THERMAL DIFFUSION IN A TURBULENT WATER STREAM WITH GAS BUBBLES

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Results are shown of an experimental study concerning the heat transfer and the average temperature fields in an effervescent gas–liquid stream. It is noted that an increase in the gas content brings about turbulent diffusion of heat across the entire stream section.

In several studies concerned with heat transfer during a turbulent flow of gas–liquid mixtures through pipes [1-3] note has been taken of the fact that the heat transfer coefficient is higher in a liquid containing gas bubbles than in a one-phase liquid flowing at the same rate. This higher heat transfer coefficient is accompanied by a depressed temperature field [4, 5], indicating a change in the turbulence characteristics of the stream.

Test data reveal that the presence of a gaseous phase has a much greater effect here than a higher velocity of the liquid. The authors of [6] have shown that the hydraulic drag in a stream increases when a gaseous phase is present. The extra pumping power to cover the drag loss in a stream with a low gas content would, however, be insufficient for also maintaining such a high heat transfer rate.

If the thickness of the thermal boundary layer δ_T in a gas–liquid stream is at a given value of the Prandtl number assumed to be proportional to the thickness of the hydraulic boundary layer δ_H, as in the case of a one-phase stream, then the heat transfer coefficient becomes approximately

$$\alpha \sim \frac{\lambda}{\delta_T} \sim \frac{\lambda}{\delta_H} \approx \frac{\lambda u_\tau}{\nu} \approx \frac{\lambda \bar{u}\sqrt{\frac{\xi}{8}}}{\nu}, \tag{1}$$

and, therefore, the ratio of heat transfer coefficients for a gas–liquid stream and a one-phase liquid stream respectively can be expressed as

$$\frac{\alpha}{\alpha_0} \approx \sqrt{\frac{\xi}{\xi_0}},$$

with ξ and ξ_0 denoting the drag coefficient in a gas–liquid stream and in a one-phase liquid stream respectively. An analysis of our results pertaining to heat transfer and of the results in [6] pertaining to hydraulic drag indicates that within the range of low gas constants the ratio α/α_0 can be several times higher than the ratio $\sqrt{\xi/\xi_0}$. For this reason, we rewrite expression (1) as

$$\alpha \approx \frac{\lambda \bar{u}\sqrt{\xi/8}}{\nu} f(\varphi), \tag{2}$$

with $f(\varphi)$ denoting some function of the gas content.

Evidently, an improvement in the diffusion characteristics of a stream is due to additional mechanical energy introduced into the stream by the relative motion of the phases under the action of Archimedes forces.

According to the author of [3], the energy dissipated in a gas–liquid stream consists of energy dissipated in the boundary layer (as in the case of a one-phase stream) and energy dissipated over the entire

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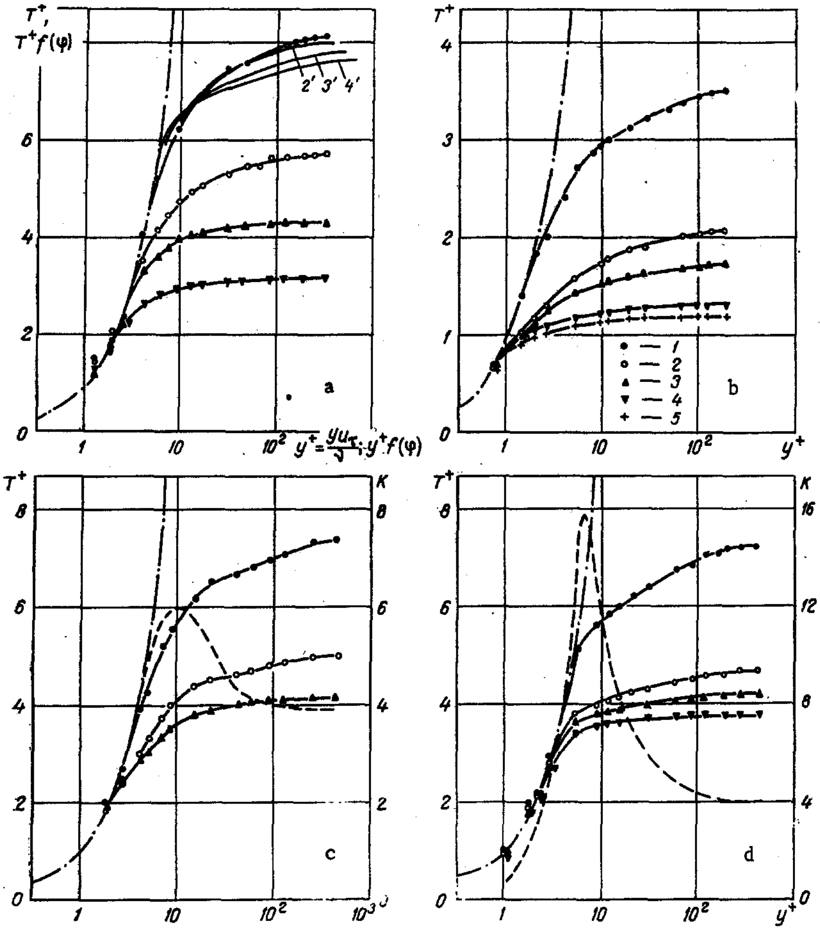

