INTENSIFICATION OF HEAT TRANSFER IN THE CONDENSATION OF WATER VAPOR ON HORIZONTAL TUBES WITH ANNULAR GROOVES

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Results are presented from an experimental study of the mean heat-transfer coefficient in the film condensation of water vapor on the external surface of tubes with transverse periodic grooves of smooth configuration.

A reduction in the weight and size of condensers used in power engineering, the chemicals industry, and other areas of technology is an important goal from the viewpoint of both science and engineering and the national economy. The most promising method of attaining this goal is to intensify heat transfer in condensers.

In tubular heat exchangers of power plants, condensation is usually observed on the outside of the tubes. Several methods are known for intensifying heat transfer on the side of the condensing vapor: the inducement of dropwise condensation by the use of hydrophobic coatings or liquid stimulators, the use of different types of finned tubes, vibration of the heat-transfer surface, the use of an inclined tube bundle, and the development of efficient schemes of air extraction. Dropwise condensation is one of the most promising methods, since it makes it possible to increase heat transfer by a factor of 5-10 compared to film condensation. Investigators have already discovered a large number of chemical compounds (organosilicon liquids, machine lubricating oil, kerosine) whose addition to a heat carrier stimulates dropwise condensation, but their effect lasts no more than several hundred hours. Coatings made of fluoroplastic, organosilicon and phenolformaldehyde resins, and Teflon can serve as hydrophobic coatings. The tasks facing researchers is to find new stimulators and coatings that will ensure long, stable operation of equipment at low additional cost. Vibration of the heat-transfer surface and the inclination of tube bundles do not result in significant intensification of heat transfer.

There are various designs of heat-exchanger tubes for intensifying heat transfer with film condensation on their external surface [1]. This approach basically amounts to knurling annular or helical projections on the surface welding longitudinal or transverse wire fins to the external surface, or cutting threads on this surface. The shortcomings of these methods are as follows: additional consumption of metal in the fins; the technical complexity of manufacturing the tubes; the absence of intensification of heat transfer inside the tubes with the flow of coolant water within the tubes. In the best case, the heat-transfer coefficient is increased by a factor of 1.4-1.6. If we consider that the thermal resistances outside and inside the tubes for condensers are comparable — and that the internal thermal resistance is sometimes higher than the external — then it becomes obvious that only a design of heat-exchanger tube that would ensure intensification of heat transfer on both surfaces of the tube would be efficient.

The Moscow Aviation Institute has developed an effective method of intensifying heat transfer in tubular heat exchangers and conducted extensive studies of the efficiency of this method in tubes, annular channels, and longitudinally washed tube bundles in flows of gases and liquids [2]. The essence of the proposed method is the knurling of periodically located annular grooves on the external surface of the tubes (Fig. 1). These grooves, along with the annular diaphragms with a smooth configuration formed on the inside surface, agitate the flow in the boundary layer and ensure intensification of heat exchange on both the outside and inside of the tubes. Here, the external diameter of the tubes remains the

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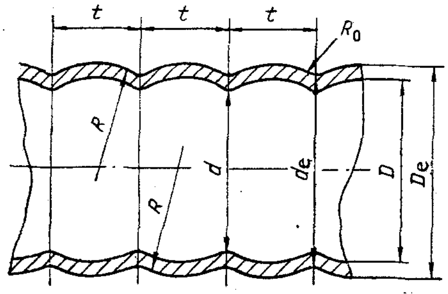

Fig. 1

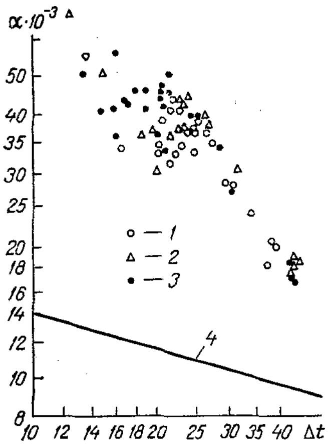

Fig. 2

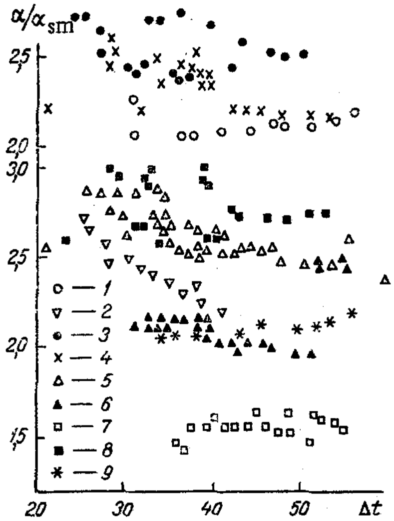

Fig. 3

Fig. 1. Tube with annular grooves.

