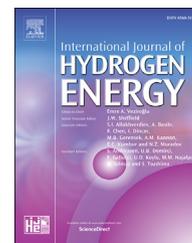


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# Adapting the zero-emission Graz Cycle for hydrogen combustion and investigation of its part load behavior

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## ARTICLE INFO

### Article history:

Received 12 September 2017

Received in revised form

9 January 2018

Accepted 26 January 2018

Available online 15 February 2018

### Keywords:

Graz Cycle

Hydrogen combustion

Power generation

Part load behavior

## ABSTRACT

A modern energy system based on renewable energy like wind and solar power inevitably needs a storage system to provide energy on demand. Hydrogen is a promising candidate for this task. For the re-conversion of the valuable fuel hydrogen to electricity a power plant of highest efficiency is needed.

In this work the Graz Cycle, a zero-emission power plant based on the oxy-fuel technology, is proposed for this role. The Graz Cycle originally burns fossil fuels with pure oxygen and offers efficiencies up to 65% due to the recompression of about half of the working fluid. The Graz Cycle is now adapted for hydrogen combustion with pure oxygen so that a working fluid of nearly pure steam is available. The changes in the thermodynamic layout are presented and discussed. The results show that the cycle is able to reach a net cycle efficiency based on LHV of 68.43% if the oxygen is supplied “freely” from hydrogen generation by electrolysis.

An additional parameter study shows the potential of the cycle for further improvements. The high efficiency of the Graz Cycle is also achieved by a close interaction of the components which makes part load operation more difficult. So in the second part of the paper strategies for part load operation are presented and investigated. The thermodynamic analysis predicts part load down to 30% of the base load at remarkably high efficiencies.

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## Introduction

In order to counteract the threatening climate change most countries regard it as virtually self-evident that they must concentrate on the development of the renewable energy resources within their national boundaries. However, there is a growing realization that the national resources are insufficient to achieve this objective. For example, MacKay [1]

showed quite convincingly that the United Kingdom cannot replace fossil-based energy generation without recourse to nuclear power generation or without importation of energy from the outside.

Germany came to a similar conclusion and therefore proposed to take advantage of the elevated solar power densities in North Africa and the Middle East by building concentrated solar power plants there and transmitting the electric energy via high voltage direct current cables. The technical challenges

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<https://doi.org/10.1016/j.ijhydene.2018.01.162>

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and the political instabilities in this region have impeded the implementation of this Desertec Initiative [2].

As an alternative in 2009 Platzer and Sarigul-Klijn [3] proposed a concept to exploit the global wind sources more easily. They suggest the use of sailing ships instead of stationary floating platforms so that the ships can be operated in areas of optimum wind conditions. In this concept the available wind power is converted into propulsive ship power which, in turn, is converted into electric power by means of ship-mounted hydropower generators. Hydrogen produced by electrolysis will be used for energy storage. In a number of papers Platzer et al. analyzed this concept in more detail, e.g. Refs. [4,5].

Due to the fluctuating nature of solar and wind power a storage system is also inevitable for land-based electricity generation from renewable energy in order to provide energy at the times of demand [6]. The limited storage potential of pumped hydroelectric storage, compressed air energy storage, flywheels and batteries, make Power-to-Gas (PtG) technology one promising option to overcome these limitations [7]. Surplus or intermittent power is used to produce hydrogen via water electrolysis. At demand, hydrogen can then be reconverted to electricity.

In order to find the optimum storage technology for electricity generated from renewable energy, in 2016 Walker et al. [8] compared Power-to-Gas with other energy storage technologies in applications ranging from residential load shifting to bulk energy storage and utility-scale frequency support. The authors found that Power-to-Gas is favorable for utility-scale energy storage where criteria such as energy portability, energy density and ability for seasonal storage are considered. PtG can provide significantly higher energy density than competing energy storage technologies. The ability of PtG for long-term storage of large amounts of energy led to studies of this concept for future renewable energy based systems in Great Britain [9], Germany [10] and Italy [11]. They showed that PtG is able to reduce the overall costs of the gas and electricity network and to improve system reliability in the case of large-scale use of renewable energy. E.g., in Great Britain electricity curtailment of 50–100 TWh in 2050 is possible without a large-scale storage technology.

Regarding electricity-to-electricity efficiency Walker et al. [8] showed that for bulk energy storage the storage efficiency of PtG is only about 35% at current technologies compared to pumped hydro with 82% and batteries ranging from 60 to 90% depending on the technology. But the batteries are far more expensive and do not have seasonal storage capability. In Ref. [12] a hybrid PtG-battery system was investigated with the result that batteries can support electrolyser operation but at too high costs.

So if hydrogen will be used as an energy storage system on a large scale, there is a need for highly efficient power plants for the re-conversion to electricity. In this sense Jericha et al. [13] proposed a hydrogen/oxygen fuelled steam power plant using fuel cells and gas turbine cycle components. The concept is based on the assumption that oxygen is provided “freely” together with hydrogen from the electrolysers. In their hybrid cycle about 20% of the net power output are generated by fuel cells, whereas the main output comes from the succeeding power plant. They predicted a net cycle

efficiency of 74% which is far above the efficiency of state-of-the-art combined cycle power plants of 60%. In Ref. [14] Platzer et al. analyzed the energy conversion chain of the energy ship concept combined with the hybrid cycle and predicted that 44% of the hydro-turbine electrical power can be regained. Other researchers also proposed novel fuel cell/gas turbine hybrid cycles with the goal of highest efficiency. So Evely et al. [15] combined the hybrid cycle with an organic Rankine cycle, Wang et al. [16] proposed the combination with a Kalina cycle and Meng et al. [17] used an additional supercritical CO<sub>2</sub> process. The achieved efficiencies varied between 64 and 70%.

But the realization of the hybrid cycle concept lacks – besides the development work needed for the turbomachinery components – the availability of fuel cells of large power output. Therefore, in this work a concept is presented which also additionally uses the oxygen from the electrolysis for the hydrogen combustion thus leading to a power cycle of remarkably high efficiency without the need for fuel cells. In contrast to Ref. [6] where the hydrogen/oxygen combustion takes place in an internal combustion engine, a power cycle based on turbomachinery technology is proposed.

