EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER WITH DROP CONDENSATION INSIDE HORIZONTAL TUBES

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The authors describe the method and present results of an investigation of heat transfer with drop condensation of water vapor, generated by injecting fluoride-containing bisulphide into the condensate of a closed system.

Drop condensation is an efficient method of achieving a manifold intensification of heat transfer [1-3]. However, the literature has no data for an important practical case, that of condensation inside a horizontal tube. In the present study we have investigated the laws of drop condensation of water vapor inside a horizontal brass tube of inside diameter d = 30 mm, and length l = 100 mm, during independent variation of the flow regime parameters. A fluoride-containing disulphide was used as a hydrophobizator. According to the data of [4], this hydrophobizator ensures stable drop condensation. Tests to evaluate the influence of the disulphide on the intensity of the process were conducted on a cleaned surface, and then on a surface washed with distilled water. We first conducted investigations on film condensation, following which the disulphide in the amount 7 mg was injected into the distillate of mass 70 gm filling the electric boiler. Figure 1 shows a schematic diagram of the experimental facility on which the investigations were made, and includes the closed vapor circuit and three independent cooling system circuits. Water vapor generated in the electric boiler 1, passing through the separator 2, is superheated by 1-2 deg K and comes to the test condenser 3. From the condenser the water vapor mixture goes to the measuring unit 5, where it is separated. The condenser from the measuring unit is poured into the condensate bath 7, and the noncondensed vapor moves into the supplementary condenser 8, which has an independent cooling system operating from the water mains. The condensate from the supplementary condenser flows into the condensate bath via the measuring unit 6.

The main element of the equipment is the test condenser, consisting of two sections: settling and measuring. The settling section is a heat exchanger, a tube within a tube, of length L = 1.3 m with the internal diameter of the copper tube equal to d = 30 mm, and it also has an independent cooling system from the water main. The flow rate of cooling liquid through the cooling liquid through the cooling system of the section was measured by a type RS-5 rotameter. The function of the settling section is to create different vapor condensation regimes in the following measuring section of the test condenser by its partial or full condensation. The measuring section is also a tube within a tube heat exchanger with the special feature that it is demountable, and the inner test brass tube is made with a thin wall δ =3 mm and length of cooling section l = 0.105 mm. The cooling system of the measuring unit is equipped with closed and forced cooling, and has a level for make-up and pouring off of excess cooling liquid. The gap of the inter-tube space of the cooling sleeve of the measuring section is 4 mm. The measurement of liquid flow rate was made with a type RS-5 rotameter, equipped with a bypass line. The ends of the test condenser had glass windows for visual observation. In order to reach low heat flux densities ($q \leq 50$ kW/m^2) the circulation cooling circuit of the experimental section had a supplementary electric heater of power N = 1.5 kW supplied via a voltage regulator.

The mean wall temperature was determined using a resistance thermometer, wound in two layers of type PELShO copper wire of diameter d = 0.05 mm. Then it was pulled through a brass capillary tube of external diameter d = 1.8 mm and positioned in a spiral with a pitch of 20 mm along the measuring section in a channel of depth 2 mm from the outer surface. Then the channel was soldered with tin, and trimmed flush with the tube surface. The temperature of the internal heat transfer surface of the tube was determined, allowing for the depth of emplacement of the resistance thermometer and the readings of a type Shch-34 ohmmeter. The temperatures of the vapor and the cooling liquid were measured with calibrated copper—constantan thermocouples and a type Shch-68003 digital millivoltmeter. The heat flux density and the amount of condensate formed were determined from the amount of heat transmitted to the cooling liquid in the corresponding sections of the condenser. The vapor rate was determined from the flow rate of condensate,

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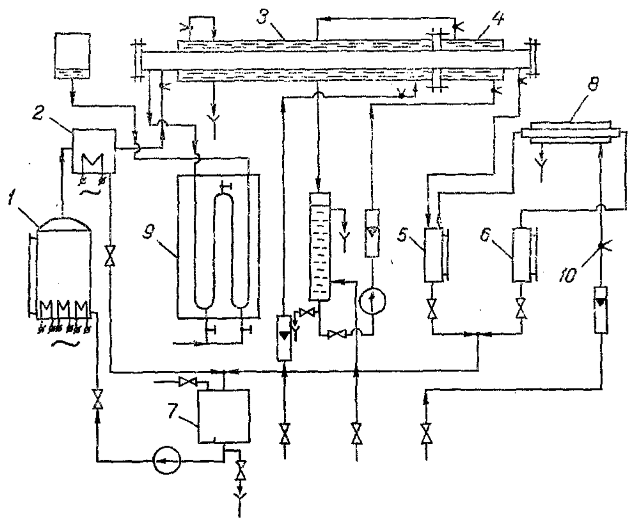

Fig. 1. Schematic of the experimental equipment: 1) electric boiler; 2) separator-superheater; 3) settling section of the condensator; 4) experimental section of the condensator; 5, 6) measuring unit; 7) condensate bath; 8) supplementary condenser; 9) differential manometer; 10) thermocouples.

