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Thermal efficiency of coal based power plants: from theoretical to practical assessments

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Abstract

The improvement in thermal efficiency for coal to power processes is increasingly important due to concerns on CO_2 emissions. This paper presents a systematic study on direct combustion coal to power processes with respect to thermodynamic, technical and economic factors. Traditional exergy analysis focuses on irreversibilities in existing processes, while the new methodology investigates the thermal efficiency from its theoretical maximum to practical values by adding irreversibilities one by one. As a result of the study presented in this paper, various measures for increasing the thermal efficiency are investigated and the corresponding improvement potential is presented. For a reference power plant, the exergy of the coal feed is calculated to be 1.08 times the lower heating value, while the actual thermal efficiency is 45.5% when irreversibilities for the combustion reaction, the heat transfer between flue gas and water/ steam, the low temperature heat losses, the steam cycle, and other

factors are included. Different measures to increase the thermal efficiency of the reference plant by 0.1% points are presented. The minimum thermal efficiency penalty related to CO_2 capture is 2.92-3.49% points within an air factor range of 1.0-1.4 when the CO_2 is 100% recovered.

Keywords: coal based power plant; benchmarking; exergy analysis; CO_2 capture.

1. Introduction

Coal will continue to be a dominant energy source also in the next decades. It was responsible for 41% of the world power generation in 2012 and is projected to be around 31% in 2040[1]. Coal based power plants have been developed for more than 100 years with respect to the capacity and thermal efficiency. The plant thermal efficiency has increased continuously from around 5% to 45% in the past century[2]. Reducing cost for power generation has always been a motivation for efficiency improvement. The increasing concerns about CO_2 emissions stimulate further improvements in thermal efficiency. In direct combustion coal to power processes, the chemical energy of coal is converted into heat and this heat is further converted into power. Considerable efforts have been made to improve the thermal efficiency, such as reducing the irreversibilities in the process that converts the chemical energy of coal into heat[3], maximizing power production from the heat[4] and minimizing the losses of low temperature heat[5]. For pulverized coal based power plants, the long-term target on thermal efficiency is above 55% by using steam with maximum temperatures around 1073 K ($800^{o}C$)[5].

The thermodynamic principles of coal based power plants (mainly steam cycles) have been described in many textbooks related to thermodynamics and power technologies[5-10]. Various measures for improving the plant performance have also been presented in these books as well as in many other publications. The primary objective of this paper is to

investigate the improvement potential in thermal efficiency and the corresponding limitations for such measures presented in literature. The paper is an extension of the work by Fu et al.[11] The study starts by calculating the maximum thermal efficiency for a specific coal feed in an ideal (reversible) power plant. This efficiency will decrease when realistic (irreversible) unit operations are added for the combustion process, the heat transfer process, the steam cycle, and the flue gas treatment (CO_2 emission control). The thermodynamic losses (irreversibilities) are caused by spontaneous processes such as combustion, as well as heat transfer at finite (often large) temperature differences, mixing, pressure drops, and turbo-machinery inefficiencies. In addition, the thermal efficiency is limited by technical and economic factors, such as excess air for combustion, maximum pressure and temperature of the main steam, and low temperature heat losses. The influence of these limiting factors on the thermal efficiency has been investigated. For the reference plant, the measures for increasing the thermal efficiency by 0.1% points are investigated. The minimum energy penalty with respect to thermodynamic limitations for capturing CO_2 at various purities and recovery rates is also studied. The results can be used as a basis for evaluating the thermal efficiency of plants where CO_2 capture will be implemented in the future, and also the efficiency improvement measures.

2. Methodology

A methodology for benchmarking and identifying improvement potentials of processes was presented by Anantharaman et al.[12] The motivation for the new methodology was to develop a systematic and consistent way to identify improvement potential and integration opportunities in power processes with CO_2 capture. To this end, three efficiencies that can be specified for a process are[12]:

(1) Thermodynamics limited: This is a scheme that requires the thermodynamically lowest possible energy input to produce the specified energy output. The resulting efficiency is the "ideal" efficiency that is the thermodynamically maximum attainable for such a process. This efficiency can never be achieved in practice since it requires perfectly reversible processes, however, it provides a thermodynamic benchmark or target for process design.

(2) Technology limited: Limitations, both technological and those inherent in unit operations, prevent achieving the thermodynamic maximum efficiency. The thermal efficiency attainable by employing state-of-the-art technology can be thought of as a technology limited efficiency, which is typically compared in different benchmarking studies[13].

(3) Economics limited: While the technology limited efficiency by definition is achievable, it may not necessarily be economical. Latest technologies are often associated with a premium, which makes utilizing them economically infeasible. Thus the economics limited efficiency is the efficiency of a process using technology that results in a process that is commercially viable.

Power plants with CO_2 capture can be benchmarked with respect to the three above-mentioned efficiencies. It must be noted that while the thermodynamic limited efficiency is fixed for a given process, the technology limited and economics limited efficiencies are subject to change over time. The difference between the thermodynamic maximum and the technology limited efficiencies quantifies the theoretical improvement potential and constitutes an additional source of information for benchmarking studies, which merits further attention. The source(s) of this difference in efficiency can point to possible future directions for technology development.

The approach in this paper consists of applying engineering thermodynamics to increase the understanding of the fundamental losses imposed on a power cycle. The first step in the methodology is to evaluate the maximum efficiency limited by thermodynamics. This limit is achieved by defining an ideal (reversible) process. A set of non-idealities in the form of technological limitations are added systematically in series to go from the thermodynamics limited to the technology limited cases. The difference between the thermodynamics limited and technology limited efficiencies can thus be attributed to the different sets of irreversibilities and quantified. This is represented visually in Figure 1.

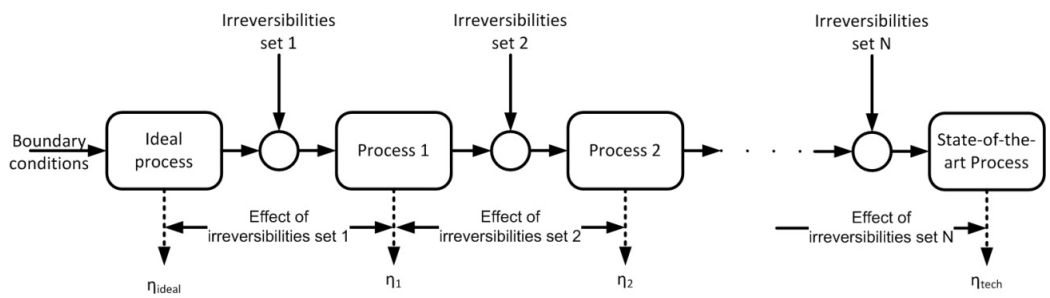


Figure 1. Representation of the systematic methodology for benchmarking

This methodology is applied to a coal fired power plant to identify and quantify the sources and scope for improvements.

3. The reference plant

A 754 MWe pulverized coal based power plant has been used as the reference for a benchmarking study [14]. A simplified flowsheet with exergy flow data are shown in Figure . The fuel is Bituminous Douglas Premium coal, and the air factor is 1.22. The main parameters of the reference plant are presented in Table A1 (Appendix). The coal characteristics and the composition of the atmospheric air are listed in Tables A2 and A3 respectively.

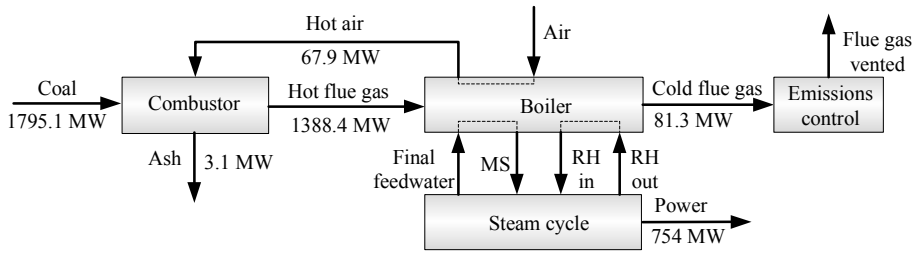


Figure 2 The reference coal based power plant with exergy flows

The procedure for calculating the chemical exergy of substances, the exergy of process streams and the exergy balances for process units are described in the literature[15, 16]. The reference state (marked as "0") is T_0=15°C, p_0=1.01 bar (i.e. 1 atm). The standard chemical exergy of pure substances, e_{ch}^0 [kJ/mole], can be found in Szargut[16] and converted into the corresponding reference state. The chemical exergy of a material stream $\dot{E}_{ch}$ [kW] is calculated using Eq. (1).

$$\dot{E}_{ch} = \dot{F}\sum_i (x_i e_{ch,i}^0) + \dot{F}\bar{R}T_0 \sum_i (x_i \ln x_i) \tag{1}$$

where $\dot{F}$ [mole/s] is the molar flow of the stream and x_i is the molar fraction of component *i*.

The physical (thermo-mechanical) exergy of a stream $\dot{E}_{ph}$ [kW] is equal to the amount of work arising when changing a stream reversibly from process conditions (T, p) to the reference conditions (T_0, p_0), and is calculated by Eq. (2) for the general case of multiple phases.

$$\dot{E}_{ph} = \left[\sum_j (\dot{F}_j h_j) - \sum_j (\dot{F}_{0,j} h_{0,j})\right] - T_0 \left[\sum_j (\dot{F}_j s_j) - \sum_j (\dot{F}_{0,j} s_{0,j})\right] \tag{2}$$

Here, h [kJ/mole] and s [kJ/(mole·K)] are the molar enthalpy and entropy of the stream, and *j* is the phase index.

The total exergy of a stream $\dot{E}_{tot}$ [kW] at process conditions (T, p) is given as:

$$\dot{E}_{tot} = \dot{E}_{ch} + \dot{E}_{ph} \tag{3}$$

The irreversibility of a unit operation, $\dot{I}$ [kW], can be calculated by:

$$\dot{I} = \dot{E}^{in} - \dot{E}^{out} \tag{4}$$

where $\dot{E}^{in}$ [kW] and $\dot{E}^{out}$ [kW] are the exergy entering and exiting the unit respectively.

The theoretical minimum work required for a unit operation, $\dot{W}_{min}$ [kW], is equal to the difference in exergy between the products and the feeds, and can be calculated by Eq. (5)[15].

$$\dot{W}_{min} = \sum_{product} \dot{E}_{tot}^{product} - \sum_{feed} \dot{E}_{tot}^{feed} \tag{5}$$

Here, $\dot{E}_{tot}^{product}$ [kW] is the exergy of a product stream and $\dot{E}_{tot}^{feed}$ [kW] is the exergy of a feed stream.

According to the methods described by Szargut[16], the chemical exergy of the coal feed (Table A2) is calculated to be 27,295 kJ/kg using Eq. (6).

$$e_{ch,coal}^{0} = \varphi(\mathrm{LHV})^{0} \tag{6}$$

Here, $(\mathrm{LHV})^{0}$ [kJ/kg] is the lower heating value of coal at the reference conditions (T_0, p_0), and φ is the ratio of the chemical exergy to the lower heating value, calculated by[16]:

$$\varphi = 1.0437 + 0.1896(h/c) + 0.2499(o/c) + 0.0428(n/c) \tag{7}$$

where c, h, o and n are the mass fractions of carbon, hydrogen, oxygen and nitrogen in the ultimate analysis of coal (dry basis) respectively.

