Proceedings of ASME TURBO EXPO 2004
Power for Land, Sea and Air
June 14-17, 2004, Vienna, Austria

GT2004-53722

THERMODYNAMIC AND ECONOMIC INVESTIGATION OF AN IMPROVED GRAZ CYCLE POWER PLANT FOR CO_2 CAPTURE

W. Sanz, H. Jericha, M. Moser, F. Heitmeir

Inst. f. Thermal Turbomachinery and Machine Dynamics
Graz University of Technology, Graz, Austria
wolfgang.sanz@tugraz.at

ABSTRACT

Introduction of closed cycle gas turbines with their capability of retaining combustion generated CO_2 can offer a valuable contribution to the Kyoto goal and to future power generation. Therefore research and development at Graz University of Technology since the 90's has lead to the Graz Cycle, a zero emission power cycle of highest efficiency. It burns fossil fuels with pure oxygen which enables the cost-effective separation of the combustion CO_2 by condensation. The efforts for the oxygen supply in an air separation plant are partly compensated by cycle efficiencies far higher than 60 %.

In this work a further development, the S-Graz Cycle is presented, which works with a cycle fluid of high steam content. Thermodynamic investigations show efficiencies up to 70 % and a net efficiency of 60 % including the oxygen supply. For a 100 MW prototype plant the layout of the main turbomachinery is performed to show the feasibility of all components.

Finally, an economic analysis of a S-Graz Cycle power plant is performed showing very low CO_2 mitigation costs in the range of 10 \$/ton CO_2 captured, making this zero emission power plant a promising technology in the case of a future CO_2 tax.

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 has 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 EU there is a strong pressure on utilities and industry to reduce the CO_2 emissions by power generation. In 2003 the European Parliament passed a directive on emission trading. In 2005 emission allowances will be 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, estimates vary between 10 and 20 €/ton CO_2 by 2010 and even more by 2015 [1].

So there is a strong driving force to develop commercial solutions for the capture of CO_2 from power plants. The main technologies are [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
- chemical looping combustion
- oxy-fuel cycles with internal combustion of fossil fuels with pure oxygen

The authors believe that oxy-fuel cycles are a very promising technology and that their Graz Cycle can be the most economic solution for CO_2 capture from fossil power generation once the development of the new turbomachinery components needed are done. Oxygen needed in a large amount for this kind of cycles can be generated by air separation plants 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]. The equivalent amount of this oxygen could feed a Graz Cycle plant of 1300 MW.

The basic principle of the so-called Graz Cycle has been developed by Jericha in 1985 [4]. Improvements and further developments since then were presented at many conferences [5-10]. Any fossil fuel gas (preferable with low nitrogen content) is proposed to be combusted with oxygen so that mainly only the two combustion products CO_2 and H_2O are generated. The cycle medium of CO_2 and H_2O allows an easy and cost-effective CO_2 separation by condensation. Furthermore, the oxygen combustion enables power cycles which are far more efficient than current air-based cycles, thus largely compensating the additional efforts for oxygen production.

At the ASME IGTI conference 2003 in Atlanta a Graz Cycle power plant with a cycle efficiency of 63 % was presented and the general layout of all components for a 90 MW prototype plant were discussed [9]. In this paper this cycle is compared with a modified cycle scheme which promises efficiencies up to 70 % thus allowing to fully compensate the efforts of oxygen supply. The general layout of all turbomachinery components of this new Graz Cycle as well as the general arrangement with gears and electric generators is discussed. An economical investigation shows that this new Graz Cycle is a very cost-effective solution of a zero emission power plant which is worth to be pursued in the future.

CYCLE CONFIGURATIONS AND THERMODYNAMIC LAYOUT

All thermodynamic simulations were performed using the commercial software IPSEpro by SIMTECH Simulation Technology [11]. This software allows to implement user-defined fluid properties to simulate the real gas properties of the cycle medium. The physical properties of water and steam are calculated using the IAPWS_IF97 formulations [12], the CO_2 properties are calculated using [13]. Furthermore, a turbine module was developed for the calculation of cooled turbine stages. A simple stage-by-stage approach similar to [14] is assumed which allows to calculate the amount of cooling steam needed per stage.

Although the Graz Cycle is suited for all kinds of fossil fuels, for natural gas fuel it seems more economical to reform CH_4 to $CO + H_2$. Hydrogen can then be separated and burned in an air breathing gas turbine, a solution which reduces the oxygen requirements considerably. But in using oxygen blown coal gas as a fuel a Graz Cycle plant is most effective in retaining CO_2 and in use of oxygen. So the thermodynamic data presented are for a cycle fired with a typical fuel gas composition from an oxygen blown coal gasification plant (syngas mole fractions: 0.1 CO_2, 0.4 CO, 0.5 H_2).

The thermodynamic simulation is based on the following assumptions on efficiencies and losses: 1) the isentropic efficiency of turbines is 92% (90% for HPT); 2) the isentropic efficiency of CO_2 compressors is 90 % and of CO_2/ H_2O compressors 88%; 3) the mechanical efficiency of the turbomachinery is 99%; 4) the generator efficiency is 98.5%; 5) HRSG: cold side pressure loss is 5 bar; hot side pressure loss is neglected; 6) the pinch point of the heat exchangers is limited to 5°C; 7) the cooling water temperature in the condenser is 20° C; 8) fuel and oxygen is supplied at 40 bar; 9) CO_2 is released at 1 bar, efforts of a further compression to 100 bar (275 kJ/kg) is considered in the power balance; 10) the power consumption of oxygen production is 900 kJ/kg (0.25 kWh/kg) and of oxygen compression is 455 kJ/kg.

Graz Cycle

At first the flow scheme of the Graz Cycle as published in [8,9] will be presented before the modification leading to enhanced efficiency will be discussed.