This cycle is more or less the Graz Cycle, an oxy-fuel cycle for CO<sub>2</sub> capture which has been developed at Graz University of Technology since 1995 [18]. Since then many further thermodynamic studies as well as component developments have been published, e.g. Refs. [19–22]. It is based on the internal combustion of fossil fuels with oxygen so that a working fluid consisting mainly of steam and CO<sub>2</sub> is generated thus allowing an easy CO<sub>2</sub> separation by condensation. Net efficiencies of more than 65% were predicted when the efforts for oxygen generation and CO<sub>2</sub> compression were not considered [22].

In this work the Graz Cycle is adapted for hydrogen/oxygen combustion so that a working fluid of nearly pure steam is available. This can be considered as a return to its origin, when Jericha firstly proposed a high-temperature steam cycle with internal combustion of hydrogen and stoichiometric oxygen [23]. A thermodynamic layout of the cycle is presented resulting in a power balance promising highest efficiency. Then a variation of important cycle parameters is performed to study the sensitivity of the plant to parameter changes. A final investigation of the part load behavior will prove the applicability of a Graz Cycle plant in a future energy system based on renewable energy and hydrogen as storage medium.

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## Thermodynamic layout

All thermodynamic simulations were performed using the commercial software IPSEpro v7 by SIMTECH Simulation Technology [24]. This software allows implementing user-defined fluid properties to simulate the real gas properties of the cycle medium as well as to add new models to the model library as the hydrogen combustion chamber. The physical properties of water and steam are calculated using the IAPWS\_IF97 formulations.

Furthermore, a turbine module was developed for the calculation of cooled turbine stages. A simple stage-by-stage approach similar to Ref. [25] is assumed which allows calculating the amount of cooling steam needed per stage. The module assumes that half of the cooling mass flow is mixed to

the main flow at stage inlet, thus contributing to the stage expansion work. The rest is added at the stage exit.

Efficiencies and losses of the components of the power cycle as well as important parameters are listed in Table 1.

### Process description of a Graz Cycle plant for hydrogen combustion

Fig. 1 shows the principle flow scheme of the Graz Cycle plant for hydrogen combustion, and Fig. 2 the associated temperature-entropy (T-s) diagram generated by the software IPSEpro. The plant is based on a proposal by Jericha [18] and consists basically of a high-temperature Brayton cycle and a low-temperature Rankine cycle – a combined cycle. The Brayton part consists of the combustion chamber (CC), the high-temperature turbine (HTT) and the compressors (C1/C2). The Rankine steam loop consists of the heat recovery steam generator (HRSG), high-pressure steam turbine (HPT), low pressure steam turbine (LPT), condenser, condensate pump, deaerator and finally the feed pump supplying high pressure water to the HRSG.

In the following, the cycle will be explained in more detail. The flow sheet used for the thermodynamic simulation can be found in the appendix (Fig. 14) and gives mass flow, pressure, temperature and enthalpy of all streams.

**Table 1 – Component efficiencies and losses as well as important parameters used in the thermodynamic simulation.**

Fuel	Pure hydrogen
Hydrogen LHV	120,0 MJ/kg
Hydrogen HHV	146.8 MJ/kg
Oxygen purity	100%
Oxygen excess	0%
Fuel and oxygen supply temperature	15 °C
Combustor pressure loss	1.7 bar
Combustor heat loss	0.25% of heat input
Turbine inlet temperature	1500 °C
Turbine inlet pressure	40 bar
Turbine isentropic efficiency	HTT: 92%, HPT, LPT: 90%
Maximum turbine metal temperature	750 °C
Compressor isentropic efficiency	88%
Pump isentropic efficiency	70%
HRSG pressure loss: cold side	3.15% per heat exchanger = 29.6 bar in total
HRSG pressure loss: hot side	2.5% per heat exchanger = 12 kPa in total
Condenser pressure loss	3%
HRSG minimum temperature difference	Economizer: 5 K Superheater: 15 K
Condenser pressure	0.025 bar
HRSG heat loss	0.5% of transferred heat
Mechanical efficiency $\eta_m$	99.6% of net power
Generator efficiency $\eta_{gen}$	98.5%
Transformer efficiency $\eta_{tr}$	99.65%
Auxiliary losses $P_{aux}$	0.35% of heat input
Oxygen production	0.25 kWh/kg = 900 kJ/kg
Oxygen compression	2.4–42 bar: 325 kJ/kg

Pure hydrogen together with a stoichiometric mass flow of pure oxygen is fed to the combustion chamber, which is operated at a pressure of 40 bar. The high purity can be obtained by producing hydrogen and oxygen with electrolyzers supplied by electricity from renewable energy as discussed above. In order to obtain reasonable combustion temperatures steam stemming from the steam compressor as well as from the HPT are supplied to form the environment for the combustion process and to cool the burners and the liner. As experiments on oxygen combustion have shown (see below) an oxygen surplus of at least 3% is necessary for nearly complete fuel conversion. In this case a small amount of oxygen would accumulate in the cycle, which is extracted in the deaerator. In the simulation stoichiometric combustion is assumed.

Steam leaves the combustion chamber at a mean temperature of 1500 °C, a value achieved by H class turbines nowadays (point 1 in the T-s diagram of Fig. 2). The fluid is expanded to a pressure of 1.2 bar and 596 °C in the HTT (point 2). Cooling is performed with steam coming from the HPT at 41.7 bar/364 °C for the high pressure section and at 15 bar/240 °C for the low-pressure section (see dashed lines in Fig. 2). Cooling is assumed to an expansion temperature of 750 °C, leading to a cooling mass flow of 21.8% of the HTT inlet mass flow.

It is quite clear that a further expansion down to condenser pressure would not end at a reasonable condensation point, so that the hot exhaust steam is cooled in the following HRSG to vaporize and superheat steam for the HPT; the pinch point of the HRSG is 5 K at the economizer outlet, the approach point is 15 K at the superheater exit. The associated temperature-heat diagram (T-Q) is shown in Fig. 3; the transferred heat is 128 MW for a heat input of 300 MW (see below). But after the HRSG (point 3) only 52% of the steam mass flow at 150 °C are further expanded in the LPT, a typical condensing turbine. For a cooling water temperature of 10 °C the LPT exit and thus condenser pressure is 0.025 bar which corresponds to a condenser temperature of 21.1 °C. The steam quality at the LPT exit is 89% (point 4).

