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# Integrating the Compression Heat in Oxy-combustion Power Plants with $CO_2$ Capture

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Oxy-combustion is a competitive technology to enable the capture of $CO_2$ from fossil fuel based power plants. Cryogenic air separation is the only commercially available technology for large volume $O_2$ production that can be applied for oxy-combustion plants. The considerable energy consumption in the air separation units is the main challenge to implement oxy-combustion technology. The compression heat in the air compression process is generally removed by cooling water. The water coolers are responsible for 17.6 % of the total exergy losses in an air separation unit. This paper investigates the possibilities of integrating the compression heat from the air separation unit with the steam cycle. The compression process of the air is studied in two cases: adiabatic (one-stage) compression and three-stage compression with interstage cooling. A decomposed methodology has been developed for studying the heat balance in the boiler feedwater heaters in the steam cycle. The Grand Composite Curve is applied to determine the heat demand and thus the amount of steam extraction. For a coal based oxy-combustion power plant with a gross power output of 792 MW, the thermal efficiency increases with 0.47 % points (net power increases by 8.8 MW) by heat integration in the adiabatic compression case.

## 1. Introduction

Several studies indicate that oxy-combustion is a promising technology for $CO_2$ capture from fossil fuel based power plants, particularly from coal based power plants (DOE/NETL, 2008, Fu and Gundersen, 2010, Wall et al., 2009). The core concept of oxy-combustion is to use high purity oxygen instead of air for the combustion process so that the flue gas is composed mainly of $CO_2$ and $H_2O$. The $CO_2$ can be separated by condensing the $H_2O$ and then purified by chilling. The main challenge for implementing this technology is that a large power penalty is caused by the air separation unit (ASU). A 500 MWe power plant will consume around 10,000 t/day $O_2$ (Higginbotham et al., 2011). Cryogenic air separation is the only commercially available technology for large volume $O_2$ production. For a coal based oxy-combustion power plant, the thermal efficiency penalty caused by the air separation unit (ASU) is around 6.6 % points (based on the higher heating value, HHV) when a traditional double-column distillation cycle is applied for $O_2$ supply (Fu and Gundersen, 2010).
The air feed compression process in a cryogenic ASU is responsible for 38.4 % of the total exergy losses, where the water coolers are responsible for 17.6 % of the total losses (Fu and Gundersen, 2012). The water coolers are used to remove the compression heat, thus reduce the compression work. Since the compression heat is low grade, it is normally regarded as having little value. A model for utilizing the low-grade heat (30 – 250 $^oC$) in site utility systems is presented by Kapil et al. (2010). For oxy-combustion power plants, the low grade compression heat could be integrated with the steam

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cycle by partially preheating the boiler feedwater, thus less steam is extracted for the preheating process. Romeo et al. (2009) investigated the possibility of utilizing the compression heat from the compression process of $CO_2$ for the preheating of boiler feedwater in the low pressure section. The total power saving is reported to be around 40 % of the power consumption for the $CO_2$ compression when the compression heat is utilized. This paper presents a study on the heat integration between the compression heat from the ASU and the steam cycle in a coal based oxy-combustion power plant. A decomposed methodology has been developed for investigating the heat balance in each boiler feedwater heater. The Grand Composite Curve (GCC) is used to determine the load (amount) and the level (temperature) of the extracted steam.

## 2. The steam cycle

The ASU and the entire oxy-combustion power plant are described by Fu and Gundersen (2010). The ambient air is compressed to 5.6 bar (absolute pressure is used in this paper). The flowsheet of a detailed supercritical steam cycle is presented in Figure 1 (DOE/NETL, 2008). The steam cycle has 4 closed feedwater heaters in the low pressure section (FWH1-4) and 3 in the high pressure section (FWH6-8). The high pressure (HP) steam is heated to 242 bar / 599 $^{o}$C with single reheat of the intermediate pressure (IP) steam to 45 bar / 621 $^{o}$C. The condenser pressure is 0.069 bar. The gross power generation from the steam cycle is 791 953 kW. The thermal input by the fuel is 1 878 556 kW (based on the higher heating value, HHV). This steam cycle is used as the reference case in the paper.

