Development of a Wood Powder Fuelled 35 kW Stirling CHP Unit

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ABSTRACT

For biomass fuelled CHP in sizes below 100 kW, Stirling engines are the only feasible alternative today. Using wood powder as fuel, the Stirling engine can be heated directly by the flame like when using a gaseous or liquid fuel burner. However, the combustion chamber will have to be much larger due to the slow burning of the fuel. To avoid sintering of ash the flame temperature in a biomass burner has to be quite low. The flame temperature can be lowered by using a high excess air rate; however this will lower system efficiency and needs a large air preheater. By using combustion gas recirculation (CGR) a smaller air preheater can be used, while system efficiency will increase compared with using excess air for flame cooling.

In a three-year project, a wood powder fuelled Stirling engine CHP unit will be developed and run in field test. The project will use the double-acting four-cylinder Stirling engine SM3D with an electric output of 35 kW. This engine is a further development of the engine SM3B that has been developed at the Technical University of Denmark. The engine heater is being adapted for use with wood powder as fuel. During a two-year period a combustion system for this engine, using wood powder as fuel will be developed at Lund University, Sweden, in cooperation with the Technical University of Denmark and with the wood powder boiler manufacturer VTS AB.

The unit is to be run in CHP operation by Vattenfall - the largest electric power company in Sweden - in a one-year field test in Älvkarleby, Sweden, and the operation of the unit will be evaluated at the end of the field test period.

The unit is intended for blocks of flats, schools, local heat production plants and the wood industry.

INTRODUCTION

The market for small scale or distributed CHP will be depending on the evolution of the price of electricity as related to the production cost of a mature commercial CHP plant. Specifically, the capital costs will be an important factor. Low capital costs will be a function of standardization, product optimisation and the size of production series.

For southern Sweden the following small scale plants' power need has been estimated:

Application:	Approximate number of plants:	Power need:
Wood industry	250	300 GWh
Local production plants	-	150 GWh
Small district heating grids	20	200 GWh
Boiler stations	200	200 GWh

This gives a total of about 850 GWh, of which 75 % could be covered by combined heat- and power production. Calculated for base load heat production this represents about 150 MW. The conclusion is that alone in southern Sweden there is a considerable potential for introducing small scale biofuel CHP technology. Including the whole of Sweden the potential is even bigger.

In Sweden small scale utilizing of biofuels is mainly through the use of refined wood fuels, such as wood pellets or wood powder. These fuels, having low water contents and efficient packing, are easy to handle and transport. Also the combustion characteristics are good, and emission levels are low.

For biomass fuelled CHP in sizes below 100 kW, Stirling engines are the only feasible alternative today. The Stirling engine is well suited for direct combustion of biofuels. Combustion chamber gas velocities are low enough to avoid substantial erosion of the heater by fuel particles, and using air preheating efficiencies in the range 20 to 25% can be reached. Using wood powder as fuel, the Stirling engine can be heated directly by the flame like when using a gaseous or liquid fuel burner, and dynamic control is easier as the mass of fuel present in the combustion chamber at a time is small. However, the combustion chamber will have to be much larger than when using liquid or gaseous fuels due to the slow burning and consumption of the wood powder fuel particles. To avoid sintering and melting of ash the flame temperature in a biomass burner has to be quite low. The flame temperature can be lowered by using a high excess air rate; however this will lower system efficiency and requires a large air preheater. By using combustion gas recirculation (CGR) a smaller air preheater can be used, and system efficiency will increase compared to when using excess air for flame cooling. The Stirling heater also has to be designed in such a way that it will allow the passage of ash particles travelling with the combustion gas flow.

PROJECT DESCRIPTION

In a three-year project, a wood powder fuelled Stirling engine CHP unit will be developed and run in field test. The development aims at the adaptation of an existing burner type. It is considered important to get operational experience and to identify problems concerning materials, technology and control. However, as current wood powder and wood pellets burners are estimated to already have acceptable exhaust emission levels, the optimisation of operation towards low emission levels is not prioritised.

