Techno-economic study of a heat pump enhanced flue gas heat recovery for biomass boilers

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ABSTRACT

An active condensation system for the heat recovery of biomass boilers is evaluated. The active condensation system utilizes the flue gas enthalpy exiting the boiler by combining a quench and a compression heat pump. The system is modelled by mass and energy balances. This study evaluates the operating costs, primary energy efficiency and greenhouse gas emissions on an Austrian data basis for four test cases. Two pellet boilers (10 kW and 100 kW) and two wood chip boilers (100 kW and 10 MW) are considered. The economic analysis shows a decrease in operating costs between 2% and 13%. Meanwhile the primary energy efficiency is increased by 3–21%. The greenhouse gas emissions in CO2 equivalents are calculated to 15.3–27.9 kg MWh\(^{-1}\) based on an Austrian electricity mix. The payback time is evaluated on a net present value (NPV) method, showing a payback time of 2–12 years for the 10 MW wood chip test case.

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1. Introduction

1.1. Background

In Europe, biomass is the dominating renewable source for residential heating and hot water production. Today 80% of the total energy consumption of households in the EU-27 is required for heating (67%) and hot water provision (13%) [1]. During the last decade, biomass boilers have made a major comeback due to increasing prices for fossil fuels and the renewable character of the fuel. In particular the introduction of automatic boilers based on pellet and wood chips has made biomass boilers convenient to use and thus a low CO\(_2\) alternative to fossil fuelled ones. In fact, from 2004 to 2010 biomass boiler sales increased by 84% in the EU [2]. In 2007 the total number of biomass boilers installed is estimated at 8 million. For example, in Austria 10,000 pellet, 6000 log wood, and 4000...
wood chip boilers with a nominal heat output up to 100 kW and 700 biomass boilers with a nominal heat output >100 kW were installed in 2011 [3].

Today small-scale biomass boilers for heating and hot water provision are highly efficient. State of the art boilers reach efficiencies of 85%–89% (pellets) and 73%–81% (wood chips) under testing conditions based on the gross calorific value [4]. Nevertheless these boilers do have losses that mainly originate from the thermal energy of the flue gas, which leaves the boiler at temperatures of 70–200 °C, depending on the boiler technology. These losses can be divided into the sensible heat of the flue gas and the latent heat of the water vapour in the flue gas. The sensible heat losses depend nearly linearly on the temperature of the flue gas. In contrast, the latent heat can only be recovered if the flue gas is cooled down below the dew temperature of the water vapour in the flue gas.

In the field of oil and especially gas burners, condensing heat exchangers are a state of the art technology. Energy efficiency enhancements of 10–14% for gas and 5–7% for oil burners based on the net calorific value (NCV) are achievable [5]. For an application of these heat exchangers in biomass boilers corrosion and fouling poses a challenge. Some condensing heat exchangers for small scale biomass boilers exist [6]. These heat exchangers are either integrated into commercially available boilers or can be retrofitted to existing ones. Ceramics, carbon and stainless steel are applied as heat exchanger materials to avoid corrosion. Most of the heat exchangers need to be cleaned periodically because of fouling. Using these heat exchangers flue gas temperatures of 5–20 °C above the return flow temperature can be reached. An example is introduced in Ref. [7], which enhances the energy efficiency to 103% (+12%) based on the NCV. This improvement is valid for a return temperature of 35 °C. For biomass boilers with a nominal thermal output >100 kW a variety of flue gas condensation systems has already been installed [6]. State of the art CHP and district heating plants usually contain an internal heat recovery system including an air preheater, economizer, etc. Additionally these systems do not only focus on the energetic efficiency but also on cleaning the flue gas from fly ash, particulate matter and condensable gaseous air pollutants with scrubbers or filters. Furthermore some systems perform a devaporization. These facilities are usually individually adapted to the actual plant design (e.g. Ref. [8]). They are cost intensive and show a large space need.

