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Calculation of the furnace exit gas temperature of stoker fired boilers

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Abstract In the paper the methodology of furnace exit gas temperature calculations by using well known normative standard method CKTI is presented. There are shown changes in methodology approach for three editions of it and in additional developments. Furnace exit gas temperature for two stoker grate boilers is calculated. By using described methods, it was possible to determine their effectiveness by comparing with measurements. Knowledge of the furnace exit gas temperature allows to define the division into irradiated and convection surfaces, which has an impact on the design features of the boiler as well as its dimensions and weight.

Keywords: FEGT; CKTI; Grate boilers calculations; Furnace

Nomenclature

A	–	ash content in fuel, wt %
a	–	equivalent emissivity, –
as	–	fraction of component, %
B	–	fuel flow, kgs^{-1}
Bo	–	Boltzmann number
Bu	–	Buger number
cp	–	specific heat capacity at constant pressure, Jkg^{-1}K^{-1}
d	–	average diameter of particle, μm
C^r	–	carbon content in fuel, wt. %
F	–	area, surface, m^2

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FEGT	–	furnace exit gas temperature, K
G	–	flue gas mass of unit fuel, kg^{-1}
k	–	coefficient of radiant absorption, $\text{m}^{-1}\text{Pa}^{-1}$
L	–	heat loss of boiler, %
LHV	–	lower heat value of fuel, MJkg^{-1}
M	–	parameter characterizing the location of the maximum flame temperature in the furnace chamber, –
p	–	pressure inside in the furnace chamber, MPa
Q	–	heat output, kW
q	–	heat flux, Wm^{-2}
r	–	fraction of element
r_H	–	volume fraction of water vapor in flue gas
s	–	effective radiation layer thickness, m
STP	–	standard temperature and pressure, normal conditions for $t = 0°\text{C}$ and $p = 0.101$ MPa
T	–	temperature, K
V	–	volumetric flow per kg of fuel, m^3kg^{-1}; volume, m^3
V_{air}^t	–	theoretical air required for combustion, $\text{Nm}^3\text{kg}^{-1}$
V_{N2}	–	volume of nitrogen related to the fuel burned, $\text{Nm}^3\text{kg}^{-1}$
V_{RO2}	–	volume of triatomic gases related to the fuel burned, $\text{Nm}^3\text{kg}^{-1}$
vol	–	volatiles content in fuel, %
x_T	–	relative position of the flame in the furnace chamber, –

Greek symbols

Π	–	furnace parameter
λ	–	excess air ratio
μ	–	dimensionless concentration
θ	–	dimensionless temperature
ξ	–	fouling factor
ρ	–	effect of radiation of the fuel layer on the grate
σ	–	Stefan–Boltzmann constant, $\text{Wm}^{-2}\text{K}^{-4}$
φ	–	heat preservation coefficient
χ	–	coke particles radiation coefficients
ψ	–	thermal efficiency coefficient

Subscripts

ad	–	adiabatic
avg	–	average
f	–	furnace
fa	–	fly ash
fc	–	coal fuel
FG	–	flue gas
fl	–	flame
g	–	gas
gr	–	grate
r	–	radiation
w	–	wall

1 Introduction

The thermal calculations of grate boilers do not differ from those for pulverized coal or fluidized bed boilers. They have a similar flow diagram of calculations and algorithms (Fig. 1). The principal differences are based on

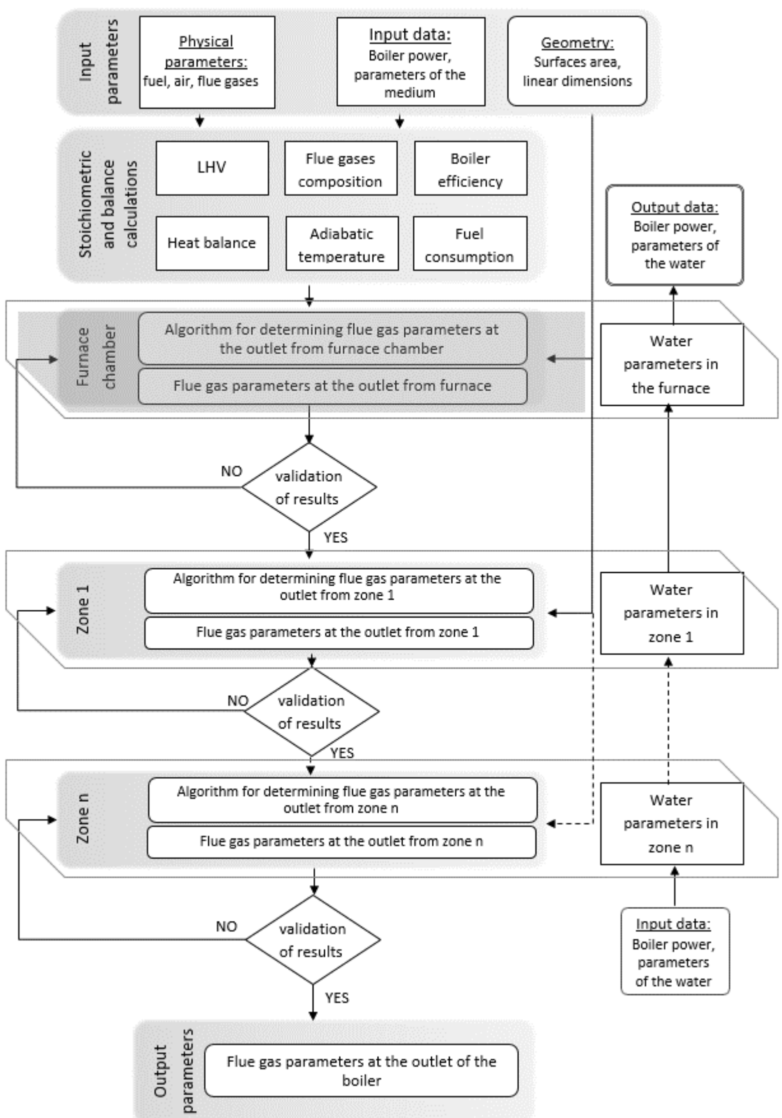

Figure 1: A simplified diagram of thermal calculations for a water boiler.

a different philosophy of combustion and the resulting mass balance of fuel and ash in the furnace. The calculation methods of combustion chambers differ mainly due to the different position of the flame inside of it and the temperature distribution along the height of the combustion chamber.

