

REFEREED PAPER

THERMAL-HYDRAULIC ANALYSIS OF HEAT RECOVERY THROUGH FLUE GAS CONDENSATION IN BIOMASS FIRED BOILERS

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Abstract

Higher thermal efficiency (plant efficiency) is becoming an ever-increasing imperative in process and power generation industries. One of the many research fields which aim to increase this efficiency is heat recovery of waste heat streams. In biomass fired boiler systems the major contributor to loss in fuel efficiency is the wet gas loss or moisture in fuel loss. Moisture loss accounts for 18-25% of losses, depending on the fuel used in the boiler system and this in turn increases the amount of fuel required to achieve a certain energy output. The only way to regain some of the wet gas loss is by condensing the moisture out of the flue gas stream at the lowest temperature of the flue gas stream, which is usually downstream of the economiser or airheater. This paper thus sets out to explain the development of a thermal-hydraulic model which simulates a flue gas condensing device for a small biomass fired boiler. The paper is divided into the following sections: explanation of applicable gas mixture thermodynamics, film condensation of binary mixtures with inert gas presence, lumped system analysis of condensing economiser and thermal-hydraulic network modelling of the condensing economiser.

Keywords: heat transfer, thermodynamics, condensation, moisture loss, thermal-hydraulic modelling, flue gas condensation

Introduction

During combustion processes of moisture rich fuel such as biomass (which includes bagasse) there is an inherent efficiency loss due to wet gas loss. This is due to the latent energy lost during vaporisation of the moisture in the fuel when the water vapor is lost to the atmosphere. These wet gas losses range anywhere from 18-25% of the fuel efficiency losses depending on the fuel used in the boiler system. A method to regain some of the wet gas loss is by condensing out the moisture from the flue gas by lowering the temperature of the gas below its dew point and bringing the gas into contact with a cooled heat transfer surface. This is done by bringing the flue gas into contact with a U-tube bank with water as the tube side cooling medium. The first section of the tube bank will be used as the 'economiser section' where the flue gas will be cooled to the dew point of the water vapor at partial pressure. The second stage is the condenser stage, where the water is condensed out of the binary gas mixture and routed to a chemical processing plant or waste water system.

This paper covers research that was conducted to determine the validity of a condensing economiser for the John Thompson Microgen boiler. The theoretical models developed in this document are a preliminary analysis of the conceptual device and the development of

such a device is still ongoing. The model utilises a lumped system analysis for sizing and calculation of thermodynamic and heat transfer properties, then uses the parameters in a quasi two-dimensional thermal-hydraulic systems computational fluid dynamics (SCFD) model.

This document is subdivided into the following sections: literature study, thermal-hydraulic modeling and results, and recommendations and conclusion.

Literature study

In this section all the various thermodynamic and heat transfer mechanisms used to model the condensing economiser are presented. The sections are: gas mixture thermodynamics, filmwise condensation of pure vapors, filmwise condensation of binary gas mixtures with inert gas present, tube bank heat transfer and internal forced convection heat transfer in pipe flow.

When components of a gas mixture do not behave as ideal gases, the analysis becomes more complex because the properties of real gas such as enthalpy, internal energy and specific heat changes as a function of pressure and temperature. Thus, when real gases are involved it is necessary to account for the effect of non-ideal behavior on the mixture's properties. One way of doing that is using the compressibility factors in conjunction with generalised equations and charts for real gases. Therefore for a real gas the enthalpy change between two states can be given by the following equation (Kay's rule):

$$h_2 - h_1 = (h_2 - h_1)_{ideal} - RT_{cr,mixture}(Z_{h2} - Z_{h1}) \quad (1)$$

where $h_{1,ideal}$ and $h_{2,ideal}$ is calculated for the specific gas mixture using:

$$h_{mixture} = \sum_{i=1}^k m f_i h_i \quad (2)$$

The enthalpies h_i in (2) are the ideal enthalpies of each of the components of the gas mixture, which is determined from the ideal gas tables (Boles and Cengel, 2006. In (2) $m f_i$ is the mass fraction of the gas mixture component. The compressibility values in (1) are determined from the compressibility chart, by calculating the gas mixture reduced pressure and temperatures using the same approach as in (2). Using (1) along with the energy conservation, equation the amount of heat transfer required was calculated which is required to condense the moisture present in the gas (Boles and Cengel, 2006).

