5

Gas Turbines in Unconventional Applications

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1. Introduction

The Chapter presents unconventional gas turbine applications. Firstly some selected non-MAST (Mixed Air and Steam Turbine) solutions are discussed – these are intended for smaller gas turbine systems, where the regenerative heat exchanger supplies energy for an additional thermal cycle, where it is utilised.

The gas turbine cycle (Brayton) may be coupled with several other thermal engines (like another Brayton, Diesel, Kalina, and Stirling). Those hybrid systems have several previously unrecognised advantages. They may find applications in some market niches.

The next section describes hydrogen-fuelled gas turbine solution. Big international programme – WE-NET – is discussed. Several hydrogen-fuelled gas turbine concepts based on those programmes are proposed: Westinghouse, Toshiba, Graz, New Rankine. The section provides description of them all, including specification of possible efficiency values. The development programmes themselves are also reviewed. This part of the text describes also potential combination of a hydrogen-fuelled gas turbine and a nuclear power generation unit which might be used to cover peak load power demands in a power system.

Last section of the chapter discusses integration of a fuel cell into a gas turbine system. High temperature fuel cells can play a role similar to a combustion chamber but simultaneously generating additional power. Fuel cell hybrid systems for both high-temperature types of fuel cells – Solid Oxide Fuel Cell (SOFC) and Molten Carbonate Fuel Cell (MCFC) – are proposed. Additionally, some specific properties of the MCFC can be used to reduce carbon dioxide emissions from the gas turbine itself.

Gas turbine systems, particularly combined cycle units, are among the most popular power systems in the modern world. This results from the very fast technical progress allowing to gradually increase the parameters at the turbine inlet as well as unit outputs. There is also a parallel development trend of searching for new unconventional solutions, which would allow to achieve efficiencies higher than enabled by a simple cycle. Scheme of the simple cycle process is presented in Fig. 1. Simple cycle efficiency in many cases is too low.

Internal power N_i of a gas turbine can be obtained by an analytical approach by using the relation 1, which is obtained with assumption that the process is real (contained losses) and working fluid is modelled as semi-ideal gas.

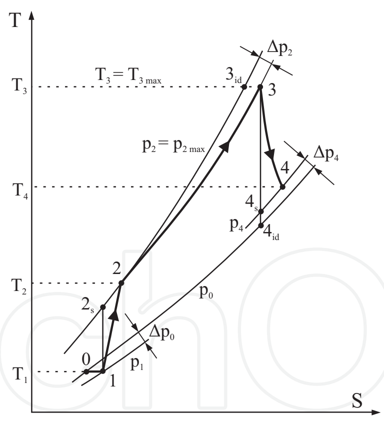

Fig. 1. Thermal process of a simple cycle gas turbine

$$
\begin{aligned}
N_i &= N_T - N_K \\
&= G_T \cdot c_{p,T} \cdot T_3 \left(1 - x_T\right) \cdot \eta_T - G_K \cdot c_{p,K} \cdot T_1 \cdot \left(x_K - 1\right) \cdot \frac{1}{\eta_K} s \\
&= G_K \cdot c_{p,K} \cdot T_1 \cdot \left[\frac{1}{1-\beta} \cdot \bar{c}_p \cdot \Theta \cdot \left(1 - x_T\right) \cdot \eta_T - \left(x_K - 1\right) \cdot \frac{1}{\eta_K}\right]
\end{aligned}
\tag{1}
$$

where: N_T, G_T, N_K, G_K stand for internal power and fluid inlet mass flow in turbine and compressor, respectively, η_T – turbine internal efficiency, η_K – compressor polytropic efficiency, $c_{p,T}$, $c_{p,K}$ – averaged isobaric specific heat capacity for the working fluid in turbine and compressor, k_K, k_T – averaged heat capacity ratios (isentropic exponents) for compressor and turbine for the flue gas and air. Other symbols used in Eq. 1 are:

$$
\begin{aligned}
\bar{c}_p &= \frac{c_{p,T}}{c_{p,K}}; \quad \Theta = \tfrac{T_3}{T_1} \\
x_T &= \frac{1}{\Pi_T^{m_T}}; \; m_T = \tfrac{k_T - 1}{k_T}; \; \Pi_T = \frac{p_3}{p_4} \\
x_K &= \Pi_K^{m_K}; \; m_K = \tfrac{k_K - 1}{k_K}; \; \Pi_K = \frac{p_2}{p_1} = \Pi \\
\Pi = \Pi_K &= \frac{p_2}{p_1}; \quad \Pi_T = \tfrac{p_3}{p_4} \quad \Pi_T = \varepsilon \Pi_K \\
\beta &= \frac{G_p - \Delta G}{G_T}
\end{aligned}
\tag{2}
$$

where: G_p – fuel mass flow delivered to a combustion chamber, ΔG – air mass flow delivered for cooling purposes of the hottest parts of gas turbine and bearings, and leakages losses, Π_T, Π_K are pressure ratios of outlet and inlet of compressor and turbine, and losses factor ε describes combination of pressure losses at compressor inlet, combustion chamber and turbine outlet.

Subscripts of working fluid parameters (pressure and temperature) indicate location in reference to the turbine diagram (Fig. 2). Specific internal power by definition is expressed as:

$$
N_j = \frac{N_i}{G_K} \tag{3}
$$

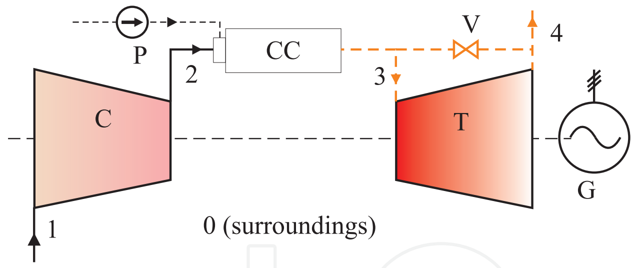

Fig. 2. Simple cycle gas turbine diagram: C – compressor, T – turbine, G – electric generator, CC – combustion chamber, P – fuel pump, V – bypass valve

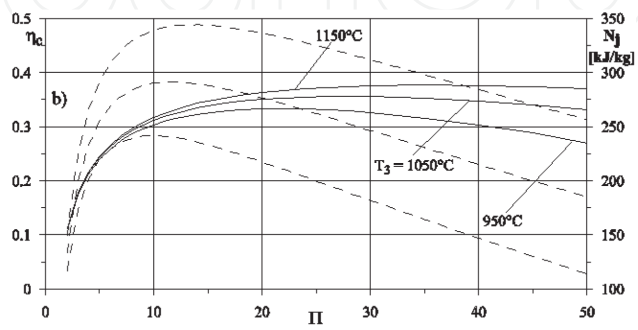

Fig. 3. Specific power output and thermal efficiency of a simple cycle gas turbine unit as functions of the pressure ratio. Dashed lines denote specific power output.

and thermal efficiency is a ratio of internal power and fuel power input:

$$\eta_c = \frac{N_i}{N_p} \tag{4}$$

Characteristics of a simple cycle gas turbine unit based on the relationships presented above are shown below. Performance calculations were carried out at variable pressure ratio Π_K and temperature T_3 (at turbine inlet, without taking into consideration cooling losses, i.e. $\Delta G = 0$). Constant values of polytropic efficiencies of turbine $\eta_T = 0.88$ and compressor $\eta_K = 0.88$, pressure losses factor $\varepsilon = 0.95$ were used. Ambient conditions (pressure and temperature) were assumed according the ISO conditions:$T_0 = 288$ K (15°C).

The first curve (Fig. 3) illustrates key performance parameters of a simple cycle gas turbine unit (specific power output and efficiency) as functions of pressure ratio and Turbine Inlet Temperature (TIT). It can be seen that in a real system maximal specific power output is achieved at lower pressure ratio then the maximal efficiency.

Another curve shows relation between efficiency and a specific power output (Fig. 4). This curve was obtained with the same relationships and assumptions as for Fig. 3. The range of investigated temperatures was extended (while keeping the assumption that there is no turbine cooling, which results with performance figures being somewhat exaggerated). Points on individual curves denote selected pressure ratio values. Analysis of the cooling system impact on performance of a gas turbine can be found for example in Nat (2006).

It is possible to considerably increase efficiency and other performance figures of a gas turbine unit at the expense of introducing new components and making the flow system more complex. Therefore, except for simple gas turbines, also complex units with more

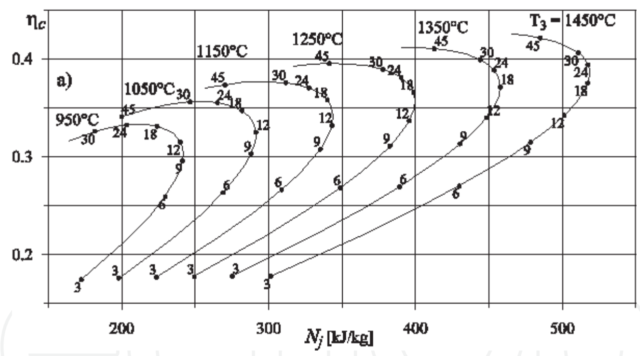

Fig. 4. Thermal efficiency of a simple cycle gas turbine unit as a function of specific power and turbine inlet temperature.

sophisticated working cycle are developed and build. These may involve for example heat regeneration, inter-stage cooling and re-heating. Such solutions require introduction of new components: regenerative (or recuperative) heat exchanger, gas coolers, combustion chamber cooling systems or inter-stage heaters. Also the compressor and turbine need to be split into high- and low-pressure parts (or even more sections).

In parallel to development of simple cycle gas turbines, also combined cycle solutions are developed, among them units with additional thermal circuit. Technical progress in gas turbine development is coupled with development of combined cycle systems, including solutions with combination of different thermal engine types. Combination of a gas turbine thermal cycle with an intermediate- or low-temeprature steam power plant cycle may be seen as the most efficient method to increase the efficiency of a process involving gas turbines used in industrial practice. Nowadays gas turbine combined cycle solutions are among the most intensely developed power generation technologies. "Conventional" Gas Turbine Combined Cycle or GTCC solutions may reach a nominal efficiency reaching 60%.

It is however not possible to achieve such high efficiencies at intermediate output GTCC units. It may prove more feasible to utilise simpler systems at lower scales. Specific investment cost for low-output GTCC unit is quite high, while the efficiency may be considerably lower. They do display high reliability though. One method to improve performance of low-output GTCCs may be utilising recuperation, inter-stage compressor cooling or working agent integration, e.g. utilising air-steam mixture as a working agent.

This last concept utilised in gas turbine systems has been proposed in various forms and under different names. One of the used names is Mixed Air and Steam Turbine (MAST). Another name is Cheng Cycle, and General Electric utilises name Steam Injected Gas Turbine – STIG. The gas turbine concepts with air humidification are known as Humid Air Turbines (HAT), Cascaded Humidified Advance Turbines (CHAT) or Wet Compression systems.

2. Gas turbines with other thermal engines

2.1 Brayton-Brayton

The Brayton-Brayton cycle (diagram shown in Fig. 5) is a combination of two simple-cycle gas turbines. Working agent in one of them is flue gas, and in the other – air. The turbines are interconnected with a high-temeprature air/flue gas heat exchanger. As the air turbine utilises exhaust heat, this solution enables to considerably improve total efficiency when compared to

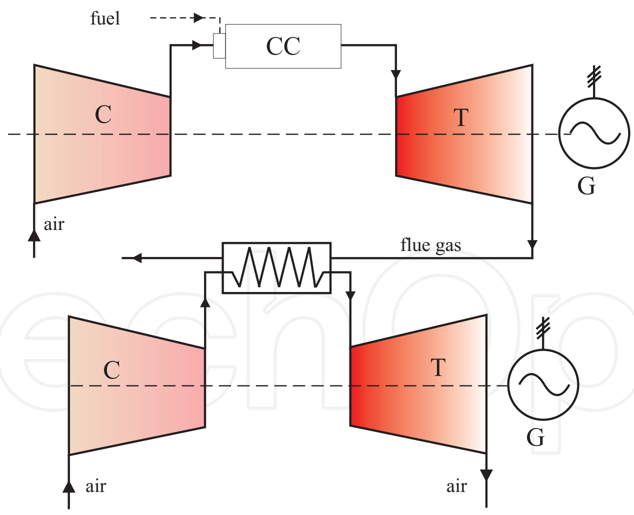

Fig. 5. Diagram of a combined cycle composed of a gas turbine and an air turbine supplied with heat recovered from the flue gas flow. (Brayton-Brayton)

a simple cycle solution. When compared to a classic GTCC, the Brayton-Brayton techenology requires less auxiliary equipment. It also needs less space and has lower investment cost. Systems of this type are not extensively analysed in the literature. Discussion presented in this chapter is based on the analyses presented in detail in Baader (n.d.). Performance of each gas turbine utilised in the system may be determined according to the rules and relations presented in the introduction.

Performance of the entire system depends on parameters (selection criteria) of the heat exchanger and air-based turbine unit. Calculations carried out in order to determine performance of a system shown in Fig. 5 were based on the following assumptions:

1. The air cycle is designed to maximally utilise energy carried with the flue gas exhausted from the gas turbine unit. The flue gas is cooled as far as possible, but to temperatures no lower than 200°C. Efficiency of the flue/air heat exchanger is 80% and the pinch point temperature is 30°C.
2. Pressure ratio of the air turbine is designed to enable achieving highest possible internal power.
3. Assumed compressor and turbine polytropic efficiencies are 88% (just like for the simple cycle).
4. Assumed pressure losses at the heat exchanger for both air and flue gas is 3.4%, which results with the pressure loss coefficient of 0.928.
5. Just like in case of previous calculations no impact of internal gas turbine cooling systems on system's performance is taken into account.

Other assumptions and the relationships used followed the rules previously presented for the simple cycle. Results are shown in the charts – characteristic curves for the Brayton-Brayton system. Fig. 7 shows relation between the system's efficiency and specific power output. Values shown here indicate that a considerable increase in reference to a simple cycle system may be expected (compare to Fig. 2).

