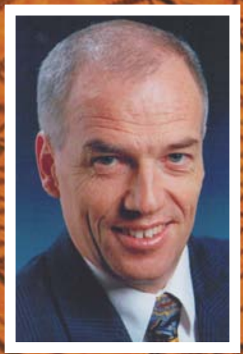
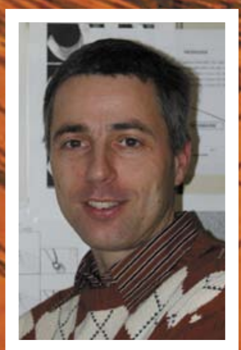
1.3.1.1

Graz Cycle – a Zero Emission Power Plant of Highest Efficiency


Franz Heitmeir

Wolfgang Sanz

Herbert Jericha

Institute for Thermal Turbomachinery and Machine Dynamics
Graz University of Technology, Graz, Austria

http://www.ttm.tugraz.at
franz.heitmeir@tugraz.at
wolfgang.sanz@tugraz.at

1.3.1.1-1 Introduction

In the last hundred years the concentration of some greenhouse gases in the atmosphere has markedly increased. There is a wide consensus in the scientific community that this seems to influence the Earth surface temperature and thus the world climate.

Therefore, in 1997 the Kyoto conference defined the goal of global greenhouse gas emission reduction of about 5% in the next years compared to the 1990 emission level. CO_2 is the main greenhouse gas due to the very high overall amount emitted by human activities, and about one third of the overall human CO_2 emissions are produced by the power generation sector. In the European Union (EU) there is a strong pressure on public utilities and industry to reduce the CO_2 emissions by power generation[1]. In 2003 the European Parliament passed a directive on emission trading. In 2005 emission allowances were assigned to about 10,000 companies in 25 countries within the EU which cover about 46% of the overall EU CO_2 emissions. Companies which do not need their full amount can sell it to companies which need more than assigned. As emission allowances become scarce, they will have an increasing value. First estimates varied between 10 and 20 €/ton CO_2 (12 and 24 \$/ton CO_2) by 2010, but in June 2005 European Union Allowances (EUA) were already being traded at 23 €/ton CO_2 (\$28/ton CO_2).

So there is a strong driving force to develop commercial solutions for the capture of CO_2 from power plants. The main technologies are as follows[2]:

- post combustion CO_2 capture, e.g. by washing of exhaust gases using amines;
- pre-combustion decarbonization of fossil fuels to produce pure hydrogen or hydrogen-enriched fuels for use in conventional power plants;
- chemical looping combustion; and,
- oxy-fuel cycles with internal combustion of fossil fuels with pure oxygen.

The authors believe that oxy-fuel cycles are a promising technology. The combustion with pure oxygen leads to a working fluid consisting mainly of steam and CO_2, which allows an easy and cost-effective CO_2 separation by steam condensation. Further advantages are the great variety of fuels which can be used (natural gas, syngas from coal or biomass gasification, etc.) and the low NO_x generation, since nitrogen is only introduced by fuel bound nitrogen or as a residue in the oxygen to the combustion chamber. The generated NO_x as well as other gases are removed together with CO_2, so that no pollutants are emitted to atmosphere. On the other hand oxy-fuel cycles need the development of new turbomachinery components and have to bear the high efforts for oxygen supply. Oxygen needed in a large amount for this kind of cycles can be generated by air separation units (ASU) which are in use worldwide with great outputs in steel making industry and even in enhanced oil recovery. The largest air separation plant already in operation for some years in the Gulf of Mexico produces nitrogen for the injection in the gas dome of a large oil field off-shore[3]. Fortunately, the new working fluid of steam and CO_2 allows new power plant cycles of highest efficiency, so that the additional efforts for oxygen supply can be largely compensated. Among them the Matiant Cycle, the Water Cycle, and the Graz Cycle are the best known[4].

History

The authors believe that the so-called Graz Cycle has the potential of highest efficiency. The basic principle was developed and published by Jericha in 1985[5]. He presented a power cycle without any emissions which was based on the internal combustion of hydrogen with oxygen in stoichiometric ratio as obtainable from solar power plants. Thermodynamically this steam cycle was an integration of a top Brayton cycle and a bottom Rankine cycle. In the nineties the hydrogen technology lost its impetus, so that the Graz Cycle was adopted for the firing of fossil fuels[6]. At this time cooperation with Japanese companies and research organizations led to the name

"Graz Cycle". The working fluid was a mixture of about three quarters steam and one quarter CO_2, the electrical efficiency was about 64%. Improvements and further developments since then were presented at many conferences[7]. In 2000, a variant of the Graz Cycle was proposed with a change of fuel from methane to oxygen blown coal gas (syngas), striving for minimum compression work[8]. All water of the cycle medium was condensed before compression, thus a minimum compression work could be obtained. In this cycle CO_2 was the main component of the working fluid. In the following years the general layout of all components for a 75 MW prototype plant of this type was presented[9].

But in 2004 there was a return to the original high steam content Graz Cycle (S-Graz Cycle), because it had become clear that the reduction in compression work of almost pure CO_2 has led to a considerable lowering of the inlet temperature to the combustion chamber[10]. So by increasing the steam content in recompression the compression work is increased, leading to a much higher combustion chamber inlet temperature. The heat input to the combustion chamber was lowered considerably thus raising the efficiency to the highest value that could be reached in the course of this cycle optimization. At the same time it turned out that much more steam for cooling could be made effective for the combustion chamber burners and the high temperature turbine (HTT) first blade rows. The resulting highest thermal efficiency of nearly 70% could be obtained if syngas was used as a fuel. The net efficiency, including the efforts of oxygen supply and compression of captured CO_2 for liquefaction, is 56%. The general layout of the components for a 75 MW prototype plant showed the feasibility of all components. In recent discussions with gas turbine industry a scale-up to a 400 MW plant was discussed for the S-Graz Cycle scheme. In 2005 further modifications of the Graz Cycle were discussed and their potential was analyzed[11]. An economic analysis of the Graz Cycle power plant showed the strong dependence of the economics on the still uncertain investment costs.

In this work the name "Graz Cycle" means the original "S-Graz Cycle," which was the more efficient variant and the one which will be pursued in the future.

1.3.1.1-2 Cycle configuration and thermodynamic layout

All thermodynamic simulations were performed using the commercial software IPSEpro by SIMTECH Simulation Technology[12]. This software allowed implementation of user-defined fluid properties to simulate the real gas properties of the cycle medium. The physical properties of water and steam were calculated using the IAPWS_IF97 formulations[13]; CO_2 was also modeled as real gas based on the correlation of Sievers[14]. Furthermore, a turbine module was developed for the calculation of cooled turbine stages. A simple stage-by-stage approach similar to the one presented by Jordal et al. was assumed. This assumption allowed for the calculation of the amount of cooling steam needed per stage[15]. Within the module, half of the cooling mass flow was mixed to the main flow at the stage inlet, thus contributing to the stage expansion work. The rest was added at the stage exit. Details of the model were presented in Luckel, 2004[16].

The thermodynamic data presented was for a cycle fired with methane, because it gave similar results as natural gas, the most likely fuel to be used in a first demonstration plant. The lower heating value was 50015 kJ/kg.

