EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN EVAPORATION AND CONDENSATION PROCESSES ON CAPILLARY SURFACES

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The article describes experimental investigations of the evaporation and condensation of water and ethyl alcohol on wetted threadlike capillary surfaces of materials with different thermal conductivities.

Evaporation and condensation processes on wetted capillary surfaces with different microgeometry are widely used in modern power engineering and technology. A new kind of heat-transfer devices — heat pipes — is based on the application of these processes. The specifics of the accompanying physical phenomena and the practical importance of the corresponding technical apparatus are the reason for the interest in investigating the above-mentioned processes of phase transformations on capillary surfaces [1-10].

Among different kinds of surface capillary structures, the threadlike (a structure with parallel triangular grooves) structure is distinguished by a number of favorable characteristics. A thread with triangular profile provides the largest evaporation surface of all the known kinds of capillary structure. Because the width of the grooves decreases downward, the capillary hydraulic head of this structure increases noticeably in stressed thermal regimes. Besides that, as was shown previously [1], the threadlike structure, being integral with the heating surface, is very efficient from the point of view of utilizing the thermal conductivity of metal for intensifying the evaporation and condensation processes of ordinary (nonmetallic) liquids. Also important is the circumstance that this kind of surface is among the most highly technological capillary structures.

An analysis of the evaporation process from a wetted threadlike surface was provided by Ratiani et al. [1]. This article also presented an equation for the heat-transfer coefficient

$$\alpha = \frac{Q}{F\Delta T} = \frac{1}{h}\sqrt{\frac{\lambda_w \lambda_l}{\sin\Theta \operatorname{tg}\varphi}}^{\dagger}. \tag{1}$$

In dimensionless writing, (1) amounts to the equality with unit modified Nusselt number whose characteristic linear dimension is $h\sqrt{\sin\Theta\tan\varphi}$, and the coefficient of thermal conductivity is equal to the geometric mean of the coefficients of thermal conductivity of the metal and the liquid:

$$\mathrm{Nu}^* = 1. \tag{2}$$

On the basis of Eq. (1), the authors of [1] predicted the possibility of using the examined process for a substantial intensification of heat exchange in the evaporation or ordinary (nonmetallic) liquids. It may also be assumed that the model of the process given in [1] and the results of the analysis are fully applicable to the condensation processes on an analogous capillary surface as well.

Below we explain the results of experimental investigation of evaporation processes of thin films of water and ethyl alcohol and of filmwise condensation of their vapors on threadlike surfaces when the coefficients of thermal conductivity of their materials vary within fairly wide limits.

Technique and Methodology of Experimental Investigation. For investigating heat ex-

†In [1], this equation contained a printing error: sin φ should read tan φ.

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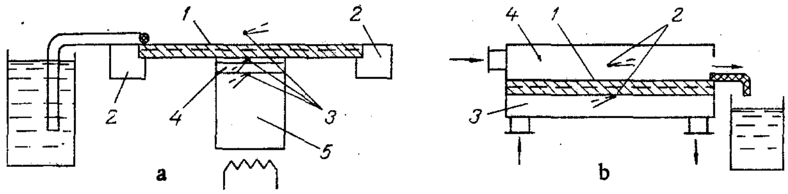

Fig. 1. Basic diagram of the chief units of the experimental devices: a) evaporation unit (1) experimental section; 2) current leads; 3) thermocouples; 4) thermal insulation; 5) copper rod with electric heater); b) condensation unit (1) experimental section; 2) thermocouples; 3) cooling jacket; 4) vapor section).

change in evaporation and condensation on threadlike surfaces, two separate experimental devices were made.

A basic diagram of the chief unit of the device for investigating the evaporation process is shown in Fig. 1a. The experimental section in the form of a reactangular plate with a system of longitudinal parallel grooves with triangular profile was mounted horizontally on current leads connected to the low-voltage side of a power transformer. Attached to the lower smooth side of the section was the end face of a measuring probe which contained the upper themocouple that was in contact with the lower surface of the section, a layer of heat insulation beneath it, the lower thermocouple under the layer, and a system for uniform compensation heating (in the form of a copper rod with an electric heater in its lower part).

