QUASISTATIONARY HEAT TRANSFER IN A ROTATING REGENERATIVE AIR HEATER CONTAINING SPHERICAL PACKING ELEMENTS

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Results are presented on quasistationary heat transfer in a layer of spheres with allowance for the thermal resistance with intersecting orientation of the heat-exchanging media in the working chambers of a rotating air heater.

Regenerative rotating air heaters are widely used at large thermal power stations and in gas-turbine systems. They are used in supplying hot air in agricultural processes (dryers, warm-air heating in greenhouses, animal-rearing buildings, etc.). Recently, nonmetallic materials have replaced the sheet-metal fittings in these exchangers: glass and ceramic spheres [1, 2].

Temperature gradients occur within the spheres during heat transfer between the gas and the ceramic spheres at high gas flow speeds. The current methods of calculating heat transfer are reasonably reliable only for regenerative air heaters with gradient-free metal filling heating [3]. Not many studies have been performed on these processes when gradients are involved, as in equipment filled with ceramic spheres characterized by cross-current orientation of the heat-exchanging media [4, 5].

Here we derive a working formula for the heat-transfer coefficient for the general case of quasistationary heat transfer in the working chambers of regenerative air heaters with allowance for the internal thermal resistance in the spheres.

To describe the process, we determined the heating (cooling) law for a sphere of radius R in a medium whose temperature varies along the time coordinate τ in accordance with

$$T_c(\tau) = T_{c\infty} - (T_{c\infty} - T_{c0}) \exp(-k\tau), \tag{1}$$

where $T_{c\infty} = T_c(\infty)$. At the initial instant ($\tau = 0$) there is the following temperature difference between the medium T_{c0} and the packing T_0:

$$b = T_{c0} - T_0. \tag{2}$$

To derive the heat-transfer coefficient, we have to find the temperature distribution in the sphere at any time τ with boundary conditions of the third kind. We assume that the problem is symmetrical, i.e., the isothermal surfaces within the sphere are concentric spheres.

The available evidence for rotating regenerative heat exchangers shows that this approach describes the actual processes quite accurately, while the formulation of the initial conditions is more general than that used in [5, 6].

A regenerative rotating air heater contains unflushed (nonworking) regions between the working chambers, in which there is adiabatic temperature redistribution over the volume of a sphere, so under certain conditions one can write the initial condition as

$$T(r, 0) = T_0 = \text{const}, \quad 0 < r < R. \tag{3}$$

The boundary condition and the condition for symmetry of the temperature pattern in a sphere correspondingly take the form

$$-\frac{\partial T(R, \tau)}{\partial r} + H[T_{c\infty} - (T_{c\infty} - T_{c0}) \exp(-k\tau) - T(R, \tau)] = 0, \tag{4}$$

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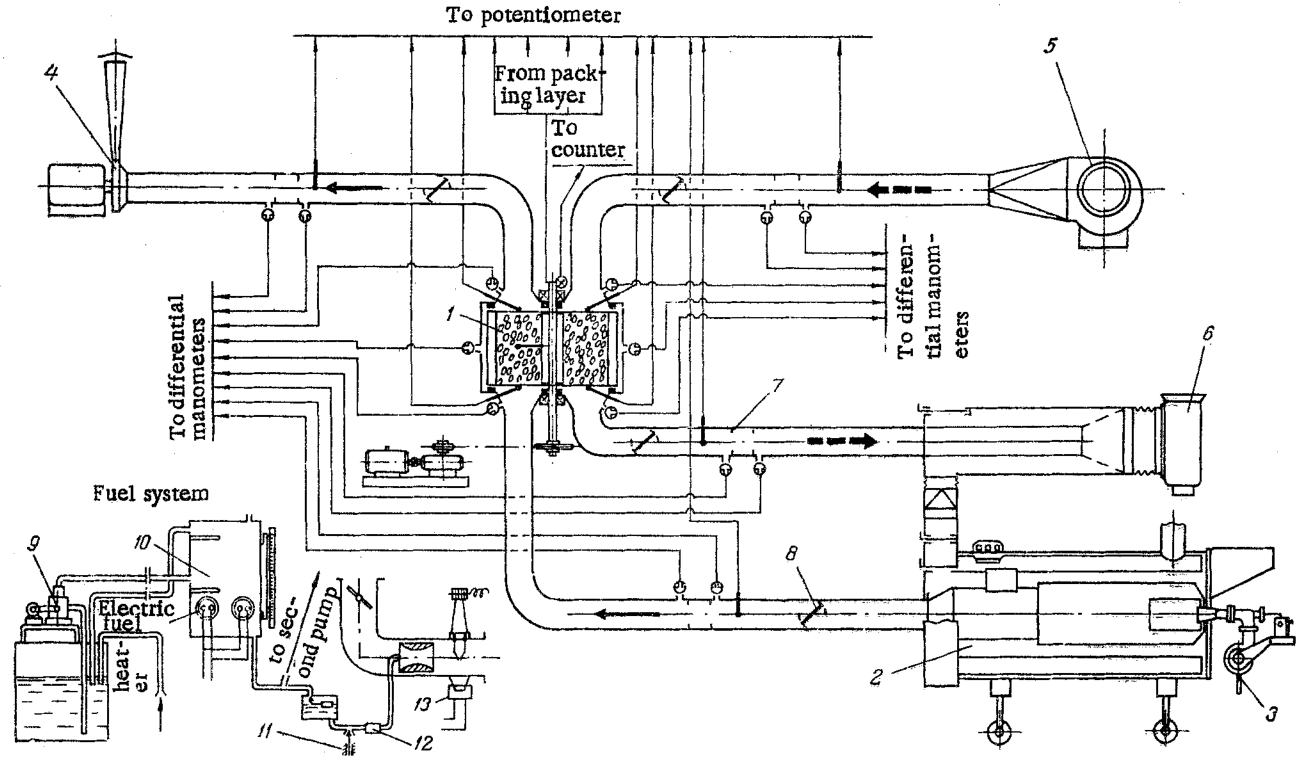

