The Externally Fired Gas Turbine (EFGT-Cycle) and Simulation of the Key Components

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Abstract

The externally fired gas turbine unites two advantages. On the one hand, the utilisation of the waste heat from the turbine in a recuperative process and, on the other, the possibility to burn "dirty" fuel. In particular, the EFGT opens a new option to utilise biomass for combined-heat-and-power and contributes to reduce greenhouse gas emissions. A micro gas turbine with 100 kW electric output is chosen as an example to study the effects of temperature difference and pressure loss in the gas-to-air heat exchanger on cycle efficiency and power. The simulation calculations are performed with the codes AspenPlus and GateCycle. In addition to cycle optimisation the effect of low-calorific biogas on the combustion air ratio and the possibility of solar energy as a heat source for the EFGT are studied.

Keywords: externally fired gas turbine, bio fuel, decentralized CHP, heat exchanger

Introduction

The future world energy supply will have to rely on all energy resources, especially renewable energies. The technologies should combine high conversion efficiency with low emissions, in particular CO_2. Modern energy conversion technologies like internal combustion engines and gas turbines demand clean fuels as the combustion gases are in direct contact with the moving parts of the machine. Indirect systems separate the combustion and the thermodynamic conversion cycle. A conventional coal fired steam cycle plant is the best example. The same principle applies to the Stirling engine. The indirectly or externally fired gas turbine (EFGT) is a novel technology under development for small and medium scale power and heat supply.

The externally fired gas turbine unites two advantages. On the one hand, the utilisation of the waste heat from the turbine in a recuperative process increases the efficiency and, on the other, the possibility to burn "dirty" fuel. In particular, the EFGT opens a new option to utilise biomass for combined-heat-and-power and contributes to reduce greenhouse gas emissions. Although the technology was studied for coal in the 1950's [1] it is still in the early stages of development.

The micro gas turbine technology with recuperator is a promising technology for introducing the EFGT-Cycle. Furthermore, due to the small power unit it is compatible with the biomass output of a single farm. The heat exchanger of the EFGT-Cycle can be based on the same recuperation technology as in the micro gas turbine, although it will work at a much higher temperature. Also, a new type of combustor is needed to burn a low-calorific and mostly unclean fuel. The aim of the paper is to study system parameters and in particular the key components the high temperature heat exchanger and the combustor.

Thermodynamic Principles

The simple **once-through gas turbine** is shown in Figure 1. The compressed air is burnt in the combustor and the hot gas expands in the turbine which drives the compressor and the electric generator. The energy content in the hot exhaust gas from the turbine is not utilised.

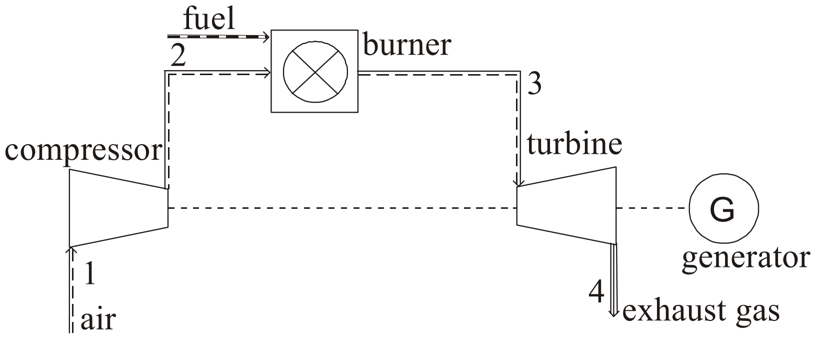
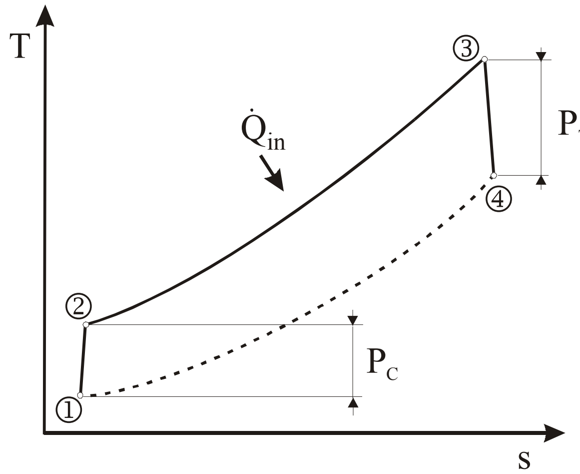


