BIOMASS CHP BASED ON GAS TURBINE PROCESSES

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INTRODUCTION

The very fast increase of renewable power in some countries, mainly based on wind and photovoltaic, causes problems for the stability of the power grids already. It is not easy to preserve the equilibrium between power production and power consumption anymore, due to the stochastic nature of wind and insolation availability. Therefore, the role of biomass for the production of power and heat becomes increasingly important. Contrary to wind and solar energy, it enables a more constant supply, e.g. of power and heat from CHP plants. In spite of very favourable feed-in tariffs in many countries, its contribution is still rather limited. The reasons lie in the high prices for biomass, as well as in the high specific investment costs. This means: developments of more efficient and simpler CHP are required. In this paper, some new developments in the usage of gas turbines with external combustion for biomass processes will be presented. Further on, the advantages of new developments towards the state-of-the-art processes will be discussed as well. This paper relates to the usage of solid biomass, with a water content up to 60%.

TRADITIONAL STATE-OF-THE-ART

The first biomass CHP facilities were based on steam boilers and steam turbines, i.e. on the Clausius-Rankine cycle. Those facilities still produce most of the biomass power. However, they are not specially developed for biomass applications, but scaled down from big power plants for applications with fossil fuels. Therefore, they are not very efficient, because a compromise between complexity and efficiency is unavoidable. Those plants usually have a power capacity under 10 MWe and therefore not a single regenerative heat exchanger, nor steam reheating. Steam parameters are usually not higher than 60bar and 400°C. Therefore, the highest electrical efficiency in condensing mode does not overcome 25%. In extraction mode, those values are considerably lower, depending on the amount of extracted steam. As

CHP facilities operate mostly in extraction mode, the mean value of electrical efficiency over the year is rather low, which has a negative impact on the economy of such units. Further on, the power capacity in extraction mode is also reduced. If the heat is used for district heating systems, which operate mostly from early in the morning till evening but not over the night, it results in a reduced power production during the day and a maximal power production during the night. That is not favourable for the grid stability.

The Organic Rankin Cycle (ORC) is more attractive in some applications. It has much lower specific investment costs and, in spite of considerably lower electrical efficiency, may reach better profitability in some cases. That is mostly the case when much more heat than power is required. Therefore, those plants are mostly heat governed; it means the stability of power production is again not given.

For many years, great efforts have been undertaken in order to obtain clean syngas from biomass gasification, which could be used for power generation in very effective gas engines. However, such gasifiers, together with gas cleaning systems, are rather expensive, while also decreasing the system efficiency. Moreover, those gasifiers require in some cases a very high quality of biomass input, which is expensive and decreases the profitability of those units.

SYSTEMS BASED ON GAS TURBINES

Very few traditional biomass CHP plants are truly competitive. The main two obstacles for a higher competitiveness are:

- high specific investment costs
- high expenditure for the logistics of biomass collection and transportation and therefore high biomass cost.

In fact, those two are reversely connected: by decreasing the plant capacity the costs of biomass decrease, but the specific investment costs increase and vice versa. Thus, the most successful biomass CHP plants are usually located near large-scale wood industry, where there is plenty of available waste biomass. The large amount of waste wood is locally available, so that it is possible to realize a high plant capacity without any additional logistics problems. However, such locations are more or less exhausted.

In order to overcome those two obstacles, a system with externally fired gas turbine, as presented in Figure 2, was proposed at the end of ‘90ties (Stevanović & Emmel, [1, 2]). Such a facility should have low maintenance costs, but at the same time a high power to heat ratio and a high thermal efficiency of power production, with the aim of reaching a good profitability. The main issue is to use the turbine outlet as preheated combustion air for the biomass combustion. In this particular case, a newly developed regenerator, the so-called Pebble-Heater, has been proposed (see Figure 1). The important role of the Pebble-Heater for the simplicity and efficiency of that proposal will be discussed in the next section of this paper.

Later on, some other concepts, based on classical recuperative heat exchangers, have been proposed and constructed. All those concepts will be presented in detail and discussed further on in this paper.