Fig. 1. Dimensionless temperature fields in the 50 mm (diameter) pipe: a) ascending stream with Re = $10 \cdot 10^3$ and 1) $\varphi = 0$, 2) 0.88%, 3) 2.25%, 4) 5.4%; b) ascending stream with Re = $5 \cdot 10^3$ and 1) $\varphi = 0$, 2) 0.4%, 3) 1.2%, 4) 3.0%, 5) 6.4%; c) ascending stream with Re = 15 $\cdot 10^3$ and 1) $\varphi = 0$, 2) 3.1%, 3) 3.5%; d) descending stream with Re = $15 \cdot 10^3$ and 1) $\varphi = 0$, 2) 0.56%, 3) 1.44%, 4) 3.2%; the dashed—dotted lines represent a constant temperature gradient near the wall; the dashed lines represent the ratio K of the coefficient of turbulent heat transfer in the stream with the highest gas content to the coefficient of turbulent heat transfer in a one-phase stream; the solid lines 2', 3', 4', represent the universal relations for the temperature fields in a gas—liquid stream at the respective levels of gas content, in $T^+f(\varphi)$, $y^+f(\varphi)$ coordinates.

volume by the relative motion of the phases. The dynamic velocity here is expressed as

$$u_\tau = \sqrt[4]{\left(\frac{\tau_0}{\rho}\right)^2 + \varkappa^4 \nu g W_{rel} \varphi (1-\varphi)^2} \quad . \tag{3}$$

The factor $(1-\varphi)^2$ has been introduced to account for the shielding effect of bubbles at the heat transfer surface, which impede the penetration of turbulent fluctuations from the mainstream to the boundary layer. With the empirical coefficient $\varkappa$ equal to 1.9–2.2, the author of [3] finds a close agreement with test results over a wide range of liquid and gas velocities.

With the problem formulated in those terms, however, it still remains unclear how the distribution of the gaseous phase over the stream section, the size of gas bubbles, and the flow pattern around the latter affect the heat transfer boosting mechanism directly.

It has been shown in [7] that the distribution of the gaseous phase over the stream section is determined by the direction and the mode of the gas—liquid mixture flow. During an ascending flow of a gas—liquid stream through a 50 mm (diameter) pipe with the Reynolds number Re > $14 \cdot 10^3$ and with

