Experimental studies of a biomass boiler suitable for small district heating systems

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Abstract

Extensive experiments have been carried out in a newly developed furnace suitable for small district heating networks. The fuel is wood-chips with moisture content in the range of 30–58%. One of the unique features of this new furnace is the broad thermal output span, which makes it possible to run the boiler down to 10% of maximum heat load, with maintained low emissions of CO and total hydrocarbons (THC). The aim of this study has been to evaluate the performance of the combustion chamber during steady-state operation in the complete thermal output range.

The experiments show very good results over the entire thermal output range. In the range 60 kW up to 500 kW, the average CO content in the stack gases is typically below 25 mg Nm$^{-3}$ (20 ppm) and the NO$_x$ concentration below 195 mg Nm$^{-3}$ (95 ppm) during steady state conditions. At lower thermal outputs, the average CO content is below 105 mg Nm$^{-3}$ (84 ppm). (All values standardised to 10 vol% O$_2$.)

Keywords: Biomass; Combustion; District heating; Emissions

1. Introduction

A study, carried out by the Swedish National Board of Industrial and Technical Development (NUTEK), shows that biomass boilers in the thermal output range 0.5–10 MW emit a disproportional amount of pollutants in form of products of incomplete combustion during low or varying thermal output [1]. Since this is a common situation in small district-heating networks, it is important to have a system that can handle these types of operation with minimal environmental impact. Large and frequently occurring heat load peaks arises in daytime, especially during the summer. The heat demand can be considered very low or non-existent while the demand for hot tap water is approximately the same, independent of season. Due to this the boiler must be able to work with a modulating thermal output in the range 10–100% of the nominal thermal output, since the average heat demand during summer is estimated to be around 10% of the demand in winter.

A new concept for small-scale heat production based on biomass, described in detail by Lundgren et al. [2], has been developed by Luleå University of Technology and a local industry company, AB Swebo Flis och Energi. The combustion chamber is designed for a maximum heat load of 500 kW and for thermal output modulation down to 50 kW, using...
wood-chips, with high moisture content, as fuel. The reason for using an unrefined fuel instead of pellets or briquettes is to reduce the fuel costs. The investment can also be reduced if the boiler can handle this wide thermal output span, since no supplementary heating source will be needed during summer.

The market for this kind of installation is expected to grow substantially, if the emission problems can be mastered. Particularly, in Northern-, Mid- and Eastern Europe and other countries with access to forestry and agricultural residues there is an increasing use of biomass fuels for central heating systems in small communities.

The aim of this experimental study has been to evaluate the performance of the combustion chamber at constant thermal output. Experiments have been carried out to verify the wide thermal output span and the furnace’s ability to handle fuel with high moisture content, with maintained low emissions of unburnt gases.

The combustion efficiency has not been measured during the experiments. However, the emission of CO can be regarded as a good indicator of the combustion quality.

2. Test facility and experimental set up

The test plant is located in the northern part of Sweden, around 70 km south of the Arctic Circle in the town of Boden, and is connected to the local district-heating network. The facility includes, besides furnace and heat transfer unit, a water heat store as one possibility to handle the heat load variations. Two cyclones are used for stack gas cleaning.

Fig. 1 shows an explanatory sketch of the fuel- and airflow in the plant.
Table 1
Chemical composition of the fuel

<table>
<thead>
<tr>
<th>Analysis</th>
<th>Method</th>
<th>Unit</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sulphur</td>
<td>SS 18 71 77:1</td>
<td>% of dry substance</td>
<td>&lt;0.01</td>
</tr>
<tr>
<td>Carbon</td>
<td>LECO-method 1</td>
<td>% of dry substance</td>
<td>49.5–49.8</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>LECO-method 1</td>
<td>% of dry substance</td>
<td>6.1–6.2</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>LECO-method 1</td>
<td>% of dry substance</td>
<td>&lt;0.1</td>
</tr>
<tr>
<td>Oxygen</td>
<td>Calculated</td>
<td>% of dry substance</td>
<td>43.5–44.0</td>
</tr>
<tr>
<td>Char residues</td>
<td>SS 18 71 77:1</td>
<td>% of dry substance</td>
<td>0.5</td>
</tr>
<tr>
<td>Volatiles</td>
<td>SS-ISO 562:1</td>
<td>% of dry substance</td>
<td>84.1–84.6</td>
</tr>
</tbody>
</table>

