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

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1 Abstract

About 60 % of the anthropogenic greenhouse effect is caused by CO_2 emissions, methane, nitrous oxide and chlorofluorocarbons (CFC) are responsible for the rest. 90 % of the CO_2 emissions originate by burning fossil fuels. According to that the power production is responsible for more than 50 % of the emitted greenhouse gases. In industrialized countries this contingent is much higher, e.g. Germany >80 %. Other responsible parties are industry, agriculture and fire clearings [17]. The reduction of CO_2 emissions during combustion of fossil fuels will play a major key role.The development of advanced fossil fuel power plants enabling CO_2 capture can help to reach the Kyoto goal. Introduction of oxy-fueled power generation within a closed cycle gas turbine provides the capability of retaining combustion generated CO_2.

Since the 90's research and development work of TTM Institute of Graz University of Technology has lead to the so called Graz Cycle, a zero emission power cycle of highest efficiency, which only uses well-established gas turbine technology enhanced by recent research results. In this work the design optimization for a prototype plant based on current technology as well as cutting-edge turbomachinery is presented. The next step should be the demonstration of operational capabilities in a prototype power plant which eventually leads to the design of much larger units with high reliability and thermal efficiency.

2 Introduction

The optimization of a thermal power plant starts with the optimization of the cycle scheme. The cycle has to be optimized regarding heat transfer, pressure losses, cooling demands and materials available.

Closed cycle power plants with their capability of capturing or retaining combustion generated CO_2 require novel cycle solutions and - depending on the concept - new components. A general comparison between different solutions of CO_2 retaining plants is given very detailed in [1]. Among them is the so-called Graz Cycle, which has been presented in several papers at previous conferences [2 - 13]. One advantage of the Graz Cycle is that most of the components are state of the art and need only further development. No completely new designed component is necessary.

The thermodynamic details of a prototype Graz Cycle plant of 92 MW power fired with oxygen blown coal fuel gas were presented to VDI in 2000 [10]. This cycle scheme is used in this paper as a baseline for the discussion of the turbomachinery optimization. A general layout of all components, especially turbomachines, combustion chamber and burners, and general arrangement with gears and electric generators was presented at ASME IGTI conference 2002 [13].

3 Survey of the Graz Cycle

Figure 1 shows the principle flow scheme of the Graz Cycle with the main components will be used to explain the main characteristics of this zero emission power cycle.
Detailed cycle data for a 92 MW pilot plant fired with gas from coal gasification, like mass flow, pressure, temperature, enthalpy or cycle fluid composition as well as the details of the thermodynamic simulation performed with the commercial code IPSEpro by SIMTECH Comp. can be found in [13].

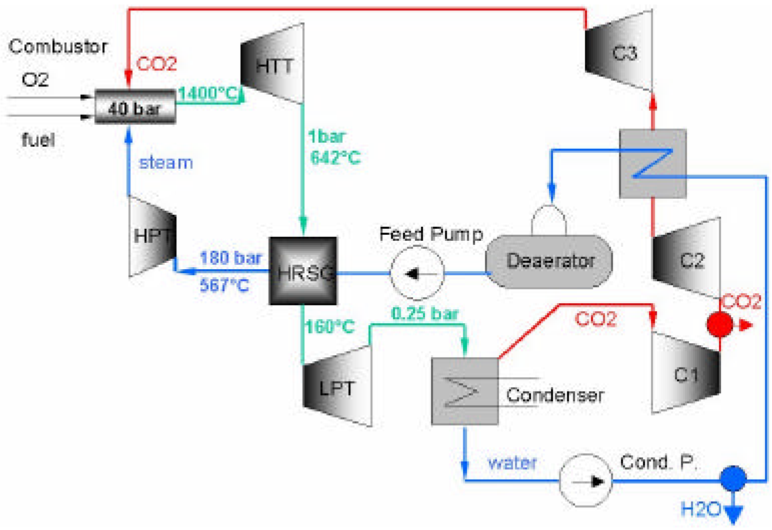

Fig. 1 Principle flow scheme of Graz Cycle

Basically the Graz cycle consists of a high temperature Brayton cycle (CO_2 compressors C2, C3, combustion chamber and High Temperature Turbine HTT) and a low temperature Rankine cycle (Low Pressure Turbine LPT, condenser, Heat Recovery Steam Turbine HRSG and High Pressure Turbine HPT).

