KTH Chemical Engineering
and Technology

Biomass Fired Steam Cycle with Heat Recovery

Question 1F
Gemma Bishop
Alex Mosquera

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1. SUMMARY

This report will analyse the efficiency and profitability of a Biomass-fired steam cycle in a paper mill. The objective of the project is to produce process steam to meet the heat demand of 50 MW. The focus is to achieve an efficient solution and to produce the steam as cost-efficient as possible. This case includes a steam cycle and flue gas condensation to allow for waste heat recovery. To do so, it will begin by calculating the relevant stream properties of the steam, air, biofuel, flue gas, and condensate streams.

In order to calculate the steam stream properties, a mass and energy balance is done around the mill condenser, which generates 50 MW of heat. From here, the mass balance around the condenser is yielded. By doing a combined mass and energy balance around the preheater 1 and preheater 2, the remaining stream properties are yielded. As such, the heat consumed by steam from the boiler is able to be calculated, equalling 65.27 MW.

In order to calculate the relative properties of the flue gas cycle, it is first required to find the mass flow rate of fuel and air going into boiler. This is done by doing an energy balance around the boiler so that the heat produced from the boiler is equal to the heat gained from the steam cycles. As such a mass flow rate of moisture and ash free fuel of 4.42 kg/s is yielded. From this, the relative properties for the flue gas, condensate, air and biofuel streams are calculated through energy and mass balances around each of the relevant mechanisms. With inclusion of the condenser, 15.76 MW of heat is able to be recovered for use in the district heating network. Furthermore, greater use of the flue gas water vapour energy is utilised to heat the combustion air streams through the humidifier.

The total efficiency of the cycle with inclusion of heat recovery is very high at 1.14 in comparison to 0.91 without heat recovery. This allows for generation of greater annual profit from the process as the additional heat generated can then be sold to the market. This is demonstrated through the six scenarios of annual profit for differing power prices and green certificates where all scenarios are expected to generate a profit.

2. INTRODUCTION

In the face of a growing population and rising energy demands, current industrial energy processes have pressure on them to become even more efficient. Combined cycles are a great solution for the integration of by-products to generate and produce new value chains. This assignment will look at the application of a combined steam cycle with flue gas condensation and measure whether the additional heat recovered is both profitable and efficient.

In Sweden, district heating networks constitute approximately 50% of the sources of energy for residential heating. As such, their ability to recovery waste heat from steam cycles makes them a very appealing solution and increase the overall efficiency of the process. Furthermore, both the steam and flue gas cycle utilise steam condensate as the source of energy. As such, the combined heat and flue gas condensation is a completely environmentally friendly process with low cost, and low fuel consumption.

3. ENERGY AND MASS BALANCE CALCULATIONS

In Figure 3.1, we can observe the flow sheet for the complete process of steam productions using a biomass-fired steam cycle that includes waste heat recovery and humidification of inlet air.

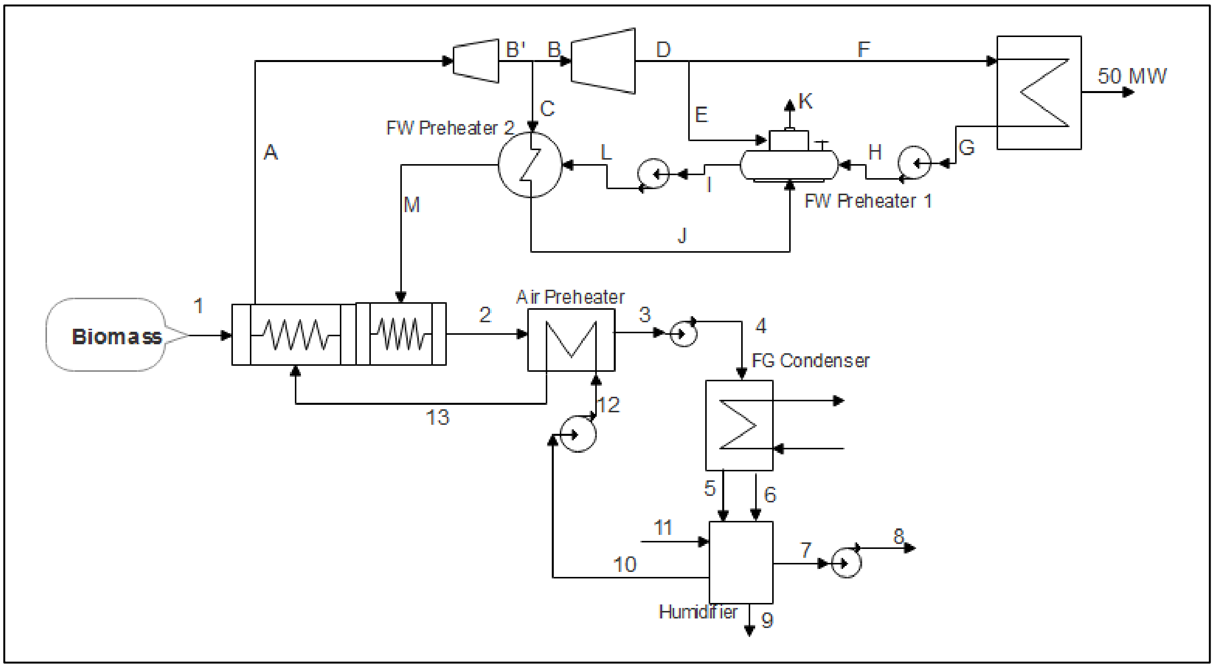

FIGURE 3.1 FLOW DIAGRAM OF BIOMASS-FIRED STEAM CYCLE WITH HEAT RECOVERY SYSTEM

3A. STEAM CYCLE

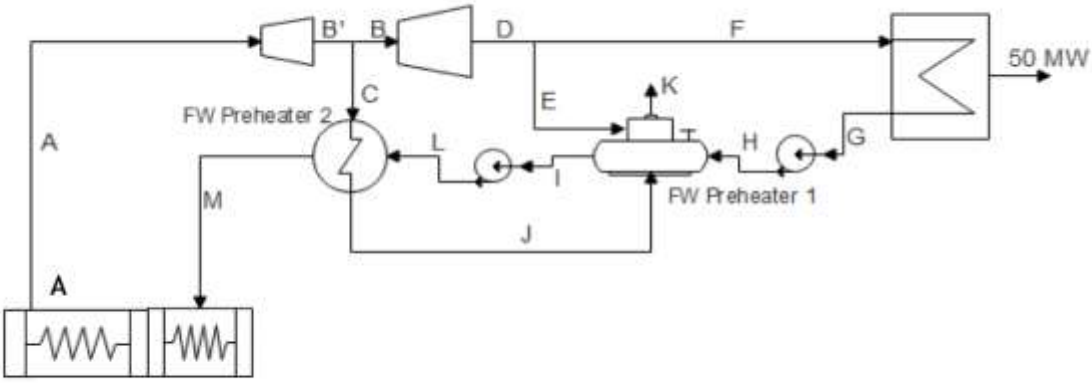

FIGURE 3.2 FLOW DIAGRAM OF THE STEAM CYCLE

In this project, we need to produce steam at 500 °C and 80 bar in the boiler. This information was used to establish the pressure of streams A and M as 80 bar and the temperature of stream A as 500 °C. The temperature and pressure from stream A can be used to obtain its other properties from "Tabeller och Diagram för energitekniska beräkningar, August 2015, Lars Wester", which will denoted as T&D in future references.

The temperature of stream M, which is the heated feed water that comes out of the 2nd preheater, is 10 °C below the condensing temperature of the steam in stream C (to allow for 10 degree driving temperature difference in the heat exchanger). Stream C is 12.0 bar superheated steam extracted from the 1st turbine. The condensing temperature of stream C, obtained from T&D, is 187.96 °C, so the temperature of stream M is 177.96 °C. Stream J is the condensate from the pre-heater. It is assumed to be saturated liquid at the boiling temperature of 187.96 °C, and its pressure is the same as stream C. It is led to the feed water tank that is part of the 1st preheater. Streams B' and B leave the same turbine as stream C and they have the same thermodynamic properties.

The enthalpy and temperature of B', B, and C was calculated using the enthalpy of stream A and the isentropic efficiency of the turbines which is 0.85. The calculations are described in section 3A (II) of this document. The same procedure was used to calculate the enthalpy and temperature of streams E, D, and F. Stream E is steam at 3.5 bar that is injected in the first preheating stage. Streams E, D, and F have the same properties since they are steam expanded in the same turbine.

The return temperature for the condensate (from the mill), which corresponds to stream G, is 80 °C and the pressure is 1.0 bar. This preheating is combined with de-aeration of the feed water. After the first preheating stage the feed water is saturated at 3.5 bar. Therefore, the pressure of stream I is 3.5 bar. A minor amount of steam, stream K, is consumed in the desorption of air but this amount can be neglected.

The pressure of stream H was estimated to be 3.5 bar, which is the operating pressure of the first preheater. The pressure of stream L was set as 80 bar, which is the pressure needed for the steam that is produced in the boiler as stream A. The temperature for these streams was calculated taking into account the work provided by the pumps located before these streams and using the increase in enthalpy to calculate the temperature after the pumps.

With this information, the temperatures and pressure for every stream are known and the enthalpy or other properties can be obtained from T&D. The mass flow rates for every stream were calculated by doing energy balances around both preheaters. Mass balances after both turbines were used to reduce the amount of unknown variables in our system of equations.

3A. (I) RESULTS

The mass flow rate, pressure, temperature and enthalpy of every stream on the side of the steam cycle are summarized in Table 3.1

STREAM PROPERTIES

TABLE 3.1 DATA FOR EACH STREAM OF THE STEAM CYCLE

Stream	Physical state	Mass flow rate (kg/s)	Pressure (bar)	Temp (°C)	Enthalpy (kJ/kg)
A	Steam	24.68	80	500	3399.50
B'	Superheated steam	24.68	12	260.68	2959.72
B	Superheated steam	22.82	12	260.68	2959.72
C	Superheated steam	1.86	12	260.68	2959.72
D	Steam	22.82	3.5	150.08	2757.22
E	Steam	2.18	3.5	150.08	2757.22
F	Steam	20.64	3.5	150.08	2757.22
G	Liquid	20.64	1	80	335.10
H	Liquid	20.64	3.5	80.05	335.44
I	Saturated liquid	24.68	3.5	138.85	584.2
J	Saturated liquid	1.86	12	187.96	798.3
K	Neglected				
L	Liquid	24.68	80	140.26	595.21
M	Liquid	24.68	80	177.96	757.73

TABLE 3.2 POWER AND HEAT PRODUCTION AND CONSUMPTION OF TURBINE, FW PUMPS, AND CONDENSER

		Type of Energy	kW
Turbines	A	Power Production (incl. Mech and Gen. Losses	10104.37
	B	Power Production (incl. Mech and Gen. losses)	4302.86
	Total	Power Production (incl. Mech and Gen. losses)	14407.23
Feed Water Pump 1		Power consumption	7.08
Feed Water Pump 2		Power consumption	271.61
Condenser		Heat production	50000
1st Feedwater preheater		Preheating	14417.47
2nd Feedwater preheater		Preheating	4010.83

3A (II) CALCULATIONS

STEP 1. PROPERTIES AROUND THE TURBINE

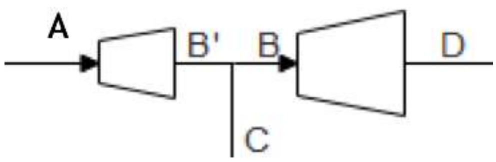

FIGURE 3.3 TURBINES

Procedure to find enthalpy after the 1st Turbine

The temperature and pressure from stream A are known and were used to obtain its other properties from T&D.

$$h_A(500°C, 80\ bar) = 3399.5\ kJ/kg$$
$$S_A(500°C, 80\ bar) = 6.7266\ \ kJ/kg\ K$$

