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OPTIMISATION OF FLUE GAS CONDENSATION AT BIOMASS COMBUSTION PLANTS THROUGH INTEGRATION OF A DIRECT EVATORATION HIGH TEMPERATURE HEAT PUMP

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ABSTRACT: Within the work described a new energy recovery technology based on flue gas condensation with a direct evaporation industrial compression heat pump was developed. In a first step the flue gas condensation unit mainly consisting of the flue gas condenser / refrigerant evaporator and the compression heat pump were designed and different variants regarding system integration have been evaluated. For this purpose a design tool was developed to simulate the heat transfer from the condensing flue gas to the evaporating refrigerant. In a second step a pilot plant was set up and a number of test runs under different conditions have been carried out. The measured data showed good agreement with the data derived from the design tool. An increase in efficiency of up to 14%-points (with respect to the primary energy input, based on NCV) or up to 34%-points (with respect to the fuel energy input) respectively, compared to a conventional biomass boiler plant without heat recovery have been achieved.
Keywords: direct evaporation heat pump, flue gas condensation, heat recovery, efficiency, biomass

1 INTRODUCTION

Flue gas condensation systems in biomass combustion plants are a well-known technology in the field of heat recovery, dust precipitation and plume removal from the flue gas which typically can be reasonably applied in plants with biomass boiler capacities of 1 MW_{th} and above.

Flue gas condensation is the more efficient the lower the flue gas can be cooled below its water dew point in order to use its sensible and in particular the latent heat (see Fig. 1 for the potential increase in thermal efficiency (heat output related to fuel energy input) by further cooling of the flue gas for different biomass fuel water contents). For practical applications the usage of heat at a low temperature level is often limited because the temperatures of the return flows from heat sinks (e.g. district heating networks: 50 up to 60°C), which are used to cool the flue gas, are too high in many cases in order to reach temperatures significantly below the water dew point which typically is also in the range of 50 to 60°C.

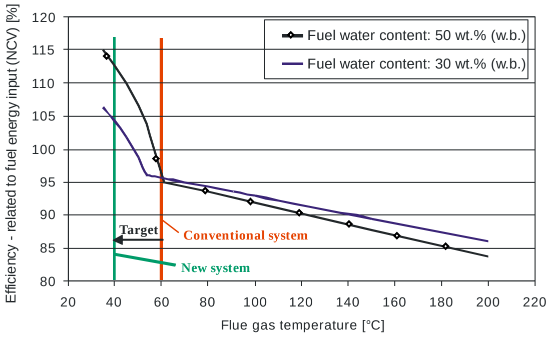

Figure 1: Potential for increase in efficiency by reducing the flue gas temperature for different biomass fuel water contents; a reduction of the flue gas temperature from 60 to 40°C leads to an increase of efficiency of about 10 to 20%-points

The target of this project was to utilize the heat of condensation in the flue gas condensation plant efficiently over the whole year through implementation of an industrial compression heat pump (HP) in a flue gas condensation unit. The heat recovered from the flue gas will be raised from its low temperature level of e.g. 40°C to a temperature level which can be used directly in the return flow of conventional district heating networks or process heat supply systems. This shall lead to an efficiency of the flue gas condensation plant twice as high and the respective savings in emissions (reduction of the necessary fuel input and the equivalent emissions by about 10 to 20%) compared to state-of-the-art flue gas condensation technologies under central European conditions. In addition the input of electric energy for the plume removal can be reduced which leads to a further increase in the efficiency of the overall plant.

To achieve this the development of a flue gas condenser suitable for the refrigerant of the heat pump was necessary. The direct integration of the heat pump into the flue gas condensation plant (flue gas condenser = refrigerant evaporator), the right choice of the refrigerant as well as the improvement of the condenser design, which is only possible in an inaccurate way with the currently available methods, were the main goals of development and optimization within this project.

For the development of the condenser calculations and simulations have been carried out in a first step while in a second step experimental measurements on a pilot plant have been performed. The developed concept included the design of the flue gas condensation plant, the heat pump and their components as well as the overall system integration, the test of the system under real-life conditions (pilot plant) as well as its ecological and economic evaluation.

2 SYSTEM INTEGRATION OF THE HEAT PUMP

During the project the integration of the new flue gas condensation technology with a heat pump into the flue gas and water system of a biomass combustion plant was investigated. The evaluation of different variants showed that there is not one standard solution, especially when retrofitting this new technology into existing plants. Fig. 2 shows one example for the integration of the new flue gas condensation technology. The heat recovery occurs in the condenser/evaporator heat exchanger of the heat pump. This variant corresponds to the setup used in the pilot plant, which will be described in section 4.1 in more detail. It is possible to implement this variant with or without the quench, also the implementation of a plume removal air pre-heater before the chimney is possible.

