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THERMO-PHYSICAL PHENOMENA OF THE METALLIC ALLOYS USED IN AUTOMOTIVE INDUSTRY AT HIGH TEMPERATURE

For the constructive-functional improvement of heat treatments equipement for metallic alloys used in automotive industry imposes first a theoretical analysis of thermo-physical phenomena into a chamber furnace heated electrically at high temperature, with resistors, in respect of the fact that this type of furnace is the most used for heat treatments applied to metallic materials.

In electric furnaces with resistors indirect heating are met all that three ways of heat transfer. For resistor furnaces of high temperature the basic ways of heat transfer are those through thermal conduction and radiation. Convection heat transfer way cannot be treated practically as a separate mechanism for these furnaces in regard to the other two mentioned-above.

Key words: *heat conductivity, heat radiation, method of boundary elements, thermal balance, equation.*

Introduction

Thermal conduction, which flows heat from a corps's part with a high temperature towards another part of the same corps, or other corps with whom is in direct contact, with a lower temperature, is marked out into a resistors furnace at heat pass through insulated walls of the furnace, through resistive elements, accessories and pieces and devices that make the heating charge.

Thermal radiation, in contradiction to thermal conduction, does not need the existence of a material propagation medium; on the contrary, it is much more efficient as vacuum is more advanced. It represents a part of electromagnetic radiation emitted by a corps at a certain temperature and assures heat exchange between different surfaces within furnace as well as between exterior walls and environment.

For the most parts, used treatment atmospheres do not participate at heat exchange through absorption and emission because they can be considered as perfectly transparent medium. This view bases on the fact that some gaseous components with molecular structure symmetrically non-polar (H_2, N_2, O_2) are transparent through their nature.

On the other hand, the surfaces within furnace are not sufficiently processed to be considered mirrors and in consequence of that they emit radiation after a direction proportional to angle cosine made by emission direction and the normal to emission surface.

Insulating refractory materials, resistors and accessories are enough harsh being characterized by emitted radiation or reflected step by step in a diffuse way. As well behave pieces for treating that besides processing state they can present surface oxidations in case of air heat treatment. The hypothesis of diffuse radiation presents the reduction advantage of representing complexity of physical phenomenon.

All real surfaces from furnace that participate to radiation heat exchange are supposed to be identically emitter or absorbing to radiation independent to the wavelength (they are called grey surfaces).

Another hypothesis taken into consideration in radiation heat exchange, in electric furnaces with resistors, refers to the thermal insulation surfaces assumed opaque and isotherms.

Radiation heat exchange represents the preponderant way of heat transfer into furnaces with indirect heating through resistors especially for high treatment temperatures or for furnaces with high power density.

Natural convection is due to fluid's flow as a result of density difference produced by heat transfer itself. It contributes to indirect transfer of the heat from warmer mediums to colder medium and together with radiation it participates to heat losses through walls towards furnace environment. In contrast with radiation heat transfer, heat convection has a certain importance to surface heat exchange only at the beginning of the heat cycle when the charge is cold. While the surface charge heats convection becomes insignificant.

Electric conduction in resistors is influenced by material’s electric resistivity hence the resistor is made and its temperature.

Use of finite element method and finite differences in solving heat transfer problems

The use of finite differences method in solving heat treatment problems must be preceded by digitization in finite elements respective the creation of a thermal model [1]. Testing of thermal model there realizes through analogy with an electric model.

Heat transfer equations are approximate through linear partial differential equations type [2]:

– parabolic, through their nature dependent of time, which need singularity conditions in time spatial (frontier) and temporal (that characterize transitory regimes);

– elliptic, through their nature dependent of spatial conditions and present solutions independent of time (that characterize stationary regimes);

On this line, the problem of creating thermal model relies on digitization principle under two aspects:

– spatial geometric digitization ($i\Delta x$, $j\Delta y$, $k\Delta z$)

– time digitization ($n\Delta t$)

Thermal node identifies with geometrical center of gravity of the created incremental finite element. The entire material mass is considered concentrated into thermal node.

– Spatial digitization

In terms of specific particularities of each application, the digitization manner of a solid corps, no matter its geometric form it can be done in much many variants. The most significant ones take into account the following basic criteria:

– To each control volume, control surface or “i” control point it is associated a thermal node;

– In each thermal node heat energy must be conserved;

For industrial electric furnaces the most adecquate digitization manner in finite elements is plan to radiation heat exchange and pointed to conduction heat transfer.

a) Triangular digitization [1]

For interior nodes:

Thermal node is created into the center of gravity of the triangle (mediators’ intersection). Thus, temperature gradient is perpendicular on separation boundary of two spatial increments under different temperatures. This kind of model assures the easy determining of thermal conductance between nodes and in addition, it creates much many freedom degrees for thermal flow. The afferent thermal balance to each node corresponds per assembly to the thermal element (figure 1).

