INITIAL HEATING PERIOD FOR A MASSIVE BODY IN A HIGH-TEMPERATURE OVEN

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The difference between the reduced values of the degree of blackness and absorptivity for the oven space is incorporated in the approximate method of [1], and a correction is made for convective heat transfer.

At high temperatures the heating of a body is determined by the radiation mechanism. There is an exact solution to the problem [2]

$$\frac{\partial T}{\partial \tau} = a\frac{\partial^2 T}{\partial x^2}$$

with the boundary condition

$$\bar{a}\sigma(T^4 - T_*^4) = -\lambda\left(\frac{\partial T}{\partial x}\right)_{x=0} \tag{1}$$

and the initial conditions

$$T(x) = T_{*\min} = \text{const.} \tag{2}$$

The nomograms of [2] give results with low accuracy on account of interpolation with respect to the numbers Bo and Fo, and especially β. The range in β required for our studies is not covered by these nomograms, and therefore extrapolation is required. There may be substantial errors from neglecting the convective component of the heat flux and the use of thermal conductivity averaged with respect to temperature [3]. This means that there is considerable interest in simple calculation methods suitable for engineering use.

In [1], an approximate method for $Fo < 1$ was proposed; unfortunately the analysis was restricted to $\beta \leqslant 0.25$, whereas frequently $\beta > 0.75$ for linings. Here we show that the method of [1] is suitable with the much more general boundary conditions

$$\sigma(\bar{\varepsilon}T^4 - \bar{a}T_*^4) + \alpha_k(T - T_*) = -\lambda\left(\frac{\partial T}{\partial x}\right)_{x=0}, \tag{3}$$

if the role of the convective term is small. In [4], which was performed with a hydrostatic integrator, it was shown that one can assume with an error of 1-2% that the heat is transferred only by radiation if we use the effective value of the Boltzmann number given by

$$\frac{1}{Bo_e} = \frac{1}{Bo} + 0.25 BiBo,\ BiBo \leqslant 1. \tag{4}$$

The studies were performed at $Bi = 0.05 - 1$; $Bo = 0.33 - 1.67$; $\beta = 0.15$. In the high-temperature zones of rotating ovens and similar devices, one can use the approximation of (4) because the convective term in (3) can be omitted. However, the principal difference between conditions (1) and (3) persists, viz. that $\bar{\varepsilon}$ and $\bar{a}$ are not equal. The value of $\bar{a}$ is dependent on the temperature T_* and we have $\bar{a} \to \bar{\varepsilon}$ only for $T_* \to T$. The solution should now be iterative with correction of T_* at each step.

The form for the approximate solution given in [1] still applies, so it is not given here. There are changes only in the dimensionless arguments $\beta = T_{*\min}/(bT)$, $z = (3/2)Fo/(bBo_e)^2$ and the function

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$$y = \frac{q}{\bar{\varepsilon}\sigma T^4} = 1 - \left(\frac{T_*}{bT}\right)^4. \tag{5}$$

The new dimensionless quantities include the number $b = \sqrt[4]{\bar{\varepsilon}/\bar{a}}$, with b → 1 for $T_* \to T$; this number may be less than or greater than one. Calculations on the blackness and absorptivity of an oven space constitute a special section in heat engineering. The absorptivities a_1, a_2, ... may directly incorporate the temperature distribution in the media [5]. However, this property is not used here. We assume ab initio that the medium is isothermal and has effective temperature T, whose calculation constitutes a separate problem. The degrees of blackness ε_1, ε_2 and the reduced quantity $\bar{\varepsilon}$ are meaningful only for the isothermal volume. The characteristics $\bar{\varepsilon}$ and $\bar{a}$ of the oven space should also incorporate the inherent radiation from the surfaces and repeated reflection.

