Indirect evaporative cooling for enhanced efficiency of industrial gas boiler application: Thermodynamic potential evaluation.

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Abstract:
In sustainable combustion processes, latent heat recovery from flue gas has emerged as a key innovation for enhancing energy efficiency. Rather than discharging flue gas carrying residual energy to the stack, recovering its waste heat and re-injecting this energy into the system leads to a higher efficiency. Indirect Evaporative Cooling (IEC) allows this approach by exploiting water evaporation at the inlet air side to directly and efficiently recover the latent heat released during flue gas water vapour condensation. In addition, water injection allows active modulation of the heat-exchanger operating curves, enabling adjustment of the effective heat capacity of the flue gas. Although the concept appears promising, the existing literature lacks a concrete evaluation of its potential, making it difficult to compare its effectiveness with that of other technologies. In this paper, we present a combined gas boiler and absorption heat pump system with an integrated indirect evaporative cooler (IEC) for flue gas latent heat recovery. To quantify the additional recovery potential enabled by the IEC configuration, the system performance is compared with that of a direct evaporative cooling approach reported in the literature. A generic IEC technique based on the M-saturator was constructed in Aspen Plus using generic performance data (i.e. dew point effectiveness). First results indicate a potential thermal capacity of 310% compared to the 11.7% of the direct evaporative cooling solution, showing the potential of IEC for waste heat recovery. The validity and feasibility of performance indicators reported in the literature are assessed for this specific case. Assuming a dew point effectiveness of 80% and a wet channel outlet temperature of 95 °C, an overall IEC effectiveness of 67.1% is obtained.

Keywords:
Indirect evaporative cooling, Latent heat recovery, Industrial waste heat recovery, Heat and mass transfer, Heat exchangers.

1. Introduction

With the growing awareness of global warming, coupled with rising fuel prices, it has become essential to reassess industrial processes to optimise energy efficiency and minimise waste heat. In recent decades, researchers have focused on reducing the heat lost through stacks into the atmosphere, so-called waste heat [1], by proposing several options to reduce the flue gas temperature by recovering this heat [2]. However, 52% of our global primary energy use still ends up as waste heat through flue gas [3], with 20-25% being latent heat [4]. The improvement of the energy efficiency of combustion systems is thus a must.

Flue gas heat recovery methods can be classified according to the level of heat recovery achieved. Sensible heat recovery is simple as flue gas is cooled down by preheating the oxidiser, and producing hot water. Latent heat recovery is more complex, as the main challenge is to recover energy below the dew point by achieving significant heat transfer at a low temperature. Several technologies have been developed for direct or indirect recovery of this waste heat [2, 5]. Although these techniques significantly reduce flue gas temperature, condensing boilers are the only commercially available technologies that can recover the latent heat in the flue gas. However, no effective solution has yet been found to directly reintegrate this latent heat into the process by condensing the water vapour in the flue gas. The main reason for this unsuccessfulness has been the lack of a cold reservoir to absorb this heat, since the latent heat is only released at low temperatures (below the dew point of, e.g. 40-60°C for natural gas combustion [6]), making a direct recovery in the process often impossible, due to the specific process temperature. The latest research is mainly focused on upgrading this remaining potential by using heat pumps [7], requiring additional energy input [8]. But neither in the passive, nor in the active method, are the temperatures low enough to condense the flue gas and thus recover the condensation heat. On the other hand, indirect methods, such as desalination plants running on waste heat from power generation or industrial processes, involve the creation of a useful by/side-product by using the waste heat [9]. However, the possible valorisation of the waste heat depends then on local opportunities and specific heat demands [10], hence not providing a general solution.

Direct heat recovery methods use heat exchangers to preheat the working fluid of the process using the remaining heat of the flue gas to limit the use of primary energy. From a process efficiency optimisation point of view, this is the preferred option [1]. Depending on the process and the used technology, high recovery rates can be achieved, but the intrinsic mismatch in heat capacities (i.e., higher heat capacity of the flue gas due to the specific water content) as well as the missing large cold sink, leads to inherent potential waste heat that cannot be recovered.

Water evaporation on the cold inlet air side can solve the problem of a lack of a cold reservoir. Indeed, applying indirect evaporative cooling (IEC) on the process air can result in the creation of a cold sink, enabling enhanced waste heat recovery and, while doing so, improved energetic performances. This technology could be used to optimise industrial processes with low-temperature and high-humidity content flue gas, which benefits from the oxidiser humidification. Combustion process flue gas contains mainly nitrogen ($>70\%$), remaining oxygen, a small CO_2 fraction ($<10\%$), and an important share of water (in vapour form) [11]. The latent heat of this water share accounts for 20-25$\%$ of the total waste heat [4]. With the expected future massive deployment of Power-to-Fuel and the usage of green hydrogen, this latent heat will become even more important as the fraction of water in the exhaust gas will increase. Moreover, humidified combustion reduces thermal NO_x formation by lowering flame temperature [12, 13], whereas waste heat recovery reduces CO_2 emissions through improved overall system efficiency [13]. Gas boilers are thus good candidates for this improvement, as they benefit from humid combustion and their flue gas temperature commonly reaches 150-200 °C [13]. However, a specific case study demonstrating the potential in a real practical application is still lacking.