Fig. 1. Essential scheme for system for examining quasistationary heat transfer: 1) rotating regenerative air heater; 2) combustion chamber; 3) blower; 4) smoke extraction; 5 and 6) fans; 7) double diaphragm; 8) slide; 9) fuel pump; 10 fuel flowmeter; 11) adjusting screw; 12) electromagnetic valve; 13) igniter.

$$\frac{\partial T(0,\ \tau)}{\partial r} = 0. \tag{5}$$

The following is the solution to the differential equation for thermal conduction in the sphere in transformed form:

$$T(r,\ s) - \frac{T_0}{s} = \frac{HR^2\left[(T_{c\infty} - T_0) + \frac{bs}{k}\right] \operatorname{sh} \sqrt{\frac{s}{a}}\, r}{rs\left(1 + \frac{s}{k}\right)\left[(HR - 1) \operatorname{sh} \sqrt{\frac{s}{a}}\, R + R\sqrt{\frac{s}{a}} \operatorname{ch} \sqrt{\frac{s}{a}}\, R\right]} = \frac{\Phi(s)}{\psi(s)}, \tag{6}$$

where $\Phi(s)$ and $\psi(s)$ are readily reduced to generalized polynomials in s.

We apply the expansion theorem to write the general solution as the sum of inverse Laplace transformations for the three roots of the polynomial in the denominator:

$$T(r,\ \tau) = T_{c\infty} - \frac{HR(T_{c\infty} - T_0 - b) \sin \sqrt{\frac{k}{a}}\, r}{r\left[(HR - 1) \sin \sqrt{\frac{k}{a}}\, R + R\sqrt{\frac{k}{a}} \cos \sqrt{\frac{k}{a}}\, R\right]} \times$$

$$\times \exp(-k\tau) - \sum_{n=1}^{\infty} \frac{A_n}{1 - \frac{a\mu_n^2}{kR^2}} \left(T_{c\infty} - T_0 - \frac{ab\mu_n^2}{kR^2}\right) \frac{R \sin \mu_n \frac{r}{R}}{r\mu_n} \exp\left(-\frac{a\mu_n^2}{R^2}\tau\right). \tag{7}$$

The values of A_n are given to four figures in [6].

If we determine the dimensionless temperatures of the medium as $\Theta_c = (T_c - T_{c0})/(T_{c0} - T_0)$, then the Predvoditelev criterion is given by

$$\mathrm{Pd} = \left(\frac{d\Theta_c}{d\,\mathrm{Fo}}\right)_{\max} = \frac{R^2}{a(T_{c0} - T_0)} \left[\frac{d(T_c - T_{c0})}{d\tau}\right] = \frac{kR^2(T_{c\infty} - T_{c0})}{a(T_{c0} - T_0)} = \frac{kR^2}{a}\Theta_\infty. \tag{8}$$

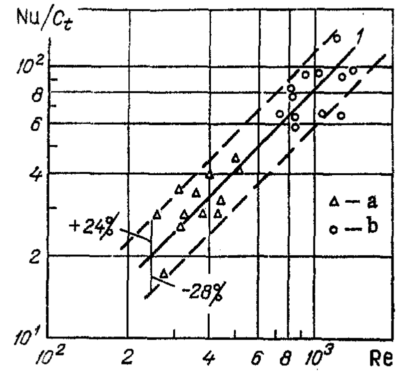

Fig. 2. Experimental data on heat transfer in working chambers of a rotating air heater having nonmetallic heating surfaces in the form of a ceramic sphere layer: a) for packing heating chamber; b) for cooling chamber; 1) $Nu/C_t = 0.081$ Re.

