EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN THE ANNULAR CHANNEL OF A CIRCULATION LOOP

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The results of an experimental investigation of heat transfer in the vertical annular channel of a closed circulation heat exchanger are given.

The design of closed circulation heat exchangers necessitates calculation of the mean heat-transfer coefficients for a viscous gravitational single-phase flow of working fluid in channels of different configurations. One of the possible practical forms of the compact heat-receiving part of a circulation loop is a vertical annular channel.

There are a fair number of studies of convective heat transfer in annular tubes, but the effect of free convection is considered in only a few of them. This case is most thoroughly investigated in [1-3].

In [1, 2] the heat transfer of the inner tube of the annular channel was investigated, and in [3] that of the outer tube was investigated. In [2, 3] there was an attempt to investigate the effect of the channel geometry on heat transfer. In the present paper we investigate the heat transfer of the outer heated wall of an annular slot to the fluid flowing through it and the effect of the channel geometry on the heat transfer.

The experimental apparatus, shown schematically in Fig. 1, was a closed circulation loop consisting of a vertical annular channel, cooler, and connecting system. This type of heat-exchanger loop is similar to the freezing columns used in hydraulic structures, where the working medium is a cheap organic fluid such as kerosene. In the experimental apparatus discussed here we used distilled water. The fluid was heated in the annular gap, entered the cooler, where it was cooled, and then returned to the heated section. At the inlet and outlet of the annular channel there were grid-type mixers.

The investigated channel was formed by two coaxial vertical tubes. The outer was of galvanized steel and was 0.1 m in diameter and 1.5 m high. The inner tube was a Viniplast cylinder, whose diameter could be varied to give different channel geometries.

The outer tube of the annular slot was heated from outside by four electric heaters made of Nichrome wire uniformly wound onto the tube over glass and asbestos cloth. Heat loss was reduced by enclosing the heater in fire clay. Each of the heaters had an independent supply and regulation system, which allowed the obtaining of a heat flux of the required profile. The supply and regulation system of each of the heaters consisted of an ac source, a voltage regulator, a regulating transformer, and a stepdown transformer.

The connecting line between the heated region and the cooler was made of 1-in. zinc-plated steel tubes and had a window for visualization of the flow.

The cooler was a countercurrent tube-in-tube 1-m-long heat exchanger. Its inner tube formed part of the connecting system, and the outer tube was made of steel and was 0.15 m in diameter. The coolant was water from the communal main supply, which was driven through the heat exchanger by a centrifugal pump. Its flow was regulated by a valve and was monitored by a measuring tank. At the inlet and outlet of the two cooler channels there were diffusion grids and screw-type mixers.

The flow of fluid through the investigated channel was measured by two methods: from the heat balance of the cooler and by a thermal flowmeter constructed and calibrated in the Lensovet Leningrad Technological Institute.

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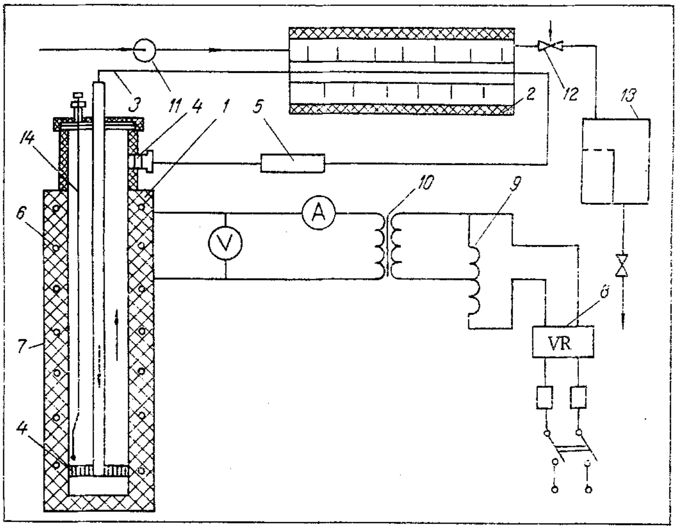

Fig. 1. Diagram of experimental apparatus: 1) annular channel; 2) cooler; 3) connecting line; 4) mixers; 5) thermal flowmeter; 6) heaters; 7) heat insulation; 8) voltage regulator; 9) autotransformer; 10) stepdown transformer; 11) centrifugal pump; 12) regulating valve; 13) measuring tank; 14) heat probe.