For nodes with boundary conditions (convection and/or radiation):

This model type take into consideration spatial (boundary) sigularity conditions. Under these conditions at the exchange boundary with environment the equationa of thermal balance will be:

$$\Phi_{i\ \mathrm{conduction}} = \Phi_{\mathrm{convection}} + \Phi_{\mathrm{radiation}} \tag{1}$$

b) Linear unidirectional digitization

In this case it adopts a pointed digitization with one liberty degree. This digitization is used frequently for thermal conduction analysis through insulated walls of the industrial furnaces whose shape and dimensions can lead to the approximation of thermal conduction through a plan plate.

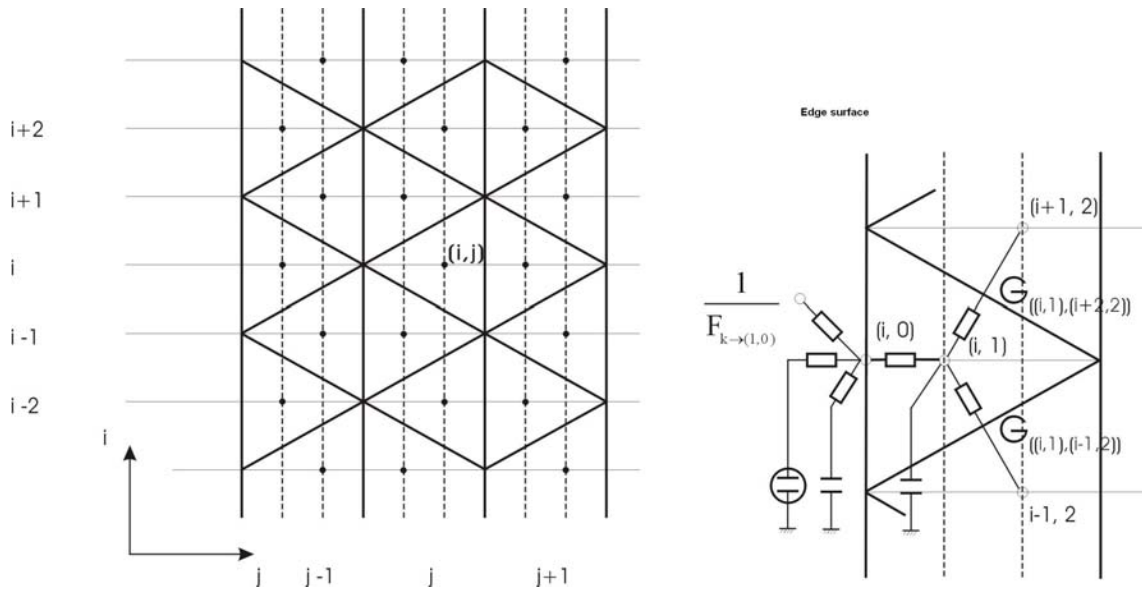

Figure 1. Spatial digitization with triangular elements and electric model analogus to thermal one [1]

– Digitization after time axis

With respect to chosen calculus methodology there distinguishes on convergence criterion the next forms of temporal digitization [1]:

a) Case of determining the value of the unknown parameter in pure explicit form:

$$\theta_i\ (t_{n+1}) = f\ [\theta_i(t_n)] \tag{2}$$

Meaning direct calculus of temperature afferent to a spatial thermal node at a certain time level (t_{n+1}) supposing known temperature values at a anterior time level (t_n) (figure 2). In this case the minimum stability condition is: $\frac{\Delta t}{(\Delta l)^2} \le \frac{1}{2}$ (for solving case with finite differences).

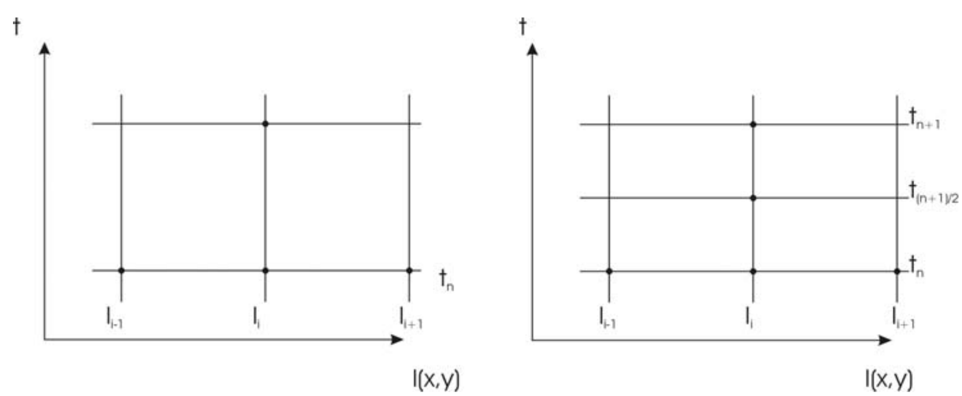

Figure 2. Time digitization [1]

b) Determining case of the unknown parameter's value in explicit form with speediness increase of convergence rapid (Crank – Nicholson method).

This method reduces dependence in regard to time increment Δt and it is independent of report's value $\Delta t / (\Delta l)^2$.

The use of explicit expression of differential equations imposes the control of the solutions' stability condition at every time turning and the eventual correction of time increment for the next time turning.

