Journal of Engineering Physics and Thermophysics, Vol. 74, No. 3, 2001

HEAT TRANSFER FROM A SURFACE WITH SPHERICAL HOLLOWS IN BOILING OF WATER AND A STEAM-WATER MIXTURE IN THE SUPERCRITICAL REGION

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UDC 536.24

The authors carried out an experimental investigation of heat transfer in the supercritical zone in flow of water and a steam-water mixture in a vertical annular channel. The interior wall heated by the current was either smooth or with hollows stamped in a straight-line order. The application of them increased the heat-transfer coefficient by a factor of 1.5–2 or more. The increase in the frictional resistance was much smaller.

Introduction. Boiling of a liquid on a heated surface is a process characterized by the intense removal of heat, which is widely used for efficient cooling of the walls of various apparatuses. In heat-power engineering and a number of other technologies, this process is used for generation of steam or evaporation of solutions. In all these cases, the limiting heat flux is the critical flux where either the transition from the nucleate regime of boiling to a film regime (burnout of the first kind) occurs or, in the angular-dispersed regime of flow of a steam-water mixture about the wall, the film boiling on it dries out due to the insufficient intensity of wetting of the wall by droplets flying in the flow (burnout of the second kind).

In the two cases, burnout is accompanied by a sharp increase in the wall temperature: a larger one in burnouts of the first kind and one markedly smaller in burnouts of the second kind. In the first case, tubes, even those manufactured of stainless steel or high-temperature nickel-chromium alloys, fail by a loss in strength or burn through, while in the second case, at high pressures, a region exists where the wall temperature does not exceed the permissible one (about 900 K). It is precisely under these conditions of supercritical heat exchange that the transition zones of single-pass boilers operate [1]; in these zones, the evaporation of the last 10% of water droplets and transition to the superheating of a steam occur. In order to restrict the temperature increase, the transition zone in single-pass boilers is removed to a convective gas duct where the heat fluxes are much smaller than in a furnace. The practical necessity of operation in the supercritical region stimulated a special study of heat exchange under these conditions. The first systematic investigations of the heat transfer to a steam-water mixture and a steam in the supercritical region of operation of steam-generating channels go back to the beginning of the 60s when Z. L. Miropol'skii [2] investigated at the G. M. Krzhizhanovskii Institute of Power Engineering the temperature regimes of steam-generating tubes 8–29 mm in diameter throughout the entire range of relative enthalpies of the heat-transfer agent X: from a subcooled water ($X < 0$) to a superheated steam ($X > 1$). The experiments were conducted for pressures P = 4–21.7 MPa and mass velocities of 200–1400 kg/(m^2·sec). Subsequently, analogous investigations were carried out at other scientific-research organizations both in this country and abroad. Their results are generalized in the form of "skeleton tables" in [3].

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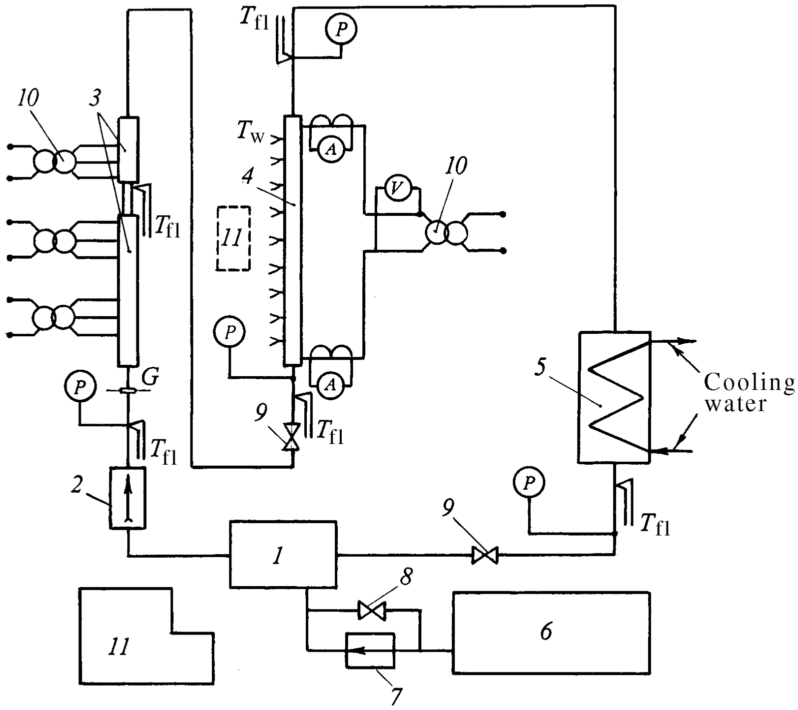

Fig. 1. Schematic diagram of a steam-generating bench.

