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Research and testing methods for establishment of fuel oil combustion flame emissivity

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Abstract

The method for combustion flame total emissivity ε_f determination is especially founded on a new deduced formula. For the total emissivity ε, of furnace inner wall surface, a determination method with developments of scientific fundaments, is conceived. By a functional particularization results the laboratory classic method. As a particular case of εf , the Schmidt formula is obtained. Knowing the εf variation can improve different combustion procedures of furnace economic operation. For the miniaturized combustion of fuel oil as singel droplets, a research-testing method is proposed., using an increased emission of soot. Thus, in different points of flame, the emissivity is near of unity. Using an IR-Camera for droplet flame thermogram obtaining, are acquired valuable results for gas oils quality establishing.

1. Introduction

Emissivity, for surface of gray bodies, is a function of the emitting surface condition, temperature and wavelength of measurement. 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, i. e. having a selective radiation. The gaseous atmosphere in any fuel-fired furnace or combustion plant comprises mainly carbon dioxide and water wapor.Thus for industrial combustion, also the flame of fuel oil, usually is composed of polyatomic combustion gases especially CO_2 and H_2O with the greater capacity for selective emission and absorption of thermal radiatiions, together with bi and monoatomic combustion gases which are transparent at thermal radiation. Also can appear unburned products as gaseous and solid components. CO_2 and H_2O, are called non-grey combustion gases because absorb and emit selective radiation at discrete wavelengths. Especially in the case of fuel oil together with solid fuel combustion, can be very important the radiation of solid carbon particles yielded in flame. In this last case, combustion gases radiation can be enough small in comparison with soot radiation [1]. Only experimental determination on furnace by an adequate method of ε_f determination, gives real values. Also, for this reason is proposed a method for determination the real value of total emissivity ε .

2. Method for εf determination

At the beginning are effected two measurement of thermal radiation fluxes F_f , F_t from inner the furnace in specific situations (with and without a small mobile by round cooled screen for incandescent wall surface S1 as in Fig. 1, but having negligible influence on combustion process development). Was used a special total radiation pyrometer, together with the measurement of the furnace inner refractory wall temperature T_w in a considered point of a small surface S_1. For this reason, when by total radiation pyrometer the temperature T is measured, can be determined a thermal radiation flux for black body $F = \sigma T^4$ where $\sigma = 10^{-8}C$, and C is the Stephan – Boltzman constant referring to the same measurement unity of surface as F. According to Fig. 1, the measured total thermal radiation flux of F_t in a direction (Δ) normal to the flame symmetry axe and unity heat receiving surface belonging to S_1 , results from the summation of the total normal thermal radiation flux from the S_1 surface with temperature T_w behind the flame and the total normal thermal radiation flux of F_f only from the flame (when is removed the refractory wall radiation by the mobile cooled screen). By writing the thermal balance equation for an unit surface belonging in center of the surface S_1, we obtain:

$$F_c + F_e = \varepsilon \sigma T_w^4 + (1 - \varepsilon) F_e + F_m \qquad (1)$$

where F_e – total thermal radiation incident flux to the receiving unit surface of S_1 (from the flame and flame surrounding refractory walls when furnace burner is in operation); F_c – thermal flux yielded, to the receiving unit surface, by convection of heat transfer due hoot combustion gases flow; F_m – thermal flux transmitted by conductibility in furnace wall, towards the ambient medium, from the receiving unit surface; From relation (1), results:

$$F_e = \sigma T_w^4 - (F_c - F_m)\, \varepsilon^{-1} \qquad (2)$$

and the normal radiation flux from the refractory wall surface placed behind the flame, is increased with the part of reflected F_e :

$$F_b = \varepsilon \sigma T_w^4 + (1 - \varepsilon) F_e \qquad (3)$$

By replacing (2) in (3) one obtains

$$F_b = \sigma T_w^4 – A \qquad (4)$$

where

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$A = (F_c - F_m)(1 - \varepsilon)\ \varepsilon^{-1}$ (5)
and
$F_t = F_f + (\sigma T_w^4 - A)(1 - \varepsilon f)$ (6)
From (6) one obtain the total emissivity of the flame:
$\varepsilon_f = 1 - (F_t - F_f)(\sigma T_w^4 - A)^{-1}$ (7)
and when A=0 ,
$\varepsilon_f < \varepsilon_{f0} = 1 - (F_t - F_f)/ \sigma T_w^4$ (8)

