PARTICULARITIES REGARDING THE THERMAL RADIATION INSIDE THE HEATING FURNACES FOR METAL FORMING

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Abstract

During the thermal processes in metallurgy, it is important in many cases to know the exactly particularities of heat transfer in a complex system. In the case of steel semi-finished products heating, the most of the models refers to the heat exchange by convection and radiation or to the heat transfer by conductivity inside the material. The heat transfer by radiation has two important components: the value of the heated surface by radiation, but also the more exactly determination of the values of the radiation coefficient. The aim of this analysis is to establish the basic relations for a model in order to help to evaluate the parameters of the heat transfer and energy consumption in the case of some metallurgical heating furnaces for billets heating. The article refers also to "the equivalent surface of heat transfer" (a new terms proposed by the author), subject to the way of billets disposal in various types of heating furnaces. Starting from considerations about the burning process of the fuels, there are established relations between the heat exchange coefficients, energy and metallic material saving. We have also to consider that it is an important difference between various types of continuous operating furnaces: while in the pusher-type furnace the billets may be disposed only one near the other, without free spaces between them, the walking-beam furnace or the rotary hearth furnace allows the disposal of the billets at the required distances. Modifying this distance there is the possibility of acting upon the billets stationary time in the furnace, upon temperature distribution in the section of the billets, upon oxidation and decarburization reduction. Saving energy and lost metal due to the oxidation process, means to have a cleaner environment.

Keywords: heat transfer, plastic deformation, billet, mathematical model, furnace.

1. INTRODUCTION REGARDING THE OBJECTIVES OF THE ARTICLE

When heating a billet or an ingot in view of forming, we have to take in consideration the main thermal phenomenon operating in the thermal space. In the figure 1 are summary presented the phenomena we have to take in consideration.

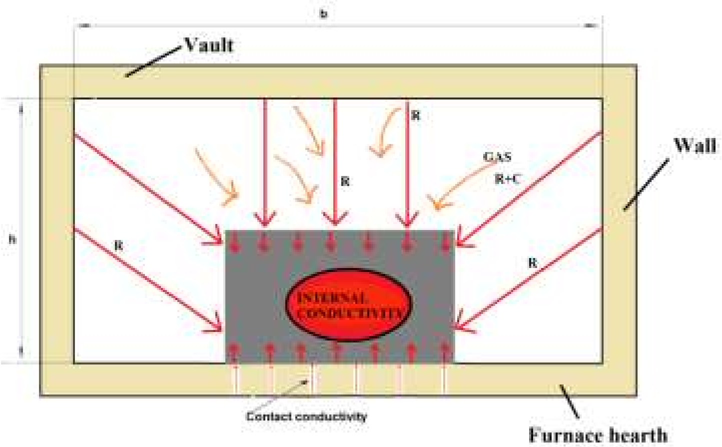

Figure 1: The main thermal phenomena operating in the furnace thermal space; GAS R+C: radiation and convection due to the flow gases; R: radiation from the thermal isolation; contact conductivity includes the effect of the oxides

The interaction between the heat exchange phenomenons is practically very complex and we have to take its complexity into account if we expect a good quality of heating and a reduction of energy and material consumption in order to reduce value of the industrial *ecological footprint* resulted from these processes.

2. HEAT EXCHANGE BY RADIATION (R) BETWEEN THE FURNACE'S THERMAL ISOLATION AND THE STEEL BILLET

The computation of the heat exchange by radiation between the thermal isolation components can be calculated using the angular coefficient of radiation, φ, recommended by Heiligenstaedt [1].

$$\varphi = \frac{1}{\pi}\left[\frac{1}{B\cdot L}\cdot \ln\frac{(1+B^2)(1+L^2)}{1+B^2+L^2} - \frac{2}{B}arctg(L) - \frac{2}{L}arctg(B) + \frac{2}{L}\sqrt{1+L^2}\,arctg\frac{B}{\sqrt{1+L^2}} + \frac{2}{B}\sqrt{1+B^2}\,\mathrm{arctg}\frac{L}{\sqrt{1+B^2}}\right] \quad (1)$$