The solution to (7) is written as follows in dimensionless variables:

$$\Theta = \frac{T(r,\ \tau) - T_{c0}}{b} = \Theta_\infty \left\{ 1 - \frac{\mathrm{Bi}\, R \sin\sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}}\, \frac{r}{R}}{r\left[(\mathrm{Bi}-1)\sin\sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}} + \sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}}\cos\sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}}\right]} \times \exp\left(-\frac{\mathrm{Pd\,Fo}}{\Theta_\infty}\right)\right\} - \sum_{n=1}^{\infty} \frac{A_n}{1 - \frac{\Theta_\infty \mu_n^2}{\mathrm{Pd}}} \frac{R \sin\mu_n \frac{r}{R}}{r\mu_n}\left(\Theta_\infty + 1 - \frac{\Theta_\infty \mu_n^2}{\mathrm{Pd}}\right)\exp(-\mu_n^2\,\mathrm{Fo}). \quad (9)$$

We put b = 0 in (7) and determine the dimensionless temperature of the medium as $\Theta_c = (T_c - T_{c0})/(T_{c\infty} - T_{c0})$, while the Predvoditelev criterion is determined as $\mathrm{Pd}_0 = kR^2/a$, which gives Lykov's solution [6].

Expression (9) together with the transcendental equation

$$\tan\mu = \frac{\mu}{1 - \mathrm{Bi}} \quad (10)$$

enables one to determine the heat-transfer coefficient α appearing in Bi.

The initial temperature difference b was determined from the surface temperature of the sphere T(R, 0) and the mean integral temperature of the medium in the packing layer. The mean integral heat-transfer coefficient may be derived by direct measurement of the temperatures of the sphere surface and the gas, and one uses (7) written in transformed form, i.e., for r = R.

When we had established the errors on discarding terms other than the first, the method of [5, 6] was used to write the transformed equation in dimensionless form with an error of less than 0.2%:

$$\Theta_s = \Theta_\infty \left[1 - \frac{\mathrm{Bi}\tan\sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}}}{(\mathrm{Bi}-1)\tan\sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}} + \sqrt{\frac{\mathrm{Pd}}{\Theta_\infty}}}\exp\left(-\frac{\mathrm{Pd\,Fo}}{\Theta_\infty}\right)\right] + \frac{A_1 \sin\mu_1}{\mathrm{Pd} - \Theta_\infty \mu_1^2}\left[\Theta_\infty \mu_1 - \mathrm{Pd}(\Theta_\infty + 1)\frac{1}{\mu_1}\right]\exp(-\mu_1^2\,\mathrm{Fo}). \quad (11)$$

This equation applies whenever

$$\mu_1^2 \gg \mathrm{Pd}_0 > 0. \quad (12)$$

A laboratory system at this institute was used in examining the heat transfer in such chambers with crossing gas flows. Figure 1 shows the essential scheme. The model for the air heater is a pilot-plant specimen of a Jungström rotating apparatus with a rotor diameter of 1.4 m. The working chambers were filled with porcelain spheres of diameter 12.8 mm made by the Rechitsa porcelain plant, and the layer depth was 0.3 m. The system was fitted with a combustion chamber, a set of extraction and blower units, a panel bearing the monitoring and measuring instruments, and a control system for the power and signaling devices.

We determined the following parameters required in estimating the mean heat-transfer coefficients by the above method.

The flow rates of the heating and heated media were measured by means of paired diaphragms; the temperatures were determined with Chromel–Alumel thermocouples, with recording with ÉPP-0.9 and KSP-4 multipoint electronic potentiometers. The temperatures of the ceramic spheres were measured with insulated Chromel–Alumel thermocouples attached to the surfaces by means of a special cement used in joining amalgam and silicate objects. To measure the gas temperatures in the packing layer, the hot thermocouple junctions were placed in short tubes with radial holes. These two types of thermocouple were close together. In this way, the temperatures of the gas and sphere surface were measured at three points at three levels. The thermo-emf's from the thermocouples rotating together with the rotor were passed to a copper–graphite contact device set up at the cold end of the rotor shaft. The secondary instrument was a KSP-4 potentiometer (accuracy class 0.25%).

The rotor speed was measured with an electrical pulse counter and special contact attached to the shaft. The surface of the packing in the working chamber was defined by

$$F_p = \frac{6(1-m)}{d_p} V_p \tag{13}$$

The volume occupied by the packing in a sector cell in the rotor was determined by calculation from the geometrical dimensions. The density of the material was found by a volumetric method.

