Biomass-fired Combined Heat and Power Plant with a Fuel Drier

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Abstract

This report investigates the thermodynamic aspects and evaluates the economic feasibility of a biomass-fired combined heat and power (CHP) plant with a fuel drier. The plant utilizes a steam cycle to generate electricity and provide district heating. The fuel drier reduces the moisture content of biomass, enhancing its combustion efficiency and maximizing energy recovery. The plant's profitability is evaluated based on scenarios with varying electricity prices and green certificate revenues. The analysis reveals that the plant is economically viable in cases with high electricity prices (>1400 SEK/MWh). However, even with lower electricity prices, the plant can achieve profitability when combined with green certificate revenues.

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1 Introduction

Biomass-fired combined heat and power (CHP) plants are a pivotal component of sustainable energy generation, providing a solution that integrates electricity and heat production. These plants are designed to harness the energy potential of biomass sources, typically wood residue pellets. CHP plants offer a unique advantage by not only generating electricity but also capturing and utilising the waste heat produced during the process for various heating and cooling applications. This approach contributes to the efficient use of resources and plays a significant role in reducing greenhouse gas emissions.

The first aim of this project is to design a complete process for a biomass-fired CHP plant with a biofuel drier. The process was designed to satisfy process conditions given in the problem description such as 50 MW heating output to district heating network, minimum temperature differences in heat exchangers, pressure drops, and pressure and temperature of turbine inlet steam. By thermodynamic calculations covered in the course and using a steam table in 'Tabeller och diagram för energiteksniska beräkningar' (T&D)[1], intensive thermodynamic properties of the streams in the process were specified in the first part of the project. Then, by defining dimensionless variables that describe the flowrates of all streams in the system, extensive properties of the streams were calculated. A degree of freedom analysis was carried out to verify whether the system is well-defined.

Another aim of this project is to assess the plant by looking at a QT-diagram for the boiler and air-preheater in the process and carrying out investment calculations to examine its economic profitability. Economic parameters given in the problem description were used as the reference.

This report will demonstrate how all the calculations were made step-by-step, starting from calculating mass and energy balances for combustion and completing the steam cycle for the basis of 1kg/s of moist and ash free fuel (MAF).

[1] L. Wester, *Tabeller och diagram för energitekniska beräkningar*. Marklund Solutions, 2015.

2 Method

2.1 Process description

The steam cycle consists of typical components of a steam power plant: a boiler supplied by air and biofuel that produces steam at 540 °C and 100 bar; three steam turbines; a condenser with a heat production of 50 MW; a low pressure pump after the condenser; a deaerator where the boiler feed water is preheated by direct steam injection at 3.5 bar; a high pressure pump that increases the pressure in order to have 100 bar at the inlet of the high pressure turbine; and a feed water preheater that uses 12 bar superheated stream extracted from the turbine to increase further the water temperature before the inlet of the boiler.

On the air side, there is an air preheater that operates to harvest and utilise waste heat from the flue gas to raise the temperature of the air stream that enters the boiler, while limiting the temperature of the flue gas exiting the preheater at 120 °C. There are also two fans that are used to transport air and flue gas throughout the system.

The wet biomass, before being fed to the boiler, passes through the drier, that is indirectly heated by 3.5 bar steam extracted from the turbine. In the first heat exchanger in the drier, most of the water content of the biomass evaporates and forms saturated vapour at 2.0 bar. Since this steam is too polluted to be directly used, it is led to a second heat exchanger where its heat is used to produce 1.0 bar saturated steam from the pure condensate that was formed in the first heat exchanger. Then the saturated steam is led to the district heating together with the excess pure condensate. Polluted condensate leaves the drier to a wastewater treatment facility.

2.2 Process flowsheet

The flowsheet of the complete process is represented in Figure 1 below. It also represents with different colour the different fluids that pass through the plant.

Based on given information, a complete process flowsheet was prepared using Microsoft Visio and presented in Figure 1. Streams are named according to following rules to facilitate reading:

- Arabic numbers (1, 2, and so on) are used for steam/water streams. The main flow that passes through the cycle is prioritised.
- Streams involved in the biomass drying process are named as D1, D2, and so on, where ‘D’ denotes ‘drier’. ‘DW’ stands for ‘dirty water’ removed from the wet fuel.
- Air and flue gas streams are named with Roman numbers (I, II, and so on).
- When there is a split of one stream into multiple streams, small-letter characters are used to represent them. (e.g., stream 2 is split into streams 2a and 2b.)

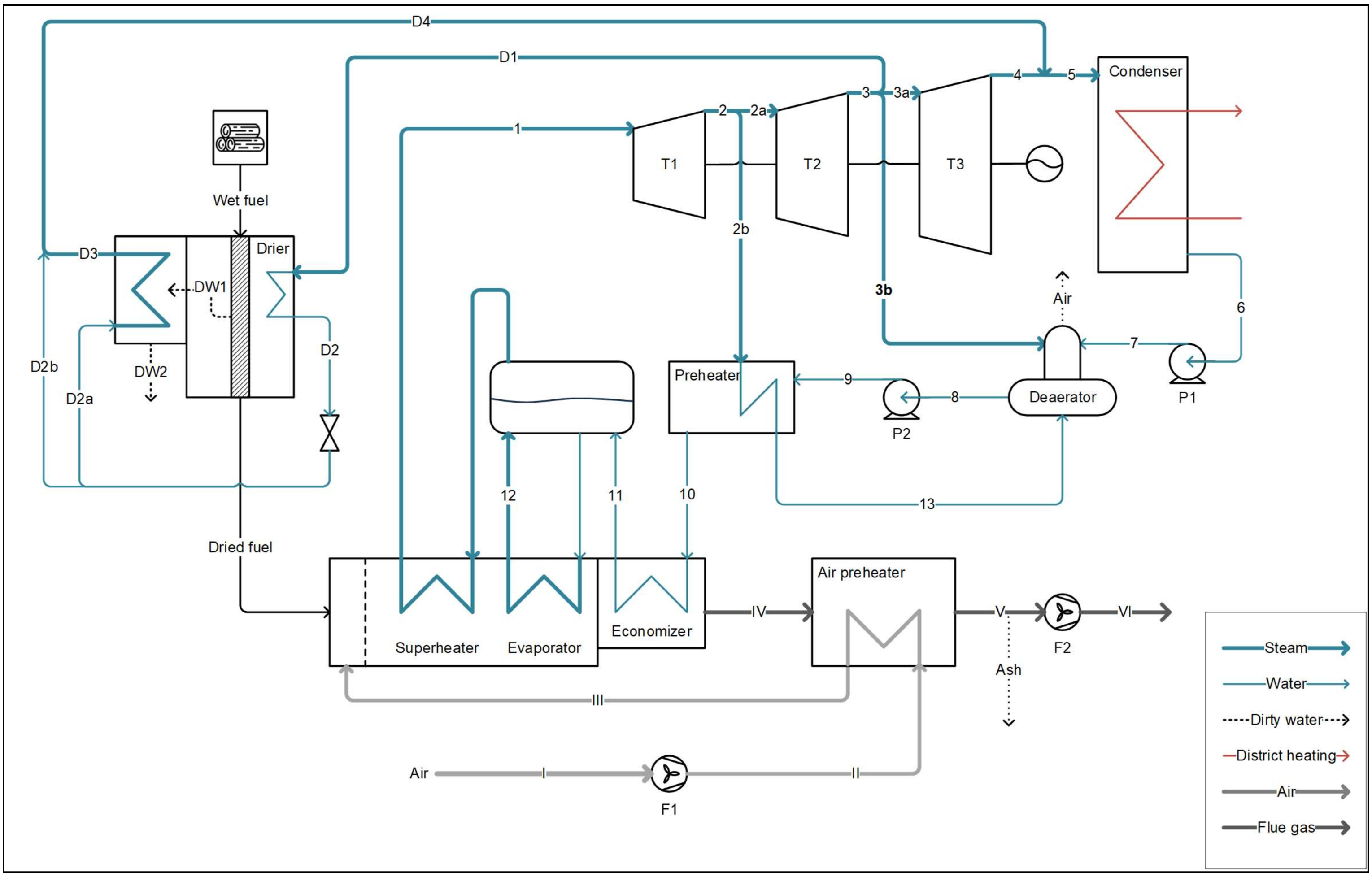

Figure 1. Process flowsheet

2.3 Calculation for the basis of 1kg MAF/s

Since the actual inlet mass flow of the fuel is unknown, all calculations in this section were carried out for the basis of 1 kg MAF/s. A dimensionless multiplication factor X, defined in eq. 1, will be evaluated in the very last step of the calculation (section 2.4.2). Then it will be multiplied to all calculation results to yield actual mass flowrates and enthalpy flows.

$$X = \dot{m}_{\mathrm{MAF,real}} / \dot{m}_{\mathrm{MAF,basis}} \tag{1}$$

To avoid confusion between calculation on the basis and real values, all extensive values calculated for the basis are denoted with a bar. For example,

$$\bar{\dot{m}} = \dot{m}/X, \qquad \bar{\dot{n}} = \dot{n}/X, \qquad \bar{H} = H/X.$$

2.3.1 Combustion

Step 1: Elemental analysis and calculation of the inlet composition

Solving the system begins with calculating mass and energy balances for the boiler section to establish proportional relationship between basis of inlet fuel and air led to the boiler. This can be done by following combustion calculations.

