

Combustion System

Government Sponsorship

This invention was made with government support under Grant No. DE-FE0009478
5 awarded by the Department of Energy. The government has certain rights in the invention.

Priority Information

This application claims priority to utility application serial number 14/479,777, filed
on September 8, 2014 and to provisional application serial number 61/880,266 filed on
10 September 20, 2013, the contents of which are incorporated herein by reference.

Background of the Invention

The present invention is a combustion system in which hot flue gases are split into at
15 least two streams, mixed with recycled gas to control the gas temperature and introduced into
a heat recovery steam generator. The split arrangement allows for lower recycling power
requirements and a smaller heat exchange area.

Reducing the capital cost and/or increasing the efficiency of power generation is
20 highly desirable especially given the ever growing market for electric power [1]. Besides
economic concerns, efficiency in power generation is also important because the dominating
majority of electric energy production is from non-renewable fossil fuels, and there are
increasing concerns and regulations regarding the associated greenhouse gas emissions [2].
Flexibility to uncertain parameters, like fuel specifications, ambient conditions, and thermal
25 load is another important characteristic required from the implemented technologies.

In thermal power generation there are several forms of unavoidable losses as well as
several operations constraints and economic considerations. For example, heat exchanger
areas are limited by capital cost considerations, resulting in less effective heat transfer and/or
30 larger pressure drops of the streams due to the packed and constrained pathways; thus, the
efficiency of the cycle decreases. Metallurgic properties pose temperature constraints on
combustors, boilers, heat recovery steam generators (HRSG), turbines, etc., [3,4], thus
resulting in energy waste and reduction of power output. For example, flue gas recycling

(FGR) in coal boilers is required in order to quench the combustion gas to limit the radiative-dominant (high temperature) heat transfer requirement and enable a convective-dominant heat transfer; flue gas recycling (FGR) is applied because radiation is more expensive than convection for the same degree of thermal energy transfer, [5]. FGR also increases the boiler efficiency and reduces emissions, and is applied in almost all relatively recent (younger than 30 years old) coal fired powerplants, [5, 6]. The FGR requires compression to compensate for losses in the boiler and the recycling pipes; moreover, throughout the heat exchange process, the temperature gradient between the hot and cold streams decreases as flue gas moves away from the inlet, which increases the heat exchange area required close to the cold end of the heat exchanger.

One of the promising concepts of carbon-capture and sequestration is pressurized oxy-coal combustion (OCC). Pressurizing the flue gas increases the effectiveness of the convection heat transfer. In [7, 8, 9, 10, 11, 12, 13, 14], a pressurized OCC concept is considered with an HRSG with relatively high FGR that relies on convective heat transfer. Note that in general the combustion process occurs in a section that may or may not be physically connected to the HRSG, referred to as a combustor. The capital cost of the HRSG is a relatively large portion of the capital cost of the powerplant, e.g., [15], and reducing its size results in significant savings.

Summary of the Invention

The combustion system of the invention includes a combustor generating a combustion gas and a splitter for receiving the combustion gas and dividing the combustion gas into first and second gas streams. A temperature controller receives the first gas stream, mixes it with recycled gas and introduces it into a heat recovery steam generator at a desired temperature. A mixer receives the second gas stream mixing it with recycled gas and introducing it into the HRSG. Cool gas exits the HRSG with the cool gas forming the recycled gas. In a preferred embodiment, the combustor is an oxy-coal combustor. A preferred embodiment further includes recycling fans to pressurize the recycled gas for mixing with the first and second gas streams. The mass flow rate of the second gas stream, temperature at an inlet of the HRSG and flow rate of the recycled gas are controlled to

optimize the system to minimize heat exchanger area. Recycling power requirements are also smaller.

Brief Description of the Drawing

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Fig. 1 is a schematic illustration of a heat recovery steam generator as part of a pressurized OCC process with a thermal recovery unit.

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Fig. 2 is a graph of temperature versus heat recovery steam generator duty transfer showing temperature profiles of four different operations of flue gas with identical cold streams.

Description of the Preferred Embodiment

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Herein, a novel split concept for the HRSG is introduced in order to enhance the rate of thermal energy transfer by increasing the average temperature between the involved streams, and reduce the compression requirements by reducing the recycling flow rates and/or pressure losses compared to the conventional operation. The concept is applicable to coal boilers and other heat exchange processes that require quenching of the hot fluid.

