CONVECTIVE HEAT EXCHANGE WITH THE WALLS OF A CYCLONE DUST SEPARATOR

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Results are presented from an experimental study of heat exchange in a TsN-II cyclone processing combustion products at temperatures up to 400°C.

The waste heat of gases leaving production equipment frequently goes unused due to complications and costs connected with the installation of additional heat exchangers. Meanwhile, cyclones have to be installed in most production lines to remove dust from the gases. At some plants, these cyclones have been refitted to also function as heat exchangers in which the gases give off their heat to water or a water–steam mixture through the cyclone wall. One of the obstacles to the broader use of such cyclone heat exchangers is the lack of methods of designing same and inadequate information on the laws of heat exchange in cyclone dust separators.

Works [1-4] studied convective heat transfer in cyclone chambers with the ignition of a fuel, while [5] investigated the operation of a cyclone with a cooled cylindrical part and exhaust flue. We investigated heat exchange in a model of the cyclone designed by the "Giprogazoochistka" state planning institute, the most widely used cyclone in domestic industry.

Figure 1 presents a diagram of the experimental unit. The water jacket of the TsN-II cyclone (inside diameter of cyclone 204 mm) was made of eight individual sections. The exhaust flue was also cooled. The difference in water temperature at the inlet and outlet of each section was measured to within 0.1°C with hyperthermocouples. Chromel–Alumel microthermocouples with a 1-mm-diameter junction were used to study the temperature field over the cyclone volume. The accuracy of location of the thermocouple junctions at the assigned sites, done with a coordinate finder, was 0.2 mm. The error of the temperature measurement did not exceed 2%. Here, the moment of contact of the junction with the wall was determined from the closure of an electrical circuit including an indicator light. The thermocouple voltage was delivered through a switch to PP-63 and ÉPP-09 potentiometers. The hot gas supplied to the cyclone was obtained from a turbulent-type burner. The presence of a lengthy thermally

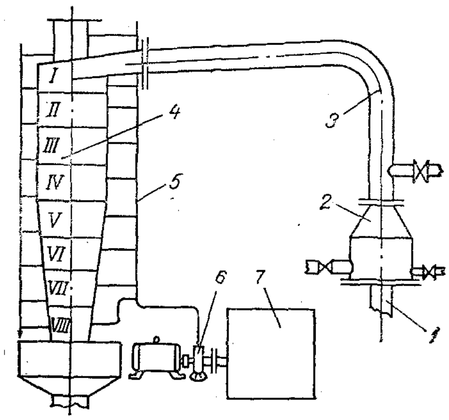

Fig. 1. Basic diagram of experimental unit: 1) air line; 2) burner; 3) stabilizing section; 4) calorimeters; 5) water jacket; 6) pump; 7) water tank; I-VIII) numbers of cyclone sections.

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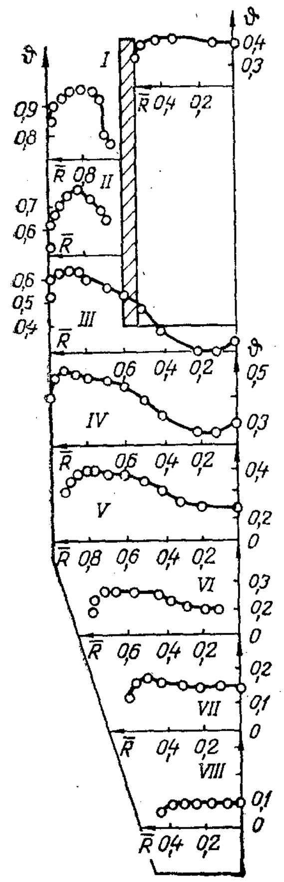

Fig. 2. Distribution of relative values of temperature over the height of the cyclone: $\vartheta = (t - t_{wa})/(t_{in} - t_{wa})$; Re = $7.16 \cdot 10^4$; t_{in} = 416°C; I-VIII) numbers of cyclone sections.

insulated section allowed us to stabilize the flow thermally and hydrodynamically. The use of a water pump and storage tank ensured a constant flow rate and temperature of coolant water (about 50°C) for the duration of the experiment. Water flow rate was determined by the gravimetric method. The temperature of the inside wall was monitored with Chromel—Alumel thermocouples caulked into the wall.

