INTENSIFICATION OF TRANSFER PROCESSES IN A NONUNIFORM GAS-MIXTURE STREAM

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An evaluation is made of the influence of periodic nonuniformity of motion of a gas-mixture stream on interphase heat transfer. Optimum conditions for this process are derived.

It is known that heat transfer between an ascending gas stream and solid particles suspended in it over a steady section is determined by the value of the velocity $V_{rel} = V_{cr} \approx$ const. Therefore increase of the gas velocity usually leads not to increase of the amount of heat transferred (as in the case of fixed particles), but to a decrease (since the value of α does not change, because the relative velocity stays constant, while the dwell time of the particles in the equipment is decreased). Figure 1 shows the main features of an equipment which allowed increase of the relative rate of motion of gas and particles and thus enhanced heat transfer. A mixture of gases moves along a channel whose area changes periodically along its length. In order to decrease the lengths of the transition sections (diffusers and convergent sections), the whole channel, or, in some cases, individual parts of it, may be subdivided into a number of parallel branches. The gas velocity alters to correspond with changes in channel cross-sectional area, increasing in the narrow, and decreasing in the wide sections. The result is that the solid particles lag behind the gas velocity in the narrow sections and are gradually accelerated, while in the wide sections they overtake the gas and are slowed down. This kind of motion of a gas mixture stream is similar to that with a fluctuating gas stream [1, 2, 3], and therefore the technique developed in [3] may be used to evaluate heat transfer in the section examined. We shall make the following simplifying assumptions:

1. The particles, of constant mass, move vertically upwards, entrained by the isothermal gas stream.
2. The limiting case $s_2 \gg s_1$ is examined, where the stream velocity is V in section l_1 and zero in section l_2.
3. No account is taken of particle collisions with the walls of the tube and with other particles.
4. The aerodynamic drag and heat transfer of a particle are quasi-steady, i.e., at every instant of time they obey laws obtained under steady conditions.
5. The particle drag coefficient does not depend on gas velocity (self-similar region) and particle orientation.
6. The gas specific weight is negligibly small in comparison with that of the particle.
7. The equipment is made up of identical sections, and the particle motion in each section is identical (so-called "stationary" regime).

It was shown in [3] that under the assumptions made and with the conditions

$$g(\tau_1 + \tau_2)/V_{cr} < 0.2 \qquad (1)$$

or

$$g(l_1 + l_2)/uV_{cr} < 0.2 \qquad (1a)$$

the approximate expression for particle velocity in the equipment

$$\begin{aligned} \bar{u} &= (\bar{V} - \sqrt{\bar{V}^2 m - m^2 + 1})/(1-m), \quad m \neq 1; \\ \bar{u} &= (\bar{V}^2 - 2)/2\bar{V}, \quad m = 1, \end{aligned} \qquad (2)$$

is valid, where $\bar{u} = u/V_{cr}$ and $\bar{V} = V/N_{cr}$ are the dimensionless velocities of the particle and air. (Since u $\approx$ const, $l_2/l_1 \approx \tau_2/\tau_1$ = m.) By letting u $\to$ 0, we may determine, from (2), the minimum gas velocity in section l_1 required to transport the particle (the analog of velocity V_{cr} in equipment with fixed gas velocity):

$$\bar{V}_{min} = V_{min}/V_{cr} = \sqrt{1+m}. \qquad (3)$$

The physical meaning of (3) is clear: deceleration of a particle in section l_2 may be compensated for by an increase of velocity $V > V_{cr}$ in section l_1.