Fig. 2. Dependence of the heat-transfer coefficient for tube No. 9 on the temperature head with different arrangements of the tubes in the bundle: 1) top tube of a horizontal bundle; 2) second tube of a vertical bundle; 3) third tube; 4) heat transfer with condensation on a smooth tube at p_s = 0.2-0.25 MPa, t_s = 120-125°C. $\alpha \cdot 10^{-3}$, W/(m²·K); Δt, °C.

Fig. 3. Intensification of heat transfer on horizontal tubes with annular grooves: 1-9) variants of tubes with the geometry indicated in the text.

same, which makes it possible to avoid significantly altering the procedure for assembly of tubular heat exchangers. Moreover, blockage and salt deposition are not significant problems with the new tubes [3, 4]. The knurled tubes are made on standard equipment.

In film condensation, the intensification of heat transfer on the outside surface of tubes is due to the effect of surface tension on the condensate film. The greater the thickness of the film and the greater its uniformity, the greater its thermal resistance. Thus, to ensure a substantial intensification of heat transfer on the outside of the tubes, the latter must have a geometry such that, with the retention of the periodically located diaphragms on the inside surface, the prevailing conditions will promote separation of the condensate film or its flow into the grooves as its thickness decreases on the remaining tube sections. As will be shown below, the proposed tube design solves this problem. This is done by reducing the distance between the annular grooves to 0.25-0.35 of the outside diameter of the tubes, making the projecting part of the tubes convex, and smoothly joining the surfaces of the grooves with the projections (Fig. 1).

If the tubes are placed horizontally, the condensate film formed on them flows over the outside surface of the tube. The flow is across the tube axis from top to bottom, and the condensate separates from the tube at the bottom. The presence of a variable cross section with smooth fillets on the tube produces additional flow into the periodically located grooves. This leads to a reduction in the thickness of the condensate film on the projecting parts of the tube, which lowers the thermal resistance between the vapor and the wall and increases the heat-transfer rate. The flow of condensate into the grooves from the projecting parts of the tube accelerates their filling, reduces the stability of the film, and leads to its separation or accelerated flow. Thus, the artificial redistribution of the condensate along the tube leads to a sharp increase in heat transfer on the projecting parts of the tube (which make up most of the tube) and ultimately significantly increases the mean heat-transfer coefficient on the external surface.

To substantiate the above choice of parameters for the grooves and projections, we conducted detailed experimental studies within a broad range of regime and geometric parameters.

The intensification of heat transfer on horizontal tubes was studied in the condensation of water vapor. The test section consisted of 16 tubes arranged in corridor fashion and separated by vertical partitioners into four sections. Each section contained four tubes with identical or different groove parameters. Cooling water was fed inside the tubes, while saturated steam was directed into the space between the tubes from above. We measured the following parameters: the temperature of the walls of the tubes at 14 points (four thermocouples were located in three different sections of the tubes, while two other thermocouples were located in another section); water discharge; water pressure and temperature at the inlet and outlet of each tube; discharge, pressure, and temperature of the heating steam; discharge and temperature of the condensate. The availability of reliable data on heat transfer inside tubes with annular diaphragms [2] allowed us to conduct some of the experiments by the heat-exchanger method, i.e., without measurement of the temperature of the wall.