The entropy of stream A is the same for streams B' and D in the reversible case. The isentropic efficiency of the turbines is 0.85.

$$S_A(500°C, 80\ bar) = S_{B's} = S_{Ds}$$

With the pressure and entropy of B’, we found its reversible enthalpy by interpolating between a higher and lower value found in T&D.

$$h_{B's} = \begin{Bmatrix} p_{B'} = 12\ bar \\ S_{B's} = 6.7266\ kJ/kg\ K \end{Bmatrix} = Steam\ Table, page\ 58 = Superheatead\ steam$$

$$h_{B's} = h_0 + (S_{B's} - S_0)\frac{(h_1 - h_0)}{(S_1 - S_0)}$$

$$h_{B's} = 2865.7 + (6.7266 - 6.6937)\frac{(2889.5 - 2865.7)}{(6.7414 - 6.6937)}$$

$$h_{B's} = 2882.12\ kJ/kg$$

We used the relation between reversible turbine work and irreversible turbine work to find the real enthalpy of B’.

$$\eta_s = \frac{h_A - h_{B'}}{h_A - h_{B's}} = 0.85$$

$$h_{B'} = h_A - \eta_s(h_A - h_{B's})$$

$$h_{B'} = 3399.5\ kJ/kg - 0.85 * (3399.5 - 2882.12)\ kJ/kg$$

$$h_{B'} = 2959.72\ kJ/kg$$

We used the real enthalpy to find the temperature of the stream using interpolation.

$$T_{B'} = T_0 + (h_{B'} - h_0)\frac{(T_1 - T_0)}{(h_1 - h_0)}$$

$$T_{B'} = 260 + (2959.72 - 2958.2)\frac{(270 - 260)}{(2980.5 - 2958.2)}$$

$$T_{B'} = 260.68\ °C$$

<u>Procedure to find enthalpy after the 2nd Turbine</u>

The entropy of stream A is the for D in the reversible case

$$S_{Ds} = S_A(500°C, 80\ bar) = 6.7266\ kJ/kg\ K$$

$$h_{Ds} = \begin{Bmatrix} p_D = 3.5\ bar \\ S_{Ds} = 6.7266\ \ kJ/kg\ K \end{Bmatrix} = Steam\ Table, page\ 47, 48$$

In T&D, there is not a steam table with values at 3.5 bar, so it is necessary to interpolate between the tables of 3.4 bar and 3.6 bar. We can find the interpolated values in Table 3.3.

From Table 3.3 we can observe that stream D in the isentropic case is in the 2-phase region, so steam quality needs to be calculated.

$$S_{Ds} = (1 - x_{Ds})\ S_{sat\ liq} + x_{Ds} \cdot S_{sat\ vap}$$

$$x_{Ds} = \frac{S_{Ds} - S_{sat\ liq}}{S_{sat\ vap} - S_{sat\ liq}} = \frac{6.7266 - 1.7273}{6.9403 - 1.7273}$$

$$x_{Ds} = 0.96$$

The steam quality was used to calculate the enthalpy in the 2-phase region.

$$h_{Ds} = (1 - x_{Ds})\ h_{sat\ liq} + x_{Ds} \cdot h_{sat\ vap}$$

$$h_{Ds} = (1 - 0.96)\ 584.2 + 0.96 \cdot 2731.9$$

$$h_{Ds} = 2643.88\ kJ/kg$$

TABLE 3.3 STEAM TABLE AT 3.5 BAR

3,5 bar		
Sat liq	584,2	1,7273
Sat vap	2731,9	6,9403
Temp (°C)	Enthalpy (kJ/kg)	Entropy (kJ/kg K)
Liquid water		
130	546,4	1,6346
Steam		
140	2734,6	6,9467
150	2757,05	7,00045
160	2778,95	7,05155

With the relation between reversible turbine work and irreversible turbine work, the real enthalpy of D was calculated.

$$\eta_s = \frac{h_A - h_D}{h_A - h_{Ds}} = 0.85$$

$$h_D = h_A - \eta_s (h_A - h_{Ds})$$

$$h_D = 3399.5\ kJ/kg - 0.85 * (3399.5 - 2643.88)\ kJ/kg$$

$$h_D = 2757.22\ kJ/kg$$

We used the real enthalpy to find the temperature of the stream.

$$T_D = T_0 + (h_D - h_0)\frac{(T_1 - T_0)}{(h_1 - h_0)}$$

$$T_D = 150 + (2757.22 - 2757{,}05)\frac{(160 - 150)}{(2778{,}95 - 2757{,}05)}$$

$$T_D = 150.08\ °C$$

Mass balance to check for equivalent streams

In order to make the mass balance easier to solve, we first checked for streams that have the same mass flow rate because they only experience a change in temperature or pressure, or they experience evaporation or condensation without changing the total amount of liquid or steam in the stream.

$$m_A = m_M = m_L = m_{B'} = m_I$$
$$m_F = m_G = m_H$$
$$m_C = m_J \quad ; \quad m_B = m_D$$

STEP 2. ENERGY BALANCE AROUND CONDENSER

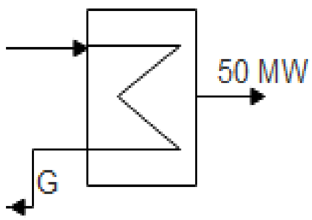

FIGURE 3.4 STEAM CONDENSER

First, we started with an energy balance over the condenser where we know that the power output is 50 MW. The enthalpies of and streams G and F have already been determined and their mass flow rate is the same.

$$50\ MW = m_F h_F - m_G h_G$$
$$m_G = \frac{50\ MW}{h_F - h_G}$$
$$m_G = \frac{50\ MW * 1000\ \ kW/1\ MW}{2757.22\ kJ/kg - 335.10\ kJ/kg}$$
$$m_G = m_F = 20.64\ kg/s$$

STEP 3. ENERGY BALANCES AROUND REMAINING COMPONENTS

Energy balance around 1st pre-heater:

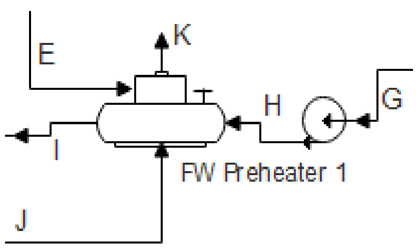

FIGURE 3.5 FEEDWATER PREHEATER 1.

$$m_E h_E + m_H h_H + m_J h_J = m_I h_I$$

$$m_E h_E + m_G h_H + m_C h_J = m_A h_I \tag{1}$$

__Energy Balance around 2nd preheater__

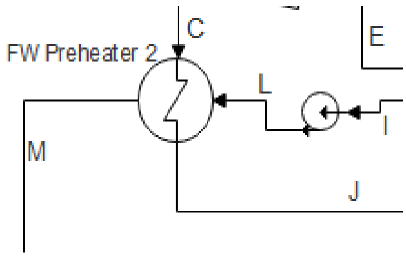

FIGURE 3.6 FEEDWATER PREHEATER 2

$$m_C h_C + m_L h_L = m_J h_J + m_M h_M$$

$$m_C h_C + m_A h_L = m_C h_J + m_A h_M \tag{2}$$

__Mass Balance after 1st Turbine__

$$m_C + m_B = m_{B'} = m_A$$

$$m_C = m_A - m_B \tag{3}$$

__Mass Balance after 2nd Turbine__

$$m_E + m_F = m_D = m_B$$

$$m_E = m_B - m_F \tag{4}$$

In order to reduce the number of unknown mass flow rates in our system of equations, we replaced equation (3) in equation (2) and solved for mass flow rate of A

$$(m_A - m_B)h_C + m_A h_L = (m_A - m_B)h_J + m_A h_M$$

$$m_A h_C - m_B h_C + m_A h_L = m_A h_J - m_B h_J + m_A h_M$$

$$m_A h_C + m_A h_L - m_A h_M - m_A h_J = m_B h_C - m_B h_J$$

$$m_A = \frac{m_B h_C - m_B h_J}{h_C + h_L - h_M - h_J} \tag{5}$$

Next, we replaced equations (3), (4) and (5) in equation (1) and solved for the mass flow rate of B

$$(m_B - m_F)h_E + m_G h_H + (m_A - m_B)h_J = m_A h_I$$

$$m_B h_E - m_F h_E + m_G h_H + m_A h_J - m_B h_J = m_A h_I$$

$$m_B h_E - m_F h_E + m_G h_H - m_B h_J = m_A(h_I - h_J)$$

$$m_B(h_E - h_J) - m_F h_E + m_G h_H = \frac{(m_B h_C - m_B h_J)(h_I - h_J)}{h_C + h_L - h_M - h_J}$$

$$m_B(h_E - h_J) - \frac{(m_B h_C - m_B h_J)(h_I - h_J)}{h_C + h_L - h_M - h_J} = m_F h_E - m_G h_H$$

$$m_B = \frac{m_F h_E - m_G h_H}{(h_E - h_J) - \frac{(h_C - h_J)(h_I - h_J)}{h_C + h_L - h_M - h_J}} \quad (6)$$

Equation (6) allowed us to calculate the mass flow rate of stream B using enthalpies of other streams and the mass flow rates of streams F and G that we calculated previously. Using the mass flow rate of stream B we calculated the mass flow rate of stream A from equation (5).

$$m_B = 22.81\ kg/s$$

$$m_A = 24.68\ kg/s$$

The mass flow rates of A, B, G, and F were used to calculate the mass flow rates of the other streams.

<u>Power produced by turbine:</u>

The mass flow rates of A and B were used to calculate the power generated by the turbines

$$W_T = m_A(h_A - h_B) + m_D(h_B - h_D)$$

$$W_T = 24.68\frac{kg}{s}(3399.5 - 2959.72)\frac{kJ}{kg} + 22.82\frac{kg}{s}(2959.72 - 2757.22)\frac{kJ}{kg}$$

$$W_T = 15475.01\ \mathrm{kW}$$

$$P_{el} = W_T * \eta_{mech} * \eta_{gen}$$

$$P_{el} = 15504.05\ \mathrm{kW} * 0.98 * 0.95$$

$$P_{el} = 14407.23\ \mathrm{kW}$$

<u>Power consumption of pumps:</u>

The work consumed by the pumps was obtained from equation (7)

$$W_{pump} = \frac{m\ \nu\ \Delta p}{\eta} \quad (7)$$

The specific volume of stream G can be obtained from T&D. This value, the pressure drop, and mass flow rate of G were used to calculate the work of the pump located before the first preheater.

$$W_{pump} = \frac{20.64\ \frac{kg}{s} * 0.001029 \frac{m^3}{kg} * (3.5 - 1) bar * \frac{100000\ Pa}{1\ bar} * \frac{N}{1\ Pa \cdot m^2}}{0.75}$$

$$W_{pump} = \frac{20.64\ * 0.001029 * (3.5 - 1) \frac{m}{s} * 100000\ N * \frac{1\ J}{1\ N \cdot m} * \frac{1\ kJ}{1000\ J}}{0.75}$$

$$W_{pump} = 7{,}08\ kW$$

A variation of equation (7) was used to establish the change in enthalpy

$$h_H = h_G + \frac{\nu\ \Delta p}{\eta}$$

$$h_H = 335.10 \frac{kJ}{kg} + \frac{0.001029 \frac{m^3}{kg}\ (3.5 - 1) bar * \frac{100000}{1000}}{0.75}$$

$$h_H = 335.44\ kJ/kg$$

With this value of enthalpy and the pressure that we already know, the temperature was obtained though interpolation.

$$T_H = T_0 + (h_H - h_0) \frac{(T_1 - T_0)}{(h_1 - h_0)}$$

$$T_H = 80 + (335.44 - 335.25) \frac{(90 - 80)}{(377.25 - 335.25)}$$

$$T_D = 80.05\ °C$$

The same procedure was used to determine the properties of stream L. The specific volume of stream I can be obtained from T&D. This value, the pressure drop, and mass flow rate of I were used to calculate the work of the pump located before the second preheater.