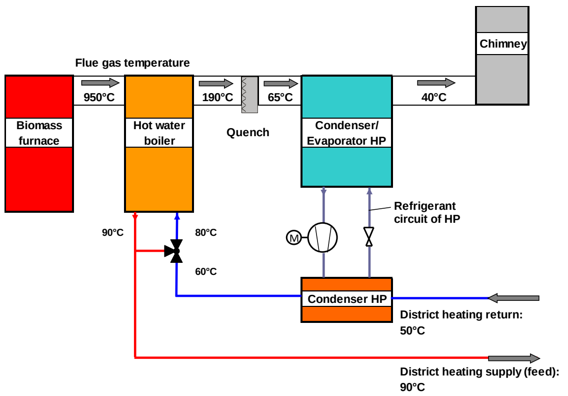

Figure 2: Example for the integration of the condensation unit and direct evaporation heat pump into the flue gas and water system of a biomass combustion plant

For the integration of the refrigerant condenser of the heat pump in the hot water system either serial or parallel integration with respect to the biomass boiler is an option. In order to achieve the highest efficiency for the system the coefficient of performance (COP = heat output of the condenser related to the electric input to the heat pump) of the heat pump must be as high as possible. The COP gets higher, the lower the temperature difference between the cold end (flue gas condensation) and the hot end (integration in the district heating network) of the heat pump gets. The integration of the heat pump in series with the biomass boiler is thus the optimal solution. To utilize the heat in this situation a small temperature difference over the heat pump is sufficient whereas for parallel integration the complete temperature difference of the district heating network between return and feed would need to be overcome.

3 HEAT EXCHANGER DESIGN TOOL

For the design of the flue gas condenser / refrigerant evaporator a tool based on heat transfer models according to the VDI heat atlas [1] was developed. In a first step the condensation on the outer surface of the heat exchanger pipes was modeled using the Nusselt theory for the calculation of condensation. This is a model for the film condensation of water vapor under presence of an inert gas or a mixture of inert gases i.e. flue gas.

In a second step the model for the evaporation of the refrigerant inside of the heat exchanger pipes was implemented in order to finally be able to describe the whole heat transfer during this process correctly.

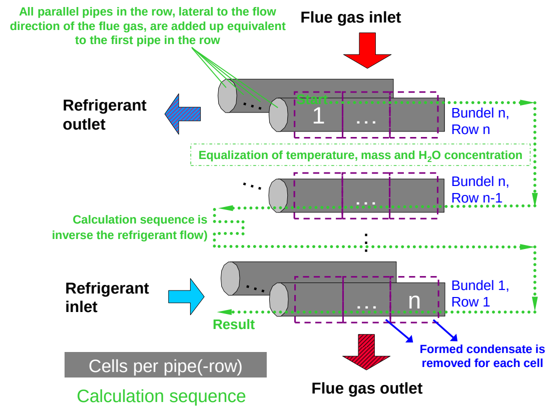

Figure 3: Cell model for the heat exchanger calculation

After definition of the geometry of the heat exchanger, the mass flow of the flue gas and the refrigerant as well as the flue gas composition and inlet temperature, the iterative calculation of the flue gas outlet and refrigerant inlet temperature was executed with an initial value for the refrigerant outlet temperature. For the calculation the heat exchanger was separated into cells (see Fig. 3) and mass and energy balances were solved for each of the cells. The condensate was removed of the mass and energy balance for every cell. After each pipe row an equalization of the temperature, mass and water vapor content of the flue gas over the pipe row occurred i.e. no cross-flow effects were considered. The order in which the cells were calculated is the inverse of the refrigerant flow. The calculation was repeated until the difference between the calculated value of the refrigerant inlet temperature and its target value was within an acceptable deviation. The results of the calculation are the outlet temperatures of both, the flue gas and the refrigerant as well as the transferred amount of heat and the total amount of condensate formed.

Before designing the heat exchanger for the pilot plant the model was validated through comparison with experimental data of conventional condensers with water as cooling medium. These values showed good agreement.

4 TEST RUNS AND RESULTS

4.1 Pilot plant

With the results of the design tool described in section 3 a pilot plant was constructed and integrated in an existing biomass boiler system. The design was based on the following conditions:

- Nominal capacity of the biomass combustion plant of 200 kW and the resulting flue gas condensation capacity
- Selection of a suitable screw-compressor for the power range of the pilot plant. This type of compressor was chosen because it is mainly used in large-scale applications (up to 1 MW heating capacity per unit) which also is the target for future demonstration plants of the new technology.
- Determination of application limits of the selected compressor regarding the refrigerant temperatures
- Use of the high temperature refrigerant R-245fa (favorable pressure levels for the high temperatures needed in this project and good safety properties)

- Water return temperature from the heat sink of about 50°C

The water outlet temperature of the refrigerant condenser (feeding temperature on the heating side of the heat pump) at the design point was defined with 60°C. This leads to a refrigerant condensation temperature of about 70°C at the design point of the heat pump. Due to the application limits of the compressor and the defined flue gas outlet temperature of 40°C, a refrigerant evaporation temperature of 30°C was defined (with 5 K superheating of the refrigerant at the evaporator outlet).

