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Gasturbine with CO2 retention – 400 MW Oxyfuel-System Graz Cycle

Mr. Herbert Jericha, Graz University of Technology, Austria
Mr. Wolfgang Sanz, Graz University of Technology, Austria
Mr. Emil Goettlich, Graz University of Technology, Austria

Abstract: The oxy-fuel system Graz Cycle is an invention of members of the Institute for Thermal Turbomachinery and Machine Dynamics at Graz University of Technology in Austria. It was presented first in 1995 at CIMAC Interlaken and offers the possibility to avoid any CO2 emissions to atmosphere from a gas turbine design of highest efficiency. In several publications this most environmentally friendly solution for thermal power generation was presented to the international community and is today recognised as a most valuable contribution to the current discussion.

The paper presents the history of development, the most modern design for a very high power combined gas-steam cycle as well as details of the flow design and rotor dynamic of all turbomachinery components. The most advanced components are the first transonic stage of the C1 compressor and the transonic single stage of the compressor turbine.

INTRODUCTION

Today the necessity to counteract the rapid increase of CO_2 and other greenhouse gases is acknowledged by science worldwide. The European Union has installed a research platform ZEP (Zero Emission Power Plants) a year ago and has held its general assembly last September.

Its urgent proposals are:

- Keep global temperature rise - for all times in the future - below 2 °C.
- Complete the necessary research work in power production till 2010.
- Make operable CO_2 retaining power plants – in sizeable proportion till 2015.

Graz University of Technology has contributed to this goal intensively during the last 20 years. The lead author has presented to CIMAC Oslo 1985 [1] a power system based on the splitting of water by solar energy to be combusted in a gas-steam-turbine of highest efficiency. Also in 1995 the regress to fossil fuel to be combusted with pure oxygen was proposed at the conference CIMAC Interlaken creating thus the scheme of the Graz Cycle [2].

With the help of the co-authors of this paper in a row of further publications improvements were added step by step in the years between 1995 and 2006 [3-12]. Aiming at minimum compression power we obtained very high efficiency so that the paper presented at the ASME Turbo Expo 2003 in Atlanta obtained the best paper award [7]. Further efficiency improvement was made possible by raising the steam content of the working gas. Introducing separation of the combustion products at 1 bar and from there on recompression of working gas to combustion chamber, the condensation of the water content was made easier. This version of the Graz Cycle (S-Graz Cycle) was presented in papers to ASME Turbo Expo in Vienna 2004 and Reno 2005 [8, 9] and helped us in presenting our work in form of valuable discussions with industry partners. The respective patent of the authors Jericha and Sanz [13] is still upheld by the Institute for Thermal Turbomachinery and Machine Dynamics (TTM) and so in the legal property of Graz University of Technology. We offer this as a contribution to further work with industry. This large amount of research work culminates in the papers of 2006 which will be cited in detail - ASME Turbo Expo Barcelona [11] and VDI Conference in Leverkusen [12].

Three methods of CO2 capture are thus discussed internationally nowadays:

- Fossil fuel pre-combustion decarbonization to produce pure hydrogen or hydrogen enriched fuel for a power cycle (e.g. steam reforming of methane)
- Power cycles with post-combustion CO_2 capture (membrane separation, chemical separation, ...)
- Oxy-fuel power generation with internal combustion with pure oxygen and CO_2 /H_2O as working fluid enabling CO_2 separation by condensationThe authors believe that the Graz Cycle has taken the lead in terms of performance prediction as well as in terms of efficiency and operability. This is partly supported by recent studies performed within the ENCAP project [14].

DESCRIPTION OF THE GRAZ CYCLE

The simplified scheme of the Graz Cycle of Fig. 1 with the main cycle data as published in [11] shall bring out the salient features of the cycle. Details of the turbomachinery and shaft connections can be seen in Fig. 2. The detailed flow sheet used for the thermodynamic simulation is included in the appendix (Fig. 13) and gives mass flow, pressure, temperature and enthalpy of all streams. All assumptions on component losses and efficiencies agreed on with an industry partner are listed in [9]. The process is split into the high-temperature cycle and a separate low temperature condensation process. The high temperature part consists of High Temperature Turbine HTT, Heat Recovery Steam Generator HRSG, C1/C2 compressors and High Pressure Turbine HPT. Condensation of the working fluid in the 1 bar range is proposed in order to avoid the problems of a working fluid

