Determinations methods of emissivities for gas or oil fuel flame and furnace inner wall surface

Part 1

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In function of total emissivity $\varepsilon$ of the furnace inner wall surface situated behind the flame and others characteristics, a general formula for the combustion flame total emissivity $\varepsilon_f$ is deduced. Also a new determination method for $\varepsilon_f$, with development of necessary specific scientific fundamentals, is established. This method functionally is founded on the adequate decrease of radiation flux received by considered wall surface, using a tubular cooled screen with a translation movement. As a particular case of $\varepsilon_f$ determination resulted the Schmidt formula, but are clarified the conditions of an efficient precision utilization. By functionally particularization of $\varepsilon$ determination method can result the classical laboratory method using the total radiation pyrometer.

1. Introduction

Theory and experiment results show a difference between the total emissivity and emissivity having a average value in an interval relative small of wavelength used by an infrared camera (IRC) or other devices. This difference can be large for combustion gases, but for solid bodies even can be very small, which explained the great use of IRC for measurement of surface temperatures. The use of total radiation pyrometer (TRP) in place of IRC give the main advantages to measure of real combustion gases thermal flux and precisely to effect the thermal balance for a heated by combustion gases of the inner furnace wall surface [1]. Indeed the IRC

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takes into account only of received radiations in an interval relative small for wavelength. Thus for combustion gases especially containing CO₂ and H₂O having selective radiation can miss a sensible quantity of radiated possible energy. However, the use of IRC is the great advantage to realize the infrared thermograms. In a more detailed analysis, it results that emissivity, for surface of most materials (so called gray bodies), is a function of the emitting surface condition, temperature and wavelength of measurement range. For gases, their emissivity depend of temperature, nature of gas, the thickness of the emitting layer, wavelength range and gas pressure. Gases possess a much smaller radiating power and their volume participates in radiation. Most solids have a continuous spectrum, i.e. they emit radiant energy of all wavelengths but gases radiate energy only within a certain wavelength range. The gaseous atmosphere in any fuel-fired furnace or combustion plant comprises mainly CO₂ and H₂O, with the greater capacity for emission and absorption of thermal radiations. Also in combustion gases can appear unburned products as gaseous and solid components. Especially in the case of fuel oil together with solid fuel combustion, can be very important the radiation of solid carbon small particles yielded in flame. In this last case, combustion gases radiation can be enough small in comparison with soot radiation. Due the great complexity and difficulties for precise calculation, only experimental determination on a real furnace by an adequate method of ε determination, gives correct values. The same situation is for total emissivity εf determination of furnace inner wall surface behind the flame, for which also a method is proposed. As important infrared thermography applications using an IRC, two methods for research an testing of gas oil and gas fuel combustion are conceived, together with procedures of establishing of necessary emissivities. These methods, developing the theoretical fundaments, are based on a scientific analysis of infrared thermogram characteristics, obtained for the combustion of a gas oil droplet in an unheated miniaturized furnace and for gas fuel in a small tubular furnace.

2. DETERMINATION METHOD FOR TOTAL EMISSIVITY εf

The main measurement are effected with a special total radiation pyrometer, determining the thermal radiation fluxes Φ₀, Φ₁ (with and without a small mobile by round cooled screen for incandescent wall surface behind the flame, but having negligible influence by cooling on combustion process development), in two specific situations. Also was used the measurement of the furnace inner refractory wall surface temperature T_w, in a considered point of S₁ small surface behind the flame. The measured total thermal radiation flux of Φ₀ in a direction (∆) normal to the flame symmetry axe and unity heat receiving a surface belonging to S₁, it results from the summation of the total normal thermal radiation flux from the S₁.
surface with temperature $T_w$ and the total normal thermal radiation flux of $\Phi_f$, only from the flame (removing the refractory wall radiation by the cooled screen). The proposed method uses the deduction of a formula for $\varepsilon_f$ calculation. By writing the thermal balance equation for an unit surface belonging in center of the surface $S_1$:

$$\Phi_c + \Phi_w = \varepsilon \sigma T_w^4 + (1 - \varepsilon) \Phi_w + \Phi_m,$$  \hspace{1cm} (1)

