BIOMASS FIRED HOT AIR GAS TURBINE WITH FLUIDIZED BED COMBUSTION

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Abstract - The prevailing demand for decentralized energy supply out of renewable energy sources, such as biomass, requires small scale Combined Heat and Power (CHP) technologies. Current developments at the TU Munich comprise the Biomass fired Hot Air Gas Turbine (BioHGT), a fluidized bed wood combustor with integrated high temperature heat exchanger for indirect firing of microturbines at 100 kW_{el}. The present work comprises a cycle proposal, allowing electrical efficiencies in the range of 30%. Simulation results showed, that enhanced heat exchanger tubes with structured tube walls allow higher heat flux without an increase in pumping power up to a PEC ratio of 3.5 compared with smooth tubes. The heat flux density within the fluidized bed was found to be a function of tube spacing and type of structure. Experiments showed shading of heat transfer area on axially knurled tubes. Literature data on the influence of the horizontal pitch could be confirmed. A novel ceramic corrosion protection layer was tested in a fluidized bed combustor for coating of freeboard air preheater tubes with no signs of damage in short period tests.

INTRODUCTION

The availability of solid biomass fuels is restricted because of a low energy yield per unit area. Transportation of these fuels over a long-distance is not economic due to its low energy density per unit weight. Therefore, a sustainable use of solid biomass fuels demands the application of micro scale and small scale combined heat and power plants in the range of 0.1 to 50 MW combustion heat performance. The net electrical efficiency of the CHP plant is one of the main parameter for economic performance. It also has effect on the output to the thermal base load of a district heat network and, hence, affects the number of potential applications. In the power range below 1 MW_{el}, power generation technologies for liquid and gaseous fuels achieve higher electrical efficiencies, than those for solid fuels. Hence, by replacing the combustion chamber of a gas turbine with a high temperature heat exchanger, the electrical efficiency of a solid biomass fuelled power plant can be increased from 15-20% to 25-30% (cf. Kautz, 2004, 2005).

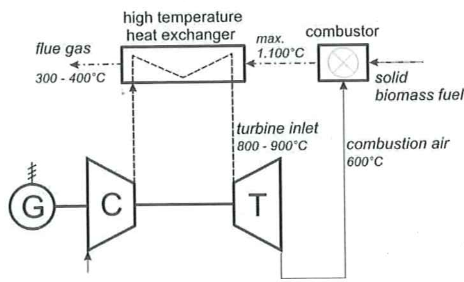

Fig. 1: Process flow diagram of the externally fired hot air gas turbine

One shaft micro turbines have proved suitable for the hot air gas turbine (HGT) process (Fig. 1). A screening of research and development activities throughout Europe showed that current HGT pilot plants make use of biomass combustion on grate furnaces and smoke tube heat exchangers (cf. Gallmetzer, 2006). The turbine exhaust gas is used as combustion air for full recuperation of the turbine exhaust waste heat. This, in return, leads to an increased excess air ratio in the combustion chamber of approx. $\lambda = 2.5$, resulting in a large exhaust gas mass stream at temperatures > 300°C. With this cycle layout, an electrical efficiency in the range of 17 - 20% is possible (Talbott's Ltd., cf. Pritchard 2005). The heat transfer coefficient k of the gas to gas high temperature heat exchanger is in the range of approx. 10 W/m^2K (cf. Pritchard 2002), which indicates a low material efficiency of the high temperature heat exchanger tubes.

To increase the electrical efficiency up to the range of 30%, a small scale organic rankine cycle (ORC) in the exhaust gas mass stream after the turbine can be used (Ökozentrum Langenbruck, cf. Schmid, M. 2006) without increasing the specific investment cost of approx. 4.5 k€/kW_{el}.

The R&D screening revealed major research necessity in optimizing the performance and cost effectiveness of the biomass combustion chamber and high temperature heat exchanger.

PROPOSAL

The subject of the present work in the investigation of two different design concepts for the high temperature heat exchanger including bubbling fluidized bed combustion: A) the finned heat pipe and B) structured tube wall heat exchanger. Furthermore, a freeboard heat exchanger for heat recovery from the combustion chamber exhaust gas will be discussed. Simulation and experiment results will be presented and a design recommendation will be discussed below.

Cycle analysis of the biomass fired HGT via a process simulation tool IPSEpro™ revealed the need for a high turbine inlet temperature with a high grade of heat recuperation for achieving electrical efficiencies in the range of 30% without ORC bottom cycle. Because of fuel ash sintering, temperatures in the biomass combustor of the HGT are limited to a maximum of approx. 1.100°C. Metallic heat exchanger tube wall temperatures are limited to approx. 925°C because of material stability. High cost and low durability due to fouling erosion and corrosion of the heat exchanger tube wall material require a high performance heat exchanger design with low material effort.