The elementary analysis for 1000 g of MAF is:

C: 540 g
H: 62 g
O: 398 g
Ash content of fuel: 10 g ($c_{P,\mathrm{ash}}$= 0.9 kJ/kg K)
Moisture content after the drier: 0.11 kg

Assuming a complete combustion takes place in the furnace, only two reactions occur in the boiler:

$$\mathrm{O_2 + C \rightarrow CO_2}$$

$$\frac{1}{2}\mathrm{O_2 + H_2 \rightarrow H_2O}$$

The combustion is performed in presence of oxygen supplied at 130% of the stoichiometric demand. Now it is possible to calculate the total amount of supplied components at the inlet of the boiler. Equations 2 and 3 were used to calculate the amount of oxygen and nitrogen entering the air stream (stream III), respectively. The result of this calculation step is presented in Table 1.

$$\bar{n}_{\mathrm{O_{2,air}}} = (1.3)\left(\bar{n}_{\mathrm{C}} + \bar{n}_{\mathrm{H_2}}/2\right) - \bar{n}_{\mathrm{O_{2,fuel}}} = 0.066\ \mathrm{kmol/s} \tag{2}$$

$$\bar{n}_{\mathrm{N_{2,air}}} = (79/21)\left(\bar{n}_{\mathrm{O_{2,air}}}\right) = 0.249\ \mathrm{kmol/s} \tag{3}$$

Table 1. Composition of biomass and air at the inlet of the boiler

Component	In [kg/s]	In [kmol/s]
C	0.540	0.045
H_2	0.062	0.031
$H_2O_{\ fuel}$	0.110	0.006
$O_{2\ fuel}$	0.398	0.012
$O_{2\ air}$	2.119	0.066
$N_{2\ air}$	6.974	0.249
CO_2	0	0

Step 2: Mass balance over the boiler

Now, considering the mass balance [In] + [Produced] = [Out], the composition of the flue gas after the combustion is determined, as shown in Table 2.

Table 2. Composition of the mixture of the biomass, air and the flue gas before and after the combustion

Component	In [kmol/s]	Produced [kmol/s]	Out [kmol/s]	Out [kg/s]
C	0.045	- 0.045	0	0
H_2	0.031	- 0.031	0	0
H_2O	0.006	+ 0.031	0.037	0.668
$O_{2,\ fuel}$	0.0124	- 0.061	0.018	0.581
$O_{2,\ air}$	0.066			
$N_{2,\ air}$	0.249	0	0.249	6.974
CO_2	0	+ 0.045	0.045	1.98

From these tables now the mass flowrates of the air entering the boiler and the flue gas exiting per kg of MAF can be calculated by equations 4 and 5.

$$\bar{\dot{m}}_{\text{air}} = \bar{\dot{m}}_{\text{O}_{2,\text{air}}} + \bar{\dot{m}}_{\text{N}_{2,\text{air}}} = 9.093\ \text{kg/s} \tag{4}$$

$$\bar{\dot{m}}_{\text{flue gas}} = \bar{\dot{m}}_{\text{H}_2\text{O}_{\text{out}}} + \bar{\dot{m}}_{\text{O}_{2\ \text{out}}} + \bar{\dot{m}}_{\text{N}_{2\ \text{out}}} + \bar{\dot{m}}_{\text{CO}_{2\ \text{out}}} = 10.2032\ \text{kg/s} \tag{5}$$

Step 3: Calculating enthalpies and temperatures of air/flue gas streams.

Knowing the composition of the inlet air, with the 'Air calculation sheet' (AC) provided it is possible to compute enthalpies and temperatures of air and flue gas streams if either enthalpy or temperature for a stream is known. Excel Goal Seek function was used when calculating temperature from enthalpy. Since we are still calculating for the basis of 1 kg MAF/s, all extensive properties calculated here are denoted with a bar (e.g., $\bar{H}$), to distinguish from actual values. It was assumed that ash is removed between the air preheater and the last fan, i.e., from stream V.

For stream I, both the composition and the temperature of 30 °C are known from the problem description, therefore the enthalpy flow of the stream, $\bar{H}_{\text{I}}$ can be specified by AC.

$$\bar{H}_{\text{I}} = 273.81\ \text{kW} \qquad \text{(from AC)} \tag{6}$$

For stream II, Adding the fan work of Fan 1 to $\bar{H}_{\mathrm{I}}$ gives $\bar{H}_{\mathrm{II}}$ as shown in equation 7. Ideal gas law was used to calculate volumetric flowrate of the air for calculation of fan works, since the streams are within moderate pressure and temperature ranges where ideal gas law gives sufficiently accurate prediction of thermodynamic properties of air.

$$\bar{H}_{\mathrm{II}} = \bar{H}_{\mathrm{I}} + \frac{(\bar{\dot{V}}_{\mathrm{air,in}})(\Delta P_{\mathrm{II-I}})}{\eta_{\mathrm{f}}} = 285.01\ \mathrm{kW} \tag{7}$$

η_{f} (efficiency of the fan) = 0.7
$\Delta P_{\mathrm{II-I}}$ (pressure drop inside the air preheater) = 1 kPa
$\bar{\dot{V}}_{\mathrm{air,in}}$ (volumetric flowrate of the air) $= \frac{(\bar{\dot{n}}_{\mathrm{air,in}})(R)(T_{\mathrm{I}})}{P_{\mathrm{I}}} = 7.843\ \mathrm{m^3/s}$

For stream III, the temperature is unknown, however, an energy balance in the air preheater can be set up as equation 8. The last term on the right-hand side of the equation takes account for the heat content of ash in the flue gas streams.

$$\bar{H}_{\mathrm{III}} - \bar{H}_{\mathrm{II}} = \bar{H}_{\mathrm{IV}} - \bar{H}_{\mathrm{V}} + (\bar{\dot{m}}_{\mathrm{ash}})\big(c_{P,\mathrm{ash}}\big)(T_{\mathrm{IV}} - T_{\mathrm{V}}) \tag{8}$$

For stream IV, the temperature has to be 20 °C above the temperature of stream 10 (T_{10}) entering the economiser to allow minimum temperature difference. T_{10} is calculated by allowing 10 °C temperature difference with the saturated condensate (stream 13) in heat exchange in the feed-water preheater. Therefore, T_{IV} can be calculated from T_{13} by a simple calculation (equations 9–11). Knowing T_{IV}, $\bar{H}_{\mathrm{IV}}$ can be now computed by AC (equation 12).

$$\mathrm{T}_{13} = \mathrm{T(sat.liq\ @12\ bar)} = 187.96\ \mathrm{°C} \qquad \text{(from T\&D)} \tag{9}$$

$$T_{10} = T_{13} - 10\ \mathrm{°C} = 177.96\ \mathrm{°C} \tag{10}$$

$$T_{\mathrm{IV}} = T_{10} + 20\ \mathrm{°C} = 197.96\ \mathrm{°C} \tag{11}$$

$$\bar{H}_{\mathrm{IV}} = 3814.57\ \mathrm{kW} \qquad \text{(from AC)} \tag{12}$$

For stream V, both the composition and the temperature of 120 °C are known, therefore H_{V} can be computed by Air calculation sheet. Now, with the properties specified for streams IV and V, previously formulated energy balance for the air preheater (equation 8) can be solved, specifying the state of stream III.

$$\bar{H}_{\mathrm{III}} = 1142.30\ \mathrm{kW} \qquad \text{(from AC)} \tag{13}$$

For stream VI, adding the second fan work to $\bar{H}_{\mathrm{V}}$ gives $\bar{H}_{\mathrm{VI}}$ as shown in equation 14.

$$\bar{H}_{\mathrm{VI}} = \bar{H}_{\mathrm{V}} + \frac{(\bar{V}_{\text{flue gas}})(\Delta P_{\mathrm{VI-V}})}{\eta_f} = 3136.61\ \mathrm{kW} \tag{14}$$

η_f (efficiency of the fan) = 0.7

$\Delta P_{\mathrm{VI-V}}$ (pressure drop inside the air preheater) = 10 kPa

$\bar{\dot{V}}_{\text{flue gas}}$ (volumetric flowrate of the flue gas) = $\frac{(\bar{n}_{\text{flue gas}})(R)(T_{\mathrm{V}})}{P_{\mathrm{V}}} = 12.504\ \mathrm{m^3/s}$

Table 3 summarises all the date calculated in this step.