The project will use the double-acting four-cylinder Stirling engine SM3D with an electric output of 35 kW. The engine heater is being adapted for use with wood powder as fuel. During a two-year period a wood powder combustion system for this engine will be developed at Lund University, Sweden, in cooperation with the

Technical University of Denmark and with the wood powder boiler manufacturer Värmeteknisk Service AB (VTS).

Most wood powder burners are considerably bigger than the 200 kW needed for the SM3D engine. However, VTS have designed and built a 150 kW wood powder burner for the Swedish Agricultural University in Umeå, which is currently used for the evaluation of different wood powder fuels. The experiences gained in the operation of the Umeå burner have been used when designing the burner for the Stirling unit.

The unit is to be run in CHP operation by Vattenfall - the largest electric utilities company in Sweden - in a one-year long field test in Älvkarleby, Sweden, and the operation of the unit will be evaluated at the end of the field test period. The main goal of the field test is to demonstrate the technology and to run the unit at varying load during the test period. The test period will be followed by an evaluation of the system performance, focusing on financial issues like market potential and cost of operation. Furthermore, Vattenfall are interested in getting operational skills and experiences of how this kind of wood powder unit will interact with the domestic heating and electrical grids.

Prior to moving the unit to the Vattenfall development laboratory in Älvkarleby, the unit will go through a test procedure at the Lund University location, including running continuously for 700 hours without unplanned stops. Planned stops should be for maintenance and adjustment only. Maintenance intervals should exceed 200 hours and the duration of each maintenance stop should not exceed 4 hours or 2% of the operating time.

The unit is intended for blocks of flats, schools, local heat production plants and the wood industry.

THE SM3D STIRLING ENGINE

The SM3D engine, a further development of the SM3B engine, has been developed at the Technical University of Denmark. This 35 kW_{el} Stirling engine has four cylinders arranged in a square with the cylinders parallel to each other. Helium is used as working gas at a maximum mean pressure of 4.5 MPa. The engine is designed as a hermetically sealed unit with the generator incorporated in the pressurised crankcase just like a hermetically sealed compressor for refrigeration, where the electric motor is incorporated in a hermetically sealed unit. Only static seals are necessary and the only connections from the inside of the hermetically sealed crankcase to the surroundings are the cable connection between the generator and the grid. The asynchronous generator, which is also used as starter motor, has 6 poles corresponding to an engine speed of 1010 rpm when coupled directly to the power grid (50 Hz AC). The engine is described in (ref. 1, 2).

A new heater has been designed specifically for the wood powder combustion system. The heater has a large surface area for transfer of heat from the combustion of the powder to the helium working gas. Narrow passages have been avoided, as the content of ashes in the fuel must be transported through the heater. The pressure loss in the heater has been kept at a very low level in order to make recirculation of hot exhaust gas possible.

The engine is mounted horizontally at the end of the powder combustor as shown in Figure 8. Figure 1 shows a photo of the 4 cylinder engine.

Figure 1   **Photo of the SM3D Stirling engine that will be used in the project**

COMBUSTION SYSTEM

Solid biofuels (e.g. wood powder) will give an ash residue when burned. Above a certain temperature he ash will start to melt and get "sticky", which will lead to build-up of ash at the Stirling heater or somewhere in the combustion chamber. To avoid this, it is imperative to keep the combustion chamber temperature below the ash melting point. Thus it has been decided to design the system for a combustion chamber temperature of 1250°C.

