HEAT EXCHANGE IN DESCENDING ANNULAR GAS—LIQUID FLOWS

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Experimental data on heat exchange in turbulent descending liquid flows with an accompanying gas flow are presented.

In various branches of industry heat and mass exchange are intensified by introducing technological processes occurring in thin films [1-4]. Such processes are used for heating, evaporation, cooling, and absorption. Much film equipment operates with an organized single direction flow of the liquid film and gas. While heat exchange in turbulent descending films has been studied quite thoroughly [1-6], there is only limited information available on the effect of a gas flow on intensification of heat exchange [4, 7].

In connection with this an attempt was made to experimentally clarify the principles of heat exchange in two-phase annular descending flows within a 30-mm-diameter tube. The studies were performed with air—water and water vapor flows at pressures of 0.1-0.4 MPa. The hydrodynamics of such flows were recently studied by some of the present authors [8, 9]. The experimental apparatus consisted of an open channel for the gas and liquid with forced motion of the phases [8]. The length of the measurement section of stainless steel tube was 2400 mm. The film was formed with the aid of concentric slot devices. Liquid flow rate was determined by the volume method. Film thickness was measured with special electrical probes moved by micrometer screws. The moment of probe tangency to the film surface was determined recorded by an electronic relay with indications recorded on chart paper. The sensor construction developed permitted film thickness measurement to an accuracy of 0.01 mm. The amplitude of waves on the film surface was found from the measured peaks and troughs of the waves. The frequency at which the mean film level intersected the probe was determined from the number of operations of the relay, which was recorded by a special device. Statistical processing of the experimental data revealed that the mean uncertainty of deviation of regression curve values from experimental data was $\pm 7\%$. The irrigation rate in the experiments was varied from 0.15 to 1.8 kg/m·sec, while gas phase velocity varied from 3 to 20 m/sec. Gas expenditure was measured with a precalibrated diaphragm. The experimental section was heated by saturated water vapor. The thermal flux was determined from the quantity of condensate collected on sections of the tube spaced 500 mm apart. Temperatures of the tube wall, gas, and film were measured by copper—constantin thermocouples and low resistance potentiometers. In experiments with air—water flows the air was first heated up to the film temperature.

In the first stage of the experiments the intensity of heat exchange was determined for single-phase flow of a turbulent film ($Re_1 > 1600$). The results of these studies are shown in Fig. 1. Also shown for comparison are similar dependences obtained by other researchers. It is evident from Fig. 1 that the present data agree completely satisfactorily with the results of previous studies in the Reynolds number range considered. The dependence shown in Fig. 1 can be approximated by the expression

$$\mathrm{Nu}_1 = 0.0233\,\mathrm{Re}_1^{0.8}\mathrm{Pr}^{0.4}, \tag{1}$$

which agrees within $\pm 5\%$ with the analytical expression of [10] obtained previously.

The principles of heat exchange for two-phase annular descending flows are shown in Fig. 2, whence the effect of the gas phase on heat exchange intensity can be judged. It was established during the experiments that the effect of the gas on heat exchange begins to appear in practice at velocities $w_2 > 6$ m/sec. In the gas velocity range of 0-20 m/sec (commencement of droplet removal from the film surface) the gas flow intensifies heat exchange by an average of 40%. The increase in the heat liberation coefficient is caused by a

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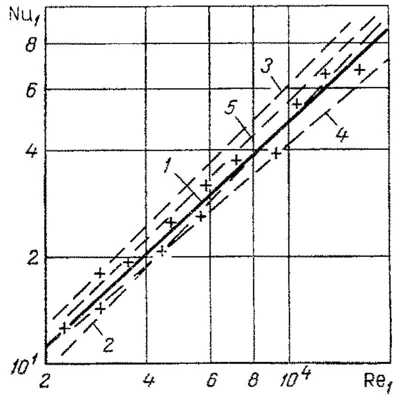

Fig. 1

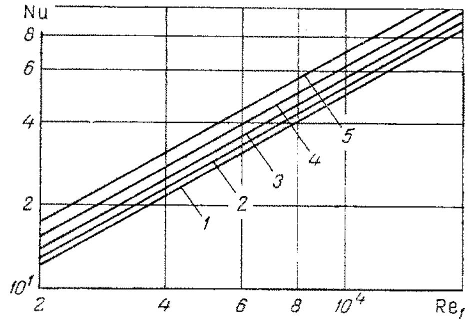

Fig. 2

Fig. 1. Heat exchange principles for single-phase flow at Pr = 2: 1) authors' experiments; 2) [2]; 3) [3]; 4) [5]; 5) [6].

