EXPERIMENTAL INVESTIGATION OF CONVECTIVE HEAT TRANSFER DURING COMBUSTION OF OPPOSED GAS–AIR JETS

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The results of experiments on the heat transfer between jets burning in an enclosed space and a cylindrical calorimeter are given. A semiempirical expression for calculating the heat transfer in cyclone and swirl chambers is proposed.

The case of heat transfer considered here, which can occur in rapid-heating industrial furnaces, has hardly been investigated. The experiments are performed in a lined chamber (Fig. 1) with a square 170 × 170 mm cross section and cut-off corners, the length of which is equal to 460 mm. The outside steel jacket of the chamber is cooled by water. The combustion products are drawn off through one of the end faces. A water-cooled sectionalized (along the periphery) calorimeter with a length of 300 mm and a diameter of 103 mm is positioned along the axis of the chamber. The calorimeter is provided with protective water-cooled calorimeters on its end faces.

A "hot" and a "cold" calorimeter, made of 1Kh18N9T steel, are both used as the central measuring calorimeter in experiments. The wall temperature of the "hot" calorimeter in experiments is equal to 500-1000°C, while the wall temperature of the "cold" calorimeter is equal to 100-500°C. The wall temperature is measured by means of eight Chromel–Alumel thermocouples, which are embedded to a depth of 3 mm along the perimeter. The surface temperature of the "hot" calorimeter is regulated by supplying hydrogen and carbon dioxide into the annular slot, packed with corundum sand, between the outside and inside water-cooled frames of the calorimeter. In order to impart stable radiation characteristics to the surface of the "hot" calorimeter, it was first oxidized, while the surface of the "cold" calorimeter was polished mechanically from time to time to ensure a low degree of blackness. The latter made it possible to reduce the error connected with the calculation of radiant heat transfer.

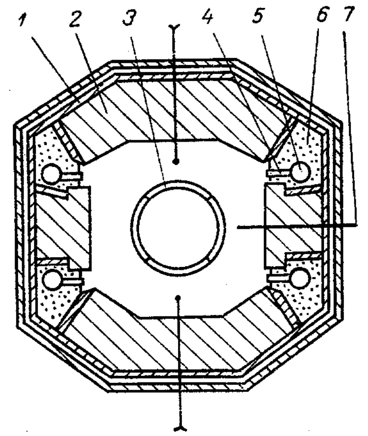

Fig. 1. Experimental chamber (transverse cross section): 1) water-cooled jacket; 2) lining; 3) calorimeter; 4) nozzles; 5) header; 6) insulation; 7) thermocouples (nine pieces).

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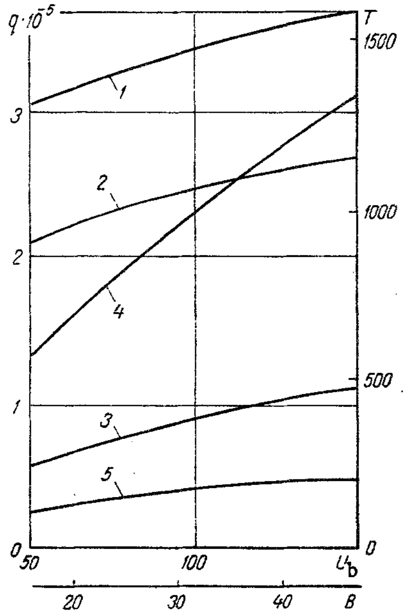

Fig. 2

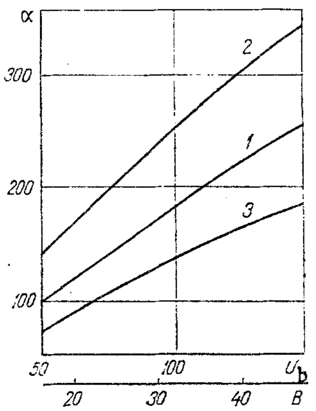

Fig. 3

Fig. 2. Dependences of the temperature T (°C) and the specific thermal flux q (W/m^2) on the jet velocity U_b (nm/sec) and the gas discharge nm^3/h) per m of furnace length: 1, 2, and 3) mean temperatures of the gas, the lining, and the calorimeter surfaces, respectively; 4) and 5) mean total and radiant heat fluxes.