There are two main methods of lowering the flame temperature and the combustion chamber temperature. Both methods incorporate the use of an inert bulk gas flow through the combustion chamber that absorbs some of the released heat, lowering the flame temperature in the process. The first method is to increase the excess air, i.e. increase the air/fuel ratio (lambda). The other method is to use recirculation of exhaust or combustion gas. In practice a combination of the two methods is commonly used. If we use combustion gas recirculation (CGR), we can tap the recirculation flow from three different locations in the exhaust path (Figure 2 below):

1. Just downstream of the Stirling engine heater
2. Downstream (exhaust-wise) of the air preheater, or
3. In the case the system has an exhaust cooler, we may take the recirculation flow from downstream of the cooler[1].

The recirculation circuit will need to be powered by some pressure-increasing device, like an electric blower or a high-speed jet flow (cf. ref. 3). In the case a blower is used, it is preferably connected at the coldest part of the

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[1] Exhaust cooler: Heat exchanger situated downstream of the preheater and that will cool the exhausts in order to deliver the heat to the heating grid.

recirculation circuit. For alternatives 2 and 3 we then have the option to let the recirculation flow once more pass through the preheater on its way to the combustor. However, for solid biofuels this may lead to the build-up of ash residues on the preheater air side, why this alternative is not considered feasible and thus is omitted from the calculations below.

The further downstream we tap the recirculation flow, the colder the flow will be and consequently the greater flame cooling power it has and the lower recirculating massflow we need. On the other hand, the further downstream of the heater we tap the recirculation flow, the lower the system efficiency will be because of the heat exchanging characteristics.
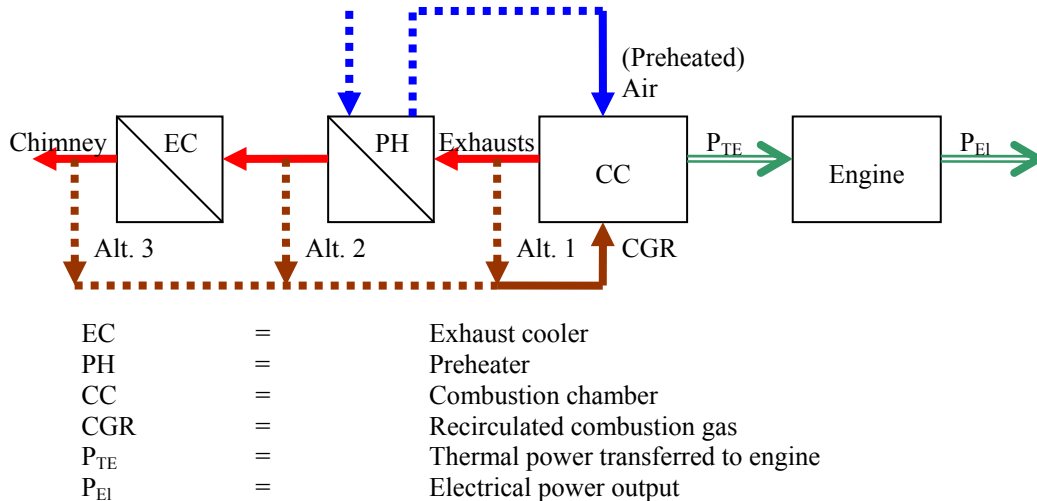

EC   =   Exhaust cooler
PH   =   Preheater
CC   =   Combustion chamber
CGR   =   Recirculated combustion gas
P_{TE} =   Thermal power transferred to engine
P_{El} =   Electrical power output

Figure 2   Alternative combustion gas recirculation arrangements

COMBUSTION SYSTEM CALCULATIONS AND RESULTS

The adiabatic flame temperature for a wood powder flame was calculated for different combinations of air preheating temperature, air/fuel ratio (lambda, λ) and combustion gas recirculation (CGR). Figure 3 shows the results for wood powder fuel with 8% water content, 0.3% ash content and a fuel effective heat value of 18.8 MJ/kg. The recirculated gas is assumed to have a temperature of 1000K when re-entering the combustion chamber. The calculations do not take into account dissociation, combustion chamber heat losses or temperature drop due to radiative heat transfer (which have considerable impact on the real flame temperature), and are to be used qualitatively only.

