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COMPUTATION OF THE TEMPERATURE DISTRIBUTION IN BIOMASS BOILERS WITH RECIPROCATING GRATE FURNACES

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ABSTRACT

To achieve a higher efficiency and emission reduction from biomass boilers, a detailed knowledge of the turbulent flow quantities, gas species, velocity, and temperature distribution in the combustion chamber is needed. The complex thermal degradation processes makes it difficult to achieve an accurate prediction in short computation time for rapid prototyping. Hence a simplified but sufficiently accurate pre-processing model for describing the solid biomass combustion on the grate is presented. This pre-processing model computes the species mass and energy fluxes as boundary conditions so that only the reactive gas phase flow in the combustion chamber have to be simulated. The model was validated through a 100kW wood pellet and a 1MW wood chip furnace system. For the turbulent reactive gas phase flow the Realizable-k-ε turbulence model, eddy dissipation model in combination with a 3-step global reaction mechanism for the methane-air mixture and the discrete ordinates model radiation were used. Boundary conditions were obtained from experimental measurements which included heating value, moisture, and the elementary contents (C, H, O, and H_2O) of the wood pellet and chips (determined by fuel analysis). To validate the computed results temperature distributions in the combustion chambers were measured with S-type thermocouples. The simulated temperature distributions showed in both combustion chambers good agreements with the experimental findings.

NOMENCLATURE

$\dot{m}$	mass flow rate [kg/s]
T	temperature [°C]
$T_{EQ\text{-}CFD}$	simulated equilibrium temperature [°C]
$T_{EQ\text{-}EXP}$	measured equilibrium temperature [°C]
$T_{FG\text{-}CFD}$	simulated flue gas temperature [°C]
T_{max}	maximum grate temperature [°C]
$T_{RAD\text{-}CFD}$	simulated radiation temperature [°C]
$u_{MAG\text{-}CFD}$	simulated velocity magnitude
x	length direction of the grate [m]
L	grate length [m]
ε	emission coefficient [-]
λ	Air–fuel equivalence ratio [-]

INTRODUCTION

The threat of climate change has resulted in increased use of sustainable energy resources, such as biomass for electricity production in place of fossil fuels. In Europe thermal biomass systems represent an essential part of the energy supply chain. The large temperature range and the varied useable biomass fuel make its operation attractive. However, EU directives and local government restrictions on fine dust, CO and NO_X emissions require continuous improvement of the system. Modern biomass boilers use an automatic fuel loader with reciprocating grate furnaces where combustion takes place. Along the grate different zones of solid fuel pellets/chips to its gas emissions (CH_4, CO, CO_2, H_2, H_2O, O_2 etc.) are established. The biomass chips or pellets enter the combustion chamber by passing through the drying zone. The chips/pellets are then heated to a temperature that produces auto-ignition in the ignition zone. This is followed by the main combustion and burnout zone.

The grate design should be compactly constructed and allow a homogeneous combustion chamber temperature, low dust and NO_X emission. The complex interaction between these requirements makes it necessary to derive such measures of performance in a real system. Therefore detailed information about the fluid- and thermodynamic transport processes and chemical reaction rates are needed.

Commercial CFD codes offer a wide range of physical transport and chemical reaction models to determine the relevant quantities - but the treatment of the pyrolysis in the grate area is still an unresolved problem (Xue et al. 2012; Yin et al. 2008, Mehrabian et al. 2011, 2012, 2014 and 2015, Shiehnejadhesar et al. 2015). Model approaches which consider the motion and conversion of each individual chip or pellet have an exorbitant computational effort and therefore are not capable for industrial applications. A promising alternative is the measurement or the determination of species release rate profiles. The profiles can be implemented as source terms or as boundary conditions in the grate region (Scharler 2001; Zhang et al. 2010; Chaney et al. 2012; Shiehnejadhesar et al. 2013). The main disadvantage of this approach is the insufficient interaction between the flue gas and the wood pellet and the chip combustion emission. Therefore this model is not able to reproduce the combustion processes far away from the ideal operating condition and reliable use of it requires comprehensive validation. However its remarkably low computational effort makes the boundary condition approach meaningful for industrial applications.