With A = 0, from (7), results as a particular case (8), so called Schmidt formula [2], also used for example to the IFRF experiments. According to (5), the condition A = 0 is valid when: a) $\varepsilon = 1$, that is refractory wall surface is admitted to be a black body. In different industrial situations, the emissivity ε is near of this condition determining a small approximation for εf determined by (8); b) $\Delta F = F_c - F_m = 0$, this case can be obtained only accidentally, because for modern industrial furnaces the heat losses towards environment are very small, and for high temperatures of combustion gases together with the increase tendency of their usual flow velocities, Fc can be sensible and thus increases ΔF; c) $F_c = 0$, $F_m = 0$, represent a case industrial impossible to be realized, but using an experimental furnace, negligible ΔF is possible to obtain. If the emissivity ε has a small value and the heat flux F_c is important especially due large speed flow in furnace of combustion gases, neglecting A, the flame emissivity may be obtained with a sensible error. In conclusion, the application of formula (8) used in numerous cases, but with a very small error is possible to be applied only when $A \rightarrow 0$, so that $\varepsilon_f \rightarrow \varepsilon_{f0}$. This favorable situation for combustion flame emissivity determination, especially is realized when combustion experiments take place into an experimental furnace, conceived in construction and combustion operation, to give a very small value for A. The researches in flame radiation problems, which are developed in prestigious institutions of the world, have scientific-technical great importance, as well for optimum intensification of useful radiation thermal transfer in different furnaces. Indeed, this important useful heat is determined essential by the value of ε_f. Also, many years ago was conceived and realized a complex experimental furnace unit at IERAB, for research-testing of natural gas and inferior fuel oil combustion and flame radiation [3].

3. Method for ε determination

During the furnace operation, due the wear and tear as consequence of interaction between combustion products and refractory bricks as well due the eventual local superheating, the total emissivity ε especially after a long time, sensible changes. Thus in different industrial cases is very useful to establish a real value for ε . The proposed method application, requires adequate experiments based on the development of same theoretic fundaments. According to (3) can be defined the so-called apparent total emissivity of furnace inner wall surface, as:

$\varepsilon_a = F_b (\sigma T_w^4)^{-1} = [\varepsilon \sigma T_w^4 + (1 - \varepsilon) F_e] (\sigma T_w 4)^{-1}$ (9)

Admitting $F_c = 0$ and $F_m = 0$ when furnace burner operation is forbidden (inner furnace without combustion gases), from (1) results:

$F_e = \sigma T_w^4$ (10)

By substituting (10) in (9) is obtained $\varepsilon_a = 1$ independent of T_w. For real furnace according to the practical experiments, εa has a value inferior of unity but however near of this. When is maintained constant the temperature T_w and decreases the heat flux F_e till F_{ed} value, the apparent total emissivity decreases at the value εad and with $\Delta\varepsilon_a = \varepsilon_a - \varepsilon_{ad}$, becoming:

$\varepsilon_{ad} = [\varepsilon + (1 - \varepsilon) F_{ed}](\sigma T_w^4)^{-1}$ (11)

where

$\Delta\varepsilon_a = (F_e - F_{ed}) (\sigma T_w^4)^{-1}$ (12)

When $F_{ed} \rightarrow 0$, from (11) results $\varepsilon ad \rightarrow \varepsilon$, and from (12) taking into account of (10) $\Delta\varepsilon_a \rightarrow 1 - \varepsilon$. The thermal flux F_e decreasing is practical realized with the add of a long but small in diameter steel tub, cooled with water circulation between the tub double steel walls. During experiments, this cooled tub as mobile screen is moved by translation at different distances y, in (Δ) direction normal at the energy receiving surface S_1 (Fig. 2). In this case, the decreased radiated flux Fbd results from (3) as:

$F_{bd} = \varepsilon\ \sigma T_w^4 + (1 - \varepsilon) F_{ed}$ and $\varepsilon_{ad} = F_{bd} (\sigma T_w^4)^{-1}$ (13)

From (11) and (13) results $\varepsilon_{ad} \geq \varepsilon$, appearing a positive systematic error on ε determination which can be eliminate. Also $F_{ed} \rightarrow 0$ when $y \rightarrow 0$. But with a suitable precision is obtained $\varepsilon_{ad} \cong \varepsilon$ when F_{ed} has a very small value however practical admissible, for a convenient angular factor or ratio y/d (established by experiments), where d is the exterior diameter of the cylindrical cooled tub.