When a vapor comes into contact with a surface, the temperature of which is below the thermodynamic equilibrium, the vapor at this surface turns from the gaseous state into the liquid state. This process is called condensation and there are variations such as condensation on a flat plate vertical, horizontal, dropwise and filmwise. The condensation mechanism used to model the heat transfer of the condensing economiser is filmwise condensation on a horizontal tube bundle. For the case of a tube bundle the effect of vapor shear stress on the heat transfer in the horizontal pipe must be taken into account by using the following equation (Numrich, 2010):

$$Nu_{F,1} = CX \left(1 + \frac{0.276 Pr_{film}}{X^4 Fr Ph} \right)^{1/4} \sqrt{Re_{GS}} \frac{L_{char}}{D} \quad (3)$$

where C is equal to 0.8 for in-line tube banks, Re is the Reynolds number and Pr is the gas mixture Prandtl number. The Froude number in (3) is calculated using:

$$Fr = \frac{V_{vapor}^2}{g \cdot D} \quad (4)$$

The Reynolds number is determined by:

$$Re_{GS} = \frac{\rho_{vapor} V_{vapor} D}{\mu_{vapor}} \quad (5)$$

The phase change number is calculated by the following equation:

$$Ph = \frac{Cp_{film}(T_{gas\ mix} - T_{wall})}{\Delta h_{fg, film}} \quad (6)$$

The other variables are determined by:

$$G = \frac{Ph}{Pr_{film}} \left(\frac{\rho_{film} \mu_{film}}{\rho_{vapor} \mu_{vapor}} \right)^{0.5} \quad (7)$$

$$X = 0.9(1 + G^{-1})^{1/3} \quad (8)$$

To take into account the effect of the liquid running from one tube to the next, also called condensate inundation, the following relation was used (Numrich, 2010):

$$\frac{Nu_{F,1,RR}}{Nu_{F,1}} = N_{rows}^{-0.25} \quad (9)$$

Gaseous substances in a specified thermodynamic state can be subdivided into vapors and inert gases. Vapors are condensable at respective specified coolant temperatures, but inert gases are not. In condensation of gas mixtures all the components may be vapors or only some. In both cases, in addition to the heat transfer resistance in the condensate film, heat and mass transfer resistances also occur in the gas phase, leading to reduction of condensation. The heat flow to be removed locally to the coolant adjacent to the heat transfer surface is:

$$q = k'(T_{film} - T_{coolant}) \quad (10)$$

where k' is composed of the various heat transfer coefficients of the condensate film, namely heat transfer coefficient of the film, conduction through pipe wall and the forced convection on the coolant side:

$$\frac{1}{k'} = \frac{1}{h_{film}} + \frac{t}{k_{wall}} + \frac{1}{h_{coolant}} \quad (11)$$

To determine the correct film heat transfer coefficient, the film temperature must be assumed and the calculation process iterated until the temperature value converges. The iterative equation used to determine the correct film temperature is (Schlunder, 2010):

$$k'(T_{film} - T_{coolant}) = h_{gas\ side} \Phi_T \left(\frac{\Delta h_{vg}}{Cp_{vapor}} + \frac{T_{gas} - T_{film}}{1 - e^{-\Phi_T}} \right) \quad (12)$$

where Φ_T is:

$$\Phi_T = \frac{Cp_{vapor}}{Cp_{gas} Le^{0.6}} \ln \left(\frac{y_{2F}}{y_{2B}} \right) \quad (13)$$

y_{2F} - Mole fraction of gas (not water included) at film temperature

y_{2B} - Mole fraction of gas (not water included) at bulk gas temperature

Le - Lewis number $\frac{k_{gas}}{n_{gas} Cp_{gas} \delta_{gas}}$

n_{gas} - Molar density in kmol/m³

δ_{gas} - Mass diffusion coefficient m²/s.