Therefore, in this paper, we present a combined gas boiler and absorption heat pump system with an integrated indirect evaporative cooler (IEC) for flue gas latent heat recovery. The system performance is compared with that of a direct evaporative cooling approach reported in the literature to quantify the additional recovery potential enabled by the IEC configuration [14]. A generic IEC model, based on the M-saturator, was constructed in Aspen Plus using generic performance data as dew point effectiveness.

2. Model description

In this section, we introduce the IEC and integrate it in a practical industrial application to compare its performance with that of a direct-contact heat exchanger (DCHE). The IEC concept and modelling are presented, and several performance indicators from the literature are defined.

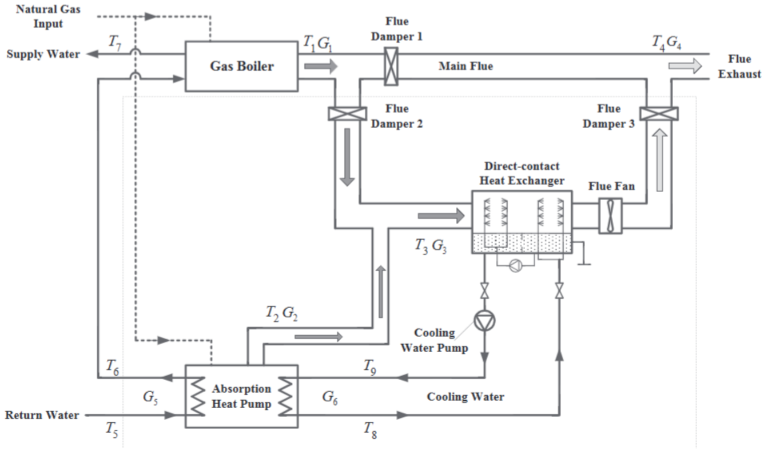


Figure 1: The reference system combines an absorption heat pump and a direct-contact heat exchanger to recover heat from flue gas. Cooling water captures this waste heat and feeds it back to the heat pump. The recovered heat is then used to preheat district heating return water, improving overall efficiency [14].

Table 1: Key values of the reference model [14].

Parameter	Value
Q_{boiler} [MW]	13
Q_{AHP} [MW]	4.1
$Q_{AHP,generator}$ [MW]	2.3
$Q_{recovery}$ [MW]	1.8
Q_{gas} [MW]	15.3
ϵ [%]	11.7
Heat recovered [MW]	1.8
Condensing heat recovered [MW]	1.2
Excess air ratio [%]	20

2.1. Absorption heat pump + DCHE

To evaluate the performance of the IEC technology, a reference industrial case from the literature is needed. Zhu et al. [14] present a technology that combines the absorption heat pump and the direct-contact heat exchanger illustrated in Fig. 1. In this reference system, a direct-contact heat exchanger is added to the gas boiler and the main flue for heat recovery. The flue gas from the gas boiler is mixed with that of the absorption heat pump before entering the DCHE. The absorption heat pump produces cooling water in its evaporator, which is then pumped into a direct-contact heat exchanger. There, the water is sprayed to cool the flue gas. This cooling water then returns to the evaporator, carrying the recovered heat and serving as a low-temperature heat source for the heat pump. In the district heating system, the return water first enters the absorption heat pump, gaining heat from the heat pump and the flue gas. Once preheated in the heat pump, the water then flows into the boiler for the main heating purpose [14]. In the article, the authors present a "design" configuration of their system. The parameters of this configuration are used to build our own model, in which the DCHE is replaced by an IEC fully integrated into the process. The key values used to reproduce the characteristics of the reference model are listed in Table 1. The results of the two models can then be compared.

