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ON THE OPTIMAL TEMPERATURE PROFILE OF SELECTIVE GASES UNDER RADIANT HEAT TRANSFER

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UDC 536.3

It is shown that a selective gas temperature profile exists for which radiant heat flux is a maximum in the heating surface. The mean, minimum, and maximum gas temperatures are determined.

The paper [1] is devoted to the question of the influence of furance gas temperature profiles on the resultant radiant heat flux q in a heating surface. The necessity is shown there for taking account of the real temperature field in evaluating the quantity q. Computations were executed taking the selectivity of the gas radiation into account. It is shown here that there exists a temperature profile for which q is a maximum. For simplicity, we assume the lining to be absolutely black and adiabatic, the heating surface nonselective, and we simulate the furnace geometry by a plane layer. The higher the emissivity of the lining, the greater the q since the intrinsic radiation of the lining is transmitted better by a selective gas than the reflected radiation containing the gas band in its spectrum.

Let Q_1 be the intrinsic gas radiation on the heating surface and Q_2 on the lining. The expression for q has the form

$$q = -Q_{01} + (1-r)\,[Q_1 + D_2(l)\,Q_{02}]. \tag{1}$$

From the adiabatic condition of the lining, we have

$$Q_{02} = Q_2 + D_1(l)\,Q_{01} + rQ_1D_3(l) + Q_{02}D_2(2l) \tag{2}$$

and we obtain from (1) and (2)

$$q = -Q_{01} + (1-r)\left\{Q_1\left[1 + r\frac{D_2(l)\,D_3(l)}{1-rD_2(2l)}\right] + Q_2\frac{D_2(l)}{1-rD_2(2l)} + \frac{Q_{01}D_2(l)\,D_1(l)}{1-rD_2(2l)}\right\}. \tag{3}$$

For a fixed mean gas temperature, the coefficients of Q_1, Q_2, and Q_{01} are practically independent of Q_1, Q_2, and r. Then, if the coefficient of Q_2 in (3) is greater than the coefficient of Q_1, then it is energetically advantageous to organize the furnace heating scheme with the greatest possible value of Q_2, i.e., when the torch is directed to the crown (indirect mode). The condition for preference of the indirect over the direct heating mode

$$\frac{D_2(l)}{1-rD_2(2l)} > 1 + \frac{rD_2(l)\,D_3(l)}{1-rD_2(2l)}, \tag{4}$$

follows from (3), and can be written as

$$r > \frac{1-D_2(l)}{D_2(2l) - D_2(l)\,D_3(l)}. \tag{5}$$

All-Union Scientific-Research Institute of Metallurgical Thermal Engineering, Sverdlovsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 39, No. 1, pp. 109-112, July, 1980. Original article submitted June 15, 1979.

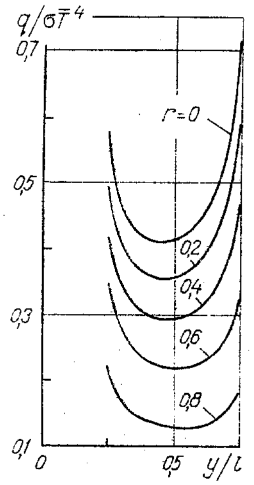

Fig. 1

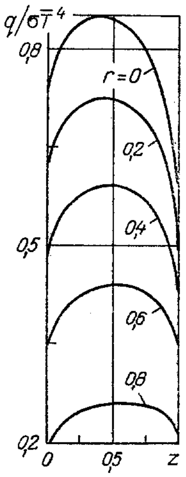

Fig. 2

Fig. 1. Dimensionless resultant radiant flux on the heating surface as a function of the dimensionless coordinate of the center of the burning zone. The burning zone thickness is $l/2$.

Fig. 2. Dependence of $q/\sigma\bar{T}^4$ on the relative thickness of the zone pressed to the lining for the case of two high-temperature zones pressed to the lining and the heating surface. Total burning zone thickness is $l/2$.