To determine the internal forced convection heat transfer coefficient for the flow inside the tubes the Dittus-Boelter equation was used (Mills, 1995):

$$Nu_D = 0.023Re_D^{0.8}Pr^{0.4} \quad (14)$$

To determine the gas side heat transfer coefficient the calculation is divided into two parts: first for the calculation of the Nusselt number for forced flow past a cylinder and then adjusting this value for forced flow past a tube bank. The Nusselt number for a single tube can be determined using (Mills, 1995):

$$Nu_D = 0.3 + \frac{0.62Re_D^{0.5}Pr^{1/3}}{[1+(0.4/Pr)^{2/3}]^{1/4}} \quad Re < 10^4$$

$$Nu_D = 0.3 + \frac{0.62Re_D^{0.5}Pr^{1/3}}{[1+(0.4/Pr)^{2/3}]^{1/4}} \left[1 + \left(\frac{Re}{282000} \right)^{0.5} \right] \quad 2 \cdot 10^4 < Re < 4 \cdot 10^5.$$

Next the Nusselt number is adjusted for a tube bank by using the following calculation procedure:

$$\varphi = 1 - \frac{\pi}{4P_T} \quad P_L \geq 1$$

$$\varphi = 1 - \frac{\pi}{4P_T P_L} \quad P_L < 1$$

The arrangement factor is then calculated for aligned tube banks:

$$\Phi_{aligned} = 1 + \frac{0.7}{\varphi^{1.5}} \frac{\frac{S_L}{S_T} - 0.3}{\left(\frac{S_L}{S_T} + 0.7 \right)^2}$$

For tube banks with more than ten rows the adjusted Nusselt number can be calculated by:

$$Nu_D^{10+} = \Phi_{aligned} Nu_D^1$$

and less than ten rows:

$$Nu_D^{10-} = \frac{1 + (N - 1)\Phi}{N} Nu_D^1$$

Thermal-hydraulic modelling and results

For the preliminary calculations it was decided to size the device for 33% moisture extraction and investigate the efficiency increase and required heat transfer area. The thermal-hydraulic calculation is divided into two sections; firstly the thermodynamic calculations to determine the amount of heat extraction required to condense 33% of the moisture and, secondly, the thermal-hydraulic modelling of the conceptual device.

For the preceding calculations the input parameters as seen in Tables 1 and 2 were used.

Table 1. Thermal-hydraulic calculation inputs.

Parameter	Value	Units
Economiser		
Duct width	1.016	m
Duct height	2.7	m
Tube diameter	0.0334	m
Tube thickness	0.00277	m
Tube length per element	175.5	m
Tube number in parallel	30	-
Condenser		
Duct width	1.016	m
Duct height	2.7	m
Tube diameter	0.0214	m
Tube thickness	0.002211	m
Tube length per element	23.625	m
Tube number in parallel	30	-

Table 2. Geometric parameters of device.

Parameter	Value	Units
Gas temperature and pressure		
Inlet flue gas temperature	179	°C
Assumed inlet flue gas pressure	101.325	kPa
Gas mixture mass fractions		
CO ₂	0.1983	kg/kg
H ₂ O	0.1362	kg/kg
SO ₂	0	kg/kg
N ₂	0.5054	kg/kg
Air	0.1579	kg/kg

Using a lumped-system approach analysis and the equations stated in the literature study the parameters were calculated for the two sections, namely the economiser section and the condenser section. For the economiser section the flue gas is cooled by direct contact heat transfer. This is done because to condense the water vapour out of the flue gas the temperature of the gas mixture must first be cooled to the dew point temperature of the water vapour at its partial pressure in the mixture. Kay's rule is used where a gas mixture is treated as a pseudo substance with defined critical temperatures and pressures. The results for the calculation are seen in Table 3.