Fig. 1 shows the efficiency (NCV based) of a boiler, counting only flue gas losses. As can be seen, the efficiency strongly increases if the flue gas is cooled below the dew point, which is between 40 and 60 °C, depending on the fuel moisture content. To regain the enthalpy contained in the flue gas, a heat sink below the dew temperature is needed, but this is usually not available because the return flow temperature of the heating system is too high. Integrating a heat pump with a sink temperature lower than the return flow overcomes this limitation.
1.2. **Active condensation**

Active condensation uses a heat pump to regain the low temperature flue gas enthalpy and gives it back at a higher temperature. With the active condensation system the heat output of the system (boiler and active condensation) is higher without increasing the fuel input. However, the concept is called “active”, because an additional energy input in the heat pump is necessary. The technical concept discussed contains a quench as heat recovery device and a compression heat pump. In contrast, the term “passive” condensation is used to describe the typical condensing boiler.

Heat recovery with heat pumps was already discussed for various fuels. Compression and absorption heat pumps were introduced in waste and biomass combustion plants by Ref. [9]. Fu et al. discuss the integration of an absorption heat pump with the flue gas of a natural gas driven internal combustion engine [10]. Garamella et al. introduce an absorption heat pump for heating and cooling driven by flue gas, but not in the condensing region [11]. Blarke et al. discuss the integration of a flue gas driven heat pump with a cold storage to improve the intermittency friendliness of natural gas combined heat and power plants [12]. Mostly condensing heat exchangers are used to recover the flue gas heat. Using a quench instead of a heat exchanger as condensation device has three advantages but also some drawbacks. First, in a quench one can expect less corrosion problems, because the quench water leads to permanently wet walls. In contrast, in a condensing heat exchanger, parts of the surface change between wet and dry when the flue gas dew point changes. This enhances corrosion [13]. Thus, a quench makes it possible to use cheaper materials, usually stainless steel with low alloy content. Second, a defined amount of water circulates in the quench, which does not change rapidly with the water content of the flue gas or load changes of the boiler. Therefore, a stable operation of the heat pump can be expected. Third, it allows using a water-to-water heat pump, which is available in the market. A disadvantage of the quench is the necessity to use a filter to avoid depositions in the quench nozzles. Furthermore, a pump for the additional water circuit needs to be installed, leading to increased investment and operating costs.

Heat pumps can be based on a variety of concepts, where the high-end energy is provided either by thermal energy at high temperatures or by electricity. The former can be adsorption or absorption machines, while the latter is a compression-based machine. Absorption and adsorption devices would be interesting, because the high temperature heat is already available from the boiler. However, heat-driven heat pumps have high investment costs and high space needs. Since our aim is to design a device that is applicable to boilers of different sizes (10 kW–10 MW) we use a standard compression heat pump.

Existing biomass fuelled active condensation devices can be found in industrial boilers using absorption heat pumps (e.g. Ref. [14]). For smaller devices research is done concerning thermal heat pumps [15]. Also economic evaluations were made which compare passive condensation to active condensation [16,17]. The present study extends the evaluation to compression heat pumps.

1.3. **Objective**

Previous work on this topic was already done concerning the efficiency [18], the economics [19] and a market survey of
available components and detailed evaluation of possible fields of application [6]. The present paper is based on this work and extended by the environmental assessment.

In the present study the economic and environmental impact of a heat pump assisted heat recovery for biomass boilers is evaluated. Both pellet and wood chip boilers are considered. The system containing a biomass boiler, a quench and a compression heat pump is thermodynamically modelled to calculate the heat output of the system. Depending on the flue gas temperature and return flow temperature the most cost efficient configuration is deduced. The economic analysis is performed on an Austrian cost basis. Then the primary energy and greenhouse gas emissions are calculated. Finally, a net present value method is used to evaluate the economic viability including installation costs.

2. Methodology

2.1 Thermodynamic model

A sketch of the active condensation system is presented in Fig. 2. The system is modelled to calculate the total heat output \( Q_{\text{tot}} = Q_b + Q_{\text{hp}} \), with \( Q_b \) and \( Q_{\text{hp}} \) being the heat output of the boiler and the heat pump respectively. The parameters used during the study are indicated in Table 1.