ROLE OF THE PEBBLE-HEATER

In 1996 Faßbinder [3] has patented a new type of regenerative heat exchanger with radial gas flow. Beside lower investment costs (bulk material is used as heat storage mass), this new type of Pebble-Heater enables higher temperatures (above 800°C) and a very high recuperation efficiency. The recuperation efficiency is defined as:

$$\varepsilon = \frac{m_1(h_{1h} - h_{1c})}{m_2(h_{2h} - h_{1c})}$$

and represents the ratio of recuperated heat towards the maximum possible recuperated heat. The maximum possible recuperated heat is defined by the hot end (inlet) temperature T_{2h} of the heating fluid (mass flow m_2) and the cold end temperature T_{1c} of the heated fluid (mass flow m_1). In cases of constant flow rates ($m_1 = m_2$) and constant specific heats of two gas streams, the recuperation efficiency is reduced to the more understandable expression:

$$\varepsilon = \frac{T_{1h} - T_{1c}}{T_{2h} - T_{1c}}$$

For the proposed biomass plant it is expected to reach the recuperation efficiency of 95%. In some other applications of the Pebble-Heater technology, extremely high values of up to 98% have been measured (Stevanović & Faßbinder [4]). The reason for such a good heat recovery lies in the small pebble size and the corresponding high heat transfer surface. For example, pebbles with a diameter of 4.5 mm give the specific heat transfer surface of 800 m^2/m^3. Therefore, the temperature difference between solid phase and gas phase is very low, in some cases less than 10K. That illustrates another extraordinary characteristic of Pebble-Heater heat exchangers: the exergy losses are lower than in the cases of other types of heat exchangers.

Those characteristics (high recuperation efficiency, low exergy losses and high temperature operation) are exactly those needed for the improvement of the Joule cycle with gas turbines. Therefore, it was a very logical idea to apply the Pebble-Heater technology to that cycle.

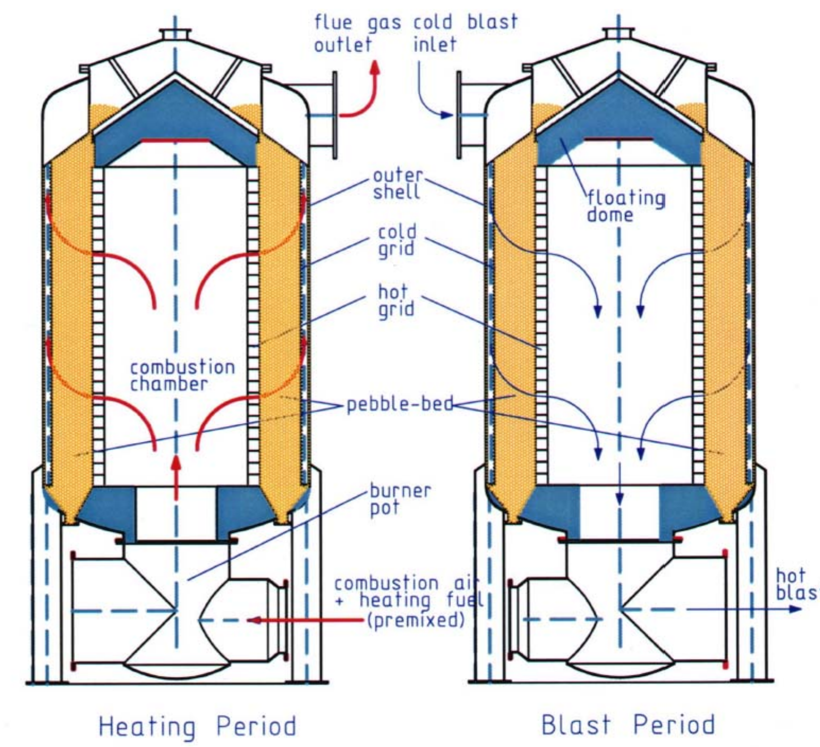

Figure 1: Regenerative heat exchanger with radial fluid flow, known as “Pebble-Heater”

GAS TURBINE CYCLE WITH EXTERNAL COMBUSTION AND PEBBLE-HEATER (SiPeb)

