Heat Transfer in Furnaces under Oxyfuel Combustion Conditions

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Abstract: - Oxyfuel technology is a promising option for carbon dioxide capture in greenfield as well as in retrofit power-plants. In the oxyfuel process the coal is burned in an atmosphere of pure oxygen and reduced flue gas. The resulting concentrations of CO_2 and H_2O in the flue gas exceed levels of conventional air firing by 3 to 4 times. This implies changes in radiant heat transfer.
The present paper evaluates the behaviour of a two zone furnace heat transfer model under oxyfuel conditions. Results will show heat transfer tendencies for a typical oxyfuel flue gas composition.

Key-Words: Oxyfuel combustion, Heat transfer, Radiative heat transfer, Convection, Flue gas recirculation

1 Introduction and problem formulation

Although the oxyfuel process has a similar process layout as conventional power plants, Oxyfuel combustion needs considerable rates of flue gas recirculation to reduce the adiabatic flame temperature to levels comparable to air combustion. This paper evalutes the impact of oxyfuel combustion on the heat transfer in a coal fired furnace. The heat transfer in a pulverized coal fired furnace is dominated by flame and gas radiation. The emissivity of the flue gas components has been quantified by [1] and [2] depending on the partial pressure and the temperature. But the levels of H_2O and CO_2 concentrations resulting from oxyfuel combustion can be so high that they are out of the range covered by these measurements. Therefore calculations need to use extrapolation equations for the gas emissivities of [3]. The furnace behaviour will be evaluated by a two-zone furnace model. A sensitivity analysis will account for the uncertainties arising from the extrapolations. Finally the implications on the heat transfer in the convective heating surfaces will be presented.

2 Greenfield and retrofit oxyfuel plants

The process layout of an oxyfuel-process is similar to that of a conventional plant enhanced by an air-separation-unit (ASU) and a CO_2-compression-and-purification-unit (CO_2-CPU). Because no changes in the water-steam cycle are required a retrofit of conventional plants to oxyfuel-combustion conditions should be possible. Coal combustion in an almost pure oxygen atmosphere would result in an adiabatic flame temperature of approximately 3500 °C. In order to maintain furnace wall temperatures at their design temperature a recirculation of cooled flue gas is necessary.

In a greenfield oxyfuel boiler the flue gas recirculation can be designed to the same adiabatic firing temperature as in an air fired boiler. Due to the high CO_2 and H_2O fraction in the flue gas, its heat capacity is about 20% higher than that of preheated air. Thus the flue gas flux will be much lower and to ensure appropriate velocities in the convective heat exchangers the boiler section has to be smaller, reducing capital costs.

The percentage of heat duty transferred in the furnace results from three main parameters: flame emissivity (A), flue gas heat capacity (B) and difference between adiabatic and furnace exit temperature (C). In a retrofit case the relative heat duty in the different sections of a steam generator needs to be kept constant in order to stay within metal temperature design limits. As a consequence, and given that both (A) and (B) increase due to increased CO_2 and H_2O contents, the furnace temperature difference (C) needs to be decreased. The only way to obtain a decrease of (C) is to decrease the adiabatic temperature. This is the reason why in most oxyfuel retrofit applications the adiabatic flame temperature needs to be decreased compared to the air case. In oxyfuel greenfield applications the steam generator can be designed to a different duty split between radiative and convective sections.

The current work is based on a process simulation of a once through single pass boiler, fired with hard coal. In addition to an overall calculation of the boiler performance a detailed analysis of the furnace was carried out.

3 Furnace modelling

The combustion and heat transfer phenomena in utility boiler furnaces are complex. Models of different complexity are in use. Zero dimensional models of the "stirred reactor" type have the disadvantage that they do not take into account temperature gradients within the furnace. Three dimensional CFD models are appropriate for detail analysis but less for overall system analysis. For this study the flame- and radiation-zone model (FRZM) as presented in [4] is used. This model subdivides the furnace into two sections: the flame-zone and the radiation-zone (Fig. 1). The flame zone is filled completely by the flame and the heat exchange is dominated by flame radiation. It is further assumed that the fuel is completely burned at the entrance of the radiation zone and that no back-radiation from the radiation-zone into the flame-zone occurs. In the radiation-zone the heat-exchange is dominated by gas- and particle radiation.