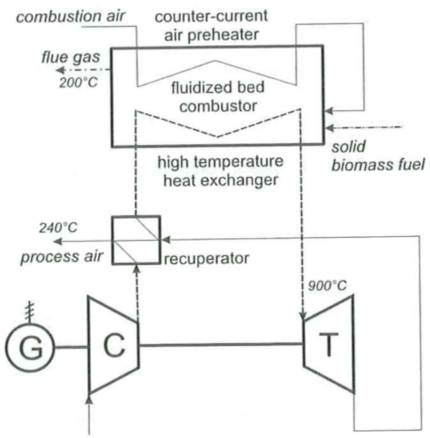

Fig. 2: Process flow diagram of the biomass fired hot air gas turbine with fluidized bed combustion

The proposal comprises the integration of a vertically immersed tube heat exchanger into a fluidized bed combustion chamber. This design resulted from the adaptation of the heat capacity streams within the hot air gas turbine process, for optimum heat transfer conditions. It includes several advantages.

- Heat flux is maximized when heat capacity streams within the counter-current heat exchangers are similar in magnitude.
- Low excess air ratio and high combustion air preheating possible due to cooling of the combustion chamber through heat release to the immersed tubes within the fluidized bed.
- High heat transfer coefficients within the fluidized bed provide a large zone for heat transfer in the range of the highest process temperatures.
- Permanent particle convection minimizes fouling and, thus, corrosion at the surface of the immersed tubes.

Design parameters of the immersed tube heat exchanger and operating parameters of the fluidized bed combustor consider heat transfer coefficients; tube wall erosion rates, heat release rates and combustion properties of solid biomass fuels (cf. Ottmann, M. 2007). Several measures should be applied, to minimize material wear due to erosion (cf. Oka, S. N., 2004). They comprise vertical immersion of tube bundles for minimizing the particle collision impetus, fluidization gas velocity $u < 0.2$ m/s, use of low hardness particle material, e.g. olivine and small particle sizes $d_P < 400$ µm for sufficient fluidization at low gas velocities, use of chromium rich austenitic steel for an increased oxide layer shielding and air excess ratio $\lambda > 1$ for a sufficient oxygen partial pressure that ensures a stable oxide layer at the tube wall. Further, tube bundles in tightly packed arrangements will limit bubble growth and, thus, bubble rise velocity that is considered as one of the main causes for erosion. Following these measures, material wear rate is estimated to a maximum in the range of 1 mm/10^4h, but only long term experiments would provide reliable data.

SIMULATION

Cycle analysis of the HGT process has been carried out in-depth with IPSEpro™. They point out the importance of systematic design of the key component high temperature heat exchanger. It is characterized by pressure loss Δp and heat conductance k*A. Measurements at the University of Rostock with smooth tube and structured tube high temperature heat exchangers showed an overall heat transfer coefficient of $k \approx 30$ W/m²K for smooth tubes and $k \approx 50$ W/m²K for structured tubes (cf. Kautz, 2005). In both cases, a gas to gas heat exchanger has been used.

Taking advantage of the high heat transfer coefficients of immersed tubes within a small particle fluidized bed in the range of $\alpha_{fb} = 700 - 800$ W/m²K, the heat conductance value k*A of the high temperature heat

exchanger can be largely improved, without increasing the pressure loss Δp of the gas flow. Two design concepts for the high temperature heat exchanger have been analyzed and evaluated and will be discussed below.

A) Finned heatpipe heat exchanger

The principle of the finned heatpipe heat exchanger consists of an improved heat conductance value k*A by an increase in heat transfer area A at the tube to pressurized gas heat transfer. The availability of space in the fluidized bed for an increased heat transfer area is low. Hence, heat is transferred via high heat conductance heatpipes into a pressure tank. The heat releasing condenser of the heatpipes within the pressure tank is finned with circular fins to provide sufficient surface for the area intensive heat transfer to the pressurized gas of the HGT. Mathematical modelling of the finned heatpipe heat exchanger showed heat conductance values in the range of k*A = 3 kW/K with a heat transfer area of 60 m². Pressure loss within the finned heatpipe heat exchanger is not a major issue.

The finned heatpipe heat exchanger requires a relatively large volume within the pressure tank. High material costs for the thick-walled high temperature pressure tank and poor availability of high temperature heatpipes are the main disadvantages that led to the conclusion not to favor this option.

