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High Temperature Biomass Fired Stirling Engine (HTBS)

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Abstract. Biomass is an important resource for the utilisation of renewable energy not only in Germany, but in many other countries. Being sufficient to provide base load it is capable to stabilise electricity grids besides contributing to climate change precaution. Additionally it is essential that the applications for heat, power and fuel feature high conversion efficiencies. Therefore the on-site generation of electricity is a crucial contribution to renewable energy supply, especially in real small-scale applications. The project "HTBioStir" - development of a high temperature heat exchanger for the coupling of biomass boiler and Stirling engine - is designed and realized as a fundamental research project focusing on important mechanisms of heat transfer. The basic approach to generate electricity and heat implies the transfer of enthalpy on high temperature levels from the flue gases of a wood chip operated fire tube boiler to the working fluid helium through the heater head of a Stirling engine generator set. Using an indirect heat transfer mechanism, air as the carrier, an innovative heat exchanger with surface structured industrial power tube bundle and upstream a similar recuperative air-preheater by which an α-type SOLO 2V is operated to generate electricity. The leaving air at a much lower temperature level serves then as preheated oxidant in the furnace, while the flue gases produce hot water in the boiler. At present measurement campaigns were carried out to establish energy and mass balances of the fire tube boiler as well as the Stirling engine.

Key words

Biomass Combustion, Fire Tube Boiler, Air-Preheater, High-Temperature Heat Exchanger, Stirling Engine.

1. Aim and Scope

In the following a new approach for the utilisation of renewable energy is presented consisting of the coupling of a wood chip fired biomass boiler with a Stirling engine for purposes of electricity generation and district heating.

Fig. 1: Wood Chip operated Fire Tube Boiler KÖB, Pyrtec Type KDZ-150.

In all applications of biomass energy - heat, power and fuel - high conversion efficiencies are essentially necessary. Biomass is highly versatile when using it for heat, power and even fuel applications. It is characterised as a storable and renewable energy carrier, which could stabilise the energy supply system and contribute to climate change requirements, significantly. In fact, the on-site electricity generation based on biomass is still a substantial challenge because of the specific problems in real applications, the availability of technology, the economy of scale and the cost-to-performance-ratio.

Therefore within the range of electric capacities below 100 kW the aims of the work reported here are targeted towards the development of a highly resource-, energy- and cost-efficient combined heat and power generation system. Moreover, this project "HTBioStir" is designed and laid out to realize fundamental research focusing on important mechanisms of heat transfer. The main interest is concentrated on the technological development of a special heat exchanger operated at high temperatures for the combination of a fire tube boiler and a Stirling engine.

In particular, Stirling engine combined heat and power generation units were assessed to have a comfortable overall system efficiency and being most appropriate in terms of operational costs [1]. Energy and mass balances of the fire tube boiler as well as of the Stirling engine should be established by conducting specific measurement campaigns.

2. Configuration of the CHP-Plant

A comprehensive literature review and evaluation of several approaches using Sterling Engines with biomass furnaces was carried out [2-4, 6]. There are some systems with direct heat transfer to the Stirling engine. For example, these are a direct fired 75 KW$_{el}$ Hermetic Eight Cylinder Stirling engine and a special low-sensitive to fouling design of a Stirling engine up to 300 kW electric capacity integrated into a biomass combustor at 850 °C flue gas temperature and the heater head temperature at 750 °C. But most concepts use an indirect heat transfer. In particular, the initial power costs of a pellet fired low-speed kinematic β-type Sterling engine up to 5 KW electric capacity was estimated to be 14 €-Ct. $\cdot$ kWh^{-1}, the installation costs of the combined heat and power plant would be 19,000 € $\cdot$ kW^{-1}. Another Guntamatic Biostar Pellet Furnace was used to drive a SOLO V161 with an electric capacity of 9 kW and working with Helium at 650 °C and 150 bar. In this case a special heater was developed and applied consisting of NiCroFer tubes moulded in a monolithic copper head. Indirect heat transfer was achieved by a Nicofer 6023H tube bundle having a thermal capacity of 7 kW inserted into a Silicon Carbide mantle at a flue gas temperature of 1,150 °C and air as the carrier at 1,070 °C.