After the condensate pump excess water stemming from the combustion process is separated, before the water is degassed in the deaerator (point 5). It is then further compressed in the feed pump and delivered to the HRSG. After preheating, evaporation and superheating steam of 170 bar and 581 °C is fed to the HPT (point 6). After the expansion it is used to cool the burners and the HTT stages as described above.

Nearly half of the cycle steam - the return flow after the HRSG - is compressed using the main cycle compressors C1 and C2 with intercooler (see Fig. 14) and is fed to the combustion chamber with a temperature of 538 °C (point 7). Intercooling is performed to keep the compressor exit temperature at reasonable levels; its heat partially superheats the high pressure steam which causes the jump in the T-Q diagram of Fig. 3. The split ratio is mainly determined by the heat balance in the HRSG and the request of having superheated steam at the compressor inlet to avoid possible condensation there. The proposed return rate of 48% is found by an efficiency optimization. Considering reasonable boundaries for combustion pressure and temperature, condenser efficiency and pinch points, the optimum efficiency was found by a Design of

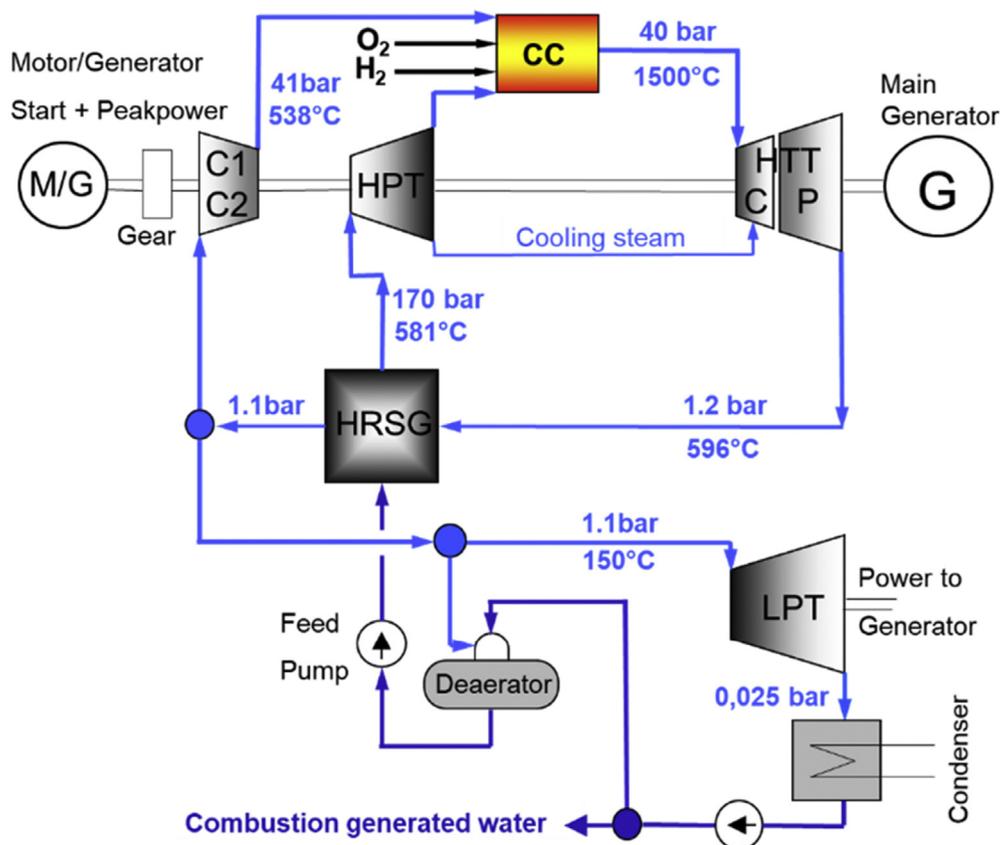


Fig. 1 – Principle flow scheme of the Graz Cycle for hydrogen/oxygen combustion.

Experiments (DoE) approach for the parameters condensation pressure, feed pump pressure, combustion pressure and temperature and pressure of compressor intercooling.

The cycle arrangement of the Graz Cycle offers several advantages: On one hand, it allows heat input at very high temperature, whereas on the other hand expansion takes place to vacuum conditions, so that a high thermal efficiency according to Carnot can be achieved. But the fact that only half of the steam in the cycle releases its heat of vaporization by condensation whereas the other half is compressed in the gaseous phase and so takes its high heat content back to the combustion chamber leads to the remarkably high efficiencies of a Graz cycle plant. But on the other hand the close integration of Brayton and Rankine cycle makes the operation more complex especially when part load is considered.

But the idea of a top Joule and a bottom Rankine cycle with the common working fluid steam was presented much earlier. Horlock [26] assigned them to the category of “doubly cyclic plants”. In the Field Cycle [27] evaporation is done by mixing with HTT exhaust gas. The complete mass flow is then compressed to the cycle peak pressure which leads to very high pressures at peak temperature if high efficiency is aimed at. Another process described by Horlock [26] is the Sonnenfeld Cycle, which can be considered as a supercritical Rankine cycle with three stages of reheat and an internal Joule cycle. Both cycles achieve efficiencies up to 55%. In order to use oxy-fuel technology for CO<sub>2</sub> separation, in 2002 Gabrielli and Singh [28] presented three cycles based also on the principle of a top Brayton and a bottom Rankine cycle. In contrast to the

Graz Cycle, steam evaporation takes place at compressor inlet pressure or at compression exit pressure at most. An additional high-pressure steam turbine as in the Graz Cycle does not exist. Efficiencies predicted are lower than for a Graz cycle plant. In 2007 Stankovic [29] presented a very intensive investigation of doubly cyclic plants also based on steam. Their common feature is again the addition of a recirculating steam compressor. His high temperature turbine operates at elevated steam turbine inlet conditions (900 °C, 300 bar), but does not apply gas turbine technology as the Graz Cycle. In the best case efficiencies up to modern combined cycle plants can be achieved. So although many efficient cycles have been proposed in the past based on the doubly cyclic concept, the Graz Cycle stands out by applying both state-of-the-art gas turbine and steam cycle technology at the same time leading to highest efficiency.

