

OPERATING EXPERIENCES WITH A SMALL-SCALE CHP PILOT PLANT BASED ON A 35 kW_{EL} HERMETIC FOUR CYLINDER STIRLING ENGINE FOR BIOMASS FUELS

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ABSTRACT:

Within the scope of the RD&D project presented a small-scale CHP plant with a hermetic four cylinder Stirling engine for biomass fuels was developed and optimised in cooperation with the Technical University of Denmark, MAWERA Holzfeuerungsanlagen GesmbH, an Austrian biomass furnace and boiler manufacturer, and BIOS BIOENERGIESYSTEME GmbH, an Austrian development and engineering company. Based on the technology developed, a pilot plant was designed and erected in Austria. The nominal electric power output of the plant is 35 kW_{el} and the nominal thermal output amounts to approx. 220 kW_{th}. The plant operates fully automatically. Currently, this plant has run for more than 5,000 hours using wood chips as fuel with very satisfactory results. Problems to be addressed are primarily regarding an enhanced electric efficiency. Major emphasis will be placed on improving the efficiencies of the hot heat exchanger of the Stirling engine, of the air preheater and of the entire combustion system. Furthermore, the optimisation of the pneumatic cleaning system to reduce ash deposition in the hot heat exchanger is of great relevance.

KEYWORDS:

Biomass combustion, combined heat and power (CHP), Stirling engine process, operating experiences.

1 INTRODUCTION AND OBJECTIVES

Efficient utilisation of biomass for energy can be achieved with a minimum of environmental impact if biomass is used for small-scale combined heat and power (CHP) production in smaller towns and villages close to biomass production sites as well as in the wood processing industries. In the range of an electric power output up to 150 kW_{el} Stirling engines are presently the only useful technology for CHP generation based on solid bio-fuels, which has the potential to meet the technical and economic demands in an environmental acceptable way [1, 2].

Stirling engines for CHP plants using biomass as fuel have been developed over a period of more than ten years at the Technical University of Denmark. As part of these activities a 9 kW_{el} Stirling engine for biogas and a 35 kW_{el} Stirling engine for wood chips were developed. Furthermore, within the

scope of a EU research project, a biomass CHP plant with a 75 kW_{el} eight cylinder engine was designed. The engines are developed as hermetically sealed units where the generator is incorporated in the pressurised crank case in order to avoid dynamic seals between the working gas and the surroundings [3].

Within the scope of the RD&D project presented a small-scale CHP plant with a 35 kW_{el} hermetic four cylinder Stirling engine for biomass fuels was designed in cooperation with the Technical University of Denmark, MAWERA Holzfeuerungsanlagen GesmbH, an Austrian Biomass furnace and boiler manufacturer, and BIOS BIOENERGIESYSTEME GmbH, an Austrian development and engineering company. Based on the technology developed, a pilot plant was erected and tested in Austria.

2 DESCRIPTION OF THE STIRLING CHP TECHNOLOGY

Stirling engines are based on a closed cycle, where the working gas is alternately compressed in a cold cylinder volume and expanded in a hot cylinder volume. The advantage of the Stirling engine over internal combustion engines is that the heat is not supplied to the cycle by combustion of the fuel inside the cylinder, but transferred from the outside through a heat exchanger in the same way as in a steam boiler. Consequently, the combustion system for a Stirling engine can be based on proven furnace technology, thus reducing combustion related problems typical of solid biomass fuels. The heat input from fuel combustion is transferred to the working gas through a hot heat exchanger at a high temperature, typically between 680 °C and 780 °C. The heat that is not converted into work on the shaft is rejected to the cooling water in a cold heat exchanger at 25 °C - 75 °C.

In order to obtain a high overall electric efficiency of the CHP plant, the temperature in the hot heat exchanger should be as high as possible. Therefore, it is necessary to preheat the combustion air with the flue gas leaving the hot heat exchanger by means of an air preheater. Typically the temperature of the combustion air is raised to 500 °C – 600 °C, resulting in very high temperatures in the combustion chamber. This can cause ash slagging and fouling problems in biomass combustion systems and in the hot heat exchanger.

