Deriving guidelines for the design of biomass grate furnaces with CFD analysis – a new Multifuel-Low-NO_x furnace as example

Robert Scharler[1], Ingwald Obernberger[1,2]
[1]Institute of Chemical Engineering Fundamentals and Plant Engineering, Graz University of Technology, Inffeldgasse 25, Austria; Tel.: +43 (0)316 481300 33, Fax: +43 (0)316 481300 4,
E-mail: scharler@bios-bioenergy.at
[2]BIOS – Bioenergy Systems, Sandgasse 47, A-8010 Graz, Austria

Abstract

In the present work, guidelines for the design of grate furnaces in the typical size range of Austrian district heating plants (0.5 – 10.0 MW_{th}) were derived from CFD analysis. For this purpose, a case study systematically investigating all important parameters for a successful design of a Low-NO_x furnace was performed.

The simulation of solid biomass combustion on the grate and turbulent reactive flow in the combustion chamber was performed via forward coupling. An empirical model was applied for thermal decomposition of the solid fuel. The calculation results were used as boundary profiles for subsequent CFD simulations of the turbulent reactive flow in the furnace. CFD modelling was based on the Realizable k-ε Model (turbulence), the Discrete Ordinates Model (radiation) and the Eddy Dissipation Model (turbulent combustion) in combination with a global 3-step reaction mechanism considering the species CH_4, CO, CO_2, H_2, H_2O and O_2. NO_x formation and reduction were not considered explicitly with a CFD postprocessor. The furnace was designed under Low-NO_x aspects, i.e., with air staging technology and a temperature controlled primary combustion zone obtained by flue gas recirculation.

As a result of the case study, a Low-NO_x biomass grate furnace for a broad fuel assortment (waste wood, wood chips and bark) was developed and implemented as a pilot-scale plant. Moreover, the modular structured CFD based furnace development allows further combustion chamber geometries to be created depending on the application.

Furthermore, guidelines for the design of biomass grate furnaces were derived. In general, a considerable reduction of investment and operating costs is possible by a compact furnace design as well as by reduced air and flue gas fluxes in the furnace. This can be achieved by an appropriate design of the nozzles for air and flue gas injection as well as by adjusting the geometry of the combustion chamber (e.g. barriers) in order to improve the mixing of unburned flue gas and air as well as the utilisation of the furnace volume. A staged secondary air supply leads to a strong reduction of CO emissions, but is linked with increased flue gas temperatures between the nozzles for secondary and tertiary air injection. The higher temperature peaks due to improved mixing conditions have to be considered concerning ash slagging and deposit formation. For dry fuels (e.g. waste wood) additional measures like furnace cooling and enhanced flue gas recirculation are necessary. A scaling of an optimised furnace geometry should be based on approximately constant ratios of the characteristic Reynolds numbers.

Keywords

CFD modelling, Eddy Dissipation Model, biomass, combustion, grate furnace.

1. Introduction and objectives

Biomass combustion represents a possibility to lower regional emissions of the greenhouse gas CO_2, especially in countries with large wood resources. In Austria about 10% of the primary energy demand is supplied by thermal utilisation of biomass, whereof decentralised heating plants represent an important sector. Despite the ecological advantages of renewable energies, a techno-economic optimisation of the combustion unit is necessary in order to make biomass-fired heating and CHP plants competitive with fossil fuel-fired systems. The major goal of the development and optimisation of biomass grate furnaces is the reduction of investment and operating costs. This can be achieved by a compact furnace design, an increased availability of the plant, by reduced emissions (CO and NO_x) as well as by reduced air and flue gas fluxes in the furnace.

The design of biomass furnaces is still usually based on experience and empirical data. An exemplary case study systematically investigating all important parameters relevant for the design of a Low-NO_x grate furnace for a broad assortment of wood derived fuels was thus performed. Moreover, the modular structured CFD based furnace development allows further combustion chamber geometries to be created depending on the application. Another major goal of the work presented was to derive guidelines for the design of grate furnaces in the typical size range of Austrian biomass district heating plants (nominal boiler capacity 0.5 – 10 MW_{th}) from the CFD analysis performed.

2. Modelling

The simulation of solid biomass combustion on the grate and turbulent reactive flow in the combustion chamber was performed via forward coupling. An empirical model was applied for thermal decomposition of the solid fuel. The calculation results were used as boundary profiles for subsequent CFD simulations of the turbulent reactive flow in the furnace.

2. 1 Modelling of fixed bed combustion

A detailed mathematical treatment of biomass fixed bed combustion is, on the one hand, too time-consuming for engineering applications and, on the other hand, not completely understood so far. An empirical model was thus applied for the thermal decomposition of the solid fuel on the grate. The model consists mainly of three parts. The definition of one-dimensional profiles along the grate concerning the degradation of the fuel components as well as fuel drying (part No. 1; based on assumptions and experimental data) is the basis. In combination with the definition of conversion parameters (based on assumptions as well as experimental and literature data), which describe the formation of the most important flue gas components CH_4, CO, CO_2, H_2, H_2O and O_2 (part No. 2), the stepwise balancing of mass and energy fluxes released from the fuel bed is possible (part No. 3). For a more detailed description of the model see Scharler [9].

2. 2 CFD modelling of turbulent reactive flow

The flow in the combustion chamber (above the fuel bed up to the entrance into the boiler) was simulated with the unstructured CFD solver Fluent 5. For modelling of the turbulent reactive flow the Realizable k-ε model (turbulence), the Discrete Ordinates Model (radiation) as well as the Eddy Dissipation Model of Magnussen and Hjertager [5] (turbulent gas phase combustion) in combination with a global methane 3-step reaction mechanism (Brink [1]) were used.