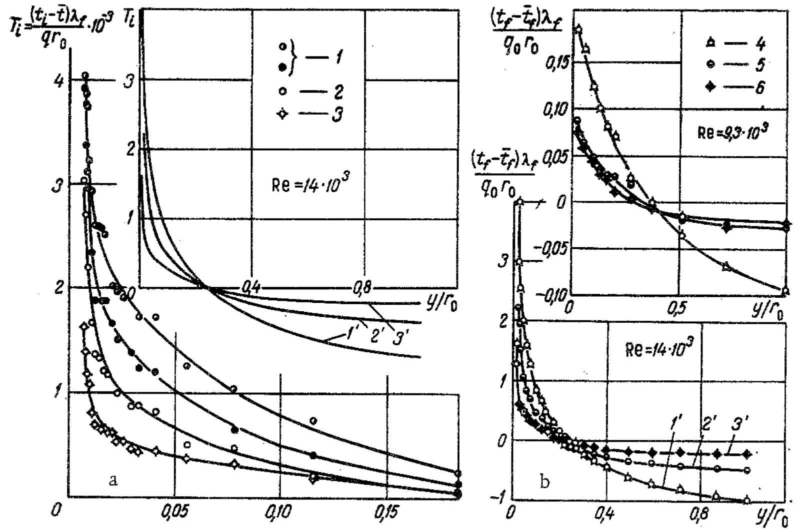

Fig. 2. Dimensionless temperature fields of an ascending stream of a gas—liquid mixture in the 28 mm (diameter) pipe: a) water—gas with 1') G_H/G_{mix} = 0 or 3%, 2') 8%, 3') 37.5%; b) mercury—gas with 1) β = 0, 2) 8%, 3) 37.5%, 4) 0, 5) 9.4%, 6) 23.4%.

effervescence, for example, the concentration of gas bubbles was highest at the wall and the nonuniformity of concentration, i. e., the ratio of concentration at the wall to concentration in the mainstream could be equal to as much as one order of magnitude. In a descending gas—liquid stream the concentration of bubbles was highest in the mainstream.

We will show here the results obtained in a study of heat transfer and average temperature fields in an anisothermal two-component stream with effervescence. The measurements were made in vertical circular pipes: one 50 mm in diameter (2.5 m long) and one 28.9 mm in diameter (2 m long). In the first pipe (50 mm) the average temperature field was determined at water flow rates corresponding to a Reynolds number Re = $5 \cdot 10^3$, $10 \cdot 10^3$, $15 \cdot 10^3$, $25 \cdot 10^3$, and $36 \cdot 10^3$ respectively, with the gas content ranging from 0.5 to 10% in an ascending stream. In a descending stream the temperature field was measured at a Reynolds number Re = $15 \cdot 10^3$, $25 \cdot 10^3$, and $36 \cdot 10^3$ respectively, with the gas content ranging up to 6.5%.

In the second pipe (28.9 mm) we measured not only the average sectional temperature fields but also the effect of the gas content level on the heat transfer coefficient, with the Reynolds number ranging from $10 \cdot 10^3$ to $30 \cdot 10^3$ and the gas content ranging up to 65%.

The heat transfer coefficient and the average temperature field were also measured in a mixture of mercury and gas bubbles flowing through the 28.9 mm pipe, with the Reynolds number ranging from $10 \cdot 10^3$ to $120 \cdot 10^3$ and the gas content ranging up to 28%.

The temperatures in the gas—water mixture were measured with Chromel—Alumel thermocouples having junctions 0.2 mm in diameter. These thermocouples had a frequency characteristic which made it feasible to record temperature fluctuations occurring at a frequency of 5-10 Hz. With the aid of a special probe, a thermocouple could be placed at any point across a channel section accurately within 0.1 mm. The temperature field was measured at a section three diameters before the exit.

Signals from the thermocouple probe were transmitted to a model ÉPP-09 potentiometer with 0.5 mV divisions and class 0.5 accuracy. The recording of a thermocouple signal lasted for 30-60 sec, so that the temperature reading could be properly averaged within an error not greater than 0.1°C.

In the mercury—gas stream the average temperature field was measured with Chromel—Alumel thermocouples inside a stainless steel capillary sheath with a 0.5 mm outside diameter. A thermocouple signal was measured with a model R 2/1 semiautomatic potentiometer with a sufficiently large inertia to allow average temperature readings to be taken at each point in the stream.

In Figs. 1 and 2 are shown the results of our studies pertaining to the effect of the gas content level on the average temperature field in a gas—water stream.