The liquid, sucked up by the thread itself from a capillary braid connected with the vessel, evaporated from the upper capillary surface of the section. It was possible to observe the evaporation process visually. The level of the liquid in the vessel was set such that the loss of head for sucking up the liquid by the thread was negligible and that there was no negative meniscus in the groove. The measurements were carried out in steady-state regimes. The specific thermal flux of evaporation was determined by the parameters of the electric current flowing through the section. The temperature of the lower surface of the section was measured by the upper thermocouple of the probe. Before the measurement, the compensation heating was set to such a capacity that the readings of the upper and the lower thermocouple were the same. The temperature of the capillary surface, corresponding to the section at the apexes of the triangular recesses, was determined by converting the temperature of the lower surface. The vapor saturation temperature was measured by the thermocouple situated above the evaporation surface. The experiments were carried out at atmospheric pressure. From the measurement data, the heat-transfer coefficients in the evaporation of two different liquids on different threadlike surfaces were determined in a fairly wide range of changes of specific thermal loads.

The basic diagram of the chief unit of the device for investigating condensation processes is shown in Fig. 1b. Like in the case of evaporation, the experimental section was made in the form of a rectangular plate. The thermocouple junction was fixed to the lower, smooth surface of the section by spot welding or caulking. The electrodes of the thermocouple, leading from the junction were attached to that same surface at some distance away. On this same underside, the section was provided with a cooling jacket through which water from the thermostat was pumped. On the upper capillary surface the condensation process was effected. For this purpose, vapor from the evaporator was conducted into the upper section of the unit. The other end of the section was directly connected with the atmosphere. The power of the evaporator in the experiments was set in such a way that part of the discharged vapor always emerged from the section into the atmosphere. The condensate forming in the section was also conducted out of the section, and from its discharge the specific thermal load of the condensation surface was determined.

To prevent wet steam getting into the unit, the steam was superheated in the steam pipe to 1-2°K above the saturation temperature. The saturation temperature of the vapor was measured with the thermocouple situated above the experimental section (at the beginning and end of each experiment with the steam superheater switched off). It was possible to observe visually the condensation process on the thread. The experimental section was inclined at a certain angle which caused the condensate to be run down the grooves. This inclination was

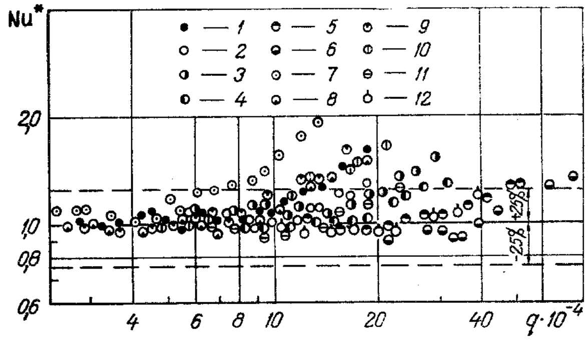

Fig. 2. Comparison of the experimental data on evaporation with Eq. (2): 1) water, surface No. 1; 2) water, No. 2; 3) water, No. 3; 4) water, No. 4; 5) water, No. 5; 6) water No. 6; 7) ethyl alcohol, No. 1; 8) ethyl alcohol, No. 2; 9) ethyl alcohol, No. 3; 10) ethyl alcohol, No. 4; 11) ethyl alcohol, No. 5; 12) ethyl alcohol, No. 6. $q \cdot 10^{-4}$, W/m².

selected in each experiment in such a way that flooding of the thread with condensate was prevented, as well as the formation of a negative meniscus inside the grooves. The temperature of the capillary surface, corresponding to the cross section through the apexes of the triangular recesses, was determined by conversion of the temperature of the lower surface measured by the thermocouple that was spot-welded from below (in the case of the copper section with the thermocouple caulked on from below, the temperature gradient in the wall was very small and was altogether disregarded). In some experiments, the thermal load, determined from the amount of condensed liquid, was checked with the aid of the thermal balance of the cooling water. The vapor pressure in the condensation section practically did not differ from the atmospheric pressure. Like in the case of evaporation, the experiments were conducted in steady-state regimes, and from the measurement data the heat-transfer coefficients were determined in different regimes on a number of threadlike surfaces. Wetting of all surfaces was ensured in accordance with the recommendations of [11] including the application of ultrasonic cleaning.