This concept, presented in Figure 2, uses a system of Pebble-Heaters to recuperate gas turbine Joule cycles. For the case of biomass, it consists of an external biomass combustor at atmospheric pressure. The process consists of following steps:

- The ambient air (15°C) is first compressed to 4.50 bar in a compressor driven by a gas turbine. Due to compression its temperature rises to 205°C.
- In an after-cooler it is cooled down to 90°C in a recuperative heat exchanger. The available heat may be used for a heat consumer, e.g. as hot water or low temperature steam. Decreasing temperature before entering the first Pebble-Heater (PH1) is important for lowering the stack losses. Lower input temperature at PH1 enables lower outlet temperature at PH2.
- Compressed air enters the first Pebble-Heater, where it is heated to 830°C.
- Hot air enters the gas turbine, where it is expanded to almost ambient pressure (1.03 bar) and to a temperature of 542°C. The released expansion work is used for compressor and generator drive.
- The most part of the expanded air is used as preheated combustion air for the biomass combustor. The rest may be used for another heat consumer, at a higher temperature level.
- Biomass is fuelled into the combustion chamber and burnt with preheated (542°C) combustion air. Due to a high air factor, combustion gases have a relatively low temperature of 870°C. That prevents a sintering of flying ash.
- Hot combustion gases enter the second Pebble-Heater (PH 2) from the hot side (the hot grid), where there is a homogeneous temperature field (870°C). If there are still some tar particles in the gas stream, they will be certainly burnt. The combustion gases are cooled down to 97°C and exhausted through the stack.

The presented cycle will result in an electric efficiency of 32.3% and total CHP efficiency of 71% (in case that the heat from the air after-cooler and from the hot air at the turbine outlet may be used). Those values depend on the characteristics of the gas turbine and the rest of the equipment (given in a table in Figure 2), and therefore may vary from approx. 30-35% electric efficiency. If an even higher electrical efficiency and higher power output are required, the plant may switch to the operational mode with compressed air cooling by water evaporation (injector or film evaporator) before entering the first Pebble-Heater PH1. Thus, the flow rate through the compressor (and so the compression work) stays constant, while the flow rate through the gas turbine increases. That operational mode may be used to adapt to the daily fluctuations of power demand and daily or even seasonal changes in heat consumption.

For a smooth operation and a stable power output, the best solution is to use three Pebble-Heaters, equipped with the valves which are required for the switching between the units.

This concept is known under the acronym “SiPeb”, given by Siemens. However, through that cooperation just one test unit was built to test some main components.