For the reference plant, the exergy of the coal feed is calculated to be 1,795.1 MW and the thermal input (LHV) is 1,655.3 MW. After combustion, the total exergy of the flue gas is 1,388.4 MW. The mass fraction of the major components of the ash is: SiO_2-0.45; Al_2O_3-0.3; CaO-0.07; Fe_2O_3-0.03; SO_3-0.035[5]. The chemical exergy of ash is calculated to be 330 kJ/kg, equal to 0.17% of the exergy of the coal feed (3.1 MW), and is thus negligible. The exergy values for main streams are presented in Figu

4. Assessments on the thermal efficiency

For the coal feed, the maximum work output is equal to its chemical exergy if the coal to power process is reversible. The chemical exergy of the coal feed is calculated to be 1.08 times the lower heating value, while the thermal efficiency of the reference plant is 45.5%. Ideal reversible processes for reaction, separation and heat transfer are presented in the literature[15]. Such processes are infeasible in practice with respect to limitations in technologies, investment cost and plant lifetime. Proper driving forces are necessary and thus causing irreversibilities. The oxidation (combustion) of coal, the heat transfer between the flue gas and the working fluid of the power cycle, and the power cycle itself are the major sources of irreversibilities. The following sections illustrate how the thermal efficiency is reduced from its theoretical maximum to practical values by thermodynamic, technical and economic factors.

4.1. Combustion losses

The chemical energy of coal is released when the coal is oxidized to CO_2 and H_2O. Direct combustion with air is the most common way for the oxidation of coal. In the case of complete stoichiometric combustion, the adiabatic flame temperature is calculated to be 2,332 K. The exergy destruction related to stoichiometric combustion is determined to be 30.4% of

the exergy of the coal. The chemical exergy of the flue gas, which corresponds to 3.5% of the exergy of the coal feed, has not yet been recovered by current technologies (unless the flue gas is further utilized as feed for other processes instead of being vented), meaning that the theoretical work that can be recovered from the flue gas is 66.1% of the exergy of the coal feed. The maximum thermal efficiency that can be obtained is thus 71.4% (determined as the ratio between the theoretical work that can be recovered from the flue gas (physical exergy) and the thermal input).

At temperatures higher than 1,250 K, components such as CO_2 and H_2O will dissociate[17]. The equilibrium temperature (the temperature when chemical equilibrium is achieved) is obtained as 2,229 K, around 100 K lower than the adiabatic flame temperature of complete combustion. The exergy destruction is the same as for the complete combustion case. However, the chemical exergy of unburned CO and H_2 in the flue gas is not expected to be recovered when the flue gas is cooled since the chemical equilibrium can not be achieved in finite time.

4.1.1. Thermodynamic analysis of the combustion processes

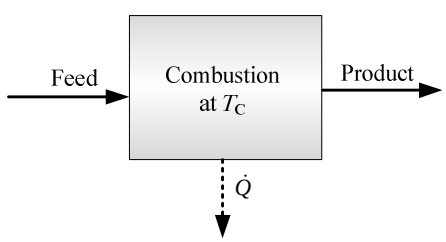


Figure 3 An isothermal combustion process

For an isothermal combustion process as shown in Figure 3, the entropy balance and energy balance can be written as:

$$\dot{S}_{\text{generation}} = \dot{S}_{\text{product}} - \dot{S}_{\text{feed}} + \dot{Q} / T_C = \Delta\dot{S} + \dot{Q} / T_C \tag{8}$$

$$\dot{Q} = \dot{H}_{\text{feed}} - \dot{H}_{\text{product}} = -\Delta\dot{H} \tag{9}$$

where $\dot{S}_{\text{generation}}$ is the entropy generation caused by combustion irreversibilities, $\dot{S}_{\text{product}}$ and $\dot{S}_{\text{feed}}$ are the entropy of the product and the feed, $\Delta\dot{S}$ is the entropy difference between the product and the feed, $\dot{Q}$ is the heat removed from the system, T_C is the operating temperature, $\dot{H}_{\text{feed}}$ and $\dot{H}_{\text{product}}$ are the enthalpies of the feed and the product, and $\Delta\dot{H}$ is the enthalpy difference between the product and the feed.

If the process is reversible, $\dot{S}_{\text{generation}} = 0$, then $T_C = \Delta\dot{H} / \Delta\dot{S}$. This temperature is defined as the combustion temperature[18]. The change in Gibbs free energy $\Delta\dot{G}$ is zero at T_C [19]. This temperature (T_C) can be understood as the one where the combustion reaction can take place reversibly, i.e. the (practically impossible) condition is necessary to be maintained for the entire combustion process. The changes in the enthalpy and entropy differences between the product and the feed with the temperature are negligible[3, 18, 19], thus $T_C \approx \Delta\dot{H}_0 / \Delta\dot{S}_0$, where $\Delta\dot{H}_0$ and $\Delta\dot{S}_0$ are the enthalpy and entropy differences between the product and the feed for a reaction at ambient temperature T_0. The combustion temperature of bituminous coal can be as high as 27,466 K[3], which is much higher than the stoichiometric adiabatic combustion temperature (2,332 K). In addition, the isothermal condition is impossible to maintain in practice. Irreversibilities will also be introduced when the reactants are heated from ambient temperature to the combustion temperature and the products are cooled from combustion temperature to ambient temperature. Thus the irreversibilities for the combustion process (air is used as oxidant) are considerable.

In the case of adiabatic combustion of H_2 and CH_4, the irreversibilities are mainly caused by: (1) combined diffusion/fuel oxidation, (2) internal thermal energy exchange, and (3) the product mixing process[20]. The internal thermal energy exchange is responsible for more than 2/3 of the total losses. Similar results have also been observed for the combustion of carbon[21], i.e. internal thermal energy exchange and the chemical reaction (fuel oxidation) are responsible for the major losses.

4.1.2. Reducing combustion irreversibilities

The following options are available to reduce the combustion irreversibilities:

(1) *Converting coal into syngas by gasification (CO and H_2).* The gasification process has lower exergy destruction than the combustion process[21]. The syngas is further converted into H_2 that can be oxidized in fuel cells. The fuel cells can achieve a high thermal efficiency at a much lower temperature compared to heat engines[19]. However, current gasification processes are not competitive to direct combustion with respect to economic considerations. Large scale implementation of fuel cells is also indeed a technical challenge.

(2) *Shifting direct combustion processes to chemical looping combustion (CLC).* Metal oxides can be used as the oxidant. The combustion temperature T_C of CLC processes can be reduced to feasible levels (e.g. lower than 1,000 K)[3], thus the irreversibilities related to the reaction are reduced. The irreversibilities caused by internal thermal energy exchange and product mixing can also be reduced. Further technology developments are required to implement the CLC technology and much research is ongoing[22].

(3) *Increasing the operating temperature (adiabatic temperature).* The exergy destruction related to combustion is lower at higher operating temperatures. Theoretically, very high

operating temperatures can be achieved by preheating the air feed. However, the temperature is limited by materials of construction. In addition, the dissociation of CO_2 and H_2O increases at high temperature. The CO and H_2 can not be completely oxidized into CO_2 and H_2O when the flue gas is cooling down due to limited time for heat exchange.

(4) *Reducing the exergy destruction related to internal thermal energy exchange*. The preheating of air can reduce such losses.

(5) *Reducing the exergy destruction related to product mixing*. Such losses are small, but can still be reduced by reducing the air factor (defined as the ratio of the actual air feed to the stoichiometric air feed) or using pure O_2 or metal oxides. Lower air factors, however, can result in incomplete combustion of coal.

The possible change in the irreversibilities of other units should also be taken into consideration when the combustion irreversibilities are reduced. For the reference power plant, the following three practical measures are investigated: (i) preheating the air (referred to items (3) and (4) above), (ii) reducing the air factor (referred to items (4) and (5) above) and (iii) using pure O_2 (referred to items (3), (4) and (5) above).

Figure 4 shows the exergy destruction of the combustion process (as percentage of the exergy of the coal feed) and the adiabatic flame temperature at different air feed temperatures and air factors (referred to the numbers for each curve in the figure). The results are obtained using two models: (1) complete combustion and (2) chemical equilibrium. The exergy destruction is almost the same for the two combustion models. This is explained by relatively low T_C for CO and H_2 (3,276.5 K and 5,454 K respectively according to literature[3]). The adiabatic flame temperature is not far away from T_C, thus the exergy destruction is very small if the unburned CO and H_2 (determined by the equilibrium model) are burned completely.

However, the chemical exergy of the unburned CO and H_2 may be lost in the stack if the CO and H_2 have not been oxidized when the flue gas is cooled.

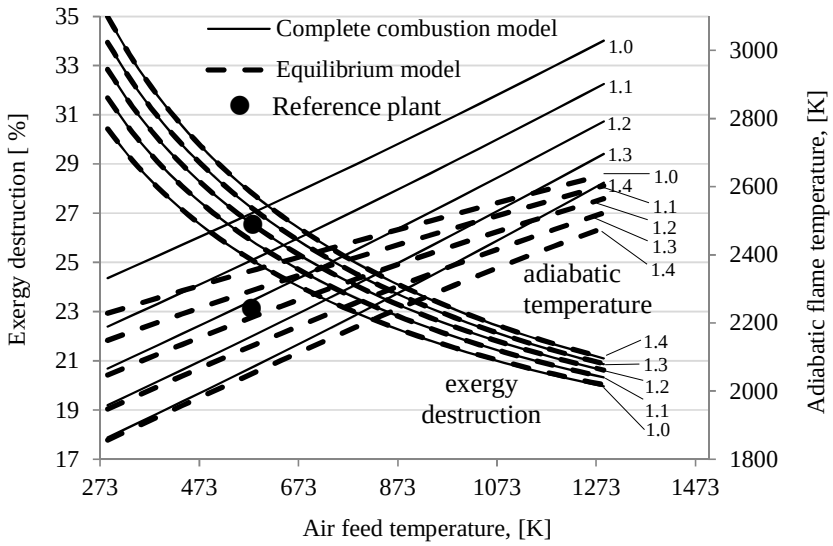


Figure 4 Exergy destruction due to combustion [% of the exergy of the coal]

Preheating of air can reduce the losses caused by internal thermal energy exchange and combustion reactions. The exergy destruction is reduced from 30.4% to 20.0% when the air is preheated from 288 K to 1,288 K in the case of stoichiometric combustion. Without any preheating of air, the exergy destruction increases from 30.4% to 35.0% when the air factor increases from 1.0 to 1.4. The excess air reduces the adiabatic flame temperature, thus increases the exergy destruction caused by combustion reactions. The exergy destruction related to mixing and the internal thermal energy exchange also increases. When the air is preheated, the influence of the air factor on the exergy destruction is reduced. When the air factor is 1.22 (the reference plant) in the case of no preheating, the exergy of the combustion product is 67.0% of the exergy of the coal feed, where the physical exergy and the chemical exergy contribute 63.0% and 4.0% respectively.