Fig. 1: The simple once-through gas turbine process

The thermodynamic cycle is shown as a T,s-diagramme in Figure 2. The comparatively poor efficiency of the open gas turbine results from the high exhaust temperature of 500 to 600 °C. A large amount of heat is released unused to the atmosphere. A first priority is to utilise the waste heat, either in a combined-cycle with a waste heat boiler and a steam turbine or in a combined-heat-and-power system. A further possibility is the recuperative air heating in the gas turbine itself.



$$\eta_{th} = \frac{P_T - P_C}{\dot{Q}_{in}}$$

Fig. 2: T,s-diagramme of the simple gas turbine process

In the **recuperative gas turbine** heat from the hot turbine exhaust gas is transferred to the colder compressed air in a heat exchanger between compressor and combustor. Due to the preheating of the air less fuel is required and the thermal efficiency increased. A precondition for the recuperative heat transfer is that the exhaust gas is hotter than the compressed air, i.e. $T_4 > T_2$. Furthermore, a temperature difference is necessary to facilitate the heat transfer so that the exhaust gas cannot be cooled down all the way to T_2 but to a slightly higher temperature T_{2R}. The efficiency of the gas turbine with recuperation is given in the formula below.

$$\eta'_{th} = \frac{P_T - P_C}{\dot{Q}'_{in}} = \frac{P_T - P_C}{\dot{Q}_{in} - \dot{m}_{air} \cdot c_{p,air} \cdot (T_{2R} - T_2)}$$

The degree of recuperation depends on the temperature of the air leaving the compressor and this again on the pressure ratio. A high pressure ratio leads to a large temperature increase in the compressor. Consequently, recuperation is most efficient in turbines with low pressure ratios. This is typical for the simpler design in the low power unit size whereas the modern turbines in the 100-MW-class have pressure ratios so high as to make them unsuitable for recuperation.

The recuperation can be carried one step further with all the heat input to the gas turbine process provided in a heat exchanger. This is an **indirect gas turbine** utilising waste heat from another process. However, the heat can also be provided by placing the combustor in the hot exhaust air stream from the turbine. The **externally fired gas turbine (EFGT)** has the thermodynamic advantage of the preheated air and that the combustion gases do not pass through the turbine. In the recuperator the gas is cooled to a temperature in excess of the compressor outlet temperature and the waste heat to the environment is minimised, cf. Fig. 3.

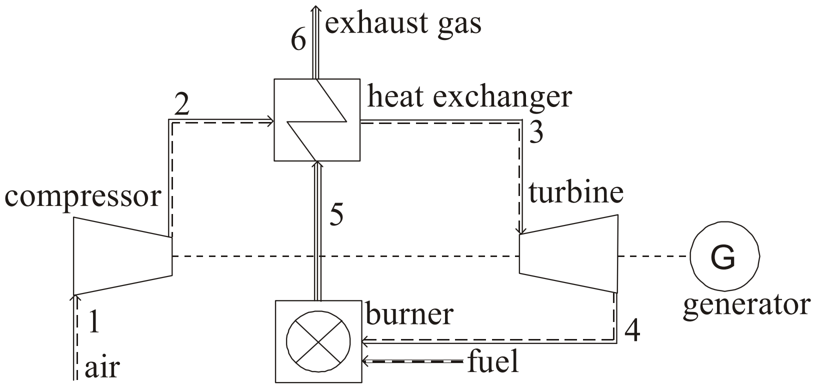
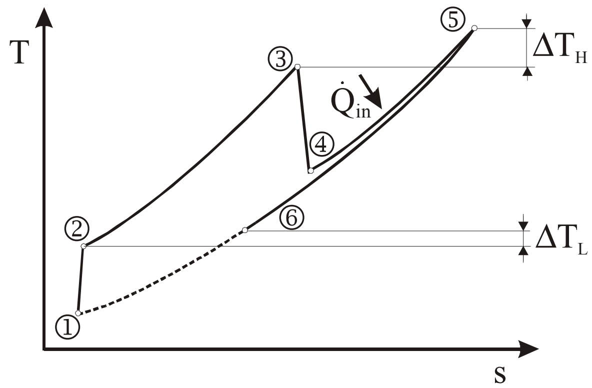


Fig. 3: The externally fired gas turbine process

In comparison with the directly fired gas turbine the EFGT sets less stringent requirements with respect to composition and cleaning of the combustion gas. It has the advantage of allowing to burn alternative, non-standard fuels, for instance biogenic fuels. The smaller unit size also enables decentralised units appropriate for the biomass output from farms and agricultural processing units.