The temperature of the fluid in the annular gap was measured by a heat probe, which could be moved transversely and along the channel axis. The wall temperature of the heated outer tube was measured by 20 Chromel–Alumel thermocouples welded on and built-in to minimize heat transfer from the junction and site of contact with the wall through the thermoelectrodes.

To measure the local heat fluxes we used laminated heat-flux sensors constructed in the Institute of Technical Thermophysics, Academy of Sciences of the Ukrainian SSR (ITT). The sensors were calibrated, after they were embedded in the heated wall, on the ITT calibration station. The emfs of all the thermocouples were recorded by an R-306 potentiometer with an M 195/1 galvanometer, and the signals of the heat-flow gauges were recorded by an R-307 potentiometer with an M 195/3 galvanometer.

The results of the experimental investigation are given below. We consider only the case of constant temperature of the outer wall of the annular channel. The total amount of heat Q carried by the fluid from the heated section was determined from the flow of liquid and from the difference in its heat content at the inlet and outlet. Knowing Q, the area of the heat-transfer surface $F = \pi d_2 h$, and $\Delta T = T_\omega - T_0$, we determine the mean heat-transfer coefficient over the height of the channel:

$$\bar{\alpha} = \frac{Q}{F\Delta T} = \frac{\bar{q}}{T_w - T_0}. \quad (1)$$

The local heat-transfer coefficients are calculated in the following way:

$$\alpha = \frac{q}{T_w - T_0}. \quad (2)$$

Here q is the heat flux in the particular section of the channel, determined from the reading of the heat-flux gauge. Integrating the values of the local heat-transfer coefficients over the length of the channel, we obtain the averaged value. The difference between the values of the mean heat-transfer coefficients calculated by the two methods did not exceed 10%. For concentric annular channels the dimensionless heat-transfer coefficient is a function of the Reynolds number, Grashof number, Prandtl number, K, and the reduced tube length. As a characteristic dimension in the convective heat-transfer criteria we use the equivalent slot diameter. The physical constants in them are taken for a mean liquid temperature $T_m = 0.5(T_0 + T_h)$.

The experimental data for the heat transfer from the heated outer wall of the annular channel to the flowing medium were correlated by the equation

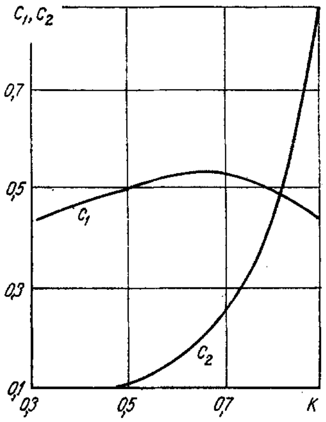

Fig. 2. Curves of $C_1 = f(K)$; $C_2 = \varphi(K)$ (C_1 and C_2 are dimensionless).

$$\overline{\mathrm{Nu}} = C_1 \mathrm{Pr}^{1/3} \left(\mathrm{Cr}\ \frac{d_E}{h}\right)^{1/4}. \tag{3}$$

Here C_1 is a constant that depends on the slot width. This dependence is shown in Fig. 2. As can be seen from Fig. 2 and Eq. (3), there is an optimum gap width at which heat transfer is greatest. A similar result was obtained in [3]. This effect can be accounted for as follows. Widening of the slot, i.e., reduction of K, leads to a reduction of resistance to the ascending flow. In addition, the broader the slot, the slower the stream core is heated, the greater the temperature gradient at the wall, and, hence, the greater the velocity gradient at the heated wall, which determines the heat transfer. Hence, the heat transfer increases. When the optimum slot width is exceeded, this trend is reversed and the heat transfer decreases with increase in K.

This change can be attributed to the onset of a secondary circulation flow due to the great acceleration of the fluid at the outer wall. The flow is maintained by a simultaneous reverse flow of the fluid around the unheated inner wall and in the stream core. As a result of this, the temperature in the stream core increases, while the upthrust and velocity gradient at the heated wall decrease. The experimental investigations showed that heat transfer was greatest at $K = 0.62$.