## Power balance

Table 2 gives the power balance of the hydrogen fuelled Graz Cycle plant for a heat input of 300 MW based on lower heating value. This corresponds to a hydrogen supply of 2.5 kg/s.

Whereas in a conventional combined cycle plant the compressor power is roughly half of the total turbine power, in the Graz Cycle plant it is only about a quart. This is a result of the compression of about half of the cycle mass flow in the liquid state. The HTT turbine, the counterpart of the gas turbine expander in a conventional combined cycle plant, is also

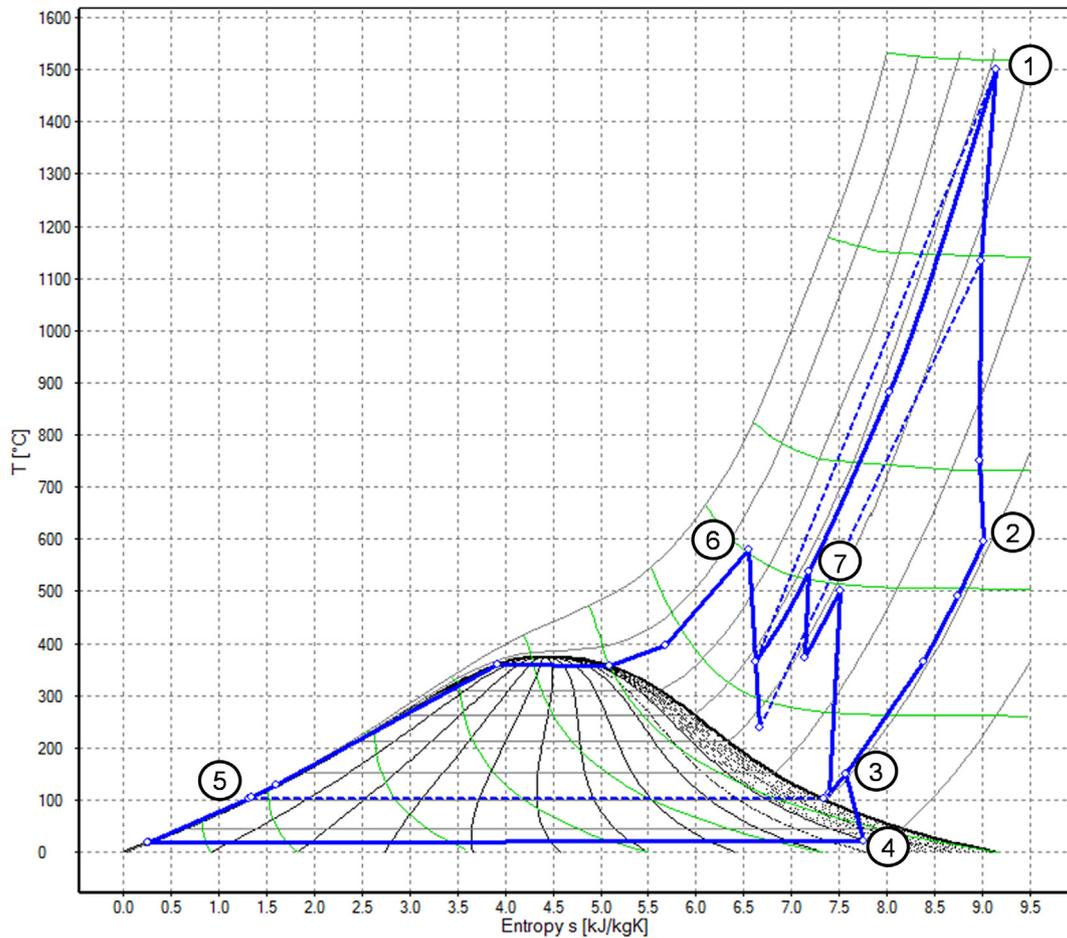


Fig. 2 – Temperature-entropy diagram of the Graz Cycle for hydrogen/oxygen combustion.

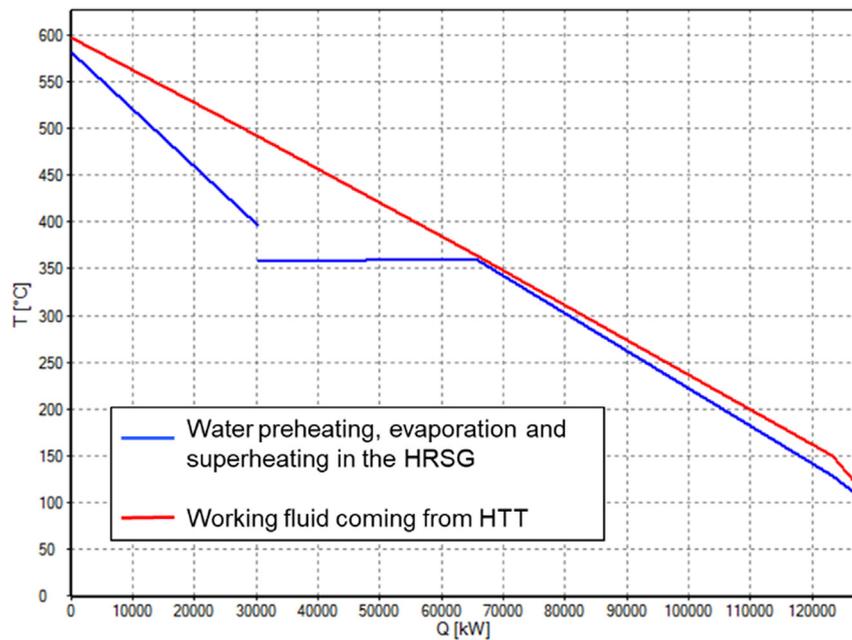


Fig. 3 – Temperature-heat diagram of the HRSG.

the dominant working machine generating more than 80% of the total turbine power. The favorable ratio of turbine to compression power leads to a remarkably high thermal efficiency of 70.35% based on LHV. If mechanical, electrical and

auxiliary losses are considered the net output is 205 MW which corresponds to a net cycle efficiency of 68.43% according to Eq. (1) which is far above the efficiency of state-of-the-art combined cycle plants.