4. Theoretical fundaments and experiments for ε determination

In order to clarify 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 ,…Ai having the

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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 unit surface of S_1 , in general is:

$$F_e = \sum_{n=2}^{n=i} F_{n1} = \sum_{n=2}^{n=i} \sigma T_n^4 f_n^1 \quad (14)$$

Where f_n^1 is the angular factor of surface Sn with area An related to the area A_1. But the surface S_j with area A_j (Fig. 2) don't radiate thermal energy towards the surface S_1 due the screening of the cooled tub, and thus F_e is decreased with thermal flux F_{j1} , and become :

$$F_{ed} = F_e - F_{j1} = F_e - \sigma T_j^4 f_j^1 \quad (15)$$

n 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, f_j^1 can be calculated from [4] :

$$f_j^{\,1} = (2\pi)^{-1} \oint c_j \cos\alpha d\beta \quad (16)$$

where α is the angle between normal N to element of area d A_1 and the normal N' at the plan determined of element dl belonging to the contour curve C_j and the centre of area d A_1 ;

$d\beta$ - the elementary plan angle under which is "seen" the element dl, from center of d A_1 . For small values the angular factor, in practical calculation with a first approximation can be admitted as constant the temperature T_j on the surface S_j.

To clarify more easy the practical variation possibilities of decreased apparent emissivity εad , first of all, will be presented two hypothetic cases. In the first case a) admitting $T_j = T_w$ = const., $F_c = 0$, $F_m = 0$, taking into account that from (10) $F_e = \sigma T_w^4$, and replacing (15) in (11) results:

$$\varepsilon_{ad} = \varepsilon + (1 - \varepsilon)(1 - f_j^1) \quad (17)$$

Relation (17) establish the variation of εad 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 (10) results that the flux of thermal energy radiates for wall surface unity, is:

$$F_b = \varepsilon\ \sigma T_w^4 + (1 - \varepsilon)\sigma T_w^4 = \sigma T_w^4 \quad (18)$$

Using first equation of (13):

$$\varepsilon = (F_{bd} - F_{ed})(\sigma T_w^4 - F_{ed})^{-1} \quad (19)$$

Can be measured, F_{bd} with a total radiation pyrometer and T_w using a thermocouple, but it is difficult to calculate with precision F_{ed}. In a first approximation, assimilating with black bodies the inner furnace surfaces which radiate thermal energy, F_{ed} can be calculated from :

$$F_{ed} = \sum_{n=2}^{n=k} \sigma T_n^4 f_n^1 \quad (20)$$

The surface Sn with area An which thermal energy radiates towards the surface S_1 , can be divided in k small areas an for which can be admitted a constant mean temperature T_n . For relative great values of f_j^1, F_{ed} is not necessary to be determined with precision , because when $f_j^1 \to 1$, $F_{ed} \to 0$. For giving in evidence the practical applying possibilities of presented method, many years ago, were effected experiments on the experimental furnaces at the SEGFT and IERAB. The main used experimental furnace at SEGFT having interior appreciative dimensions 7 m × 1.5 m × 1.5 m, presented an important decrease of inner wall temperature, towards of furnace exit for combustion gases. The furnace inner wall, was made of silimanit refractory bricks with a great content of alumina. The cooled tube as screen with a length of 3 m, had the exterior diameter of 70 mm and interior diameter of 50 mm. This tube (having the symmetry axe perpendicular on flame symmetry axe) was introduced into the furnace in a high temperature zone, at approximate 1800 mm distance of furnace burner. F_{bd} was measured with a total radiation pyrometer type IFRF (having an adequate calibration) with recording apparatus. Also a recorder has the thermocouple Pt. – Pt. Rh. used to measure the temperature T_w. The total radiation pyrometers were calibrated with and without cooled tube. Experimental researches were effected in the following conditions:

- with the cooled tube introduction into furnace volume at different distances y (Fig.2), for obtaining different values f_j^1 ;
- without the cooled tube introduction , to determine the apparent emissivity ε_a ;
- experiments without and with natural gas burner operation.