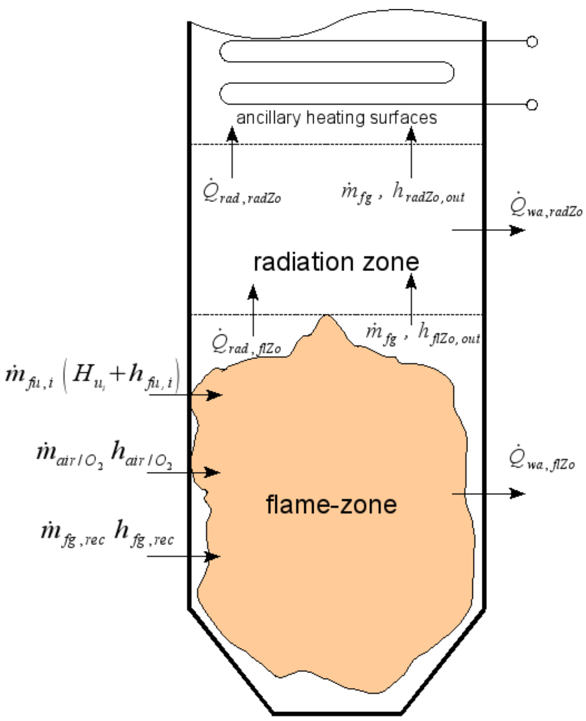

Fig. 1: flame- and radiation-zone of the furnace-modell

To calculate the transferred heat in the flame-zone as well as the flame-zone exit temperature the following parameters have to be determined:

- the emissivity of the flame
  $\varepsilon_{fla} = f(l_{mb}, fuel, p_{CO_2/H_2O})$
- the emissivity of the wall
  $\varepsilon_{wa} = f(material_{wa}, T_{wa})$
- the fouling factor
  $f_{foul} = f(fuel)$

The value of the flame emissivity coefficient is, according to [5], set to 0.81. In the first approach this value will be kept constant between air- and oxyfuel operation. But because available data on the flame emissivity did not consider oxyfuel conditions, an increase of 20% of the flame emissivity is studied as well.

The approximation of the wall temperatures was based on the steam temperature at the furnace outlet, which originates from the overall simulation of the boiler, mentioned above. In the flame-zone the wall temperature was set to 5 K above the steam exit temperature. In the radiation-zone the wall temperature was set to 15 K above the steam exit temperature. These calculation wall temperatures in both sections were kept constant for all calculations. The real wall temperature are of course higher. In the two zone model this is accounted for by the fouling factor. The emissivity of the wall was specified according to [3] to 0,81. The value of the fouling factor was, based on empirical data, set to 0,7. Using the equation for the mean flame-zone emissivity (1) and the specific adiabatic enthalpy (2), the exit temperature from the flame-zone can be calculated from (3) and (4).

$$\bar{\varepsilon}_{fla} = \frac{1}{\frac{1}{\varepsilon_{fla}} + \frac{1}{\varepsilon_{wa}} - 1} \tag{1}$$

$$h_{FlZ,i} = \frac{\sum \dot{Q}_{FlZ,i}}{\dot{m}_{fg}} = \dot{m}_{fg}{}^{-1}\left[\dot{m}_{air}h_{air} + \dot{m}_{fg,rec}h_{fg,rec} + \sum_{i=1}^{k} \dot{m}_{fuel,i}\left(H_{u_i} + h_{fuel,i}\right)\right] \tag{2}$$

$$T_{fla,out} = \sqrt[4]{\frac{\dot{m}_{fg}h_{FlZ,i} - h_{FlZ,o}}{\bar{\varepsilon}_{FlZ} f_{foul} \sigma_{rad} A_{wa,F} + A_{FlZ,out}} + T_{wa,F}^4} \tag{3}$$

$$h_{FlZ,o} = \bar{c}_{p,f}\vartheta_{FlZ,o} \tag{4}$$

The heat transferred within the flame-zone walls can be calculated according to [4].

In the radiation-zone the emission and absorption coefficients of the flue gas suspension have to be determined to calculate the heat transfer coefficient.

$$\alpha_{RaZ} = f(\bar{T}_{fg}, \bar{T}_{wa,RaZ}, \varepsilon_{fg}, \alpha_{fg}, \varepsilon_{wa}) \tag{5}$$

The emissivity of the flue gas itself depends on the content of three or more atomic molecules. The relations between gas temperature, mean beam length and partial pressure of CO_2 and H_2O are based on measurements, see [1] and [2]. Because the CO_2 and H_2O concentrations under oxyfuel operation exceed the boundaries, for which experimental validated relations for the determination of the emissivities exist, the data have to be extrapolated. For this study the equations presented in [3] were used. The emissivity of the gas composition was calculated as the sum of the emissivity of each separate component, reduced by the band overlap. The inaccuracy caused by the extrapolation is accounted by a variation of the resulting values in the range of ±20%. Finally the emissivity of the fly ash was considered according to [6]. The gas-temperature at the exit of the radiation-zone has to be calculated from equation (8) using the temperature dependence of the heat transfer coefficient (6).