**Table 2 – Graz Cycle power balance.**

HTT power [MW]	231.85
Total turbine power $P_T$ [MW]	283.80
Total compression power $P_C$ [MW]	72.75
Net shaft power [MW] without mechanical losses	211.05
Total heat input $Q_{zu}$ [MW]	300.0
Thermal cycle efficiency [%]	70.35
Electrical power output [MW] incl. mechanical, electrical & auxiliary losses	205.28
Net efficiency $\eta_{net}$ [%]	68.43
O2 generation & compression [MW]	24.30
Additional power by preheat of hydrogen and oxygen with ASU compressor intercooler to 150 °C [MW]	4.70
Net efficiency considering O2 supply $\eta_{O2,net}$ [%]	61.89

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

But one should be aware that the difference between higher and lower heating value is with 18% the highest for the fuel hydrogen. Compared to natural gas this leads to an increase in efficiency by 7%, if the efficiency definition is based on the lower heating value. The jump in efficiency compared to a fossil fuel fired Graz Cycle is also caused by this effect.

In this power balance, it was assumed that oxygen of high purity is available “free” from the hydrogen producing electrolysis. If this is not the case, oxygen has to be provided by an air separation unit (ASU). The oxygen demand of the presented plant is 1715 tons/day, which is the capacity of large ASU nowadays. A typical value for the oxygen generation is 0.25 kWh/kg. Since oxygen is usually provided at an elevated pressure, an effort for the intercooled compression from 2.4 to 42 bar of 325 kJ/kg is taken into account. This leads to a total power consumption of 24.3 MW for the oxygen supply. On the other hand, it was found that the heat of compression of ASU air and oxygen can be used to preheat the fuel, so that hydrogen and oxygen can be fed to the combustion chamber at a temperature of 150 °C resulting in a power increase of 4.7 MW. With these assumptions the net efficiency of the Graz Cycle plant is found with 61.89% if oxygen has to be generated on-site. This value is still above current plant efficiencies.

## Turbomachinery layout

Since the working fluid is pure steam, new turbomachinery components have to be developed for this plant. The high temperature turbine with steam cooling and steam as working fluid is expected to require the highest development efforts.

The combustion chamber for the nearly stoichiometric combustion of hydrogen with oxygen in a steam environment also demands development work. Experiments for oxy-fuel combustion in a steam vapor environment were performed for natural gas as fuel. Chorpeneing et al. [30] and Richards et al. [31] operated a combustion chamber for 1 MW thermal output at 10 bar and an exhaust temperature of 1200 °C. Oxygen was mixed with 80% total steam entering the primary combustion zone. 20% total steam entered through dilution ports downstream in order to operate the primary zone at higher

temperatures. Based on their CO measurements they suggest an oxygen surplus of 6%; and they concluded that the concept of oxy-fuel combustion using steam dilution is viable. Griffin et al. [32] also investigated oxy-fuel combustion of natural gas in an inert steam/CO<sub>2</sub> environment. They concluded that an excess amount of oxygen of 3% is necessary to ensure a nearly full fuel conversion. Experience can also be taken from Clean Energy Systems, Inc., who conducted combustion tests in a steam environment using a combustor can of a GE J79 engine [33].

The development work needed for the steam compressors and the HRSG are considered to be small. All other components are regarded as state-of-the-art. As Fig. 14 in the appendix shows, about one third of the cooling steam is expanded in a small steam turbine of 4 stages running at 20000 rpm. Omitting it would lead a decrease of efficiency of 0.65 %-points so that it is a matter of economics if it will be installed.

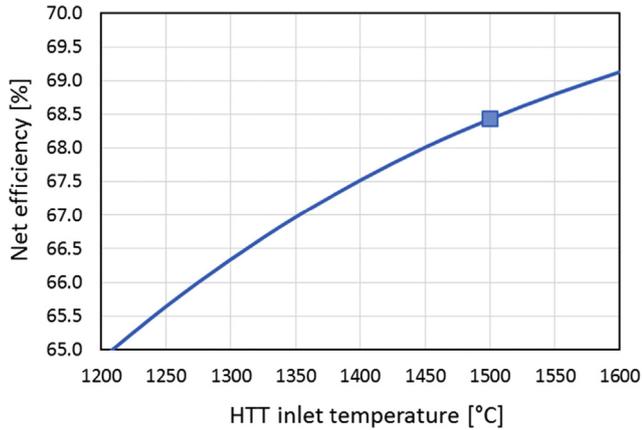
Since the heat capacity of steam is about twice the value of the working gas of an air-breathing gas turbine, the HTT and the steam compressors have to cope with a larger enthalpy drop for the same pressure ratio. In order to keep the number of stages low it was suggested in previous publications on the Graz Cycle (e.g. Refs. [21,22]) to arrange the compressors and the first two stages of the HTT on a fast-running shaft. The larger speed of sound of steam also allows a higher rotational speed without surpassing a relative tip Mach number of 1.3 at the compressor inlet. A faster speed is also advantageous for the HPT due to its relatively small volume flow. Therefore, it is suggested to group the compressors, the compressor turbine HTT-C and the HPT on a fast running shaft, which is connected via a gear box with a generator/motor indicated in the cycle scheme of Fig. 1. This electrical machine has to cope with the difference in power and can also be used as motor for start-up. The HTT power turbine and the LPT run at 3000 rpm and can be connected to the same generator.

## Parameter study

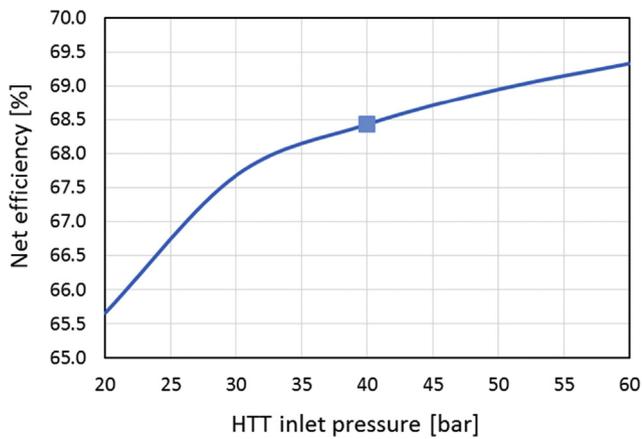
The main parameters of the plant are chosen to obtain high efficiency at realistic values. In order to see the chances for improving the process efficiency and the sensitivity of the cycle if design variables cannot be met a parameter study is conducted. HTT inlet temperature and pressure, condenser pressure and HPT feed pressure are varied to see their influence on the cycle net efficiency. In the following Figs. 4–7 the square marks the design value.