Thermal calculations of the water boiler are carried out in the following order: determination of input parameters, stoichiometric and balance calculations, calculations of the combustion chamber and thermal calculations in the convective sections outside the combustion chamber.

Thermal calculations methods are based on the principles of the heat transfer theory and on coefficients determined empirically and experimentally during the operation of similar boilers. Calculations determine the fuel consumption and air requirement, pressures, temperatures, velocity, and composition of the exhaust gases along the flow through the boiler. This allows for subsequent calculations, such as flow resistance and walls surface temperatures.

The purpose of the combustion chamber calculations is, among others, to determine the temperature at its outlet and at the same time at the inlet to the convective section or cooling zone. This temperature should meet two conditions:

- should be sufficient for the combustion of solid particles and devolatilization products in the flue gas (800–1100°C),
- should be below the softening point of the ash due to necessity to prevent contamination of the heating surfaces by slag deposits.

2 Furnace exit gas temperature calculating methodology

The methodology of boilers calculation was evolving with the development of research and better understanding of the physical and chemical processes occurring during fuel combustion in the boiler, as well as the heat exchange between the flame and the heating surfaces inside the combustion chamber. The most popular methods are those based on the work of Central Boiler urbine Institute – CKTI (Russian: Центральный Котлотурбинный Институт, ЦКТИ). The first methodology study was published in 1937 and revised in 1945 under the name of "Thermal calculations" (Russian: "Тепловой расчет"). In 1951 All-Russia Thermal Engineering Institute – VTI (Russian: Всероссийский Теплотехнический Институт, ВТИ) published the "Thermal Calculation of Boiler Units. Normative Method" [1].

Initially, it was based on the analysis of dependencies describing the heat transfer process based on the theory of similarity. It allowed to omit unknown parameters as a function of dimensionless criteria describing the combustion process. In their works [1, 2], Authors relied on the experimental results obtained from the tests of furnace chambers of steam boilers. In the following years, efforts were made to develop more practical methods of thermal calculations for furnace chambers based on the results of measurements of existing units and laboratory tests. In 1973, the second edition of the normative method was prepared under the editorship of N.V. Kuznetsov [2]. In 1984 (English version 1988) A.G. Blokh presented his proprietary method of calculating boilers as an extension of the CKTI method, but only large steam boilers were included [3,4]. In 1998, the third edition of the normative method [5] was published, introducing numerous changes in the approach to calculations. In 2003, guidelines for the calculation of industrial boilers were issued in China [6], which were an extension of the CKTI method (1973), the methodology was also presented in 2016 by Y. Zhang [7] with some exceptions.

In 2006 B.Ya. Kamenetskii proposed some changes where a rational parameter of heat transfer in the case of grate boilers is the temperature of the fuel layer, not the gas flame as described in the normative method [8,9]. The final formula for the furnace exit gas temperature of the method is

$$\mathrm{FEGT} = \frac{T_{ad}}{\dfrac{M}{\left(\dfrac{\mathrm{Bo}}{a_f}\right)^{0.6}} + 1}, \tag{1}$$

where M is a constant value for most furnaces, a_f is the degree of blackness of the furnace (equivalent emissivity), and Boltzmann's criterion is

$$\mathrm{Bo} = \frac{\varphi B_{fc} \sum V_{FG} cp_{avg}}{\sigma \xi F_r T_{ad}^3}. \tag{2}$$

Both values are described later in the text.

2.1 CKTI method 1st (1951) and 2nd edition (1973)

The input parameters of the method are geometric data of the furnace chamber and grate, physical parameters of the fuel and oxidizer taking part in the combustion process in the furnace chamber, as well as a number of empirical factors. Intermediate calculations of the furnace and heat exchange are used to determine the flue gas temperature. The calculations are then repeated until the required accuracy of the iterative calculus is achieved. The simplified algorithm is depicted in Fig. 2.

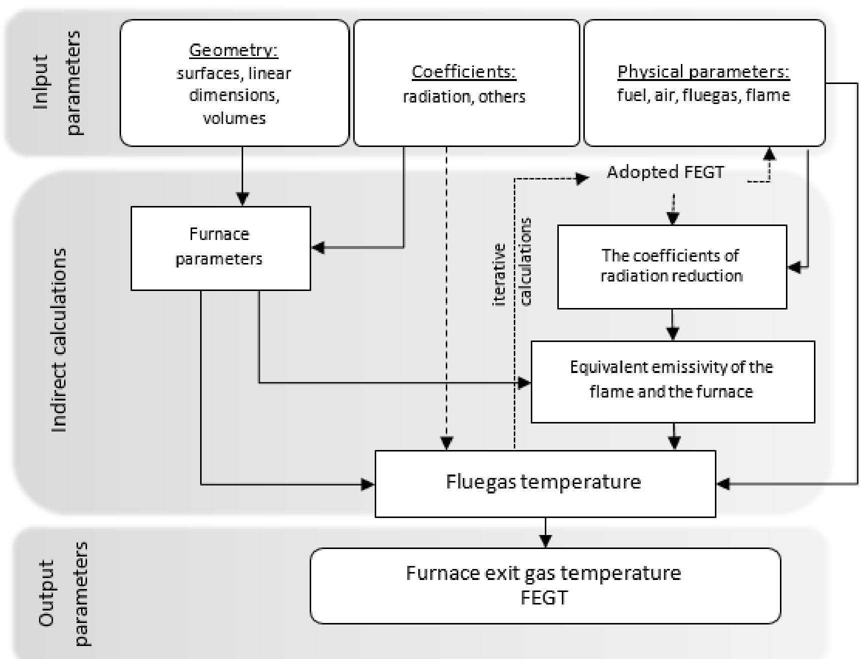

Figure 2: A simplified algorithm of CKTI method 1st and 2nd edition.

