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Change of existing circulating fluidized bed boilers to oxy-firing conditions for CO_2 capture

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ARTICLE INFO	ABSTRACT
	This work investigates a circulating fluidized bed boiler, originally designed for air-firing, retrofitted to oxy-firing with the purpose of removing the CO_2 emission from coal combustion. Previous studies have shown that the heat balance on the gas-particle side can be satisfied without changes in the boiler, but then the volume flow of gas is reduced. To retain the operation like that during air-firing, the volume flow, that is the fluidization velocity, in oxy-firing should be equal to that in air-firing. It is the main purpose of this work to determine the conditions for the transition from air to oxy-firing, while the heat transfer conditions are maintained at a constant fluidization velocity. Measures to achieve this, such as adjusting the supply of additional gas and the heat transfer surface, are analysed. The fulfilment of the furnace's heat balance requires extra fuel or reduction of the heat-transfer surface in the furnace. These changes affect the performance of the back pass, which must be modified to accommodate the change in gas composition and the higher sensible heat content of the flue gas. Strategies to deal with these circumstances in CFB boilers are discussed.

1. Introduction

In some countries, decisions are taken to reduce the use of coal, while coal still is an important part of the energy supply and coal combustion will continue. Then, the question of CO_2 capture from combustion becomes of great significance, and the development of the related technologies is urgent. One such technique is oxy-fuel combustion. Its application to existing circulating fluidized bed (CFB) boilers, originally designed for air combustion, is investigated in this work.

Oxy-fuel combustion, combustion in oxygen with a gaseous moderator added to replace the air-nitrogen, has been studied thoroughly [1,2, 3]. In a new design of an oxy-combustion CFB boiler there is a freedom to arrange the fluidization velocity and the heat transfer surfaces to suit the oxygen concentration in the added gas, and to choose the size of the furnace. Beside this case, oxy-fuel can be applied in two other ways for CO_2 capture in CFB: 1) The "retrofit case", an existing air-fired CFB boiler is transformed to oxy-combustion with the least possible modification, and 2) The "ready to convert case", a new boiler is built to be operated with either oxy-combustion or air-combustion.

In the latter cases mentioned, there are some restrictions to consider, as was already mentioned in a previous analysis [4]. Because of the different physical properties of nitrogen and the substitute gas, the oxygen concentration in the input gas must be adjusted to achieve the same operation condition as with air (under the conditions of the same fuel supply and heat extraction) when converting an existing CFB boiler from air to oxy-firing. The suitable oxygen concentration can be determined by a heat balance on the furnace. It was found [4] that the fulfilment of the heat balance results in a lower volume flow in the retrofit oxy-fuel case, and then, consequently, of a lower fluidization velocity. The reduction of the volume flow is known from [4] and from similar investigations on pulverized fuel (PC) boilers [3]. In the latter type of boiler, it can be handled by modifying the burners. In contrast, in CFB boilers, it disturbs the operation, since the reduced velocity affects the vertical solids distribution and concentration. The particle concentration, in turn, affects the heat transfer.

The purpose of the present work is to extend the previous work [4] by: (i) determining the conditions for the transition from air to oxy-firing by analysing different strategies to satisfy the heat and volume requirements in existing CFB boilers built for combustion with air; and (ii) discussing the modifications needed in the furnace and in the back pass to achieve the new operation condition. Specially investigated are the adjustment of the volume flow (the fluidization velocity in the furnace)

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to attain the desired similarity with air and the impact of the circulating bed particles on the heat balance. The focus is on the boiler itself, assuming that oxygen production and CO_2 compression can be handled.

2. Modelling

2.1. Systems considered

The commonly proposed flue-gas recirculation, either dry (if the water is removed from the gases) or wet (if water is not removed), is the principal arrangement investigated. For comparison, an arrangement where the flue-gas recirculation is replaced by water injection is also studied. Water injection to replace nitrogen is simpler than flue-gas recirculation and has other advantages, but also disadvantages, such as the energy and exergy demand for evaporation of the water and then the difficulty in utilizing the same energy amount removed as low-quality energy (low exergy) in the condensation of the flue gas when removing the water before the CO_2 compression. The three systems concerned are shown in Fig. 1.

2.2. Assumptions and specifications

The purpose of the calculations is to determine the basic features of the processes rather than including the multitude of details that characterize CFB boiler design. To simplify, the following assumptions are made: No air ingress; complete burnout, that is, a combustion efficiency of 100%; no impurities in the added oxygen; the temperature of the gas entering the furnace is taken as reference; the impact of the sensible heat of the ash and the heating of the fuel (the sensible heat counted from the input temperature to the reference temperature) are small quantities and can be neglected; heat losses from the boiler are not included in the heat balance; calculations, in some cases iterative, are run to an error of <1%; the furnace is regarded a well-mixed reactor for particles, because of its considerable back-mixing of particles and fast gas-particle heat transfer. This means that the bed (furnace) temperature is constant in the furnace and that there are no variations over its cross section.

2.2.1. Fuel and media properties

The fuel flow is the same and, thus, cancelled out in the comparison between the air and oxy cases (see Eq. (9)). Therefore, for simplicity, the combustion calculations refer to a combustible fuel flow of $F = 1$ kg/s. Fuel data are given in Table A1, found in Appendix A. The data represent typical bituminous coal and lignite. The property data for gases and solids are presented in Table A2, Appendix A.
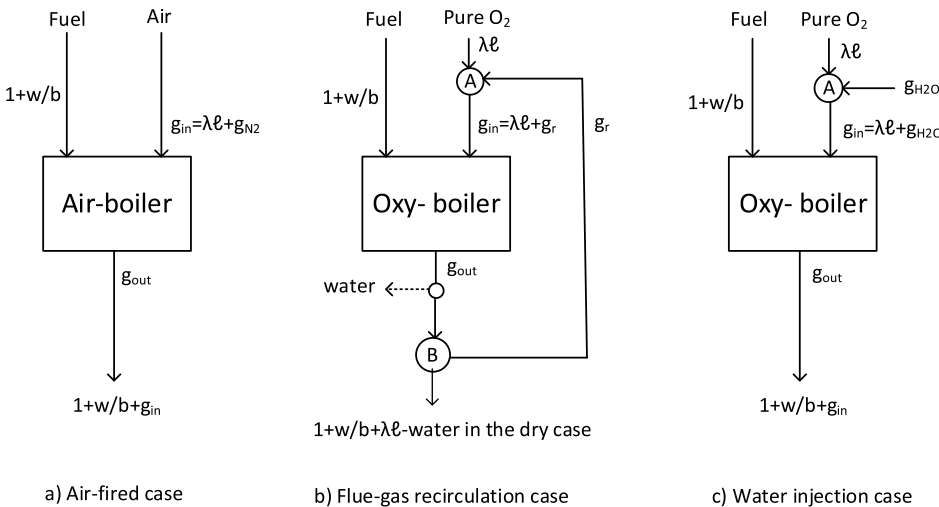
2.2.2. Boiler data and operational specifications