The T,s-diagramme for the EFGT-cycle is presented in Figure 4. The temperature difference in the heat exchange ΔT_H and ΔT_L is an important parameter in the design optimisation. In general, a small temperature difference improves the utilisation of the heat and the efficiency but increases the size of the heat exchanger and the costs.



$$\eta''_{th} \approx \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_5 - T_4)}$$

Fig. 4: T,s-diagramme of the externally fired gas turbine

Simulation Studies

A micro gas turbine Turbec T100 (ABB/Volvo) with 100 kW electric output is chosen as the design basis to study the effects of temperature difference and pressure loss in the gas-to-air heat exchanger on cycle efficiency and power. The Turbec machine consists of a one stage radial compressor and a radial turbine with an external combustor. A recuperator is used to raise the net electric efficiency from 16 to 30 %. The technical data are given in Table 1 above.

Tab. 1: Design data for Turbec T100 micro gas turbine

Net electric output	100	kW
Thermal power input	333	kJ/s
Turbine power	281.89	kW
Compressor power	158.97	kW
Net electric efficiency, ISO	30.0	%
Fuel type	Natural gas	
Exhaust gas temperature	650	°C
Air temperature compressor outlet	214	°C
Gas temperature turbine inlet	950	°C
Gas temperature after recuperator	270	°C
Mass flow air	0.7833	kg/s
Mass flow natural gas	0.0067	kg/s
Mass flow exhaust gas	0.79	kg/s
Pressure ratio	4,5	
Efficiency compressor	0.7680	
Efficiency turbine	0.8261	
Heat exchanger area in the recuperator	164	m²

In the simulation all components with the respective flows of process media are depicted. The input parameters and the design points, e.g. pressure ratio and gas turbine inlet temperature can be varied. Different options for utilising the waste heat exist, e.g. drying of biomass. The simulation calculations are performed with the codes AspenPlus and GateCycle.

For the externally fired gas turbine the Turbec combustor is replaced by a heat exchanger and a new atmospheric burner at the turbine outlet. The recuperator becomes part of the heat exchanger. The inner efficiency of the compressor stage of $\eta_{i,\,C}$ = 0,7680 and of the turbine stage of $\eta_{i,\,T}$ = 0,8261 have been derived from the general data of Turbec machine and are used in the simulation of the EFGT. The first series of calculations are for natural gas and subsequently other gases are investigated. All calculations are performed for ISO standard conditions (15 °C, 1.013 bar and 60% humidity). The pressure ratio has the largest impact on power and efficiency and the results for varying π_C between 2 and 8 are shown in the graph, Figure 5. The curves have individual maxima and the highest output is achieved with a pressure ratio of 6.0, whereas the highest efficiency is found for 2.9. The design pressure ratio of 4.5 is a compromise.

The key component in the EFGT is the counter flow heat exchanger and the most important parameter is the temperature difference between the hot and the cold gas. At the hot end of the heat exchanger where the hot gas enters and the heated air exits the temperature difference is defined by $\Delta T_H = T_5 - T_3$ (Fig.4). The result of a parameter variation between 10 and 150 K is shown in Figure 6. The efficiency is a function of turbine inlet temperature and temperature difference in the heat exchanger; the pressure ratio is constant at 4.5.