As can be seen from Figure 3, preheating of the inlet air will have a major impact on the amount of excess air needed to lower the flame temperature down to 1250°C. If we use no recirculation at all (Figure 3, top left) and have a preheated air temperature of 727°C (1000K) we have to use a λ of about 5 to reach a flame temperature of 1250°C. However, if we use a recirculation rate of 100% (compared to massflow exhausts leaving the chimney) we can use the same λ as in the case of no preheating and no recirculation.

How, then, does the use of recirculation rather than excess air affect the system efficiency? We want the combustion system to deliver the power P_{TE} (see Figure 2) to be transferred from the combustion system into the Stirling cycle through the engine heater. The engine can not distinguish how this power was generated and

transferred; it is only sensitive to the amount of energy transferred and at what temperature it is transferred. In this way we may detach the combustion system from the engine and calculate a *combustor* or *combustion system efficiency*, η_{CS}, i.e. the fraction of the supplied fuel energy that is transferred to the Stirling cycle. The combustor efficiency can then be optimised independent of the engine. The values given in Table 1 are assumed to be representative of the engine operation and of the combustion system, and are used as input for the calculations.
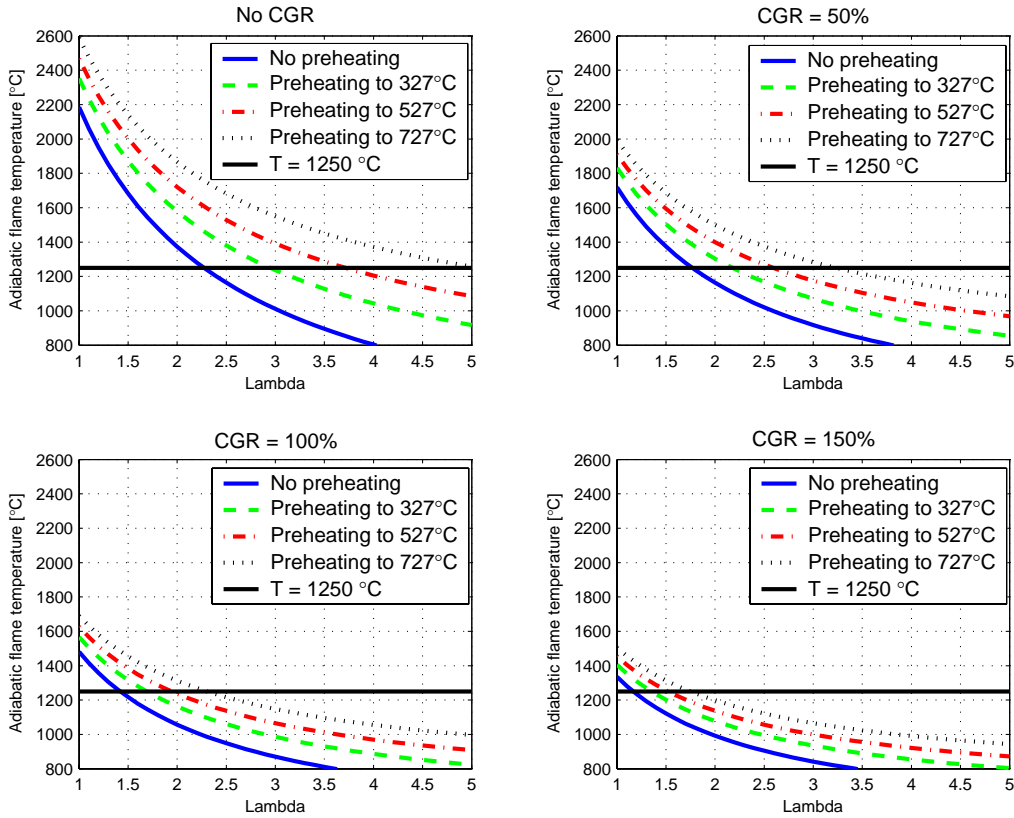
Figure 3   **Calculated adiabatic flame temperatures (assuming no dissociation or radiation) for a wood powder flame at varied λ, preheating and CGR**