It is assumed for the air case: fluidization velocity $u = 5$ m/s, bed temperature $T_f = 850\,°C$, and global oxygen-excess factor $\lambda = 1.10$. These data are commonly used in CFB boilers except, possibly, the oxygen-excess factor that is low and requires a well-developed control system or an even operation. In an oxy-fired boiler operating with flue-gas recirculation, a low over-all oxygen excess is reasonable because the contribution of oxygen from the recirculating loop will increase the oxygen supply to the furnace [4]. First, a heat balance on the furnace will specify the oxygen concentration in the input gas $x_{O2,in}$ that makes the air and oxy-combustion cases equivalent from the thermal stand-point. Two operational modes are considered (Fig. 1): either the moisture in the flue gas is condensed before recirculation or wet gases are recirculated. There are another two cases to treat: operation with or without an external heat-exchanger (EHEX). The entrance temperature of the circulating particle-flow differs depending on how much heat that is extracted in the EHEX, Q_{ext}, see Fig. 2. If there is no EHEX (the usual case for boilers with a power of < 300 MW$_e$), the entrance temperature of the solids to the furnace is equal to their exit temperature $T_{s,in} = T_{s,out}$ that is, there is no temperature change in the particle flow in the circulation loop. The particles play a role locally, though, in terms of a
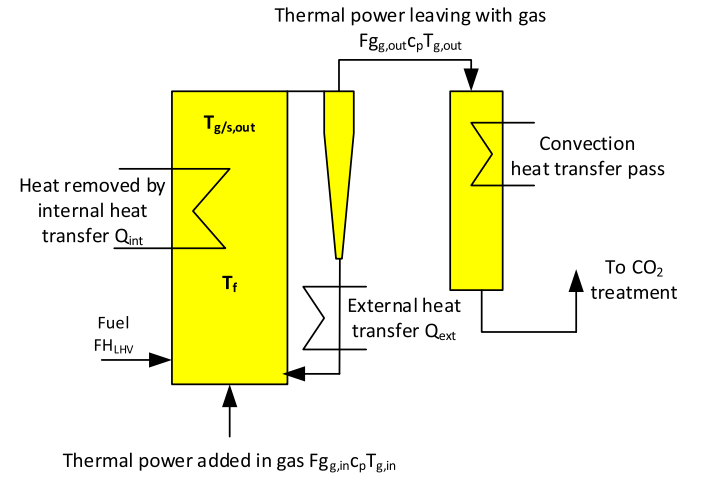
Fig. 2. Heat streams in a CFB boiler. The heat balance includes the furnace and, if active, the EHEX. (The water and flue-gas recirculation systems are shown on Fig. 1). The energy flows are referred to an input of combustibles of $F = 1$ kg/s (later, F is included as a variable to balance the volume flow) and the inlet temperature of the gas, $T_{g,in}$.

Fig. 1. Three alternative arrangements. a) air combustion b) flue-gas recirculation c) H_2O injection. The calculations are carried out for the fuel feed of $F = 1$ kg/s combustible matter. Most quantities shown are defined in Appendices A and B. Symbols: b kg combustibles/kg fuel, g kg gas/kg combustibles, ℓ kg O_2/kg combustibles, w – kg moisture/kg fuel, λ – excess-oxygen ratio. Indices: in – entering the furnace, out – exiting the furnace, r – recirculation of gas, $H2O$ – water addition, $N2$ – nitrogen addition (with air).

"thermal fly-wheel", to even out minor disturbances with local redistribution of heat, but they do not affect the global balance over the furnace, since the particles do not change temperature when they flow in or out of the system. If there is an EHEX in operation, the circulating particles are cooled. In a hypothetical extreme situation, the particles (index s) are cooled to the entrance temperature of the gas (index g), $T_{s,in}= T_{g,in}$. During more normal operation, the particle return-temperature is $T_{s,in} \geq T_{g,in}$. The temperature difference in the particle flow will be related to the total temperature difference in the furnace as $T_{s,out} - T_{s,in} = \varphi(T_{g,out} - T_{g,in})$ where the degree of cooling is expressed by the coefficient φ.

2.3. Model formulation

2.3.1. Heat balance

The total heat balance over a control volume that includes the furnace and, if active, the EHEX in Fig. 2, expresses that, besides heating of the gas Q_f, the energy in the fuel released in the furnace is removed through the heat exchangers located in the furnace Q_{int}, and externally Q_{ext} through the EHEX. The heat-balance is (since the particles and the gas are well-mixed in the furnace $T_f=T_{g/s,out}$ and $T_s=T_g$):

$$FH_{LHV} = Q_{int} + Q_f + Q_{ext} \tag{1}$$

Where:

$$Q_{int} = f\left(\overline{\rho}_s, T_f\right) A_{int,h}\left(T_f - T_h\right) \tag{2}$$

$$Q_f = F g_{out} c_{pm,g}\left(T_f - T_{g,in}\right) \tag{3}$$

$$Q_{ext} = A_f G c_{pm,s} \varphi\left(T_{g,out} - T_{g,in}\right) \tag{4}$$

Here, G is the solids recirculation flux (kg solids/m^{2}cross-section furnace-area,s) and, g_{out} is the specific mass flow of combustion gas through the furnace in kg gas/kg combustibles. $A_{int,h}$ is the internal heat receiving surface area and T_h its temperature, A_f the furnace cross section, and c_{pm} are the mean specific heats of the gas (index g) and the solids (index s). $f(\overline{\rho}_s, T_f)$ is the coefficient of heat transfer between the bed and the heat transfer surfaces in the furnace, dependant of the average suspension density in front of the heat transfer surfaces $\overline{\rho}_s$and the corresponding furnace temperature T_f.

The heat absorbed by the gas in the furnace Q_f is equal to that extracted in the back pass, $Q_f = Q_{bp}$.

2.3.2. Gas and solids fluxes in the boiler

The fluidization gas-flow in the furnace (at T_f) is equal to the gas flow produced by combustion,

$$u_* A_f = F_* g_{out,*} / \rho_* \tag{5}$$

Here, $g_{out,*}$ is the specific mass flow of combustion gas through the furnace (where * stands for the air and oxy-combustion cases, but in most cases below only the oxy-options are denoted by *). The solids recirculation flux G is expressed like the gas quantities, related to the fuel flow by a specific quantity g_s (kg solids/kg combustible fuel) with Eq. (5) and the vertical particle flow,

$$A_f G = g_s F \quad \text{where} \quad g_s = G g_{out,*} / (u_* \rho_*) \tag{6}$$

In the upper part of the combustion chamber the particle flux is $G \approx \overline{\rho}_{s,exit}(u - u_t)$where u_t is the terminal velocity of a single particle. The particle density at the exit $\overline{\rho}_{s,exit}$is uncertain to determine, so another way is tried, namely, the calculations are based on solids recirculation fluxes measured in boilers, Fig. 3.