Figure 1 shows the principle flow scheme with the main components and will be used to explain the main characteristics of this zero emission power cycle. Detailed cycle data for a 92 MW pilot plant, like mass flow, pressure, temperature, enthalpy or cycle fluid composition as well as the details of the thermodynamic simulation can be found in [9].

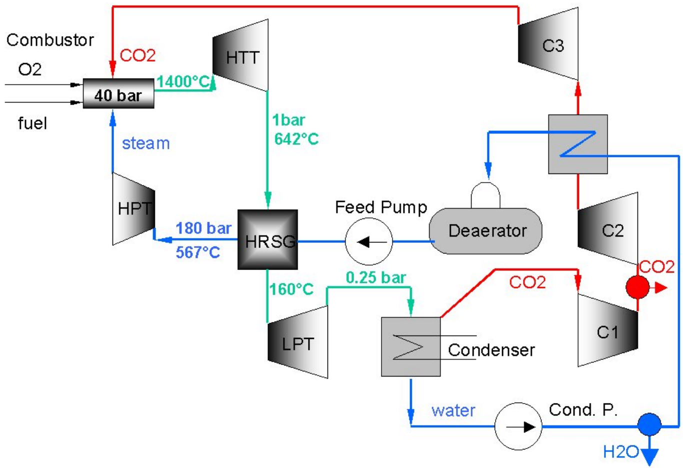

Fig. 1: Principle flow scheme of Graz Cycle power plant

Basically the Graz cycle consists of a high temperature Brayton cycle (compressors C2, C3, 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 stoichiometric mass flow of oxygen is fed to the combustion chamber, which is operated at a pressure of 40 bar. Steam as well as CO_2 is supplied to cool the burners and the liner. A mixture of about three quarters of CO_2 and one quarter of steam leaves the combustion chamber at a mean temperature of 1400°C. The fluid is expanded to a pressure of 1 bar and 642 °C in the HTT. Due to the addition of cooling steam in the HTT, the steam content in the cycle medium increases to 31 %. 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 vaporise and superheat steam for the HPT. Then it is further expanded in the LPT to a condenser pressure of 0.25 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. Assuming a minimum cooling water temperature of 20° C and condensation of 93 % of the water content results in a minimum condenser pressure of 0.25 bar due to the steam content of 31%.

In the condenser the separation of CO_2 and H_2O takes place by water condensation. The water is preheated and in the

HRSG vaporised and superheated while doing the necessary cooling of the HTT exhaust flow. The steam is then delivered to the HPT with 180 bar and 567 °C, after the expansion it is used to cool the burners and the first and second HTT stage. The CO_2 from the condenser is compressed to atmospheric pressure, the combustion CO_2 is then separated for further use or storage at 1 bar. The remaining CO_2 is compressed and fed to the combustion chamber to cool the liners.

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 till to vacuum conditions, so that a high thermal efficiency according to Carnot can be achieved. The dual medium CO_2 and H_2O results also in very low compression work. With both medium components in use over the full temperature range, the cycle gives this beneficial effect since only the gas CO_2 requires turbo compressors (C1, C2, C3) whereas feed water can be pumped to form high pressure steam. High pressure steam can be expanded to generate additional power in the high pressure turbine HPT, before being united with the CO_2 flow in the combustion chamber. Further beneficial effects are the possibility to create burner vortices and to cool the hottest nozzles and blades of the HTT first and second stage using the exhaust steam of the HPT which is of suitable pressure and temperature.

Table 1 gives an overview of the power balance of the Graz Cycle. For a 92 MW pilot plant the total turbine power adds up to 111 MW, 90 MW of it are supplied by the HTT [9]. On the other side the total compression power mostly used for CO_2 compression is 18.8 MW. Considering generator mechanical and electrical losses, an overall thermal cycle efficiency of 63.3 % can be evaluated. But considering the efforts for oxygen production and compression from atmosphere to combustion pressure results in an efficiency reduction to 55.0 %. For comparison a combined cycle with a single pressure steam cycle was calculated using the same main cycle data and assumptions. The resulting thermal efficiency of 53 % is below the Graz Cycle efficiency considering the efforts of oxygen production and compression. If CO_2 is compressed to 100 bar for liquefaction, the efficiency further reduces to 52.5 %.

Table 1: Graz Cycle Power Balance

Total turbine power [MW]	111
Total compression power [MW]	18.8
Net shaft power [MW]	92.2
Total heat input [MW]	143.4
Thermal cycle efficiency [%]	**64.3**
Electrical power output [MW]	90.4
Electrical cycle efficiency [%]	**63.3**
O_2 generation and compression [MW]	11.95
Net efficiency [%]	**55.0**
CO_2 compression to 100 bar [MW]	3.7
Net efficiency if CO_2 at 100 bar [%]	52.5

High Steam Content Graz Cycle (S-Graz Cycle)

The authors have tested various compositions of the cycle medium (about 20 % CO_2 and 80 % H_2O) in the publications of 1995 [5,6], where only approximate designs for turbomachinery arrangement, rotor and blading configuration were shown. In the following years intensive work on an innovative cooling system and a test stand for transonic turbine stages were done, so that in publications [7-9] an optimized compressor design and a first transonic turbine stage could be presented. These design features are associated with a cycle medium composition of about 75 % CO_2 and 25 % H_2O being on the other boundary of the admissible medium composition range.

The work presented here aims again for a much higher water content - now called "High **S**team Content **Graz Cycle**" or shortly "**S-Graz Cycle**". This cycle differs from the 1995 cycle [5,6] only by different cycle parameters, which result in a remarkable efficiency increase and promises a still more viable turbomachinery arrangement. When previous solutions are referred to in the following, the result of last years` ASME publications [8,9] are nominated as the **original Graz Cycle**.