The thermodynamic simulation was based on the following assumptions on efficiencies and losses:

- The isentropic turbine efficiency is 90.3% for the High Temperature Turbine (HTT), 90% for the High Pressure Turbine (HPT) and 88% for the Low Pressure Turbine (LPT);
- The isentropic efficiency of CO_2 compressors is 78% and of CO_2/ H_2O compressors 88%;
- The isentropic efficiency of pumps is 75%;
- The mechanical efficiency of the turbomachinery is 99.6% of net power;
- The generator efficiency is 98.5%;
- The transformer efficiency is 99.65%;
- Auxiliary losses are 0.25% of heat input;
- The combustor heat loss is 0.25%, the pressure loss 4%;
- The oxygen excess is 3% of the stoichiometric ratio in order to keep CO generation low;
- The minimum temperature difference at Heat Recovery Steam Generator (HRSG) economizer is 5 K, at superheater 25 K;
- HRSG: cold side pressure loss is 28 bar (including 5 bar for HPT pipe); hot side pressure loss is 4 kPa;
- The pressure loss of all other heat exchangers is 3%;
- Fuel is supplied at 41.7 bar and 150°C;
- The cooling water temperature in the condenser is 10°C;
- CO_2 is released at 1 bar, efforts of a further compression to 100 bar including the remaining steam content at 1 bar (350 kJ/kg) is considered in the power balance; and
- The power consumption of oxygen production is 900 kJ/kg (0.25 kWh/kg) and of oxygen compression from an ASU exit pressure of 2.4 bar to combustor pressure is 325 kJ/kg.

Figure 1 shows the principle flow scheme of the S-Graz Cycle with the main components and main cycle data.

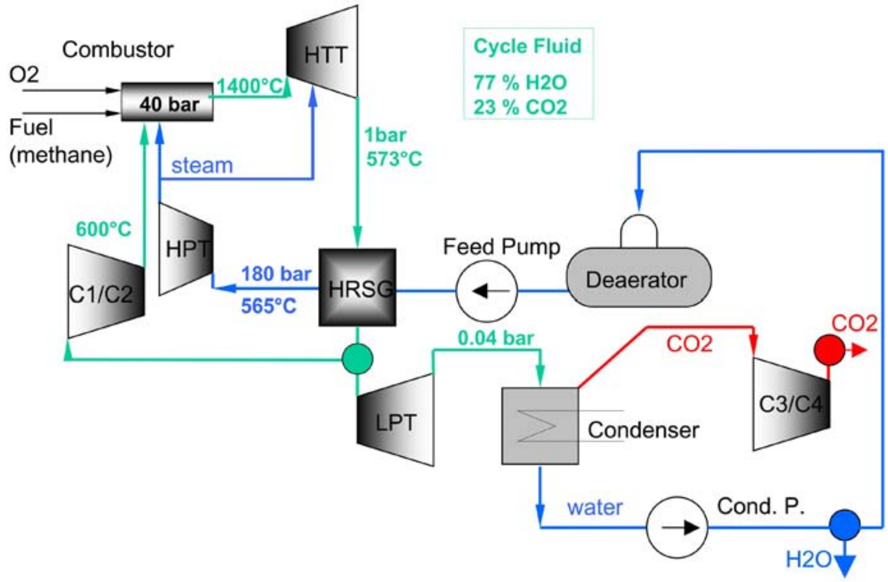


Fig. 1. Principle flow scheme of Graz Cycle power plant

Basically, the Graz Cycle consists of a high temperature Brayton cycle (compressors C1 and C2, combustion chamber and High Temperature Turbine HTT) and a low temperature Rankine cycle (Low Pressure Turbine LPT, condenser, Heat Recovery Steam Generator HRSG and High Pressure Turbine HPT). The fuel together with the nearly stoichiometric mass flow of oxygen is fed to the combustion chamber, which is operated at a pressure of 40 bar. Steam as well as a CO_2/ H_2O mixture is supplied to cool the burners and the liner.

A mixture of about 74% steam, 25.3% CO_2, 0.5% O_2 and 0.2% N_2 (mass fractions) leaves the combustion chamber at a mean temperature of 1400°C. The fluid is expanded to a pressure of 1.053 bar and 579°C in the HTT. Cooling is performed with steam coming from the HPT (13.7% of the HTT inlet mass flow), increasing the steam content to 77% at the HTT exit. It is quite clear that a further expansion down to condenser pressure would not end at a reasonable condensation point for the water component, so that the hot exhaust gas is cooled in the following HRSG to vaporize and superheat steam for the HPT, the pinch point of the HRSG is 25°C at the superheater exit. But after the HRSG, only 46% of the cycle mass flow is further expanded in the LPT. The LPT exit and thus condenser pressure is 0.043 bar.

For a mixture of a condensable (steam) and a non-condensable gas (CO_2) the condensation temperature depends on the partial pressure of steam, which continuously decreases during the condensation. For a given condensate exit temperature the condenser pressure determines the amount of steam condensed. In order to maximize the LPT power, the condenser pressure should be reduced as far as possible, but this is counteracted by an increased effort for compressing the gaseous steam/CO_2 mixture to atmospheric pressure. So for a given condensate exit temperature of 18°C (for a cooling water temperature of 10°C) the optimum condenser pressure is 0.043 bar, where about half of the combustion water is condensed (see figure 2).

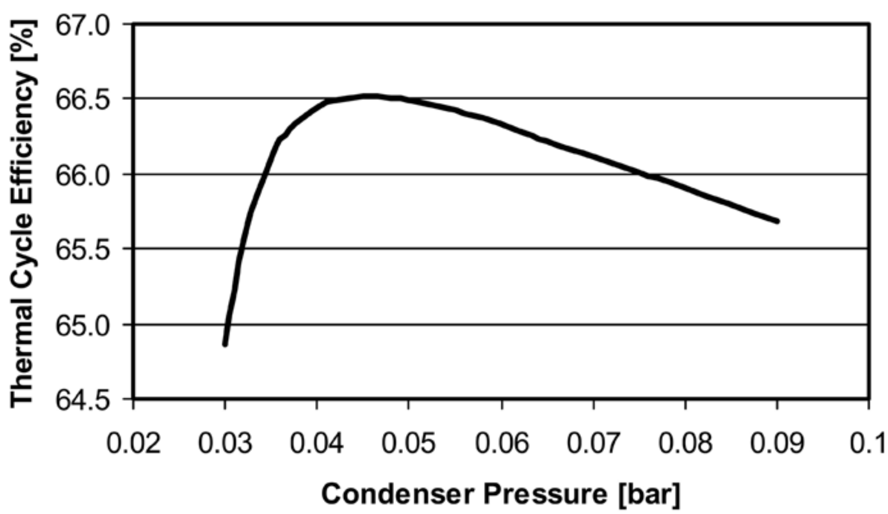


Fig. 2. Influence of the condenser pressure on thermal cycle efficiency

Gaseous and liquid phases are separated in the condenser. From there on, the gaseous mass flow, which contains the combustion CO_2 and half of the combustion water, is compressed to atmosphere (C3/C4) with intercooling and extraction of condensed water, and supplied for further use or storage. At atmosphere the CO_2 purity is 94%, further water extraction is done during further compression for liquefaction.

After segregating the remaining combustion H_2O, the water from the condenser is preheated, vaporized, and superheated in the HRSG. The steam is then delivered to the HPT at 180 bar and 549°C. After the expansion it is used to cool the burners and the HTT stages.

The major part of the cycle medium, which is separated after the HRSG, is compressed using an intercooled compressor (C1/C2) and fed to the combustion chamber with a maximum temperature of 600°C. The detailed flow sheet used for the thermodynamic simulation is included in the appendix and gives mass flow, pressure, temperature, and enthalpy of all streams.

In order to achieve a high thermal efficiency the heat extracted from a power cycle should be small compared to the input. The cycle arrangement of the Graz Cycle achieves this on the one hand by a very high peak temperature enabling large heat input, and on the other hand by feeding only the smallest possible mass flow of working fluid to the condenser (main heat extractor from the cycle) which has to contain the CO_2 generated in the combustor. The major part of the working fluid is compressed in the gaseous phase and so takes its high heat content back to the combustion chamber.

Graz Cycle for syngas from coal gasification

The Graz Cycle is suited for all kinds of fossil fuels. Best results regarding net cycle efficiency can be obtained for syngas firing from coal gasification. For this investigation it was assumed that syngas is bought from an external gasification plant at elevated costs, so that the production effort was not considered in the thermodynamic balance. Syngas was provided at 500°C, because coal gasification takes place at very high temperatures. This temperature was chosen due to material restrictions. The syngas composition was typical for an oxygen blown coal gasification plant (syngas mole fractions: 0.1 CO_2, 0.4 CO, 0.5 H_2).