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Fig. 1 illustrates the split concept applied to the HRSG of a pressurized OCC process, [11, 12, 13, 14]. In pressurized OCC, oxygen is delivered to the combustor at an elevated pressure. Primary recycling flue gas, FG-Rec-pri, is mixed with the oxygen stream for dilution in order to control the temperature of the combustor to acceptable levels, [3]; a temperature of 1550° is considered here. This temperature is entirely exemplary. Combustion gas 10 is mixed with secondary recycling flue gas, FG-Rec-sec, to achieve an acceptable temperature at the entry of the HRSG 12. In the HRSG 12, thermal energy is transferred to the working fluid 14 of a Rankine cycle. Flue gas is recycled for controlling the critical temperature of the combustors and the HRSG in two possible configurations, wet or dry recycling. In wet recycling flue gas is recycled directly after the HRSG exit, while in dry

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recycling flue gas is recycled after condensing and separating the water. Fig. 1 illustrates the wet recycling case, but the split design disclosed herein can be equivalently applied to dry recycling. Recycling fans 16 are used to compensate for the pressure losses encountered by the flue gas mainly in the HRSG and the recycling pipes.

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The concept disclosed herein splits the hot combustion gas prior to its dilution by the secondary recycling stream. The flue gas entering the HRSG 12 decreases in temperature as it exchanges thermal energy with the working fluid 14 of the Rankine cycle; flue gas acid condensation in the HRSG is not allowed as discussed later. In essence, the mixing of the split stream in the HRSG 12 is intended to elevate the flue gas temperature. The primary flue gas is mixed with a recycling stream similar to the HRSG without splitting. The split flue gas is (potentially) mixed with another secondary recycling stream. The secondary recycling to the split allows larger ranges and larger feasible combinations of the mixing positions and the mixing temperatures. The specific amount of recycling to the split, if any, is determined by optimization. Minimal area does not require any recycling to the split, whereas minimal compensation power is required. The additional split pipe, the pipe of the recycling to the split (if used), and the recycling to the split recycling fan (if used) add some capital cost, however, it is insignificant compared to the savings in the HRSG. Note that even when two recycling streams are used, a single fan 16 can be installed by introducing some throttling at the outlet of the split recycling stream; obviously this results in higher compensation power requirements. For a given HRSG thermal load, the splitting process can increase the overall temperature difference between the streams of the exchanger, particularly avoiding small temperature differences which require the most heat transfer area. Moreover, the split allows for lower rates of recycled flue gas compared to regular HRSG: the required amount of recycling to the inlet of the HRSG is smaller because it needs to dilute a smaller amount of the combustion gas. Also, at the mixing position, the flue gas within the HRSG acts as a diluent to the split stream, thus a relatively small amount of the recycling to the split is required (if any).

30 The higher average temperature differences imply smaller exchanger area, and the lower recycling requirements imply lower pressure drops and lower compensation power requirements (CPR). The CPR consists of two components; the first is the power needed by

the recycling fans 16 to re-pressurize the recycling streams to compensate for the pressure losses encountered in the HRSG 12 and the recycling pipes. The second component of the CPR is the power needed to maintain the main flue gas flow as it faces pressure losses while passing through the HRSG. For simplicity, only one split is illustrated and tested here.

5 Multiple splits, that would be introduced sequentially at different intermediate locations, are possible and would further increase performance and decrease the heat transfer area, but would add structural complexity. Note that in general the split can be extracted from any point along the HRSG and not limited to the inlet combustion gas, and the recycling can also be withdrawn from any point along the HRSG and not limited to the outlet Cool-Gas;
10 however, such processes are less practical to implement and more complicated to model and optimize.

To illustrate the possible advantages, Fig. 2 shows the profiles of four cases of HRSG operation as a standalone unit with an identical set of input parameters. As shown in Table 1,
15 the input parameters are the specifications of the hot and the cold input streams, the outlet specifications of the cold stream, and fractional thermal losses; the fixed specifications signify the total amount of thermal energy transferred in the HRSG is fixed. The numbers in Table 1 are exemplary only. Table 1 shows two sets of input parameters as obtained from an optimized pressurized OCC with wet recycling, [13, 14]; the specifications titled CoalA are
20 the operating conditions that the HRSG encounters when the basecase OCC process utilizes a high quality coal, while those titled CoalB are relevant to combusting a lower quality coal, where CoalA and CoalB are identical to those utilized in [13]. Both sets of input specifications of the streams are for the nominal full load operation. The cold stream profile is that of the main feedwater and the reheat streams. The maximum allowed HRSG
25 temperature is 800°C equal to that considered in [11, 12, 13, 14]. Similar to the basecase the HRSG is considered to face a 0.75% fractional heat duty losses. First, note that in Fig. 2 although the input streams specifications, CoalA specifications of Table 1, and the total duty transfer in the HRSG are constant among all four operations, the temperature of the flue gas at the exit of the HRSG, Cool-Gas, might not be. The different pressure losses and recycling
30 requirements for each operation result in different CPR which causes different amounts of compression enthalpy rise (CER) carried by the flue gas, [11]; therefore, the temperature of the Cool-Gas exiting the HRSG is slightly different between the four profiles.

