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EXTERNALLY WITH BIOMASS AND INTERNALLY WITH NATURAL GAS FIRED MICRO GAS TURBINE – SYSTEM, FURNACE AND HIGH TEMPERATURE HEAT EXCHANGER DESIGN AS WELL AS PERFORMANCE DATA FROM FIRST TEST RUNS

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ABSTRACT: The work presented in this paper is based on the EU FP6 project “Innovative small-scale polygeneration system combining biomass and natural gas in a micro gas turbine (BIO-MGT)”, which aims on the development of a small-scale biomass CHP technology based on an externally with biomass and internally with natural gas fired micro gas turbine (MGT) with a nominal electric capacity of about 100 kW_{el}, where the MGT exhaust gas should be used as efficiently as possible and the biomass contribution should be as high as possible (target: > 50% based on NCV). This new technology aims to close the gap in the power range between 70 and 200 kW_{el}, in which no biomass CHP technology is currently available (i.e. the gap between the Stirling engine (< 70 kW_{el}) and the ORC (> 200 kW_{el}) technology). The development of a system which complies with these requirements took place. A new biomass furnace technology was developed for the complete and exclusive utilisation of turbine exhaust gas as oxidising agent for combustion. A high temperature heat exchanger was designed to transfer heat from the flue gas of the biomass furnace to the pressurised air of the MGT cycle. The plant was started up in January 2010. First operating experiences could already be gained. Test runs at various loads are scheduled for the next half year.
Keywords: combined heat and power (CHP), combustion, demonstration, gas turbine, microturbine, small scale applications

1 INTRODUCTION

The work presented in this paper is based on the EU FP6 project “Innovative small-scale polygeneration system combining biomass and natural gas in a micro gas turbine (BIO-MGT)”. The main aim of the project is the development of a small-scale biomass CHP technology based on an externally with biomass and internally with natural gas fired micro gas turbine (MGT) with a nominal electric capacity of about 100 kW_{el}, where the MGT exhaust gas should be used as efficiently as possible and the biomass contribution should be as high as possible (target: > 50% based on NCV). This new technology aims to close the gap in the power range between 70 and 200 kW_{el}, in which no biomass CHP technology is currently available (i.e. the gap between the Stirling engine (< 70 kW_{el}) and the ORC (> 200 kW_{el}) technology) [1; 2; 3; 4]. The focus of this paper is on the optimisation of the system configuration as well as on the furnace and high temperature heat exchanger design. Overall plant and MGT related issues are presented in [5]. Previous publications related to this project can be found in [6; 7; 8].

2 SYSTEM DESIGN AND OPTIMISATION

The basic idea of the BIO-MGT concept is to heat up the exhaust gas from a MGT (turbine exhaust gas, TEG) to a certain temperature by mixing it with flue gas from a biomass furnace and to transfer heat via a high temperature heat exchanger (HT-HE) from this gas mixture to the pressurised air before entering the combustion chamber of the MGT where natural gas is used to further increase the turbine inlet temperature and thus the electric efficiency.

Different configurations, how the turbine exhaust gas could be mixed with flue gas from the biomass furnace most efficiently, have been investigated and an optimisation based on thermodynamic calculations with regard to the share of biomass related to the total primary energy input and the electric efficiency (= net electric power produced / fuel power input related to NCV) achievable took place. The configurations investigated were:

- Configuration 1: operation of the biomass furnace with ambient combustion air and mixing the TEG with the flue gas in a specially developed mixer.
- Configuration 2: like configuration 1 but with combustion air pre-heating via a heat exchanger utilising a part of the exhaust heat of the flue gas after the HT-HE.
- Configuration 3: use of ambient air as combustion air in the primary combustion zone and injection of the TEG into the secondary combustion zone.
- Configuration 4: use of the total amount of TEG in the biomass furnace as oxidising agent (no use of ambient air).

The different configurations had to be evaluated with regard to the plant engineering point of view (required space, required components) and the materials which can be used (in particular in the primary combustion zone). Another important issue is related to the efficiencies and the biomass contribution achievable.

Configurations 1 and 2 must be regarded as disadvantageous in terms of required space and components, because both configurations need a separate mixer, configuration 2 requires an additional air pre-heater. In contrast, configurations 3 and 4 do neither require an external mixer nor an air pre-heater, as the mixing zone is integrated in the biomass furnace and no air pre-heating is done. Thus, the latter two save space and investment costs and reduce heat losses as a hot gas duct between furnace and mixer is not needed. Although the integration of the mixing zone into the furnace required a new design of the biomass furnace it can not be seen as a disadvantage, because the mixer would also have to be newly designed. On the other hand, an advantage of configurations 1 and 2 is the fact, that a conventional biomass furnace and control system could be used. An additional challenge arises in configurations 3 and 4, as the biomass furnace must always use the amount of TEG provided from the MGT. Thus, the control system must be adapted accordingly, so that the fuel feed controls the load (without influence of the combustion air).