After the amount of heat stated in Table 3 is extracted from the flue gas stream any further cooling will cause condensation of the water vapour out of the gas mixture. The calculation for this section is shown in Table 4.

From these calculations the required heat transfer rates and the inlet/outlet temperatures for the gas side were calculated. Next these values were used to determine the size required to transport this amount of heat for each section and the corresponding heat transfer coefficients and temperature changes.

Table 3. Thermodynamic calculations for the economiser section.

Parameter	Variable	Equation	Value	Unit
Inlet ideal enthalpy of CO ₂	$h_{in,ideal,CO_2}$	Ideal gas tables	15576.1	kJ/kmol
Inlet ideal enthalpy of H ₂ O	h_{in,g,H_2O}	Steam and water tables	50028.9	kJ/kmol
Inlet ideal enthalpy of N ₂	$h_{in,ideal,N_2}$	Ideal gas tables	13168.2	kJ/kmol
Inlet ideal enthalpy of air	$h_{in,ideal,air}$	Ideal gas tables	13152.3	kJ/kmol
Outlet ideal enthalpy of CO ₂	$h_{out,ideal,CO_2}$	Ideal gas tables	10749.8	kJ/kmol
Outlet ideal enthalpy of H ₂ O	h_{out,g,H_2O}	Steam and water tables	47056.7	kJ/kmol
Outlet ideal enthalpy of N ₂	$h_{out,ideal,N_2}$	Ideal gas tables	9731.48	kJ/kmol
Outlet ideal enthalpy of air	$h_{out,ideal,air}$	Ideal gas tables	9704.9	kJ/kmol
Gas mixture critical temperature	$T'_{cr,m}$	$\sum y_i * T_{cr,i}$	259.858	K
Gas mixture critical pressure	$P'_{cr,m}$	$\sum y_i * P_{cr,i}$	7904.93	kPa
Reduced inlet temperature	$T_{R,inlet}$	$T_{m,inlet}/T'_{cr,m}$	1.73999	
Reduced outlet temperature	$T_{R,outlet}$	$T_{m,outlet}/T'_{cr,m}$	1.28771	
Reduced pressure	P_R	$P_m/P'_{cr,m}$	0.01282	
Inlet compressibility factor	$Z_{h,inlet,m}$	Nelson-Obert chart	1	
Outlet compressibility factor	$Z_{h,outlet,m}$	Nelson-Obert chart	1	
Inlet mixture ideal enthalpy	$h_{inlet,ideal,m}$	$\sum y_i * h_i$	21269.5	kJ/kmol
Outlet mixture ideal enthalpy	$h_{outlet,ideal,m}$	$\sum y_i * h_i$	17754.3	kJ/kmol
Heat per kmol extracted	q_{out}	$\frac{h_{in,ideal} - h_{out,ideal} - R_u T_{cr} (Z_{h,in} - Z_{h,out})}{M_m}$	3515.25	kJ/kmol
Molecular weight of mixture	M_m	$\sum y_i * M_i$	28.0642	kg/kmol
Heat per kg extracted		q_{out}/M_m	125.257	kJ/kg
Heat extraction	Q_{out}	$(q_{out}/M_m) * m_{flue\ gas}$	1277.51	kW

The heat transfer coefficient correlations for the internal, external tube bank and condensation flows shown in the literature study were utilised to calculate the average values across the economiser and condenser section. The fluid properties were evaluated at the bulk fluid temperature. The calculated results that is used in the SCFD simulation are the forced convection heat transfer coefficient over the tube bank for the economiser section (167.285 W/m²), minimum film condensation heat transfer coefficient for the condenser section (18843.5795 W/m²) and the condensation flux (0.013643 kg/m²s). These values are then implemented as constants in the numerical model.

