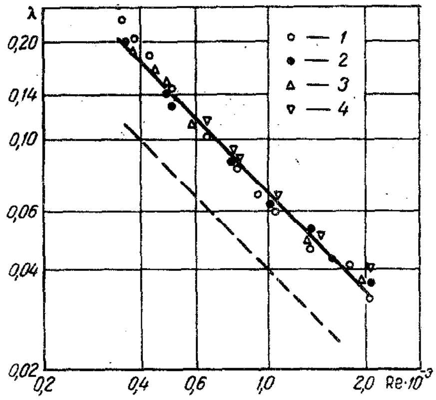

Fig. 3. Hydraulic drag in a two-phase stream ($\lambda_{2\varphi}$) through narrow channels: h = 1.5 mm (1), 3.0 mm (2), 5.0 mm (3), 8.0 mm (4).

The measurements included also hydraulic pressure losses during droplet condensation of vapor from an air stream.

The hydraulic losses in a two-phase stream through a channel consist of several components: friction loss at the interphase boundary, loss on droplet separation from the condensation surface, and loss on droplet acceleration.

By analogy to a one-phase stream, the hydraulic pressure losses were calculated here according to the formula

$$\Delta P = \lambda_{2\varphi} \frac{\rho_{mix} w_{mix}^2}{2} \frac{l}{h}, \tag{2}$$

where $\lambda_{2\varphi}$ denotes the drag coefficient for the two-phase stream.

The test data on the drag coefficient are shown in Fig. 3 as a function of the Reynolds number. These data fit the curve

$$\lambda_{2\varphi} = \frac{69}{Re_{mix}}. \tag{3}$$

The dashed line in Fig. 3 represents the hydraulic pressure losses during a forced flow of dry air.

The test data were evaluated on the basis of the distance between plates as the characteristic dimension. The temperature of the vapor-air mixture at the entrance to the heat exchanger was regarded as the governing temperature.

During droplet condensation of vapor from a humid air stream through a narrow channel, according to these data, the hydraulic pressure losses are much higher than in a one-phase stream.

NOTATION

p_V	is the partial pressure of vapor in air;
p_s	is the saturation pressure of vapor at the plate surface temperature;
p_r	is the partial pressure of vapor above the droplet surface;
p_f	is the partial pressure of vapor above the flat surface;
ρ_{mix}	is the density of humid air at the heat exchanger entrance;
w_{mix}	is the velocity of humid air at the heat exchanger entrance;
l	is the length of channel;
Re	is the Reynolds number;
μ_{mix}	is the dynamic viscosity of the vapor-air mixture;
σ	is the surface tension in a droplet;
R	is the radius of a droplet;
ρ_1	is the density of saturated vapor;
ρ	is the density of liquid.

LITERATURE CITED

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