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CHALLENGES IN THE THERMAL DESIGN OF LOW-CAPACITY GRATE COMBUSTION CHAMBERS

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Abstract: *The article deals with the issue of the use and reliability of the standard thermal design calculation of grate combustion chambers of low-capacity local heat sources. This standard analytical calculation based on dimensionless characteristic numbers tends to fail when applied to smaller furnaces even though it gives good results for centralized sources of higher capacity. However, when designing low-capacity boilers, it is still appropriate and advantageous to use this standard (empirical-analytical) calculation, which is able to give at least indicative results in an incomparably shorter time and with significantly lower costs than an alternative and demanding numerical analysis based on CFD (Computational Fluid Dynamics) analysis.*

Keywords: Grate combustion, thermal design calculation, low-capacity combustion chambers

1. Introduction

Grate combustion chambers are being used to obtain thermal energy by burning solid fuels in piece or bulk form with a sufficiently large fraction. Currently, grate furnaces are primarily used in facilities burning wood biomass and combustible wastes. The construction of the grate combustion chamber typically consists of the grate itself, on which the fuel burns in a layer, and the surrounding volume above the grate, which is bounded by a temperature and chemically resistant lining (see *Fig. 1*). This first part of the combustion chamber (the so-called primary combustion chamber, PCC) is subsequently followed by the second part (the so-called secondary combustion chamber, SCC), which can be equipped with a membrane wall, or a similar heat exchange system serving to use the radiant heat of the generated flue gases and reducing their temperature at the exit from the combustion chamber, or SCC (Jegla et al., 2010).

The outlet temperature of the flue gas from the combustion chamber (T_{bw}) is limited during the design stage mainly due to the negative impact on the production of thermal NOx and problems with fouling of the heat exchange surfaces with molten ash. In the case of large and medium capacity chambers, the outlet temperature of the flue gas is calculated during the design stage from the energy balance of the heat released by fuel combustion and the heat removed by the membrane wall or other heat exchange surface in the SCC and checked for the above-mentioned technological limitations (Basu et al. 2000).

However, in the case of low-capacity furnaces, it is difficult to use with sufficient accuracy the standard analytical-empirical calculation apparatus derived for common (i.e., high and medium) industrial capacities. The reason is that as the capacity of the device decreases, the active volume decreases with the third power and the lining surface decreases with the second. Although this is to some extent compensated for in the formulas used, the general shape of the combustion chamber walls and the deviation from the ideal state assuming the simplified radiation model considering an infinite surface can

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lead to a significant calculation inaccuracy. The mentioned problems with the inaccuracy of the standard thermal calculation are already starting to appear during the design and operation of combustion chambers of medium capacities.

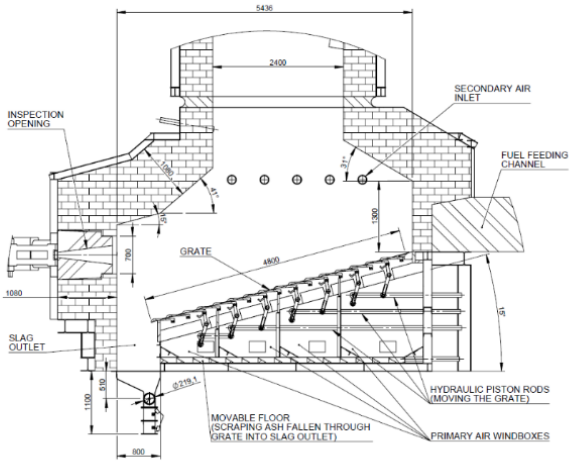

Fig. 1: Sample design of an industrial primary combustion chamber with a grate for a medium-capacity boiler with a thermal output of 9.6 MW (Zabloudil, 2022)

2. Methods

The standard recognized method of thermal calculation of boiler combustion chambers is a globally widespread and used calculation method using a characteristic number, adiabatic combustion temperature and parameters expressing radiation in the furnace (Basu et al. 2000). This so-called Gurvich method is based on the following semi-empirical formula (Basu et al. 2000):

$$\frac{T_{bw}}{T_{ad}} = \frac{Bo^{0,6}}{M \cdot a_f^{0,6} + Bo^{0,6}} \#(1)$$

where:

- T_{bw} $[K]$ is the set temperature of the flue gas at the exit from the combustion chamber (otherwise called bridgewall temperature)
- T_{ad} $[K]$ is the analytically calculated adiabatic flame temperature of the given fuel
- Bo $[-]$ is the Boltzmann characteristic number (dependent on T_{ad}, radiation properties of surfaces in the furnace, heat capacity of flue gases, rate of the combustion)
- M $[-]$ coefficient characterizing the position of the flame in the given furnace
- a_f $[-]$ characteristic furnace emissivity

The method is very suitable for fluidized bed furnaces and other combustion chambers designed for higher capacities, but due to the circumstances described in the previous chapter, it tends to fail with low-capacity devices. It is therefore currently necessary to design them using measurements on prototypes, or with the use of time- and capacity-consuming numerical CFD models.

The analytical approach for an adequately accurate determination of T_{ad} and the mean heat capacity of flue gases is based on the chemical balance of the production of gaseous substances by oxidation of individual elements in the fuel and estimation of the lower heating value (LHV) of the fuel, also depending on its composition. The approximate value of T_{ad} is determined by interpolation from known values of the flue gas components' temperature-enthalpy relations. A description of the theory of such a calculation can be found in specialized literature, for example in (Basu et al. 2000).

The very method of thermal calculation of the combustion chamber according to equation (1) was first presented by Gurvich and Bloch in 1956 (Gurvich and Bloch, 1956) and for the needs of initial thermal design of combustion chambers it still has not been surpassed.

As a semi-empirical method, however, in its empirical part, it uses the selection of tabular values of constants that characterize entire groups of combustion devices, according to their typical construction. The values of the calculated results are therefore again most accurate for large-capacity furnaces firing powdered fuels, whose combustion chambers can be described as almost regular cuboids. On the other hand, devices with a more complex characteristic geometry thus acquire additional introduced inaccuracy during the calculation.

3. Case study – thermal calculation vs. actual operation:

The calculation method described above was applied to two operating industrial furnaces of similar construction (*Fig. 1*) as an assessment of its predictive capabilities. Both are grate combustion chambers, the design and continuously measured operational data of which the author received from the operators of the facilities that operate them. Due to the non-disclosure agreement, it is possible to present only the necessary parameters required by the calculation model and the design of the combustion chambers. In the following text, these devices are therefore referred to only as *device 1* and *device 2*.

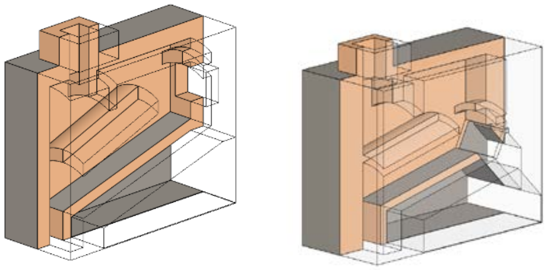

Fig. 2: Sections of reference combustion chambers used in the assessment of the calculation method (left: device 1; right: device 2)

Device 1 incinerates hospital waste in an EU member state. PCC is roughly cuboidal, mainly countercurrent and without a significant ignition arch. The main dimensions of the inner volume are approx. 4.6 x 3 x 1.3 m. The PCC ceiling is vaulted. The grate is reciprocating, divided into two air sections, each of which has a separate fan and the possibility of independent movement of the grate bars. The gradient of the grate is 16°. The countercurrent arrangement is ensured by a vaulted insert made of fireclay blocks at a height of 1 m above the plane of the grate, smeared with a layer of refractory concrete. The rest of the inner layer of the PCC lining is also made of the same material. The secondary air is led through a channel in the casing into the space above the fireclay insert. The retention length between the last air supply vent and the exit from the combustion chamber is approx. 3 m. The exit from the PCC is followed by a cylindrical horizontal SCC equipped with a natural gas stabilization burner and a fire-tube boiler.