Analysis of the physics of processes which accompany burnout of the second kind [4] shows, in particular, that the liquid film does not terminate instantaneously and sharply and its boundary alternately executes a downstream movement and returns. Thus, the readings of a thermocouple installed at a concrete site vary within wide limits from a value close to the saturation temperature (nucleate boiling in a liquid film) to a maximum value which corresponds to a situation where the removal of heat is provided by the flow of a vapor phase, without droplet wetting. In the interval, we observe partial, to a degree, removal of heat by droplets incident onto the wall and evaporating instantaneously (or recoiling from it if the temperature of the wall is above the Leidenfrost temperature).

The instantaneous local values of the heat-transfer coefficients calculated from the instantaneous readings of the thermocouple vary within 2 orders of magnitude: from hundreds of $kW/(m^2 \cdot K)$ to unities. As the heat flux increases, the zone of burnout proper shifts upstream and the supercritical regime with a stable temperature is established at this site. The values of α characteristic of the supercritical zone are 1 $kW/(m^2 \cdot K)$ for the parameters of the present work.

Highly diverse intensifiers are used with the aim of increasing the intensity of heat transfer to the heat-transfer agent, including the cases in boiling. However, the increase in heat transfer in channels is accompanied, as a rule, by a stronger increase in hydraulic resistance, often making it unprofitable economically. In the 80s, one applied for this purpose spherical recesses ("hollows") in which tornadolike flows occur in the case of turbulent flow along the surface (when $Re > 10^4$). The tornadolike flows, flowing actively out of the hollows, break down the wall layer, disturbing the known Reynolds analogy between heat exchange and momentum exchange. As a result, the increase in hydraulic resistance markedly lags behind the increase in heat transfer and sometimes is simply absent. Hollows and effects which accompany them are described in [5] in detail.

It was shown experimentally that the lower the intensity of heat transfer from a smooth surface, the more pronounced the effect of hollows. Under the conditions of strongly developed turbulence with large and high-power vortices, the effect from tornado transfer is reduced. That is why, for example, in developed boiling the intensification of the heat transfer by hollows must be insignificant. This is confirmed by experiment.