Hence, in this paper a suggested approach is presented and validated against experimental findings. The validations were carried out by comparing temperature profiles with two independent investigations of reciprocating grate furnaces – one for a 100kW wood pellet and one for a 1MW wood chip.

MODEL DESCRIPTION

Empirical fixed-fuel bed combustion model

The basic concept of the empirical conversion model was developed by Scharler (2001) and Klasen (2003). Since then it has been applied in two-dimensional modelling of combustion of fixed fuel beds (Hermansson and Thunman, 2012; Shiehnejadhesar et al. 2013). This considers the biomass burn-out behaviour and the conversion from solid to gas components by mass and energy balances. The local temperature distribution, species and mass flow rates are obtained in the grate furnace, just above the fuel layer. In this work a similar empirical model which considers the burn out of the solid biomass as a “black box” was used. Hence, the solid to gas conversion model and the CFD combustion simulation are decoupled.

In principle the empirical model consists of three parts which are executed sequentially:

i. Determination of the thermal degradation profiles along the grate for the fuel components C, H, O and H_2O as well as the primary air and the temperature distribution.
ii. Determination of the conversion of the fuel components C, H, O to the flue gas components CH_4, CO, CO_2, H_2, H_2O and O_2 in dependent on the local stoichiometric air ratio.
iii. Balancing of the mass and energy rates along the grate.

The thermal degradation profiles of the C component along the normalized grate was carried out by a fit-function (cf. Scharler, 2001) developed from experimental investigations on a flat reciprocating grate furnace. In this study four different positions samples were retrieved and analysed from the grate during nominal load. In addition two fixed points were assumed – the starting point of the degradation (x = 0, C/C_{in} = 0) and the end of the burnout zone ($x = L$, C/C_{in} = 0). With this data a curve fit-function was determined which approached the degradation of the C component along the grate furnace (Figure 1).

Experimental investigations showed that the fuel decomposition started at the same time and the C component can be used as the reference indicative parameter for the decomposition processes of H, O and H_2O. Hence, the decomposition rate of each component can be simply determined by use of its gradients.

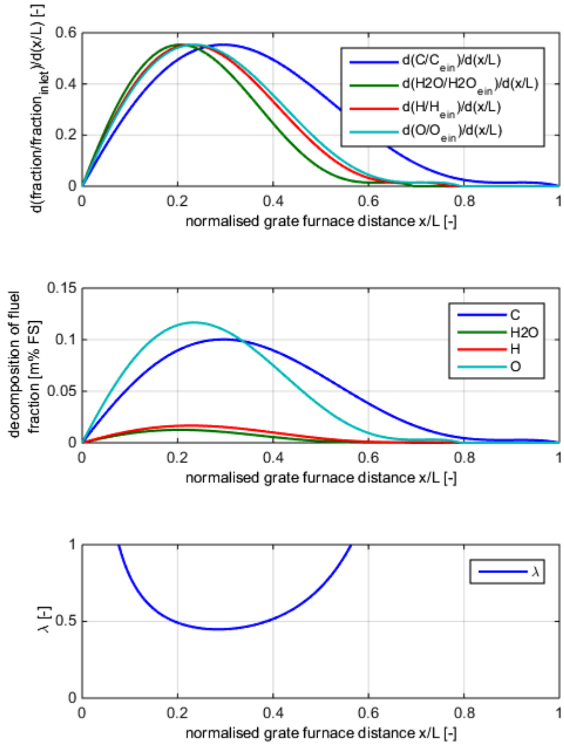


Figure 1: Determined decomposition of the fuel fraction and the air–fuel equivalence ratio along the normalised grate furnace for the 100kW pellets boiler.

