A new biomass based boiler concept for small district heating systems

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ABSTRACT

A new bio fuel based boiler concept for small district heating systems has been developed. The boiler enables heat load variations from 100% down to 10% of nominal heat power output fulfilling the most rigorous environmental restrictions. To obtain as low fuel costs as possible, an unrefined fuel in the form of wood-chips with moisture content in the range of 35-55% is used.

The primary combustion chamber is partitioned to be able to maintain the required combustion temperature even at low thermal outputs. The thermal output span is 50 kW to 500 kW, with one module operating in the range of 50-150 kW and the other in the range of 150-350 kW. The smaller module will operate during summer time when heat is demanded only for hot tap water production, which on average amounts to about 10% of the maximum heat demand during one season. Because of the wide thermal output span that is possible, the boiler can run continuously without starts and stops, which reduces emissions of unburnt gases radically. As another possibility to handle fast heat load variations that frequently occur in smaller district heating networks, a water heat store has been installed in the system.

The introductory experiments in the larger module are very promising. Initially, the primary air was supplied partly through the grate with CO peaks around 2500 mg/nm³ as a result. After some modification of the primary air supply system, the average CO content in stack gases are now typically below 20 mg/nm³ during steady state conditions. To achieve these results it is necessary to have a combustion temperature above 800 °C and oxygen content in the stack gases not below 5 vol%.

INTRODUCTION

The Swedish Parliament has decided to restructure the energy supply system. One of the objectives in the energy policy decision is to gradually phase out the nuclear power and replace it with domestic non-polluting energy sources. The nuclear power production was 1999 amounted to 69.5 TWh or 46% of the total electricity production in Sweden.

The most common type of heating in detached houses in Sweden is electric heating, being the main heating source in approximately 40% of such dwellings. In 1998, the total electricity use in detached houses for heating purposes was amounted to 20.5 TWh. The reason for this high proportion of electric heating is low installation costs and convenient operation. In order
to make a nuclear phase out possible, the demand for electricity must be reduced and other energy sources must be used to a larger extent.

One way to reduce the electricity demand is to convert from electric heating to district heating. This is an ongoing activity in areas covered by the main district heating networks. However, there are a lot of smaller communities outside these areas in the northern, mid- and eastern Europe and other countries with access to forestry and agricultural residues, which are very well suited for smaller district heating networks where the heat production is based on bio fuels. It is also expected that the market for small and middle sized biomass fired boilers will grow substantially if the emission problems can be mastered.

Due to great load variations in smaller district heating networks, boilers have to work with a modulating thermal output of 10 to 100% of their nominal output. A study financed by the Swedish National Board for Industrial and Technical Development (NUTEK) shows that boilers in the power output range of 0.5-10 MW emit substantial amounts of harmful emissions like CO, NO$_x$ and THC at low thermal output or at fast load changes.

Therefore, Luleå University of Technology together with a local industry company has developed a new bio fuel based boiler concept, which will make it possible to produce heat in small district heating networks at a competitive price and with little impact on the environment.

The emphasis of this paper has been placed on describing the new concept and presenting the results of introductory experiments.

PROJECT OBJECTIVITIES

The main objective has been to design a biomass boiler with a maximum thermal output of 500 kW.

The boiler must fulfil the most rigorous restrictions on emissions considering CO, NO$_x$ and THC in the whole thermal output span, that is down to 10% of maximum heat power. It must also be able to handle fast heat load variations with maintained low emissions.

To keep the operating costs low, the fuel should be unrefined.

DESCRIPTION OF THE TEST PLANT

The test site is located in Boden, about 36 km from Luleå, and is connected to the local district-heating network through a heat exchanger.

The plant consists of a fuel store, fuel conveyors, combustion equipment, a boiler, two cyclones and a water heat store.

The combustion is performed in two stages, in the primary and secondary zone. After the secondary combustion chamber, the flue gas flows to a conventional convection boiler where the heat is transferred to the water. The cooled gases continue through one or both of the cyclones to the chimney. (See figure 1).
Fuel and fuel feeding

The fuel is wood-chips with moisture content in the range of 35-55%. The wood-chips are stored in two containers outside the boiler house. (See figure 2) Each container can supply requisite fuel for 48 hours at maximum heat load. A feeding screw transports the fuel from the two wood-chips containers in to an intermediate store before it is fed into the combustion chamber.
A mixer is installed in the intermediate fuel store to secure the fuel feeding. Wood-chips have tendency to build caves above the rotating feeding screws in the bottom of the store, which decreases the fuel supply radically. Therefore, the mixer is needed to destroy the formed caves.

Three feeding screws supply the fuel to the combustion chamber, with two of them feeding the larger chamber. Experiments have shown that at least two feeding screws are necessary to get a uniform distribution of the fuel in the chamber. Two different ways to run the feeding screws will be investigated, frequency speed control and on/off control of the electrical engines.