Table 1: Fixed input parameters for the HRSG. Two different flue gas conditions are presented, each relevant to a different coal type.

Input Parameter	With CoalA	With CoalB
Flue Gas Conditions		
Combustion gas flowrate	402kg/s	394kg/s
Combustion gas temperature	1550°C	
Combustion gas pressure	7.41bar	9.67bar
Combustion gas mole fraction	H ₂ O = 0.479; O ₂ =0.030; H ₂ O = 0.478; O ₂ =0.030;	
Composition	N ₂ =0.008; CO ₁ =0.457; N ₂ =0.009; CO ₂ =0.458;	
	SO ₂ =0.001; AR=0.025; SO ₂ =0.001; AR=0.024;	
Maximum allowed HRSG tempera- ture	800°C	
HRSG fractional heat loss	0.75%	
Flue Gas flowrate to recovery unit	120kg/s	132kg/s
Feedwater and Reheat Conditions		
Feedwater flowrate	306kg/s	301kg/s
Feedwater inlet temperature	322°C	322.0°C
Feedwater inlet pressure	265bar	
Feedwater outlet temperature	600°C	
Feedwater outlet pressure	250bar	
Reheat flowrate	233kg/s	260kg/s
Reheat inlet temperature	358°C	
Reheat inlet pressure	53.5bar	
Reheat outlet temperature	610°C	
Reheat outlet pressure	53.1bar	
Compressors Specifications		
Primary recycling fan	$\eta_{\text{isentropic}} = 0.83$ $\eta_{\text{mechanical}} = 0.99$ Thermal spec = Adiabatic	
Secondary recycling fan(0&1)	$\eta_{\text{isentropic}} = 0.8338$ $\eta_{\text{mechanical}} = 0.99$ Thermal spec = Adiabatic	
Main stream compensation compressor	$\eta_{\text{isentropic}} = 0.90$ $\eta_{\text{mechanical}} = 0.98$ Thermal spec = Adiabatic	

The first profile in Fig. 2 is without splitting or recycling, i.e., all the combustion gas enters the exchange directly without dilution; this operation violates the maximum temperature constraint on the HRSG and is only shown for illustration. The infeasible operation of Profile 1 theoretically requires the smallest heat exchanger area due to the largest

temperature differences between the hot and cold streams, and requires zero recycling and zero second component of the CPR.

The second profile in Fig. 2 represents the standard basecase operation where all the combustion gas is mixed with enough recycling to result in Hot-Gas entering the HRSG 12 at precisely the maximum allowed temperature. The flue gas temperature then drops to the exit temperature as thermal energy is transferred to the working fluid. The flue gas has approximately constant thermal capacity as inferred by the nearly linear temperature profile versus thermal energy transferred. A lower inlet temperature for the Hot-Gas into the HRSG required higher recycling flowrate, which results in larger flue gas flowrates in the HRSG, a flatter temperature profile on the hot stream, and smaller temperature differences between the streams of the HRSG. If thermal energy transfer and pressure losses are independent of the flow conditions in the HRSG, then lower inlet temperature, leading to smaller temperature differences and larger recycling flow rates, are clearly unfavorable regarding both exchanger area and the CPR. However, the heat transfer, pressure losses, and flow conditions of the flue gas are not independent, so larger flows and smaller entry temperatures might be favorable in some cases, especially when CPR is of a higher priority than area, since larger flowrates may contribute in smaller HRSG pressure losses.

The third profile in Fig. 2 represents a theoretical operation while respecting the maximum temperatures constraint; this graph is only for illustrative purposes. The profile is achieved by an infinite number of splits, and an infinitesimal recycling to the inlet required to decrease the temperature of the infinitesimal inlet combustion gas from 1550°C to 800°C. The split combustion gas is introduced infinitesimally into the HRSG 12 maintaining for as long as possible a constant temperature equal to the maximum allowed. When all the combustion gas is introduced, the temperature profile is only infinitesimally flatter than that of Profile 1, and the two profiles seem indistinguishable.