Fig. 3. Dependences of the coefficients of convective heat transfer α (W/m$^2\cdot$deg) on the jet velocity U_b (nm/sec) and the gas discharge B (nm^3/h) per m of furnace length: 1, 2, and 3) heat-transfer coefficient, averaged along the perimeter, coefficient of heat transfer to the upper section, and coefficient of heat transfer to the lateral section, respectively.

Four headers, using interchangeable nozzles with diameters of 3 and 4.5 mm, spaced at 30 and 60 mm, are mounted in the side walls of the chamber. A previously prepared gas–air mixture (natural gas and air) is supplied to the chamber through the nozzles, whereby two systems of opposed jets, above and below the calorimeter, are produced.

The temperature of the chamber inside surface is measured by means of nine PP-1 thermocouples, while the gas temperature in the chamber is measured by means of nine PR-30/6 thermocouples with bare junctions, the diameter of which is equal to 0.17 mm. The gas temperature is measured at three sections along the chamber length, above, on the side of, and below the calorimeter. The thermocouple readings are adjusted by introducing corrections obtained by comparison with readings of a draw-off thermocouple. It is known that combustion at the surface of a platinum junction provides an exaggerated gas temperature. Therefore, the measurements are performed with a draw-off thermocouple whose junction is covered with quartz with a thickness of 5-10 μ (the shock wave deposition of quartz was performed at the Gas Institute, Academy of Sciences of the Ukrainian SSR).

By processing the experimental data, we determined the total thermal fluxes to the calorimeter sections (with respect to the heat absorbed by the cooling water) and calculated the radiant heat fluxes (with respect to the experimentally measured temperatures of the calorimeter, the lining, and the gas and the degrees of blackness, determined on the basis of the material, its finish, and its temperature according to data from [5-7]). The coefficient of heat transfer was calculated by means of the expression

$$\alpha = (q_\Sigma - q_r)/(T - T_r). \tag{1}$$

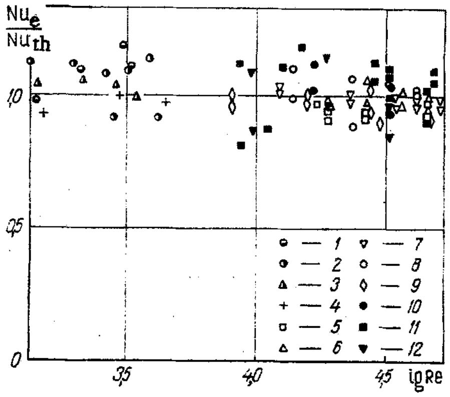

Fig. 4. Comparison between experimental data and calculations based on (6) concerning the heat transfer in swirl and cyclone chambers; our experiments: 1) chamber dimensions (cross section and length), 170 × 170 × 460 mm; calorimeter diameter, 103 mm; number of calorimeters, 1; burner (nozzle) diameter, 3 mm; number of burners (nozzles), 56; surface temperature of calorimeter, 500-1000°C; 2) 170 × 170 × 460; 103; 1; 3; 56; 200-500; 3) 170 × 170 × 460; 103; 1; 4.5; 28; 200-500; 4) 170 × 170 × 460; 100 × 100; 1; 12.5, 24; 100-1000; respectively; experiments in [2]: 5) 150 diameter ×209; 47; 2; 50; 1; 100, respectively; 6) 150 × 209; 23; 5; 2; 42; 5; 1; respectively; 7) 300 × 396; 90; 2; 100; 1; 100 respectively; 8) 300 × 396; 46.5; 2; 100; 1; 100, respectively; 9) 450 × 594; 109; 2; 150; 1; 100, respectively; experiments in [3]: 10) 265 × 504; 42; 2; 108; 1; 11) 265 × 504; 42; 2; 108; 2; 12) 265 × 504; 42; 2; 87; 1.