For the equation (1), it is noted: h- height of the heating space; b – bright and l – length of the heating space; $B = h/b$ and $L = l/b$ (figure 1). In the case of heat exchange between the thermal isolation and the billets, the coefficient φ is [2]:

$$\varphi = \frac{1}{2\pi}\left(\frac{B}{\sqrt{1+B^2}}\cdot \arcsin\frac{L}{\sqrt{1+B^2+L^2}} + \frac{L}{\sqrt{1+L}}\cdot \arcsin\frac{B}{\sqrt{1+B^2+L^2}}\right) \quad (2)$$

If all the thermal energy radiated by the isolation, Q_{pm}, is receipted by the heated metal, it is possible to write:

$$Q_{pm} = \alpha_{pm}\cdot \varepsilon_{pm}\cdot S\cdot(\theta_p - \theta_s) \quad (3)$$

θ_s: temperature of the metallic billet at the surface, °C (figure 2)
S : heated surface of the metallic material (billets); here it is necessary to calculate the *„equivalent surface of heat exchange*", m² [3]
θ_p: temperature of the thermal isolation, inside the furnace, °C
α_{pm}: heat exchange coefficient by radiation between the thermal isolation and the billet, $kJ\cdot m^{-2}\cdot h^{-1}\cdot K^{-1}$
ε_{pm} : thermal emissivity reciprocal coefficient for metal and refractory isolation

But, a part of this radiation is absorbed by the flue gases. The absorption process depends on the partial pressure of CO_2 and H_2O. The absorbed thermal energy by radiation, Q_{abs}, is equal with the quantity of energy which the metal could receive from the flue gases if the temperature of the gases is equal with the temperature of the thermal isolation:

$$Q_{abs} = \alpha_{gpm}\cdot \varepsilon_p\cdot S\cdot(\theta_p - \theta_s) \quad (4)$$

α_{gpm} : heat exchange coefficient from the gases to the metallic material, if it is considerate that the temperature of the gases is the same with the temperature of the thermal isolation, $kJ\cdot m^{-2}\cdot h^{-1}\cdot K^{-1}$
ε_p: thermal emissivity coefficient of the isolation

So, the real value of Q_{pm} is:

$$Q_{pm} = S(\alpha_{pm}\cdot \varepsilon_{pm} - \alpha_{gpm}\cdot \varepsilon_p)\cdot(\theta_p - \theta_s) \quad (5)$$

It is possible to write:

$$S_p\cdot(\alpha_{gp}\cdot \varepsilon_p + \alpha_c)\cdot(\theta_g - \theta_p) = S\cdot(\alpha_{pm}\cdot \varepsilon_{pm} - \alpha_{gpm}\cdot \varepsilon_p)\cdot(\theta_p - \theta_s) + S_p\cdot q_{ex} \quad (6)$$

where:
S_p: surface of the thermal isolation
θ_{ga}: temperature of the flue gases at the exit from the furnace, °C
α_{gp}: radiation heat exchange coefficient between the gases and the thermal isolation, $kJ\cdot m^{-2}\cdot h^{-1}\cdot K^{-1}$

ε_p: thermal emissivity coefficient of the isolation (refractory material)

α_c : convection heat exchange coefficient between the gases and the thermal isolation, $kJ \cdot m^{-2} \cdot h^{-1} \cdot K^{-1}$

θ_g: temperature of the flue gases, °C

q_{ex} : thermal flow thru the furnace's isolation , $kJ \cdot m^{-2} \cdot h^{-1}$

If it is noted the ratio between the „equivalent surface of heat exchange“and the surface of the thermal isolation $\sigma = {}^{S}/_{S_p}$, the equation (6) will be:

$$\theta_g \cdot (\alpha_{gp} \cdot \varepsilon_p + \alpha_c) = \theta_p \cdot (\alpha_{gp} \cdot \varepsilon_p + \alpha_c + \sigma \cdot \alpha_{pm} \cdot \varepsilon_{pm} - \sigma \cdot \alpha_{gpm} \cdot \varepsilon_p) - \theta_s \cdot \sigma(\alpha_{pm} \cdot \varepsilon_{pm} - \alpha_{gpm} \cdot \varepsilon_p) + q_{ex} \quad (7)$$

It is clear that, together with the value of the emissions coefficients, the *equivalent surface of heat exchange* has also an important impact on the value of the heat transmitted to the metallic body (billet).