The method is based on solving Eq. (1) under the conditions Bo $< 10a_f$ and $\dfrac{\text{FEGT}}{T_{ad}} < 0.9$ for the first edition, while for the second edition there is only the criterion $\dfrac{\text{FEGT}}{T_{ad}} < 0.9$.

After substituting the Boltzmann criterion and transformations, one can obtain the dependencies for determining the furnace exit gas temperature at the outlet from the furnace chamber

$$\text{FEGT} = \frac{T_{ad}}{M\left(\dfrac{\sigma a_f \psi_{avg} F_r T_{ad}^3}{\varphi B_{fc}\sum V_{FG} c p_{avg}}\right)^{0.6} + 1} . \tag{3}$$

Determining the equivalent emissivity of the furnace (the so-called degree of blackness) for grate boilers is determined by the formula:

$$a_f = \frac{0.82\left[a_{fl} + (1 - a_{fl})\rho\psi'\right]}{1 - (1 - \psi'\psi_{avg})\,(1 - a_{fl})\,(1 - \rho\psi')} \qquad \text{1st edition,} \tag{4}$$

$$a_f = \frac{a_{fl} + (1 - a_{fl})\rho}{1 - (1 - \psi_{avg})\,(1 - a_{fl})\,(1 - \rho)} \qquad \text{2nd edition.} \tag{5}$$

The thermal efficiency coefficient of the screens

$$\psi = x\xi \tag{6}$$

is the multiplication of the shape factor depending on the geometry x (equal to 1 for a membrane walls) and the contamination coefficient of the surfaces irradiated in the furnace chamber (values presented in Table 1). The fouling factor (ξ) of the heating surface takes into account the reduction in the amount of heat transferred as a result of contamination or covering the screens with a layer of insulation.

Table 1: Values of the fouling factor (ξ) of irradiated surfaces.

Edition	Walls made of standard tubes, membrane walls	Walls covered with refractory mass	Walls covered with fireclay bricks
First	0.7	0.2	0.1
Second	0.6		

If the furnace walls consist of tubes of different diameters or covered with lining, the average thermal efficiency coefficient of the screens is determined as

$$\psi_{avg} = \frac{\sum \psi_i F_{w,i}}{F_w} \tag{7}$$

taking into account the differences in the contamination coefficient of the irradiated surfaces.

The φ factor taking into account boiler radiation losses in the combustion chamber (L_r) (according to the EN12952–15 standard [10]), can be determined using

$$\varphi = \left(1 - \frac{L_r}{100}\right). \tag{8}$$

The degree of screening of the furnace for grate furnaces is a changed form of formula (8) for the thermal efficiency of the irradiated surfaces (F_r)

$$\psi' = \frac{F_r}{F_w - F_{gr}}\,. \tag{9}$$

The parameter ρ determines the effect of the radiation of the fuel layer on the grate on heat exchange in the furnace chamber and is defined as the ratio of the grate's active surface (F_{gr}) to the total irradiated area in the furnace chamber:

$$\rho = \frac{F_{gr}}{F_r}\,. \tag{10}$$

The flame equivalent emissivity (a_{fl}) depends on the coefficients that weaken its radiation (k), the thickness of the radiating layer (s) and the pressure inside the furnace chamber ($p = 0.101$ MPa (1 atm) is assumed). Coefficient depicted in both editions:

$$a_{fl} = \beta\left(1 - e^{-kps}\right) \qquad \text{1st edition,} \tag{11}$$

$$a_{fl} = 1 - e^{-kps} \qquad \text{2nd edition,} \tag{12}$$

in which

$$s = 3.6\frac{V_f}{F_w}, \tag{13}$$

$$k = k_g r_p + k_{fa}\mu_{fa} \qquad \text{1st edition,} \tag{14}$$

$$k = k_g r_p + k_{fa}\mu_{fa} + k_{coke}\chi_1\chi_2 \qquad \text{2nd edition.} \tag{15}$$

Here $k_g r_p$ is the factor of attenuation of radiation with triatomic gases with a fraction r_p, $k_{fa}\mu_{fa}$ is the factor of attenuation of the radiation with ash particles in concentration μ_{ash}, $k_{coke}\chi_1\chi_2$ is the coefficient of attenuation of coke particles radiation, depending on the type of fuel χ_1 and its combustion χ_2: $\chi_1 = 0.5$ – fuels with high volatile matter content, $\chi_2 = 0.03$ – grate furnaces, $k_{coke} = 1$. The β factor for solid fuel and luminous flame is 0.65. The individual components are calculated using the following formulae:

$$k_g = \left(\frac{0.8 + 1.6r_H}{\sqrt{p_p s}}\right)\left(-0.38\frac{\text{FEGT}}{1000}\right) \qquad \text{1st edition,} \tag{16}$$

$$k_g = \left(\frac{0.78 + 1.6r_H}{\sqrt{p_p s}} - 0.1\right)\left(1 - 0.37\frac{\text{FEGT}}{1000}\right) \qquad \text{2nd edition,} \tag{17}$$

$$p_p = r_p p, \tag{18}$$

$$k_{fa} = 7\sqrt[3]{\frac{1}{d_{fa}^2\,\text{FEGT}^2}} \qquad \text{1st edition,} \tag{19}$$

$$k_{fa} = \frac{4300\rho_{FG}}{\sqrt[3]{\text{FEGT}^2 d_{fa}^2}} \qquad \text{2nd edition,} \tag{20}$$

where is assumed the flue gas density $\rho_{FG} = 1.3$ kg/m^3 and the average diameter of the ash particle $d_{fa} = 20$ μm for grate furnaces.