The literature carries only particular calculations on $\bar{\varepsilon}$ and $\bar{a}$, in which the energy dissipation is usually not incorporated. If, on the other hand, all the surfaces are black and cold, then

$$b = \frac{T_*}{T}\left[\int_0^\infty I_{0\omega}(T)\,A_\omega d\omega \Big/ \int_0^\infty I_{0\omega}(T_*)\,A_\omega\,d\omega\right]^{1/4},$$

where the spectral absorptivity A_ω is calculated for the temperature of the medium T. Here b is largely determined by the factor T_*/T; the quantities ε_1 and a_1 diverge most notably for carbon dioxide. For steam, although this is much grayer, they may differ by a substantial factor [6]. In [7], allowance was made for the inherent radiation from the surfaces and repeated reflection. A system of three isothermal bodies was considered: the volume of the real medium and two gray surfaces. Here we supplement the calculations on the heat transfer for the heating surface with temperature T_0 with an analogous calculation for the lining. In Eq. (3) of [7] we assumed equality of the temperatures for the two surfaces, $T_0 = T_*$. Usually, the surface area of the heated workpiece is relatively small. In a rotating oven, the surface area of the layer of material is several times less than the surface of the lining. Therefore, the assumption does not introduce a large error, and we get $\bar{\varepsilon} = \varepsilon_1 K$, $\bar{a} = a_1 K$, K = K'' + (K' − K''), where K' and K'' are coefficients found on the assumption of gray and extremely selective properties for the medium, while c is an interpolation vector, which defines the place of the actual spectrum between these extreme ones, and

$$\frac{K'}{A_*} = \frac{1 + R_0(1-\varphi)(1-a_1)}{1 - R_*\varphi(1-a_1) - R_0R_*(1-\varphi)(1-a_2)};\quad \frac{K''}{A_*} = 1;$$

$$c = \frac{\varepsilon_2 - \varepsilon_1}{\varepsilon_1(1-\varepsilon_1)}.$$

Analysis shows that there are only minor differences between K' and K'', so the calculation of c does not involve substantial error.

The first step consists in estimating $\bar{a}$ and deriving b, β, and z. Then the approximate formula of [1] is used to calculate y. The quantities T_* and q are then found from (5). The temperature distribution in the body is approximated by $T_{**}(x) = T_{*\min} + \frac{q}{2\lambda\Delta}(\Delta - x)^2$.

If necessary, $\bar{a}$ can be refined from the result for T_* and the calculation cycle repeated.

Table 1 compares the results for b = 1 and β = 0.75 with the corresponding data of [2]; the approximate method is quite applicable up to Fo = 0.5. At Fo = 1, the approximate values are appreciably too low. One can take $\Delta = 3.54\sqrt{a\tau_0}$ for a rotating oven, and then the maximum Fourier number, when the oven filling is reduced to zero, is $Fo_{max} = 1/(3.54)^2 \approx 0.08$; the accuracy of the approximate method is then high.

In [1] we find a simplified method suitable for desk calculators. We construct two curves (Fig. 1): for $z \gg 1$

$$y = 1 - [\beta + (1-\beta^4)\sqrt{z}]^4, \tag{6}$$

and for $z \gg 1$ $(y \ll 1)$

TABLE 1. Comparison of the Dimensionless Temperature $\theta_* = (T_* - T_{*min})/(T - T_{*min})$ for $\beta = 0.75$: the Upper Numbers Are From the Nomogram of [2] and the Lower Ones From the Approximate Formula of [1]

Fo	Bo 0,5	1	2	3	5	10
0,1	0,77	0,60	0,40	0,30	0,19	0,09
	0,77	0,60	0,40	0,29	0,19	0,10
0,25	0,86	0,74	0,53	0,41	0,28	0,14
	0,85	0,72	0,53	0,41	0,28	0,15
0,5	0,92	0,82	0,65	0,53	0,37	0,20
	0,90	0,80	0,63	0,51	0,37	0,21
1	0,96	0,92	0,80	0,68	0,52	0,30
	0,93	0,85	0,72	0,61	0,46	0,28

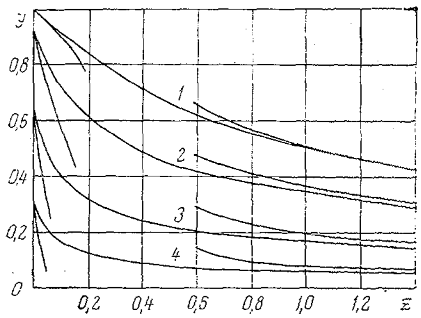

Fig. 1. Curves constructed from the unified approximate formula of [1] and the very simple formulas of (6) and (7): 1) $\beta = 0.25$; 2) 0.5; 3) 0.75; 4) 0.9.

$$y = \frac{2(1-\beta)}{\sqrt{8z + 1 - (\beta/2) - 4\beta^2}}. \qquad (7)$$

The tangent to the curves together with the parts of the curves away from the tangent constitutes the approximate solution. Figure 1 shows the line for the unified approximate solution from the formula of [1] and the lines for the particular solutions from (6) and (7). The curve for (6) is close to a straight line. The curve from the unified solution runs below the curve of (7) without touching it. This set of curves is sufficient to interpolate the particular solutions of (6) and (7) obtained for any values of z. Here the accuracy of the calculation may be higher than that from interpolating the curves in the nomograms of [2].

