EFFECT OF MAIN FACTORS ON MEAN HEAT TRANSFER OF STEADY AND OSCILLATING DUST–GAS FLOWS

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Experimental data relating to external heat transfer of an air–graphite mixture in different hydrodynamic conditions are examined.

The difference in the mechanisms of momentum and energy transfer in laminar and turbulent flows leads to a different effect of external hydrodynamic pulsations on them [1].

We can expect a similar result for disperse through flows. In view of this it is worthwhile to examine separately the effect of imposition of pulsations on a laminar flow (pulsating laminar flow) and on a turbulent flow (pulsating turbulent flow). The change in flow rate is recurrent and, hence, on the average the heat transfer picture is constant in time and is characterized by a mean heat transfer coefficient for the particular flow regime. Thus, a periodically oscillating disperse flow can be regarded as quasistationary.

Investigations on heat transfer to a flow of air containing particles of natural graphite with a weighted mean diameter of 9.3 μm were made on a closed apparatus (Fig. 1), which provided for combined flow of the phases in the circuit and regulation of the particle concentration in a wide range.

The two-component mixture was impelled by a rotary blower from a YaMZ-205 motor with an adjustable electric drive. The air–graphite suspension was driven through a settling section and the test channel, which was heated on the outside by an alternating electric current, and then entered a shell-and-tube heat exchanger 6, where it was cooled with water. After the heat exchanger the gas suspension passed through a heat-labeling section 9 and returned to the blower.

The test channel 4 consisted of a tube of 1Kh18N10T stainless steel with internal diameter 26 mm, length 1250 mm, and wall thickness 2 mm. The outer diameter of the channel, including the layer of heat insulation, was 130 mm. The flow temperature was measured with C–C thermocouples in eight sections along the channel. At a distance of 150 mm from the exit section there was a movable fast C-A thermocouple with outside insulation diameter 1 mm and a thermoelectrode wire of diameter 0.1 mm, which enabled us to measure the temperature distribution over the tube cross section.

The temperature of the channel walls was measured with C-A thermocouples in the same cross sections as the flow temperature. As the measurements showed, there were no pulsations of the wall temperature in the experimental conditions, since the steel tube used as the test channel, together with the external insulation and heater, constituted a massive body with large bulk heat capacity. The emf of the thermocouples was measured to an accuracy of 0.02 mV with a potentiometer. In all the experiments the heat flux density was constant over the length of the channel. Heat loss to the surroundings was compensated by an additional heater and was 8-14%. The power consumption of the heaters (5 kW) was varied by means of an RNO-250 and was measured through a UTT-6M current transformer by D 566T ammeters and ASTV voltmeters. The flow rate of the cooling water passing through the heat exchanger was measured with a double diaphragm and was kept constant during the experiment.

To test the measurement technique we first conducted experiments with pure air. Flow pulsations with a frequency of 0.7-4 Hz were produced by a pulsator mounted after the test channel at a distance of 0.65 m (25 diameters). The pulsator consisted of a sealed case containing a rotating cylinder with a radial hole equal to the internal diameter of the tube. With this type of pulsator we could obtain almost sinusoidal oscillations of pressure and flow rate.

