

HEAT AND MASS TRANSFER DURING DROPLET
CONDENSATION OF WATER VAPOR FROM A
STREAM OF RAREFIED HUMID AIR IN NARROW
RECTANGULAR CHANNELS

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An experimental study has been made concerning the heat and mass transfer process during vapor condensation from humid air in narrow channels.

Heat exchangers with narrow channels have found wide applications in many branches of modern technology.

The process of heat and mass transfer during vapor condensation from humid air in such channels has not been sufficiently well explored yet. There are hardly any data available in the technical literature pertaining to heat and mass transfer during the condensation of water vapor from a stream of humid air under pressures below atmospheric. The data which are available pertain only to the effect which a small quantity of noncondensing gas has on vapor condensation in vacuum. Under those conditions, certain laws governing the condensation of "pure" vapor, namely the dependence of the heat transfer coefficient on the heat load, remain in force. When the vapor content in the mixture is low, however, the rate of vapor condensation is determined mainly by the velocity at which vapor is transported through the boundary layer with a lower vapor concentration, by the condensation mode (droplet air film), and by the hydrodynamic characteristics of condensate and humid air flow.

We will present here the results of a study concerning the process of heat and mass transfer during a forced flow of rarefied air through a channel $h = 1.5, 3.0, 5.0,$ and 8 mm wide between two parallel copper plates. The width of the plates was in each case the same and equal to $b = 100$ mm.

The experiment was performed on a test stand and by a procedure all shown in [4, 5].

For separating the phases in the two-phase stream and collecting the condensate at the heat exchanger exit, we used a device with a hydrophilic porous cylinder shown schematically in Fig. 1.

The porous cylinder 3, 130 mm long, wrapped in a layer of polyvinylformal 5, was placed at the heat exchanger exit. During the test, a pressure drop ΔP_p built up between cavities 1 and 4 which, according to the requirement of liquid permeability only, should not have exceeded the "limiting" pressure drop ΔP_l due to the capillary effect, i. e., the condition $\Delta P_p \leq \Delta P_l$ was to prevail. This "limiting" pressure drop across the porous element was measured directly in the vapor separator system at the wet layer of polyvinylformal, periodically throughout the test. The condensate droplets carried away from the surfaces of the copper plates 2 by the air stream were taken up by the polyvinylformal and transported by capillary action to the surface of the porous cylinder, through which the condensate then oozed into the measuring container 6. In this way, the phases were separated and the liquid condensate was collected effectively at all velocities of the humid air stream within the test range ($v = 0.8-20$ m/sec).

The state parameters of the humid air were varied during the test over the following ranges: temperature $t_{\text{mix}} = 20-100^\circ\text{C}$, volume vapor content $\varepsilon = P_v/P_{\text{mix}} = 0.03-0.40$, pressure $P_{\text{mix}} = (0.133-1.0) \cdot 10^5$ N/m², Reynolds number $Re = 360-2010$.

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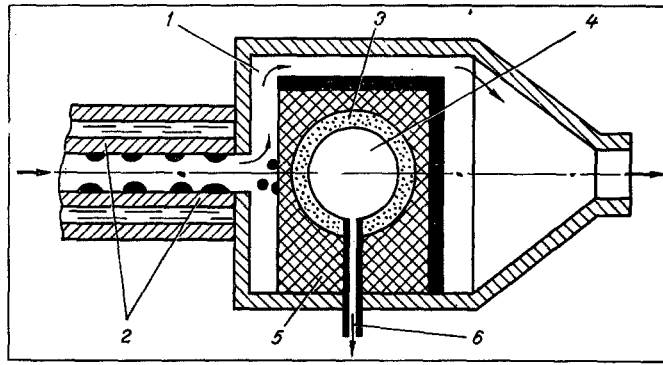


Fig. 1. Device for separating the phases in a two-phase stream and collecting the condensate.

The test procedure was set up so that both the coefficients of convective heat transfer and the coefficients of mass transfer could be determined. The coefficients of convective heat transfer were determined from the change of temperature in the humid air through the test segment, just as in the study of heat transfer in dry air [6], while the mass transfer coefficients were determined from the quantity of collected condensate. In order to obtain reliable data on the condensate flow rate, each test was performed under steady-state conditions – after the parameters of the humid air at the channel entrance have remained constant for a long period of time (up to 5 h).