$$W_{pump2} = \frac{24.68\ \frac{kg}{s} * 0.001079 \frac{m^3}{kg} * (80 - 3.5) bar * \frac{100000\ Pa}{1\ bar} * \frac{N}{1\ Pa \cdot m^2}}{0.75}$$

$$W_{pump2} = \frac{24.68\ * 0.001079 * (80 - 3.5) \frac{m}{s} * 100000\ N * \frac{1\ J}{1\ N \cdot m} * \frac{1\ kJ}{1000\ J}}{0.75}$$

$$W_{pump2} = 271{,}61\ kW$$

$$h_L = h_I + \frac{\nu\ \Delta p}{\eta}$$

$$h_L = 584.2\frac{kJ}{kg} + \frac{0.001079\frac{m^3}{kg}\ (80 - 3.5)bar * \frac{100000}{1000}}{0.75}$$

$$h_L = 595.21\ kJ/kg$$

$$T_H = T_0 + (h_H - h_0)\frac{(T_1 - T_0)}{(h_1 - h_0)}$$

$$T_H = 140 + (595.21 - 594.1)\frac{(150 - 140)}{(636.9 - 594.1)}$$

$$T_D = 140.26\ °C$$

3B. COMBUSTION AND HEAT RECOVERY

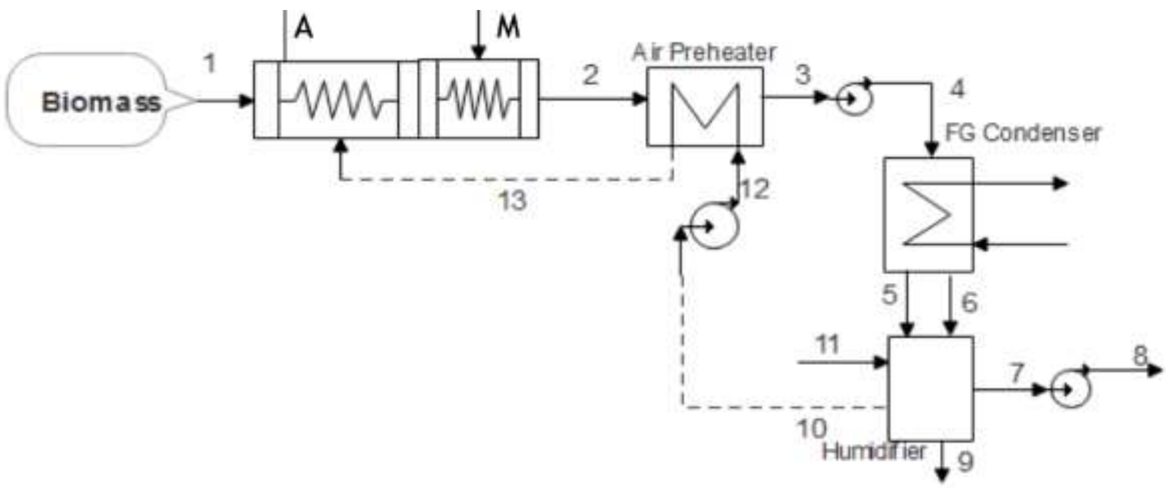

FIGURE 3. 7 FLOW DIAGRAM OF COMBUSTION AND HEAT RECOVERY FROM FLUE GAS

The combustion side of the plant is using wet bio-fuels as fuel and air to produce flue gas. The boiler is able to generate heat that is used to heat up the alternate steam streams. To improve the efficiency of the process, the flue gas is fed through a condenser. The flue gas downstream from the boiler is cooled by the flows of district heating and as such, heat used to evaporate water content of the flue gas can be recovered. As such, the use of a condenser allows for an increased production of heat of 15.85 MW in addition to the 50 MW produced in the steam flow condenser mill.

To improve the efficiency of the cycle further, the ambient air is humidified through the humidifier. Here, the combustion air is able to be heated by exchange with warmer flue gas and condensate downstream from the flue gas condenser.

3B. (I) RESULTS

STREAM PROPERTIES

BIOFUEL (STREAM 1)

TABLE 3.4 TEMPERATURE, PRESSURE, MASS FLOW RATE, HHV, AND ENERGY OF STREAM 1

Temperature (° C)	25
Pressure (bar)	1.013 (assumed)
Mass Flow Rate of MAF (kg/s)	4.437
HHV (MJ/kg MAF)	19.6
Energy (kW) (Appendix A)	2328.95

TABLE 3.5 COMPOSITION OF STREAM 1

	In (mol/s)	*In (kg/s)*
C (fuel)	*199.67*	*2.40*
H2 (Fuel)	*137.55*	*0.28*
O2 (fuel)	*55.19*	*1.77*
H20 (fuel)	*246.50*	*4.44*
Ash		*0.04*
Total Mass Flow Rate		*8.92*

AIR STREAMS

The combustion air (Stream 11) is taken from the boiler room and has a temperature of 30 C into the humidifier. The humidity of the ambient air can be neglected, therefore the mass flow rate of H_2O entering the humidifier is equal to 0. Furthermore, a pressure of 1 atm (1.013 bar) is assumed.

The inlet air humidification is producing saturated air at a dew point temperature of 40 C (stream 10). The internal energy consumption is assumed to have a 2000 Pa pressure drop around the humidifier due to internal energy consumption. As such, the exiting pressure of the air is at 0.993 bar (1.013-0.02 bar). Furthermore, a relative humidity of 100% is assumed as the air will leave completely humidified.

The air fan is then placed to increase the pressure of the air back up to 1.013 bar in stream 12. The efficiency of the combustion air fan is assumed to be around 70%. Assuming the air to be incompressible at moderate pressure, the new pump work equation can be used yielding the temperature of stream 12 to be around 42.56 C.

The combustion air is then passed through the air pre-heater, heating the air stream to 147.28 C, whilst remaining at 1.013 bar (Stream 13).

TABLE 3.6 PROPERTIES OF AIR STREAMS. THE ENERGY CONTENTS OF AIR STREAMS ARE TAKEN FROM THE PROVIDED EXCEL SPREAD SHEET.

	Temperature (°C)	Pressure (bar)	N2 (kg/s)	O2 (kg/s)	H2O (kg/s)	$\dot{n}_{total}$ (mol/s)	$\dot{m}_{total}$ (kg/s)	Energy (KW)
10	40	0.993	30.95	9.40	2.02	1511.30	42.37	6828.0
11	30	1.013	30.95	9.40	0	1398.98	40.35	1214.9
12	42.55	1.013	30.95	9.40	2.02	1511.30	42.37	6941.2
13	147.28	1.013	30.95	9.40	2.02	1511.30	42.37	11609.0

FLUE GAS STREAMS

Around the boiler, the flue gas is expected to have a 3000 Pa pressure drop between stream 13 and 3. As such, pressure of stream 3 is 0.983 bar. Accordingly, pressure of stream 2 will be equal to stream 3 at 0.983 bar.

In addition, the flue gas temperature leaving the boiler is going to be 20 °C higher than the water temperature at the cold end of the economizer. As such, the temperature of Stream 2 will be at 197.96 °C.

After leaving the boiler, ash in the fuel is assumed to be filtered out before entering the air preheater. Furthermore, the temperature of the flue gas stream 3 will be reduced to 120 °C.

After the preheater, the flue gas passes through the first flue gas pump. A pressure increase of 3000 Pa is assumed, giving a pressure of 1.013 for Stream 4. By solving for the Win required by the pump, a temperature of 124.39 °C is yielded for Stream 4. The reversible pump equation can be used for fans given that the pressure increase is moderate so that air may be considered incompressible.

The flue gas is then passed through the condenser to recover heat from the vapour water. An estimated 15.76 MW of heat is produced. The temperature of the flue gas leaving the condenser is assumed to be at 53 °C (Stream 6). Furthermore, a pressure drop of 2000 Pa in the condenser is assumed, giving a pressure of 0.993 bar for stream 6.

Stream 6 is then passed through the humidifier to allow for extra use of the vapour energy in flue gas to heat and humidify the combustion air. A temperature of 42 °C is yielded for the exiting flue gas stream (Stream 7). Furthermore, a pressure drop of 2000 Pa around the humidifier is assumed, giving a pressure of 0.973 bar for stream 7.

Finally, the flue gas will pass through the second flue gas fan in order to bring the pressure back up to atmospheric, giving a temperature of 44.23 °C in stream 8.

TABLE 3.7 PROPERTIES OF FLUE GAS STREAMS. THE ENERGY CONTENTS OF GAS STREAMS ARE TAKEN FROM THE PROVIDED EXCEL SPREAD SHEET.

	Temperature (°C)	Pressure (bar)	O2 (kg/s)	H2O (kg/s)	N2 (kg/s)	CO2 (kg/s)	$\dot{n}_{total}$ (mol/s)	$\dot{m}_{total}$ (kg/s)	Energy (kW)
2	197.96	1.013	2.6	8.9	30.9	8.4	1881.75	50.89	34045.7
3	120	0.983	2.6	8.9	30.9	8.4	1881.75	50.89	29362.3
4	124.39	1.013	2.6	8.9	30.9	8.4	1881.75	50.89	29622.5
6	53	0.993	2.6	4.2	30.9	8.4	1618.65	46.12	13109.2
7	42.7	0.973	2.6	2.4	30.9	8.4	1581.76	44.32	7987.8
8	44.78	1.013	2.6	2.4	30.9	8.4	1581.76	44.32	9781.4

CONDENSATE STREAMS

Using a mass and energy balance around the condenser, 4.86 kg/s of condensate (H_2O) will be produced in Stream 5. Given there is a pressure drop of 2000 Pa in the condenser, it will be also assumed a pressure of 0.993 bar.

From here, the condensate is used to humidify the combustion air stream and further condense the flue gas stream. As such, the condensate will decrease to 4.68 kg/s in Stream 9. The condensate is assumed to leave the humidifier at 30 °C and a pressure of 0.973 due to a 2000 pa pressure drop in the humidifier.

TABLE 3.8 PROPERTIES OF CONDENSATE STREAMS

	Temperature (°C)	Pressure (bar)	$\dot{m}_{total}$ (kg/s)	H (sat. liq, T (C°))	Energy (kW)
5	53	0.993	4.74	221.9	1050.90
9	30	0.973	4.50	125.7	565.80

POWER AND HEAT PRODUCTION AND CONSUMPTION

TABLE 3.9 POWER AND HEAT PRODUCTION AND CONSUMPTION VALUES

	Type of Energy	kW
Flue Gas Fan 1	Power consumption	268.16
Flue Gas Fan 2	Power consumption	243.94
Air Fan	Power consumption	113.21
Condenser	Heat production	15462.43

3B. (II) CALCULATIONS

This part of the report will illustrate the steps taken to define all stream properties of biofuel, air, flue gas, and condensate whilst calculating the total heat production from the condenser, power consumption of the FG and air fans, and the relative heat loads of the humidifier and air pre-heater.

Part 1. Of these calculations will find the required mass flow rate of MAF.

In order to calculate the mass flow rate of the wet biofuel, the air required for combustion, and subsequently, the amount of flue gas produced, an energy balance must be taken around the boiler so that the heat delivered from the boiler to the water/steam stream is equal to the heat uptake of the water/steam stream.

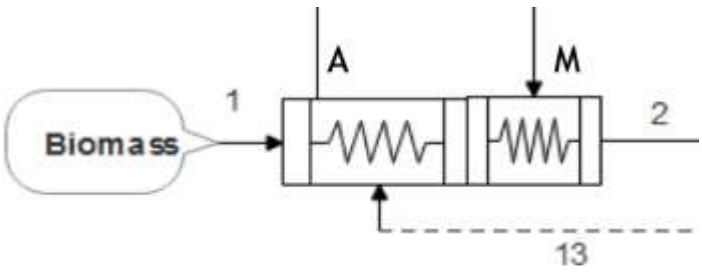

FIGURE 3.8 BOILER

The *heat intake* of the steam stream has already been calculated in Part 1. Where the Q of the boiler is equal to 65.27 MW.

