INVESTIGATION OF HEAT TRANSFER AND FLUID FRICTION IN A VISCOUS GRAVITATIONAL FLOW OF WATER IN A HORIZONTAL TUBE AT q_W = const

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Results of an experimental investigation of local and mean heat transfer and fluid friction are given. It is shown that the heat transfer and fluid friction coefficients in the viscous gravitational regime depend on the heat supply law.

At present there are no reliable data for calculating heat transfer and fluid friction in viscous gravitational flows in horizontal tubes at constant q_W. The experimental investigations, known to the authors, that deal with processes in the above flows have generally been carried out at or near constant tube wall temperature [1–3]. The paper by Id [4] is an exception, but the calculations recommended there can scarcely be used because of the great scatter of the experimental data and the absence of limits of variation of the governing parameters.

Our tests were done on a horizontal 1Kh18N9T steel tube of internal diameter 9.6 mm and wall thickness 0.5 mm. The heat supply at q_W = const was provided by passing an alternating electric current directly along the tube over a section of 120 diameters, 16 diameters from the tube entrance and exit. Static pressure taps were provided over a length of 130 diameters.

Viscous regime. In order to obtain a more reliable figure for the influence of free convection, relations were obtained on the same equipment for heat transfer and resistance coefficients at the minimum possible heat flux (from the viewpoint of reliable measurement), as well as for the resistance coefficient in isothermal laminar flow, in other words, in conditions close to the viscous regime.

The heat transfer averaged over the length from these experiments may be approximated to within an error of 6.5% by the formula

$$\overline{\mathrm{Nu}} = 1.64\,(\overline{\mathrm{Re}}\,\overline{\mathrm{Pr}}\,d/L)^{1/3}\,(\mu_l/\mu_W)^{1/3}\,.$$

The reference temperature is the mean liquid temperature along the length. In the factor $(\mu_l/\mu_W)^n$ we chose $n = 1/3$ as giving the best grouping of the points, and the constant factor is reduced somewhat (to 1.64) as compared with $n = 0.14$, as recommended by Seider and Tate [5], and also with $n = 0.125$ to 0.17, as recommended by Petukhov et al. [6]. This factor generally has no great influence in the viscous gravitational regime, since as the heat flux increases free convection plays a decisive role.

Calculation according to [1] gives results 6% greater than for Petukhov's formula [6], and 13% less than for Seider and Tate's [5].

The local heat transfer data were reduced for the six sections: x/d = 8, 13, 24, 42, 65, and 106. The results show good agreement with the formula of Filimonov and Khrustalev [7],

$$\mathrm{Nu}_x\,(\mathrm{Pr}_W/\mathrm{Pr}_l)_x^{1/4} = 4.36 + 0.36\,X^{-0.5}\cdot 10^{-18X}, \qquad (1)$$

where

$$X = \frac{x/d}{\mathrm{Re}_{in}\,\mathrm{Pr}_{in}^{0.8}}\,.$$

Here all the quantities with subscript x are referred to the liquid temperature at section x, Pr_W to T_W, and Re_{in} and Pr_{in} to the liquid temperature at the entrance.

The experimental results to determine the mean resistance coefficient were corrected for lack of hydraulic stabilization as recommended by Filimonov and Khrustalev, and are then described satisfactorily by the relation

$$\xi = (64/\overline{\mathrm{Re}})\,(\mu_W/\mu_l)^{0.14}.$$

The Reynolds number was calculated in terms of the mean liquid temperature along the length.

Viscous gravitational regime. Three series of experiments were conducted, with fixed Re_{in} = 840, 1170, 1600 and variable heat flux.

It can be seen from Fig. 1 that the character of the wall temperature distribution depends strongly on the heat loading, whose influence we shall evaluate in terms of the group

$$\mathrm{Gr^*Pr} = \frac{g\beta d^4 q}{\nu^2\lambda}\,\frac{\nu\gamma c_p}{\lambda} = \frac{g\beta d^4\gamma c_p q}{\nu\lambda^2}\,.$$

Thus, with $\overline{\mathrm{Gr^*Pr}} = 5.25\cdot 10^6$ a decrease in heat transfer coefficient along the tube is apparent, and there must also be one in the stabilizing section. With $\overline{\mathrm{Gr^*Pr}} = 2.6\cdot 10^7$ however, the situation is reversed, since the convective flux increases along the tube and leads to increasingly strong mixing of the liquid in the form of double spiral flow.