B) Structured tube wall heat exchanger

The principle of the structured tube wall heat exchanger consists of an improved heat conductance by an increase of the gas to tube wall heat transfer coefficient. Heat exchanger tubes are vertically immersed in the fluidized bed. Heat is transferred from the fluidized bed through the tube wall into the pressurized gas.

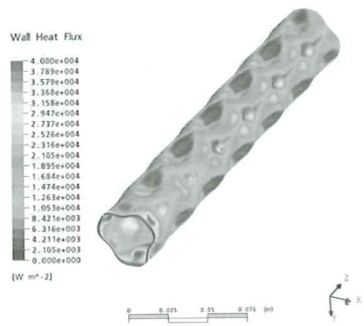

Fig. 3: Simulation results of the local heat flux density through a structured tube wall with 5.25 mm dent depth

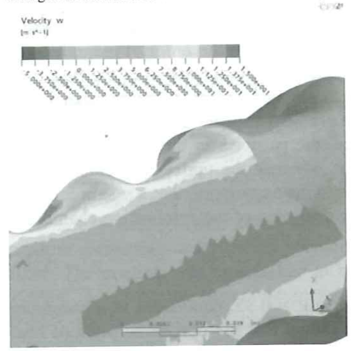

Fig. 4: Simulation results of the averaged local gas velocity in a structured tube wall with 6 mm dent depth and 40 mm inner diameter

The tube to gas heat transfer coefficient dominates heat transfer. It differs in approx. one order of magnitude from the fluidized bed heat transfer coefficient. An immersed tube heat exchanger with smooth tube walls would require an elevated number of immersed tubes and, hence, exceed the maximum tube density μ within the fluidized bed. Cycle and material efficiency can be improved by an increase in the inner gas to tube wall heat transfer coefficient.

The tube walls can be structured by cold deformation to show an axially recurring pattern of radial dents (Fig. 3 and 4). These dents force a thinner boundary layer in the gas flow and cause an increase in the average heat transfer number of up to 150% in comparison to smooth tube walls (e. g. Mitrovic, J. 2004). Dent depth has a major impact on the heat transfer performance, but there are little data available about a correlation of structured tube wall geometry and heat transfer performance.

A CFD simulation has been carried out to visualize and understand the influence of the tube wall geometry on the gas flow and to determine the overall pressure loss and local heat transfer coefficients. The structured surfaces of a series of tubes with different dent depths have been captured to a digital model by an optical three-dimensional scan. The CFD calculation applied the boundary conditions of the HGT process streams and temperatures. The influence of the tube wall geometry on the local heat flux density can be shown (Fig. 3) to range over one order of magnitude. The average heat flux shows an increase of the overall heat transfer coefficient. It can be explained by two effects:

- The gas velocity at the dent tube wall is much higher than the average (Fig. 4). This decreases the height of the gas boundary layer at the tube wall and, hence, increases the heat flux through it.
- Enhanced turbulence of the gas flow causes an increased mass transport of gas from the hot tube wall to the cooler core flow of the tube and, hence, increases heat flux.

Higher turbulence and high local gas velocities cause an increased flow resistance of the structured tube. By structuring of the tube surface, heat transfer can be greatly improved, but pressure drop of the gas flow along the structured tubes increases, too. An instrument for evaluating the overall gain is the Performance Evaluation Criteria PEC (cf. Webb, R. L. 1992). It is a measure for evaluating the quality of the gas flow including the gain in heat transfer and the increase in pressure drop and pumping power respectively.

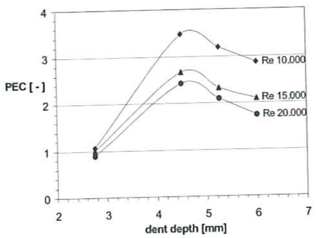

Fig. 5: Performance Evaluation Criteria PEC for the evaluation of heat transfer gain with different dent depths in the structured tube wall with a diameter of 40mm

$$PEC = \frac{\left(\dfrac{St_{structured\ tube}}{St_{smooth\ tube}}\right)^3}{\dfrac{\zeta_{structured\ tube}}{\zeta_{smooth\ tube}}} \quad [-], \quad St = Nu/(Re \cdot Pr) \qquad (1)$$

St is the Stanton number, a measure for heat transfer, calculated from the heat flux results; ζ is the average friction factor of the gas flow, calculated from the pressure loss results of the CFD-Simulation. The dent depth of the structured tube and the Reynolds number are the main parameter for influencing heat transfer and pressure loss. Fig. 5 shows the correlation of the PEC with the dent depth at different flow conditions. At dent depths below 3 mm and 15% of the tube radius respectively there can be seen no gain in flow quality. A clear maximum in flow quality is achieved at a dent depth of 4.6 mm and 23% respectively. It decreases sharply with further increase in dent depth. Different flow conditions have no influence on the position of the maximum at the dent depth of 23% of the tube radius but clearly show the best quality of gas flow at the Reynolds number $Re = 10^4$.