The *heat produced* by the boiler can be calculated using the below equation, where production is equal to the inlet energy plus the higher heating value (HHV), minus the outlet energy and any convection losses.

$$Q_{boiler} = In + HHV - Out - Convection\ Losses$$

As such, in order to do this, properties around Stream 1, 13 and 2 must be yielded to find their relative energies. Thus, in Part 1, we will initially find the energy produced per kg MAF or dry biomass. From here, a conversion factor can be used to find the correct amount of MAF needed to match the Q_{boiler} gained by the steam streams to the Q_{boiler} produced from the combustion streams.

<u>Part 2. Will calculate the real energy and mass properties of the streams.</u>

Accordingly, Part 2. Will then adjust the calculations with the new MAF flow rate. From here, the real heat production from the condenser, power consumption of the FG and air fans, and the relative heat loads of the humidifier and air pre-heater are calculated.

PART 1.

The *heat intake* of the steam stream has already been calculated in Part 1. Where the Q of the boiler is equal to 65.27 MW.

$$\dot{Q}_{gain\ boiler} = Energy_A - Energy_M$$

$$\dot{Q}_{boiler} = \dot{m}(h_A - h_M) = 65.27\ MW \ldots (1)$$

The *heat produced* by the boiler can be calculated using Equation 2, where production is equal to the inlet energy plus the higher heating value (HHV), minus the outlet energy and any convection losses.

$$Prod = In - Out + HHV - Convection\ Losses$$

$$\dot{Q}_{boiler} = (Energy_1 + Energy_{13} + HHV) - Energy_2 - Convection\ Losses$$

$$Energy =$$

$$\dot{Q}_{\ boiler} = \dot{m}_{MAF}\big(h_{1,MAF} + h_{1,H20} + h_{13} + HHV + \ h_{1,ash} - h_2 - h_{2,ash} - Convection\ Losses\ \big) \ \ldots.(2)$$

The enthalpies, h, are the enthalpies of each stream per kg of MAF [kJ/kg MAF] and m_{MAF} is expressed in kg of MAF per second [kg MAF/s].

By solving equation (1) with equation (2), a mass flow rate of MAF can be found.

$$65.20\ MW = \dot{m}_{MAF}\big(h_{1,MAF} + h_{1,H20} + h_{13} + HHV + h_{1,ash} - h_2 - h_{2,ash} - Convection\ Losses\big)$$

STEP 1. BIOFUEL COMPOSITION

TABLE 3.10 BIOFUEL TEMPERATURE, PRESSURE, REAL MASS FLOW RATE, AND HHV

Temperature (° C)	25
Pressure (bar)	1.013 (assumed)
Real Mass Flow Rate (kg/s)	$\dot{m}_1 = \text{total mass flow rate} * \dot{m}_{MAF}$
HHV (MJ/kg MAF)	19.6

TABLE 3.11 COMPOSITION OF BIOFUEL PER 1KG MAF. VALUES WERE YIELDED FROM T&D AT 25 °C AND 1 BAR.

	[g/kg MAF]	Enthalpy (h) [kJ/kg MAF]
C	540	$h_{1,MAF} = Cp_{DB}T_1 = 1.4\frac{kJ}{kg.K} * (25 - 0) = 35\frac{kJ}{kgMAF}$ where DB = dry biomass
H	62	
O	398	
H20	1000	$h_{1,H20\,(25\,C)} = 104.8$ from T&D at 25 C and 1.0 bar.
Ash	10	$h_{1,ash} = Cp_A T_{1,} = 0.9\frac{kJ}{kg.K} * (250 - 0) = 0.0225\frac{kJ}{kgMAF}$
Total	**2010**	

STEP 2. AIR STREAM COMPOSITIONS

This step will analyse the air stream compositions per kg of moisture and ash free fuel.

Oxygen demand for combustion:

To find the composition of both the flue gas and air streams, it is required to find the total amount of oxygen demanded by the combustion unit in comparison to the total amount of oxygen supplied by both air and the wet biofuel.

Combustion Reaction Formula:

$$2H + O \rightarrow H_2O$$

$$C + 2O \rightarrow CO_2$$

For each mole of C, one mole of O_2 is required

For each mole of H, ½ mole of of O_2 is required

Note that $\dot{n} = \frac{\dot{m}}{M}$

Amount of oxygen required:

$$\dot{n_{O_{2,req}}} = \dot{n}_{1,C} + \frac{\dot{n}_{1,H}}{2} = \frac{540\frac{g}{s}}{12\frac{g}{mol}} + \frac{\frac{62\frac{g}{s}}{2}}{2\frac{g}{mol}} = 60.50\ \left[mol\frac{O_2}{s}\right]$$

Total amount of oxygen supplied:

Considering the stoichiometric demand of 130% for combustion:

$$\dot{n_{O_{2,total}}} = ST_{O_2} * \dot{n}_{O_{2,req}} = 1.3 * 60.5 = 78.65\left[mol\frac{O_2}{s}\right]$$

Amount of oxygen in air streams:

As the fuel is consisting partially of oxygen, not all oxygen is required form the airstream.

$$\dot{n}_{13,O_{2,}} = \dot{n_{O_{2,total}}} - \dot{n}_{1,O_{2,}} = 78.65 - \left(\frac{398}{12}\right) = 66.21\left[mol\frac{O_2}{s}\right]$$

The amount of oxygen in the air stream will remain the same in Stream 10, 11, 12, and 13.

$$\dot{n}_{13,O_2} = \dot{n}_{12,O_2} = \dot{n}_{11,O_2} = \dot{n}_{10,O_2}$$

Amount of O_2 and N_2 in Air Streams:

After calculating the amount of O_2 in the air stream in Step 2., we are able to calculate the amount of N_2 based on the fact that dry air contains 21 mol% O_2 and 79 mol% N_2 .

$$\dot{n}_{N_{2,air}} = \dot{n_{O_{2,air}}} * \frac{79}{21} = 249.09\ \frac{mol}{s}$$

The amount of N_2 and O_2 remains the same in all air streams.

$$\dot{n}_{13,N_2} = \dot{n}_{12,N_2} = \dot{n}_{11,N_2} = \dot{n}_{10,N_2}$$

Amount of H_2O in air leaving the humidifier (Stream 10=12=13):

In order to calculate the amount of H_2O in the air stream entering the boiler for combustion (Stream 13), we must do an analysis around the humidifier to find the amount of water vapour leaving the humidifier.

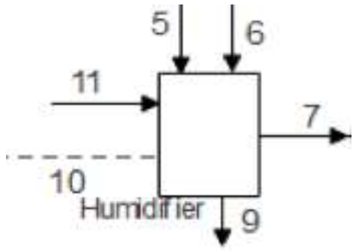

FIGURE 3.9 HUMIDIFIER

P (sat. vap, 40 °C) = 0.0738 [bar]

$$\dot{n}_{10,H_2O} = \frac{p_{H_2O}}{p_{10,tot} - p_{H_2O}} * \left(\dot{n}_{10,tot} - \dot{n}_{10,H_2O,}\right) = \frac{p_{H_2O}}{p_{tot} - p_{H_2O}} * \left(\dot{n}_{10,N_2} + \dot{n}_{O_2}\right)$$

$$\dot{n}_{10,H_2O} = \frac{0.0738}{0.993 - 0.0738} * (249.09 + 66.21) = 25.31 \frac{mol}{s}$$

The amount of H_2O remains the same in air streams 10, 12, and 13.

$$\dot{n}_{10,H_2O} = \dot{n}_{12,H_2O} = \dot{n}_{13,H_2O}$$

STEP 4. FLUE GAS STREAM COMPOSITIONS (UNTIL HEAT RECOVERY)

This section will analyse the flue gas stream compositions per kg of moisture and ash free fuel before reaching the waste heat recovery section: The condenser and humidifier. As such, this will analyse Streams 2, 3 and 4.

Mass balance over boiler:

Now that we know the composition of stream 1 (biofuel) and stream 13 (air), a mass balance around the boiler can be done in order to calculated the compositions of the flue gas leaving (stream 2).

2H + O → H_2O

C + 2O → CO_2

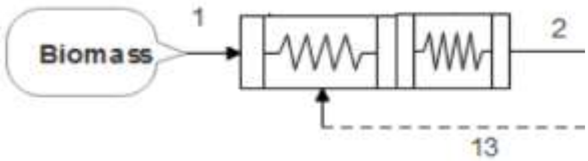

FIGURE 3.10 BOILER

The composition of the flue gas will remain the same, minus the ash assuming that the ash is filtered out after the boiler. As such, streams 2, 3, and 4 will have the same compositions.

TABLE 3.12 MASS BALANCE AROUND THE BOILER

	In (mol/s)	In (kg/s)	Produced (mol/s)	Out (mol/s)	out (kg/s)	mol %
C (fuel)	45.00	0.54	-45.00			
H2 (Fuel)	31.00	0.06	-31.00			
O2 (fuel)	12.44	0.40	-60.50	18.15	0.58	4%
H20 (fuel)	55.56	1.00	31.00	111.87	2.01	26%
H20 (air)	25.31	0.46				
N2 (Air)	249.09	6.97		249.09	6.97	59%
O2 (Air)	66.21	2.12				
CO2	0.00	0.00	45.00	45.00	1.89	11%
Ash	-	0.01		-	0.01	
TOTAL				**424.11**	**11.47**	

STEP 5. TEMPERATURES AND ENERGIES OF AIR STREAMS

Energy balance around the air fan:

To find the temperature of Stream 12 entering the air pre-heater, the Win required by the fan must be calculated.

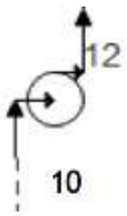

FIGURE 3.11 AIR FAN

The volumetric flow rate of the air can be calculated using the Ideal gas law:

$$\dot{V_{10}} = \frac{\dot{n_{10}}RT}{P_{10}} = \frac{340.61 * 8.313 * (40 + 273.15)}{0.993 * 100000} = 8.93\frac{m^3}{s}$$

From here, the work required by the fan can be calculated, assuming an efficiency of 70% (given).

$$Win\ [kW] = \frac{\dot{V_{10}} * (P_{12} - P_{10})}{\eta}$$

$$Win = 25.52\frac{kW}{kg\ MAF/s} \dots (1)$$

Temperature of Stream 12:

$$Win = Energy_{12} - Energy_{10} \dots (2)$$

Using the solver and subbing the value generated from equation (1) into equation (2), a temperature of 42.55 C was retrieved for Stream 12 (using values from the excel spread sheet).

$$T_{12} = 42.55\ C$$

Energy Balance around the aIr preheater

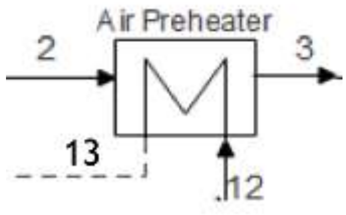

FIGURE 3.12 AIR PREHEATER

Temperatures of the two flue gas streams, Streams 2 and 3 are given at 197.96 C and 120 C, respectively. From this, an energy balance can be done around the pre-heater to analyse the respective heat loads and to find the exiting temperature of the combustion air in stream 13.

$$(Energy_2 - Energy_3) = (Energy_{13} - Energy_{12}) \dots (1)$$

By using values from the excel spreadsheet, $Energy_2$ = 7668.9 kW/ (1 kg MAF/s) and $Energy_3$ = 6617.6 kW/ (1 kg MAF/s) (See Appendix A for sample calculations).

$$(Energy_2 - Energy_3) = 7668.9 - 6617.6 = \; 1052\frac{kW}{kg\ MAF/s} \dots (2)$$

subbing (1) into (2) and $Energy_{12}$ = 1564.4 kW/ (1 kg MAF/s), calculated from the excel spreadsheet (See Appendix A for sample calculations).

$$1052\frac{kW}{kg\ MAF} = (Energy_{13} - Energy_{12}) = (Energy_{13} - 1564.4)$$

$$Therefore, \qquad Energy_{13} = 2616.4\frac{kW}{kg\ MAF/s}$$

Using solver, we are able to find the temperature of stream 13 to be 147.28 C.

$$T_{13} = 147.28\ C$$

STEP 6. SOLVING FOR REAL FLOW RATES

Energy Balance around the Boiler:

Now that we have the stream properties around the boiler, we are able to calculate the amount of fuel and air entering the boiler and as such, the amount of flue gas produced.