2.1.1. Boiler and gas model

A basic combustion calculation was used to determine the chemical composition of the flue gas. Boie formula [4] was used to estimate the NCV as a function of the fuel composition. A dry fuel composition with 50% carbon, 44% oxygen and 6% hydrogen was assumed. GCV \( J/\text{kg}_{\text{dry}} \) is calculated from the NCV via \( \text{GCV} = \text{NCV} + (2.44 \times 10^6 (\text{g}_{\text{H}_{2}O} + \text{g}_{\text{H}_{2}O})) \) with the fuel moisture content \( \text{g}_{\text{H}_{2}O} \) from Table 1.

Knowing the GCV and the excess air ratio, the flue gas mass flow \( m_{\text{fg}} \) can be calculated. Defining a flue gas outlet temperature \( T_{\text{fg}} \) results in a heat output of the boiler \( Q_b \) of

\[
\dot{Q}_b = \dot{m}_{\text{fg}} \text{GCV} - \dot{m}_{\text{fg}} x_{\text{fg}} (T_{\text{fg}})
\]

This calculation neglects other losses to the ambient except the outgoing flue gas enthalpy. The heat transfer in the boiler itself is not modelled. Therefore no feedbacks due to the higher return temperature are considered. Since boilers usually have an internal loop to keep water temperatures always above minimum 50 °C (pellets) or 60 °C (wood chips) to avoid the dew point, this limitation does not affect the efficiency for return temperatures lower than this limit. At return flow temperatures higher than this limit, the effect on the boiler is still very low, because the water temperature only changes marginally. Air humidity is neglected. Both fuel and air are assumed to have 25 °C at the inlet. All gas components are modelled as ideal gases, water is the only component for which condensation is considered. The enthalpies, heat capacities etc. are taken from Ref. [20]. The enthalpy of the dry flue gas is calculated with a constant specific heat capacity, which is averaged over the temperature range \((0-120 \, ^\circ\text{C})\) for the specific flue gas composition. Pressure is assumed to be standard pressure.

2.1.2. Quench

In a quench cold water droplets are injected into hot flue gas. Due to the high surface area of the water droplets a fast energy and mass transfer between water and flue gas takes place. As a result the flue gas is cooled down while the water droplets are heated up. Depending on the temperature of both fluids and the humidity of the flue gas, this leads to evaporation or condensation of water and thereby to an enthalpy exchange. Our aim is to condense the water vapour of the flue gas. We assume to get two output fluids at thermodynamic equilibrium: liquid water and flue gas with 100% relative humidity at the same temperature. The equilibrium assumption is valid if the residence time of the fluids is long enough. Thus, the incoming flows are flue gas which consists of the dry mass flow rate \( \dot{m}_{\text{fg}} \) with a water content \( x_{\text{fg}} \) at the temperature \( T_{\text{fg}} \) and liquid water with mass flow \( \dot{m}_{\text{cw,hot}} \) at the temperature \( T_{\text{cw,hot}} \). The outgoing flows are flue gas which consists of the same dry mass flow rate \( \dot{m}_{\text{fg}} \) but now with the water content \( x_{\text{fg}}^{\text{sat}} \), and liquid water with mass flow \( \dot{m}_{\text{cw,cold}} \). Both outgoing fluids are at thermodynamic equilibrium at the same outlet temperature \( T_q \).