Maximum specific power occurs at a lower pressure ratio value than for a simple cycle system, but at higher values than for a system with regeneration. Specific power is much higher than in case of a simple cycle or a regenerative system.

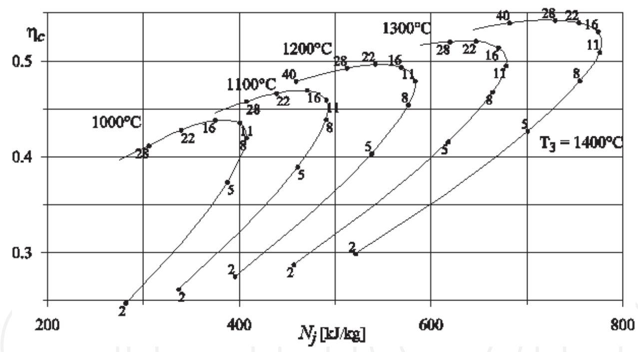

Fig. 6. Thermal efficiency of a Brayton-Brayton system as a function of specific power output and Turbine Inlet Temperature. Numerical values shown in the chart denote the gas turbine pressure ratio.

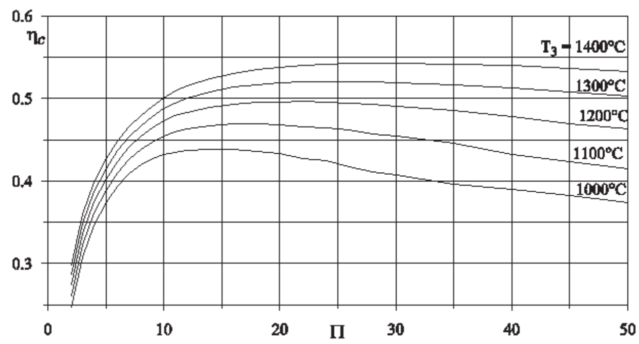

Fig. 7. Thermal efficiency of a Brayton-Brayton system as a function of gas cycle's pressure ratio and turbine inlet temperature

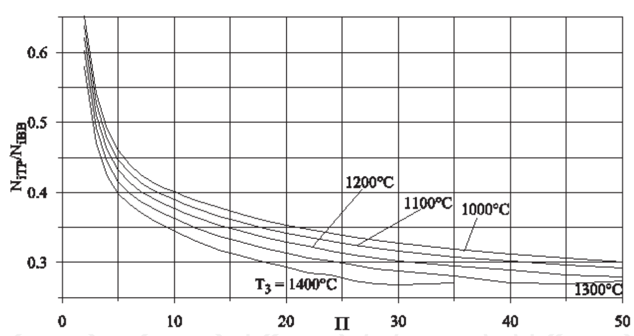

Fig. 8. Share of the internal power generated in the air turbine in a total internal power of a Brayton-Brayton system as a function of gas cycle's pressure ratio and turbine inlet temperature.

Fig. 7 shows relation between the efficiency of a Brayton-Brayton system and the pressure ratio in the gas cycle. The next figure (Fig. 8) illustrates distribution of internal power between the gas and air cycles. High share of the air cycle at low gas cycle pressure ratio values results from the high temperature of exhaust gas delivered to the heat exchanger downstream from the gas turbine unit in such a case. This allows to achieve high pressure ratio in the air cycle – as already mentioned this value is optimised to achieve highest possible output. As shown in Fig. 9 the pressure ratio of the air cycle at low pressure ratio of gas cycle is high Π_{KP}/Π much lower than one). The results in this area should be seen as exaggerated, as the model does not include any limit of maximum (achievable) air temperature downstream from the heat

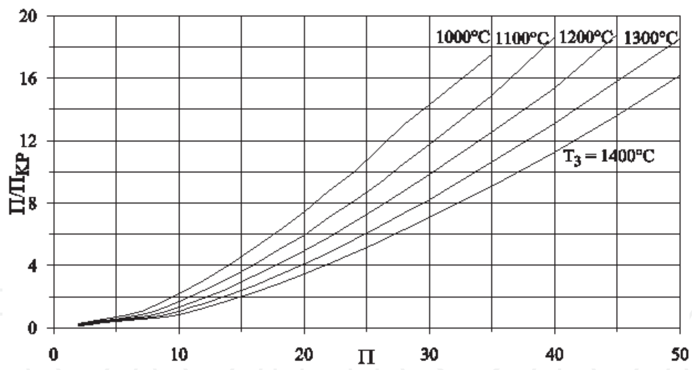

Fig. 9. Ratio between the gas cycle pressure ratio and air cycle pressure ratio in a Brayton-Brayton system as a function of the gas cycle pressure ratio.

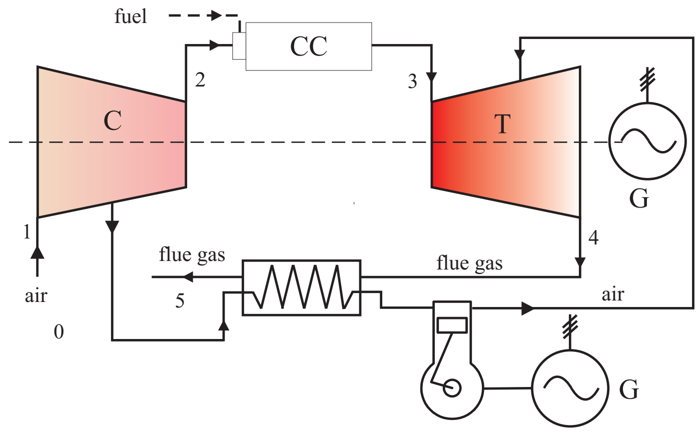

Fig. 10. Brayton-Diesel system flowchart

exchanger. In fact such a restriction would result from the material properties – this effect would practically rule out systems with low gas cycle pressure ratio. Practically only systems where the gas cycle parameters are similar to those used in simple cycle systems, i.e. those close to the parameters range enabling maximum specific power outputs and efficiencies (see Fig. 3 and Fig. 4), should be seen as reasonable.

2.2 Brayton-Diesel

One of the interesting proposals presented in the literature is the so-called Brayton-Diesel system Poullikkas (2005).

The Brayton-Diesel system (Fig. 10) is a combination of a simple cycle gas turbine with a heat exchanger and a reciprocating expander. Working agents are exhaust gas and air. Part of the air flow from the compressor is supplied to the combustion chamber, while some air is transferred into a heat exchanger where it is heated by the gas turbine exhaust. Then this flow expands in a reciprocating expander and is further fed into the low-pressure stages of the gas turbine, mixing with the exhaust gases. It was assumed that the mixing process is isobaric. The resulting mixture further expands to the turbine exhaust pressure.

Application of the Brayton-Diesel system allows to increase maximum power output of a single unit by some 6–12% and increase maximum efficiency by 1.3–3.6% in the investigated temperature range (900–1,300°C).

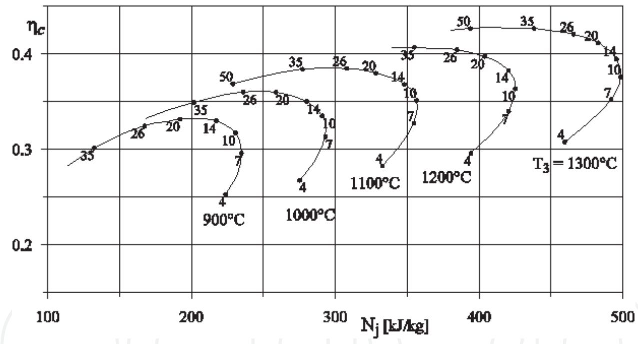

Fig. 11. Thermal efficiency of a combination of a gas turbine and reciprocating expander (configuration as in Fig. 10) as a function of specific power and TIT (T_3). Numerical values shown in the chart denote gas turbine pressure ratios.

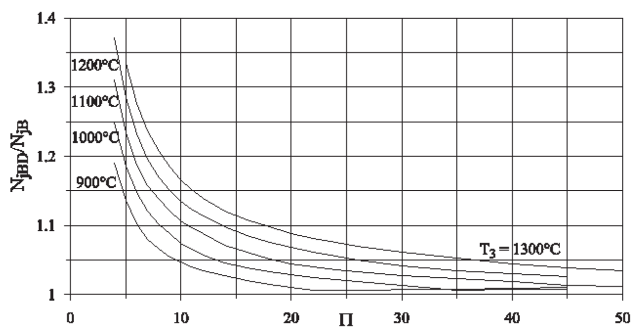

Fig. 12. Increase of a specific power output of a system with a reciprocating expander in comparison to the specific power output of a simple cycle gas turbine as a function of gas turbine pressure ratio and TIT (T_3).

An interesting solution could be also supplying fuel to the reciprocating expander, which would also require combustion process there. This case is currently a subject of further investigation.

The Brayton-Diesel system can also be used for combined heat and power applications. Mixture of exhaust gas and air at the outlet of the heat exchanger has a temperature in range of 170–330°C, so it can be used as a source of process heat.

Internal power of the combined cycle N_{iz} was calculated according to the rules and relationships given in Miller (1984), by binding it to the parameters of the thermal process according to equations:

$$N_{iz} = N_{Tz} + N_R - N_{Kz} \tag{5}$$

where: N_{Tz}, N_R, N_{Kz} – internal power of the turbine, reciprocating expander and combined-cycle compressor, respectively.

Specific power of the system was defined as follows:

$$N_{jz} = \frac{N_{iz}}{G_K} \tag{6}$$

Analysis of the results obtained for the Brayton-Diesel system model allows to draw following conclusions:

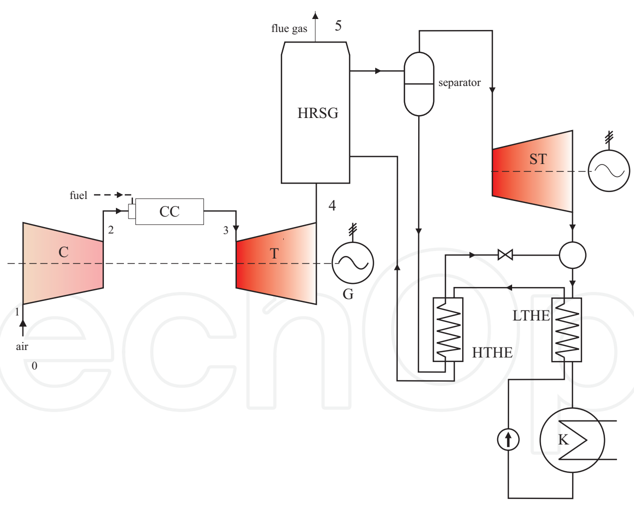

Fig. 13. Process diagram of a Brayton-Kalina system: C – compressor, T – turbine, G – generator, CC – combustion chamber, ST – steam turbine, K – Condenser, HRSG – heat recovery steam generator, HTHE – high temperature heat exchanger, LTHE – low temperature heat exchanger.

1. Combination of a simple-cycle gas turbine with a heat exchanger and a reciprocating expander increases thermal efficiency and specific power of the system.
2. Maximal thermal efficiency and maximal specific power increase; the higher the maximum temperature in the system – the higher growth of those parameters.
3. Maximal thermal efficiency grows slightly, the growth of the specific power is more significant.
4. Optimum values of the simple cycle pressure ratio are lower for the combined cycle than for the simple cycle.
5. Using reciprocating expander is most profitable for possibly high maximal temperature of the system and possibly low pressure ratio of the simple cycle.
6. Heat exchanger used in this system has to provide significant amount of heat to the air ($\Delta T_{pow} = 173 \ldots 470°C$) so it needs to have quite large heat exchange area which makes it expensive and bulky.
7. Discussed system can also be used in combined heat and power applications as the air-exhaust gas mixture flowing out of the heat exchanger has a quite temperature in range 170...330°C.

While selecting design parameters of the Brayton-Diesel system also the economical analysis is very important. Increasing TIT and using very large heat exchanger could prove too expensive when compared to the benefits resulting from the higher efficiency.

2.3 Brayton-Kalina

The Kalina system is a variant of the Organic Rankine Cycle (ORC). It utilises a mixture of water and ammonia as a working agent. Ratio between ammonia and water may be changed, depending on the process, which enables adjusting boiling and condensation temperatures. The Kalina cycle is based on the Rankine cycle with added distilling and absorption components (Fig. 14). Its main advantage is already mentioned ability to adjust boiling and

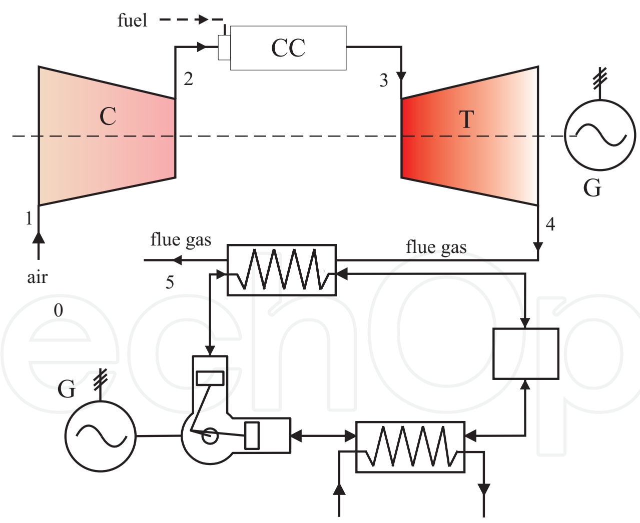

Fig. 14. Process diagram of a Brayton-Stirling system. C – compressor, T – turbine, G – generator, CC – combustion chamber

condensation temperatures of the working agent, allowing to adjust entire cycle to the variable temperature of the heat source. Changing boiling and condensation temperatures during plant operation provide an additional degree of freedom when compared to a traditional steam cycle (including ORC solutions). Variable water content in the ammonia solution allows to optimise operational parameters of the cycle and thus improve the efficiency which may be close to the Carnot cycle.