In terms of materials that can be used in the primary combustion zone, configurations 1, 2 and 3 must be regarded as more advantageous than configuration 4. In the configurations 1 to 3 ambient air (in case of configuration 2 pre-heated air with 300°C) is used as combustion air, which enables the application of conventional combustion chamber materials. In contrast, configuration 4 uses TEG as oxidising agent in the primary and secondary combustion zone, which requires the use of higher quality combustion chamber and grate materials, which are able to cope with combustion air (TEG) temperatures of 645°C.

In terms of the efficiencies and biomass contribution achievable some important framework conditions must be mentioned, before these parameters can be evaluated. The HT-HE inlet temperature at the flue gas side has been limited to 925°C due to restrictions in terms of mechanical and creep strength of stainless steels at such high temperatures. The temperature in the primary combustion chamber has been limited to 1,200°C and in the secondary combustion chamber to 1,100°C in order to avoid slagging problems. Moreover, the power output from the MGT is the same for all configurations and amounts to about 100 kW_{el}. Additionally, also the mass flow rate and the temperature of the TEG are the same for all configurations and amount to about 0.8 kg/s, respectively 645°C (all data are related to nominal load). These framework conditions together with the given HT-HE design and the required turbine inlet temperature of 950°C determine for each configuration the required biomass fuel and natural gas input as well as the excess air ratio lambda (for configuration 1 and 2).

For configuration 1 the biomass contribution (based on the NCV) would be around 93.5%. However, the electric efficiency would only be around 7% and thus is quite low.

Due to the air pre-heating in configuration 2 the air ratio would be increased in order to keep the same combustion chamber temperature and the biomass fuel input will decrease to keep the same HT-HE inlet temperature. Consequently, the biomass contribution slightly decreases to 92.1% and the electric efficiency slightly increases to 8.4%.

Due to the substitution of the primary air by TEG in configuration 3, the amount of biomass fuel needed to reach the same HT-HE inlet temperature is reduced. This leads to a decrease of the biomass contribution to about 88% and to an increase of the electric efficiency to about 13%.

If the whole amount of combustion air is substituted by TEG, which has been done in configuration 4, the electric efficiency increases to 21.2% at a biomass contribution of more than 75%.

Taking all aspects (complexity and space demand of the system, materials to be used for primary combustion zone and the grate, efficiencies and biomass contribution) into account, it was decided to further follow configuration 4, since it provides by far the highest electric efficiency at acceptably high biomass contributions and the lower complexity of the whole system. Against the background of these advantages the disadvantages regarding the higher quality materials needed for the construction of the grate and the primary combustion zone are of minor relevance.

In parallel to the investigation of these different configurations the HT-HE was designed (see Section 4). During these activities it turned out, that the inlet temperature in the HT-HE must be limited to 850°C (instead of the 925°C previously taken as a basis for configurations 1 to 4). Moreover, in order to reduce the flue gas velocities (and thus particle entrainment from the fuel bed) and temperature peaks in the primary combustion zone, the possibility to operate with a lower primary air ratio, namely 0.5, has been investigated by CFD (Computational Fluid Dynamics) simulations. The CFD simulations have shown that the adiabatic temperature in the primary combustion chamber is increased slightly to about 1,250°C by this measure, which can, however, be handled by a proper design of the primary combustion zone so that temperature peaks at the arch and the combustion chamber walls do not occur. With regard to the charcoal burnout no problems are expected due to the high temperature of the TEG used as an oxidising agent in the furnace. Therefore, these measures have been considered for the final design and this led to a new operating mode of configuration 4 with reduced flue gas inlet temperature in the HT-HE and lower primary air ratio. In addition to the advantages already mentioned for configuration 4 the development risk for the HT-HE and its investment costs got reduced, as the thermal stress on the material is lower. Moreover, this configuration shows the highest electric (23.5%) and overall (76.3%) efficiency at an acceptable biomass contribution (50%).

An overview of the relevant technical data of the micro CHP system is shown in Table I. A scheme of the final design is shown in Figure 1.