Fig. 4 shows the influence of the cycle peak temperature on the net efficiency, which is quite strong. Reducing the temperature to 1400 °C leads to a decrease in efficiency of nearly 1 %-point. A further reduction goes along with an increasing penalty in efficiency with a value of 64.9% at 1200 °C, about 3.5 %-points lower. If an increase to 1600 °C could be done, the net efficiency would slightly increase by 0.7 %-points.

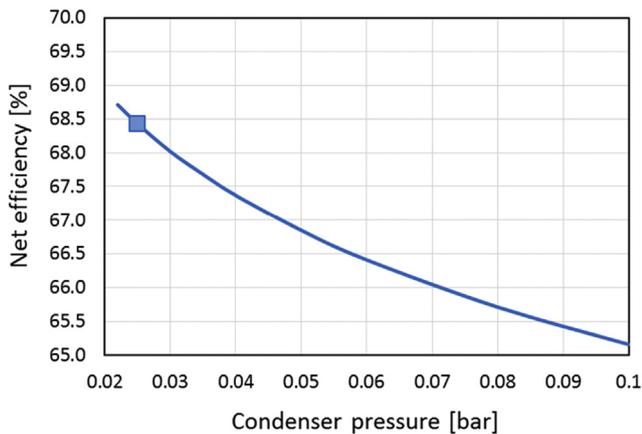
Increasing the HTT inlet pressure from 40 bar to 60 bar leads to a nearly linear increase in efficiency as shown in Fig. 5, with a value of 69.38% at 60 bar. The increasing efficiency indicates that the optimum cycle pressure has not yet been achieved. On the other hand, efficiency decreases considerably with decreasing pressure; a reduction by 20 bar leads to a loss in efficiency by 2.8 %-points.



**Fig. 4 – Net efficiency as a function of HTT inlet temperature.**



**Fig. 5 – Net efficiency as a function of HTT inlet pressure.**



**Fig. 6 – Net efficiency as a function of condenser pressure.**

The cycle reacts very sensible to a change in condenser pressure as shown by Fig. 6. Increasing the condenser pressure to 0.1 bar and thus the condensation temperature to 46 °C, which is typical for hot regions like India, reduces the cycle efficiency by 3.2 %-points to 65.3%.

The influence of the HPT feed pressure is relatively small. Varying it in a relatively wide range of 120–190 bar leads only to a moderate change in efficiency of 1.4 %-points as shown in Fig. 7.

## Part load performance

Since a Graz Cycle plant is characterized by a closer interaction between Brayton and Rankine part than a conventional combined cycle plant, the question arises to what extent a part-load operation is possible. Therefore, in the following the results of a part-load simulation are presented.

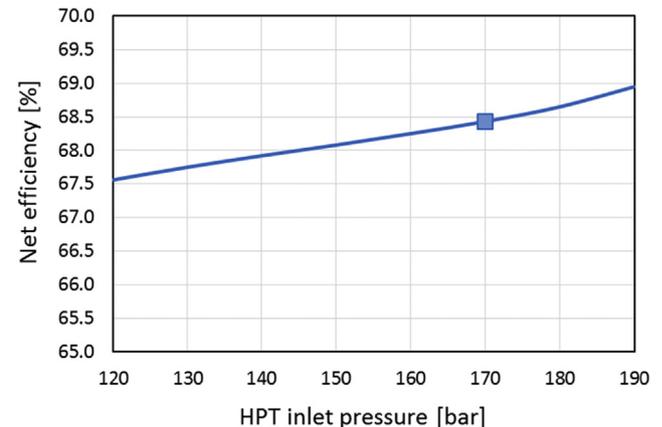
There are several ways to control the power output of a combined cycle plant in part load. First, mostly the mass flow in the gas turbine is reduced with the help of variable guide vanes at the compressor inlet. The peak temperature is kept nearly constant to maintain a high efficiency. In this case the gas turbine exhaust temperature also remains nearly constant which has a positive impact on the bottoming steam cycle. If the load is further decreased the peak temperature has to be reduced additionally. For the steam cycle the power is controlled by throttling the feed mass flow or by floating pressure operation which promises higher efficiency.

For the simulation of the part load behavior of the Graz Cycle plant, as a first guess the main cycle parameters, i.e. the mass flow to the HTT as well as HTT inlet pressure and temperature are varied simultaneously whereas a floating pressure control is assumed for the HPT. This control strategy is mostly dictated by the need to maintain reasonable temperature differences in the heat exchangers.

In order to simulate the part load behavior of the turbines and compressors Stodola's law [34] is applied which relates mass flow ( $\dot{m}$ ), inlet and outlet pressure ( $p_{in}$  and  $p_{out}$ ) and inlet temperature  $T_{in}$  between design point (DP) and part load (PL). Eq. (2) gives the formula for a turbine, for a compressor the pressure difference between outlet and inlet is used.

$$\frac{\dot{m}_{PL}}{\dot{m}_{DP}} = \frac{\sqrt{\frac{p_{in,PL}^2 - p_{out,PL}^2}{T_{in,PL}}}}{\sqrt{\frac{p_{in,DP}^2 - p_{out,DP}^2}{T_{in,DP}}}} \quad (2)$$

A simple relation between isentropic efficiency and mass flow as shown in Fig. 8 considers the drop in efficiency for the turbomachinery components in part load. For the heat exchangers the pressure loss is assumed proportional to the square of the mass flow according to Eq. (3). The change in heat transfer is considered by relating the heat transfer coefficient  $k$  to the transferred heat ( $q_{trans}$ ) according to Eq. (4).



**Fig. 7 – Net efficiency as a function of HPT feed pressure.**

$$\frac{\Delta p_{PL}}{\Delta p_{DP}} = \left( \frac{\dot{m}_{PL}}{\dot{m}_{DP}} \right)^2 \tag{3}$$

$$\frac{k_{PL}}{k_{DP}} = \left( \frac{q_{trans,PL}}{q_{trans,DP}} \right)^{0.6} \tag{4}$$

For the evaporator and superheater the full load model was used. All other parameters, i.e. combustor pressure and heat

loss, pump efficiencies, mechanical and electrical efficiencies as well as auxiliary losses were kept constant for simplicity reasons.