3. RADIATION HEATING SURFACE OF THE STEEL BILLETS

In order to analyse the radiation surface of the billets (the equivalent surface of heat transfer) there were taken in consideration some frequent cases for the heating furnaces [3]. In figures 2 and 3 there are presented the cases of square and rectangular sections. Next, it is a comparison between the two cases.

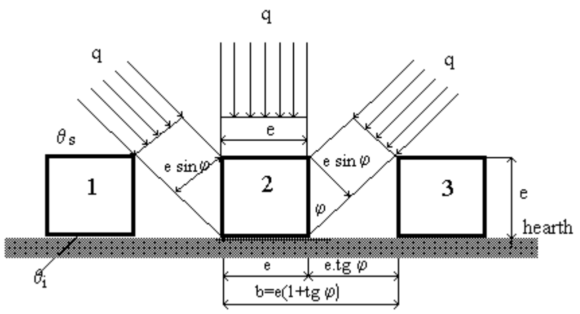

Figure 2 : Heating of the billets with square section on the hearth of the furnace; q - thermal flow ; φ– angle of radiation; θ_s – temperature of the upper surface; θ_i – temperature of the inferior surface of the billet; l – length of the billet

$$S = e \cdot l + 2 \cdot e \cdot l \cdot sin\varphi = e \cdot l(1 + 2sin\varphi) \quad (8)$$

The temperature difference between the hot and cold surface, $\Delta\theta$, will be:

$$\Delta\theta = \theta_s - \theta_i = \frac{q}{\lambda} \cdot (1 + 2\sin\varphi) \quad (9)$$

λ – thermal conductivity of the steel

If the billets are stuck, as in case of pusher-type furnace with heating by the upper face, then sin φ=0

If the billets are very far one to the other ($e << e.tg\ \varphi$), then *lim(sin* φ) *=1*. In this situation it is obtained a better uniformity of temperature on the billet section than in case of the both faces heating in the pusher-type furnace. The heating mode equivalent to the situation of the both faces heating in the pusher-type furnace is obtained for the case of the walking beam

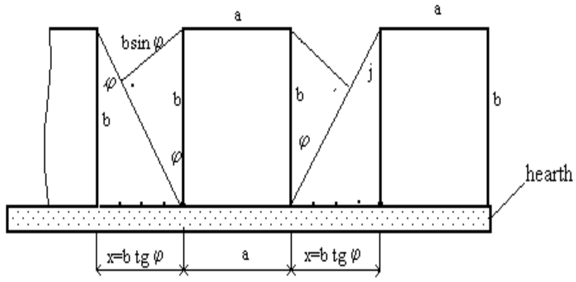

Figure 3: Heating of billets with rectangular section on hearth of the furnace (same notation as for figure 2); ${}^{a}/_{b} = f$

$$S = l \cdot b \cdot (j \cdot tg\varphi + 2\sin\varphi) \quad (14)$$

where j = 2 ÷0.4, depending on the distance between the billets

Examples:

x=0.5·a : *S=l·b*·(2·tg φ+*e*·sin φ)

x=2.5·a : *S=l·b*·(*0,4*·tg φ+*2*·sin φ)

Analyzing the obtained data, we can remark:

- maximal values of equivalent surface in the conditions *a =ct*, there are obtained at an incidence angle of the thermal radiation of 30 degrees; from these, the biggest value is obtained in case of x=2.5a , for ratio of a/b = 0.25
- the smallest values of the coefficient of optimum distance, z, are obtained for a distance between billets x=1.5a........2a and for values of incidence angle of thermal radiation of 45......60 degree; under these conditions the optimum value of the ratio between the section sides of billet must be 0.6.......1.2 ; at the distance

furnace when φ=60 degree

The most favourable situation from the thermal point of view, in case of heating square billets, would be when φ=60°. Having in view the requirement to provide a high degree of furnace hearth charging as well the same productivity, it must be taken into consideration the case when φ=45°. This represents, on basis of established data, practically supposed the optimum situation from the point of view of uniformity heating, heating time and furnace productivity.