The EDM in combination with reaction kinetics serves to calculate the rate of fuel consumption R_{br} as the lowest (limiting) step of mixing rates proportional to the decay of fuel, oxygen and product swirls as well as a kinetic rate $R_{br,kin}$ without consideration of the influence of turbulent fluctuations:

$$\overline{R_{br}} = MIN\left(\underbrace{A_{mag}\,\overline{\rho}\frac{\widetilde{\varepsilon}}{\widetilde{k}}\widetilde{Y}_{br}}_{decay\ of\ fuel\ eddies},\ \underbrace{A_{mag}\,\overline{\rho}\frac{\widetilde{\varepsilon}}{\widetilde{k}}\frac{\widetilde{Y}_{ox}}{r_f}}_{decay\ of\ oxygen\ eddies},\ \underbrace{A_{mag}B_{mag}\,\overline{\rho}\frac{\widetilde{\varepsilon}}{\widetilde{k}}\frac{\widetilde{Y}_{prod}}{\left(1+r_f\right)}}_{decay\ of\ product\ eddies},\ \underbrace{\overline{R_{br,kin}}}_{kinetic\ term} \right),$$ *Equation 1*

with A_{mag}, *and* B_{mag} as empirical constants of the EDM, the Favre-averaged values of k (turbulent kinetic energy), ε (dissipation rate of turbulent kinetic energy) and Y, the mass fractions of fuel (f), oxidiser (ox) or product (prod) as well as r_f, the mass-weighted stoichiometric coefficient of the fuel.

With the reaction scheme of Brink [1]:

$$CH_4 + 1.5\,O_2 \xrightarrow{r_1} CO + H_2 + H_2O\,,$$ *Equation 2*

$$CO + 0.5\,O_2 \xrightarrow{r_2} CO_2\,,$$ *Equation 3*

$$H_2 + 0.5\,O_2 \xrightarrow{r_3} H_2O\,,$$ *Equation 4*

the oxidation of methane (*r1*) and hydrogen (*r3*) is described with the assumption of fast kinetics and the CO oxidation (*r3*) is calculated with an additional Arrhenius expression (see Howard [3]). The EDM is reasonably accurate for most industrial applications, numerically robust and applicable to premixed, non-premixed and partially premixed combustion but cannot account for strong coupling between turbulence and multi-step chemistry. Another disadvantage is the fact that the empirical parameter A_{mag} of the EDM is not universally valid. In the literature, values between 0.5 and 4.0 are usually proposed. Calculations of a test case as well as the comparison with in-situ measurements of species concentrations and flue gas temperatures at a biomass grate furnace have shown that the originally proposed value of 4.0 (Magnussen and Hjertager [5]) is too high for the designated application (see [6, 9, 8]). A value of 0.6 – 1.0 is proposed for the simulation of biomass grate furnaces. With a value of A_{mag} = 0.6 the reaction rates, especially in the flue gas burnout zone, are slightly under-predicted. In the present case study this value was thus used as safety margin, in order to ensure compliance with the CO emission limit, which is of special importance concerning the design of biomass grate furnaces. Moreover, the predicted CO concentrations should be regarded as trends and not as accurate values. For this reason, the predicted CO emissions which were used as indicators for flue gas burnout are stated in percent of the emission limit.

NO_x formation and reduction were not considered explicitly. However, the furnace was designed under Low-NO_x aspects, i.e., with air staging technology and a temperature controlled primary combustion zone obtained by flue gas recirculation. Moreover, investigations at a pilot-scale furnace (see Weissinger [10] and Scharler [9]) showed a reduction of NO_x emissions due to improved mixing of unburned flue gas with recirculated flue gas. Mixing conditions in the primary combustion zone under oxygen lean conditions were therefore, also taken as an indicator for a NO_x reduction by primary measures. Besides the parameters CO distribution as well as O_2 distribution, calculations of flue gas residence time distributions in the primary combustion zone (as well as in the whole combustion chamber) were made in order to evaluate the mixing quality. To calculate residence times in the combustion chamber, virtual tracer particles, moving with the flue gas, were injected above the fuel bed. The particle trajectories were calculated with a Lagrangian particle tracing procedure (see FLUENT documentation [2]). The calculation of flue gas residence time histograms is valuable especially for the primary combustion zone, where due to oxygen lean conditions, CO concentrations and O_2 distribution supply only limited information.

3. Case study – methodology and explanations

The furnace type investigated (see Figure 1) is designed for Low-NO_x combustion and therefore divided into two combustion zones. Whereas the primary combustion zone is operated as an air lean hot reduction zone with sufficient residence time for the flue gas (0.6 – 0.8 s at nominal power) to reduce NO_x emissions by primary measures, the secondary combustion zone is designed as an air rich burnout zone. A further important feature is the staged flue gas recirculation (flue gas supply below and above the grate) in order to optimise temperature control and turbulent mixing in the primary combustion zone.

In order to perform a case study, a basic geometry of the combustion chamber (see Figure 1) in combination with representative operating conditions (see Table 1) had to be defined as reference case. For this purpose, a geometry of a furnace with a nominal boiler load of 2.1 MW_{th} was designed and analysed through simulation based on results from former CFD simulations of biomass grate furnaces (see [6, 7]). Starting from the reference case, relevant design parameters were summarised in groups (primary and secondary combustion zone) and investigated systematically for one representative fuel (wood chips). Concerning the development of the primary combustion zone, the design of the recirculation nozzles is of special importance and was therefore investigated in detail (arrangement and diameter of the nozzles as well as exit velocity of recirculated flue gas, see section 4. 2). The investigation of the secondary combustion zone was performed in subsections. Besides the design of the secondary air nozzles (arrangement and diameter of the nozzles as well as the exit velocity of the secondary air), double air staging as well as the geometry of the secondary combustion zone were analysed and optimised (see section 4. 3). Based on the results of this sensitivity analysis a concept for Multifuel-Low-NO_x biomass grate furnaces consisting of a group of pre-optimised variations was tested and evaluated for a broader wood fuel assortment with different moisture contents (waste wood, wood chips and bark see section 4. 4).