Besides that initially even a regenerative solid matter storage and finned sodium filled heat pipes were discussed in the project, but finally a recuperative solution with surface structured industrial power tube bundle made of high-alloyed steel materials No. 1.4841 and No. 1.4828 was promoted.

A. Wood Chip operated Fire Tube Boiler

On this basis a cascade of the enthalpy content of the flue gas at several stages of the furnace and the first, second and third pass of the boiler plus the water side including temperature and pressure levels were determined.

The main value for subsequent analysis is the temperature distribution along the combustion zone and the fire tubes. A sufficient number of measurements were executed at different locations along the flame zone, the radiation and convection sections of the furnace. Wood chips are charged via a screw conveyer and ignited with primary air on the grate. Secondary air is supplied above the grate and the maximum portion after the first ignition parts nearly at half way of the flue gases before entering the heat exchanger section of the final water heater (Fig. 2). Tertiary air is supplied before the baffle in front of the tube section to achieve complete combustion. The hot section is coated with refractories sufficient for the operation of the boiler at high temperatures. At different locations thermocouples were installed and recorded by an automatic data collection system.

In the biomass boiler a maximum temperature of some 1,250 °C could be reached at the end of the combustion chamber. Slightly more than 1,150 °C were reached in the passage between the combustion chamber and the fire tubes as well as above the grate after primary air inlet. In addition during the last test series the first part of the fire tubes of the boiler was covered with an insulation.

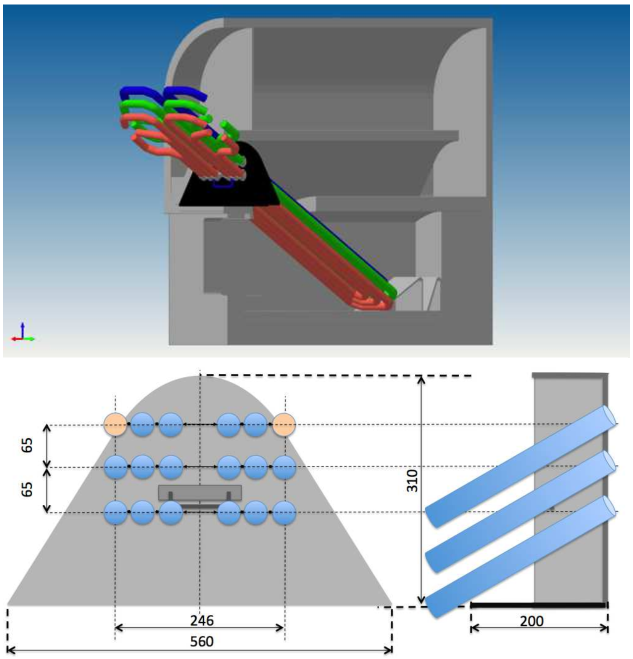

Fig. 2: Integration of the industrial power tube bundle heat exchanger into the furnace door and the radiation section.

Based on these initial measurements of the flue gas temperatures and composition and that on the water side an approach of an enthalpic balance was established. The fuel enthalpy input of wood chips amounts 221 kW, while the enthalpy values for the additional air are small in comparison. The wood chips burn out in different stages - on the grate, at the primary and secondary air inlet and the afterburning unit, which leads to dissipation and finally an enthalpy term equal to 145 kW is leaving the combustion compartment.

B. High-Temperature Heat Exchanger and Recuperative Air-Preheater

Based on the results of theoretical, experimental and numerical analyses two heat exchangers with surface structured industrial power tube bundles were designed, calculated and built [7, 8]. For obvious reasons the design was done according to principles of indirect, recuperative heat transfer.

The primary high-temperature heat exchanger made of alloyed steel material No. 1.4841 features a special arrangement of eight surface structured tubes with an effective length $l = 1.685$ m, a diameter $d = 0.0037$ m and a wall thickness $s = 0.002$ m (Fig. 2, 3). In this context the surface structuring of the tubes has favourable effects not only by means of a significantly increased heat transmittance, but also due to considerably reduced deposit formation [9, 10].