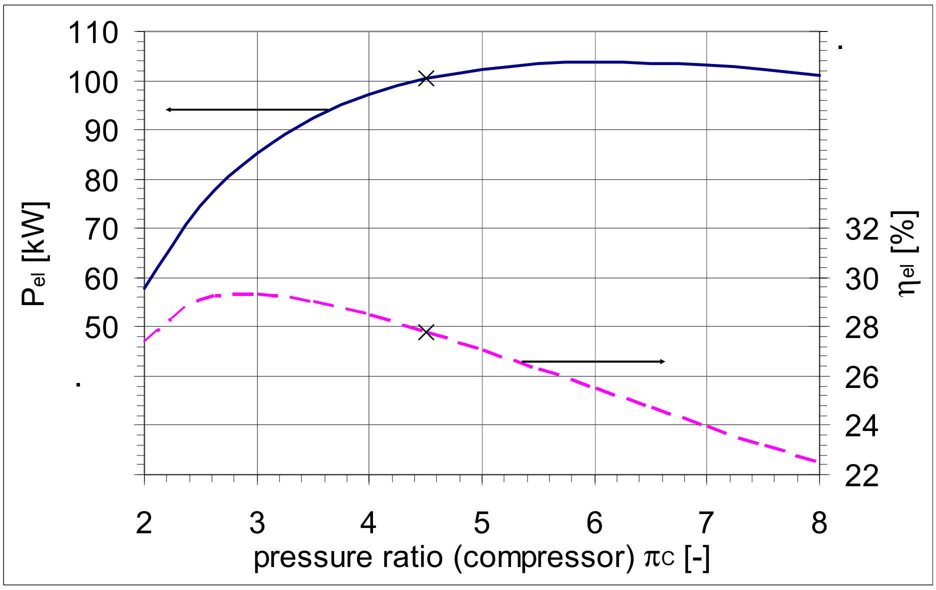


Fig. 5: Power and efficiency of the EFGT as a function of pressure ratio (x design point)

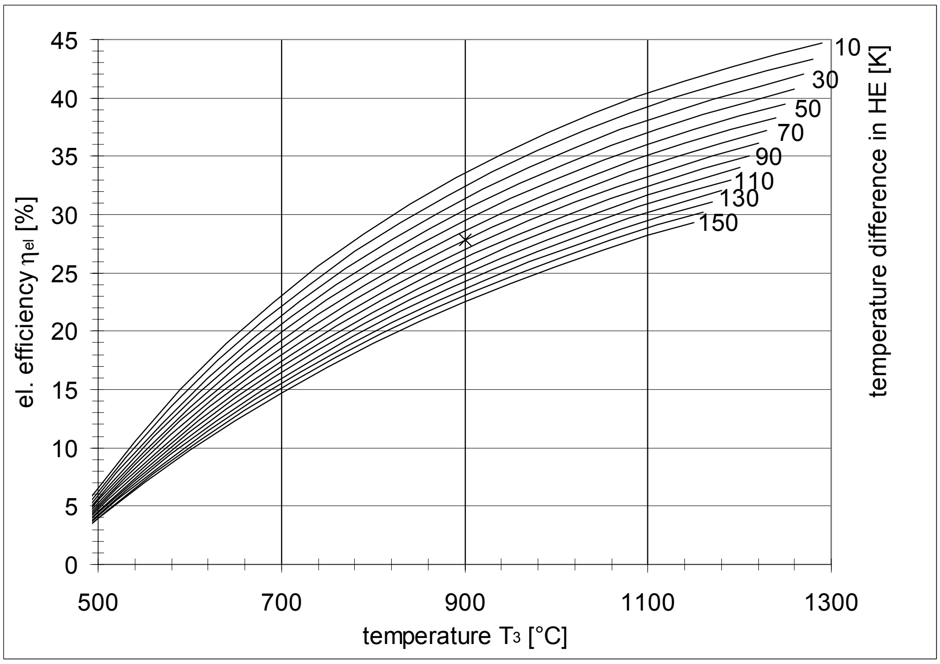


Fig. 6: Efficiency as a function of temperature difference in the heat exchanger and turbine inlet temperature (x design point)

e.g. $\Delta T = 70$ K & T3 = 900 °C => $\eta_{el} = 27{,}8$ %

For a realistic difference of 70 K and a turbine temperature of 900 °C the net electric efficiency reaches 27.8 %. The importance of the temperature difference is demonstrated by doubling ΔT from 30 to 60 K which reduces the efficiency by 3 % points. This is understandable as a decrease in temperature difference in the heat exchanger implies that the exhaust gas leaving the system is cooler, approaching the air temperature at the compressor outlet, and less heat is wasted to the environment. A higher turbine inlet temperature will in general improve the cycle performance.

Another important parameter is the pressure drop in the heat exchanger. An increase in the backpressure of the turbine reduces the pressure difference in expansion and directly impacts on the power output. The total pressure drop in the system consists of the pressure drop in the airflow on the cold side and in the gas flow on the hot side. The effect of both on the net efficiency is shown in Figure 7.