The heat and mass transfer is described with two balance equations. The inlet and outlet fluids fulfil a water mass balance

\[
\dot{m}_{\text{fg}} x_{\text{fg}} + \dot{m}_{\text{cw,cold}} = \dot{m}_{\text{fg}} x_{\text{fg}}^{\text{sat}} + \dot{m}_{\text{cw,hot}}
\]

and an energy balance

\[
\dot{m}_{\text{fg}} (h_{\text{fg}}(T_{\text{fg}}) + x_{\text{fg}} h_{\text{fg,0}}(T_{\text{fg}})) + \dot{m}_{\text{cw,cold}} h_{\text{fw,cold}}(T_{\text{cw,cold}})
\]

\[
= \dot{m}_{\text{fg}} (h_{\text{fg}}(T_{\text{fg}}) + x_{\text{fg}}^{\text{sat}} h_{\text{fg,0}}(T_{\text{fg}})) + \dot{m}_{\text{cw,hot}} h_{\text{fw,hot}}(T_q)
\]

where \( h \) indicates the specific enthalpy.

2.1.3. Heat pump

The heat pump transfers heat from the water circuit leaving the quench to the higher temperature return flow by an inverted Carnot cycle. The process is driven by electric energy, which is also converted to thermal energy. Neglecting losses to the ambient, the energy balance for the heat pump yields

\[
\dot{Q}_{\text{hp}} = \dot{Q}_{\text{cw}} + P_{\text{hp}}
\]

where \( \dot{Q}_{\text{hp}}, \dot{Q}_{\text{cw}} \) and \( P_{\text{hp}} \), describe the heat flow to the return flow, the heat flow from the water circuit, and the electric power respectively. Thus, the heat pump cools down the water circuit and heats up the return flow. The temperature change of the return flow is calculated via

\[
\dot{Q}_{\text{hp}} = \dot{m}_{\text{fg}} (h_{\text{fg},0}(T_{\text{bw}}) - h_{\text{fg},0}(T_{\text{bw}})) \quad \text{with} \quad T_{\text{bw}}
\]
Inflation rates for economic analysis.

<table>
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<tr>
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<tr>
<td>Fuel</td>
<td>4 or 5</td>
</tr>
<tr>
<td>Electricity</td>
<td>4 or 5</td>
</tr>
<tr>
<td>Interest rate</td>
<td>4 or 10</td>
</tr>
</tbody>
</table>
combination of a boiler in non-condensing mode with a direct electric heating is used for the purpose of comparison in the NPV analysis. Boiler plus direct electric heating would be the case if the actual boiler is undersized and the additional heat is produced by electricity. Then, the additional heat output of the active condensation system replaces the direct electric heating.

3. Results and discussion

The integration of active condensation into a biomass boiler was evaluated for different flue gas and return flow temperatures. Four different test cases (2× pellets and 2× wood chips as described in Section 2.2) were analysed. A discussion concerning the operating costs, primary energy demand and greenhouse gas emissions is given below for all flue gas and return flow temperatures. Additionally an economic viability based on a net present value is carried out.

The heat output increases for all test cases due to the integration of the heat pump. In the case of pellets the increase is 7–10% for low flue gas temperatures and 13–18% for high flue gas temperatures. For wood chips the increase of the heat output is 8–27%, which is higher than the increase of the heat output for pellets. The heat pump needs an electric input of 1–5% of the heat output. The calculated COP varies between 4.3 and 7.0 for pellets and 5.5 to 7.6 for wood chips. At one typical operating point (120 °C flue gas and 50 °C return flow temperature) the detailed results of the model are given in Table 5. Fig. 4 shows a zoom of the pinch analysis in the low temperature regime with and without heat pump. It can be seen that with the heat pump also the low temperature heat of the flue gas is utilized.

3.1. Operating costs

Fig. 5 shows the operating costs for the operation of pellet boilers (both cases) with and without active condensation.