In a closed circulation loop the flow of fluid is not an independent quantity, but varies with the heat load and channel geometry. Hence, the experimental data for the average dimensionless velocity of the fluid in the investigated annular channel were correlated by the equation

$$\mathrm{Re} = C_2 (\mathrm{Gr})^{1/2}, \tag{4}$$

which expresses the relationship between Re and Gr, and characterizes the hydrodynamics of the flow. Here C_2 is a constant whose dependence on the gap width is shown in Fig. 2. Equations (1) and (2) are valid for values

$$20 \leqslant \frac{h}{d_E} \leqslant 120; \quad 0.3 \leqslant K \leqslant 0.9; \quad 10^6 \leqslant \mathrm{Gr} \leqslant 10^8.$$

NOTATION

d_1, d_2, diameters of inner and outer tubes of annular channel; $d_E = d_1 - d_2$, equivalent diameter; h, height of annular slot; T_0, T_h, temperatures of heat-transfer medium at inlet and outlet of channel; T_ω, temperature of heated wall; q, $\bar{q}$, local and mean (over length of channel) heat fluxes; $K = d_1/d_2$, parameter characterizing gap width; $\mathrm{Pr} = \nu/a$, Prandtl number; $\mathrm{Gr} = [g\beta(T_W - T_0)/\nu^2]d_E^3$, Grashof number; h/d_E, reduced tube length; $\bar{u}$, mean velocity of heat-transfer medium in investigated gap; $\mathrm{Re} = \bar{u}d_E/\nu$, Reynolds number; $\overline{\mathrm{Nu}} = \bar{\alpha}d_E/\lambda$, mean Nusselt number over length of channel.

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FORMATION OF THE DROP-SIZE SPECTRUM IN A GAS – LIQUID FLOW

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The deformation of the surface waves on a liquid film, the breakaway of liquid from the film, and the drop-size spectrum formed during wave destruction are investigated by using a holographic method.

The moisture suspended in the gas core of a dispersed annular gas–liquid flow is formed as a result of part of the liquid breaking away from the film on the channel wall. Up to now it has been established that the breakaway of liquid is realized principally from the crests of the largest, so-called "perturbing" waves. The liquid stripped from the wave crest is subsequently taken into the gas core of the flow. The drop spectrum in the core of a dispersed annular flow is formed as a result of breakup and coagulation of the drops stripped from the film.

In this paper, the breakaway of liquid from a film running over the outer surface of a 6.0-mm-diameter pipe is studied using a holographic method [1, 2]. The liquid is inserted through an annular ~5-mm gap between two coaxial pipes. The tests were conducted with water, ethyl alcohol, and glycerin solutions (the viscosities of the aqueous glycerin solutions are $20 \cdot 10^{-6}$ and $140 \cdot 10^{-6}$ m^2/sec) at air speeds of 35-110 m/sec. The liquid flow rates per unit perimeter were 0.07-0.3 kg/m · sec for water, 0.2-0.5 kg/m · sec for alcohol, and 0.08-1.0 kg/m · sec for the glycerin solutions.

Deciphering the holograms showed that the disintegration of the surface waves is analogous to breakup of a liquid jet in a gas. The nature of drop formation around the film surface, like breakup of a jet, depends essentially on the viscosity of the liquid. In conformity with this, all liquids can be grouped into low-viscosity (water, alcohol) and high-viscosity (glycerin) liquids depending on the magnitude of the parameter $a\sigma/\rho_l \nu_l^2$. A study of the holograms showed the following sequence of wave destruction. The smooth monotonic character of the wave surface is spoiled first. Individual perturbations appear which grow with time and eventually develop a rather complex shape. The presence of one or more projections is characteristic for such a perturbed surface. One such projection can evolve into a jet flowing out of the film into the gas flow. The most characteristic, fully developed jet is shown in Fig. 1a. Waves and necks appear in the jet, and in the long run it dissociates into individual drops (Fig. 1b).

The formation of not one, but several, individual jets on the surface of one wave is possible at higher gas velocities.

It should be emphasized that in all these cases we speak of perturbations in the scale of the wave itself, when the wave as a whole takes part in the deformation and when we can speak about the internal motion of the whole liquid mass included in the wave. We refer to this type of wave surface destruction as dissociation. Dissociation is the main type of wave destruction at relatively low gas velocities. These velocities reach approximately 50 m/sec for water.

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