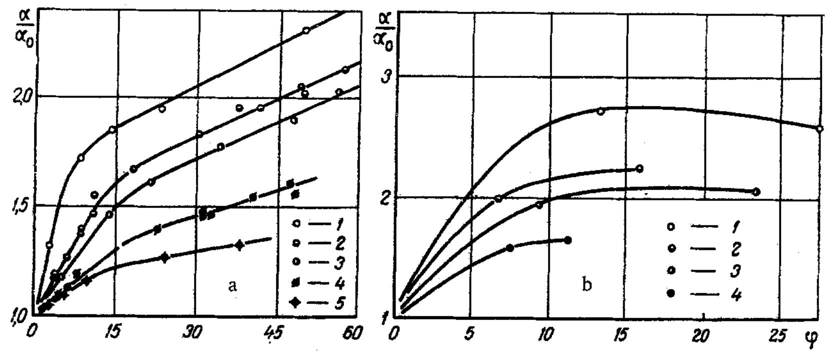

Fig. 3. Ratio of the heat transfer coefficient in a gas—liquid stream to the heat transfer coefficient in a one-phase stream, as a function of the gas content (φ, %), at various values of the Reynolds number: a) water—gas mixture with 1) Re = $11.9 \cdot 10^3$, 2) $14 \cdot 10^3$, 3) $15 \cdot 10^3$, 4) $20.5 \cdot 10^3$, 5) $30 \cdot 10^3$; b) mercury—gas mixture with 1) Re = $9.3 \cdot 10^3$ and W = 0.03 m/sec, 2) Re = $18.6 \cdot 10^3$ and W = 0.48 m/sec.

A comparison between the temperature fields in two-phase streams with various gas content levels indicates the basic trend: the temperature gradient decreases at all points in the stream, as the gas content increases. This decrease in the temperature gradients has to do with an increase in the turbulent heat transfer in a stream with gas bubbles.

As a result of these studies, we have established the effect of a gaseous phase on the heat transfer coefficient. There appear two ranges here. The heat transfer coefficient increases with increasing gas content, but faster when the gas content rises up to approximately 10% than when it continues to rise above that. The heat transfer improves, however, with increasing gas content φ throughout the entire test range (up to φ = 60%) (Fig. 3a).

The effect of a gaseous phase becomes weaker, as the flow rate of liquid increases, i. e., the additional turbulence in the stream due to the relative motion of the phases becomes less pronounced against the ambient turbulence of the stream.

With an increasing content in the mercury—gas mixture, the temperature field in our test varied along the same trend as in the case of the water—gas mixture.

The mercury—gas tests have revealed, however, that in certain flow modes the heat transfer coefficient increases to some maximum value and then decreases again with further increasing gas content (Fig. 3b).

For plotting the temperature fields, we have introduced coordinates which seem to us most universal.

Since the equation of the temperature field within the thermal boundary layer can be written as

$$(t - t_w) \approx \frac{q_0}{\lambda} y, \tag{4}$$

and the thickness of the thermal boundary layer is

$$\delta_T \approx \frac{\nu}{u_\tau f(\varphi)},$$

hence dividing (4) by δ_T and a few subsequent transformations will yield the required universal expression

$$\frac{(t - t_w)\, c_p \rho u_\tau f(\varphi)}{q_0 \,\mathrm{Pr}} \approx f_1 \left[\frac{y u_\tau}{\nu} f(\varphi) \right], \tag{5}$$

for plotting the temperature fields in Fig. 1.

In Fig. 4 are shown the values of $f(\varphi)$ at various values of the Reynolds number, based on the condition of approximate equality between the temperature field in a gas—liquid stream and that in a one-phase liquid stream. At low values of the Reynolds number the temperature field is a stronger function of the gas content than at rather high values of the Reynolds number. This effect depends on the direction of flow and on the diameter of the test channel.