Unfortunately, published curves from boilers differ considerably from each other, as shown in Fig. 3a, and furthermore, the details in the measurements are not revealed. Both correlations in Fig. 3a are determined for typical CFB conditions, using ashes/sand in the order of 200 μm size at 5 to 6 m/s. However, the Tsinghua data are referred to repeatedly by the Tsinghua authors [7] and are claimed to be used for the design of Chinese CFB boilers. Therefore, these data are used here, represented by the fitting correlation, shown in Fig. 3b,

$$G = 0.20u^2 + 0.94u + 0.43 \tag{7}$$

Dividing by the velocity u yields the ratio G/u to be used in Eq. (6)

$$G/u = 0.20u + 0.94 + 0.43/u \tag{8}$$

In Fig. 3b, it is shown that with $u = 5$, $G/u = \xi$, where $\xi \approx 2$ kg/m^3 (not much dependant on minor velocity variations, but, of course, dependant on the solids fluxes as in Fig. 3a).

3. Results and discussion

Maintaining the performance means to maintain the heat transfer in the boiler, but as it was discussed in the Introduction and in our previous work [4], this is not possible without modifications. Section 3.1. contains an analysis of several operational modes, carried out to search for this equivalent performance: in the first one (Mode 1) the conditions are adjusted to maintain an equal performance in air- and oxy-firing without changing the boiler, its main operational parameters, or the heat load. The conditions required to reach this goal are identified. In Mode 2, the equivalent performance is achieved by modifications in the fuel feed rate (Mode 2a) or in the internal heat extraction (Mode 2b). The influencing parameters are discussed in Section 3.2. The proposed measures lead to the conservation of most parameters in the furnace, but the
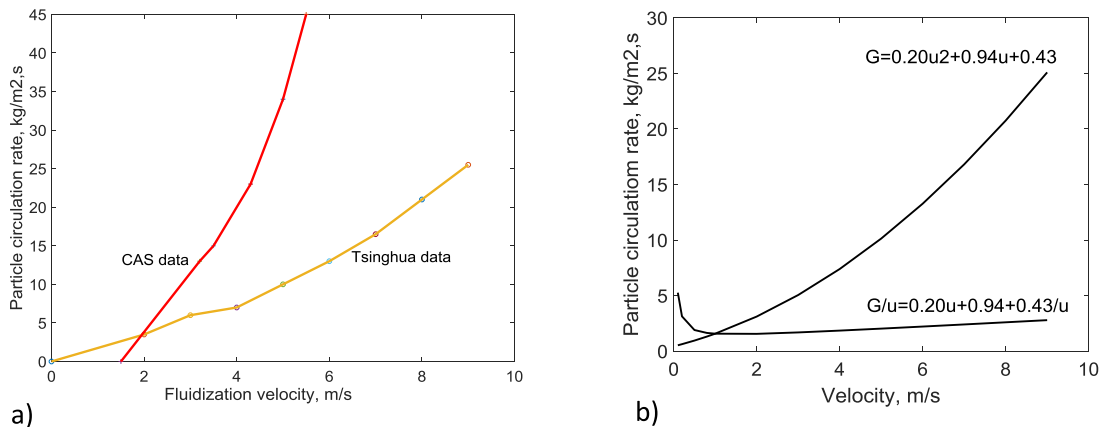

Fig. 3. (a). Solids recirculation fluxes measured by the Chinese Academy of Science (CAS) [5] and by the Tsinghua University [6] on CFB boilers.(b), the Tsinghua data, represented by a fit to the measured curve, Eq. (7), and transformed to G/u in Eq. (8).

composition of the gas and the heat (sensible heat) carried by the gas differ from those in air-firing. As a result, the back pass must be modified, as discussed in Section 3.3.

3.1. Search for equivalent performance modes in air and oxy firing

In the following, first, expressions are developed to identify the operation conditions that are equivalent to air combustion, and then the different solutions are analysed.

A comparison between the heat balances in the same boiler, operated under the same conditions with air or oxygen, can be formulated from Eq. (1), using Eqs. (5 and 6) and including the heating of particles

$$\left. \begin{array}{l} F_{air}H_{LHV} = Q_{\text{int},air} + F_{air}g_{out,air}c_{pm,air}\Delta T_f + F_{air}g_{out,air}c_{pm,s}\xi\varphi\Delta T_f/\rho_{air} \\ F_*H_{LHV} = Q_{\text{int},*} + F_*g_{out,*}c_{pm,*}\Delta T_f + F_*g_{out,*}c_{pm,s}\xi\varphi\Delta T_f/\rho_* \end{array} \right\} \quad (9)$$

With the effective specific heat $c_{pm,*,eff} = c_{pm,*} + c_{pm,s}\xi\varphi/\rho_*$ Eqs. (9) can be further simplified,

$$\left. \begin{array}{l} F_{air}H_{LHV} = Q_{\text{int},air} + F_{air}g_{out,air}\Delta T_f c_{pm,air,eff} \\ F_*H_{LHV} = Q_{\text{int},*} + F_*g_{out,*}\Delta T_f c_{pm,*,eff} \end{array} \right\} \quad (9a)$$

Mode 1: The operation conditions are adjusted to maintain equality between air- and oxy-firing without changes in the boiler

If F, Q_{int}, Q_{ext}, and ΔT_f are the same in air and oxy combustion, Eq. (9) gives the heat balance,

$$g_{out,*}c_{pm,*,eff} = g_{out,air}c_{pm,air,eff} \text{ or } g_{out,*} = g_{out,air}\frac{c_{pm,air,eff}}{c_{pm,*,eff}} \quad (10)$$

The input oxygen concentration in oxy-firing $x_{O2,in,*}$ and u_* are the desired results, as developed in detail by the relationships in Appendix B.

Fig. 4 shows $x_{O2,in,*}$, calculated from Eq. (10) with data from Appendix B inserted. The accompanying diagram shows the velocities, obtained from Eq. (5), written for the air and oxy cases with $g_{out,*}$ from Eq. (10). The curves start with the case where no heat was exchanged in the EHEX (indicated by the markers on the curves) and show the further development for various effective specific heats (which are related to the amount of heat extracted in the EHEX).

Fig. 4 shows that when no heat is removed externally in the EHEX (the points given by the markers), the fluidization velocities are around 3.6 to 3.8 m/s (Fig. 4a), which is far below the target of 5 m/s, employed in the air-fired case. The corresponding input oxygen concentrations range from 27 to 37 mass% (Fig. 4b). The heat balance yields higher gas velocity in the furnace as more heat is exchanged in the EHEX. For

maximum EHEX operation, marked in Fig. 4 with a vertical line, corresponding to an effective specific heat of about 3 kJ/(kg,K), the input oxygen concentrations fall to 23 to 33% and the velocities increase to 4.2 to 4.4 m/s, still not sufficient for acceptable operation.