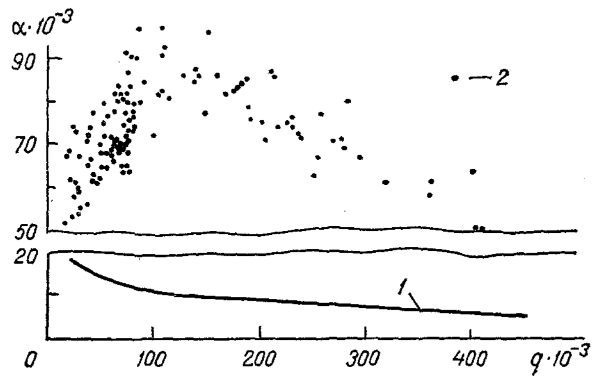

Fig. 2. Influence of heat flux density on the heat removal intensity with drop condensation for $t_s = 100°C$ and $W_v < 2$ m/sec: 1) film condensation [5]; 2) drop condensation.

measured by a volume method. The investigations were conducted with condensation of water vapor at atmospheric pressure. Analysis of the measurement errors showed that with the measuring apparatus and the investigative technique the heat transfer coefficient (α) could be determined with an accuracy of not less than 7.2% at a confidence level of 0.96.

The heat transfer laws were investigated for a variation of heat flux density of $\overline{q}_\ell = 20$—500 kW/m^2, mean vapor velocity of $W_{vapor} = 0.2$—11 m/sec, measured at the exit of the measuring section, and $Re_{film} = qL/r\mu = 2$—200. In computing the heat transfer coefficient and the heat flux density the total heat flux was reference to the internal surface of the tube in which the wall temperature was measured. Visual observations, taken during the experimental investigations, showed that the concentration of disulphide used in the tests achieved stable drop condensation inside a brass horizontal tube of length $l = 0.1$ and a copper tube of length $L = 1.3$ m. During six months of practically daily operation the equipment did not show a reduction of the numerical values or the heat transfer coefficients, nor did a mixed regime of condensation appear, even on a negligible part off the heat transfer surface. The composition of the disulphide used ensured stable drop condensation over the whole perimeter of the horizontal tube, apart from the perimeter of the stick of condensate. The maximum radius of the fixed drops was $R_k = 1.5$ mm. The frequency of sliding of drops was determined by the heat flux density, and increased when it increased. For a vapor velocity of $W_{vapor} \leq 2$ m/sec the drops ran off preferentially along the perimeter of the horizontal tube. we did not observe separation of the drops formed from the top generator of the tube surface. Repeated cleaning and surface preparation during the potting of hydrophobizator into the condensate did not show an influence on the heat transfer intensity.

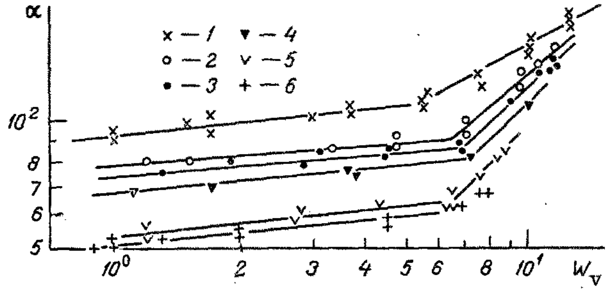

Fig. 3. Dependence of the heat transfer coefficient on the flow of condensate and the vapor velocity for ℓ = 120 kw/m^2 and T_s = 373 K: 1) Re_f = 2; 2) 28; 3) 53; 4) 80; 5) 107; 6) 130. α, kw/(m^{2}K); W_v, m/sec.

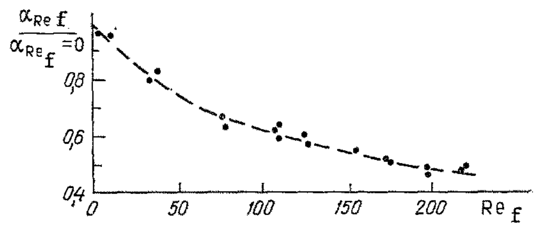

Fig. 4. Influence of flow of condensate on heat transfer in drop condensation inside a horizontal tube with W_v < 2 m/sec .

influence of the heat flux density on the heat transfer coefficient $\alpha = f(q)$ for $W_{vapor} \leq$ 2 m/sec and P = 0.1 MN/m^2 is shown in Fig. 2. For comparison Fig. 2 also shows a line correlating the test data on film condensation of a fixed vapor (W_{vapor} < 0.5 m/sec), obtained in the same test section before the stimulator of drop condensation was inserted into the water boiler. The dependence of α on q shows a clearly pronounced maximum. The maximum heat transfer coefficient is reached for q = 80 = 130 kW/m^2, which corresponds to a temperature head value of ΔT = 0.8 - 1.4 K. Further increase of the heat flux density causes the heat transfer coefficient to decrease. Here the degree of intensification of the process varies from factors of 7-10 to 6-7 compared with film condensation.