2.1. Fuel

Wood-chips with moisture content in the range of 30–58% have been used during the experiments. To determine the moisture content, fuel samples were dried in an electrical oven for at least 24 h at approximately 105°C.

An accredited laboratory (SLU, Umeå, Sweden) has performed analysis of the chemical composition of the wood-chips on several occasions. The results are shown in Table 1.

According to Hellwig [3], the size of the wood-chips can be divided into fine- and rough wood-chips with side-lengths from 5 to 50 mm and 50 to 100 mm, respectively. The wood-chips used in most of the performed experiments can be classified as rough wood-chips with a typical size between 40 and 100 mm.

2.2. The combustion chamber

The combustion chamber, shown in Fig. 2, is designed in a way that enables use of wood-chips with high moisture content, up to at least 55%, and to have a wide thermal output span, 10–100%, still fulfilling the most rigorous emission restrictions. It should also be able to handle large and fast heat load variations. It is however known that a conventional biomass boiler out on the market in Sweden can not manage a thermal output below 30% of its nominal without producing large amounts of pollutants like CO and THC. One of the main reasons is the difficulty to keep the required combustion temperature at lower heat loads. Therefore, the primary combustion chamber is divided in two modules, where one of them is designed to handle the thermal output span 50–150 kW and the other 150–350 kW. For operation over 350 kW, both modules are running together.

The fuel feeding system consists of three feeding screws, where two of them are connected to the larger module and one to the smaller. Experiments have shown that it is necessary to have at least two feeding screws to get a uniform distribution of the fuel. The feeding system also includes a hydraulic piston inside each chamber, which is used to transport the burning fuel bed forward. The displacement and the speed of the piston stroke can be chosen freely.

The wood-chips enter the back of the primary chamber on the upper horizontal plane and are pushed slowly towards a slope. The intention with the plane and the slope is to dry the fuel before the combustion starts using heat transfer by radiation and convection from the burning gases. Pyrolysis takes place at the end of the slope and on the beginning of the lower horizontal plane. Final char combustion takes place on the lower horizontal plane and at the steps shown in Fig. 2.

The primary air is pre-heated in a double wall arrangement outside the ceramics in each module. Primary air is introduced partly through slotted steel pipes in the sidewalls and partly through a pipe in the front of the furnace. (See Fig. 2). In the present design, no air is supplied through the grate, due to problems with large peaks of CO in earlier experiments when this was tried, see Lundgren et al. [2]. The ratio between front- and side primary air can be set manually. Experiments with different settings of this ratio and the ratio between total primary- and secondary air have been performed in order to study the effects on emissions and excess air. The results are shown in this paper.
The combustible gases produced by the combustion process continue to the cylindrical ceramic secondary chamber. The design was developed on basis of computational fluid dynamics (CFD) simulations and results from previous experiments. The secondary air is pre-heated partly outside the cylinder and partly between the two primary chambers to a few hundred degrees C. The temperature is a function of the final thermal output, with preheating increasing as the power of the furnace increases. CFD simulations were used to find the best possible way to supply the secondary air in order to obtain as good mixing between secondary air and combustible gases as possible. The design of the secondary combustion chamber is described in detail in a paper written by Lundgren et al. [4].