In [2], the temperatures of the body and the medium were compared for $z \to \infty$. When the lining is heated, allowance must be made for the heat losses through it. The temperature distribution in the lining can be represented as the sum of constant and variable components. The first is found from the conditions for stationary heat transfer through the wall into the surrounding space. The variable component is determined subject to the condition of ideal thermal insulation.

The initial condition of (2) causes the main objection when the method is used for the lining heating period in a rotating oven. This clearly does not correspond to the instant when the lining is cleared from contents as the oven rotates. Measurements on an experimental 8 × 1.14 m oven [8] indicate that the surface temperature of the lining takes a constant (equilibrium) value within 1 min with a period of rotation of 4.63 min. From the maximum gas temperature in the study, $z \approx 0.5$. Our calculation gives a temperature at this point fairly far from the maximum one. The conflict is partly explained by the curve for the initial temperatures in the lining not corresponding to condition (2). In fact, the temperature rise occurs more rapidly and the present calculation gives only the extreme reference curve for the lining surface temperature.

NOTATION

a, thermal diffusivity, m^2/sec; a_1 and a_2, absorptivities for the incident flux with a black spectrum at a temperature T_*, single and double passes (double pass includes total reflection after the first pass); ε_1, volume emmissivity; $\bar{\varepsilon}$ and $\bar{a}$, reduced emissivity and absorptivity of the furnace space; $I_{0\omega}$, Planck intensity, $W \cdot cm/(m^2 \cdot sr)$; A_ω, spectral absorptivity, dimensionless; A, surface absorptivity; $R = 1 - A$; ω, wave number, cm^{-1}; α_k, heat-transfer coefficient from medium to heating surface, $W/(m^2 \cdot K)$; λ, thermal conductivity, $W/(m \cdot K)$; τ, heating time, sec; Δ, penetration depth of varying temperatures, m; τ_0, period of furnace rotation, sec; φ, angular coefficient from lining to lining without absorption; $\sigma = 5.67 \cdot 10^{-8}\ W/(m^2 \cdot K^4)$; T and T_*, temperatures of the medium and lining surface, °K; T_{*min}, initial temperature constant over the depth, °K; $Bi = \alpha_k\Delta/\lambda$; $Bo = \lambda/(\bar{\varepsilon}\sigma T^3\Delta)$; $Fo = a\tau/\Delta^2$; $\beta = T_{*min}/(bT)$; $z = 1.5Fo/(bBo_e)^2$; $b = \sqrt[4]{\bar{\varepsilon}/\bar{a}}$; $\Theta_* = (T_* - T_{*min})/(T - T_{*min})$; $y = q/(\bar{\varepsilon}\sigma T^4)$. Subscripts * and 0, lining surface and charge surface; A, surface absorptivity.

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RADIATIVE CHARACTERISTICS OF COKE PARTICLES OF A COAL-DUST FLAME

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An experimental investigation is made of the complex index of refraction of the cokes of solid fuels. The spectral coefficients of absorption and scattering needed to calculate the thermal emission of coke particles in a coal-dust flame are determined on the basis of the results obtained.

The thermal emission of a coal-dust flame is determined by the emission of the triatomic gases CO_2 and H_2O and of the ash and coke particles contained in the stream of these gases. As volatiles escape and the coke residue burns up, the sizes and concentration of the coke particles vary, whereas for ash particles they remain approximately constant over the height of the furnace box. In connection with the foregoing, the radiative properties of the solid disperse phase vary considerably in the process of combustion.

By now the radiative properties of triatomic gases and particles of ash dust have been studied in sufficient detail. But as for coke particles, all the available data are confined to the results of investigations [1] of the optical constants of coke in the visible region of the spectrum at $\lambda = 0.546\ \mu m$. And yet the main transfer of radiant energy in the furnace boxes of boiler plants takes place mainly in the infrared region of the spectrum in the wave-

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