Table 3. Air and flue gas data per kg of MAF

Stream (without ash)	Temperature [°C]	Pressure [kPa]	Enthalpy flow ($\bar{H}$) [kW]	Mass flowrate ($\bar{\dot{m}}$) [kg/s]
I	30	101.325	273.807	9.093
II	31.2	102.325	285.012	9.093
III	124.7	101.325	1142.299	9.093
IV	197.96	-	3814.568	10.2032
V	120	91.325	2957.983	10.2032
VI	136.4	101.325	3136.610	10.2032

2.3.2 Steam Cycle

Step 1: Turbine expansion, isentropic efficiency calculation

In order to specify properties of all the streams in the steam cycle, calculations have been carried out starting from the superheated steam after the boiler (stream 1) of which properties are already defined by the problem description. Specific enthalpy and entropy of the stream at 540 °C and 100 bar can be found in T&D [1].

$$h_1 = 3476.9\ \mathrm{kJ/kg},\ s_1 = s_{2s} = s_{3s} = s_{4s} = 6.7278\ \mathrm{kJ/kg\,K}$$

The isentropic specific enthalpy of point 2 ($h_{2,s}$) can be calculated by a simple interpolation on the steam table. This is possible because stream 2 is still superheated vapor, above the two-phase region. Given the isentropic efficiency of the steam turbine ($\eta_s = 0.85$), it is possible to compute the real specific enthalpy of stream 2 (h_2) by an interpolation (equation 15), again. Because streams 2a and 2b are split out from stream 2, their specific enthalpies are identical to h_2 as well.

$$h_2 = h_1 - \eta_s(h_1 - \mathrm{h}_{2,s}) = 3476.9 - 0.85(3476.9 - 2882.7) = 2971.8\ (\mathrm{kJ/kg}) \tag{15}$$

$$h_2 = h_{2\mathrm{a}} = h_{2\mathrm{b}} \tag{16}$$

For stream 3, the isentropic expansion ends up in the two-phase region at 3.5 bar and 138.85 °C. Therefore, the isentropic enthalpy of steam 3s can be calculated from interpolation between saturated liquid and saturated vapor properties of steam at 3.5 bar. From there, the enthalpy of stream 3 can be calculated in the same way as equation 15, taking the isentropic efficiency into account.

$$h_3 = h_1 - \eta_s\left(h_1 - h_{3,s}\right) = 3476.9 - 0.85(3476.9 - 2644.4) = 2769.3 \text{ kJ/kg} \tag{17}$$

$$h_3 = h_{3a} = h_{3b} = h_{D1} \tag{18}$$

Isentropic expansion from stream 1 to stream 4 also ends up in the two-phase region, at 0.5787 bar and 85 °C (equal to the condensing pressure of condensate leaving the condenser). Therefore, the same method is used to calculate $h_{4,s}$ and h_4.

$$x_{4,s} = \frac{s_{4,s} - s_l(85°\text{C})}{s_v(85°\text{C}) - s_l(85°\text{C})} = 0.873 \tag{19}$$

Now $h_{4,s}$ can be calculated as a weighted average of its liquid and vapor contents as in equation 20. Equation 21 calculated h_4 by the same way as in equation 17.

$$h_{4,s} = h_v(85°\text{C})x_{4,s} + h_l(85°\text{C})\left(1 - x_{4,s}\right) = 2359.2 \text{ kJ/kg} \tag{20}$$

$$h_4 = h_3 - \eta_s\left(h_3 - h_{4,s}\right) = 2526.85 \text{ kJ/kg} \tag{21}$$

Step 2: Finding saturated properties from the steam table

For a single stream of pure water/steam, information about two of its thermodynamic properties (e.g., pressure, temperature, steam quality, specific enthalpy, etc.) is sufficient to specify its state. In this step, thermodynamic properties for each stream that are known by problem description are identified and their specific enthalpies found from T&D are listed.

Many streams in the steam cycle are either saturated vapor or liquid at certain temperature or pressure, which allows specifying their thermodynamic properties. Streams 6 and 8 are saturated liquids at 85 °C and 3.5 bar, respectively.

$$h_6 = 356.0 \text{ kJ/kg}$$

$$h_8 = 584.2 \text{ kJ/kg}$$

Temperatures of streams 13 and 10 have already been found from T&D in equations 9 and 10, by setting minimum temperature difference of 10 °C. While stream 13 is a saturated liquid, stream 10 is subcooled liquid at 177.96 °C and 107 bar, considering pressure drop in the economiser (3 bar).

$$h_{10} = 759.1 \text{ kJ/kg}$$

$$h_{13} = 798.3 \text{ kJ/kg}$$

Streams entering and exiting the steam drum, streams 11 and 12, are assumed to be saturated liquid and vapor, respectively. In addition, by assuming there is no pressure drop in the evaporator, both streams are at 104 bar, which allows specifying their states.

$$h_{11} = 1425.1 \text{ kJ/kg}$$

$$h_{12} = 2718.0 \text{ kJ/kg}$$

Step 3: Pump work calculation

Specific enthalpies of streams 7 and 9 were calculated using the pump work equation, assuming an incompressible fluid and a pump efficiency of 0.75. The condenser pump increases the pressure of the condensate to 3.5 bar and the value of the specific enthalpy of the subcooled stream 7 can be calculated as shown in equation 22:

$$h_7 = h_6 + \frac{v_6(P_7 - P_6)}{\eta_{\text{p}}} = 356.402 \text{ kJ/kg} \tag{22}$$

η_{p} (efficiency of the pump) $= 0.75$
$P_7 - P_6 = 3.5 \text{ bar} - 0.5787 \text{ bar} = 2.9213 \text{ bar} = 292.13 \text{ kPa}$
v_6 (specific volume of the stream) $= 0.001032 \text{ m}^3/\text{kg}$ (from T&D)

The second feed water pump after the deaerator increases the pressure of stream 8 to 110 bar because there is 10 bar pressure drop from the high-pressure pump to the inlet of the high-pressure turbine. Equation 23 gives the value of h_9 by adding the feedwater pump work to h_8:

$$h_9 = h_8 + \frac{v_8(P_9 - P_8)}{\eta_{\text{p}}} = 599.522 \text{ kJ/kg} \tag{23}$$

η_{p} (efficiency of the pump) $= 0.75$
$P_9 - P_8 = 110 \text{ bar} - 3.5 \text{ bar} = 106.5 \text{ bar} = 10650 \text{ kPa}$
v_8 (specific volume of the stream) $= 0.001079 \text{ m}^3/\text{kg}$ (from T&D)

Table 4 summarises all the data that have been used and calculated in this section.

Table 4. Steam cycle data and unknown

Stream no.	Temperature [°C]	Pressure [bar]	Phase	Specific enthalpy [kJ/kg]	Specific volume [m^3/kg]
1	540	100	Superheated	3476.9	
2	266.12	12	Superheated	2971.84	
2a	266.12	12	Superheated	2971.84	
2b	266.12	12	Superheated	2971.84	
3	138.85	3.5	Two-phase	2769.25	
3a	138.85	3.5	Two-phase	2769.25	
3b	138.85	3.5	Two-phase	2769.25	
4	85	0.5787	Two-phase	2526.85	
5	85	0.5787	Two-phase	Unknown	
6	85	0.5787	Sat. liq.	356.0	0.001032
7	85.02	3.5	Subcooled	356.402	
8	138.85	3.5	Sat. liq.	584.2	0.001079
9	140.80	110	Subcooled	599.522	
10	177.96	107	Subcooled	759.05	
11	313.88	104	Sat. liq.	1425.1	
12	313.88	104	Sat. vap.	2718.0	
13	187.96	12	Sat. liq.	798.3	

The only unknown intensive property in the table is the specific enthalpy of the stream 5, since at this stage there is no sufficient information on the stream from the fuel drier that joins stream 4 at the mixing point.

2.3.3 Fuel Drier

Properties of all the streams involved in the drier unit are already well-defined by the problem description. Intensive properties of stream D1 have been calculated already, for they are identical to those of stream 3 from which it is split. Since the steam trap is isenthalpic, specific enthalpy of stream D2 is identical to that of D2a and D2b, which are saturated liquid at 1.0 bar.

$$h_{\mathrm{D2}} = h_{\mathrm{D2a}} = h_{\mathrm{D2b}} = h_l(1.0\ \mathrm{bar}) = 417.5\ \mathrm{kJ/kg}$$

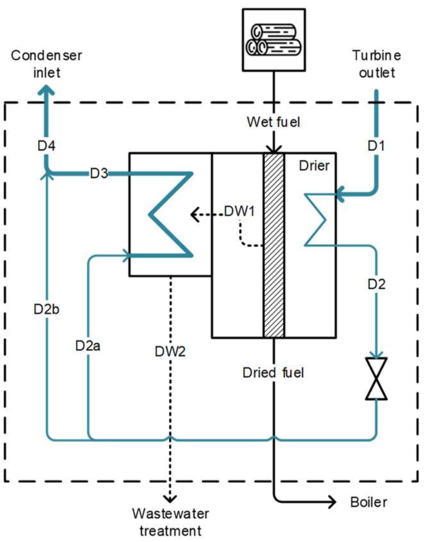

Figure 2. Flow diagram of the biofuel drier. HX1 is on the right and HX2 on the left.

Stream DW1, the water removed from the fuel, is saturated vapor at 2.0 bar leaving the first heat exchanger in the drier (HX1), and stream DW2 is saturated liquid at 2.0 bar, leaving the second heat exchanger (HX2) where it behaves as a condensing hot stream.

$$h_{\mathrm{DW1}} = h_v(2.0\ \mathrm{bar}) = 2706.2\ \mathrm{kJ/kg} \quad \text{(from T\&D)}$$

$$h_{\mathrm{DW2}} = h_l(2.0\ \mathrm{bar}) = 504.7\ \mathrm{kJ/kg} \quad \text{(from T\&D)}$$

Stream D3 is saturated vapor at 1.0 bar, evaporated by stream DW1. Lastly, stream D4 requires information on mass flowrates to be specified like stream 5.

$$h_{\mathrm{D3}} = h_v(1.0\ \mathrm{bar}) = 2674.9\ \mathrm{kJ/kg}$$

The calculation results above are summarised in Table 5.