Due to the higher carbon content of the fuel, the composition of the working fluid at HTT exit was 69% steam and 31% CO_2 (mass fractions). Then, half of the cycle mass flow was expanded in the LPT and fed to the condenser, where the lower steam content led to a slightly higher optimum pressure of 0.05 bar. But in general the main cycle parameters did not change considerably.

Power balance

Table 1 gives the power balance of the Graz Cycle plant for methane and syngas firing for 143.8 MW heat input. For syngas two variants with syngas at 150°C and 500°C are given in order to better understand the differences to the methane fired version. The net cycle efficiency shown in the last row was calculated according to Equation 1.

Table 1 Graz Cycle power balance

	Methane fired Graz Cycle	Graz Cycle with syngas at 150°C	Graz Cycle with syngas at 500°C
HTT power [MW]	119.4	115.7	119.7
Total turbine power P_T [MW]	141.8	138.0	143.0
Total compression power P_C [MW]	46.5	43.0	43.6
Net shaft power without mechanical losses [MW]	95.3	95.0	99.4
Total heat input Q_{zu} [MW]	143.8	143.8	143.8
Thermal cycle efficiency [%]	**66.3**	**66.1**	**69.1**
Electrical power output [MW] incl. mechanical, electrical & auxiliary loss	93.2	92.9	97.2
Net electrical cycle efficiency [%]	**64.8**	**64.6**	**67.6**
O_2 generation & compression P_{O2} [MW]	14.7	11.5	11.5
Efficiency considering O_2 supply [%]	**54.6**	**56.6**	**59.6**
CO_2 compression to 100 bar P_{CO2} [MW]	3.0	5.0	5.0
Net power output [MW]	75.5	76.4	80.7
Net efficiency η_{net} [%]	**52.5**	**53.1**	**56.1**

$$\eta_{net} = \frac{(P_T - P_C) \cdot \eta_m \cdot \eta_{gen} \cdot \eta_{tr} - P_{aux} - P_{O2} - P_{CO2}}{Q_{zu}} \qquad (1)$$

Looking at the methane-fired version of the Graz Cycle, we see that the HTT was the major turbomachinery component in the cycle. The thermal cycle efficiency was 66.3%, and accounting for the electrical, mechanical and auxiliary losses, the net electrical cycle efficiency was 64.8%, a value far higher than that which is typical of state-of-the-art combined cycle plants. If considering the efforts for oxygen production and compression to combustion pressure, a net efficiency of 54.6% could be evaluated. If the cycle were to be penalized for the CO_2 compression to 100 bar needed for liquefaction, the net efficiency would further reduce to 52.5%, a value still higher than that of most alternative technologies.

In comparison of the methane-fired version with the Graz Cycle fired with syngas provided at the same temperature as methane, i.e. 150°C, the turbomachinery power reduces due to the less steam content with a lower heat capacity. The cycle efficiency was slightly reduced by 0.2 %-points due to higher condenser losses. Syngas demands less oxygen per heat input, so that the penalty of oxygen supply decreases considerably. But this gain is partly offset by a larger amount of CO_2 generated by syngas firing which then has to be compressed for liquefaction. Finally, the syngas fired version has a net cycle efficiency of 53.1%, 0.6 %-points higher than the methane-fired version. If the heat of the syngas production can be used in the Graz Cycle plant for free (it is considered only in the fuel price), the net cycle efficiency would increase by 3 %-points up to 56.1%.

Sensitivity study of HTT performance

The significance of the thermodynamic simulation was based on the choice of reasonable data for component efficiency and losses. The two key parameters for the Graz Cycle were the HTT efficiency and HTT cooling mass flow because of the very high contribution of this turbine to the overall power generation. Figure 3 shows the influence of the HTT isentropic efficiency. The effect of an improved HTT efficiency was counteracted by the decreased HTT outlet temperature resulting in a decrease of the HPT power output. If we assume an HTT isentropic efficiency of 92% instead of 90.3%, the net cycle efficiency would reach only 53% instead of 53.8%, the value expected if we do not account for the above mentioned effect of the reduced HTT exit temperature on the overall cycle.

On the other hand, the HTT cooling mass flow had a more significant influence on the cycle efficiency. It was estimated to be 13.7% using a model evaluated by comparison with conventional gas turbines, but a percentage-point increase in cooling mass flow decreased the net efficiency by 0.22 %-points. These considerations showed that the HTT performance had a decisive influence on the overall cycle efficiency.

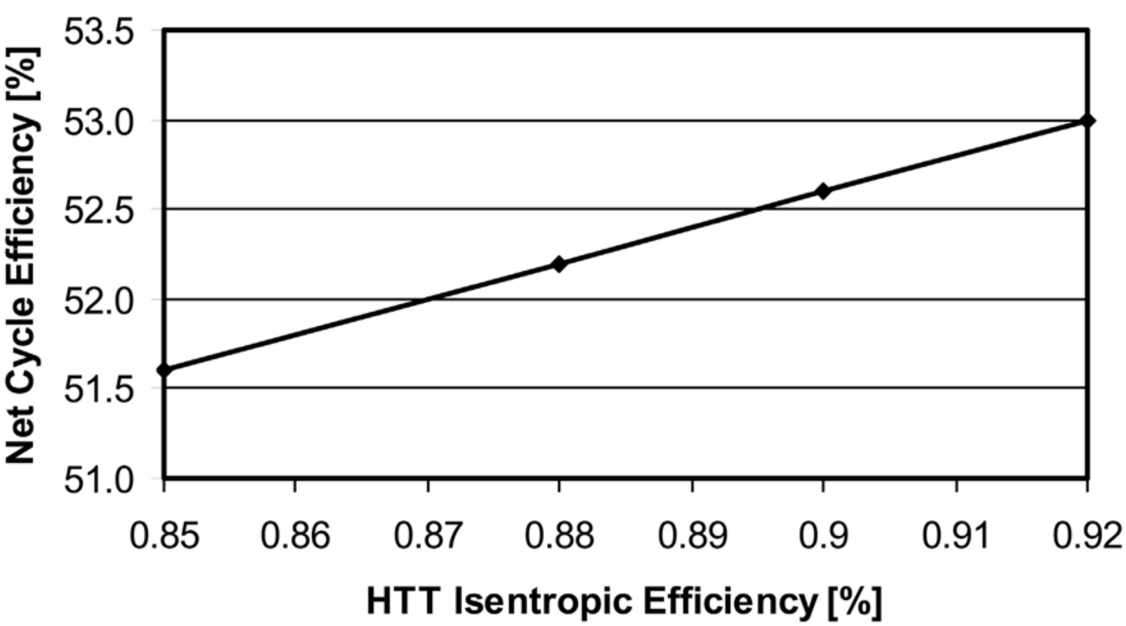


Fig. 3. Influence of HTT isentropic efficiency on net cycle efficiency

Modifications of cycle configuration

In order to improve the efficiency of the Graz Cycle, several modifications were investigated. The following cycle variants will be discussed in this work:

- condensation of the cycle working fluid at 1 bar and re-vaporization of the separated water; and,
- heat supply to the deaerator by the cooling heat of the CO_2 compression intercoolers.

2.4.1. Condensation at 1 bar and water re-vaporization

The working fluid containing the mass flow of CO_2 generated in the combustor was expanded in the LPT to a condenser pressure of 0.043 bar. There the steam was condensed, allowing separation of the gaseous CO_2. But this configuration had some deficiencies:

1) The condenser was very large and thus expensive, because of the high volume flow and the anticipated reduced heat transfer due to the high inert gas content; and

2) The CO_2 mass flow was expanded in the LPT together with steam and afterwards recompressed to atmosphere, so that due to the higher compression effort and additional losses a net loss was generated by this mass flow.