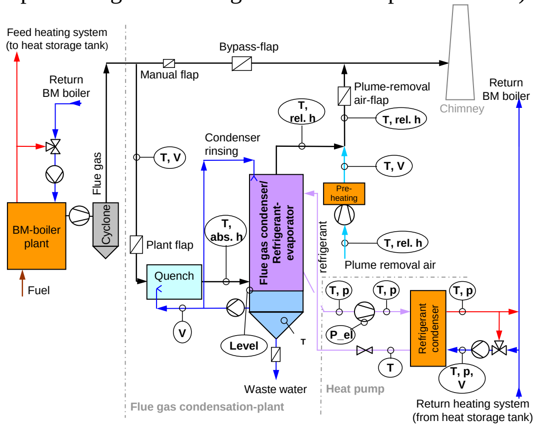

Figure 4: Simplified scheme of the pilot plant; measurement points: T…temperature, p…pressure, V…volume flow, rel. h… relative humidity, abs. h… absolute humidity, P_el…electric power

Fig. 4 shows a simplified scheme of the pilot plant including all measurement points used for the mass and energy balance calculations. Fig. 5 shows a picture of the installed pilot plant.

Figure 5: Pilot plant; (1): biomass boiler, (2): flue gas condenser, (3): heat pump

4.2 Test runs

During the period from October 2013 until April 2014 the following test runs have been carried out on the pilot plant by BIOS:

- Test runs with fuel water content of about 50%
    - Operation of the heat pump with and without ECO (the economizer loop is an internal heat recovery loop of the heat pump) at different return flow temperatures (58, 68, 78°C) at 100% power level (power level corresponds to the internal flow valve position of the compressor)
    - Variation of the flue gas volume flow (determination of the minimum admissible boiler capacity for the operation of the heat pump with 10 K of superheating and within the application limits of the compressor respectively)
        - Nominal flue gas flow (about 420 Nm³/h)
        - Reduced flue gas flow (about 250 Nm³/h)
    - Operation of the heat pump at 75% partial load of the compressor (without economizer loop)
    - Operation of the heat pump without quench
    - Operation of the heat pump with permanent rinsing of the condenser (water injection at the top of the flue gas condenser in counter flow to the flue gas) for a more effective heat exchanger cleaning
- Dust measurements (total dust and PM_1 emissions) before and after the energy recovery plant. The total dust emissions have been measured with equipment according to EN13284-1 while a Berner-type low pressure impactor (BLPI) was used for the PM_1 emissions. The flue gas was preheated for the measurements in order to avoid condensation problems.
- Repetition of relevant test runs with a biomass fuel water content of about 30%
- Optimization of the heat pump cycle

4.3 Results of the test runs

Out of the collected data of all the test runs performed 40 cases of about 1 hour of constant operation have been extracted. For these cases energy and mass balances have been calculated in order to evaluate the application limits, the efficiencies and the performance of the new flue gas condensation technology with direct evaporation heat pump.

The thermal energy transferred at the evaporator as well as the COP have been calculated based on the energy balances for the flue gas and for the water side of the heat pump unit. The difference between these calculations was an indicator for the plausibility of the data. For all the analyzed cases this difference was within an acceptable range of +/– 10%.

Fig. 6 shows the envelope (limits of operation) of the compressor used in the heat pump of the pilot plant. The envelope is characterized by the dependency of the condensation temperature of the refrigerant from its evaporation temperature. The necessary boiler capacity and electric input to the heat pump to operate within the envelope as well as the resulting COP are also shown in the diagram.

Measured data for the condensation temperature (at different flue gas flows) and COP are also shown in the chart. The evaporation temperatures were in the range from 30 to 40°C even though the envelope extending up to 55°C. The reason for this is that the achievable envelope, which only depends on the selected compressor and the used refrigerant, is limited by the actual design of the refrigerant circuit (mainly the heat exchanger surfaces).