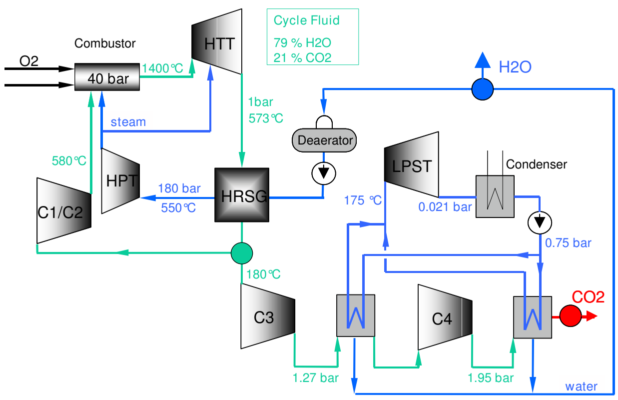

Figure 1 - Scheme of Graz Cycle

condenser at vacuum conditions. The heat content in the flow segregated after the HRSG for condensation is still quite high so re-evaporation and expansion in a bottoming cycle with the Low Pressure Steam Turbine LPST is mandatory.

A wide variety of fuels together with the nearly stoichiometric mass flow of oxygen can be 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 mainly steam and CO_2 leaves the combustion chamber at a mean temperature of 1400°C. The fluid is expanded to a pressure of 1.053 bar and 579°C in the HTT. Cooling is performed with steam coming from the HPT increasing the steam content to 79 % at the HTT exit. The hot exhaust gas is cooled in the following HRSG to vaporize and superheat steam for the HPT. After the HRSG about 55 % of the cycle mass flow are segregated and re-compressed to combustion chamber using the compressors C1 and C2 with intercooling.

The remaining mass flow is fed to a condenser, where the steam content is condensed allowing to separate the combustion generated CO_2. The condenser works at the same time as an evaporator providing steam for the bottoming cycle. The auxiliary compressors C3 and C4 aid the low pressure steam evaporation for the bottoming steam cycle by counteracting the effect of sinking H_2O partial pressure due to condensed water extraction from working fluid and thus allowing a reasonably high constant re-evaporation pressure of 0.75 bar for the bottoming steam cycle. A detailed heat – temperature diagram for this condensation/ evaporation process can be found in [11].

The advantage of the Graz Cycle over other proposals of oxy-fuel systems is the composition of the working gas. The high steam content is obtained by feeding high pressure steam from the HRSG via the HPT in the combustion chamber and into the cooling passages of the HTT. In the combustor the steam forms a vortex at each burner keeping the reactants fuel and O_2 close together and so lowering the flame temperature below values of dissociation. The low pressure core of each burner vortex is thus active to obtain a high combustion efficiency.

Feeding steam into the cooling passages of the HTT provides a very effective means of cooling by carefully cleaned steam thus avoiding any danger of deposition of particles in the serpentine passages. These are under a radial acceleration of up to 50.000 g in such a high speed gas turbine shaft. Of great advantage can be the use of the Innovative Cooling System ICS developed at the institute within the EU research project DITTUS [10].

The additional power generated by the HPT increases efficiency. Also the power from LPST is added to a high value of total turbine power. The fact that steam is directly fed to the combustor and the HTT cooling passages reduces the required total compression power greatly.

Power balance of 400 MW Graz Cycle Plant

Table 1 gives the power balance of the modified Graz Cycle plant of about 400 MW net power output as a result of the thermodynamic simulations described in [11]. Further data on power export and mass flow rates are found in Fig. 13 in the appendix:

Table 1 - Graz Cycle Power Balance

Total turbine power P_T [MW]	739.4
Total compression power P_C [MW]	234.7
Net shaft power [MW] without mechanical losses	504.7
Total heat input Q_{zu} [MW]	758.6
Thermal cycle efficiency [%]	66.52
Electrical power output [MW] incl. mechanical, electrical & auxiliary loss	490.7
Net electrical cycle efficiency [%]	64.68
O_2 generation & compression P_{O2} [MW]	74.7
Efficiency considering O_2 supply [%]	54.83
CO_2 compression to 100 bar P_{CO2} [MW]	13.0
Net power output [MW]	403.0
Net efficiency incl. environmental obligations[%]	53.12

This Power balance shows that an oxy-fuel plant needs a 490 MW turboset to deliver 400 MW to the electrical grid. As will be shown later on, this difference in power serves to collect CO_2 and to transport it, so that in the case of Enhanced Oil Recovery (EOR) a considerable economic gain can be achieved.