Engine input power, P_{TE}	120 kW
Stirling heater combustion gas inlet temperature, $T_{heater,in}$	1250 °C
Stirling heater combustion gas outlet temperature $T_{heater,out}$	780 °C
Air inlet temperature, $T_{air,in}$ (just downstream of air blower)	50 °C
Setpoint flame temperature (= $T_{heater,in}$)	1250 °C
Exhaust gas outlet (chimney) temperature, $T_{exh,out}$	130 °C
Combustion chamber heat transfer loss, $Q_{CC,loss}$	8 kW
Preheater efficiency, ε_{PH}	80 %

Table 1   **Input values for calculations**

The preheater has been simulated using the ε-NTU method (see e.g. ref. 3) for a counter-flow heat exchanger, with an assumed preheater efficiency ε of 0.8. Keeping the preheater efficiency constant (at 0.8) is equivalent to keeping the preheated air temperature constant, in this case at 642 °C (a consequence of the fact that we have

specified the preheater air inlet temperature and the heater outlet temperature). Some characteristics of the preheater and combustion system for different λ, using combustion gas recirculation (CGR) according to the three alternatives given in Figure 2, have been calculated and are shown in Figure 4 below. Comparative energy balances for the three alternatives of Figure 2 was calculated, and are shown in Figure 6 below. Combustion gas mass- and volume flow at the Stirling heater inlet are shown in Figure 5. Figures 4-6 all refer to the same set of calculations, and thus share the same inputs and restrictions given in Table 1.
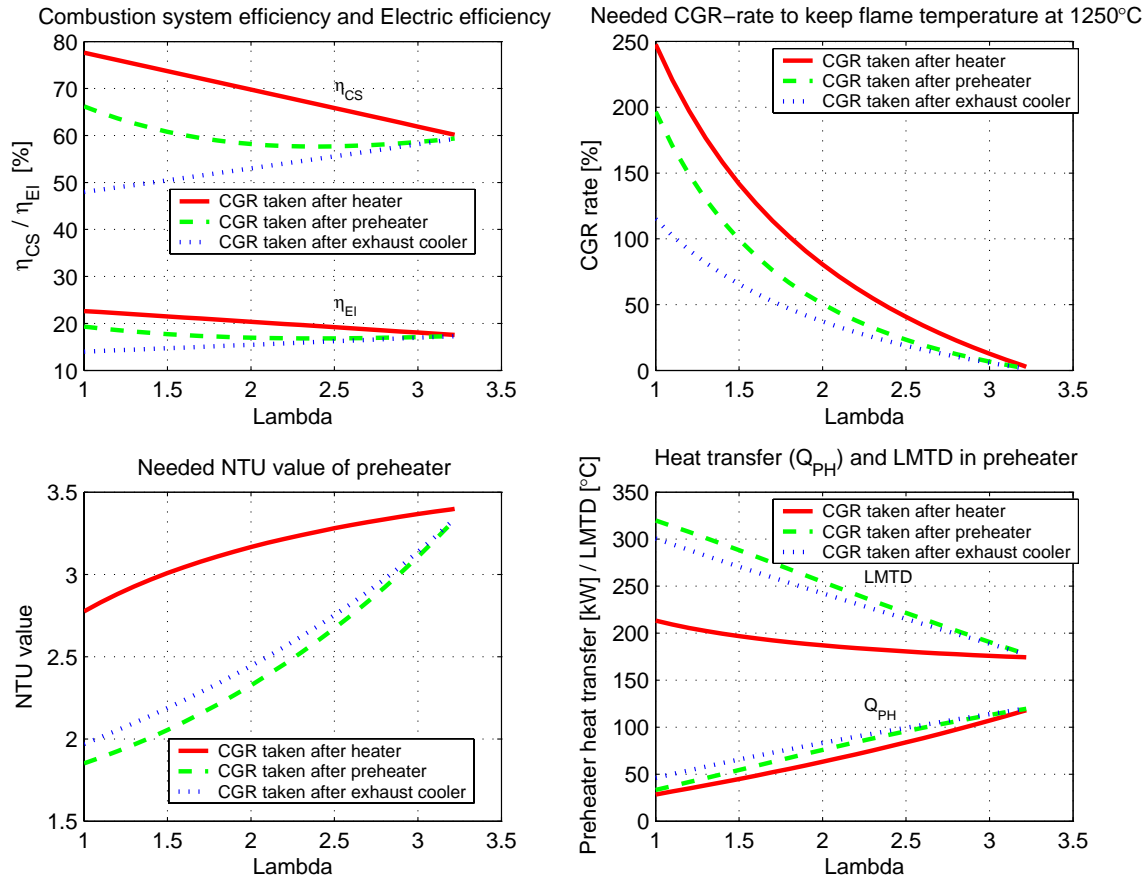