Based on the curve fit-function of the C component, the H, O and water component species decomposition were computed by a linear correlation. Figure 1 shows the relative species concentration (ratio between the local discharged concentration and the fuel input concentration) along the normalized grate distance (*x/L*). The area under each curve corresponds to the respective discharged fuel components. With the biomass inlet mass flow rate the discharged mass flow rate of each component can be calculated for local regions/sectors of the furnace grate.

Oxygen was provided from the primary air supply and used for continuous conversion of the fuel components C, H, O, and the water drying process. It was assumed that the primary air stream is homogeneously distributed over the bottom side of the grate. Hence, the flue gas components CH_4, H_2O, H_2, CO_2, CO and its flow rates were computed from the combustion calculations with respect to the local air–fuel equivalence ratio. Figure 2 shows the determined discharge species along the grate exemplified for the 100kW pellets boiler.

The required chemical analysis and heating value for the wood pellets and chips were carried out in a combustion laboratory. The mass flow rates and the energy fluxes were calculated for each discrete section along the grate so that the flue gas mass flow rates could be used directly for the boundary conditions. All other mass flow rates relevant to the combustion processes (e.g. secondary air) were included directly in the CFD model. It should be pointed out that the empirical fixed-fuel bed combustion model does not consider additional leaked air into the grate area.

The boundary condition computations for the CFD simulation was developed in Matlab (Mathworks, 2014) which automatically determines the discharged species fraction and mass flow

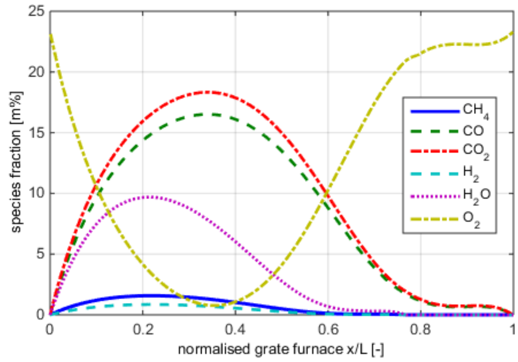


Figure 2: Determined discharges species fraction along the normalized grate distance *x/L* for the 100 kW pellet boiler.

rates dependent of the fuel composition and heating value. In addition, the Matlab code generated a boundary condition profile which was directly read into the CFD code ANSYS Fluent.

The temperature distributions along the grate were found by experimental measurements. The temperatures were measured at discrete sample points just above the firebed for several biomass chips and pellets boilers. If the measurement results are normalised by the grate length and the maximum temperature all temperature profiles shows a similar distribution. Hence a fit-function according to Figure 3 was used to determine the temperature boundary condition along the grate.

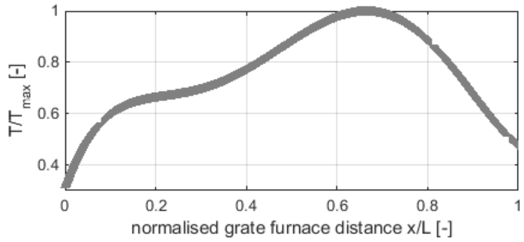


Figure 3: Determined non-dimensional temperature profile from the experiments.

Modelling the fluid phase and combustion

The commercial CFD software ANSYS-Fluent was used for the simulations. The turbulent momentum exchange, heat transfer and species mixing were carried out using the linear eddy viscosity approach and the Realizable k-ε closure model (Shih et al., 1995) with standard wall function. The radiation heat transfer between the surface and the flue gas was determined by the Discrete Ordinates Radiation Model (angular discretization 4x4). The Eddy Dissipation Model from Magnussen and Hjertager (1976) in combination with the 3-step reaction approach according to Brink (1998) were used for the gas phase reaction (cf. equation 1-4). For the oxidation of methane and hydrogen it was assumed that chemical reactions occur much faster than turbulent mixing process. The oxidation of the carbon monoxide was controlled from both – the turbulent mixing rate and the chemical reaction rate. The slowest rate was then used to determine the reaction rate.