In the layout presented a proposed fuel with a typical composition from oxygen blown coal gasification plant is taken, Table 2. 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 discharges from 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. The hot exhaust gas is used in the following Heat Recovery Steam Generator (HRSG) to vaporize and superheat H_2O for the HPT. Then it is further expanded in the LPT to a condenser pressure of 0.25 bar. In the condenser the separation of CO_2 and H_2O takes place by water condensation. The water is preheated, vaporized and superheated in the HRSG. The steam is then delivered to the HPT with 180 bar and 567 °C, after expansion it is used to cool the HTT and the burners. The CO_2 from the condenser is compressed to atmospheric pressure, the surplus amount of CO_2, resulting from the combustion is separated for further use or storage. The remaining CO_2 is compressed and fed to the combustion chamber to cool the liners.

The advantages of the Graz cycle are:
- allowing heat input at very high temperature,
- expansion into near vacuum, so that a high thermal efficiency can be achieved.
- the dual medium fluid (CO_2 and H_2O) has a very low compression work.
- with both components in use over the full temperature range, only the gaseous CO_2 requires turbo compressors (C1, C2, C3) whereas the feed water can be pumped in a liquid phase.

Further beneficial effects are the possibility to create burner vortices and to cool the thermally heavily loaded nozzles and blades of the HTT first stage with steam, because the exhaust steam of the HPT is of suitable pressure and temperature and can be passed through the hollow blades.

Table 1 Main turbomachinery data

Turbines Total Turbine Power 111081 kW							**Compressors and Pumps** Total Compr. Power 18830 kW						
Turbine Name		HPT	HPT cool	HTT hp	HTT lp	LPT	Compressor Name		C1	C2	C3	Cond. Pump	Feed Pump
m	kg/s	21.36	3.0	88.48	91.48	91.48	m	kg/s	66.33	52.48	51.15	25.16	21.36
p_1	bar	179.9	40.0	39.99	10.0	1.0	p_1	bar	0.25	1	2.7	0.25	5
p_2	bar	40.0	10.0	10.0	1.0	0.25	p_2	bar	1	2.7	40	5	180
t_1	°C	567.7	339.7	1312	1002	160	V_1	m^3/s	150.83	39.13	10.6o	0.0252	0.0233
t_2	°C	339.7	183.7	1042	642.4	63.4	V_2	m^3/s	49.46	17.43	1.169	0.0252	0.022
V_1	m^3/s	0.4125	0.195	9.264	31.50	106.67	P	kW	5498	3957	8953	12	410
V_2	m^3/s	1.387	0.590	30.65	226.05	331.71	n	rpm	3000	12000	20000	3000	3000
P	kW	8544	837	39374	51595	10731	z	-	7	5	7+1rad		
n	rpm	20000	20000	20000	12000	3000	D_{o1}	m	1.47	0.528	0.274		
z	-	1rad+ 2axi	1part. adm.	1	2	2	L_1	m	0.304	0.137	0.068		

D_{m1}	m	0.468	-	0.496	0.800	1.640	D_i/D_o	-	0.586	0.481	0.504		
L_1	m	0.01	-	0.064	0.170	0.338	M		1.00	1.31	1.39		
D_{m2}	m	0.227	-	0.510	0.880	1.640	D_{o2}	m	1.47	0.462	0.280		
L_2	m	0.027	-	0.070	0.250	0.513	L_2	m	0.084	0.071	0.014		