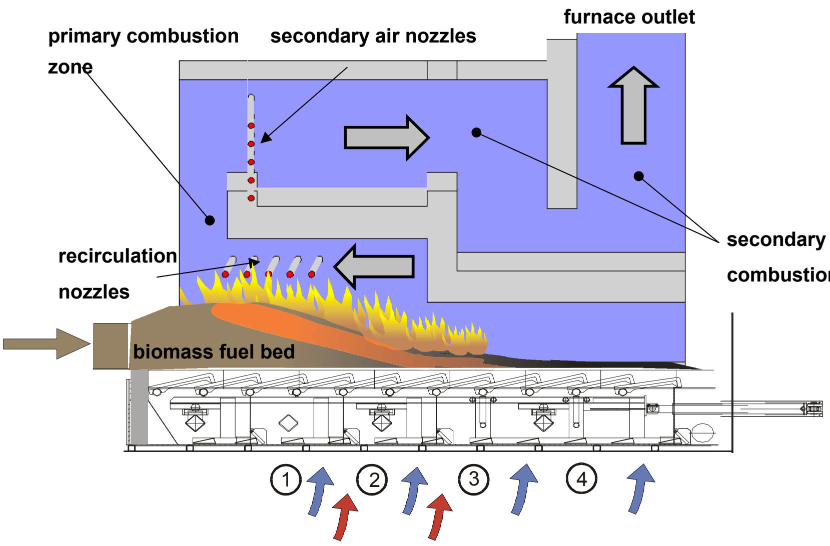

Figure 1: Modelled part of the biomass furnace equipped with a horizontally moving grate (basic geometry)

Explanations: staged air supply below the grate (blue arrows…primary air) and through air nozzles (secondary air); staged flue gas recirculation below the grate (red arrows) and through recirculation nozzles above the grate

Table 1: Representative operating data

Explanations: nominal boiler load; recirculation ratio…mass of flue gas recirculated / mass of total flue gas in the furnace; λ_{prim}…primary air ratio; $\lambda_{fuel\ bed}$…air ratio in the fuel bed based on the amount of primary air and recirculated flue gas supplied below the grate; $\lambda_{eff\ prim}$…effective primary air ratio based on the amount of primary air and the total amount of recirculated flue gas

operating parameter	value unit
nominal boiler load	2,100 kW
fuel	wood chips
gross calorific value	19.5 MJ/kg
net calorific value	12.0 MJ/kg
water content	30 wt% w.b.
adiabatic furnace temperature	995 °C
flue gas recirculation ratio	0.3
ratio flue gas below / above the fuel bed	1.0
lambda prim	0.62
lambda fuel bed	0.75
lambda eff prim	0.89
lambda tot	1.62

4. Discussion of results

4. 1 Reference case

Examples of the flow pattern for the reference case are shown in Figure 2. The injection of recirculated flue gas through nozzles in addition to the supply below the grate resulted in a high turbulent mixing of unburned flue gas and a sufficient utilisation of the furnace volume in the primary combustion zone (CO profiles see Figure 2, residence time distribution see Figure 4, left histogram). Both effects indicate a good NO_x reduction potential. Recirculation swirls due to strong curvatures of the flow channels were especially observed in the secondary combustion zone. Furthermore, the penetration depth of the secondary air jets was sufficient but the air injection led to an uneven velocity distribution and a reduction of the effective hydraulic diameter. Contrary to a satisfactory turbulent mixing of unburned flue gas, a moderate utilisation of the furnace volume in the secondary combustion zone was the result (Figure 2). The flue gas composition (CO concentration see Figure 2) after the secondary air nozzles is sufficiently homogeneous. The calculated CO concentrations at boiler inlet amount to about 65 % of the emission limit (100 mg/Nm^3 dry flue gas related to 13 Vol% O_2). Temperature peaks occur especially between the flue gas recirculation nozzles and the secondary air nozzles (see Figure 2). The maximum values in the near-wall region are between 1,100 and 1,250 °C and restricted to a small area, which is acceptable for wood chip fuel in terms of ash slagging.

As already mentioned, a compact furnace design is one of the key factors for a reduction of investment costs. With the geometry of the reference case, a vertical chamotte duct implemented in the corresponding forerunner model (same nominal boiler load, similar furnace length and width – see [7, 8, 9]) was no longer necessary, but a further optimisation concerning the targets stated in chapter 1 is possible.

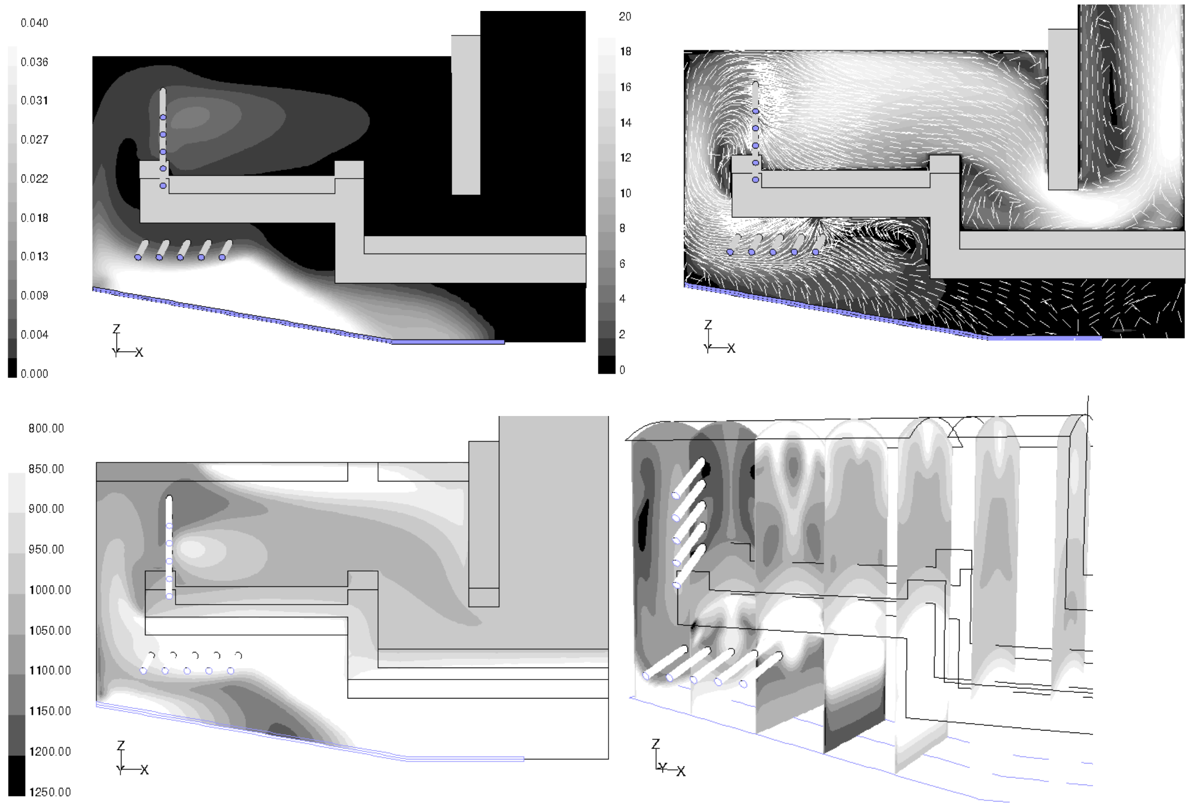

Figure 2: Flow field in the biomass furnace (reference case)