Zhu et al. [14] introduce the notion of heat capacity improvement. This parameter is used as the

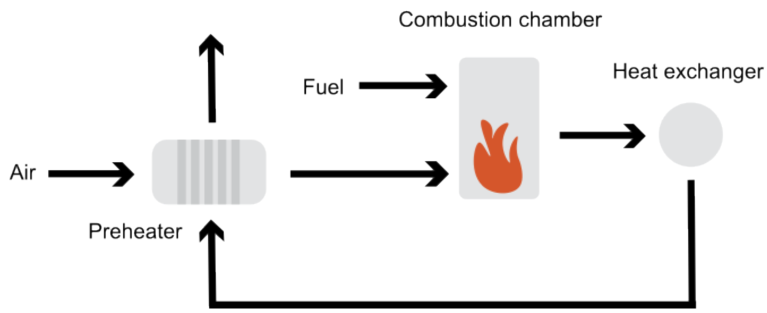


Figure 2: In a conventional combustion process, atmospheric air is preheated before entering the combustion chamber. The resulting flue gases are then utilised for various applications, such as steam generation, before passing through the preheater and being discharged via the stack.

main performance indicator for the comparison with the system using the DCHE. To avoid any misunderstanding with the notion of heat capacity, we prefer referring to this parameter as the thermal capacity, to emphasise the capacity to absorb and release heat. It compares a useful flux coming from the flue gas to a cost-effective heat flux linked to the fuel consumption. This thermal capacity is defined as:

$$\varepsilon = \frac{\dot{Q}_{recovery}}{\dot{Q}_{gas}}. \tag{1}$$

where $\dot{Q}_{recovery}$ is the useful heat flux recovered from the flue gas, and $\dot{Q}_{gas}$ is the cost-effective heat flux related to the use of natural gas. The first one can be distinguished into sensible and latent heat. Those are computed as shown in 2.

$$\dot{Q}_{recovery} = \dot{Q}_{sens} + \dot{Q}_{lat} = \dot{m}_{fg} c_{p_{fg}} (T_{in,fg} - T_{out,fg}) + \dot{m}_{fg} L_{v_{H_2O}} (x_{in,fg} - x_{out,fg}), \tag{2}$$

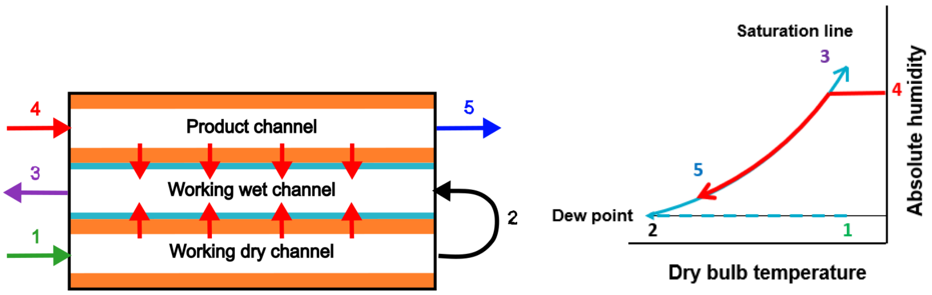
where $\dot{Q}_{sens}$ and $\dot{Q}_{lat}$ represent the sensible and latent heat recovered from the flue gas, respectively. The other flux in 1 is defined as:

$$\dot{Q}_{gas} = \dot{Q}_{boiler} + \dot{Q}_{AHP} = (\dot{m}_{NG} LHV_{NG})_{boiler} + (\dot{m}_{NG} LHV_{NG})_{AHP} \tag{3}$$

2.2. Indirect Evaporative Cooler

A basic industrial process, illustrated in Fig. 2, can be modelled as a preheater, a combustion chamber and a heat exchanger. Atmospheric air is preheated before entering the combustion chamber. The preheater uses the hot flue gases to increase the temperature of the working fluid. This device is classically used to reduce the amount of fuel needed to reach a certain temperature at the outlet of the combustion chamber, and thus increasing the system efficiency. The flue gases are then sent through a heat exchanger, where they are cooled down for a designated purpose.

The indirect evaporative cooler, represented in Fig. 3a, consists of a three-fluid counterflow heat exchanger in which atmospheric air is cooled by the wet channel, which also extracts heat from the product channel. This technology is directly derived from the air conditioning domain [15, 16]. For this paper, as an IEC technique, we selected the M-saturator of the M-power cycle [17], and modelled it in Aspen Plus. This software has been chosen because of its extensive component panel including a wide range of pre-built models, thermodynamic data, and physical properties. Aspen Plus offers accurate process simulations allowing the development of models used, in our case, for the evaluation of waste heat recovery potential. In Aspen Plus, the "REFPROP" model is used as a property method, allowing the computation of physical properties, as well as the dew point value [18]. REFPROP was



(a) (b)

Figure 3: Atmospheric air is cooled towards its dew point through the dry channel. This cooled air is then humidified to saturation while exchanging heat with the adjacent channels. The creation of a cold sink reservoir allows the extraction of heat from the product channel, and thus cooling the flue gases passing through. In the psychrometric chart of Fig. 3b, the blue curve illustrates the path of the atmospheric air through the IEC, while the red curve shows the cooling of the flue gases.

selected, as after analysis, it appeared to provide the closest correspondence with the psychrometric diagram for the dew point assessment.