Working agent properties allow to decrease the temperature difference in reference to the exhaust gas temperature in the heat recovery steam generator (the pinch point effect in evaporator is missing).

The Kalina cycle has a number of features making it competitive against the Rankine cycle (both "classic" and organic):

1. It may produce 10–30% energy more than a Rankine cycle.
2. It has lower (by some 40%) space requirements than a corresponding system with a steam turbine.
3. Exhaust steam pressure is higher than ambient pressure so it is not required to create vacuum in the condenser – this enables to shorten the start-up time.
4. The Kalina cycle can be easily optimised for changed ambient conditions by adjusting working agent composition.

The licence for construction of Brayton-Kalina systems (process illustrated in Fig. 14) has been owned by General Electric since 1993. While further technology development was declared and a commercial power plant based on it was supposed to be completed by 1998, in fact the programme was suspended.

2.4 Brayton-Stirling

The combination of Brayton and Stirling cycles can have different configurations. Heat to the Stirling engine may be delivered in two different points of the cycle. One possibility is heat transfer through a heat exchanger installed directly inside the combustion chamber of a

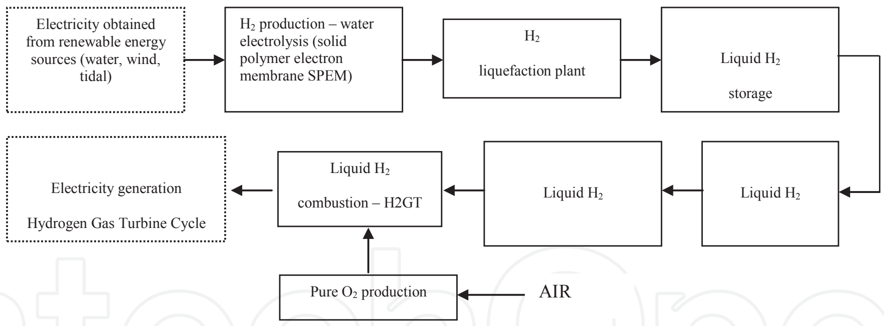

Fig. 15. Configuration of a hydrogen-based power generation system

gas turbine unit. The other possibility is recovering exhaust gas heat downstream from the gas turbine (as shown in the Fig. 14). Performance of this configuration may be optimised, with one of the variables being materials used for construction of the main heat exchanger. A Brayton-Stirling plant is able to achieve relatively high efficiency values. In a system built some years ago and utilising a Rolls-Royce RB211 gas turbine installation of a Stirling engine allowed to increase the power output by 9 MW (from the original 27.5 MW). This increased the total efficiency Poullikkas (2005) to 47.7%. And advantage of such a solution is its compact form and simplicity of waste heat utilisation.

3. Hydrogen gas turbine

The issue of environment pollution caused by human social and economical activities in recent years, has caused an increase of interest in environment-friendly (zero-emission) technologies of energy generation and utilisation. This approach is commonly seen as a way to solve global environmental problems that have recently become more and more intense. Environmental footprint of classic energy generation technologies based on fossil fuels – which are currently in use – is an argument for developing alternative technologies for energy generation. Power generation based on release of chemical energy of combusted hydrocarbon fuels is one of the main causes of the biosphere degradation. Combustion of fossil fuels (coal, lignite, oil, gas) causes an environmental footprint: emission of combustion to the atmosphere or the problem of combustion waste management which still remains unsolved.

An increase of CO_2 concentration in the atmosphere intensifies the greenhouse effect and thus influences climate changes. Moreover, SO_x and NO_x emissions caused by many industry processes, especially combustion used in energy industry, causes acid rains falling in industrial areas.

Hydrogen as a clean fuel is a subject of interest for many research institutions all over the world. National and international research projects aimed at utilising hydrogen generated by renewable energy sources (e.g. wind or solar) have been proposed in various parts of the world Gretz (1995); Kaya (1995).

3.1 Review of the WE-NET system concept with hydrogen turbine

The Japanese World Clean Energy Network programme using hydrogen conversion (WE-NET) started in 1993 as a part of larger "New Sunshine Program", whose main goal was to develop environment-friendly energy generation technologies that could meet constantly increasing energy demand and at the same time allow to solve arising environmental problems (n.d.). The main target of the WE-NET project is to construct a worldwide energy network for effective supply, transmission and utilisation of non-carbonaceous renewable energy (water, solar, wind, sea) using hydrogen as a clean secondary energy carrier. The energy conversion system proposed in the WE-NET programme includes:

- Hydrogen production through water electrolysis
- Conversion of gaseous hydrogen into forms enabling easy transport
- Storage and maritime transport, and
- Utilisation in multiple areas (including power generation by hydrogen combustion turbine systems).

Diagram of such a power generation system is shown in the Fig. 15.

Alternative energy generation technologies based on hydrogen and proposed by the WE-NET programme could limit emissions of CO_2 and other greenhouse gases to the atmosphere. Additionally this technology would enable using renewable energy sources on large industrial scale. It is due to the fact that H_2 would be produced from water by electrolysis process powered by renewable energy. Present feasibility of the utilisation of renewable energy sources depends on a specific project site and scale, ergo depends on a local energy demand and is also limited by the losses in electricity transformation and distribution processes.

Last but not least, CO_2 emission has recently become one of the most important subjects of the environment protection. In order to meet the newly defined environmental requirements new and more efficient technologies need to be researched. Performance of an ordinary GTCC unit is approaching the maximum efficiency level limited by NO_x emission problem. Thus a Hydrogen Combustion Turbine System could be seen as an interesting research subject, because it does not cause any NO_x or CO_2 emissions and could be able to achieve over 60% (HHV) efficiency.

Various cycles for the hydrogen combustion turbine system could be proposed to obtain high thermal efficiency (over 60% HHV). 500 MW class gas turbine system with turbine inlet temperature of 1,700°C was proposed as a result of the preliminary research. High performance parameters are crucial, as hydrogen is still a very expensive fuel (due to high cost of production, storage and transport).

The Japanese WE-NET Program NEDO (1997) predicts an implementation of the Hydrogen-Fuelled Combustion Turbine Cycle (HFCTC) as a new energy source for power generation sector. In this respect, a configuration and performance study of the HFCTC was conducted Miller et al. (2001). The WE-NET Program aims at establishing a global energy network using renewable energy based on utilisation of hydrogen – a secondary clean energy carrier. Production cost of hydrogen is relatively high, so it is required to design a power generation system with thermal efficiency superior to that of existing conventional power units. Thermal efficiency above 60% HHV is required (it means above 71% LHV).

Nowadays, thermal efficiency of the most advanced combined cycles (natural gas-fuelled) is close to 60% LHV (54% HHV) and it is comparable to 50.4% HHV if hydrogen fuel is used.

It is obvious that efficiency should be increased by about 10 percentage points, which is an equivalent of about 20% in comparison to the most efficient contemporary power plant units. It is a very serious technological challenge (a qualitative change, not only quantitative).

The development target was to build a full-scale 500 MW_e power generation unit. Besides increasing working agent's temperature at the turbine inlet to 1,700°C (at present it is a level of 1,500°C), it is essential to implement a new approach (different than traditional one) to both conceptual design of the system (its configuration and working parameters) and detailed design solutions. The issue was challenging and potentially beneficial.

Taking the above into account, the main problem was to identify correct cycle concepts and evaluate their performance characteristics. Due to its advantageous features, the GRAZ cycle was treated as a basic concept in the WE-NET Program Mouri (1999). However, during the research conducted by the authors several other concepts were analysed Iki et al. (1999); Miller et al. (2002). The research included, among others, the following main items:

- Cycle identification and assessment,
- Cycle selection,
- Comparison of selected cycles' performance in nominal state under the same reference conditions.

Until now several concepts of the HFCTC have been proposed, out of which the most important are:

1. Combined Steam Cycle with Steam Recirculation developed by Prof. H. Jericha (Technical University of Graz), commonly referred to as the GRAZ cycle Iki et al. (1999); Jericha (1984); Moritsuka & Koda (1999).
2. Direct-Fired Rankine Steam Cycle (New Rankine Cycle) which was studied in the following variants:
   (a) proposed by Toshiba Co., commonly referred to as the TOSHIBA cycle Funatsu et al. (1999); Moritsuka & Koda (1999);
   (b) proposed by Westinghouse Electric Co., commonly referred to as the WESTINGHOUSE cycle Bannister et al. (1997);
   (c) Modified New Rankine Cycle, as authors' own concept, commonly referred to as the MNRC cycle.

A common feature of the abovementioned cycles is that only one working medium (steam) is used for both the topping and bottoming cycles. This is enabled by replacing external firing system (as in the Rankine steam cycle) with direct firing concept (similar to gas turbines or reciprocating engines). Main assumption made here is stoichiometric combustion of hydrogen and oxygen mixture. This combustion takes place inside a stream of a cooling steam, which cuts down combustion temperature to 1,700°C. It is also assumed that hydrogen and oxygen are available at the ambient temperature and at pressure level that allows to inject them into a combustor. It means that hydrogen would need to be provided as cryogenic liquid, however cryogenic energy could be utilised for pure oxygen production in an air-separator unit.

GRAZ, TOSHIBA, WESTINGHOUSE and MNRC cycles were analysed in comparable conditions to evaluate their performance. The analysis was undertaken for the same specific conditions with the same assumptions and property tables. Results of the research for abovementioned cycles published so far, do not provide an opportunity for such a comparison because of incomparable and/or not clear conditions and assumptions of individual studies. Cycles without cooling system were only taken into account in order to create an opportunity for an explicit evaluation.

Fuel type	η_{LHV}/η_{HHV}
Hard coal	1.05
Natural gas	1.11
Hydrogen	1.19

Table 1. Relation between LHV and HHV efficiency for different types of fuel.

3.2 Performance analysis of WE-NET systems with hydrogen turbine

3.2.1 Theory

At the beginning of this section some remarks on cycles' efficiency definition are presented. There are four kinds of efficiency definition used for the comparative analysis:

1. Carnot efficiency which defines theoretical limits for the efficiency of the specific cycle:

$$\eta_C = \left(1 - \frac{T_{bottoming}}{T_{topping}}\right) \cdot 100\% \tag{7}$$

   where: T – absolute temperature, K.

2. Rankine cycle efficiency related to the simple steam Rankine cycle:

$$\eta_R = \frac{P}{Q_{in}} \cdot 100\% = \frac{P}{P + Q_{cond}} \cdot 100\% = \frac{1}{1 + \frac{Q_{cond}}{P}} \cdot 100\% \tag{8}$$

   where: P – output power of the cycle, MW, Q – heat flow, in – input, $cond$ – output.

3. HHV efficiency related to higher heating value:

$$\eta_{HHV} = \frac{P}{m_{fuel} \cdot HHV} \cdot 100\% \tag{9}$$

   where: m – mass flow, kg/s; HHV – Higher Heating Value of the fuel, MJ/kg.

4. LHV efficiency related to lower heating value:

$$\eta_{LHV} = \frac{P}{m_{fuel} \cdot LHV} \cdot 100\% \tag{10}$$

   where: m – mass flow, kg/s; LHV – Lower Heating Value of the fuel, MJ/kg.

HHV is LHV plus heat of evaporation of water content in the fuel (2,500 kJ/kg). The larger amount of hydrogen in the fuel is, the higher the difference between HHV and LHV efficiencies gets. If coal is the fuel then the heat from hydrogen content is practically impossible to recover. This is why LHV is commonly used for thermodynamic calculations of coal fired cycles. But when the fuel is the pure hydrogen, then steam content in combustion gases is 100% and HHV is more convenient to use.

The relative difference between η_{LHV} and η_{HHV} is about 20%. Relation between LHV and HHV efficiency is shown in the Table 1.

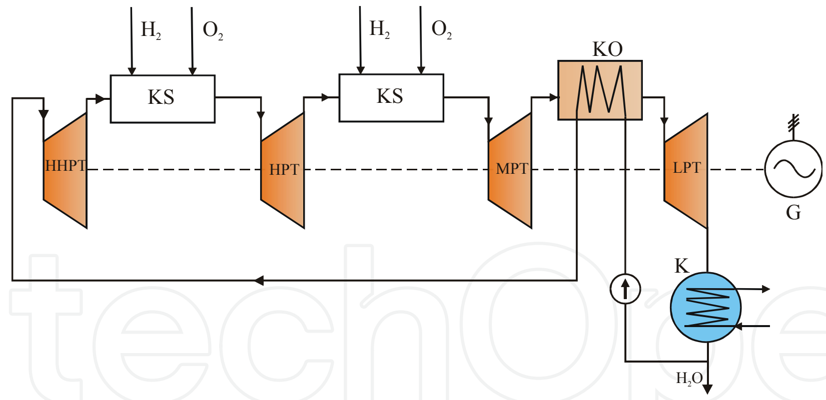

Fig. 16. TOSHIBA' cycle diagram

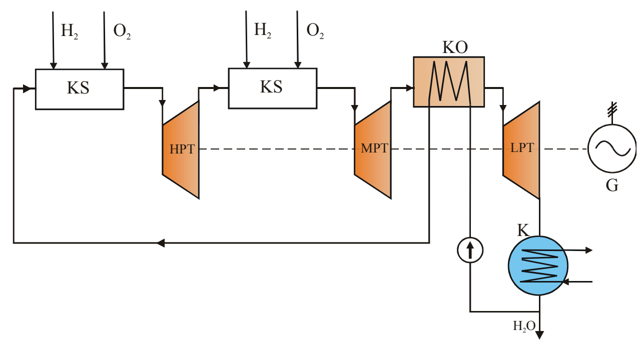

Fig. 17. WESTINGHOUSE cycle diagram

3.2.2 Toshiba cycle

Toshiba Co. proposed the cycle as the company's contribution to the WE-NET project Funatsu et al. (1999). The concept is based on standard Rankine cycle. The regeneration heat exchanger replaced the boiler of the Rankine cycle. Reheating is carried out by two combustors with H_2-O_2 combustion in a steam flow. An implementation of four turbine parts and two combustors is characteristic for the TOSHIBA cycle. The TOSHIBA cycle thermal diagram is shown on the Fig. 16.