Device 2 burns wood waste and wood chips and is operated in a non-EU country, therefore the operation is governed by less strict rules. PCC is also roughly cuboidal, purely countercurrent and without a significant ignition arch. The main dimensions of the inner volume are approx. 4.1 x 3.2 x 1.4 m. The PCC ceiling is flat. The grate is reciprocating, divided into two air sections, each of which has a separate fan and an adjustable supply of recirculated flue gas. The gradient of the grate is 12°. The countercurrent arrangement is ensured by a vaulted insert made of fireclay blocks at a height of 1.3 m above the plane of the grate. The rest of the inner layer of the PCC lining is also made of the same material. Secondary air is led through three channels into the space above the grate and into the space above the fireclay insert. The retention length between the last air supply vent and the exit from the PCC is approx. 3.2 m. The exit from the PCC is directly followed by the SCC, designed also in this case to be followed by a fire-tube boiler.

The values shown in the following *Table 1* were obtained from the operating data measured at steady state and from the documentation provided. The data was then used for the verification thermal calculations of both devices. The composition of the fuels was estimated based on analyzes of fuels of the same nature, available in specialized literature.

Tab. 1: Operating and design parameters of device 1 and device 2 in steady state operation

Parameter		Device 1	Device 2	Unit
Temperatures:	Combustion air temperature	29	20	°C
	Mean furnace temperature	990,4	-	°C
	Furnace exit gas temperature (T_{bw})	927,6	700	°C
	Ash deformation temperature	1100	1038	°C

Parameter		**Device 1**	**Device 2**	**Unit**
	Stack gas temperature	134,6	132	°C
Combustion air:	Excess air coefficient	1,5	2	-
	Flue gas recirculation	No	5	vol. %
Grate:	Grate area	5,16	4,51	m^2
	Grate gradient	16	12	°
	Number of air zones (windboxes)	2	2	-
Furnace:	Relative flue gas pressure	-64	-38	Pa
	Furnace walls area	45,23	49,59	m^2
	Refractory lining area	38,43	42,88	m^2
	Active volume of the furnace	12,22	14,24	m^3
Fuel:	Type	Hospital waste	Waste wood, woodchips	-
	Mass flow of the fuel	352	600	kg/h
	Combustibles content	60,00	49,60	w. %
	Moisture content	34,00	50,00	w. %
	Ash content	6,00	0,40	w. %
	Lower heating value (LHV)	15547	8583	kJ/kg
Intermediate	Adiabatic flame temperature (T_{ad})	1053,1	947,5	°C
Calculations:	Boltzmann char. number (Bo)	19,0708	22,2257	-
	Coefficient M	0,59	0,59	-
	Characteristic emissivity	0,0434	0,0383	-

The main result of the thermal calculation performed according to equation (1) for each of the combustion grate devices with parameters from *Table 1* is the determination of the temperature of the flue gas at the exit from the SCC (i.e., T_{bw}). Compared to the measured value of this temperature during operation, this calculated temperature is a necessary indicator of the accuracy of the calculation method. The calculated and measured values of T_{bw} are shown in the following *Table 2*.

Tab. 2: Comparison of the calculated and measured T_{bw}

Device	**Parameter**	**Unit**	**Measured value**	**Calculated value**
1	Furnace flue gas exit temperature (T_{bw})	°C	927,6	943
2	Furnace flue gas exit temperature (T_{bw})	°C	700	860

4. Conclusions

It is clear from *Table 2* that the result of the thermal calculation for *device 1* is satisfactorily close to the measured value, while excellent agreement (corresponding to reality) is considered to be a deviation of the calculated T_{bw} up to 50 K from its measured value (Basu et al. 2000). In contrast, for *device 2*, the measured temperature is significantly different from the calculation result. Considering that the same kind of gauges were used for both devices and the same tabular coefficient values were also used in both cases, this case study shows that even at medium processing capacities there are significant inaccuracies in the thermal calculation of grate combustion chambers. Refinement of the thermal calculation, especially in the sense of taking into account the influence of individual designs of grate PCC on heat exchange rates will be the subject of future research work.

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