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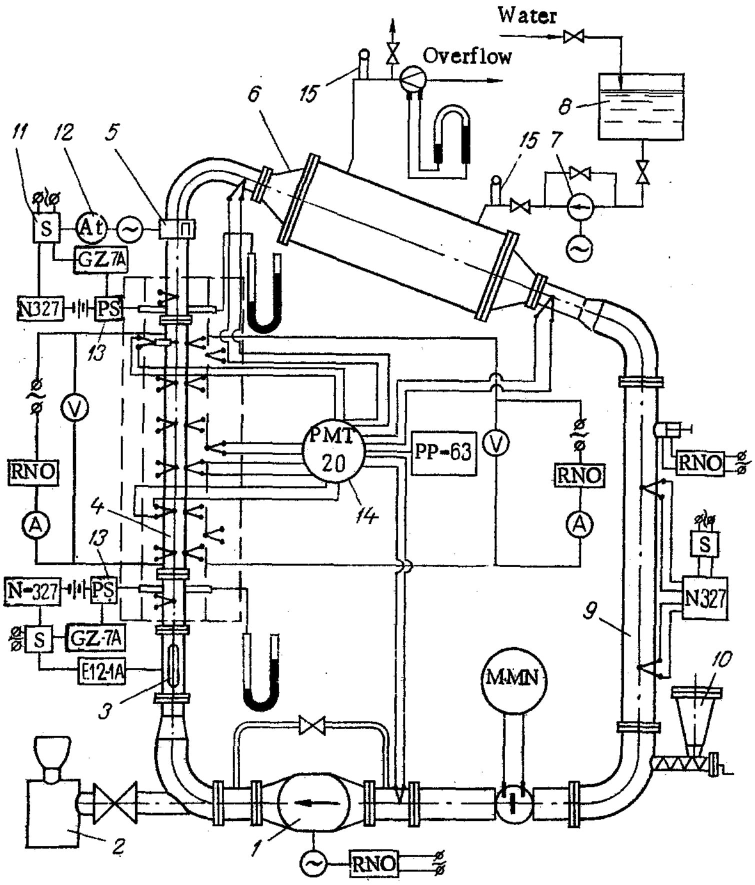

Fig. 1. Schematic of experimental apparatus and measurements of main quantities: 1) blower; 2) dust collector; 3) concentration gauge; 4) test channel; 5) pulsator; 6) heat exchanger; 7) air pump; 8) water tank; 9) heat-labeling section; 10) screw feeder; 11) voltage stabilizer; 12) autotransformer; 13) capacitive pressure sensor; 14) thermocouple switch; 15) mercury thermometer.

Special vibration tests of the experimental apparatus by the impact method with the aid of an INA-3 frequency meter showed that the natural vibration frequency of the apparatus was two orders greater than the frequency of the artificially imposed flow pulsations and was 120 Hz.

The amplitude of the flow pressure oscillations was measured at the entrance and exit of the experimental channel by capacitive sensors, which were powered by GZ-7A high-frequency signal generators. The recording instrument was an N-327 automatic recorder. The sensors were calibrated in static conditions.

The mixture flow rate was determined from the heat-balance equation written for the cooler, and the air flow rate was determined from the known velocity of the particles and air by the heat-label method [2] with due allowance for the velocity slip of the components. The velocity slip was determined in special experiments and did not exceed 13%. In addition, for control of the dust content in the test channel we measured the true volumetric concentration with a VNIIMT concentration gauge, the main component of which was a transducer in the form of two coaxial cylinders 400 mm long.

The diameter of the inner cylinder, which had tapering ends, was half the diameter of the outer cylinder, whose diameter was equal to the internal diameter of the test channel. The transducer was calibrated by installing in it a paper cylinder, which was filled with air-dry wadding thoroughly mixed with graphite particles. Using an E-12-1A capacitance and inductance meter we measured the capacitance corresponding to the amount of dust present in the transducer. From the known value of β and the velocity of the components it was easy to calculate μ.

A comparison of the two methods of concentration determination showed good agreement of the measurement results up to $\mu = 40$ kg/kg and thus confirmed that μ could be reliably determined by reduction of the heat balance. When $\mu > 40$ kg/kg the difference in the results reached 30-40%, which can be attributed to the nonlinearity of the calibration relationship (capacitance against volumetric concentration, $\beta > 0.02$) of the concentration gauge transducer.

In the treatment of the experimental data we introduced into the calculations the mass flow concentration μ, which was defined as the ratio of the mass flows of solid particles and air. Henceforth in the text the term "concentration" means the mass flow concentration.