Heat transfer within heat treatment equipment with electric heating with resistors

An analysis of heat transfer in any closed system uses the equation of energy balance that satisfies the principle of conservation of energy:

$$\frac{dE_i}{dt}+\frac{dE_g}{dt}=\frac{dE_e}{dt}+\frac{dE_a}{dt} \tag{3}$$

dE_i / dt – energy that enters the system in time unit; dE_g / dt – energy generated within the system in time unit; dE_e / dt – energy that gets out the system in time unit; dE_a / dt – energy accumulated by the system in time unit.

– Conduction heat exchange

The equation of thermal conduction in a material medium of λ thermal conductivity, γ density and c massic heat has the expression:

$$\text{div}\ (- \lambda\ \text{grad}\ T) + \gamma\ c\ dT/dt = P \tag{4}$$

where P represents volume power dissipated into material.

By integrating this equation on an “i” elementary volume, surrounded by “j” vicinities, obtains the digitization form that represents thermal balance of “V_i”:

$$C_i \frac{d\theta_i}{dt} = \Phi_i + \sum_j G_{ij}(\theta_i - \theta_j) \tag{5}$$

where: G_{ij} – thermal conductance between elements “i” and “j” [W/K]; $C_i = \gamma_i\ c_i\ V_i$ – thermal capacity of ‘i” element respectively stored heat quantity when its temperature increases with a degree [Wh/k] ; $\Phi_i = P_i\ V_i$ – energy produced by “i” element [W].

Thermal node identifies geometrically with the element whose material mass is concentrated in element's center of gravity. For plan representation this node is mediators' intersection of the geometric element's sides. Mediators' choice permits the generation of a net of exact thermal conductance. On the other hand temperature gradient generated through temperature difference between the center of the node and the middle of boundary side is always perpendicular on the boundary side that separates the two elements.

Thermal balance, considered to be in thermal node, corresponds to the balance on incremental element in its assembly, and thus the used method is a nodal method which conserves thermal energy.

Thermal nodes are connected between them through conductance that can be conductive, convective or radiant.

Heat sources are situated in thermal node. The positioning of the thermal node is dependent on geometric form of the element, fact that can be an inconvenient if the element is a right-angle triangle (thermal node will be distributed on the hypotenuse) or for obtuse (thermal node will be distributed outside the element). The most used elements are equilateral triangles as seen in figure 3:

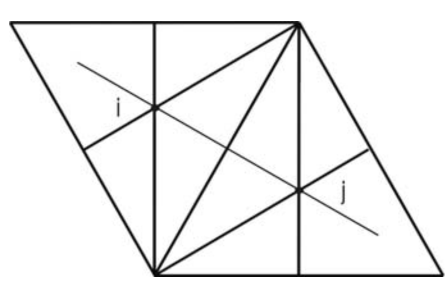

Figura 3. Positioning of thermal node for plan digitization through triangular elements [1]

Conductance is calculated between adjacent nodes that have a common boundary (figure 4). Conductance between center and the considered boundary is developed for each node:

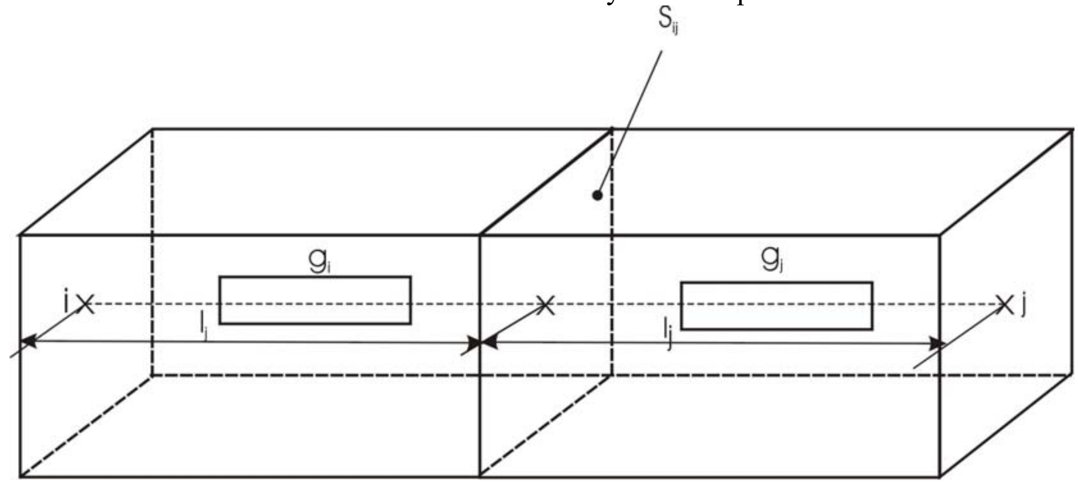

Figura 4. Presentation of conductive conductance [3]

$$\frac{1}{G_{ij}} = \frac{l_i}{\lambda_i S_{ij}} + \frac{l_j}{\lambda_j S_{ij}} = \frac{1}{g_i} + \frac{1}{g_j} \tag{6}$$

where: G_{ij} – conductance between elements centers “i” and “j” [W/K]; g_i, g_j – thermal conductance between nodal centers and separation boundary between two adjacent elements [W/K]; l_i, l_j – distance between the nodes of the elements “i”, “j” and common boundary [m]; λ_i, λ_j – thermal conductivities of elements’ nodes “i” and “j” [W/mK]; S_{ij} – separation surface between two adjacent elements [m^2].