where $\sigma = 10^{-8} \text{C}$, and $C$ is the Stephan-Boltzman constant; $\Phi_w$ – total thermal radiation incident flux to the receiving wall unit surface of $S_f$ from the flame and flame surrounding refractory walls, (when furnace burner is in operation); $\Phi_c$ – thermal flux yielded, to the receiving unit surface, by convection of heat transfer due hot combustion gases flow; $\Phi_m$ – thermal flux transmitted by conductibility in furnace wall, towards the ambient medium, from the receiving unit surface. For numerous modern industrial furnaces having a good exterior thermal insulation, $\Phi_m$ has a very small value. From relation (1), it results:

$$\Phi_w = \sigma T_w^4 - (\Phi_c - \Phi_m) \varepsilon^{-1},$$  \hspace{1cm} (2)

and the normal radiation flux from the refractory wall surface placed behind the flame is increased with the part of reflected $\Phi_w$

$$\Phi_b = \varepsilon \sigma T_w^4 + (1 - \varepsilon) \Phi_w.$$  \hspace{1cm} (3)

Replacing (2) in (3) it results:

$$\Phi_b = \sigma T_w^4 - \psi,$$  \hspace{1cm} (4)

with

$$\psi = (\Phi_c - \Phi_m)(1 - \varepsilon) \varepsilon^{-1}.$$  \hspace{1cm} (5)

Thus, the measured total thermal radiation flux, above presented, is:

$$\Phi_t = \Phi_f + (\sigma T_w^4 - \psi)(1 - \varepsilon_f).$$  \hspace{1cm} (6)

Taking into account of (6) it results the flame total emissivity in ($\Delta$) direction:

$$\varepsilon_f = 1 - (\Phi_t - \Phi_f)(\sigma T_w^4 - \psi)^{-1}.$$  \hspace{1cm} (7)

When $\psi = 0$, $\varepsilon_f$ decreases, becoming

$$\varepsilon_{fd} = 1 - (\Phi_t - \Phi_f)(\sigma T_w^4)^{-1}.$$  \hspace{1cm} (8)

Admitting $\Phi_1 = \sigma T_1^4$ and $\Phi_f = \sigma T_f^4$

$$\varepsilon_{fd} = 1 - (T_1^4 - T_2^4)/T_w^4.$$  \hspace{1cm} (9)

The formula (9) can be used when the total radiation pyrometer measures the conventional temperatures $T_1$ and $T_2$ corresponding to the fluxes $\Phi_1$ and $\Phi_f$ taking the used apparatus emissivity, equal with unity. Also formula (8) is used, but when the total radiation pyrometer is calibrated and measures direct the fluxes $\Phi_t$ and $\Phi_f$
(as specific thermal power). Indeed a total radiation pyrometer generally comprises
the main two parts: a sensor (optical system focusing the received radiation on a
detector especially as a thermopile) providing an electrical signal proportional the
received thermal radiation flux, and a secondary part converting the sensor output
signal into temperature or thermal radiation flux readings. With $\psi = 0$, from (7), it
results as a particular case (8), so called Schmidt formula [2], also used for
example to the IFRF experiments. According to (5), the condition $\psi = 0$ is valid in
three cases: when $\varepsilon = 1$, that is refractory wall surface is admitted to be a black
body. In different industrial situations, the emissivity $\varepsilon$ can be near of this
condition, thus determining a small approximation for $\varepsilon_f$ determined by (8); when
$\Delta \Phi = \Phi_c - \Phi_m = 0$, this case can be obtained only accidentally, because for modern
industrial furnaces for high temperatures of combustion gases together with the
increase tendency of their usual flow velocities, $\Phi_c$ can be sensible and thus
increases $\Delta \Phi$; and when $\Phi_c = 0, \Phi_m = 0$, which represent a case industrial
impossible to be realized. With a very small error is possible to be applied (8) only

![Fig. 1 – Schematic representation of experimental furnace complex unit, from IERAB [3], used for research-testing of the natural gas and heavy fuel oil combustion, together with the combustion flame radiation:](image-url)