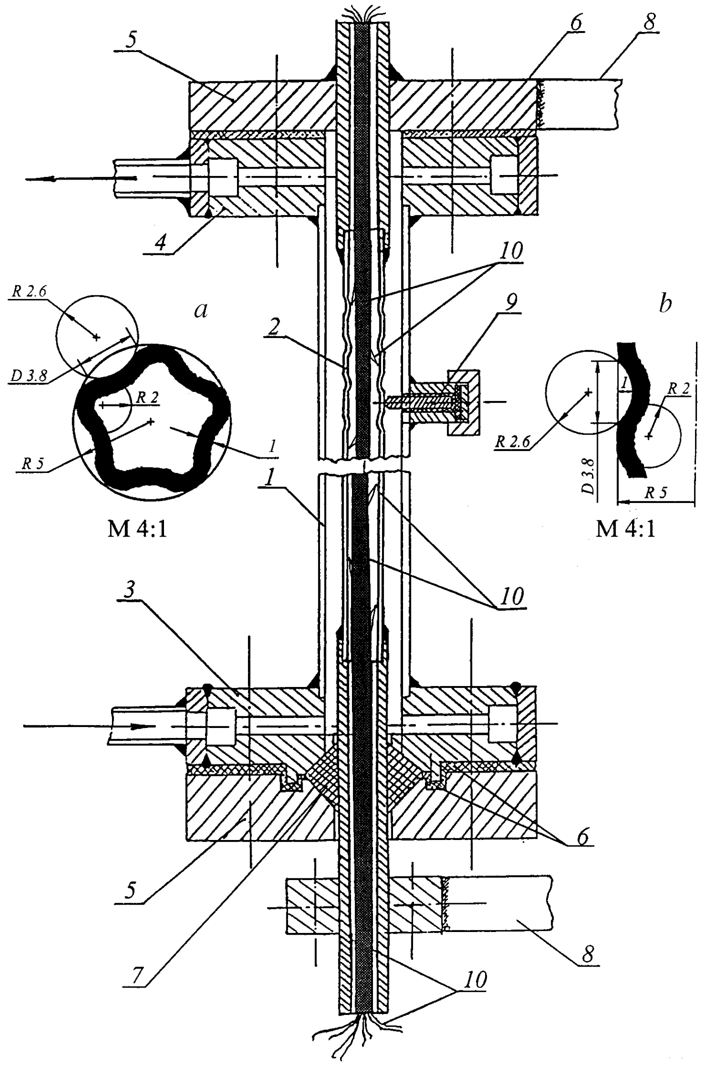

Fig. 2. Scheme of an annular experimental area.

The magnitude of the critical heat fluxes increased by only 20–50% [6]. More suitable conditions for the manifestation of the "effect of hollows" are available in the supercritical region. Here tornadolike structures self-generated in the hollows must break down a steam boundary layer, eject the steam mass into the flow, and in place of the mass draw in the heat-transfer agent, containing droplets or at least a substantially colder steam, from the steam-water flow. This was confirmed already by the first experiments, which were conducted at the Institute of High Temperatures of the Russian Academy of Sciences in the first half of 1999 [7] and showed a twofold (or more) increase in the heat-transfer coefficients for pressures P = 17.7 and 21.7 MPa and a mass velocity of ρw = 300–500 kg/(m^2·sec) in a wide range of X.

New experimental data are given below. Over an almost one-year period, we manufactured an experimental area and conducted experiments on it for pressures of 17.7, 19.7, and 21.7 MPa and mass velocities of 350 and 230 kg/(m^2·sec) in a wide, up to a superheated steam, range of relative steam fractions. Furthermore, we carried out comparative hydraulic tests of annular channels with all three types of internal tubes (including a smooth one).

Experimental Setup and Experimental Procedure. The experiments have been carried out on a bench available at the Institute of High Temperatures of the Russian Academy of Sciences and somewhat modernized by us. The heat output of the bench is 500 MW; the heat-transfer agent is desalinated water, a steam-water mixture, and superheated steam. The limiting parameters are as follows: pressure 25 MPa and

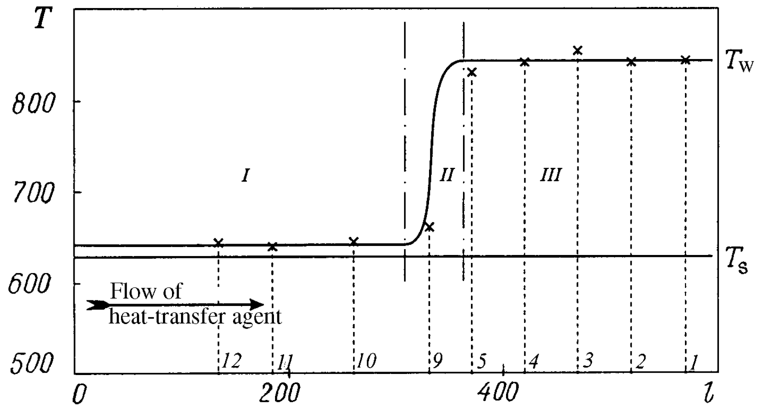

Fig. 3. Change in the wall temperature of a steam-generating tube with burnout in its middle part and with the subsequent supercritical region. $P = 17.7$ MPa; $\rho w = 430$ kg/(m^2·sec); $q = 322$ kW/m^2; $X_{inl} = 0.01$. T, K; l, mm.

temperature 780 K. The bench is described in [7] in detail. Here we give a schematic diagram of the modernized bench and describe the main details required for understanding the procedure of the investigations.