For precision of measurement, before the two cycles of experiments, the calibration of total radiation pyrometers was performed. Will be presented same characteristic results obtained in the two cycles of experiments (without and with combustion: for $f_j1 = 0$, T_w = 1303 K, ε_a = 0.965; for f_j1 = 0.16, T_w = 1250 K, ε_{ad} = 0.75 (0.79);for f_j^1 = 0.48,T_w = 1173 K, ε_{ad} = 0.72 (0.74); and for f_j1 = 0.80, T_w = 1089 K, ε_{ad} = 0.70 (0.71). The results written in round brackets are referring to ε_{ad} when the furnace gas natural burner was in function producing combustion gases with low emissivity. The value for each experimental point represents the rounded arithmetical mean for three measurements of ε_{ad}. The combustion gases presence in this cycle of experiments for f_j^1 = 0.80 increase ε_{ad} with smaller as 2%. Numerous experimental results for this large value of f_j1, mark that using new refractory bricks inner furnace, are obtained near values for surface wall total emissivity ε, as by electric heating of the wall brick is realized in laboratory experiments. Indeed in this last case, the silimanit brick radiates only the flux $\varepsilon\ \sigma T_w^4$ because Fe don't exist. Will be presented two characteristic experimental results obtained in laboratory and on furnace (written in round brackets), as follows: for f_j^1 = 0.80 (y = 2,61 cm), T_w = 1089 K, ε = 0.68 (0.70); and with T_w = 1303 K, ε = 0.63 (0.64). According to the numerous effected experiments, by decrease with approximate 3% of experimental obtained value εad for f_j^1 = 0.8 , using (13) give practice the real value of ε.