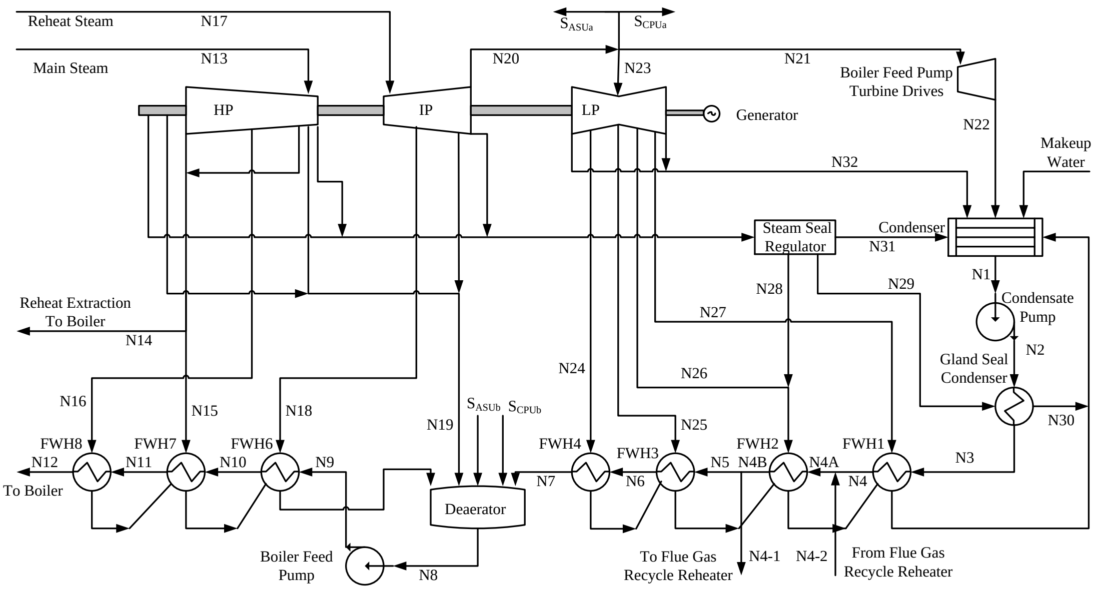


*Figure 1: The supercritical steam cycle*

## 3. Heat integration

When the compression heat is integrated with the steam cycle, the heat is transferred from the compressed gases to the boiler feedwater. The minimum temperature difference for the gas/water heat exchangers is specified to be 10 $^{o}$C, and for the steam/water heat exchangers it is assumed to be the same as in the reference case. The steam seal regulator (SSR) and the deaerator are assumed not to be influenced by the heat integration. Thus the temperature of the boiler feedwater to FWH1 (N3) is maintained (39.1 $^{o}$C). The compressed gases can be cooled to around 49.1 $^{o}$C by the boiler feedwater. The integration study is performed by the following steps:

(1) Determine the supply temperature of the compression heat and thus the FWHs by which the compression heat can be integrated.

(2) Draw the Grand Composite Curve (GCC) when assuming that no steam is extracted for the FWHs that can utilize the compression heat.
(3) Use the GCC to determine the demand of the extracted steam at different pressure levels (the pressure levels are maintained as in the reference case). The condensed water in each FWH flows into the next lower pressure FWH (the deaerator and the gland seal condenser can be regarded as special FWHs), thus supplies a small portion of heat for the lower pressure FWH. This portion of heat is neglected when determining the demand of extracted steam, and primarily used when designing the FWHs. Thus, the compressed gases will be cooled to temperatures higher than 49.1 $^oC$, resulting in larger temperature driving forces in the gas/water heat exchangers.
(4) Further cooling of the compressed gases to 35.1 $^oC$ by cooling water to reduce the compression work or cooling duty in other heat exchangers.

In order to determine the mass flow of the extracted steam based on the GCC, the heat loads for the FWHs are decomposed based on the heat supplied by each extracted steam flow. The decomposed heat loads in the reference case are shown in Figure 2. The external drain coolers (flash type) are used (STEAM PRO 21.0, 2012). The condensate from each FWH is flashed to a lower pressure and condensed again at the lower saturation temperature. Thus the temperature of each extracted steam does not decrease continuously.