DESIGN OF GRAZ CYCLE COMPRESSORS AND TURBINES

The shaft arrangement of Fig. 2 has been elected according to the task of each turbomachine in terms of volume flow and required pressure difference. The HTT gas turbine is split into a compressor turbine HTTC connected with the two

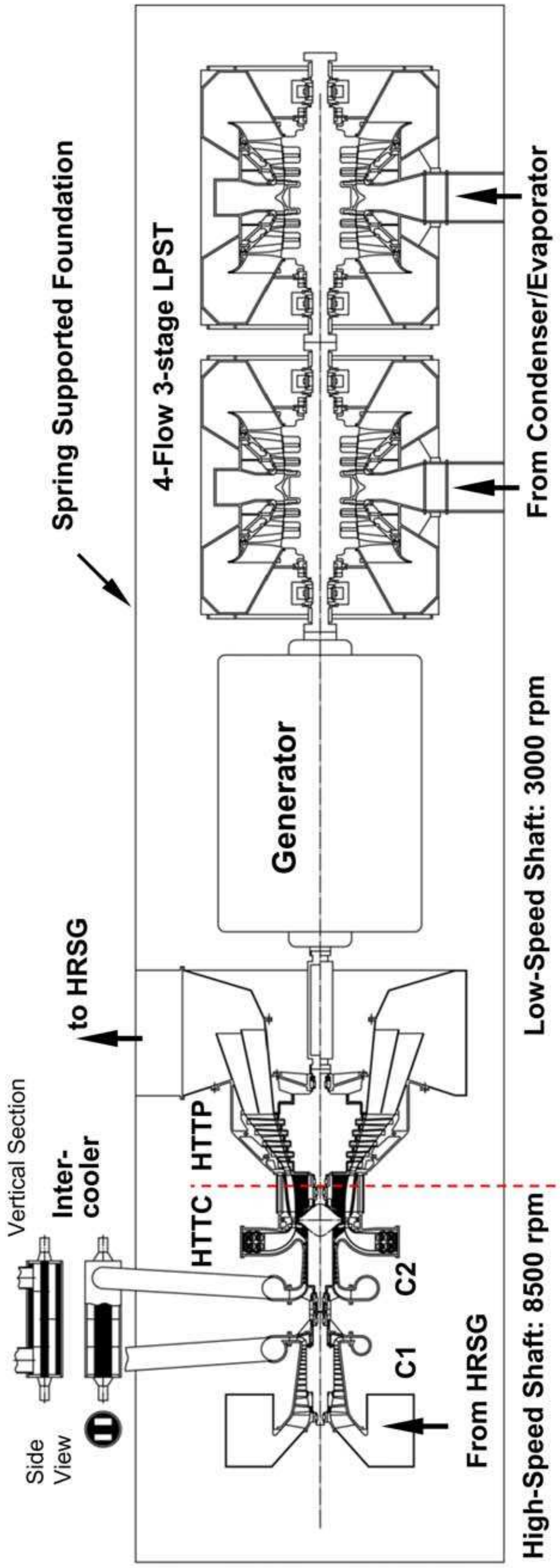

Figure 2 - Arrangement of main turboshaft with generator on foundation plate.

main compressors C1 and C2 and a power turbine HTTP. The compressor shaft is a very high speed shaft running at 8.500 rpm at full power, the total driving power is 214 MW. The first compressor stage is to be a transonic blisk and the single stage driving turbine is also a transonic stage with steam cooling developed at the institute. The power turbine HTTP and the intermediate bearings arranged in the same casing allow building a high efficiency diffuser for crossover of the high temperature flow to the power shaft. The five-stages 3.000 rpm power turbine is coupled to the main generator, on the other side of which the condensing steam turbine LPST is arranged on the common spring supported foundation plate.

The heat recovery steam generator is thought to be arranged on the upper side in parallel to this foundation plate on solid ground together with the intercooler between compressors C1 and C2 serving for additional heat input to the flow of superheated steam to the high pressure steam turbine HPT. The HPT is designed as a standard high-pressure high-speed steam turbine driving a separate generator to add its power to the electricity delivered. It is also arranged sidewise to the main power shafts. Since several standard designs for this task are available on the turbine market it is not shown in Fig. 2. Just the inflow from the HRSG and the flow delivery to combustor and HTT casing is indicated.

The high speed design of the compressor shaft was selected because power input, mass flow and volume flow are much lower than in a comparable air-breathing gas turbine. The number of stages in the compressors can be kept at a minimum and the last blade length of C2 is kept at a maximum even at the relatively high combustor pressure of 40 bar. The speed is limited by the admissible Mach Numbers of first compressor stage and driving transonic single stage. Fig. 3 shows the high speed shaft consisting of compressors C1 and C2 and compressor turbine HTTC. Radial outflow to intercooler and radial inflow serve to improve flow condition in compressor stages. The arrangement of four bearings serves for vibration damping.