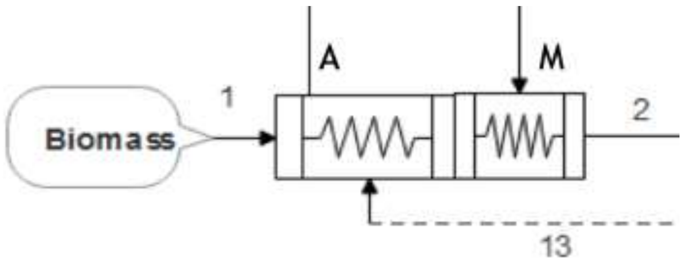

FIGURE 3.13 BOILER

$$\dot{Q}_{boiler} = (Energy_1 + Energy_{13} + HHV) - Energy_2 - Convection\ Losses$$

$$\dot{Q}_{\,boiler} = \dot{m}_{MAF}\big(h_{1,MAF} + h_{1,H20} + h_{1,ash} + h_{13} + HHV - h_2 - h_{2,ash} - Convection\ Losses\,\big)$$

where

$$\dot{Q}_{boiler} = 65.20\ MW$$

$$65.27\ MW = \dot{m}_{MAF}\big(h_{1,MAF} + h_{1,H20} + h_{1,ash} + h_{13} + HHV - h_2 - h_{2,ash} - Convection\ Losses\,\big)$$

Example of calculations to find energy and LHV for the wet fuel are in Appendix A and B.

TABLE 3.13 ENERGY BALANCE AROUND THE BOILER. THE ENERGY CONTENTS OF GAS AND AIR STREAMS ARE TAKEN FROM THE PROVIDED EXCEL SPREAD SHEET.

Source	Energy (In) [kW/ (kg MAF/s)]	Energy (Out) [kW/ (kg MAF/s)]
Stream 1	524.89	
Stream 13	2616.39	
HHV	19600	
Stream 2		7673.13
SUM	**22425.37**	7673.13
Convection Losses (2 % of LHV)	$= 0.02 * 15795.83$	315.92

$$\dot{Q}_{boiler} = (Energy_1 + Energy_{13}) - Energy_2 - Convection\ Losses \ldots (1)$$

$$\dot{Q}_{boiler} = 22425.37 - 7673.18 - 315.92$$

$$\dot{Q}_{boiler} = 14.75 \frac{MW}{kg\ MAF/s}$$

As such,

$$\dot{Q}_{boiler} = 65.20\ MW = 14.75 \frac{MW}{kg\ MAF/s}$$

$$\dot{m}_{MAF} = 4.437 \frac{kg}{s}$$

As such, the new mass flow rates of flue gas (Stream 2), air (Stream 13), and biofuel (Stream 1) are generated using the conversion equations stated in Step 1.

Furthermore, the W_{in} required by the air fan and the heat load of the air preheater were adjusted with the new flow rates.

The new values can be seen in in table 3.4, 3.5, 3,6, 3.7, and 3.9.

PART 2.

The new mass flow rates can be seen in the following tables.

WET BIOFUEL (STREAM 1)

TABLE 3.14 TEMPERATURE, PRESSURE, MASS FLOW RATE, HHV, AND ENERGY OF STREAM 1

Temperature (° C)	25
Pressure (bar)	1.013 (assumed)
Mass Flow Rate of MAF (kg/s)	4.437
HHV (MJ/kg MAF)	19.6
Energy (kW) (Appendix A)	2328.95

TABLE 3.15 COMPOSITION OF STREAM 1

	In (mol/s)	*In (kg/s)*
C (fuel)	*199.67*	*2.40*
H2 (Fuel)	*137.55*	*0.28*
O2 (fuel)	*55.19*	*1.77*
H20 (fuel)	*246.50*	*4.44*
Ash		*0.04*
Total Mass Flow Rate		*8.92*

AIR STREAMS

TABLE 3.16 PROPERTIES OF AIR STREAMS. THE ENERGY CONTENTS OF AIR STREAMS ARE TAKEN FROM THE PROVIDED EXCEL SPREAD SHEET.

	Temperature (°C)	Pressure (bar)	N2 (kg/s)	O2 (kg/s)	H2O (kg/s)	$\dot{n}_{total}$ (mol/s)	$\dot{m}_{total}$ (kg/s)	Energy (KW)
10	40	0.993	30.95	9.40	2.02	1511.30	42.37	6828.0
11	30	1.013	30.95	9.40	0	1398.98	40.35	1214.9
12	42.55	1.013	30.95	9.40	2.02	1511.30	42.37	6941.2
13	147.28	1.013	30.95	9.40	2.02	1511.30	42.37	11609.0

FLUE GAS STREAMS

TABLE 3.17 PROPERTIES OF FLUE GAS STREAMS. THE ENERGY CONTENTS OF GAS STREAMS ARE TAKEN FROM THE PROVIDED EXCEL SPREAD SHEET.

	Temperature (°C)	Pressure (bar)	O2 (kg/s)	H2O (kg/s)	N2 (kg/s)	CO2 (kg/s)	$\dot{n}_{total}$ (mol/s)	$\dot{m}_{total}$ (kg/s)	Energy (kW)
2	197.96	1.013	2.6	8.9	30.9	8.4	1881.75	50.89	34045.7
3	120	0.983	2.6	8.9	30.9	8.4	1881.75	50.89	29362.3
4	124.39	1.013	2.6	8.9	30.9	8.4	1881.75	50.89	29622.5
6	53	0.993	2.6	4.2	30.9	8.4	1618.65	46.12	13109.2
7	42.7	0.973	2.6	2.4	30.9	8.4	1581.76	44.32	7987.8
8	44.78	1.013	2.6	2.4	30.9	8.4	1581.76	44.32	9781.4

STEP 1. FG FAN 1 WORK AND EXITING TEMPERATURE

Energy balance around the FG fan:

To find the temperature of Stream 4 entering the FG condenser, the Win required by the fan must be calculated.

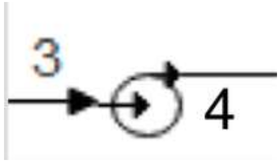

FIGURE 3.14 FG FAN 1.

The volumetric flow rate of the air can be calculated using the Ideal gas law:

$$\dot{V_3} = \frac{\dot{n_3}RT}{P_3} = \frac{1881.75 * 8.313 * (120 + 273.15)}{0.983 * 100000} = 62.57\frac{m^3}{s}$$

From here, the work required by the fan can be calculated, assuming an efficiency of 70% (given). Pressure of Stream 3 is at 0.983 bar and Stream 4 is 1.013 bar (Given in description).

$$Win\ [kW] = \frac{\dot{V_3} * (P_4 - P_3)}{\eta} = 62.57 * \frac{101.3 - 98.3}{0.7}$$

$$Win = 268.16\ kW \ldots (1)$$

Temperature of Stream 4:

$$Win = \ Energy_4 - Energy_{\ 3} \ldots\ (2)$$

Given we know the composition of FG streams 3 and 4, by using the energy values in the excel spreadsheet, the temperature of Stream 4 can be calculated. Using the solver and subbing the value generated from equation (1) into equation (2), a temperature of 124.4 C was given for Stream 4.

$$T_4 = 124.4\ C$$

STEP 2. WASTE HEAT RECOVERY

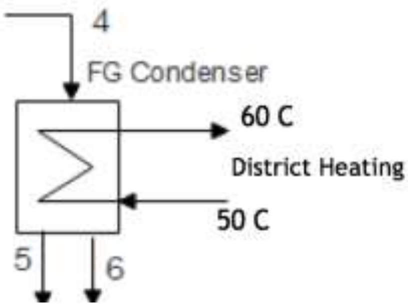

FIGURE 3.15 CONDENSER

In order to calculate the heat production from the flue gas condenser, a water mass and energy balance must be done over the streams.

Water Mass Balance:

$$\dot{m}_{4,H2O} = \dot{m}_{5,H2O} + \dot{m}_{6,H2O}$$

$\dot{m}_4$ has already been calculated:

$$\dot{m}_{4,H2O} = 8.93\frac{kg}{s}$$

To find out how much water is left in the flue gas stream, one first considers the saturation pressure of the flue gas with given temperature of 53 °C.

$$p(sat, 53\,°C) = 0.1431$$

Therefore, the $\dot{m}_{6,H2O}$ can be calculated:

$$\dot{m}_{6,H2O} = \frac{p_{H_2O}}{p_{tot} - p_{H_2O}} * \left(\dot{n}_{tot} - \dot{n}_{H_2O}\right) * M_{H_2O}$$

$$\dot{m}_{6,H2O} = \frac{0.1431}{0.993 - 0.1431} * (1881.75 - 496.37) * \frac{18}{1000} = 4.20\frac{kg}{s}$$

Therefore, $\dot{m}_5$ can be calculated:

$$\dot{m}_5 = \dot{m}_{4,H2O} - \dot{m}_{6,H2O}$$

$$\dot{m}_5 = 8.93 - 4.19 = 4.74\frac{kg}{s}$$

Condenser Output:

Energy of stream 5 can be calculated by using the saturated liquid enthalpy at 53 °C.

$$Energy_5 = h(sat.liq, 53\,°C) * \dot{m}_5$$

$$h(sat.liq, 53\,°C) = 221.9\frac{kJ}{kg}$$

$$Energy_5 = 5.17 * 221.9 = 1146.84\ kW$$

TABLE 3.18 ENERGY BALANCE AROUND THE CONDENSER. THE ENERGY CONTENTS OF GAS STREAMS ARE TAKEN FROM THE PROVIDED EXCEL SPREAD SHEET.

Source	Energy (In) kW	Energy (Out) kW
Stream 4	29622.49	
Stream 5		1050.90
Stream 6		13109.2
SUM	29622.49	14160.06

$$Prod = In - Out$$

$$\dot{Q}_{condenser} = Energy_4 - (Energy_5 + Energy_6)$$

$$\dot{Q}_{condenser} = 15.46\ MW$$

STEP 3. TEMPERATURE OF EXITING FLUE GAS

Mass and Energy Balance around the Humidifier:

In order to find all stream properties around the humidifier, it is required to solve for the temperature of the leaving flue gas stream (Stream 7), and subsequently, the amount of H2O in the condensate (Stream 9). This was done through a reiteration of mass and energy balances around the FG humidifier.

Knowing the mass flow rates and properties for Streams 5, 6, 10 and 11, energy and mass balances can be performed to find the mass flow rates and properties of stream 7 and 9. To do so, initially, a temperature for stream 7 Is assumed. If the assumed temperature of Stream 7 does not give a proper energy balance over the humidifier, a new T_7 is used and repeated until an energy balance that works for the humidifier is yielded.