The fourth profile in Fig. 2 represents an operation with a single split. First a certain amount of combustion gas is split, and just enough recycling to the inlet of the HRSG is used to obtain a Hot-Gas temperature of, for example, 800°C. The split is then introduced to the

HRSG without any recycling at a point where the resulting flue gas mixture in the HRSG attains a temperature of, for example, 800°C. Compared to the conventional operation of Profile2, the split can provide larger average temperature differences between the streams of the HRSG, therefore, smaller areas. Moreover, lower recycling flowrates, and possibly smaller CPR, are required since respecting the maximum temperature constraints are attained not only by recycling but also by the gradual heat transfer (note that pressure losses in the HRSG, which might increase, are another factor in determining the CPR). Lower flue gas flowrates can also be inferred from the steeper sloped of Profile4 compared to those of Profile2. Introducing the split further downstream and/or adding recycling to the split, neither of which are shown here but considered later for optimization, result in a lower mixture temperature inside the HRSG. Also, adding recycling to the split can allow earlier mixing positions while satisfying the maximum temperature constraint.

It can be proven geometrically that for a given split flowrate, the largest temperature differences between the hot and cold streams are attained when the inlet and the mixing temperatures are at the maximum allowed and when there is no recycling to the split. Also, by comparing the slopes of the temperature profiles, it can be proven that for any split flowrate, operating with the mixing temperatures at the maximum possible value minimizes the flowrate of the recycling streams. It is tempting to say that for a given split flowrate the least area requirements and the least recycling requirements are obtained when the constraints of the maximum allowable temperatures are active; as the optimization shows, in fact the area is minimized when the maximum temperature constraints are active, however, this is not always true for the CPR.

It also can be proven that any split operation with a certain total amount of recycling, regardless of the number of splits, the splits flowrates, or the mixing positions, can at most reach the borders outlined by a profile with no split and the same total amount of recycling introduced all at once to the inlet; as an example Profile3 and Profile1 with an infinitesimal recycling to the inlet, respectively.

The purpose of the split disclosed herein is to reduce the capital cost by utilizing smaller surface area of the heat exchanger and reduce the CPR by reducing pressure drops and/or recycling flowrates. The two objectives are neither equivalent, nor mutually exclusive,

and have to be accounted for simultaneously; smaller exchangers in general lead to larger pressure losses, and the flow properties affect the heat transfer coefficient, the HRSG pressure drop, and the recycling losses. The HRSG area may be calculated according to the logarithmic mean to temperature differences and heat exchanger elements.

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The CPR consists of two components: the first component of the CPR is the power needed by the recycling fans to re-pressurize the recycling streams after experiencing pressure losses from the HRSG and the recycling pipes. The second component is the power needed to maintain the main gas flow and overcome the HRSG pressure losses. The extra
 10 compression needed to overcome the main flow losses can be introduced prior to combustion or after the HRSG. For example, in standard coal powerplants, the same fans or compressors that drive the inlet streams to the combustor force the flow through the main stream pressure losses. But in pressurized OCC combustors, since a compressor is already present after thermal recovery and needed for the carbon sequestration unit (CSU), the compensation
 15 power requirement is less costly to be accounted for by compressing a cooler stream with a lower flowrate post the thermal recovery.

As aforementioned, a single split is considered herein. The independent decision variables are chosen to facilitate optimization, since they allow satisfying some constraints by
 20 properly setting their ranges. Also, these variables are considered the simplest to monitor and to set their desired values during operation. The optimization variables are: i- the split flowrate, $\dot{m}_{\text{Split},1}$, ii- the temperature at the inlet of the HRSG $T_{\text{Mix},0}$, iii- the flowrate of the recycling stream to the split, $\dot{m}_{\text{Rec},1}$, and finally iv- the temperature of the flue gas in the HRSG after introducing the split, $T_{\text{Mix},1}$. The variables are illustrated in Fig. 1 and the ranges
 25 are defined in Table 2.

Based on this choice of independent variables, important variables are now dependent. For example, the flowrate of the recycling stream to the inlet, $\dot{m}_{\text{Rec},0}$, is dependent once the split flowrate and the inlet temperatures are specified; i.e., the stream entering the HRSG is fully specified. Further, the position of introducing mixture of the split
 30 and its recycling in the heat exchanger, $\text{Mix-Pos} = \frac{A_{\text{before mixing}}}{A_{\text{total}}}$ is dependent once the split flowrate, the flowrate of the recycling to the split, and the mixing temperature are specified.