3.2.3 Westinghouse cycle

Westinghouse Electric Co. proposed the cycle as the company's contribution to the WE-NET project Bannister et al. (1998). The concept is based on standard Rankine cycle. The regenerative heat exchanger and one H_2-O_2 combustor replace the boiler. Re-heat is performed via one H_2-O_2 combustor. The WESTINGHOUSE cycle consists of two H_2-O_2 combustors and three turbine parts. The WESTINGHOUSE cycle thermal diagram is shown in the Fig. 17.

3.2.4 The GRAZ cycle

A combined steam cycle with steam recirculation, proposed by prof. Herbert Jericha from Graz University of Technology (Austria) Jericha (1991) is called the GRAZ cycle. Graz cycle consists of one combustor, three turbine parts, heat recovery steam generator, condensing part and compressor. The GRAZ cycle thermal diagram is shown in the Fig. 18. The GRAZ cycle is an interesting composition of Brayton and Rankine systems. The Brayton cycle is used as

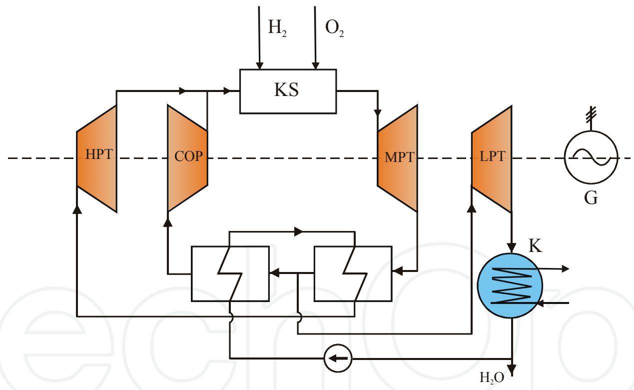

Fig. 18. Flow diagram and calculation results for the GRAZ cycle

a topping cycle (high parameters zone) in semi-closed arrangement, and is coupled to the Rankine cycle operating as the bottoming cycle (low parameters zone).

Basically, the cycle consists of one combustor, three turbine parts, heat recovery steam generator, condensing part and compressor. Thermal efficiency achieved in the cycle amounts to 61% HHV (version without cooling). A flow diagram and basic flow parameters of the GRAZ cycle are shown in Fig. 18.

As in other cycle concepts of this type, the single working medium (steam) is used in both topping and bottoming cycle. Replacement of an external firing (as in the Rankine steam cycle) by a direct firing (similar to gas turbines or reciprocating engines) is also a characteristic feature. The combustion takes place inside a cooling steam flow, which reduces combustion temperature to 1,700°C. Combustion process of hydrogen-oxygen mixture was assumed to be stoichiometric.

An original concept, which distinguishes the GRAZ cycle from others, was a bleed of partially cooled working fluid from the Brayton cycle (point 8 of flow diagram, Fig. 18) and its utilisation as working fluid in the Rankine cycle (points 8, 9, 12, Fig. 18). Thus, this cycle is sometimes called the topping-extraction cycle. Efficiency benefit in this case comes from substantial decrease of compression load in the Brayton cycle, because large share of working fluid (more than a half) reaches high pressure in a feed water pump (points 12, 10: Fig. 18), instead of being compressed in a compressor.

Nominal conditions

Analysis of the GRAZ cycle under nominal conditions was conducted for the following values of parameters:

- Compressor stages group internal efficiency: 90%
- Turbine stages group internal efficiency: 90%
- Combustor efficiency: 99%
- Heat exchanger pressure loss: 4.3%
- Combustor pressure loss: 5%
- Pump efficiency: 90%
- Electric generator efficiency: 99%
- Cycle overall mechanical efficiency: 99%

- Overall power output: 500 MW_e
- Temperature after combustor: 1,700°C
- Condenser pressure: 0.005 MPa
- Condensate temperature: 33°C.

Working medium parameters were calculated based on the NIST/ASME steam property tables.

The GRAZ cycle performance under nominal conditions was analysed basing on maximal overall thermal efficiency criteria (minimum 60% HHV for the 500 MW_e class unit).

It was possible to increase the GRAZ cycle efficiency by adding a compressor inter-stage cooling and recuperation (high-temperature regeneration) to the basic configuration. Nevertheless, it should be pointed out that implementing the GRAZ cycle would require introducing, in a different range, cooling systems of the hottest elements, including turbine blades. Usually implementation of such cooling radically decreases overall cycle efficiency, so it is especially important that the cycle has certain efficiency "reserves" (above 60% HHV).

During this research, a modified version of the GRAZ cycle was chosen. The GRAZ cycle was additionally equipped with the following items:

- An inter-stage cooling system by condensate injection in the steam compressor
- Additional heat exchanger – regenerator – to heat steam flow at the inlet to the combustor
- "Classic" two-stage regenerative heating system with a deaerator.

In order to analyse cycle performance under changed conditions, a simple version of the system was chosen (Fig. 18). This way, we can concentrate on basic issues. Operation of cooling system and additional regeneration system under changed conditions can be taken into account in the next step.

Off-Design Analysis

Part-load analysis was an important issue and it should be taken into account when designing and defining operational characteristics. Results of the cycle behaviour analysis under part-load conditions should support defining the cycle structure and its nominal parameters, as well as design solutions and characteristics of a given subsystem. It could happen that calculation results based on nominal conditions analysis (e.g. by use of maximum efficiency criteria) are not useful from the operating point of view. Some specific working conditions of the cycle performance could be assumed, making proper operation possible only in a very narrow range of parameters, different from nominal ones, making the cycle not adaptable to power output changes. Thus, in the extreme case, starting up the cycle would practically not possible at all. It should be stressed that no complex analysis of the GRAZ cycle part-load operation has been published so far.

Part-load operation characteristic research can be reduced mainly to conditions of co-operation between turbomachinery, the heat exchange element and other equipment. The regenerative heat exchangers included in the GRAZ cycle provide a kind of link between low- and high-pressure part of the cycle. A specific feature of this study was the existence of many bonds and limits. Bonds were defined mainly by the cycle configuration and properties of devices, together with their characteristics. Limits usually result from boundary values of operating parameters. Thus, studying conditions of co-operation of the cycle can be reduced to description and analysis of all possible operational conditions for which bonds and limits were fulfilled.

Basic off-design analysis of the GRAZ cycle developed here was limited only to a single-shaft version with constant rotational speed. So it was necessary to stress that a condensing turbine module (LPT) was formally isolated, though according to the concept of dividing the GRAZ cycle into low-pressure and high-pressure part, the LPT module was assigned to the low-pressure part (LP). Isolating the LPT module was done for two reasons:

1. The analysed GRAZ cycle concept was a single-shaft variant, so the LPT turbine was placed in sequence power generation in mechanical sense, and was one of elements when summing up mechanical power generated on shaft-ends of particular turbine stage group.
2. Because of using the same elementary mathematical model (turbine stage group), describing the module's performance.

Mathematical model of the GRAZ cycle was formulated as a very complex and strongly non-linear algebraic equation set. Due to the complex structure of the GRAZ cycle, solving this model was difficult. There were interconnections between main flow streams (collectors, distributors in the flow stream paths), while a split ratio was unknown and needed to be found.

The equation set should be divided in such a way that equation sub-sets would be connected by possibly minimal number of common variables – so called coordination variables. Additionally, equation sub-sets should be coherent with mathematical models of particular devices. In this way it is possible to formulate general modules modelling devices' performance which makes a possibility to study different structures made of typical modules.

Mathematical models of individual elements of the system (modules) were presented by Kiryk (2002) and Miller et al. (2001). In particular, a model of turbine stage group was proposed by Miller et al. (2000).

Mathematical model of a steam compressor requires some additional remarks. Such a compressor is a new and virtually unknown element introduced in the GRAZ cycle. To our knowledge, this type of compressor has not been used until now, and its implementation would require separate design work. For the purpose of mathematical modelling, several assumptions were made, on the basis of the analogy of flow properties of superheated steam and gas, taking into consideration the possibility of applying several design ideas known from air compressors. Fig. 19 shows compressor characteristics used in the research.

In case of a compressor operating with constant, rated rotational speed – and simulating GRAZ cycle operation – the compressor characteristic is needed only to find out, whether operating conditions being examined (reduced flow and compression) are attainable by changing pitch of blades and vanes and how it will reduce the compressor efficiency.

It seems that under the circumstances assuming compressor characteristic of Fig. 19 is acceptable.

Performance static characterisitcs

In order to solve the mathematical model of the GRAZ cycle, a specific sequence-iteration program was developed based on modelling experience for "classic" condensing turbine sets.

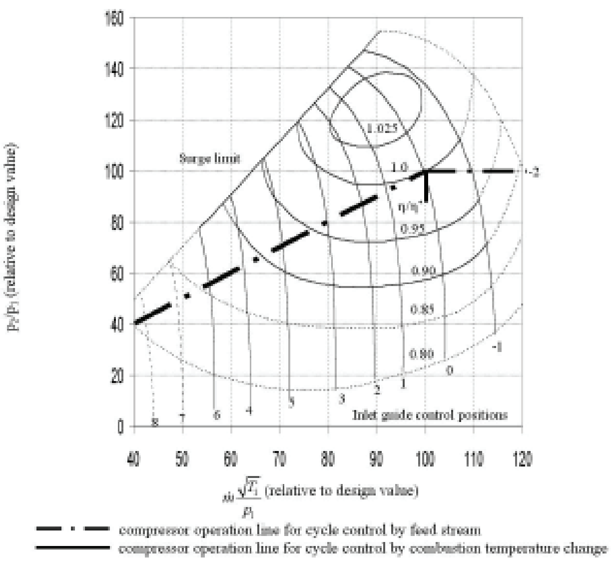

Fig. 19. Compressor characteristic: variable inlet vane angle

The GRAZ cycle power output was changed affecting main circulating mass flow (feed steam flow change) or changing temperatures after combustor (combustion temperature change), or both (combined method). The first case could be treated as a quantitative control method, the second one as a qualitative one and the third as a combined quantitative and qualitative.

Control was possible on the working medium mass flow side (by compressor outlet mass flow) and by changing flow of hydrogen-oxygen mixture supplied to combustors (fuel mass flow). When changing the main circulating mass flow, pressures change accordingly. To follow these pressure changes, the cycle should be equipped with two valves – at the inlets of HHPT and LPT turbines, respectively. There is a kind of analogy here with the accumulation operating mode. The influence of this control concept on the GRAZ cycle performance was simulated using the programme under different part-load and overload conditions. Calculations, if possible, were performed within the range 20–120% of the nominal point.

When performing calculations, the following constraints and limits were taken into consideration:

1. Compressor characteristics working range for the variable inlet vane angle.
2. Maximal temperature of the cycle (combustor outlet temperature 1973 K).
3. Minimal temperature difference in the heat exchanger, to ensure proper heat exchange conditions.
4. Stoichiometric ratio of fuel and oxygen, to ensure stability of the combustion process.

Input data for these calculations were:

1. Compressor outlet mass flow.
2. Combustor outlet temperature.
3. Condenser cooling conditions.

Apart from sequential calculations, iteration loops were required to define operating conditions of heat exchangers (EX1, EX2) and operating conditions between the compressor (COMP) and the high-high-pressure turbine (HHPT).

Combustion temperature change characteristics show that it was possible to operate only within very limited range of changes ($P/P_0 \subset (0.65, 1.0)$). Change of combustion temperature with constant feed steam flow causes almost-linear power output change.

Feed steam flow change with constant combustion temperature was accompanied by virtually linear change of power output, as well as linear pressure change in the cycle. In this case, control was possible down to 37% of P/P_0 and in overload conditions. Overall thermal efficiency varies only slightly within a wide range of the power output values (it was almost constant). Stable overall thermal efficiency is a very important characteristic of the GRAZ cycle. The overall thermal efficiency change is higher when control is exercised by feed steam flow change.

Combustion temperature change has a smaller impact on changes of the overall thermal efficiency. However, it could be applied within very limited operational range of the cycle performance. Hence a combined method could be considered – the main control would be done by changing feed steam flow together with changing combustion temperature for compensation regulation.

Cooling system concept

Possible solutions of turbine cooling systems for the GRAZ cycle are intensively researched by Hitachi, Mitsubishi, Toshiba and other R&D centres Desideri et al. (2001). According to Hitachi Kizuka et al. (1999) the efficiency of the cycle without cooling system is 61.3% HHV, for the cycle with closed water-steam cooling system is 60.1% HHV, for the cycle with closed steam cooling system is 58.7% HHV and for the cycle with open steam cooling system is 54.7% HHV. Cooling system implementation in those cases has decreased the HHV efficiency respectively about 1.2, 2.6 and 6.4 percentage point. Negative impact of cooling implementation is especially big for the open cooling system.

The Mitsubishi Co. has assessed the efficiency of the cycle with the cooling system as 61.8% HHV Sagae et al. (1998) which was then corrected to 60.8% HHV Mouri (1999) and the Toshiba Co. has set this value to 60.1% HHV Okamura et al. (1999). Estimates, which have been recently presented, indicate a slightly higher numbers 61.1–61.8% HHV. Some technology solutions of turbine blades' cooling and results of their analysis were published in Mouri & Taniguchi (1998); Okamura et al. (1999); Sagae et al. (1998).

Summary

Specific calculation programme has been developed for the GRAZ cycle to define static characteristics and part-load analysis under different off-design working conditions. This analysis was done for the single-shaft version with constant rotational speed. It was found that GRAZ cycle has good operational properties for part-load, stable efficiency and acceptable properties for overload. Results obtained for the GRAZ cycle basic off-design performance characteristics seem to be, in our opinion, the first results published in this field.