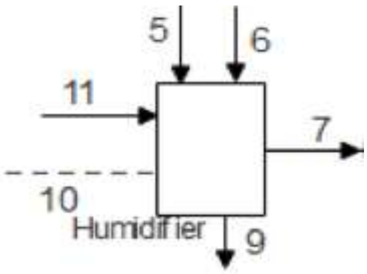

FIGURE 3.16 HUMIDIFIER

Mass Balance of H20:

$$\dot{m}_5 + \dot{m}_6 + \dot{m}_{11} = \dot{m}_{10} + \dot{m}_7 + \dot{m}_9$$

Referencing the mass flow rates calculated in Table 3.14, 3.15, and 3.16…

$$\dot{m}_7 + \dot{m}_9 = 4.74 + 4.2 - 2.02 \ldots (1)$$

To find out how much water is left in the flue gas stream, one first considers the saturation pressure of the flue gas with an unknown temperature of T C.

$$p(sat, T\,°C) = ?$$

Therefore, the $\dot{m}_7$ can be calculated:

$$\dot{m}_7 = \frac{p(sat, T\,°C)}{0.973 - p(sat, T\,°C)} * (1618.65 - 233.26) * \frac{18}{1000} = ?\ \frac{kg}{s} \ldots (2)$$

By subbing equation (1) into equation (2), an equation for m_9 can be yielded.

$$\dot{m}_9 = \underbrace{(\dot{m}_7 + \dot{m}_9)}_{(1)} - \underbrace{\dot{m}_7}_{(2)} \ldots (3)$$

Energy Balance:

$$Energy_{in} = Energy_{out}$$

$$(Energy_5 + Energy_6 + Energy_{11}) = (Energy_{10} + Energy_7 + Energy_9)$$

$$1050.90 + 13109.2 + 1214.9 = 6828 + Energy_7 + Energy_9$$

$$Energy_7 + Energy_9 = 8547 \quad \ldots. (4)$$

Energy of stream 9 can be calculated by using the saturated liquid enthalpy at 30 C and the calculated mass flow.

$$h(sat.liq, 30\,°C) = 125.7\frac{kJ}{kg}$$

$$Energy_9 = 125.7 * \dot{m}_9 \ldots (5)$$

By subbing equation (3) into Equation (5)

$$Energy_9 = 125.7 * \left(5.17 + 4.30 - 2.14 - \frac{p(sat, T\,°C)}{1.003 - p(sat, T\,°C)} * (1706.17 - 238.66) * \frac{18}{1000}\right) \quad \ldots (5)$$

Energy of Stream 7 can be seen as:

$$Energy_7 = \dot{m}_{7,H2O} * h_{7,H2O} + \dot{m}_{7,N2} * h_{7,N2} + \dot{m}_{7,O2} * h_{7,O2} + \dot{m}_{7,CO2} * h_{7,CO2} + \dot{m}_{7,Ash} * Cp_{Ash} * T$$

$$Energy_7 = \frac{p(sat, T\,°C)}{1.003 - p(sat, T\,°C)} * (1706.17 - 238.66) * \frac{18}{1000} * h_{7,H2O} + 32.8 * h_{7,N2} + 2.7 * h_{7,O2} + 8.9 * h_{7,CO2} + 0.047 * Cp_{Ash} * T \quad \ldots (6)$$

Solving equations 4, 5, and 6 iteratively will give a temperature of flue gas leaving of around 42.7 *C.

$$T_7 = 42.7\ C$$

TABLE 3.19 ENERGY BALANCE AROUND THE HUMIDIFER

Source	Energy (In) kW	Energy (Out) kW
Stream 5	1079.49	
Stream 6	12699.5	
Stream 11	1211.3	
Stream 10		6802
Stream 7		7607.99
Stream 9		580.2

STEP 3. FLUE GAS FAN 2

Energy balance around the FG fan:

To find the temperature of Stream 8 exiting the FG fan 2, the Win required by the fan must be calculated.

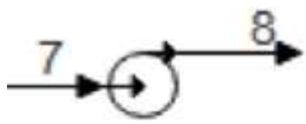

FIGURE 3.17 FG FAN 2.

The volumetric flow rate of the air can be calculated using the Ideal gas law:

$$\dot{V}_7 = \frac{\dot{n}_7 RT}{P_7} = \frac{1572.64 * 8.313 * (42.5 + 273.15)}{0.973 * 100000} = 42.42 \frac{m^3}{s}$$

From here, the work required by the fan can be calculated, assuming an efficiency of 70% (given). Pressure of Stream 7 is at 0.973 bar and Stream 8 is assumed to return back to atmospheric pressure at 1.013 bar.

$$Win\,[kW] = \frac{\dot{V}_7 * (P_8 - P_7)}{\eta} = 42.42 * \frac{101.3 - 97.3}{0.7}$$

$$Win = 242.38\,kW \ldots (1)$$

Temperature of Stream 8:

$$Win = Energy_8 - Energy_7 \ldots (2)$$

Given we know the composition of FG streams 8 and 7, by using the energy values in the excel spreadsheet, the temperature of Stream 7 can be calculated. Using the solver and subbing the value generated from equation (1) into equation (2), a temperature of 44.78 C was given for Stream 8.

$$T_8 = 44.78\,C$$

4. QT DIAGRAM

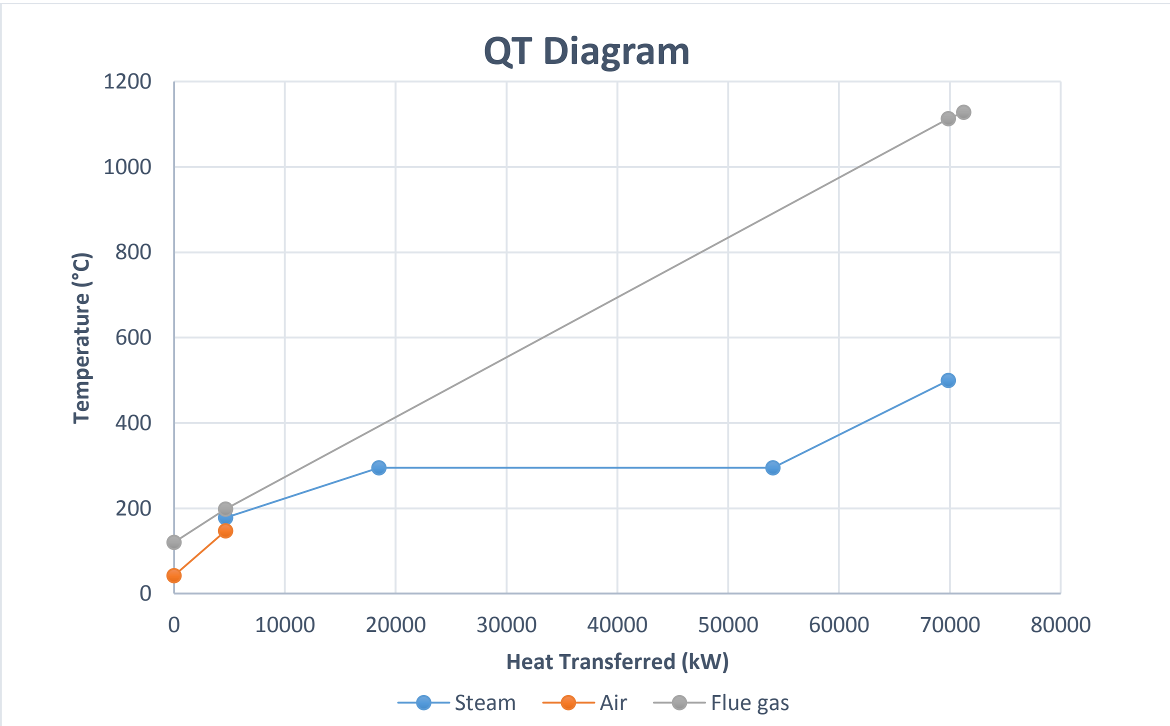

FIGURE 4.1 QT DIAGRAM OF AIR, STEAM, AND FLUE GAS AROUND THE BOILER

Figure 4.1 illustrates the heat transferred between flue gas and the streams of steam and air at different temperatures. We can observe that it takes around 4600 kW to heat air in the preheater from 42.55 °C to 147.28 °C. In the preheater, the flue gas temperature decreases from 197.96 °C to 120 °C. Flue gas leaves the boiler at 197.96 °C, and water enters it at 177.96 °C. This allows to have a minimum temperature difference or pinch point between flue gas and steam/water of 20 °C as seen on the diagram. In the boiler, water is preheated, evaporated and superheated to 500 °C at 80 bar. From the diagram, we can observe that this process requires approximately 65000 kW. This matches the calculated value of 65.20 MW transferred to steam in the boiler.

To analyse all the heat the fuel can transfer to its surroundings, the adiabatic flame temperature for flue gas was calculated. The diagram shows that in adiabatic condition, flue gas can reach a temperature 1113.1 °C and transfer approximately 70 MW of heat. Convection losses of 2% around the boiler are expressed at the top of the flue gas stream in the diagram. Considering this, flue gas should reach a temperature of 1128.0 °C in order to provide all the necessary heat to satisfy the demands of the project.

CALCULATIONS

The relative enthalpies and temperatures are calculated using the numbers yielded from the mass and energy balances in Part 3. The adiabatic flame temperature of the flue gas was also calculated for the QT diagram. This temperature was obtained with an energy balance around the boiler assuming that no heat is transferred to its surroundings.

$$h_{Flue\ gas} = h_{Flue\ gas\ at\ 197.96\ °C} + Q_{boiler}$$

$$h_{Flue\ gas\ at\ 197.96\ °C} = h_2 = 34026.88\ kW$$

$$Q_{boiler} = \dot{m}_A(h_A - h_M)$$

$$Q_{boiler} = 24.68\frac{kg}{s}\left(3399.5\frac{kJ}{kg} - 757.73\frac{kJ}{kg}\right) = 65196.32\ kW$$

$$h_{Flue\ gas} = 34026.88\ kW + 65196.32\ kW = 99223.2\ kW$$

From this equation, we get the enthalpy of flue gas in adiabatic condition. We can calculate the temperature of the flue gas using the solver option from Microsoft Excel in the sheet provided to calculate the enthalpy of flue gas.

$$h_{Flue\ gas} = 99223.2\ kW$$

$$T_{Ad} = 1113.1\ °C$$

The temperature needed to account for convection losses can be calculated in a similar way.

$$h_{Flue\ gas\ including\ convection\ losses} = h_{Flue\ gas\ at\ 197.96\ °C} + Q_{boiler} + Q_{convection\ losses}$$

$$Q_{convection\ losses} = 0.02 \cdot LHV$$

$$Q_{convection\ losses} = 0.02 * 15.79\frac{MJ}{kg} * 4.44\frac{kg}{s} * \frac{1000\ kJ}{1\ MJ}$$

$$Q_{convection\ losses} = 1401.72\ kW$$

$$h_{Flue\ gas\ including\ convection\ losses} = 34026.88\ kW + 65196.32\ kW + 1401.72\ kW$$

$$h_{Flue\ gas\ including\ convection\ losses} = 100624.92\ kW$$

$$T_{Flue\ gas\ including\ convection\ losses} = 1128.0\ °C$$

5. EFFICIENCY OF THE CYCLE

TABLE 5.1 EFFICIENCIES OF THE PROCESS WITH AND WITHOUT HEAT RECOVERY

	Net Electrical Efficiency	Heat Efficiency	Total Efficiency
With Waste Heat Recovery	0.194	0.937	1.130
Without Waste Heat Recovery	0.197	0.716	0.913

$$\eta_{el} = \frac{Net\ Electric\ Output}{LHV\ of\ fuel} \qquad \eta_{heat} = \frac{Heat\ delivered\ to\ users}{LHV\ of\ fuel}$$

$$\eta_{tot} = \eta_{el} + \eta_{heat}$$

The inclusion of the flue gas condenser allows for energy recovery of the process in which heat of the water condensate is recovered for use in district heating. As shown in table 5.1, this has allowed an increase of around 22.5% to the total and heat efficiency of the biomass-fired steam cycle in a paper mill.

Since the biofuel entering the boiler is wet, the flue gas condensation is able to recover the energy of the water vapour as the moist flue gas is cooled by the counter-current district heating streams. The district heating return temperature of 50 C is lower than the saturation temperature of the flue gas vapour of 53 C and therefore, the water is able to condense allowing for transfer of heat to the district heating network. The flue gas is cooled from 124. 39 C to 53 C, releasing a heat of 15.76 MW to the district heating network.