$$C_xH_y + \left(\frac{x}{2} + \alpha\frac{y}{4}\right)\cdot O_2 \rightarrow \tag{1}$$

$$x\cdot CO + (1-\alpha)\cdot\frac{y}{2}\cdot H_2 + \alpha\cdot\frac{y}{2}\cdot H_2O$$

with

$$\alpha = \frac{[H_2O]_{eq}}{[H_2O]_{eq} + [H_2]_{eq}} \tag{2}$$

$$CO + \frac{1}{2}\cdot O_2 \rightarrow CO_2 \tag{3}$$

$$H_2 + \frac{1}{2}\cdot O_2 \rightarrow H_2O \tag{4}$$

With the proposed assumptions, faster calculation times were produced for the combustion modelling which is critical for industrial applications.

The proposed approach was tested with two different biomass combustion chambers, a 100kW wood pellet boiler, and a 1MW wood chip boiler. For the discretization of the convective terms the second order upwind scheme and for the pressure-velocity coupling the SIMPLE algorithms were used. The computational domain and the used boundary conditions are shown in Figure 4 and 5. For the 100kW boiler a grid with 6 mill. cells and for the 1MW boiler a grid with 21 mill cells were carried out (Figure 6).

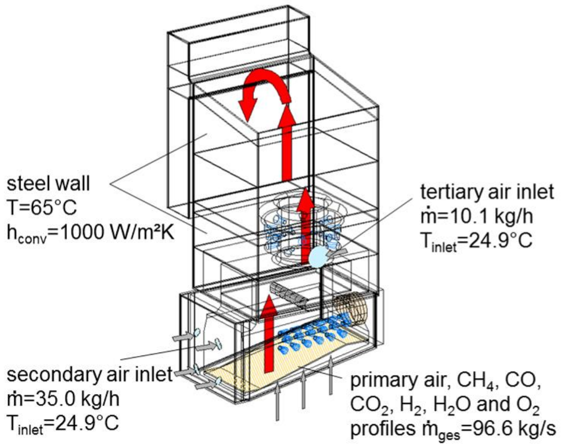


Figure 4: Computational domain and used boundary condition for the 100 kW pellet boiler.

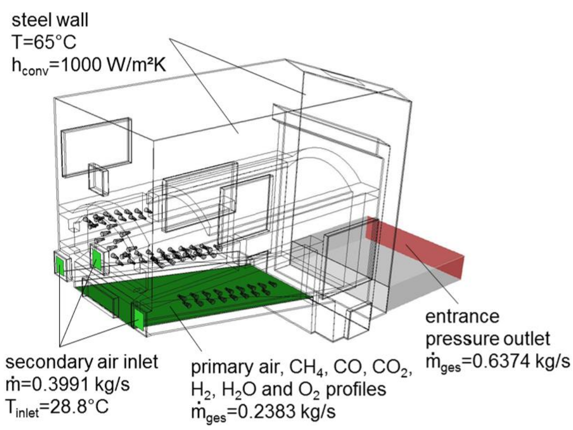


Figure 5: Computational domain and used boundary condition for 1MW wood chip boiler.

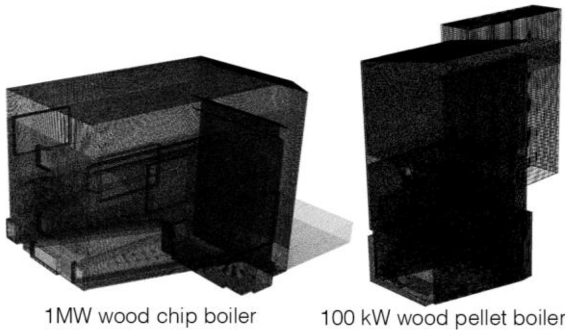


Figure 6: Computational mesh used for 1MW wood chip boiler and 100kW wood pellet boiler