The condensing economiser consists of a duct and U-tube bank, with cold water flowing inside the tubes and the hot flue gas flowing across the bank. The discretisation of this heat exchanger system is seen in Figure 1. Each vertical element, in turn, consists of two one-dimensional control volumes, the hot flue gas and cold water stream labeled 1,2,3,...,i,...,N. The energy equation for steady-state conditions is given by:

$$0 = \dot{m}H_{0,i} - \dot{m}H_{0,e} + Pq''\Delta s + -g(\dot{m}z_e - \dot{m}z_i) \quad (15)$$

H_0 is the total enthalpy, which is the state enthalpy plus the kinetic energy of the control volume for the hot and cold streams. Equation (15) is applied to each of the three separate control volumes. In applying the conservation of energy, however, a number of assumptions are needed:

- Flue gas flows uniformly across the tube banks.
- The condensate drops are immediately removed from the control volume on formation (no evaporative heat transfer between the flue gas and the droplet as it falls).
- Pressure drop across the tube banks are negligible.

- Constant film condensation heat transfer coefficient (heat transfer rate is limited by the internal forced convection heat transfer coefficient due to $h_{film} \gg h_{int\ conv}$).
- Constant external forced tube bank heat transfer coefficient for the economiser section.

Table 4. Thermodynamic calculations for condenser section.

Parameter	Variable	Equation	Value	unit
Inlet conditions				
CO ₂ inlet mass flow rate	$m_{CO_2,inlet}$	$mf_{CO_2} * m_{flue\ gas}$	2.022913	kg/s
Air inlet mass flow rate	$m_{Air,inlet}$	$mf_{Air} * m_{flue\ gas}$	1.610586	kg/s
N ₂ inlet mass flow rate	$m_{N_2,inlet}$	$mf_{N_2} * m_{flue\ gas}$	5.15493	kg/s
Water inlet mass flow rate	$m_{H_2O,inlet}$	$(mf_{H_2O}) * m_{flue\ gas}$	1.389558	kg/s
Inlet mixture ideal enthalpy	$h_{inlet,ideal,m}$	$\sum y_i * h_i$	17754.28	kJ/kmol
Molecular weight of mixture	M_m	$\sum y_i * M_i$	28.06424	kg/kmol
Inlet enthalpy per kg	$h_{inlet,m}$	$h_{inlet,m} / M_m$	632.6301	kJ/kg
Energy inlet	$E_{in,flue\ gas}$	$h_{inlet,m} * m_{fluegas,inlet}$	6438.902	kW
Outlet parameters				
Molecular mass of dry gas	M_{gas}	$\sum mf_{i,drygas} * M_i$	31.95495	kg/kmol
Partial pressure of water at exit	$P_{water\ vap\ out}$	Steam tables	15.763	kPa
Water vapour mass/kg flue	$m_{water\ per\ fg}$	$\phi * 18.02 * P_{water} / M_{drygas} * (P_{total} - P_{water})$	0.10389	kg/kg flue
Water vapour in flue gas outlet	$m_{water\ fg\ out}$	$m_{water\ per\ fluegas} * (m_{CO_2,inlet} + m_{N_2,inlet} + m_{Air,inlet})$	0.913032	kg/s
Ideal enthalpy of CO ₂	$h_{in,ideal,CO_2}$	Ideal gas tables	240.1727	kJ/kg
Ideal enthalpy of H ₂ O	h_{in,H_2O}	Steam tables	2591.3	kJ/kg
Ideal enthalpy of N ₂	$h_{in,ideal,N_2}$	Ideal gas tables	341.6193	kJ/kg
Ideal enthalpy of air	$h_{in,ideal,air}$	Ideal gas tables	330.34	kJ/kg
Energy outlet in flue gas	$E_{out,flue\ gas}$	$\sum mfi * hi * m_{fluegas,out}$	5144.852	kW
Enthalpy of outlet condensate	h_{cond}	Steam tables	209.34	kJ/kg
Condensate mass flow rate	m_{cond}	$(mf_{H_2O}) * m_{flue\ gas,in} - m_{water\ fluegas\ out}$	0.476526	kg/s
Energy out in condensate	$E_{out,cond}$	$h_{cond} * m_{cond}$	99.75606	kW
Heat transfer out of CV	Q_{out}	$E_{in,fluegas} - E_{out,cond} - E_{out,fluegas}$	1194.293	kW