**Primary combustion chamber**

The primary combustion chamber is partitioned to maintain the required combustion temperature even at lower thermal output. The maximum thermal output of the chamber should be 500 kW and the lowest about 50 kW. A conventional combustion chamber out on the market in Sweden can not manage a thermal output below 30% of maximum without producing large amounts of unburnt gases due to the resulting lower combustion temperature. This means that a reasonable way to divide the combustion chamber is between one module of about 350 kW and one module of about 150 kW. Figure 3 shows a sketch of the partitioned primary combustion chamber.

![Figure 3. Sketch of the partitioned primary combustion chamber](image)

The fuel enters the combustion chamber at a horizontal plane and moves slowly towards an inclined plane. The purpose with the two planes is to dry the wood-chips, before the combustion, using heat transfer by radiation and convection from the combustible gases. Pyrolysis starts in the slope and the beginning of the horizontal plane after the slope. Final char combustion takes place on the horizontal plane and on the steps shown in the figure.

Two rectangular hydraulic pistons placed in the lower horizontal section, pushes the fuel forward towards the steps. It is possible to vary the length and velocity of the piston stroke.

The primary combustion air is pre-heated in the double wall construction outside the ceramics in each module. The air is supplied partly above the fuel bed in the front end of the combustion chamber (front air) and partly from pipes along the sidewalls.
In this prototype, the ash is removed manually. In the final design, the ash will be removed by use of a worm conveyor.

**Secondary combustion chamber**

The design of the secondary combustion chamber is based on previous experiences and CFD simulations. It is designed to get as good mixing as possible between the secondary air and the combustible gases. The secondary combustion air is pre-heated to a few hundred degrees Celsius depending on the actual thermal output.

The partly oxidised combustion gases from the primary zone are mixed with the pre-heated combustion air, which is supplied in the throat between the primary- and secondary stage. The mixture enters the horizontal cylindrical secondary chamber above the throat where the combustible gases are completely oxidised if the right conditions are achieved. Figure 4 shows the design of the ceramic body of the secondary zone. The air supply arrangement is not shown in the picture.

![Figure 4. The secondary combustion chamber](image)

**Cyclones**

Two cyclones of different size have been installed to separate particles from the combustion gases. The gases will pass through one or both of the cyclones, depending on the thermal output. For example, at low thermal output the gases will pass only the smaller cyclone to maintain the gas velocity and secure the separation.

**Water heat store**

As one way to handle heat load variations in the district-heating network, an unpressurised water heat store is installed. It also works as an expansion volume and, in case of failure of the circulation pumps, as a self-circulated cooling buffer for the heat recovery unit. The heat store has a water volume of about 35 m$^3$.

**Control strategy**

The purpose of the introductory experiments was to learn how the system works. Important parameters for the combustion process as fuel feeding, primary- and secondary air supply needed to be studied. Therefore, all units in the heating central were manually controlled and the set point values adjusted continuously during the experiments.
Two Glow Guards, developed by AMFT, Graz Austria, are installed to measure the intensity of light in the fuel bed. The idea is to use the Glow Guards to control the fuel feeding. If, for example, the Glow Guard at the end of the combustion chamber registers a high intensity of light, the fuel-feeding rate has to decrease.

As mentioned earlier, heat storage is one way to handle the heat load variations, which arises in the district-heating network, especially during mornings and evenings. These variations will be simulated with power output from the test site matching these demands either with the partitioned combustion chamber only or with the larger combustion chamber together with the heat store. Both ways will be compared concerning efficiency and emissions.

**Measuring equipment**

The concentrations of CO₂, CO, O₂, NOₓ, THC and gas temperature are measured in the stack gas after the boiler. The gas analysing system consists of a multi-component gas analyser (Maihak) for continuous measurements of NO, CO, CO₂ and O₂, together with a NO₂/NO converter (JNOX) to be able to measure NOₓ. A heated total hydrocarbon (THC) analyser (JUM) is also included. The gas sample probes are installed immediately after the convection boiler. The measuring ranges and methods are shown in table 1.

<table>
<thead>
<tr>
<th>Component</th>
<th>Range</th>
<th>Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>O₂</td>
<td>0-25 vol%</td>
<td>P.m.c*</td>
</tr>
<tr>
<td>CO</td>
<td>0-1000/10 000 ppm</td>
<td>FTIR**</td>
</tr>
<tr>
<td>CO₂</td>
<td>0-20 vol%</td>
<td>FTIR**</td>
</tr>
<tr>
<td>NO</td>
<td>0-500 ppm</td>
<td>FTIR**</td>
</tr>
<tr>
<td>THC</td>
<td>0-10/10³/10⁴/10⁵ ppm</td>
<td>FID***</td>
</tr>
</tbody>
</table>

*Paramagnetic O₂ cell, **Fourier Transform Infra Red, ***Flame Ionisation Detector

The temperatures in the combustion chamber and temperature of the flue gas are measured by radiation shielded thermocouples.

The data acquisition equipment consists of two loggers, with 24 channels each. Data’s are sampled at a frequency of 0.2 Hz and recorded every 30 second.

