AN EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN THE DROPWISE CONDENSATION OF WATER VAPOR ON A VERTICAL TUBE

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The results of an experimental study of heat transfer are presented for the dropwise condensation of vapor on vertical tubes over wide intervals of variation in tube length and specific heat load. It is demonstrated that the condensate runs off in jets on long tubes.

There are extremely few experimental data on heat transfer for the dropwise condensation of water vapor on the long vertical tubes generally used in industry (4 m and longer). We know that Kirschbaum [1] tested a 1.5-meter heat-exchanger tube and found a heat-transfer coefficient on the order of 20–35 kW/m^2·deg, i.e., one smaller by a factor of 2–5 than obtained by other researchers [2–6] on installations with limited surface height (less than 0.6 m). Fitzpatrick and his co-workers [7] found similar results with vertical tubes 1.8 and 3 m long. Thus, in dropwise condensation, the heat-transfer coefficients for a tube 3 m in length are smaller by a factor of approximately three than for a tube 1.8 m in length. We assume [6, 8, 9] that the comparatively low values for the heat-transfer coefficient measured on long tubes are a result of the merging of the dripping condensate drops into a continuous film and a changeover to a film-condensation regime in the lower sections of the vertical heat-exchanger surface. However, no special investigation of this problem has yet been undertaken—from the practical standpoint, a most important next step.

This particular study was undertaken for the condensation of vapor on the outside surface of a vertical tube. The experimental condenser consisted of a "tube-in-a-tube" heat exchanger made up of a water-cooled copper tube with an outside diameter of 40 mm, 4.63 m in length, within a casing. The vapor was led into the condenser through three connecting tubes uniformly located along the height of the apparatus. The length of the heating surface was altered (between 0.5 and 4.63 m) by shifting the location of the point at which the condensate run off from the tube, and this was done by means of a movable cup attached to the tube. To ensure dropwise condensation, the surface of the tube was covered with a thin (0.2–0.3 μm) hydrophobic polyethylhydrosiloxane film [10].

Hydrodynamic and thermal stabilization of the cooling water flowing into the heat-exchanger tube was provided for in these tests.

The flow rate for the condensate was measured by means of a volume meter. The quantity of transmitted heat was calculated on the basis of the condensate-flow rate. The water-flow rate through the heat-exchanger tube was measured by means of a membrane. The temperature of the vapor and the water was measured with a mercury thermometer calibrated for 0.1 deg.

The specific heat loads were regulated by varying the temperature of the heated water, i.e., by altering the over-all temperature head.

The coefficient for the transfer of heat from the condensing vapor was determined from the familiar formula

$$\alpha = \left[\frac{1}{k} - \frac{d_o}{2\lambda}\ln\frac{d_o}{d_i} - \frac{1}{\alpha_w}\frac{d_o}{d_i}\right]^{-1}. \quad (1)$$

The total heat-transfer coefficient k was determined on the basis of experimental data: the coefficient α_w for the transfer of heat to the water was calculated from the following equation [11]:

$$\mathrm{Nu} = \frac{0.14\,\mathrm{Re}\,\mathrm{Pr}\sqrt{\xi}}{\ln\dfrac{\mathrm{Re}\sqrt{\xi}}{760} + 2\ln\dfrac{1+5\mathrm{Pr}}{1+0.2\,\mathrm{Pr}} + 2.4\,\mathrm{Pr}\,\varphi(\mathrm{Pr})}. \quad (2)$$

The experiments were carried out under conditions in which the vapor was condensed at atmospheric pressure and at heat loads between 40 and 300 kW/m^2. The maximum velocity for the vapor in the condenser section did not exceed 1 m/sec, i.e., it was not great. The velocity of water motion through the heat-exchanger tube was kept constant in all of the tests and amounted to 25–26 m/sec. The need for high-speed water was methodically dictated by the following considerations. First of all, it provided for greater sensitivity on the part of the over-all heat-transfer coefficient to changes in the intensity of the heat transfer on vapor condensation (the coefficient for the transfer of heat to the water ranged from 85 to 105 kW/m^2·deg, i.e., it was of the same order as in the case of dropwise condensation). Secondly, the use of high-speed water in the tube ensured virtually isothermal vapor-condensation conditions over the height of the surface (the water in the tube was heated 0.2–1.5 deg).