The secondary recuperative air-preheater is made of alloyed steel material No. 1.4828 featuring a compact hexagonal arrangement of 217 tubes with a nominal length $l = 3.1$ m, a diameter $d = 0.01$ m and a wall thickness $s = 0.0005$ m (Fig. 4). Due to this arrangement

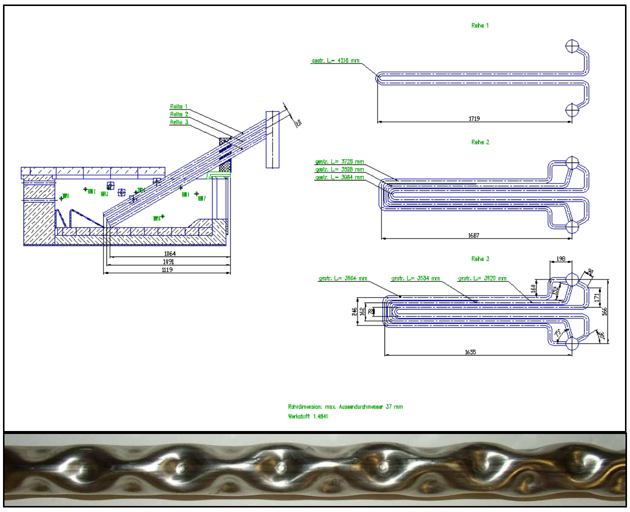

Fig. 3: High-Temperature Heat Exchanger with a surface structured industrial power tube bundle (d = 0.0337 m, s = 0.002 m).

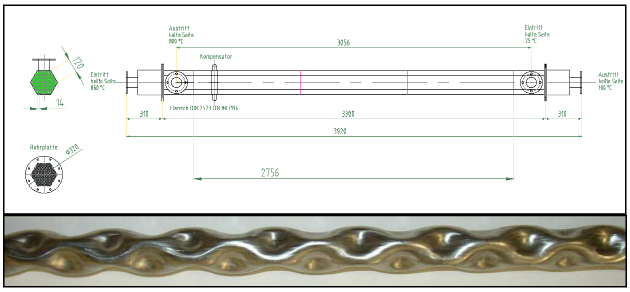

Fig. 4: Recuperative Air-Preheater with a surface structured industrial power tube bundle (d = 0.01 m, s = 0.0005 m).

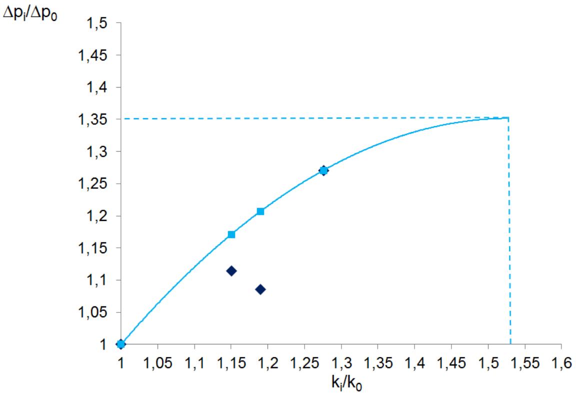

Fig 5: Effectiveness $\varepsilon^* = (k_i/k_0) \cdot (\Delta p_i/\Delta p_0)^{-1}$ of the measures to enhance heat transfer and corresponding pressure loss with increasing surface structuring (functional) of a gas-gas-heat exchanger system at 700 °C and Re = 1,500.

of counter-current air flows the functional structuring of the surface of industrial power tube becomes most effective regarding the heat transmittance k. Variously a significant increase of the overall effectiveness ε* could be demonstrated, because the heat transmittance is enhanced in excess of the corresponding pressure drop (Fig. 5) [9].

On the whole the configuration of the functional units is arranged in the following way: By the secondary preheater the process air taken from the environment by a fan is conditioned before entering the primary heat exchanger and flowing on to the heater head of the Stirling engine, from which it passes the preheater again to serve as heated up combustion air for the furnace of the boiler.