It is observed that the indirect heating mode can be energetically advantageous for high reflexivities of the heating surface. Starting with a certain $l = l_1$, the inequality (4) is not satisfied so that for $l > l_1$ the indirect heating mode is energetically disadvantageuus compared to the direct mode when the torch is applied to the heating surface.

In the approximation of an equivalently-gray gas

$$D_1(l) = D_2(l) = D_3(l) = 2E_3(\tau_0),$$

where τ_0 is the optical thickness of the layer. Let us show that in this case the indirect heating mode is always energetically more favorable than the direct. From the obvious inequalities

$$1 - 2rE_3(2\tau_0) \geqslant 1 - 2rE_3(\tau_0), \qquad 1 > 2E_3(\tau_0) \quad \text{for} \quad \tau_0 > 0$$

there follows

$$1 - 2rE_3(2\tau_0) > 2E_3(\tau_0)\,[1 - 2rE_3(\tau_0)].$$

We rewrite this last inequality in the form

$$\frac{2E_3(\tau_0)}{1 - r2E_3(2\tau_0)} < 1 + \frac{r\,[2E_3(\tau_0)]^2}{1 - r2E_3(2\tau_0)}, \tag{6}$$

which contradicts (4). Therefore, the indirect heating mode can be energetically more advantageous than the direct mode only in a selective medium, when the gas radiation is converted into continuous radiation of the ing which easily passes through the gas.

The question of the optimal temperature profile in a selective gas medium is of great interest. If the temperature of the medium has no lower bound, but a mean temperature is determined, then the solution of the variational problem formulated is evident. The temperature will be zero everywhere, with the exception of the area of the torch, but the torch can be located anywhere since the gas does not absorb radiation at zero temperature. If the temperature has no upper bound, then the solution of this problem will be in the form of a δ-function. The problem of a gas with a fixed mean $\bar{T}$ and both upper and lower temperature bounds is of practical interest. It is clear from the nonlinear dependence of the radiation flux on the temperature that the solution will have the form of one or more zones with maximum temperature, while the temperature of the rest of the

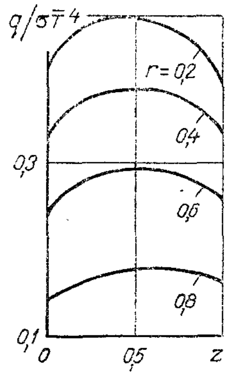

Fig. 3. Dependence of $q/\sigma\bar{T}^4$ on the relative thickness of the zone pressed to the lining for two burning zones. Total burning zone thickness is $l/4$.

gas will take on a minimal value. As an illustration, we performed a computation by the method in [1] for a medium consisting of a mixture of carbon dioxide, steam, and nitrogen with a carbon dioxide partial pressure of 0.12 atm, a steam partial pressure of 0.19 atm, and a total pressure of 1 atm for a layer thickness of $l = 1.5$ m. The heating surface temperature was $T_1 = 800°K$, the temperature of the burning zone was $T_m = 2000°K$, while $\bar{T} = 1400°K$. The dimension of the burning zone was half the height of the layer and its location varied.

Results of computing the quantity $q/\sigma\bar{T}^4$ are presented in Fig. 1 as a function of the coordinate of the center of the burning zone for different metal reflexivities. It follows from Fig. 1 that if the torch is remote from the lining or the heating surface, its efficiency drops. This suggests dividing the burning zone into two and pressing them to the walls. Under given constraints the temperature distribution in the form of two high temperatures zones pressed to the lining and the heating surface is optimal for a definite ratio between the thicknesses of these zones. Indeed, if there is just one high-temperature zone pressed to the metal, then its upper part will operate inefficiently since only a small fraction of the radiation reaches the metal because of self-absorption of the radiation in the burning zone and a small part will reach the lining because of radiation absorption in the gas. If the high-temperature zone is split into two and the upper part is pressed to the lining, then the lining will afterwards receive more energy and reradiate it in a continuous spectrum to the heating surface. The continuous lining radiation is absorbed considerably more weakly by a selective gas than is the radiation produced by this gas, hence the flux in the metal grows as compared with the initial case. Analogous reasoning can be performed for the case when the high-temperature zone is initially at the lining. It is hence clear that the version with a definite ratio between the thicknesses of the high-temperature zones squeezed to the surfaces will be optimal.