Table 5. Stream data in the fuel drier and unknown

Stream no.	Temperature [°C]	Pressure [bar]	Phase	Specific enthalpy [kJ/kg]
D1	138.85	3.5	Two-phase	2769.3
D2	99.56	3.5	Subcooled	417.5
D2a	99.61	1	Sat. liq.	417.5
D2b	99.61	1	Sat. liq.	417.5
D3	99.61	1	Sat. vap.	2674.9
D4	99.61	1	Two-phase	Unknown
Dw1	120.21	2	Sat. vap.	2706.2
Dw2	120.21	2	Sat. liq.	504.7

2.4 Mass and Energy balances, calculation of extensive properties

2.4.1 Degree of freedom analysis

With intensive properties of the system all calculated except for some that depend on mass flowrates, a degree of freedom analysis was carried out to identify unknown variables and available equations to solve for all the mass flowrates.

All mass flowrates can be expressed by combinations of 9 dimensionless variables listed below.

$X\ (= \dot{m}_{\mathrm{MAF,real}}/\dot{m}_{\mathrm{MAF,basis}})$

$\bar{\dot{m}}_1\ (= \dot{m}_1/X)$

$x_{2a/2}(= \dot{m}_{2a}/\dot{m}_2)$ $\quad$ $x_{2b/2}(= \dot{m}_{2b}/\dot{m}_2)$

$x_{3a/3}(= \dot{m}_{3a}/\dot{m}_3)$ $\quad$ $x_{3b/3}(= \dot{m}_{3b}/\dot{m}_3)$ $\quad$ $x_{\mathrm{D1}/3}(= \dot{m}_{\mathrm{D1}}/\dot{m}_3)$

$x_{\mathrm{D2a/D2}}(= \dot{m}_{\mathrm{D2a}}/\dot{\mathrm{m}}_{\mathrm{D2}})$ $\quad$ $x_{\mathrm{D2b/D2}}(= \dot{m}_{\mathrm{D2b}}/\dot{m}_{\mathrm{D2}})$

By a simple mass balance at each split, following summations must hold true, giving 3 equations (MB–1–3), leaving 6 degrees of freedom.

$$x_{2a/2} + x_{2b/2} = 1 \qquad \text{(MB 1)}$$

$$x_{3a/3} + x_{3b/3} + x_{\mathrm{D1}/3} = 1 \qquad \text{(MB 2)}$$

$$x_{\mathrm{D2a}/D2} + x_{\mathrm{D2b/D2}} = 1 \qquad \text{(MB 3)}$$

Six equations for the system can be derived from energy balances over process units:

Boiler
Feed-water preheater
First heat exchanger in the drier
Second heat exchanger in the drier
Deaerator
Condenser

Therefore, it is shown that the degree of freedom for the system is zero, which suggests the system is well-defined, hence solvable.

$$\mathrm{DOF} = 9 - (3 + 6) = 0$$

2.4.2 Solution of the system

The system of mass and energy balances was solved step-by-step starting from simple to more complex equations. Although an analytical solution of the system is possible, the calculation was carried out numerically using Microsoft Excel Goal Seek function, with tolerance of 10^{-9} kW to ensure precision. Each equation is listed and expressed below in terms of previously listed 9 dimensionless unknowns, in the order of calculation. For most equations, enthalpies 'taken from' hot streams are placed on the left-hand side and energy 'taken by' cold streams right-hand side. At the end of each step, the calculated value of the variable the equations are solved for are presented.

Step 1. EB Boiler

$$X\left(\bar{E}_{\text{fuel,in}} + \bar{H}_{\text{III}} - \bar{H}_{\text{IV}} - 0.02(\text{LHV})\right) = \dot{m}_1 h_1 - \dot{m}_{10} h_{10} \tag{24}$$

or
$$\left(\bar{E}_{\text{fuel,in}} + \bar{H}_{\text{III}} - \bar{H}_{\text{IV}} - 0.02(\text{LHV})\right) = \bar{m}_1 (h_1 - h_{10}) \tag{25}$$

Solving this equation tells us the proportion of the mass flowrate of steam circulating in the steam cycle to that of the fuel inlet ($\bar{m}_1$).

$$\boldsymbol{\bar{m}_1 = 6.1789\ \text{kg}\ \text{H}_2\text{O}/\text{kg MAF}}$$

Step 2. EB Feed-water preheater

$$\dot{m}_{2b} h_{2b} - \dot{m}_{13} h_{13} = \dot{m}_{10} h_{10} - \dot{m}_9 h_9 \tag{26}$$

or
$$\dot{m}_1 x_{2b/2} (h_{2b} - h_{13}) = \dot{m}_1 (h_{10} - h_9) \tag{27}$$

By MB 1, $x_{2a/2}$ can be calculated as well.

$$\boldsymbol{x_{2a/2} = 0.92660, \qquad x_{2b/2} = 0.07339}$$

Step 3. EB HX1 (first heat exchanger in the drier)

$$\dot{m}_{D1} h_{D1} - \dot{m}_{D2} h_{D2} = \dot{m}_{\text{DW1}} h_{\text{DW1}} + \bar{E}_{\text{dried fuel}} - \bar{E}_{\text{wet fuel}} \tag{28}$$

or
$$\bar{m}_1 (\dot{m}_1 x_{2a/2} x_{\text{D1/3}})(h_{\text{D1}} - h_{\text{D2}}) = (\bar{m}_{\text{DW1}} h_{\text{DW1}} + \bar{E}_{\text{dried fuel}} - \bar{E}_{\text{wet fuel}}) \tag{29}$$

$$\boldsymbol{x_{\text{D1/3}} = 0.18519}$$

Step 4. EB HX2 (Second heat exchanger in the drier)

$$X(\bar{m}_{\text{DW1}} h_{\text{DW1}} - \bar{m}_{\text{DW2}} h_{\text{DW2}}) = \dot{m}_{\text{D3}} h_{\text{D3}} - \dot{m}_{D2a} h_{D2a} \tag{30}$$

or
$$(\bar{m}_{\text{DW1}} h_{\text{DW1}} - \bar{m}_{\text{DW2}} h_{\text{DW2}}) = \bar{m}_1 \left(x_{2a/2} x_{\text{D1/3}}\right)\left(x_{\text{D2a/D2}} h_{\text{D3}} - x_{\text{D2b/D2}} h_{D2a}\right) \tag{31}$$

Plugging in MB 3 in the equations 31 gives the values of both $x_{\text{D2a/D2}}$ and $x_{\text{D2b/D2}}$. This allows calculating specific enthalpy of the stream that returns to the turbine network (h_{D4}) by equation 32.

$$h_{\text{D4}} = x_{\text{D2a/D2}} h_{\text{D3}} + x_{\text{D2b/D2}} h_{D2b} \tag{32}$$

$$\boldsymbol{x_{\text{D2a/D2}} = 0.81861, \qquad x_{\text{D2b/D2}} = 0.18139, \qquad h_{\text{D4}} = 2265.4\ \text{kJ/kg}}$$

Step 5. EB Deaerator

$$\dot{m}_8 h_8 = \dot{m}_{3\mathrm{b}} h_{3\mathrm{b}} + \dot{m}_7 h_7 + \dot{m}_{13} h_{13} \tag{33}$$

where $\dot{m}_{3b} = \dot{m}_1 x_{2\mathrm{a}/2} x_{3\mathrm{b}/3}, \quad \dot{m}_7 = \dot{m}_1 x_{2\mathrm{a}/2}\left(1 - x_{3\mathrm{b}/3}\right),$

$$\dot{m}_8 = \dot{m}_1, \quad \dot{m}_{13} = \dot{m}_1 x_{2\mathrm{b}/2}$$

Now, with $x_{\mathrm{D}1/3}$ calculated in *Step 3*, $x_{3\mathrm{a}/3}$ can be calculated by MB 2 as well:

$$x_{3\mathrm{a}/3} = 1 - x_{\mathrm{D}1/3} - x_{3\mathrm{b}/3} \tag{MB 2}$$

It is reminded that h_5 was not presented in Table 4 (section 2.3.2) due to insufficient information about the mixing point where stream D4 and 4 joins. Now it can be calculated by an energy balance equation over the mixing point (equation 34):

$$\dot{m}_5 h_5 = \dot{m}_4 h_4 + \dot{m}_{\mathrm{D}4} h_{\mathrm{D}4} \tag{34}$$

or $\bar{\dot{m}}_1 x_{2\mathrm{a}/2}\left(x_{3a/3} + x_{\mathrm{D}1/3}\right) h_5 = \bar{\dot{m}}_1 x_{2\mathrm{a}/2}\left(x_{3a/3}\right) h_4 + \bar{\dot{m}}_1 x_{2\mathrm{a}/2}\left(x_{\mathrm{D}1/3}\right) h_4$

hence, $h_5 = \left(x_{3a/3} h_4 + x_{\mathrm{D}1/3} h_4\right) / \left(x_{3a/3} + x_{\mathrm{D}1/3}\right)$

$$\boldsymbol{x_{3\mathrm{b}/3} = 0.72743}, \quad \boldsymbol{x_{3\mathrm{b}/3} = 0.08738}, \quad \boldsymbol{h_5 = 2473.8\ \mathrm{kJ/kg}}$$

Step 6. EB Condenser

$$\dot{m}_5 h_5 - \dot{m}_6 h_6 = 50\ \mathrm{MW} \tag{35}$$

$$\bar{\dot{m}}_1\left(x_{2\mathrm{a}/2}\right)\left(1 - x_{3b/3}\right)\left(h_5 - h_6\right) X = 50\ \mathrm{MW}$$

Scaling-up the entire system by the factor of *X* to match the heat demand in the condenser for district heating network completes the mass and energy balance calculation.

$$\boldsymbol{X = 4.5185}$$

This means the plant takes 4.5185 kg MAF every second.