The experimental capillary surfaces were produced by cutting threads with the required profile in a cylindrical surface with a relatively large diameter (on the order of magnitude of 150 mm) on a lathe. A thin ring, 15 mm wide, was cut off the cylinder, cut open along a generatrix, and straightened out into a rectangular strip with longitudinal triangular grooves on one surface. Since the diameter of the original cylinder was sufficiently large, the thread was not deformed when the ring was straightened out. In the experiments with evaporation and condensation, sections were used that had been cut from the same strips. In all sections, a thread of triangular profile with apex angle (2φ) of 55° was cut. The surfaces were made from three different materials: stainless steel 08Kh18N10T, steel 20, and copper; this made it possible to carry out the experiments with the coefficients of thermal conductivity of the wall varying by a factor of almost 25. The threads were cut in two sizes: with a pitch of 0.3 and 0.5 mm. Altogether six surfaces were used: No. 1, stainless steel, pitch of the thread 0.5 mm; No. 2, stainless steel, pitch 0.3 mm; No. 3, steel 20, pitch 0.5 mm; No. 4, steel 20, pitch 0.3 mm; No. 5, copper, pitch 0.5 mm; No. 6, copper, pitch 0.3 mm. To increase the accuracy of comparing the experimental data with Eq. (2), the wetting angles of the surfaces of the used materials by the liquids were also determined. For each combination of a material with a liquid, the wetting angle was determined by the maximum height of the hanging liquid column maintained by a cylindrical hole of 0.52 mm diameter, and by the specific surface energy of the liquid according to handbook data. In all cases, the heat-transfer coefficients were determined for the main surface of the section in accordance with the left-hand part of Eq. (1). In the experiments, the maximum error of determining them amounted to 15%. In determining Nu*, the values of the wetting angles obtained by us were used. In the system water—steam—stainless steel, this angle was 52 ± 3°, in the system water—steam—steel 20 it was 45 ± 3°, in the system water—steam—copper it was 43 ±

3°. For ethyl alcohol on stainless steel this angle was 33 ± 5°, on steel 20 it was 30 ± 5°, on copper 38 ± 5°. The error of determining the modified Nusselt numbers from the experimental data did not exceed 25%.

Discussion of the Results. A comparison of the experimental results of evaporating water and ethyl alcohol on the mentioned six surfaces with Eq. (2) in coordinates $Nu^* = f(q)$ is presented in Fig. 2.

It follows from the figure that on each experimental curve there are two regimes of heat exchange: the regime of pure evaporation from the surface of the liquid (Nu^* = const) and the regime with boiling of the liquid in the grooves (Nu^* increases with increasing q). The heat-transfer coefficients in the regime of pure evaporation of the liquid changed within fairly wide limits in our experiments in dependence on the combination surface—liquid. In the case of evaporation of ethyl alcohol on surface No. 1, the heat-transfer coefficient in the mentioned regime was $6.5 \cdot 10^3$ W/m² · °K, in the evaporation of water on surface No. 6 it was $8.5 \cdot 10^4$ W/m² · °K. It follows from the comparison that the theory of [1] within the limits of the effected processes is fully confirmed by the experiments.

The obtained experimental data are also of certain interest from the point of view of the attained intensification of the heat transfer. Even if the heat-transfer coefficients are calculated according to the full area of the threadlike surface (which exceeds in our experiments the area of the main surface by a factor of 2.2), the maximally attained heat-transfer coefficient in pure evaporation of water amounts to $4.0 \cdot 10^4$ W/m² · °K. Considering that this magnitude does not increase with decreasing thermal load, it is easy to see that in the range of small thermal loads the intensity of the heat exchange in evaporation on a threadlike surface of highly heat-conducting material is one order of magnitude or more higher than the intensity of the heat exchange with a large volume of boiling water at the same pressure. Even with a fairly large load of $1 \cdot 10^5$ W/m², the increase in the heat-transfer coefficient amounts to four orders of magnitude. It is also obvious that the summary effect including the effect of developing the surface by the thread is much higher. The conclusion of [1], that the investigated process can be used for intensifying heat exchange, is consequently experimentally fully confirmed.