The tests were conducted on nine tubes of brass L68 with an external diameter D_e = 18.3-18.92 mm and a length ℓ_e = 1859 mm. The parameters of the grooves were as follows: 1) $d_e/D_e = 0.856$, $t_e/D_e = 0.37$, $R/D_e \cong 1$, $R_0/D_e = 0.051$; 2) 0.955, 0.37, 0.5, 0.092; 3) 0.931, 0.283, 0.5, 0.095; 4) 0.910, 0.37, 0.5, 0.093; 5) 0.933, 0.37, 0.5, 0.092; 6) 0.913, 0.32, 0.5, 0.094; 7) 0.948, 0.37, 1, 0.053; 8) 0.927, 0.53, 1, 0.053; 9) 0.893, 0.37, 0.76, 0.062. The radius of curvature of the annular grooves was 0.95-1.75 mm. For the sake of comparison, we also studied smooth tubes.

The main parameters in the tests were measured within the following ranges: vapor pressure p_s = 0.157-0.323 MPa, saturation temperature t_s = 112.7-136.6°C, wall temperature t_w = 75.2-98.3°C, temperature head $\bar{\Delta}t$ = 7.2-44.2°C, vapor velocity no more than 5 m/sec. The Reynolds number for the film was Re_{fi} = 25-150. The velocity of the coolant water changed within the range 0.2-3 m/sec, while the temperature at the inlet varied within the range 4-40°C.

Nearly the same results were obtained in regard to the heat-transfer coefficients in the case of vapor condensation on the tubes when they were arranged in different rows. Figure 2 shows the dependence of the heat-transfer coefficient in the condensation of vapor on tube No. 9 as a function of the temperature head with different arrangements of the tubes in a bundle. The data for the smooth tube agrees satisfactorily with the relation [5]

$$\alpha_{sm} = 0.728 \sqrt[4]{\frac{\lambda_q^3 \rho_q^2 g^2}{\mu_q (t_s - t_w) D_e}}, \qquad (1)$$

obtained for the film condensation of stationary vapor on a horizontal tube. The values of thermal conductivity λ_q, absolute viscosity μ_q, and density ρ_q were determined for the temperature t_s.

Figure 3 presents data on heat transfer for the investigated tube variants in the form of the dependence of the ratio of the heat-transfer coefficients α/α_{sm} on the temperature head $(t_s - t_w)$. The intensification of heat transfer is nearly independent of the temperature head. The presence of the annular grooves increases the coefficient by a factor of 1.8-2.65, with the increase being greater, the deeper the grooves and the smaller the distance between them. Comparison of the data in Fig. 3 shows that given identical value of d_e/D_e and t/D_e, a reduction in the radius of curvature of the projecting parts of the tubes R intensifies heat transfer. A reduction in R/D_e from 1 to 0.5 increases the heat-transfer coefficient by 40%. The data obtained is generalized by the relation

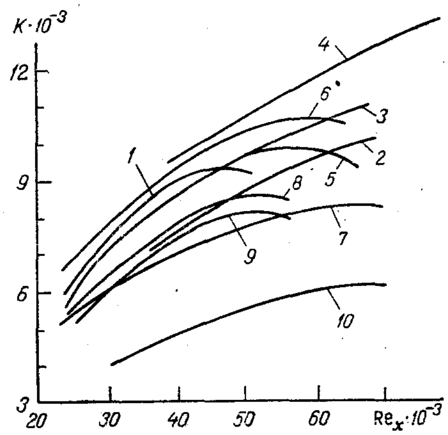

Fig. 4. Dependence of the heat-transfer coefficient on the Reynolds number in water, with a constant value Re_{fi} = 60; 1-9) variants of tubes; 10) smooth tube. $K \cdot 10^{-3}$, W/(m²·K).

$$\frac{\alpha}{\alpha_{sm}} = 2.469\left(1 - 0.2445\frac{R}{D_e}\right)\left(1 - 0.379\frac{t}{D_e}\right)\exp\left[(3.65)\left(1 - \frac{d_e}{D_e}\right)\right], \tag{2}$$

which is valid for d_e/D_e = 0.89-0.95; t/D_e = 0.283-0.37 R/D_e = 0.5-1. Since the temperature head was the same in the comparison of α and α_{sm}, the ratio α/α_{sm} characterizes the increase in heat flux.

It should be noted that in the tubes we investigated, the internal heat-transfer coefficient increased by a factor of 2.5-3 compared to the smooth tube. Thus, the overall heat-transfer coefficient obtained with the use of the given tubes is increased by a factor of 2-2.2 (Fig. 4).