As illustrated in Fig. 4, the basic industrial process from Fig. 2 is adapted to include the water injection in the IEC module. The IEC is represented by three heat exchangers and a mixer. The "Cooler" and the "Flue Gas Cooler" remove heat from their respective streams, whereas the "Reheater" raises the temperature of the humidified air. The dew point effectiveness, ε_{dew} [19], defined in 4, quantifies how closely the air approaches its dew point without changing the absolute humidity in the dry channel. This corresponds to transformation 1–2 in Fig. 3b and occurs in the Cooler shown in Fig. 4. The specific outlet dry channel temperature is computed by:

$$T_{out,dry} = T_{atmo} - \varepsilon_{dew}(T_{atmo} - T_{dew,atmo}), \tag{4}$$

This process allows the creation of a large cold sink reservoir to extract heat from the product channel. Once cooled, this air goes through the wet channel, where it gets humidified to saturation and heated by both adjacent channels. This corresponds to transformation 2-3 in Fig. 3b and takes place through both the Mixer and the Reheater shown in Fig. 4. Indeed, the first law of thermodynamics can be written on the wet channel as:

$$\Phi_{Wet} = \Phi_{Dry} + \Phi_{FG} = \dot{m}_{wet} c_{p_{wet}} (T_{out,wet} - T_{in,wet}) + \dot{m}_{wet} L_{v_{H_2O}} (x_{out,wet} - x_{in,wet}), \tag{5}$$

where Φ_{Dry} and Φ_{FG} represent the heat fluxes exchanged between the dry channel and the wet channel, and between the product channel and the wet channel, respectively. Φ_{Wet} corresponds to the heat flux exchanged between the inlet and the outlet of the wet channel.

As discussed in [20], several performance indicators can be defined. However, their validity and feasibility have to be assessed for the particular case of an IEC applied to industrial waste heat recovery. First, the wet bulb effectiveness that compares the actual evolution with an adiabatic humidification, for which the minimum achievable temperature is the wet bulb temperature:

$$\varepsilon_{wb} = \frac{T_{in,dry} - T_{out,dry}}{T_{in,dry} - T_{wb,in,dry}}. \tag{6}$$

Then, the dew point effectiveness is defined in 4. It compares the actual evolution with sensible cooling until saturation, which would ideally occur in indirect evaporative coolers. To evaluate the effectiveness of the heat recovery in the flue gas, the dew point effectiveness can also be expressed on the product channel:

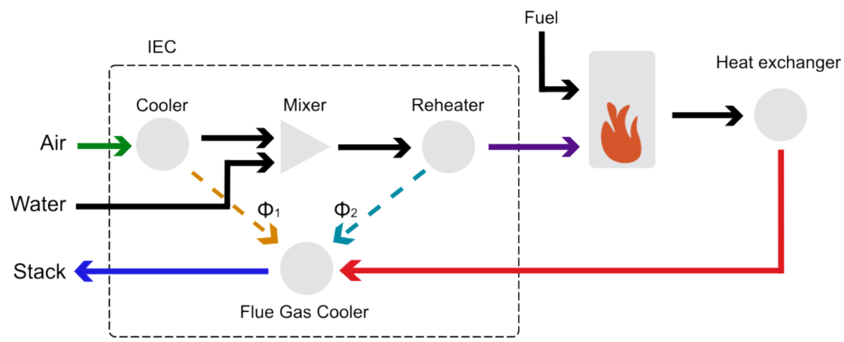


Figure 4: The IEC is modelled using three heat exchangers and a mixer for the water injection. Cooling and reheat fluxes are used to determine the flue gas cooling heat flux in the designated cooler.

$$\varepsilon_{dew} = \frac{T_{in,fg} - T_{out,fg}}{T_{in,fg} - T_{dew,in,fg}}. \tag{7}$$

The dew point and wet bulb effectiveness do not necessarily reflect the actual performance of IECs, as they only account for operating conditions on the primary side. Furthermore, the performance of M-saturator type IECs may be overestimated when using dew point effectiveness, since the minimum theoretical temperature is the dew point temperature of the secondary air. To better assess the true performance of evaporative coolers, a more general definition of effectiveness can be introduced:

$$\varepsilon_{IEC} = \frac{T_{in,fg} - T_{out,fg}}{T_{in,fg} - T_{min}}. \tag{8}$$

where T_{min} for an M-saturator type of IEC is the dew point temperature of the fluid at the inlet of the dry channel ($T_{dew,in,dry}$) [20].

In this first design, the dew point effectiveness (ε_{dew}) is fixed at 0.8 and allows to determine the outlet temperature of the dry channel. Then, the outlet temperature of the wet channel (T_{hum}) is imposed to compute Φ_{Wet} in 5. Knowing Φ_{Dry} and Φ_{Wet} (respectively Φ_1 and Φ_2 in Fig. 4), the heat flux extracted to the flue gas (Φ_{FG}) can be deduced. The resulting outlet temperature of the product channel can be used to compute the dew point effectiveness on the product channel. The parameter ε_{IEC} then serves as an indicator of the validity of the obtained value.