The mass flow rate of the packing was given by

$$M_p = 30\, n\, (D_1 - D_2)\, h_b\, \rho_{pa}\, \pi D. \tag{14}$$

To find the numerical value of the exponent and the maximum temperature of the medium, the time spent by a packing element in the working chamber was split up into four equal intervals τ_0. The mean integral temperatures at the boundaries of the intervals were found graphically. The values were used with (1) to calculate the exponent describing the variation in mean integral gas temperature and the maximum value $T_{c\infty}$:

$$k = \ln \Delta t / \tau_0. \tag{15}$$

Similarly, we determined the exponent describing the mean integral temperature of a packing element. The definitive gas and packing temperatures were taken as the mean integral values in the working chambers. The physical constants of the gas medium and packing were determined for these temperatures by means of the data given in [1, 3, 7]. The linear dimension was taken as the sphere diameter d_p, while the characteristic velocity was taken as the gas speed in a section free from packing.

Equations (10) and (11) with given values of $\Theta_s, \Theta_\infty, \mathrm{Pd}, \mathrm{Fo}$ enable one to determine the heat-transfer coefficients between the media in the working chambers from the known value of the Biot criterion.

Figure 2 shows the results in terms of the Nusselt and Reynolds numbers. For Reynolds numbers between 250 and 1400, the approximation is

$$\mathrm{Nu} = 0.081\, \mathrm{Re}\, C_t. \tag{16}$$

The values of C_t varied within narrow limits: 0.86-0.95 for the packing heating chamber and 1.06-1.15 for the cooling chamber. For practical calculations with an error of up to 5%, one can take C_t = 0.9 for the heating chamber and C_t = 1.1 for the cooling one.

It should be mentioned that the scope for using the (16) derived from the solution of (11) should be established in each case by means of a calculation designed to check the condition stated in (3). This condition is met fairly fully if the calculated value of the

Fourier number for the unflushed (nonworking) part of the air heater is not less than 0.45-0.50. It should be noted that the cross section of each unflushed part in the experimental heat exchanger was 20% of the total cross section of the exchanger, while the rotor speed was 0.3-0.4 rpm and the values of the packing constants [1] were $\lambda_p = 1.5$ W/m·deg, C_p = kJ/mg·deg, and $\rho_{pa} = 2350$ kg/m³.

The design and working parameters of the prototype were very close to those of commercial regenerative air heaters.

Therefore, this solution on the heating and cooling of a single packing element can be used for conditions of initial temperature difference with the most general case of exponential gas temperature variation to examine the heat-transfer processes under conditions of crossing flow for the heat-exchanging media in the presence of a radial temperature gradient in the packing element, and one gets the criterion relationship described by (16).

These relationships have been used in calculating industrial air heaters having nonmetallic heating surfaces.

NOTATION

$Nu = \alpha d_p/\lambda$, Nusselt number; $Re = \omega d_p/\nu$, Reynolds number; $Fo = a\tau/R^2$, Fourier number; $\theta_0 = (T_{c\infty} - T_{c0})/(T_{c0} - T_0)$ dimensionless temperature of the medium for $\tau \to \infty$; $\theta_s = [T(R, \tau) - T_{c0}]/[T_{c0} - T(R, 0)]$, dimensionless surface temperature; $m = 1 - (G_p/V_p\gamma_p)]$, packing bed voidage; $C_t = \overline{T}_{pa}/\overline{T}_{fl}$, temperature factor for thermal resistance of the boundary layer $\Delta t = (t_{c2} - t_{c1})/(t_{c3} - t_{c2})$; $Pd_0 = kR^2/a$, Predvoditelev number α, heat-transfer coefficient, λ_p, λ, thermal conductivity of packing and gas, respectively; d_p, diameter of a sphere, $d_p = 2R$; a, thermal diffusivity of the packing material; Bi/R, reduced heat transfer coefficient ($Bi/R = H = \alpha/\lambda_p$; ω, gas flow velocity in the section free of packing; ν, kinematic viscosity; $T_{c\infty}$, maximum medium temperature; k, exponent; $\overline{T}_{pa}$, $\overline{T}_{fl}$, mean integral temperatures in the working chamber for packing and flow; t_1, t_2, t_3, mean integral temperatures for sections; τ_0, interval; interval; V_p, packing volume in chamber; F_p, packing surface participating in heat transfer; G_p, packing weight in the chamber; γ_p, density of packing; D, D_1, D_2, mean, maximum, and minimum diameters of rotor shell, respectively) h_b, ρ_{pa}, M_p, bed height, packing density, and packing mass flow rate, respectively.

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