2.3. Measuring equipment

The analysis of the stack gas composition is carried out immediately after the heat transfer unit. A multi-component gas analyser for online measurements of NO, CO, CO$_2$ and O$_2$ (Maihak) and a heated THC analyser (JUM) are installed. In addition, a NO/NO$_2$ converter (JNOX) is installed to be able to measure total NO$_x$. The temperatures are measured by radiation shielded thermocouples of type N. Temperatures in the primary zone are measured in the larger module, but not in the smaller one. The only temperature measured when the smaller module is in operation, is the temperature after the secondary combustion chamber.

The values are recorded every thirtieth second in two loggers.

2.4. Measuring uncertainties

Calculations show that the measuring uncertainties for CO, CO$_2$ and NO$_x$ are $\pm 5.2\%$ of the measured value for a newly calibrated gas analysis instrument and $\pm 5.4\%$ after one week of operation. For O$_2$ measurements, the uncertainty is $\pm 5.1\%$ and $\pm 5.3\%$ respectively. The uncertainty of the THC measurements is calculated to $\pm 2.7\%$ after calibration and $\pm 3.4\%$ respectively.
Table 2
Measuring methods, source of errors and its inaccuracy for different measuring methods

<table>
<thead>
<tr>
<th>Gas</th>
<th>Method</th>
<th>CO, CO₂, NOₓ</th>
<th>O₂</th>
<th>THC</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FTIR</td>
<td></td>
<td>P.m cell&lt;sup&gt;c&lt;/sup&gt;</td>
<td>FID</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Source of errors</th>
<th>Inaccuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calibration (reference gas)&lt;sup&gt;a&lt;/sup&gt;</td>
<td>±2%</td>
</tr>
<tr>
<td>Zero point drift&lt;sup&gt;a&lt;/sup&gt;</td>
<td>&lt;1%/week</td>
</tr>
<tr>
<td>Span gas drift&lt;sup&gt;a&lt;/sup&gt;</td>
<td>&lt;1%/week</td>
</tr>
<tr>
<td>Noise&lt;sup&gt;a&lt;/sup&gt;</td>
<td>&lt;1%</td>
</tr>
<tr>
<td>Residual moisture in gas sample&lt;sup&gt;b&lt;/sup&gt;</td>
<td>&lt;1%</td>
</tr>
<tr>
<td>Interfering substances&lt;sup&gt;b&lt;/sup&gt;</td>
<td>&lt;4%</td>
</tr>
<tr>
<td>Linearity deviation&lt;sup&gt;b&lt;/sup&gt;</td>
<td>&lt;2%</td>
</tr>
<tr>
<td>Pressure- and temperature changes&lt;sup&gt;b&lt;/sup&gt;</td>
<td>&lt;1%</td>
</tr>
<tr>
<td>Oxygen synergism&lt;sup&gt;a&lt;/sup&gt;</td>
<td>—</td>
</tr>
<tr>
<td>Data logger (analog input)&lt;sup&gt;a&lt;/sup&gt;</td>
<td>±0.03%</td>
</tr>
</tbody>
</table>

<sup>a</sup>According to the manufacturer.
<sup>b</sup>Maximal acceptable deviations according to the Swedish Environmental Protection Agency.
<sup>c</sup>Paramagnetic O₂ cell.

after 24 h. The gas analysers have been calibrated at least once a week.

The reason for the small difference in uncertainty between newly calibrated equipment and after one week of operation is due to the large contribution from interfering substances, which always occurs. The error sources considered are presented in Table 2.

3. Experimental results

3.1. The larger combustion chamber (150–350 kW)

Most of the experiments have been performed in the larger combustion chamber, mainly because it has more measuring equipment than the smaller module.

The introductory experiments showed that it was possible to obtain very low emissions of CO and THC in the complete thermal output span of the larger chamber. The results, presented by Lundgren et al. [2], showed that primary air supplied through the grate caused a more inefficient combustion than when the primary air is supplied from above the fuel bed. It was also found that to obtain a good combustion process, the gas temperature must exceed at least 800°C before the entrance to the secondary zone.