The schematic diagram of the bench is presented in Fig. 1. The water from a feed vessel 1 is supplied, by a circulating pump 2, to a straight-through three-section heater – a steam generator 3. Heated water, a steam-water mixture, or superheated vapor arrive, from below, at the vertical experimental area 4 and next at the condenser-cooler 5. The cooled condensate is discharged into the feed vessel 1. Makeup with a chemically desalinated water is carried out to the vessel 1 from a vessel 6 by a makeup pump 7. Discharge of the water from the circuit is effected through a valve 8. The pressure, temperature, and flow rate of the heat-transfer agent are measured and recorded by standard methods. The pressure of the heat-transfer agent on the experimental area and its flow rate are controlled using valves 9.

The prescribed enthalpy of the heat-transfer agent at the inlet to the experimental area is provided using the heater – the steam generator 3 – in which the heating and partial evaporation of the water are carried out by direct resistance heating of the tubes by a low-voltage alternating current from transformers 10. The working tube of the experimental area is heated in the same manner. The supplied electric power is monitored by precision ammeters and voltmeters. The readings of all instruments are recorded by a system of data collection and processing 11.

The experimental area (Fig. 2) is an annular channel with a constant smooth external tube 1 18/14 mm in diameter and a changeable internal tube of diameter 10/8 mm. The changeable tube 2 was manufactured in several versions: smooth and with applied systems of spherical hollows. In this case, use was made of internal tubes with systems of hollows with their straight-line arrangement with a longitudinal step of 6 mm and a transverse step of 6.3 mm (over the exterior surface). The hollows were made by impressing bearing balls 4.8 mm in diameter; 1-mm-deep hollows with an outside diameter of 4 mm with rounded edges (the edge radius is 2 mm) were produced. The longitudinal and cross-sectional views of the wall of the internal tube (the section is through the center of the hollows) are given in Fig. 4. Flanges 3 and 4 with pipes for supply and removal of the heat-transfer agent are welded to the external tube. The corresponding flanges 5, using spacers 6 and stuffing box seal 7, provide the air-tightness and heat insulation of the external tube from the internal one at which electric energy from copper current leads 8 arrives. In order to avoid contact of the external and internal tubes in the annular channel, on the external tube we mounted spacer heat-insulated pins 9 that fix the internal tube on three sides (in 120^o) in three cross sections along its length. Provision is made for changing the central tube both in the case of its failure in burnout and in replacement by a tube of another structure.

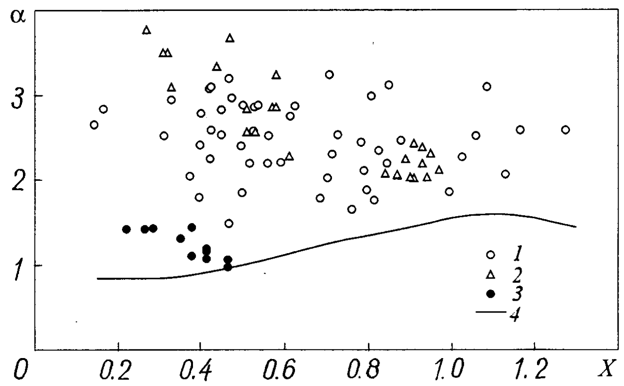

Fig. 4. Comparison of the coefficient of heat transfer α from tubes with hollows with the values of α from smooth tubes (P = 17.7 MPa; ρw = 350 kg/(m^2·sec); q = 270–410 kW/m^2): 1) tube with hollows throughout the length (l_h = 590 mm); 2) the same in the outlet part (l_h = 250 mm) [7]; 3) smooth tube [7]; 4) the same [3].

The temperature of the wall of the heated internal tube was measured by nine Chromel–Copel thermocouples inserted into the internal cavity of the tube in the form of two rigid bunched conductors 10 through its open upper and lower ends. The pressing of the thermojunctions to the tube wall is ensured from inside by the intrinsic elasticity of short (up to 10 mm) segments of thermocouple wires (0.3 mm in diameter) coming out from the bunched conductor. Thus, we measured the temperature of the interior surface of the heated tube T_w^{int}.