Number	Variable	Range	Base-case and/or default value
1	$\dot{m}_{\text{Split},1}$	[0 - 300] kg/s	0 set to 100 kg/s
2	$T_{\text{Mix},0}$	$[T_{\text{FW},\text{out}} - T_{\text{HRSG},\text{max}}]$ °C	$T_{\text{HRSG},\text{max}} = 800^\circ\text{C}$
3	$\dot{m}_{\text{Rec},1}$	[0 - 600] kg/s	NA/0
4	$T_{\text{Mix},1}$	$[T_{\text{FW},\text{in}} - T_{\text{HRSG},\text{max}}]$ °C	NA set to $T_{\text{HRSG},\text{max}} = 800^\circ\text{C}$

Table 2: Optimization variables, their ranges and the basecase default values, for a single split. Because most of the basecase variables values are far from the optimum (zero split, no recycling to split, and no mixing within the exchanger), several initial guesses are implemented in order to exclude suboptimal convergence. The boundaries of the ranges of the temperature variables are set to avoid constraint violations. $T_{\text{Mix},0}$ lower bound is set to the maximum temperature of the cold streams, i.e. the reheat stream exiting the HRSG at 610°C; also, the upper bound on mixing temperatures $T_{\text{Mix},0}$ & $T_{\text{Mix},1}$, are set to the maximum allowable temperature in the HRSG 800°C; and the lower bound of $T_{\text{Mix},1}$ is set to the temperature of the feedwater entering the HRSG 321.7°C.

The operation of the heat exchanger is subject to physical limitations. Introducing the split provides better performance while still satisfying these constraints. The temperatures of the streams inside the heat exchanger are a major concern. Defined by the metallurgic properties, the maximum temperature allowed in the HRSG is limited to $T_{\text{HRSG},\text{max}}=800^\circ\text{C}$. In essence the constraint has to be satisfied at every point within the heat exchanger; but because the temperature monotonically decreases along the HRSG (apart from the mixing point), the constraint needs only be imposed at the inlet and at the mixing position. For the addressed values of the input streams, the temperature of the cold stream entering the HRSG is safely above the acid condensation temperature of the flue gas, [11], therefore, there is no need to include the constraints on the minimum allowed temperature for avoiding condensation of acids in the flue gas or on the feedwater tubes in the flue gas. In contrast, it is beneficial to include constraints on the MTA to avoid temperature crossovers

and ensure a realistic operation. The physical limit of MITA is zero, but a value of 0.5°C is used to speed the optimization process. In other words, the intuition that small MITAs are clearly undesirable since they result in huge area requirement, is communicated to the optimizer.

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A coal powerplant utilizes different types of coals during its lifetime. The advantages of the split concept are not limited to the type of coal used; for any coal utilized for power generation, the HRSG-split can be designed accordingly in order to reduce the area and/or the CPR. Since during the operation of the powerplant the utilized coal is expected to change, it is extremely important that the optimal design of the HRSG-split for one coal is flexible to changing the coal; i.e., the design is also optimal for the other coal.

Assessing the flexibility of the HRSG-split in general requires characterizing the variables as design or operation valuables, [13, 14]. Design variables are fixed upon design while operation variables can change with the different operations of the HRSG. The four independent variables for the optimization of the standalone HRSG-split $TMIX0$, $mSplit1$, $TMix1$, & $mRec1$, are all operation variables. However, dependent variables of the standalone model, in particular the split mixing position, $Mix-Pos$, and the resulting HRSG area $\frac{Aa}{Ab}$, are design variables and have to be common between the different operations. Luckily, based on the following approach, there is no need to reformulate the problem to include design variables as decision variables, which pose a lot of difficulties in solving for the objective function which would require complicated numerical methods, and result in large domains of infeasible operations.

The ideal flexibility is demonstrated herein by a much simpler approach; the same optimization above is performed on a standalone HRSG-split but with the new specifications of the input streams which are relevant to a coal different from the original. The input streams specifications used above are a result of the pressurized OCC process designed for ideal flexibility to uncertainties when operating with a typical bituminous coal with composition similar to Venezuelan and Indonesian coals (referred to as CoalA), while the new streams' specifications result from operating with a lower quality South African coal almost identical

to Douglas Premium or Kleincopje coal (referred to as CoalB), as presented in [13]. The multi-objective optimization of the HRSG-split operating with the streams conditions of CoalB results in a Pareto front curve very similar to that seen for the HRSG-split multi-objective optimization (presented above) operating with the conditions of CoalA. More specifically, equal areas of the two Pareto fronts have identical mixing positions, therefore, a given Pareto optimal design of the HRSG-split for one coal is also a Pareto optimal design for the second, therefore, the HRSG is thermodynamically ideally flexible. Moreover, equal areas between the original and new Pareto curves are a result of equal weight vectors for the multi-objective optimization, therefore, the HRSG-split is also economically ideally flexible; determining the most profitable design does not require to consider the coal distribution because any optimal design for one operation is optimal for the other, and has the same tendencies/preferences towards each of the two objectives. A similar behavior to that of the coal variation is encountered for the other uncertainties. As a conclusion, the HRSG-split is ideally flexible to uncertainties, and at least capable of maintaining the flexibility of the process it is incorporated in.