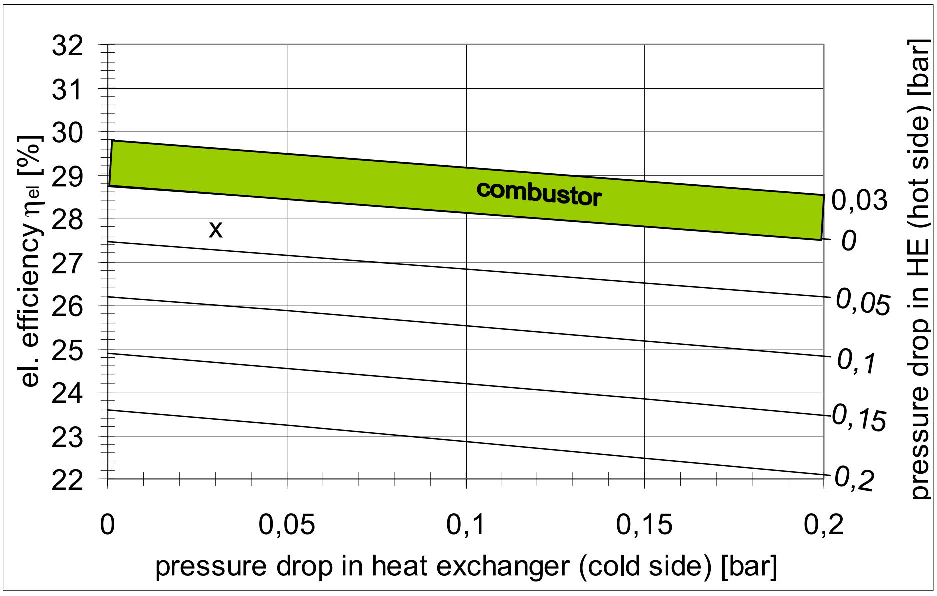


Fig. 7: Net electric efficiency as a function of pressure drop in heat exchanger (HE) (x design point) e.g. Δp (hot side HE & burner) = 0,06 bar & Δp (cold side) = 0,03 bar => η_{el} = 27,8 %

In a realistic design the total pressure drop should not exceed 0.1 bar and a reasonable operating point is drawn in the Figure. The actual pressure drop is determined by the heat exchanger area, again a result of the temperature difference, and the geometric design.

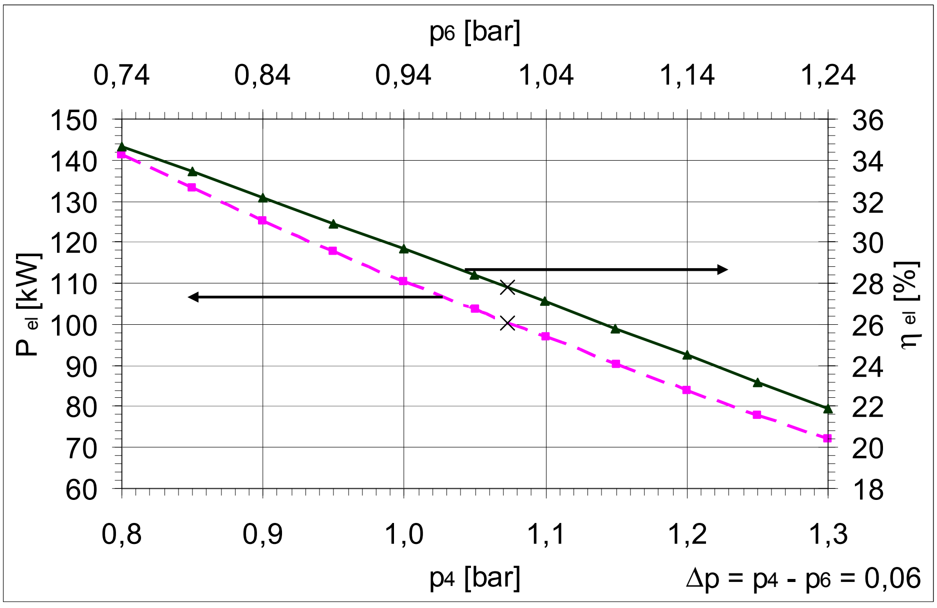


Fig. 8: Variation of the turbine exit pressure p_4 (x design point p_4 = 1.073 bar)

In further calculations the turbine exit pressure p_4 has been varied, Figure 8. The installation of an additional heat exchanger below the recuperator utilising the waste heat for drying would increase p_4 = 1.1 to 1.3 bar. On the other hand, placing a suction blower at the exit of the heat exchanger could decrease the pressure. This implies lowering p_4 = 0.8 to 1.1 bar and would increase the system efficiency. However, the suction blower demands additional power and would only be justified when the efficiency is higher than the turbine.