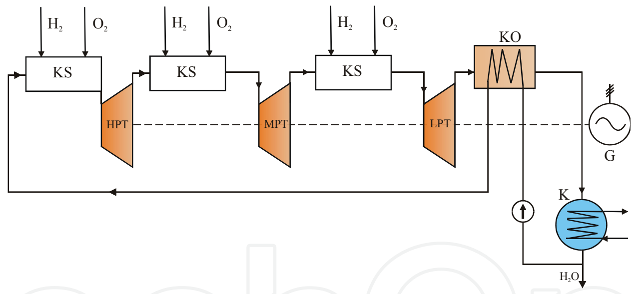

Fig. 20. The MNRC cycle diagram.

3.2.5 Modified New Rankine Cycle – MNRC

The MNRC is proposed by the authors after reconciliation of the abovementioned solutions. The main features increasing the cycle's efficiency are single energy input points (exchanger and one H_2-O_2 combustor) and two-stage re-heat (two H_2-O_2 combustors).

The MNRC cycle is a new concept developed as a result of analysis of previously developed solutions. Contrary to other cycles, in the MNRC solution the heat recovery steam generator (HRSG) is located downstream from the last turbine stage group, within the low-pressure part. This concept has been already mentioned in elaboration Bannister et al. (1997). Arrangement of the cycle allows adding another reheat stage before the low-pressure turbine stage group. As a result a very high thermal efficiency of the cycle is achieved. Flow diagram of the MNRC cycle is shown in Fig. 20. Pressure split between high-pressure and mid-pressure turbine stage groups was optimised according to maximum overall HHV thermal efficiency. Working medium parameters were calculated using the NIST/ASME steam property tables.

The MNRC cycle is relatively insensitive to variation of main parameters, in particular it allows to reduce combustion temperature from 1,700°C to 1,300°C, while maintaining the required 60% HHV overall thermal efficiency.

Static characteristics

Part-load analysis is an important issue for any type of cycle including the HFCTC and it should be taken into account during designing and defining operating characteristics. Results of the cycle's behaviour analysis in part-load conditions should support defining of the cycle's structure and its nominal parameters as well as design solutions and characteristics of components. It could happen that calculation results based on nominal conditions analysis (e.g. by use of maximum efficiency criteria) are not useful from the operating point of view. Some specific working conditions of the cycle performance could be assumed, making proper operation possible only in a very narrow range of parameters, different from nominal ones, making the cycle not adaptable to power output changes. Thus, in the extreme case, starting up the cycle would practically not possible at all. It is to be stressed that no complex analysis on the HFCTC's part-load operation has been published so far.

Part-load operation characteristic research of the HFCTC can be reduced mainly to study conditions of co-operation among rotary machines and the heat exchange element (it means the HRSG or regenerative heat exchangers set), which is a link between low- and high-pressure parts of the cycle, and other equipment. A characteristic feature of this study is existence of many bonds and limits. Mainly the cycle's configuration and properties of devices together with their characteristics define bonds. Limits arise usually from boundary values of working parameters. Study on conditions of co-operation of the HFCTC can be reduced then to description and analysis of all possible operational stages in which bonds and limits are fulfilled.

Mathematical models of the HFCTC are formulated as very complex and strongly non-linear equation set. In order to solve effectively the MNRC cycle's model a specific sequence-iteration program was developed based on modelling experience for "classic" condensing turbosets.

Control over main circulating mass flow or adjustments of temperatures after combustors (combustion temperatures) or both (a combined method) may be used to change the MNRC cycle's power output. The first case could be treated as a quantitative control method, the second one as a qualitative and the third as a combined quantitative and qualitative. These control methods could be practically be implemented without any valves operating within the main circulating mass flow because of its high temperature. Control is possible on the feed water side (by feed water pump) and by change of hydrogen-oxygen mixture supplied to combustors. When changing the main circulating mass flow, pressures change accordingly. There is a kind of an analogy here to accumulation work mode. An influence of this control concept onto the MNRC cycle's performance was simulated using the programme in different part-load and over-load conditions.

As it is shown, a change of the main circulating mass flow at constant (nominal) combustion temperatures, is accompanied with a practically linear change of an power output (both overall and on individual turbine stage groups) as well as linear pressure change in the cycle. Internal efficiencies of the HPT and MPT are constant while the internal efficiency of the LPT varies. In this area the HFCTC displays behaviour similar to the "classic" steam cycle. However the cycle's overall thermal efficiency behaves differently. It varies only slightly in a wide range of the power outputs (it is almost constant). A stable overall thermal efficiency is very important characteristic of the HFCTC.

Changes of combustion temperatures, with the main circulating mass flow set constant, also result with almost-linear changes of the power output (both overall and on individual turbine stage groups). The overall thermal efficiency change is higher than in case of flow-control mode. A combined method is possible to consider then – the main control by the main circulating mass flow with additional combustion temperature control for larger power output increments.

Cooling system concept

The high-pressure turbine should be primarily analysed when considering engineering feasibility of the MNRC cycle. The middle-pressure turbine's working conditions are close (slightly lower) to the high-pressure turbine in the GRAZ cycle that had been already analysed in the WE-NET Program working-task frame.

An identification of possible cooling systems for the MNRC cycle, and their influence on the efficiency, requires defining dimensions – specifically those of turbines – in order to evaluate a surface that would need to be cooled down. An example of blading design for the high-pressure turbine (HPT on Fig. 20) is proposed. While this solution should not be considered final or even optimal, it still could be a good basis for assessments of HPT turbine design possibilities. The HPT turbine should be a high-speed machine (7,000–12,000 rpm), with 5–10 stages, installed in one-, two- (most probably) or three-casing mount. It seems that it should be a very compact machine of blading's diameter below 1.0 m and length 1.5–2.0 cm.

It is necessary to underline that high inlet parameters of the working medium allow to significantly increase the turbine stages' load – isentropic enthalpy drop of 160–230 kJ/kg (compared to 40–80 kJ/kg for conventional turbine stages) – while keeping Mach numbers at reasonable levels of 0.4–0.5. Such conditions would result with power of 12–17 MW per stage. An interesting feature is that as opposed to traditional turbines, there is no radical temperature drop of the working medium along the HPT turbine flow part, so cooling is required on the whole length of the HPT turbine. Blade channel divergence is small (inlet/outlet blades' height ratio is 3.5–4).

A concept of the cooling system for the MNRC-HPT turbine was based on results obtained for the GRAZ cycle by Hitachi Ltd. Kizuka et al. (1999); Mouri (1999). Results were transformed to higher parameters and smaller dimensions of the MNRC-HPT turbine flow part.

Following assumptions were made when evaluating required output of the cooling system for the first turbine stage:

- The highest accepted temperature of blade metal is 1,000°C,
- Temperature drop on TBC (thermal barrier coating) is 400°C.

an average heat flow on TBC's border is 2 MW/m^2, initial temperature of medium used for cooling is 250–300°C. Next stages require adequately lower cooling capacity. Total demand for cooling medium is about 12% of the working flow. Taking into account results achieved by Hitachi, Mitsubishi and Toshiba Kizuka et al. (1999); Mouri (1999); Okamura et al. (1999); Sugishita et al. (1996); Uematsu et al. (1998) it seems realistic to achieve similar parameters in a short-term run. Mass and energy balance of the MNRC cycle with cooling system was defined according to the abovementioned evaluation. It was assumed that, similar to other concepts, cooling is carried out with steam obtained from condensate in the regenerative heat exchanger (Heat Regenerative Steam Generator – HRSG). Cooling steam streams have different pressure levels adequate to individual demands of particular turbine stage groups. As it turns out, there are certain limits of balanced cooling steam amounts, it means 12% – HPT, 13% – IPT and 15% – LPT. When larger flows are needed, water injection into high-pressure combustor is necessary, additionally affecting overall cycle efficiency. A closed counter-current cooling system was not investigated here because of difficulties in obtaining required hermetic conditions of the HPT turbine's cooling system and resulting big pressure differences. Therefore, depending on a cooling system variant, its presence has an important impact on the cycle's efficiency drop. For example:

- Cooling with 300°C steam and steam supply to the high-pressure combustor with the following cooling steam ratios HPT – 12%, IPT – 13%, LPT – 15% (those are the highest amounts of cooling steam that are possible to generate) and the cycle's efficiency is 60.9% HHV ($\Delta = 5.5$ percentage points)
- Cooling with 300°C steam and water injection (70°C) into the high-pressure combustor with the following cooling steam ratios HPT – 13%, IPT – 20%, LPT – 26% and the cycle's efficiency is 57.9% HHV ($\Delta = 8.5$ percentage points),
- Cooling with 300°C steam and with the following cooling steam ratios HPT – 13%, IPT – 20%, the low-pressure combustor is switched off and the cycle's efficiency is 60.4% HHV (above 60%HHV requirement),

- Cooling with 300°C steam with the following cooling steam ratios HPT – 13%, cooling stream IPT – 20% is reverted in counter-flow to the intermediate-pressure combustor, the low-pressure combustor is switched off and the cycle's efficiency is 61.9% HHV which is a very promising result showing a direction of the future research.

Technology and design possibilities of MNRC cycle implementation (concerning turbines design) seem to be fully realistic, especially bearing in mind achievements for the GRAZ cycle, and at the same time enable achieving very high efficiency of the cycle that ensures its competitiveness comparing to other concepts.

3.2.6 Comparison of HFCT cycles in nominal conditions

Comparison and cycles' evaluation could be done from different points of view. Basic criterion here is maximal overall thermal efficiency, at least 60% HHV for a 500 MW class unit. Main performance parameters of analysed cycles are shown in Table 2.

Parameter	Graz	Toshiba	Westinghouse	MNRC
$p_{\max}$, bar	350	380	250	250
$t_{\max}$, °C	1,700	1,700	1,700 / 1,600	1,700
Inner Power, MW	513	513	513	513
η_{LHV}, %	70.8	71.2	74.0 / 72.8	79.0
η_{HHV}, %	59.5	59.8	62.2 / 61.2	66.4
Specific Power, kJ/kg	2,202	3,331	3,489	4,706
Electrical Power, MW	500	500	500	500
$\eta_{el,LHV}$, %	69.0	69.4	72.2 / 71.0	77.0
$\eta_{el,HHV}$, %	58.0	58.3	60.6 / 59.7	64.7
Temperature at the most thermally loaded element, °C	1,700	1,700	1,700 / 1,600	1,700
Pressure at the most thermally loaded element, bar	50	73	250	250
Pressure at the most pressure loaded element, bar	350	343	277	277
Temperature at the most pressure loaded element, °C	650	876	517	463
Heat exchanged / HRSG heat load, MW	315	329	256	165

Table 2. Main performance parameters of the HFCT Cycles

As it is shown, efficiencies of GRAZ and TOSHIBA cycles are practically on the limit of the WE-NET Program requirements. The WESTINGHOUSE cycle fulfils these requirements with some margin and the MNRC cycle is far above needed values. It is possible to increase the GRAZ cycle's efficiency by adding compressor inter-stage cooling and recuperation (high-temperature regeneration) to the basic configuration.

Nevertheless, it should to be pointed out that implementation of the HFCT Cycles would require introducing cooling systems of the hottest elements, including turbine blades cooling, operating in a new range of parameters. Usually implementation of such cooling radically decreases overall cycle's efficiency so it is especially important that the proposed concept of the HFCTC has a certain "reserves" of its efficiency (above 60% HHV). It should be said that the MNRCycle has the highest chances to be practically realised due to achieved efficiency. A modified version of the GRAZ cycle was chosen as a leading concept for the Japanese WE-NET programme. The GRAZ cycle was additionally equipped with the following items:

- An inter-stage cooling system by condensate injection in the steam compressor
- Additional heat exchanger – regenerator – for heating of a steam flow at the inlet to the combustor
- "Classic" two-stage regenerative heating system of combustor outlet flow with a deareator.

These versions of the GRAZ cycle were named: Inter-cooled Topping Extraction Cycle and Inter-cooled Topping Recuperation Cycle. Evaluation of efficiencies of these GRAZ cycle's versions (without low-pressure regeneration) has resulted with HHV efficiencies 60.13% and 61.45% respectively Desideri et al. (2001); Uematsu et al. (1998).

All cycles discussed achieve very high specific power (related to a maximal mass flow in the cycle). Those values are much higher (even by an order of magnitude) than ones achieved by contemporary heavy-duty gas turbines (450–500 kJ/kg), GTCC units (600–700 kJ/kg) or steam turbo-sets (1,200–1,400 kJ/kg). It would be possible then to build extremely compact power units with minimal usage of structural materials.

When evaluating technical feasibility of analysed cycles the first step should concern nodes (elements) working at the highest levels of steam parameters. A bottom part of the Table 2 refers just to the most loaded elements' analysis. As it is shown, for the WESTINGHOUSE and MNRC cycles (the most efficient cycles) the highest working medium's temperatures occur together with the highest pressures. This refers to high-pressure combustors and high-pressure turbines and these elements seem to be the "critical" ones. Bearing that in mind, the high-pressure combustor was omitted in the TOSHIBA cycle what resulted, however, with setting very high pressure of the cycle, adding high-temperature heat exchangers and lowering the cycle's efficiency. An advantage of the GRAZ cycle, without any doubts, is relatively low pressure in the highest temperature range – as in a "classic" gas turbine unit. On the other hand big disadvantages are:

- The steam compressor, which is a completely unknown element so far
- Very complex system of heat exchangers and
- Generally high complexity of the cycle.

Summarising the above and considering performance in nominal conditions, the most interesting proposals are the GRAZ cycle and the MNRC.