EXPERIMENTAL INVESTIGATIONS

Two biomass combustion chambers, 100kW wood pellet boiler, and the 1MW wood chip boiler were investigated and the used test rigs are described below:

Investigated biomass combustion chamber systems

The construction design of the biomass boilers are shown in Figure 4 and 5. Both boilers feed the biomass fuel towards the burning chamber with a feeding screw and a reciprocating grate furnace. In the considered cases, the primary combustion air was supplied from the bottom of the grate and secondary air was supplied in the main combustion chamber with multiple distributed jet nozzles. Both boilers used a non-adiabatic combustion chamber and a downstream mounted heat exchanger module. To ensure an optimized heat radiation distribution and an ideal gas mixture time the combustion chambers were equipped with different shaped fire-proof cupola constructions. To enable well defined boundary conditions during the experimental investigations a separate control system (fuel and air mass flow rates etc.) was developed and integrated in the test facilities. The temperature sensors for the CFD model validations were installed in the post-combustion chamber of the 100kW wood pellet boiler and in the intersection between the main and the post-combustion chamber of the 1MW wood chip boiler.

Test Rigs

Two different test rigs were developed to determine the efficiency and biomass boiler emissions.

The first rig was a boiler with a thermal output up to 100 kW (Figure 7) and wood pellets as the fuel. Continuous measurement of the heating power input was made using a digital monitor. The heating value, chemical composition, and water content of the biomass fuel were measured with a calorimeter, elementary chemical analyser, and a drying chamber. The efficiency was determined by the boiler fuel supply and heat power output. On the test rig, more than 60 continuously measuring sensors were installed. The primary and secondary air mass flow rates, temperature and humidity were measured separately for three inlet zones. The data recorded and visualization were carried out with A/D-converter technology and a LabView program from which the data was exported for further processing. Gas emission measurements were found from industrial gas sensors. More detailed information of the used test rig can be found in Krail et al. (2015).

Figure 7: Experimental test rig used for the 100kW wood pellet boiler.

The second test rig was similarly arranged but developed for higher thermal loads so that it was used for a thermal output up to 2MW. This test rig was used for the 1 MW wood chip boiler.

Additional temperature measurement method

To analyse the temperature distribution in both combustion test chambers additional S-type (PtRh-Pt) thermocouples were installed (Figure 8) According to IEC 60584-1 (2013) the accuracy class of the used sensors is 1 (± 1.0 °C or [1+0.003(T-1100°C)]). The thermocouples had a ceramic-built protective shell with a radiation emission coefficient between 0.3 - 0.43 (emission temperature 1000°C). The sensor emission properties had significant relevance. The non-adiabatic combustion chambers caused a deviation between the flue gas and radiation temperatures. If the radiation and flue gas temperatures were significantly different then the sensors measured the thermodynamical equilibrium temperature T_{EQ}. In terms of the CFD validation this issue was taken into account.

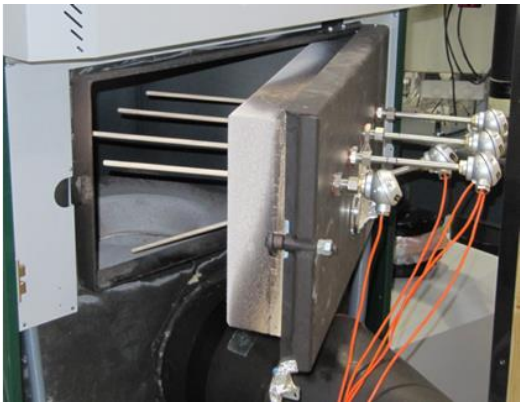


Figure 8: Temperature sensors of the post-combustion chamber from the 100 kW wood pellet boiler.