When pure O_2 is used for combustion (oxy-combustion), the exergy destruction related to the combustion reaction, mixing (N_2 is not present) and internal thermal energy exchange (no heating of N_2) is reduced. In case of stoichiometric combustion without preheating of air, the exergy destruction for the combustion process is calculated to be 20.5% of the coal feed. However, the extremely high adiabatic flame temperature (5,495 K) requires dilution. In addition, the production of pure O_2 introduces air separation units and thus new losses. When CO_2 capture is included, however, oxy-combustion is a promising alternative[23].

4.2. Low temperature heat losses

In the reference power plant, the exhaust flue gas temperature is 393 K. The physical exergy of the flue gas is calculated to be 0.96% of the thermal input. The reduction in low temperature heat wasted or recovery of this heat is an important way to improve the thermal efficiency. The low temperature heat losses can also be reduced by decreasing the air factor and the flue gas temperature. Organic Rankine Cycles (ORCs) and some other cycles have been proposed for recovering low temperature heat[24]. This topic is, however, not discussed in detail in this paper.

The influence of the air factor (denoted by *f*) on low temperature heat losses can be explained using Figure 5. The energy balance is assumed to be satisfied for the case of stoichiometric combustion (*f*=1), and is represented by the streams in solid lines. Note that the heat losses related to surface radiation and convection and the steam losses due to boiler blowdown and surface blowoff for impurity removal, and any other steam losses are neglected. The heat loss caused by the ash is also neglected. If the air factor increases from 1 to *f*, more coal ($\dot{m}'_{coal}$) is burned in order to heat the additional flue gas ($\dot{m}'_{FG}$) from ambient temperature to the flue gas temperature. The dashed lines show the combustion of the additional coal. Note that the flue gas represented by dashed lines includes the products of the combustion of the additional

coal and the excess air. The mass balance should be satisfied for the dashed streams, as shown by Eq. (10).

$$\dot{m}'_{\text{coal}} + \dot{m}'_{\text{air}} = \dot{m}'_{\text{FG}} + \dot{m}'_{\text{ash}} \tag{10}$$

Here $\dot{m}'_{\text{air}}$ is the mass flow of the additional air to be used, $\dot{m}'_{\text{air}} = f\omega\dot{m}'_{\text{coal}} + (f-1)\omega\dot{m}_{\text{coal}}$, where ω is the stoichiometric ratio ($\omega = 8.8122$ for the coal feed). $\dot{m}'_{\text{ash}}$ is the mass flow of the additional ash produced, thus $\dot{m}'_{\text{ash}} = 0.1415\dot{m}'_{\text{coal}}$ (according to Table A2). Eq. (10) can be rearranged as:

$$\dot{m}'_{\text{FG}} = (0.8585 + 8.8122f)\dot{m}'_{\text{coal}} + 8.8122(f-1)\dot{m}_{\text{coal}} \tag{11}$$

According to the definition of the lower heating value (LHV), the following energy balance can be obtained:

$$\dot{m}'_{\text{coal}}(\text{LHV}) = \dot{m}'_{\text{FG}}(h'_{\text{FG},T_{\text{FG}}} - h'_{\text{FG},T_0}) \tag{12}$$

where $h'_{\text{FG},T_{\text{FG}}}$ and h'_{FG,T_0} are the specific enthalpies of the additional flue gas at exhaust temperature (T_{FG}) and ambient temperature (T_0).

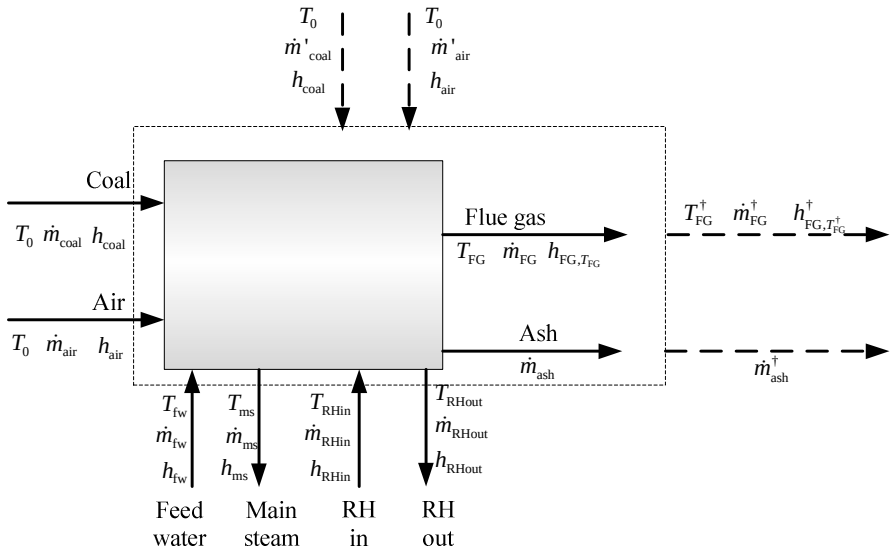


Figure 5 Mass and energy balances for investigating the influence of the air factor (*f*)

For a given T_{FG}, the influence of the air factor on the thermal efficiency can be investigated based on Eqs. (11) and (12). Similarly, the influence of the exhaust flue gas temperature on the thermal efficiency can also be investigated. Figure 6 shows the mass and energy balances when the flue gas temperature changes from T_{FG} (the box in solid lines) to $T^{\dagger}_{FG}$ (the box in dashed lines). Note that the flue gas represented by the dashed lines includes the gas products resulting from the burning of the two portions of coal feed ($\dot{m}_{coal}$ and $\dot{m}'_{coal}$). More (or less) coal is consumed when the flue gas temperature increases (or decreases) in order to maintain the energy balance. The following energy balance can easily be obtained from Figure 6.

$$LHV(\dot{m}'_{coal}) = \dot{m}^{\dagger}_{FG}(h^{\dagger}_{FG,T^{\dagger}_{FG}} - h_{FG,T_0}) - \dot{m}_{FG}(h_{FG,T_{FG}} - h_{FG,T_0}) \quad (13)$$

where $h_{FG,T_{FG}}$ and h_{FG,T_0} are the specific enthalpies of the flue gas at exhaust temperature (T_{FG}) and ambient temperature (T_0). For a given *f*, the influence of the flue gas temperature on the thermal efficiency can then be determined.

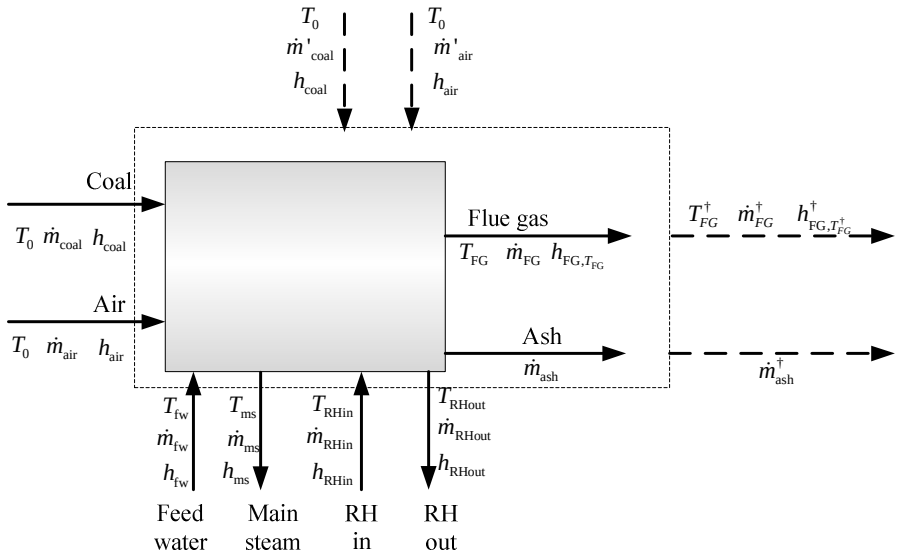


Figure 6 Mass and energy balances for investigating the influence of the flue gas temperature

The results are illustrated in Figure 7. When the flue gas temperature is 393 K (the reference plant), the thermal efficiency increases from 45.2% to 45.9% when the air factor is reduced from 1.4 to 1.0. The thermal efficiency increases by 0.1% points on average when the air factor is reduced by 0.057. When the air factor is 1.22 (the reference plant), the thermal efficiency increases from 45.5% to 46.6% if the flue gas temperature decreases from 393 K to 345 K. Such reduction in flue gas temperature can be achieved by using flue gas as heat source in air preheaters that can withstand the acid corrosion[5]. The spread and distribution of the flue gas from the stack limits the flue gas temperature to 345 K[5]. If the flue gas could be further cooled to 308 K (this temperature is limited by the temperature driving forces of heat exchangers), the thermal efficiency would increase to 47.4%. It is found that for every reduction of 4.5 K in flue gas temperature, the thermal efficiency increases by ~0.1% points. The air factor has negligible influence on the thermal efficiency when the flue gas is cooled to ambient temperature. This is reasonable since the heat losses are very small when the flue gas exits at around ambient temperatures.

It should be noted that the thermal efficiency increases by around 2% points (according to Figure 7) when the low temperature heat is completely recovered in the boiler system, but by a maximum of 0.96% points (corresponding to the physical exergy of the flue gas) if the low temperature heat is converted into work by additional ORCs. This difference is explained by the very low quality (exergy) of the low temperature heat and thus the very low efficiency of ORCs. When the low temperature heat is recuperated in the boiler system, less coal is burnt and thus the irreversibilities are reduced. The capital cost is also a considerable challenge for implementing ORCs. Thus, it is reasonable to put efforts into reducing low temperature heat losses from the boiler system before using ORCs to recover work from the low temperature heat. This can be implemented simply by increasing the size of the air preheater (reducing the temperature difference at the pinch point of the air preheater), resulting in more low temperature heat being recirculated into the burner.