Explanations: top left...CO-profiles in the symmetry plane [ppmv]; top right...flue gas velocity profiles [m/s] combined with vector plots of the flue gas velocity; bottom left...profiles of flue gas temperature in the symmetry plane [°C]; bottom right…profiles of flue gas temperature in various vertical cross-sections (0.5 m steps)

4. 2 Investigation of the primary combustion zone with main emphasis on the design of the flue gas recirculation nozzles

Without staging of flue gas recirculation (same geometry as the reference case, only flue gas supply below the grate), simulation results showed a significantly worse mixing of unburned flue gas with oxygen (see Figure 3, left illustration) compared to the reference case (see Figure 2). Results of the poor mixing quality in the primary combustion zone were a CO strain arising from the fuel bed (see Figure 3, left illustration) as well as increased CO emissions, which were around the level of the emission limit.

Moreover, turbulent mixing as well as the utilisation of the furnace volume (higher residence times of the flue gas, see Figure 4, right histogram) could be improved compared to the reference case by means of an optimised arrangement of the flue gas recirculation nozzles (see Figure 3, right illustration). The nozzles were placed at different locations in the furnace (three nozzle pairs below the ceiling of the first duct and two nozzle pairs at the transition between the first and the second duct) in order to achieve a more even penetration of the unburned flue gas (see Figure 3, right illustration). As a consequence, the CO emissions decreased to about 55% of the emission limit. Moreover, the improved mixing conditions are considered to result in a higher NO_x reduction potential. Furthermore, due to increased reaction rates, the temperature peaks in the near-wall region of the primary combustion zone extended to an area still acceptable for wood chip fuel. A further increased exit velocity of the flue gas jets (fixed amount of recirculated flue gas, reduced number of nozzles) resulted in slightly lower CO concentrations but led to further increased temperature peaks in the primary combustion zone (50 – 150 °C) and therefore was not further investigated in the present case (results are not shown here).

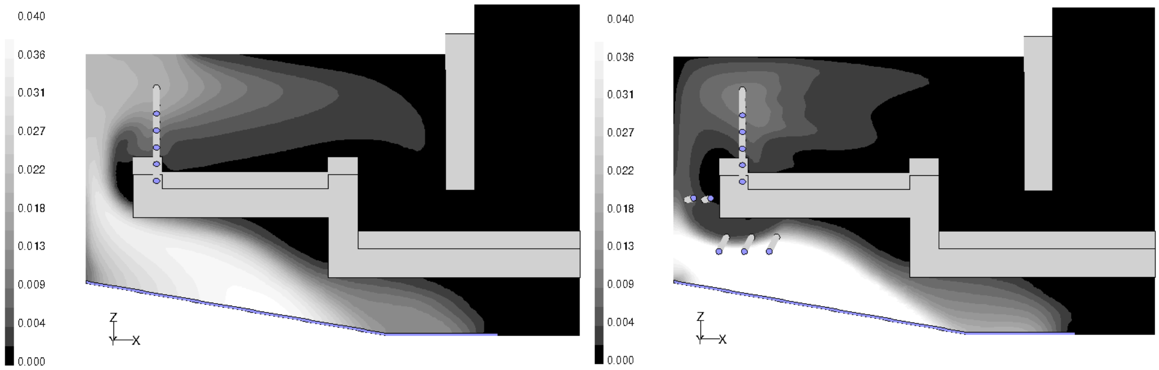

Figure 3: CO profiles [mole fraction] in the symmetry plane of two different furnace variations

Explanations: left figure…results for the variation with supply of recirculated flue gas below the bed only; right figure…results for the variation with the optimised design of flue gas recirculation nozzles

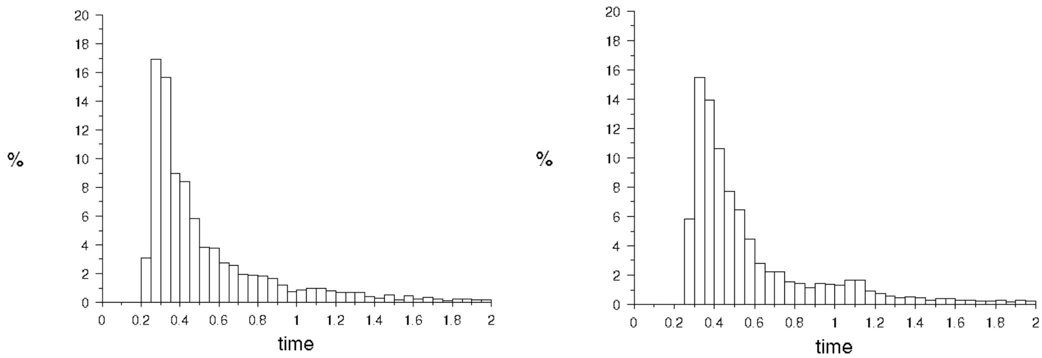

Figure 4: Calculated residence time distribution (RTD) of the flue gas released from the biomass fuel bed in the primary combustion zone calculated for different furnace variations

Explanations left figure: results for the reference case (minimum value: 0.2 s; maximum value: 14.7 s; mean value [0-6 s]: 0.7 s; standard deviation [0-6 s]: 0.9 s;
Explanations right figure: results for the best case of the recirculation nozzle variations (minimum value: 0.3; maximum value: 21.7 s; mean value [0-6 s]: 0.8 s; standard deviation [0-6s]: 0.8 s

In general, improved mixing of unburned flue gas in the primary combustion zone reduces CO and NO_x emissions. An injection of recirculated flue gas above the fuel bed zone where the major amount of flue gas is released is recommended as primary option. However, the jets should not blow into the fuel bed in order to avoid increased fly ash emissions or a disturbance of solid biomass combustion. An optimised flow distribution can be achieved by the installation of different nozzle diameters or by different exit velocities of the recirculation nozzles. The influence of nozzle diameter and exit velocity of the flue gas jet on the flow field as well as recommendations for scaling are discussed in more detail for the secondary air nozzles (see section 4. 2) but are valid for the recirculation nozzles as well. However, a higher turbulent mixing of unburned flue gas with oxygen leads to raised temperature peaks. This should be considered especially in the primary combustion zone, where the possible temperatures are relatively high due to the stoichiometry and composition of the flue gas. Moreover, it should be avoided that the hot reaction front around the flue gas jet hits the furnace walls in order to prevent slagging. A possibility for cooling the furnace walls, especially suitable for round cross-sections of the flow channel, is a flue gas injection tangential to the walls.