– Radiation heat exchange

Heat exchange preponderant in electric furnaces with resistors indirect heating, with work temperatures and high energy densities is radiation. Resistive systems brought to temperatures of 950 – 1200 °C can be perfectly assimilated with infrared radiations emitters.

On the other hand, in the paper is treated heat processing on some charges made of pieces thermally thin, which change preponderantly radiation heat energy with surfaces from the furnace interior and with pieces surfaces from their vicinity.

From these reasons it treats in detail the problems connected with the specific of radiation heat transfer.

Thermal nodes are considered of surface, wherefore in treating radiation heat transfer in furnaces there are formulated the following hypotheses [1]:

– Furnace chamber is closed made of isotherm elementary surfaces assumed opaque with limited emission to a zone from wall’s vicinity and with a thickness sufficiently reduced (to be assimilated with geometric surfaces);

– All surfaces are grey their emittance is independent of wavelength;

– These surfaces are the locus of some emissions and diffuse reflections according to Lambert law;

– The medium from the chamber is perfectly transparent and does not participate to radiation heat exchange.

Radiation heat exchange treats under the next three aspects: heat exchange exclusively between two surfaces, heat exchange between more that two surfaces and determination of form factors (view).

a) Radiation heat exchange between two surfaces

In thermodynamic analysis energy density represents radiated energy from a surface in time unit and from surface unit. Stefan and Boltzmann established the of radiation net exchange law between two dark corps:

$$\Phi_{ij} = \sigma C_0 S_i (T_i^4 - T_j^4) \quad [W] \tag{7}$$

C_0 = 5,67 x 10^{-8} [W/m^2k^4], representing Stefan – Boltzmann constant; T_i, T_j – corps temperatures that exchange heat energy between them [K]

A corps heating depends on its form, its emissive and absorbent properties of heat radiation. Thus, radiation heat transfer equation between two real grey surfaces, diffuse, gets the form:

$$\Phi_{ij} = C_0 F(\varepsilon) S_i F_{i \to j} (T_i^4 - T_j^4) \quad [W] \tag{8}$$

where:

Φ_{ij} – net heat flow between two area surfaces S_i and S_j, in time; T_i – temperature of emitting corps [K]; T_j – temperature of receiving corps [K]; F (ε) – function that depends on emission factors ε_i and ε_j of the two material surfaces that exchange energy through thermal radiation. This function takes into account the fact that real surfaces are not dark corps but grey corps and respectively not all the incident energy will be absorbed but a part will be reflected towards other surfaces or outside the system in addition, reflectivity phenomenon between surfaces will be able to take place many times or repeatedly in both directions. F(ε) is a surface characteristic and can be assimilated by a surface thermal resistance. $F_{i \to j}$ – form factor (view) of the corps “i” in regard to corps “j” considers the fact that only a part of the emitted radiation by corps “i” will be received by corps “j”. This factor is due to surface orientation and can be assimilated through a spatial thermal resistance.

F(ε) and $F_{i \to j}$ can be coupled in an interdependency form with the help of grey form factor, F_{fg}, that signifies the part of heat flow that leaves the surface of element ”i” and is absorbed by element “j” after many diffuse reflections on other nodes of the transfer domain.

$$F_{fg} = \overline{F(\varepsilon) F_{i \to j}} = f(F_{i \to j}, f(\overline{\varepsilon_i, \varepsilon_j}), \overline{S_i, S_j}) \tag{9}$$

In practical calculi uses the corrected emissive concept of a corps system that takes into consideration incomplete absorption of radiant energy of grey corps and reflected flows [4].

$$\Phi_{ij} = C_0 F(\varepsilon_c) \rho F_{i \to j} (T_i^4 - T_j^4) \tag{10}$$

where ρ – reflection index;

Due to the fact that at the surface of a corps submitted to a radiant flow takes place besides reflection also an absorption phenomenon and transmissivity an it can be written the following relation between ponderosity coefficients of dividing the incident flow on that surface:

$$\rho + a + \tau = 1 \tag{11}$$

where:

a – absorption factor; τ – transmission factor.

For opaque corps it can be written:

$$\rho + a\tau = 1, \tag{12}$$

According to Kirchhoff’s law:

$$\rho = 1 - a = 1 - \varepsilon \tag{13}$$

In figure 5 is represented the scheme of the flows implied in radiation heat exchange at the surface of a corps.