Cycle	η_{HHV}, %	Deviation from reference value ($\eta_{HHV} = 60\%$), %
Graz cycle	59.466	-0.534
Toshiba cycle	59.780	-0.220
Westinghouse cycle	61.125	1.125
Modified New Rankine cycle	65.098	5.098

Table 3. Comparison of HHV-efficiencies for different cycles' configuration

3.3 Summary

All the HFCTC cycles with direct firing which were studied within this research project (500 MW class units) have shown very high thermal efficiency (minimum 60% HHV, 71% LHV) which is far above the performance of the currently used cycles.The possibility to achieve the efficiency at least 10 percentage points higher than the efficiency of the most efficient contemporary power units (which is about 20% more) has been fully confirmed. Similarly all the HFCTC cycles have very high specific power (2,200–4,700 kJ/kg), which is many times higher (up to an order of magnitude) than performance of the contemporary gas or steam turbines or combined cycles. Thus the new concepts could allow to construct extremely compact power units with minimal usage of construction materials. Considering, additionally, a fact that the HFCTC cycles almost totally eliminate CO_2 and NO_x emissions, this solution can be recognised as an interesting alternative for the future power technology development trend when compared to conventional power technologies.

The concept favoured by WE-NET Program at present – the GRAZ cycle – has certain advantages. One of them is relatively low pressure level in high temperature zones: 5 MPa at 1,700°C. However, the GRAZ cycle has also serious disadvantages, like usage of a steam compressor which is a completely unknown element so far, very complex system of heat exchangers and generally high complexity of the cycle.

The MNRC cycle – proposed in the this research – could be attractive in this context, especially because its efficiency is much higher (66% HHV, 77% LHV) and its general structure is simple. A comparative study has shown that it seems fully realistic to implement an effective cooling system for this case.

A sensitivity study undertaken for the MNRC cycle has shown its big "resistance" to any changes of operating parameters. For example, it is possible to reduce working temperature even to 1,300°C while adhering to 60% HHV efficiency requirement.

Specific calculation programmes have been developed for both the GRAZ cycle and the MNRC cycle to define static characteristics and carry out part-load operation analysis in different off-design working conditions. The GRAZ cycle's operating features have proved to be "acceptable", while the MNRC cycle has displayed high operational and control flexibility combined with very stable thermal efficiency at the same time. Taking the above into account, it seems that the MNRC cycle should be more thoroughly investigated in the future.

It is to be stressed that, in opinion of authors, obtained results of the HFCTC's off-design performance characteristics seem to be the first results published in this field.

Further research should focus on general modification of the HFCTC in order to utilise natural gas as a fuel with CO_2 separation. This research direction seems to be natural due to some delay in research on industrial-scale hydrogen production technologies (including hydrogen-fuelled power sector needs). In more detailed scope – it would be correct to continue research on the GRAZ cycle as well, in order to improve its operational flexibility (multi-shaft cycle concept, etc.).

Hydrogen combustion power systems could be an important alternative to the conventional fossil fuel combustion power systems in the future. Hydrogen could be available in large quantities and its utilisation in power generation systems will have minimal influence on the environment.

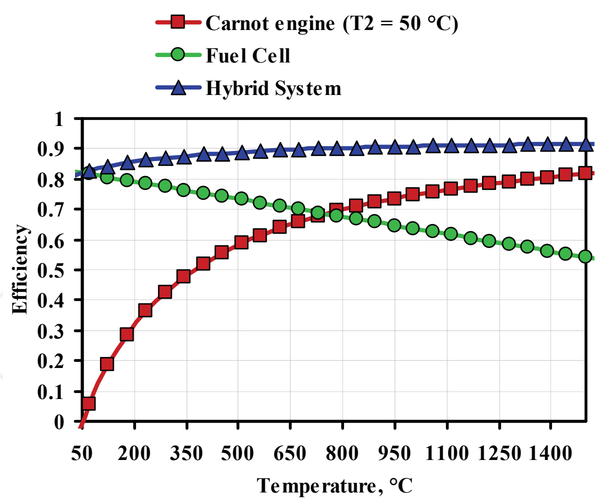

Fig. 21. Theoretical efficiency of a fuel cell – gas turbine system Milewski et al. (2011)

However development of the hydrogen utilisation technologies for energy generation requires R&D work to be undertaken in many fields. Results of the research completed so far show that power generation systems with hydrogen combustion can become competitive in the power technologies market when compared to today's state-of-the-art conventional systems. Nevertheless, high costs of hydrogen production cause make the cycle overall thermal efficiency over 60% (HHV) an essential requirement for this technology.

4. Fuel cell – gas turbine hybrid systems

The main advantage of combining a fuel cell with a gas turbine is that one can create a binary system which can potentially achieve ultra-high efficiencies (see Fig. 21). This task is fulfilled through the other system using the fuel cell exhaust heat.

A typical Fuel Cell-Gas Turbine Hybrid System (FC-GT) consists of the following elements:

- Air Compressor
- Fuel Compressor
- Gas Turbine
- Air Heater
- Fuel Heater
- Fuel Cell.

The Fuel Cell is not the only power source in the SOFC-GT hybrid system (additional power is produced by the gas turbine unit). FC-GT hybrid system efficiency is defined by the following relation:

$$\eta_{HS} = \frac{P_{FC} + P_T - P_C - P_{fuel}}{\dot{m}_{fuel} \cdot LHV} \tag{11}$$

FC-GT hybrid systems can be classified according to their fuel cell module operating pressure. Systems in which the pressure of leaving the fuel cell is comparable to atmospheric pressure

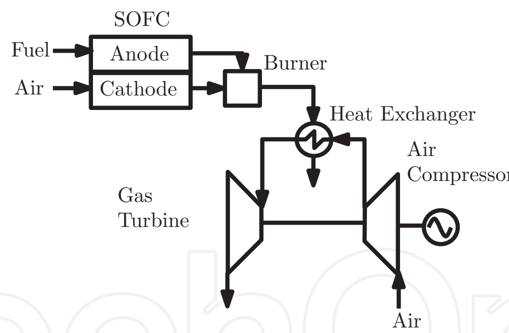

Fig. 22. Diagram of an atmospheric SOFC with a bottom cycle based on an air turbine.

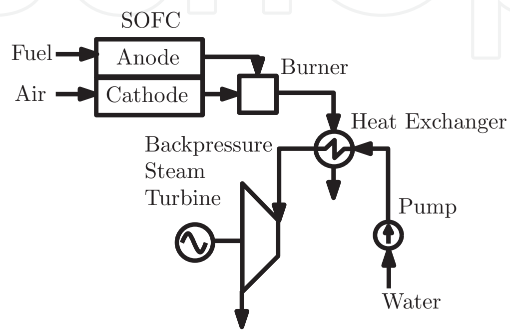

Fig. 23. Diagram of an atmospheric SOFC with bottoming cycle based on an open steam cycle

are called atmospheric FC systems. The second group consists of systems in which the pressure of the exhaust gas leaving the fuel cell is significantly higher than atmospheric; those kinds of systems are called systems with pressurised FC.

This classification determines the location of the fuel cell in a power system. Generally, a fuel cell can perform function similar to a classic combustion chamber, i.e. oxidise fuel supplied to the system, which results in relatively large quantities of electricity being taken from the fuel cell itself. A combustion chamber works with lower amounts of fuel and does not require a large excess of air in order to reduce the temperature of gas getting to the turbine.

4.1 SOFC-GT hybrid systems

4.1.1 Atmospheric SOFC-HS

In case of an atmospheric SOFC, increased system efficiency can be achieved only by heat recuperation through a bottoming cycle. In systems containing an atmospheric SOFC, additional equipment is needed to recover part of the exhaust heat. The simplest bottom cycle is based on an air turbine subsystem (composed of an air compressor and air turbine). This solution is shown in Fig. 23. Heat is transferred to the bottoming cycle by adequate heat recuperative heat exchangers. In this case, SOFC flue gas is considered as the upper heat source for the air turbine cycle.

Regenerative heat exchangers can be used both on the air and fuel flows. One limitation in this solution is the air compressor, because the temperature of compressed air is increased. Therefore, the desirability of a regenerative heat exchanger at the flow depends on the temperature of the exhaust leaving the gas turbine. This problem can be solved by introducing an additional combustion chamber, located upstream from the exchangers. This makes the fuel cell operation independent from temperature rise before the heat exchangers.

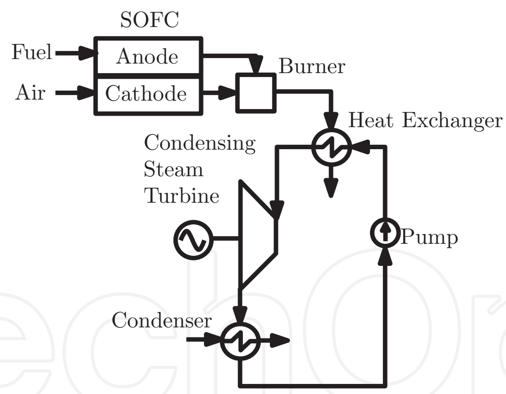

Fig. 24. Diagram of an atmospheric SOFC with bottoming cycle based on a closed steam cycle.

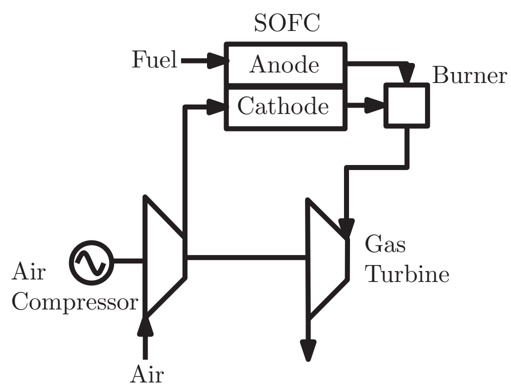

Fig. 25. SOFC-GT hybrid system with the fuel cell placed instead of the combustion chamber.

This system suffers from relatively low efficiency due to the relatively large amount of power consumed by the air compressor. To increase the efficiency of the system, water (steam) can be substituted for compressed air as a working agent. Much less energy is required to increase water pressure than to compress air to the same pressure. Such a system system is illustrated in Fig. 23. The system with a bottoming cycle based on an open steam turbine cycle has better performances than the one based on an air turbine due to the reduced compression work and better heat transfer in the heat recovery steam generator (HRSG).

Steam at a temperature of more than 100°C is discharged from the system. Firstly, the system in this configuration will consume huge amounts of water. Secondly, all the water evaporation heat is lost to the environment. Part of this heat can be recovered using a system working in the Rankine cycle. This solution is presented in Fig. 24 and is comparable to the classic gas turbine combined cycle (GTCC) in which the SOFC is placed instead of a gas turbine.

4.1.2 pressurised SOFC-HS

From the gas turbine system point of view, the pressurised SOFC can be substituted for the combustion chamber.

Simple addition of a gas turbine subsystem to the SOFC raises efficiency to 57%. The power generated by the gas turbine subsystem represents about 30% of the total system power.

The upgrade of the SOFC-based system is the addition of a heat exchanger, which is placed between the fuel cell stack and the air compressor, similarly to upgrade of an open cycle gas

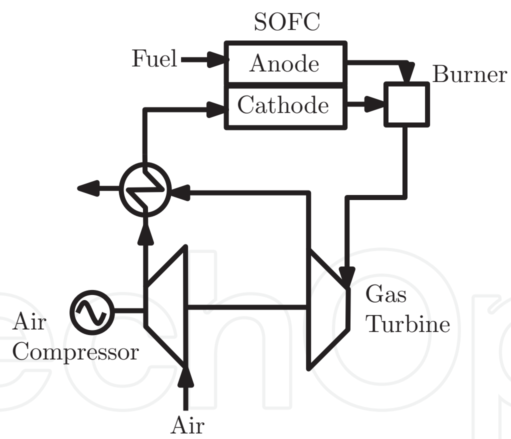

Fig. 26. SOFC-GT hybrid system with a heat exchanger placed upstream from the fuel cell.

Parameter	GT	SOFC	SOFC-GT
Fuel	CH_4	H_2	CH_4
Efficiency, %	20 (27*)	37	66
TIT, °C	1,100	–	1,100
Fuel cell temperature, °C	–	800	800
GT pressure ratio	11	–	9.6
Power given by GT in relation to total system power, %	100	0	22

Table 4. Main parameters of a SOFC-GT Hybrid System.
*with heat recuperation

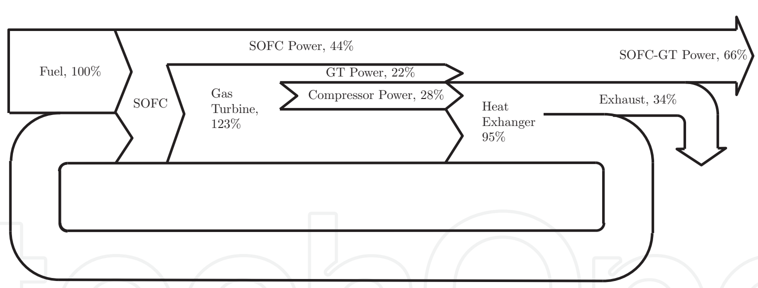

Fig. 27. Sankey's diagram of energy flows of SOFC-GT system.

turbine system (see Fig. **??**). The heat exchanger is fed by the gas turbine outlet stream. An adequate heat exchanger may increase the efficiency by recovering part of the gas turbine exhaust heat. This solution increases efficiency to 66% and decreases gas turbine pressure ratio to 8.2. TIT is still relatively high at 1,100°C.

The addition of a SOFC to a gas turbine system can boost efficiency to 57%. By changing the fuel from hydrogen to methane efficiency is increased to 63% due to reforming reactions which convert thermal energy into the chemical energy of fuel. The addition of a heat exchanger increases the efficiency even further, to 66%.

Energy flow paths in a pressurised SOFC-GT hybrid system are shown in a form of Sankey diagram in Fig. 27.

Selection of the size, design and working parameters of the SOFC is crucial when seeking to obtain a highly-efficient hybrid system. Additionally, both the gas turbine and the air compressor should be designed for operation with SOFC.

4.1.3 Control issues

It should be underlined that in case of a system which contains both a pressurised SOFC and a gas turbine, varying the amount of fuel injected is not the only way to control the power output. Fuel cell voltage and current are quite dependent on the variable rotational speed of the compressor-turbine unit. This is accompanied by varying system efficiencies. Hence there is a need to formulate an appropriate control concept (control strategy logic) and its approach for technical realisation.