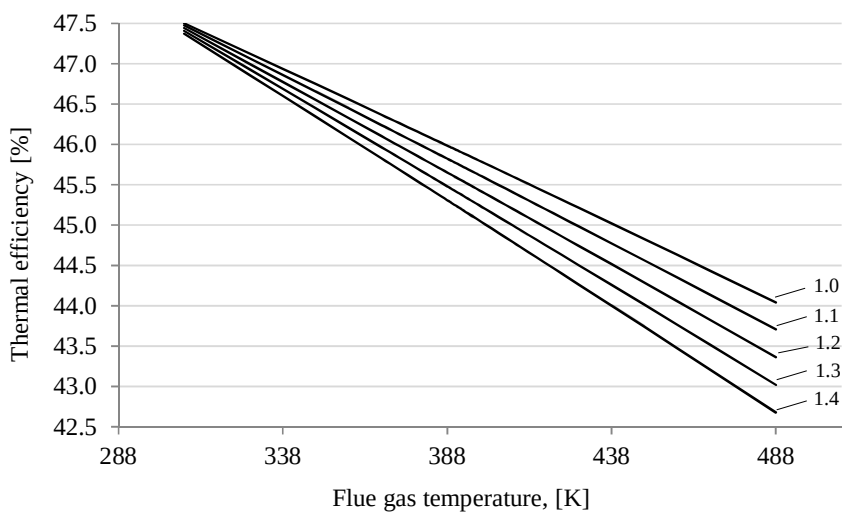
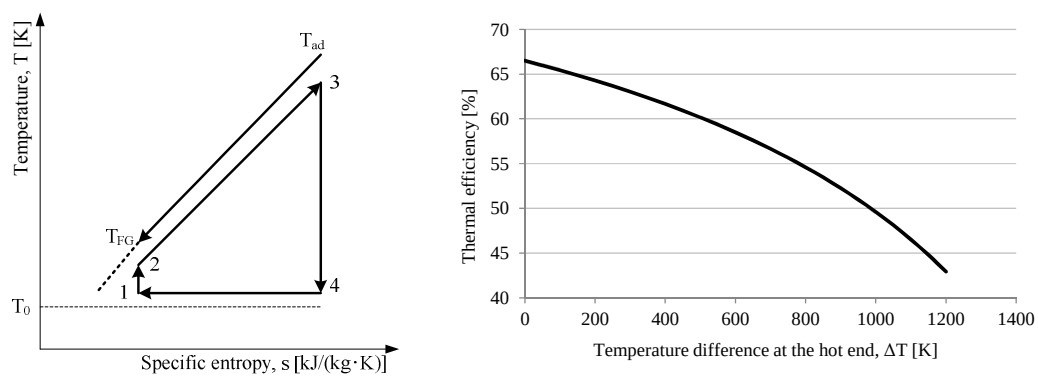


Figure 7 Influence of the air factor and flue gas temperature on thermal efficiency

4.3. Heat transfer between the flue gas and the working fluid

Combustion heat is normally converted into work by heat engines. The efficiency of heat engines is limited by the Carnot efficiency. An ideal working fluid receives heat from the flue

gas with small temperature differences and rejects it to the ambient environment (normally to cooling water). Such a process is illustrated in Figure 8(a) and explained as following: the fluid is reversibly heated by the flue gas (2-3) after being isentropically compressed (1-2), and then reversibly cooled (4-1) at ambient temperature after being isentropically expanded (3-4). In the case of reversible heat transfer, the maximum work output is equal to the changes in exergy for the flue gas between the adiabatic flame temperature (T_{ad}) and the exhaust temperature (T_{FG}). For the reference plant, the physical exergy of the flue gas is 63.0% of the exergy of the coal, thus the maximum thermal efficiency (referred to LHV) is 68.3%.



(a) (b)

Figure 8 Ideal heat engine: (a) T-s diagram, (b) Thermal efficiency

If the heat capacities of the flue gas and the working fluid are assumed to be constant, and assuming $T_2 = T_1 = T_0$ and $T_{\mathrm{FG}} = 308.15$ K, the plant thermal efficiency is determined by the temperature difference in the hot end ($\Delta T = T_{\mathrm{ad}} - T_3$). Figure 8(b) illustrates the influence of ΔT on the thermal efficiency. The thermal efficiency decreases from 66.5% to 44.8% when ΔT increases from 0 K to 1,150 K. For $\Delta T = 1150$ K, T_3 is equal to 878.6 K. This is very close to the main steam temperature (873 K) in the reference plant, the thermal efficiency of which is 45.5%. When T_3 is 1,073 K (800°C, corresponding to ultra-supercritical steam cycles), the thermal efficiency is 51.0%.

In order to reduce the exergy destruction caused by heat transfer, it is useful to explore an ideal working fluid for the cycle “1-2-3-4-1”. Such working fluids are, however, not available. One alternative is to use a mixture of components as the working fluid (e.g. the Kalina cycle[25]). The boiling points are not constant in Kalina cycles, thus the exergy destruction related to heat transfer can be reduced. Another alternative is to use a combination of several cycles: liquid metal such as potassium and mercury is used for the high temperature range[4]; water is used for the medium temperature range; organic substances or CO_2 are used for the low temperature range. Gas turbine combined cycles are more commonly used, where the air/exhaust flue gas is used as working fluid for the higher temperature range, and water is used for the lower temperature range.

4.4. Thermal efficiency improvement for the steam cycle

Steam Rankine cycles are most commonly used in coal based power plants. The heat transfer between the flue gas and the steam causes considerable exergy destructions due to large temperature differences. Other irreversibilities are caused by the inefficiencies of steam turbines, pumps, and the heat transfer with finite temperature differences in the regenerative feedwater preheaters and the condenser.

The maximum temperature and pressure of the steam are mainly limited by the construction materials of steam generators and steam turbines[5]. The steam temperature is expected to reach 973 K (700°C) in the near term and 1,073 K (800°C) in the long term. With reference to Figure 5, the heat supplied to the steam cycle, $\dot{Q}_{\mathrm{SC}}$, is calculated by:

$$\dot{Q}_{\mathrm{SC}} = \dot{m}_{\mathrm{ms}} h_{\mathrm{ms}} + \dot{m}_{\mathrm{RHout}} h_{\mathrm{RHout}} - \dot{m}_{\mathrm{fw}} h_{\mathrm{fw}} - \dot{m}_{\mathrm{RHin}} h_{\mathrm{RHin}} \tag{14}$$

where $\dot{m}_{\mathrm{fw}}$, $\dot{m}_{\mathrm{ms}}$, $\dot{m}_{\mathrm{RHin}}$ and $\dot{m}_{\mathrm{RHout}}$ are the mass flows of boiler feedwater, main steam, cold reheating steam and hot reheating steam; h_{fw}, h_{ms}, h_{RHin} and h_{RHout} are the corresponding specific enthalpies of the streams. According to the data presented in Table A1 for the reference plant and steam properties, $\dot{Q}_{\mathrm{SC}}$ is calculated to be 1,567.6 MW. The boiler efficiency (the ratio of $\dot{Q}_{\mathrm{SC}}$ to the thermal input of the coal feed) is thus 94.7%.

The maximum work that can be produced from the steam cycle, $\dot{W}_{\mathrm{SC}}$, is calculated by:

$$\dot{W}_{\mathrm{SC}} = \dot{m}_{\mathrm{ms}} e_{\mathrm{ms}} + \dot{m}_{\mathrm{RHout}} e_{\mathrm{RHout}} - \dot{m}_{\mathrm{fw}} e_{\mathrm{fw}} - \dot{m}_{\mathrm{RHin}} e_{\mathrm{RHin}} \tag{15}$$

where e_{fw}, e_{ms}, e_{RHin} and e_{RHout} are the specific exergies of boiler feedwater, main steam, cold reheating steam and hot reheating steam. $\dot{W}_{\mathrm{SC}}$ is calculated to be 916.9 MW for the reference plant. Thus, the theoretical maximum thermal efficiency of the steam cycle (cycle efficiency) for the given parameter values for feedwater and steam is 58.5%. The maximum thermal efficiency of the entire plant is then calculated to be 55.4% (by including the boiler efficiency). This value is higher than the value (44.8%) predicted in Figure 8(b), since the boiler feedwater and the reheating steam are fed at temperatures much higher than ambient, while the boiler feedwater is fed at ambient temperature for the case shown in Figure 8.

4.4.1. Influence of main steam parameters on thermal efficiency

By considering Eq. (14), and assuming that reheating is not applied, the mass flow of the main steam is determined by the parameters (temperature and pressure) of the boiler feedwater and the main steam. The maximum work output from the steam cycle is calculated by Eq. (15). Thus, the maximum thermal efficiency of the entire plant can be calculated. The influence of the main steam parameters on the thermal efficiency is shown in Figure 9. Two feedwater temperatures are investigated: 308 K (corresponding to the outlet temperature of the

condenser in the reference plant; represented by the dashed lines) and 581 K (corresponding to the final feedwater temperature, FFWT, in the reference plant; represented by the solid lines). Since the benchmarking methodology in this paper is to evaluate the thermal efficiency by stepwise investigation from theoretical to practical values, the condensate from the condenser is assumed to be reversibly brought to final feedwater conditions. The work calculated by the exergy difference between the final feedwater and the condensate is included in the calculation of thermal efficiency. The pressure losses are neglected.

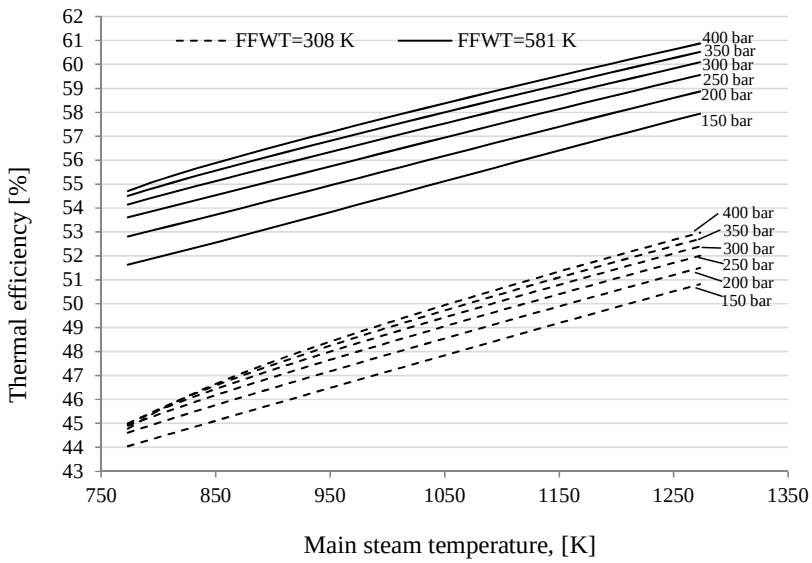


Figure 9 Influence of the main steam parameters on the thermal efficiency

The results in Figure 9 clearly illustrate the benefit from feedwater preheating that is today commonly applied in steam power plants. It can also be noticed that the effect of pressure is that the thermal efficiency decreases with increasing pressure. This can be explained by the smaller changes in specific entropy with pressure at higher pressures. Eq. (16) shows how specific entropies of the main steam and boiler feedwater (s_{ms} and s_{fw}) affect the physical (thermo-mechanical) exergy. The heat transferred to the steam cycle is fixed,

$\dot{Q}_{\mathrm{SC}} = \dot{m}_{\mathrm{ms}}(h_{ms} - h_{fw})$, thus the exergy transferred to the steam cycle, $\dot{E}_{\mathrm{SC}}$, is mainly influenced by the entropy differences of the main steam and the feedwater.

$$\dot{E}_{\mathrm{SC}} = \dot{m}_{\mathrm{ms}} e_{\mathrm{ms}} - \dot{m}_{\mathrm{fw}} e_{\mathrm{fw}} = \dot{m}_{\mathrm{ms}}[(h_{\mathrm{ms}} - h_{\mathrm{fw}}) - T_0(s_{\mathrm{ms}} - s_{\mathrm{fw}})] \tag{16}$$

When the feedwater temperature is 308 K (dashed lines in Figure 9) and the steam temperature is about 773 K, the steam pressure has negligible influence on the thermal efficiency in the case that the pressure exceeds 300 bar. The thermal efficiency increases almost linearly with temperature when the feedwater temperature is 581 K (solid lines in Figure 9). It increases from 54.1% to 60.0% when the steam temperature increases from 773 K to 1,273 K for a pressure of 300 bar. An average increase of 0.1% points in thermal efficiency is obtained for every increment of 8 K in the steam temperature. The thermal efficiency increases from 52.8% to 56.2% when the pressure increases from 150 bar to 400 bar for a steam temperature of 873 K. The thermal efficiency increases by approximately 0.1% points for every increment of 10 bar in the pressure range of 250-350 bar.