Table 1 shows the main turbomachinery data for a 92 MW pilot plant. Table 2 summarizes the power balance. It shows a total turbine power of 111 MW, from which 90 MW are supplied by the HTT. On the other side the total compression power, mostly used for CO_2 compression, is 18.8 MW. Considering component pressure losses as well as mechanical and electrical losses, an overall thermal cycle efficiency of 63.0 % can be evaluated which is significantly beyond most advanced combined cycle power plants of 60 %. But considering the efforts for oxygen production (0.25 kWh/kg O_2) the efficiency is reduced to 57.5 %, the effort for the oxygen compression from atmosphere to combustion pressure results in a net efficiency of 55 % (see Table 2). This efficiency penalty compared to combined cycle plants has to be balanced by the savings from a future tax on CO_2.

In a sensitivity analysis a comparison with a combined cycle power plant fired with Syngas from coal gasification (IGCCPP) with same power output was performed. An electricity selling price of 6 €-c/kWh and a Graz cycle net efficiency of 55 % were assumed. For the capital costs (similar erection costs, no costs for new developments, no costs for ASU), and the O&M costs the same order of magnitude between a conventional power station and a Graz Cycle power plant were assumed. Figure 2 shows the CO_2 avoidance costs in relation to the cycle efficiency of such an IGCC power plant. For example this means for an IGCC efficiency η = 58 % (excluding gasification), which is state of the art, that if the CO_2 avoidance costs are higher than 5.3 €/t the Graz Cycle is more economic. Assuming a CO_2 tax of 30 €/t CO_2 additional costs of 1.4 €-c/kWh could be covered, i.e. additional investment costa of 1500 €/kW (15 years x 7000 hrs)

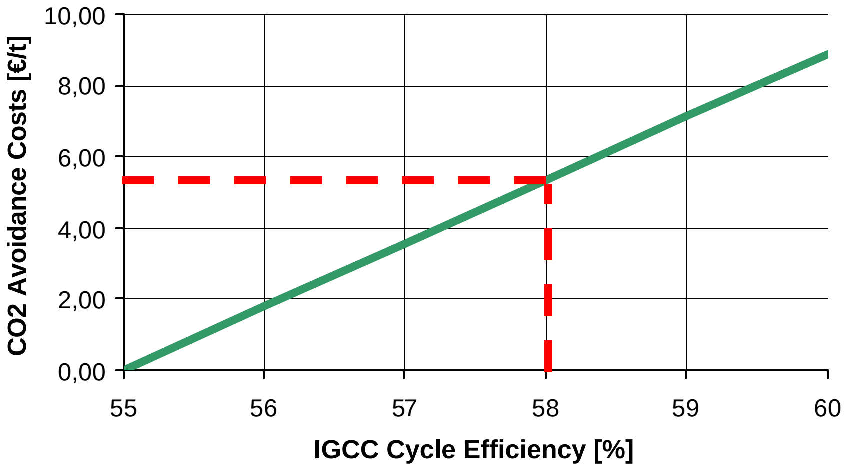

Fig. 2 Sensitivity analysis of CO2 avoidance costs vs. IGCC cycle efficiency

Table 2 Power Balance

Net power: 92251 kW
Total combustion heat input: 143342 kW
Mech. and generator efficiency: 0.98
Thermal cycle efficiency: 63.0 %
O_2 generation by air separation (0.25 kWh/kg): 7940 kW
O_2 compression (atmosphere to burner): 3440 kW
Net cycle efficiency: 55.0 %
Net Power: 111081 – 18830 = 92251 kW
Fuel gas mole fractions:
0.1 CO_2, 0.4 CO, 0.5 H_2, LHV: 14.121 MJ/kg

4 Design of the Components

Figure 2 shows the schematic arrangement of the turbomachinery components for a 92 MW pilot plant. Mass flow, power, speed, dimensions and number of stages of the turbo-machines are given in Table 1.