2.3. Absorption heat pump + IEC

The reference system is modified by replacing the direct-contact heat exchanger (DCHE) with an indirect evaporative cooler (IEC). The integration of the IEC aims to achieve effective flue gas cooling and improved heat recovery by avoiding direct contact between the cooling medium and the flue gas, thereby reducing exergy destruction. To ensure consistency with the article, the boiler and absorption heat pump powers are matched by adjusting the fuel mass flow rates of the components. Excess of air remains at 20%. The outlet temperature of the wet channel (T_{hum}) is varied from 35 °C to 95 °C to highlight the impact on the amount of heat recovered from the flue gas and the effectiveness of the IEC.

2.3.1. System description

The IEC does not simply replaces the DCHE, it impacts the whole process. Atmospheric air first enters the IEC where it gets cooled towards its dew point with a given dew point effectiveness ε_{dew}. It is then simultaneously heated and humidified near saturation, before entering the gas boiler. The water used for this humidification is supplied by the evaporator of the absorption heat pump. The flue gas from the boiler and the heat pump are then mixed before re-enter the IEC, where is gets cooled in the

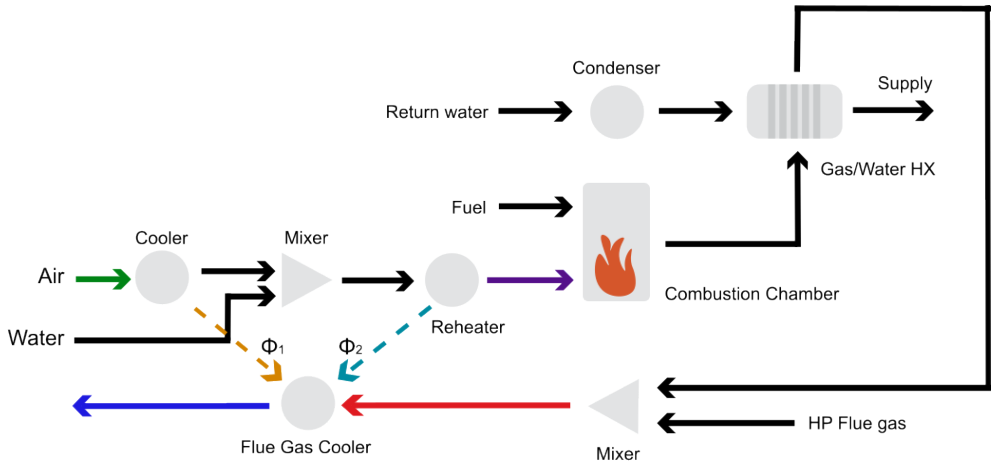


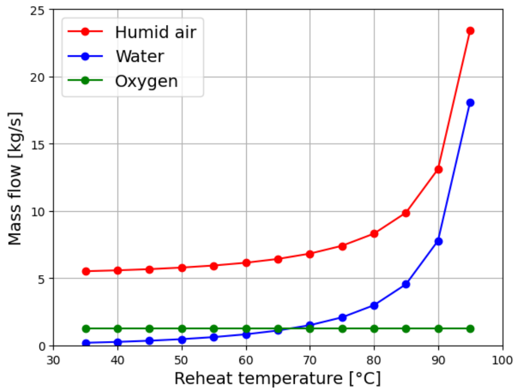
Figure 5: The Aspen flowchart represents the modelling of a coupled gas boiler and absorption heat pump system, in which the DCHE is replaced by an IEC. Since these components are not directly available as built-in blocks in Aspen, they are modelled using heaters, mixers, and reactors. The absorption heat pump itself is not explicitly modelled, as its outputs are known. It is therefore treated as a black box.

product channel. There, the water vapour contained in the inlet flue gas condenses and returns to the evaporator, carrying the recovered heat and serving as a low-temperature heat source for the heat pump. As in the reference system, the return water first enters the absorption heat pump, gaining heat from the heat pump and from the flue gas. Once preheated in the heat pump, the water then flows into the boiler for the main district heating purpose.