The simulations demonstrate that for the chosen operation strategy part load down to 30% of the base load can be achieved. Figs. 9–12 show the change of HTT mass flow, inlet temperature and pressure as well as HPT feed pressure over the load. The HTT inlet mass flow is decreased strongly down to about 91.5% at 75% load whereas the peak temperature is reduced more

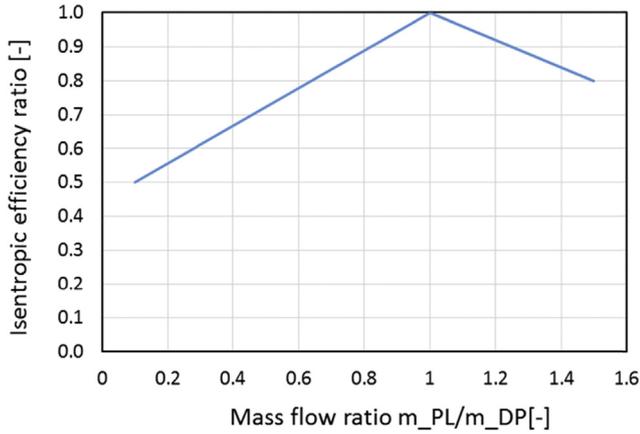


Fig. 8 – Isentropic efficiency ratio vs. mass flow ratio.

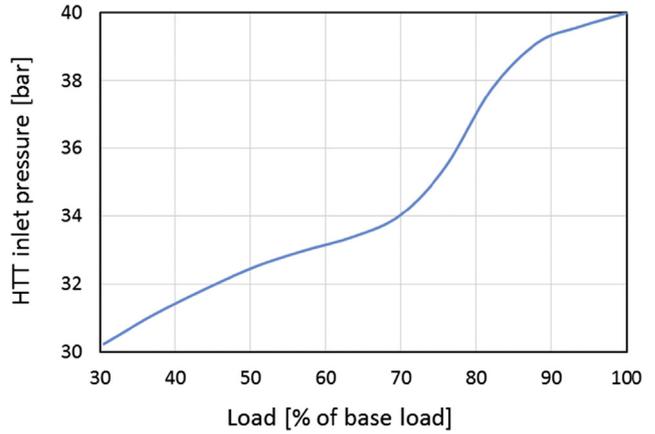


Fig. 11 – HTT inlet pressure at part load operation.

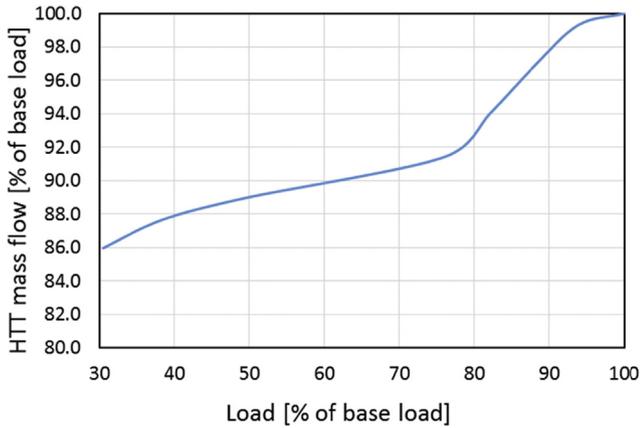


Fig. 9 – HTT inlet mass flow at part load operation.

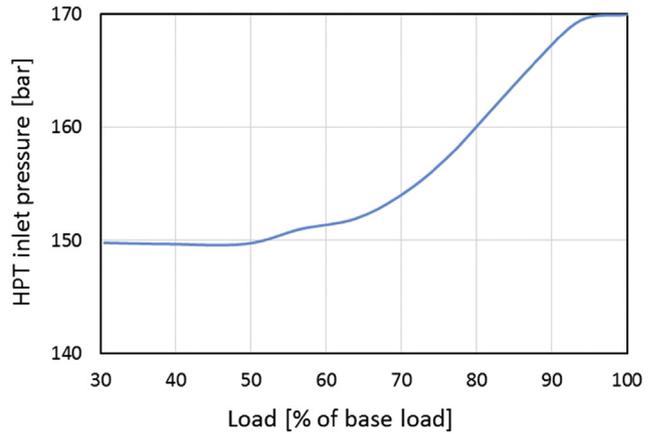


Fig. 12 – HPT inlet pressure at part load operation.

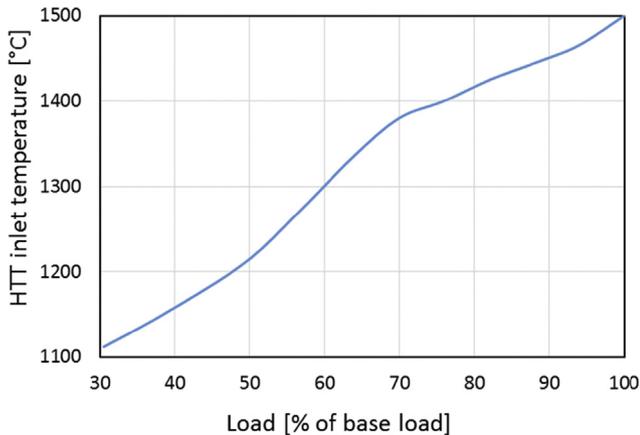


Fig. 10 – HTT inlet temperature at part load operation.

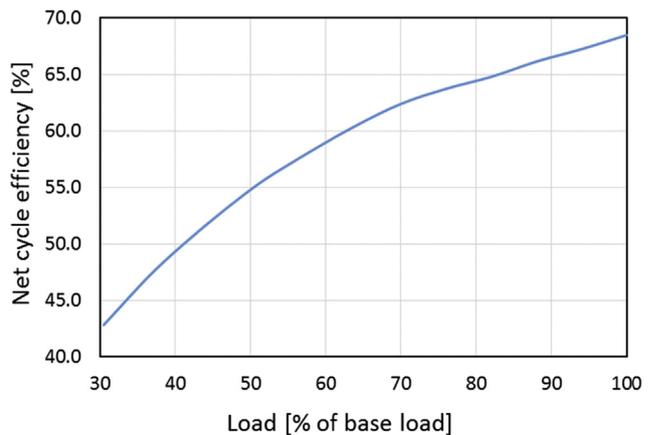


Fig. 13 – Net cycle efficiency at part load operation.

slightly to about 1400 °C. Then the control strategy is changed. The mass flow decline is less pronounced and it reduces slowly to 86% at 30% load, whereas the temperature drops significantly to 1100 °C in the same load interval.