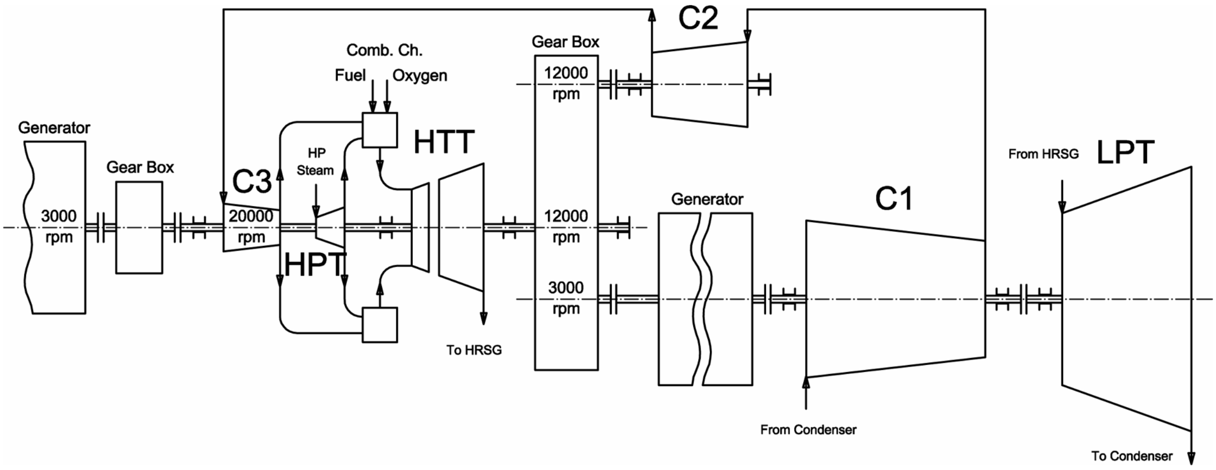

Fig. 3 Schematic arrangement of turbomachines

4.1 Compressor Design

The compressor concept mainly defines the speed requirements. After separation of the cycle media by condensation of water, compression of CO_2 is realized by three turbo compressors (C1, 3000 rpm, C2, 12000 rpm and C3, 20000 rpm). The necessity of different speeds is given by the compressibility of CO_2 because the volume flow changes from 150 m^3/s in the inlet of C1 to 1.17 m^3/s at the outlet of C3. The requirements for long first stage blades of C2 and C3 and a maximum admissible blade tip Mach number of 1.35 [15-16] lead to three different compressor speeds. This design allows achieving a reasonable last stage blade length. The last stage of C3 is built as radial stage with a radial diffuser for efficient transfer to the combustion chamber.

4.2 High Temperature Turbine HTT

The high temperature turbine is the most advanced component. Compared to an air-breathing gas turbine the cycle fluid has an 11 % smaller gas constant R and a 23 % larger heat capacity c_p. This results in nearly the same enthalpy drop for a given pressure ratio, but in higher temperatures, so that cooling is more important. The design features of the HTT make full use of the possibilities offered by the cycle. Even after some steam is fed into the Combustion chamber there is still sufficient steam of low temperature and suitable pressure available to cool the blades and vanes of the HTT.
The high pressure ratio of 40:1 in the HTT together with the strong volume change of the media results in a very high ratio of outlet volume flow to inlet volume flow. Therefore it is optimal to split the HTT and run it with two different speeds. The first stage runs with 20.000 rpm and is directly connected to HPT and C3 thus giving also optimal speed for the last CO_2 compressor, Fig. 3.