Therefore it was suggested in the Austrian patent of the Graz Cycle to condense this mass flow at atmospheric pressure, separate the combustion CO_2 and re-vaporize the water at a reduced pressure level using the condensation heat (see figure 4)[17]. The steam was then fed to the LPT and could be expanded to a condenser pressure lower than that for the working fluid mixture[18]. Advantages of this configuration are the avoidance of the difficult condenser for the working fluid at vacuum conditions and the saving of the relatively large CO_2 compressors C3 and C4 needed for compression to atmosphere. Instead of using a standard condenser, an additional condensation/re-vaporization unit was needed. This condensation/revaporization unit worked at atmospheric conditions and was similar to distillers used for conversion of sea and brackish water into high purity water by vacuum vapor compression.

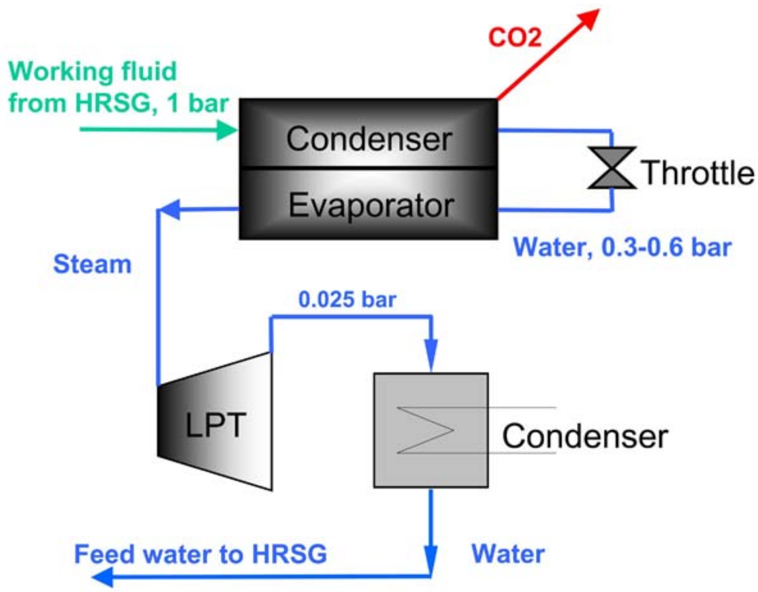


Fig. 4. Scheme of condensation/re-vaporization

If the saving of CO_2 compression power and the advantage of a lower condenser pressure exceeded the power loss of the LPT due to the reduced mass flow, a net gain in efficiency could be achieved. A thermodynamic study found an optimum for a dual pressure vaporization at the pressure levels 0.55 bar and 0.3 bar. The losses assumed for vaporization were 0.18 bar for the higher pressure level and only 0.07 bar for the lower pressure level. If these relatively low losses could be met, the efficiency for this new configuration would remain the same at 52.5%. So this configuration could lead to reduced plant costs and an even greater efficiency, if the original low condenser pressure cannot be kept for the working fluid condensation.

As a second alternative currently investigated, the condensation heat could be utilized in a bottoming steam cycle. It has the advantage of more flexibility, of an easier start-up of the plant and has an easier water make-up.

2.4.2. Deaerator heating by CO_2 compression intercoolers

In order to remove dissolved gases (N_2, O_2 and CO_2) in the HRSG feed water, a deaerator was arranged in front of the feed pump. Since there was no pure steam at an appropriate pressure available for heating, the feed water was heated close to saturation temperature in a surface heat exchanger that utilized the working fluid extracted in front of the LPT. This fluid passing by the LPT caused a reduction in its power output. To avoid this configuration and the resulting power reduction, it was investigated to supply the necessary heat for the deaerator from the CO_2 compression coolers instead of the working fluid by passing the LPT (figure 5).

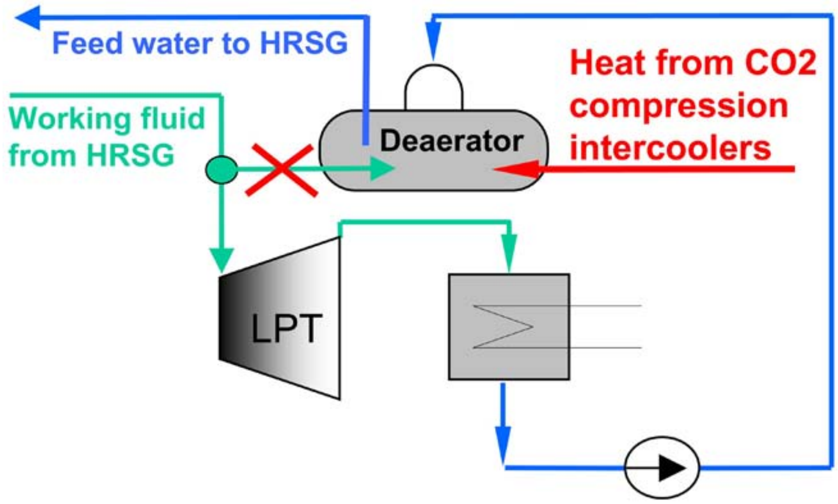


Fig. 5. Scheme of deaerator supplied with heat from CO_2 compression intercoolers

The thermodynamic simulation showed that the heat from the CO_2 intercoolers can completely replace the extraction in front of the LPT. So the mass flow and thus the power output of the LPT increased by 8.5%, resulting in an increase of net cycle efficiency by 0.8 %-points up to 53.3%. This improvement showed that there was still room for efficiency improvement of the Graz Cycle, but often in trade-off with higher complexity.

1.3.1.1-3 Turbomachinery Design

Prototype plant of 75 MW output

Compression and expansion in large power cycles can only be affected with modern turbomachinery. The gases we have to deal with in our case, CO_2 and H_2O steam, are very compressible at the given high enthalpy heads or pressure ratios. The resulting high changes in volumetric flow in the individual compressors and turbines require a multi-shaft arrangement connected by gears.

The design decision to have the high temperature flow channel with minimum surface area and minimum heat loss and also with minimum cooling flow supply leads to the arrangement of turbomachinery as given in figure 6, which also includes the compressors C3 and C4 used to compress the separated CO_2 to atmospheric pressure. There it is delivered to a final compression up to 100 bar for liquefaction.

The HTT turbine needs 4 stages due to the high heat capacity of the steam-rich cycle medium. The HTT is split into two shafts, where the first stage runs at 23 000 rpm, the other three stages at 12 000 rpm. The two overhang disks of different speed provide the shortest possible high temperature annular flow channel. A bearing is arranged between the second and third stages. In order to reduce the number of generators, the power of all four compressors is balanced with the HTT first stage and the HPT. Both turbines drive the cycle medium compressors C1 and C2 and in normal operation they also drive the CO_2 delivering compressors C3 and C4. These compressors are connected via a self-synchronizing clutch and are disconnected from the main high-speed shaft during start-up. Then they are driven by a separate electric motor in a mode similar to the vacuum pump in a steam plant. This arrangement needs two gear boxes, because the compressors C1 and C4 run at 12 000 rpm and the compressor C3 at 3 000 rpm.

The stages 2, 3, and 4 of the HTT run at 12 000 rpm and deliver their power via the main gear to the generator, which is driven on the other side by the LPT in a way that is similar to very large steam turbines.

The main turbomachinery data and their dimensions for a prototype plant are given in[19]. Due to the small volumetric flow of the HPT it is designed in the form of a 4-stage partial admission impulse steam turbine. Its arrangement immediately ahead of the HTT allows for cooling of the HTT first stage disk in an effective way. Exhaust steam is fed via labyrinth seals to the front side of the disk thus holding the shaft and the disk at a temperature of around 300°C. The disk is bell shaped with broad width in the center leading to a strong fir-tree root blade attachment which contains the cooling steam inlet ports to the hollow blades.