1 – experimental furnace with inner refractory walls and different slits with volume $1m \times 1m \times 4m$; 2 – generator of steam; 3 – condensate of steam; 4 – recuperator for combustion air preheating; 5 – air centrifugal fan; 6 – furnace burner; 7 – air by-pass of recuperator; 8 – central tank of heavy fuel oil (HFO); 9 – heater with steam of HFO; 10 – HFO pump; 11 – HFO filter; 12 – daily thank with inner heating; 13 – daily HFO pump; 14 – HFO electric preheater; 15 – fine filter; 16 – natural gas pressure regulator; 17 – chimney for combustion products exit; 18 – water pump of steam generator; 19 – additional water pump for cooling; 20 – water thank for the safety of measure devices cooling; 21 – device for flow measure of combustion air; 22 – regulator of combustion air flow; 23 – device for air temperature measure; 24 – cooled screen; 25 – cooled transversal slit; 26 – device for gas natural flow measure; 27 – thank for visual testing of injector; 28 – exhaustor for combustion gases.
when \( \psi \to 0 \), so that \( \varepsilon_f \to \varepsilon_{fd} \). This favorable situation especially is realized when combustion experiments take place into an experimental furnace, conceived in construction and combustion operation, to give a very small value to \( \psi \). The researches in flame radiation problems, are developed in prestigious institutions of the world, having a great scientific-technical importance, especial to obtain an efficient combustion and optimum intensification of the useful radiation thermal transfer in different furnace types. For this reason, many years ago was conceived and realized a complex experimental furnace unit at IERAB, schematic presented in Fig. 1 with annexed equipments for research-testing of natural gas and inferior fuel oil combustion together with the study of flame radiation [3].

3. DETERMINATION METHOD FOR TOTAL EMISSIVITY \( \varepsilon \)

The total emissivity \( \varepsilon \), during the long furnace operation, can change due the wear or tear as consequence of interaction between combustion products and refractory bricks as well in addition, due the eventual frequent local superheating. Thus, can appear differences in comparison with the laboratory experimental obtained value of \( \varepsilon \), which usual it results using the brick electric heating. These considerations recommends the application of proposed method, giving an efficient determination of \( \varepsilon \) real value. This method uses an adequate experiment based on the development of specific theoretic fundaments. According to (3) can be defined the apparent total emissivity of furnace inner wall surface:

\[
\varepsilon_a = \left[ \varepsilon \sigma T_w^4 + (1 - \varepsilon) \Phi_w \right] (\sigma T_w^4)^{-1} = \Phi_b (\sigma T_w^4)^{-1}. \tag{10}
\]

When \( \Phi_c = 0 \), \( \Phi_m = 0 \) and furnace burner operation is forbidden (without combustion gases in furnace volume), using (1) it results:

\[
\Phi_w = \sigma T_w^4. \tag{11}
\]

Substituting (11) in (10) is obtained \( \varepsilon_a = 1 \) independent of \( T_w \). According to the effectuated experiments, \( \varepsilon_a \) has a value inferior of unity but however near of the unity (Fig. 4). When it is maintained constant the temperature \( T_w \) and decreases the heat flux \( \Phi_w \) till \( \Phi_{wd} \) value, the apparent total emissivity decreases at the value \( \varepsilon_{ad} \) and with \( \Delta \varepsilon_a = \varepsilon_a - \varepsilon_{ad} \), becoming:

\[
\varepsilon_{ad} = \varepsilon + (1 - \varepsilon) \Phi_{wd} (\sigma T_w^4)^{-1}, \tag{12}
\]

with \( \Delta \varepsilon_a = (1 - \varepsilon ) (\Phi_w - \Phi_{wd} ) (\sigma T_w^4)^{-1} \). \tag{13}

From (12), when \( \Phi_{wd} \to 0 \), it results \( \varepsilon_{ad} \to \varepsilon \), and from (13) taking into account of (11) \( \Delta \varepsilon_a \to 1 - \varepsilon \). The thermal flux \( \Phi_w \) decrease is experimentally realized with the add of a long but small in diameter steel tub, cooled by water.
circulation between the tub double steel walls. This tub is moved by experiments development, at different distances \( z \) (Fig. 2), in (\( \Delta \)) direction normal at the energy receiving surface \( S_1 \). But, the decreased radiated flux \( \Phi_{bd} \), it results from (3):

\[
\Phi_{bd} = \varepsilon \sigma T_w^4 + (1 - \varepsilon) \Phi_{wd}.
\]  

(14)