Fig. 2 shows the principal flow scheme with the main components. Similar to the original Graz Cycle, the fuel together with the 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 62 % steam and 38 % CO_2 leaves the combustion chamber at the same mean temperature of 1400°C. The fluid is expanded to a pressure of 1 bar in the HTT. But the turbine exit temperature is now about 60°C lower due to the different cycle medium. It contains 66 % steam after the HTT. The hot exhaust gas is also used in the following HRSG to vaporise and superheat steam for the HPT, the pinch point of the HRSG is 9.4°C at the superheater exit.. But after the HRSG only 43 % of the cycle mass flow are further expanded in the LPT. The exit and thus condenser pressure is 0.085 bar and thus significantly lower than for the Graz Cycle, because of the higher steam content of cycle medium (see above).

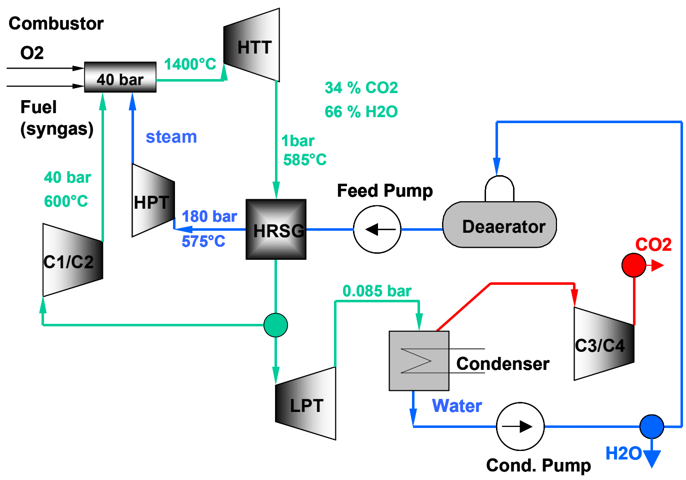

Fig. 2: Principle flow scheme of S-Graz Cycle power plant

CO_2 and steam are separated in the condenser. From there on the CO_2 mass flow, which is equal to the combustion CO_2, is compressed to atmosphere and supplied for further use or storage. After segregating the combustion H_2O, the water is preheated and in the HRSG vaporised and superheated. The steam is then delivered to the HPT with 180 bar and 575 °C, after the expansion it is used to cool the burners and the HTT stages.

The cooling mass flow for the HTT turbine is 12.6 % of the HTT inlet mass flow.

The major part of the cycle medium, which is separated after the HRSG, is compressed using an intercooled compressor 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 (Fig. 9) and gives mass flow, pressure, temperature and enthalpy of all streams.

Table 2 gives the power balance of the S-Graz Cycle for the same heat input as before. Turbine and compressor power are 150.4 MW and 50.5 MW, respectively, and thus are considerably higher than in the original Graz Cycle. The high compressor power of the S-Graz Cycle results from the compression of more than half of the cycle steam in the gaseous phase which is avoided in the Graz Cycle. The resulting net shaft power of 99.9 MW leads to a thermal cycle efficiency of 69.6 % and electrical cycle efficiency of 68.6 % including generator losses. If considering the efforts for oxygen production and compression from atmosphere to combustion pressure a net efficiency of 60.3 % can be evaluated which is nearly 6 percentage points higher than for the original Graz Cycle, higher than for state-of-the-art combined cycles and far beyond the efficiencies which are reported for other technologies of a zero emission power plant. If CO_2 is compressed to 100 bar for liquefaction, the efficiency further reduces to 57.7 %.

Table 2: S-Graz Cycle Power Balance

Total turbine power [MW]	150.4
Total compression power [MW]	50.5
Net shaft power [MW]	99.9
Total heat input [MW]	143.4
Thermal cycle efficiency [%]	**69.6**
Electrical power output [MW]	98.4
Electrical cycle efficiency [%]	**68.6**
O_2 generation and compression [MW]	11.95
Net efficiency [%]	**60.3**
CO_2 compression to 100 bar [MW]	3.7
Net efficiency if CO_2 at 100 bar [%]	57.7

The original Graz Cycle strives for minimum compression work which is achieved in so far as only the non-condensable gas CO_2 in its return flow to the combustion chamber is compressed by compressors. The second component liquid water is only mixed into the combustion chamber by way of evaporation, superheating and expansion in the HPT. The advantage of minimal compression work is counter-acted by the disadvantage of the higher heat input to the combustion chamber which is thus required.

The second solution, the S-Graz Cycle, uses a very high steam content, from which less than the half releases its heat of vaporization by condensation. The major part is compressed in the gaseous phase and takes its high heat content back to the combustion chamber. A second advantage of the high steam content is the lower possible condenser pressure for the same cooling medium temperature. This leads to a higher power output of the LPT despite of a much lower mass flow. Both features contribute to the remarkable efficiency increase compared to the original Graz Cycle.

Methane-fired S-Graz Cycle: If instead of syngas methane is fired in a S-Graz Cycle power plant, the cycle medium contains nearly 75 % H_2O at the combustion chamber exit. This different composition leads to an electrical cycle efficiency of 67.6 %, which is 1 percentage point below the syngas-fired version. Methane needs more oxygen per heat input to the combustion chamber, so that the effort for oxygen production and compression is 15.5 MW. This results in a net efficiency of 56.8 %, about 4.5 percentage points below the syngas-fired version. If CO_2 compression to 100 bar is considered, the efficiency further decreases to 55.3 %.

TURBOMACHINERY DESIGN

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 volume flow in the individual compressors and turbines require a multi-shaft arrangement connected by gears.

The design decision of having the high temperature flow channel with minimum surface and minimum heat loss and also with minimum cooling flow supply leads to the arrangement of turbomachinery for the original Graz Cycle as given in Fig. 3 which is intensively discussed in [8, 9]. Two overhang disks of different speed provide the shortest possible high temperature annular flow channel. So power end drive has to be on opposite sides. At 20 000 rpm HPT and C3 can be optimally connected. At the 12 000 rpm side main power is delivered via gears to the main generator. On the other side of the generator C1 and low pressure turbine LPT are arranged running at 3 000 rpm.