To analyze easier the heating mode of the billets, it will be introduced the notion "*specific time of internal heating -STIH*" representing the time necessary to heat a billet with effective thickness (which corresponds to the geometric thickness and is different from the *thermal thickness* which is reported to the Biot criteria; in case of square section, *X=e*), reported to the thermal diffusion "a_0", suitable to the heating temperature:

$$t = \frac{X^2}{a_0} = \frac{X^2 \cdot c \cdot \rho}{\lambda} \tag{10}$$

For modelling the heat transfer by radiation, we propose to use the main relations (table 1), in the case of square section.

Table 1: Functions for modelling the heat transfer by radiation for the square section billets

function of equivalent surface of heat exchange	$k_1 = 1 + 2sin\varphi$
function of heating duration	$k_2 = \frac{(1+4\sin^2\varphi)}{(1+2\sin\varphi)}$
function of the specific time of internal heating	$i = \frac{1}{(1+4\sin^2\varphi)}$
criteria of optimum distance between billets	$z = \frac{(1+tg\varphi)}{(1+4\sin^2\varphi)}$
specific time of internal heating	$t = X^2 \cdot \frac{i}{a_0}$

Specific time of internal heating:

$$t = \frac{e^2 \cdot c \cdot \rho}{(1+4\sin^2\varphi)\cdot\lambda} \tag{11}$$

The heating duration:

$$\tau = t \cdot \frac{\theta_f - \theta_i}{\Delta\theta} \cdot \frac{1+4\sin^2\varphi}{1+2\sin\varphi} \tag{12}$$

The productivity "P" could be calculated using the relation:

$$P = \frac{m \cdot n \cdot \Delta\theta}{t \cdot k_2 \cdot (\theta_f - \theta_i)} = \frac{m \cdot L_c \cdot \Delta\theta}{t \cdot k_2 \cdot (\theta_f - \theta_i) \cdot e} \tag{13}$$

where L_c is the length of the furnace

x>2.5a, practically the coefficient „z" remains constant.

- the specific time of heating is first of all influenced by the shape of the billet section: the minimum value is obtained for flat billets (f=5.5; i=0.05); for value of the ratio very close, the value of the STIH decreases by the increase of the distance x;

-for the interval considered optimum (x=1.5a.......2a), i has the value (0.3....0.4) for f=(0.6....0.5), (φ=45 degree) and (0.18....0.22) for f=(1.1....0.9), (φ=60 degree)

-for x>2a, the STIH for the same values of the incidence angle is very little modified

For modelling the heat transfer by radiation, we propose to use the main relations presented in table 2 in the case of rectangular section:

STIH - representing the time necessary to heat a billet with effective thickness (which corresponds to the geometric thickness and is different from the *thermal thickness* which is reported to the Biot criteria; in this case section, *X=b*), reported to the thermal diffusion "a_0", suitable to the heating temperature is the same as in equation (10).

For modelling the heat transfer by radiation, we propose to use the main relations (table 2), in the case o rectangular section.

Table 2: Functions for modelling the heat transfer by radiation for the rectangular section billets

function of equivalent surface of heat exchange	$k_1 = j \cdot tg\varphi + 2 \cdot \sin\varphi$
function of heating duration	$k_2 = \frac{j \cdot tg^2\varphi + 4\sin^2\varphi}{j \cdot tg\varphi + 2\sin\varphi} = \frac{1}{k_1 \cdot i}$
function of the specific time of internal heating	$i = \frac{1}{j \cdot tg^2\varphi + 4\sin^2\varphi}$
criteria of optimum distance between billets	$z = \frac{b\cdot(f+tg\varphi)}{j\cdot tg^2\varphi + 4\sin^2\varphi} = b\cdot(f+tg\varphi)\cdot i$
specific time of internal heating	$t = X^2 \cdot \frac{i}{a_0}$

The heating duration:

$$\tau = t \cdot \frac{\theta_f - \theta_i}{\Delta\theta} \cdot \frac{j \cdot tg^2\varphi + 4sin^2\varphi}{j \cdot tg\varphi + 2sin\varphi} \tag{15}$$

θ_f and θ_i: final and initial average temperatures of the heated material

The productivity "P" will be calculated in this case with the relation:

$$P = \frac{m \cdot L_c \cdot \Delta\theta}{t \cdot k_2 \cdot (\theta_f - \theta_i) \cdot b \cdot f} \tag{16}$$

4. HEAT EXCHANGE BY CONVECTION AND RADIATION OF THE burned GASES

Equation (7) correlates the temperature of the flue gases, temperature of the thermal isolation and the temperature of the billets (θ_s). But, the establishing of the values of the heat exchange coefficients put yet some dificulties.