The larger combustion chamber is designed to handle thermal outputs in the range of 150–350 kW. It is, however, of a great interest to find the absolute minimum and maximum thermal output with maintained low emissions.

The results presented in Fig. 3 show that it is possible to run the combustion chamber between 430 kW down to around 60 kW. However, it is most likely difficult to run the boiler at such low thermal output for a longer time with low emissions, due to the resulting temperature decrease in the combustion chamber. Noticeable is that the combustion process is more sensitive for piston strokes at lower thermal outputs considering CO peaks, see Fig. 3. At lower thermal outputs, the temperature of the combustible gases decreases, which reduces the combustion intensity in the secondary combustion chamber.

The moisture content of the fuel during the experiment was 50%. The average CO and NOₓ concentration in the stack gases were 20 and 138 mg Nm⁻³, respectively. The average O₂ content was 10 vol%.

Several tests have been performed at different thermal outputs with different moisture contents of the wood-chips. Figs. 4 and 5 show the results from two test runs at 350 kW with moisture contents 35% and 58%, respectively.

As shown in Figs. 4 and 5, the start up phase is significantly longer when using a fuel with higher moisture content. However, during steady-state conditions, the heat transfer rate from the ceramics is high enough in both cases to cause an efficient drying process. It is therefore possible to keep the gas temperature level above 800°C and thereby obtain an efficient combustion process. Noticeable is that the start up
Fig. 3. The actual thermal output span of the larger combustion chamber (emissions standardised to 10 vol% O₂).

Fig. 4. Temperature before the secondary zone, O₂ content and CO emissions at 10 vol% O₂. Moisture content 35%.

Fig. 5. Temperature before the secondary zone, O₂ content and CO emissions at 10 vol% O₂. Moisture content 58%.
phase in the experiment with the drier fuel shown in Fig. 4 is very fast.

However, other experiments show that emission problems can occur if the wood-chips are fine and at the same time wet. In this case, the fuel bed gets very compact, which makes it difficult for the primary air jets to penetrate into the fuel bed. This results in a lower combustion temperature, higher emissions of CO and an increased amount of unburnt fuel.

Three different distributions of primary- and secondary air supplies at two different thermal outputs have been studied. The studies also included three different ratios between front- and side primary air with the total volume flow of primary air kept constant.

The moisture content of the wood-chips was 45% during the experiments. Each experiment lasted for at least one hour during steady state conditions. Table 3 shows the average excess air ratio and emissions for different settings.

The study shows that the NO$_x$ content can be slightly reduced if the amount of primary air is reduced. It also shows that the excess air ratio decreases if more side- than front primary air is supplied, due to a more effective use of the primary air. The content of CO is not considerably affected.

It can be discussed whether the levels of primary-/secondary air ratios should have been chosen with larger difference to be able to draw any definitive conclusions. Therefore, further investigations are necessary.

Table 4 shows a summary of performed test runs in the whole thermal output span of the larger module.
Table 5
Average thermal outputs and emissions at 10 vol% O$_2$ during steady-state conditions in the smaller combustion chamber

<table>
<thead>
<tr>
<th>Module 2 (50–150 kW)</th>
<th>Thermal output</th>
<th>Moisture content$^b$</th>
<th>Excess air ratio $\lambda$</th>
<th>CO (mg nm$^{-3}$)</th>
<th>NO$_x$ (mg nm$^{-3}$)</th>
<th>THC (mg nm$^{-3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>50 kW</td>
<td>—</td>
<td>2.15</td>
<td>104</td>
<td>160</td>
<td>0.3</td>
</tr>
<tr>
<td></td>
<td>100 kW</td>
<td>—</td>
<td>1.87</td>
<td>97</td>
<td>181</td>
<td>—</td>
</tr>
<tr>
<td></td>
<td>125 kW</td>
<td>—</td>
<td>1.64</td>
<td>44</td>
<td>172</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>150 kW</td>
<td>—</td>
<td>2.1</td>
<td>101</td>
<td>167</td>
<td>—</td>
</tr>
</tbody>
</table>

$^b$The moisture content of the fuel was not measured during these experiments, but was estimated to be between 30% and 40%.