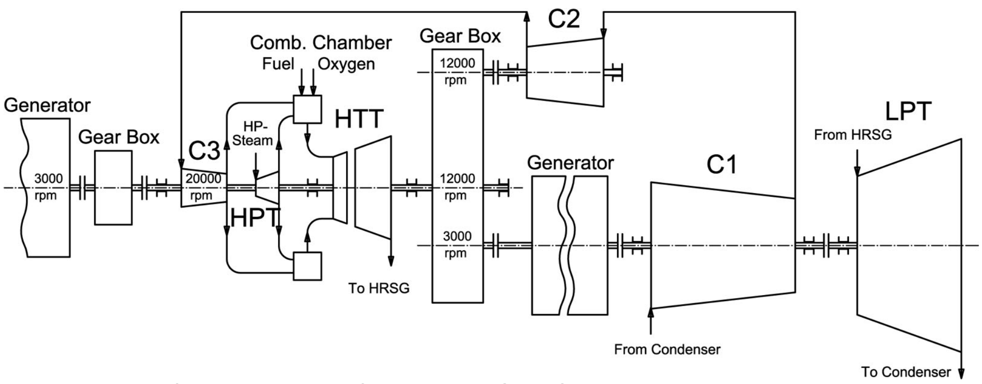

Fig. 3: Schematic arrangement of turbomachinery for a 92 MW Graz Cycle power plant

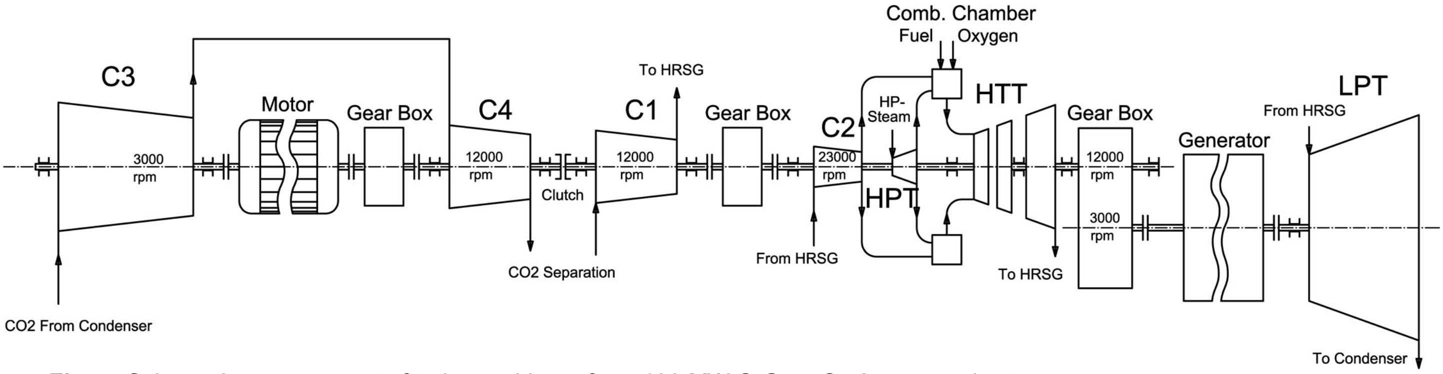

Fig. 4: Schematic arrangement of turbomachinery for a 100 MW S-Graz Cycle power plant

The power produced by HTT first stage and HPT greatly surpasses the power demand of C3, so a second electric generator is necessary. The only alternative would have been an outside gear shaft connecting to the main gear box and the main generator. The electric shaft proposed here seems to be a superior solution since for much larger units the same type of arrangement can be kept. Since gears with a power of about 100 MW are in successful operation in standard gas turbine units, the output of this Graz Cycle prototype unit could be doubled, still retaining high speed turbomachinery design.

The S-Graz Cycle differs from the original Graz Cycle in so far that only two compressors of high power (C1 and C2) are needed for the compression of the cycle medium and that the effort for the CO_2 delivery is much reduced. This leads to a different turbomachinery arrangement as shown in Fig. 4, which also includes the compressors C3 and C4 compressing the separated CO_2 to 100 bar.

The HTT turbine needs 4 instead of 3 stages due to the higher heat capacity of the now optimized cycle medium. Again the HTT is split into two shafts, where the first stage runs at 23 000 rpm, the other 3 stages at 12 000 rpm. A bearing is arranged between second and third stage. 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 the CO_2 delivering compressors C3 and C4 also. 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 quite similar to very large steam turbines.

Table 3 gives the main turbomachinery data and their dimensions for the S-Graz Cycle. Corresponding data for the original Graz Cycle are given in [9]. Due to the small volume 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 to affect cooling of the HTT first stage disk in a very 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

Table 3: Main turbomachinery data of S-Graz Cycle

Turbines: Total Turbine Power 150387 kW						
Turbine Name		HPT	HPT cool	HTT stage 1	HTT st. 2-4	LPT
m	kg/s	21.31	1.79	82.26	90.81	37.12
V_{inlet}	m³/s	0.416	0.117	12.37	30.78	81.05
V_{exit}	m³/s	1.401	0.349	30.78	298.74	589.21
P	kW	8553	508	40835	86919	13572
n	rpm	23000	23000	23000	12000	3000
z	-	4		1	3	4
$D_{m,inlet}$	m	0.190	-	0.389	0.554	1.29
L_{inlet}	m	0.02	-	0.050	0.118	0.27
$D_{m,exit}$	m	0.192	-	0.477	0.958	1.85
L_{exit}	m	0.02	-	0.066	0.272	0.58
Compressors and Pumps: Total. Power 50524 kW						
Compressor Name		C1	C2	C3	C4	Feed Pump
m	kg/s	52.34	52.34	14.24	14.24	21.31
V_{inlet}	m³/s	78.12	11.40	109.5	54.35	0.0221
V_{exit}	m³/s	12.49	6.56	67.81	10.71	0.0219
P	kW	27157	20466	608	1864	427
n	rpm	12000	23000	3000	12000	3000
z	-	9	7+1rad	8	4	
$D_{o,inlet}$	m	0.842	0.304	1.071	0.659	
L_{inlet}	m	0.168	0.053	0.214	0.132	
D_i/D_o	-	0.60	0.65	0.60	0.60	
$M_{rel\ tip}$	-	1	1	1	1.3	
$D_{o,exit}$	m	0.754	0.322	0.956	0.607	
L_{exit}	m	0.080	0.025	0.100	0.080	