4. 3 Investigation of the secondary combustion zone with main emphasis on the design of the secondary air nozzles

Design of the secondary air nozzles

Various designs of secondary air nozzles, aiming at a high turbulent mixing of the unburned flue gas with secondary air as well as an efficient utilisation of the combustion chamber were tested. Based on the results of preceding works (see [6, 7]), the nozzles of the reference variation were symmetrically arranged in vertical rows and were inclined at an angle of 30° to the top of the duct (see Figure 5, left illustration). The nozzles of the best design case (see Figure 5, right illustration) were symmetrically arranged in vertical rows of horizontal nozzles normal to the furnace walls. This variation reduces the CO emissions to about 55% of the emission limit. Due to the shape of the nozzle geometry a secondary flow could be induced and the unburned flue gas was effectively mixed with secondary air. The temperature peaks calculated were in the same range as for the reference variation and are acceptable for wood chip fuel. Both geometries were taken as a basis for the investigation concerning the influence of the exit velocity of the secondary air on flow profiles and mixing conditions in the furnace. By reducing the number of nozzles from 10 to 8 (the nozzle pair in the bottom region of low flue gas mass flow was eliminated) the exit velocity was increased from 30 to 38 m/s. In both cases the higher turbulent mixing resulted in a significant reduction of CO emissions. For the variation with the horizontal nozzles, the trend was more distinctive (reduction from 55 % to 25 % of the emission limit) compared to the reference variation with the nozzles inclined 30° to the top of the flow channel (reduction from 65% to 50% of the emission limit). The explanation can be found in a more distinctive swirled flow, leading to an improved mixing of unburned flue gas with secondary air. For this variation, the temperature peaks in the near-wall region slightly increased, but are acceptable for wood chip fuel. Furthermore, this variation was taken as a basis for the variation of the (inner) nozzle diameter (reduction to the next smaller nominal size) at a fixed exit velocity (38 m/s) of secondary air (i.e. the number of nozzles was increased to 14). As a result, the flow patterns in the secondary combustion zone became a little bit more even, but did not change significantly due to the small variation in nozzle diameter. Moreover, the CO emissions were about 10 % lower and the temperature peaks were slightly lower.

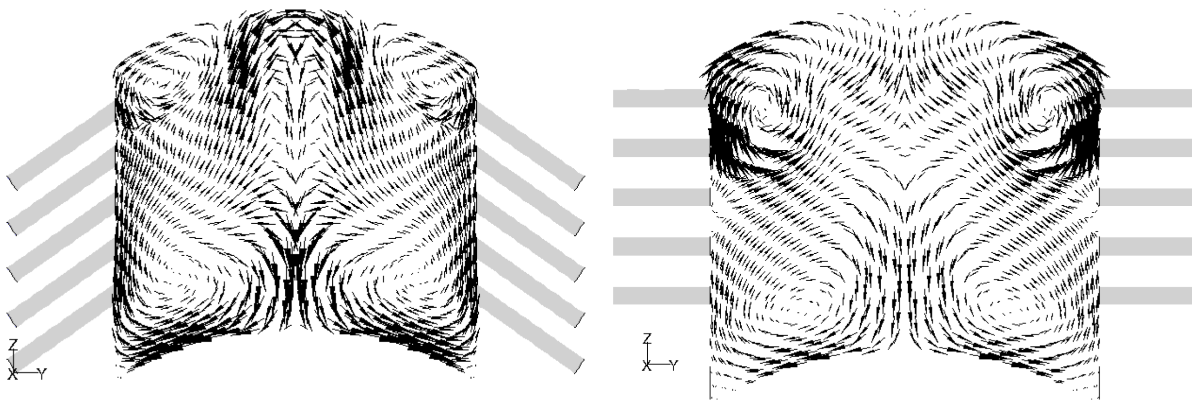

Figure 5: Flow pattern (vector plot) of flue gas velocity in a cross-section immediately after the secondary air jets for two different designs of the secondary air nozzles

<u>Explanations:</u> left figure…vector plot as characteristic flow patterns of the variations with nozzles inclined 30° to the top of the flow channel; right figure…vector plot as characteristic flow pattern of the variations with horizontal nozzles

In general, the design of secondary air nozzles is of great importance in order to optimise both flue gas burnout and the utilisation of the furnace volume. By means of a suitable arrangement of the nozzles the flue gas stream can be homogenised and strains get destroyed. In order to fulfil these demands, the induction of a secondary/swirling flow is highly recommended. A higher exit velocity of the air jets leads to higher mixing rates and lower CO concentrations/emissions but is linked with increasing temperature peaks in the region after the nozzles. The effect of high temperatures can be partially diminished by a swirling flow as well. However, a higher exit velocity is linked with a quadratic increase in pressure loss and therefore, restricted by the power demand of the air fans. In case of a sufficient penetration depth of the secondary air jets, the flue gas distribution after the nozzles is getting more even and flue gas burnout is improved by a reduced nozzle diameter (as well as a higher number of nozzles). Otherwise, a larger nozzle diameter is needed to lower CO emissions as well as temperature peaks due to a complete penetration of the unburned flue gas flow. The optimum nozzle diameter depends on the shape of the flow channel, the nozzle arrangement as well as the exit velocity of the air jets. Therefore, an absolute value cannot be specified. A CFD analysis is recommended in order to determine the optimum nozzle configuration (nozzle arrangement and diameter as well as exit velocity of the secondary air). Simulations at partial load, which are not discussed in this paper, indicated similar flow profiles and species concentrations in case of fixed ratios of flow velocities in the furnace (e.g. exit velocity of secondary air on the one hand and mean flue gas velocity in the flow channel on the other hand). Hence, a scaling of an optimised furnace geometry based on fixed ratios of characteristic Reynolds numbers is recommended.