The velocity of the air flow at the cyclone inlet was changed during the experiment from 20 to 40 m/sec, while the temperature was changed from 100 to 400°C. This corresponded to a change in the Reynolds number, calculated from the inlet velocity and the diameter of the cyclone, from $7 \cdot 10^4$ to $30 \cdot 10^4$.

The measurements were made under steady-state conditions, which were established 1-2 h after the startup of the unit. The quantity of heat received by the walls was determined from the flow rate and increase in temperature of the water to within 1.5%, including a correction for heat loss to the environment. The quantities of heat found from the heating of the water in all of the sections and the cooling of the combustion products in the cycline did not differ by more than 3-7%.

The laws of heat transfer were determined by the character of motion of the flows in the cyclone and the gas temperature distribution in same. Figure 2 shows the dimensionless-temperature field in the vertical axial plane parallel to the walls of the intake flue. It is apparent here that the maximum temperature of the swirled flow decreases as the flow descends. In the region of the main vortex, i.e., roughly to the bottom edge of the exhaust flue, the maximum temperature zone is located next to the outer wall of the cyclone. Below the edge of the exhaust flue, the temperature field across the cone gradually smooths out. The temperature in the exhaust flue is typically markedly higher than in the bottom part of

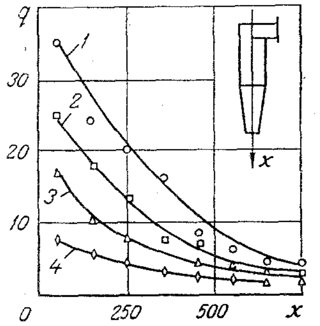

Fig. 3

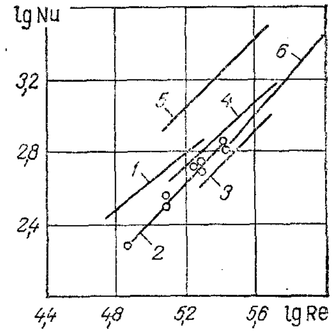

Fig. 4

Fig. 3. Change in heat flux over the height of the cyclone: 1) t_{in} = 413°C, w = 29.5 m/sec; 2) 312 and 29; 3) 206 and 28; 4) 121 and 30. q, kW/m²; x, mm.

Fig. 4. Comparison of experimental data on convective heat exchange: 1) data from [5]; 2) our data; 3) [7]; 4) [1]; 5) [3]; 6) [2].

the cone. This indicates that there are at least two circulation contours in the cyclone. Initially the main vortex, descending along the cyclone walls, carried with it the hot layers entering from the intake flue. Once having reached the edge of the exhaust flue, part of the hot air is caught by the outgoing portion of the vortex and ejected from the cyclone. The wall portion of the vortex continues to descend into the conical part of the cyclone, cooling and gradually being drawn toward the central flow ascending into the exhaust flue. The above-described pattern of flow corresponds to that in [6].

The distribution of local heat flux (Fig. 3) also corresponds to the above pattern of temperatures over the height of the cyclone (the readings begin from the point of intersection of the cyclone course and the uppermost plane of the intake flue — Fig. 2). It is clear from Fig. 2 that the dimensionless excess temperature in the lowest section is roughly an order lower than the same value in the highest section. Approximately the same difference exists in the top and bottom sections (Fig. 3). With a constant gas velocity at the cyclone inlet, the heat fluxes — at least toward the top courses — are nearly proportional to the difference between the temperatures of the gas and wall. Figure 3 does not show the heat flux toward the exhaust flue, which was cooled both externally and internally. Evidently, the gas flow at the cyclone outlet retains much of its vortical character, which promotes intensive heat exchange between it and the walls of the exhaust flue. According to our data, the heat removed from the exhaust flue accounts for about 30% of all of the heat removed from the cyclone.

