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CALCULATION OF THE EFFECT OF EVAPORATING DROPS ON THE RELATIVE LAW OF HEAT EXCHANGE WITH A DISPERSE MIST FLOW

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On the basis of a two-layer scheme of wall turbulence, a relative law of heat exchange with a disperse mist flow is calculated. It is shown that the influence of drops on heat exchange leads to a finite "stepwise" increase in heat transfer compared to the case of a single-phase vapor flow.

As is known [1], post-dryout heat transfer depends substantially on the characteristics of the spectrum of liquid drops dispersed in a turbulent channel flow.

When a mathematical description of post-dryout heat transfer involves a continuous (vapor) phase energy equation based on the parameters of the spectrum of drops (size distribution functions, dependences of the characteristic diameter on the radial coordinate), mass sources and heat sinks are determined that owe their origin to the evaporation of saturated drops of liquid in a superheated vapor flow [2, 3]. In this case, in the absence of a rigorous theory for the fragmentation of drops in a turbulent continuous-phase flow various empirical expressions for the corresponding source terms are usually used.

In [4] a "resonance" model of the fragmentation of drops in a turbulent gas (vapor) flow in a channel is suggested according to which the fragmentation of a drop occurs when the "Kolmogorov" frequency of turbulent oscillations in a continuous phase

$$\omega_{\mathrm{K}} \approx \sqrt{\left(\frac{\varepsilon}{\nu}\right)} \tag{1}$$

exceeds the "Rayleigh" frequency of the natural oscillations of the drop:

$$\omega_{\mathrm{R}} \approx \sqrt{\left(\frac{\delta}{\rho_{\mathrm{d}}\, d_{\mathrm{d}}^{3}}\right)} . \tag{2}$$

Thus, those drops are susceptible to fragmentation that have a diameter larger than a certain "maximum stable size":

$$d_{\mathrm{d\,max}} = \beta \left(\frac{\delta \nu}{\rho_{\mathrm{d}}\, \varepsilon}\right)^{1/3} . \tag{3}$$

Here ρ_{d} is the density of the liquid drop; ν is the kinematic viscosity of the continuous phase (vapor); σ is the surface tension coefficient; ε is the turbulent energy dissipation density; β is a numerical constant.

The use of the hypothesis of a turbulent energy generation-dissipation balance and the Prandtl logarithmic law for the turbulent flow core leads to the following expression for the most stable diameter of a drop:

$$d_{\mathrm{d\,max}}^{+} = \beta_1 \left(\frac{\delta y^{+}}{\rho_{\mathrm{d}}\, \nu\, u_{*}}\right)^{1/3} . \tag{4}$$

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Here $d^+_{d\,max} = u_* d_{d\,max}/\nu$ is the dimensionless diameter of a drop; $y^+ = u_* y/\nu$ is the dimensionless transverse coordinate reckoned from the wall; $u_* = \sqrt{\xi/8}\,\bar{u}$ is the wall scale of velocity; $\bar{u}$ is the mean mass velocity of the continuous phase (vapor); ξ is the coefficient of hydraulic resistance.

Within the framework of the resonance model [4] it is assumed that drops of size $d_d < d_{d\,max}$ coalesce on collision and, as a result, attain the most stable size. At the same time, drops of size $d_d > d_{d\,max}$, conversely, are unstable and are susceptible to fragmentation until they attain size $d_{d\,max}$. Thus, according to Eq. (4), for each value of the transverse coordinate y the dispersed phase is represented by drops with an identical (most stable) diameter $d_{d\,max}$.

The expression for the heat sink q_v, which appears due to evaporation of saturated liquid drops and which is "smeared" over the vapor phase has the following form [2, 3]:

$$q_v = 12\,\frac{1-x}{x}\,\frac{\rho}{\rho_d}\,\frac{\lambda\,(T(y) - T_s)}{d^2_{d\,max}(y)}. \tag{5}$$

Here λ is the thermal conductivity of the vapor; ρ and ρ_d are the densities of the vapor and liquid, respectively; $T(y)$ is the vapor temperature at the current value of the transverse coordinate; T_s is the saturation temperature; x is the vapor quality of the mist flow.

Let us consider Prandtl's classical two-layer scheme [5] for the case of turbulent flow of a disperse mist flow in a tube. We will make the following assumptions: 1) the influence of the dispersed phase on heat transfer is manifested in "smearing" of the heat sink, determined from Eq. (5), over the entire vapor volume, with no drops being present inside of the viscous sublayer; 2) dependence (4) of the characteristic diameter of a drop on the transverse coordinate of form $\sim y^{1/3}$ is replaced approximately by a dependence of form $\sim y^{1/2}$; 3) drops do not exert any influence on the velocity field of the continuous phase; 4) the Prandtl number for the vapor is equal to unity.

With allowance for the assumptions made, the energy equation for the turbulent core can be written as

$$y^+ \frac{d}{dy^+}\left(y^+ \frac{d\vartheta^+}{dy^+}\right) = m^2 \vartheta^+ . \tag{6}$$

Here $\vartheta^+ = (T - T_0)/T_*$ is the dimensionless temperature difference; T_* is the wall scale of temperatures defined by the relation [5]

$$\frac{T_*}{T_w - T_0} = \frac{\mathrm{St}}{\sqrt{\xi/8}}, \tag{7}$$

where T_w and T_0 are the vapor temperature on the wall (at $y = 0$) and in the tube center (at $y = R_0$), respectively; ξ is the friction factor; St is the Stanton number.

The dimensionless parameter m, which determines the effect of drops on heat transfer has the form

$$m^2 = \beta_2\,\frac{1-x}{x}\,\frac{\rho\bar{u}}{\delta^{2/3}}\left(\frac{\nu D_0}{\rho_d}\right)^{1/3}. \tag{8}$$

The numerical constant β_2 is determined by the corresponding constant β_1 in relation (4). As estimates show, $\beta_1 \sim 1$, $\beta_2 \sim 1$.