Fig. 2. Intensity of two-phase flow heat exchange at Pr = 2: 1) $Re_2 \cdot 10^{-4} = 2$; 2) 0.4; 3) 0.7; 4) 1.6; 5) 2.8.

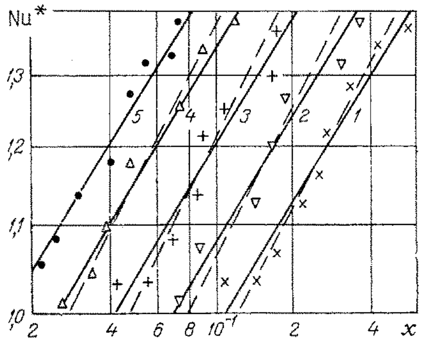

Fig. 3

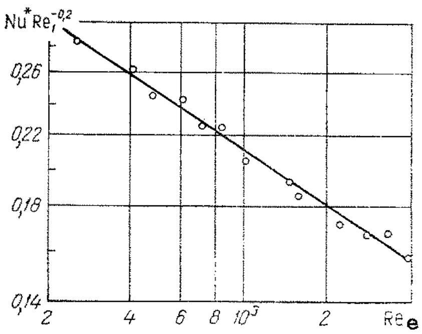

Fig. 4

Fig. 3. Effect of mass gas content on heat exchange intensity: 1) $Re_1 \cdot 10^{-3} = 2.02$; 2) 4.038; 3) 7.62; 4) 10.78; 5) 16.16.

Fig. 4. Function $Nu^* Re_1^{-0.2} = f(Re_e)$.

decrease in film thickness with increase in w_2 [8] and a change in the structure of the film surface [9]. The dependences presented in Fig. 2 can be described by the equations

$$Nu = 5.5 \cdot 10^{-3} Re_1^{0.8} Re_2^{0.175} Pr^{0.4}, \tag{2}$$

$$Nu^* = 0.236 Re_2^{0.175}, \tag{3}$$

which are convenient for practical calculations of heat exchange in two-phase annular flows.

In the previous studies [4, 7] it was recommended that the effect of the gas flow on intensification of heat exchange be considered with the aid of the Martinelli parameter or the value of the mass gas content, which in our opinion, would complicate the computation expressions. Figure 3 shows the value of the relative Nusselt number as a function of the mass gas content of the flow. The dashed lines here indicate data calculated for the experimental conditions of [7]. It is evident from comparison of these dependences that their divergence over the x range considered does not exceed ±8%. To an accuracy of ±10% the data of Fig. 3 can be generalized with the expression

$$Nu^* = 1 + x^{1.2} \exp(0.151 \cdot 10^{-3} Re_1 - 0.2), \tag{4}$$

which is more cumbersome and less accurate than Eqs. (2) and (3).

As has already been noted, an increase in gas flow rate and gas content changes the wave structure on the film surface [9]. It is the structure of the small-scale waves which determines the roughness of the phase boundary and produces the major effect on heat exchange intensity and hydraulic resistance. The character of the wave surface was considered in [9] by means of the film wave Reynolds number Re_w, which relates the length, frequency, and amplitude of the waves. It was shown that pressure losses for the flows considered correlate well with the value of Re_w. In determining the structure of a turbulent annular flow, where the waves on the film surface are irregular, the value of Re_w permits establishment of regimes with optimum heat exchange for both single-phase and two-phase flows. The values of Re_w are easily determined analytically or from the graphs of [9]. The relationship between the criterion Nu* and the wave Reynolds number is shown in Fig. 4. It is described by the expression

$$Nu^* = Re_1^{0.2} Re_w^{-0.23} \tag{5}$$

very convenient for calculations.

In conclusion, we will note that the relationships obtained are valid for film surface evaporation conditions in the absence of droplet removal.

NOTATION

Γ, mass irrigation density; d, tube diameter; δ, film thickness; w, velocity; ρ, density; ν, kinematic viscosity; α, λ, and A, coefficients of heat liberation, thermal conductivity, and thermal diffusivity; G, mass flow rate; f, λ_w, and A, frequency, length, and amplitude of wave; Nu, Nusselt number for two-phase annular flow; $Nu^* = Nu Nu_1^{-1}$; $Nu_1 = 4\alpha_1 \delta \lambda_1^{-1}$; $Re_1 = 4\Gamma(\rho_1 \nu_1)^{-1}$; $Re_2 = w_2 d \nu_2^{-1}$; $Re_w = f \lambda_w \nu_1^{-1} A$; $Pr = \nu_1 a_1^{-1}$; $x = G_2 (G_1 + G_2)^{-1}$. Subscripts: 1, liquid; 2, gas.

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