Upon application of the hollows to the exterior surface of the internal tube, this surface increased by approximately 10%. The volume of the slot in its part covered with hollows increased by nearly the same magnitude, i.e., the averaged slot width was to increase to 2.2 mm. However, as is usually done in the case of employment of other methods for intensification of heat exchange, in calculations of the flow velocity, heat fluxes, and heat-transfer coefficients, these changes are not taken into account, i.e., all refer to the initial size.

In the previous series of experiments [7], in addition to a smooth tube, we used a tube in which hollows were applied on its 250-mm-long inlet portion; on the 340-mm-long preceding portion the tube remained smooth. In the new experiments, the object of investigation was an internal tube covered with the same system of hollows but now throughout its length, i.e., 590 mm. Since for pressures of 17.7 and 21.7 MPa covered by the previous series of experiments the character of burnouts differed somewhat, it seemed of interest to conduct the same experiments with ρw = 350 kg/(m^2·sec) for an intermediate pressure of 19.7 MPa.

Finally, data on the influence of hollows on the intensity of heat transfer for other mass velocities are of undeniable interest. Therefore, we conducted experiments for ρw = 230 kg/(m^2·sec). The estimative calculations showed that for this mass velocity, too, the Reynolds numbers remained higher than 10^4 for the entire range of P and X in question, i.e., the restrictions on the region of efficient operation of the hollows were not obeyed.

The experimental procedure was standard. We increased pressure in the circuit and set the prescribed flow rate of the heat-transfer agent. The electric energy required for obtaining the prescribed enthalpy at the inlet to the experimental area was supplied to the steam generator, after which the current was supplied to it. The thermal load on the experimental area was increased in steps up to the point of attainment of the critical heat flux and subsequent passage to the supercritical region where the basic calculations of P, q, and T_w and also of ρw were done. And in the same manner up to a wall temperature T_w^{perm}, the maximum permissible for the tube. We adopted in the experiments that T_w^{perm} = 850 K. Then the load on the experimental area was decreased until the regime of boiling was converted to a nucleate one; one of the parameters (P, q, or X_{inl}) was changed, and the experiment was repeated.

Upon passage to the lower mass velocity ρw = 230 kg/(m^2·sec), the absolute error in determining X increased: for P = 21.7 MPa it attained ±0.08, while for P = 17.7 MPa it was markedly lower – it attained ±0.03; the absolute error of measuring the wall temperature was ±2 K. The relative error of determining the heat-flux density is estimated at ±5%.

The temperature curve typical of the experiments conducted is shown in Fig. 3. The length of the annular channel with indication of the fixed sites of installation of the thermocouples that measure the temperature of the interior wall of the heated tube is plotted on the X axis, while the temperature T is plotted on the Y axis. Nucleate boiling with a wall temperature similar to the boiling point of water T_s is retained on the inlet portion I. Next is the transition region II of the burnout proper with a sharp increase in the wall temperature and with usually unstable readings of the thermocouples. Finally comes the high-temperature supercritical region III. In the supercritical region, T_w pulsations are practically absent. With increase in q, the transition and supercritical regions shift to the left toward the flow of the heat-transfer agent.

In calculating the coefficients of heat transfer to the steam-water mixture in the supercritical region, we processed just the points known to be lying in region III (Fig. 3). These are the highest temperatures T_w lying on the upper plateau. They correspond to the lowest stable values of the heat-transfer coefficients, which are of the greatest practical interest.

The sought quantity – the coefficient of heat transfer from the exterior surface of the internal tube of the annular channel to a boiling heat-transfer agent – is determined as

$$\alpha = q/(T_w^{ext} - T_s) , \tag{1}$$

where T_w^{ext} is the temperature of the exterior surface of the heated wall, K and T_s is the boiling point of the heat-transfer agent. In the case of operation in the region of superheated steam, one uses, instead of T_s, the temperature of this steam that corresponds to the mass-mean enthalpy of the flux. Since (see Fig. 3) the longitudinal temperature gradient in the supercritical region is small, we disregard longitudinal spreadings of heat over both the tube and the thermocouple electrodes. The temperature difference across the wall thickness ΔT_w of the tube heated by the internal heat sources (ohmic heating) is calculated from the existing formulas and is subtracted from the temperature of the interior surface of the wall measured by the thermocouples: $T_w^{ext} = T_w^{int} - \Delta T_w$. In the investigated range of heat fluxes q, the magnitude of ΔT_w did not exceed 15 K. Thus, its computational error even at the level of 10% could not have an appreciable effect on the error of determining α (estimated at ±10%) in the supercritical region with characteristic temperature differences of the order of 100 K.