From (12) \( \varepsilon_{ad} = \Phi_{bd} (\sigma T_w^4)^{-1} \) and using (14), it results \( \varepsilon_{ad} > \varepsilon \), appearing a positive systematic error on \( \varepsilon \) determination. With \( z \to 0 \) also \( \Phi_{wd} \to 0 \). The decreased flux \( \Phi_{wd} \) with a small value can give a very small difference between \( \varepsilon_{ad} \) and \( \varepsilon \) (but with a best operation), for a relative great value of reference angular factor \( f_j \) (which will be by experiments determined) corresponding to a small ratio \( z_r d^4 \). Notation \( d \) represents the exterior diameter of the cylindrical cooled tub. The obtained systematic small positive error on \( \varepsilon \), can be eliminated. By particularization of \( \varepsilon \) determination method when \( \Phi_{wd} = 0 \) (but with refractory brick electric heating in place of combustion flame and also measuring by a thermocouple the temperature \( T_w \)) can result \( \varepsilon = \Phi / (\sigma T_w^4)^{-1} \) using the classical laboratory method. Indeed the emitted thermal flux \( \Phi \) is measured with a total radiation pyrometer.

4. CONCEPTION AND ANALYSIS OF THEORETICAL FUNDAMENTS FOR \( \varepsilon \) DETERMINATION

First of all, to establish and clarify by an scientific analysis the theoretical fundaments of this method, are made some simplifying assumptions. Thus, the inner furnace without combustion gases is considered, and also the inner furnace hot surfaces radiating thermal energy, as black bodies with uniform temperature are assimilated. At the same time, is admitted that a small cooled tub introduction into working space of furnace don’t sensible modify the furnace thermal equilibrium and the heat losses are negligible. Are considered the surfaces \( S_2 \), \( S_3 \), …, \( S_i \) with areas \( A_2 \), \( A_3 \), …, \( A_i \) having the temperatures \( T_2 \), \( T_3 \), \( T_i \), which radiate thermal energy towards the surface \( S_1 \) with area \( A_1 \) and temperature \( T_w \). The total thermal radiation incident flux to the receiving wall unit surface of \( S_1 \), in general is:

\[
\Phi_{wd} = \sum_{n=2}^{n=i} \sigma n^4 f_n^{-1},
\]  

(15)

where \( f_n^{-1} \) is the angular factor of surface \( S_n \) with area \( A_n \) related to the area \( A_1 \). But the surface \( S_i \) with area \( A_i \) (Fig. 2) don’t radiate thermal energy towards the surface \( S_1 \) due the screening of the cooled tub, and thus \( \Phi_w \) is decreased with thermal flux \( \Phi_{j1} \), and become:

\[
\Phi_{wd} = \Phi_w - \Phi_{j1} = \Phi_w - \sigma T_j^4 f_j^{-1}.
\]  

(16)
In hypothesis when the surface $S_1$ is with very small dimensions so that all the points of surface $S_j$ to be practically “seen” of this under the same solid angle, with variation between 0 and 1, the angular factor $f_j^1$ can be calculated [4] from:

$$f_j^1 = (2\pi)^{-1} \int_{C_j} \cos \alpha \, d\beta,$$  \hspace{1cm} (17)

where $\alpha$ is the angle between normal ($n_A$) to element of area $dA_1$ and the normal ($n_l$) at the plan determined of element $dl$ belonging to the contour curve $C_j$ and the centre of area $dA_1$; $d\beta$ – the elementary plan angle under which is “seen” the element $dl$, from center of $dA_1$. For small values the angular factor, in practical calculation in a first approximation can be admitted as constant the temperature $T_j$ on the surface $S_j$.

![Fig. 2 – Schematic-conventional representation of the furnace working space, emphasizing a cooled tubular mobile screen and the thermal fluxes $\Phi_{wd}$, $\Phi_{bd}$, $\Phi_c$, $\Phi_m$: 1 – special total radiation pyrometer; 2 – furnace incandescent walls; 3 – Pt.-Pt. Rh – thermocouple measuring the temperature in symmetry centre of area $A_1$ corresponding to the wall small surface $S_1$; 4 – cooled tubular screen with $d$ exterior diameter; 5 – cold water entrance; 6 – water evacuation.](image)

When $T_j$ appreciably changes on the surface $S_j$, the flux $\Phi_{j1}$ is given by the quadruple integral and is more difficult for calculation, as follows:

$$\Phi_{j1} = \sigma \int_{S_j} \int_{S_1} T_j^4 \, dA_1 \, dA_j \, \cos \theta_i \, dA_1 \, \cos \theta_j \left( \pi x^2 \right)^{-1},$$  \hspace{1cm} (18)

where $dA_1$ and $dA_j$ represent the elementary areas belonging to areas $A_1$ and $A_j$; $\theta_i$ and $\theta_j$ – angles between the $x$ segment which unites the centers of elementary areas $dA_1$ and $dA_j$ with the normal direction at determined planes by $dA_1$ and $dA_j$. 