Figure 2 shows the results of local heat transfer experiments, expressed as the ratio of the measured value of Nu_q to that determined from (1) as $q \to 0$. The heat flux varied in the range

$$q = 4.76\cdot 10^3 - 5.7\cdot 10^4\ \ \mathrm{W/m^2}.$$

The experimental results break down independently of $\mathrm{Pe}(x/d)$ into three regions, and are well described by the equation

$$\mathrm{Nu}_q/\mathrm{Nu}_{q\to 0} = C\,(\mathrm{Gr^*Pr})^n, \qquad (2)$$

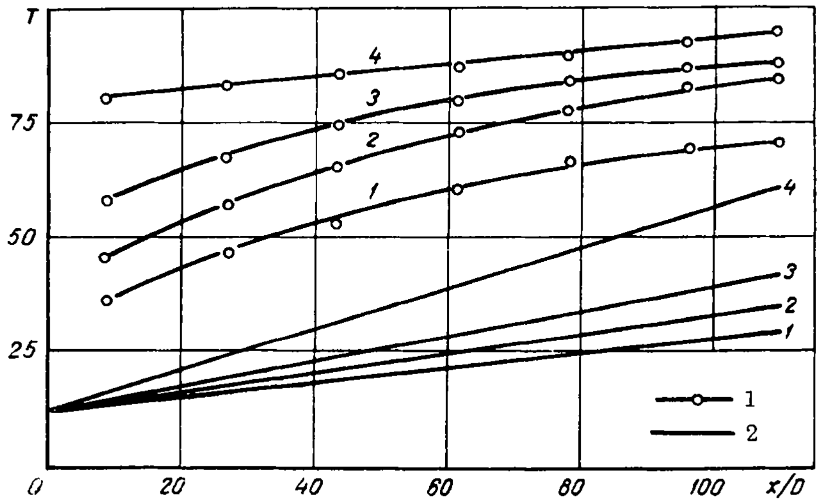

Fig. 1. Variation in temperature T, °C, of the wall (1) and of the liquid (2), along the tube (Re_{in} = 1150): 1) with $\overline{Gr^{*}Pr}$ = 5.25 · 10^6; 2) 8.5 · 10^6; 3) 1.3 · 10^7; 4) 2.6 · 10^7.

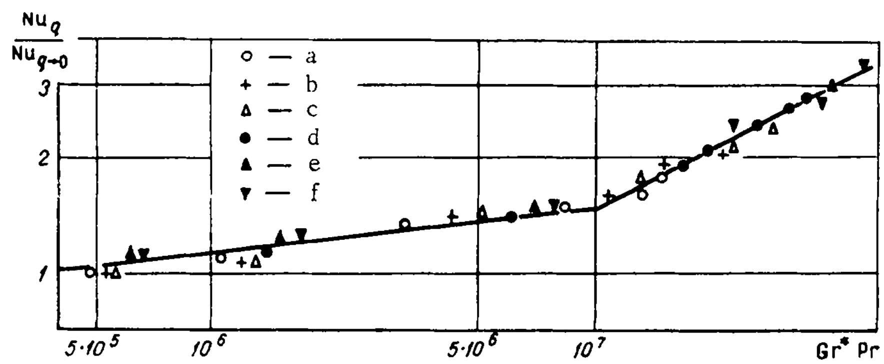

Fig. 2. Local heat transfer in the viscous gravitational regime: a) when x/d = 26; b) 43; c) 61; d) 78; e) 96; f) 106.

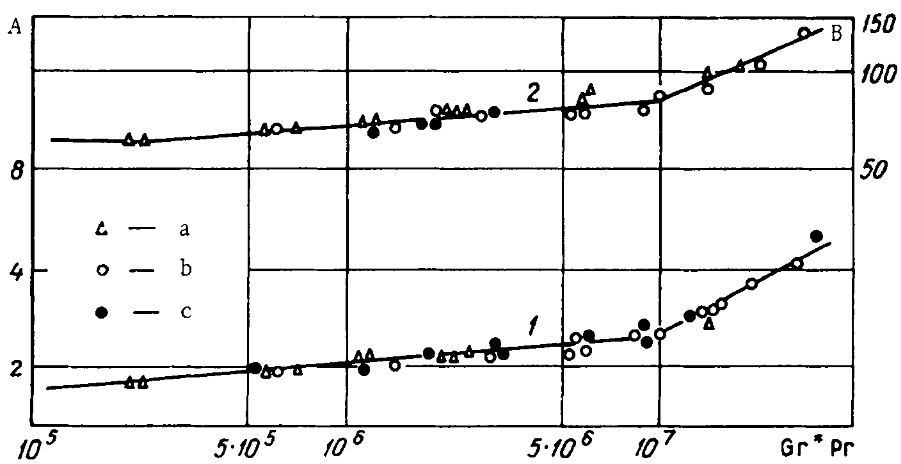

Fig. 3. Mean heat transfer coefficient (1) and resistance coefficient (2) in the viscous gravitational regime (A = $\overline{Nu}/(\overline{Pe}\, d/L)^{1/3}$; B = $\xi Re/(\mu_w/\mu_l)^{0.14}$): a) with $\overline{Pe}$ d/L = 57.5; b) 90; c) 127.5.