The reading of each thermocouple correlates with the local value of the relative enthalpy (steam quality) determined with allowance for an increase in the enthalpy along the length of the experimental tube:

$$X = X_{inl} + \frac{\pi q d l_i \eta}{Gr} , \tag{2}$$

where X_{inl} is the dimensionless relative enthalpy (steam quality) at the inlet to the heated channel, q is the density of the heat flux from the exterior surface of the internal tube, kW/m^2, l_i is the distance from the inlet to the heated channel to the given cross section (site of installation of the ith thermocouple), m, and $\eta \approx 1$ is for the annular channel with an intensely heated internal tube.

Results of the Experiment. As has already been mentioned, the main part of the experiments in this series is conducted on a tube with hollows applied over its entire surface. Figure 3 presents the characteristic temperature curve that reflects the instant at which the burnout occurred at the end of the tube shifted up-

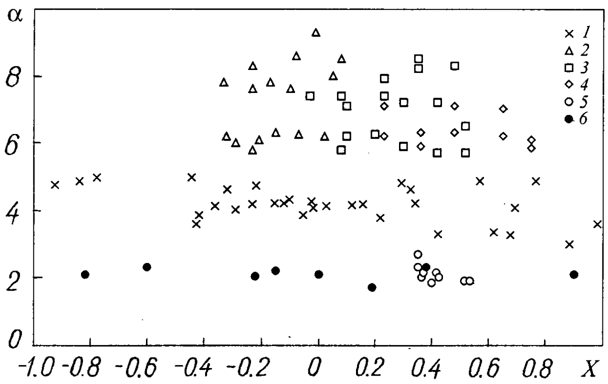

Fig. 5. Comparison of the coefficient of heat transfer α from tubes with hollows with the values of α from smooth tubes (P = 21.7 MPa; ρw = 350 kg/(m^2·sec)): 1) tube with l_h = 590 mm (q = 411–762 kW/m^2; ρw = 370–380 kg/(m^2·sec)); 2) tube with l_h = 250 mm [7] (350–480 and 330–345); 3) the same (550–630 and 330–345); 4) the same (q = 722 and 330–345); 5) smooth tube [7] (318–370 and 330–345); 6) the same [8] (220–300 and 400–480).

stream after several successive steps of increase in q. The curve corresponds to the following actual experimental conditions: P = 17.7 MPa, ρw = 430 kg/(m^2·sec), X_{inl} = 0.01, and q = 322 kW/m^2. Since the present work seeks to investigate heat transfer in the supercritical region *III*, we determined the local coefficients of heat transfer and the corresponding values of q, ρw, and X for five points. The results of other experiments were processed in a similar manner.

The data obtained are presented in Figs. 4–6 in the form of the dependence α = $f(X)$ for P and ρw as parameters. Figure 4 gives results of the present investigation 1 (in the range X = 0.10–1.30). The same figure shows the data [7] obtained on a combined tube in the part with hollows 2 and on a totally smooth tube 3. Although the latter correspond to a somewhat higher mass velocity ρw = 510 kg/(m^2·sec), the corresponding values of α turn out to be 1.5–2 times lower than for the tube with hollows. Nearly the same difference is observed in comparing the values of α 1 and 2 obtained by the authors with the data 4 of the skeleton table in [3].