The closed Stirling cycle makes it possible to use a working gas, which is better suited for heat transfer to and from the cycle than air. The use of Helium or Hydrogen is most efficient, but utilisation of these low molecular weight gases makes it difficult to design a piston rod seal, which keeps the working gas inside the cylinder and prevents the lubrication oil from entering the cylinder. Many solutions have been tested, but it is still a delicate component in the engine. An attractive possibility is to bypass the problem by designing the engine as a hermetically sealed unit with the generator incorporated in the pressurised crankcase, just like the electric motor in a hermetically sealed compressor for refrigeration. Only static seals are necessary and the only connections from the inside to the outside of the hermetically sealed crankcase are the cable connections between the generator and the grid.

The problems concerning utilisation of biomass fuels in connection with a Stirling engine are concentrated on transferring the heat from the combustion of the fuel into the working gas. The temperature must be high in order to obtain an acceptable specific power output and efficiency, and the heat exchanger must be designed so that problems with fouling are minimised.

Because of the high temperatures in the combustion chamber and the risk of fouling, it is not possible to utilise a Stirling engine designed for natural gas, as narrow passages in the hot heat exchanger are blocked after less than an hour of operation with biomass fuels. The risk of fouling in biomass combustion processes is mainly due to aerosol formation and condensation of ash vapours when the flue gas gets cooled [4].

3 SPECIFICATIONS AND DESCRIPTION OF THE 35 kW_{EL} STIRLING ENGINE

A Stirling engine especially designed for CHP plants using biomass fuels has been developed at the Technical University of Denmark. The design of the Stirling engine is based on numerical optimisation of more than 20 parameters describing cylinders, heat exchangers, regenerators etc. [5]. The engine, which is designed for a nominal electric capacity of 35 kW, 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 four hot heat exchangers (one for each cylinder) are designed as panels forming a square combustion chamber, where radiation from the combustion is transferred directly to the panels. Narrow passages in the hot heat exchanger sections are avoided in order to adapt the system to flue gases from combustion systems fired with solid fuels.

The engine is designed as a hermetically sealed unit. The built-in asynchronous generator, which is also used as starter motor, has 6 poles corresponding to an engine speed of approximately 1000 rpm when coupled directly to the power grid (50 Hz AC).

The design of the SM3C Stirling engine for the plant described here is based on experiences from a first and a second prototype, which have already been tested. The first prototype, SM3A, was tested for more than 1,400 hours with wood chips as fuel, before the tests had to be stopped due to mechanical problems [3]. The second engine, SM3B, was improved considerably compared to the first engine, and the test results were very satisfactory [6].

Table 1: Specifications of the 35 kW_{el} Stirling engine

Nominal electric power	kW	35
Bore	mm	142
Stroke	mm	76
Number of cylinders		4
Speed	rpm	1,010
Mean pressure	Mpa	4.5
Working gas		Helium
Temperature of hot heat exchanger	°C	750
Engine weight	kg	1,600

The specifications of the new Stirling engine are shown in Table 1. The improvements of this engine compared to the first and the second prototype are improved reliability of the crank mechanism, improved cooling of the piston rod seals and a new design of the hot heat exchanger. The dimensions of the yokes in the patented crank mechanism were increased in order to improve rupture strength. Furthermore, several bearings were replaced with a larger type, thus improving the lifetime of the engine.

A new piston rod cooling system was tested on engine No. 2 (SM3B), but the design was very complicated. Cooling of the piston rod improves the lifetime of the seal and considerably extends service intervals, as the piston rod seal is a dynamic seal and thus subjected to the highest load. The new engine is also provided with piston rod coolers, but the system has been considerably improved and the results have been very satisfactory. The objective is to obtain replacement intervals of 15,000 hours of operation.