The application of the HRSG-split is not limited to the OCC process. The split concept can be applied to any heat exchanger process that requires a recycling stream to control the temperature of the main stream; for example, in conventional boilers, both the FGR rates and the heat exchange area, particularly radiative, can be reduced. The concept can be readily applied to new powerplants; moreover, the retrofit of existing plants is conceivable.

Although not shown here, the benefits of the HRSG-split in subcritical power cycles can have larger magnitudes than those obtained here for a supercritical. Recall that the smaller the temperature differences between the streams, the larger the area required for the same amount of heat transfer, and therefore, the exchanger area is directly related to the value of the MITA. In the case study above, the feedwater conditions are those from an optimized supercritical Rankine cycle, where the pinch in the HRSG is located at the cold end; the feedwater temperature entering the HRSG is relatively large, by utilizing the FWHs regeneration, allowing higher rates of feedwater through the HRSG while respecting the MITA specifications, and thus larger flowrates through the expansion line to increase the

power output and efficiency. Since the pinch is at the cold end, introducing the HRSG-split cannot avoid the pinch because the flue gas temperature at the exit of the HRSG varies only slightly due to the variations in the CPR. Also, the feedwater temperature profile in the HRSG is smooth due to the absence of phase transition. On the other hand, with subcritical
5 feedwater, the transition from the subcooled liquid state to the two-phase state is marked by a sharp kink, and usually the pinch point occurs at that location rather than at the cold end. Since in an HRSG-split the temperature of the flue gas after mixing is larger than that of the basecase operation, except very close to the exit of the HRSG where the temperature might be slightly lower depending on the CPR, then the temperature difference at the location of the
10 pinch is larger than that of the basecase. Now since in subcritical operations the pinch is alleviated, the reduction in area and pressure losses are significantly larger compared to the supercritical scenario. Note that the increase in the temperature approach due to the split in the subcritical process allows, upon process optimization, for even larger feedwater flowrate through the HRSG which increases the power output and the efficiency; i.e., compared to the
15 supercritical scenario, in a subcritical process the savings on area and CPR are larger, and there is a possibility of increased power output and further in the efficiency.

The split concept disclosed herein is applicable to heat exchangers that require recycling of the hot stream for temperature control, e.g., coal boilers and HRSGs. The
20 concept proposes splitting the hot stream, which has a temperature higher than that allowed in the heat exchanger, before its dilution and its introduction into the heat exchanger. At the inlet of the flue gas, a lower amount of dilution is required to control the temperature of the now smaller fluid flowrate. The split fraction is then introduced into the heat exchanger at an intermediate point downstream, increasing the temperature of the hot stream and enhancing
25 the temperature gradient of the heat exchange process. The concept is able to reduce the cost by reducing the area requirements and/or increase the efficiency by decreasing the power required to compensate for the pressure losses of the flue gas. The concept is illustrated in a standalone model of an HRSG in the context of a pressurized oxy-coal combustion process. Multi-objective optimization is performed by constructing the Pareto front of minimal area
30 and minimal power requirements. Both the heat exchange area and the compensation power requirements are shown to be reduced compared to the conventional operation; in the illustrated case, the area can be reduced down to 0.63 the original size, and the compensation power requirements can be reduced down to 0.82 the original requirements. The design and

operation is not limited to new heat exchangers and retrofitting in considered easily possible because no changes in the internal structure of the heat exchanger is required.

Moreover, facing uncertainty in input parameters and operating conditions, the split
5 concept is shown to be ideally flexible and preserves the flexibility of the process it belongs to.

Herein, the heat exchanger process is enhanced by a modification to the design of heat
exchanger while holding the input streams and the total transferred thermal energy constant.
10 However, the input streams to the exchanger are variables of the process it belongs to.
Therefore, the overall performance of the process and the performance of the HRSG and the
capital cost savings can be enhanced by simultaneous optimization of the HRSG-Split and the
powerplant design.