Control strategy is an important element in designing any system of this kind and it constituted a significant part of the modelling works done. Off-design (part-load) analysis is an important issue for any type of system involving an SOFC-GT hybrid solution and should be taken into account when designing and defining the operational characteristics. A proper off-design performance map underscores control strategy design. Results drawn from system behaviour analysis under part-load conditions should aid in defining the system structure and its nominal parameters, as well as the design solution and characteristics of a given subsystem.

Part-load operation characteristics research regarding SOFC-HS can be reduced mainly to study of the conditions of co-operation among the SOFC, turbomachinery and other equipment. A specific feature of this study is the existence of many bonds and limits. Bonds are defined mainly by the system configuration and properties of devices that make up the system, together with their characteristics. Limits usually result from boundary values of working parameters. Thus, studying the conditions of co-operation of the SOFC-GT can be reduced to describing and analysing all possible operational conditions. Part-load and overload performance characteristics of SOFC-GT were calculated and analysed to show control possibilities of the cycle.

Off-design (part-load) analysis is an important issue for any type of system including an SOFC-GT and should be taken into account during designing and defining operational characteristic. Results of the system behaviour analysis under part-load conditions should aid in defining the system structure and its nominal parameters, as well as the design solution and characteristics of a given subsystem.

Implementation of a control system depends on its structure, which is subject to regulatory processes at work in varied conditions. Two sources of electricity may exist in hybrid systems. Usually one of these sources is to provide the main energy flux while the other plays auxiliary role. There is therefore a need to identify possible structures for systems analysis of power distribution to the various subsystems. A multi-stage procedure is used to determine the control strategy of SOFC-GT hybrid systems, as follows:

1. Internal constraints of the system are determined based on the chosen structure and parameters of the nominal SOFC-GT systems (i.e. mechanical, flow and electrical connections between the elements and characteristics of specific elements).
2. Mathematical model of the system is built at the level of off-design operation.
3. Based on the model all possible operating conditions are determined. A database of those steady-state points of the system is created.

4. Based on the database an adequate searching algorithm is used to find the most convenient operation line of control strategy for chosen criteria (e.g. highest system efficiency).
5. The control strategy is realised by the first level of a multi-layer control system.

The control strategy of the system should allow to quickly and accurately follow the load profile, while maintaining good system efficiency. Safe operation of the system is an obvious requirement. Based on mathematical modelling and numerical simulations, the control strategy for an SOFC-GT is presented.

An SOFC-GT hybrid system has three degrees of freedom, and with regard to m-fan up to four; which means that at least three free parameters can vary independently within certain ranges. Any combination of them should define a certain state of the system. An SOFC-GT hybrid system has only one control variable, which is generated power, and three manipulated variables, i.e. the current cell module, the flow of fuel and the electric generator load. The dependencies and relationships occurring in the hybrid system can be divided into four groups:

1. Correlations defined by a mechanical scheme of the system (mechanical linkage of turbines, compressors, generators).
2. Correlations defined by the flow chart, (i.e. flow relationship between the compressor, turbine, fuel cell, heat exchangers and other components of the system and the order of the working flow direction through system elements).
3. Depending on specific characteristics of turbines and compressors (average parameters at the inlet and outlet of the rotating machine are closely linked through its characteristics).
4. Depending on the characteristics associated with other system elements.
5. Depending on specific electrical connections inside the system.

An SOFC-GT hybrid system can be controlled using the following parameters:

- Electric current taken from SOFC stack by external resistance (load)
- Fuel mass flow through a valve
- Rotational speed of the compressor-turbine subsystem by power output of electric generator with adequate power electronic converter.

Ensuring safe operation of the system requires elimination of events (operating conditions) which could damage the system or its components. This represents at the same time a limit on the scope of permissible working conditions (states) of the system. Typical constraints normally are a result of:

- Acceptable working fluid parameters (mainly the highest temperature and pressure)
- Acceptable electrical parameters
- Compressor limits (surge line)
- Critical frequencies of rotating machines
- Acceptable torque values.

Those limitations can be defined arbitrarily, and the values presented below are exemplary only. Based on data calculated for all technically possible conditions, additional limits and bonds were applied on the system efficiency map, and they are as follows:

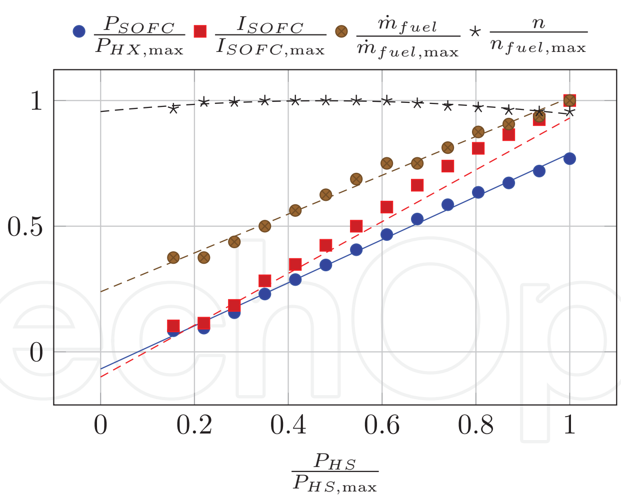

Fig. 28. Relationships between control parameters of a SOFC-GT system. Legend: n – GT shaft rotational speed, I - SOFC stack current, $\dot{m}$ – fuel mass flow, P – overall power output

- Average solid oxide fuel cell temperature < 1,000°C
- SOFC stack temperature difference < 320°C
- Turbine Inlet Temperature (TIT) < 1,000°C
- Air compressor surge limit curve.

Other conditions reflect the more general requirement to maintain fuel cell stability, fostered by keeping the cell temperature as constant as possible and reducing (limiting) charged current (limited local heat source). The amount of steam at the anode inlet must be monitored to avoid carbon deposition during reforming processes. Deposition takes place when the temperature is too low and/or there is too little steam at the inlet to the stack (low s/c ratio). A reverse flow of gases from the combustion chamber to the anode channels can result in anode coming into contact with oxygen – anode reverse flow can occur with rapid increases in pressure, hence the need to limit the increase in pressure over time. And finally, too low fuel cell voltage can result in unstable fuel cell stack operation.

Taking into account controllable parameters, the control strategy can be based on the following three functional relationships:

$$\begin{aligned} \dot{m}_{fuel} &= f\left(P_{HS}\right) \\ n &= f\left(P_{HS}\right) \\ I_{SOFC} &= f\left(P_{HS}\right) \end{aligned} \tag{12}$$

Choosing the highest possible efficiency of the system, the relevant functional dependencies 12 of all controllable parameters take the form of adequate functions of the system power. The controllable parameters of the system as a function of system power are presented in Fig. 28. The figures were normalized to their maximum values. It is evident that both the current cell stack and the amount of fuel supplied are close to the linear trend. The rotational speed of the gas turbine is relatively constant (+/- 10%), reaching a maximum at 50% system load.

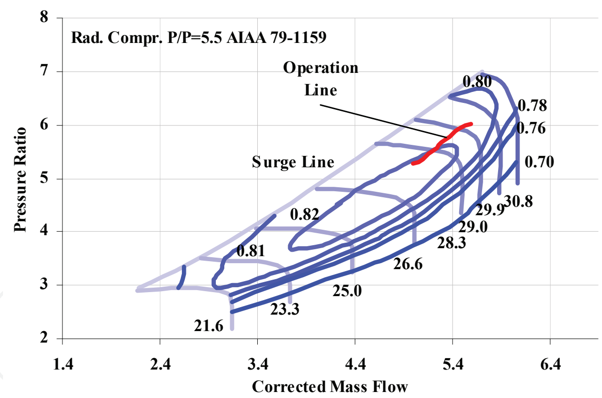

Fig. 29. Operation line of control strategy indicated on air compressor map used during simulations.

Exemplary of the functional relationships (Eq. 12) are as follows:

$$\begin{aligned} \frac{\dot{m}_{fuel}}{\dot{m}_{fuel,\max}} &= 0.7725 \cdot \frac{P_{HS}}{P_{HS,\max}} + 0.2391 \\ \frac{n}{n_{\max}} &= -0.1915 \cdot \left(\frac{P_{HS}}{P_{HS,\max}}\right)^2 + 0.1812 \cdot \frac{P_{HS}}{P_{HS,\max}} + 0.9561 \\ \frac{I_{SOFC}}{I_{SOFC,\max}} &= 1.0304 \cdot \frac{P_{HS}}{P_{HS,\max}} - 0.1001 \end{aligned} \tag{13}$$

It can be seen that the fuel cell stack current should be increased more rapidly than the fuel mass flow. The GT shaft speed should be increased until system power reaches 50% and, thereafter, the shaft speed has to be decreased. The SOFC-GT hybrid system maintains good efficiency even at partial loads. For example, while reducing the system load to about 40% the efficiency is still over 80% of the nominal value.

Normal operation of the system is possible in a wide power range, from approximately 17% of the rated power. The operation line is located preferably on the characteristics of the compressor – far from the surge limit (see Fig. 29).

Determination of reasonable parameters (due to optimised efficiency) of an SOFC-GT hybrid system also defines the necessary parameters for rotating machinery and other system components. Needed mass flow of both compressor and turbine is in the range 5–9 kg/s at a pressure ratio (both compression and expansion) of 5–6.2. The parameters of rotating machinery needed for the SOFC-GT can be achieved by a single-stage radial compressor and a two-stage axial turbine.

4.2 MCFC-GT hybrid systems

A molten carbonate fuel cell-hybrid system consists of the following elements:

- Air compressor
- Fuel compressor

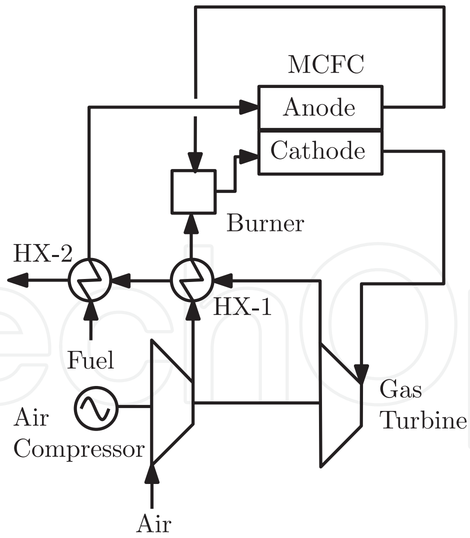

Fig. 30. Pressurised GT-MCFC Hybrid System.

Parameter	Value
Overall Efficiency (LHV), %	58
Re-cycle Factor, %	44
Excess Air Factor	1.68
Average Cell Voltage, mV	659
DC/AC inverter efficiency, %	95
Electric generator efficiency, %	99
Mechanical efficiency of the GT, %	99
Electric motor efficiency, %	95
Turbine Inlet Temperature (TIT), °C	650
Compressor pressure ratio	8.35
Electrolyte material	Li/Na
Matrix thickness, µm	1,000

Table 5. Nominal parameters of a MCFC-HS system.

- Gas turbine
- MCFC module
- Air heater
- Fuel heater
- MCFC module.

Based on mathematical modelling the design point parameters of the system presented in Fig. 30 were estimated. The main system parameters are presented in Table 5.

4.3 Reducing CO_2 emission of gas turbine by implementation of MCFC

The European Union has imposed limits on CO_2 emissions by Member States as a part of its Emission Trading Scheme. This affects fossil fuel power plants to a significant degree as their emissions are governed by the number of emission allowances they receive from the Member State allocation. Excess CO_2 emissions have to be covered by purchasing extra allowances, which is in effect a financial penalty (EUR100/Mg). In contrast, undershooting emission limits enables the operator to sell CO_2 allowances. The selling price of a traded allowance is estimated at EUR20–30/Mg_{CO_2}.

There is a variety of methods available to remove CO_2 from a fossil fuel power plant system Gottlicher & Pruschek (1997). The idea of adopting a molten carbonate fuel cell to reduce CO_2 emissions was developed by Campanari Campanari (2002). An estimation made by Campanari shows that a reduction of 77% in CO_2 emissions can be achieved in a steam turbine power plant.

Fuel cells generate electricity through electrochemical processes Milewski et al. (2011). There are many types of fuel cells, two of them – the Molten Carbonate Fuel Cell (MCFC) and the Solid Oxide Fuel Cell (SOFC) – are high-temperature types. They operate at temperatures ranging from 600–1,000°C. Amorelli et al. Amorelli et al. (2004) described an experimental investigation into the use of molten carbonate fuel cells to capture CO_2 from gas turbine exhaust gases. During experiments performed using a singular cell, an emission reduction of 50% was achieved.

Lusardi et al. Lusardi et al. (2004) investigated the application of a fuel cell system for CO_2 capture from thermal plant exhaust. They found that even without CO_2 separation, the relative emission of carbon dioxide could be reduced below the Kyoto Protocol limit. If a separator is used, emissions could be reduced by 68%.

Use of an MCFC as a carbon dioxide concentrator was investigated by Sugiura et al. Sugiura et al. (2003). In this work the experimental results of CO_2 capture using an MCFC are given. One key conclusion from this work is that the CO_2 removal rate can be obtained by making calculations using electrochemical theory.

Novel methods whereby carbonates were used as an electrochemical pump in carbon dioxide separation from gases were described by Granite et al. Granite & O'Brien (2005).

Based on the review of abovementioned literature, a reduction of at least 50% in CO_2 emissions could be expected.

Hydrogen, natural gas, methanol or biogas may be used as fuels for MCFCs. On the cathode side, a mixture of oxygen and carbon dioxide is required.

An MCFC may work as a carbon dioxide separator/concentrator because the CO_2 is transported from the cathode side to the anode side through molten electrolyte.