The thermal efficiency is 56.6-57.1% in the pressure range of 300-350 bar when the steam temperature is 973 K. This is the common target in the very near future based on the development of nickel-based alloys[24]. If the steam temperature reaches around 1,073 K, the thermal efficiency is 57.8-58.2% for the same pressure range. The thermal efficiency achievable is 60.9% at 400 bar and a steam temperature of 1,273 K (the very long-term target). However, the thermal efficiency presented in Figure 9 is the *theoretical* maximum efficiency without reheating. In practice the steam from the last stages of the low pressure (LP) turbine should neither be too wet nor too hot. Thus the temperature and pressure of the main steam should be matched with each other.

4.4.2. Final feedwater temperature

Figure 9 shows the established fact that the feedwater temperature has a significant influence on the thermal efficiency. Higher feedwater temperature increases the mean temperature of heat addition from the flue gas to the steam cycle. Thermal efficiency of Rankine cycles increases when the average temperature of heat addition is higher. However, when the amount of heat from the flue gas and the main steam temperature are fixed, the mass flow of the feedwater and thus the size and capital cost of equipment increases. High feedwater temperature is normally achieved by regenerative pre-heating. The final feedwater temperature (FFWT) should be optimized with respect to the thermal efficiency and investment cost. For subcritical cycles, the maximum FFWT is the boiling point of the main steam. The maximum thermal efficiency is obtained by optimizing the heat loads of each feedwater. An infinite number of heaters is required to achieve the maximum thermal efficiency when the feedwater is supplied at the boiling point[26]. For supercritical and ultra-supercritical cycles, there is no transition between vapor and liquid. The maximum feasible FFWT is limited by the pinch temperature differences of the economizer and the air preheater, as illustrated in Figure 10. When the temperatures in the cold end of the preheater (the air inlet temperature $T_{\mathrm{air,c}}$ and the flue gas exhaust temperature T_{FG}) are fixed, the FFWT is limited by the pinch temperature differences of the economizer (ΔT_{eco}) and the preheater (ΔT_{pre}).

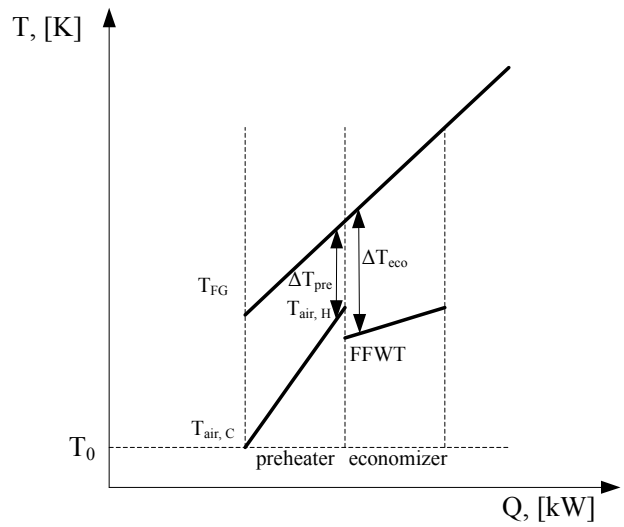


Figure 10 Pinch temperatures of the preheater and economizer

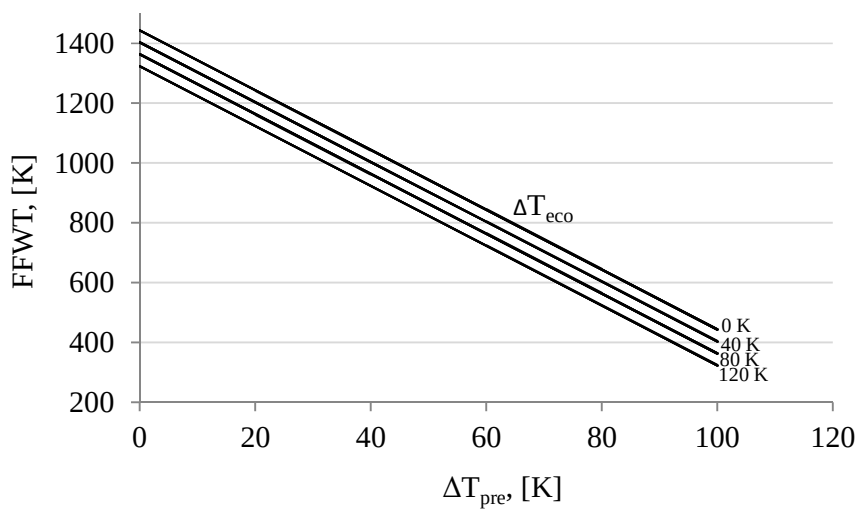


Figure 11 Influenceof pinch temperatures for the preheater and economizer on the FFWT

For the reference plant, T_{FG} =393 K, $T_{air,c}$ =288 K and f=1.22. The influence of ΔT_{eco} and ΔT_{pre} on the FFWT can then be investigated when assuming constant specific heat capacities of the air and the flue gas, as illustrated in Figure 11. The maximum FFWT is 1,443 K when both ΔT_{eco} and ΔT_{pre} are 0 K. This temperature is far beyond the maximum temperature of the superheated steam limited by materials of construction. When ΔT_{eco} and ΔT_{pre} are fixed, a

higher FFWT will increase the flue gas exhaust temperature (T_{FG}), thus the low temperature heat losses will increase.

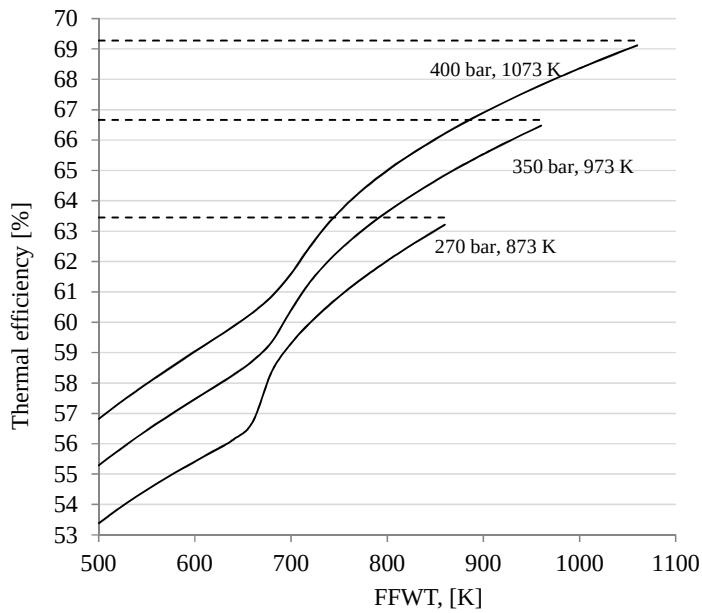


Figure 12 Influence of FFWT on the thermal efficiency

Similar to the investigation on the influences of steam main parameters presented in Section 4.4.1, the maximum work output is calculated by Eq. (15), and the energy consumption for the feedwater heating process is calculated by the exergy differences between the final feedwater and the condensate. The net work production and thus thermal efficiency can then be determined when the FFWT and steam parameters are known. Figure 12 shows the influence of the FFWT on the thermal efficiency of the entire plant for typical main steam parameters (without reheating). The dashed lines just above each curve represent the Carnot efficiency (the mass flow of the main steam is infinitely large). When the FFWT gets close to the main steam temperature, the thermal efficiency is close to the Carnot efficiency. However, the high FFWT increases the mass flow of boiler feedwater and also the number of feedwater heaters. The capital cost thus considerably increases. The improvement is economics limited.

The FFWT is typically around 500-600 K[27]. For this range, the thermal efficiency increases almost linearly by 0.1% points for an average increment of 5 K in FFWT. Beyond this range (FFWT>600K), evaporation may take place, thus the curves are non-linear.

4.4.3. Reheating and turbine efficiency

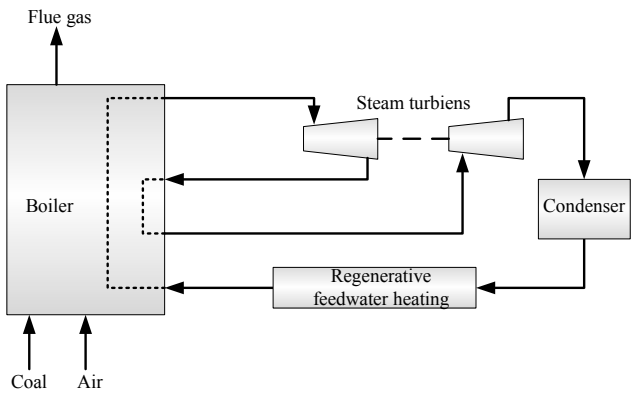


Figure 13 A steam Rankine cycle with reheating

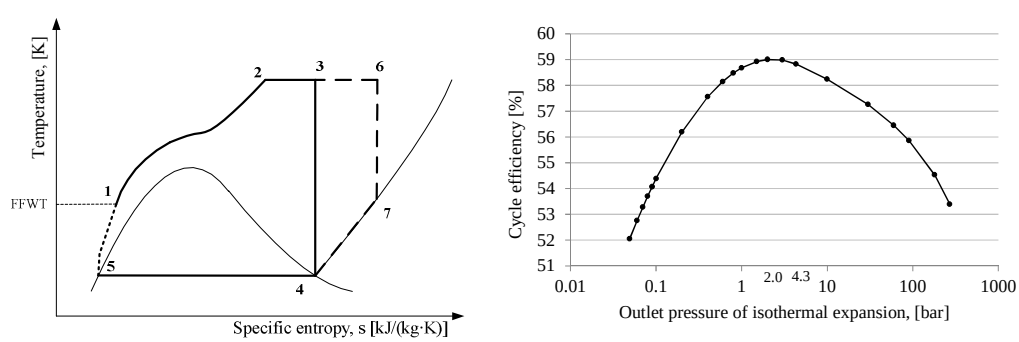
The main steam can be reheated against the flue gas in the boiler area before it is expanded to the condenser pressure, as illustrated in Figure 13. Reheating is applied in steam cycles for two reasons: (i) reducing the moisture content in the last stages of the turbine and (ii) increasing the mean temperature of heat addition. Reheating with more than two stages has been less discussed in literature since the additional investment cost and the complexity are not expected to be justified by the gain in thermal efficiency. Without considering the investment cost, an infinite number of reheating stages can be imagined. Then the reheating process is an isothermal expansion process, as illustrated by process 2-3 in Figure 14(a). The boiler feedwater (BFW; 1) is heated to the main steam condition (2). The steam is expanded to state 3 at constant maximum temperature. Point 3 has the same entropy as the saturated vapor at the condenser pressure (4; 0.048 bar). The process 3-4 is an isentropic expansion process. The steam is then condensed (4-5), pumped and heated to the feedwater conditions

(5-1). The process 5-1 is assumed to be reversible, thus the work consumed in this process is calculated by the exergy difference of the two states. No steam is extracted for regenerative preheating. The cycle efficiency is calculated by Eq. (17).

$$\eta = (\dot{W}_{23} + \dot{W}_{34} - \dot{W}_{51}) / (\dot{Q}_{12} + \dot{Q}_{23}) \tag{17}$$

where $\dot{W}$ and $\dot{Q}$ are the work and heat for the processes.