15 Further details of the invention are included in "A Split Concept for HRSG (Heat
Recovery Steam Generation) with Simultaneous Area Reduction and Performance
Improvement," Zebian and Mitsos, Vol. 71, pp. 421-431, Energy (2014).

The contents of the references listed herein are incorporated by reference in their
20 entirety.

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What is claimed is:

1 1. Combustion system comprising:

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3 a combustor generating a combustion gas;

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5 a splitter for receiving the combustion gas and dividing the combustion gas into first
6 and second gas streams;

7

8 a temperature controller for receiving the first gas stream, mixing it with recycled gas
9 and introducing it into a heat recovery steam generator at a desired temperature;

10

11 a mixer for receiving the second gas stream, mixing it with recycled gas and
12 introducing it into the heat recovery steam generator;

13

14 wherein cool gas exits the heat recovery steam generator, the cool gas forming the
15 recycled gas.

1 2. The combustion system of claim 1 wherein the combustor is an oxy-coal combustor.

1 3. The combustion system of claim 1 further including recycling fans to pressurize the
2 recycled gas before mixing with the first and second gas streams.

1 4. The combustion system of claim 1 wherein the mass flow rate of the second gas stream,
2 temperature at an inlet of the heat recovery steam generator, and flow rate of the recycled gas
3 are controlled to optimize the system.