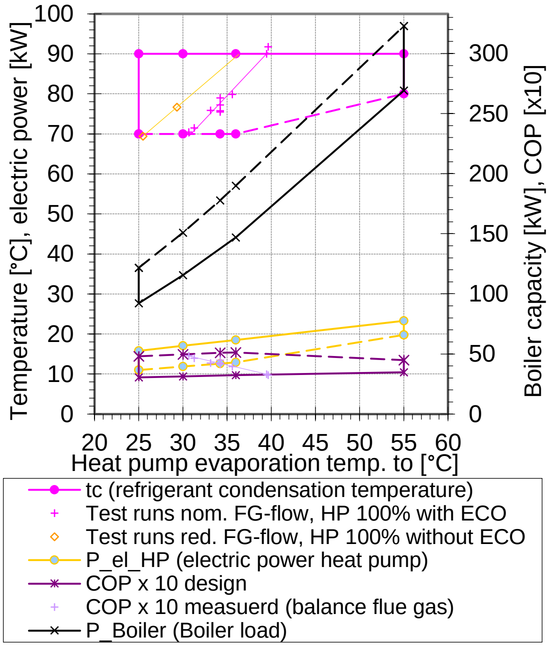

Figure 6: Envelope (limits of operation) of the compressor used in the heat pump of the pilot plant for a fuel water content of 50% and running the compressor at 100% power level as well as the measured values for condensation temperature and COP; FG…flue gas, HP…heat pump

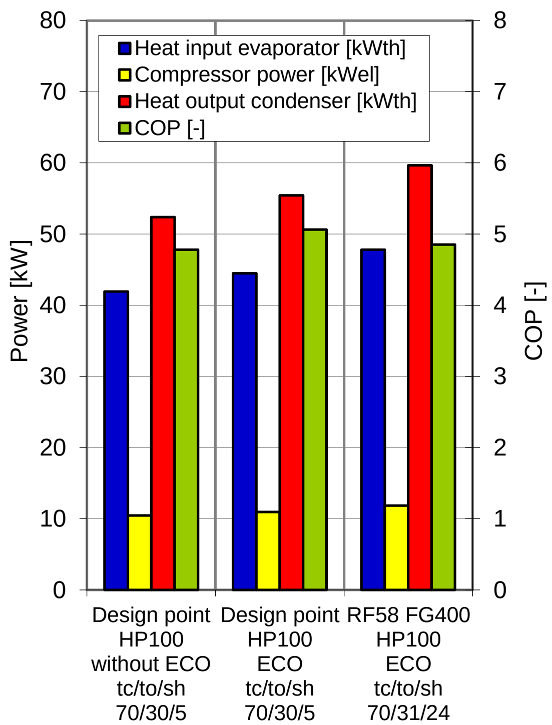

Figure 7: Measured power and COP of the heat pump at nominal load compared to the design figures; RF58 FG400 HP100 ECO…return temperature 58°C, flue gas volume flow 400 Nm³/h, power level of the compressor 100%, ECO loop active; tc…refrigerant condensation temperature in °C, to…refrigerant evaporation temperature in °C, sh…refrigerant superheating after evaporator in K

Fig. 7 shows a comparison of the heat input to the evaporator, the heat output of the condenser of the heat pump, the electric power consumed by the heat pump and the COP of a test run with 58°C return temperature at the inlet of the heat pump to the design figures with and without economizer loop.

During the test run shown in Fig. 7 the highest heat output and COP of all test runs were achieved. The test run was carried out at nominal flue gas flow (about 400 Nm³/h) and a fuel water content of 49%. The compressor was running at 100% and the economizer loop was active. The measured heat input at the evaporator of about 48 kW is above the design value of 44.5 kW but the measured COP of 4.9 is slightly below the design value of 5.1. The electric power consumption of the heat pump (11.8 kW) is slightly above the design value of 10.9 kW.

According to the heat pump supplier the efficiency of the heat pump could be further improved by optimizing the economizer loop.

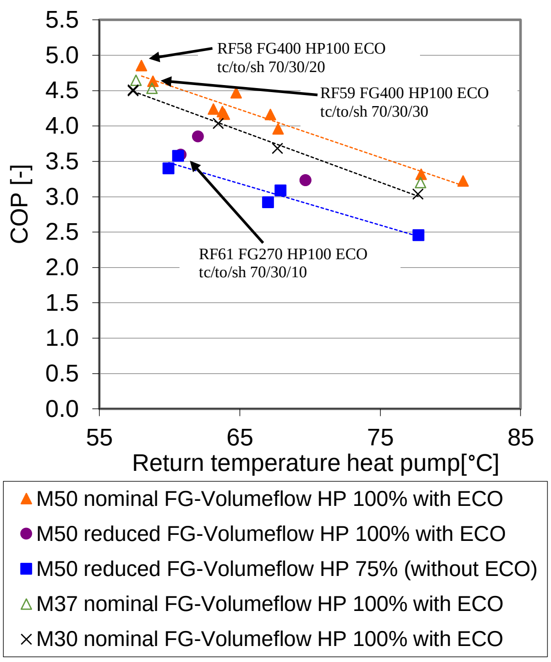

Figure 8: Measured COP values at different return temperatures, fuel water contents, flue gas volume flows and power levels of the compressor; RFxx…return temperature in °C, FGxx…flue gas flow in Nm³/h, Mxx…fuel water content in % (w.b.), HPxx…power level of the compressor in %, with ECO…economizer loop active, without ECO…economizer loop inactive, tc…refrigerant condensation temperature in °C, to…refrigerant evaporation temperature in °C, sh…refrigerant superheating after evaporator in K

Fig. 8 shows the measured values of the COP over the return temperature at different fuel water contents, flue gas flows and power levels of the compressor.