Results of computing such a version are represented in Fig. 2 for the conditions of the previous example with different relationships between the lengths of these zones. Plotted along the horizontal axis is the ratio z between the length of the burning zone pressed to the lining, and the total length of the burning zones. The maximum of the flux to the heating surface is obtained for the value $z = z_0$, where z_0 depends on many parameters. Results analogous to Fig. 2, but with a total high-temperature zone width of $l/4$ are presented in Fig. 3 for $\bar{T} = 1600°K$, $T_m = 2200°K$, and $T_1 = 1000°K$. The remaining parameters are the same as in the preceding versions. It is seen that the maximums of the curves differ somewhat from the results in Fig. 2 for the same values of r.

The possible advantage of the combined heating (crown and direct, jointly) is mentioned in [2, 3].

The estimate of the contribution of the heat conduction for the moving medium, which we performed by standard methods, does not alter the general conclusion.

NOTATION

q, resultant radiant heat flux at the heating surface; Q_1, Q_2, flux of intrinsic gas radiation on the heating surface and the lining, respectively; r, reflexivity of the heating surface; Q_{01}, Q_{02}, intrinsic radiation fluxes of the heating surface and the lining; l, layer thickness; D_1, D_2, D_3, transmissivity of the gas layer for radiation of the heating surface, lining, and layer of the same gas; E_3, integral exponential function of third order; $\bar{T}$, mean gas temperature; T_1, heating surface temperature; T_m, maximum temperature; σ, Stefan–Boltzmann constant; y, coordinate of the layer measured from the lining.

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CALCULATION OF EFFECTIVE THERMAL RADIATION ABSORPTION COEFFICIENT OF A CAVITY WITH DIFFUSELY REFLECTING WALLS

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UDC 536.3

The system of integral equations of radiation heat exchange in a closed cavity is solved numerically.

One of the basic requirements of a calorimeter for radiant heat fluxes is the total absorption of all radiation incident on its entrance opening, independently of the spectral composition and direction of the radiation. The most effective method of increasing the absorption of radiation is the use of cavities of different configurations to collect the radiation. The geometry of a cavity can be changed so as to make its radiation characteristics approach those of a black body as closely as possible. The actual characteristics of the cavity can be determined either experimentally or theoretically, but the experimental arrangements for determining the absorptance of a cavity are so complex that only the theoretical solution of this problem is practical.

The effective thermal radiation absorption coefficient of a cavity of any configuration is defined as the ratio

$$\varepsilon_{\text{eff}} = 1 - \frac{Q_{\text{ref}}}{Q_{\text{in}}}, \tag{1}$$

where

$$Q_{\text{ref}} = \sum_{i=1}^{N} \int_{A_i} \int_{A_0} (\varphi_i(\mathbf{r}_i) - f_i(\mathbf{r}_i)) K_{i0} dA_i dA_0 \tag{2}$$

is the reflected heat, and

$$Q_{\text{in}} = \int_{A_0} \varphi_0(\mathbf{r}_0)\, dA_0 \tag{3}$$

is the incident heat. Here $\varphi_i(\mathbf{r}_i)$ is an unknown function characterizing the flux density of effective radiation from the i-th zone of the cavity surface (the subscript o refers to the opening); $f_i(\mathbf{r}_i)$ is a known function which characterizes the self-radiation of the cavity surface.

In order to find the unknown function $\varphi_i(\mathbf{r}_i)$, and consequently to determine the radiation characteristics of the cavity, it is necessary to solve the radiation heat-exchange problem in a closed cavity. By using the generalized zonal method this problem is reduced to the solution of a system of integral equations of the form [1]

$$\varphi_i(\mathbf{r}_i) = g_i(\mathbf{r}_i) + \lambda_i \sum_{j=1}^{N} \int_{A_j} \varphi_j(\mathbf{r}_j) K_{ij} dA_j \quad (i = 1, 2, \ldots, N), \tag{4}$$

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 39, No. 1, pp. 113–117, July, 1980. Original article submitted May 28, 1979.