The HTT inlet pressure follows more or less the tendency of the mass flow with a sharp decrease at the beginning down to 34 bar at 70% load. Then the pressure falls slowly to 30 bar at 30% load. The pressure of the HPT steam turbine decreases from 170 bar at base load to 150 bar at 50% load and then remains constant.

The resulting change of the net cycle efficiency is displayed in Fig. 13. There is a nearly linear decrease from 68.5% at 70% part load. Then the efficiency drops more substantially to 42.8% at 30% load. This behavior is mainly caused by peak temperature, which shows a similar trend and influences strongly the cycle efficiency.

The part load efficiency can be considered as remarkably high, which allows an economic operation of the Graz Cycle plant even at part load. This is a valuable feature since the fluctuating nature of renewable energy forces power plants more often to operate at part load. But it has to be kept in mind that this result is based on a relatively rough assumption of the part load behavior of the main components. A more thorough study with different control strategies can lead to a more negative, but also to a more positive part load behavior.

### Summary and conclusions

The Graz Cycle, a power plant of highest efficiency, is proposed for the energy conversion of hydrogen to electricity in a future energy system based on renewable energy.

The Graz Cycle in this work is based on the internal combustion of hydrogen with pure oxygen, so that a working fluid of nearly pure steam is obtained. The thermodynamic layout at the design point assumes state-of-the-art gas turbine technology with a peak cycle condition of 1500 °C and 40 bar. At design point the net cycle efficiency is 68.5% which is remarkably higher than the efficiency of modern power plants.

The high efficiency is obtained amongst others by the recompression of about half of the cycle fluid thus reducing the heat extraction out of the process. But this leads to a close interaction of the components so that the feasibility of part-load operation is studied.

For the proposed control system part load down to 30% of the base load could be achieved at remarkably high efficiencies. These high efficiencies at part and full load make the Graz Cycle to a promising candidate for the re-conversion of hydrogen in a future energy system based on hydrogen as storage medium.

### Appendix

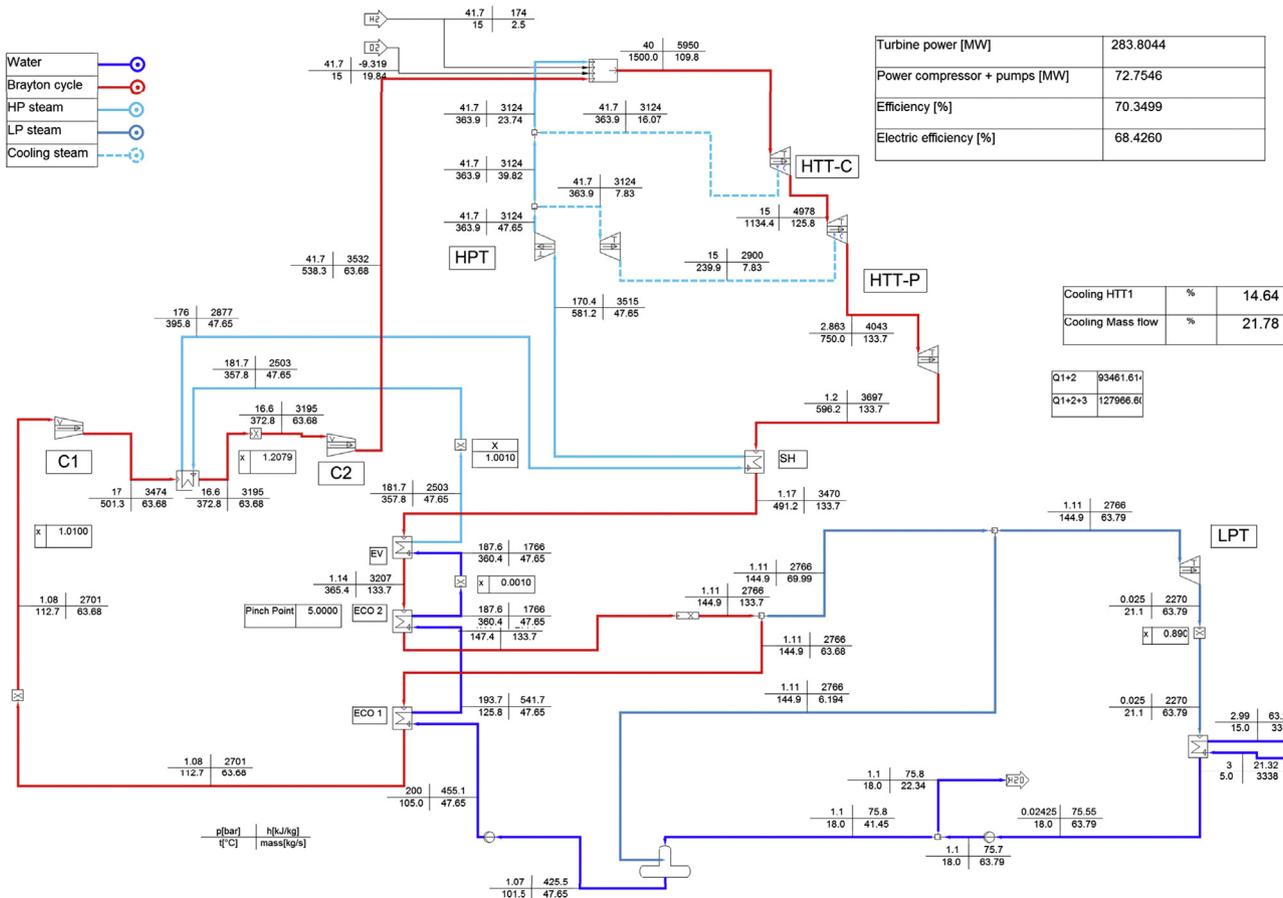


Fig. 14 – Detailed thermodynamic cycle data of the hydrogen/oxygen fuelled Graz Cycle plant.

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