The ash concentration coefficient in the flue gas depends on the fraction of fly ash (a_{fa}) and ash content in the fuel and the flue gas mass (G_{FG}):

$$\mu_{fa} = \frac{10A^r a_{fa}}{V_{FG}} \qquad \text{1st edition,} \tag{21}$$

$$\mu_{fa} = \frac{A^r a_{fa}}{100G_{FG}} \qquad \text{2nd edition,} \tag{22}$$

where

$$G_{FG} = 1 - \frac{A^r}{100} + 1.306\lambda V^t_{air}\,. \tag{23}$$

The M parameter is a function of the relative coordinate of the maximum temperature in the furnace and the type of fuel burned. In the first edition $M = 0.445$ was assumed, regardless of the type of furnace or fuel. The second edition introduced dependencies for the determination of this parameter mainly depending on the fuel burned, but also on the relative position of the flame in the furnace chamber (x_T).

$$M = 0.59 - 0.5x_T\,. \tag{24}$$

According to the standards, the value of x_T for grate boilers is 0.14, therefore the constant value of the M factor for grate boilers is 0.52.

2.2 Development of the 2nd edition of CKTI method by Blokh

The input parameters of the method are geometric data of the furnace chamber and grate, physical parameters of the media taking part in the combustion process in the furnace chamber, as well as a number of empirical factors. Intermediate calculations and heat exchange are used to determine the flue gas temperature. The calculations are then repeated until the required accuracy of the iterative calculus is achieved. The simplified algorithm is depicted in Fig. 3.

The calculation methodology derives indirectly from the normative method, with the general equations and relationships having a different structure and form. The expressions for ψ_{avg} and M have the same values.

The main difference is the introduction of the furnace parameter as

$$\Pi = \frac{1}{a_f}\frac{1}{\psi_{avg}}\mathrm{Bo}\,, \tag{25}$$

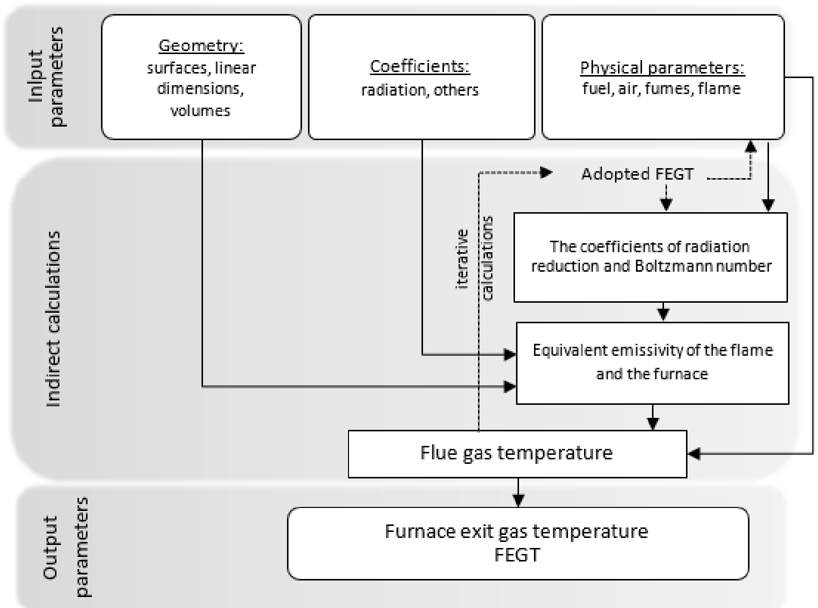

Figure 3: A simplified algorithm of CKTI method by Blokh.

The furnace emissivity has the form

$$a_f = \frac{a_{fl}}{a_{fl} + (1 - a_{fl})\psi_{avg}} \, . \tag{26}$$

The form of the temperature simplex proposed by the author is as follows:

$$\frac{T_{ad}}{\mathrm{FEGT}} - 1 = \frac{M}{\Pi^{0.6}} \, . \tag{27}$$

Formula (27) (similarly to the formulae from normative methods) allows to determine the temperature at the outlet from the furnace chamber without taking into account the influence of heat exchange on the temperature field in the chamber. Depending on the dimensions of the chamber, i.e. height (H) and depth (D), this effect varies. For very small D/H values (infinitely high chambers) the temperature field along the height is the least uniform with relatively little heterogeneity in the gas flow across the chamber cross-section. As the D/H ratio increases, the decrease in temperature field heterogeneity along the height of the chamber goes hand in hand with an increase in the exhaust gas temperature heterogeneity in the cross–section. This leads to a reduction in heat transfer in the furnace.

The increase in the thickness of the radiating layers causes an increase in the flame temperature heterogeneity in the cross-section. Radiation from the high-temperature flame zone in the boiler axis is blocked by colder layers of flue gas. This is so-called thermal radiation suppression and leads to a reduction in heat transfer to the boiler walls and ultimately to an increase in the exhaust gas temperature at the outlet of the combustion chamber.

I.E. Dubovsky as a result of his research on this issue, proposed a changed form of the dependence on temperature at the outlet from the furnace chamber [3, 4]

$$\frac{T_{ad}}{\mathrm{FEGT}} - 1 = 0.96M\left(\frac{T_0}{T_{ad}}\right)^2\left(a_f\psi_{avg}\frac{1}{\mathrm{Bo}}\right)^{0.6}. \tag{28}$$

The formula is valid for the reference temperature $T_0 = 1530$ K.