5. Thermographic research-testing method of miniaturized combustion

Applying the proposed method for ε_f determination at combustion of inferior fuel oils, in many experiments with heavy fuel oil are obtained total emissivity even 0.9 in diffusive combustion front of flame, due the abundant release of soot particles. Especially, this released soot, is the consequence of necessary combustion air miss in mentioned zone, and relative great content of carbon in fuel oil. But the burner by his injector produces the atomized oil droplets which ignite and burn giving combustion gases and soot. For this reason is special useful to research the miniaturized combustion for a single droplet. In consequence, the proposed research-testing method use the emissivity near unity for some characteristics points of a single fuel oil droplet flame by miniaturized diffusion combustion in operation conditions. Thus was emitted a great quantity of soot particles. This thrermographyc method also determines the combustion infrared thermography (ITH) development, founded on infrared thermogram (IRT) analyze of the burning fuel oil droplet flame. For applying the method is realized an un-cooled mini-furnace so-called diffusive combustion simulator [4], which permits the droplet combustion at a inner furnace temperature relative low, near the environmental air temperature T_e. Was yielded an abundant soot quantity, giving emissivity near of black body in different characteristic points of flame. For an experimented fuel oil droplet is measured the initial mean diameter d0 and his temperature T0. Was used an infrared camera IRC operating in the wavelenght infrared band Δλ = 3.4 – 5 μm, in order to obtain the IRT with thermal images. This IRT give the fields of apparent temperatures T_f of burning droplet at a real - time τ of the diffusive combustion process development. The values of T_f are function especially of: diameter d0, time τ and fuel oil properties. From the IRT initial analyze has resulted that for a given fuel oil the apparent temperatures are greater when d0 is greater and the τ time is nearer of value corresponding to the droplet flame maximum volume DFMV which was a reference value. For example, in Fig. 3 is presented beside, the flame detail with schematically representation at DFMV, of a superior gas oil GO droplet and the IR thermogram for the some fuel droplet (with d_0 = 1.7mm, T_e = T_0 = 295K) at DFMV . Using an appropriate IRC, the mean real temperature T_r corresponding to T_f, can significant characterize the fuel oil combustion process development. T_r temperature results when is know the emissivity εe in direction of normal incidence of xAy plane of Fig. 3, within the Δλ band. The point A represents the symetry center for liquid of burning droplet. Theory and practical experience show that in the majority of IRT points regarding the droplet flame, the emissivity εe is variable in function of numerous parameters which are presented. But these parameters are variable in the time of droplet burning. For this reason it is impossible to determine with precision, even by special methods, the variable value of εe. To obtain a small variation of εe in normal direction at x'x axe for the interval of burning droplet , with a reduce difference between T_f and T_r , it is necessary to burn in ambient air, relative large droplet releasing abundant soot by diffusion combustion and the environment giving a very low emissivity. Thus, the released fine particles of soot are rich and spectral emissivity ε(λ) slowly varies with wavelength λ, for some droplet flame zones, similar as for solid objects. In normal direction to xAy plane surface for the point A, where the released soot is larger, and the liquid droplet represents a background screen, the emissivity εe for an inferior fuel oil diffuse combustion has a great value near to black body emissivity. Apparent temperature TA for the point A is determined according to radiation energy of the referent flame layer thickness with abundant soot, and the burning fuel droplet liquid. In direction from droplet towards the droplet flame ended, the emissivity decreases especially due soot concentration and gaseous layer thickness decrease, but increases with temperature diminution. Such as, if we admit εe = 0.9, for which in IRT the temperature T_f is determined, but in reality εe is decreased with 0.10 in the considered point of flame, for this point results T_r = 1.0298 T_f. Thus, is resulted a relative small error on T_r temperature value determination. The influence of εe variation is much less important as temperature T_f variation in the validity field of Stephan Boltzmann's law, which it is admitted in a first approximation. For this reason, the analysis and comparison of T_f temperature fields from IRT, according to εe estimation give a valuable qualitative information on diffusion combustion process development, very dependent on T_r temperature. Also by use of the proposed method, analyzing IRT obtained by experiments performed in the same initial combustion conditions of droplets, for a tested fuel oil, and a standard fuel oil, new scientific criteria of the fuel combustion quality determination, are established. Thus to characterize and compare, the gas oil combustion quality at DFMV, where defined the specific criteria: T_{mx} – average of flame apparent temperatures in normal direction on the xx' axe and xAx' plane; T_t – average of total flame apparent temperatures in normal direction on xAy plane obtained by conversion of total radiation energy in Δλ band, from the burning droplet ; T_A – apparent temperature in normal direction on the xAy plane, for the point A. Also can be established secondary qualitative selection criteria in connection with the shape of curves for apparent temperature corresponding to xx' and yy' axes, namely T_{fx}=f(x) and T_{fy}=F(y), as well as for apparent temperature of filament extremity suspending the burning droplet. Experimental researches and testings were effected using the presented combustion simulator, and different types of gas oils, intermediate and heavy fuel oils, in the same initial conditions, characterized by: temperatures T_e and T_0; natural draft for combustion air feeding; cylindrical un-cooled simulator and and especial initial mean diameter of droplet d_0 = 2 mm . During the majority of fuel oil testing, variation of above mentioned criteria, for ε_e = 1, was: T_{mx} = 295 - 357°C; T_A = 387 - 447°C and T_t =263 -

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293°C. The conclusion of numerous experiments and according theory result that referring to the gas oil quality for combustion intensification, is better when the specific criteria T_{mx}, T_A and T_t have larger values.

6. Conclusions

In the first part, was presented a research-testing method for determination of industrial fuel oil combustion flame total emissivity, founded on a deduced formula of ε_f calculation, $\varepsilon_f = 1 - (F_t - F_f)(\sigma T_w^4 - A)^{-1}$ where $A = (F_c - F_m)(1 - \varepsilon)\varepsilon^{-1}$. With A = 0, from (7), results as a particular case (8), so called Schmidt formula [2]. Thus, the formula (8) used in numerous cases, with a very small error it is possible to be applied only when $A \rightarrow 0$. For experiments, long ago was conceived and realized an experimental furnace with annexed equipments at IERAB [3].