The flow design for the first stage of C1 was obtained by development work at University of Darmstadt [15], where intensive numerical calculation and flow tests were done leading to an optimised blade design. The test rotor there is scaled up to form the blisk (outer radius 0,7 m, tip velocity 632 m/s, relative tip Mach number of 1,36). All the blades of C1 are carried on a solid drum shaft with the Titanium blisk at the entry and a Nimonic radial wheel at the high pressure exit.

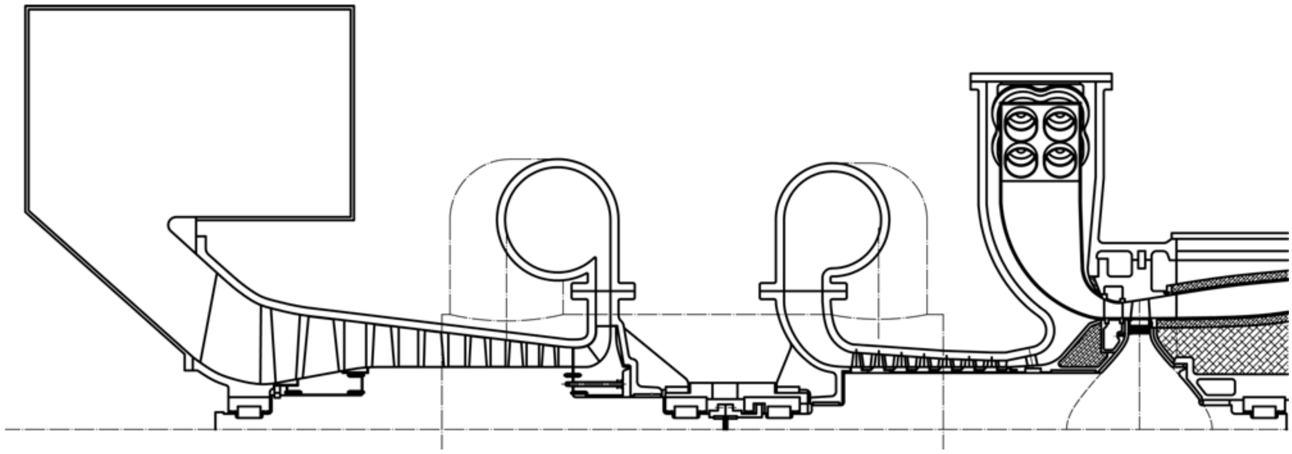

Figure 3 - Design of high speed shaft consisting of compressors C1 and C2 and compressor turbine HTTC

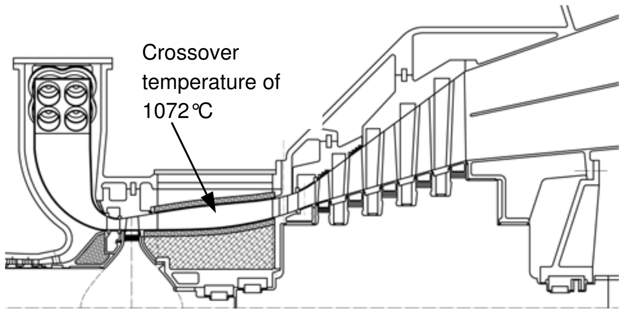

Figure 4 - Compressor turbine HTTC (8500 rpm) and crossover to power turbine HTTP (50 Hz).

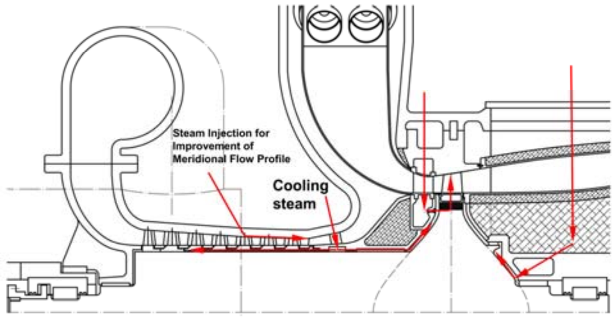

Figure 5 - Details of transonic stage of HTTC with steam cooling by ICS.

Compressor C2 and HTTC blading are also on a drum like solid shaft. In the high temperature region 300°C temperature steam cools all accessible surface areas. The disk carrying the transonic blading of HTTC is built according to a disk of constant stress [16].