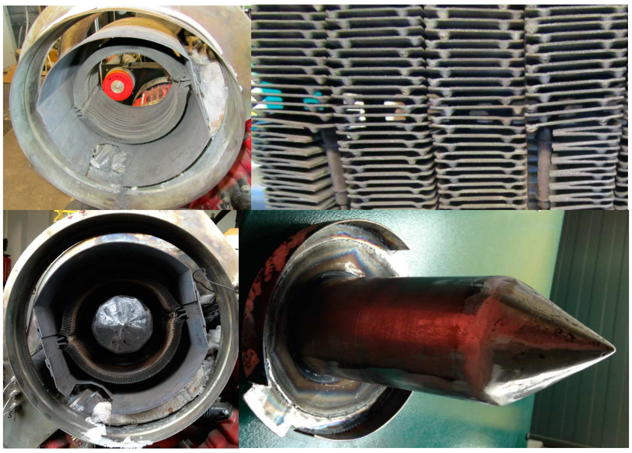

Fig. 6: Finned tube heat exchanger section of the Stirling engine heater head (top) and modified section with mixing tube, baffle plate and end caps (bottom) [11].

C. Stirling Engine

The Stirling engine was dismantled and the temperature niveaus of the burner flame, in front and behind the V-cylinder configuration and the CHP-unit were estimated. In addition the corresponding pressure ratings of the engine were recorded, so that a T-S-diagram of the working process as well as an energy balance could be derived from the data, and served as development tools.

Taking the α-type SOLO 2V Sterling engine and rearrange it for the operation with hot air instead of the flue gas from the original natural gas combustor the burner and heat exchanger section of the engine was modified, significantly. The original natural gas combustor, mixing tube, baffle plate and end caps were removed and replaced by a redesigned heat exchanger section (Fig. 6) [11, 12]. In order to ensure an almost complete perfusion of the hot air flow through the finned tube heat exchanger and to improve the contact between the fluid and the surface a shortened cylindrically shaped mixing tube with a conical head was constructed and installed in the centre of the heat exchanger. Besides that a reconfigured baffle plate with a circular throat at the centre was inserted, through which the fluid flow is focused on the surface of the heat exchanger. To complete the reinstallation of the section resized end caps were attached to it acting as headers in the flow of the hot air cycle. Finally the modified Stirling engine heater head section was reassembled and adjusted to the working machine (Fig. 7).

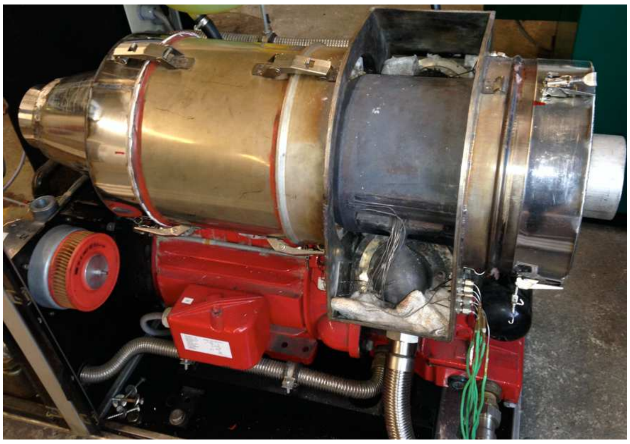

Fig. 7: Modified Stirling engine heater head section (right) [11].

3. Results

Based on the results of the measurement campaign mentioned above calculations of the high-temperature heat exchanger and the recuperative air-preheater were carried in some detail (Table I., II.).

Table I. - Detail Calculation of the Heat Exchanger

High-Temperature Heat Exchanger			
Quantity	Symbol	Unit	Value
Air Mass Flow	$\dot{m}_{Air}$	$kg \cdot s^{-1}$	0.0805
Inlet Temperature	$T_{Air,i}$	K	1071
Outlet Temperature	$T_{Air,o}$	K	1308
Average Velocity	v_{Air}	$m \cdot s^{-1}$	30.3
Pressure Loss	Δp_{Air}	Pa	2348
Surface Area	A_{Sur}	m^2	27
Heat Transmittance	k	$W \cdot m^{-2} \cdot K^{-1}$	37.6
Thermal Power	P_{th}	kW	21.02
Referring to process conditions			