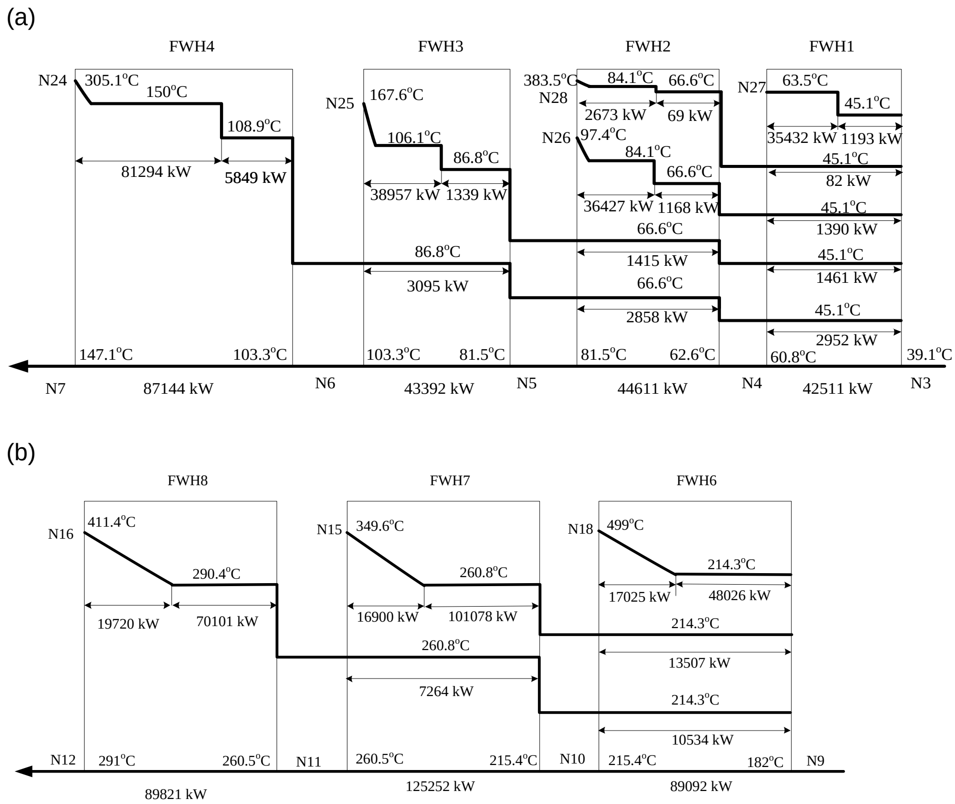


*Figure 2: Decomposed heat loads in the FWHs: (a) FWH1-4; (b) FWH6-8*

**3.1 Case 1: Three-stage compression in the ASU**

The excess $O_2$ in the combustor is around 10 wt%. The ambient air (1.013 bar, 25 $^oC$, 78 617 kmol/h) is compressed to 5.6 bar by a three-stage compression process with interstage cooling. The compression ratio for each stage is equal (1.768). The three heat streams from the outlet of each compression stage are: Q11 (88.9→49.1 $^oC$), Q12 (101.1→49.1 $^oC$) and Q13 (101.1→49.1 $^oC$). The heat capacity is assumed constant and the same for the three heat streams. The mean heat capacity flowrate is calculated to be 641 kW/$^oC$. Based on the supply temperatures, the compression heat can be used to reduce the steam consumption in FWH1-3. Assuming that no steam is extracted for FWH1-3, i.e. the heat is supplied by the compression heat, the condensed water from FWH4, and the steam from the SSR (N28; the sensible heat is very small and thus not included in the GCC), the GCC is shown in Figure 3.