From a theoretical point of view, the larger the particle term $c_{pm,s}\xi\varphi$, the closer the ratio in Eq. (10) is to unity, and the conditions approach those of air combustion. However, there is a limit for the extraction of heat from an EHEX. Eq. (9) shows how the heat input is distributed to the three sinks: the heat-receiving surfaces of the furnace Q_{int} (Eq. (2)), the heat carried with the gas Q_f (Eq. (3)), and the heat removal from the particles in the EHEX, Q_{ext} (Eq. (4)),

$$H_{LHV} = Q_{\text{int},air}/F_{air} + g_{air,out}c_{pm}\Delta T_f + g_{air,out}c_{pm,s}\xi\varphi\Delta T_f/\rho_{air} \quad (9b)$$

For the bituminous coal, the terms of Eq. (9b) in MW/kg become, if all heat supplied is extracted in the EHEX, 28= 0 + 11 + 17 giving φ=0.25 at ξ=2. This is an upper limit, corresponding to a refractory-lined combustion chamber without heat transfer surfaces, where Q_{int}=0. This could be called the "chemical reactor limit" where combustion and heat transfer are separated from each other: combustion takes place in the furnace and all heat transfer in the EHEX. The heat extraction in the EHEX in this retrofit case is limited by φ=0.25, which is obtained from the above Eq. (9b) for the case when Q_{int}=0. If, on the other hand, heat extraction in the EHEX is zero, φ=0, then in the above example, the heat extraction in the furnace should be Q_{int}=17 because the heating of the gases remains the same if the temperature is constant. The range of 0<φ<0.25 is more common in boilers with heat transfer surfaces in the furnace, although CFB boilers with separated combustion and heat transfer sections have been built [8]. However, in all cases shown on Fig. 4 the fluidization velocity is too low.

If the solids circulation rate is higher because of the uncertainty, shown in Fig. 3a, and if it can be assumed to follow a similar relationship as in the investigated case, ξ would be greater than 2. Then the coefficient φ must be reduced to satisfy the heat balance of Eq. (9b) and the result would not otherwise differ from the previous one.

The differences between wet and dry recirculation, and water injection, with respect to gas velocity and inlet oxygen concentration, shown in Fig. 4 and Table 1, are not of great importance in the present context, and other reasons could motivate the choice of one or the other option. Also, it is seen that the water injection case is slightly less favourable, having lower velocity and higher oxygen concentration than those in the cases with flue-gas recirculation.

The conclusion is that, when the heat and mass balance is satisfied and the operation conditions are unchanged, the extraction of heat in an
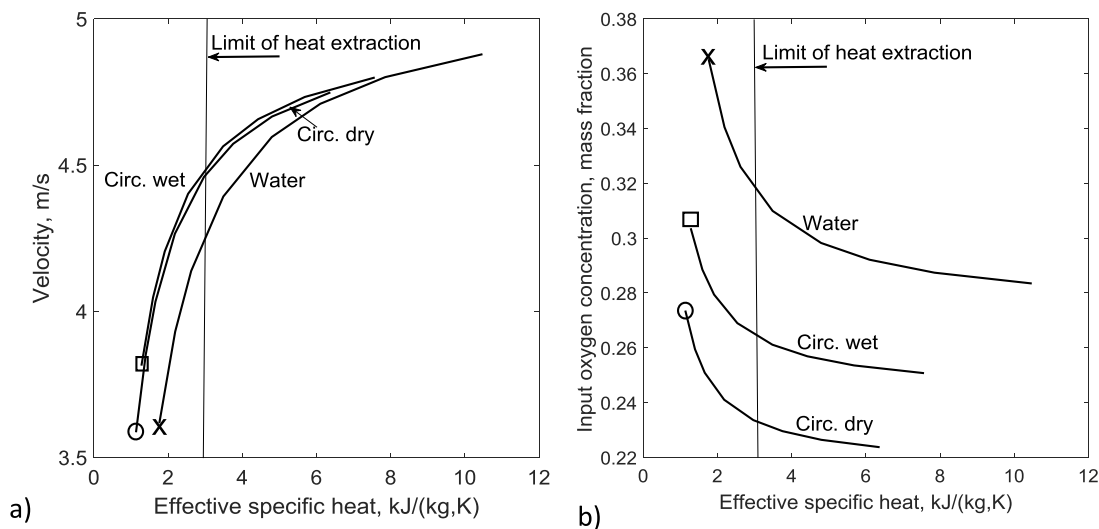

Fig. 4. (a)Velocity and (b) input oxygen concentration $x_{O2,in,*}$ in an oxy-CFB boiler for the fulfilment of the heat and mass balance with different amounts of heat extraction in an EHEX. The markers on each curve are cases without external heat extraction. The limit of heat extraction is described in the text.

Table 1

The fuel ratio η at constant fluidization velocities $u_* = u_{air}$ and constant internal heat transfer $Q_{int,*} = Q_{int,air}$, at a given furnace temperature difference between entering and leaving gas/particle flows of $\Delta T_f = 700$ °C .

	Bituminous coal H_{LHV}=28 MW/kg				Lignite H_{LHV}=25 MJ/kg		
Case	$g_{out\_v,air}$m³/kg	η	ΔQ_{int} or ΔQ_f kW/kg from Eq. (11)		$g_{out\_v,air}$ m³/kg	η	ΔQ_{int} or ΔQ_f kW/kg from Eq. (11)
Air	41.9	—	—		33.8	—	—
Recirculation, dry	–	1.15	4200		–	1.14	3500
Recirculation, wet	–	1.12	3400		–	1.10	2500
H₂O,addition	–	1.14	3900		–	1.13	3300

EHEX gives velocities that approach the velocity of air combustion somewhat, while the oxygen concentration required in the input gas falls. However, the EHEX heat extraction is limited by the heat balance of the furnace. As mentioned, it ranges from the situation when no EHEX is needed, because all heat is extracted in the furnace, to the case when all heat is extracted in the EHEX. It is seen from the diagrams that the limit does not reach very far into the range of effective specific heats, and the impact of EHEX in this retrofit case is not great.

Mode 2: The similarity of performance between air- and oxy-firing is achieved by changes in the fuel feed or internal heat extraction at $u_* = u_{air}$

To maintain the heat transfer in the furnace (Eq. 2), not only the mass balance but also the solids suspension density and the fluidization velocity should be kept the same as in air combustion. With $u_* = u_{air}$ in Eq. (5), $F_* g_{out\_v,*} = F_{air} g_{out\_v,air}$, where index v stands for volume and $g_{out\_v,*} = g_{out,*}/\rho_*$. This is inserted into the two Eqs. (9) before they are subtracted one from the other, which results in Eq. (11),

$$(\eta - 1) = (Q_{int,*} - Q_{int,air})/(H_{LHV}F_{air}) + g_{out\_v,air}(c_{pm,*,eff}\rho_* \\ - c_{pm,air,eff}\rho_{air})\Delta T_f/H_{LHV} \tag{11}$$

Here, $\eta = F_*/F_{air}$ is the desired result (when $\eta > 1$ more fuel should be added) if $x_{O2,in,*}$ is the quantity determined previously and Q_{int} is constant and equal in the air and oxy combustion cases,

$$(\eta - 1) = g_{out\_v,air}(c_{pm,*,eff}\rho_* - c_{pm,air,eff}\rho_{air})\Delta T_f/H_{LHV} \tag{11a}$$

Alternatively, if the fuel feed F is constant and $\eta = 1$, the heating of the added gas requires a corresponding reduction of the internal heat-exchanger area A_h, hidden in the expression of Q_{int},

$$(Q_{int,air} - Q_{int,*})/F_{air} = g_{out\_v,air}(c_{pm,*,eff}\rho_* - c_{pm,air,eff}\rho_{air})\Delta T_f \tag{11b}$$

In Eqs. (11), the difference $(c_{pm,*,eff}\rho_* - c_{pm,air,eff}\rho_{air}) \equiv c_{pm,*}\rho_* - c_{pm,air}\rho_{air}$ is independent of the particle suspension and equal in the two modes concerned if the external heat extraction is constant.