Table II. - Detail Calculation of the Air-Preheater

Air-Preheater			
Quantity	Symbol	Unit	Value
Inner Air Mass Flow	$\dot{m}_{Air,i}$	$kg \cdot s^{-1}$	0.0806
Inlet Temperature	$T_{Air,ii}$	K	1133
Outlet Temperature	$T_{Air,io}$	K	370
Average Velocity	$v_{Air,i}$	$m \cdot s^{-1}$	12.9
Pressure Loss	$\Delta p_{Air,i}$	Pa	2757
Outer Air Mass Flow	$\dot{m}_{Air,o}$	$kg \cdot s^{-1}$	0.0806
Inlet Temperature	$T_{Air,oi}$	K	298
Outlet Temperature	$T_{Air,oo}$	K	1070
Average Velocity	$v_{Air,o}$	$m \cdot s^{-1}$	10.53
Pressure Loss	$\Delta p_{Air,o}$	Pa	1881
Surface Area	A_{Sur}	m^2	22.5
Heat Transmittance	k	$W \cdot m^{-2} \cdot K^{-1}$	44.1
Thermal Power	P_{th}	kW	66.84
Referring to process conditions			

Due to the temperature difference ΔT_{HE} = 237 K of the heat carrier air between the inlet and outlet the thermal power of the heat exchanger amounts $P_{th,HE}$ = 21 kW, while the corresponding values of the air-preheater are ΔT_{PH} = 772 K and $P_{th,PH}$ = 67 kW. The heat transmittance of the former one is $k_{HE} = 37.6 \; W \cdot m^{-2} \cdot K^{-1}$ and of the latter one $k_{PH} = 44.1 \; W \cdot m^{-2} \cdot K^{-1}$. Because of the utilisation of industrial power tube bundles effecting enhanced heat transfer the corresponding increase of the heat transmittance equals $\Delta k_{HE} = k_{HE}/k_{Ref} = 1.24$ in the case of the heat exchanger and $\Delta k_{PH} = k_{PH}/k_{Ref} = 2.55$ in the case of the air-preheater in comparison to conventional reference models with plain tubes. In the furnace the radiation term, which is proportional to the fourth power of temperature, is dominant, while convection contributes more to the heat transfer in the tube bundle of the air-preheater.

Referring to the process conditions with maximum temperature of T_{max} = 1520 K and pressure p_{max} = 0,11 MPa it is possible to use well known weldable and highly temperature resistant alloyed austenitic steel materials No. 1.4828 and 1.4841 for the manufacturing of the devices instead of nickel-base alloys or ceramics. The integration of the industrial power tube® bundle heat exchanger is effected by welded flanges mounted at the furnace door, thereby extending in the radiation section of the biomass boiler.

4. Conclusion

The high-temperature heat exchanger represents a somehow unorthodox design of an apparatus exposed to combined flows of different proportions of counter-current, co-current and cross-current character in the radiation section of the biomass boiler. The air-preheater constitutes a kind of ideal design with counter-current flows and enhanced heat transfer by a combination of increased Reynolds number, induced turbulence, disturbance of the boundary layer at the tube walls and a superposition of effects of angular as well as lateral stream components.

In the project reported here the demonstration of the proof of concept combining a wood chip operated fire tube biomass boiler KÖB Pyrtec Type KDZ-150 with an α-type Stirling SOLO 2V engine working with Helium at 12 MPa and 700 °C was of prior concern. Because of the restricted space in the furnace the thermal power of the high-temperature heat exchanger is probably limited to P_{th} = 21 kW, although the temperature levels are sufficient, wherefore the maximal pressure of the Stirling engine could not be reached, so that the nominal load would possibly be reduced to 80 %. Then the combined heat and power generation system has a capacity of 4.5 kW electricity and 12.7 kW thermal power, if condensation and re-cooling effects are considered. In combination with combustion air heated up the thermal power of the water heater amounts about 111 KW, the over-all power output of the system equals 128.2 kW.