As in [3], from the condition $\alpha \sim V_{rel}^n$ we find the ratio of the mean heat transfer coefficients in sections l_1 and l_2

$$\frac{\alpha_m}{\alpha_0} = \frac{1}{m+1}[(\bar{V} - \bar{u})^n + m\bar{u}^n] \qquad (4)$$

of the particle dwell times in the equipment

$$\tau_{dw}/\tau_{dw_0} = (\bar{V}_0 - 1)/\bar{u} \qquad (5)$$

and of the amounts of heat taken in or given out by the particle

$$\frac{q_m}{q_0} = \frac{\alpha_m \tau_{dw}}{\alpha_0 \tau_{n_0}} = \frac{1}{m+1}[(\bar{V} - \bar{u})^n + m\bar{u}^n]\frac{\bar{V}_0 - 1}{\bar{u}}. \qquad (6)$$

From (6) we may compare the efficiency of heat transfer in the apparatus examined and in that with

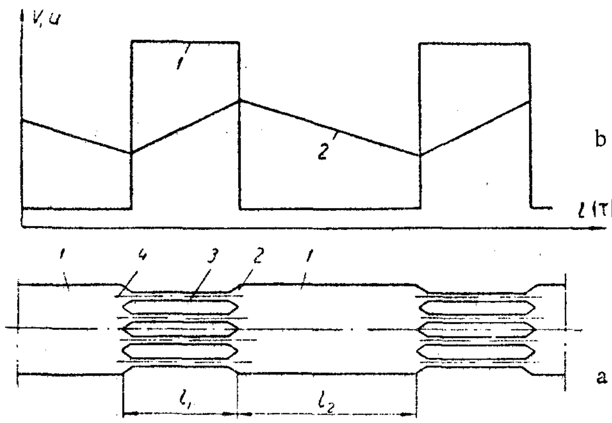

Fig. 1. a) Main features of the equipment [1) large-area section; 2) diffuser; 3) small-area section; 4) convergent channel] and b) graph of velocities [1) gas velocity; 2) particle velocity].

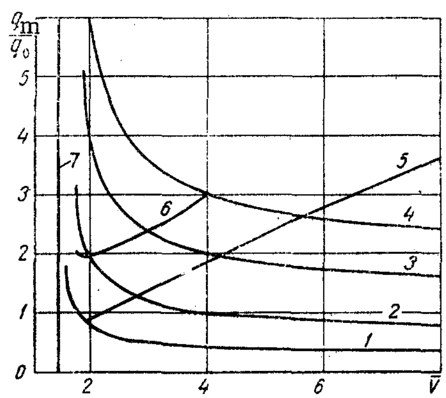

Fig. 2. Influence of gas velocity on heat transfer intensification ($l_2/l_1 = 1$): 1, 2, 3, 4, 5, and 6) with $\bar{V}_0 = \sqrt{2}$; 2; 3; 4; $\bar{V}/\sqrt{2}$ and $\bar{V}$, respectively; 7) with $\bar{V} = \bar{V}_{min}$.

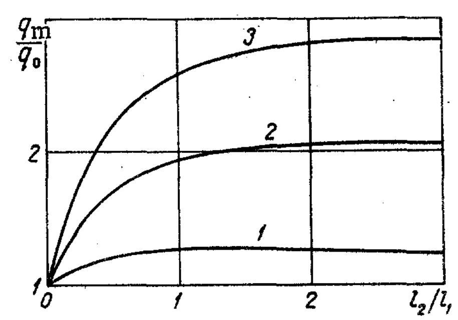

Fig. 3. Influence of the parameter $l_2/l_1 = m$ on heat transfer intensification ($\bar{V} = \bar{V}_0\sqrt{1 + l_2/l_1}$): 1, 2, 3) with $\bar{V}_0 = 2$, 3, and 4.

constant section area. To determine the most suitable value of velocity $\overline{V}$, values of $q_m/q_0 = f(\overline{V})$ were found from this formula, taking n = 0.8 and various fixed values of $\overline{V}_0$ and m (the results for m = 1 are shown in Fig. 2; curves for m = 2 and m = 3 are similar in shape). It may be seen from Fig. 2 that, for given $\overline{V}_0$, the ratio q_m/q_0 increases as $\overline{V}$ decreases. Lowering of $\overline{V}$, however, as is also true in the constant-area equipment, involves the possibility of the particles settling out, and is therefore restricted by the need to maintain a definite margin as regards the ratio to velocity V_{min}. Further calculations were therefore carried out for the condition $\overline{V}/\overline{V}_0 = \sqrt{1+m}$, i.e., with the same critical velocity safety factor in the two cases compared (Fig. 3). The calculations for $\overline{V}_0 = 2$ and $\overline{V}_0 = 3$ have already been done in [3], while those for $\overline{V}_0 = 4$ were done in the present work.