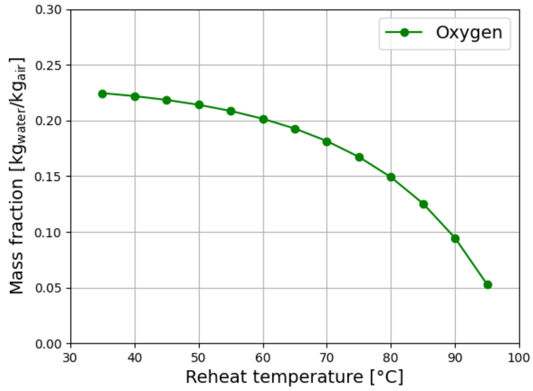
2.3.2. System modelling

The Aspen modelling of the system is presented in Fig. 5. Atmospheric air at 15 °C and 40% relative humidity enters the system before getting cooled down towards its dew point, with a dew point effectiveness imposed at 80%. This air is then humidified by a water stream in the mixer, and reheated to a given outlet temperature of the wet channel by the reheater. Saturation at the outlet of the wet channel of the IEC is ensured with a design-spec, to adapt the injected water mass flow. After that, this humid air enters the combustion chamber of the gas boiler, where a 20% excess of air is ensured by another design-spec. The gas boiler is modelled in Aspen as a Gibbs reactor, coupled with a water-flue gas HeatX. The flue gas of both the gas boiler and the absorption heat pump is mixed before being cooled down in the flue gas cooler. The flue gas is then separated from the water condensed during gas cooling, which can be reused afterwards. Initially, the absorption heat pump is not explicitly modelled. However, since all design parameters are known, the latter can be seen as a black box where only the outlets are implemented in the model (i.e. the water mass flow entering the gas boiler and the flue gas leaving the heat pump generator). To ensure consistency with the reference article, the power values listed in Table 1 are matched by adjusting the fuel mass flow rates in both the gas boiler and the heat pump.

Several key pieces of information were not specified in the reference article, so assumptions were made. First, the composition of the natural gas used was not provided. Therefore, the Beijing natural gas composition reported in [21] was adopted, which corresponds to 93.77% CH_4, 3.36% C_2H_6, 0.59% C_3H_8, 0.21% C_4H_{10}, 0.06% C_5H_{12}, 1.49% CO_2, and 0.52% N_2. The fuel is injected at a temperature of 20 °C and 1 bar. Secondly, it is assumed that the same air excess of 20% is applied in the absorption heat pump generator. Finally, the sprayed water cycle, which serves as the heat source for the absorption heat pump, is considered a black box and has not been modelled. The water mass



(a)



(b)

Figure 6: Operating with a constant excess air ratio in the combustion chamber results in a constant oxygen mass flow rate in the oxidiser. However, since the combustion air is close to saturation before entering the gas boiler, a significantly higher water injection rate is required as T_{hum} increases. This increase in the H_2O mass fraction leads to a corresponding decrease in the O_2 mass fraction in the combustion air.

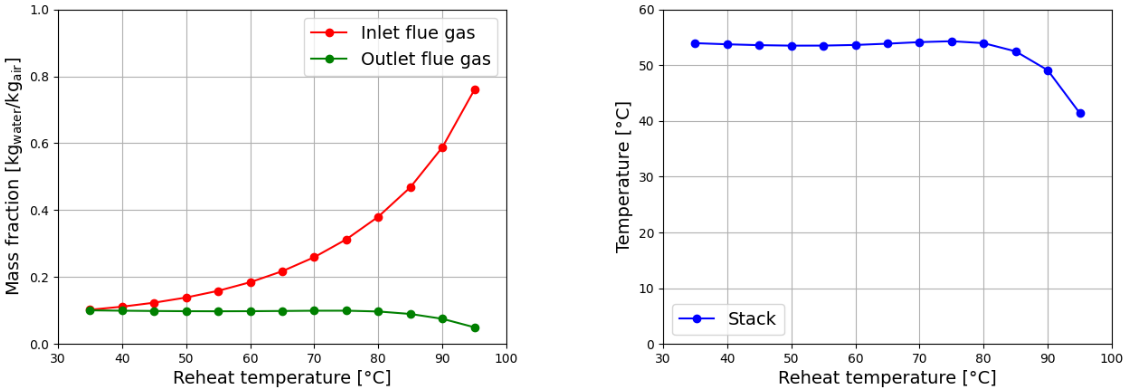
flow through the condenser is simply imposed to match the gas boiler power. Those assumptions were verified by replicating the reference system of Fig. 1 in Aspen Plus. Results were showing a relative error of less than 1% on the volume flow rates of the flue gases. Relative error on the recovered heat reaches 20.5% because of a lack of information on the DCHE unit. In Aspen Plus, a RADFRAC component working in "Equilibrium" was used in accordance with the methodology suggested in [22] to model the DCHE. Two stages are implemented, but as no information on the Murphree efficiencies of those stages has been given, the latter were set to 93% and 85% for the first and the second stage, respectively. This lack of information on the configuration leads to a lower condensing rate than the 70% given in the article.