By introducing a modified formula for the furnace parameter

$$\Pi_* = 0.1268M^{\frac{5}{3}}\left(\frac{T_{ad}}{1000}\right)^2\left(\frac{1}{a_f}\frac{1}{\psi_{avg}}\mathrm{Bo}\right) = 0.1268M^{\frac{5}{3}}\left(\frac{T_{ad}}{1000}\right)^2\Pi\,. \tag{29}$$

Equation (28) can be represented by the following relation, which is more practical in the calculations

$$\theta = \frac{\mathrm{FEGT}}{T_{ad}} = 1 - \frac{0.44}{\Pi_*^{0.6}}\,. \tag{30}$$

This formula can also be written as

$$\frac{T_{ad}}{\mathrm{FEGT}} - 1 = \frac{0.96}{\left(\frac{\mathrm{FEGT}}{T_0}\right)\left(\frac{T_{ad}}{T_0}\right)^{0.2}}\frac{M}{\Pi^{0.6}}\,, \tag{31}$$

which is analogous to the basic formula (27). The relationship takes into account the influence of heat exchange in the furnace chamber on the temperature field determined by the function:

$$f(\mathrm{FEGT}, T_{ad}) = \frac{0.96}{\left(\frac{\mathrm{FEGT}}{T_0}\right)\left(\frac{T_{ad}}{T_0}\right)^{0.2}}\,. \tag{32}$$

A further simplification of Eq. (31) is the introduction of a more convenient form of Boltzmann number that takes into account the heat load on the walls:

$$\mathrm{Bo}_f = \frac{q_f}{\sigma T_{ad}^4}\,, \tag{33}$$

where

$$q_f = \frac{B_{fc}\mathrm{LHV}}{F_r} \tag{34}$$

and eventually

$$\frac{\mathrm{FEGT}}{T_{ad}} = 1 - 0.85M\left(\frac{T_0}{T_{ad}}\right)^{1.2}\left(a_f\psi_{avg}\frac{1}{\mathrm{Bo}_f}\right)^{0.6}, \tag{35}$$

with $\mathrm{Bo}_f \cong 0.812\mathrm{Bo}$.

The optimal value of the heat load intensity of the walls to ensure proper boiler operation without slagging can be determined from the following equation:

$$q_f = a_f\psi_{avg}\left(\frac{T_{ad}}{100}\right)^2\left(\frac{M}{1-\Theta}\right)^{5/3}. \tag{36}$$

Equivalent emissivity of the flame (a_{fl}) has the same physical form as in the normative method, i.e. it depends on the coefficients that weaken its radiation k and the pressure inside the furnace chamber assumed as $p = 0.101$ MPa,

$$a_{fl} = 1 - e^{-k}, \tag{37}$$

in which

$$k = k_g r_p\sqrt{p_p s} + k_{fa} + k_{coke}\,, \tag{38}$$

$$k_g = \left(\frac{0.78 + 1.6r_H}{\sqrt{p_p s}} - 0.1\right)\left(1 - 0.37\frac{\mathrm{FEGT}}{1000}\right), \tag{39}$$

$$p_p = r_p p, \tag{40}$$

$$k_{fa} = \frac{4.1}{\sqrt[3]{\mathrm{FEGT}^2 d_{fa}^2}}\left[1 - \frac{b_2}{1 + 30\cdot 10^3\left(\mu_{fa}^{\mathrm{STP}}s\right)^{-2}}\right]\mu_{fa}^{\mathrm{STP}}s\,, \tag{41}$$

$$\mu_{fa}^{\mathrm{STP}} = \mu_{fa}\frac{\mathrm{FEGT}}{273}\frac{10A^r a_{fa}}{V_{FG}}\,. \tag{42}$$

$$k_{coke} = \frac{10}{\sqrt[3]{\mathrm{FEGT}^2 d_{fa}^2}}\mu_{coke}^{\mathrm{STP}}s, \tag{43}$$

$$\mu_{coke}^{STP} = \frac{55C^r\left(10 + L_r\right)}{\left(100 + vol\right)V_{FG}}\,, \tag{44}$$

with $b_2 = 0.6{-}0.7$. Here $k_g r_p$ is the factor of attenuation of radiation with three-atomic gases with the fraction r_p, k_{fa} is the factor of attenuation of radiation with ash particles with a concentration of μ_{fa}, k_{coke} is the coefficient of attenuation of radiation with coke particles with a concentration of μ_k, μ_{fa} is the mass concentration of ash, a_{fa} is the fly ash fraction, and d_{fa} is the average diameter of the ash particle (20 μm for grate furnaces). The STP index (standard temperature and pressure) means normal conditions for $T = 273.15$ K (0°C) and $p = 0.101$ MPa.

2.3 Development of the 2nd edition of normative method by Chinese researchers

In the standards and technical recommendations for Chinese boiler designers [6, 7], the provisions of the Russian normative method from 1973 are still used. In the case of grate boilers, the methodology was modified to assume that the flame temperature in the furnace chamber changes with its height but the algorithm is the same as in Fig. 2:

$$T_{fl} = \mathrm{FEGT}^n T_{ad}^{(1-n)}, \tag{45}$$

where n is the influence of combustion conditions on the temperature field in the furnace chamber. During the tests, the value of n for grate boilers was determined as 0.6.

Temperature at the outlet from the combustion chamber can be calculated using the following relationship:

$$\frac{\mathrm{FEGT}}{T_{ad}} = k\left[\mathrm{Bo}\left(\frac{1}{a_{fl}} + m\right)\right]^p, \tag{46}$$

The m factor takes into account the influence of temperature of the contamination layer on the screen wall on heat transfer. The value of m is constant in order to simplify the calculations and depends on the operating pressure of the boiler, values are presented in Table 2. The k and p coefficients are presented in Table 3.