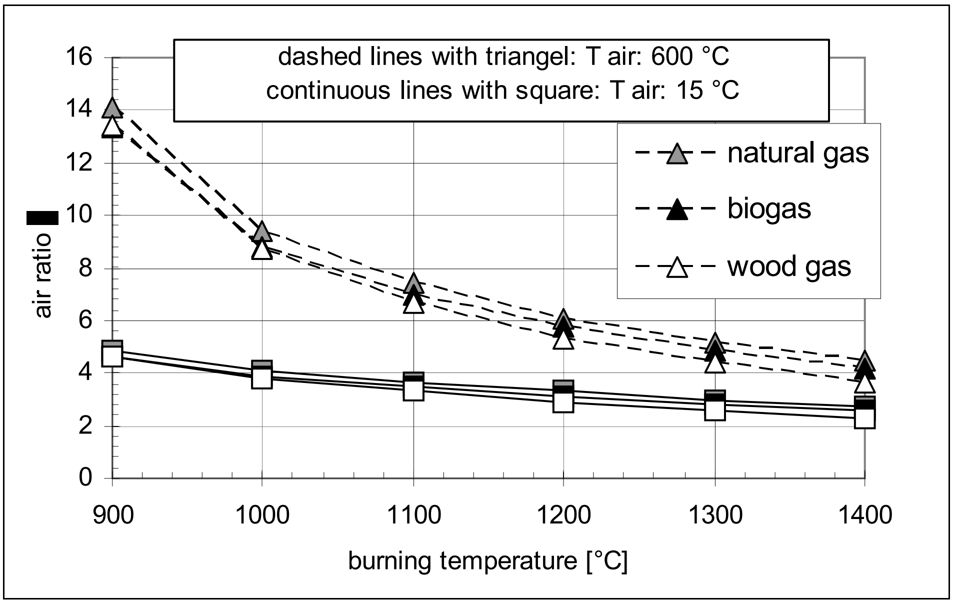


Fig. 9: Air ratio as function of combustion temperature

As already mentioned, the preheating of the air in the recuperator leads to less fuel, e.g. in the standard Turbec micro gas turbine the air enters the combustor with 580 °C. The air ratio is roughly 7.5, which signifies a large amount of surplus air compared to the stoichiometric combustion. In the externally fired turbine the temperature of the air leaving the turbine is 650 °C and a similar high air ratio exists in the combustor. With large surplus air there may be a danger of instable combustion conditions. In Figure 9 the air ratios are calculated for air at 15 °C and 600 °C as a function of the combustion temperature.

Tab. 2: Composition of fuel gases in Fig. 9

		Natural Gas H	Wood Gas	Biogas
CH_4	Fraction in %	0,930		0,600
C_3H_8	Fraction in %	0,013		
C_4H_{10}	Fraction in %	0,025		
CO_2	Fraction in %	0,002	0,106	0,380
N_2	Fraction in %	0,030	0,414	0,005
H_2O	Fraction in %		0,073	0,010
H_2	Fraction in %		0,205	
CO	Fraction in %		0,202	
O_{min}	-	2,088	0,204	1,200
H	kJ/kg	47015	4534	17800

Natural gas is compared to biogas from fermentation and wood gas from thermal gasification, cf. Table 2. With increasing combustion temperature the air ratio falls to values between 8 and 5. For the turbine inlet temperature of 950 °C the combustion temperature is in the range 1000 to 1050 °C. Only for air ratios of 10 and more the stability of combustion needs to be studied in more detail. In the case of too much surplus air some of it may bypass the combustion chamber and either be added to the combustion gas before entering the heat exchanger or used for other purposes. One possibility is a drying process demanding clean hot air as for instance in the food industry.