The new design of the hot heat exchanger was necessary in order to adapt it to a maximum flue gas temperature in the combustion chamber of 1,300°C (the first and the second prototype were designed for flue gas temperatures of 1,600 °C). As a consequence of the reduction of the maximum temperature in the combustion chamber, the mass flow of flue gas had to be increased. Furthermore, less heat is transferred as radiation compared to the previous design.

The hot heat exchanger consists of 23 tubes with an outside diameter of 13.7 mm. The tubes are U-formed connecting the cylinder manifold with the regenerator manifold. In the old design, half of these tubes formed the radiation panel of the hot heat exchanger, while the other half were responsible for the convective heat transfer. As shown in Figure 1 the new design has only 12 half-tubes for the radiation part, while the remaining 36 half-tubes are used for convective heat transfer. Fins are used for enhancing the heat transfer area. Figure 2 shows the engine with the hot heat exchanger ready for installation in the furnace. The water-cooled generator, which is part of the hermetically sealed design, can be seen in front.

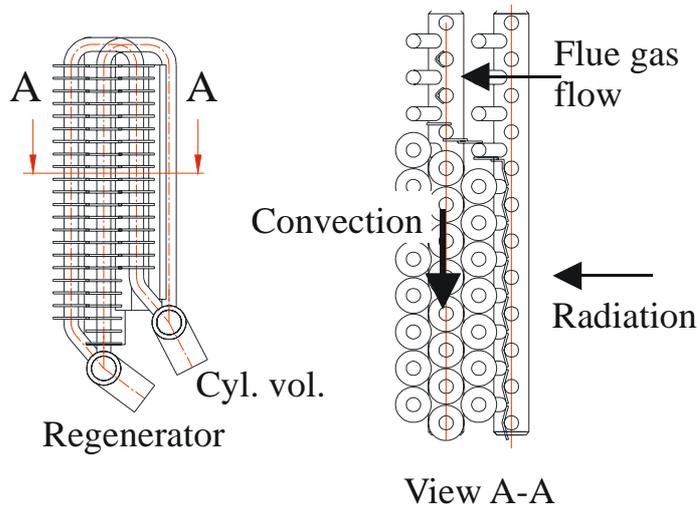


Figure 1: Sketch of hot heat exchanger panel



Figure 2: 35 kW Stirling engine before installation in the furnace

4 CFD SUPPORTED DESIGN AND OPTIMISATION OF THE FURNACE

The design of the furnace and its adaptation to the special requirements of a CHP plant with a 35 kW_{el} Stirling engine is an important and difficult task. The plant should operate at a high temperature level to gain a high electric efficiency from the Stirling engine but temperature peaks in the furnace should be impeded in order to reduce ash slagging and fouling. The plant is designed for temperature levels in the furnace of about 1,300 °C.

The new combustion system was developed and optimised using CFD simulations. The results achieved showed that it is a very important task to optimise the design of the furnace geometry, of the secondary air nozzles and the nozzles for flue gas recirculation in order to reduce temperature peaks in the furnace as well as CO emissions. In addition, the CFD simulations performed improved an equal distribution of the flue gas flow into the 4 different sections of the hot gas heat exchanger and thus ensured an equal heat transfer to the 4 cylinders of the Stirling engine.