The completed tests showed that when the tubes are positioned horizontally, the chosen tube profile helps organize flowoff of the condensate, reduces the thickness of the film on the projecting parts of the tubes, and accordingly increases the rate of heat transfer during vapor condensation.

The above data was substantiated by commercial trials of a heater for mains water. The heater consisted of 214 tubes with a length of 4.08 m and a diameter D_e = 16 mm. The wall thickness was 0.85 mm. The tubes were located horizontally. The parameters of the grooves d_e/D_e = 0.932, t/D_e = 0.31 (inside the tubes, d/D = 0.932; t/D = 0.357). Tests of the heater and a similar smooth-tube heater were conducted at the same stream pressures. Replacement of the smooth tubes by knurled tubes made it possible to more than double the thermal load and the heat-transfer coefficient. Here, the internal heat-transfer coefficient increased by a factor of 2.47-2.55, while the external coefficient increased by up to 2.5 times. There was a marked increase in the efficiency of intensification with an increase in the pressure of the heating steam.

It should be noted that the annular-grooved tubes studied here are more efficient when used in condensers than tubes with a helical knurled surface [6, 7]. Tubes with a multiple helical knurling make it possible to increase the heat-transfer coefficient by at most 25%.

Thus, the completed tests showed that tubes with annular grooves made by the method developed at the Moscow Aviation Institute can significantly intensify heat transfer in condensers in which the tubes are positioned horizontally.

NOTATION

D, internal diameter of tube; d, diameter of annular diaphragm; D_e, external diameter of tube; d_e, diameter of annular groove; g = 9.8 m/sec²; p_s, saturation pressure; R, radius of curvature of the projecting part of the tube in the axial direction; R_0, radius of curvature of the annular groove; Re_{fi}, Reynolds number of the condensate film; t, spacing of the annular grooves; t_s, saturation temperature; $\overline{t}_w$, mean temperature of the wall; $\overline{\Delta t} = t_s - t_w$, temperature head; α, mean heat-transfer coefficient; ρ, density; ρ_q, μ_q, λ_q, density, absolute viscosity, and thermal conductivity of the condensate film. Indices: sm, smooth.

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EVALUATING THE SPECTRUM OF DIMENSIONS OF VAPOR OCCLUSIONS BY THE ELECTRIC PROBING METHOD

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This article examines aspects of the use of conductimetric sensor-probes to study the structure of steam–water flows.

One of the main parameters governing heat transfer and fluid dynamics in gas–liquid systems is the volume concentration of the lighter phase – the true volumetric gas or vapor content φ_V. The value of φ_V is measured empirically using various integral methods, such as from the reduction in the intensity of radioactive radiation or the change in the mass or relative electrical conductivity of a layer of the two-phase mixture.

The distribution of occlusions of the light phase in the liquid can be characterized by the notion of the local probabilistic gas content φ. By this, we mean the probability of finding the light phase at a given control point. The value of φ in a steady flow can be obtained by probing with a sensor having a sensitive element whose dimensions are considerably smaller than the dimensions of occlusions of the components of the mixture. Knowing the total time the sensitive element is in the light phase $\Sigma\tau_i$ during the chosen exposure T, we can find the probabilistic gas content from the relation

$$\varphi = \frac{\Sigma\tau_i}{T}. \tag{1}$$

For a given velocity of vapor occlusions w, the total time over which the sensitive element passes through the bubble along the chosen streamline is determined by the chords of interaction. With a nominal spherical shape, the maximum length of interaction corresponds to the diameter. The actual spectrum of dimensions of the spherical or spheroidal occlusions unambiguously determines the probability density function of the duration of the above events $\rho(\tau)$ [1]. If there is an equal probability that the bubbles of the actual dimensional spectrum will be distributed over the volume of the mixture, then data obtained by the integral method will agree with the values of φ_V calculated from the diagram of φ in the control section [2].

In practice, the validity of the integral and local methods of measuring gas content is often questionable in large volumes, in dense tube bundles, and in channels of complex geometry. In these cases, the agreement just noted rarely occurs, so the results of probing take on an independent value. This is particularly so in the study of the dispersion and structure of two-phase flows. The present investigation is devoted to evaluating the above-mentioned structural characteristics of flows in power-plant equipment at a pressure of 6.0-6.4 MPa.

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