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 last blades result in low clearance loss and low deterioration of 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 delivers the medium radially outwards making the inflow to the combustion chamber easier.

The number of stages of the LPT increased from 2 for the original Graz Cycle to 4 for the S-Graz Cycle, due to the lower condenser pressure and the higher heat capacity of the cycle medium.

Start-up of a S-Graz Cycle power plant

At start-up all machines are thought to be filled with steam from a foreign source fed into the turbomachinery high pressure side. All labyrinths are sealed with steam, the turning gears on both shafts are operating. The electric motor of C3 and C4 starts turning and creates vacuum in the condenser. Now the self-synchronizing clutch is coupled and thus the electric motor drives the high speed shaft. Steam flows throughout the interior of all turbomachinery and is condensed in the condenser. The feed water flow is started towards the feed water tank. This should go so far that the turbo-sets run out of turning gear and reach about 25 % nominal speed. Now oxygen and fuel gas can fed to the combustion chamber and ignition of the combustion chamber burner pilot flames is tried. Putting-in more fuel and oxygen should allow to run the machines up to speed. In case of flame out or ignition failure oxygen and fuel supply stop immediately, steam supply is restarted, compressors C3 and C4 are disconnected and repeat ventilating the interior of the combustion chamber and all turbomachinery.

When ignition is successful, the turbo-set will run up to higher speed and the generator can be synchronized. Compressors C3 and C4 are again connected to the high-speed shaft and start delivering non-condensable gases outside first to a gas cleaning unit and then into the CO_2 delivery line at 100 bar.

ECONOMIC EVALUATION

Despite the high efficiency and the positive impact on the environment of a S-Graz Cycle power plant, a future application of this technology and an erection of a power plant mainly depends on the economical balance. Therefore, an economic comparison with a high efficiency state-of-the-art combined cycle power plant is performed, whose economical data are reported in [15].

The economical analysis is based on following assumptions: 1) the yearly operating hours is assumed at 6500 hrs/yr; 2) the capital charge rate is 15%/yr; 3) fuel is either syngas or methane; 4) methane fuel costs are 1.3 ¢/kWh$_{th}$ according to [15]; 5) syngas is supplied by a syngas producer at 3.5 ¢/kWh$_{th}$, so no efficiency penalty for the production is considered; 5) an efficiency decrease of the reference plant due to syngas firing and a higher fuel compression effort for the Graz Cycle due to a higher combuster pressure are not considered (but these influences are discussed by the sensitivity analysis of Fig. 7 below); 6) the investment costs per kW are the same for the reference plant and the S-Graz Cycle plant; 7) additional investment costs are assumed for the air separation unit and for additional equipment (see Table 4, [16]); 8) additional investment and operating costs for the compression of CO_2 to 100 bar for liquefaction are considered alternatively (the effort for CO_2 compression is 275 kJ/kg CO_2); 9) the costs of CO2 transport and storage are not considered because they depend largely on the site of a power plant.

Table 4: Estimated investment costs

Component	Scale parameter		Specific costs
Reference Plant [15]			
Investment costs	Electric power	\$/kW$_{el}$	414
S-Graz Cycle Plant			
Investment costs	Electric power	\$/kW$_{el}$	414
Air separation unit [16]	O_2 mass flow	\$/(kg O_2/s)	1 500 000
Other costs (Piping, CO_2-Recirc.) [16]	CO_2 mass flow	\$/(kg CO_2/s)	100 000
CO_2-Compression system [16]	CO_2 mass flow	\$/(kg CO_2/s)	450 000

Three indicators characterizing the economical 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

Tables 5 and 6 show the result of the economical evaluation for syngas and methane firing, respectively. Compared to the reference plant, the capital costs are about 50 % higher only by considering the additional components for O_2 generation and CO_2 compression. The fuel costs have the major influence on the COE, especially for syngas firing. The S-Graz Cycle has lower fuel costs in most cases due to its high efficiency. The O&M costs are assumed 15 % higher for a S-Graz Cycle plant due to the operation of additional equipment.

Table 5: Economical data for syngas firing

	Reference plant [15]	**S-Graz Cycle**	**S-GC + CO_2 at 100 bar**
Reference Plant			
Plant capital costs [\$/kW$_{el}$]	414	414	414
Addit. capital costs [\$/kW$_{el}$]		148	209
CO_2 emitted [kg/kWh$_{el}$]	0.629	0.0	0.0
Net plant efficiency [%]	56.2	60.3	57.7
COE for plant amort. [¢/kWh$_{el}$]	0.96	1.3	1.44
COE due to fuel [¢/kWh$_{el}$]	6.22	5.8	6.06
COE due to O&M [¢/kWh$_{el}$]	0.7	0.8	0.8
Total COE [¢/kWh$_{el}$]	**7.88**	**7.9**	**8.30**
Comparison			
Differential COE [¢/kWh$_{el}$]		**0.02**	**0.42**
Mitigation costs [\$/ton CO_2 capt.]		**0.3**	**6.7**