As expected the COP gets lower as the temperature difference from flue gas to hot water over the heat pump gets higher when the return temperature into the heat pump is increased. The results show a quite linear relation for the COP from 4.7 at 58°C to 3.2 at 80°C return temperature for a fuel water content of 50% at nominal load. With lower fuel water content or reduced flue gas flow (with respect to the nominal case) equivalently lower heat input at the evaporator and heat output were measured. In these conditions the compressor was operating at partial load (75% power level) which enabled the operation of the compressor within its application limits (envelope) for the lower amounts of energy available at the evaporator. However the electric power consumption of the heat pump stayed almost the same (due to reduced internal compressor efficiency), which led to lower COP values for partial load on the flue gas side at comparable return temperatures.

The partial load performance has been determined experimentally through reduction of the flue gas flow, and thus the amount of energy available at the flue gas condenser, until automatic shutdown of the heat pump occurred due to envelope monitoring. A partial load of about 65% could be achieved for a fuel water content of 50% (heat output of the heat pump of about 40 kW compared to 60 kW at nominal load). Stable operation of the heat pump was possible down to 30 kW transferred at the evaporator. The COP values for partial load were lower compared to nominal load. The determined partial load performance of the system is a big challenge for the large-scale application of this technology.

Regarding the operation of the plant with and without quench the results show no major differences i.e. the heat input to the evaporator does not change whether the quench is active or not. Without quench no rise of the refrigerant temperatures at the inlet of the compressor was measured and startup without quench was also successful. Thus the quench is not necessary from a safety point of view. During the experiments with permanent rinsing of the evaporator a reduction of the heat input at the evaporator and heat output of about 10% with respect to the normal operation with active quench were calculated. Whereas the permanent rinsing had a positive effect on the evaporation pressure of the refrigerant thus a more stable operation of the heat pump was achieved.

The total amount of dust in the flue gas at the inlet to the flue gas condensation system was 100 mg/Nm³ (related to dry flue gas and 13% O_2) on average. At the outlet of the flue gas condensation system 37 mg/Nm³ have been measured on average. This corresponds to a precipitation efficiency of about 63% (shown in Fig. 9). The separation efficiency for the coarse fly ash (particle diameter above 1 µm) amounted to 83% on average (about 76 before and 13 mg/Nm³ after the flue gas condensation plant). As expected no separation of fine dust (particle diameter below 1 µm) could be measured, as the particles follow the path lines of the flue gas and can get precipitated efficiently only if formation of water droplets in the bulk flow occurs, which normally cannot be achieved in flue gas condensation units.

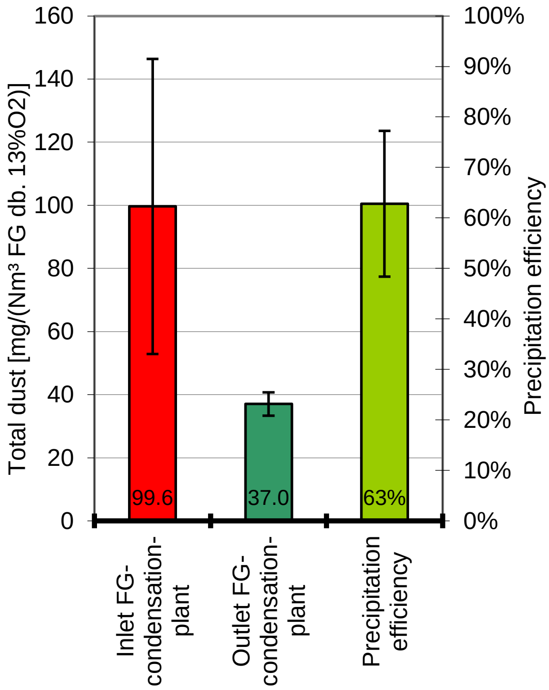

Figure 9: Measured amounts of total dust at the inlet and outlet of the flue gas condensation pilot plant as well as related precipitation efficiency; FG…flue gas