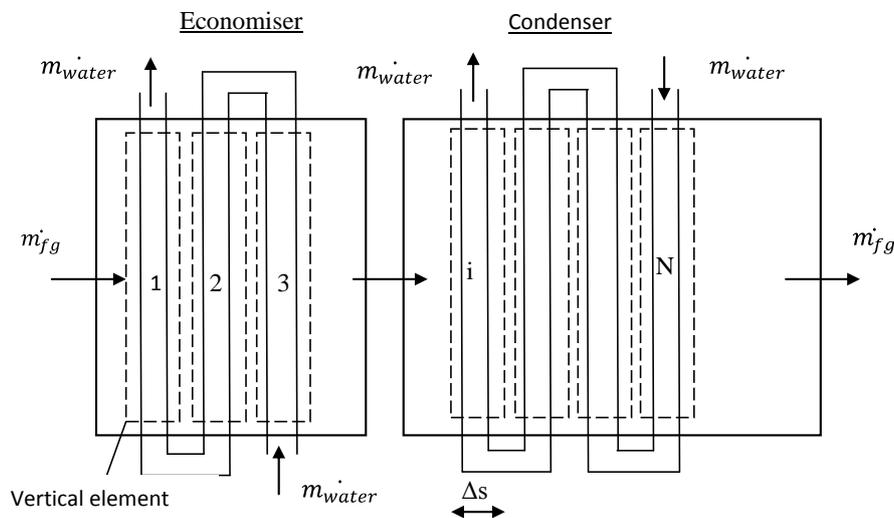


Figure 1. Discretisation of heat exchanger device.

Applying the conservation of energy, as given by (15), to each of the two control volumes per vertical element is shown in Figure 2.

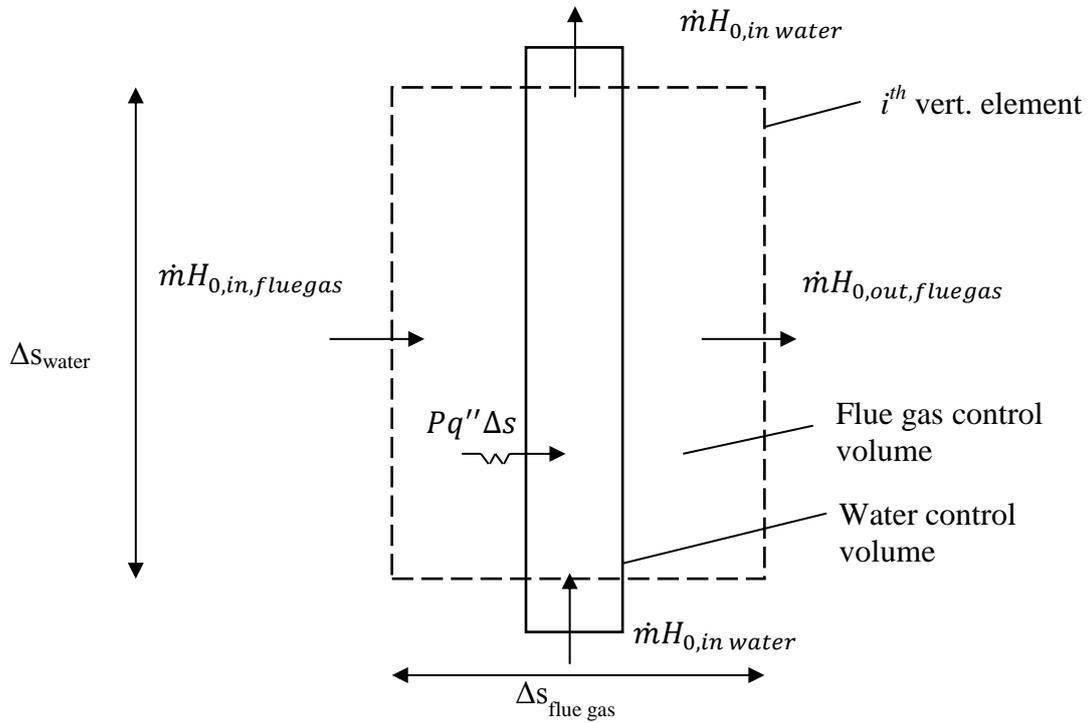


Figure 2. Depiction of two control volumes constituting the *i*th element.