Figure 5 gives, for P = 21.7 MPa, the values of α of the present investigation 1 and of the previous one [7] (points 2, 3, and 4)[*)] and also the data for a smooth tube from [7] 5 and the values of α from [8] 6. Unfortunately, in the skeleton table [3] there are no data on heat transfer in the supercritical region for P = 22 MPa. The values of α 1 lie nearly 25% lower than the similar values (2, 3, and 4) from [7] but nonetheless exceed at least twofold the values of α from smooth tubes [7, 8] extended as a narrow band at the level 2 kW/(m^2· K). Figure 6 gives experimental values of α for the pressure P = 19.7 MPa for which investigations in [7] were not conducted. These data are obtained for two mass velocities ρw = 350 and 230 kg/(m^{2}sec). Points 1, which correspond to the higher velocity, are located nearly 20% higher than points 2, which suggests the influence of the velocity on the heat transfer in the case of flow along hollows, somewhat stronger than in the case of flow along a smooth surface. The latter is demonstrated by curves 3 and 4 constructed by interpolation of the data of the skeleton tables in [3] to mass velocities of 350 and 230 kg/(m^2·sec), respectively, and running close to each other in a wide range of X. The experimental values in

*) Since the experiments [7] for P = 21.7 MPa covered the widest range of variation of q – from 350 to 722 kW/m^2 – by subdivision of the points into three ranges in q (points 2, 3, and 4) we show a weak influence of q on α that makes it possible to disregard it in analyzing the data obtained and demonstrating them in the figures of the present paper.

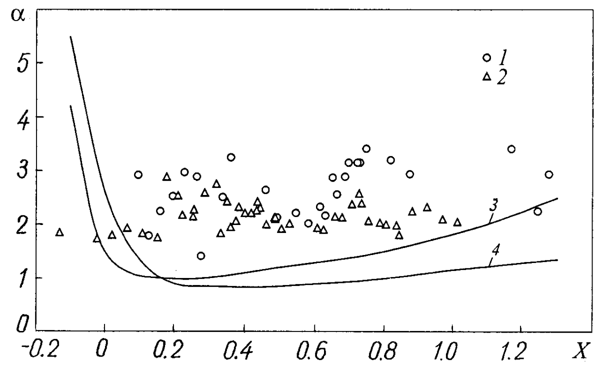

Fig. 6. Comparison of the coefficient of heat transfer α from a tube with hollows throughout the length (l_h = 590 mm) and from smooth tubes (P = 19.7 MPa; q = 250–400 kW/m^2): 1) tube with l_h = 590 mm (ρw = 310–390 kg/(m^2·sec)); 2) the same (202–270); 3) smooth tube [3] (ρw = 350 kg/(m^2·sec)); 4) the same (230).

the coordinates $\alpha = f(X)$ lie practically parallel to the X axis, without corresponding to the slight slope of curve 3 in the zone of X from 0.4 to 1.3 and less so to a sharp increase in the values of α from the skeleton tables for $X < 0.1$.

On the main portion of X from 0.1 to 1.0 we observed a substantial (twofold or more) increase in the heat-transfer coefficients due to the effect of hollows. This agrees well with the experimental data obtained for pressures of 17.7 and 21.7 MPa.

Having no way, for such high pressures and temperatures, of not only studying the character of the flow of boiling water in hollows and their neighborhood but of at least measuring hydraulic resistance under experimental conditions, the authors compared the hydraulic characteristics of the annular experimental areas by blowing them with air under room conditions. The experimental area in the full assembly was installed strictly horizontally, and compressed air at nearly atmospheric pressures was fed to the inlet. The presence of four pressure taps along the length of the annular area made it possible to measure the pressure differences ΔP on two portions 150 mm in length each. The pressure, temperature, and flow rates of the air were also determined. We tested experimental areas with the installed smooth tube, tube covered with hollows throughout the length, and a "combined" tube, which was used in [7]. In the latter case, under practically the same conditions, we measured simultaneously the hydraulic resistances of the smooth portion and the portion covered with hollows: first a 120-mm stabilization portion was followed by the 150-mm measuring portion with a smooth tube, next was a stabilization portion without determination of ΔP, and then again the 150-mm measuring portion with hollows.

The experiments were conducted in the ranges of Reynolds numbers from 10^4 to 10^5. The coefficients of friction ξ were determined from the formula

$$\Delta P = \xi \frac{l}{d_e} \frac{\rho w^2}{2}, \tag{3}$$

where ΔP is the pressure difference on the investigated portion of the annular channel, Pa, l and d_e are the length of the portion in question and the equivalent diameter of the annular channel, m, and ρ and w are the density and velocity of the air under experimental conditions, kg/m^3 and m/sec, respectively.