Double air staging

The best case of investigations concerning the secondary air nozzle design (see Figure 6, left illustration) was used as a basis for a furnace variation in order to study double air staging. The second row of the air nozzles (on both furnace sides) was moved downstream in a horizontal position to the lower edge of the vertical wall (see Figure 6, right illustration). As a consequence, the CO emissions were dramatically reduced to the absolute minimum value of about 10 % of the emission limit. The flow profiles in the secondary combustion zone (see Figure 6, right illustration) were similar to the ones achieved for the basic geometry (optimum secondary air nozzle design see Figure 6, left illustration), but more even between the air nozzles due to the staging of air supply. The flue gas temperatures between the nozzle rows were higher due to the lower amount of excess air in this region. In the particular case, the investigated furnace variation with double air staging was not considered concerning an implementation in an optimised furnace geometry due to the good flue gas burnout already achieved with the reference geometry. The findings of the simulation performed were rather taken as a basis for the simplification of the complete furnace geometry (see section *Geometry of the combustion chamber*).

It is evident that multiple air staging is an efficient measure to achieve an efficient flue gas burnout. Remaining flue gas strains after secondary air injection are efficiently mixed and burned after the supply of tertiary air. The increased residence times in the secondary combustion zones resulting from the locally lower mass fluxes also contribute to an improved flue gas burnout. But increased temperatures in the region between the air nozzles should be considered especially for biomass fuels, which have a tendency to slagging (e.g. waste wood or straw). Besides flue gas recirculation, additional measures like water cooled walls are recommended for temperature control in such cases.

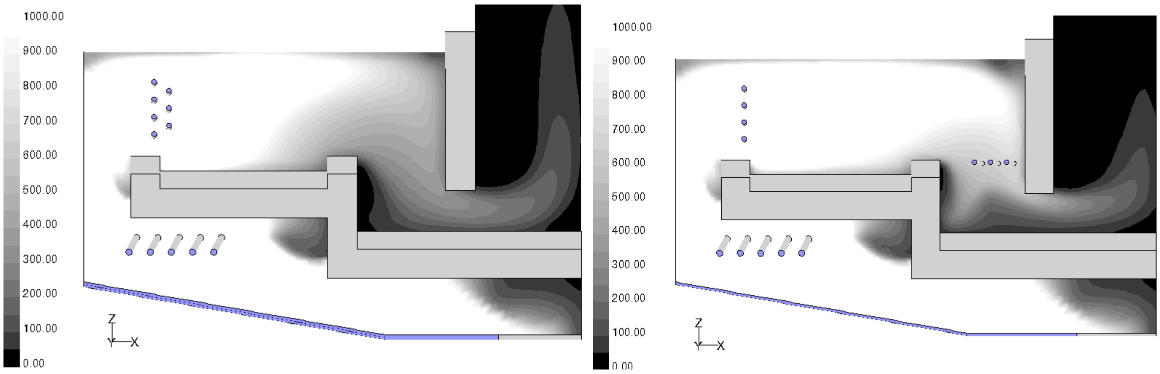

Figure 6: CO profiles [ppmv] in the symmetry plane of two different geometry variations

<u>Explanations:</u> left…results for the best design case of the secondary air nozzles (base geometry for a variation with double air staging); right…results for a variation with double air staging

Geometry of the combustion chamber

The results of the case study already achieved concerning the development of the secondary combustion zone were used as a basis for simplifications of the combustion chamber geometry rather than for a reduction of CO emissions, which are already below the emission limit for the base geometry. After a removal of the vertical wall in front of the entrance into the boiler the CO emissions strongly increased (see Figure 7, left illustration) compared to the reference case (see Figure 2). A variation of the combustion chamber in combination with double air staging and one column of fire bricks after each air nozzle row (see Figure 7, right illustration, and Figure 8) led to excellent results. The flue gas flow, penetrated with secondary air, was divided and mixed at the first pillar, passed to the tertiary nozzles, penetrated with tertiary air and post-mixed at the second pillar. Due to an efficient mixing, the flue gas flow was homogenised (distribution of mass flow, species concentrations as well as flue gas temperature), strains were broken up and CO emissions were reduced to 30% of the emission limit. Furthermore the smaller distance between the air nozzle rows reduced the area of the near-wall temperature peaks compared to the variation concerning the investigation of double air staging and is acceptable for the fuel investigated (wood chips). Another important outcome was that the velocity peaks occurring in the region of the vertical wall (erosion effects) disappeared after this wall had been removed. The variation with secondary air staging and vertical barriers was taken as a basis for the implementation of an optimised furnace geometry (secondary combustion zone) because of the good results achieved,.

It can be stated that an intelligent design of the combustion chamber geometry and mainly the nozzles for flue gas and air injection results in a compact furnace with a minimised volume, low emissions and a low excess air ratio as well as a high availability resulting in low investment and operating costs. An indiscriminate installation of noses and walls for an enhancement of turbulent mixing leads to higher costs for material and maintenance due to an increased complexity of the furnace geometry.