For the reference power plant, the influence of the outlet pressure from the isothermal expansion (p_3) on the cycle efficiency is shown in Figure 14(b). If no reheating is applied, $p_3 = p_2 = 270$ bar, the cycle efficiency is 53.4%. In this case, the steam quality is 0.736 at the outlet of the last stage of the turbine. The thermal efficiency is calculated to be 50.5% by including the boiler efficiency previously calculated to be 94.7%. Note that if the condensation process is reversible (all of the condensation heat can be recovered as equivalent work), the cycle efficiency is calculated to be 58.2%. The corresponding thermal efficiency is 55.1% (by including a boiler efficiency of 94.7%), which is the same as the value obtained in Figure 12 (FFWT = 581 K). Thus the irreversible condensation process has reduced the thermal efficiency by 4.6% points (0.551-0.505).



(a) (b)

Figure 14 Isothermal expansion: (a) T-s diagram, (b) Cycle efficiency

If the main steam is isothermally expanded to 4.3 bar and then isentropically expanded to 0.048 bar, the steam at the outlet of the last stage of the turbine is saturated. The cycle

efficiency is 58.8%. However, this is not the maximum efficiency that can be achieved. Instead the main steam can be isothermally expanded to p_6, and then isentropically expanded to p_7. In this case, superheated steam (7) is condensed in the condenser. Such a process is shown by dashed lines in Figure 14(a). This process increases the temperature of heat dissipation but also increases the temperature of heat addition. An optimal p_6 can be obtained by iteration when the cycle efficiency is equal to the efficiency of a Carnot cycle operating between the two temperatures T_6 and T_7, i.e. $(1-T_7/\mathrm{T}_6)$. For the given process conditions, the optimal pressure is 2.0 bar and the corresponding cycle efficiency is 59.0%. The temperature of the steam from the last turbine stage is 371 K. The observation is useful for the optimization of the reheating pressures if the condenser can withstand superheated steam.

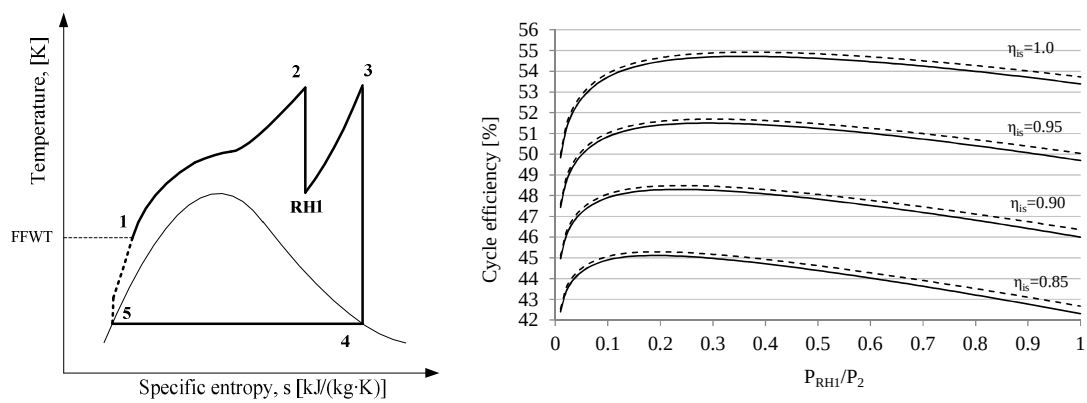
The isothermal expansion process can not be implemented in practice which is a practical irreversibility of the steam power cycle. A one-stage reheating process is shown in Figure 15(a). Assuming that regenerative preheating is not used (the process 5-1 is reversible), the near optimal reheating pressure is obtained by a sensitivity analysis. Figure 15(b) shows how the cycle efficiency varies with the ratio between the reheating pressure (p_{RH1}) and the main steam pressure (p_2). The solid lines represent the cases in which the reheating temperature is equal to the main steam temperature. The dashed lines are for the cases in which the reheating temperature is 20 K higher than the main steam temperature. The numbers on the lines are the isentropic efficiencies (η_{is}) for the steam turbines (assumed to be the same for all turbine stages). The following conclusions can be observed:

(1) The maximum cycle efficiency increases by around 0.2% points when the reheating temperature is 20 K higher than the main steam temperature for various turbine efficiencies.

(2) The cycle efficiency could be smaller than the value without reheating if the reheating pressure is too low, since the mean temperature of the heat addition for the reheating process in such cases is too low.

(3) Without reheating, the cycle efficiency is reduced from 53.4% to 46.0% if the turbine isentropic efficiency decreases from 1.0 to 0.9. The corresponding drop in the thermal efficiency of the plant is 7.01% points.

(4) When the turbine isentropic efficiency is 0.90 (close to state-of-the-art technology), the optimal pressure ratio between the reheating pressure and the main steam pressure is 0.24. This optimal value is almost equal to the value (0.237) in the reference plant where regenerative feedwater preheating is used[5].



(a) (b)

Figure 15 One-stage reheating process: (a) *T-s* diagram, (b) Cycle efficiency

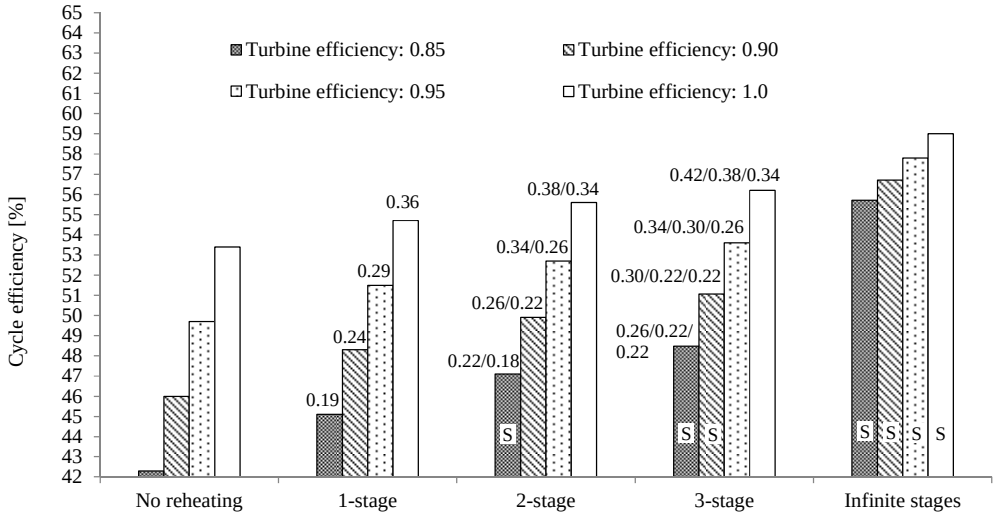


Figure 16 Influence of number of reheating stages on the steam cycle efficiency

Figure 16 shows the influence of the number of reheating stages on the cycle efficiency for different turbine efficiencies. The efficiencies for all the stages are assumed to be the same. The reheating temperatures are assumed to be the same as the main steam temperature. The cycle efficiencies are obtained by sensitivity analysis and are thus near optimal values. When more than one-stage reheating is applied, the steam from the last stage of the turbine is superheated for some cases (marked as 'S' on the bars) in order to get the maximum efficiency. The numbers above each bar in the reheating cases (with finite stages) are the optimal ratio of the reheating pressure to the inlet pressure of the upstream turbine. Obviously, the improvement in cycle efficiency diminishes with an increasing number of reheating stages. When the turbine efficiency is 0.9, the cycle efficiency increases 2.3% points by one-stage reheating, 1.6% points by a second reheating stage and 1.2% points by a third reheating stage. The improvement in cycle efficiency with increasing number of reheating stages is more notable with lower turbine efficiencies.

4.4.4. Other practical limitations on the thermal efficiency

In the previous sections, the condensate is assumed to be reversibly brought to the final feedwater conditions. In practice, regenerative heating and pumping are used to lift the temperature and pressure. For modern steam cycles, 8-10 feedwater heaters are commonly used, resulting in very small irreversibilities. The irreversibilities in the feedwater heaters (the deaerator included) and pumps are reported to be responsible for around 1.2% of the thermal input[28, 29]. The same value is assumed for the reference power plant. The steam extraction reduces the irreversibilities in the turbines and the condenser. The condensate flow is around 75% of the main steam flow[28, 29]. Thus the irreversibilities in the condenser can be reduced by 1.15% of the thermal input compared to the value presented in Section 4.4.3 (4.6%). Table 1 shows the changes in thermal efficiency. When all the components are included, the thermal efficiency is very close to the value for the reference power plant (45.5% as presented in Table A1).

Table 1. Consequent changes in plant thermal efficiency (%)

Turbine efficiency	1.0	0.95	0.9
Cycle efficiency for one-stage reheating (refer to Figure 13)	54.7	51.5	48.3
Reheating temperature is 20 K above the main steam temperature	54.9	51.7	48.5
Cycle efficiency is converted into thermal efficiency (by a factor of boiler efficiency, 94.7%)	52.0	49.0	45.9
Irreversibilities for the feedwater heating and pumping included	50.8	47.8	44.7
Irreversibilities in the condenser is reduced due to steam extraction	52.0	49.0	45.9
Auxiliary power is included (1.7% of the thermal input[5])	51.3	47.3	44.2

5. Emissions control: CO_2 capture

Technologies for the control of SOx, NOx and particles in thermal power plants are relatively mature. For the reference plant, both low NOx burners and a selective catalytic reduction (SCR) DeNOx plant are used for the control of NOx emissions. The flue gas desulphurization (FGD) unit is assumed to remove 98% of SOx in the flue gas an finally the electrostatic precipitator (ESP) is used for the removal of particles[5]. The corresponding energy

consumption for the control of SOx, NOx and particles is small and included in the auxiliary power in Table 1.

The control of CO_2 emissions from thermal power plants is a possible measure to mitigate climate change. Various approaches for CO_2 capture are under investigation. They are normally classified into three categories: post-combustion, pre-combustion and oxy-combustion[30]. For post-combustion, the CO_2 is separated directly from the flue gas, thus the main separation is between N_2 and CO_2. For pre-combustion, the fuel is converted into syngas (CO and H_2) by gasification or reforming, then the CO is further converted into CO_2 by the water gas shift reaction, and finally CO_2 is separated from H_2 prior to combustion. Thus the main separation in pre-combustion is between H_2 and CO_2. The key idea of oxy-combustion is to use pure O_2 or other oxidants such as metal oxides (CLC[3]) instead of air for the combustion, resulting in concentrated CO_2 in the flue gas, thus the main separation is between O_2 and N_2.