Table 6: Economical data for methane firing

	Refer-ence plant [15]	S-Graz Cycle	S-GC + CO_2 at 100 bar
Reference Plant			
Plant capital costs [\$/kW$_{el}$]	414	414	414
Addit. capital costs [\$/kW$_{el}$]		177	221
CO_2 emitted [kg/kWh$_{el}$]	0.37	0.0	0.0
Net plant efficiency [%]	56.2	56.8	55.3
COE for plant amort. [¢/kWh$_{el}$]	0.96	1.36	1.47
COE due to fuel [¢/kWh$_{el}$]	2.31	2.29	2.35
COE due to O&M [¢/kWh$_{el}$]	0.7	0.8	0.8
Total COE [¢/kWh$_{el}$]	**3.97**	**4.45**	**4.62**
Comparison			
Differential COE [¢/kWh$_{el}$]		**0.48**	**0.65**
Mitigation costs [\$/ton CO_2 capt.]		**13.0**	**17.5**

Based on these assumptions, the COE of a syngas fired S-Graz Cycle plant is with 0.02 ¢/kWh$_{el}$ and 0.42 ¢/kWh$_{el}$, respectively, only slightly higher than for the reference plant. The mitigation costs are 0.3 \$/ton CO_2 and 6.7 \$/ton of CO_2 captured, if the CO_2 liquefaction is considered, and thus are far below the height of a CO_2 tax currently discussed.

If methane is used as fuel, the mitigation costs are higher because of the reduced CO_2 emission of methane (0.37 kg/kWh$_{el}$ compared to 0.629 kg/kWh$_{el}$), the higher need of oxygen per heat input and the lower efficiency of the S-Graz Cycle as discussed above. Assuming lower fuel costs as for the syngas cycle (1.3 to 3.5 ¢/kWh$_{el}$), the mitigation costs are 13.0 \$/ton CO_2 and 17.5 \$/ton of CO_2 captured, if the CO_2 liquefaction is considered. These values are still promising and show the potential of the S-Graz Cycle. But as discussed above, it has to be investigated if for methane the steam reforming and subsequent H_2 separation is the more economical option.

The results of the economic study depend mainly on the assumptions about investment costs, fuel costs and capital charge rate as well as on the choice of the reference plant. Therefore, a cost sensitivity analysis is performed in order to study the influence of the different parameters. The following Figs. 5-8 show the resulting changes of the mitigation costs for the S-Graz Cycle considering also the CO_2 liquefaction costs.

The capital charge rate considers the payback period and the capital interests. If it varies between 5 % and 25 %, the mitigation costs change from 1.5 to 12 \$/t CO_2 captured for the syngas fired S-Graz Cycle and from 8 to 27 \$/t CO_2 captured for the methane fired version (Fig. 5). If a CO_2 tax of at least 20 \$/ton is assumed, an investment in a syngas-fired S-Graz Cycle plant is reasonable even for a strongly varying capital charge rate.

The variation of the fuel costs (Fig. 6) between 1 and 10 ¢/kWh$_{th}$ does not strongly change the mitigation costs, because of the small efficiency difference between S-Graz Cycle and reference plant. Therefore, the investment decision is not influenced by the fuel costs. Because of higher electrical efficiency of the syngas-fired S-Graz Cycle compared to the reference plant, the mitigation costs decrease with increasing fuel costs.

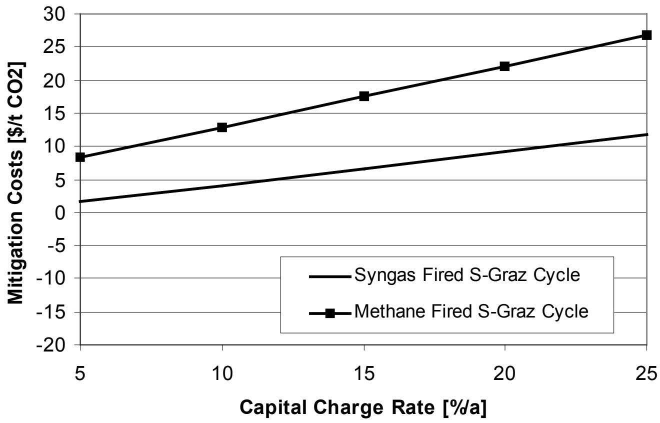

Fig. 5: Influence of capital charge rate on the mitigation costs (CO2 provided at 100 bar)

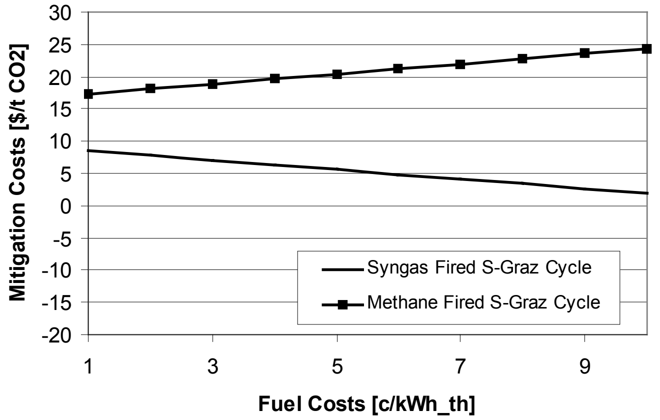

Fig. 6: Influence of fuel costs on the mitigation costs (CO2 provided at 100 bar)