Figure 4   Calculated efficiencies, needed CGR-rate and preheater parameters

What can be seen from Figure 4 is that the combustion system efficiency η_{CS} increases with decreasing λ, or – rather – increases with increasing recirculation rate, which can be explained by the fact that the CGR-flow does not have to leave at an elevated temperature through the chimney, "leaking" energy from the system. The exception is when the CGR-flow is taken from downstream the exhaust cooler (where some of the exhaust energy has already left the system through the exhaust cooler), however this alternative has such low efficiency as to make the use of it unrealistic. Best efficiency can be reached when the CGR-flow is tapped directly downstream of the Stirling engine heater, which can be intuitively understood if we regard the CGR-flow as a bulk gas flow circulating through the combustion chamber and the heater, at the same time moving heat from the flame to the heater at 100% efficiency as this (hypotetical) flow will never pass the preheater nor leave through the exhaust pipe.
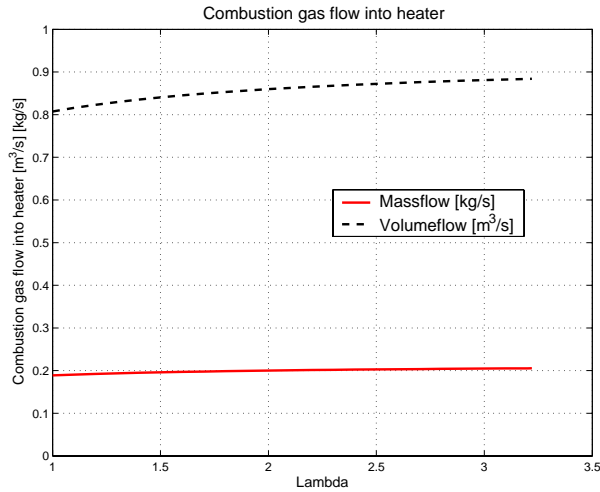
Figure 5   Calculated combustion gas mass and volume flows at the heater inlet