The test results are shown in Fig. 1. We see from the curves that for heat loads on the order of 40–180 kW/m^2—characteristic for industrial heat-exchanger equipment—the heat-transfer coefficient drops with increasing heat load and with increasing tube length. With low heat loads (40–60 kW/m^2) the effect of the length of the heat-exchanger tube on the intensity of the heat transfer is particularly pronounced. Thus, with an increase in the tube length from 0.5 to 4.63 m there is a corresponding drop in the heat-transfer coefficient from 150 to 65 kW/m^2·deg (q = 60 kW/m^2). An increase in the heat load leads to a leveling of the effect exerted by the length of the heat-exchanger tubes on the intensity of

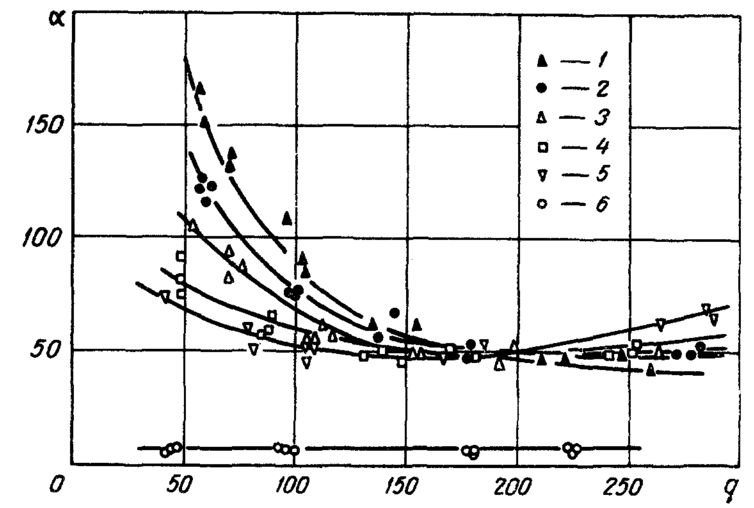

Fig. 1. Heat-transfer coefficient α (kW/m^2 ; deg) versus specific heat load q (kW/m^2) for the following tube lengths, 1) 0.5, 2) 1.5, 3) 2.5, 4) 3.5, 5) 4.63 (1–5 show dropwise condensation), 6) 4.63 (film condensation.

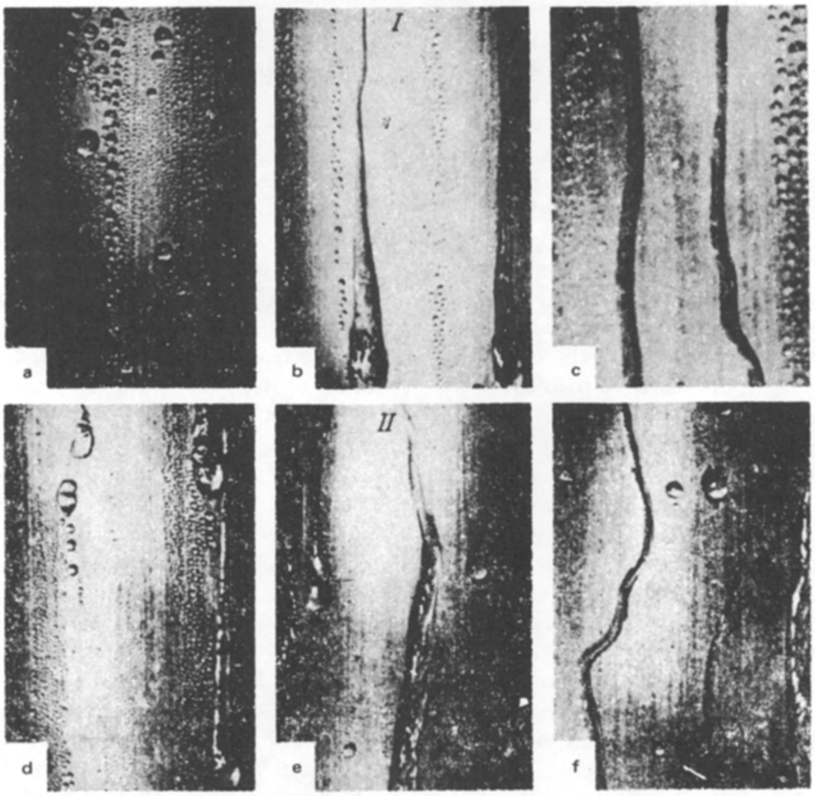

Fig. 2. Dropwise condensation in various regions of the vertical tube (I, q = 60 kW/m^2; II, 250) with the following distances of the region from the upper end of the tube: a, d) 0.2 m; b, e) 2.5 m; c, f) 4.5 m.

the heat transfer, and for heat loads on the order of 150–200 kW/m^2 the heat-transfer coefficients in the case of dropwise vapor condensation become virtually identical for both long and short tubes (about 50 $kW/m^2 \cdot deg$).