The presented experimental data also make it possible to determine the thermal loads at which the boiling process in the grooves begins to affect the intensity of the heat transfer. In the investigated processes these loads also change within fairly wide limits (from $0.5 \cdot 10^5$ W/m² in the evaporation of alcohol on surface No. 1 to $0.5 \cdot 10^6$ W/m² in the evaporation of water on surface No. 6). The regimes of heat exchange obeying the regularities revealed in [1] consequently encompass a fairly wide and practically important range of thermal loads.

A comparison of the experimental results on the evaporation of water and ethyl alcohol vapors with Eq. (2) is presented in Fig. 3. As was to be expected, in the case of condensation, the intensity of the heat transfer is maintained constant in the entire investigated range of thermal loads. The deviation of the experimental data from Eq. (2) does not exceed the error of determining Nu^* from the experimental data. Thus, Eqs. (1) and (2) were confirmed in a fairly wide range of changes of the parameters contained in them, and they may be recommended for calculating the resepctive evaporation and condensation processes.

From the point of view of the obtained results, it is of certain interest to examine the known data on heat exchange during condensation on finely ribbed capillary surfaces [6-10]. With great confidence it may be assumed that the main qualitative peculiarity revealed in [1], viz., the great role of thermal conductivity of the metallic component of the capillary liquid—metal layer in the intensity of the heat transfer, is also characteristic for this process of condensation. In the experiments carried out, the best agreement was found with the model of the process according to [1] with condensation on horizontal pipes with transverse trapezoidal grooves which are very similar to the profile with triangular thread [9, 10]. The interest in the described experiments is also due to the circumstance that they were intended to study surfaces that differed strongly in heat-transfer coefficients (copper and stainless steel) and thermal conductivity (water and Freon-21). A comparison of these experimental data with Eq. (2) is presented in Fig. 4.

In the comparison we used the data of [9, 10] on the height of the protrusions of the profile and the slope of the lateral surfaces of the grooves. The wetting angle of the surfaces by Freon-21 was taken as 40° (the error of determining this value, equal to ± 10°, leads in the given range of angles to an error of ± 10% in determining Nu^*). For water we used the

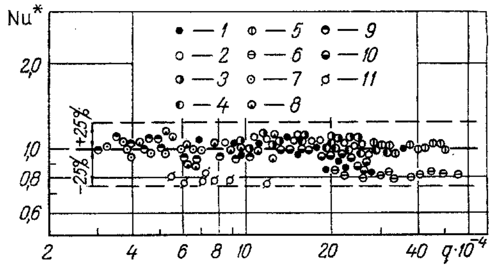

Fig. 3

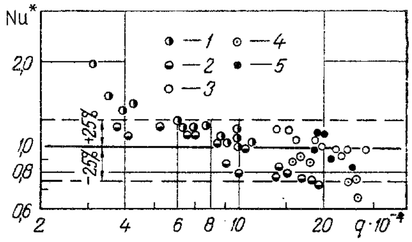

Fig. 4

Fig. 3. Comparison of the experimental data on condensation with Eq. (2): 1) water, surface No. 1; 2) water, surface No. 2; 3) water, surface No. 3; 4) water, surface No. 4; 5) water, surface No. 5; 6) water, surface No. 6; 7) ethyl alcohol, surface No. 1; 8) ethyl alcohol, surface No. 2; 9) ethyl alcohol, surface No. 3; 10) ethyl alcohol, surface No. 5; 11) ethyl alcohol, surface No. 6.