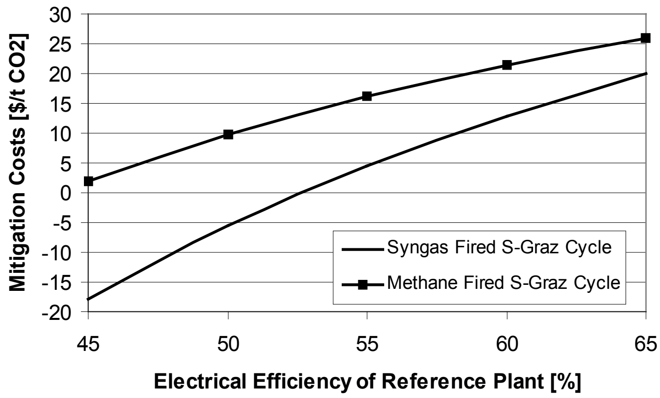

Fig. 7: Influence of reference plant efficiency on the mitigation costs (CO2 provided at 100 bar)

On the other hand the electrical efficiency of an alternative plant has a higher impact on the investment decision (Fig. 7). This parameter also covers the influence of an efficiency decrease of the reference plant due to syngas firing or of the S-Graz Cycle plant due to a higher fuel compression effort or component efficiencies lower than assumed. A reference plant of 60 % efficiency, which has just been achieved by a H-technology combined cycle plant, increases the mitigation costs to 13 \$/ton CO_2 captured for the syngas fired S-Graz Cycle and to 26 \$/ton CO_2 captured for the methane fired S-Graz Cycle.

It is rather difficult to estimate the capital costs for a S-Graz Cycle power plant due to some new turbomachinery components, i.e. the HTT and the fuel-oxygen combustion chamber. If only the investment costs for the air separation unit and the CO_2 compression is considered, the capital costs increase by approximately 50 %. For the syngas fired S-Graz Cycle investment costs twice as high as for a reference combined cycle plant lead to mitigation costs of 15 \$/ton CO_2, three times higher investment costs to 30 \$/ton CO_2 captured (see Fig. 8). Considering that currently a CO_2 tax of this height is discussed, then a syngas fired S-Graz Cycle power plant seems a reasonable investment even at remarkably higher capital costs compared to a conventional power plant. The economical situation is less promising for the methane fired S-Graz Cycle, double investment costs lead to mitigation costs of 30 \$/ton CO_2 captured.

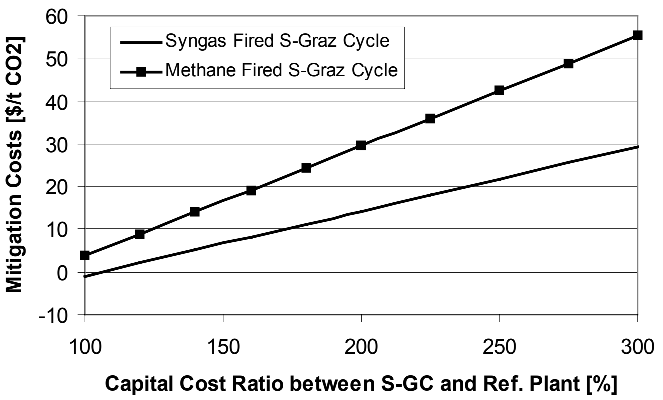

Fig. 8: Influence of capital costs on the mitigation costs (CO2 provided at 100 bar)

In these considerations about the height of additional investment costs, a further advantage of the S-Graz Cycle, the almost NOx-free combustion was not evaluated. According to [17] 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.

CONCLUSION

The Graz Cycle is an oxyfuel power cycle with the capability of retaining all the combustion generated CO_2 fur further use. In this work a further development, the S-Graz Cycle, has been presented, which works with a cycle fluid of higher steam content and which promises thermal efficiencies nearly up to 70 %. Even considering the costs of oxygen supply the net efficiency is in the range of most modern combined cycle power plants.

In the S-Graz Cycle about one third of the total turbine power is used for compression, so that a new arrangement of the turbomachinery components is proposed. The general layout of all components for a 100 MW prototype plant is presented verifying the feasibility of all components.

In an economical analysis a S-Graz Cycle power plant is compared with a reference plant. Due to the very high efficiency of the S-Graz Cycle, the CO_2 mitigation costs of the syngas fired version are less than 10 \$/ton CO_2 captured, so that the investment in such a zero emission power plant seems very reasonable considering stronger CO_2 taxation in the future.

ACKNOWLEDGMENTS

Support came from project Y57-TEC “Non-Intrusive Measurement in Turbomachinery”, founded by the Austrian Science Foundation (FWF, START-program, J. Woisetschläger).

NOMENCLATURE

Latin

D	[m]	diameter
L	[m]	blade length
$M_{rel,tip}$	[-]	relative tip Mach number
m	[kg/s]	mass flow
n	[rpm]	speed
P	[kW]	power
V	[m³/s]	volume flow
z	[-]	number of stages

Subscripts

o	outer, tip
m	mean
i	inner, hub

REFERENCES

[1] **Strömberg, L.,** 2003, “Overview of CO_2 Capture and Storage – Technology and Economics for Coal Based Power Generation”, VGB Congress 2003, Copenhagen

[2] **Gabbrielli, R., Singh, R.,** 2002, "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] **Turanskyj, L., Keenan, B.A.,** 2001, "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

[4] **Jericha, H.,** 1985, "Efficient Steam Cycles with Internal Combustion of Hydrogen and Stoichiometric Oxygen for Turbines and Piston Engines", CIMAC Conference Paper, Oslo, Norway

[5] **Jericha, H., Sanz, W., Woisetschläger, J, Fesharaki, M.,** 1995, " CO_2 - Retention Capability of CH_4/O_2 – Fired Graz Cycle", CIMAC Conference Paper, Interlaken, Switzerland

[6] **Jericha, H., Fesharaki, M.,** 1995, "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

[7] **Jericha, H., Lukasser, A., Gatterbauer, W.,** 2000, "Der “Graz Cycle” für Industriekraftwerke gefeuert mit Brenngasen aus Kohle- und Schwerölvergasung" (in German), VDI Berichte 1566, VDI Conference Essen, Germany