Table 2: The value of the m coefficient.

Working pressure, MPa	0.7	1.0	1.25	1.6	2.5	3.8
m coefficient	0.13	0.14	0.15	0.16	0.18	0.21

For grate boilers, the equivalent emissivity of the furnace is determined by the formula

$$a_f = \frac{1}{\dfrac{1}{0.8} + x\left(\dfrac{1}{M} - 1\right)}, \tag{47}$$

Table 3: The value of the k and p coefficients.

$\mathrm{Bo}\left(\frac{1}{a_{fl}}+m\right)$	k	p
0.6–1.4	0.6711	0.2144
1.4–3.0	0.6755	0.1714

where

$$M = a_{fl} + \rho\,(1 - a_{fl})\,.$$

The parameter ρ, given by

$$\rho = \frac{F_{gr}}{F_w - F_{gr}}\,, \tag{48}$$

determines the effect of radiation of the fuel layer on the grate on heat exchange in the furnace chamber – similar to (10), but it is defined as the ratio of active surface of the grate to the total area of the furnace chamber excluding the grate.

The x factor is the ratio of the irradiated area to the total area of the furnace chamber, excluding the grate:

$$x = \frac{F_r}{F_w}\,. \tag{49}$$

Equivalent emissivity of the flame a_{fl} is determined as for the 2nd edition by Eq. (12).

Particular components have a different form depending on the references. According to [6], dependencies (17), (20), and (22) can be used. According to [7], these expressions have the following form:

$$k_g = \left(\frac{2.49 + 5.11 r_H}{\sqrt{p_p s}} - 1.02\right)\left(1 - 0.37\frac{\mathrm{FEGT}}{1000}\right), \tag{50}$$

$$k_{fa} = \frac{7752}{\sqrt[3]{\mathrm{FEGT}^2}}\,, \tag{51}$$

$$k_{coke}\chi_1\chi_2 = 0.153. \tag{52}$$

2.4 CKTI method 3rd edition (1998)

The input parameters of the method are geometric data of the furnace chamber and grate, physical parameters of the media taking part in the combustion process in the furnace chamber, as well as a number of empirical factors. Intermediate calculations of the furnace and heat exchange are used to determine the flue gas temperature. The calculations are then repeated until the required accuracy of the iterative calculus is achieved.

The methodology of calculating the temperature at the outlet from the furnace chamber does not differ significantly from that presented in the previous edition of the standard from year 1973. It is still based on a simplex of temperatures and appropriately selected coefficients and parameters. The simplified algorithm is depicted in Fig. 4. An additional parameter Bũ was introduced as a function of the Buger number (Bu), which in this case is equivalent to the flame emissivity.

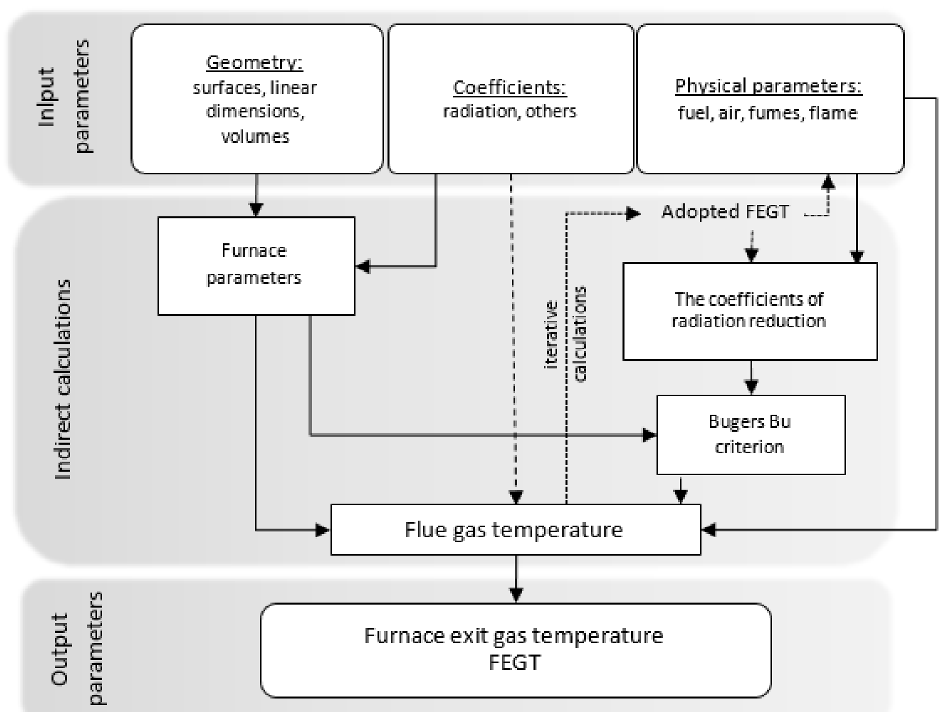

Figure 4: A simplified algorithm of CKTI method 3rd edition.

Equation for determining the furnace exit gas temperature at the outlet is presented below

$$\mathrm{FEGT} = \frac{T_{ad}}{M\mathrm{B\tilde{u}}^{0.3}\left(\dfrac{\sigma\psi_{FG}F_r T_{ad}^3}{\varphi B_{fc}\sum V_{FG}cp_{FG}}\right)^{0.6} + 1}\,. \tag{53}$$

The dependence on the temperature simplex, which corresponds to Eq. (1), is as follows:

$$\frac{\mathrm{FEGT}}{T_{ad}} = \frac{\mathrm{Bo}^{0.6}}{M\mathrm{B\tilde{u}}^{0.3} + \mathrm{Bo}^{0.6}}\,. \tag{54}$$

The formula is valid under the conditions of $\dfrac{\mathrm{FEGT}}{T_{ad}} < 0.9$.