**FIRST EXPERIMENTAL RESULTS**

The first tests were carried out in the larger combustion chamber (150-350 kW), since this chamber has more measuring equipment than the smaller one. The moisture content of the wood chips was in the range of 42-50% during the experiments.

The emissions of CO have been used as an indicator of the effectiveness of the combustion and changes of the design have been made with the purpose to minimise these emissions without increasing the amount of NOₓ in the stack gases.

In the first design of the combustion chamber, the primary air was supplied partly through pipes in the sidewalls and partly through the grate in the steps.
The first experiments showed that the CO content in the stack gases was high, especially after a piston stroke. When the air supply through the grate was closed, the emissions of CO were radically reduced. See figure 5. The average CO concentration during this run was 506 mg/nm$^3$.

![Graph of CO emissions over time](image)

**Figure 5. Emissions of CO standardised to 10 vol% O$_2$. The thermal output was ~140 kW**

Several experiments were made with grate- and side air, but the best results were achieved when the primary air was supplied only through the sidewalls. However, without combustion air fed through the grate, problems with increased amount of unburnt fuel occurred. Therefore, it was decided to redesign the primary air system. Instead of supplying the air through the grate, the air is, in the present design, supplied above the fuel bed in the front end of the combustion chamber, front air. The reconstruction resulted in a decrease of the average stack gas content of CO and the amount of unburnt fuel. One experiment with front- and side primary air is shown in figure 6. The average CO concentration was here 22 mg/nm$^3$.

![Graph of CO emissions over time](image)

**Figure 6. Emissions of CO standardised to 10 vol% O$_2$. The thermal output was ~150 kW**

It has been observed that the ratio between front- and side primary air affects excess air ratio. It seems as the concentration of O$_2$ decreases if more side- than front air is supplied, probably due to a more effective use of the primary air. Further experiments are however necessary to investigate this dependence more closely.
Figure 7 shows the emissions of CO as a function of O\textsubscript{2} content in the stack gases. In the present design, the O\textsubscript{2} content must not be lower than 5 vol.% to avoid local deficit of O\textsubscript{2} with resulting high peaks of CO.

![Figure 7. The emissions of CO as a function of O\textsubscript{2} content in the stack gases](image)

Emissions of CO also occur if the temperature in the combustion chamber is too low. Figure 8 shows that it is important that the temperature before the entrance to the secondary zone exceeds at least 800°C to get as efficient combustion as possible.

![Figure 8. Emissions of CO standardised to 10 vol.% O\textsubscript{2} as a function of the temperature before the secondary zone](image)

Very good results have been achieved after the modification of primary air supply. The results show that the combustion in the larger chamber is very effective in the whole thermal output span. See Table 2.

| Table 2. Average thermal outputs and emissions (standardised to 10 vol% O\textsubscript{2}) during steady-state conditions in the larger combustion chamber. |
|---|---|---|---|
| Module 1 (150-350 kW) |
| Thermal output | 150 kW | 250 kW | 350 kW |
| Moisture content | 50.4% | 42.8% | 44.7% |
| O\textsubscript{2} (vol%) | 7.6 | 7.5 | 7.1 |
| CO (mg/nm\textsuperscript{3}) | 14 | 11 | 11 |
| NO\textsubscript{x} (mg/nm\textsuperscript{3}) | 142 | 193 | 191 |
| THC (mg/nm\textsuperscript{3}) | 0.3 | 0.6 | 0.8 |
The content of CO and THC in the stack gases is extremely low. However, the NO\textsubscript{x} content is increasing with increasing thermal output, which indicates that thermal NO\textsubscript{x} is formed.

DISCUSSION

This far the results seem very promising. However, a lot of work is of course still remaining.

The large combustion chamber works very well in the tested thermal output span. Though, it is most likely possible to get even better results with some modifications. For example, air leaks through the piston inlet. If this leakage could be reduced, it is possible to decrease the excess air ratio and thereby increase the total thermal efficiency.

CONCLUSIONS

It is possible to obtain very low emissions of CO in the whole thermal output range in the larger combustion chamber using fuel with high moisture content. It is known that low values of CO also means minimal emissions of VOC and PAH.

The combination of grate- and side primary air yields ineffective combustion in this type of combustion chamber design. The gas residence time probably gets too short due to the very intensive gasification above the grate when the piston strikes. However, without air from the grate the amount of unburnt fuel increases. Therefore, primary air should be supplied through the sidewalls, and instead of grate air, from the front above the fuel bed.

It is of great importance to keep the O\textsubscript{2} content above 5 vol.\% in the stack gases, to avoid local deficit of O\textsubscript{2} in the combustion chamber. However, this limit can probably get lower when the air leakage, discussed earlier, is reduced.

A combustion temperature above 800 °C is necessary to completely burn the CO. Otherwise emissions of unburnt occurs due to low temperatures in the combustion chamber.

ACKNOWLEDGEMENTS

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