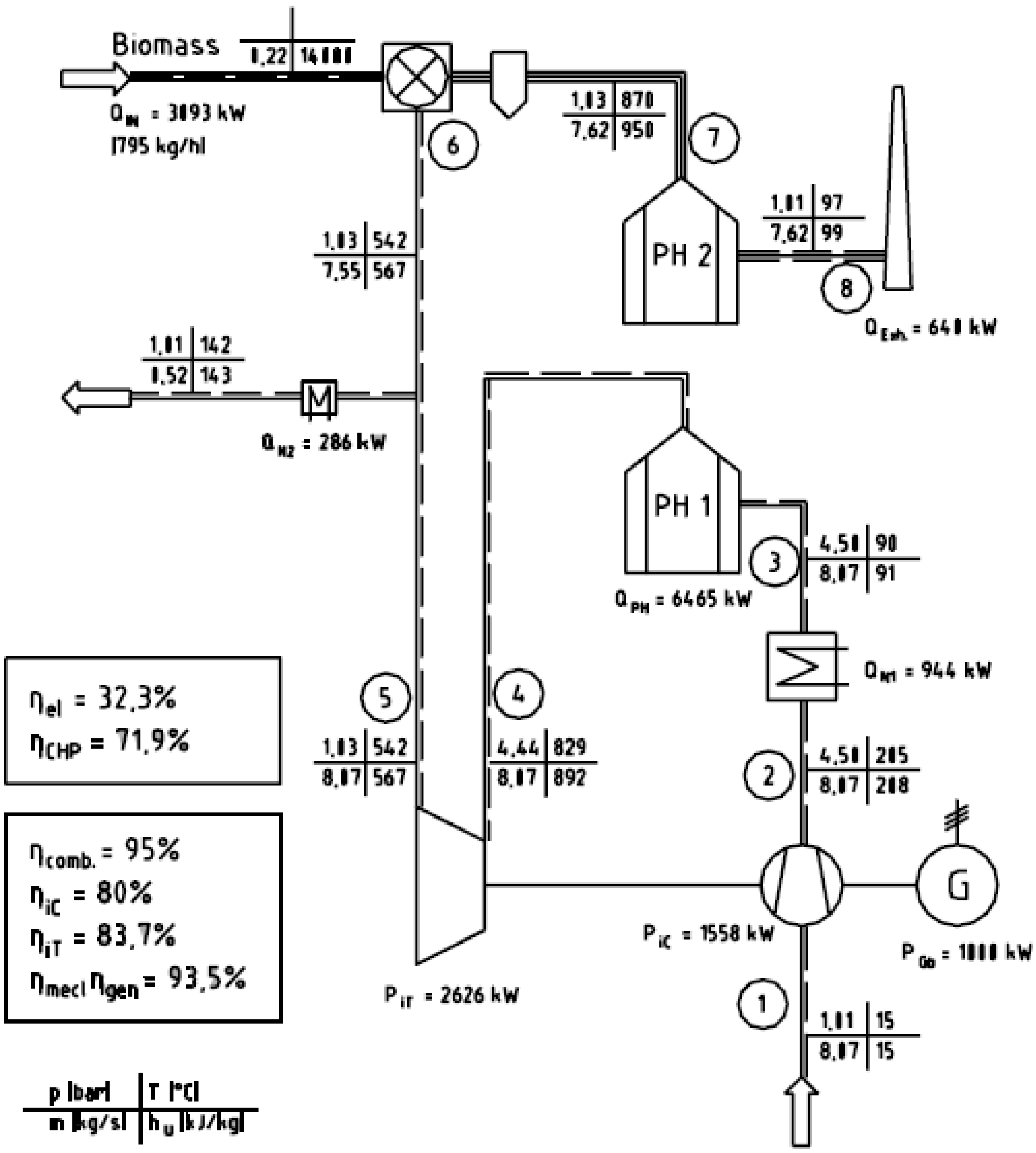

Figure 2: Gas turbine cycle with external combustion and Pebble-Heater (SiPeb)

CONCEPTS WITH EXTERNAL COMBUSTION WITHOUT PEBBLE-HEATER

As the Pebble-Heater technology is patented and requires specific know-how, there were many efforts to apply classical recuperators instead. However, the main advantages of Pebble-Heaters are the main obstacles for the usage of recuperators: the maximum temperature is clearly limited, the recuperation efficiency is not great and the exergy losses are too high. All those disadvantages result in a significantly lower electric efficiency of such systems.

Figure 3 shows one such concept, which was built in several facilities in UK and Switzerland (Schmid [5]). It is based on a micro gas-turbine, but there are solutions with the usage of adapted turbo-chargers, too. It is immideately clear, that the temperature differences between the cold and hot side of the presented recuperator are 100K and that the maximum gas (air) temperature entering the turbine is only 720°C. That is, of course, too low to achieve higher efficiencies. At the beginning, efficiencies were in the range from 10-15%, but were later improved towards 22%.