Therefore the equivalent equilibrium temperatures in the CFD simulations were computed with the energy conservation equation during post processing. The Nusselt-numbers for the convective heat transfer were determined with the Engineer Equation Solver (Klein, 2015) for flow around a cylinder. The radiation heat transfer were determined by the use of the computed radiation temperature and a sensor emission coefficient of 0.3 and 0.43.

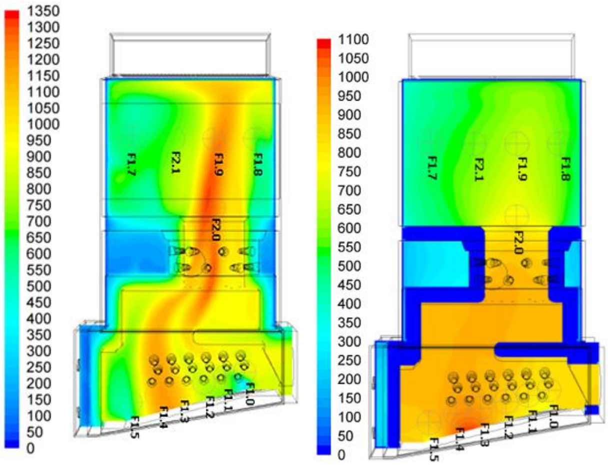
RESULTS AND DISCUSSION

100 kW wood pellet boiler

Five thermocouples were mounted in the post combustion chamber to record the temperatures for validation of the computed temperature distribution in the 100kW wood pellet boiler. Figure 9 shows the computed flue gas temperature and the computed radiation temperature distribution along the cross section. The positions of the five thermocouples (F1.7, F2.1, F1.9, F1.8 and F2.0) are illustrated.

Comparison of both temperature distributions show differences between the flue gas and the radiation temperature which confirms the need to use the thermodynamical equilibrium temperature. Based on the measured points, the computed values are directly compared and given in Table 1.

Accordingly, the differences in temperatures can reach several hundred degrees. In terms of the comparison, the computed and measured equilibrium temperatures agree well.



(a) Flue gas temperature [°C] (b) radiation temperature [°C]

Figure 9: Computed (a) flue gas temperature [°C] and (b) radiation temperature [°C] distribution in the 100 kW wood pellet boiler.

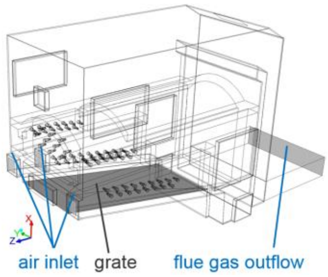
	F1.7	*F2.1*	*F1.9*	*F1.8*	*F2.0*
$T_{FG\text{-}CFD}$ [°C]	716	700	1195	754	1250
$T_{RAD\text{-}CFD}$ [°C]	526	577	663	581	798
$u_{MAG\text{-}CFD}$ [m/s]	1.4	1.3	6.0	2.0	7.3
$T_{EQ\text{-}CFD\ \varepsilon=0.43}$ [°C]	**609**	**626**	**895**	**656**	**974**
$T_{EQ\text{-}CFD\ \varepsilon=0.30}$ [°C]	**625**	**637**	**935**	**671**	**1008**
$T_{EQ\text{-}EXP}$ [°C]	**627**	**656**	**943**	**679**	**1029**

Table 1: Comparison of the measured and computed temperatures. Subscript definitions: FG = flue gas, RAD = radiation; EQ = equilibrium temperature; ε = emission coefficient of the temperature sensors

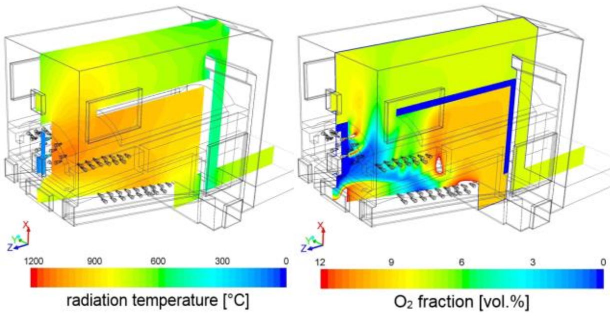
1MW wood chip boiler

Figure 10 shows the (a) CFD wire frame model, (b) the computed radiation temperature and (c) computed oxygen distribution in the symmetry plane. The maximum radiation temperature reaches just under 1100°C at the air inlet region, while the computed oxygen fraction in the outlet area of the furnace is about 7.5 vol.%. Under the same steady state boundary conditions 7.4 vol.% was measured.