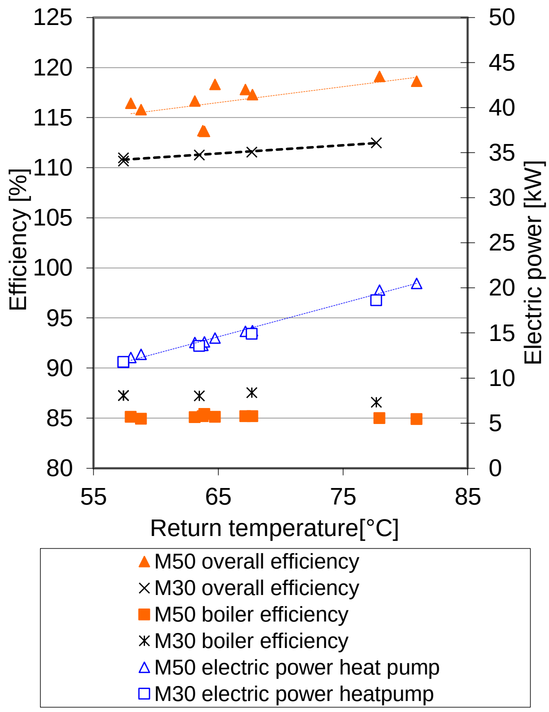

Figure 10: Calculated efficiencies (related to the net calorific value, NCV, of the biomass fuel) of the biomass boiler and the overall system, as well as electric power consumption of the heat pump over the return temperature at different fuel water contents at nominal load; Mxx…fuel water content in % (w.b.), boiler efficiency = boiler heat output / fuel power input (NCV), overall efficiency = (heat output of boiler + heat pump) / fuel power input (NCV)

Fig. 10 shows the calculated values for the efficiency of the biomass boiler and the overall plant related to the net calorific value (NCV) of the biomass fuel used.

The boiler efficiency shows no impact of the return temperature while the overall efficiency rises slightly with increasing return temperature (from about 116% at 58°C to about 119% at 81°C return temperature). The reason for this behavior is simply that at higher return temperatures the COP of the heat pump gets lower, therefore more electric energy is needed at the compressor. This electric power input contributes to the heat output of the heat pump which increases. Since the fuel power input does not change the efficiency rises. Compared to the boiler efficiency of about 85% an increase of 31 to 34%-points is possible through the heat recovery system with the integrated heat pump.

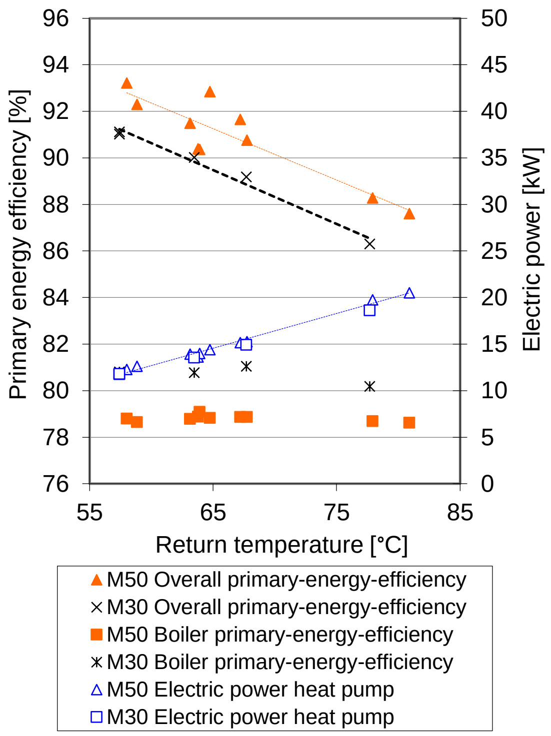

Figure 11: Calculated primary energy efficiency of the biomass boiler and the overall system as well as electric power consumption of the heat pump over the return temperature at different fuel water contents at nominal load; Mxx…fuel water content in % (w.b.), primary energy factor (PEF) used according to OIB-guideline 6 [2]: PEF biomass 1.08, PEF electric energy 2.62, boiler primary-energy-efficiency = heat output boiler / (fuel power input * PEF biomass), overall primary-energy-efficiency = (heat output of boiler + heat pump) / (fuel power input * PEF biomass + electric power heat pump * PEF electric energy)

Fig. 11 shows the calculated values for the primary energy efficiencies for the biomass boiler and the overall plant as well as the electric power consumption of the heat pump for different fuel water contents at nominal load over the return temperature. To rate the primary energy inputs primary energy factors (PEF) were used. These factors are defined as the relation between the amount of primary energy used and the amount of final energy gained. The primary energy for production, conversion, storage, transmission, distribution and all other production steps is considered, i.e. besides the energy content of the energy source the primary energy demand of the whole upstream chain is taken into account. For the calculations within this project a PEF of 1.08 for biomass and 2.62 for electric energy have been used (according to OIB-guideline 6 [2]).

The primary energy efficiency gets lower as the return temperature increases since the COP values get worse with increasing refrigerant condensation temperatures. The overall primary-energy-efficiency at nominal load and a fuel water content of 50% is about 93% at 58°C and drops to about 88% at a return temperature of 81°C. Compared to the boiler-primary-energy-efficiency of about 79% an increase of efficiency through heat recovery of 9 to 14 %-points could be achieved. For a fuel water content of 30% the boiler primary-energy-efficiency increases to about 81% while the overall primary-energy-efficiency drops by 2% because of the lower amount of recovered heat at the evaporator for equal flue gas flows (lower latent heat at the condenser due to lower fuel water content).

