ANALYTICAL AND EXPERIMENTAL STUDIES OF
RADIATIVE HEAT EXCHANGE IN THE MUFFLE
CHAMBERS OF MULTICHANNEL FURNACES

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An analysis is given for the process of radiative heat exchange between the element of a muffle furnace and the fired material. On the basis of the analysis and experimental data we have derived a system of equations which uniquely define the conditions of heat transfer within the furnace.

In the heat treatment of thin ceramics we note a trend toward the use of multichannel furnaces with the fired materials placed in a single row, in preference over the use of muffle-tunnel furnaces. Efficient utilization of multichannel furnaces makes it necessary to know the conditions of heat transfer which prevail in the muffle chamber of the furnace.

The purpose of this investigation is the study of the transfer of heat between the muffle chamber and the material to develop methods of calculating the optimum heating rate, proceeding from the physico-mechanical properties of the fired material and the structural features of multichannel furnaces.

The muffle chambers of multichannel furnaces are rectangular elongated tunnels lined with shaped refractories forming a number of narrow parallel channels (slots) through the height of the tunnel, with the material sliding along the bottoms of these channels. The muffle chamber is broken down lengthwise into zones of heating, firing, and cooling.

Two external side surfaces are used to transmit heat to the muffle chamber and this heat is propagated by conduction and radiation. The transfer of heat from the muffle chamber to the material is accomplished by radiation about the inside perimeter of the muffle-chamber element.

Figure 1 shows the diagram for the muffle-chamber element for a single channel.

Heat is radiated to the surface of the material from the two side walls on the inside and from the cover on the top, which is a double extended surface whose base is formed by the side walls of the channel.

We make the following assumptions in this analysis: 1) the process of heat transfer from the channel perimeter to the material is accomplished by radiation exclusively; 2) the emissivities of the muffle-chamber materials and of the fired materials are assumed equal to unity; 3) the heat flow within the fired material in the x-direction is negligibly small; 4) the channel is divided lengthwise into individual segments. Within the limits of each segment the temperature $T_2$ of the side surface of the muffle chamber is a constant, i.e., the smooth variation in temperature over the length of the channel changes into a stepwise variation, making it possible to change from a three-dimensional to a two-dimensional heat-transfer scheme.

The starting point of our analysis is the law of the conservation of energy, which in this case assumes the form of a system of balance equations for the transfer of heat by pure conduction within the volume of the muffle-chamber element and by radiation within the space of the channel bounded by the inside perimeter.

On the surface of the fired material we will isolate an area $dx$ which receives heat exclusively as a consequence of radiation from the bases and the extended surface. Heat is spent on the natural radiation and on the internal heat requirement $q_x$.

Let us compile the first balance equation of the system
\[ \sigma_0 T^4_x dx + q_x dx = \sigma_0 T^4_1 \phi_{1, dx} + \sigma_0 T^4_2 \phi_{2, dx} + \frac{1}{\delta} \int T^4_y dy \phi_{dy, dx}, \]  
(1)
where \( \phi_{1, dx}, \phi_{2, dx}, \) and \( \phi_{dy, dx} \) are angle factors.

According to the principle of angle-factor reversibility
\[ h \phi_{1, dx} = dx \phi_{1, dx}, \]
\[ h \phi_{2, dx} = dx \phi_{2, dx}, \]
\[ dy \phi_{dy, dx} = dx \phi_{dy, dx}. \]

We find the values of the angle factors \( \phi_{dx, dy}, \phi_{dx, 1}, \) and \( \phi_{dx, 2} \) from the formulas proposed by Jakob [1] to determine the angle factors between materials of infinite length in a direction perpendicular to the plane of the drawing:
\[ \phi_{dx, dy} = \frac{1}{2} \left( \frac{1}{h^2 + (y-x)^2} \right) dy; \]
\[ \phi_{dx, 1} = \frac{1}{2} \left( 1 - \frac{1}{h^2 + (l-x)^2} \right); \]
\[ \phi_{dx, 2} = \frac{1}{2} \left( 1 - \frac{1}{v^2 + x^2} \right). \]

We will substitute the values of the angle factors into (1) and we will reduce terms of the equation by dx:
\[ \sigma_0 T^4_x dx + q_x = \sigma_0 T^4_1 \left( 2 - \frac{l-x}{h^2 + (l-x)^2} \right) \]
\[ + \frac{1}{\delta} \int T^4_y \left( \frac{h^2}{h^2 + (x-y)^2} \right) dy. \]  
(2)

To compile the second equation, let us examine the conditions of heat exchange between the double extended surface and the ambient medium.

On the plane of the extended surface we will isolate an area \( dy \), situated at a distance \( y \) from the channel wall. The heat reaches \( dy \) as a consequence of radiation from the first and second bases and from the fired material. In addition, heat is supplied to the area \( dy \) as a consequence of heat conduction through the extended surface. Heat is expended on natural radiation. Using the basic principle of conservation of energy and evaluating the term characterizing the thermal conductivity, using the Fourier law for an extended surface of unit width, we obtain in the usual fashion
\[ \sigma_0 T^4_y \frac{dy}{dy^2} = \lambda \delta \frac{dy}{dy^2} + \sigma_0 T^4_y \phi_{dy, dy} + \sigma_0 T^4_1 \phi_{gy, dy} + \sigma_0 \int \frac{T^4_y}{\delta} \phi_{dy, dy}. \]  
(3)

Applying the principle of angle-factor reversibility and substituting the values of these factors into (3), we obtain the following expression for the second balance equation:
\[ \sigma_0 T^4_y = \lambda \delta \left( 2 - \frac{l-y}{h^2 + (l-y)^2} - \frac{y}{h^2 + y^2} \right) + \frac{1}{\delta} \int T^4_y \phi_{dy, dy}. \]  
(4)