To explain the resulting relationships, let us examine the hydrodynamic regimes of condensate runoff. The visually observable pattern of dropwise condensation at various sections of the vertical tube is shown in Fig. 2 (for heat loads of 60 and 250 $kW/m^2 \cdot deg$, respectively), and here we can note two basically distinct regimes of condensate runoff: the "dropwise" regime (a, b, d) and the "jet" regime (c, e, f). The "dropwise" regime of condensate runoff is briefly described as follows: The condensation surface is coated with drops increasing in size (Fig. 2a) and these, on reaching a critical dimension (2–4 mm), bead up, picking up the smaller drops on their way, and increasing in size. As the volume of the beading drop increases they gradually assume the form of a jet and the regime of the entire condensate runoff motion then becomes exclusively "jet-like" in nature.

In the dropwise condensation characterized by jet-condensate runoff, the drops on the surface are unable to grow to critical size, since they are removed by the jet. The nature of the jet motion over the heat-exchanger surface is determined by the amount of condensate runoff, i. e., by the heat load and the tube length. For small quantities of condensate runoff, the motion of the jet is virtually rectilinear (Fig. 2c); in the case of large quantities the jet meanders chaotically (Fig. 2f). In the last case, the jets—covering the greater portion of the surface—are considerably more effective in removing the condensate drop from the surface (compare Fig. 2c and f). In evaluating the effect of the runoff jet on heat transfer in the case of dropwise-vapor condensation we should bear in mind at least two competing consequences. On the one hand, the jets of the condensate covering a portion of the surface impair the transfer of heat; on the other hand, however, they remove the condensate drops from the heat-transfer surface and promote an increase in the intensity of the heat transfer. The relationship between the cited processes apparently defined the nature of heat-transfer intensity as a function of the quantity of condensate runoff.

Thus, with dropwise condensation on a vertical surface, the latter can be divided into two zones: an upper zone with a dropwise mode of condensate motion, and a lower zone, with a jet condensate motion. With increasing heat load the upper zone is reduced and the lower zone is enlarged (compare Fig. 2b and d, as well as Fig. 2c and e). The established hydrodynamic pattern of condensate runoff makes it possible to provide the following explanation for the results that were derived (Fig. 1): With increasing heat load there is an increase in the fraction of the surface covered by jet condensate motion, thus resulting in a corresponding reduction in the heat-transfer coefficient. On reaching specific values for the heat load (150–200 kW/m^2) the zone of the cited regime becomes decisive both for long and short tubes, thus leading to the corresponding equalization of the heat-transfer coefficients.

The weak relationship between the intensity of heat transfer and the heat load for a long tube indicates that the earlier-cited competing consequences: 1) the covering of the surface by the runoff jet and 2) the cleansing of the drops from the surface by the jets—are in approximate balance. The slight rise in the heat-transfer coefficients noted in the experiments for the condensation of vapor on long tubes in the case of heat loads in excess of 200 kW/m^2 are apparently also governed by a change in the cited relationship. With an intense jet-condensate motion (Fig. 2c) the effect of removing the condensate drops from the surface evidently begins to predominate over the effect of surface coverage by the runoff jet.

In conclusion, it is appropriate to note that the intensity of heat transfer in the case of vapor condensation under conditions of jet-condensate runoff (Fig. 1, curve 5) remains considerably higher, as before, than under the conditions of a turbulent-film condensate (Fig. 1, curve 6).

NOTATION

α is the heat-transfer coefficient in liquid condensation of vapor; k is the total heat-transfer coefficient; α_w is the coefficient of heat-transfer to the moving water; λ is the thermal conductivity; d_0 and d_i are the outside and inside diameters of the heating tube; Nu, Re, and Pr are the Nusselt, Reynolds, and Prandtl numbers; ξ is the hydraulic resistance factor; q is the specific heat load.

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