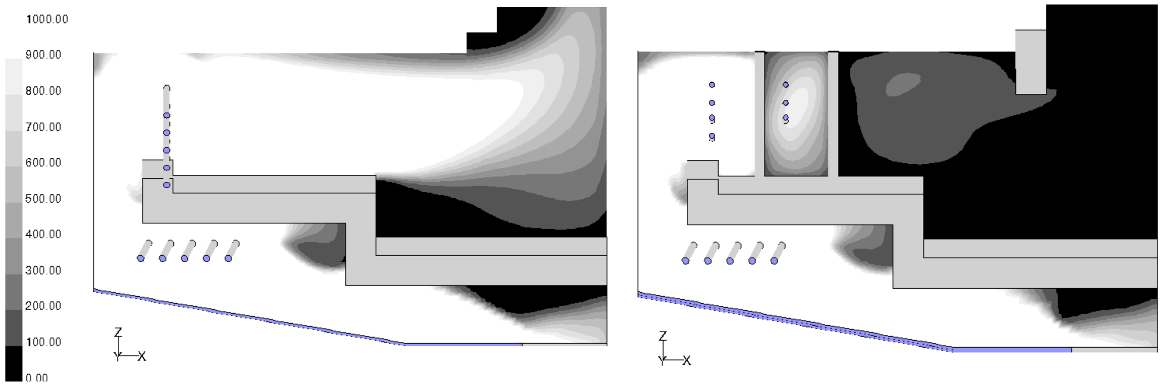

Figure 7: CO profiles [ppmv] in the symmetry plane of two different geometry variations

Explanations: left…results for a variation without a vertical wall in front of the entrance to the boiler; right…results for a variation with double air staging and columns of fire bricks as well as a removed/shortened vertical wall in front of the entrance to the boiler

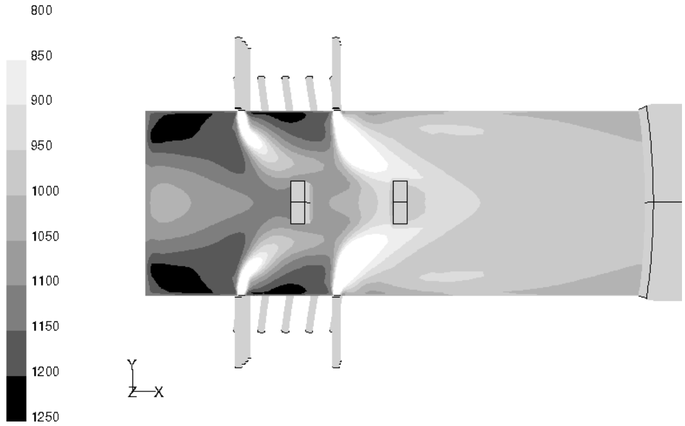

Figure 8: Temperature profiles [°C] in a horizontal cross-section between the secondary and tertiary air nozzles of a furnace variation with double air staging and columns of fire bricks

4. 4 Investigation of an optimised furnace for various biomass fuels

Based on the results of the case studies performed a concept for Multifuel-Low-NO_x biomass grate furnaces consisting of various pre-optimised furnaces was developed. The furnaces are based on the best design cases of the primary (see Figure 3) and secondary combustion zone (see Figure 7, right illustration and Figure 8). Various variations, which can be converted by simple measures were tested for wood chip fuel (combination of the variations see Figure 9). Single air staging or multiple air staging in combination with or without columns of fire bricks inserted in the secondary combustion zone are possible options.

The simulation results for the pre-optimised furnaces showed no significant differences to the flow patterns of the furnace variations taken as design basis in the different furnace sections. A variant with double air staging and without vertical pillars was found to be the best case under technological (mixing conditions, flue gas burnout, temperature distribution) and economic aspects (investment and operation costs as well as availability of the furnace). For this geometry, additional investigations for a very dry fuel (waste wood, water content 10 w% w.b.) and a very moist fuel (bark, water content 50 w% w.b.) were performed under the same operating conditions as for wood chips (nominal boiler load, adiabatic furnace temperature as well as primary and secondary air ratios). For different moisture contents, no significant differences concerning the flow profiles were found.

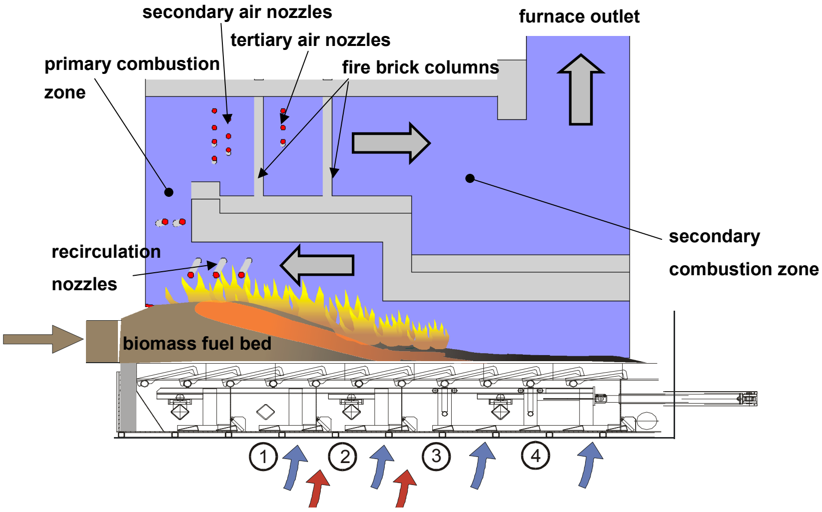

Figure 9: Concept of an optimised Multifuel-Low-NO_x biomass grate furnace

Explanations: staged air supply below the grate (blue arrows…primary air) and through air nozzles (secondary air and tertiary air); staged flue gas recirculation below the grate (red arrows) and through recirculation nozzles above the grate

The CO concentrations and emissions slightly increased with rising water content of the fuel (not shown here), but were acceptable for the furnace investigated. Higher CO emissions for moist fuels can be explained by a lower effective air ratio as well as a smaller amount of recirculated flue gas injected above the grate (deterioration of turbulent mixing). The injection of an increased amount of recirculated flue gas through nozzles is recommended as an efficient strategy for improving flue gas burnout. An increased power demand of the flue gas fans has to be considered, yet the amount of secondary air can be reduced and therefore, a higher thermal efficiency can be achieved.