[8] **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

[9] **Jericha, H., Göttlich, E., Sanz,W., Heitmeir, F.,** 2003, "Design Optimisation fo the Graz Cycle Prototype Plant", ASME Paper 2003-GT-38120, ASME Turbo Expo 2003, Atlanta, USA

[10] **Heitmeir, F., Sanz,W., Göttlich, E., Jericha, H.,** 2003, "The Graz Cycle – A Zero Emission Power Plant of Highest Efficiency", XXXV Kraftwerkstechnisches Kolloquium, Dresden, Germany

[11] **SimTech Simulation Technology,** 2003, "IpsePro Overview", http://www.simtechnology.com/IPSEpro

[12] **Wagner, W., Kruse, A., "**Properties of Water and Steam", Springer-Verlag Berlin Heidelberg New York, 1998

[13] **Span, R., Wagner, W.,** 1996, "A New Equation of State for Carbon Dioxide Covering the Fluid Region from the Triple-Point Temperature to 1100 K at Pressures up to 800 MPa", Journal of Physical and Chemical Reference Data, Vol. 25, No. 6, pp. 1509-1596

[14] **Jordal, K., Bolland, O., Klang, A.,** 2003, "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

[15] **IEA,** 1999, "Assessment of Leading Technology Options for Abatement of CO_2 Emissions", IEA Greenhouse Gas R&D Programme and Stork Engineering Consultancy B.V.:

[16] **Göttlicher, G.,** 1999, "Energetik der Kohlendioxidrückhaltung in Kraftwerken" (in German), Fortschritt-Berichte VDI, Reihe 6, Energietechnik, Nr. 421. Düsseldorf: VDI Verlag

[17] **Gas Turbine World,** 2003, "2003 Handbook for Project Planning, Design and Construction", Pequot Publishing Inc.

APPENDIX

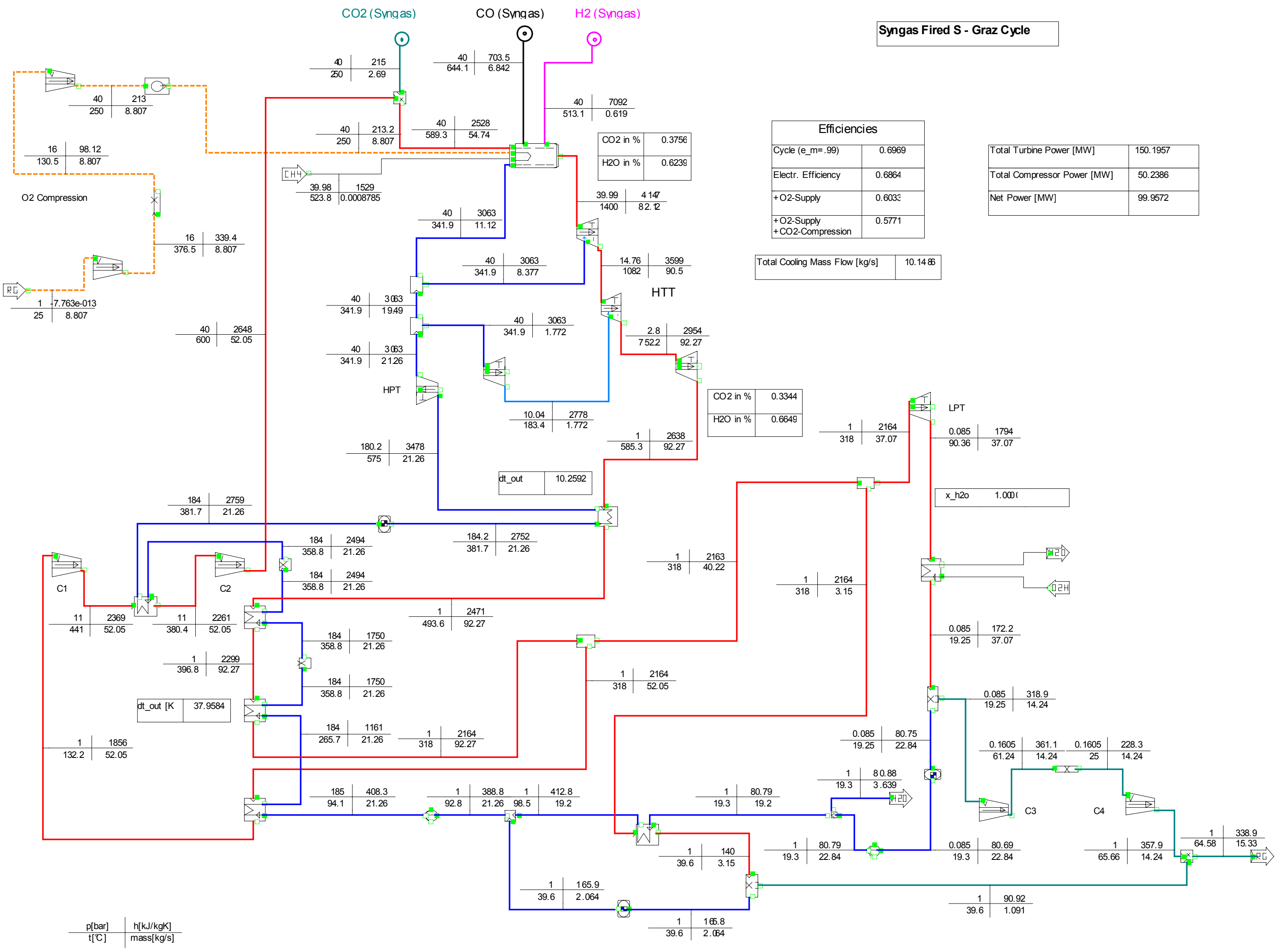

Fig. 9: Detailed thermodynamic cycle data of a 100 MW S-Graz Cycle Power Plant fired with syngas