3. Results

In this section, the performance of the IEC system will be compared to the reference system with a DCHE to evaluate the amount of heat that can be recovered from the flue gas. A sensitivity analysis on the outlet temperature of the wet channel (T_{hum}) is performed by varying its value from 35 °C to 95 °C. The upper limit is deliberately set below 100 °C, corresponding to the boiling temperature of water at 1 atm. Exceeding this threshold could lead to phase change within the wet channel, making it difficult to maintain fully saturated conditions at the end of the heating process and potentially affecting the reliability of the model. First, the impact of this variation on the combustion air is examined, focusing on the conditioning of atmospheric air before it enters the gas boiler. Then, the system performance is evaluated by examining the amounts of sensible and latent heat recovered and by computing the resulting global efficiency of the system. Finally, the indicators of performance defined in the methodology are evaluated, and their validity and feasibility are discussed.

3.1. Impact on the combustion air

The outlet temperature of the wet channel in the IEC is a critical parameter governing the performance of this type of heat exchanger, as it directly determines the amount of heat that can be extracted from the product channel. As T_{hum} increases, the amount of water contained in the oxidiser also rises, to reach saturation at the outlet of the wet channel. This trend is illustrated in Fig. 6a, where the total humid air mass flow rate increases due to additional water injection, while the oxygen flow rate remains constant because a fixed excess air ratio is maintained. Operating at a constant excess air ratio implies



(a) (b)

Figure 7: The introduction of water into the combustion air increases the H_2O mass fraction in the flue gas. This enhances the potential for latent heat recovery, as a larger amount of water vapour becomes available for condensation. Consequently, this additional heat recovery leads to a decrease in the stack temperature.

that the quantity of oxygen entering the combustion chamber does not change. However, the increasing water content leads to a dilution effect, reducing the oxygen mass fraction in the combustion air, as shown in Fig. 6b. For instance, while the oxygen mass fraction is about 22.5% at 35 °C, it decreases significantly to approximately 5.3% when the air temperature reaches 95 °C. To ensure saturation at the outlet of the wet channel, the amount of water needed grows exponentially as 70 °C is reached, as shown in Fig. 6a. As the fuel mass flow in the boiler is constant, the adiabatic flame temperature will be reduced due to the part of the energy used to heat the water content. Nevertheless, this effect does not compromise the system performance, as the flue gas temperature at the boiler outlet remains sufficiently high to supply the hot source of the heat pump. When T_{hum} is set to 85 °C, the flue gas temperature at the outlet of the combustion chamber still reaches 1534 °C.

3.2. Performance of the system

The increased water content in the flue gas leads to a rise in its absolute humidity, as illustrated in Fig. 7a. This enhances the potential for water vapour condensation and, consequently, latent heat recovery. This potential is represented by the area between the red and green curves in Fig. 7a, which expands exponentially with increasing T_{hum}. This phenomenon results in a noticeable decrease in the stack temperature, as depicted by the blue curve in Fig. 7b.
The heat recovered from the flue gas is computed and shown in Fig. 8, where the contributions of sensible and latent heat are detailed. The thermal capacity defined in [14] is also included. The reference DCHE system achieves a total heat recovery of 1.8 MW, consisting of 1.2 MW of latent heat and 0.6 MW of sensible heat, corresponding to a thermal capacity of 11.7%. As shown in Fig. 8, this reference is exceeded for T_{hum} = 60 °C, where the recovered heat reaches 2.3 MW, including 1.8 MW of latent heat and 0.5 MW of sensible heat. This results in a thermal capacity of 14.8%. Beyond this point, latent heat recovery increases sharply, reaching up to 46.2 MW if T_{hum} = 95 °C is achieved. The evolution of the thermal capacity closely follows the trend of the recovered heat. Under the assumption that T_{hum} can reach 95 °C, the thermal capacity could increase to as much as 310.1%.

3.3. Feasibility of the performance

The feasibility of the heat recovery can be assessed by looking at the different effectivenesses presented earlier. However, their validity have to be confirmed regarding to our particular case of industrial waste heat recovery with a M-saturator type IEC. On the dry channel, the dew point effectiveness (ϵ_{dew}) has been fixed to 80%. The whole concept of using indirect evaporative cooling over direct evaporative

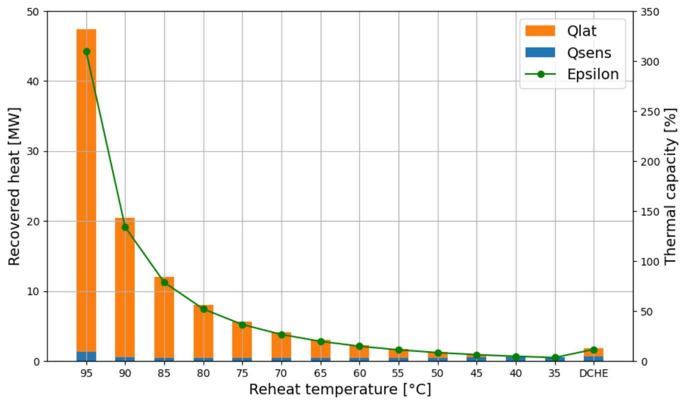


Figure 8: As T_{hum} increases, the amount of latent heat recovered also rises, since a larger quantity of water vapour is available for condensation in the flue gas. This results in a higher recovered heat flux for the same natural gas input, thereby leading to an increased thermal capacity.