The surfaces of the copper plates had been treated to a grade $\nabla 7$ finish. They were unwettable by liquid. Visual observations of condensation from humid air revealed that this was a droplet condensation in each case, as had been reported earlier in [1-3].

In determining the coefficients of heat and mass transfer, the temperature of the channel surfaces, measured with thermocouples, was regarded as the temperature of the condensation surface. The saturated-vapor pressure corresponding to this temperature was also determined. In a strict sense, the thermal resistance of the condensate at the channel surfaces should also have been taken into account and used in the evaluation of test data for the determination of the real surface temperature. However, while the temperature of a liquid film during film condensation can still be calculated, it is practically very difficult to calculate the temperature of an intricately shaped surface during droplet condensation. Rough estimates have shown that the true surface area of a condensate can be about 1.5 times larger than the area of a dry plate surface.

The variation in the coefficients of heat and mass transfer along the channel length was determined in channels of various lengths ($l/h = 20, 40, 80, 133, 266$).

An evaluation and analysis of test data has yielded the following criterial relations:

for convective heat transfer

$$Nu = 0.065 Re^{0.66} k_l, \quad (1)$$

for mass transfer

$$Nu_D = 0.084 Re^{0.92} \Pi_q^{0.25} \left(\frac{l}{h} \right)^{-0.45}, \quad (2)$$

with the correction factor k_l accounting for the change in the heat transfer rate over the length of the hydrodynamic stabilization segment.

All physical quantities in the expressions for the Nusselt number and the Reynolds number were determined from the vapor-air state parameters at the heat exchanger entrance.

Test data on convective heat transfer and mass transfer in humid air are shown in Fig. 2a, b. For comparison, a dashed line is also shown in Fig. 2a representing generalized test data on convective heat transfer in dry air. The values of the correction factor in formula (1) for $l/h = 20, 40, 60, 80$, and 100 are, respectively, $k_l = 1.56, 1.18, 1.08, 1.04$, and 1.00 ; when $l/h > 100$, the Nusselt number does not depend on the channel length.

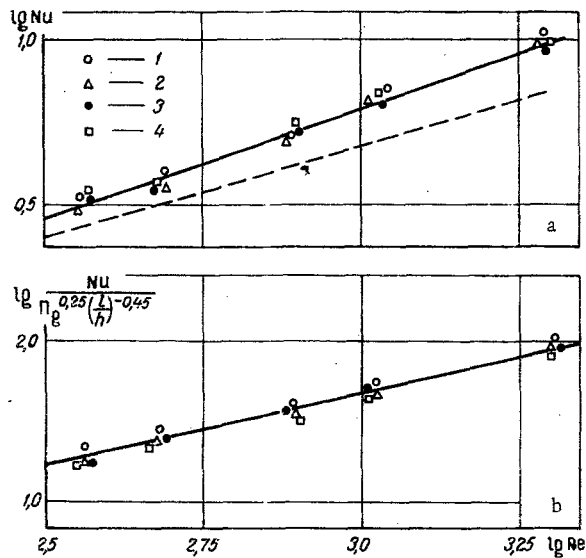


Fig. 2. Test data on convective heat transfer (a) and mass transfer in humid air (b) ($L/h > 100$): 1) $h = 8$ mm; 2) 5; 3) 3; 4) 1.5.

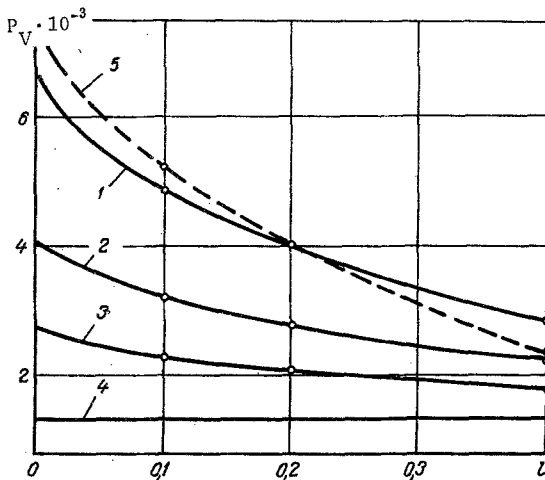


Fig. 3. Variation of partial vapor pressure (1, 2, 3), of saturated-vapor pressure (4), and of saturation pressure (5) along the $h = 5$ mm test channel: 1) $P = 6.72 \cdot 10^3$ N/m²; 2) $4.05 \cdot 10^3$; 3) $2.7 \cdot 10^3$.