On the other side, the space between the HTT first and second stage disks is again filled with cooling steam from outside, cooling both disks and providing in a form of a stationary steam bearing additional damping to both shafts. Again from here cooling steam is fed into the second disk and its blades.

The compressors C1 and C2 have to act on a medium consisting of CO_2 and steam. The high volume change requires a change of speed (C1 at 12 000 rpm, C2 at 23 000 rpm) with relatively high Mach numbers at the tip of their respective first blades. But relatively long lasting blades result in low clearance loss and low deterioration of the meridional flow profile. In order to keep the high-speed shaft short and in order to reduce the number of stages in C2 a radial final stage is proposed which can replace 3 or 4 axial stages due to its higher diameter and at the same time can deliver the medium radially outwards, making the inflow to the combustion chamber easier.

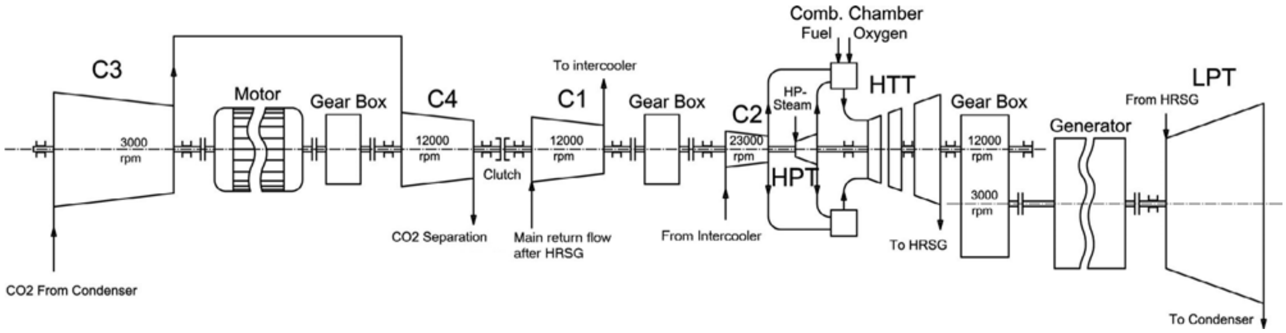


Fig. 6. Schematic arrangement of turbomachinery for a 100 MW S-Graz Cycle power plant

Graz Cycle plant of 400 MW output

For a Graz Cycle power plant of 400 MW net power output, a different but also very feasible design is suggested which works without gear boxes in the main power shafts. The turbomachinery arrangement consists of three independent shafts. The first shaft is a free running high speed shaft of 7600 or even 8500 rpm. It consists of the working fluid compressors C1 and C2 and the first two stages of the HTT turbine working as a compressor turbine. The turbine power is balanced with the power demand of the two compressors. The second shaft consists of the HTT power turbine and the low pressure turbine LPT. Both turbines run at 3000 rpm and drive the electrical generator. The third shaft consists of the CO_2 compressors C3 and C4 and the HPT turbine. It runs at 3000 and preferably 7,600 rpm connected via a gear box. An additional motor/generator delivers power for start-up and is driven in full load by the HPT power which exceeds the demand of the compressors.

Development work needed for a Graz Cycle plant

Most components of the Graz Cycle are well known but they have to work with an unusual working fluid of steam and CO_2. More critical components are as follows:

- The combustor for a nearly stoichiometric combustion with oxygen and use of steam and CO_2 as cooling medium;
- The HTT with a working fluid of about three quarts of steam and one quart of CO_2 and steam cooling;
- The condenser, condensing steam in the presence of a high content of inert gas.

All other components (LPT, HPT, all compressors, HRSG and heat exchangers) can be considered as standard components and do not pose any difficult design problems.

Combustion chambers for firing oxygen with methane in a steam environment have already been tested in the USA, in Japan, and Europe[20]. Recent tests at the National Energy Technology Laboratory (NETL) were performed for a 1 MW combustor, working at 10 bar and an exit temperature of 1200°C[21]. The combustor was no more difficult to operate than a regular combustor and performed well in terms of CO generation, if a small oxygen surplus of 3% was provided. In summary, all investigations showed that the concept of oxy-fuel combustion using steam dilution is viable.

The design of the HTT was studied carefully at Graz University of Technology and discussed at several conferences[22]. Recent studies by a main gas turbine manufacturer also confirmed the technical feasibility of the HTT, but experience has to be gathered for the behavior of the high-temperature alloys in the steam/ CO_2 environment of HTT hot sections.

The condenser has to deal with a large volume flow due to its very low pressure and the difficulty of reduced heat transfer in the presence of inert gas. But little experience has yet been gathered for steam condensation at a 20% CO_2 content, so that very little data on heat transfer in this environment is available. Further research work is necessary and condensation at 1 bar as discussed above is a reasonable option if very large heat transfer surfaces are required.

1.3.1.1-4 Economic Evaluation

Despite the high efficiency and the positive impact on the environment by a Graz Cycle power plant, a future application of this technology and an erection of a power plant mainly depend on the economic balance. The main indicator characterizing the economic performance of a power plant for CO_2 capture is the mitigation costs. They represent the increased capital and operational costs incurred by new and additional equipment and lower cycle efficiencies in relation to the CO_2 mass flow avoided. The CO_2 captured has an economic value of about $10/ton, if it can be used for enhanced oil recovery (EOR) or of about $30/ton in the future CO_2 emission trading scenario. These prices show the momentary threshold for the economic operation of zero emission power plants, although it is very difficult to foresee future trends.

In order to estimate the mitigation costs for a Graz Cycle plant, an economic comparison with a state-of-the-art combined cycle power plant of 58% efficiency is performed. The economic balance is based on the following assumptions:

- The yearly operating hours is assumed at 8500 hrs/yr;
- The capital charge rate is 12%/yr;
- Methane fuel costs are 1.3 ¢/kWh$_{th}$;
- Syngas is supplied by a syngas producer at 3.5 ¢/kWh$_{th}$, so no efficiency penalty for the production or additional investment costs are considered;
- The investment costs per kW are the same for the reference plant of about 400 MW net power output and the Graz Cycle plant (see below);
- Additional investment costs are assumed for the air separation unit (ASU), for additional equipment and CO_2 compression to 100 bar (see Table 2[23]); and,
- The costs of CO_2 transport and storage are not considered because they depend largely on the site of a power plant.

Table 2 Estimated investment costs

Component	Scale parameter		Specific costs
Reference Plant			
Investment costs	Electric power	$/kW$_{el}$	414
Graz Cycle Plant			
Investment costs	Electric power	$/kW$_{el}$	414
Air separation unit [24]	O_2 mass flow	$/(kg O_2/s)	1 500 000
CO_2-Compression system [24]	CO_2 mass flow	$/(kg CO_2/s)	450 000
Other costs (Piping, CO_2-Recirc.) [24]	CO_2 mass flow	$/(kg CO_2/s)	100 000

The assumption of similar investment costs for a conventional and a Graz Cycle power plant is based on a comparison with typical turbomachinery sizes for a 400 MW combined cycle plant as given in table 3. It shows that the turbine power and the HRSG are of similar sizes, whereas the compressor power is remarkably smaller. On the other hand the Graz Cycle needs a larger generator due to the additional power consumption for ASU and CO_2 compression. Developmental efforts are needed especially since the HTT and combustor were not considered in the investment costs.

Table 3 Comparison of equipment size for a 400 MW plant in terms of power

	Conventional CC plant	Graz Cycle plant
turbine of "gas turbine"/HTT	667 MW	638 MW
compressor of "gas turbine"/C1+C2+C3+C4	400 MW	251 MW
steam turbine/HPT+LPT	133 MW	121 MW
HRSG	380 MW	365 MW
Generator	400 MW	500 MW

Three indicators characterizing the economic performance of a power plant for CO_2 capture are estimated:

- The costs of electricity (COE) for both plants;
- The differential COE representing the additional costs of electricity due to CO_2 capture;
- The mitigation or capture costs representing the additional costs incurred by CO_2 capture per ton CO_2.