Overall heat transfer coefficient values k in the range of 150 W/m²K can be achieved. The heat conductance value k*A of the high temperature heat exchanger can be doubled from 1 – 1.5 kW/K for smooth tubes to 2 – 3 kW/K using structured tubes. The impact on the HGT process is an increase in turbine inlet temperature and, thus, an increase in electrical efficiency in the range of 3 – 5 percentage points with a raise of the investment costs below 1%.

EXPERIMENTAL

Heat transfer coefficient in fluidized bed with packed arrangement of heat exchanger tubes

Solid particle dispersion within the bubbling fluidized bed combustion chamber is impeded by a packed arrangement of vertical tubes. Based on experimental results from different authors and his own theoretical examination of the process of solid particle mixing, M. O. Todes suggested, that the order of magnitude of solid particle axial dispersion is proportional to $L^{3/2}$ where L is the characteristic dimension of the bed, e. g. the horizontal pitch of the vertical tubes (Todes, M. O. 1981). Hence, a reduced vertical dispersion rate in the biomass combustion chamber would lead to an elevated temperature gradient between the lower bed and the bed surface, where char combustion takes place. As a consequence, the particles in the lower bed region would lose their sensible heat to the heat exchanger and undercool, whereas the upper bed region would overheat and run the risk of ash agglomeration. This correlation leads to the assumption that a maximum tube density μ within the fluidized bed is critical for heat transfer and save operation of the combustion chamber.

$$\mu = 1 - \frac{A_{tube\ bundle}}{A_{total}} \quad [-] \qquad (2)$$

During experiments, (cf. Metz, 2007) this value has been estimated to $\mu = 0.85$, equivalent to a minimum average horizontal pitch of $S_h = 50$ mm. Further experiments have shown that the horizontal pitch of heat exchanger tubes is a key parameter for influencing heat transfer to immersed tube bundles (cf. Gel'perin, Ainstein, 1971).

$$Nu_{p,\max} = 0{,}75 Ar^{0{,}22} \left(1 - \frac{D_T}{S_h}\right)^{0{,}14} \quad , \quad S_h / D_T = 1.25 - 5 \qquad (3)$$

The axial and radial deformation of structured tubes can have a relevant influence on heat transfer and particle dispersion in the fluidized bed. Due to the relevance of this cognition in regard to the design of the

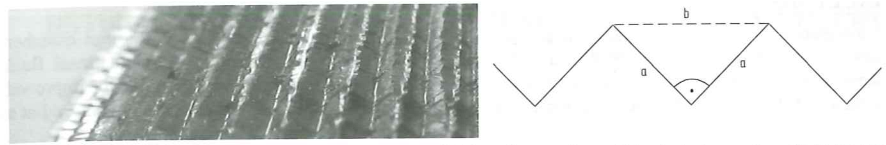

Fig. 6: Close-up of the surface structure of axially knurled tubes. By cold deformation, an orthogonal triangle structure can be applied to the tube surface resulting in a surface area increase of 41% with b = 2 mm.

immersed tube heat exchanger, further investigations have been carried out. The experiments were performed in a cold model BFB unit with a diameter of 0.4 m and a height of 0.6 m. The probes consisted of 40mm diameter tubes with smooth surface, axially knurled structured surface (Fig. 6) and structured surface with an axially recurring pattern of radial dents (Fig. 3 and 4). Probes were surrounded by an annulus of smooth tubes with flexible horizontal pitch. Experimental results (Fig. 7) show, that there is a relevant change in heat transfer coefficient with the horizontal pitch. Within a tolerance of +-10% expression (3) could be confirmed. Radially structured tubes (Fig. 3 and 4) showed no relevant effect on the heat transfer coefficient. However, axially knurled tubes (Fig. 6) showed a decrease in heat transfer coefficient of 20 – 40% compared with radially structured and smooth tubes. Their gain in heat transfer area of 41% is partly compensated by low particle heat transfer coefficients at low horizontal tube distances. The reason may be shading of the inner triangle area from particle convection by the limitation of bubble size and, thus, bubble rise velocity at low tube distances. However, heat transfer through radiation is not affected by tube spacing of the heat exchanger. At a bed temperature of 900°C heat transfer takes place by radiation to a relevant extend of 30 – 50 %. This part can fully be increased by the factor of surface area gain of axially knurled tubes.