During the furnace operation, due the wear and tear as consequence of interaction between combustion products and refractory bricks as well due the eventual local superheating, the total emissivity ε especially after a long time, sensible changes. These considerations recommend to give a method for establishment of ε real value especially based on the development of specific theoretic fundaments. By relation (10) was defined the so-called apparent total emissivity of furnace inner wall surface εa. When is decreased the heat flux F_e till F_{ed} value, also decrease the heat flux F_b till F_{bd} . Thus ε_a decreases at the value $\varepsilon ad = F_{bd} (\sigma T_w^4)^{-1}$which can be calculated from (13) measuring F_{bd} with a total radiation pyrometer and Tw using a thermocouple. With condition $F_{ed} \rightarrow 0$, results $\varepsilon ad \rightarrow \varepsilon$. The Fe decreasing is practical realized with the add of a long but small in diameter steel tub, cooled with water circulation between the tub double steel walls. This tub, in experiments is moved at different distances y, in (Δ) direction normal at the heat receiving surface S_1. In general from (19) . results ε when are measured F_{bd} and T_w , but laborious F_{ed} is calculated. According to the numerous effected experiments, by decrease with approximate 3% of experimental obtained value εad for $f_j^1 = 0.8$, give practical the real value of ε, when in furnace are combustion gases with low flame emissivity (especial CO_2 and H_2O radiate energy). For great values of ε_f, however the supplementary increase of the ε_{ad}, but for relative great fj1 this growth is small, because especial Fed is very small in comparison with σT_w^4. In experiments without furnace burner operation for similar conditions, the positive systematic error on ε determination was approximate 2%, and practically can be eliminate. On principle, from the proposed method results as a particular case (when $F_{ed} = 0$ and $f_j^1 = 1$), the laboratory usual method for ε determination. The proposed thermographic research-testing method uses the emissivity near of unity, for different points of a fuel oil droplet flame, by miniaturized diffusion combustion in operation conditions giving a great quantity of soot particles. It is realized an un-cooled mini-furnace as combustion simulator, which permits the droplet diffusion combustion at a inner furnace temperature near the environmental air temperature T_e. Was used an infrared camera IRC, in order to obtain the IRT with thermal images. This IRT give the fields of apparent temperatures T_f of burning droplet at a real - time τ of the combustion process development. From the IRT initial analyze has resulted that for a given fuel oil the apparent temperatures are greater when the initial droplet diameter d_0 is greater and the τ time is nearer of value corresponding to the droplet flame maximum volume DFMV which was the reference value. Using an appropiate IRC, the mean real temperature T_r corresponding to T_f, can significant characterize the fuel oil combustion process development. Indeed, for great values εe even an important variation of this factor, have not a great influence on T_f variation. Experimental researches were effected using different types of fuel oils. The conclusion of numerous experiments effected in above mentioned conditions and according theory, result that referring to the gas oil quality for combustion intensification, is better when the new criteria T_{mx}, T_A and T_t have larger values

NOMENCLATURE AND ABBREVIATIONS

$F = \sigma T^4$ - radiation thermal flux of black body,[W/cm^2] at temperature T [K], where $\sigma = 10^{-8}$ C; C = 5.67 . 10^{-4} [W/cm^2 K^4] is Stephan – Boltzman constant for black body; S_1 – reference surface, is considered a small surface of furnace inner refractory wall which radiates in direction (Δ), the received thermal energy, [cm2]; ε_f ,ε_e - combustion flame total emissivity and droplet flame emissivity in infrared band of Δλ = 3.4 – 5 μm ; ε - total emissivity of furnace inner refractory wall surface; ε_a - apparent total emissivity of furnace inner wall surface; y - distance till the surface S_1; λ – wavelength, [μm]; τ - time variation, [s]; f_n^1 - angular factor of surface Sn with area An related to the area A_1 , with variation of n = 2 – i ; T_w - wall surface temperature behind the flame [K]; F_e – total thermal radiation incident flux to the receiving 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 miss the flame, [W/cm^2]; F_c – thermal flux yielded, to the receiving unit surface, by convection of heat transfer due hoot combustion gases flow, [W/cm^2]; F_m - thermal flux transmitted by conductibility in furnace wall, towards the ambient medium, from the receiving unit surface, [W/cm^2]; F_f - total normal thermal radiation flux only from the flame, in direction (Δ) normal to the flame symmetry axe, [W/cm2]; F_t - total thermal radiation flux in direction (Δ), summating F_f with the total normal thermal radiation flux from the unity of S_1 surface, [W/cm^2]; T_e, T_0 – environmental air and fuel oil temperatures, [°C or K]; T_a, T_r – apparent and real temperatures of flame, [°C or K]; T_{mx}, T_t – average of apparent temperatures, on xx' axe and total flame, [°C or K]; d_0 – initial mean diameter of droplet, [mm]; IFRF - International Flame Research Foundation (IJmuiden); SEGFT – Station d'Essais de Gas de France, Toulouse; IERAB - Institute of Energetics from Rumanian Academy, Bucharest ; DFMV – droplet flame maximum volume; IRC – infrared camera; ITH – infrared thermography; FIT – flame infrared thermogram.