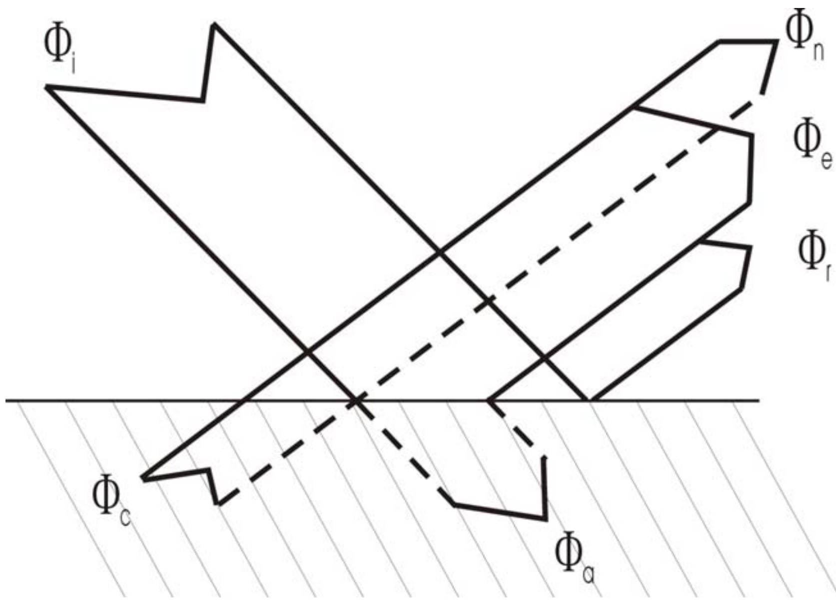

Fig. 5. Flows implied in surface heat exchange through radiation [2]

Φ_i – is the flow emitted of all the surfaces from the chamber which come in direct contact with the considered surface or after one more multiple reflections; Φ_a – is the flow absorbed by surface; ($\Phi_a = \varepsilon\Phi_i$); Φ_r – is the flow reflected by surface ($\Phi_r = (1-\varepsilon)\Phi_i$); Φ_e– emitted flow at the surface due to corps temperature ($\Phi_e = \varepsilon C_0 S T^4$); Φ_n – radiated net flow, effective loss through radiation at surface ($\Phi_n = \Phi_e - \Phi_a$); Φ_c – conducted flow.

For opaque – grey corps, absorption factor depends on spectral composition of the incident flow.

b) Radiation heat exchange simultaneous between more surfaces

in case of furnace made of closed chambers, of different geometric forms in whose work spaces are placed charges of different forms and configurations, the surfaces that exchange energy through radiation are more than two, thermal exchange being realized through direct radiation and indirect too by the reflections of these surfaces. The analysis of heat transfer in this situation makes considering energetic radiance notion of each surface. The analysis imposes the assumption that all surfaces are considered grey – diffuse, uniform in temperature and emissive and reflective properties are constant on the entire surface.

Thus, there define two notions:

I – irradiation respectively total incident radiation in time unit and surface unit [W/m^2];

B – energetic radiance respectively total emissive radiation in time unit and surface unit [W/m^2].

On the grounds of calculi simplification supposes that energetic radiation and irradiation are uniform on the entire surface, fact that can introduce an error because grey – diffuse surfaces do not strictly submit to this condition.

Thus, for a certain material surface “i”, is written an energetic balance:

Energetic radiance =emitted energy + sum of reflected radiations

It gets:

$$B_i = \varepsilon_i E_{ni} + \rho_i I_i = \varepsilon_i E_{ni} + (1 - \varepsilon_i) I_i \quad [\mathrm{W/m^2}] \tag{14}$$

Radiant energy flow that leaves a surface “S_i” is:

$$\Phi_i = (B_i - I_i)\, S_i = \varepsilon_i S_i\, (E_{ni} - B_i) / (1 - \varepsilon_i) \quad [\mathrm{W}] \tag{15}$$

And expresses the speed of heat transfer from surface “S_i”.

Considering energy exchange between two surfaces S_i and S_j, then:

$B_i\ S_i\ F_{i\to j}$ – quantity from total radiant energy that leaves from surface S_i and reach surface S_j;

$B_j\ S_j\ F_{j\to i}$– quantity from total radiant energy that leaves from surface S_j and reach surface S_i.

Energy net exchange between the two surfaces is [4]:

$$\Phi_{ij} = B_i S_i F_{i\to j} - B_j S_j F_{j\to i} = S_i F_{i\to j}(B_i - B_j) \quad [W] \tag{16}$$

Considering that surface “i” flows towards surface “j”, total irradiation is the sum of all irradiations I_j from the other “j” surfaces:

$$\Phi_i = \varepsilon_i S_i (E_{ni} - \sum_j I_j) \tag{17}$$

Because irradiation can be expressed as $S_j\ B_j\ F_{i\to j} = I_j S_i$ and keeping in mind reciprocity property: $S_j F_{j\to i} = S_i F_{i\to j}$ gets:

$$\Phi_i = \varepsilon_i S_i\ (E_{ni} - \sum_j F_{i\to j} B_j\ ) = S_i\ B_i - \sum_j F_{i\to j} B_j \tag{18}$$

where:

$$B_i = (1 - \varepsilon_i) \sum_j F_{i\to j} B_j + \varepsilon_i E_{ni} \tag{19}$$

In this case taking into account the fact that:

$$\Phi_{ij} = S_i F_{i\to j} B_i - S_j F_{j\to i} B_j = S_i F_{i\to j}(B_i - B_j)\ [W] \tag{20}$$

It can define:

$G_{ij}(fg) = S_i F_{i\to j}$, as a conductance of geometric form in radiant heat exchange [4].