The capture of CO_2 from coal based power plants causes considerable thermal efficiency penalties, typically varying from 6.5 to 15% points (energy consumption for CO_2 compression is included) depending on the technology pathways[30, 31]. There is a minimum energy penalty which is limited by thermodynamics. When a process is assumed to be thermodynamically reversible, the minimum work consumption (or production) is equal to the difference between the exergy of the product streams and the exergy of the feed streams. In the case of reversible removal of CO_2 from the reference plant, the CO_2 is assumed to be directly separated from the flue gas at ambient temperature and is then isothermally compressed to transportation pressure ($p_{\mathrm{transportation}}$ =110 bar, according to literature[5]), as shown in Figure 17. Both separation and compression processes are assumed to be reversible.

The minimum work for the separation and compression of CO_2 can be calculated[12], thus the thermal efficiency including CO_2 capture can be obtained.

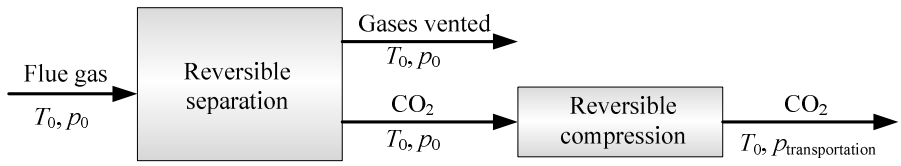


Figure 17 A reversible CO_2 capture process

Figure 18(a) shows the thermal efficiency for various CO_2 recovery rates (referred to the numbers for each curve on the figure) at different air factors for the reference power plant. The purity of the captured CO_2 is assumed to be 1.0. The dashed curves represent cases when only CO_2 separation work is included, while the solid curves represent cases when both CO_2 separation work and compression work are included. When the CO_2 recovery rate is zero, the thermal efficiency is actually equal to the value when CO_2 capture is not applied (the temperature of the flue gas vented is 393 K). The following conclusions can be observed from Figure 18(a):

(1) Within the given range of air factor for a CO_2 recovery rate of 1.0, the thermal efficiency penalty related to CO_2 capture is 2.92-3.49% points, where the reversible separation of CO_2 contributes 1.24-1.55% points (around 43% of the total penalty).

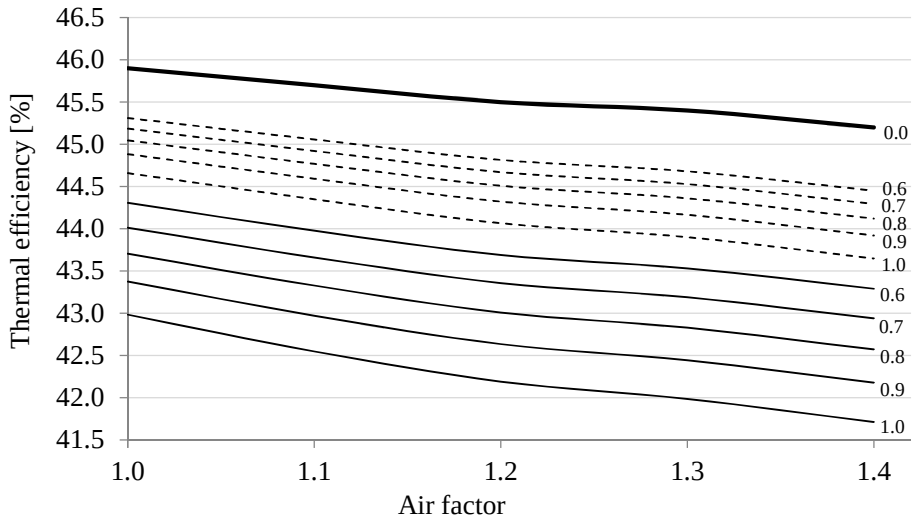
(2) Not very surprisingly, an obvious reduction in efficiency penalty can be observed when the CO_2 recovery rate is reduced. For the reference plant where the air factor is 1.22, the efficiency penalty related to CO_2 capture reduces from 3.31 to 1.81% points when the CO_2 recovery rate decreases from 1.0 to 0.6. The penalty related to CO_2 separation reduces from 1.43 to 0.69% points. Thus, partial capture of CO_2 in power plants may be attractive when the investment cost of equipment is taken into consideration[32]. This will depend on the future cost of emitting CO_2.

(3) The changes in efficiency penalty (also in work consumption which is not shown in the figure) with the CO_2 recovery rate become smaller when the recovery rate decreases. The more the recovery rate is decreased, the lower the relative efficiency gain will be. Thus energy saving by reducing the recovery rate should preferably be implemented at high recovery rates.

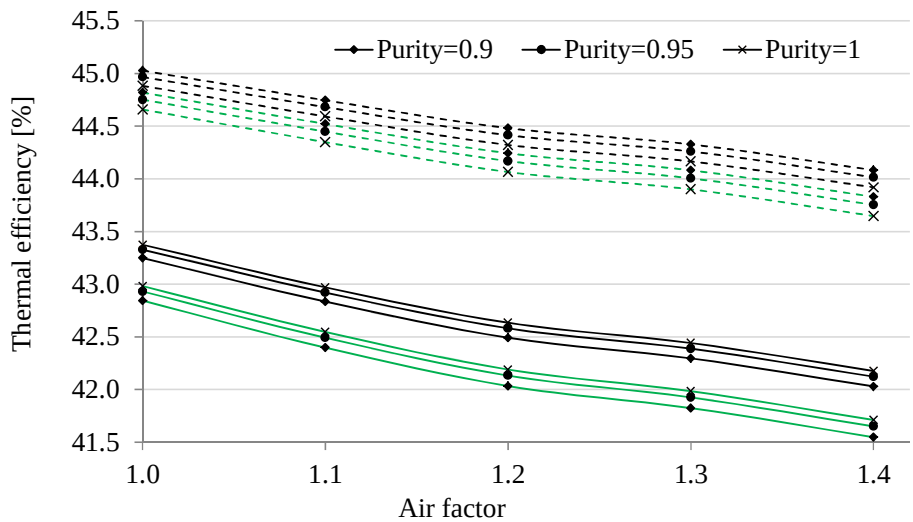
Figure 18(b) shows the thermal efficiency for different CO_2 purities (mole fraction: 0.9, 0.95 and 1.0). Since the main separation is between N_2 and CO_2, the impurity in the captured CO_2 is assumed to be N_2. Two recovery rates are investigated: 0.9 and 1.0. The following two conclusions can be made:

(1) The thermal efficiency does not change much with the purity of CO_2. For a recovery rate of 1.0, the thermal efficiency increases about 0.15% points at various air factors when the purity of CO_2 increases from 0.9 to 1.0.

(2) The separation work increases when the purity of CO_2 increases, however, the compression work decreases since fewer impurities are compressed. According to Figure 18(a), the compression work contributes more to the total efficiency penalty than the separation work, thus the total efficiency penalty related to CO_2 capture decreases. Therefore it is more favorable in the reversible case to capture CO_2 with high purity.



(a)



(b)

Figure 18 Thermal efficiency including ideal CO_2 capture processes: (a) various CO_2 recovery rates when the CO_2 purity =1, (b) various CO_2 purities when the CO_2 recovery rate = 1 (curves in green color) and 0.9 (curves in black color).

The above observations from Figures 18(a) and 18(b) are based on the assumption that the capture processes are reversible. The thermal efficiency shown in the two figures are thus theoretical maximum values. Reversible processes are difficult to realize in practice, however, the observations can somewhat guide practical improvement measures of CO_2 capture processes, e.g. the improvement potential by developing advanced solvents such as ion liquids

is limited by the minimum energy penalty. For the reference power plant[5], the thermal efficiency penalty is 11.7% points when a monoethanolamine (MEA) based solvent is used for capturing CO_2. This value is more than 3 times the theoretical minimum (thermodynamics limited). The difference is mainly caused by the technology route for CO_2 capture.

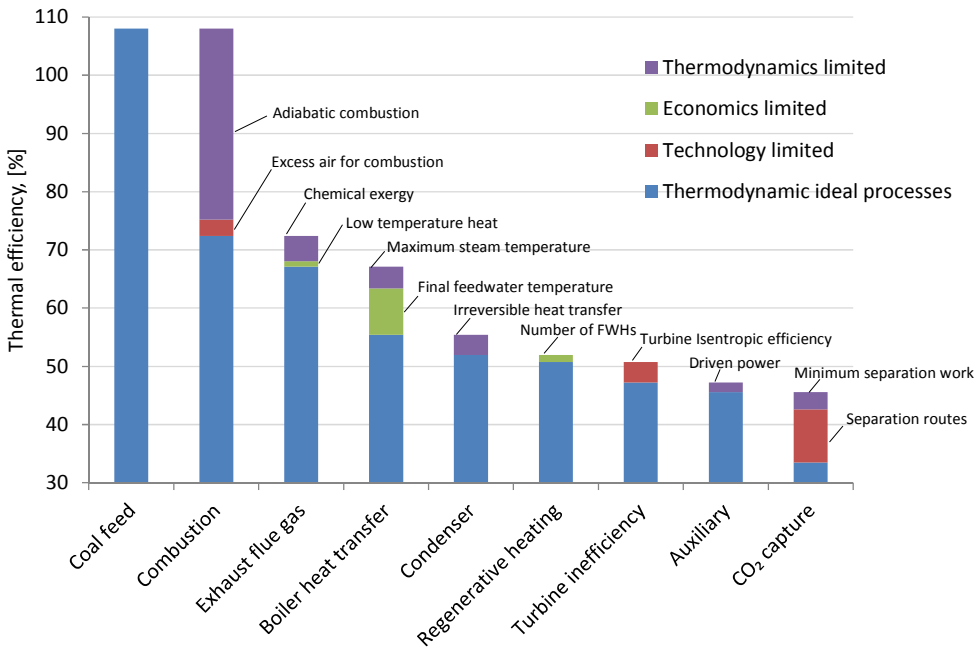


Figure 19 Decrease of the plant thermal efficiency caused by various factors

A summary on the decrease of thermal efficiency from theoretical maximum to pratical values for the reference plant is shown in Figure 19. The thermal efficiency is reduced by adding limitations in each unit operation. The dominant factors are listed on the figure and are classified into thermodynamic, technical and economic categories. The potential for improvement on in each unit can easily be determined without detailed exergy analysis on the entire power plant. Figure 19 shows that adiabatic combustion of coal is the most inefficient process and thus a technology shift from adiabatic combustion to new oxidation routes such as chemical looping combustion and fuel cells is necessary for significant efficiency improvements. It can also be observed that the technical factors are more dominant than

economic factors, thus efforts should focus more on technology development such as reducing air factors for combustion, developing more efficient steam turbines, and exploring advanced separation routes for CO_2 capture. Note that the MEA capture of CO_2 causes the second largest loss of efficiency after combustion irreversibilities. The final boiler feedwater temperature has also caused a considerable reduction in thermal efficiency. This temperature is economics limited that determines the average temperature of heat addition to the steam cycle.