From Figure 5 we see that the massflow through the Stirling heater, though almost constant, is a function of lambda (and engine power) but independent of the choice of recirculation alternative. This is due to the fact that the flame temperature (i.e. the heater inlet temperature) as well as the heater outlet temperature are kept constant (in the calculations), as is the engines power consumption. However, if we change λ we change the heat capacity of the combustion gas, and thus the massflow changes to keep the power delivered to the engine constant.
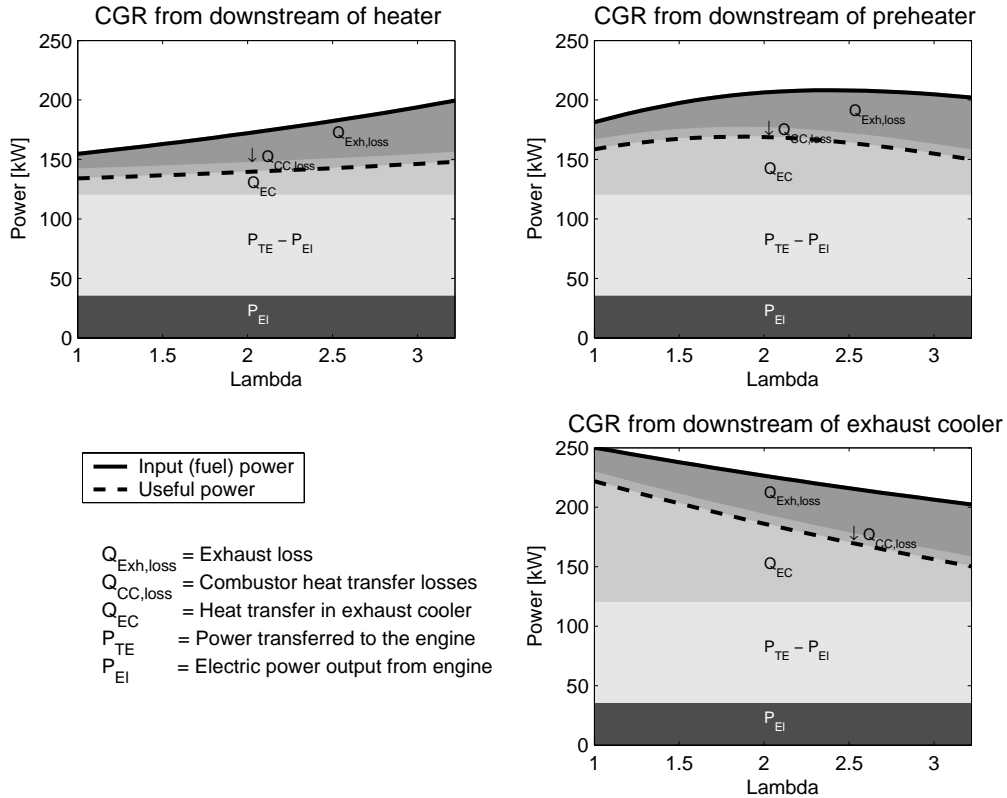

Figure 6   Calculated energy distribution for a wood powder burner system at varied λ, preheating and CGR

SYSTEM DESIGN

From the results of the qualitative system calculations described above, a suitable system design point was chosen for further evaluations. Then hardware specific calculations were made in order to decide on the final design. Calculation results for the final design are shown in Table 2 below.

Lambda, λ:	1.3	
Fuel water content:	8	%
Combustion gas massflow	0.225	kg/s
Combustion gas recirculation rate:	180	%
Maximum adiabatic flame temperature:	1331	°C
Heater inlet temperature:	1255	°C
Heater outlet temperature:	920	°C
Heater power:	123	kW
Radiative part of heater power:	23	kW
Fuel power:	165	kW
Pressure drop across heater:	4	mm H_2O
Preheater efficiency:	65	%

Table 2   Calculation inputs and results for final hardware design

The unit is designed with a set-up according to Figure 2, using air preheating and recirculation alternative 1. Initially, the unit will use a high-temperature CGR blower to recirculate combustion gas from the heater outlet directly to the combustion chamber. Later other methods for driving the recirculation flow will be evaluated. An air preheater and an exhaust cooler have been designed and adapted for using with exhausts containing ash. To give optical access to the flame, a number of pairs of small windows have been fitted on the combustion chamber in such a manner as to give multiple lines of sight through the flame at different locations and directions. First engine operation is scheduled for early spring, 2004. Figure 8 shows the engine with attached combustion chamber, air preheater and exhaust cooler. Figure 7 shows a photo of the combustion chamber.