Thus, when the vapor-air pressure changes from $7.98 \cdot 10^4$ to $1.33 \cdot 10^4$ N/m² as a result of a higher molecular diffusivity, the vapor flow density should presumably increase 6 times but, according to our test data, it increases 9 times. Such an increase of the mass transfer coefficient (in addition to its normal increase on account of the higher molecular diffusion) can be explained by a higher mass fraction of vapor in the mixture under a lower total vapor-air pressure at the same Reynolds number. As the velocity of the vapor-air stream increases under a reduced total pressure, furthermore, the thermal resistance of the condensate at the channel surfaces decreases, because an air stream with a high velocity head can easily detach and carry away droplets even when they are small.

Increasing the forced-flow velocity of humid air offers one way of improving the mass transfer. With the same partial vapor pressure at the channel entrance, an increase of the stream velocity will result in a merely insignificant increase of the partial vapor pressure at the exit and, according to Eq. (2), the mass transfer coefficient is an almost linear function of the stream velocity ($Re^{0.92} \sim Re$).

Evidently, the rate of convective heat transfer during vapor condensation from humid air is much higher than the rate of heat transfer in dry air. Certainly, the transverse flow of vapor has a definite effect on both the heat and the mass transfer. However, the boost in the rate of convective heat transfer under our test conditions can hardly be attributed to a transverse vapor flow. The presence of rather large condensate droplets on the surfaces of narrow channels does, in effect, alter the channel dimensions and affects the hydrodynamic structure of the condensate. These, in our opinion, are the factors which determine the boosting of convective heat transfer in humid air.

It has been established experimentally that the mass transfer rate is a function of the velocity, the pressure, and the temperature of humid air, as well as of the difference between partial pressures $P_V - P_{sw}$.

As the total pressure of humid air decreases, while the difference between partial pressures $P_V - P_{sw}$ and the Reynolds number remain unchanged, the vapor flow density toward the condensation surface increases. It increases at a faster rate than the molecular dif-

As was to be expected, the coefficients of heat and mass transfer are much higher in shorter channels. The high condensation rate in the entrance stage is explained by the still low resistance to diffusion when humid air enters the channel. Subsequently, as the humid air moves forward, diffusion boundary layers form and build up in the channel. The concentration of air in the mixture increases near the condensation surface, which causes the resistance to vapor diffusion to increase and, consequently, the mass transfer rate to decrease.

Interestingly enough, in several tests the air at the heat exchanger exit was supersaturated. Its maximum relative humidity there reached up to 150%. As an example, we show in Fig. 3 the variation in the partial vapor pressure at $Re = 2010$ and $P_{mix} = 7.98 \cdot 10^4 \text{ N/m}^2$ along the test channel with $h = 5 \text{ mm}$ thick walls. According to the diagram, already at an entrance vapor pressure $P_V = 6.72 \cdot 10^3 \text{ N/m}^2$ the air saturates approximately at the center of the test channel. From the instant of saturation till the exit of the humid air from the channel, however, no fog (volume condensation) could be seen. Neither was fog seen in any other test.

This is explained by the use of thoroughly filtered air by the supersaturation level much below the critical supersaturation at which fog appears.

NOTATION

$Nu_D = gh/D_p \Delta P$	is the Nusselt diffusion number;
g	is the vapor flow density toward the condensation surface;
D_p	is the molecular diffusivity of water vapor in air;
P_V	is the vapor pressure in air at the heat exchange entrance;
P_{sw}	is the pressure of saturated vapor at the wall temperature;
$Re = (v_{mix} \rho_{mix})h/\mu_{mix}$	is the Reynolds number;
v_{mix}	is the velocity of humid air;
ρ_{mix}	is the density of humid air;
μ_{mix}	is the dynamic viscosity of humid air;
$\Pi_g = (P_V - P_{sw})/P_{mix}$	is the dimensionless group describing the molecular flow;
P_{mix}	is the vapor-air pressure
l	is the channel length;
h	is the distance between flat channel plates.

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