![Fig. 3 — Operating costs depending on the quench outlet temperature and corresponding coefficient of performance (COP) of the heat pump. System at 120 °C flue gas and 50 °C return flow temperature.](image)

![Fig. 4 — Pinch analysis for the 10 kW pellet case at 120 °C flue gas and 50 °C return flow temperature. Top: without heat pump. Bottom: with heat pump. Dash-dotted: hot composite curve, solid: cold composite curve. (Flue gas is shown with a maximum of 200 °C to enable a detailed view on the low temperature regime).](image)
Comparing the results one can see the strong dependence on both flue gas and return flow temperature. Through active condensation a cost reduction of 5–10% can be gained at a flue gas temperature of 200 °C, and 2–5% at a flue gas temperature of 70 °C. At low return flow temperatures the operating costs can be decreased more than for high return flow temperatures, because the temperature lift in the heat pump is lower and thus the COP higher. For instance at 120 °C flue gas outlet temperature the heat pump outlet temperatures are 23 °C on the cold and 37 °C on the hot side at 35 °C return flow temperature compared to 39 °C and 67 °C for cold and hot heat pump outlet at 65 °C return flow temperature. Thus the temperature difference increases from 14 °C to 28 °C.

It can be noted that passive condensation would lead to lower operating costs than active condensation at 200 °C flue gas temperature but not for lower flue gas temperatures.

Fig. 6(a) shows the operating costs for the 100 kW wood chips test case, while Fig. 6(b) represents the 10 MW test case. In the first case a reduction of the costs by 2–7% can be achieved through active condensation, compared to 5–13% in the second case. As opposed to the operating costs for pellets, which decrease strongly with decreasing return flow temperature, in the case of wood chips the decrease is more flat (Fig. 6(b)) or shows a minimum at a specific return flow temperature (Fig. 6(a)). This difference is caused by the higher flue gas water content which leads to a condensation of water at already higher temperatures (see Fig. 1). The highest heat recovery takes place between 50 °C and 60 °C.

The costs for active condensation for the 10 MW wood chip test case are lower than for passive condensation, if the return flow temperature is high, but higher if the return flow temperature is low. The trend is quite similar for the 100 kW wood chip case, but, due to the higher electricity to fuel cost ratio, the cost reduction comparing active condensation to the boiler only system is lower. Therefore the passive condensation system is the most cost efficient system in a broader range of operation than for the 10 MW wood chip test case.

3.2. Primary energy

The primary energy based on Austrian and European electricity generation is calculated as indicated in subsection 2.2.2. The primary energy efficiency is higher for active condensation than for the boiler only system for all evaluated temperatures and it increases with lower return flow temperatures. Fig. 7 shows the primary energy efficiency for all test cases. The wood chip 100 kW case is very similar to the wood chip 10 MW case, but it shows slightly higher efficiencies for high return flow temperatures (up to 3%).

It is interesting to note that for the pellets case the primary energy efficiency is higher for active condensation than for passive condensation with both the Austrian and European electricity mixture. In the wood chips cases the primary energy efficiency is only lower for active condensation than for passive condensation at low return flow temperatures and only for the European electricity mixture.

3.3. Greenhouse gas emissions

In contrast to the primary energy efficiency the GHG emissions are higher for active condensation than for the boiler.
only system (see Fig. 8), because of the partly fossil production of electricity. The emissions in CO₂ equivalents for the boiler only system are 21.9–23.8 kg MWh⁻¹ for the pellets cases and 13.5–14.9 kg MWh⁻¹ for the wood chips cases. With the Austrian electricity mix the greenhouse gas emissions for active condensation are 23.2–27.8 kg MWh⁻¹ for the pellets cases, and 15.2–23.8 kg MWh⁻¹ for the wood chips cases. The higher relative increase for wood chips is due to the lower GHG emissions during the production of wood chips compared to pellets. The European electricity mix leads to greenhouse gas emissions of 17.7–34.4 kg MWh⁻¹ for all cases.

3.4. Payback time

As was shown, the integration of an active condensation device lowers the operating costs between 2 and 13%. Nevertheless, the installation is only economically viable, if the installation and maintenance costs equal the gain through the lower operating costs. Fig. 9 shows the net present value for the 10 MW wood chip boiler at 120 °C flue gas and 50 °C return flow temperature. The costs are calculated for different inflation and interest rates. Since the yearly savings are approx. 100 k€ and the estimated installation costs for an active condensation system in this size are 400 k€; a payback time around 4 years can be accomplished.