In reference to the enthalpy transferred from the heat exchanger to the Stirling engine the electric efficiency of the developed system should be of the order of $\eta_{el,HE,St}$ = 0.2, while the over-all efficiency then would be $\eta_{oa,HE,St}$ = 0.8, respectively. These values tend to be some 10 % lower than in cases of being fuelled up with natural gas and are in good agreement with data taken from literature.

In order to achieve the highest net electric efficiency the main interest was focused on the realisation of maximum heat transfer rates and temperature levels by means of a high-temperature heat exchanger and an air-preheater well-equipped with specific surface structured industrial power tube bundles. As a consequence the effected pressure loss of both units was of much less importance as well as it was below 3000 Pa, at all.

Prospectively carrying on with the work it, should be attempted to gain an increased effectiveness and a further evaluation of the definite measures of heat transfer enhancement.

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References

[1] K. Uzuneanu, I. Diaconescu, Analysis of New Technologies used for CCHP Systems. Proc. 2nd Int. Conf. on Energy and Environment Technology and Equipment, 1rst - 3rd June 2013, Brasov, Romania, pp. 230-34.

[2] G. Marinitsch, J. Bovin, A. Lund, C. Ebbesen, Development of a Pre-Combustor for Small-Scale Biomass Stirling Engine CHP Plants. Proc. 20th EUBC&E, 18th - 22nd June 2012, Milano, Italy, pp. 750-55.

[3] R. Swaminathan, Cost effective, Low Capacity, Biomass fired Power Plant. Energy and Power 3(2013)1, pp. 1-6, DOI: 10.5923/j.ep.20130301.01.

[4] G. Schaumann, W. Sehn, H. Gerber, Untersuchung von Möglichkeriten zur Reinigung von Rauchgasen holzbefeuerter Biomasse-Stirling-BHKW. AZ 15340, Federal German Environmental Foundation 2003.

[5] B. Thomas, Stirling-Motoren zur direkten Verwertung von Biobrennstoffen in dezentralen KWK-Anlagen. Status Colloquium BWPLUS, 7th March 2007, Karlsruhe, Germany.

[6] F. Biedermann, H. Carlsen, I. Obernberger, M. Schöch, Small-scale CHP-plant based on a 75 KW_{el} Hermetic Eight Cylinder Stirling Engine for Biomass Fuels – Development, Technology and Operational Experiences. 2nd WCEB, 10th-14th May 2004, Rome, Italy, pp. 1-4.

[7] N. Senkel, W. Siemers, Combining Solid Biomass Combustion and Stirling Technology. Proc. 21th EUBC&E, 3rd - 7th June 2013, Copenhagen, Denmark, pp. 1313-19.

[8] N. Senkel, W. Siemers, A new approach for combining Solid Biomass Combustion and Stirling Technology. Proc. 19th EU BC&E, 6th - 10th June 2011, Berlin, Germany, pp. 1305-13.

[9] Y. Neubauer, A. Kölling, P. Schröder, F. Behrendt, U. Hellwig, N. Sachno, H. Liu, W. Zhang, I. Ul Hai, S. Riffat, H. Oldenburg, Gas Treatment of Fuel Gases from Biomass Gasification with Structured Tube Gas Cooling, Quench, Electrostatic Precipitator and Mop-Fan filter (EMF) - Effectiveness and Viability of Components developed in ERA-Net Bioenergy Project. Proc. 20th EUBC&E, 18th - 22nd June 2012, Milano, Italy, pp. 844-49.

[10] J. Peterseim, U. Hellwig, K. Endrullat, Parallel flow boiler designs to minimise erosion and corrosion from dust loaded flue gases. Proc. ASME Power, 29th July - 1rst August 2013, Boston, Mass., USA, No. 98123.

[11] N. Senkel, W. Siemers, Entwicklung eines Hochtemperaturwärmeübertragers für die Kopplung von Biomassekesseln mit Stirlingmotor. 5. Statuskonferenz Energetische Biomassenutzung, 14th November 2013, Leipzig, Germany.

[12] N. Senkel, W. Siemers, Combining Solid Biomass Combustion and Stirling Technology. Proc. 20th EUBC&E, 18th - 22nd June 2012, Milano, Italy, pp. 1345-49.

https://doi.org/10.24084/repqj12.341