In an indirectly heated gas turbine any heat at a sufficient high temperature can be used. **Solar energy** as a heat source is one possibility. In a solar thermal-electric system the EFGT has the same role as the Stirling engine. The solar heat from the collector is focussed on a hot air heat exchanger and either alone or together with a backup fuel serves as the energy input to the gas turbine process. The process is shown in Figure 10. A parameter optimisation has been performed with the GateCycle programme. For smaller power units parabolic or dish collectors would be suitable whereas for larger units solar towers are more appropriate. With bio fuels to make-up for solar deficiencies and the diurnal variation, the system would rely fully on renewable energy resources.

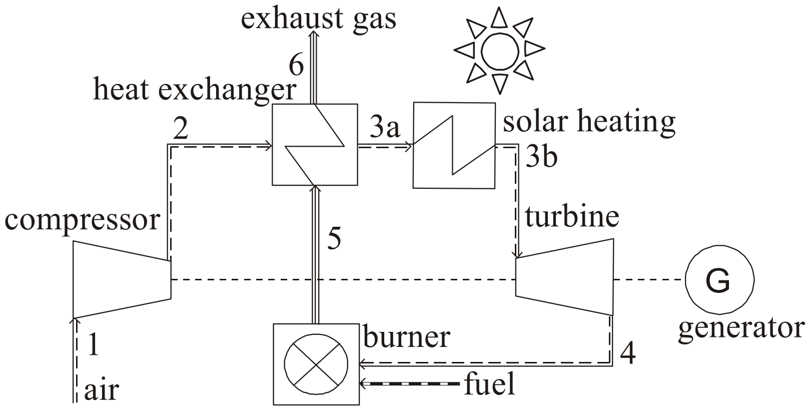


Fig. 10: Simplified process diagramme of the EFGT with solar energy input

Problems

Most of the components of the EFGT are standard parts. The only new items are the high temperature heat exchanger and the combustor for alternative fuels. For combusting biomass in an EFGT-Cycle two alternatives are possible. In both cases, the combustor works at atmospheric pressure and problems with feeding solid fuel to a burner at high pressure or compression of the gas are avoided.

- For solid biomass, a combustor burning finely ground material would have to be coupled with a cyclone to reduce particles in exhaust gas, and in case of a low ash melting point also to remove liquid slag.
- The raw gas from a thermal gasification process may be utilised without tar removal as the higher hydrocarbons are cracked at the prevailing high temperature in the combustor. The biogas is passed through a filter to remove solid particulates before being fed to a standard gas burner. Waste heat from the recuperator can be used for drying and preheating biomass.

The key component in the process is the heat exchanger. In principle, it may be of either regenerator or recuperator type. For reason of better cleaning and for avoiding gas leakage between the clean air and the gas side, a recuperator design has been chosen. The present recuperators operate in the range 200 to 600 °C. Building on the technology for recuperators, it is necessary to have an additional heat exchanger for compressed air in the temperature range from 600 to 1000 °C. To withstand high temperatures of more than 1000 °C, the heat exchanger requires special materials on the basis of either meatallics or ceramics. However, the mechanical loads are not high as the pressure difference between the air and the gas side is moderate, ca. 5 bar. The challenge is to construct a heat exchanger with minimal temperature difference (large heat exchanger area) and minimal pressure loss (small heat exchanger area). The estimated heat flux could be twice as high as in a standard recuperator. Because of more expensive material for high temperatures the price could be around three times as much as for a standard recuperator. For experimental studies the combustion test facility of the Institute for Energy- and Environmental Technology is used to test a high temperature resistant stainless steel heat exchanger module, cooled by compressed air. The design is of the tube-and-shell-type to facilitate easy cleaning of the heat surfaces.

The combustion gases from bio fuels are frequently rich in particulates and may be chemically aggressive and cannot without cleaning be used in combustion engines or a directly fired gas turbine. The indirect cycle shifts the problem to the heat exchanger which can more easily accommodate high impurity loads. This is common for both the Stirling engine and the steam cycle with which the EFGT competes in the market for small to medium size CHP-units. A tentative comparison of the net electric efficiency of the three technologies burning solid biomass is given in Figure 11.