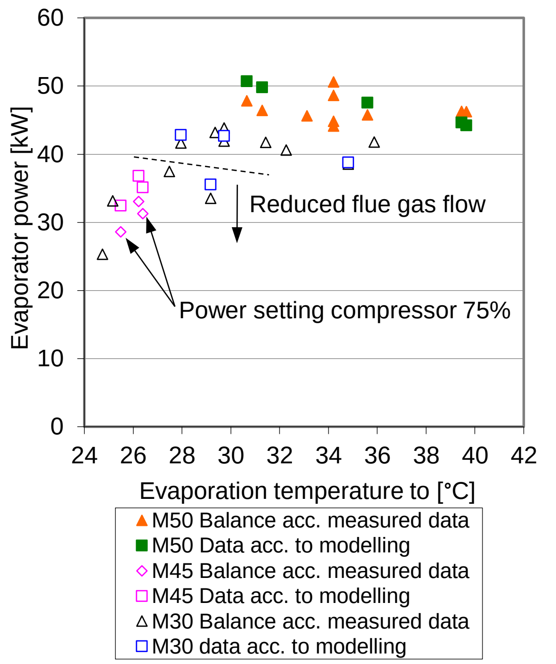

Figure 12: Comparison of the values measured during the test runs with the values calculated with the model at different return temperatures, flue gas volume flows (data above the dashed line are measured at nominal flue gas flow, data below are measured at reduced flue gas flow), compressor power level (the points indicated were measured at a compressor power level of 75% all other points were measured at 100%) and fuel water contents, Mxx…fuel water content in % (w.b.)

Fig. 12 shows a comparison of values measured during the test runs at the pilot plant with values calculated with the design tool described. The measurements cover a wide range of refrigerant evaporation temperatures (about 24 to 40°C) and fuel water contents (30, 45 and 50%). The measured values are in good agreement with the values predicted by the design tool over the whole spectrum of varied parameters (deviations in the order of typically <10%). This makes the model very useful for designing evaporators for this new flue gas condensation technology.

5 ECONOMICS

In order to be able to pre-evaluate the plant design economically, first calculations have been carried out during this project. These included calculations of costs and dynamic amortization periods. In addition sensitivity analysis have been performed in order to determine the effect of relevant parameters on the economics.

Based on the results of the test runs of the pilot plant the economic feasibility of the new flue gas condensation technology with direct evaporating heat pump was investigated and compared to the following alternative variants:

- **Variant 0:** Base scenario with a biomass boiler with a heat capacity of about 3,700 kW without heat recovery installed.
- **Variant I:** Biomass boiler plant with heat recovery based on a flue gas economizer (ECO) and the direct evaporation heat pump (as developed in this project) with a return temperature of 58°C.
- **Variant II:** Same as variant I but with a heat pump with an intermediate water circuit between the flue gas and the heat pump evaporator.
- **Variant III:** Biomass boiler plant with a flue gas economizer and a conventional flue gas condensation system with a return temperature of 58°C.

All variants were based on a biomass district heating plant with a peak load of 8,700 kW. The base load is generated by the biomass system (biomass boiler incl. heat recovery system). The remaining peak load coverage is done in all variants with an alternative heat production system (e.g. fossil fired peak load boiler) with the same annual heat production. Tab. I shows the parameters used in the calculations.

The economic calculations for the variants I to III have been carried out in comparison to variant 0 (differences in costs and savings used for cash flow calculations).

Fig. 13 shows the cumulated discounted cash flow (CDCF) for the variants I to III. Variant I has a payback time of about 3.2 years, compared to the reference variant 0, even without subsidies. Variant II has higher investment costs due to the intermediate circuit and the resulting higher difference between flue gas and evaporation temperature which leads to a longer payback time of about 4 years. The payback period for variant III with the same return temperature of 58°C is significantly higher (about 5 years) than for variant I.

Furthermore the calculations showed that the COP of the heat pump as well as the price for the biomass fuel have the strongest impact on the economic viability of the variants analyzed within the sensitivity analysis performed.