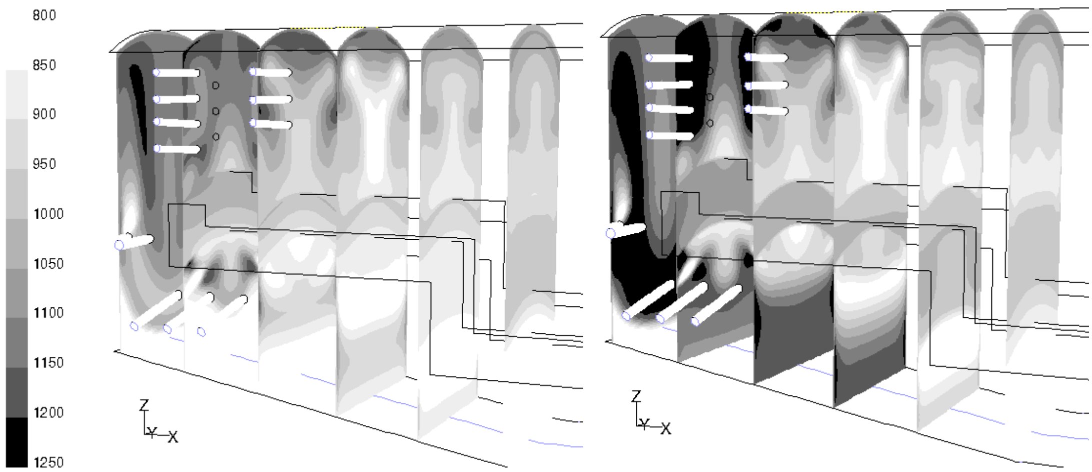

Figure 10: Temperature profiles in vertical cross-sections (0.5 m steps) of the optimised furnace geometry for a dry fuel (waste wood) and a wet fuel (bark)

Explanation: left figure…results for bark, water content 50 wt% w.b. as fuel; right figure…results for waste wood, water content 10 wt% w.b. as fuel

The furnace temperatures in the region up to the tertiary air nozzles raised with decreasing water content due to the higher amount of oxygen (from the recirculated flue gas) available as well as due to the high heat capacity of the water which lowers temperature peaks (see Figure 10). For dry fuels with a tendency to slagging (e. g. waste wood, straw) the temperatures, especially in the primary combustion zone, are too high. Additional measures besides flue gas recirculation are therefore recommended for such fuels. One simple measure is the reduction of the primary air ratio (if possible), in order to lower the temperature level in the primary combustion zone, since a modification of the operation conditions is not sufficient in most cases, wall cooling with water or thermo-oil in the high temperature regions up to the secondary air nozzles is suggested as an efficient method.

5. Summary and Conclusions

A concept for optimised Multifuel-Low-NO_x biomass grate furnaces in the size range of 0.5 – 10 MW_{th} was developed using CFD analysis. This concept is based on a combustion chamber with two horizontal chamotte ducts. The furnace can be operated with single or double air staging and has the additional option that a fire brick column can be inserted after each group of air nozzles. A staged flue gas recirculation above and below the grate in order to achieve improved mixing conditions in the primary combustion zone (flue gas burnout, NO_x reduction) is a further important feature of the plant. A first prototype was implemented as a pilot-scale furnace and is being successfully tested. Besides the combustion plant developed, the systematic investigation of relevant geometric parameters, which were grouped into modules, allows further combustion chamber geometries to be created depending on the application. Finally, guidelines for the design of biomass grate furnaces were derived from the CFD analysis performed.

In general, a considerable reduction of investment and operating costs is possible by a compact furnace design as well as by reduced air and flue gas fluxes in the furnace. This can be achieved by an appropriate design of the nozzles for air and flue gas injection as well as by adjusting the geometry of the combustion chamber (e.g. barriers) in order to optimise the mixing of unburned flue gas and air as well as the utilisation of the furnace volume. The injection of recirculated flue gas above the fuel bed zone, where the main amount of unburned flue gas is released, is recommended for lower CO and NO_x emissions by improved mixing conditions. However, the jets should not blow into the fuel bed in order to avoid increased fly ash emissions or a possible disturbance of solid biomass combustion. An optimisation of the mixing quality in the primary combustion zone is possible by the injection of recirculated flue gas through nozzles with various nozzle diameters and/or exit velocities. A higher turbulent mixing of unburned flue gas may lead to raised temperature peaks, which should be considered especially in the primary combustion zone, where the possible temperatures are relatively high due to stoichiometry and composition of the flue gas. The secondary air nozzles should be arranged so as to induce a secondary/swirled flow in order to homogenise the flow and to effectively utilise the furnace volume. With an increased exit velocity of the jets, penetration of unburned flue gas and turbulent mixing are improved, but the disadvantages of higher temperature peaks as well as a raised power demand of the corresponding fans have to be considered. If the penetration of unburned flue gas is sufficient, a reduction of the nozzle diameter (or an enhanced number of air nozzles) has the advantages of a higher mixing rate and more even flow profiles. Otherwise, the nozzle diameter must be enlarged in order to achieve a complete penetration of the flow, linked with lowered temperature peaks and a good flue gas burnout. The optimum nozzle diameter therefore depends on the nozzle configuration (arrangement, exit velocity of the jets and shape of the flow channel) and should be determined using CFD analysis. Moreover, a staged secondary air supply is recommended as a powerful method for a distinct reduction of CO emissions. However, this results in increased flue gas temperatures between the nozzles for secondary and tertiary air injection. Finally, a scaling of an already optimised furnace geometry should be based on approximately constant ratios of the characteristic Reynolds numbers.

In general, the furnace temperatures in the region up to the secondary (tertiary) air nozzles rise with decreasing water content of the fuel due to the higher amount of oxygen available as well as due to the high heat capacity of the water which lowers temperature peaks. Higher flue gas temperatures due to improved mixing conditions especially for dry fuels like waste wood, have therefore to be considered concerning ash slagging and deposit formation. Additional measures like wall cooling with water or thermo-oil, enhanced flue gas recirculation as well as a lower amount of primary air are recommended for the lowering of furnace temperatures. Furthermore, CO emissions increase with rising water content of the fuel. This can be explained with a lower effective air ratio as well as a smaller amount of recirculated flue gas injected above the grate (worse turbulent mixing). For very moist fuels like bark a higher amount of recirculated flue gas injected in the combustion chamber instead of an increased excess air ratio is recommended as an efficient strategy in order to improve flue gas burnout.

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Nomenclature

ε	dissipation rate of turbulent kinetic energy [m²/s³]
ρ	density [kg/m³]
A_{mag}	empirical constant of the Eddy Dissipation Model
B_{mag}	empirical constant of the Eddy Dissipation Model
k	turbulent kinetic energy [m²/s²]
R_{br}	rate of fuel consumption [kg/m³s]
$R_{br,kin}$	rate of fuel consumption, kinetic rate [kg/m³s]
r_f	stoichiometic coefficient fuel
Y	mass fraction [-]

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