2.5 Efficiencies

The electrical efficiency of the plant can be computed by equation 36:

$$\eta_{\text{electrical}} = \frac{\dot{W}_{\text{el,net}}}{(\dot{m}_{\text{MAF,real}})(\text{LHV})} \tag{36}$$

Where $\dot{W}_{\text{el,net}}$ represents the net power output of the plant, calculated by equation 37.

$$\dot{W}_{\text{el,net}} = \Sigma\dot{W}_{\text{turbine}} - \left(\Sigma\dot{W}_{\text{fan}} + \Sigma\dot{W}_{\text{pump}}\right) \tag{37}$$

In particular, the power produced by the turbines can be calculated by equations 38–40 and the power consumption of fans and pumps can be computed by equations 41–44. From the calculations in the previous sections, it is possible to compute all these terms:

$$\dot{W}_{\text{turbine 1}} = \eta_{\text{mech}}\eta_{\text{gen}}\left(\dot{m}_1(h_1 - h_2)\right) = 13.13\ \text{MW} \tag{38}$$

$$\dot{W}_{\text{turbine 2}} = \eta_{\text{mech}}\eta_{\text{gen}}\left(\dot{m}_3(h_{2a} - h_3)\right) = 4.88\ \text{MW} \tag{39}$$

$$\dot{W}_{\text{turbine 3}} = \eta_{\text{mech}}\eta_{\text{gen}}\left(\dot{m}_4(h_{3a} - h_4)\right) = 4.25\ \text{MW} \tag{40}$$

$$\dot{W}_{\text{fan 1}} = \frac{(\dot{V}_{\text{air,in}})(\Delta P_{\text{II-I}})}{\eta_{\text{f}}} = 0.05\ \text{MW} \tag{41}$$

$$\dot{W}_{\text{fan 2}} = \frac{(\dot{V}_{\text{flue gas}})(\Delta P_{\text{VI-V}})}{\eta_{\text{f}}} = 0.81\ \text{MW} \tag{42}$$

$$\dot{W}_{\text{pump 1}} = \dot{m}_6\frac{v_6(P_7 - P_6)}{\eta_{\text{p}}} = 0.01\ \text{MW} \tag{43}$$

$$\dot{W}_{\text{pump 2}} = \dot{m}_8\frac{v_8(P_9 - P_8)}{\eta_{\text{p}}} = 0.43\ \text{MW} \tag{44}$$

$\dot{W}_{\text{el,net}} = 13.13\ \text{MW} + 4.88\ \text{MW} + 4.25\ \text{MW} - (0.05\ \text{MW} + 0.81\ \text{MW} + 0.01\ \text{MW} + 0.43\ \text{MW}) =$ $20.96\ \text{MW}$While the lower heating value per kg MAF can be calculated by equation 45.

$$\text{LHV} = \text{HHV} - \left(\overline{m}_{\text{H}_2\text{O,out}}\right)\left(\Delta h_{\text{evap}}(25°\text{C})\right) = 17968.94\frac{\text{kW}}{\text{kg MAF}} \tag{45}$$

$\text{HHV (higher heating value of the fuel)} = 19600\ \text{kW/kg MAF}$
$\overline{m}_{\text{H}_2\text{O,out}} = 0.11\ \text{kg} + 0.558\ \text{kg} = 0.668\ \text{kg}$
$\Delta h_{\text{evap}} = 2447.1\ \text{kJ/kg H}_2\text{O (from T\&D)}$

Now, plugging in the values obtained from equations 37 and 45 to equation 36, the electrical efficiency of the plant can be evaluated:

$$\eta_{\text{electrical}} = \frac{(20.95\ \text{MW})\left(\frac{1000\ \text{kW}}{1\ \text{MW}}\right)}{(4.5185\ \text{kg MAF})(17968.94\ \text{kW/kg MAF})}(100\ \%) = 25.81\ \%$$

The heat efficiency can be calculated in a similar way, just replacing the numerator with the useful heat as the numerator, that in this case corresponds to the heat produced in the condenser.

$$\eta_{\text{heat}} = \frac{\dot{Q}_{\text{cond}}}{(\dot{m}_{\text{MAF,real}})(\text{LHV})} = \frac{(50\ \text{MW})\left(\frac{1000\ \text{kW}}{1\ \text{MW}}\right)}{(4.5185\ \text{kg MAF})(17968.94\ \text{kW/kg MAF})}(100\ \%) = 61.58\%$$

2.6 Q-T diagram

Step 1. Completing air and flue gas data table with the actual flowrates.

By multiplying X(section 2.4.2) to the '$\overline{H}$' column of Table 3 (section 2.3.1), Table 6 can be completed, now including the actual mass flowrates and enthalpy flows. For the Q-T diagram, what we are particularly interested in Table 6 are streams I, II, III, and IV.

Table 6. Air and flue gas data

Stream no.	Pressure [bar]	Temperature [°C]	Enthalpy flow (H) [kW]	Flowrate [kg/s]
I	1.01325	30	1237.20	41.0875
II	1.02325	31.23	1287.83	41.0875
III	1.01325	124.74	5161.52	41.0875
IV	-	197.96	17236.11	46.1030
V	0.91325	120	13365.63	46.1030
VI	1.01325	136.39	14172.74	46.1030

Step 2. Identifying hot and cold streams, calculating temperatures after convection loss.

In order to draw a complete QT-diagram for the boiler and the air preheater, it is necessary to first identify the hot and the cold streams that pass through the boiler and the air preheater. It was assumed that a complete combustion takes place at the very first part of the boiler, and convection heat loss (2% of the LHV) is considered at the highest temperature of the flue gas, before any heat exchange with the cold stream. These assumptions allow calculating temperatures of "imaginary" states of streams that are not identifiable otherwise: H_0, the flue gas after combustion before any convection loss or heat exchange, and H_1, the flue gas after convection loss before any heat exchange with the cold side. The temperature of H_1 is the highest temperature that is available for heat exchange with the cold stream. Plugging in equations 46–48 as well as previously presented values of HHV and $\overline{H}_{\text{III}}$ to equation 49 calculates $\overline{H}_{\text{H}_0}$, from which deducting the convection loss gives $\overline{H}_{\text{H}_1}$ (equation 50).

$$\overline{H}_{\text{MAF}} = (\dot{m}_{\text{MAF,basis}})(c_{P,\text{MAF}})(T_{\text{dried fuel}} - 0°\text{C}) \tag{46}$$

$$\overline{H}_{\text{ash}} = (\overline{m}_{\text{ash}})(c_{P,\text{ash}})(T_{\text{dried fuel}} - 0°\text{C}) \tag{47}$$

$$\bar{H}_{\mathrm{H_2O,dried\ fuel}} = \dot{\bar{m}}_{\mathrm{H_2O,dried\ fuel}}\left(h_{l@T_{\mathrm{dried\ fuel}}}\right) \tag{48}$$

$$\bar{H}_{\mathrm{H_0}} = \bar{H}_{\mathrm{MAF}} + \bar{H}_{\mathrm{ash}} + \bar{H}_{\mathrm{H_2O}} + \mathrm{HHV} + \bar{H}_{\mathrm{III}} = 20967.1\ \mathrm{kW} \tag{49}$$

$$\bar{H}_{\mathrm{H_1}} = \bar{H}_{\mathrm{H_0}} - \dot{Q}_{\mathrm{loss}} = 20608.4\ \mathrm{kW} \tag{50}$$

To determine the temperatures of streams H_0 and H_1, Goal Seek feature on Excel was used to find the temperatures where the total heat content of the flue gas and the ash matches the values presented in equations 49 and 50. Equations 51 and 52 are the equations.

$$\bar{H}_{\mathrm{H_0,without\ ash}} + \bar{m}_{\mathrm{ash}} c_{P,\mathrm{ash}}(T_0 - 0°\mathrm{C}) = 20954.2\ \mathrm{kW} + 12.9\ \mathrm{kW} = 20967.1\ \mathrm{kW} = \bar{H}_{\mathrm{H_0}} \tag{51}$$

$$\bar{H}_{\mathrm{H_1,without\ ash}} + \bar{m}_{\mathrm{ash}} c_{P,\mathrm{ash}}(T_1 - 0°\mathrm{C}) = 20595.6\ \mathrm{kW} + 12.8\ \mathrm{kW} = 20608.4\ \mathrm{kW} = \bar{H}_{\mathrm{H_1}} \tag{52}$$

Five hot streams and four cold streams can be identified in the system. States of streams denoted with subscripts, for example H_0 and H_1, are not to be confused with hot streams with non-subscript numbers, such as H0 and H1, that are the full streams. H_0 and H_1 are the initial states of the hot streams H0 and H1. The hot and cold streams are displayed in the boiler and air preheater flowsheet in Figure 3.