The experimental procedure was as follows: The apparatus was first heated with pure air for 40-60 min, then the graphite was admitted, and the compensation heater (and in the case of a pulsating flow, the pulsator too) was switched on. When the mean temperatures of the tube wall and inside insulation (thickness $\delta = 30$ mm) became equal we took the readings of the instruments and determined the particle velocity in the heat-labeling section. The duration of one experiment was 2.5-3 h. At the end of the experiment the graphite was blown out and weighed.

The mean heat transfer coefficient over the length of the test channel for steady and pulsating air–graphite suspensions was determined from the relation

$$\alpha = Q_{\text{heat}}/\pi D l\,(\bar{t}_{\text{w}} - \bar{t}_{\text{f}}), \tag{1}$$

where

$$\bar{t}_{\text{w}} = \sum_{i=1}^{n} t_i\,(l_i + l_{i-1}) \Big/ \sum_{i=1}^{n} l_i. \tag{2}$$

As a characteristic temperature we took the flow temperature averaged over the length of the test channel, and as a characteristic length we took the internal diameter of the tube.

To determine the effect of channel length on heat transfer we determined the wall temperature in five cross sections as the weighted mean up to the particular cross section, and the flow temperature as the arithmetic mean temperature. The calculated heat flux was assumed to be constant over the entire heating surface.

The general organization of the measurements of the primary quantities enabled us to calculate the experimental values of the heat transfer coefficients in relation to the hydrodynamic conditions with an error of 5 to 9%.

In experiments on heat transfer of air and air–graphite steady and pulsating flows the main parameters were in the following ranges: the time averaged air flow rate 5-21 m/sec; the solid-particle concentration 0-95 kg/kg; the mean flow temperature 38-80°C; the channel wall temperature 48-230°C; the heat flux density $(1.43\text{-}11.3)\cdot 10^3$ W/m²; Re 6600-31500; frequency of flow and pressure pulsations 0-4 Hz; relative amplitude of oscillations 20-90%.

The experimental investigation of the heat transfer of a steady suspension in a wide range of flow rates enabled us to find the known relationship for the turbulent regime: $\alpha_m/\alpha \sim \text{Re}^{-0.3}$, previously obtained in [3-5]. Hence, irrespective of the orientation of the tube in space the value of α_m in dust–gas flows increases with increase in Re, though to a smaller extent than for pure air.

Treatment of the experimental data in the form of the relation $\text{Nu}_m/\text{Nu} = \varphi(\mu)$ confirmed [3] the existence of a critical concentration ($\mu \approx 60$ kg/kg) at which there was a significant change in the effect of concentration on heat transfer. In addition, the index of the power of the concentration in the second region ($\mu > 60$ kg/kg) agreed exactly with that in [3] and was 1.33.

This kind of variation of heat transfer with increase in concentration can be attributed to the fact that in the fluid flow with $\mu > \mu_{cr}$ the flow structure is qualitatively altered owing to its confinement. The sharp reduction of the thermal resistance of the wall zone is of great significance, since the particles penetrate into it. The number of collisions of the particles with the wall increases, which leads to equalization of their temperatures [3,4].

As a result, the intensification of the heat transfer sharply increases (when $\mu = 94$ kg/kg the ratio $\alpha_m/\alpha = 27.5$) in comparison with a gas suspension, where $\alpha_m/\alpha = 7\text{-}13$, depending on the degree of saturation of the flow with solid particles.

Thus, by determining the effect of such main factors as the velocity of the carrier gas and the solid phase concentration on the rate of heat transfer of a suspension we were able to obtain general empirical correlations and use them to obtain a clearer picture of the effect of flow rate oscillations on the heat transfer coefficient.

We first conducted experiments on heat transfer of a pulsating flow of pure air.

We found, as in a previous investigation [6], that the increase in heat transfer due to imposition of periodic oscillations on a turbulent flow of pure air was 2-7% and depended on the rate of rotation of the pulsator and the flow velocity. This increase lies within the limits of accuracy of the experiment. However, when the imposed frequency is equal to the frequency of ejection of eddy structures from the wall zone ($\omega = 10^{-7}\ Re^{1.75}$) [7] we can expect a shift in the heat transfer maximum with increase in Re towards higher frequencies and an increase in its rate up to 15-20%, since the generated turbulence is not localized in the wall zone, but spreads to the tube axis [8], which was apparently observed in Agadzhanyan's experiments [9].