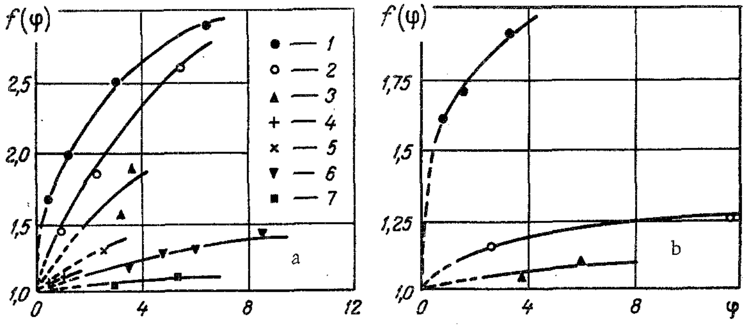

Fig. 4. Values of $f(\varphi)$ at various values of the Reynolds number: a) for an ascending stream through the 50 mm (diameter) pipe at 1) Re = $5 \cdot 10^3$, 2) $10 \cdot 10^3$, 3) $15 \cdot 10^3$, 4) 25 $\cdot 10^3$, through the 28.9 mm (diameter) pipe at 5) Re = $12 \cdot 10^3$, 6) $14 \cdot 10^3$, 7) $30 \cdot 10^3$; b) for a descending stream through the 50 mm (diameter) pipe at 1) Re = $15 \cdot 10^3$, 2) $25 \cdot 10^3$, 3) $36 \cdot 10^?$. Gas content φ (%).

Selecting the proper function $f(\varphi)$ for a stream with the respective gas content level means introducing an average correction factor K to the heat transfer coefficient in a one-phase stream:

$$\varepsilon_{aG} = \varepsilon_a K. \tag{6}$$

Coefficient K is related to $f(\varphi)$ and α/α_0 approximately as follows:

$$\frac{\varepsilon_{aG}}{\varepsilon_a} = K \approx f^3(\varphi) \tag{7}$$

and

$$\frac{\varepsilon_{aG}}{\varepsilon_a} = K \approx \left(\frac{\alpha}{\alpha_0}\right)^3. \tag{8}$$

These relations are based on the assumption that the coefficient of turbulent heat transfer is proportional to the distance from the wall cubed [8] and that this proportion remains valid in a gas—liquid stream. A comparison between Fig. 3 and Fig. 4 indicates that, on the average, the diffusion characteristics of a gas—liquid stream improve much more than the heat transfer.

A closer examination of the temperature fields has revealed that the coefficients of turbulent heat diffusion in a gas—liquid stream increase differently at different points in the stream.

In Fig. 1c, d has been plotted the ratio of temperature gradients in a stream without gas and in a stream with gas respectively. With gas in the stream, according to the graph, the coefficient of turbulent heat transfer increases most near the wall, whether the flow is ascending or descending.

Thus, the average gas content in a gas—liquid mixture is the main factor which governs the heat transfer, while the sectional distribution of phases, i. e., the inner structure of the stream has a relatively weak effect on the heat transfer rate.

Let us now consider a few possible mechanisms of heat transfer in an effervescent gas—liquid stream.

When a bubble of moderate size moves through a real liquid stream, then in the wake there appears a turbulent zone whose width is of the order of $Re_b^{-1/2}$ [9] and where the rate of velocity fluctuations is $W_{rel} Re_b^{-1/2}$.

The extra turbulent heat transfer due to these fluctuations is, according to Taylor's theory [10],

$$\varepsilon_a \approx \frac{1}{Pr_T} \sigma_V L m, \tag{9}$$

where σ_V denotes the rate of velocity fluctuations in the medium, L is a fluctuation scale factor assumed here equal to the width of the turbulence zone behind a bubble, m denotes the fraction of liquid volume flowing turbulently, $Re_b = W_{rel} d_b/\nu$, $d_b = 2R_b$ denotes the bubble diameter, W_{rel} denotes the velocity of bubble relative to the liquid, and Pr_T is the Prandtl number under turbulence conditions. An increase in the turbulent heat transfer due to turbulent flow behind bubbles is not an adequate explanation for our test results.

Our experiments with single bubbles in still water have shown that behind a bubble there appears not only a turbulence zone but also large-scale perturbations larger than such a bubble by one order of magnitude. Experiments with a two-phase anisothermal stream [4] have also revealed large-scale perturbations during effervescence, the dimension of these perturbations being several times larger than the average dimension of eddies in a one-phase stream.

On the basis of test data, we will assume that in an effervescent gas—liquid stream the turbulence due to the relative motion of bubbles represents an array of eddies with the average dimension L_0 and the maximum dimension equal to the distance between bubbles when the gas content is φ^* (transitional gas content level, at which the latter ceases to have a strong effect and begins to have a weak effect on the heat transfer) (Fig. 4).