Table I: Relevant technical data of the micro CHP system

Parameter	Value	Unit
Biomass fuel power input	219.5	kW_{NCV}
Natural gas fuel power input	221.1	kW_{NCV}
Electric capacity	103.7	kW_{el}
Thermal capacity (at 100°C flue gas temperature)	249.8	kW_{th}
Electric efficiency (gross)	23.5	%
Thermal efficiency (at 100°C flue gas temperature)	56.7	%
Total efficiency (at 100°C flue gas temperature)	80.2	%
Biomass contribution	49.8	%

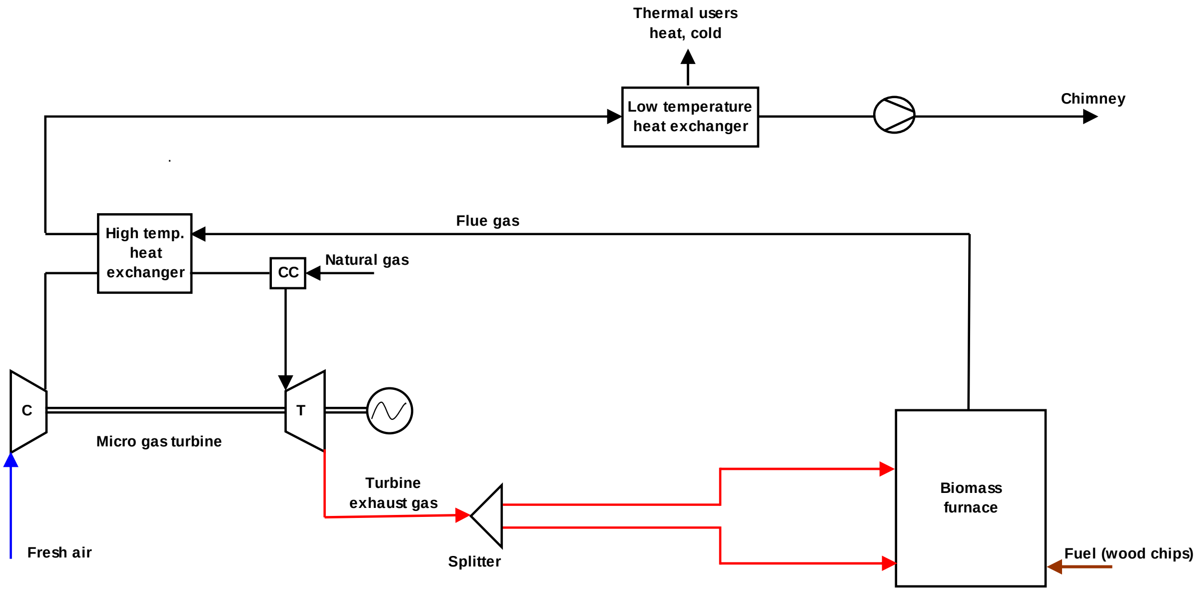

Figure 1: Final configuration of the micro CHP system
Explanations: C…compressor; T…turbine; CC…combustion chamber

3 BIOMASS FURNACE DESIGN

The geometry of an underfeed stoker with two horizontal ducts (i.e. with secondary and tertiary combustion zone) has been taken as a starting point. The feed of TEG as an oxidising agent into the primary combustion zone takes place below the grate and the injection into the secondary combustion zone should take place at the crossover between primary and secondary combustion zone. The flue gas passes through the secondary combustion zone before it leaves the furnace and enters the HT-HE.

The challenges concerning the design of the biomass furnace were related to the comparatively high mass flow of turbine exhaust gas, which has to be used as an oxidising agent, and its quite high temperature (645°C at the design point). The average temperature in the secondary combustion zone should be kept below 1,000°C and no temperature peaks should occur.

During the basic design phase CFD simulations have been performed and several potential optimisation measures could be identified.

Firstly, due to the intensive mixing of TEG and flue gas from the primary combustion zone a rapid gas phase burnout takes place and very low CO and organic carbon emissions are expected at furnace outlet. Thus, the tertiary combustion zone, which has primarily been considered for safety reasons to ensure low CO and organic carbon emissions, could probably be removed again.

Both, in the primary as well as in the secondary combustion zone temperature peaks have been identified by the CFD simulations. To reduce these temperature peaks an increase (in height) both of the primary and secondary combustion zone has been seen as a possible solution.

At the edge of the arch between primary and secondary combustion zone comparatively high flue gas velocities were identified, which would cause high material wear in this region. In order to reduce these velocity peaks, the possibility to reduce the length of the arch between primary and secondary combustion zone has been considered. Moreover, the change of the positions of the secondary air nozzles as well as their number has been considered to further reduce the flue gas velocity.

The aforementioned optimisation potentials have been implemented in the optimised design, for which again CFD simulations have been performed.

The removal of the tertiary combustion zone led to a slight increase of the CO emissions at furnace outlet, which are, however, still very low (see Figure 4). The temperature peaks in the primary and secondary combustion zone could be reduced. Therefore, both the removal of the tertiary combustion zone as well as the increase of the height of the primary and secondary combustion zone have been adopted for the final design.