We detected no effect of amplitude of the flow pulsations in the investigated range on heat transfer and in a first approximation we can neglect such an effect. Indirect confirmation of the weak effect of the amplitude of velocity pulsations on the heat transfer of through disperse media is provided by investigations of external heat transfer in an inhomogeneous fluidized bed, which is a self-oscillatory system, whose inhomogeneity is manifested in periodic low-frequency oscillations of the void fraction and heat transfer rate. Kulikov et al. [10] found that when the amplitude of the pulsations of the gas velocity was less than its mean value the instantaneous values of the heat transfer coefficients differed from the mean values by not more than 6-8%.

The main factors affecting the heat transfer coefficient averaged in time and over the length of the tube (q_ω = const) for a pulsating disperse flow are the solid-particle concentration, gas velocity (constant component), channel diameter, particle size, frequency of flow (pressure) pulsations, and the thermophysical properties of the particles and gas.

We found that the adverse effect of increase in velocity of the carrier medium on the intensifying role of the particles is more pronounced in a pulsating flow ($\alpha_{p.m}/\alpha \sim Re^{-0.4}$) than in a steady flow, for which the power of Re is −0.3. It is apparent from this that the presence of small particles in a pulsating flow leads to greater laminarization of the flow, i.e., to greater suppression of the turbulent pulsations of the carrier medium ($\alpha_{p.m} \sim Re^{0.4}$).

The effect of solid-particle concentration (in the region $\mu < \mu_{cr}$) on the relative heat transfer rate $\alpha_{p.m}/\alpha$ is proportional to $\mu^{0.375}$, and for a steady flow to $\mu^{0.435}$. It should be noted that the critical value of the concentration in a pulsating turbulent flow ($\mu_{cr} \approx 47$ kg/kg) is appreciably lower than for a steady flow (60 kg/kg); when $\mu > 47$ kg/kg the ratio $\alpha_{p.m}/\alpha \sim \mu^{1.13}$. Hence, the effect of concentration on $\alpha_{p.m}/\alpha$ is much weaker than for a steady flow.

It was shown in [11] that the greatest value of the ratio $\alpha_{p.m}/\alpha_m = 1.3$-$1.4$ is obtained mainly in the region of frequencies up to 5 Hz and concentration up to 10 kg/kg. Hence, for a highly concentrated suspension forced low-frequency pulsations of the flow were created.

In this case, however, a slight increase in heat transfer (up to 10%) due to forced pulsations is detected up to the critical concentration, but with further increase in concentration of particles in the flow the effect of pulsations on the external heat transfer is reduced and at a concentration close to optimal ($\mu \approx 100$ kg/kg) this effect is barely detectable owing to the fact that the increase in flow density at f = 0 is accompanied by large-scale unsteady flow in the stream core [12] and the artificial imposition of forced oscillations does not lead to intensification of the propagation of such a flow towards the tube wall.

The effect of Sh and the confinement factor D/d_T on heat transfer was considered in conjunction with the experimental data that we previously obtained in an investigation of heat transfer in a flow of air carrying 60-μm electrocorundum particles in a tube with D = 8 mm and f varying from 0 to 24 Hz. This correlation of the experimental data revealed that in the case of turbulent flow $\alpha_{p.m}/\alpha \sim Sh^{-0.057} \cdot (D/d_T)^{0.5}$ and $\alpha_m/\alpha \sim (D/d_T)^{0.47}$. The latter is in good agreement with the result obtained in [3], where the experimental data of many authors was correlated by means of the relation $\alpha_m/\alpha \sim (D/d_T)^{0.43}$. It should be noted that with increase in the channel diameter the absolute values of the heat transfer coefficients