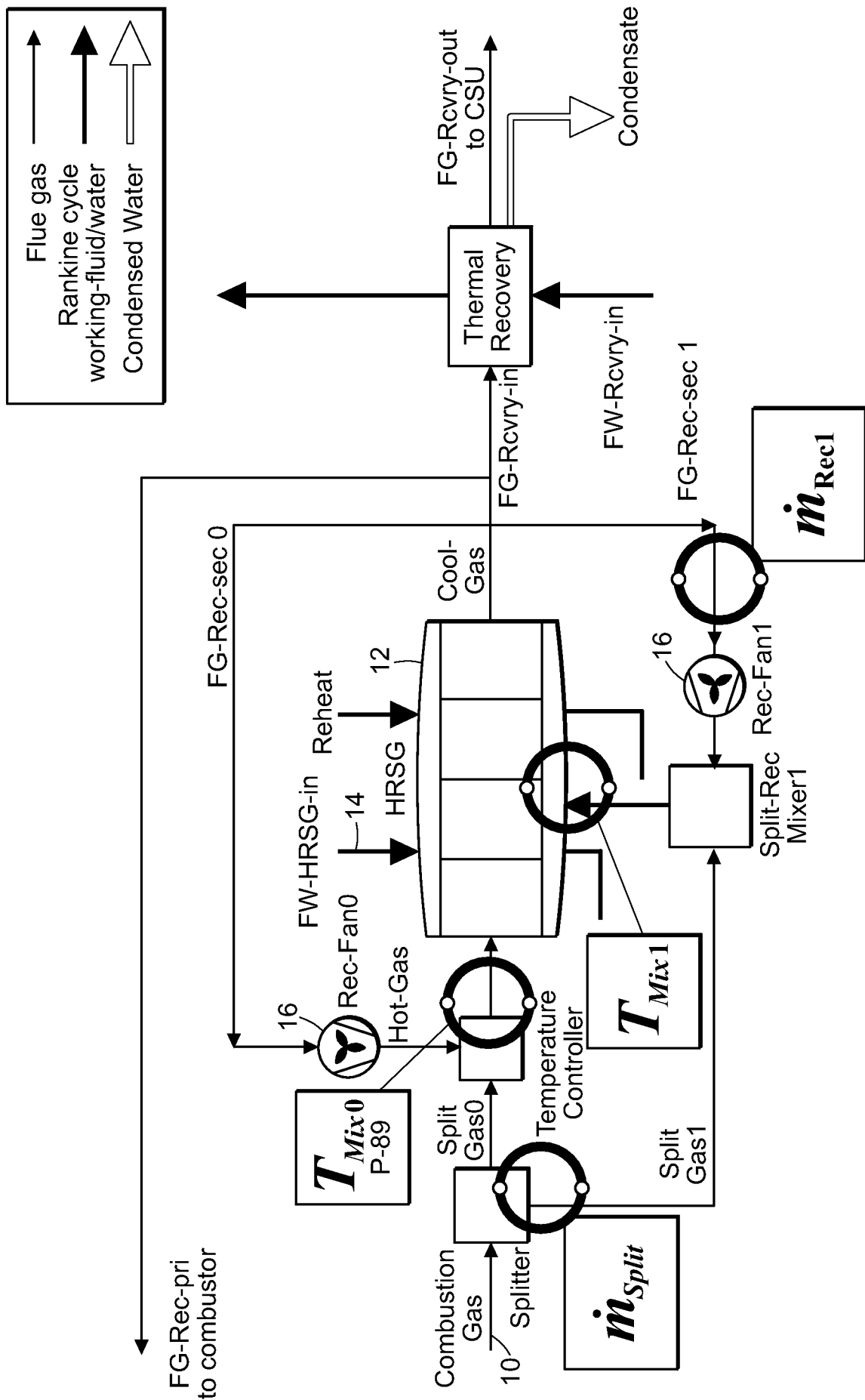


FIG. 1

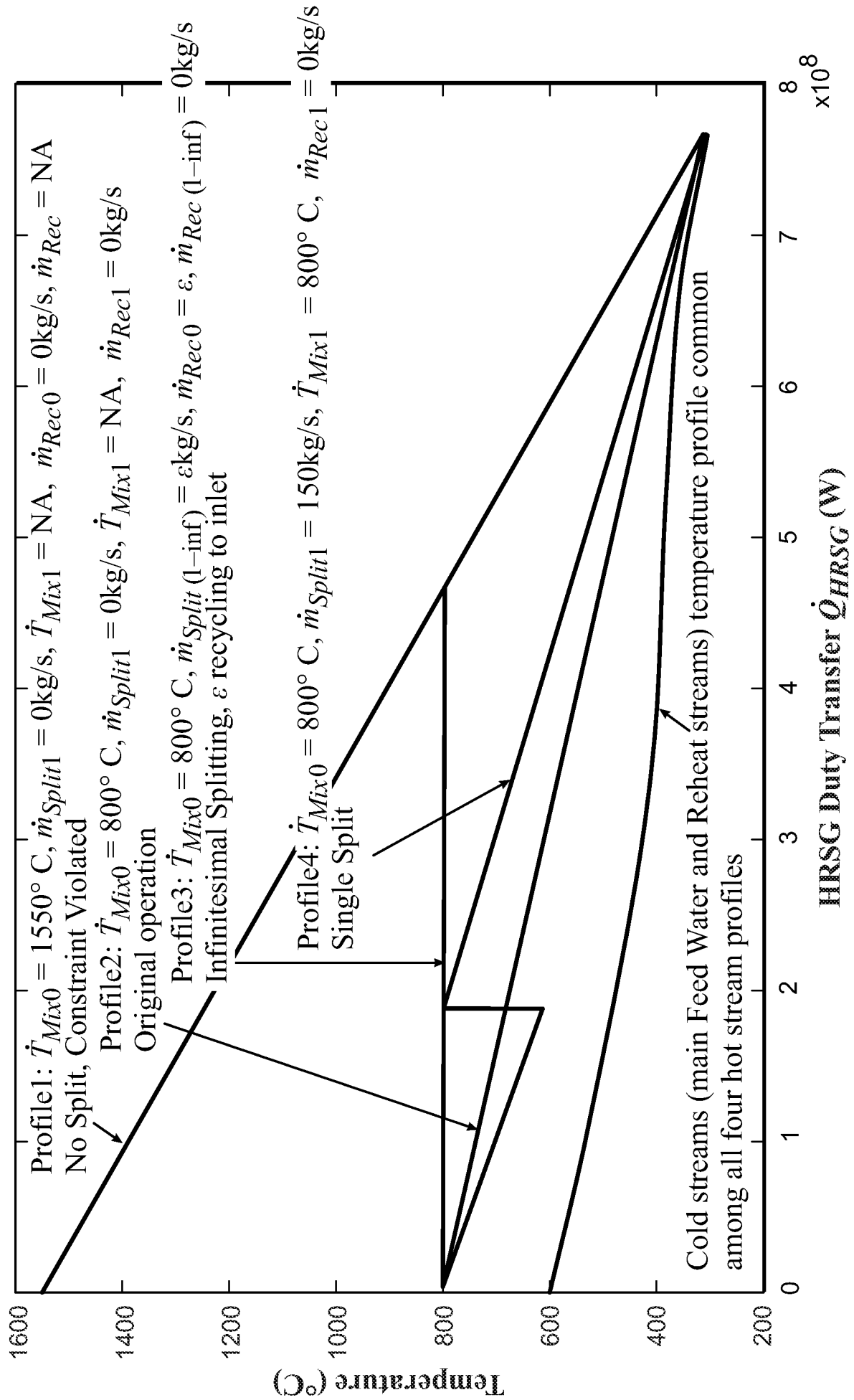


FIG. 2