Table 4 shows the result of the economic evaluation for methane and syngas firing, respectively. For syngas firing, the reference plant is also syngas-fired without considering an efficiency decrease. The syngas plant has slightly smaller additional investment costs because of the smaller ASU needed.

Compared to the reference plant, the capital costs are about 60% – 70% higher by considering only the additional components for O_2 generation and CO_2 compression. So they contribute mostly to the difference in COE. The fuel costs have the major influence on the COE, especially for syngas firing, but they do not differ largely between reference and Graz Cycle plant. The O&M costs are assumed 15% higher for a Graz Cycle plant due to the operation of additional equipment.

Table 4 Economic data for methane and syngas fired Graz Cycle

	Methane fired ref. plant	Methane fired Graz Cycle		Syngas fired ref. plant	Syngas fired Graz Cycle
Plant capital costs [$/kW$_{el}$]	414	414		414	414
Addit. capital costs [$/kW$_{el}$]		288			258
CO_2 emitted [kg/kWh$_{el}$]	0.342	0.0		0.583	0.0
Net plant efficiency [%]	58.0	52.5		58.0	56.1
COE for plant amort. [¢/kWh$_{el}$]	0.58	0.99		0.58	0.95
COE due to fuel [¢/kWh$_{el}$]	2.24	2.47		6.03	6.24
COE due to O&M [¢/kWh$_{el}$]	0.7	0.8		0.7	0.8
Total COE [¢/kWh$_{el}$]	**3.52**	**4.26**		**7.31**	**7.99**
Comparison					
Differential COE [¢/kWh$_{el}$]		**0.74**			**0.68**
Mitigation costs [$/ton CO_2 capt.]		**21.6**			**11.7**

Due to the more expensive fuel, the COE for syngas firing is by far larger than for methane firing (COE due to fuel). But regarding the differential COE, the difference is 0.74 ¢/kWh$_{el}$ for the methane-fired Graz Cycle and 0.68 ¢/kWh$_{el}$ for the syngas-fired version compared to the respective reference plant. But due to the higher carbon content in syngas, the mitigation costs are only $11.7/ton CO_2 for the syngas plant compared to $21.6 /ton CO_2 for the methane-fired plant. These values are clearly below the threshold value of $30/ton showing the economic potential of the Graz Cycle.

The results of the economic study depended mainly on the assumptions about investment costs, fuel costs and capital charge rate as well as on the choice of the reference plant. A cost sensitivity analysis was performed and showed that a variation of the capital costs had the main influence on the economics, since they contributed most to the mitigation costs[24]. Unfortunately, there was a large uncertainty of these costs. A survey of the ASU costs vary in the range of $230 to $400/kW$_{el}$ (the same price as for a complete power plant). Considering this variation solely, the mitigation costs varied between $21.6 and $29.0/ton CO_2 for the methane-fired plant (see figure 7).

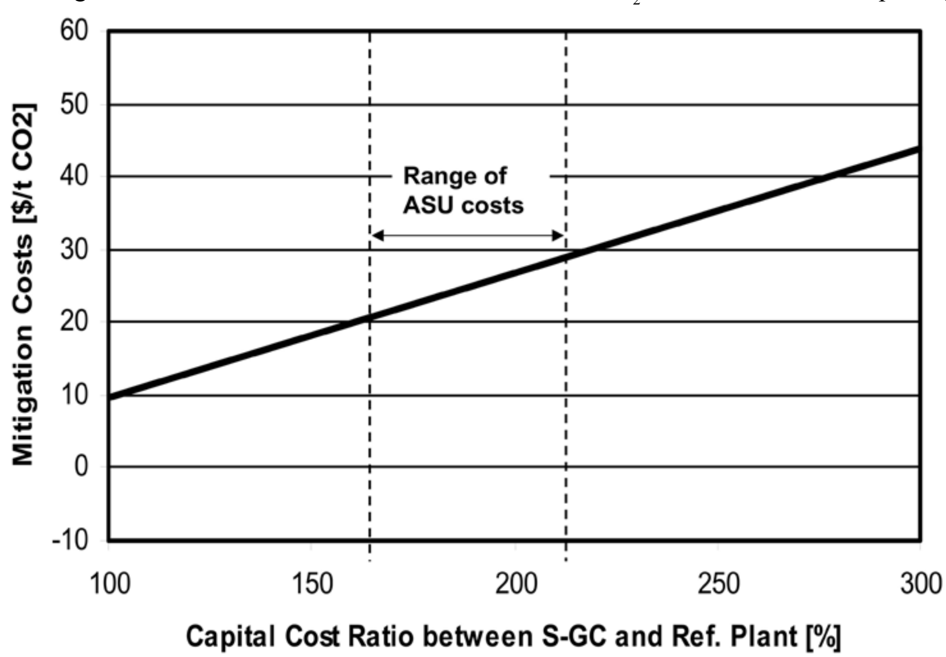


Fig. 7. Influence of capital costs on the mitigation costs (methane-fired Graz Cycle)

This high sensitivity to the capital costs showed the dilemma in performing an exact economic evaluation, since their estimation for a Graz Cycle power plant was very difficult because of the new turbomachinery components. But the authors claimed that their design of high-speed transonic stages with innovative steam cooling allowed a cost-effective manufacture. In these considerations about the height of additional investment costs, a further advantage of the Graz Cycle, the almost NOx-free combustion was not evaluated. According to exhaust flow NOx and CO catalytic reduction to achieve single-digit emissions (in strict attainment areas) can increase gas turbine genset plant costs by 40 to 50 percent[25].

1.3.1.1-5 Conclusions

The Graz Cycle is an oxy-fuel power cycle with the capability of retaining all the combustion generated CO_2 for further use. Its cycle configuration aims at highest efficiency by reducing the heat extraction in the condenser to a minimum. A thermodynamic investigation of the Graz Cycle fired with methane shows a net efficiency of 52.5%, if the efforts for oxygen supply and CO_2 compression to liquefaction are considered. If syngas can be used from an external syngas plant at 500°C, efficiencies can rise up to 56%. Studies show that further efficiency improvements and simplification of the cycle are possible.

A layout of all turbomachinery components for a 75 MW prototype plant as well as a 400 MW plant showed the technical feasibility of the Graz Cycle, although some development work is needed for the main components. But the authors claim that their proposed design of high-speed transonic stages with innovative steam cooling allows a cost-effective manufacture.