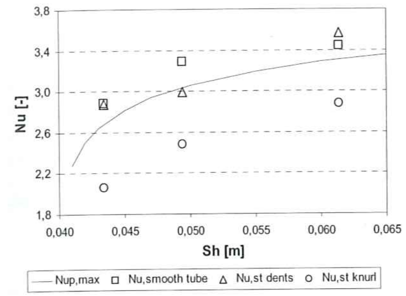

Fig. 7: Heat transfer influenced by distance between tubes for smooth tubes, structured tubes with dents and structured tubes with axial knurles compared with data from literature (Eq. 3)

Corrosion protection layer on freeboard combustion air preheater

Heat release to the immersed heat exchanger tubes mainly takes place in the fluidized bed. Great parts of the sensible heat of the flue gas and radiative heat of the fluidized bed surface are not available for heat transfer with the high temperature heat exchanger (Fig. 2). Parts of that heat can be recovered by preheating of the combustion air over a freeboard heat exchanger. Unlike tubes within the fluidized bed, heat exchanger tubes immersed in the freeboard are clearly prone to fouling and chlorine-corrosion. A coating of these tube surfaces can be an appropriate method for avoidance of material wear. The wear and corrosion resistance of steel is much lower than those of ceramic materials. So, the development of ceramic coatings on steel for protection purposes is an ongoing matter of research of scientists and engineers for many years. A number of deposition techniques such as arc-discharge plasma, gas-flame spray, vacuum deposition methods, plasma electrolytic oxidation (PEO) (Gu, W.-C. 2007) and high temperature glass enameling have been investigated to produce ceramic coatings on steel. Most of these methods are elaborate, costly or not applicable on more complex surfaces. To provide the corrosion exposed surfaces of the freeboard heat exchanger with a durable and heat conductive protection layer, a novel coating system based on a polymer-derived SiSiC ceramic (Cromme, Scheffler, Greil 2002) was tested. The precleaned austenitic steel samples were dip-coated with a mixture containing two different polysiloxanes, a crosslinking catalyst, a solvent and SiC powder as reactive filler. The coating was crosslinked at room temperature and pyrolyzed at temperatures between 800 and 1200°C in inert atmosphere and transformed into a dense homogeneous ceramic layer with good durability in erosive and corrosive atmosphere. The layer resistance was tested in a 25 kW_{th} biomass fuelled fluidized bed combustion chamber. Samples were exposed to the freeboard during a period of 7 h at approx. 900 - 950°C. The fuel consisted of wood pellets with and without the addition of 0.4 weight-% chlorine. The samples were characterized by XRD, optical end electron microscopy. No mass loss, damage or erosion marks on the ceramic layer could be detected.

CONCLUSIONS

The integration of a high temperature heat exchanger into a fluidized bed biomass combustion chamber allows a high performance hot air gas turbine cycle with moderate material effort. Computational fluid dynamic simulation has shown that the quality of gas flow through the heat exchanger tubes can be improved by a factor of 2 - 3.5 by radial deformation of the tube surface. The maximum flow quality is achieved at a dent depth of approx 23% of the tube radius.

The horizontal pitch of heat exchanger tubes is limited to a minimum of approx. 50 mm. Experiment results approved the decrease in heat transfer coefficient with decreasing horizontal pitch. Though, radially structured tubes showed no difference compared with smooth tubes, axially knurled tubes showed a superposed effect of surface shading at small horizontal pitches.

Tests in a biomass fuelled fluidized bed combustion approved the working principle of a novel ceramic corrosion protection layer for a freeboard heat exchanger. No mass loss, damage or erosion marks on the ceramic layer could be detected.

NOTATION

A	cross sectional area, m²
L	length, m
k	overall heat transfer coefficient, W/m²K
k*A	heat conductance, W/K
d_P	average bed particle size, µm
Nu	Nusselt number, -
$Nu_{p,max}$	maximum particle Nusselt number, -
Re	Reynolds number, -
St	Stanton number, -
Pr	Prandtl number, -
Ar	Archimedes number, -
S_h	horizontal pitch, m
D_T	tube diameter, m
u	fluidization gas velocity, m/s
λ	excess air ratio, -
α_{fb}	heat transfer coefficient in fluidized bed, W/m²K
μ	tube density, -
PEC	Performance Evaluation Criteria, -

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