$$\alpha_{rad} = \frac{2\varepsilon_{wa}\psi}{1+\varepsilon_{wa}\psi} \frac{\sigma_{rad}\left[\varepsilon_g \bar{T}_g^4 - \alpha_{g,\mathrm{abs}} T_{wa,RaZ}^4\right]}{T_g - T_{wa,RaZ}} \tag{6}$$

$$\bar{T}_g = 273{,}15K + \frac{\vartheta_{FlZ,out} + \vartheta_{RaZ,out}}{2} \tag{7}$$

$$\vartheta_{RaZ,out} = \left(\bar{c}_{p,g} + \frac{A_{wa,RaZ}}{2\dot{m}_g}\alpha_{RaZ}\right)^{-1} \Big(h_{RaZ,\mathrm{in}} - \frac{A_{wa,RaZ}}{\dot{m}_g}\alpha_{RaZ} 0{,}5\vartheta_{FlZ,out} - \vartheta_{Wa,RaZ}\Big) \tag{8}$$

The calculation of the heat transfer coefficients to the wall surfaces of the radiation-zone and to the successive heat exchangers is described detailed in [4].

4 Parameters of the analyzed furnace

The dimensions of the furnace are depicted in Fig. 2. While the ash hoppers volume is not taken into account it's projected surface is included in the surface calculation.

The subdivision between flame- and radiation zone was done one vertical distance between two burner levels above the top burner level.

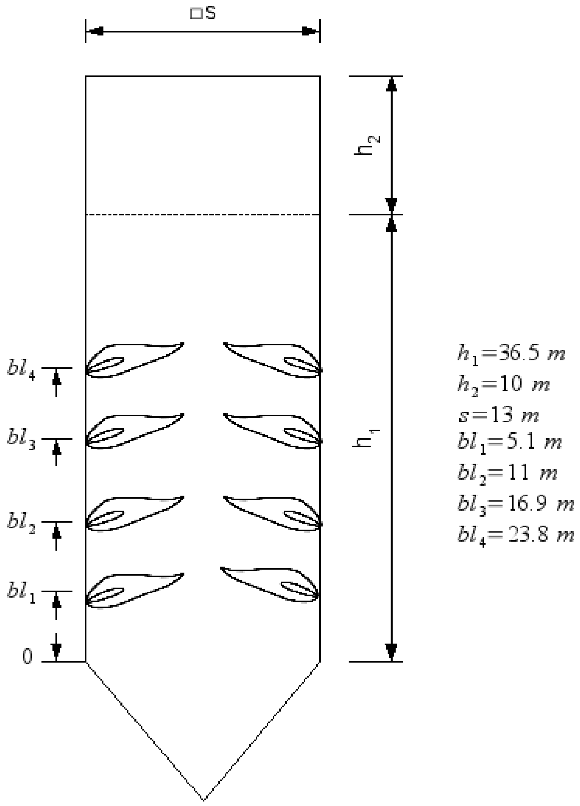

Fig. 2: furnace dimensions

Flame-zone		
Burner level 1 bl_1	m	5,1
Burner level 4 bl_4	m	23,8
Cross section	m^2	619
Volume	m^3	6168
Mean beam length	m	10
Radiation-zone		
Height h_2	m	10
Cross section	m^2	169
Volume	m^3	1690
Mean beam length	m	6,7

Table 1: Geometry of the furnace

The particle emissivity was calculated according to [6].

The flue gas exit temperature of the radiation zone was calculated iteratively using equations (6) and (8). The temperature dependency of the emissivity-coefficients was considered and therefore its calculation was embedded in the iterative loop. To consider uncertainties arising from the extrapolation of the gas emission coefficients, the calculated gas-emissivity was varied by +/- 20%. The objective of the variation is to point out the influence of the gas emissivity. That is why the flue gas recirculation was not adjusted to the impact of this variation. All four oxyfuel-variations share the same flue gas compositions (Tab. 2).