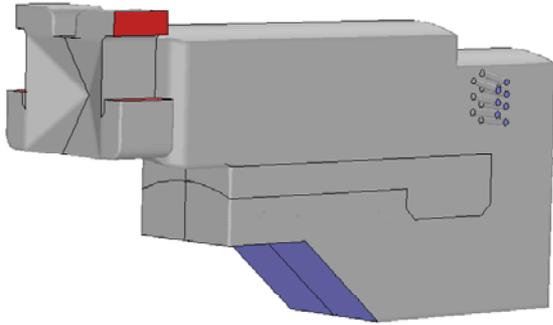


Figure 3: Schematic illustration of the furnace design – conventional design

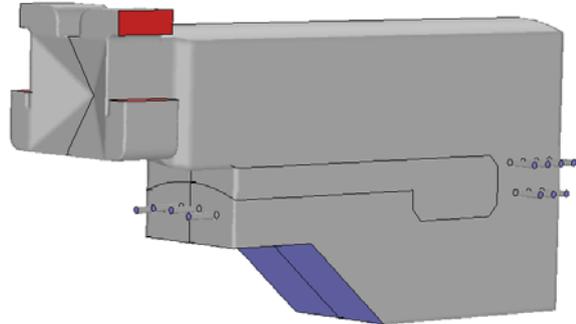


Figure 4: Schematic illustration of the furnace design – optimised design

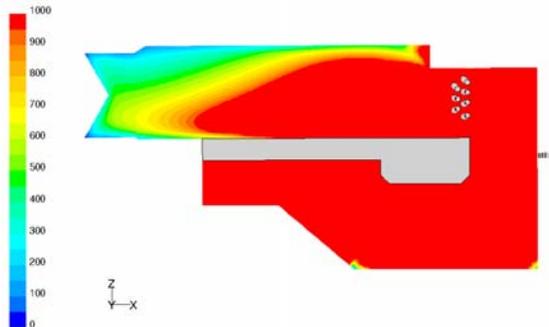


Figure 5: Contours of CO in mg/Nm^3 at the symmetry plane of the furnace – conventional design (CO emissions at furnace outlet: approx. $100 \text{ mg}/\text{Nm}^3$ related to dry flue gas and 13 Vol% O_2)

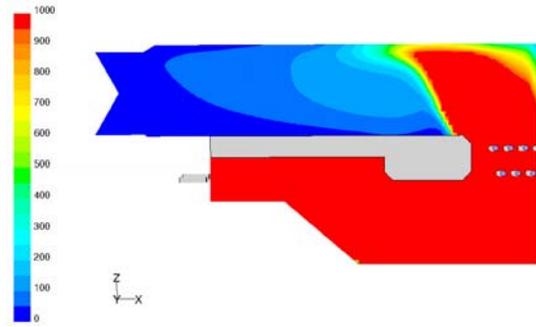


Figure 6: Contours of CO in mg/Nm^3 at the symmetry plane of the furnace – optimised design (CO emissions at furnace outlet: approx. $15 \text{ mg}/\text{Nm}^3$ related to dry flue gas and 13 Vol% O_2)

Figure 3 shows the geometry of the furnace with conventional nozzle design and placement. The secondary air nozzles are placed at the inlet of the secondary combustion chamber. The results of the CFD simulations performed for this geometry show that the flue gas burn out in the secondary combustion chamber is not efficient (see Figure 5). The CO emissions at outlet according to CFD simulations are about $100 \text{ mg}/\text{Nm}^3$ (dry flue gas, 13 Vol% O_2).

Figure 4 shows a furnace geometry with optimised nozzle design and placement. The secondary air nozzles are arranged horizontally in the transition zone between primary and secondary combustion chamber. With this configuration the combustion air is efficiently mixed with the flue gas and a swirl flow is established in the secondary combustion chamber. Consequently, the resulting CO emissions are low. Figure 6 shows the contours of CO in mg/Nm^3 calculated for the geometry with optimised air nozzles. For the optimised geometry, CFD simulations predict CO emission of approx. $15 \text{ mg}/\text{Nm}^3$ (dry flue gas, 13 Vol% O_2). The results demonstrate that an efficient turbulent flow enhances the combustion process and reduces CO emissions, which stresses the relevance of an optimisation of the combustion system supported by CFD analyses.