Fig. 10 shows the payback time for all different flue gas and return flow temperatures for the 10 MW wood chip case. Integrating active condensation into a 10 MW wood chip boiler leads to payback times between 2 and 4 years at 200 °C flue gas temperature and up to 12 years for lower flue gas temperatures. Compared to the 10 MW boiler, all other test cases have much higher installation costs per kW. Therefore, the payback time is longer. With estimated installation costs of 15 k€, the minimum payback time for the 100 kW pellets case is 10–12 years for 4% interest rate and 14–19 years for 10% interest rate. However, these payback times can only be reached at low return flow temperatures and high flue gas temperatures. For the 100 kW wood chips case, the minimum payback time is 19–30 years for 4% interest rate (at 200 °C flue gas temperature) and higher than 38 years for all cases with 10% interest rate. Since energy costs are subject to high fluctuations, the real payback time can vary strongly. For the 10 kW pellets test case, it appears that the yearly savings are around 50 €/year, compared to installation costs of 4 k€. The lowest payback time can be reached for 35 °C return flow and 200 °C flue gas temperature, which leads to a payback time of minimum 36 years for 4% interest rate. All other temperatures lead to higher payback times. Thus the integration of active condensation is not feasible for this test case.

In contrast, if the costs are not compared to the boiler only system but to boiler plus direct electric heating a lower payback time can be reached. Then the payback time for the 100 kW boilers would be 2–8 years for all different rates. Even for the 10 kW pellet boiler the payback time is 8–28 years for 4% interest rate.

4. Conclusion

The integration of an active condensation system into biomass boilers of different fuels and sizes was evaluated. Active condensation is a heat recovery of the sensible and latent heat of the flue gas with the help of a heat pump. The integration was discussed for four test cases: 1. 10 kW pellet boiler, 2. 100 kW pellet boiler, 3. 100 kW wood chip boiler, 4. 10 MW wood chip boiler. Based on a thermodynamic model the operating costs were calculated. Depending on the flue gas
temperature after the boiler (70–200 °C) and the return flow temperature (35–65 °C) the optimal heat pump size was deduced to get the lowest operating costs. The analysis shows that the operating costs of the system with active condensation are between 2% and 13% lower than the boiler only operating costs. In the case of pellets, the highest savings are for low return flow temperatures and high flue gas temperatures. Wood chip fuelled boilers with active condensation are less dependent on the return flow temperature, but also show the increase of cost reduction for higher flue gas temperatures.

The primary energy efficiency increases by 4–21% based on the Austrian electricity mix and by 3–16% based on the European electricity mix. However, the greenhouse gas emissions in CO₂ equivalents increase to 15.3–27.9 kg MWh⁻¹ based on the Austrian electricity mix and to 17.7–34.4 kg MWh⁻¹ based on the European electricity mix compared to the boiler only system with 13.5–23.8 kg MWh⁻¹. Yet, the greenhouse gas emissions for the active condensation system are still much lower than the Austrian average of 206.7 kg MWh⁻¹.

The payback time was discussed for all different cases based on a net present value calculation. For the 10 MW boiler...
the payback time is 2–12 years compared to the boiler only system. For the case of the 100 kW boilers the payback time is approx. 10–30 years for well suited operating points, while the 10 kW pellet test case does not lead to an economically viable system. For the case of the 100 kW boilers the payback time is 27%. Thus, if an existing boiler is undersized, the costs of the active condensation system have to be compared to other kinds of auxiliary heaters. Comparing to direct electric heating leads to 2–8 years payback time for the 100 kW test cases and to payback times below 28 years for the 10 kW pellets test case at low interest rates.

Concluding, the integration of an active condensation system into biomass boilers is definitely economically viable for biomass boilers >10 MW. But also for boilers with approx 100 kW, the integration is interesting, depending on the operating points and the fuel, especially if the increase in the maximum heat output can replace existing electric heating. Further reasons for active condensation systems can be the particle emission reduction which will be studied in future research work.

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