The main reasons are severe air leakage through the grate and lack of primary air distribution control. In this design, it is unfortunately not possible to control the front-/side air ratio. Most of the air is supplied through the front pipe, due to less pressure drops than in the side pipes. The velocity of the front air gets so high, that the fuel is blown away. When the piston strikes and pushes new fuel to the empty part of the fuel bed, the gasification gets too intensive, which causes large CO peaks. The excess air ratio increases since the primary air is not efficiently used.

Air leakage through the grate appears in several locations in the furnace causing considerably larger peaks of CO when the piston strikes. In addition, higher NO$_x$ content occurs when the air is blown through the fuel bed.

It also seems like the smaller combustion chamber is more sensitive to the fuel moisture content than the larger module. Figs. 6 and 7 show the temperature after the secondary combustion chamber and the CO- and O$_2$ contents in two different experiments at a thermal output of 50–150 kW.
output of approximately 150 kW. The fuel moisture contents were 52% and 35%, respectively.

The CO content is considerably higher in the experiment with high moisture content. It is observed that a CO peak occurs at the same time as an O₂ peak. This means that the combustion temporarily stops in the secondary zone, due to too low temperature level. As mentioned earlier, the temperature before the secondary combustion chamber must exceed at least 800°C to achieve as good combustion as possible. (Provided that the air supply is sufficient). When the fuel is wet, this temperature is more difficult to retain.

The average CO contents in the experiments were 242 and 39 mg Nm⁻³, respectively.

3.3. Summary of results in the complete thermal output span

Several experiments at different levels of constant thermal output, during at least three hours of steady state, have been performed. The complete thermal output span has been tested, that is between 50 kW and 500 kW. Some typical results are shown in Fig. 8.

The average O₂ level at different thermal outputs is overall acceptable. The O₂ content in the stack gas is decreasing at higher thermal outputs due to better mixing between the combustible gases and the secondary air. The combustion intensity in the secondary zone is reduced at lower heat loads, which means that the secondary air partly dilutes the stack gases. The results also show that at higher thermal outputs, above 350 kW, it is possible to decrease the content of O₂ below 5 vol% without producing higher emissions of CO.

The overall CO concentration level is very low in the complete thermal output span. The test run at 50 kW was carried out in the smaller combustion chamber, and as discussed before, this chamber does not work as well as the larger combustion chamber. The best results are obtained from 60 kW up to 350 kW when the larger combustion chamber is in
operation. Good results are also achieved when the two chambers operate together at 500 kW. Low emissions of CO also mean low emissions of THC, which is confirmed in the performed experiments.

3.4. Fuel burnout

An analysis of the amount of unburnt carbon in the solid residue from the furnace has been performed by an accredited laboratory (SLU, Umeå). The result show that the amount of unburnt carbon is below 0.1% of dry substance. This result is valid for both combustion chambers.

3.5. Thermal efficiency

Only one calculation of the thermal efficiency has been performed since the fuel consumption is not measured continuously. Therefore, calculations can only be performed when a new fuel container with a known weight is delivered. Then, when the container is empty and the boiler has been running continuously, it is possible to calculate the efficiency. Unfortunately, the surface temperature of the furnace is not measured, which makes it impossible to perform a complete heat balance.

The calorific heat value of the fuel can be calculated as

$$(19.22 - 21.7^*(f/100))*0.278 \text{ (kWh kg}^{-1})$$

where $f$ is the moisture content of the fuel in percent.

During the experiment, the larger combustion chamber was operating with a modulating thermal output in the range 150–350 kW. The moisture content of the fuel was 50.4%, which gives the calorific heat value 2.3 kWh kg$^{-1}$. Totally, 8.5 tonnes wood-chips were consumed during the test corresponding to 19.55 MWh.