Experiments show that combustion of a gas—air mixture in a system of opposed jets ensures intensive gas combustion and large values of the heat-transfer coefficient. Ignition in such a system is ensured not so much by recirculation (as in ordinary burners) as by powerful forced supply of high-temperature combustion products to the roots of the opposed jets. Temperature measurements along the axes of the jets have shown that the initially cold jet (300-400°) is heated to 800-1000°C after moving through a distance equal to 7-10 calibers (25-30 mm), so that an intensive combustion process ensues, which ends at a distance equal to 20-25 calibers (60-70 mm). Temperature measurements along the chamber above the upper and below the lower calorimeter generatrices have not revealed marked temperature differences over distances of the order of the nozzle spacing (30 mm). For a 60-mm spacing, the jet combustion process is prolonged, and the heat transfer deteriorates (therefore, only the results for the 30-mm spacing are given below). In all cases, gas analysis of the combustion products leaving the chamber has shown that the combustion is virtually complete for air excess factors of 1.02-1.4.

The thermal stress in the chamber reaches $30 \cdot 10^6$ W/m^2, while the maximum thermal flux (directed toward the upper section) is equal to $4.3 \cdot 10^5$ W/m^2 for a wall temperature of the calorimeter section equal to 550°C. Figure 2 shows the dependences of the mean temperature and of the total and the radiant heat fluxes (averaged along the perimeter) on the jet velocity and the gas discharge per lineal meter. The radiant heat flux in the experiments amounted to 20-55% of the total thermal flux for the "hot" calorimeter and 10-20% for the "cold" calorimeter.

Figure 3 shows the experimental data on the coefficients of convective heat transfer from the gases to the calorimeter, calculated on the basis of the mean gas temperature above a calorimeter section. The coefficient of convective heat transfer, averaged along the perimeter, is by 15-20% lower than the heat-transfer coefficient reduced to the theoretical temperature of combustion products obtained from the equation of heat balance.

The coefficient of convective heat transfer increases from 80-100 to 230-250 W/m$^2 \cdot$deg as the velocity increases from 50 to 150 nm/sec, respectively, which corresponds to gas discharge values of 16 and 48 nm^3/h per m of furnace length.

The maximum heat-transfer coefficient along the perimeter of the calorimeter occurs in the zone where the burning jets meet (i.e., in the lower and upper calorimeter sections); it exceeds by a factor of 1.4-1.6 the value in the lateral sections. The wall temperature of the calorimeter in the 100-1000°C range does not affect materially the coefficient of convective heat transfer; the effect of this temperature is described unambiguously by the ratio Pr_c/Pr_w.

As the amount of excess air increases, the coefficient of convective heat transfer diminishes, which is connected with changes in the thermophysical characteristics of the combustion products.

The heat transfer is not affected by changes in the nozzle diameter for equal discharges of the gas–air mixture and total nozzle areas.

We shall now introduce a dimensionless expression for the processing of experimental data. The flow in the chamber under consideration is very complex, and it obeys different laws in different parts of the chamber. The convective heat transfer at any section depends on both the interaction between the jet and the calorimeter and the mean velocity of combustion products at the given section, so that the convective heat transfer along the length and the perimeter of the chamber cannot be calculated. For practical calculation of such furnaces and combustion chambers, it is extremely important to be able to estimate at least the mean (with respect to all the heat-transfer surfaces of the chamber) coefficients of heat transfer.