		air	oxyfuel
Flue gas massflow	kg/s	368,0	323,5
Recirculated flue gas	kg/s	8,0	222,7
Spec. enthalpy	kJ/kg	1430	1624
Volumetric flow	m³/s	1508	1199
Density	kg/m³	0,2	0,3
Heat capacity cp	kJ/kgK	1,3	1,6
Gas composition:			
CO_2	% vol	13,8	49,4
N_2	% vol	73,9	6,3
O_2	% vol	3,9	4,9
H_2O	% vol	7,4	37,3
Dust charge	g/nm³	16,6	20,9

Table 2: flue gas properties at the exit of the flame-zone

One air and four oxyfuel cases were studied:
Case 1: air
Case 2: oxyfuel: uses the same flame emissivity as in 1. and the calculated gas emissivity without correction
Case 3: (ε_{gas} +20%) considers, compared to 1. and 2., a rise in the flame-emissivity by 20%.
Case 4: In (ε_{gas} +20%) the calculated gas emissivity is increased by 20% while the flame emissivity is kept at the value of 1. and 2.
Case 5: (ε_{gas} -20%) is similar to case 4 but the calculated gas emissivity is reduced by 20%.

Flame-zone		Case 1	2	3	4	5
T_{ad}	°C	1970	1890	1890	1890	1890
ε_{wall}	--	0,81	0,81	0,81	0,81	0,81
ε_{flame}	--	0,81	0,81	**0,97**	0,81	0,81
$f_{fouling}$	--	0,7	0,7	0,7	0,7	0,7
T_{wall}	°C	408,8	399,4	399,4	399,4	399,4
$T_{gas\text{-}out}$	°C	1282	1237	1207	1237	1237
Radiation zone						
$\varepsilon_{H2O\text{-}CO2}$	--	0,34	0,53	0,53	**0,63**	**0,42**
ε_{dust}	--	0,35	0,43	0,43	0,43	0,43
T_{wall}	°C	418,8	409,4	409,4	409,4	409,4
$T_{gas\text{-}out}$	**°C**	**1197**	**1153**	**1134**	**1142**	**1163**

Table 3: Calculated heat transfer parameters

5 Results and discussion

In order to obtain in the oxyfuel operation the same steam temperatures as in air operation approximately 70% of the flue gas is re-circulated to the furnace (Fig. 3). This reduces the adiabatic flame temperature in oxyfuel case by 80 K compared to the air fired operation (case 1 and case 2).

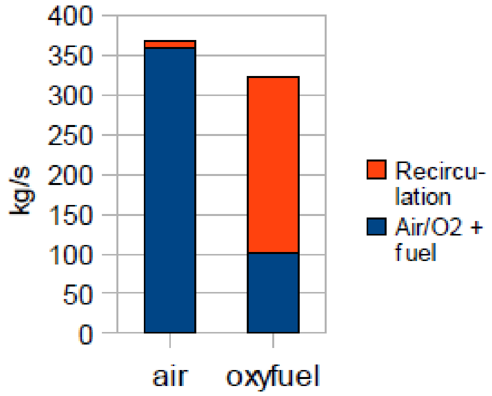

Fig. 3: Flue gas mass flow

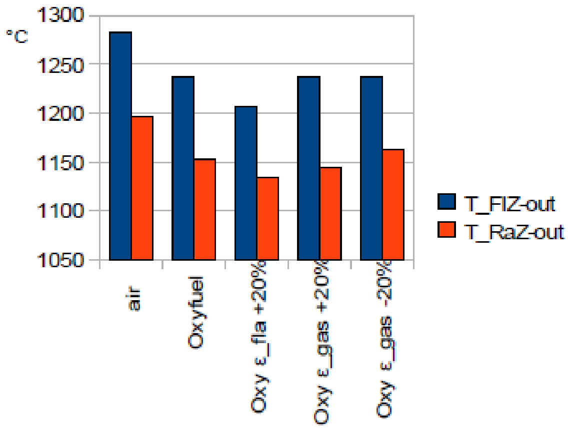

Fig. 4: Gas temperature at the flame and radiation zone outlet of the furnace

As shown in Fig. 4 the furnace exit temperature drops by 40 K from case 1 to case 2.

In all convective heating-surfaces the flue gas temperature is in oxyfuel operation lower than in air operation. The main conclusion is that the furnace exit temperature ($T_{RaZ,out}$) is not very sensible to uncertainties in the determination of emissivity coefficients. From case 2 to case 5 the dispersion of calculated furnace exit temperature remained in a window of 29 K, which is a rather low value. Still the heat absorption in the convective heating surfaces equals or exceeds the values attained at air operation. This surprising effect is related to the changed composition of the flue gas. First the specific heat capacity increases by 20%, due to which the total enthalpy flux remains constant although the mass flux decreases by 12% (Fig. 5). Second the radiant heat transfer coefficient increases as presented in the furnace calculation (Fig. 6).