where

$$C=1,\quad n=0 \text{ when } \mathrm{Gr^*Pr}<4\cdot 10^5,$$
$$C=0.28,\quad n=0.1 \text{ when } 4\cdot 10^5<\mathrm{Gr^*Pr}<10^7,$$
$$C=0.000465,\quad n=0.5 \text{ when } 10^7<\mathrm{Gr^*Pr}<3\cdot 10^7.$$

The governing temperature is the mean for the liquid at the given section. It is pertinent to stress that in the range of heat fluxes examined the relation between both the local and the mean heat transfer coefficients and the group Pe(x/d) is independent of the value of the heat flux, and remains the same as in the viscous flow regime. This permits us to allow for the influence of heat loading by introducing only the group Gr*Pr into the calculation formulas for the viscous regime.

Figure 3 shows the experimental results for mean heat transfer, which also fall into three regions and may be described by the formula

$$\overline{\mathrm{Nu}}=1.64\,(\overline{\mathrm{Pe}}\;d/L)^{1/3}[C_1(\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}})^n], \tag{3}$$

where

$$C_1=1,\quad n=0 \text{ when } \overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}<2\cdot 10^5,$$
$$C_1=0.293;\quad n=0.1 \text{ when } 2\cdot 10^5<\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}<10^7,$$
$$C_1=0.000464,\quad n=0.5 \text{ when } 10^7<\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}<3\cdot 10^7,$$

and the governing temperature is the mean of the liquid along the length.

Introduction of the factor $(\mu_l/\mu_w)^{1/3}$ into (3) leads to considerable scatter of the points when $\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}>2\cdot 10^5$ and reduces the scatter when $\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}<2\cdot 10^5$.

Figure 4 shows our results reduced according to the Petukhov method [1], the continuous line indicating the relation obtained by Petukhov with $T_w \approx$ const. In this method $\overline{\alpha} = Q/F(T_w - T_l)_{in}$. It may be seen from the figure that, other conditions being equal, the mean heat transfer for the case q_w = const is greater than for the case T_w = const, if $\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}<3\cdot 10^6$. When $\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}}>3\cdot 10^6$ the influence of the different means of heating does not appear and the heat outputs coincide.

The organization of the heat supply has a decisive influence on the variation of heat flux along the channel. Evidently, when heat is supplied according to the T_w = const law, the greatest heat flux and temperature head occur at the beginning of the heated section, i.e., in conditions in which the thermal boundary layer begins to form. The local heat transfer coefficients are large there even without this, and there is therefore no basis for expecting a noticeable influence of free convection on the flow temperature field, especially since the influence of a field with temperature gradient is confined to the region of the thin annular thermal boundary layer. As the distance from the beginning of the heater increases, there is a sharp drop in the temperature difference between wall and liquid, and the free convection flux is reduced. Heat flux at T_w = const is therefore unfavorable from the viewpoint of development of free convection flow along the channel.

The picture is different with heat supply according to the law q_w = const, when the free convection flux increases with distance from the beginning of the heater, i.e., it increases in the same direction in which a sharp decrease in heat transfer coefficient is observed to result from stabilization of the thermal boundary layer in the viscous regime. It may be seen from Fig. 1 that at large heat fluxes the total influence of free convection increasing along the tube leads to a result that is directly at variance with the usual picture of variation of local heat transfer coefficient in the viscous regime: when q_w = const the local heat transfer coefficient increases with distance from the beginning of the heater.

Thus, the difference of the local heat transfer coefficients for T_w = const and q_w = const, other conditions being equal (identical mean heat flux density, identical distance from the beginning of the heated section, etc.), results mainly from the different local hydrodynamic conditions, since the variation of the local group Gr*Pr along the tube length is different in the two cases, and may even be in the opposite sense.

A matter of independent interest is the local heat transfer coefficient with different conditions of heat supply, but with the same values of Gr*Pr at a given section, and, of course, with the other conditions: Re the same, x/d the same, etc. Unfortunately, we have no experimental data available on local heat transfer coefficients with T_w = const; we are therefore limited to qualitative considerations.