The inclusion of the air humidification further improves the total efficiency of the combined heat and power plant. The residual energy in the flue gas leaving the condenser is able to be transferred back into the combustion air. As such, with direct contact with the flue gas and condensate streams, the combustion air is heated from ambient air of 30 C to 40 C with 100% relative humidity. Not all of the condensate from the condenser will evaporate into the combustion air. A fraction of water will leave the humidifier as liquid water.

On the other hand, the net electrical efficiency of the process was calculated by dividing the net electricity produced by turbine, minus the power consumption of pumps and fans over the the lower heating value of fuel (LHV). For both instances, with and without the heat recovery, the electrical efficiency stayed relatively the same. This is due to the fact that the inclusion of the waste heat recovery does not generate any extra electricity. The only source of the electricity is from the generation of the turbine in the steam cycle. Furthermore, the extra waste heat recovery only requires one extra flue gas fan to transport the flue gas out of the humidifier at atmospheric pressure.

Possibilities to further increase the efficiency of the cycle exist for the use of district heating. Lowering the exiting temperature of the district heating further allows for an increased heat generation to the DH network. By installing a heat pump that can be used to recover heat for flue gas at even lower temperature than the current DH water will cause more water to condense.

5A CALCULATIONS

TABLE 5.2 TURBINE POWER PRODUCTION

	Pel (kW)
Turbine A	$Pel = \eta_{mech}\eta_{gen}\dot{m}_A(h_A - h_B)$ $Pel = 0.98 * 0.95 * 24.68 * (3399.5 - 2959.72)$ $Pel = 10104.37$
Turbine B	$Pel = \eta_{mech}\eta_{gen}\dot{m}_D(h_B - h_D)$ $Pel = 0.98 * 0.95 * 22.82 * (2959.72 - 2757.22)$ $Pel = 4302.86$
Total Power Produced	*14407.23*

TABLE 5.3 PUMPS AND FANS (EXAMPLE CALCULATIONS CAN BE FOUND IN PART 3B (II) CALCULATIONS).

	Power Consumption (kW)
FW Pump 1.	7.08
FW Pump 2.	271.61
Air Fan 1.	113.21
FG Fan 1.	268.16
FG Fan 2.	243.94
Total Power Consumption	*904.01*

TABLE 5.3 HEAT DELIVERED TO USERS

	Heat Delivered (kW)
Steam	50000
Flue Gas	15462.43
Total Heat Delivered to Users	*65462.43*

Efficiency calculations

Calculations for LHV are found in Appendix B.

With Heat Recovery

$$\eta_{el} = \frac{Total\ power\ produced - Total\ power\ consumed}{LHV\ of\ fuel} = \frac{14417.23 - 901.31}{(15.79 * 4.437 * 1000)}$$
$$= 0.194$$

$$\eta_{heat} = \frac{Total\ Transferred\ Heat}{LHV\ of\ fuel} = \frac{65403.43}{(15.79 * 4.437 * 1000)} = 0.934$$

$$\eta_{tot} = 1.130$$

Without Heat Recovery:

Total power consumed now does not include FG Fan 2.

$$\eta_{el} = \frac{Total\ power\ produced - Total\ power\ consumed}{LHV\ of\ fuel} = \frac{14417.23 - 658.07}{(15.79 * 4.437 * 1000)}$$
$$= 0.196$$

Total Heat Delivered now does not include the Flue Gas Condenser

$$\eta_{heat} = \frac{Total\ Heat\ Delivered}{LHV\ of\ fuel} = \frac{50000}{(15.79 * 4.437 * 1000)} = 0.713$$

$$\eta_{tot} = 0.713 + 0.196 = 0.913$$

6. CAPITAL BUDGETING

To analyse the financial viability and potential of this cycle, 9 profit cases were analysed: electricity prices of 200 SEK/MWh, 400 SEK/ MWh and 600 SEK/MWh + additional income of 50 SEK/MWh, 150 SEK/MWh and 300 SEK/MWh from green certificates.

The investment cost is calculated from the net electricity output and it is only in the case with pure heat production that is based on the heat output of 50 MW (excluding the waste heat recovery). The additional investment cost for waste heat recovery is based on the waste output of 15.76 MW. Table 6.1 below illustrates the operation time, heat price, and additional investment required for both heat productions. Calculations can be seen in Appendix C.

TABLE 6.1 ECONOMIC VALUE FOR HEAT AND POWER PRODUCTION WITH WASTE RECOVERY AND WITHOUT WASTE HEAT RECOVERY

	Without waste heat Recovery	With Waste Heat Recovery
Operation time (h/year)	7500	4000
Heat Price (SEK/MWh)	200	150
Investment (MSEK/MW)	29	2

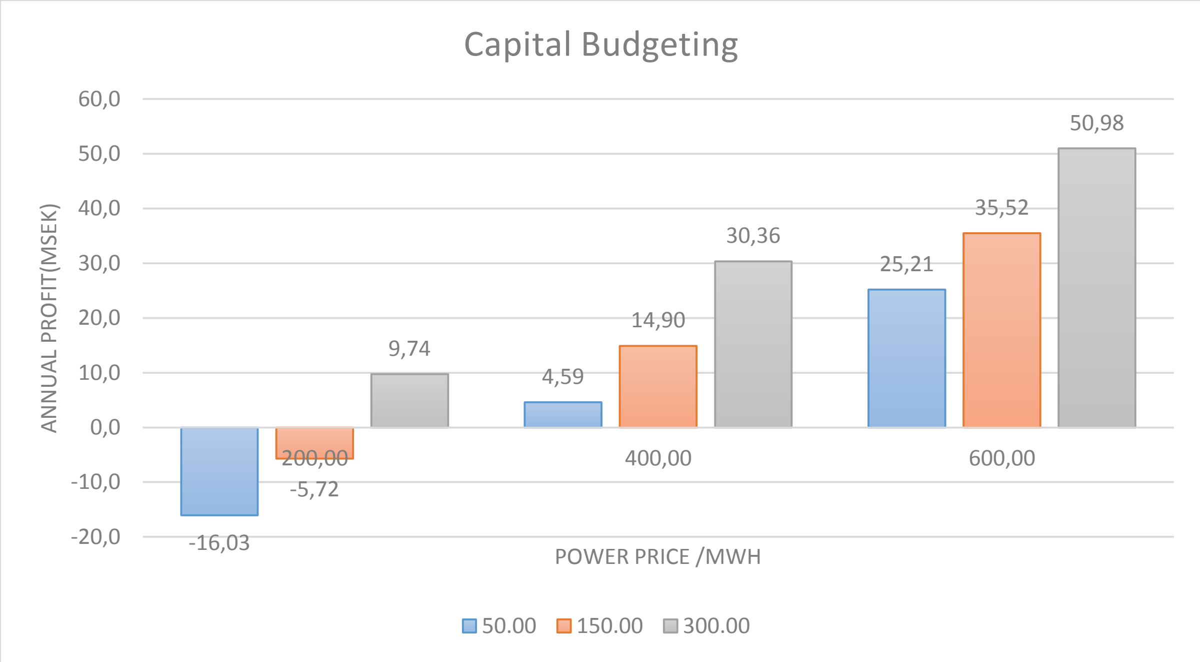

FIGURE 6.1 PROFIT VALUES FOR CYCLE WITHOUT WASTE HEAT RECOVERY

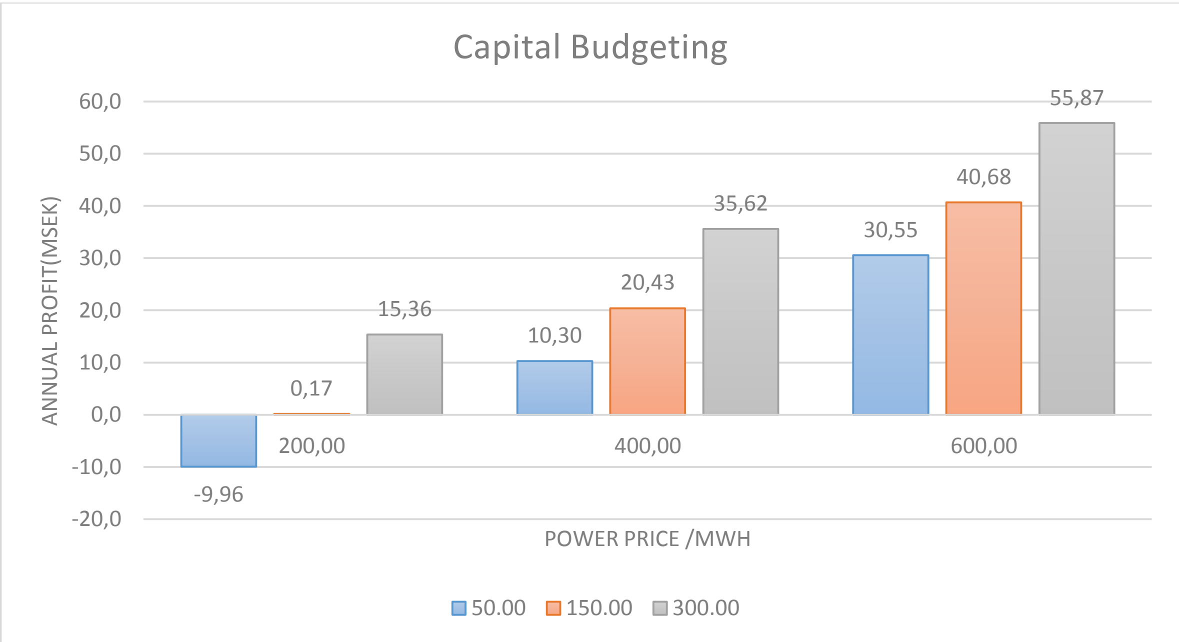

FIGURE 6.2 PROFIT VALUES FOR CYCLE WITH WASTE HEAT RECOVERY

DISCUSSION

As the result show in Figure 6.1, the cycle without heat recovery will generate profit for all cases except when the certificates equal 50 and 150 SEK/MWh and the power price is equal to 200 SEK/MWh. In comparison, Figure 6.2 shows that the inclusion of the waste heat recovery allows for profit in all cases except when the certificates equal 50 SEK/MWh and the power price is equal to 200 SEK/MWh. As such, the inclusion of the waste heat source allows for a low cost and low fuel consumption process to be implemented.

Looking at the cycle, beyond its efficiency and the point of view of investment and operation, this combined cycle is a profitable solution allowing it to generate higher profits from the recovery of waste heat in flue gas. By keeping the complexity of the system low, the configuration is able to minimise the additional electrical consumption required for additional heat recovery generation. As such, this cycle is an attractive investment.

CONCLUSION

In order to satisfy the heat of 50 MW, 24.71 kg/s of steam were needed to leave the boiler at 500 °C and 80 bar and go through the steam cycle. This allowed to have 20.64 kg/s of steam that delivered heat to the paper mill using a condenser. The fuel demand to produce that amount of steam at those conditions was 4.42 kg/s of moisture and ash free biomass. The heat delivered by the fuel in the boiler was 65.27 MW, which was used in the steam cycle. In addition, the condensation of 50.74 kg/s of flue gas provided 15.76 MW of heat recovered and available for use in district heating.

From this report, we are able to see that the biomass-fired steam cycle with flue gas condensation to recover waste heat is a favourable process. It has an overall efficiency above unity of 1.137, and it is able to generate profit for all six cases of capital budgeting. The inclusion of the flue gas condenser and humidifier allow a huge increase in the thermal efficiency of 22.5% without requiring too much extra power consumption for operation with a relatively stable electrical efficiency of 0.197 to 0.196 from the process without and with heat recovery, respectively. As such, this process cycle is a reliable investment.

APPENDICES

APPENDIX A. EXAMPLE OF ENERGY CALCULATIONS

$$Energy = \dot{m}_i Cp_i T_i = \dot{m}_i h_i$$

Wet Biofuel: Stream 1

$$Energy_1 = \left(\dot{m}_{1,DB} Cp_{DB}(25-0)K + \dot{m}_{1,H20} h_{H20\,(25\,C)} - \dot{m}_{1,A} Cp_A (25-0)K\right)$$

Where DB is dry biomass (MAF), H_2O is liquid water, A is ash. Temperature is in Kelvin with a reference temperature of 0 C and mass flow rates are in kg/s.