Fig. 4. Comparison of the experimental data of [9, 10] on condensation with Eq. (2): 1) Freon-21, copper, pipe No. 2; 2) Freon-21, copper, pipe No. 3; 3) water, stainless steel, pipe No. 2; 4) water, stainless steel, pipe No. 3; 5) water, stainless steel, pipe No. 4.

values obtained by ourselves. The results of the comparison show that the bulk of the experimental data agrees satisfactorily with Eq. (2). On the other hand, in analyzing the results, the following circumstance must be borne in mind: in the general case in condensation on a horizontal pipe, two deviations from the scheme of model [1] may occur: a negative meniscus in the groove in the upper part of the pipe under the effect of a liquid column, causing increased intensification of the heat exchange, and flooding of the grooves with condensate in the lower part of the pipe leading to decreased intensity of the heat exchange. It is obvious that the former process predominates at low thermal loads, and the latter at high thermal loads; this can be clearly seen from the disposition of the experimental dots. The comparison as a whole enables us to speak not only of qualitative, but in the first approximation also of quantitative agreement between the experimental data of [9, 10] and the results of the analysis in [1]. It may be assumed that the article [1] could be the basis for devising a more accurate theory of the condensation process of fine-ribbed horizontal pipes. On the other hand, it should be pointed out that unfortunately the role of thermal conductivity of the metallic component of the capillary layer in the intensity of heat transfer has been ignored for a long time by researchers investigating the condensation process on fine-ribbed pipes. An exception was [10] where low absolute values of the heat-transfer coefficient obtained in experiments were perfectly convincingly explained by the low thermal conductivity of the material of the surfaces used.

NOTATION

h, height of the thread; φ, half the apex angle of the thread; Θ, wetting angle of the material of the surface by the liquid; Q, total amount of heat transferred per unit time; F, area of main surface (area of the projection of the thread surface); q, specific heat flux over the main surface; T_s, saturation temperature; T_w, wall temperature corresponding to the cross section through the vertices of the triangular recesses; ΔT, difference between T_s and T_w; α, heat-transfer coefficient; λ_w, coefficient of thermal conductivity of the material of the surface; λ_l, coefficient of thermal conductivity of the liquid; $Nu^* = \alpha h \sqrt{\sin\Theta \tan\varphi}/\sqrt{\lambda_w \lambda_l}$, modified Nusselt number.

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MASS TRANSFER FROM THE WALL TO THE STREAM OF AN AXISYMMETRIC FLUID JET

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The results of measurements are presented pertaining to friction and mass transfer during interaction of an axisymmetric vertical one-phase or two-phase fluid jet and a horizontal plane wall.

In technological processes of surface-chemical treatment of materials, one now uses more extensively progressive methods where the active fluid is force fed to the surface, particularly in the form of jets. The laws governing this method of treatment as, e.g., in dimensional etching of metallic surfaces are, however, little known.

The hydrodynamics and the mass transfer in the case of a single axisymmetric one-phase fluid jet impinging on a plane barrier have already been theoretically analyzed in great detail [1, 2]. Theoretical relations have been derived [2] for determining the local hydrodynamic frictional stresses τ in all principal flow regions in this setup and calculating the mass-transfer coefficient β in the region of laminar flow. In the same study [2], the theoretical relations were verified experimentally by the electrodiffusion method. Unlike the flow pattern in [1, 2], real processes of metal dissolution include also an evolution of hydrogen in the form of bubbles so that the fluid medium becomes a two-phase system and, furthermore, solutions are often fed to the treated surface from below. Under these additional conditions, i.e., in the presence of a gaseous phase subject to forces of gravity in the stream these factors do not influence the hydrodynamics and the mass transfer in an obvious manner. Meanwhile, however, it is well known that in some flow patterns such as the one studied by other authors [3], introduction of a gaseous phase in the form of bubbles contributes to an appreciable intensification of the mass-transfer processes.

In this study the authors have experimentally verified the validity of theoretical conclusions arrived at earlier [2] in the case of a fluid jet impinging on a wall from below, and at the same time examined the effect of a gaseous phase on the hydrodynamics and the mass-transfer processes.

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