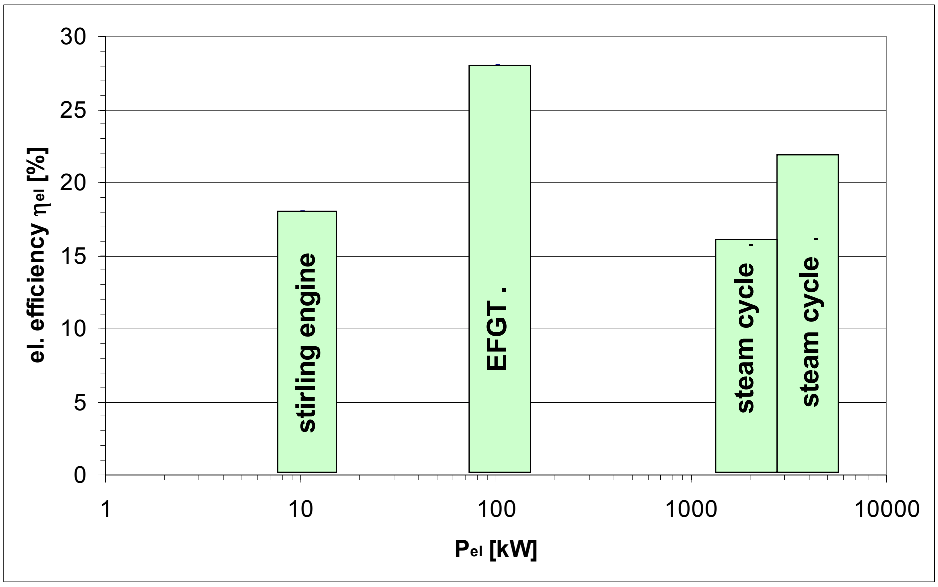


Fig. 11: Typical unit size and efficiency of small Combined-Heat-Power plants burning solid biomass [3, 4]

Steam cycle power plants are well established but need to be in the MW-range to enable a reasonable efficiency [3]. The water-steam technology is complex and expensive and is only justified in large units. At the other end of the scale, the Stirling engine achieves high efficiencies even in the kW-range [4]. The technology is in the demonstration stage with units operating on natural gas and in combination with a solar dish. The EFGT relies on standard gas turbine components. The atmospheric biomass combustor is under development and the gas-to-gas heat exchanger can draw on a wide experience in the chemical and power industry. The costs of a future commercial EFGT is difficult to predict at the present stage but a rough idea may be gleamed from Martelli [2] who suggests a 50 % increase compared with a standard gas turbine. With the cost of micro gas turbines in the range € 1000 – 1200 per kW-el this would lead to € 1500 – 1800 per kW for the EFGT. A small steam unit with 2 MW-el is reported to call for an investment of € 2200 per kW-el [3]. Also for the Stirling engine the cost estimates are still uncertain. Due to the general cost-size relation a small unit is specifically more expensive than a large one, and the investment costs for the 10 to 30 kW range will likely be higher than the two other technologies.

It is still impossible to say with any certainty which technology will yield the lowest generating costs as this entails operation, maintenance and availability. However, the EFGT promises to become a competitive energy conversion technology for bio fuels, in particular for solid biomass.

Conclusions

In general, fuels causing problems in conventional gas turbines can be burned in the EFGT. The possible power spectrum from 30 to 2.000 kW_{el} is very suitable for decentralized CHP units utilizing biomass and contributing to minimizing CO_2-emission. The cycle can be operated more economically, if waste biomass is used.

The external firing may be realised either with solid, liquid or gaseous biogenic fuels. The combustion technology needs to be developed but can draw on experience and designs of combustors for fossil fuels. At present, the most interesting option is low calorific gas from gasification, which has difficulties in finding appropriate conversion technologies due to a large amount of impurities.

The implicit utilisation of the waste heat from the turbine in the indirectly heated cycle improves the efficiency, which is close to the high efficiency realised in the micro gas turbine with recuperation. The optimisation of the heat exchanger and the recuperator is a prime objective. Whereas the recuperator uses conventional materials, the high temperature heat exchanger requires special materials, either metallic or ceramic. In the longer time perspective, a combination with solar energy is a challenging option.

The additional costs of the heat exchanger and the biomass combustor will have to be compensated for by the lower price of biomass and waste materials in comparison with premium fuels like natural gas in standard gas turbine power plants. In comparison with a combined-cycle where the waste heat from the turbine is utilised in a steam cycle, the cost of a gas-gas heat exchanger is lower than for a complete water-steam cycle. However, the EFGT-Cycle is not in direct competition with a steam power plant or a Stirling engine, because its power range fills the gap between the two. The efficiency and the investment cost of the EFGT promise advantages over the other technologies.

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