Analysis of the empirical data to obtain design formulas is connected with certain problems. The main problem is that the temperature of the gases leaving the cyclone-cooler turns out to be higher than the minimum temperature of the gases in the cyclone (Fig. 2). This prevents us from selecting a determining temperature difference between the gas and wall, by analogy with the heat exchange that takes place during motion in tubes. It would be simplest to consider as the total amount of heat taken up by the entire heat-absorbing surface of the cyclone the mean excess temperature of the gas, with the latter calculated as the arithmetic mean of the gas temperatures at the cyclone inlet and outlet. Strictly speaking, the results obtained can be used only for geometrically similar cyclones. The heat transfer coefficient may be different in other cyclones (such as those with a smaller conical portion, with a bottom gas outlet, etc.).

Line 2 in Fig. 4 shows our data analyzed in this manner. The equation of this line Nu = 0.0028Re approximates our experimental data with a relative error of 6%. Agreeing closely with these results are the results from [1, 2] (lines 4 and 6), obtained in a study of heat exchange in swirl chambers with a smooth wall for the case of undirectional air delivery. These results, in turn, are close to those obtained in [5] (line 1). The value of α turned out to be somewhat higher in this work, evidently because the cone was not cooled, i.e., its inefficient working surface was not included in the quantity F used to calculate the values of α. Finally, the value of α turned out to be almost twice as great in [3] (line 5), apparently because of the presence of a large number of projections on the heat-exchanging surface of the cyclone chamber, i.e., because of a significant enlargement of the actual heat-exchanging surface. Also shown here are data from [7] (line 3), where the authors studied heat exchange in a swirl chamber in which the air completed only one revolution.

It follows from the above that

$$\alpha\nu/\lambda w = 0.0028,$$

i.e., the heat-transfer coefficient is independent of the diameter of the cyclone and increases linearly with an increase in the velocity of the gases in the cyclone. Despite the fact that this formula was obtained within a fairly narrow range of Reynolds numbers (from $7 \cdot 10^4$ to $30 \cdot 10^4$), it gives grounds for optimism that the heat transfer coefficient in commercial cyclones up to 2-3 m in diameter will not be significantly lower than the values obtained in our experiment and the other works cited, in which the range of cyclone diameters was 60-650 mm. Given typical gas velocities at the inlet in commercial cyclones (20-30 m/sec), the mean heat-transfer coefficient will be 60 $W/m^2 \cdot K$ for air with a temperature of 400°C and 35 $W/m^2 \cdot K$ for gases with a temperature of 1000°C.

The tests showed that the dust content of the gases entering the cyclone only slightly increases the heat-transfer coefficient at levels up to 0.5 kg/kg (higher dust contents are rarely encountered in industry).

All of this provides a basis for recommending cyclone dust separators as heat exchangers to make use of waste gas heat.

NOTATION

x, vertical coordinate; w, gas velocity in intake flue; d, cyclone diameter; $\alpha = Q/F \cdot (\bar{t} - t_{wa})$, mean heat-transfer coefficient; λ, thermal conductivity; ν, kinematic viscosity; $\bar{t}$, mean temperature in cyclone; $Q = \sum_i G_i \Delta t_i C_i$, heat flux through cyclone cooling surface; F, cooled surface of cyclone; Δt_i, heating of water in the sections; G_i, water flow rate through section; C_i, heat capacity of water; $\bar{t}_{wa}$, mean wall temperature; t_{in}, mean integral temperature at cyclone inlet; q, heat flux, $\bar{R}$, dimensionless radius; ϑ, dimensionless temperature head; $Nu = \alpha d/\lambda$, Nusselt number; $Re = wd/\nu$, Reynolds number.

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