These two forms of Eq. (11), Eqs. (11a) and (11b), can be joined to give,

$$(\eta - 1)H_{LHV}F_{air} = (Q_{int,air} - Q_{int,*}) \tag{11ab}$$

showing the equivalence between the actions related to F and Q_{int}.

Table 1, obtained from Eq. (11), shows that the fuel feed-rate must be increased by a little more than 10% to heat the excess gas required to maintain the fluidization velocity at the desired level (5 m/s in this case) under the condition that Q_{int} and T_f are constant. Eq. (11ab) reveals that the effect is the same if the fuel feed-rate is maintained constant and instead the heat-transfer surface is reduced by a similar quantity (a little more than 10%). The table shows that, in both cases, a quantity of heat ΔQ_f is added to the gas, corresponding the amount of gas needed to bring the gas to the volumetric conditions of the original gas flow at a constant bed temperature T_f. This will be further evaluated in Section 3.3, but first various measures to convert from air to oxygen are discussed and summarized.

3.2. Summary of influencing factors for the different operation modes

The heat balance should not be affected when changing from air to

oxygen firing. However, the gas-volume flow in the furnace changes, and that decreases the fluidization velocity, which influences the fluid-dynamics and heat transfer. The reduction of the fluidization velocity cannot be tolerated, and some action should be taken as shown above, and summarized in Table 2.

Mode 1. The results of the heat balance, Fig. 4, Mode 1, demonstrate the insufficient velocities, even though the heat transfer was fictitiously assumed to be the same as in the air-fired boiler. In practice, this mode fails because the fluidization velocity is not sufficient, and the mode is not operational.

If the fluidization velocity u is kept constant and equal to that during air combustion, $u_* = u_{air}$, an additional gas mass-flow $F\Delta g_{out,*}$ and a corresponding additional heat flow ΔQ_f are needed, and either F, T_f, λ, or A_h or some other parameter must change, Mode 2 and 3 in Table 2.

Mode 2a. In this mode the fuel feed should be increased by a little above 10% to heat the additional gas produced (Eq. (11a) and Table 1) while the internal heat extraction is constant. This heat is delivered by the flue gas to the back pass because the total heat flow is equal in the furnace and back pass $Q_f = Q_{bp}$.

Mode 2b. The fuel feed is kept constant while the internal heat extraction is reduced by removing or inactivating the heat transfer surface A_h (e.g. by covering by refractory) to instead heat the additional gas ΔQ_f and to fulfil the heat balance, Eq. (11b). Just like in Mode 2a the heat is transported to the back pass to be removed there. The effect is the same as in Mode 2a, but there is an important difference: when F increases (Mode 2a), the load increases. In contrast, during a reduction of A_h there is no change of load, only a displacement of heat from the furnace to the back pass.

Mode 3. If u=const, F=const, A_h= const, then T_f decreases, which is not acceptable, because the temperature not only affects the heat transfer in the furnace and back pass, but also the combustion. Instead, it would be desirable to raise the temperature and so the volume flow of the gas and the fluidization velocity u_*. However, to increase the velocity up to the desired level of the air combustion velocity involves a significantly higher temperature than in the air case, and then A_h must be reduced considerably. In addition, increase in temperature is accompanied by a danger for agglomeration of the bed material.

There are several ways to influence the heat balance in connection to what is explained above or independently from those measures. The quantitative prediction of some of these measures is less certain, although there is empirical information to support a discussion. Table 3 lists various parameters and their usefulness.

Table 2

Modes treated (C stands for constant and equal to air combustion in the retrofit case).

Parameter/Mode	1	2a	2b	3
Fuel feed, F	C	Increases	C	C
Hear-transfer surface, A_h	C	C	Decreases	C
Furnace temperature, T_f	C	C	C	Decreases
Heat-transfer coefficient, f	C	C	C	Decreases
O	(fictitiously)			
Fluidization velocity, u	Decreases	C	C	Decreases
Excess-oxygen ratio, λ	C	C	C	C
Input oxygen concentration, $x_{O2,in}$	C	C	C	C

Table 3
Parameters affecting the change from air to oxy-combustion in a CFB boiler.

Parameter	Effect/Comment
Boiler dimensions and power	Cannot be changed, according to the prerequisites for the retrofit case.
Bed inventory	Can be changed, but the vertical density profile in the heat transfer section is mostly influenced in the transition region (between the bottom and transport zones) by changing the bed inventory [9]. The authors of Ref. 9 found a rise in bed temperature of about 10° for a reduction of the bed inventory corresponding to a pressure drop of 1 kPa. It seems difficult to increase the bed temperature substantially because the change in the bed inventory becomes too large.
Type and size of bed material	The bed material depends on the ashes and, consequently, on the fuel and its size. Larger particles tend to remain in the bottom region, and also the suspension density in front of the heat transfer surfaces may fall when the total pressure drop is assumed constant. This reduces heat transfer and increases furnace temperature. The recirculation to an EHEX also decreases in the case of coarser bed material. See also the above item.
Temperature	The bed temperature is affected by the above measures, but a better controlled method to modify the temperature is to change (remove) part of the heat transfer surface or to cover part of the surface with refractory material to increase the bed temperature. This was commented in Mode 2b.
Primary/secondary gas velocities	It is qualitatively known that higher primary velocity increases the descending fluxes at the walls [10] and thereby the particle density profile and the heat transfer. However, the split in primary and secondary gas also affects the emissions, and this aspect should be carefully considered before modifications are carried out.
The oxygen distribution	The distribution of oxygen to the primary or secondary gas flows and their oxygen concentrations can be manipulated, but such modifications require detailed studies to evaluate the consequences.
Stoichiometry	The circulating gas flow enhances the oxygen concentration in the circulation loop (furnace). Hence, in air combustion the air ratio could be 1.10 just as in the present calculations, but in the flue-gas recirculation case the ratio could be lower, for instance 1.05, at least in the flue-gas recirculation case, where additional oxygen is supplied by the circulating gas. This may be beneficial for the boiler's efficiency, but it does not significantly influence the problem discussed here.
The air/flue-gas preheat	This will be further commented in the following section, dealing with the back pass.

Even if the bed material and gas supply can be manipulated, it seems simplest and most predictable to cover heat transfer surface as suggested in Mode 2b. The other methods can be tried, based on trial and error or subjected to further investigations.

3.3. Performance of the back pass

The above balances are carried out across the furnace, but their consequences must be handled by the back pass. During the closure of the mass balance, the volume flow was reduced compared to air combustion (Mode 1). The convective heat transfer to the heat exchangers in the back pass will be affected by the corresponding reduction of velocity. When $u_* = u_{air}$ (Modes 2a and 2b), the volume flows are identical, but the flue gas of oxy-combustion carries more heat to be transferred. So, it is of interest to analyse the thermal performance of the back pass.