The thermal-hydraulic model (Figure 3) was created using the commercial SCFD package Flownex (2013). To simulate the removal of the condensate out of the flue gas stream a small computer program was created which discharges the condensate mass from the flue gas flow rate between the five elements (Figure 3).

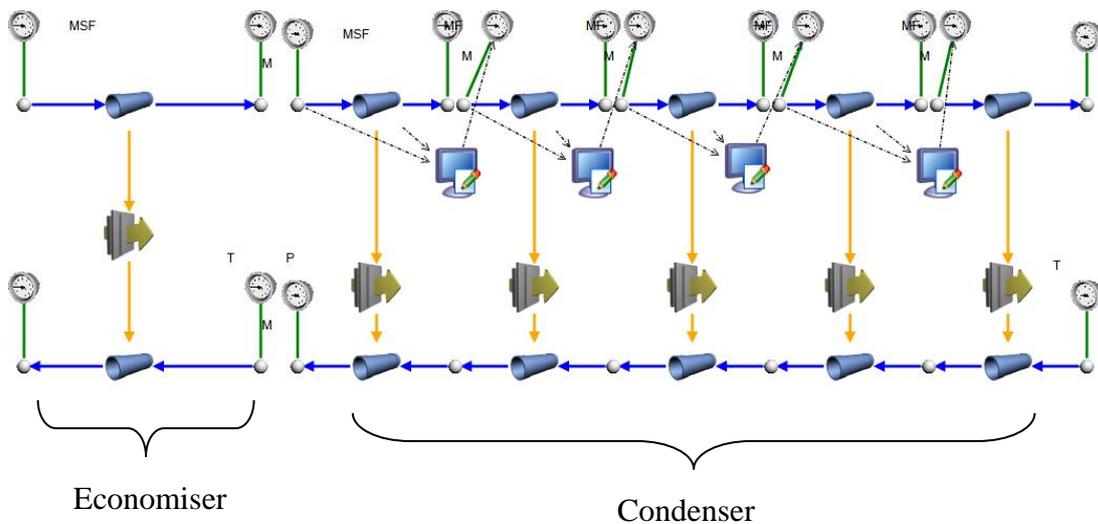


Figure 3. Condensing economiser Flownex thermal-hydraulic model.

The results of the thermal model which was set up for a Microgen boiler burning biomass intended to condense 33% of the flue gas moisture is shown in Table 5. It is seen that the required amount of heat calculated in the thermodynamic calculations is extracted out of the system, and thus 33% moisture. Further, it is seen that 7.15 kg/s water at 111.179°C is generated, which can be used in factory processes. Figures 4 and 5 depict the temperature distributions for the flue gas and the water as a function of the length from the inlet of each section.

Table 5. Thermal-hydraulic model results.

Parameter	Value	Unit
Economiser section		
Water inlet temperature	58.6707	°C
Water outlet temperature	111.179	°C
Fluegas inlet temperature	179	°C
Fluegas outlet temperature	63.2035	°C
Maximum fluegas velocity	14.248	m/s
Maximum water velocity	0.23452	m/s
Gas side convection heat transfer coef.	163.651	W/m ² .K
Water side convection heat transfer coef.	1715.6	W/m ² .K
Total heat transfer rate	1344.11	kW
Condenser section		
Water inlet temperature	20	°C
Water outlet temperature	58.6707	°C
Fluegas inlet temperature	63.2035	°C
Fluegas outlet temperature	51.9355	°C
Water massflow rate	7.15	kg/s
Inlet fluegas massflow rate	10.199	kg/s
Outlet fluegas massflow rate	9.811	kg/s
Gas side heat transfer coef.	31400	W/m ² .K
Water side convection heat transfer coef.	4108.32	W/m ² .K
Total heat transfer rate	1156.17	kW