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**Emissivities for gas or oil fuel flame on furnace inner wall surface.**

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To give in evidence the practical variation possibilities of decreased apparent emissivity $\varepsilon_{ad}$, first of all, will be presented two hypothetic cases. In the first case, admitting $T_j = T_w = \text{const.}$, $\Phi_c = 0$, $\Phi_m = 0$, taking into account that from (11), and replacing (16) in (12) it results:

$$\varepsilon_{ad} = \varepsilon + (1 - \varepsilon) (1 - f_j^1). \quad (19)$$

Relation (19) establish the variation of $\varepsilon_{ad}$ for a furnace haven’t inner combustion gases, without heat losses in environment, with uniform temperature of furnace inner walls and without influence of cooled tub introduction. In this case, referring to the emissivity afferent of furnace inner wall, the surface walls radiate similar as a black body. Indeed, from (3) and (11) becoming $\Phi_b = \varepsilon \sigma T_w^4 + (1 - \varepsilon) \sigma T_w^4$, it results the flux of thermal energy radiates for wall surface unity:

$$\Phi_b = \sigma T_w^4. \quad (20)$$

Using (19), in a rectangular system of coordinate axes by straight lines ($L_1$) and ($L_2$) can be represented the variations of $\varepsilon_{ad}$ in ordinate, function of $f_j^1$ in abscise, when for example, the temperatures of inner wall surface are $T_{w1}$ and $T_{w2} > T_{w1}$. When $f_j^1 = 1$ are obtained the total surface wall emissivities $\varepsilon_1$ for $T_{w1}$ and $\varepsilon_2$ for $T_{w2}$. Will be admit $\varepsilon_1 > \varepsilon_2$, because in numerous cases this inequality is valid for the refractory walls used inner furnaces, working at high temperatures. For $f_j^1 = f_d^1$ where $0 < f_d^1 < 1$, and constant temperature $T_{w1}$, it is obtained $\varepsilon_{ad1}$ that if it is considered equal to $\varepsilon_1$, it results an absolute error in addition of $\Delta \varepsilon_1 = \varepsilon_{ad1} - \varepsilon_1$. According to (19), the relative error is:

$$\Delta \varepsilon_{r1} = 100 \frac{(1 - \varepsilon_1)(1 - f_d^1)}{\varepsilon_1} \cdot (21)$$

In consequence, $\Delta \varepsilon_{r1}$ is smaller when the emissivity $\varepsilon_1$ is larger and the considered angular factor $f_d^1$ taken for $\varepsilon_1$ measurement is more near of unity. Admitting that the cooled tube introduction inner furnace determines only the decrease of temperature considered uniform inner furnace, from $T_{w2}$ for $f_j^1 = 0$ till $T_{w1}$ for $f_j^1 = 1$. In this case $\varepsilon_{ad}$ will have a variation according to another curve ($C_w$) comprised between straight lines ($L_1$), ($L_2$), and thus in reality the resulted error will be decreased. In the second case, admitting $T_j > T_w$, $\Phi_c = 0$ and $\Phi_m = 0$, and by introduction of cooled tube, is decreased only the temperature of surface $S_1$, but in variation interval $0 \leq f_j^1 \leq 1$ with a constant temperature $T_j$ of surface $S_j$ is considered. In this case, similar substituting (16) in (12), follows:

$$\varepsilon_{ad} = \varepsilon + (1 - \varepsilon) \left(1 - f_j^1 \frac{T_j^4}{T_w^4}\right). \quad (22)$$

From (22) it results that will be obtained smaller values for $\varepsilon_{ad}$ as in the analyzed first case. According to (16), in a rectangular system of coordinate axes (Fig. 3) by straight line ($\Omega_1$) can be represented the variation of $\Phi_{wd}$ in ordinate, function in abscise of $f_j^1$, and for $T_j = \text{constant}$, this flux has a linear decrease with
increase of \( f_j^1 \). Thus \( \Phi_{wd} = \Phi_w \) for \( f_j^1 = 0 \) (miss the screen) and \( \Phi_{wd} = 0, z = 0 \), for \( f_j^1 = 1 \) (total screening). For the real industrial furnace the above mentioned decrease is more large according to a curve \( (\Omega_i) \), situated much under \( (\Omega_e) \), especial due the cooling of tubular screen introduction and others technological necessities of furnace operation.