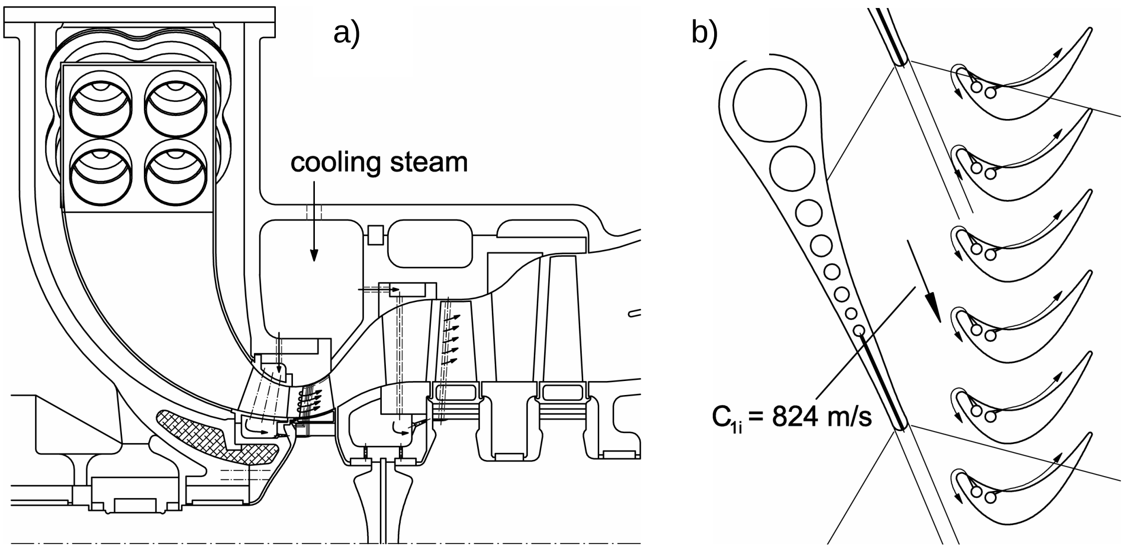

Fig 4 a) HTT cooling flow in two stages and b) ICS cooling of first rotor blades

4.3 Cooling of the High Temperature Turbine

The first stage of the HTT is a film cooled transonic stage cooled by steam, Fig. 4a. The film cooling is performed with an innovative cooling system (ICS) developed and patented for TU Graz [5-9]. The innovative feature is the use of the behavior of an underexpanded jet which means a jet from a choked convergent nozzle. This jet has the strong tendency to bend towards a convex surface like the leading edge of a turbine blade. In the present case steam of high total pressure is ejected partly against the main flow direction from slits of small height, Fig. 4b. Experimental investigations showed that the ICS film is able to cover the thermally heavily loaded leading edge safely and to provide a very good resistance to the trailing edge shocks of the previous vanes. Although the cooling medium is ejected only on the pressure side the films are able to cover the suction side as well, no further trailing edge cooling is necessary.

4.4 High Pressure Turbine HPT and Low Pressure Turbine LPT

A high speed back pressure turbine is proposed here. Various well proven designs are in use for similar tasks in industry.

The radial first stage shown in this paper is designed with the intention to keep the high speed shaft short and with a minimum number of bearings, Figs. 3 and 4.

A conventional low pressure turbine is proposed with two stages and axial outflow. The rotor has a common shaft with the first compressor C1, Fig. 3.

4.5 Combustion Chamber

In a conventional air breathing machine the fluid is oxidizer and cooling medium in one. Care has to be taken to avoid formation of nitrogen oxides and to attain high combustion efficiency at the same time. Within ignition limits any fuel particle coming close to the combustion chamber wall has still the chance to burn due to the high amount of oxygen in the "cooling medium".

The situation in the here proposed combustion chamber is different. Oxygen is provided in a stoichiometric ratio, the cooling medium is steam and CO_2. In this environment special focus has to be put on the complete combustion of the fuel without loss of reaction partners. Very low content of nitrogen together with operation in closed cycle mode alleviates the NO_X problem.

Combustors of such design are studied by several Japanese research institutions and companies (CRIEPI, NEDO; HITACHI, MHI), recent results were published by [14].