The reduction of the length of the arch between primary and secondary combustion zone, the re-positioning of the secondary air nozzles as well as the increase of the height of the combustion zones contributed to a significant decrease of the flue gas velocities. Therefore, also these measures have been adopted for the final furnace design.

In addition, in the optimised design a better mixing of secondary air with the flue gas from the primary combustion zone could be achieved, which is of great importance in order to homogenise the temperature profiles in the secondary combustion zone and to avoid temperature peaks. Therefore, no excessive thermal stress compared to conventional biomass furnaces is expected.

Besides these geometric changes the reduction of the primary air ratio to 0.5 has been investigated with the aim to reduce the flue gas velocities in the primary combustion zone and thus particle entrainment from the fuel bed and to reduce temperature peaks. The simulations showed positive impacts and therefore this operation mode has been taken into account for the final design.

Figure 2 shows a comparison of the temperature distributions in the biomass furnace for the basic and the optimised design. It can be seen, that the temperature peaks in the primary combustion zone could clearly be reduced by the optimisation measures taken.

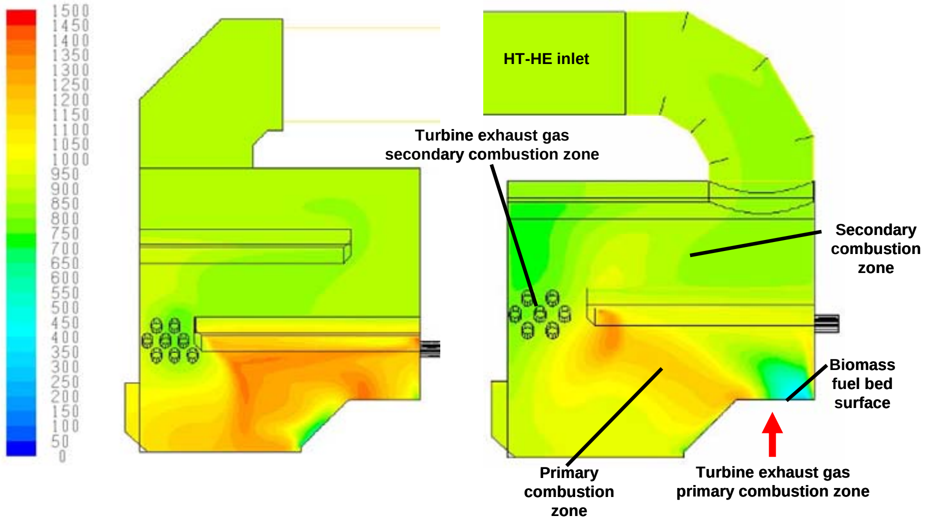

Figure 2: Comparison of the temperature distributions in the biomass furnace for the basic and optimised design
Explanations: left: basic design; right: optimised design; temperatures in °C

The increase of the height of the primary and secondary combustion zone contributed both to the reduction of temperature peaks (see Figure 2) as well as flue gas velocity peaks (see Figure 3). Moreover, not only the velocity peaks are reduced in the primary combustion zone but also the general level of the flue gas velocities. The reduction of the length of the arch between primary and secondary combustion zone as well as the removal of the lower right secondary air nozzle led to a clear reduction of the flue gas velocities in this section. Thus, material wear in this region can be minimised.

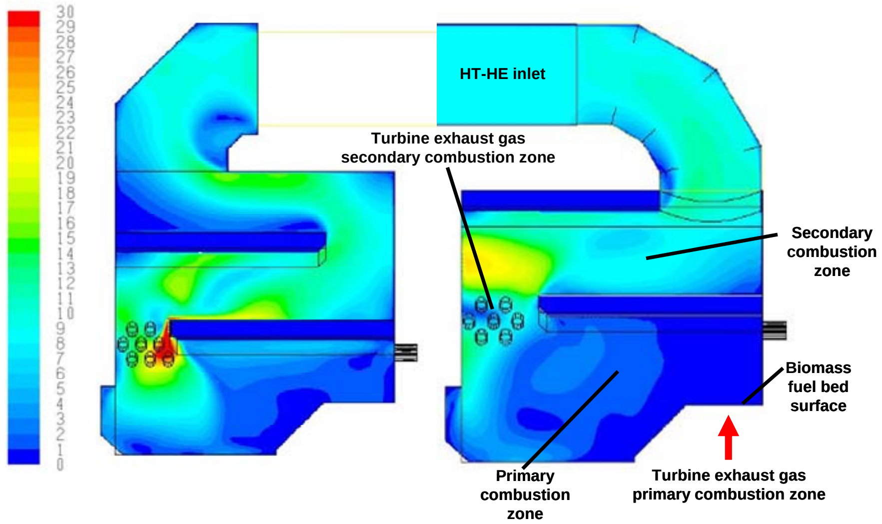

Figure 3: Comparison of the flue gas velocities in the biomass furnace for the basic and optimised design
Explanations: left: basic design; right: optimised design; flue gas velocities in m/s

An evaluation of the entrainment of particles from the fuel bed by CFD simulations showed, that directly after the fuel bed ash particles are settled and at the end of the primary combustion zone particles are entrained. Therefore, the de-ashing screw has been placed directly after the fuel bed in order to reduce ash particle entrainment with the flue gas. Moreover, the lower temperatures and flue gas velocities in the primary combustion zone lead in total to a reduced fly ash entrainment, which is of particular importance to realise a low-dust combustion and to protect the HT-HE from fouling.