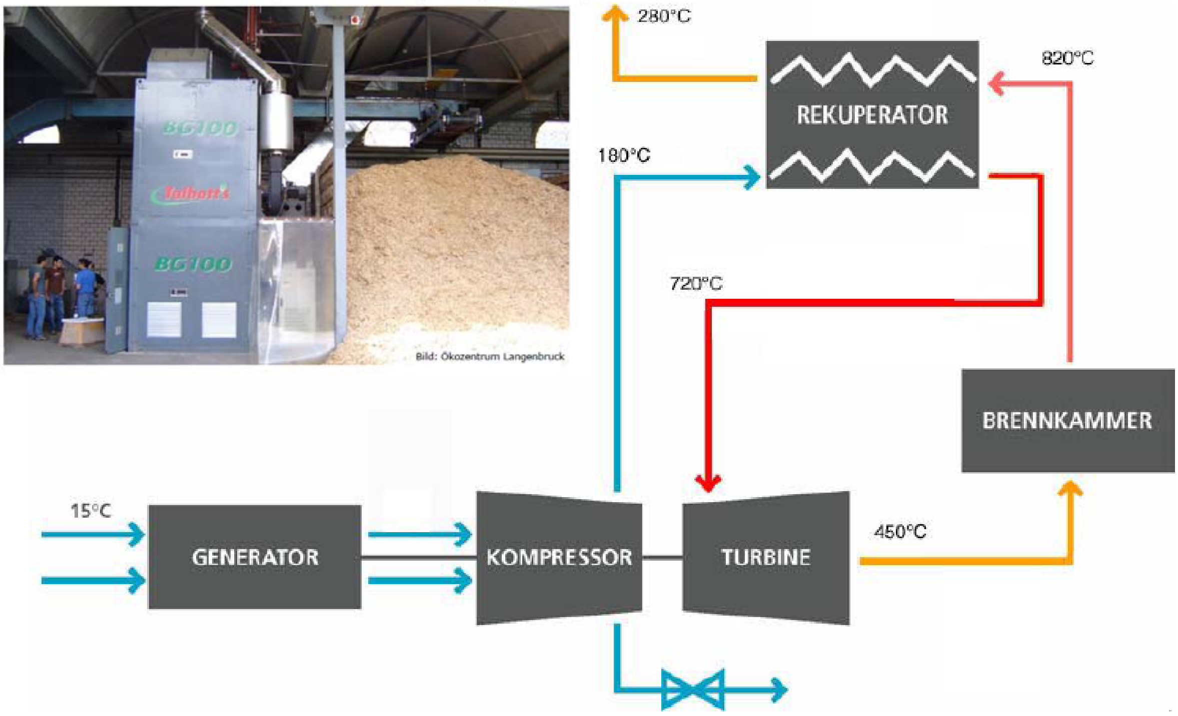

Figure 3: Gas turbine cycle with external combustion and classical heat recuperator [5]

To increase the temperature at the turbine inlet and to decrease the thermal load of the recuperative heat exchanger, it was proposed to introduce an additional combustion chamber on gas or liquid fuel. In that way, only one part of the energy is introduced by biomass (usually 50-55%) and the rest by the heat value of secondary fuel. Depending on the existing regulations on feed-in tariffs in a particular country, the secondary fuel may also be biogenic, such as bio-gas or bio-diesel. Such a concept is presented in Figure 4 (Martelli et al. [6]). The temperature of combustion gases at the recuperator inlet (position 6) is 780°C and 312°C at the outlet (position 7). On the other hand, the compressor air temperature at the recuperator inlet (position 2) is 194°C and 692°C at the outlet (position 3). In the additional combustion chamber (CC1) through additional gas combustion, the gas temperature is further increased, so that it is already 910°C at the turbine inlet. In that way the turbine inlet temperature stays the same as in the original design of the foreseen micro gas turbine, but the efficiency drops from 31% to 22% [6]. This system is in further development, known as BIO-MGT [7].

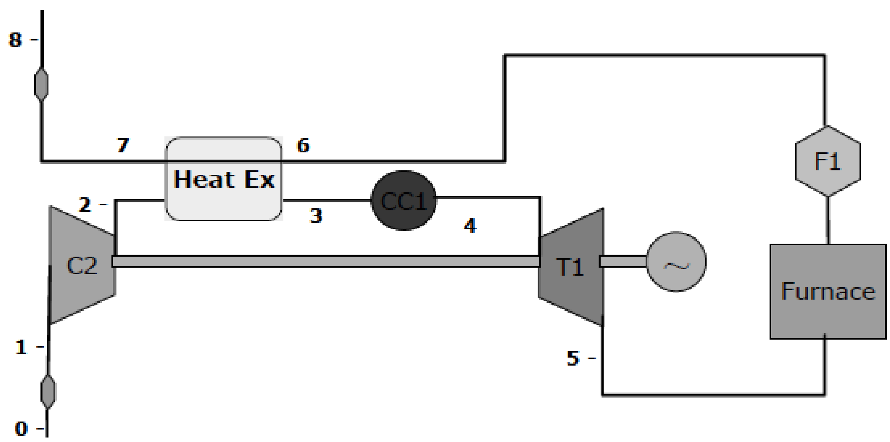

Figure 4: Gas turbine cycle with external/internal combustion [6]