Careful research is done to obtain optimal combustion efficiency. Figure 5 shows the proposed design of burners and combustion chamber. At the burner inlet a conical head is providing slots ejecting sheet like jets of fuel gas and oxygen in close proximity. Steam is fed tangentially into the burner tube forming a strong steam vortex which is wrapped around the burner head. Thus providing high vorticity in the core for intimate mixing and ignition. At the same time it cools the burner tube at the inside and holds the flame together. At the exit into the free space after the end of it a vortex breakdown is expected. The low pressure in the center induces a strong backflow from the hot flame gases so that continuous ignition and complete combustion can be achieved.

At 6 locations around the annular combustion chamber 4 burner tubes in parallel are mounted injecting into the inner part of the flame casing. The high mixing involved should result in the desired high combustion efficiency. For each quadruple of burners an ignition flare and suitable crossovers are installed.
Cooling medium is introduced to the hot flame gases in conventional manner through slots and holes in the annular flow guiding insert. A strong rotation around the turbine axis is provided in order to offer additional flow path length for better mixture. The first stage nozzles have a lower flow turning angle and can thus save blade cooling medium.

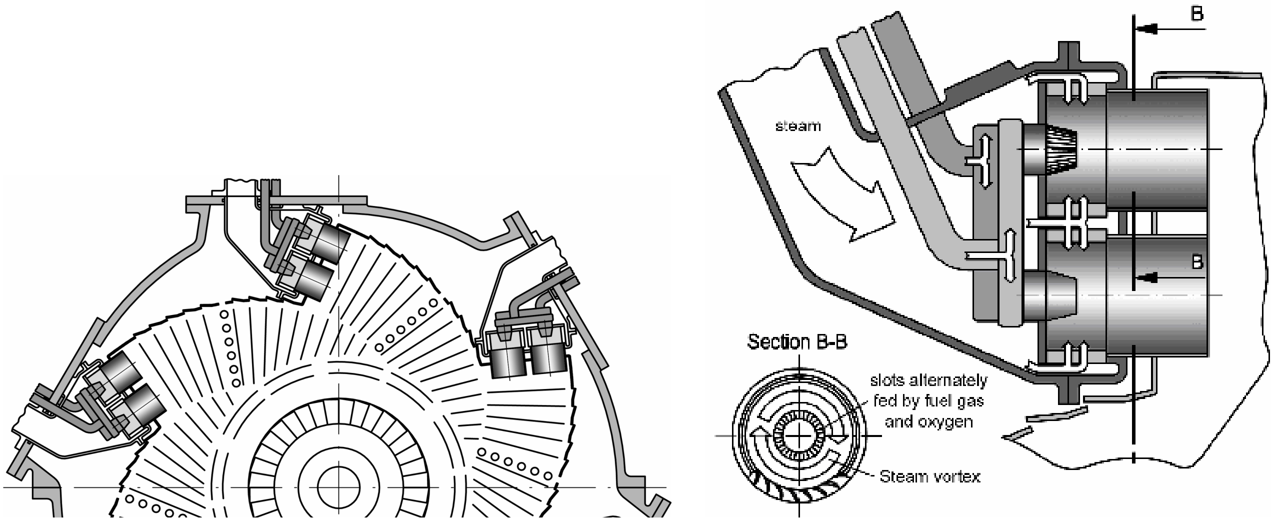

Fig. 5 Combustion chamber and burner details

5 Summary

The Graz Cycle is a novel type of a closed cycle gas turbine plant. It has the capability of retaining all the combustion generated CO_2. The Graz Cycle is based on the internal stoichiometric combustion of fossil fuels with oxygen and offers a very high efficiency. The cycle uses only turbomachinery components. A general layout for all the components was done. Out of this it has proved technically feasible so that a pilot power plant is a realistic next goal to verify the feasibility of the system in greater detail.

The next work at the Institute for Thermal Turbomachinery and Machine Dynamics is a more detailed investigation into the business case of such a zero emission power plant, which depends largely on the amount of a future CO_2 tax. So far it proves also economically reasonable that a Graz Cycle power plant could be put into operation within a few years based on the vast experience of successful gas turbine operation and research now well under way.

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