Air: Stream 13 (using the excel spreadsheet)

$$Energy_{13} = \left(\dot{m}_{1,H20} h_{H20} + \dot{m}_{1,N2} h_{N2} + \dot{m}_{1,O2} h_{O2}\right)$$

Where H_2O is the water vapour content (dependent on the relative humidity of the air stream), and N_2 and O_2 is the nitrogen and oxygen content of air which is 79 mol% N_2 and 21 mol% O_2.

Flue Gas: Stream 2 (using the excel spreadsheet)

$$Energy_2 = \left(\dot{m}_{1,H20} h_{H20} + \dot{m}_{1,N2} h_{N2} + \dot{m}_{1,O2} h_{O2} + \dot{m}_{1,CO2} h_{CO2}\right)$$

Where H_2O is the water vapour content gained from combustion, N_2 is the nitrogen content (not reacted), O_2 is the remaining oxygen content, and CO_2 is the carbon dioxide produced from combustion of biofuel and air.

APPENDIX B. LHV

$$\%moisture = 1\frac{kg\ H_2O}{kg\ MAF}$$

$$LHV = HHV - Heat\ of\ condensation$$

The heat of condensation is for H_2O in the fuel and H_2O formed during combustion.

$$LHV\ (wet) = 19.6\frac{MJ}{kg\ MAF} - 2.4417\frac{MJ}{kg\ H_2O} * (55.56+31) molH_2O * \frac{18\ g}{molH_2O} * \frac{1\ kg}{1000\ g} * \frac{1\ kg\ H_2O}{kg\ MAF}$$

$$LHV\ (wet) = 15.79\frac{MJ}{kg\ MAF}$$

TABLE C1 CAPITAL BUDGETING FOR BIOMASS-FIRED STEAM CYCLE WITH WASTE HEAT RECOVERY

Heat Production		65.46	MW	
Fuel Demand		70.09	MW	
Power Production		13.50	MW	
Investment		422.52	MSEK	
Annual Costs and income				
Power price / MWh	200	400	600	SEK/MWh
Heat income	84.28	84.28	84.28	MSEK/y
Electricity	20.25	40.51	60.76	MSEK/y
Green certificates 1	5.06	5.06	5.06	MSEK/y
Green certificates 2	15.19	15.19	15.19	MSEK/y
Green certificates 3	30.38	30.38	30.38	MSEK/y
Operation and maintenance	6.34	6.34	6.34	MSEK/y
Operation and maintenance	13.14	13.14	13.14	MSEK/y
Capital cost	42.25	42.25	42.25	MSEK/y
Annual fuel cost	57.82	57.82	57.82	MSEK/y
Annual Profit 1	-9.96	10.30	30.55	MSEK/y
Annual Profit 2	0.17	20.43	40.68	MSEK/y
Annual Profit 3	15.36	35.62	55.87	MSEK/y

Calculations with waste heat recovery

$$Fuel\ demand = \frac{Heat\ delivered}{Heat\ efficiency}$$

$$Fuel\ demand = \frac{65.46\ MW}{0.934} = 70.09\ MW$$

$$Power\ Production = Fuel\ demand * electrical\ efficiency$$

$$Power\ Production = 70.09\ MW * 0.193 = 13.50\ MW$$

$$Investment = Power\ Production * Specific\ Investment$$

$$Investment = 13.50\ MW * 29\ \frac{MSEK}{MW_e} + 15.46 * 2\ \frac{MSEK}{MW_e} = 422.52\ MSEK$$

$$Heat\ income = Heat\ delivered * Heat\ price * Operating\ time$$

$$Heat\ income = \left(50\ MW * 7500\frac{h}{y} * 200\frac{SEK}{MWh}\right) + \left(15.46\ MW * 4000\frac{h}{y} * 150\frac{SEK}{MWh}\right)$$

$$Heat\ income = 84.28\ \ MSEK/y$$

Power income: Economic calculations for an electricity price of 200 SEK/MWh, 400 SEK/MWh

$$Electricity = Power\ Production * Electricity\ price * Operating\ time$$

$$Electricity = 13.50\ MW * 200\frac{SEK}{MWh} * 7500\frac{h}{y} = 20.25\ MSEK/y$$

$$Electricity = 13.50\ MW * 400\frac{SEK}{MWh} * 7500\frac{h}{y} = 40.51\ MSEK/y$$

$$Electricity = 13.50\ MW * 600\frac{SEK}{MWh} * 7500\frac{h}{y} = 60.76\ MSEK/y$$

Additional income of 50 SEK/MWh, 150 SEK/MWh and 300 SEK/MWh from green certificates

$$Green\ certificate\ income = Power\ Production * Green\ certificate\ price * Operating\ time$$

$$Green\ certificate\ income = 13.50\ MW * 50\frac{SEK}{MWh} * 7500\frac{h}{y} = 5.06\ MSEK/y$$

$$Green\ certificate\ income = 13.50\ MW * 150\frac{SEK}{MWh} * 7500\frac{h}{y} = 15.19\ MSEK/y$$

$$Green\ certificate\ income = 13.50\ MW * 300\frac{SEK}{MWh} * 7500\frac{h}{y} = 30.38\ MSEK/y$$

$$Fixed\ Operation\ and\ Maintenance = 1.5\%\ of\ Investment$$

$$Fixed\ Operation\ and\ Maintenance\ Cost = 0.0015 *\ 422.52\ MSEK = 6.34\ MSEK/y$$

$$\begin{aligned} &Variable\ Operation\ and\ Maintenance \\ &\qquad = Fuel\ demand * Operation\ \&\ Maintenance\ Cost * Operating\ time \end{aligned}$$

$$Variable\ Operation\ and\ Maintenance\ Cost = 70.09\ MW * 25\frac{SEK}{MWh} * 7500\frac{h}{y}$$

$$Variable\ Operation\ and\ Maintenance\ Cost = 13.14\ MSEK/y$$

$$Capital\ Cost = Investment * Annuity\ factor$$

$$Capital\ Cost = 422.52\ MSEK * 0.1 = 42.25$$

$$Annual\ Fuel\ Cost = Fuel\ Demand * Fuel\ cost$$

$$Annual\ Fuel\ Cost = 70.09\ MW * 110\frac{SEK}{MWh} * 7500\frac{h}{y} = 57.82\ MSEK/y$$

$$Annual\ Profit = Annual\ income - Annual\ Cost$$

$$\begin{aligned} Annual\ Profit = {} & Heat\ Income + Power\ income + Green\ certificate\ income \\ & - Fixed\ Operation\ and\ Maintenance\ Cost \\ & - Variable\ Operation\ and\ Maintenance\ Cost - Capital\ cost \\ & - Annual\ Fuel\ Cost \end{aligned}$$

Example calculation for an annual profit with an electricity price of 200 SEK/MWh and green certifate income of 150 SEK/MWh

$$Annual\ Profit = 84.28 + 20.25 + 5.06 - 6.34 - 13.14 - 42.25 - 57.82 \quad MSEK/y$$

$$Annual\ Profit = -9.96\, MSEK/y$$

TABLE C2 CAPITAL BUDGETING FOR BIOMASS-FIRED STEAM CYCLE WITHOUT WASTE HEAT RECOVERY

Heat Production		50.00	MW	
Fuel Demand		70.09	MW	
Power Production		13.75	MW	
Investment		398.67	MSEK	
Annual Costs and income				
Power price / MWh	200	400	600	SEK/MWh
Heat income	75.00	75.00	75.00	MSEK/y
Electricity	20.62	41.24	61.86	MSEK/y
Green certificates 1	5.16	5.16	5.16	MSEK/y
Green certificates 2	15.47	15.47	15.47	MSEK/y
Green certificates 3	30.93	30.93	30.93	MSEK/y
Operation and maintenance	5.98	5.98	5.98	MSEK/y
Operation and maintenance	13.14	13.14	13.14	MSEK/y
Capital cost	39.87	39.87	39.87	MSEK/y
Annual fuel cost	57.82	57.82	57.82	MSEK/y
Annual Profit 1	-16.03	4.59	25.21	MSEK/y
Annual Profit 2	-5.72	14.90	35.52	MSEK/y
Annual Profit 3	9.74	30.36	50.98	MSEK/y

Calculations without waste heat recovery

$$Fuel\ demand = \frac{Heat\ delivered}{Heat\ efficiency}$$

$$Fuel\ demand = \frac{50\ MW}{0.713} = 70.09\ MW$$

$$Power\ Production = Fuel\ demand * electrical\ efficiency$$

$$Power\ Production = 70.09\ MW * 0.196 = 13.75\ MW$$

$$Investment = Power\ Production * Specific\ Investment$$

$$Investment = 13.75\ MW * 29\ \frac{MSEK}{MW_e} = 398.67\ MSEK$$

$$Heat\ income = Heat\ delivered * Internal\ heat\ price * Operating\ time$$

$$Heat\ income = 50\ MW * 200 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 75\ \ MSEK/y$$

Power income: Economic calculations for an electricity price of 200 SEK/MWh, 400 SEK/MWh

$$Electricity = Power\ Production * Electricity\ price * Operating\ time$$

$$Electricity = 13.75\ MW * 200 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 20.62\ MSEK/y$$

$$Electricity = 13.75\ MW * 400 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 41.24\ MSEK/y$$

$$Electricity = 13.75\ MW * 600 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 61.86\ MSEK/y$$

Additional income of 50 SEK/MWh, 150 SEK/MWh and 300 SEK/MWh from green certificates

$$Green\ certificate\ income = Power\ Production * Green\ certificate\ price * Operating\ time$$

$$Green\ certificate\ income = 13.75\ MW * 50 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 5.16\ MSEK/y$$

$$Green\ certificate\ income = 13.75\ MW * 150 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 15.47\ MSEK/y$$

$$Green\ certificate\ income = 13.75\ MW * 300 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 30.93\ MSEK/y$$

$$Fixed\ Operation\ and\ Maintenance = 1.5\%\ of\ Investment$$

$$Fixed\ Operation\ and\ Maintenance\ Cost = 0.0015 *\ 398.67\ MSEK = 5.98\ MSEK/y$$

$$Variable\ Operation\ and\ Maintenance = Fuel\ demand * Operation\ \&\ Maintenance\ Cost * Operating\ time$$

$$Variable\ Operation\ and\ Maintenance\ Cost = 70.09\ MW * 25 \frac{SEK}{MWh} * 7500 \frac{h}{y}$$

$$Variable\ Operation\ and\ Maintenance\ Cost = 13.14\ MSEK/y$$

$$Capital\ Cost = Investment * Annuity\ factor$$

$$Capital\ Cost = 398.67\ MSEK * 0.1 = 39.87$$

$$Annual\ Fuel\ Cost = Fuel\ Demand * Fuel\ cost$$

$$Annual\ Fuel\ Cost = 70.09\ MW * 110 \frac{SEK}{MWh} * 7500 \frac{h}{y} = 57.82\ MSEK/y$$

$$Annual\ Profit = Annual\ income - Annual\ Cost$$

$$\begin{aligned} Annual\ Profit = {} & Heat\ Income + Power\ income + Green\ certificate\ income \\ & - Fixed\ Operation\ and\ Maintenance\ Cost \\ & - Variable\ Operation\ and\ Maintenance\ Cost - Capital\ cost \\ & - Annual\ Fuel\ Cost \end{aligned}$$

Example calculation for an annual profit with an electricity price of 200 SEK/MWh and green certifate income of 150 SEK/MWh

$$Annual\ Profit = 75 + 20.62 + 5.16 - 5.98 - 13.14 - 39.87 - 57.82\quad MSEK/y$$

$$Annual\ Profit = -16.03\ MSEK/y$$