IMPROVED CONCEPT WITH PEBBLE-HEATERS

It was clear that the Pebble-Heater technology has extraordinary advantages for the applications with a Joule cycle. Therefore it was continued with this development, but with some further alternations and improvements. The main change is that atmospheric combustion has been replaced by a biomass gasification system (Stevanović [8], Stevanović et al. [9]), as presented in Figure 5.

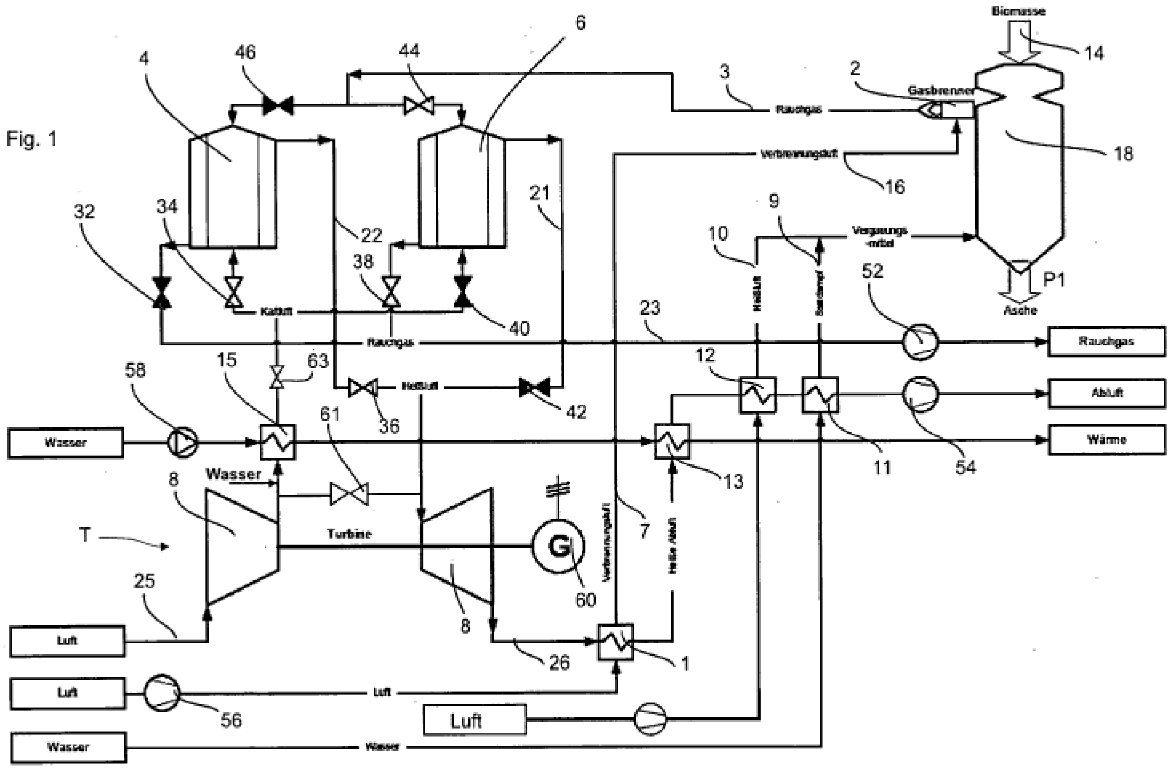

Figure 5: Gas turbine cycle with biomass gasification and Pebble-Heater

With the up-draft gasifier (18) the dust load was considerably lower than in the case of the biomass combustion. All problems with tars have disappeared after the combustion chamber (2), where at high temperatures and long enough residence time the conditions for a complete combustion were given. Moreover, the size was reduced and the conditions for avoiding NOx formation were easier to control. Depending on the gas turbine parameters, the calculated electric efficiency is in the range between 32-37%.