Fig. 3 – Decrease of thermal flux \( \Phi_{wd} \) in function of angular factor \( f_j^1 \), when \( T_j = \) constant.

In general for \( \varepsilon \) obtaining using a general formula, it is necessary also to determine \( \Phi_{wd} \), and so from (14) it results:

\[
\varepsilon = (\Phi_{bd} - \Phi_{wd})(\sigma T_w^4 - \Phi_{wd})^{-1}. \tag{23}
\]

Can be measure, \( \Phi_{bd} \) with a total radiation pyrometer when miss the flame, together with \( T_w \) using a thermocouple, and \( \Phi_{wd} \) can be laboriously calculated from:

\[
\Phi_{wd} = \sum_{n=2}^{n=k} \sigma T_n^4 f_n^{-1}. \tag{24}
\]

The surface \( S_n \) with area \( A_n \) which thermal energy radiates towards the surface \( S_i \), can be divided in \( k \) small areas \( a_n \) for which, in a first approximation, can be admitted a constant mean temperature \( T_n \). Also, with approximation, the surfaces which radiate thermal energy, can be assimilated with black bodies. According to the theory, for relative great values of \( f_j^1 \), to determine by calculation the flux \( \Phi_{wd} \) is not necessary a great precision. Indeed, for this case \( \Phi_{wd} \) represents a small part of \( \sigma T_w^4 \), according to \( \Phi_{wd} \rightarrow 0 \) when \( f_j^1 \rightarrow 1 \). (will continue with the PART 2, of this paper).

NOMENCLATURE AND ABBREVIATIONS

IERAB – Institute of Energetics from Rumanian Academy, Bucharest; IFRF – International Flame Research Foundation (IJmuiden); SEGFT – Station d’Essais, Gas de France – Toulouse; \( \Phi = \sigma T^4 \) – radiation thermal flux of black body, [W/cm\(^2\)]
at temperature $T \ [K]$, where $\sigma = 10^{-8} \ \text{C}^\circ; \ C = 5.67 \cdot 10^{-4} \ [\text{W/cm}^2 \ \text{K}^4]$ is Stephan-Boltzmann constant for black body; $S_1$ – reference surface, is considered a small surface of furnace inner refractory wall which radiates in direction ($\Delta$), the received thermal energy, [cm$^2$]; $\varepsilon_r$ – combustion flame total emissivity; $\varepsilon$ – total emissivity of furnace inner wall surface; $z$ – distance between tubular screen extremity and the surface $S_1$; $\tau$ – time variation, [s]; $f_n^1$ – angular factor of surface $S_n$ with area $A_n$ related to the area $A_1$, with variation of $n = 2 - i$; $T_w$ – temperature of all surface $S_1$, behind the flame [K]; $\Phi_w$ – total thermal radiation incident flux to the receiving wall unit surface of $S_1$ from the flame and from furnace incandescent refractory inner walls when the furnace burner is in operation, and only from inner walls when misses the flame, [W/cm$^2$]; $\Phi_c$ – thermal flux yielded, to the receiving unit surface, by convection of heat transfer due to hoot combustion gases flow, [W/cm$^2$]; $\Phi_m$ – thermal flux transmitted by conductibility in furnace wall, towards the ambient medium, from the receiving unit surface, [W/cm$^2$]; $\Phi_f$ – total normal thermal radiation flux only from the flame, in direction ($\Delta$) normal to the flame symmetry axe, [W/cm$^2$]; $\Phi_t$ – total thermal radiation flux in direction ($\Delta$), summating $\Phi_f$ with the total normal thermal radiation flux yielded from the unity of $S_1$ surface, [W/cm$^2$].

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