The back pass contains heat exchangers heating steam, water, and air, while cooling the flue gas. It may be arranged in different ways depending on the type of boiler and its steam cycle (steam data, number of feed-water heaters, etc.), the sulphur content of the fuel (acid dew point), type of air preheater (tubular or regenerative), etc. To generalize this, data are chosen to give a rough idea about the operation, and the back pass is modelled as a single heat exchanger. The temperature of the entrance gas is equal to the exit temperature from the furnace T_f, assuming the cyclone to be well insulated and without heat release from combustion. At the exit from the back pass, the gas temperature is set to 150 °C, the same temperature as downstream of the particle filter from where the flue gas is recirculated to the entrance of the furnace $T_{g,in}$. The entrance temperature in the air-combustion case is most likely around 30 to 50 °C, and then it may be heated in an air preheater to a level of about 150 to 250 °C. The recirculated flue gas or the added steam could have been pre-heated to the same temperature as that of air. In fact, this temperature level is not important for the present discussion if it is the same in all cases, such as assumed here: 150 °C. The remaining difference between the cases treated is in the heating of air from 50 °C to the entrance temperature, a gap of about 100°: the entire amount of air heats up, whereas in the oxy-fuel cases only the oxygen is heated, that is, about 30% of the flow, so there remains a thermal gap, corresponding to about 5–7% of the heat release in the back pass; less heat is consumed in the oxy-combustion recirculation case because the recirculated flue gas already has a temperature of 150 °C.

The overall heat transfer coefficient U depends on the inside (index i) and outside (index o) coefficients if the thermal resistance of the tube walls is neglected,

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} = \frac{h_i + h_o}{h_i h_o} = \left(\frac{2}{h}\right)_{h_i \approx h_o} ; U = h \bigg/ 2 \tag{12}$$

The coefficients h_o and h_i are assumed to be similar and equal to h (justified by the fact that the dominant heat exchangers are gas-gas and only a minor part, the economizer, is gas-water). The thermal power of the equivalent back-pass heat-exchanger can be written,

$$Q_{bp} = UA_{bp}\Delta\overline{T}_{bp} \tag{13}$$

Here, A_{bp} is the total heat-receiving surface of all heat exchangers in the back pass, the same in all operational modes. $\Delta\overline{T}_{bp}$ is the related mean temperature-difference. The modes with substitute gas (*) and air combustion are compared.

There are several versions of correlations for the heat transfer coefficient to a tube bundle during turbulent flow. Here a simplified form of the well-known correlation of Žukauskas [11] is useful. It is applied for fully developed turbulent region (power 0.8 on Re) and without corrections for the Prandtl number during heating or cooling. Instead, the conventional form $Pr^{0.33}$ is used,

$$Nu \sim Re^{0.8} Pr^{0.33} \tag{14}$$

The dimensionless numbers are Nusselt number $Nu = hd/\lambda$, Reynolds number $Re = ud\rho/\mu$, and Prandtl number $Pr = \mu c_p/\lambda$.

From Eq. (13), expressed as a ratio of the heat transfer in the oxy and the air combustion cases, and employing Eq. (14), one obtains,

$$\frac{h_*}{h_{air}} = \frac{u_*^{0.8}}{u_{air}^{0.8}} \frac{\lambda_*^{0.77}}{\lambda_{air}^{0.77}} \frac{\rho_*^{0.8}}{\rho_{air}^{0.8}} \frac{\mu_{air}^{0.47}}{\mu_*^{0.47}} \frac{c_{p,*}^{0.33}}{c_{p,air}^{0.33}} \tag{15}$$

where the length scale is constant (the heat exchangers are the same in all modes). Noting that the amount to be cooled is the same as that absorbed by the gas in the furnace $Q_f = Q_{bp}$, the ratio of the heat flows delivered from the combustion chamber to the back pass, obtained from Eqs. (3) and (9) is,

$$\left(\frac{Q_{bp,*}}{Q_{bp,air}}\right)_{hcap} = \frac{F_* g_{out,*} c_{pm,*} \Delta T_{bp,air}}{F_{air} g_{out,air} c_{pm,air} \Delta T_{bp,air}} \tag{16}$$

The gases are cooled by heat transfer from the furnace temperature T_f to the outlet temperature T_{out}. The heat uptake in the back pass, Q_{bp}, can be written for the air combustion and for the oxy-combustion cases with $\Delta \overline{T}_{bp}$ being the mean-temperature difference of the heat exchanger,

$$\left(\frac{Q_{bp,*}}{Q_{bp,air}}\right)_{htr} = \frac{U_* A_{bp} \Delta \overline{T}_{bp,*}}{U_{air} A_{bp} \Delta \overline{T}_{bp,air}}; \tag{17}$$

As discussed above, in the first case investigated (Mode 1), the fuel feed and the heat capacities (Eq. (10)) were constant, maintaining the furnace temperature provided that the heat transfer in the furnace is constant (imposed fictitiously). The heat flows then coincide (Eq.18), just as was determined for the furnace,

$$\left(\frac{Q_{bp,*}}{Q_{bp,air}}\right)_{hcap} \approx 1 \tag{18}$$

$$\left(\frac{Q_{bp,*}}{Q_{bp,air}}\right)_{htr} \approx 1 \tag{19}$$

There is an improvement in heat transfer due to the change in the flue-gas composition, as derived from Eq. (15). However, this improvement is counteracted by the reduction of heat transfer because of the lower velocity in the oxy-combustion case, $u_* \approx 0.7 u_{air}$. The improvement and the reduction happen to result in a value around 1.

Eqs. (18) and (19) show that in this mode the heat flows are almost the same in oxy and air combustion in the back pass. So, in a pulverized combustion boiler where the retrofit case is feasible without extensive further arrangements (except the burners), the heat is well recovered. This contrasts with a CFB boiler, where, despite the favourable situation in the back pass, the insufficient fluidization velocity makes the operation in Mode 1 less useful.

In the other situation, the velocities were imposed to be constant and equal in the air and oxy cases (Mode 2a). Then, the mass flow increases by $F\Delta g_{out,*}$. This additional flow is heated to maintain the temperature in the furnace, and the fuel flow F increases according to Eq. (11a). In this case of fluidized bed combustion, with $u_* = u_{air}$, we get, as shown in Table 1:

$$\left(\frac{Q_{bp,*}}{Q_{bp,air}}\right)_{hcap} = \frac{F_*}{F_{air}} \approx 1.10 \text{ to } 1.14 \tag{20}$$

and Eq. (17) with Eq. (15) gives,

$$\left(\frac{Q_{bp,*}}{Q_{bp,air}}\right)_{htr} = \frac{U_* \Delta \overline{T}_{bp,*}}{U_{air} \Delta \overline{T}_{bp,air}} \approx 1.25 \frac{\Delta \overline{T}_{bp,*}}{\Delta \overline{T}_{bp,air}} (1.23; 1.30; 1.25) \tag{21}$$

The fuel feed rate increases (or equivalently, the furnace's heat-exchanger area is reduced), as indicated in Eq. (20) (with data taken from Table 1). The increment in the convective heat transfer coefficient of the back-pass heat-exchanger, is shown in Eq. (21). The heat removal in the back pass is too efficient and must be reduced somewhat, down to a factor of about 1.10 - 1.14 instead of 1.25, to coincide with Eq. (20). Even in such a case it is uncertain how to take care of the excess heat on the steam side. Boiler-turbine systems can be overloaded temporarily, but overload may not be suitable for continuous operation. Furthermore, de-superheating (attemperation) is required to maintain the desired steam temperature, which is also not a good solution during continuous operation. To avoid increase in boiler load, the option of removing some heat transfer surface in the furnace (by cutting away or covering the tubes), according to Eq. (11b) and Mode 2b in Table 2, and related to the measures indicated in Table 3, is a viable solution to the problem of excess capacity, but the amount of heat extraction in the back pass remains high, and some re-allocation of heat transfer surfaces seems necessary after balancing evaporation and superheating in the furnace and back pass.