Table 7. Hot and cold streams through each unit and their initial/final states

Hot/cold streams	Initial and final states	Unit
H0	$H_0 \rightarrow H_1$	Convection loss
H1	$H_1 \rightarrow H_{1a}$	Superheater
H1a	$H_{1a} \rightarrow H_{1b}$	Evaporator
H1b	$H_{1b} \rightarrow H_2$ (= IV)	Economizer
H2	H_2 (= IV) $\rightarrow$ V	Air Preheater
C1	10 → 11	Economizer
C2	11 → 12	Evaporator
C3	12 → 1	Superheater
C4	II → III	Air Preheater

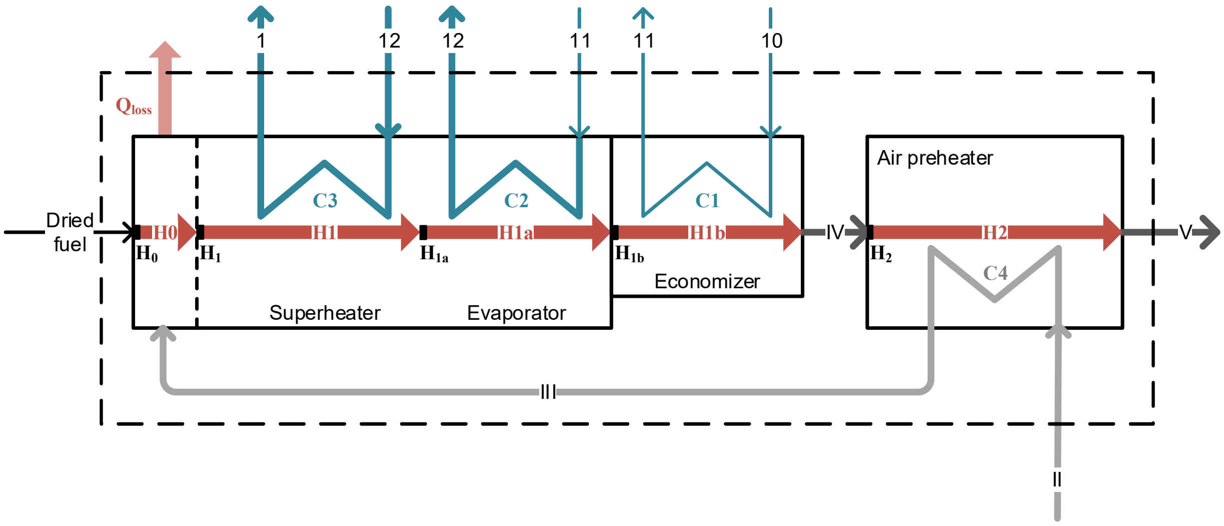

Figure 3. Boiler section scheme

Step 3. Temperatures of the hot streams at sections in the boiler.

Then, using the energy balances in the sections inside the boiler (Figure 3), the temperature and enthalpy of the flue gas at each step stage in the boiler can be calculated.

EB Economizer: $H_{\mathrm{H_{1b}}} = H_{\mathrm{IV}} + \dot{m}_{10}(h_{11} - h_{10}) = 31962.11\ \mathrm{kW}$

EB Evaporator: $H_{\mathrm{H_{1a}}} = H_{\mathrm{H_{1b}}} + \dot{m}_{11}(h_{12} - h_{11}) = 68057.16\ \mathrm{kW}$

Dividing these values by X, the specific enthalpy per kg of MAF can be obtained in order to calculate the temperature of these streams:

$$\bar{H}_{\mathrm{H_{1b}}} = \frac{H_{\mathrm{H_{1b}}}}{X} = 7930.8\ \mathrm{kW}$$

$$\bar{H}_{\mathrm{H_{1a}}} = \frac{H_{\mathrm{H_{1a}}}}{X} = 15919.1\ \mathrm{kW}$$

Now, Goal Seek feature is once again used to find the temperatures, giving following results.

$$\bar{H}_{\mathrm{H_{1b,without\ ash}}} + \bar{m}_{\mathrm{ash}} c_{P,\mathrm{ash}} \left(T_{\mathrm{H_{1b}}} - 0°C\right) = 7925.9\ \mathrm{kW} + 5.0\ \mathrm{kW} = 7930.8\ \mathrm{kW} = \bar{H}_{\mathrm{H_{1b}}}$$

$$\bar{H}_{\mathrm{H_{1a,without\ ash}}} + \bar{m}_{\mathrm{ash}} c_{P,\mathrm{ash}} \left(T_{\mathrm{H_{1a}}} - 0°C\right) = 15908.8\ \mathrm{kW} + 10.3\ \mathrm{kW} = 15919.1\ \mathrm{kW} = \bar{H}_{\mathrm{H_{1a}}}$$

Table 8 presents the temperatures and enthalpies of the hot/air steams.

Table 8. Hot stream data inside the boiler

Hot stream point	Temperature [°C]	Enthalpy flow ($\overline{H}$) [kW]
H_0	1438.9	20967.1
H_1	1420.7	20608.4
H_{1a}	1145.6	15919.1
H_{1b}	550.6	7930.8
H_2 (= IV)	198.0	3816.3

Step 4. Initial and final temperatures of the cold streams.

Regarding the water/steam that passes through the boiler section, specifying the initial and final temperatures of the streams is done by considering that the streams entering and exiting the evaporator are either saturated liquid and vapor, respectively. Table 9 shows the thermodynamic conditions of the cold streams.

Table 9. Cold stream data in the boiler section

Cold stream	Initial temperature [°C]	Final temperature [°C]	Enthalpy change (ΔH) [kW]
C1	177.96	313.88	18596.6
C2	313.88	313.88	36095.0
C3	313.88	540.00	21188.4
C4	31.23	124.74	3873.7

With the results calculated so far, it is now possible to determine Q-T coordinates for a complete Q-T diagram, which will be presented in section 3.4.

2.7 Investment calculations

The profitability of the plant was evaluated, analysing the incomes and the expenses using 2 different fuel cost, 2 different green certificate incomes and three different electricity prices, for a total of 12 scenarios. Before comparing the different scenarios, the incomes and the expenses must be evaluated as follows:

Electricity and heat production for an operating time of 5000 h/yr

Net electricity sold has been calculated by equation 37 in section 2.5 as 20.96 MW.

Electricity: $$(5000\ \mathrm{h/yr})(20.96\ \mathrm{MW}) = 104796\ \mathrm{MWh/yr}$$

Heat: $$(5000\ \mathrm{h/yr})(50\ \mathrm{MW}) = 250000\ \mathrm{MWh/yr}$$

Annual incomes

The heat income is always the same for the different scenarios (with a district heating income of 300 SEK/MWh).

Heat income: $$(250{,}000\ \mathrm{MWh/yr})(300\ \mathrm{SEK/MWh})\left(10^{-6}\,\frac{\mathrm{MSEK}}{\mathrm{SEK}}\right) = 75\ \mathrm{MSEK/yr}$$

The additional income from the green certificates (30 SEK/MWh or 150 SEK/MWh) can be calculated as:

GC income: $$(104796\ \mathrm{MWh/yr})\left(\mathrm{GC\ value}\ \left[\frac{\mathrm{SEK}}{\mathrm{MWh}}\right]\right)\left(\frac{10^{-6}\ \mathrm{MSEK}}{1\ \mathrm{SEK}}\right)$$

The electricity income must be calculated for three different electricity prices (200 SEK/MWh, 600 SEK/MWh and 1400 SEK/MWh) as follows:

Electricity: $$(104796\ \mathrm{MWh/yr})\left(\mathrm{Electricity\ price}\left[\frac{\mathrm{SEK}}{\mathrm{MWh}}\right]\right)\left(\frac{10^{-6}\ \mathrm{MSEK}}{1\ \mathrm{SEK}}\right)$$

Annual costs

With a specific investment of 44 MSEK/MWe and an annuity of 10 %/yr, the annual capital cost can be calculated as:

Investment: $$(20.96\ \mathrm{MW_e})\left(44\frac{\mathrm{MSEK}}{\mathrm{MW_e}}\right) = 922.2\ \mathrm{MSEK}$$

Capital cost: $$(20.96\ \mathrm{MW_e})\left(44\frac{\mathrm{MSEK}}{\mathrm{MW_e}}\right)(0.1/\mathrm{yr}) = 92.22\ \mathrm{MSEK/yr}$$

Operation and maintenance costs consisted of one cost based on the percentage of the total capital cost (1.5 %) and one cost based on the annual fuel consumption (40 $\mathrm{SEK/MWh_{fuel}}$), therefore considering the energy content of the fuel. The calculations are shown below:

O&M 1: $$(0.015/\mathrm{yr})(922.2\ \mathrm{MSEK}) = 13.83\ \mathrm{MSEK/yr}$$

O&M 2: $$\left(40\frac{\mathrm{SEK}}{\mathrm{MWh_{fuel}}}\right)\left(5000\frac{\mathrm{h}}{\mathrm{yr}}\right)\left(4.5185\frac{\mathrm{kg\ MAF}}{\mathrm{s}}\right)\left(17.969\frac{\mathrm{MJ}}{\mathrm{kg\ MAF}}\right)\left(\frac{1\ \mathrm{MSEK}}{10^6\ \mathrm{SEK}}\right) = 16.24\ \mathrm{MSEK/yr}$$