Shortly afterwards, some systems with a recuperator and biomass gasification have been proposed and installed. However, the same problems as before have been experienced and the efficiency is still only about 20%.

CONCLUSIONS

Since the year 2000, there is an intensive development of gas turbine usage with biomass. Those cycles are much simpler and have lower investment costs compared to the steam (Clausius -Rankine) cycle. Especially in the case of small (less than 10 MWe) and very small capacities (less than 500 kWe) those advantages are evident. However, if state-of-the-art recuperators are used, the problems with high temperatures and exergy losses diminish the effects of recuperation, so that the electrical efficiency of such units drops under the values achievable with steam turbines. However, those values may be of interest for the application of micro gas turbines in the range of 100 kWe.

On the other hand, the Pebble-Heater technology has extraordinary advantages for these kinds of applications: it is suitable for very high temperatures (even above 1200°C), has very high recuperation efficiency and low exergy losses. These advantages result in an electrical efficiency of at least 30%. With those values, such concepts are above the efficiency of gas engines with internal combustion. Moreover, they have no need for demanding and energy-inefficient gas cleaning systems and are more suitable for long-term power production.

The disadvantages of this technology lie in the fact that those units are regenerators, so that at least 2 units (better 3 or even 4) are required for continuous operation. It means that several valves should be applied, some of them suitable for high temperature and therefore very expensive. The possible problems with dust have been examined very thoroughly and the conclusion is that it will not have an influence on the long-term reliability, especially not in the case of the biomass gasification in an up-draft gasifier. However, all those advantages still have to be demonstrated in a first industrial facility, not just in test facilities.

LIST OF REFERENCES

1. Stevanović D., Emmel A., 2002, „Verfahren zur Umwandlung von thermischer Energie in mechanische Arbeit “, Deutsches Patent- und Markenamt, Patent DE 100 39 246, Munich, Germany
2. Stevanović D., Emmel A., 2004, “Method for Converting Thermal Energy into Mechanical Work”, United States Patent and Trademark Office, Patent No. US 6,799,425 B2, U.S.A.
3. Faßbinder H.-G., 1996, “Verfahren und Regenerator zum Aufheizen von Gasen”, Deutsches Patent- und Markenamt, Patent DE 42 36 319, Munich, Germany
4. Stevanović D., Faßbinder H.-G., 2000, “Regenerative thermal oxidizers based on Pebble-Heater technology”, Proceedings of the 5th European Conference on Industrial Furnaces and Boilers INFUB 2000, Volume 2, Porto, 11-14 April 2000, Portugal,
5. Schmid M., 2004, “Rolle von Mikro-Gasturbinen und Heißluft-Turbinen in der nahen Zukunft – Neue Entwicklungen und Praxistauglichkeit“, Ökozentrum Langenbruck, Switzerland
6. Martelli F., Chiaramonti D. and Riccio G., 2004, “Biomass fed micro gas turbine plant for decentralised energy generation in Tuscany”, LAMNET – Latin America Thematic Network on Bio-energy, Vila del Mar, 8-11 November 2004, Chile
7. Thek G., Brunner Th. and Obernberger I., 2010, “Externally with biomass and internally with natural gas fired micro gas turbine”, Proceedings of the 18th European Biomass Conference & Exibition, Mai 2010 Lyon, France, ETA-Renewable Energies (Edt.), Italy
8. Stevanović D., 2009, “Verfahren und Vorrichtung zur Umwandlung thermischer Energie aus Biomasse in mechanische Arbeit“, Deutsches Patent- und Markenamt, Patentschrift DE 10 2009 038 322, Munich, Germany
9. Stevanović D., Johannssen S. and Pritscher R., 2012, “Method and Device for Utilising Biomass”, United States Department of Commerce Patent and Trademark Office, Application No. 2012/0137702, U.S.A.

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