Another challenge related to the biomass furnace was the control system, because a more or less constant turbine exhaust gas mass flow must be used and heated up to a certain temperature in the biomass furnace, which requires a completely different control strategy compared to conventional biomass furnaces. In conventional biomass furnaces with hot water boilers load control takes place with the feed water temperature as a guiding value, which determines fuel and primary air feed. Combustion control is usually achieved by the regulation of the secondary air feed and is usually guided by the O_2 concentration in the flue gas. In this concept, the guiding value for the load control is the flue gas temperature at the outlet of the biomass furnace, respectively the inlet temperature at the HT-HE, which must be controlled with the fuel feed. As the mass flow rate of TEG is more or less constant at nominal load and has to be used completely as an oxidising agent in the biomass furnace only the distribution of the TEG between primary and secondary combustion zone can be controlled.

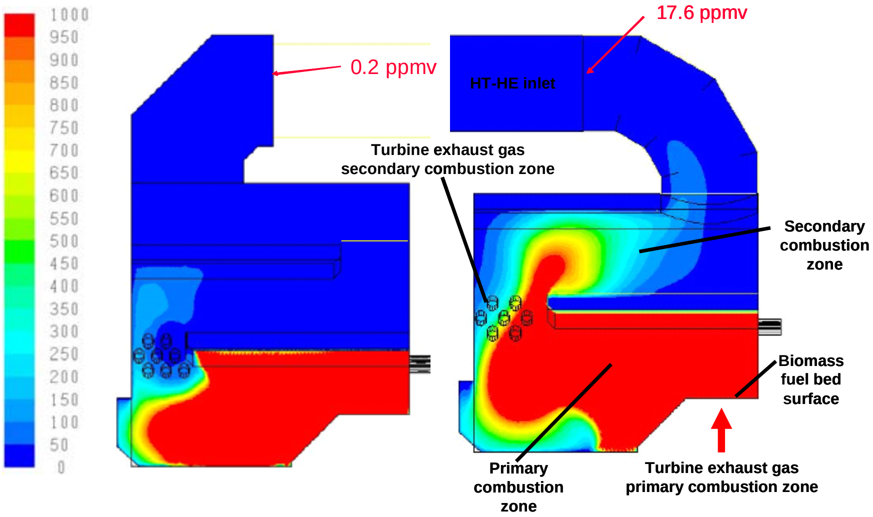

Figure 4: Comparison of the CO concentrations in the biomass furnace for the basic and optimised design
Explanations: left: basic design; right: optimised design; CO concentrations in ppmv

The fact that the load control of the biomass furnace is done by the biomass feed rate of course makes the control system slow with respect to load changes, however, this is of minor relevance since it is foreseen to run the turbine without frequent load variations (as the plant should provide base load heat).

4 HIGH TEMPERATURE HEAT EXCHANGER DESIGN

The function of the HT-HE is to transfer heat from the flue gas of the biomass furnace to the compressed air. The compressed and pre-heated air enters the combustion chamber of the MGT, where it is heated up to 950°C with natural gas before it enters the turbine.

The HT-HE has been designed as a cylindrical tube bundle heat exchanger, where the hot flue gas at atmospheric pressure flows in the tubes and the pressurised air (about 4.3 bar at nominal conditions) passes through the shell of the heat exchanger. The cylindrical construction has advantages compared to a rectangular form with regard to the high temperature levels and differences and the resulting material stress. The flue gas flow inside the tubes has been chosen as the tubes can easily be cleaned pneumatically, which enables the use of an automatic cleaning system.