6. Conclusions

A systematic methodology has been presented in this paper for the assessment in efficiency losses for direct combustion coal to power processes step by step and potential means of reducing these losses. The thermal efficiency was decreased to practical values by adding limitations such as thermodynamic, technical and economic factors. The primary advantage of the methodology is that all possible improvement measures can be covered and the improvement potential can be quickly predicted without knowing process details. As an illustrative example, in order to increase the thermal efficiency by 0.1% points for the reference power plant, the following measures can be implemented: (1) reducing the air factor by 0.057, (2) reducing the flue gas temperature by 4.5 K, (3) increasing the final feedwater temperature by 4 K, (4) increasing the main steam temperature by 8 K, and (5) increasing the main steam pressure by 10 bar.

The following four observations are directly derived when applying the methodology: (1) the low temperature heat of the flue gas should primarily be recovered by the boiler system before using Organic Rankine Cycles to recover work; (2) when reheating is implemented (particularly for two or more stages), higher thermal efficiency may be achieved if the steam at the outlet of the last turbine stage is superheated; (3) it is concluded that as turbine isentropic efficiency improves, the gain in thermal efficiency from reheating decreases; and

(4) the second largest loss after combustion in a coal based power plant with MEA capture of CO_2 is caused by this capture. Hence, the tremendous efforts being spent on developing improved CO_2 capture technologies are indeed justified.

The minimum thermal efficiency penalty related to direct CO_2 capture from the flue gas is calculated to be 2.92-3.49% points, where the separation and the compression of CO_2 contribute about 43% and 57% respectively. Considerable energy savings can be achieved by decreasing the CO_2 recovery rate, particularly at high recovery rates. The CO_2 is preferably captured at higher purities. Practical limitations on CO_2 capture are subject to further investigations.

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NOMENCLATURE

$\dot{E}$	exergy, kW
e	specific exergy, kJ/kg or kJ/mole
$\dot{F}$	molar flow, mole/s
f	air factor
$\dot{G}$	Gibbs free energy, kW
$\dot{H}$	enthalpy, kW
h	specific enthalpy, kJ/kg or kJ/mole
$\dot{I}$	irreversibility, kW

$\dot{m}$	mass flow, kg/s
p	pressure, bar
$\dot{Q}$	heat, kW
$\bar{R}$	universal gas constant, kJ/(mole·K)
$\dot{S}$	total entropy, kW/K
s	specific enthropy, kJ/(kg·K)
T	temperature, K or ^{o}C
$\dot{W}$	work, kW
x	molar fraction

Geek Letters

Δ	symbol of differences
η	efficiency
φ	ratio of the chemical exergy to the lower heating value
ω	stoichiometric ratio for combustion

Sub and superscripts

0	reference state
ad	adiabatic
C	combustion; cold end
ch	chemical
eco	economizer
FG	flue gas
fw	feed water
H	hot end
i	component index
is	isentropic
j	phase index
min	minimum
mix	mixing
ms	main steam

ph	physical
pre	preheater
RH	reheating steam
SC	steam cycle
tot	total

Abbreviations

BFW	boiler feedwater
CLC	chemical looping combustion
ESP	electrostatic precipitator
FFWT	final feedwater temperature
FGD	flue gas desulphurization
HHV	higher heating value
HP	high pressure
LHV	lower heating value
LP	low pressure
MS	main steam
ORC	organic Rankine cycle
RH	reheating
S	superheated
SCR	selective catalytic reduction

Appendix

Table A1. Main parameters of the reference plant [14]

Parameters	Values
Ambient conditions	288.15 K, 1.01 bar and 60% relative humidity
Main steam (MS)	600.0 kg/s, 270 bar, 873 K
One-stage reheating (RH) steam	485.2 kg/s, 60 bar, 893 K
Feedwater heaters	5 LP heaters and 3 HP heaters
Final feedwater	320 bar, 581 K
Exhaust flue gas	393 K

Condenser pressure	0.048 bar
Gross electrical output	819 MW
Auxiliary power consumption (feedwater pumping is included)	65 MW
Coal feed	65.765 kg/s
Lower heating value (LHV) of coal	25,170 kJ/kg
Thermal input (LHV)	1,655.3 MW
Thermal efficiency (LHV)	45.5%

Table A2. Coal characteristics [14]

	As received	Dry
Proximate analysis, %		
Moisture	8.00	0
Volatile matter	22.90	24.9
Ash	14.15	15.4
Fixed carbon	54.90	59.7
Total sulphur	0.52	0.56
Ultimate analysis, %		
Carbon (C)	66.52	72.31
Hydrogen (H)	3.78	4.11
Nitrogen (N)	1.56	1.70
Sulfur (S)	0.52	0.56
Chlorine (Cl)	0.01	0.01
Ash	14.15	15.38
Moisture (H_2O)	8.00	0
Oxygen (O)	5.46	5.93
High heating value (HHV), kJ/kg	26,230	28,500
Low heating value (LHV), kJ/kg	25,170	27,573
Chemical exergy, kJ/kg	27,295	
Stoichiometric air/coal ratio	8.8122	

Table A3. Compositions of atmospheric air

Component	Volume fraction (dry)	Volume fraction at 60% relative humidity
N_2	78.09	77.3
CO_2	0.03	0.03

H_2O	0	1.01
Ar	0.93	0.92
O_2	20.95	20.74

REFERENCES

[1] International Energy Agency, World Energy Outlook 2014.

[2] G.J. Silvestri, R.L. Bannister, T. Fujikawa, A. Hizume. Optimization of advanced steam condition power plants. J Eng Gas Turbines Power. 114 (1992) 612-20.

[3] N.R. McGlashan. Chemical-looping combustion - a thermodynamic study. Proc IMechE, Part C: J Mechanical Engineering Science. 222 (2008) 1005-19.

[4] G. Angelino, C. Invernizzi. Binary and ternary liquid metal - steam cycles for high-efficiency coal power stations. Proc IMechE, Part A: J Power and Energy. 220 (2006) 195-205.

[5] H. Spliethoff. Power generation from solid fuels. Springer-Verlag Berlin Heidelberg, Berlin, Heidelberg, Germany, 2010.

[6] A. Bejan. Advanced engineering thermodynamics. 2nd ed. John Wiley & Sons, New York, US, 1997.

[7] M.M. El-Wakil. Powerplant technology. McGraw-Hill, New York, US, 2002.

[8] J.K. Salisbury. Steam turbines and their cycles. John Wiley & Sons, New York, US, 1950.

[9] Y.A. Cengel, M.A. Boles. Thermodynamics: an engineering approach. 6th ed. McGraw-Hill, New York, US, 2006.

[10] M.J. Moran, H.N. Shapiro, D.D. Boettner, M.B. Bailey. Principles of engineering thermodynamics. 7th ed. John Wiley & Sons, New Jersey, US, 2012.

[11] C. Fu, R. Anantharaman, K. Jordal, T. Gundersen. Thermal efficiency for coal to power: from theoretical to practical assessments. The 26th International Conference on Efficiency, Costs, Optimization, Simulation and Environmental Impact of Energy Systems (ECOS 2013), Guilin, China, July 2013.

[12] R. Anantharaman, K. Jordal, T. Gundersen. CO_2 capture processes: novel approach to benchmarking and evaluation of improvement potentials. Energy Procedia. 37 (2013) 2536-43.

[13] H.M. Kvamsdal, K. Jordal, O. Bolland. A quantitative comparison of gas turbine cycles with capture. Energy. 32 (2007) 10-24.

[14] Franco F, Anantharaman R, Bolland O, Booth N, van Dorst E, Ekstrom C, Fernandes ES, Macchi E, Manzolini G, Nikolic D, Pfeffer A, Prins M, Rezvani S, Robinson L. European Best Practice Guidelines for Assessment of CO_2 Capture Technologies. European Benchmarking Task Force, 2011. Available at: <http://www.energia.polimi.it/news/D%204_9%20best%20practice%20guide.pdf> [accessed 17.04.2015].

[15] T.J. Kotas. The exergy method of thermal power plant analysis. Exergon Publishing Company, London, UK, 2012.

[16] J. Szargut. Exergy method: technical and ecological applications. WIT Press, Southampton, UK, 2005.

[17] R.C. Flagan, J.H. Seinfeld. Fundamentals of air pollution engineering. Prentice-Hall, New Jersey, US, 1988.

[18] A.J. Appleby, F.R. Foulkes. Fuel cell handbook Van Nostrand Reinhold, New York, US, 1989.

[19] A.E. Lutz, R.S. Larson, J.O. Keller. Thermodynamic comparison of fuel cells to the Carnot cycle. Int J Hydrogen Energy. 27 (2002) 1103-11.

[20] W.R. Dunbar, N. Lior. Sources of combustion irreversibility. Combust Sci and Tech. 103 (1994) 41-61.

[21] M.J. Prins, K.J. Ptasinski. Energy and exergy analyses of the oxidation and gasification of carbon. Energy. 30 (2005) 982-1002.

[22] M.M. Hossain, H.I. de Lasa. Chemical-looping combustion (CLC) for inherent separations - a review. Chemical Engineering Science. 63 (2008) 4433-51.

[23] C. Fu, T. Gundersen. Exergy analysis and heat integration of a coal-based oxy-combustion power plant. Energy & Fuels. 27 (2013) 7138-49.
[24] A.A. Lakew, O. Bolland. Working fluids for low-temperature heat source. Appl Therm Eng. 30 (2010) 1262-8.
[25] A.I. Kalina. Combined cycle cystem with novel bottoming cycle. J Eng Gas Turbines Power. 106 (1984) 737-42.
[26] R.W. Haywood. A generalized analysis of the regenerative steam cycle for a finite number of heaters. Proc IMechE. 161 (1949) 157-64.
[27] A. Seltzer, Z. Fan, H. Hack. A method to increase oxyfuel power plant efficiency and power output. The 37th International Technical Conference on Clean Coal & Fuel Systems, Clearwater, Florida, US, June 2012.
[28] I. Dincer, M.A. Rosen. Exergy: energy, environment and sustainable development. Elsevier, Amsterdam, Netherlands, 2007.
[29] Y. Yang, L. Wang, C. Dong, G. Xu, T. Morosuk, G. Tsatsaronis. Comprehensive exergy-based evaluation and parametric study of a coal-fired ultra-supercritical power plant. Applied Energy. 112 (2013) 1087-99.
[30] J. Davison. Performance and costs of power plants with capture and storage of CO_2. Energy. 32 (2007) 1163-76.
[31] M. Kanniche, R. Gros-Bonnivard, P. Jaud, J. Valle-Marcos, J.-M. Amann, C. Bouallou. Pre-combustion, post-combustion and oxy-combustion in thermal power plant for CO_2 capture. Applied Thermal Engineering. 30 (2010) 53-62.
[32] A.N. Hildebrand, H.J. Herzog. Optimization of carbon capture percentage for technical and economic impact of near-term CCS implementation at coal-fired power plants. Energy Procedia. 1 (2009) 4135-42.