The thermal flow sanded to the metallic material (billets) includes:

- radiation thermal flow from the thermal isolation

$$q_{pm} = \left(\alpha_{pm} \cdot \varepsilon_{pm} - \alpha_{gp} \cdot \varepsilon_p\right)\left(\theta_p - \theta_s\right) \quad [\mathrm{kJ \cdot m^{-2} \cdot h^{-1}}] \tag{17}$$

- radiation and convection thermal flow from the flue gases

$$q_{gm} = \left(\alpha_{gm} \cdot \varepsilon_m + \alpha_c\right)\left(\theta_g - \theta_s\right) \tag{18}$$

- the conductive thermal flow from the furnaces hearth to the heated metal (important especialy at the beginning of the heating process) [4], when the billet or the ingot is introduced in the thermal space of the aggregate (figure 1):

$$q_{vm} = -\lambda grad(T) \tag{19}$$

or

$$q_{vm} = \kappa\left(\theta_v - \theta_s\right) \tag{20}$$

κ : global coefficient of heat transfer from the hearth to the billet (or ingot)

The total thermal flow received by the billets is:

$$q = q_{pm} + q_{gm} + q_{vm} \tag{21}$$

There were obtained the following expressions regarding the complex heat exchange by radiation and convection in the analysed furnace:

1. The heat exchange coefficient between the thermal isolation and the billets:

$$\alpha_1 = \frac{\alpha_{gm} \cdot \varepsilon_m + \alpha_c}{\alpha_{gp} \cdot \varepsilon_p + \alpha_c} \cdot \left(\alpha_{gp} \cdot \varepsilon_p + \alpha_c + \sigma \cdot \alpha_{pm} \cdot \varepsilon_{pm} - \sigma \cdot \alpha_{gm} \cdot \varepsilon_p + \frac{q_{ex}}{\theta_p - \theta_s}\right) [\mathrm{kJ \cdot m^{-2} \cdot h^{-1} \cdot K^{-1}}] \tag{22}$$

2. The heat exchange coefficient between the flue gases and the billets:

$$\alpha_2 = \frac{\alpha_{pm} \cdot \varepsilon_{pm} - \alpha_{gpm} \cdot \varepsilon_p}{\alpha_{gp} \cdot \varepsilon_p + \alpha_c + \sigma \cdot \left(\alpha_{pm} \cdot \varepsilon_{pm} - \alpha_{gpm} \cdot \varepsilon_p\right)} \cdot \left(\alpha_{gp} \cdot \varepsilon_p + \alpha_c - \frac{q_{ex}}{\theta_g - \theta_s}\right) + \alpha_{gm} \cdot \varepsilon_m + \alpha_c [\mathrm{kJ \cdot m^{-2} \cdot h^{-1} \cdot K^{-1}}] \tag{23}$$

3. The heat transfer coefficient between the between the furnace hearth and the billet:

$$\kappa = \frac{k_3}{2\sqrt{\tau}} \left\{ \frac{e^{n^2}}{n}\left[1 - \Phi(n)\right] - \frac{1}{n} + \frac{2}{\sqrt{\pi}} \right\} \tag{24}$$

where:

$$\Phi(n) = \frac{2}{\sqrt{\pi}} n \left(1 - \frac{n^2}{3 \cdot 1!} + \frac{n^4}{5 \cdot 2!} - \frac{n^6}{7 \cdot 3!} + \ldots\right) \text{ and } k_3 = \frac{\lambda_3}{\sqrt{a_3}} = \sqrt{\lambda_3 \cdot c_3 \cdot \rho_3}$$ (refers to the hearth properties) [5], [6]