Three thermocouples were mounted in the post combustion chamber for validation of the computed temperature distribution. The positions of the temperature measured points as well as the simulated flue gas temperature, radiation temperature and the velocity magnitude are shown in Figure 11.



(a) CFD wire frame model



(b) (c)

Figure 10: CFD model of the 1MW wood chip boiler (a) CFD wire frame model, (b) computed radiation temperature distribution, (c) computed oxygen distribution [vol.%]

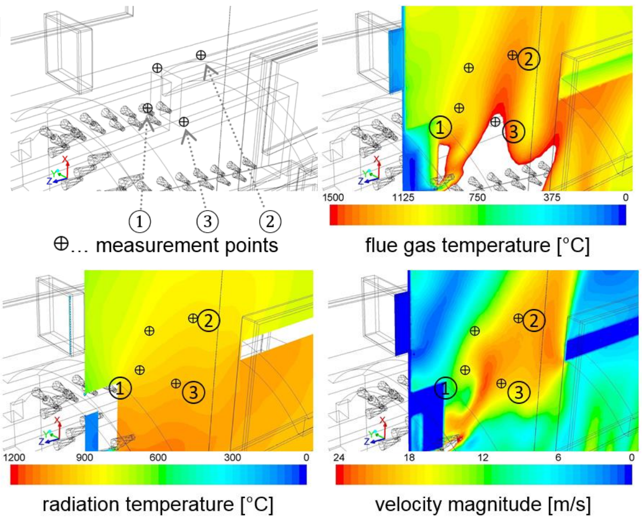


Figure 11: Position of the temperature measurement points in the post combustion chamber. Distribution of the flue gas temperature, radiation temperature and velocity magnitude in the measurement plane

The computed values are directly compared and given in Table 2 which showed the measured temperatures agreeing very well with the simulated temperatures (point ①, ②, and ③).

	①	②	③
$T_{FG\text{-}CFD}$ [°C]	1126	1296	1494
$T_{RAD\text{-}CFD}$ [°C]	910	881	990
$u_{MAG\text{-}CFD}$ [m/s]	16.2	18.6	19.1
$T_{EQ\text{–}CFD\ \varepsilon=0.43}$ [°C]	**1006**	**1066**	**1183**
$T_{EQ\text{–}CFD\ \varepsilon=0.30}$ [°C]	**1024**	**1099**	**1222**
$T_{EQ\text{–}EXP}$ [°C]	**966**	**1079**	**1143**

Table 2: Comparison of the measured and computed temperatures. Subscript definitions: FG = flue gas, RAD = radiation; EQ = equilibrium temperature; ε = emission coefficient of the temperature sensors

CONCLUSION

For industrial applications which demand fast and efficient computations, a solid biomass combustion simulation using an empirical fixed-fuel bed combustion model was investigated. The thermal degradations of the fuel components and the species release rates were determined over normalized profiles along the grate furnace distance. The profiles were computed with an external Matlab program and implemented as boundary conditions in the grate region. As a result of this approach only the gas phase combustion in the grate furnaces needed to be simulated and therefore the computation effort can be significantly reduced. The presented computational results showed that the method provided good agreement with the experimental findings when the equilibrium temperature was considered. Hence, the simulation-based development of primary measures e.g. for combustion efficiency and emission reduction can be rapidly performed with this approach. However, the application of this method assumes that the thermal degradation profiles of the fuel components are known along the grate.

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