On the other hand, radiated net flow from a radiant surface „i” is given by relation (15) and can be defined as surface conductance at radiant heat exchange [4]:

$$G_{is}(fs) = \varepsilon_i S_i/(1-\varepsilon_i) \tag{21}$$

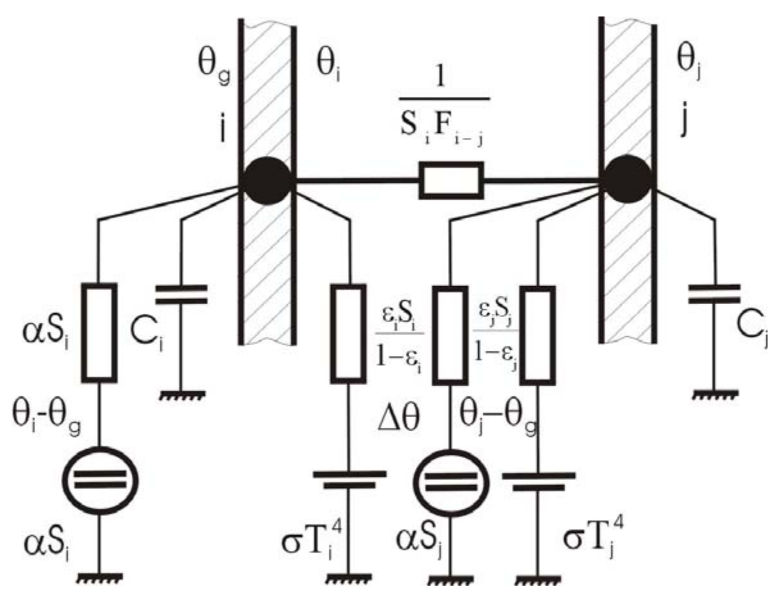

Fig. 6. Analogic electric presentation at radiation heat exchange [4]

In this case explicit equation for radiation heat exchange between more surface becomes:

$$B_i = (1 - q_i)\sum_j F_{i\to j}B_j + \varepsilon_i E_{ni} = \frac{1-\varepsilon_i}{S_i}\sum_j S_i F_{i\to j}B_j + \varepsilon_i S_i E_{ni} =$$

$$= \frac{1-\varepsilon_i}{S_i}\sum_j G_{ij}B_j + (1-\varepsilon_I)G_{is}E_{ni} \quad (22)$$

b) Determination of form factors (view)

Net flow emitted from surface "i" towards surface "j" is:

$$d\Phi_i = \frac{C_0 T_i^4}{\pi r^2}\cos\psi_i \cos\psi_j ds_i ds_j \quad (23)$$

and from surface "j" to surface "i" is:

$$d\Phi_j = \frac{C_0 T_j^4}{\pi r^2}\cos\psi_i \cos\psi_j ds_i ds_j \quad (24)$$

Net flow between the two surfaces "i" and "j" is:

$$d\Phi_{ij} = d\Phi_i - d\Phi_j = \frac{C_0(T_i^4 - T_j^4)}{\pi}\frac{\cos\Psi_i \cos\Psi_j ds_i ds_j}{r^2} \quad (25)$$

By integrating:

$$\int d\Phi_{ij} = \frac{C_0(T_i^4 - T_j^4)}{\pi}\int_{s_i}\int_{s_j}\frac{\cos\Psi_i \cos\Psi_j ds_i ds_j}{r^2} \quad (26)$$

It gets:

$$S_i F_{i\to j} C_0 (T_i^4 - T_j^4) = -\frac{C_0(T_i^4 - T_j^4)}{\pi}\int_{s_i}\int_{s_j}\frac{\cos\Psi_i \cos\Psi_j ds_i ds_j}{r^2} \quad (27)$$

It defines as a thermal function of heat exchange the multiply:

$$S_i F_{i\to j} = \frac{1}{\pi}\int_{s_i}\int_{s_j}\frac{\cos\Psi_i \cos\Psi_j ds_i ds_j}{r^2} \quad (28)$$

Through radiation reverse analysis of a surface S_j towards a surface S_i determines similarly:

$$S_j F_{j\to i} = \frac{1}{\pi}\int_{s_i}\int_{s_j}\frac{\cos\Psi_i \cos\Psi_j ds_i ds_j}{r^2} \quad (29)$$

It results reciprocity property:

$$S_i F_{i\to j} = S_j F_{j\to i} \quad (30)$$

which means that radiation heat transfer function is the same no matter the form and corps arrangement.

Into a system surrounded by corps and radiant mediums, thermal flow emitted through radiation by one of the corps towards all the other corps is equal to radiation of the implied corps:

$$\Phi_i = \sum_{j=1}^{n}\Phi_{ji} = \sum_{j=1}^{n}\Phi_i F_{i\to j} \quad (31)$$

It gets the next control relation:

$$\sum_{j=1}^{n} F_{i\to j} = 1,\ \text{unde}: F_{i\to j} \in [0,1] \tag{32}$$

For some surfaces arranged in simple geometric configurations, specialty literature presents simplifying calculus formulas or the form factors (view) [5].