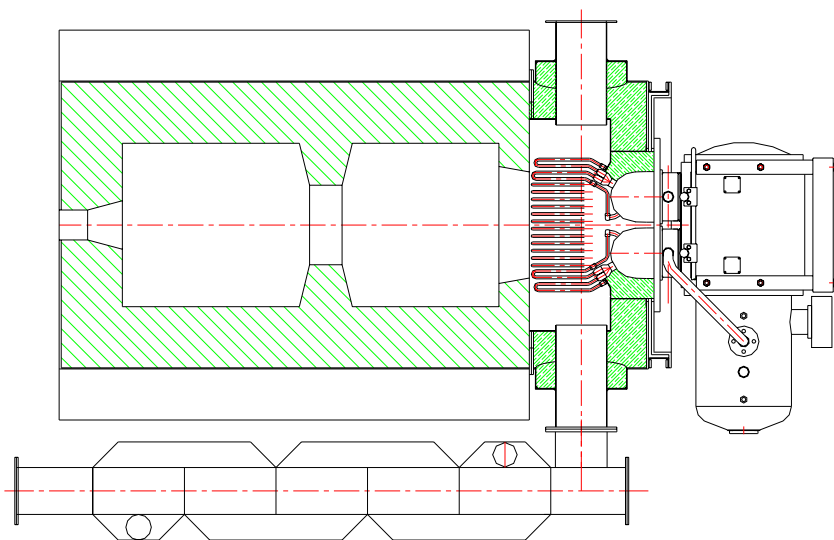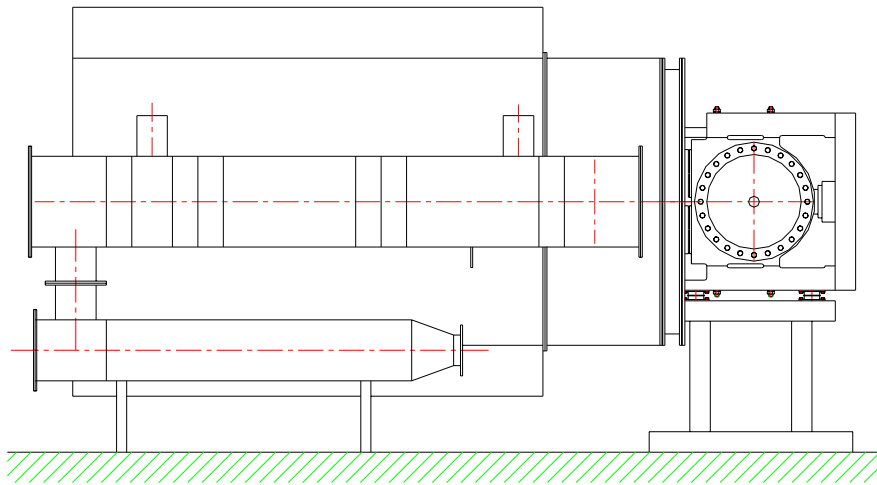**Figure 7   The combustion chamber**

Figure 8   **Combustion chamber and engine with preheater and exhaust cooler (mounted below the preheater)**

REFERENCES

1. Carlsen, Henrik, Bovin, Jonas, Ammundsen, Niels: "*Field Test of 40 kW Stirling Engine for Wood Chips*". 8th International Stirling Engine Conference Conference, ISEC97073, May 27 - 30, 1997, Ancona, Italy

2. Carlsen, Henrik, Bovin, Jonas: "*Progress Report – 35 kW Stirling Engines for Biomass*", European Stirling Forum, Febr. 2000, Osnabrück, Germany

3. Pålsson, Magnus: "*On Premixed Gas Fuelled Stirling Engine Combustors with Combustion Gas Recirculation*", Doctoral Thesis, Lund University, May 2002, Lund, Sweden, ISBN 91-628-5227-2