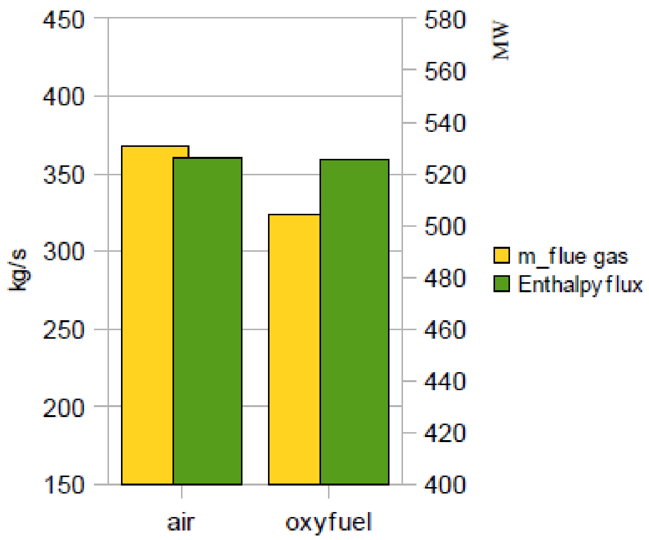

Fig. 5: Flue gas and enthalpy flux

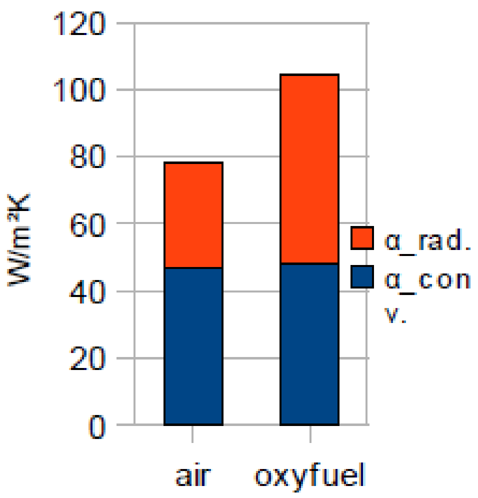

Fig. 6: Heat transfer coefficient in the lower part of the second reheater

6 Conclusion

The high CO_2 and H_2O concentrations in the oxyfuel flue gas have a significant effect on heat transfer. Variations of the emission coefficients calculated according to [6] showed that the resulting spread in furnace temperatures is small and well below the impact which can be generated by variations in fouling. As a conclusion in the analyzed steam generator no major design changes of pressure parts are necessary to permit an oxyfuel retrofit. Flue gas recirculation in combination with spray cooling gives sufficient degrees of freedom to can adjust for uncertainties which arise the calculation of a emission coefficient. In most retrofit cases, conversion to oxyfuel should be possible without modification of pressure parts.

It has to be pointed out that the current paper focuses only on the heat transfer. To ensure reliable combustion, burner modifications are necessary.

Further investigations and measurements of CO_2 and H_2O emissivities at high partial pressures will be necessary for an accurate design of a greenfield oxyfuel power plant. At greenfield power plants the cross section of the flue gas ducts can be reduced to attain higher adiabatic flame temperatures. Higher adiabatic flame temperatures reduce the required mass flux of flue gas, which results in lower material expenses. In this regard wet bottom and circulating fluidized bed furnaces could outperform dry bottom furnaces.

7 Nomenclature

A	Surface area [m^2]
$\bar{c}_p$	Spec. heat capacity [-]
h	Spec. enthalpy [J/kg]
H_u	Lower heating value of the fuel [kJ/kg]
$\dot{m}$	Mass flux [kg/s]
m_{bl}	Mean beam length [m]
p	Pressure [Pa]
$\dot{Q}$	Heat flux [kW/kg]
T	Temperature [K]
α	Heat transfer coefficient [W/m^{2}K]
ε	Emissivity [-]
ϑ	Temperature [°C]
σ_{rad}	Stefan Boltzmann coefficient [W/m^{2}K^4]
ψ	Geometrical sight factor [-]

Indizes:

ad	Adiabatic
air	Ambient air
conv	convective
fg	Flue gas
fla	Flame
FLZ	Flame zone
foul	Fouling
g	Gas
in	Inlet
O_2	Oxygen from ASU
out	Outlet
rad	Radiation
RaZ	Radiation zone
rec	Recirculation
wa	Wall

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