We will thus assume, to a rough approximation, that at a gas content φ^* the average dimension of eddies is

$$L_0 = 2R_b \sqrt[3]{\frac{\pi}{6\varphi^*}} \tag{10}$$

when the bubbles are located at vertices of a cubic lattice. Considering that the average dimension of eddies remains equal to L_0 at a lower gas content $\varphi < \varphi^*$, we can write for the coefficient of turbulent heat transfer according to Taylor [10]:

$$\varepsilon_a \approx \frac{1}{Pr_T} \bar{V}_R L_0 \frac{V_{edd}}{V_{tot}} = \frac{1}{Pr_T} W_{rel} L_0 \frac{\varphi}{\varphi^*}, \tag{11}$$

and will note here that this coefficient increases linearly with the gas content.

It is to be expected that at a higher gas content $\varphi > \varphi^*$ the bubble dimensions will decrease as the distance between bubbles decreases. The coefficient of turbulent heat transfer is then

$$\varepsilon_a \approx \frac{1}{Pr_T} \bar{V}_R L = \frac{1}{Pr_T} W_{rel} R_b \varphi^{-\frac{1}{3}}. \tag{12}$$

Here $\bar{V}_R$ denotes the average radial velocity of bubbles, equal to W_{rel}. The coefficients of turbulent heat transfer based on these estimates are of the same order of magnitude as those obtained from temperature field measurements by graphical differentiation.

One could hardly expect, however, the average dimension of eddies in a real gas—liquid stream to vary exactly as the average distance between bubbles. Therefore, the relations

$$\varepsilon_a \sim \varphi \quad \text{when} \quad \varphi < \varphi^*,$$

$$\varepsilon_a \sim \varphi^{-\frac{1}{3}} \quad \text{when} \quad \varphi > \varphi^*$$

represent extreme variations in the coefficient of turbulent heat transfer ε_a as a function of the gas content φ. It appears from the test data that in a water—gas stream the coefficient of heat transfer continues to increase when $\varphi > \varphi^*$, although much slower than when $\varphi < \varphi^*$. This may mean that the eddies decrease in size at a slower rate than the distance between bubbles, as the gas content increases. The heat transfer coefficient decreases in a mercury—gas stream, however, according to the test data.

In an ascending gas—liquid stream under certain flow conditions the gas bubbles aggregate at the wall to such an extent that their concentration here may become by one order of magnitude higher than in the mainstream.

Tests have shown, however, that the turbulizing effect of these bubbles is more significant than their shielding effect.

Another mechanism of heat transfer from wall to mainstream is possible in such circumstances. Since gas bubbles at the wall move with a stream where the velocity gradient is large, hence in the stream past every bubble there may appear vapor eddies with opposite vorticity (analogous to edge eddies at a wing of a finite span). Such eddies separate from the wall and can facilitate the heat transfer from a hot wall to the mainstream.

NOTATION

δ_T	is the thickness of the thermal boundary layer;
δ_H	is the thickness of the hydraulic boundary layer;
α	is the heat transfer coefficient in a gas—liquid stream;
α_0	is the heat transfer coefficient in a one-phase stream;
u_τ	is the dynamic velocity;
$\bar{u}$	is the mean-over-the-section stream velocity;
q_0	is the thermal flux at the heat transfer surface;
λ	is the thermal conductivity of the liquid;
ν	is the kinematic viscosity of the liquid;
t	is the temperature of the liquid;
t_w	is the wall temperature;
y	is the distance from the wall;
$y^+ = yu_\tau/\nu$	is the dimensionless distance from the wall;
$T^+ = [(t-t_w)c_p\rho u_\tau]/q_0 Pr = \Delta t \lambda u_\tau/q_0\nu$	is the dimensionless temperature;
ε_{aG}	is the coefficient of turbulent heat transfer in a two-phase stream;
ε_a	is the coefficient of turbulent heat transfer in a one-phase stream;
K	is the mean-over-the-section ratio;
$\varepsilon_{aG}/\varepsilon_a$ and V_{edd}/V_{tot}	are the ratio of eddies volume to total liquid volume.

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