For more complex configurations or more precise determinations it is necessary the achievement of a calculus of surface double integral (figure 7):

$$F_{i\to j} = \frac{1}{s_i}\int_{s_i}\int_{s_j} \frac{\cos\psi_i \cos\psi_j}{\pi r^2} ds_i ds_j \tag{33}$$

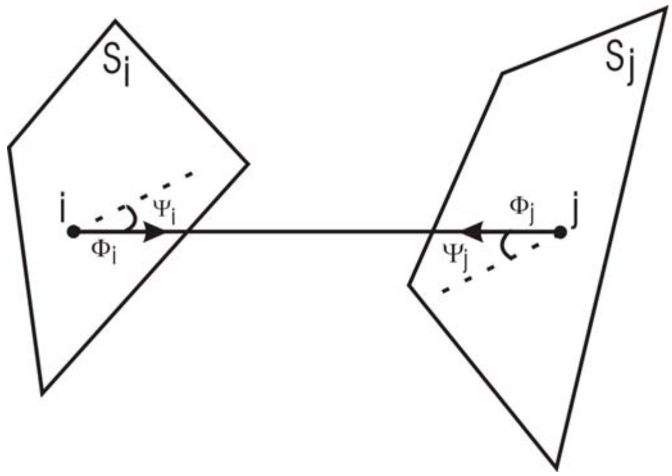

Fig. 7. Determination of view factors through surface double integral [2]

Sparrow demonstrated that form factors (view) can be determined with the help of a double circulation starting from Green–Gauss transformation formula of a surface double integral into a simple circulation (figure 8):

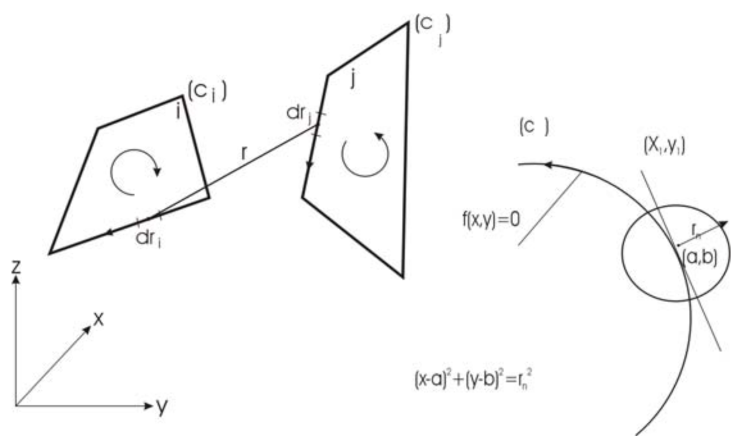

Fig. 8. Determination of view factors through double circulation [2]

$$F_{i\to j} = \frac{1}{2\pi S_i}\int_{C_i}\int_{C_j} \ln(r)\, dr_i\, dr_j \tag{34}$$

C_i, C_j – represents contours that limit surfaces S_i and S_j; dr_i, dr_j – represents the elementary lengths on the contours C_i respective C_j; r – distance between elementary lengths dr_i and dr_j .

The adopted method for determining view factors is Romberg method which consists in getting an approximation of an integral value at the limit of a time step that goes to zero. In adopted calculi the procedure consists in essence in realizing two distinct steps:

In the first step realizes an approximation of the double circulation by repeatedly applying trapeze rule, with steps lengths of h, h/2, h/4, …, for each segment of the contours. This method replaces the double integral with double sum:

$$IK = h1h2 \Bigg[\frac{f(A,C)+f(A,D)+f(B,C)+f(B,D)}{4} + \sum_{j=1}^{2^N-1}\frac{(f(A,C)+jh_2)+f(B,C+jh_2)}{2} + \frac{1}{2}\sum_{\substack{i=1\\ \Delta j=2}}^{2^N-1}\frac{f(A+ih_1,C)+f(A+ih_1,B)}{2} + \sum_{\substack{i=1\\ \Delta i=2}}^{2^N-1}\sum_{\substack{j=1\\ \Delta j=2}}^{2^N-1}(f(A+ih_1,C+jh_2)) \Bigg] \quad (35)$$

A, B – integration limits after direction r_1 (on C_1 contour); C, D – integration limits after direction r_2 (on C_2 contour); $h_{iN} = (B - A)/2^N$ – integration step after contour C_1; $h_{jN} = (D - C)/2^N$ – integration step after contour C_2; k = i x j – applications number of the double integral; $2^N = n$ – represents the number of steps or repeated applications of the trapeze rule; dr_1, dr_2 – vectors direction unit, after direction r_1 on contour C_1, respective r_2 on contour C_2.

$f = 0,5 \ln [(x_2 - x_1)^2 + (y_2 - y_1)^2 + (z_2 - z_1)^2]$

$i \in [1 \ldots k]$

x_1, y_1, z_1 – coordinates of the unitary vector dr_1; x_2, y_2, z_2 – coordinates of the unitary vector dr_2;

In next step realizes the Romberg augmentation using recursively formula that uses values I(n,1) determinate through anterior step:

$$I(m,n) = I(m-1,n+1) + \frac{\{I(m-1,n+1) - I(m-1,n)\}}{4^m - 1} \quad (36)$$

where:

n – number of rows of a diagonal matrix; m – number of columns of a diagonal matrix.