cooling results in the use of this dew point effectiveness, instead of the wet bulb effectiveness. To achieve the same cooling temperature with a DCHE, its effectiveness would have to be higher, as shown in Fig. 9, since the temperature can only be decreased to the wet bulb temperature.

The dew point effectiveness can be expressed on the product channel to evaluate the performance of the cooling process. Values exceeding 100% are reached, with a maximum effectiveness of 429.8% when T_{hum} is set to 95°C. This is due to the condensation occurring in the product channel. In the definition of the dew point effectiveness, the minimum attainable temperature is the dew point temperature of the fluid. However, when condensation occurs, the outlet temperature of the product channel can drop below this minimum temperature, resulting in a value greater than 100%. This effectiveness must thus be seen as a performance parameter of the heat recovery and not of the IEC.

To asses the true performance of the IEC, ϵ_{IEC} can be considered, where the minimum temperature is defined as the dew-point temperature of the atmospheric air. The value ranges from a maximum of 67% to a minimum of 57%. Compared with the results reported in [20], the obtained values of ϵ_{IEC} are somewhat lower. This could be improved in future work by remodelling the wet channel and introducing staggered injection and reheat.

4. Conclusion

In this paper, we present a combined gas boiler and absorption heat pump system with an integrated indirect evaporative cooler (IEC) for flue gas latent heat recovery, and compare it with a direct contact heat exchanger (DCHE). The reference system is presented, and the thermal capacity is introduced. The modelling of the heat exchanger is then detailed, highlighting the key role of the wet channel outlet temperature in determining IEC performance. The system with an IEC included is then modelled to compare its performance with the reference technology. Performance indicators such as thermal capacity and IEC effectiveness are evaluated through a sensitivity analysis on the wet channel outlet temperature (T_{hum}). The results show that increasing this temperature leads to a higher amount of injected water and, consequently, an increase in the absolute humidity of the flue gas. As a result, the potential for latent heat recovery becomes more significant as T_{hum} increases. The performance of the DCHE system is exceeded for T_{hum} = 60 °C, where 2.3 MW of heat is recovered, including 1.8 MW of latent heat and 0.5 MW of sensible heat. When the temperature reaches 95 °C, up to 47.4 MW of waste heat can be recovered. Under these conditions, the thermal capacity reaches 14.8% and 310.1%,

exceeding the reference value of 11.7% for the DCHE. The validity of these results is supported by the IEC effectiveness, which reaches 57% and 67.1%, respectively.
Future work will focus on a more physical and realistic modelling of the behaviour of the wet channel. For the moment, a single injection and reheat were implemented in Aspen Plus. This will be modified to use staggered injection to ensure saturation throughout the entire process. The dew point (ϵ_{dew}), the thermal capacity (ϵ) and the IEC effectiveness (ϵ_{IEC}) will also be reevaluated to ensure physical feasibility of the heat exchanger. After that, a black box method will be developed to identify the feasibility limit and the potential of a given industrial application. In the long term, the development and manufacturing of a new heat exchange device will be studied.

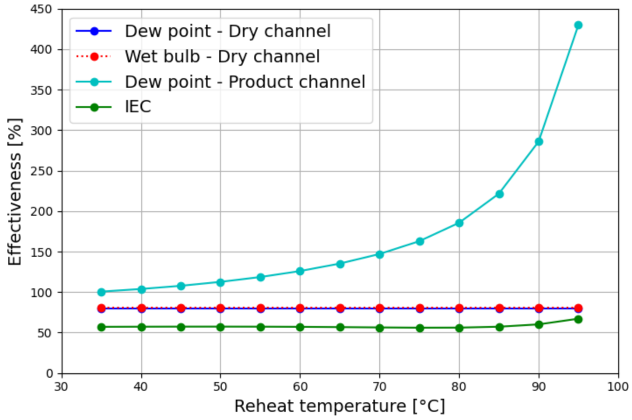


Figure 9: The dew point effectiveness remains constant, as it has been fixed by assumption. To achieve the same cooling with a DCHE, the wet bulb effectiveness has to be higher. The dew point effectiveness written on the product channel highlights the condensation occurring and must be seen as a performance parameter of the heat recovery, and not of the IEC. The IEC effectiveness provides a clearer image of the true performance of the heat exchanger.

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