Under the above assumptions that, in the calculations, the surface of flow and the equivalent diameter of the tubes with hollows are taken to be the same as in the initial smooth tube, we determined the values of ξ for channels with internal tubes of the types enumerated above. Results are presented in Fig. 7 in the form

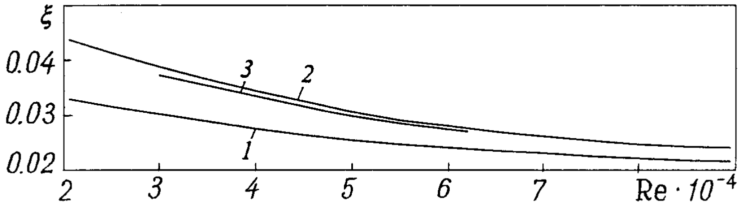

Fig. 7. Coefficients of frictional resistance ξ for annular channels with a smooth internal tube and a tube covered with hollows in the case of air flow in them ($P \approx 0.1$ MPa; $T \approx 290$ K).

of the dependence $\xi = f(\mathrm{Re})$. As could be expected, the values of the coefficient of friction ξ obtained in the experiments with the annular channel with a smooth tube and with a combined tube on its smooth part lay together and the averaging line for them is curve 1. The resistance of the annular channel with the internal tube covered with hollows throughout the length (curve 2) turned out to be somewhat higher than in the previous case: by nearly 30% for Re = $3 \cdot 10^4$ and by 12% for Re = $8 \cdot 10^4$. Almost the same values of ξ are obtained in the experiments with a combined channel on the portion with a tube covered with hollows (curve 3). This is in good agreement with the statements [5] that in the case of flow along the surface with hollows the increase in the frictional resistance lags behind the rate of increase in the heat transfer.

Conclusions. Investigations of heat transfer in the supercritical region carried out on the experimental area with an annular channel and a heated internal tube whose surface was covered with a system of spherical hollows throughout the slot length showed that:

(1) for pressures of 17.7, 19.7, and 21.7 MPa and a mass velocity of 350 kg/(m^2·sec), the presence of the hollows leads to an increase of 1.5–2 times or more in the local coefficients of heat transfer;

(2) for a pressure of 19.7 MPa, the decrease in ρw to 200 kg/(m^2·sec) led to a certain decrease in the absolute values of α, but their excess over the values of α for smooth tubes remained, as previously, at the same level (1.5–2 times).

(3) the increase in the length of the tube covered with a system of hollows from 250 to 590 mm did not change the values of q for P = 17.7 MPa but led to a decrease of 25% in α for P = 21.7 MPa, thus reducing the excess over the value of q for smooth tubes from 2.6 times [7] to twofold.

Hydraulic tests of the experimental area with a smooth tube and tubes with hollows on a length of 250 and 590 mm in blowing it with cold air showed a relatively slight increase in the frictional-resistance growth by the hollows as compared to the increase predicted by a thermohydraulic analogy, which corresponds to [5, 6].

This work was carried out with support from the Russian Foundation for Basic Research, grant No. 98-02-17323.

NOTATION

d, outside diameter of the internal tube of the experimental area or diameter of the tube in the works of other authors, m; f, function; G, mass flow rate of the heat-transfer agent, kg/sec; l, length of the portion of the channel (tube), m; P, pressure, MPa; ΔP, pressure difference on the portion of the annular channel, MPa; q, heat-flux density, kW/m^2; r, heat of vaporization, kJ/kg; T, temperature, K; X, dimensionless relative enthalpy (for $0 < X < 1$, steam quality); α, heat-transfer coefficient, kW/(m^2·K); ρ, density of the heat-transfer agent, kg/m^3; ρw, mass velocity, kg/(m^2·sec); η, dimensionless coefficient that allows for heat losses; ξ, dimensionless coefficient of friction. Superscripts: ext and int, exterior and interior surface of the heated tube; perm, permissible. Subscripts: inl, inlet; cr, critical; h, hollows; fl, flow of the heat-transfer agent; w, wall, e, equivalent; i, ordinal number; s, saturation.

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