The most important parameter for the HT-HE is the inlet temperature of the flue gas. With increasing flue gas inlet temperature the outlet temperature of the pressurised air increases as well and thus the heat transfer to the turbine is also increased which in turn reduces the demand of natural gas to reach the required 950°C at turbine inlet. Therefore, the aim is to reach an as high as possible flue gas inlet temperature at the HT-HE. However, with increasing inlet temperature material related problems increase. On the one hand, increased material temperatures lead to reduced mechanical strength. On the other hand, higher temperatures increase the risk for the formation of ash deposits. At the beginning a flue gas inlet temperature of 925°C has been considered, together with an air outlet temperature of 850°C. However, in-depth evaluations of possible steels have shown, that such high temperatures significantly reduce the creep strength. In combination with the high pressure at the shell side of the HT-HE it turned out to be most probably not possible to realise such a HT-HE with reasonable lifetime. Alternatively, ceramic could be used instead of steel or a ceramic supporting structure could be implemented in the steel made HT-HE. However, this would render the realisation of the HT-HE impossible from an economic point of view. Thus, it has been decided to reduce the flue gas inlet temperature at the HT-HE to 850°C. The design data of the HT-HE are summarised in Table II.

Table II: Design data of the HT-HE

Parameter	Value	Unit
Flue gas inlet temperature	850	°C
Flue gas temperature after first duct	634	°C
Flue gas outlet temperature	385	°C
Air inlet temperature (second duct)	219	°C
Air temperature after second duct	490	°C
Air outlet temperature after first duct	723	°C
Pressure air side	438	kPa
Mass flow rate flue gas	0.820	kg/s
Mass flow rate air	0.801	kg/s

The HT-HE has been designed as a two stage heat exchanger (2 ducts). Thus, the use of different materials for the first and second duct was possible. The first duct, which is exposed to high temperatures, has been made of a higher quality and more expensive material and the second duct of a lower quality and cheaper material. The perforated plates and the tubes of the first duct are made of the material with the highest quality, i.e. stainless steel 1.4876. The shells of both parts and the perforated plates of the low temperature part (second duct) are made of stainless steel 253MA. This material could also be used for the tubes of the low temperature part. However, no tubes are produced from this material and so stainless steel 1.4828 has been used (a stainless steel with similar quality like 253MA).

Moreover, due to the split of the HT-HE into two parts, the design got more compact and the total length of the plant could significantly be reduced. Furthermore, the design with two ducts requires a return chamber, where an automatic cleaning system for both parts of the HT-HE can easily be implemented.

For the final design of the HT-HE a finite element analysis has been performed. As an example, Figure 5 shows the wall temperature distribution of the first duct of the HT-HE. The finite element analysis turned out, that at the flue gas inlet between perforated plate and shell unacceptable mechanical stress occurs, which can not be handled from commercially available steels. This made a re-design necessary, and the solution was found in a second perforated plate, located around 200 mm behind the first (main) perforated plate, which can move in the shell (not welded with the shell) of the HT-HE and stabilise it. Between first and second perforated plate the shell is compensated.

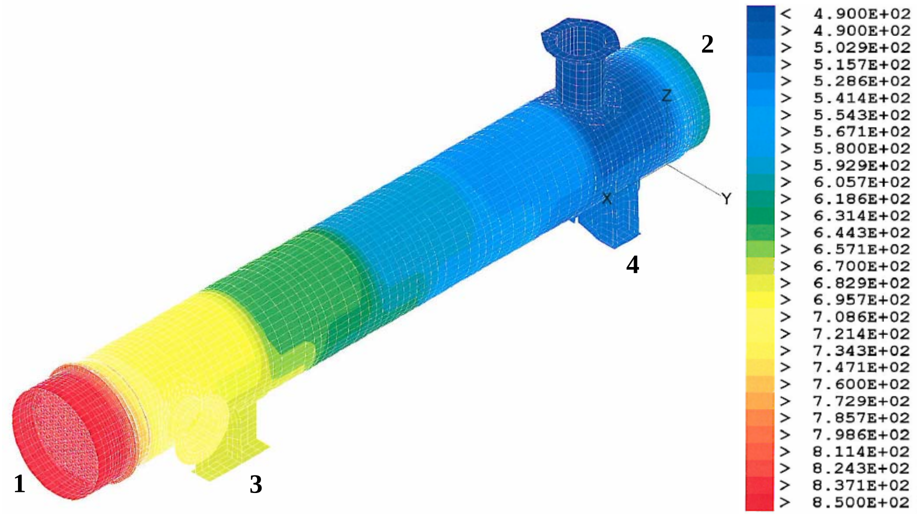

Figure 5: Distribution of the wall temperatures of the first duct of the HT-HE
Explanations: temperatures in °C; 1…flue gas inlet (850°C); 2…flue gas outlet (634°C); 3…air outlet (723°C); 4…air inlet (490°C)

For a proper operation and function of the HT-HE it is important to reach an evenly distributed inflow stream (in terms of temperature and velocity distribution). In order to optimise the flue gas inflow CFD simulations of the connection between furnace and HT-HE have been performed. Figure 6 shows the flue gas velocity distribution at the HT-HE inlet for the basic and the optimised design. It can be seen, that the adaption of the geometry of the connecting channel leads to a more equal velocity distribution at the HT-HE inlet. This is also obvious from the flue gas path lines as shown in Figure 7.