Based on the results achieved from the CFD calculations performed a small-scale CHP plant for biomass fuels based on the Stirling engine described above was developed and designed in cooperation between MAWERA Holzfeuerungsanlagen GmbH, an Austrian Biomass furnace and boiler manufacturer, the

Technical University of Denmark and BIOS BIOENERGIESYSTEME GmbH, an Austrian development and engineering company. Figure 7 shows a picture of the new CHP pilot plant. The furnace is equipped with underfeed stoker technology. The Stirling engine is mounted in a horizontal position downstream of the secondary combustion chamber for convenient maintenance (see Figure 7). The air preheater and the economiser are placed on top of the furnace in order to achieve a compact design of the plant. The CHP plant should not require substantially more space than a normal biomass combustion plant with the same heat output. To remove fly ash particles from the hot gas heat exchanger, a pneumatic and fully automatic cleaning system was developed and installed.

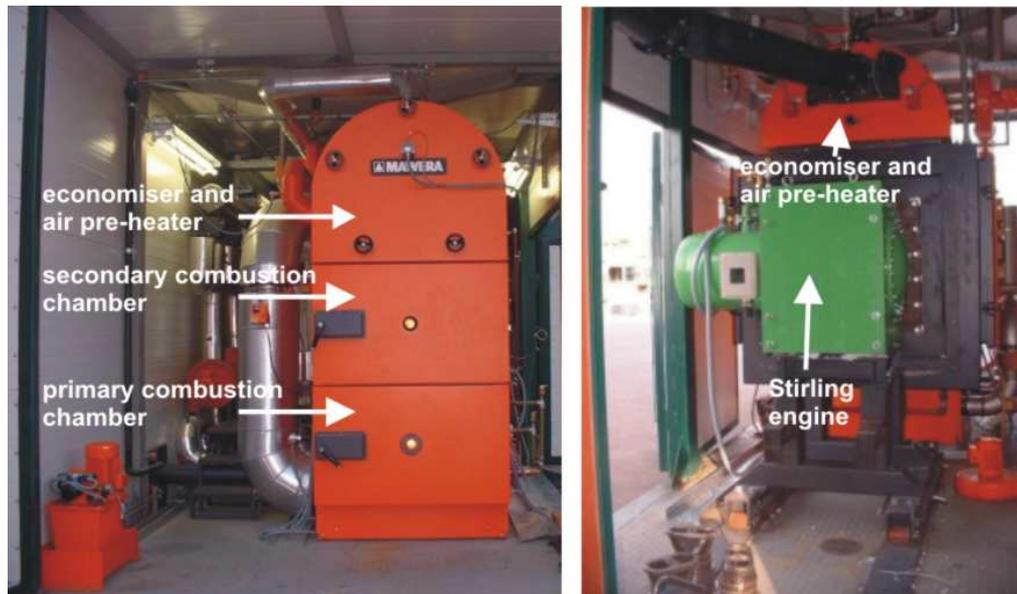


Figure 7: Pictures of the small-scale CHP pilot plant with a 35 kW_{el} Stirling engine

To keep the operating costs as low as possible, small-scale plants will have to run in unmanned operation for days or weeks. The system has therefore been fully automated. Especially, start-up and shutdown of the plant is performed automatically (except heat-up procedure). If any engine failure is detected the combustion system will immediately be shut down without operator intervention necessary.

5 OPERATING EXPERIENCES AT THE PILOT PLANT WITH A 35 kW_{EL} STIRLING ENGINE

The plant was put into operation at the end of summer 2002 and has been running fully automatically since autumn 2002. The engine has run for more than 5,000 hours until the end of July 2003 and for more than 3,600 hours from the beginning of February to the end of July 2003 at a high load level. In this period, the average availability of the pilot plant was more than 80% , in May and June 2003 even more than 92%.