4. Conclusion

Change between air and oxy-combustion without changes in fuel feed, heat transfer, or furnace conditions is not possible in a CFB, because even if the heat balance were fulfilled, the change in the flue-gas volume-flow gives too low a fluidization velocity. To maintain both heat and gas flows in the furnace when changing from air to oxy-combustion, additional measures must be taken in a CFB to increase the fluidization velocity to the level used in air-firing. Maintaining the volume flow of gas equal in both operational modes requires more gas than that needed to maintain the heat balance in the furnace, leading to a higher heat load in the gas that flows through the boiler in the oxy-combustion case. Therefore, measures must be taken in the back pass to handle the excess heat. The increase in velocity can be achieved by increasing the fuel feed (a little above 10%) at constant excess oxygen or by reducing the heat extraction through inactivation or removing heat transfer surface in the furnace. In both cases excess heat is transferred to the back pass. The increase in fuel supply over-loads the boiler-turbine system, whereas the alternate remedy, the reduction of heat transfer surface in the furnace, only leads to the need for some re-allocation of evaporator and superheater surfaces.

Because of the change in composition of the flue gases from air to oxy-combustion, the convective heat transfer in the back pass is improved, and the back-pass heat-exchangers can accommodate the increased heat rate. However, the convective heat transfer is more efficient during oxy-combustion and some attention is required to adjust the heat extraction in the back pass to the steam cycle and other heat duties of the power station. This should be studied for each system concerned.

Funding

Declaration of Competing Interest

There are no conflicts of interest.

Appendix A. Fuels and media data

Characteristic data for the two types of fuel included in the analysis are presented in Table A1.

The property data of participating gases and solids are listed in Table A2. The mean specific heat of bed material 1.2 kJ/kg,K is an average value for several minerals, adjusted to the furnace temperature range [A2].

The densities are given at P_o=100 kPa and T_o=273.15 K and for boiler conditions P_x=100 kPa and $T_x = T_f = 850+273$ K (where the pressure in the furnace has been set equal to the atmospheric pressure as an approximation). The densities are calculated from,

Table A1
Fuel properties, proximate and ultimate analyses.

kg/kg fuel	Lignite	Bituminous coal
Combustibles, b	0.8	0.8
Ashes, a	0.1	0.1
Moisture, w	0.1(dried)	0.1
kg/kg combustibles	**Lignite**	**Bituminous coal**
Carbon, c	0.70	0.85
Hydrogen, h	0.05	0.05
Oxygen, o	0.25	0.10
Lower heating value, MJ/kg	25	28
Volatile content, kg/kg	0.50	0.30

Table A2
Media data.

Quantity	Density kg/m^3		Specific heat c_p, kJ/kg,K [a]			M, molecular mass
	0 °C	850 °C	0 °C	850 °C	150–850 °C[b]	kg/kmol
Air	1.273	0.310	1.007	1.163	1.091	28.9
N_2	1.233	0.300	1.038	1.187	1.116	28.0
O_2	1.410	0.343	0.909	1.103	1.039	32.0
CO_2	1.938	0.471	0.848	1.261	1.142	44.0
H_2O	0.793	0.193	1.864	2.385	2.137	18.0
Solids	-	-	-	-	1.2	-

[a] Calculated by EES [A1], which uses the JANAF database.
[b] Calculated by integration between 150 and 850 °C.

$$\rho_k = \frac{M_k}{v_M} \frac{T_0 P_x}{P_0 T_x} \tag{A1}$$

From the base data of Table A2 the corresponding data for gas mixtures ($k = N_2, O_2, CO_2, H_2O$) can be calculated.
Densities

$$\rho = 1 / \sum_k (x_k / \rho_k) \tag{A2}$$

and specific heats

$$c_{pm} = \sum_k c_{pm,k} x_k \tag{A3}$$

Data for thermal conductivities and dynamic viscosities are taken from polynomial correlations in VDI Heat Atlas [A3] and the dynamic viscosity of water vapour [A4] for the single gas components and from mixing rules to derive the values for gas mixtures [A5].
Symbols are found in the nomenclature list.

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Appendix B. Calculation details and definitions, particularly the oxygen concentration

Eq. (10), for simplicity written for gas only (otherwise the effective c_{pm} should be used), (10)
The specific gas yield $g_{out,*}$ is defined at the exit of the combustion chamber, including the overall mass balance

$$g_{out,*} = 1 + \frac{w}{b} + g_{in,*} \equiv \frac{cM_{CO_2}}{M_C} + (\lambda - 1)\ell + \frac{hM_{H_2O}}{2M_H} + \frac{w}{b} + g_{add,*} \equiv g_d + g_w + g_{add,*} \tag{B1}$$

where the stoichiometry and the fuel properties from Table A1 give the specific oxygen demand,

$$\ell = \left(\frac{c}{M_C} + \frac{h}{4M_H} - \frac{o}{M_{O_2}}\right)M_{O_2}\,(\mathbf{kg\,/\,kg\ combustibles}) \tag{B2}$$

and the dry and wet gas components,

$$g_d = \frac{cM_{CO_2}}{M_C} + (\lambda - 1)\ell;\ g_w = \frac{hM_{H_2O}}{2M_H} + \frac{w}{b} \tag{B3}$$

The specific mass flow at the inlet to the system outside of the gas-recirculation loop, Fig. 1, is equal to that at the exit from the boiler during air combustion and steam addition,

$$g_{out,exit} = 1 + w\big/b + \lambda\ell = g_d + g_w + g_{add,N_2/H_2O} \tag{B4}$$