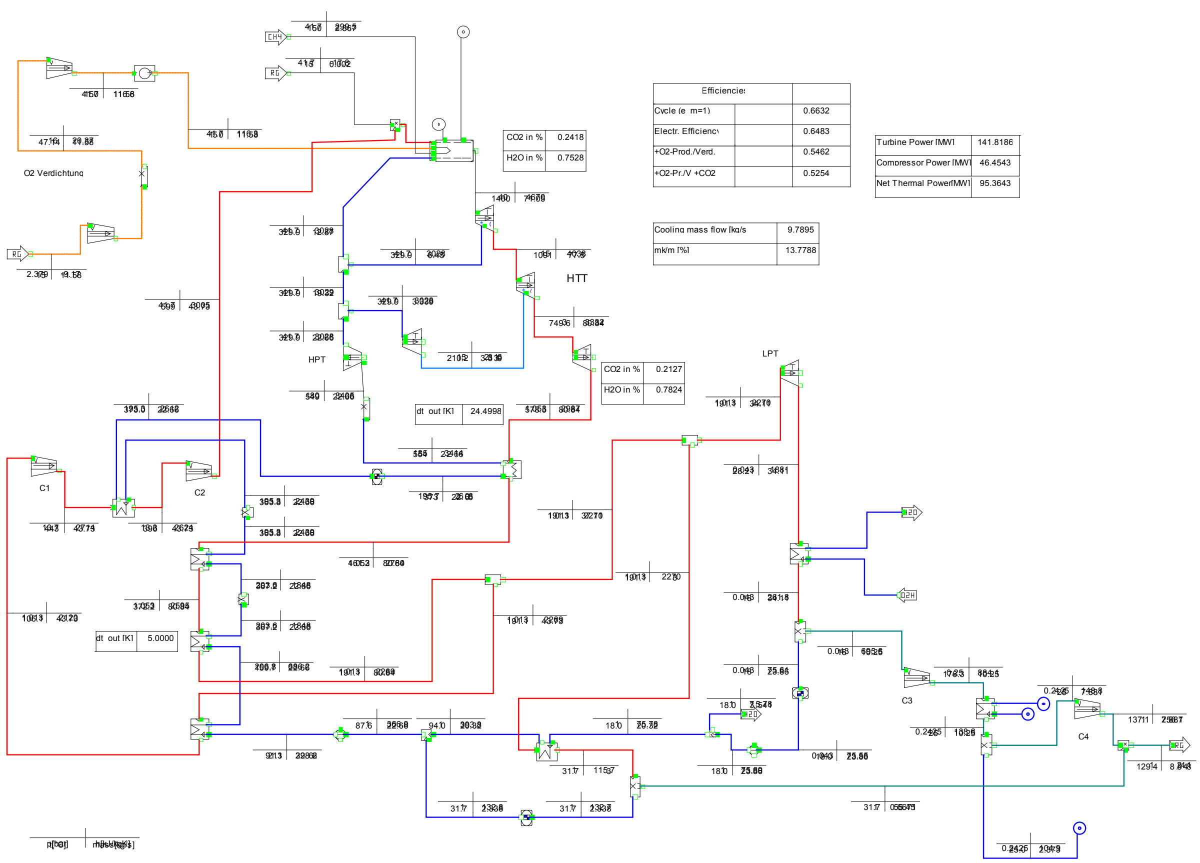
In an economic analysis the Graz Cycle power plant is compared with a state-of-the-art combined cycle plant. The resulting mitigation costs of 22 \$/ton CO_2 are below a threshold value of 30 \$/ton CO_2 (assumed for CO_2 emission trading), but this value mainly depends on the investment costs assumed. If syngas is used as fuel, the mitigation costs are only about 12 \$/ton CO_2 due to the higher carbon content of syngas.

All investigations done up to now confirm the high efficiency and technical feasibility of the Graz Cycle. In the scenario of increasing costs of CO_2 emissions, the investment in such a zero-emission power plant seems very reasonable in the near future.

1.3.1.1-6 Abbreviations and Appendix

ASU - air separation unit
COE - cost of electricity
HPT - high pressure turbine
HRSG - heat recovery steam generator
LPT - low pressure turbine

Appendix: Detailed thermodynamic cycle data of a Graz Cycle power plant fired with methane.

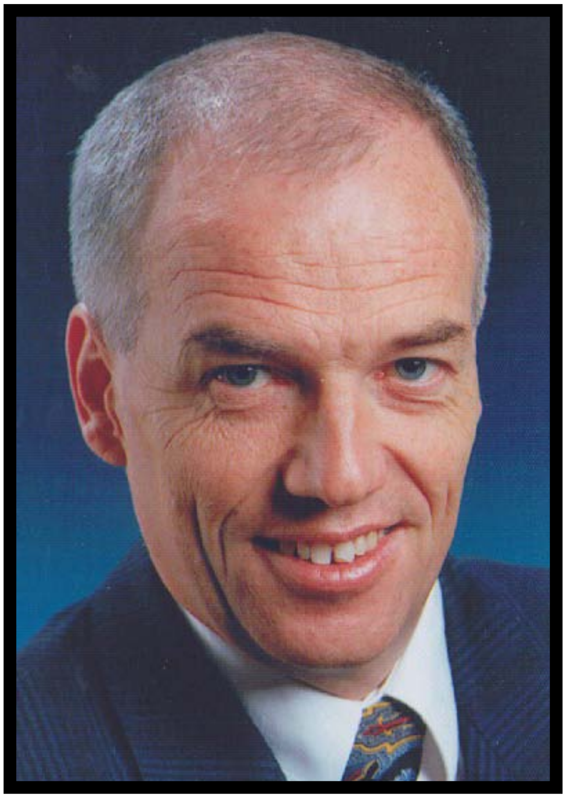


1.3.1.1-7 Notes

1. L. Strömberg, “Overview of CO_2 Capture and Storage – Technology and Economics for Coal-Based Power Generation,” (VGB Congress 2003, Copenhagen).
2. R. Gabbrielli and R. Singh, “Thermodynamic Performance Analysis of New Gas Turbine Combined Cycles with no Emissions of Carbon Dioxide,” *ASME Paper GT-2002-30117*, (ASME Turbo Expo 2002, Amsterdam, The Netherlands).
3. L. Turanskyj and B.A. Keenan, “Turbomachinery for the World’s Largest Nitrogen Plant: Enhanced Oil Recovery to Increase the Output in the Cantarell Oil Field, Mexico,” (Paper at the Exposición Latinoamericana del Petróleo, Maracaibo, Venezuela [2001]).
4. P. Mathieu and E. Iantovski, “Highly Efficient Zero Emission CO2-based Power Plants,” *Energy Conversion and Management* 38 no. 1 (1997): S141-146; R. A. Anderson, S.E. Doyle, and K.L. Pronske, “Demonstration and Commercialisation of Zero-Emission Power Plant,” (29th Int. Techn. Conference on Coal Utilization & Fuel Systems, Clearwater [2004]).
5. See note 1 above.
6. H. Jericha et al., “ CO_2 - Retention Capability of CH_4/O_2 – Fired Graz Cycle,” (CIMAC Conference Paper, Interlaken, Switzerland [1995]).
7. H. Jericha and M. Fesharaki, “The Graz Cycle – 1500°C Max Temperature Potential $H_2 – O_2$ Fired CO_2 Capture with $CH_4 – O_2$ Firing,” *ASME Paper 95-CTP-79*, (ASME Cogen-Turbo Power Conference, Vienna, Austria [1995]).
8. H. Jericha, A. Lukasser, and W. Gatterbauer, “Der “Graz Cycle” für Industriekraftwerke gefeuert mit Brenngasen aus Kohle- und Schwerölvergasung” (in German, VDI Berichte 1566, VDI Conference Essen, Germany [2000]).
9. Jericha, H., Göttlich, E., 2002, „Conceptual Design for an Industrial Prototype Graz Cycle Power Plant“, ASME Paper 2002 GT-30118, ASME Turbo Expo 2002, Amsterdam, The Netherlands; H. Jericha et al. „Design Optimisation of the Graz Cycle Prototype Plant,“ *ASME Paper 2003-GT-38120*, (ASME Turbo Expo 2003, Atlanta, USA and ASME Journal of Engineering for Gas Turbines and Power, October 2004, 126: 733-740); F. Heitmeir et al., “The Graz Cycle – A Zero Emission Power Plant of Highest Efficiency,” (XXXV Kraftwerkstechnisches Kolloquium, Dresden, Germany [2003]); F. Heitmeir and H. Jericha, 2005, “Turbomachinery Design for the Graz Cycle: An Optimized Power Plant Concept for CO_2 Retention,” *Proc. ImechE* 219, Part A: J.Power and Energy (2005): 147-155.
10. W. Sanz et al., “Thermodynamic and Economic Investigation of an Improved Graz Cycle Power Plant for CO2 Capture,” *ASME Paper GT2004-53722*, (ASME Turbo Expo 2004, Vienna, Austria, accepted for publication in ASME Journal of Engineering for Gas Turbines and Power).
11. W. Sanz, et al., “A Further Step Towards a Graz Cycle Power Plant for CO_2 Capture,” *ASME Paper GT2005-68456*, (ASME Turbo Expo 2005, Reno, USA).
12. SimTech Simulation Technology, “IpsePro Overview,” http://www.simtechnology.com/IPSEpro (accessed 2003).
13. W. Wagner and A. Kruse, *Properties of Water and Steam* (Springer-Verlag Berlin Heidelberg New York, 1998).
14. U. Sievers, „Die thermodynamischen Eigen-schaften von Kohlendioxid (in German),“ VDI Verlag, Düsseldorf. Fortschritt-Berichte VDI, Reihe 6, Energietechnik, Nr. 155 (1984).
15. K. Jordal, O. Bolland, and A. Klang, „Effects of Cooled Gas Turbine Modelling for the Semi-Closed O2/ CO_2 Cycle with CO_2 Capture,” *ASME Paper 2003-GT-38067* (ASME Turbo Expo 2003, Atlanta, USA).
16. F.Luckel, „Weiterentwicklung des Graz Cycle und der Vergleich mit anderen CO2-Rückhaltekonzepten“ (in German, Diploma Thesis, Graz University of Technology [2004]).
17. H. Jericha and W. Sanz, “Wärmekraftanlagen mit Verbrennung von Kohlenwasserstoffen mit reinem Sauerstoff zur Stromerzeugung bei Rückhaltung von Kohlendioxyd” (in German, Austrian Patent No. AT 409 162 B [2001]).
18. See note 12 above.
19. See note 11 above.
20. B. T. Chorpening, K. H. Casleton, and G. A. Richards, „Stoichiometric Oxy-Fuel Combustion for Power Cycles With CO_2 Sequestration,” (Proceedings of the Third Joint Meeting of the U.S. Sections of The Combustion Institute, Chicago, IL [2003]); B. T. Chorpening et al., “Demonstration of a Reheat Combustor for Power Productin With CO_2 Sequestration,” *ASME Paper 2003-GT-38511* (ASME Turbo Expo 2003, Atlanta, USA); H. Inoue, N. Kobayashi, and T. Koganezawa, “Research and Development of Methane-Oxygen Combustor for Carbon Dioxide Recovery Closed-Cycle Gas Turbine,” *CIMAC*, Hamburg, Germany (2001); T. Griffin et al., “Staged Catalytic Combustion Method for the AZEP,” *ASME paper GT2004-54101* (ASME Turbo Expo, Vienna, Austria [2004]).
21. See note 21 (first two references).
22. H. Jericha et al., “Konstruktion der ersten Stufe der HTT-Gasturbine für den Graz Cycle” (in German), VDI Berichte 1857, VDI Tagung “Stationäre Gasturbinen: Fortschritte und Betriebserfahrungen”, Leverkusen (2004); also see note 10 (2nd and 3rd reference).
23. G. Göttlicher, G., „Energetik der Kohlendioxidrückhaltung in Kraftwerken“ (in German), Fortschritt-Berichte VDI, Reihe 6, Energietechnik, Nr. 421. Düsseldorf: VDI Verlag (1999).
24. See note 11 above.
25. *Gas Turbine World,* “2003 Handbook for Project Planning, Design and Construction,” Pequot Publishing Inc. (2003).