The equivalent of the furnace's equivalent emissivity from the 2nd edition of method, Eq. (5), is the relationship

$$\mathrm{B\tilde{u}} = 1.6\ln\left(\frac{1.4\mathrm{Bu}^2 + \mathrm{Bu} + 2}{1.4\mathrm{Bu}^2 - \mathrm{Bu} + 2}\right). \tag{55}$$

In this case, the type of furnace does not affect the form of the equation.

The mentioned Buger number, which corresponds to the flame equivalent emissivity, Eq. (12), has the form

$$\mathrm{Bu} = kps, \tag{56}$$

where the individual components are similar in structure to formulae (15), (17), (20), and (22) from 2nd edition, with the difference, however, that the unit size for the attenuation factors has changed:

$$k = k_g r_p + k_{fa}\mu_{fa} + k_{coke}\mu_{coke}\,, \tag{57}$$

$$k_g = \left(\frac{7.8 + 16r_H}{\sqrt{10p_p s}} - 1\right)\left(1 - 0.37\frac{\mathrm{FEGT}}{1000}\right), \tag{58}$$

$$k_{fa}\mu_{fa} = \frac{10^4 0.8}{\sqrt[3]{\mathrm{FEGT}^2}}\frac{\mu_{fa}}{1 + 1.2\mu_{fa}s}. \tag{59}$$

The coke particle radiation attenuation factor depends only on the type of fuel and has a constant value. For hard coal it is

$$k_{coke}\mu_{coke} = 0.2. \tag{60}$$

The M parameter is a function of the ratio of the grate area to the total wall area, exhaust gas recirculation (r) – if present – and the fuel combustion method:

$$M = M_0 \left(1 + \varrho\right) \sqrt[3]{r_v}\,, \tag{61}$$

where parameter ρ determines the effect of the radiation of the fuel layer on the grate on heat exchange in the furnace chamber Eq. (10), and

$$r_v = \frac{V_{FG}\left(1 + r\right)}{V_{N2} + V_{RO2}}\,. \tag{62}$$

For grate furnaces, parameter $M_0 = 0.46$.

The degree of contamination of the heating surface, as in the previous standard, takes into account the reduction of the amount of heat transferred as a result of contamination or covering the screens with a layer of insulation. It recommends adopting the same values as in the 2nd edition.

3 Measurements of boilers units

The comparison of the presented methods was carried out for two selected real plants, water grate boilers WR10 and WR40 with a capacity of 10 MW and 40 MW, respectively. The data for the calculations were taken from the boiler's distributed control system (DCS) as well as direct results of warranty measurements carried out by specialized companies.

Measurements for both boilers were carried out in accordance with the guidelines of PN–EN 12952 standard [10] and in accordance with the procedures of the integrated management system certified by the Polish Center for Testing and Certification S.A., certified by The International Certification Network, for compliance with the requirements of the quality management system according to PN–EN ISO 9001:2015 standard [11], the environmental management system according to PN–EN ISO 14001:2015 [12] and the occupational health and safety management system according to PN–N–18001:2004 [13].

The results of calculations using individual methods are presented in the table. The variants directly result from the performed warranty tests of the boilers:

For the WR40 boiler:

Variant 1 – peak operation,
Variant 2 – basic operation.

For the WR10 boiler:

Variant 1 – operation with nominal efficiency of 100%,
Variant 2 – operation with efficiency of 68% of nominal power,
Variant 3 – operation with efficiency of 135% of nominal power,
Variant 4 – operation with efficiency of 35% of nominal power.

The obtained calculation values were compared with the actual values obtained directly from the temperature measurement at the outlet from the furnace chamber (for the WR10 boiler) and indirectly from the balance calculations (for both boilers), respectively Measured and Calculated (Balance) in Table 4.

Table 4: Results of temperature calculations at the outlet from the combustion chamber, in K.

Calculation method	Variant					
	WR40_1	WR40_2	WR10_1	WR10_2	WR10_3	WR10_4
CKTI1973 [2]	1202.5	1208.5	1062.0	931.0	1137.8	779.1
CKTI1998 [5]	1160.5	1165.3	1014.5	892.5	1076.3	736.5
CKTI–Blokh [3, 4]	1241.4	1238.5	863.1	553.9	1048.6	197.8
CKTI1951 [1]	1289.6	1288.6	1189.8	1063.5	1263.5	887.8
CKTI_China1 [7]	1327.4	1333.6	1211.2	1106	1281.1	943.8
CKTI_China2 [6]	1185.3	1188.2	1069.9	977.7	1137.5	836.2
Calculated (Balance)	1188.2	1185.2	1011.7	895.6	1060.9	726.4
Measured	–	–	1012.7	874.7	1026.9	698.3

When measuring the flue gas temperature, in addition to heat dissipation by conduction, there is also the phenomenon of heat dissipation by radiation from the measuring sensor to the colder walls of the combustion chamber. When measuring the flue gas temperature in boiler sections, the flue gas temperature sensor is often located near the heating surfaces of the boiler. The temperature of the walls of the boiler tubes, intensively cooled by flowing water, is always much lower than the temperature of the flue gas and the influence of these cool surfaces can distort the measurement results. An effective way to protect from the harmful effects of radiation is so-called shielding the sensor, consisting in the use of an additional shield, which will cause the heat exchange by radiation to take place between the sensor and the shield, the temperature of which is close to the measured temperature of the exhaust gases.

The intensity of the heat exchange between the medium and the sensor is largely dependent on flow velocity of the tested medium, so to avoid the so-called dead spaces and install sensors where the gas flow velocity is as high as possible. In the case of measurements for the WR10 boiler, a class 1 NiCr–Ni/K thermocouple with a nominal length of 1.2 m was used, which places the measurement in the boiler axis. The sensor is made as shockproof and the head has a temperature transmitter 4–20 mA programmed for the range 0–1200°C. The width of the WR40 boiler made it impossible to use a thermocouple to measure the flue gas temperature at the outlet from the combustion chamber, therefore it was decided to determine the temperature indirectly using balance equations for which the input data were measurements carried out under warranty tests for each of the boilers.