In the case of gas recirculation, the corresponding quantity outside of the loop, is

$$g_{out,exit} = 1 + w/b + \lambda\ell = g_d + g_w \tag{B5}$$

because the quantity $g_{add,rec}$ (corresponding to g_r in Fig. 1) only exists inside the gas recirculation loop. Downstream of the addition of recirculated gas in junction A, the gas entering the furnace is

$$g_{in,*} = \lambda\ell + g_{add,*} \tag{B6}$$

A balance on junction A (Fig. 1 and more in detail in Fig. B1) involves the input oxygen ($\lambda\ell$) and the added recirculation flow ($g_{add,*}$) and gives

$$x_{O_2,in,rec}\big(\lambda\ell + g_{add,rec}\big) = \lambda\ell + x_{O_2,rec}g_{add,rec} \tag{B7}$$

which results in the added flow components $g_{add,*}$ listed below together with the other added flows in the cases of nitrogen, recycled gas, and water addition,

$$g_{add,N_2} = \lambda\ell\frac{1 - x_{O_2,in,N_2}}{x_{O_2,in\,,N_2}};\ g_{add,rec} = \lambda\ell\frac{1 - x_{O_2,in,rec}}{x_{O_2,in,rec} - x_{O_2,rec}};\ g_{add,H_2O} = \lambda\ell\frac{1 - x_{O_2,in,H_2O}}{x_{O_2,in,H_2O}} \tag{B8}$$

The outlet gas properties, such as concentration, density, and specific heat, do not change in junction B (Fig. 1) where they are split into a recirculated flow to Junction A and an exit flow to the CO_2 treatment. In the recirculation case, the exit flow consists of the combustion gas as shown by Eq. (B5). In the cases of air or steam additions they are expressed by Eq. (B4). The concentrations depend on the fuel composition and on the excess oxygen ratio. They are chosen for the gas after the combustion has taken place and are independent of the input oxygen concentration, $x_{O2,in}$. In the recirculation case the gas concentrations are,

$$x_{O_2,rec} = \frac{(\lambda - 1)\ell}{g_d + g_w};\ x_{H_2O,rec} = \frac{g_w}{g_d + g_w};\ x_{CO_2,rec} = \frac{cM_{CO_2}/M_C}{g_d + g_w} \tag{B9}$$

while in the air and H_2O injection cases there is no recirculation, and N_2 or H_2O leave the furnace together with the combustion gas,

$$x_{O_2} = \frac{(\lambda - 1)\ell}{g_d + g_w + g_{add,*}};\ x_{H_2O} = \frac{g_w}{g_d + g_w + g_{add,*}};\ x_{CO_2} = \frac{c\,M_{CO_2}/M_C}{g_d + g_w + g_{add,*}};\ x_* = \frac{g_{add,*}}{g_d + g_w + g_{add,*}} \tag{B10}$$

As seen from Eq. (B3) the concentrations sum up to unity, as they should.

The two sets of concentrations (Eqs. (B9) and B10) appear different, but they are not: in the furnace there is clearly a term corresponding to the added flow in the numerator and the denominator of Eqs. (B9). However, in the outlet from the furnace the concentrations must be the same in the branches towards recirculation and towards the exit. The latter are the easiest ones to evaluate. They are formulated in Eqs. (B9), which are, hence, equivalent to Eqs. (B10). This identity can be shown, accounting for the circulating flow in the furnace, which gives Eqs. (B9),

$$x_{O_2,rec} = \frac{(\lambda - 1)\ell + x_{O_2,rec}g_{add,*}}{g_d + g_w + g_{add,*.}}$$
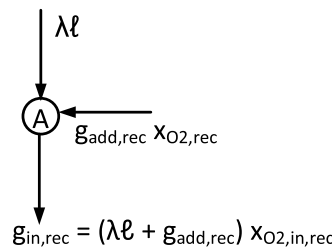

Fig. B1. Oxygen mass balance on the junction A in Fig. 1 in the gas-recycle case.

That is, $x_{O_2,rec}(g_d + g_w) + x_{O_2,rec}g_{add,*.} = (\lambda - 1)\ell + x_{O_2,rec}g_{add,*}$, where $x_{O_2,rec}g_{add,*}$ is cancelled out.

The fluidization velocities are evaluated for the flow inside the circulation loop in the furnace. The input oxygen concentration $x_{O_2,in,*}$ is given by Eq. (B8) where $x_{O_2,in,N_2} = 0.23$, of course.

Symbols and acronyms

A	Surface area; Af furnace cross section area, Ah heat exchanger surface area, m2
a	Ash content in fuel, kg/kg fuel
b	Combustibles in fuel, kg/kg fuel
CFB	Circulating fluidized bed
C	Constant
c	Char kg/kg combustibles
c	Carbon content in fuel, kg/kg combustibles
c_p	Specific heat at constant pressure; c_{pm} mean over a temperature range, kJ/kg,K
d	Dimension in Re, m
EHEX	External heat exchanger
f()	Heat transfer coefficient, kW/m^{2}K, dependant on average particle concentration $\bar{\rho}_s$ and T_f.
F	Mass flow rate of combustible matter in the fuel, kg combustible/s
FB	Fluidized bed
G	Circulation solids flux, related to furnace cross section, kg/m^2,s
g	Gas amount, kg/kg combustibles; index v, m^3/kg combustibles
H	Heating value, (MJ)kJ/kg combustibles
h	Heat transfer coefficient, kW/m^2,K
h	Hydrogen content in fuel, kg/kg combustibles
ℓ	Stoichiometric amount of oxygen, kg/kg combustibles
M	Molecular mass, kg/kmol
o	Oxygen content in fuel, kg/kg combustibles
P	Pressure, kPa
Q	Heat power, kW/kg combustible
T	Temperature, K or °C
U	Overall heat transfer coefficient, kW/m^{2}K
u	Fluidization velocity, m/s
v_M	Molar volume 22.7 m^3/kmol at 0 °C and 1 bar.
w	Moisture content in fuel, kg/kg fuel
x	Gas (mass) concentration, -

Greek symbols

Δ	Difference; Temperature increase in the furnace $\Delta T_f = T_f - T_{in}$; in the back pass $\Delta T_{bp} = T_f - T_{out}$. Mean temperature difference between the heat exchanger and gas in the back pass $\Delta\bar{T}_{bp}$. ΔQ is power added with the increment in gas quantity Δg to achieve $u = u_{air}$.
η	Ratio of fuel feed-rates, F_*/F_{air}
λ	Stoichiometric oxygen ratio, -; Thermal conductivity, kW/m,K
μ	Dynamic viscosity in Pr or Re, kg/(m,s)
ξ	Ratio G/u, kg/m^3
ρ	Density, $\bar{\rho}$ average value. kg/m^3
φ	Temperature ratio of heating in EHEX, $\varphi = (T_{s,out} - T_{s,in})/(T_{g,out} - T_{g,in})$

Indices

air	Air combustion
add	Additional quantity: nitrogen in air, recirculation gas, water added
bp	Back pass
char	Char
d	Dry
eff	Effective
exit	At the exit of the furnace
ext	External to furnace
f	Furnace
h	Heat exchanger in furnace
H_2O	water/stem
g	Gas
htr	Heat transfer
hcap	Heat capacity
i	Inside
in	Entering
int	Internal in furnace
k	Components
LHV	Lower heating value
N_2	air (nitrogen)
o	Initial, outside

oxy	Oxy-combustion
out	Exiting
pm	Index in c_{pm} denoting mean c_p over a temperature range
r, rec	Recycle
s	Solids
t	Terminal velocity
v	Volume
vol	Volatiles
w	Wet
x	Any value
*	Denotes air or oxy combustion

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