The combination of GT unit with MCFC results with a hybrid system (HS) with increased efficiency and decreased carbon dioxide emission. The exhaust flue gas of gas turbine power plant consists mainly of nitrogen, oxygen, steam and carbon dioxide. This mixture can be used as oxidant in the MCFC (cathode feeding). The temperature of the exhaust gas and electric efficiency of GT unit are around 550°C and 35%, respectively. The fuel cells in turn can achieve higher electric efficiency of 50–60%.

Negative ions are transferred through the molten electrolyte. Each ion is composed of one molecule of carbon dioxide, one atom of oxygen and two electrons. This means that an adequate ratio of carbon dioxide to oxygen is 2.75 (mass based) or 2.0 (mole based).

The typical gas turbine flue gas composition is shown in Table 6. The ratio of CO_2 to oxygen is hence 0.25 (mole based) and 0.34 (mass based). This means that flue gas contains an insufficient quantity of oxygen to slow trapping all CO_2.

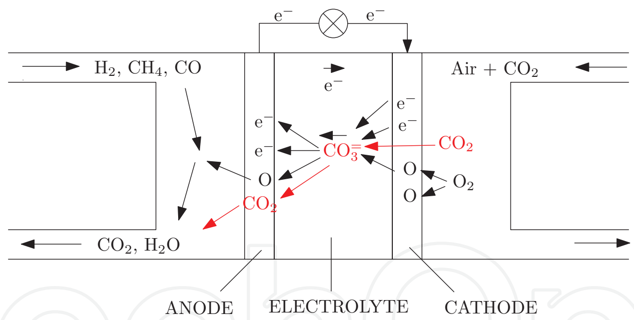

Fig. 31. Working principles of an MCFC.

Name	Value
Air compressor inlet pressure, MPa	0.1
Air compressor inlet temperature, °C	15
Pressure ratio	17.1
Fuel	Natural gas
Fuel mass flow, kg/s	4.0
Turbine inlet temperature, °C	1210
Exhaust gas mass flow, kg/s	213
Turbine outlet temperature, °C	587
GT Power, MW	65
GT Efficiency (LHV), %	33
CO_2 annually emission, Gg/a	250
Relative emission of CO_2, kg/MWh	609
CO_2 mass flow, kg/s	11

Table 6. Nominal parameters of GT Power Plant Granser & Rocca (1996)

The hybrid system (fossil fuel power plant + MCFC) has no obvious objective function of the optimising process, the system performances were estimated based on CO_2 reduction emission factor, which is defined as follows:

$$\eta_{CO_2} = 1 - \frac{m_{CO_2,out}}{m_{CO_2,in}} \tag{14}$$

where: m – mass flow, kg/s; *out* – MCFC outlet cathode stream; *in* – MCFC inlet cathode stream.

A GT unit consists of the following elements:

1. Air compressor,
2. Gas turbine
3. Combustion chamber.

Mathematical model of a GT unit was based on the following assumptions:

- Air compressor isentropic efficiency: 79%
- Gas turbine isentropic efficiency: 88%
- No pressure drops across the combustion chamber.

Component	Mass fraction, %	Mole fraction,
CO_2	5.2	3.4
H_2O	4.1	6.6
O_2	15.3	13.6
N_2	74.0	75.4
Ar	1.4	1.0
CO_2/O_2	0.34	0.25

Table 7. Exhaust gas composition.

A commercial gas turbine unit was selected for analysis Granser & Rocca (1996). Nominal parameters of the GT unit and exhaust gas composition are shown in Tables 6 and 7, respectively.

To create a CO_3^{2-} ion, it is needed to split a half mole of O_2 with one mole of CO_2. Adequate mass and molar ratios of CO_2 to O_2 (for capture all carbon dioxide) are 1.38 and 2, respectively. Data given in Table 7 shows that, theoretically, all CO_2 could be captured.

Two cases of gas turbine power plant with the MCFC were investigated. Case 1 concerns a situation when there the GT cycle stays unchanged. It means that MCFC is added at gas turbine's outlet stream. Case 2 concerns the situation when heat exchangers before combustion chamber are added to the gas turbine unit. These heat exchangers are fed by MCFC exhaust streams. Relatively low CO_2 content in flue gas results with low MCFC efficiency. The MCFC efficiency is about 34% (based on Lower Heating Value, LHV). It seems to be unreasonable to combine a low-efficiency MCFC with a high-efficiency Gas Turbine Combined Cycle unit (with efficiency about 55%) so this case was not investigated.

Proper objective function of the optimisation process is not apparent. The MCFC is installed to capture the CO_2, and from this point of view the quantity of captured CO_2 should be maximised. But from the other point of view, the MCFC uses the same fuel as the gas turbine to produce electricity. Both analysed cases were optimised to obtain maximum system efficiency.

In both cases apart from MCFC, following devices should be installed:

1. CO_2 separator (water condensing unit)
2. Catalytic burner
3. DC/AC converter with efficiency of 95%.

The CO_2 separator is cooled by water. When steam condenses, water is separated from the carbon dioxide stream.

The catalytic burner is fed by pure oxygen to utilise the rest of methane, hydrogen and carbon monoxide. An oxygen production (e.g. extraction from air) requires energy input. The production of one kilogram of oxygen at atmospheric pressure requires from 200 to 300 kJ energy input. The value of 250 kJ was taken into calculations, which decreases the system efficiency depending on the amount of consumed oxygen.

Installation of an MCFC at gas turbine outlet means back pressure drop of about 1%. It decreases the efficiency of the gas turbine from 33% to 32%.

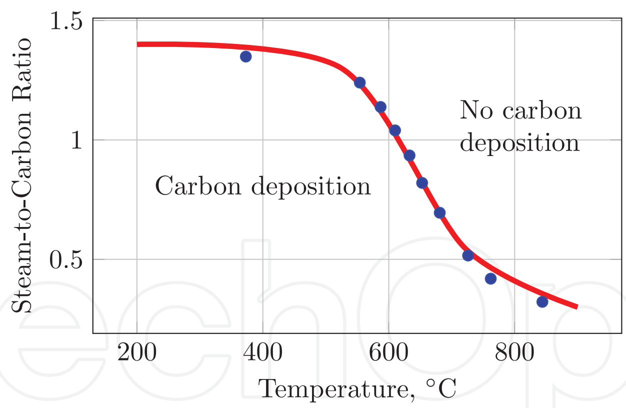

Fig. 32. Minimum temperature and required ratio of steam-to-carbon (s/c) above which no carbon deposition takes place herle et al. (2004)

Name	GT plant	Case 1	Case 2
GT-MCFC power (total power), MW	65	80	77
GT power/total power, %	100	81	82
MCFC power/total power, %	0	19	18
GT-MCFC efficiency (LHV), %	33	33	40
CO_2 emission reduction factor, %	0	73	91
Annual CO_2 emission, Gg/a	250	67	18
Relative CO_2 emission, kg/MWh	609	132	37
MCFC efficiency (LHV), %	–	34	36
GT efficiency (LHV), %	33	32	41
Fuel utilization factor Milewski et al. (2006), %	–	90	90
Average cell voltage, mV	–	513	486
Current density, mA/cm^2	–	29.5	29.6
Oxygen mass flow, kg/s	–	0.2	0.2
MCFC/GT fuel ratio	–	0.52	0.65

Table 8. Nominal parameters of GT-MCFC system.

Case 1

The MCFC is fed by two streams: GT exhaust gas at cathode side and a mixture of methane and steam at anode side.

The MCFC anode outlet stream is directly delivered to the CO_2 separator.

The system was optimised to obtain maximum system efficiency. Primary (adjusted) variables of the optimizing process were:

- Cell current density
- MCFC/GT fuel ratio.

Parameters obtained during the optimizing process are given in Table 8.

A simple combination of the MCFC with GT gives:

- CO_2 emission reduction of 73%
- Unchanged electrical efficiency

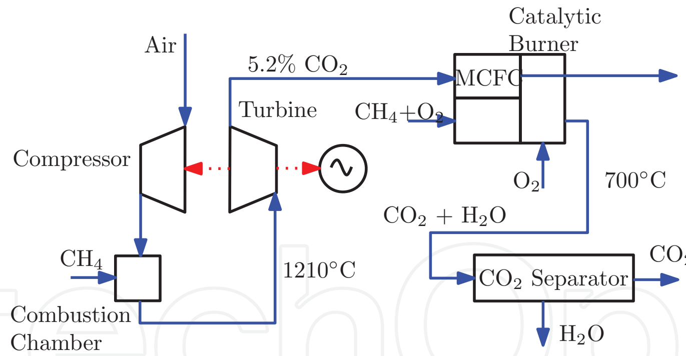

Fig. 33. GT-MCFC system – Case 1

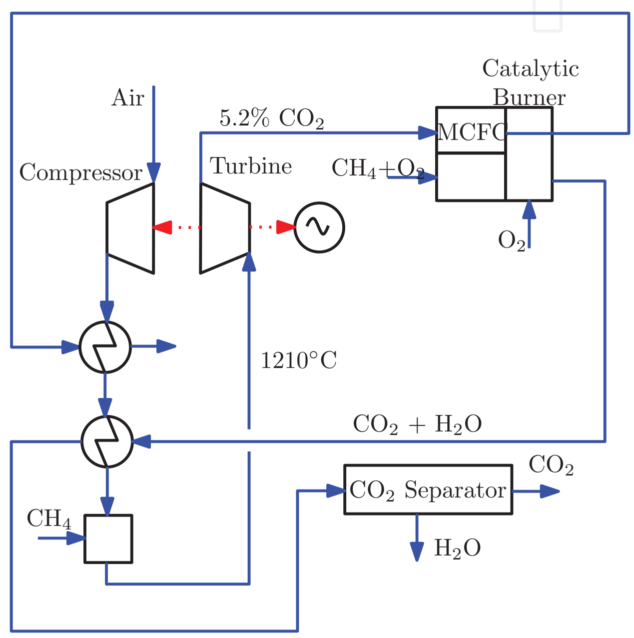

Fig. 34. GT-MCFC system – Case 2

- Power output increase of 23%.

Case 2

The MCFC-GT Case 2 was created by adding two heat exchangers. The heat exchangers are installed to recover exhaust heat from MCFC outlet streams. Note that the GT efficiency would increase with a recuperative heat exchanger when no MCFC is installed as well.

The system was optimised with the same conditions like Case 1. Nominal parameters of Case 2 of GT-MCFC system are given in Table 8.

GT-MCFC Case 2 generates slightly less power in comparison with Case 1. During the simulations a constant value of Turbine Inlet Temperature (TIT) was assumed. The implementation of heat exchangers means lower fuel mass flow demanded by the combustion chamber.

A reduction of the CO_2 emission of 91% is obtained. Simultaneously, electric efficiency is increased to 40% (LHV) what gives the relative emission of CO_2 of 37 kg/MWh.

Summary

The CO_2 emission reduction factor and CO_2 relative emission were used to compare the systems. The MCFC could reduce the CO_2 emission by more than 70% from gas turbine power plant exhaust. The relative CO_2 emission decreases more significant because the MCFC produces additional power.

Relatively low efficiency of the MCFC is caused by low CO_2 content at gas turbine exhaust, which limits maximum cell voltage.

A combination of MCFC with a GT unit means higher investment costs. Other devices like water separator and heat exchangers increase the total investment cost as well. It should be noted that typical CO_2 separation methods also increase the investment costs.

Application of the MCFC in a Gas Turbine plant gives a relatively high reduction in CO_2 emissions. The relative CO_2 emission of the GT unit is estimated at 609 kg_{CO_2}/MWh while in contrast the MCFC-GT hybrid system has an emission rate of 135 kg_{CO_2}/MWh. The quantity of CO_2 emitted by the MCFC-GT is 73% lower than is the case with the GT plant.

Application of the MCFC in a Coal Fired Power Plant gives a relatively high reduction in CO_2 emissions. The relative CO_2 emission of the coal plant is estimated at 1137 kg_{CO_2}/MWh while in contrast the MFCF-CFPP hybrid system has an emission rate of 300 kg_{CO_2}/MWh. The quantity of CO_2 emitted by the MCFC-CFPP is 60% lower than is the case with the CFPP.

As mentioned earlier, all cases were optimised to achieve maximum power generation efficiency. However, this might be changed, if it is assumed that the main task of the MCFC is to limit CO_2 emissions, which would result in the CO_2 emission reduction factor being used as the objective function of the optimisation process. If this factor is optimise,d the cell voltage at last cell can fall below zero and the MCFC will work as a CO_2 concentrator. At the very least, the MCFC would generate no power, and might even consume some. However, the main task of a power plant is power generation; hence hybrid system efficiency was chosen as the objective function for optimisation task.

Important technical issues such as sulphur or dust resistances of the MCFC fell outside the remit of this chapter, although they can evidently limit the application of MCFCs in both coal fired boiler and gas turbine power plants.

MCFCs could be profitably used in existing power plants which have been assigned emission CO_2 limits. MCFCs could potentially decrease CO_2 emissions, leaving the power generation capacity of the system at least the same, if not greater.

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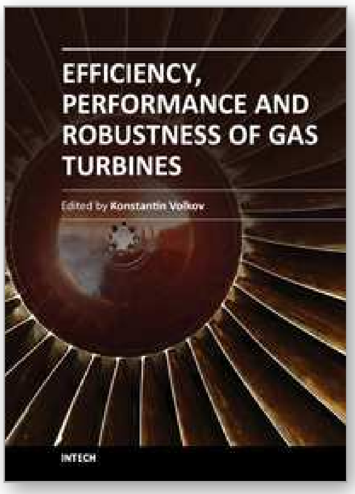

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A wide range of issues related to analysis of gas turbines and their engineering applications are considered in the book. Analytical and experimental methods are employed to identify failures and quantify operating conditions and efficiency of gas turbines. Gas turbine engine defect diagnostic and condition monitoring systems, operating conditions of open gas turbines, reduction of jet mixing noise, recovery of exhaust heat from gas turbines, appropriate materials and coatings, ultra micro gas turbines and applications of gas turbines are discussed. The open exchange of scientific results and ideas will hopefully lead to improved reliability of gas turbines.