It may be seen from Fig. 3 that the quantity q_m/q_0 increases noticeably with increase of the ratio $m = l_2/l_1$ in the range $0 < l_2/l_1 < 1$. With further increase of l_2/l_1, the ratio q_m/q_0 remains practically constant. It is therefore expedient to use equipment with the ratio $l_2/l_1 = 1{-}1.5$. The results of the calculations presented in Fig. 3 also allow the conclusion to be drawn that the use of this method of enhancement of interphase heat transfer is expedient when the relative gas velocity $\overline{V}_0 > 2$.

The above analysis has explained in principle how interphase heat transfer in a uniform gas mixture stream may be enhanced. The region of practical use of this method will depend on the ratio between the gain in q and the additional expenditure of energy entailed by the increased flow friction in the system, and may be calculated with sufficient accuracy in each concrete case.

The authors have made an approximate comparison of the hydraulic losses in equipment with variable and constant section area. Using data from [4], we find the ratio

$$\frac{\Delta p}{\Delta p_0} = \frac{\lambda l_1/d + \zeta_d + \zeta_c}{\lambda_0 l_0/d_0} \left(\frac{V}{V_0}\right)^2. \tag{7}$$

Let $l_1 = l_2 = l$. Then, from the above equality $\overline{V}/\overline{V}_0 = \sqrt{1+m}$, it follows that when m = 1, $(V/V_0)^2 = 2$. We shall neglect the length of the convergent and divergent sections and put $l_1 + l_2 = 1$, $\lambda = \lambda_0$. When the gas volume flow rates are equal in two cases compared, we must have $d_0/d = \sqrt{kV/V_0}$ or $d_0/d = \sqrt{k}\sqrt{2}$. Hence,

$$\frac{\Delta p}{\Delta p_0} = \sqrt{k\sqrt{2}} + \frac{\zeta_d + \zeta_c}{\lambda l/d_0}. \tag{8}$$

It follows from (8) that $\Delta p/\Delta p_0$ increases with the number k of parallel sections. If we take k = 1,

$$\frac{\Delta p}{\Delta p_0} \approx 1.2 + \frac{\zeta_d + \zeta_c}{\lambda l/d_0}.$$

As an example, let $\zeta_d = 0.4$, $\zeta_c \approx 0$, $l = 1$ m, $d_0 = 0.1$ m, $\lambda = 0.03$. Then $\Delta p/\Delta p_0 \approx 2.5$.

Thus, under the assumed initial conditions, the heat transfer intensity may increase (Fig. 3) roughly in proportion to the increase of flow friction in the system. This result may be acknowledged as a very favorable one, since with other means of enhancing heat transfer, the increase in hydrodynamic resistance usually substantially outstrips the improvement in heat transfer. It is clear that the use of this method proves favorable mainly owing to the reduction in the size of the equipment, and hence in capital costs. From this point of view we should also look at the question of optimum number of channels in the increased velocity zone. According to (8), as the number of channels increases, the hydrodynamic resistance of the system increases, while the size and cost of the equipment decreases.

NOTATION

g—acceleration due to gravity; k—number of parallel narrow-section channels; l and s—length and cross-sectional area of a channel section; m—ratio of lengths l_2 and l_1; n—exponent; Δp—pressure drop of gas over channel; q—amount of heat transferred in the equipment; u—particle velocity; V—gas velocity; V_{cr}—critical velocity; ζ—local friction coefficient; λ—flow friction coefficient; τ—transit time of particles in section of channel; τ_{dw}—mean dwell time of a particle in the equipment; α—heat transfer coefficient. Subscripts: 0—in flow through constant-area equipment; 1 and 2—narrow and wide sections, respectively; d—diffuser; c—convergent section; rel—relative velocity; m—mean over length $l_1 + l_2$; min—value when particle drift is zero; bar over a letter—dimensionless quantity.

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