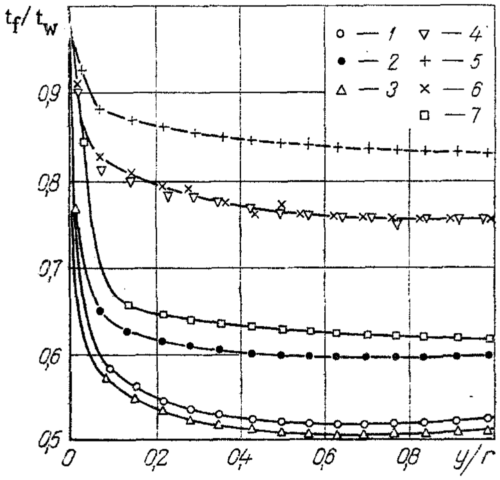

Fig. 2. Change in temperature of two-component flow over tube radius. Steady flow: 1) μ = 33.2, Re = 11210; 2) 52 and 10190. Pulsating flow f = 4 Hz; 3) μ = 26.1, Re = 17300; 4) 72.9 and 10400; 5) 91.5 and 9820. Pulsating flow, f = 1 Hz: 6) μ = 71.2, Re = 11210; 7) 52 and 9600.

for pulsating and steady flows decrease, to a greater extent in the first case: $\alpha_{p.m} \sim D^{-0.157}$ and $\alpha_m \sim D^{-0.03}$. Thus, the effect of channel diameter on heat transfer is much less for dust—gas flows than for pure air, where $\alpha \sim D^{-0.2}$.

The nature of the variation of the mean heat transfer coefficients of the suspension over the length of the experimental tube indicates that with increase in concentration of the graphite particles the relative heat transfer at the start of the tube is significantly higher, and to a greater extent for a steady flow. The greatest change in relative heat transfer coefficients occurs when z/D < 20. At a distance of almost 30 diameters from the start of the heated experimental channel thermal stabilization of the flow occurs, irrespective of the concentration of solid phase. For a better understanding of the mechanism of heat transfer of pulsating turbulent flows we investigated the temperature fields in the cross section of the tube, which showed that the introduction of particles into the flow led to leveling out of the temperature profile (Fig. 2). The intercomponent temperature difference in the stream core is practically zero. Thus, intercomponent heat transfer is fairly intense (this is confirmed by the theoretical data of [13]), and the increase in thermal resistance of the core of a fluid suspension, as usually happens when f = 0, does not take place, since the low-frequency oscillations also create a large-scale transverse motion of the particles and improves the mixing in the central part of the tube.

The results of the investigations of heat transfer of turbulent dust—gas flows are satisfactorily represented by the following empirical relations:

$$\alpha_m/\alpha = c\,\mathrm{Re}^{-0.3}\mu^m (D/d_T)^{0.47}, \tag{3}$$

$$\alpha_{p.m}/\alpha = c\,\mathrm{Re}^{-0.4}\mu^m (D/d_T)^{0.5}\,\mathrm{Sh}^{-0.057}, \tag{4}$$

where the heat transfer coefficient of pure air is found with an error of up to 13% from the known approximating correlation [14]

$$\mathrm{Nu} = 0.018\,\mathrm{Re}^{0.8}; \tag{5}$$

in Eq. (3) for region I μ = 1-59 kg/kg, Re = 5000-17000, with an error of 14% c = 0.86, m = 0.435; for region II 59 < μ < 95, Re = 9800-16800, and with error 12.5% c = 0.0219, m = 1.33.

Using Eq. (3) we correlated the empirical heat transfer data of other authors for μ varying from 0.6 to 45 kg/kg, Re $\geqslant$ 5000, D/d_T = 117.5-2790. The mean deviation of the experimental points from the curve represented by Eq. (3) was ±15.7% and the maximum deviation was 30%.