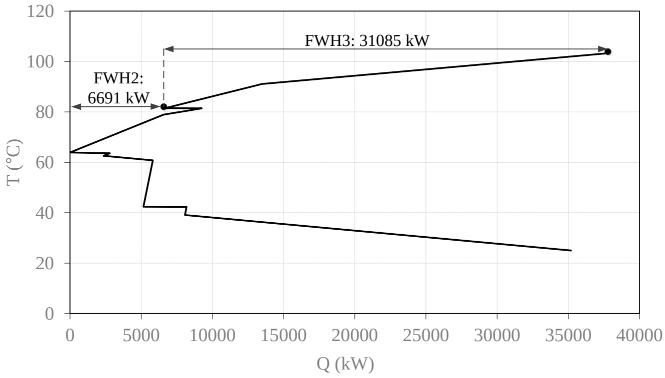


*Figure 3: The GCC for FWH1-3 in Case 1*

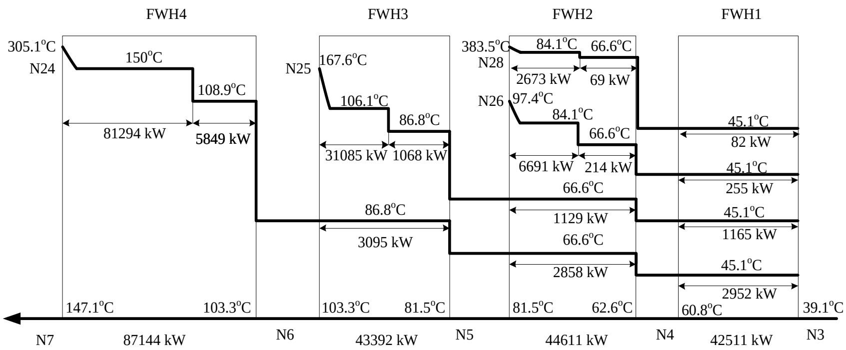


*Figure 4: Decomposed heat loads for FWH1-4 in Case 1*

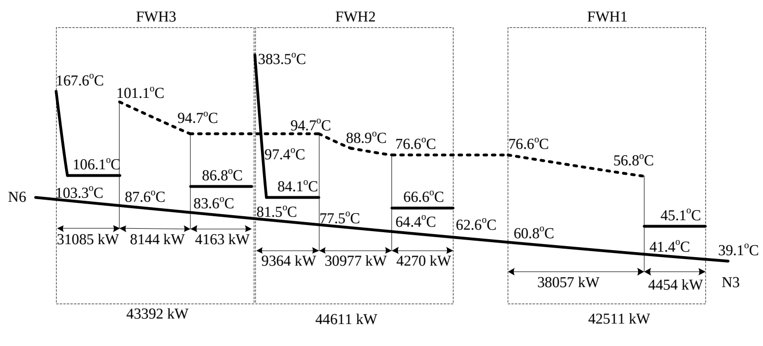


*Figure 5: Heat balance for FWH1-3 in Case 1*

Based on the GCC, the heat demands for the feedwater heaters are 31 085 kW (FWH3), 6 691 kW (FWH2), and 0 kW (FWH1). The mass flow of the extracted steam can thus be determined based on Figure 2, and is shown in Table 1. The new decomposed heat loads in FWH1-4 are illustrated in Figure 4. When the compression heat is added, heat balance can be achieved in FWH1-3. One possible thermal configuration of the boiler feedwater preheating process is shown in Figure 5. The compression heat is illustrated by the dotted lines. In this case, the compressed air is first cooled to 56.8 °C. The heat is used to preheat the boiler feedwater. The air is then cooled to 35.1 °C by cooling water (not shown in Figure 5). The boiler feedwater in each FWH is first slightly heated in an external drain cooler (flash type) by the condensate in the FWH (including the condensate from higher pressure FWHs), then heated by the compressed gases, and finally heated by the extracted steam (if necessary). Based on Figure 5, the heat exchanger network can be directly obtained and is shown in Figure 6. The multi-stream gas/water heat exchangers can be easily separated into regular two-stream heat exchangers by splitting the boiler feedwater into several streams.