Table I: Basic data for economic calculations

Variant		0	I	II	III
Return temp. DH-network	[°C]	58	58	58	58
Flue gas temp. at boiler outlet	[°C]	180	180	180	180
Fuel water content	[%(w.b.)]	50	50	50	50
Peak load DH-network	[kW]	8,700	8,700	8,700	8,700
Heat capacity base load plant	[kW]	3,674	3,674	3,674	3,674
Boiler capacity	[kW]	3,674	2,750	2,864	3,355
Capacity flue gas ECO	[kW]	-	290	303	313
Capacity flue gas condenser	[kW]	-	507	406	5.6
Heat output heat pump	[kW]	-	634	508	-
Elec. power heat pump	[kW]	-	127	102	-
COP	[-]	-	5.0	5.0	-
Full load operating hours	[h/a]	5,000	5,000	5,000	5,000
Total invest	[k€]	133	160	148	100
Electricity price	[€/MWh]	110	110	110	110
Biomass price	[€/MWh]	25	25	25	25
Disposal cost ash	[€/t]	100	100	100	100
Disposal cost waste water	[€/m³]	1.5	1.5	1.5	1.5
Price indexation	[% p.a.]	3.0	3.0	3.0	3.0
Maintenance costs	[% p.a.]*	2.0	2.0	2.0	2.0
Other costs	[% p.a.]*	0.5	0.5	0.5	0.5
Utilization time	[a]	20	20	20	20
Credit period	[a]	20	20	20	20
Interest rate	[% p.a.]	5.0	5.0	5.0	5.0

* % p.a. are related to the total invest

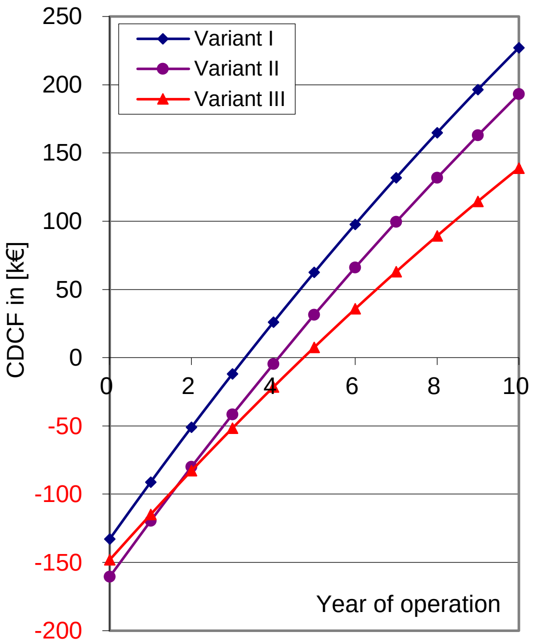

Figure 13: Cumulated discounted cash flow (CDCF) for the Variants I to III

The new flue gas condensation technology shows better performance for high return temperatures than conventional systems. If the flue gas condensation is considered during engineering of a plant the biomass boiler size can be reduced which leads to similar or lower investment costs for the plant using the new flue gas condensation technology compared to a conventional heat recovery system. Proper design of the heat pump leads to high COP values and thus high efficiency of the system. Even though electric energy is needed for the heat pump the savings in fuel are higher which leads to a better performance and shorter payback periods.

6 CONCLUSIONS

The most important results of the project are that on the one hand the technological feasibility of the new technology developed and analyzed could be proven and on the other hand the potential for increasing the efficiency and economic feasibility of biomass heating plants with a thermal capacity of 2 MW and above with return temperatures above 50 to 55°C (which is quite usual in Austria and central Europe) is quite high due to the short payback period of just about 3 years. In addition the efficient heat recovery and respective fuel savings (resource management) and the related reduction of CO_2 as well as other emissions (e.g. dust) are very important.

The results of the test runs performed show an increase in efficiency of up to 14%-points (with respect to the primary energy input) compared to a conventional biomass boiler plant without heat recovery. The new flue gas condensation technology shows higher efficiency at high return temperatures compared to conventional heat recovery systems. In case the new concept can be already considered during the design phase of a project the investment costs for the new system are similar or even lower than for conventional systems. Proper design of the heat pump and flue gas condenser gives a high flexibility regarding the return temperature of the system. The achievable fuel saving overcompensate the costs for electric energy needed for the heat pump which leads to shorter payback periods. These results make further development and demonstration of this new technology reasonable.

The design tool developed for the flue gas / refrigerant heat exchanger of the direct evaporating heat pump showed good agreement with the measured data. Thus the tool is suitable for future engineering applications.

Based on the work done in this project a list of criteria for utilization and optimization of this technology in large-scale applications was defined. The main aspects to consider are:

- Changes concerning the control concept of the heat pump: priority on staying within the limits of operation (envelope) of the compressor and thus avoiding the heat pump to switch off automatically during operation
- Improvement of the partial load performance of the system (e.g. infinitely variable control of the compressors power level, cascading multiple heat pump units)
- Improvement of the performance of the heat pump and thus the economic feasibility by adopting the envelope (choice of refrigerant and compressor design) according to the expected return temperatures and flue gas water dew points.

Based on the work performed within this project and the positive results gained the acquisition of an adequate follow up development and demonstration project is planned.

The project results are of interest for Austrian and European heat pump manufacturers as well as for the manufacturers of biomass furnaces, flue gas condensation plants and plant designers.

7 REFERENCES

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9 LOGO SPACE