For curve surfaces the problems is to determine the optimum value of the contour increment linear assimilated. To this end is necessary to know the equation and digitization expression of the curved contour. Thus, for curved contours, assimilable with a circle centered in point A(a, b):

$$(x - a)^2 + (y - b)^2 = r_n^2 \quad (37)$$

It is given contour tangent in coordinate point (a,b) that intersects the circle centered in point (x_1, y_1) and:

$$\begin{cases} \delta_1 = x_1 - a \\ \varphi_1 = y_1 - b \end{cases} \quad (38)$$

Starting from the fact that the tangent will intersect the circle in two antipodal points (δ_1,φ_1), chosen so that they have the same sign:

$$\nabla fx\{\delta_1 \vec{i} + \varphi_1 \vec{j}\} \quad (39)$$

It is (δ_1,φ_1) the initial solutions for intersection point of the circle with the contour and (δ_2,φ_2) the best approximation:

$$\begin{cases} \varphi_2 = \varphi_1 + g \\ \delta_2 = \delta_1 + k \end{cases} \tag{40}$$

where $g << \delta_1$, respective $k << \varphi_2$

It gets:

$$(\delta_1 + g)^2 + (\varphi_1 + k)^2 = r_n^2 \tag{41}$$

Which by Taylor series development becomes:

$$f(a + \delta_2, b + \varphi_2) = f_0(a,b) + \delta_2 \frac{\partial f}{\partial x}\bigg|(a,b) + \varphi_2 \frac{\partial f}{\partial y}\bigg|_{(a,b)+....} \tag{42}$$

Supposing f(a,b) = 0, it gets:

$$\left(g\frac{\partial f}{\partial x} + k\frac{\partial f}{\partial y}\right)\bigg|_{(a,b)} = -\left(\delta_1\frac{\partial f}{\partial x} + \varphi_1\frac{\partial f}{\partial y}\right)\bigg|_{(a,b)} \tag{43}$$

Solving equation system there are found the unknowns g and k, achieving (δ_2,φ_2). Replacing (δ_1,φ_1) with (δ_2,φ_2) the process can go on until convergence, (δ_n,φ_n), through (m-1) iterations, and the point on the contour will become ($a + \delta_n$) and ($b + \varphi_n$).

For plane contours that intersect themselves, in order to avoid mathematical incompatibilities given by possible situations of ln(0) (when the distance between the two contour increments is zero) for these contour parts applies formula [2]:

$$\varepsilon = -\frac{1}{2\pi S_1}\int_{r_1=0}^{L}\int_{r_2=0}^{L}|r_2 - r_1|dr_1dr_2 = -\frac{1}{2\pi S_1}L^2[\ln(L) - 1,5] \tag{44}$$

which will be added to the sum of view factors calculated for the other functions.

The matrix of view factors will be completed with values of the view factors that are not calculated through integrals but using reciprocity formula:

$$F_{i\to j}\, S_i = F_{j\to i}\, S_j \tag{45}$$

For the walls equipped with resistors it can be admitted with approximation that parts from the flow emitted by resistors that reach the charge, after reflections of the refractory wall, makes to appear a form factor as a result of two multiplied factors:

$$F_{R\text{-}S} = F_{R\text{-}P}\, F_{R\text{-}S} \tag{46}$$

$F_{R\text{-}P}$ – View factor between resistor and wall; $F_{R\text{-}S}$ – view factor between wall surface and charge’s calculated through integration method.

It results the view factor between two resistive adjacent elements:

$$F_{R\text{-}R} = 1 - F_{R\text{-}P}\, F_{P\text{-}S} \tag{47}$$

Conclusion

Convection effect is taken into consideration when the resistors are at relative low temperatures, situation which is generally of short duration. Generally the constructors of electric furnaces meant for heat treatments equip the furnaces with electrically operated agitators for leveling the atmosphere distribution within furnace chamber. These mechanic training devices of the furnace gases improve in some manner convection heat exchange especially at the charge surface but the precise determination of convection heat exchange coefficient remains very difficult to set.

The difficulties of determining convection heat exchange coefficient are due mostly to the ignorance of fluid’s speed; that crosses different zones from furnace and to thermo hydraulic resistance variation which the charge is introducing by its configuration and distribution. Lab studies has shown that neglecting convection heat exchange that takes place in the furnaces with temperatures over 800°C does not produce significant calculus errors because over this temperature the ponderosity of radiation heat exchange is over 90%. On the other hand, convective thermal exchange reduces at high temperatures due to the decrease of gases viscosity within the furnace chamber.

Heat exchange preponderant in electric furnaces with resistors indirect heating, with work temperatures and high energy densities is radiation. Resistive systems brought to temperatures of 950 – 1200 °C can be perfectly assimilated with infrared radiations emitters.

On the other hand, in the paper is treated heat processing on some charges made of pieces thermally thin, which change preponderantly radiation heat energy with surfaces from the furnace interior and with pieces surfaces from their vicinity.

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