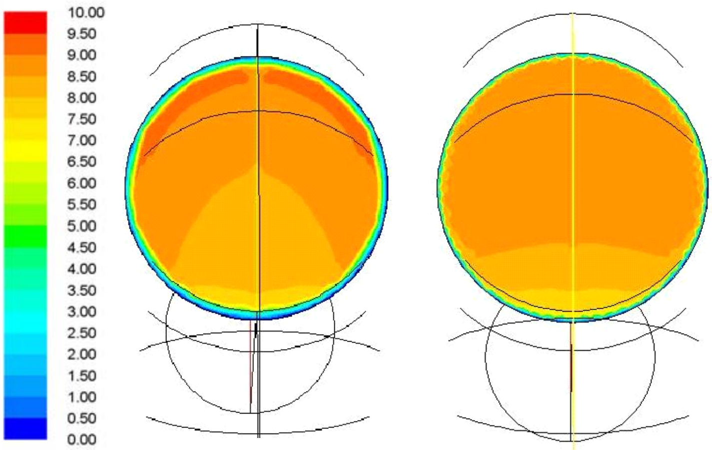

Figure 6: Distribution of the flue gas velocities at the HT-HE inlet
Explanations: left: basic design; right: optimised design; iso-surfaces of the flue gas velocities in m/s in a vertical cross section directly before HT-HE inlet

As already mentioned, in the turning chamber between the first and second duct of the HT-HE an automatic cleaning system is implemented, based on pressurised air. Each cleaning nozzle cleans a group of heat exchanger tubes and will be operated in intervals between 4 and 6 hours. The automatic cleaning is of great relevance, as the HT-HE must be cleaned regularly from deposits formed by coarse fly ashes as well as the condensation of ash vapours (in particular K_2SO_4 and K_2CO_3). This is especially of relevance for the inlet section of the HT-HE in order to avoid ash sintering at the compared with conventional heat exchangers very high surface temperatures in this section.

For the mechanical stability of the HT-HE not only the steady state operation at nominal load is of great relevance, but also the transient operation during start-up and shutdown. Too fast start-up or shutdown processes as well as abrupt temperature changes could lead to unacceptably high mechanical stress on the HT-HE materials. In order to be able to follow these recommendations, the overall control system of the plant has been designed in a way that the HT-HE inlet temperature can be increased stepwise and a by-pass valve at the HT-HE has been installed, so that the flue gas can be by-passed in case of emergencies.

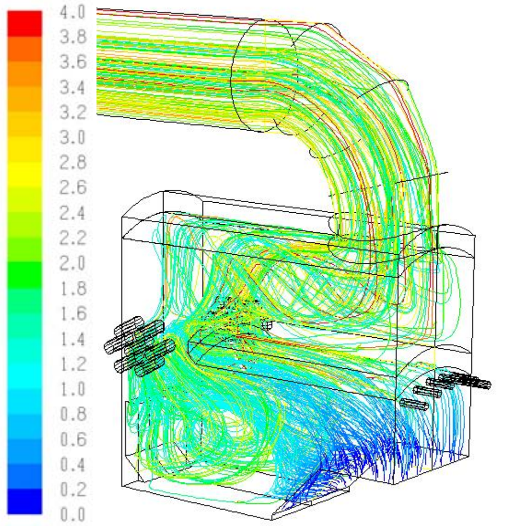

Figure 7: Path lines of the flue gas released from the fuel bed in the biomass furnace
Explanations: three-dimensional view; path lines colored according to their residence time in the furnace in s

5 LOCATION OF THE PLANT, FIRST OPERATING EXPERIENCES AND OUTLOOK

The plant is located at Il Forteto in Vicchio, region of Tuscany, Italy. Il Forteto is an agricultural cooperative with heating, cooling and electricity demand, which operates among others a cheese dairy, a super market and apartments. A picture of the location can be found in Figure 8.

Figure 8: Location of the BIO-MGT plant at Il Forteto

During 2009 the BIO-MGT plant has been assembled. A picture of the biomass furnace and the HT-HE is shown in Figure 9. The MGT is placed in a separate room, as the turbine must be protected from high room temperatures. A picture of the MGT is shown in Figure 10.