BIOGRAPHY

1.3.1.1 Graz Cycle – a Zero Emission Power Plant of Highest Efficiency



Franz Heitmeir

Institute for Thermal Turbomachinery
and Machine Dynamics
Graz University of Technology,
Graz, Austria

email: franz.heitmeir@tugraz.at
http://www.ttm.tugraz.at

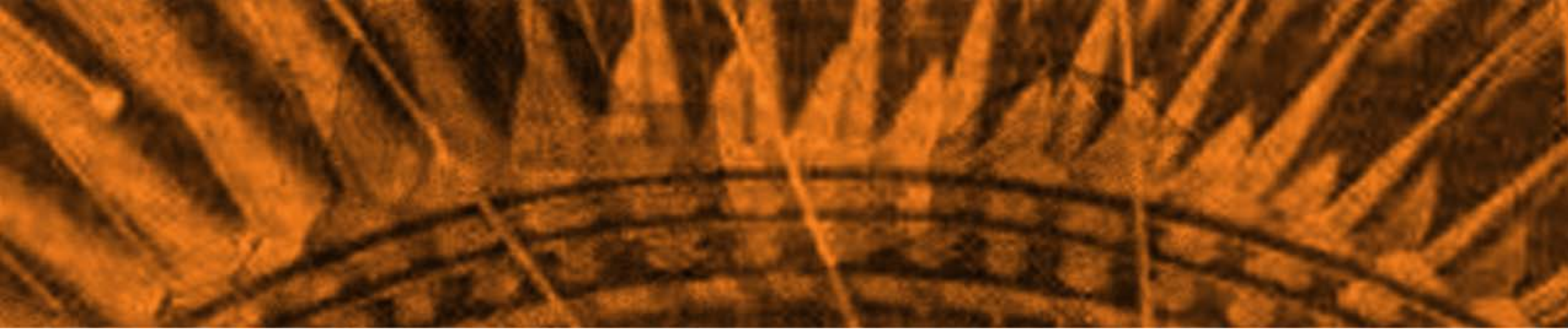
Professor Heitmeir studied Automotive and Airplane Engineering at the Munich College of Applied Engineering and Sciences (with excellence) and Mechanical Engineering and Aerospace Science at the Technical University Munich, Germany, (with excellence).

In 1987 he got his PhD in Mechanical Engineering at the University of the Armed Forces in Neubiberg, Germany, (with excellence). The Doctoral thesis was about the burning rates of graphite in high enthalpy flows.

From 1987 until 2001 he worked with MTU Munich, a leading gas turbine manufacturer in Germany.

At MTU he had a long career in different positions in the divisions research, development and testing as well as in the marketing and sales division. In his position he was head of the two engine programs RB 199 (Tornado fighter airplane) and MTR 390 (Tiger helicopter) and at the same time head of development departments for these two engine programs.

Since 2001 he has been head of the Institute for Thermal Turbomachinery and Machine Dynamics at the Graz University for Technology.



Wolfgang Sanz

Institute for Thermal Turbomachinery
and Machine Dynamics
Graz University of Technology,
Graz, Austria

http://www.ttm.tugraz.at

Professor Sanz studied Mechanical Engineering and Economics in Graz. In 1989 he got his Diploma in Mechanical Engineering (with excellence), and in 1993 his PhD in Mechanical Engineering (with excellence), both at Graz University of Technology. Since 1990 he started working as an assistant at the Institute for Thermal Turbomachinery and Machine Dynamics.

From 1994-1995, he was a visiting scientist at Naval Postgraduate School, Monterey, CA, where he worked with Max Platzer on unsteady aerodynamics.

In 1998 he made his habilitation for “Thermal Turbomachinery” and became associate professor at Graz University of Technology.

He is in charge of national funded projects on CFD and has published over 50 scientific papers, mainly on CFD and CO_2 retention. He is a member of the ASME and the Cycle Innovations Committee of the International Gas Turbine Institute IGTI.

Herbert Jericha

Institute for Thermal Turbomachinery and
Machine Dynamics
Graz University of Technology,
Graz, Austria

http://www.ttm.tugraz.at

Professor Jericha can look back over a working period of 51 years. His start as university assistant in 1954 allowed advanced studies in gas turbine technology at Farnborough, UK, leading to cooperation with the World Power Conference 1956 in Vienna. After his PhD in 1957 he worked in the US with Ingersoll Rand, where he, at that time, already worked on computer programs. Later at Elin Weiz, Austria, he became the leading designer of steam and gas turbines manufactured there.

In 1970 Graz University of Technology called him to lead the new founded Institute for Thermal Turbomachinery and Machine Dynamics. By invention, theoretical work, and the establishment of a unique laboratory, he made it known world wide. Repeated authorship in ASME conferences and ASME IGTI contributions as European coordinator and liaison chairman were the path to this success.

The most important design proposal is the so called Graz Cycle - a gas turbine system without any emissions.