For normative methods for WR40, the differences in values for both variants are at the same level as the temperature difference (below 7 K) determined from the balances – Table 5.

Table 5: Differences in temperature calculated and measured values for WR40, in K.

Method	1951 [1]	1973 [2]	1998 [5]	Blokh [3, 4]	Ch1 [7]	Ch2 [6]
Difference	1.0	6.0	4.8	2.9	6.2	2.9

For WR10 when analyzing the influence of the load on the results obtained for individual methods, significant differences can be observed, which were not possible in the case of the WR40 boiler. For the normative methods, the differences in values increase with the decrease in the boiler load below the level of the nominal value of 100%. For the higher than nominal load, the temperature difference also increased, Table 6.

Table 6: Differences in temperature calculated and measured values for WR10, in K.

Load (%)	Calculation method					
	1951 [1]	1973 [2]	1998 [5]	Blokh [3, 4]	China1 [7]	China2 [6]
135	236.6	110.9	49.4	21.7	254.2	110.6
100	170.0	39.4	–10.2	–166.6	195.7	55.7
68	188.8	56.3	17.8	–320.8	231.3	103.0
35	189.5	80.8	38.2	–500.5	245.5	137.9

For normative methods, the temperature differences between the calculated temperature and the temperature determined from the balance for WR40_-1 and WR40_2 are shown in Table 7.

Table 7: Differences in temperature calculated and measured values for both variants of WR40, in K.

Variant	Calculation method					
	1951 [1]	1973 [2]	1998 [5]	Blokh [3,4]	China1 [7]	China2 [6]
WR40_1	101.4	14.3	–27.7	53.2	139.2	–2.9
WR40_2	103.4	23.3	–19.9	53.3	148.4	3.0

For normative methods, the temperature differences between the calculated and measured temperature for WR10_1, WR10_2, WR10_3 and WR10_4 are shown in Table 8.

Table 8: Differences in temperature calculated and measured values for all variants of WR10, in K.

Variant	Calculation method					
	1951 [1]	1973 [2]	1998 [5]	Blokh [3,4]	China1 [7]	China2 [6]
WR10_1	170.0	39.4	–10.2	–166.6	195.7	55.7
WR10_2	188.8	56.3	17.8	–320.8	231.3	103.0
WR10_3	236.6	110.9	49.4	21.7	254.2	110.6
WR10_4	189.5	80.8	38.2	–500.5	245.5	137.9

4 Summary

Calculations of the WR40 boiler for both operating states: peak operation (variant 1, WR40_1) and basic operation (variant 2, WR40_2) slightly differ for each of the methods considered. The measurements were carried out for similar boiler thermal efficiencies, therefore a convergence of temperatures at the outlet from the furnace chamber, determined from the boiler heat balances, can be noticed. It is impossible to unequivocally assess the influence of load change on temperature at the outlet of the chamber for both variants for the WR40 boiler due to the lack of measurement data for other loads during warranty tests.

In the case of the WR10 boiler, 4 variants of operation were analyzed for different boiler loads. The measurements were carried out with different thermal load of the boiler, therefore the temperatures at the outlet from the furnace chamber measured by means of an appropriate measurement have different values. When analyzing the influence of the load on the results obtained for individual methods, significant differences can be observed, which was not possible in the case of the WR40 boiler.

Calculations with the use of all the methods described do not allow to determine the temperature at the outlet from the furnace chamber for boilers at different capacities for the same level of difference of the calculated and expected value as for the nominal capacity. Boilers are designed for one capacity value – the nominal capacity for which they will be used for the majority of its operation period. The remaining load conditions, although often also the boiler operating parameters, are determined using other input data, which in turn introduces inaccuracies.

The CKTI methods are based on the theory of similarities of dependencies describing the heat transfer process in the furnace chamber. From a comparison of all editions of the methodology, i.e. from 1951 [1], from 1973 [2–4, 6, 7] and from 1998 [5], the method became more and more accurate with the development of science, increasing computational capabilities or better understanding and examination of real objects. Calculations carried out using the methodology from the 1951 edition showed the highest relative differences between the temperature calculated at the outlet of the furnace chamber and its measured counterpart, compared to other official editions (1973 and 1998). The methodology presented in the 1998 edition was characterized by the best accuracy for all tested methods.

Taking into account the method from 1973, which is currently the most common in the Polish literature in the field, its basic version allows obtaining accuracy acceptable to boiler manufacturers. The calculations performed for the 1973 methodology supplemented by Chinese researchers [7] increase the obtained temperature difference due to the application of the same dependencies for the furnace emission with the simultaneous changed dependence on the furnace substitute emissivity. For the methodology presented in the form of design guidelines [6], the dependencies on the individual components of the flame equivalent emissivity also have a changed form. For both cases, the temperature discriminant and the related temperature at the outlet from the furnace chamber also changed.

The obtained results for the methodology presented by Blokh [3, 4], as an extension of the CKTI methodology from 1973, were the most underestimated for the low loads of the WR10 boiler. The methodology itself is intended for steam boilers, mainly fired with coal dust. Therefore, its use in the case of boilers of small capacity and power is not justified.

In the case of determining the parameters at the outlet from the furnace chamber for boilers operating for different thermal outputs, the disproportions between the calculated and measured values increase. The authors of the methodology accept the absolute error at the level of ± 100 K. Water and steam power boilers are designed for nominal parameters and these should be the main determinant of the correctness of thermal calculations. Other operating modes (including other capacities) should be recalculated and checked, but taking into account possible disproportions.

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