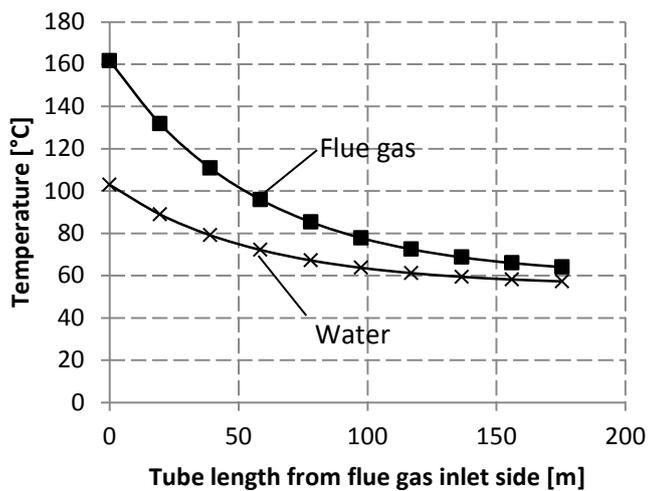


Figure 4. Economiser section temperature distribution.

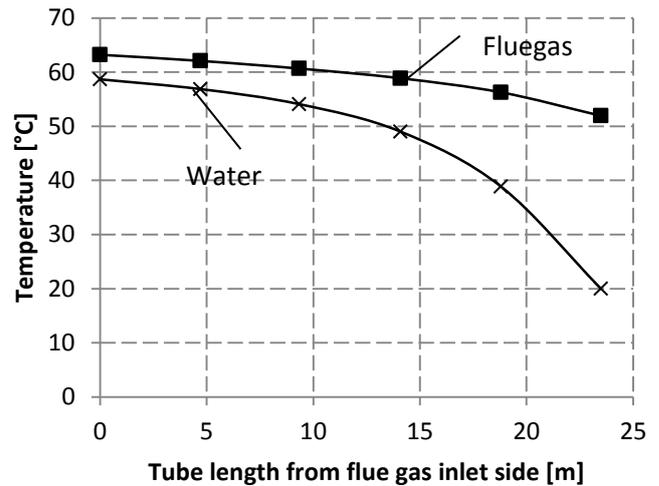


Figure 5. Condenser section temperature distribution.

Recommendations and Conclusions

From the results it can be seen that, for 30 transverse tubes with an outside diameters of 33.4 and 21.4 mm (economiser and condenser section respectively), a total number of 73.75 passes (2.7 m per pass) is required to remove 34% of the moisture in the flue gas. This gives efficiency on GCV of 80.65% and on NCV of 93.91% which is a 5.6% increase in efficiency compared to a system without a condensing economiser. By removing the moisture in the flue gas an extra 2.4 MW energy is transferred from the flue gas to water, with a final temperature of about 115°C. From the preliminary analysis, it can be deduced that flue gas condensation is definitely a viable waste heat recovery system - if a use can be found for the generated low temperature energy in the process.

Future work will focus on minimising the effective heat transfer surface by unlocking the potential in latent heat energy transfer (boiling and condensation) by building a test rig and modelling a waste heat recovery device (a submerged single pool boiling condensation thermosyphon (SSPBCT)) prototype.

REFERENCES

- Boles Y and Cengel M (2006). *Thermodynamics and Engineering Approach*. McGraw Hill, NY, New York, USA.
- Flownex (2013). *Flownex Theory Manual*. M-TEch Industrial, Johannesburg, South Africa.
- Mills AF (1995). *Heat Transfer*. 2nd Edition, Prentice Hall.
- Numrich R (2010). Filmwise condensation of pure vapors. In: *VDI Heat Atlas*. Springer, Paderborn, Germany.
- Schlunder EU (2010). Film condensation of binary mixtures with or without inert gas. In: *VDI Heat Atlas*. Springer, Germany.