Fuel cost

The fuel costs can be calculated by multiplying the specific fuel cost (150 SEK/MWh and 250 SEK/MWh) with the operating time and the energy of the fuel:

Fuel cost 1: $$\left(150\frac{\mathrm{SEK}}{\mathrm{MWh}}\right)\left(5000\frac{\mathrm{h}}{\mathrm{yr}}\right)\left(4.5185\frac{\mathrm{kg\ MAF}}{\mathrm{s}}\right)\left(17.969\frac{\mathrm{MJ}}{\mathrm{kg\ MAF}}\right)\left(\frac{1\ \mathrm{MSEK}}{10^6\ \mathrm{SEK}}\right) = 60.89\frac{\mathrm{MSEK}}{\mathrm{yr}}$$

Fuel cost 2: $$\left(250\frac{\mathrm{SEK}}{\mathrm{MWh}}\right)\left(5000\frac{\mathrm{h}}{\mathrm{yr}}\right)\left(4.5185\frac{\mathrm{kg\ MAF}}{\mathrm{s}}\right)\left(17.969\frac{\mathrm{MJ}}{\mathrm{kg\ MAF}}\right)\left(\frac{1\ \mathrm{MSEK}}{10^6\ \mathrm{SEK}}\right) = 101.49\frac{\mathrm{MSEK}}{\mathrm{yr}}$$

At the end the annual profit is calculated as:

$$\text{Annual profit} = \text{Heat income} + \text{Electricity income} + \text{GC income} - \text{Capital cost} - \mathrm{O\&M_{tot}} - \text{Fuel cost}$$

3 Results and Discussion

3.1 Complete stream table

All the intensive and extensive properties for streams in the steam cycle are now presented in Table 10.

Table 10. Complete property table for the water streams

Stream no.	Pressure [bar]	Temperature [°C]	Specific enthalpy [kJ/kg]	Flowrate [kg/s]	Enthalpy flow [kW]	Phase
1	100	540	3476.9	27.9192	97072.23	Superheated
2	12	266.12	2971.8	27.9192	82971.43	Superheated
2a	12	266.12	2971.8	25.8700	76881.41	Superheated
2b	12	266.12	2971.8	2.0492	6090.02	Superheated
3	3.5	138.85	2769.3	25.8700	71640.41	Two-phase
3a	3.5	138.85	2769.3	18.8185	52113.13	Two-phase
3b	3.5	138.85	2769.3	2.2606	6260.05	Two-phase
4	0.5787	85	2526.9	18.8185	47551.49	Two-phase
5	0.5787	85	2473.8	23.6094	58404.94	Two-phase
6	0.5787	85	356.0	23.6094	8404.94	Sat. liq.
7	3.5	85.02	356.402	23.6094	8414.43	Subcooled
8	3.5	138.85	584.2	27.9192	16310.39	Sat. liq.
9	110	140.80	599.5	27.9192	16738.16	Subcooled
10	107	177.96	759.1	27.9192	21192.27	Subcooled
11	104	313.88	1425.1	27.9192	39788.76	Sat. liq.
12	104	313.88	2718.0	27.9192	75883.80	Sat. vap.
13	12	187.96	798.3	2.0492	1635.91	Sat. liq.
D1	3.5	138.85	2769.3	4.7909	13267.24	Two-phase
D2	3.5	99.56	417.5	4.7909	2000.21	Subcooled
D2a	1	99.61	417.5	3.9219	1637.38	Sat. liq.
D2b	1	99.61	417.5	0.8690	362.82	Sat. liq.
D3	1	99.61	2674.9	3.9219	10490.63	Sat. vap.
D4	1	99.61	2265.4	4.7909	10853.45	Two-phase
Dw1	2	120.21	2706.2	4.0215	10882.88	Sat. vap.
Dw2	2	120.21	504.7	4.0215	2029.63	Sat. liq.

Table 11 summarises properties or air and flue gas streams considering the real flowrates.

Table 11. Complete property table for air and flue gas streams

Stream no.	Pressure [bar]	Temperature [°C]	Flowrate [kg/s]	Enthalpy flow [kW]
I	1.01325	30	41.0875	1237.20
II	1.02325	31.23	41.0875	1287.83
III	1.01325	124.74	41.0875	5161.52
IV	-	197.96	46.1030	17236.11
V	0.91325	120	46.1030	13365.63
VI	1.01325	136.39	46.1030	14172.74

3.2 P-h Diagram of the steam cycle

The data on Table 9 can be also presented on a P-h diagram of steam, as shown in Figure 4.

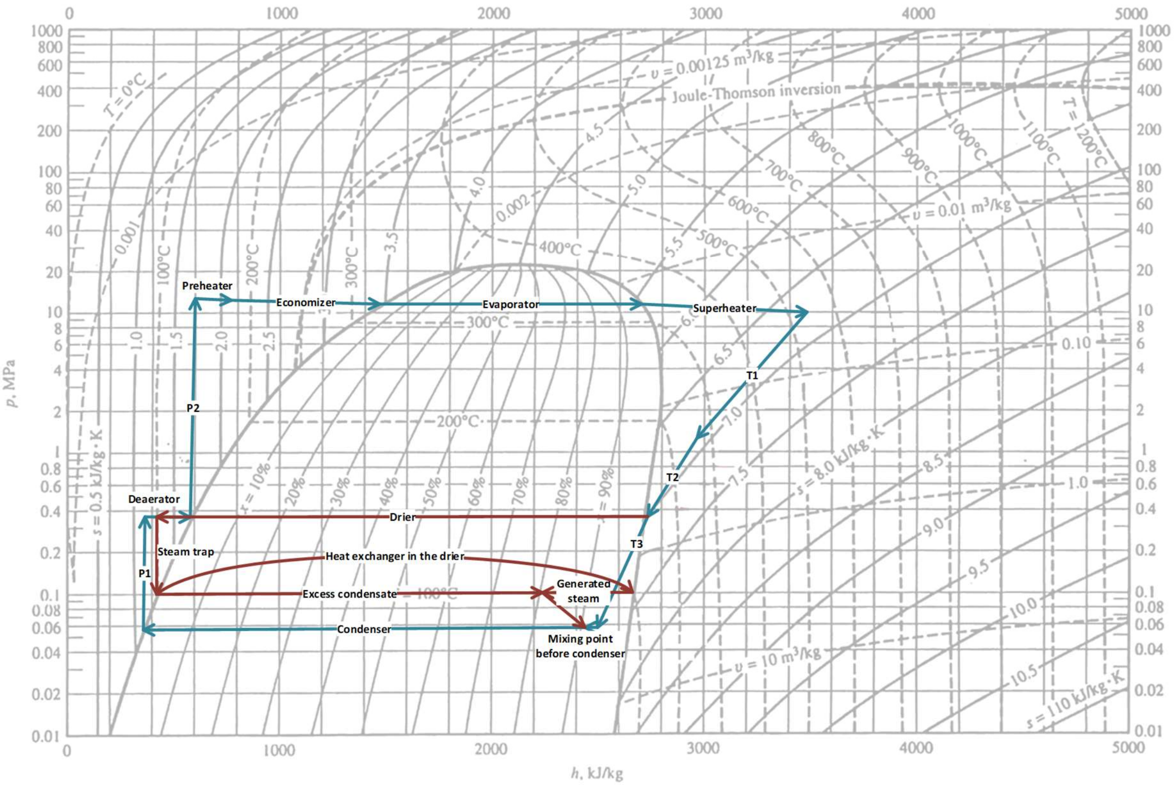

Figure 4. ph diagram of the steam cycle

3.3 Efficiencies

Electrical and heat efficiencies of the plant have already been evaluated in section 2.5

$$\eta_{\text{electrical}} = \frac{(20.95\ \text{MW})\left(\frac{1000\ \text{kW}}{1\ \text{MW}}\right)}{(4.5185\ \text{kg MAF})(17968.94\ \text{kW/kg MAF})}(100\ \%) = 25.81\ \%$$

$$\eta_{\text{heat}} = \frac{\dot{Q}_{\text{cond}}}{(\dot{m}_{\text{MAF,real}})(\text{LHV})} = \frac{(50\ \text{MW})\left(\frac{1000\ \text{kW}}{1\ \text{MW}}\right)}{(4.5185\ \text{kg MAF})(17968.94\ \text{kW/kg MAF})}(100\ \%) = 61.58\%$$

3.4 Q-T diagram

In order to determine all the Q-T coordinates, it is necessary to calculate the heat exchanged between the hot and cold streams in the different components.

Starting from the air preheater, heat is exchanged between air (II → III) and flue gas (IV → V) and it can be calculated as:

$$\dot{Q}_1 = H_{\text{III}} - H_{\text{II}} = 3873.7\ \text{kW}$$

After that, in the economiser, heat is exchanged between water (10 → 11) and flue gas (H_{1b} → IV) and it can be calculated as:

$$\dot{Q}_2 = \dot{m}_{10}(h_{11} - h_{10}) = 18596.5\ \text{kW}$$

Then there is the evaporator where the heat is exchanged between water in phase transition (11 → 12) and flue gas (H_{1a} → H_{1b}) and can be computed as:

$$\dot{Q}_3 = \dot{m}_{10}(h_{12} - h_{11}) = 36095\ \text{kW}$$

The last component is the superheater where heat is exchanged between steam (12 → 1) and flue gas (H_1 → H_{1a}) and can be calculated as:

$$\dot{Q}_3 = \dot{m}_{10}(h_1 - h_{12}) = 21188.4\ \text{kW}$$

Now, summing all the heats and knowing the temperature of each interval (summarised in section 2.6), it is possible to draw the Q-T diagram (Figure 5).