2 The oxidation process and the furnace's output

In the case of the heating process in furnaces using the combustion, the source of energy can be analysed from tow points of view:

a) as component which can reduce the material looses due to the oxidation process
b) as component which assure the technological conditions for the heating process

The presence of the oxides layers on the surface of the billets or ingots influences the transfer of the heat by contact conduction. In the Figures 4 and 5 are presented the values of the thermal conductivity of the ferrous oxides [4]. The values of the thermal conductivity of the oxides, together with the value of the thermal conductivity of the refractory material of the hearth influences the transfer coefficient, κ (equation 24)
The values of the thermal flow, q, which determines the value of Δθ [figure 1 and eq. (13) and (16)] and by this the furnace's output are presented in the figure 6.

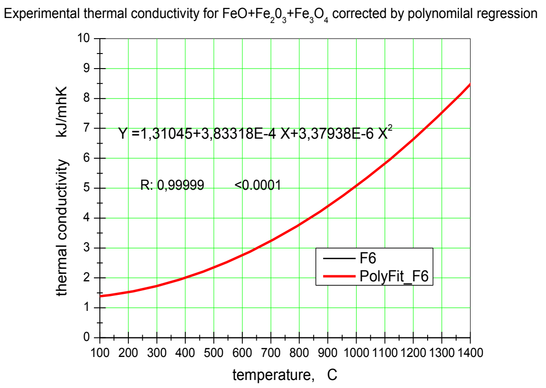

Figure 4: values of the thermal conductivity for ferrous oxides obtained from experimental analyzes

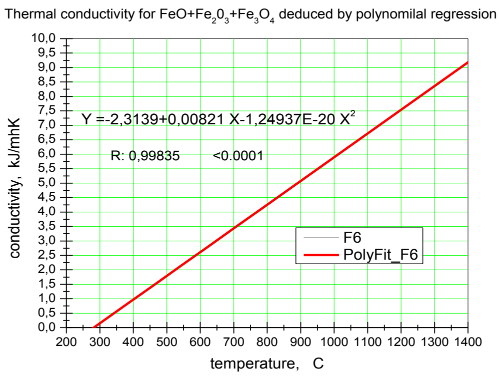

Figure 5: values deduced for the thermal conductivity for ferrous oxides from theoretical data

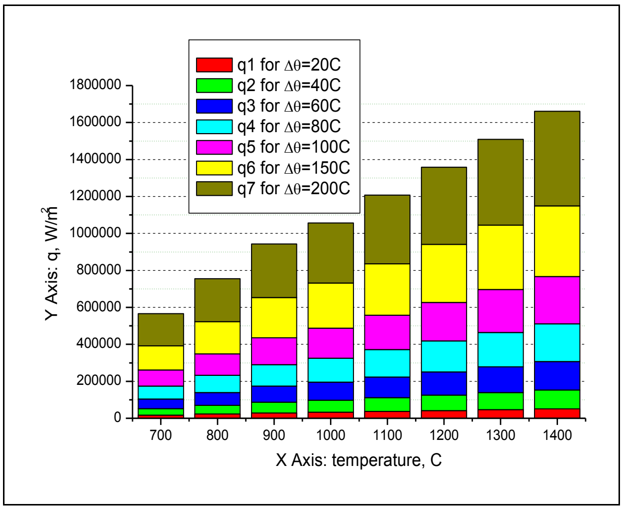

Figure 6: Thermal flow, statistics (function of Δθ, thermal conductivity and temperature of the refractory hearth)

CONCLUSIONS AND DISCUSSION

Using the proposed general solutions for the remodelling of the thermal regime it can be obtained a better control of the temperatures in each heating zone of the furnace and to correlate it with the necessary temperatures of the billets. It is also possible to control the temperature of the thermal isolation, and by this to save thermal energy. By the established equations it is possible to control the flue gases temperature in each heating zone of the furnace in correlation with the temperature of the billet or ingot. The coefficients α_1, α_2, κ as well as the function Φ(n) and the disposal mode of the billets or ingots are at the basis of the control process of heat exchange between the flue gases, metallic material and the thermal isolation.
For the next steps of studies we have in view to establish some correlations between the heating technology, the energy consumption and the ecological footprint of the thermal aggregates on the environment.

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