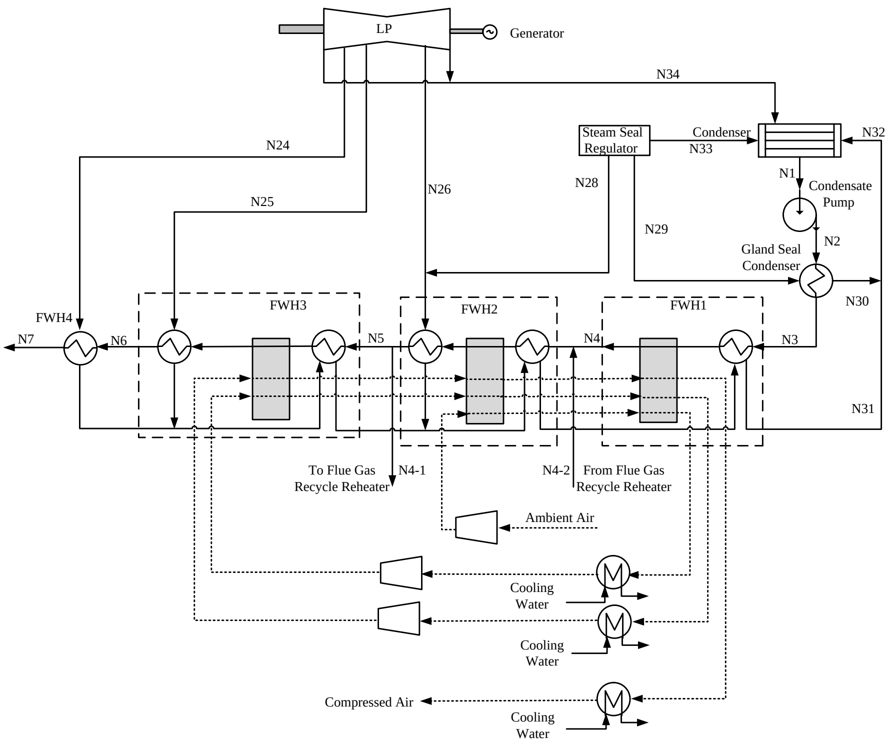


*Figure 6: A possible integration scheme for Case 1*

### 3.2 Case 2: Adiabatic compression in the ASU

The ambient air is compressed to 5.6 bar by a one-stage compressor. Thus there is only one heat stream: 250.2→49.1 °C, 650 kW/°C. The temperature of the compression heat is lifted due to the adiabatic process, thus the compression heat can be used to reduce the steam consumption in FWH1-4 and FWH6-7. The resulting extra steam from the outlet of the HP steam turbine is assumed to expand to the condenser pressure through a one-stage turbine. Following the procedure described in the beginning of Section 3, the plant performance can be investigated.

### 3.3 Plant performance comparison

The plant performance for the reference case and the two integration cases is shown in Table 1. In the reference case, the compression process in the ASU is the same as in Case 1, however, the compression heat is removed by cooling water and thus wasted. When the low grade compression heat is used for preheating the boiler feedwater, the power generated by the steam turbines increases by 7 180 kW, resulting in an increase of the thermal efficiency by 0.38 % points. In the adiabatic compression case, more work (20 999 kW) is consumed in the ASU. However, the compression heat is upgraded (the temperature is lifted) and can be used to reduce the steam extraction in the high pressure section, resulting in an increase of the thermal efficiency by 0.47 % points.

The compressor efficiency is assumed constant and the same for the three cases in this paper. In practice, the compressor efficiency will change with the operating temperature. The temperature differences of the gas/water heat exchangers are also different in the three cases. In addition, the plant performance is influenced by the compression ratio of each compression stage and the supply/target temperatures. The choice of compression scheme in the ASU tailored for heat integration is a complex optimization problem (including investment cost) and needs further investigation.

*Table 1: Results for the heat integration between the ASU and the steam cycle*

| | Reference case | Case 1 | Case 2 |
|---|---|---|---|
| Hot end temperature of the compressed gases, $^oC$ | 101.1 | 101.1 | 250.2 |
| Cold end temperature of the compressed gases, $^oC$ | / | 56.8 | 69.9 |
| Work consumption in the ASU, kW | 119,029 | 119,029 | 140,038 |
| Power generated from steam turbines, kW | 791,953 | 799,133 | 821,730 |
| Net power increment, kW | / | 7,180 | 8,768 |
| Thermal efficiency increment (HHV), % points | / | 0.38 | 0.47 |
| Mass flow of extracted steam, kg/s | | | |
| N15 | 60.995 | 60.995 | 53.814 |
| N16 | 47.57 | 47.57 | 47.57 |
| N18 | 25.515 | 25.515 | 16.753 |
| N24 | 33.281 | 33.281 | 15.956 |
| N25 | 16.477 | 13.148 | 12.98 |
| N26 | 15.674 | 2.87 | 13.327 |
| N27 | 15.481 | 0 | 12.511 |

## 4. Conclusions

The energy penalty related to $CO_2$ capture in oxy-combustion power plants is mainly caused by the energy intensive cryogenic air separation unit. The air is normally compressed by multi-stage compressors in the air separation unit. The compression heat is removed by cooling water in order to reduce the temperature of the compressed gases and thus the compression work. A decomposed methodology has been developed in this paper for integrating the compression heat with the steam cycle. For a coal based oxy-combustion power plant with $CO_2$ capture (gross power output: 792 MW), the thermal efficiency increases by 0.38 % points (net power increases by 7.2 MW) when the compression heat is integrated with the steam cycle. If an adiabatic compression scheme is applied, the thermal efficiency increases by 0.47 % points (net power increases by 8.8 MW). Further investigations on the optimal compression scheme in the air separation unit are required.

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