The total energy production, calculated by integrating the measured thermal output delivered to the district-heating network over the time of operation, amounted to 16.192 MWh. The thermal efficiency calculated as the ratio between delivered- and supplied energy, amounts to approximately 83%.

Calculations show that the stack gas losses amount to 11%, assuming an ambient temperature of 25°C. The loss due to CO emissions is calculated to 0.02%. The losses from unburnt carbon in the fly ash and in the solid residue from the furnace as well as THC emissions are however not considered.

4. Discussion

The results of the experiments at constant thermal outputs in the complete heat load range are very satisfactory. It has been shown that the furnace can operate at thermal outputs around 10% of maximum load, without producing large amounts of unburnt gases. Experiments with fast and large heat load fluctuations will also soon be performed. Need for capacity to cope with large load fluctuations can arise, for example during summer seasons, when the heat demand is equal or close to zero and only peaks for hot tap water occur. The boiler has then to be started during the peak period and then stopped until next peak arises. In this case, it will take a while for the furnace to heat up and when it finally has reached steady state, the peak is most likely over. The solution could be to use the heat store for the load variations. It will allow the furnace to run at an almost constant thermal output, while the heat store takes care of the hot tap water peaks. Experiments where the furnace operates together with the heat store will be performed.

It has been observed that when the combustion chamber is started and stopped, that is heated up and cooled down, air leakages appear in new places. The steps after the ceramic fuel bed are made out of steel to facilitate tests with different arrangements for the primary air supply. The steel steps have a high coefficient of thermal expansion, which results in dimensional changes during starts and stops. This affects the air leakage between the steel steps and the ceramic fuel bed and the sidewalls, which gets larger at every new start up.

The smaller combustion chamber appears to be more sensitive for high moisture contents of the wood-chips than the larger module, which indicates that the smaller combustion chamber is not optimally dimensioned. For example, the distance from the primary- to the secondary zone is too large and should be reduced in order to decrease the cooling rate of the combustible gases.

Furthermore, as shown in Tables 4 and 5, the NO$_x$ content is often higher in the smaller chamber than in the larger. It is very difficult to give a reasonable
explanation for this observation and it is therefore ne-
cessary to carry out further experiments in both cham-
bbers with exactly the same fuel quality to be able to
eliminate the effect of differences in fuel bound nitro-
gen in the experimental results.

5. Conclusions

Experiments at constant thermal output, in the range
60–500 kW, show very good results considering emis-
sions of CO and THC. The average CO content in
the stack gases during the presented tests is below
25 mg Nm$^{-3}$.

Moisture contents of the fuel up to around 58% do
not seem to affect the combustion process negatively,
when the large combustion chamber is in operation.
The only observed difference between using a wet and
dry fuel, is the time to heat up the furnace. The exper-
iments in the smaller chamber show, on the contrary,
that the moisture content of the fuel has a strong in-
fluence on the combustion process.

The large combustion chamber has a much broader
thermal output span than it was designed for. The ex-
periment showed that it is possible to operate it be-
tween 60 kW up to 430 kW, with maintained low
emissions of CO and THC. It is however not likely that
it is possible to operate at that low thermal output for
a longer period, due to a decreasing gas temperature.

The smaller combustion chamber does not work as
good as the larger module. However, the results are
anyway satisfactory. The CO concentrations are typ-
ically below 105 mg Nm$^{-3}$ and NO$_x$ content below
182 mg Nm$^{-3}$ (standardised to 10 vol% O$_2$) during
steady state conditions.

Further experiments are necessary to be able to draw
any major conclusions concerning the thermal effi-
ciency.

The main conclusions are that it is possible to run
the boiler down to 10% of maximum load with low
emissions of CO and THC even when wet wood-chips
are used. This can reduce both the investment and
operating costs for small district heating plants.

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