Equation (6) has an exact solution:

$$\vartheta^+ = \frac{(R_0^+)^{2m} - (y^+)^{2m}}{\kappa m\,[(R_0^+)^{2m} + 1]\,y^+}, \tag{9}$$

which satisfies the condition $\vartheta^+ = 0$ at $y^+ = R_0^+$.

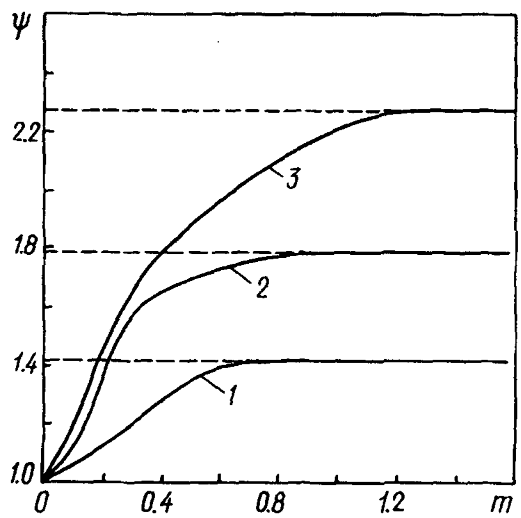

Fig. 1. Dependence of the relative law of heat transfer on the parameter m: 1) Re = 10^4; 2) 10^5; 3) 10^6; dashed lines, the limiting relative laws of heat transfer for $m \to \infty$.

In the limit $m \to 0$, the effect of the drops degenerates and solution (9) passes into Prandtl's classical law [5]:

$$\vartheta^+ = \frac{1}{\kappa} \ln \frac{R_0}{y}. \tag{10}$$

The usual procedure of "joining" profile (9) for the turbulent core with the linear profile for the temperature sublayer (which at Pr = 1 coincides with the viscous sublayer) gives the following expression for the Stanton number St

$$\mathrm{St}^{-1} \sqrt{\left(\frac{\xi}{8}\right)} = \delta^+ + \vartheta_\delta^+ - \langle \vartheta^+ \rangle , \tag{11}$$

where $\delta^+ = 11.5$ is the dimensionless thickness of the viscous sublayer; $\vartheta_\delta^+ = (T_\delta - T_0)/T_*$ is the dimensionless temperature difference for the turbulent flow core determined from Eq. (9) at $y^+ = \delta^+$; $\langle \vartheta \rangle^+$ is the dimensionless temperature difference averaged over the cross-section, which is defined by the relation

$$\langle \vartheta^+ \rangle = \frac{2}{\kappa m \, (R_0^+)^2 \, [(R_0^+)^{2m} + 1]} \left\{ \frac{(R_0^+)^{2m+1}}{1 - m} \left[(R_0^+)^{1-m} - (\delta^+)^{1-m} \right] - \right.$$

$$\left. - \frac{R_0^+}{1 + m} \left[(R_0^+)^{1+m} - (\delta^+)^{1+m} \right] - \frac{(R_0^+)^{2m}}{2 - m} \left[(R_0^+)^{2-m} - (\delta^+)^{2-m} \right] + \frac{(R_0^+)^{2+m} - (\delta^+)^{2+m}}{2 + m} \right\} . \tag{12}$$

Here $\kappa = 0.4$ is the Karman constant; $R_0^+ = R_0 u_* / \nu$ is the dimensionless radius of the tube ($R_0 = D_0/2$).

In the limit $m \to 0$, Eqs. (11) and (12) give the Prandtl classical law [5] (with allowance for the Reynolds analogy $\mathrm{St} = \xi/8$):

$$1/\sqrt{\xi} = 2 \log (\mathrm{Re}/\sqrt{\xi}) - 0.8 . \tag{13}$$

In this case, the effect of drops on heat transfer degenerates into

$$\mathrm{St} \to \mathrm{St}_0 = \xi/8 , \tag{14}$$

and the "relative law of heat transfer" $\psi \equiv \mathrm{St}/\mathrm{St}_0$ becomes equal to unity

$$\psi \to 1 . \tag{15}$$

In the limit $m \to \infty$ we have

$$\mathrm{St}^{-1} \to 11.5 \sqrt{8/\xi} . \tag{16}$$

This corresponds to the maximum "improvement" of heat transfer

$$\psi \to \psi_{max} = \frac{\sqrt{8/\xi}}{11.5} . \tag{17}$$

Figure 1 presents the results of calculation of the relative law of heat transfer depending on the parameter m, which determines the effect of drops on heat transfer to a disperse mist flow. As seen from the figure, the maximum "improvement" in heat transfer occurs already at $m \approx 1$.

Thus, the effect of drops dispersed in a vapor flow on heat transfer leads to a finite "stepwise" increase in heat transfer compared to the case of a single-phase vapor flow at the same values of the total temperature difference $T_w - T_0$ and the vapor phase Reynolds number Re. This effect has a clear physical explanation, consisting in a decrease in the thermal resistance of the turbulent core of the flow after the "introduction" into it of heat sinks appearing due to the evaporation of drops.

As heat sinks attain a certain level of intensity (increase in the concentration of drops in the vapor flow), complete "switching-off" of the thermal resistance of the turbulent core of the flow occurs and heat transfer follows the mechanism of "pure" heat conduction through a temperature (viscous) sublayer

$$q_{max} = \frac{\lambda (T_w - T_0)}{\delta} . \tag{18}$$

We calculated the influence of drops dispersed in a turbulent vapor flow in a channel on heat transfer, which can be used in determining post-dryout heat transfer.

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