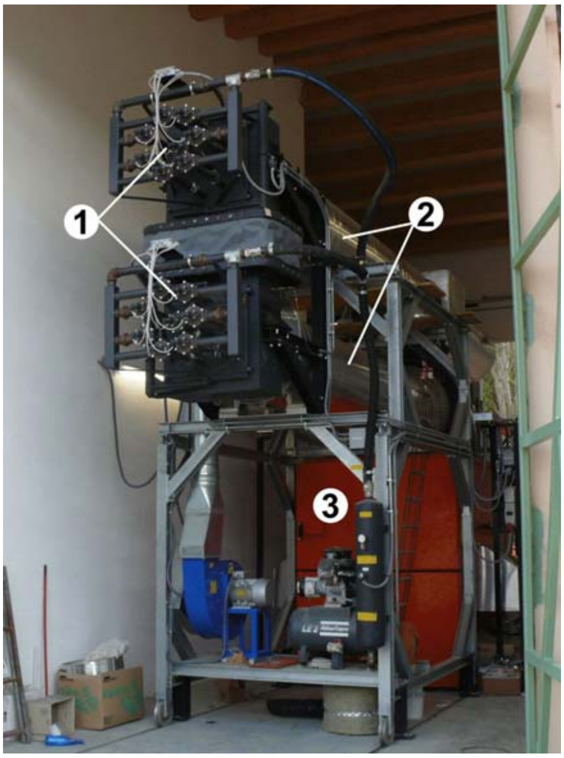

Figure 9: Biomass furnace and HT-HE of the BIO-MGT plant at Il Forteto during installation
Explanations: 1…automatic cleaning system; 2…HT-HE (first duct below, second duct at the top); 3…biomass furnace

Figure 10: MGT of the BIO-MGT plant at Il Forteto
Explanations: pipework during the installation phase before isolation; 1…ambient air inlet; 2…pressurised air to the HT-HE; 3…pressurised air from the HT-HE; 4…MGT combustion chamber; 5…MGT

The first start-up of the MGT was done in January 2010. After extensive tests on the MGT the plant was put into operation in dual fuel mode, where both MGT and biomass furnace are in operation, in February 2010.

From February to April 2010 tests of the control and security system have been performed. From May on test runs at various load in dual fuel mode are foreseen. The first operating experiences can be summarised as follows:

- The envisaged start-up procedure works properly, i.e. first to start the MGT and to reach a steady state operation at a certain load (approximately 60 kW_{el}). Due to the high temperature of the turbine exhaust gas the biomass ignites automatically as soon as biomass fuel is fed into the furnace.
- The operation of the biomass furnace during start-up works also properly and a slow, smooth and continuous increase of the furnace outlet temperature, respectively the HT-HE inlet temperature is possible.
- Although the plant is able to operate at part load, fast load changes will not be possible with the system, due to the high heat storage capacity of the HT-HE.
- The security and safety system of the plant has been checked successfully.

No major problems occurred up to now, which are particularly related to the new technology. The control system is now able to ensure stand alone operation which was a basic requirement before first test runs can be started. Within the next five months comprehensive test runs and evaluations of the new system are foreseen.

6 CONCLUSIONS

Within the EU FP6 project "Innovative small scale polygeneration system combining biomass and natural gas in a micro gas turbine (BIO-MGT)" a small-scale biomass CHP technology based on an externally with biomass and internally with natural gas fired micro gas turbine (MGT) with a nominal electric capacity of about 100 kW_{el} has been developed, built and put into operation in January 2010. This new technology aims to close the gap in the power range between 70 and 200 kW_{el}, in which no biomass CHP technology based on biomass combustion is currently available (i.e. the gap between the Stirling engine (< 70 kW_{el}) and the ORC (> 200 kW_{el}) technology).

The most important design data of the new system are an overall electric efficiency of 23.5%, an overall total efficiency of 80.2% and a biomass contribution of about 50%.

A new fixed bed biomass combustion technology for the exclusive use of turbine exhaust gas as oxidising agent was developed (no use of ambient combustion air). The design of the biomass furnace was supported by CFD simulations and an optimised design without unacceptable temperature and flue gas velocity peaks could be found. Moreover, a staged and low-dust combustion could be realised. In addition, the control system of the biomass furnace has been adapted to the need of using always the complete amount of turbine exhaust gas as an oxidising agent in the biomass furnace and to control the load, guided by the furnace outlet temperature (respectively the HT-HE inlet temperature) by the amount of biomass fuel fed into the furnace.

Moreover, a high temperature heat exchanger for the heat transfer from the flue gas to the pressurised air cycle of the micro gas turbine was designed, supported by a finite element analysis, which is able to cope with the requirements of this plant configuration (i.e. significantly reduced material strengths and creep strength of the HT-HE materials due to the high temperature levels).

Between February and April 2010 tests of the control and security system have been performed. The envisaged start-up procedure for MGT and biomass furnace worked well and a slow, smooth and continuous increase of the furnace outlet temperature, respectively the HT-HE inlet temperature is possible. The security and safety system of the plant has been checked successfully. The control system has been fine-tuned and is now able to ensure stand alone operation. From May 2010 on comprehensive test runs and evaluations of the new system are foreseen.

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9 LOGO SPACE

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