The dimensionless relationships obtained by different authors [2-4] for the heat transfer in swirl and cyclone chambers show that, in spite of the complexity of the process, convective heat transfer in such chambers is on the whole determined by few parameters, which include, in the first place, ratios of the chamber diameter to the burner diameter and to the calorimeter diameter. The drawback of these purely empirical relationships is that they are in poor agreement with each other. We shall attempt to combine them into a single correlation by considering a simplified physical process in a channel. Actually, consider the chamber as a straight channel with the equivalent diameter D, where the gas flow has a certain mean velocity, determined by the expression

$$U_{me}=\frac{G}{\rho S_c};\quad S_c=\frac{1}{\sqrt{(K_G/S_t)^2+(K_s/S_L)^2}}, \tag{2}$$

where K_G is a part of the total discharge passing through the middle of the chamber (K_G = 1/2 for a discharge uniformly distributed along the length and supplied at the center, while K_G = 1 for a discharge supplied at the end face), and K_s is a coefficient accounting for the discharge distribution over the longitudinal section (K_s = 1 for flows in opposition, and K_s = 2 for one-sided tangential flow).

If we now use the well-known expression for heat transfer in channels [1],

$$\mathrm{Nu}=0.021\left(\frac{U_{me}D}{\nu_c}\right)^{0.8}\mathrm{Pr}_c^{0.43}\left(\frac{\mathrm{Pr}_c}{\mathrm{Pr}_w}\right)^{0.25}, \tag{3}$$

it can happen that the experimental coefficient of convective heat transfer exceeds by one order of magnitude the value calculated by means of this expression.

This can be explained by the fact that there are velocities in the chamber which greatly exceed the mean velocities (the initial jet velocities U_b in the burner), as a result of which the intensity of convective heat transfer increases. From the various tentative variants, the following definition of the theoretical velocity proved to be the most acceptable one:

$$U_{th}^2=K_1U_{me}^2+K_2U_{me}(U_b-U_{me}). \tag{4}$$

By substituting U_{th} from (4) for U_{me} in (3) and using the equation of continuity $\rho_b U_b S_b = \rho_c U_{me} S_c$, we obtain

$$\mathrm{Nu}_{th}=\mathrm{Nu}\left[K_1+K_2\left(\frac{S_cT_b}{S_bT_c}-1\right)\right]^{0.4}. \tag{5}$$

Comparison with experimental data suggests that the ratio S_c/F should figure in K_2, which would result in the following relationship:

$$\mathrm{Nu}_{th}=\mathrm{Nu}\left[1+267\frac{S_c}{F}\left(\frac{S_c}{S_b}\frac{T_b}{T_c}-1\right)\right]^{0.4}. \tag{6}$$

In Fig. 4, the results obtained by means of expression (6) are compared with our experimental data and the experimental data obtained by other authors [2-4] in the range

$$4 < \frac{S_c T_b}{S_b T_c} < 100.$$

In this article, the absolute values of the chamber dimensions and of the other parameters vary in fairly wide ranges: chamber diameter, 0.17-0.45 m; length, 0.2-0.6 m; burner (nozzle) diameter 3-150 mm; number of burners, 1-56; outflow velocity, 20-150 m/sec; jet temperature, 20-1500°C.

The discrepancies between the experimental data and the results obtained by means of (6) amount to less than 20%, which can be considered as satisfactory if the considerable ranges of parameters and their ratios are taken into account.

NOTATION

T	is the temperature;
q	is the thermal flux density;
α	is the coefficient of convective heat transfer;
G	is the mass discharge;
U	is the velocity;
ρ	is the density;
ν	is the kinematic viscosity;
D	is the equivalent chamber diameter;
F	is the total surface area of heat-transfer surfaces;
S	is the cross-sectional area of the chamber;
K	is the proportionality factor;
$Nu = \alpha D/\lambda$	is the Nusselt number;
Pr	is the Prandtl number;
$Re = U_c D/\nu = GD/S_c \rho\nu$	is the Reynolds number.

Indices

Σ	is the total;
r	is the radiant;
w	is the wall;
t	is the transverse;
L	is the longitudinal (axial));
b	is the burner (nozzle);
c	is the chamber;
th	is the theoretical;
e	is the experimental.

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