During the investigations we determined the influence of vapor velocity and the amount of flowing condensate in the settling section on the intensification process. The results of the investigations are shown in Figs. 3 and 4 in the form of the dependences $\alpha = f(W_v)$ and $\alpha_{Re_f}/\alpha_{Re_f=0} = f(Re_f)$ (where $\alpha_{Re_f=0}$ is the heat transfer in the absence of flow; α_{Re_f} is the same thing with flow of condensate), with fixed values of the other flow regime parameters. The influence of these parameters was estimated for a change of Re_f = 2—200 in the range of vapor velocity W_v = 0.1—11 m/sec. Of the flow regime parameters the vapor velocity proved to have the greatest influence on the heat transfer intensification. It follows from the data shown in Fig. 3 that the nature of the dependence of the heat transfer coefficient on for different values of W_v did not change in the main. Here the intensity of the process under the conditions Re_f is determined by the vapor flow velocity. For Re_f = const. the influence of the velocity can be neglected at all values of Re_f.

Reduction of the data shows that the degree of influence of the vapor velocity on the intensity of the process depends on the velocity value. For example, in the range of values $W_v \leq$ 5 m/sec the exponent for the vapor velocity is 0.1. With further increase of the vapor velocity the exponent oscillates from 0.5 to 1.0. Thus, an increase of velocity during drop condensation of water vapor lead to a greater increase of the heat transfer coefficient than in the case of film condensation inside a horizontal tube [5].

The influence of increased amount of condensate on the law of the process was studied by increasing the heat load in the settling section of length L = 1.3 m. The investigations showed (Fig. 4) that the degree of influence of the condensate flow rate on the heat transfer intensity was practically independent of the heat flux density in the following measurement section. An increase of condensate rate, as in the case of film condensation, causes a fall of α. Reduction of the data taken, and shown in Fig. 4, indicates that in the condensation of vapor moving with velocity $W_v \leq$ 2 m/sec, an increase of Reynolds number in the range Re_f = 2—200 reduces the heat transfer coefficient in inverse proportion to the cube root of the value of the Reynolds number.

NOTATION

α, heat transfer coefficient, W/(m^2·K); q, heat flux density, W/m^2; T, t, temperature of the medium, deg K; W, velocity of the medium, m/s; L, length of tube, m; d, diameter of tube, m; δ, thickness, m; R, radius, m. The dimensionless groups: $Re_f = \bar{\ }\ell l/r\mu$. Subscripts: v, vapor flow; f, film of liquid; 0, initial value of a parameter; int, internal; Δ, differences; s, saturation; d, drop.

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SOLUTION OF THE INVERSE PROBLEM ON DETERMINING THREE FIBER COMPOSITE CHARACTERISTICS

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A method is proposed to determine the characteristics of a bonded composite: the fiber and matrix heat conductivity coefficients and the heat transfer coefficient between them, from the solution of the inverse problem of heat conduction.

Extended utilization of bonded composite materials evokes the necessity to investigate heat propagation processes in such media. From the thermophysical viewpoint, these materials are quite definitely of heterogeneous configuration [1]. The matrix can be considered homogeneous and isotropic while the bonding fiber in a beam or rod in structure is highly anisotropic. Under nonstationary heat transfer conditions, different thermophysical characteristics (TPC) of the material components specify their distinct temperature, that appears especially strongly in the composite surface layer [2, 3].

A multitemperature theory of heat conduction [4] has been developed to model heat transport processes in heterogeneous media. Taking the average of the temperature field over the section of each component results in a system of interrelated heat conduction equations that is closed by using the Henry law that sets up a connection between the thermal flux density between the components q_{ij} and their mean temperatures

$$q_{ij} = \alpha(\hat{T}_i - \hat{T}_j). \tag{1}$$

The practical lack of data about the TPC of the components and α hinders extensive utilization of the multitemperature theory.

When producing methods and apparatus to determine fiber and matrix TPC the tendency to raise the informativity [5] that is achieved by the development of fast-response, highly productive methods of complex nature that give information about a set of properties from one experiment should be taken into account. The possibility is examined in this paper, of determining the fiber and matrix heat conduction coefficients as well as the heat transfer coefficient between them from thermograms of a pulse experiment (the "laser burst" method).

The "burst" method was developed to determine the thermal diffusivity and specific heat coefficients of homogeneous materials [6] under the assumption that the TPC of the material are independent of the temperature. At this time it is utilized to determine the effective thermal diffusivity coefficients of definite classes of heterogeneous media [7]. Distinctive features of the method are the rapidity of executing an experiment and the accuracy that is associated with determining the relative and not the absolute quantities. The time behavior of the relative temperature

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