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KEYWORDS: Flame total emissivity, wall total emissivity, fuel oil combustion, IR-thermography, furnaces, research-testing.

REFERENCES

[1] V.V. Ghiea (Ghia), C. Dumitraşcu, Method using infrared thermography for combustion testing of fuel oil., Rev.Roum.Sci.Téchn. Électrotechn. et Énerg., 47, nr.2, 2002, p. 219-228, Bucharest.

[2] J. Cheddaille, Y. Brand, Mesure du transfert de chaleur. Fondation de Recherches Internationales sur les Flammes-Ijmuiden, Doc. No. 20/a/40, 1968.

[3] V. V. Ghiea (Ghia), D. Grecov, I. Spahiu, M. Biener, Research furnace for the combustion and flame radiation of liquid or natural gases fuels, Studies and researches of energetics, Seria B, Tome XII, 1962, p. 255 – 260, Bucharest.

[4] A. Gauffe, Transmission de la chaleur par rayonnement. Ed. Eyrolles. Gauthier Villars, Paris, 1968.

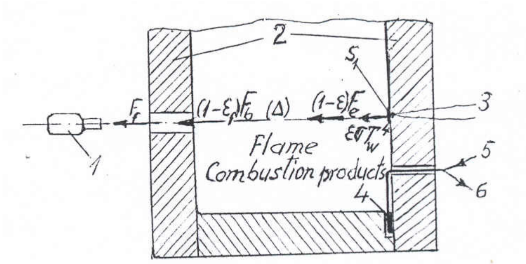

Fig. 1. *Schematic representation of the furnace working space with a cooled screen mobile by round and the total radiation measurement. 1 – Special total radiation pyrometer ; 2 – Furnace incandescent walls ; 3 – Pt.-Pt. Rh. thermocouple measuring the temperature T_w of the wall surface S_1; 4 – cooled mobile screen; 5 – cold water entrance; 6 – water evacuation.*

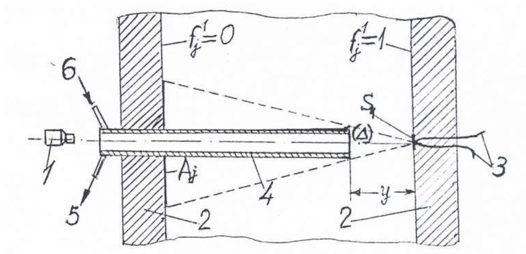

Fig. 2. *Schematic representation of the furnace working space with a tubular cooled screen mobile by translation. 1 – Special total radiation pyrometer ; 2 – Furnace incandescent walls; 3 – Pt.-Pt. Rh. thermocouple measuring the temperature of S_1 wall surface; 4 – cooled tubular screen having mobility by translation movement, with d exterior diameter; 5 – water evacuation; 6 – cold water entrance*

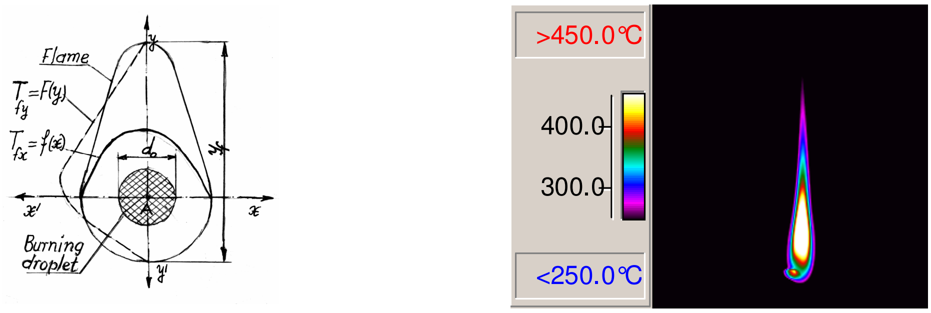

Fig. 3. *Flame detail with schematically representation at DFMV, of a superior gas oil GO droplet and IR thermogram for the some fuel droplet (d_0 = 1.7mm, T_e = T_0 = 295K) at DFMV*