In spring 2003 comprehensive test runs were performed. Table 2 shows the results of the test runs compared to design targets. The average electric power output of 31 kW_{el} is less than the expected nominal output of 35 kW_{el} achieved in tests with natural gas. The decrease in power is partly due to a lower efficiency of the hot heat exchanger with wood chip fuel compared to natural gas. Furthermore, the temperature of the cooling water at the cold heat exchanger inlet of the Stirling engine was higher than initially foreseen, which also results in a reduced electric power output. The electric efficiency of the Stirling engine is also lower than expected (20 instead of 25%), which is mainly due to the fact that the Stirling engine is significantly less efficient at partial load than at full load operation. Furthermore, the air preheater does not work satisfactorily. The test runs performed show that the temperature of the preheated combustion air is about 190 °C lower than initially foreseen (see Table 2). This is due to

high heat losses to the water cooled walls enclosing the air preheater. The overall electric efficiency of the CHP plant measured during the test runs amounts to approx. 9.2 %, which is about 25% less than expected. The overall efficiency of the CHP plant (electric + thermal) is higher than the design target, which shows that the economiser works very satisfactorily.

During the test runs different wood chip qualities were used, and the plant was running well with water contents ranging from 10 to 55 wt.% (w.b.).

Table 2: Average results of test runs performed at the 35 kW_{el} pilot plant compared to design targets

		Design target	Obtained during test runs
Temperature of the preheated air	°C	550	360
Temperature of the cooling water at Stirling cooler inlet	°C	55	61
Electric power output	kW	35	31
Thermal power output - Stirling engine	kW	105	124
Thermal power output - CHP plant	kW	215	272
Fuel power input	kW	291	337
Fuel consumption (w.b.) (water content approx. 30 wt%)	kg/h	85	96
Electric efficiency - Stirling engine	%	25.0	20.0
Overall electric efficiency - CHP plant	%	12.0	9.2
Overall efficiency - CHP plant	%	85.9	90.0

The automatic cleaning system was improved several times during the test period. At the moment, manual cleaning of the hot heat exchanger is necessary after more than one month of operation. It is expected that further improvements of the automatic cleaning system will increase these intervals to 2 - 3 months.

The Stirling engine was disassembled for inspection after 5,000 operating hours. The results of this inspection showed that the major parts of the Stirling engine have only small or no wear but the design of one bearing and of the sealings of the piston and the piston rods has to be improved. These parts will be adapted and replaced. It is planned to reinstall the plant in the end of October 2003 and continue with field tests.

6 CONCLUSIONS

With more than 5,000 hours of successful operation the pilot plant described above can be considered as a breakthrough in the utilisation of Stirling engines for small-scale CHP plants utilising wood chip fuels. At the moment, the plant works fully automatically and the development of the control system is almost completed. But there are several problems to be addressed in future, primarily regarding enhanced electric efficiency. In this context, major emphasis will be placed on improving the efficiencies of the hot heat exchanger, of the air preheater and of the entire combustion system. Furthermore, the optimisation of the pneumatic cleaning system to reduce ash deposition in the hot heat exchanger and thus to achieve a higher availability of the whole system is of great relevance. This work is continued in an Austrian R&D project with international partners performed within the frame of the Austrian Bioenergy Centre.

In addition to these development goals, future activities should further prove the reliability and low service demand of the plant. Furthermore, the specific price of the plant has to be decreased by implementing serial production on a stage-by-stage basis in order to make the technology competitive. A small series production of Stirling engines is planned to be launched in 2004, and it is expected that

the price of manufacture can be considerably reduced, once the first two or three small series have been built.

In an ongoing EU research project (project “BIO-STIRLING”; project No. NNE5-1999-00097) a CHP plant with a 75 kW_{el} eight cylinder Stirling engine has been developed. This plant was put into operation in summer 2003 and comprehensive field tests at this plant are planned for autumn 2003.

7 ACKNOWLEDGEMENTS

The project presented was supported by the Austrian Innovation and Technology Fund (ITF project No. 801.848) and by the European Commission under the Fifth Framework Programme (project “BIO-STIRLING”; project No. NNE5-1999-00097).

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