In Eq. (4): for region I μ = 0.7-47.0 kg/kg, Re = 4000-18000, Sh = $(0.45\text{-}31.5)\cdot 10^{-3}$, and with an error of 5.8% $c = 1.58$, $m = 0.375$; for region II $47 < \mu < 95$, Re = 9200-13900, Sh = $(3.4-16.5)\cdot 10^{-3}$, and with an error of 6.7% $c = 0.087$, $m = 1.13$.

It should be noted in conclusion that the factors having the greatest effect on the mean heat transfer of a pulsating dust—gas flow, and also on a steady flow, are the time-averaged gas phase velocity, the concentration and size of the solid particles, and the channel length.

The imposition of periodic flow oscillations leads to greater suppression of the turbulence of the carrier flow, to reduction of the effect of concentration, to an increase in the effect of the confinement factor on heat transfer, and also to more uniform distribution of the heat transfer coefficients along the tube. At the same time, an increase in the frequency of induced pulsations probably leads to an increase in the thermal resistance of the flow core due to reduction of the turbulent heat transfer coefficient [11].

Thus, as a heat-transfer agent a pulsating flow has several special features, which have to be taken into account in practical calculations.

NOTATION

α, heat transfer coefficient of gas; α_m, heat transfer coefficient of steady flow of gas suspension; $\alpha_{p.m}$, heat transfer coefficient of pulsating flow of gas suspension; D, channel diameter; d_T, particle diameter; μ, mass flow concentration of particles; w, gas velocity; f, frequency of flow (pressure) pulsations; $Re = wD/\nu$, Reynolds number; ν, kinematic viscosity of gas; $\underline{Sh} = fD/w$, Strouhal number; $Nu = \alpha D/\underline{\lambda}$, Nusselt number; Q, amount of heat; l, channel length; $\overline{t}_w$, weighted-mean wall temperature; $\overline{t}_f$, arithmetic mean flow temperature; β, true volumetric concentration of particles.

LITERATURE CITED

1. B. M. Galitseiskii, Yu. A. Ryzhov, and E. V. Yakush, Thermal and Hydrodynamic Processes in Oscillating Flows [in Russian], Mashinostroenie, Moscow (1977).
2. G. P. Katys, Volume Flowmeters [in Russian], Énergiya, Moscow—Leningrad (1965).
3. N. I. Syromyatnikov and V. S. Nosov, Dokl. Akad. Nauk SSSR, 163, No. 3 (1965).
4. Z. R. Gorbis, Heat Transfer and Hydromechanics of Disperse Through Flows [in Russian], Énergiya, Moscow (1970).
5. V. V. Mamaev, V. S. Nosov, and N. I. Syromyatnikov, Inzh.-Fiz. Zh., 31, No. 4 (1976).
6. V. V. Mamaev, V. S. Nosov, and N. I. Syromyatnikov, Izv. Vyssh. Uchebn. Zaved., Energ., No. 9 (1975).
7. E. R. Corino and R. S. Brodkey, J. Fluid Mech., 37, No. 1 (1969).
8. Mizushina Tokuro et al., J. Chem. Eng. Jpn., 8, No. 3 (1975).
9. G. G. Agadzhanyan, in: Theory of Similarity and Simulation [in Russian], Izd. Akad. Nauk SSSR (1951).
10. V. M. Kulikov, V. S. Nosov, V. N. Korolev, and N. I. Syromyatnikov, Inzh.-Fiz. Zh., 30, No. 6 (1976).
11. V. V. Mamaev, V. S. Nosov, and N. I. Syromyatnikov, Inzh.-Fiz. Zh., 31, No. 5 (1976).
12. R. Busroid, Flow of Gas with Suspended Particles [in Russian], Mir, Moscow (1975).
13. L. D. Kapusta, M. I. Rabinovich, and A. A. Shraiber, Inzh.-Fiz. Zh., 10, No. 6 (1966).
14. V. P. Isachenko, V. A. Osipova, and A. S. Sukomel, Heat Transfer [in Russian], Énergiya, Moscow (1970).