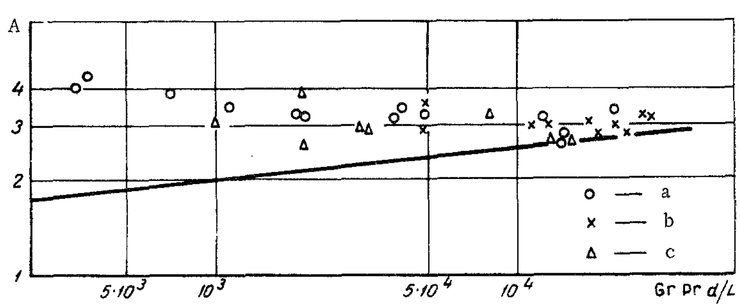

Fig. 4. Mean heat transfer coefficient in the viscous gravitational regime for various methods of heat supply. Continuous line—according to [1] ($A = \overline{\mathrm{Nu}}/0.35 \times (\overline{\mathrm{Pe}}\;d/L)_b^{0.6}$):
a) $\mathrm{Pe_b}$ = 6000; b) 9000; c) 13 000.

From the theoretical solutions for heat transfer in laminar flow in a channel in the absence of free convection, it is known that the temperature profile in the case q_W = const is fuller than for T_W = const.

If therefore convective fluxes have not been generated, then for an equal increase in local temperature head, i.e., for equal local values of Gr*Pr, they appear earlier for q_W = const, than for T_W = const, since the temperature gradient at the wall is greater in the first case.

In the case when the viscous gravitational regime has already been developed, and under heating with T_W = const, the group Gr*Pr decreases along the channel, while with q_W = const it is approximately constant (in the case of gases it decreases slowly, and for liquids it increases because of change of μ).

Therefore, the equality of local values of Gr*Pr at any section of the tube, under different heat supply laws, cannot mean equality of heat transfer at that section, since convective flow at a given place depends on its prehistory, and the nature of variation of local values of Gr*Pr is determined by the heat supply law as well as the temperature field in the absence of natural convection.

Therefore, local heat transfer in the viscous gravitational regime depends appreciably on the law of heat supply along the channel. For example, for equal local Gr*Pr at T_W = const, the influence of natural convection on heat transfer must be greater than at q_W = = const, since the values of Gr*Pr upstream will be greater in the first case, and less in the second. This means that even the convective fluxes develop to a greater extent at T_W = const.

Figure 3 shows results of experiments to determine the mean resistance coefficient at different heat loadings. These results are well described by the formula

$$\xi = (64/\overline{\mathrm{Re}})(\mu_w/\mu_l)^{0.14}[C_2(\overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}})^n], \tag{4}$$

where

$$C_2 = 1, \quad n = 0 \text{ when } \overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}} < 2\cdot 10^5,$$

$$C_2 = 0.415, \quad n = 0.07 \text{ when } 2\cdot 10^5 < \overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}} < 10^7,$$

$$C_2 = 0.002, \quad n = 0.4 \text{ when } 10^7 < \overline{\mathrm{Gr}}^*\overline{\mathrm{Pr}} < 3\cdot 10^7.$$

Here the governing temperature is the mean liquid temperature along the length.

NOTATION:

q—heat flux density at wall; T_w—temperature of tube wall; $\overline{\mathrm{Nu}} = \overline{\alpha}d/\lambda$—mean Nusselt number along length; $\overline{\alpha}$—mean heat transfer coefficient, referred to temperature difference $t_w - t_l$; t_w and t_l—mean integral wall temperature along length, and arithmetic mean liquid temperature along length, respectively; $\mathrm{Nu}_x = \alpha_x d/\lambda$—local Nusselt number; α_x—local heat transfer coefficient, referred to "wall-liquid" temperature head at section x; $\overline{\mathrm{Re}}$, Re_{in}—Reynolds number at governing mean liquid temperature and liquid temperature at tube entrance; $\overline{\mathrm{Pr}}$, Pr, Pr_{in}—Prandtl number at governing mean liquid temperature, liquid temperature at section x, and liquid temperature at tube entrance, respectively; Gr*—Grashof number, expressed in terms of heat flux density; Gr—Grashof number, expressed in terms of mean "wall-liquid" temperature head; Q—heat flux (W); F—internal surface area of heated section of tube; Pe_b—Peclet number at temperature of boundary layer $t_b = (t_w + t_l)/2$.

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