Table 12. Q-T coordinates

$\dot{Q}$ [MW]	$\dot{Q}$ [kW]	Temperature, cold streams [°C]	Temperature, hot streams [°C]
0.0	0	31	120
3.9	3873.7	125	198
3.9	3873.7	178	198
22.5	22470.2	314	551
58.6	58565.2	314	1146
79.8	79753.7	540	1421
81.4	81374.7	-	1439

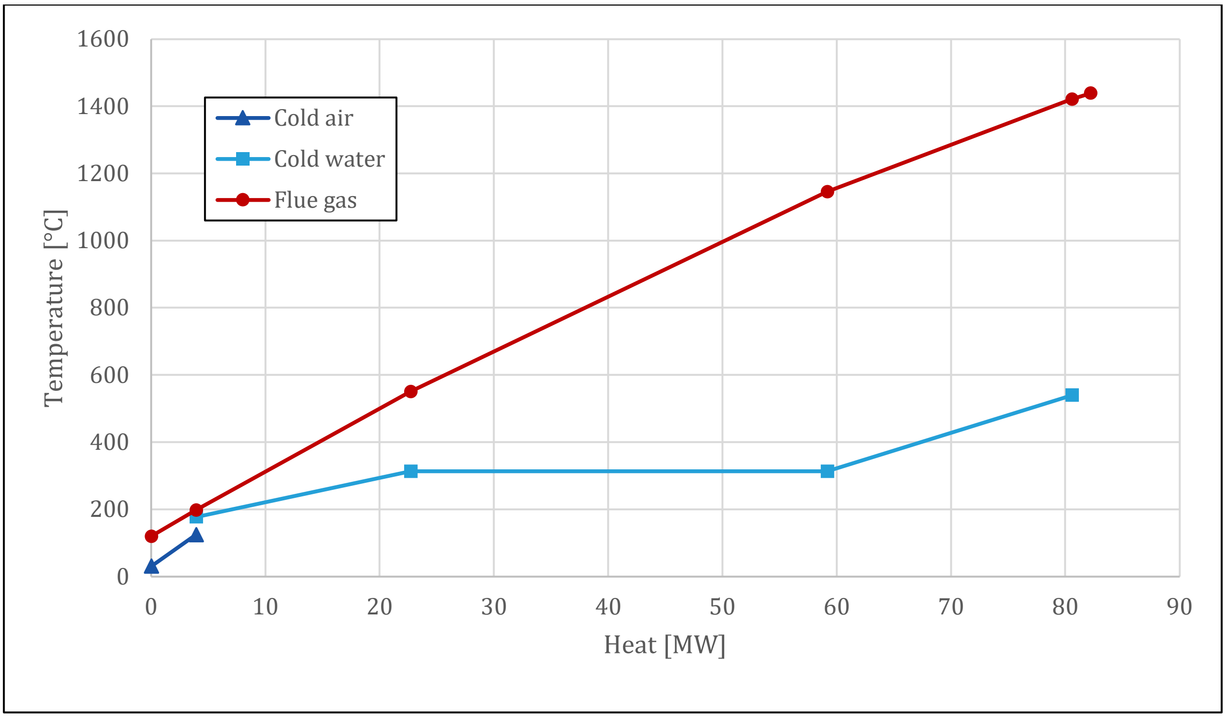

Figure 5. Q-T diagram for the air preheater and the boiler

The red curve represents the heat that is lost from the flue gas in the different components, starting from the air preheater to the superheater. It is possible to notice that the slope of the curve gradually decreases, since the c_P of air is higher at higher temperatures. The blue curves represent the cold streams. The first one on the bottom left inlet air going through the air preheater, and the second curve is water in the boiler section (economiser, evaporator, superheater from left to right).

Two points on the hot stream curve on the top right of the diagram represent the convection loss in the boiler room. It is not entirely correct to place them there because the heat loss takes place in the whole temperature interval at which the boiler and the air preheater are operating. However, it was placed there for illustration purposes.

The minimum temperature difference is found between cold water stream entering the economiser and the flue gas stream, where the temperature difference of 20 °C was considered when calculating stream properties.

3.5 Investment calculations

The results of the investment calculations for the 12 different cases are shown in the Tables 13, 14 and 15 below.

Table 13. Annual costs and incomes with a power price of 200 SEK/MWh

Fuel price	150	250	[SEK/MWh]
Fuel cost	- 60.9	- 101.5	MSEK
Electricity income	21.0	21.0	MSEK
Heat income	75.0	75.0	MSEK
GC income (30)	3.1	3.1	MSEK
GC income (150)	15.7	15.7	MSEK
Capital cost	- 92.2	- 92.2	MSEK
OP and MAINT 1	- 13.8	- 13.8	MSEK
OP and MAINT 2	- 16.2	- 16.2	MSEK
Annual profit (with add. 30)	- 84.1	- 124.7	MSEK
Annual profit (with add. 150)	- 71.5	- 112.1	MSEK

Table 14. Annual costs and incomes with a power price of 600 SEK/MWh

Fuel price	150	250	[SEK/MWh]
Fuel cost	- 60.9	- 101.5	MSEK
Electricity income	62.9	62.9	MSEK
Heat income	75.0	75.0	MSEK
GC income (30)	3.1	3.1	MSEK
GC income (150)	15.7	15.7	MSEK
Capital cost	- 92.2	- 92.2	MSEK
OP and MAINT 1	- 13.8	- 13.8	MSEK

OP and MAINT 2	- 16.2	- 16.2	MSEK
Annual profit (with add. 30)	- 42.2	- 82.8	MSEK
Annual profit (with add. 150)	- 29.6	- 70.2	MSEK

Table 15. Annual costs and incomes with a power price of 1400 SEK/MWh

Fuel price	150	250	[SEK/MWh]
Fuel cost	- 60.9	- 101.5	MSEK
Electricity income	146.7	146.7	MSEK
Heat income	75.0	75.0	MSEK
GC income (30)	3.1	3.1	MSEK
GC income (150)	15.7	15.7	MSEK
Capital cost	- 92.2	- 92.2	MSEK
OP and MAINT 1	- 13.8	- 13.8	MSEK
OP and MAINT 2	- 16.2	- 16.2	MSEK
Annual profit (with add. 30)	41.7	1.1	MSEK
Annual profit (with add. 150)	54.2	13.7	MSEK

Table 16 summarises annual profits for all considered scenarios.

Table 16. Annual profits for all considered scenarios

Fuel cost		150		250	
GC income		150	30	150	30
Electricity price	200	- 71.5	- 84.1	- 112.1	- 124.7
	600	- 29.6	- 42.2	- 70.2	- 82.8
	1400	54.2	41.7	13.7	1.1

From table 16, it is visible that the system is profitable when the electricity price is 1400 SEK/MWh, however, in the Figure 7 below, it is computed the annual profit as a function of the electricity price for the different combinations of fuel costs and green certificates. It is possible to notice that, in the best case, the system could be profitable if the electricity price is above 890 SEK/MWh. In addition, it can also be seen that the fuel cost affects the profit more than the income from the green certificates. This can be seen through a comparison of the gap between the 150/30 and the 250/30 line (representing the effect of the increase of the fuel cost with the same green certificate income) and between the 150/30 and the 150/150 line (representing the effect of the increase of the green certificate income keeping the same fuel cost).

Figure 6. Annual profit as a function of the electricity price by (fuel cost/GC income).

4 Conclusion

Combined heat and power (CHP) is a technology characterised by the simultaneous production and utilisation of heat and electricity. A CHP plant, particularly together with district heating, is an important part of greenhouse gas (GHG) emissions reduction strategies, due to higher overall plant efficiency and a reduced need for fuels. In particular, in our case, the renewable electricity produced (based on wet biofuel) with an electrical efficiency of 25.8% and a heat efficiency of 61.6% competes with fossil power generation in the Nordics and Europe. In addition, integration of the fuel drier reduces the moisture content of biomass, allowing the system to keep a high combustion efficiency (lowering pollution), a high LHV and to increase the temperature of biomass entering the boiler.

Our investment calculations concluded that the plant is profitable only if the electricity price is very high (1400 SEK/MWh) compared to today's electricity price that is around 250 SEK/MWh[2]. However, constant heat demand for district heating networks, possible decrease in the price of biomass fuels, and possible policy instruments such as green certificates still contribute to economic feasibility of such plants.

Today, only few plants have the fuel drier as additional component (as in Skellefteå, Sveg or Borås) because it is a costly process. However, integration of a flue gas condenser (to extract extra heat from the flue gas) that works independently from the fuel drier could potentially increase the profitability. Using this solution, strategic choices can be made depending on the electricity price level: when the electricity price is low enough, the plant can work with the flue gas condenser switched on to gain extra profit, and when the electricity price is too high, the flue gas condenser can be switched off and the plant can work with just the fuel drier.

[2]Statista. Monthly wholesale electricity prices in Sweden 2019-2023.
https://www.statista.com/statistics/1271491/sweden-monthly-wholesale-electricity-price/