Fig. 4 shows the compressor turbine HTTC running at 8500 rpm and the crossover to power turbine HTTP running at 3000 rpm. Intermediate bearings are arranged in a common bearing casing which is supported on main casing ribs. Temperature is shielded by metallic insulation. Proper alignment is assured by assembly of the horizontally split main casing. Crossover conditions are 13.78 bar and 1072°C.

The blade arrangement shown ensures high flow efficiency in the blades of the HTTP, which are also steam cooled. Thrust equalisation is done by a balance piston with steam on the outlet side. The intensive steam cooling of all blade roots allows to build the rotor also in solid drum design.

The innovative cooling system developed in the EU research program DITTUS [10] is applied to the transonic stage of HTTC. The test results done in TTM laboratory published in several previous papers are fully applied here. Scaling up to a mean radius of 750 mm at a circumferential speed of 668 m/s allows a degree of reaction of 0,4 and thus an optimal blade profile. Steam is transferred via the hollow nozzle blades into an inner chamber and is

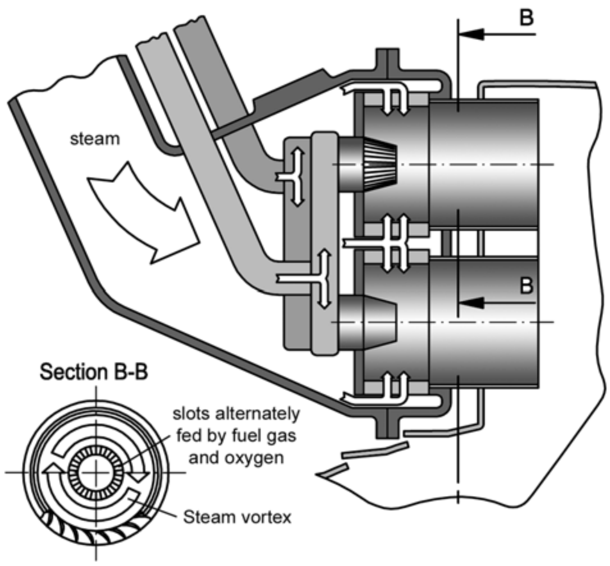

Figure 6 - Graz Cycle combustor consisting of 6 quadruples of 4 burners each in circumference

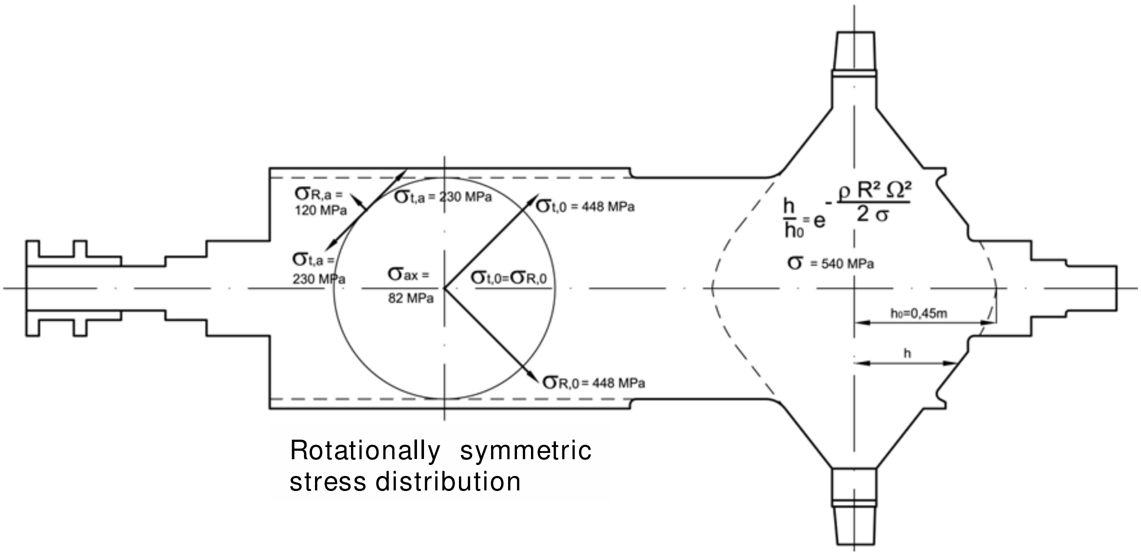

Figure 7 - C2-HTTC shaft with analytic solutions for three-dimensional stress indicated

blown at the correct angle to enter the fir tree roots in axial direction (see Fig. 5). From there on the cooling steam flows out in radial direction to the blade shell slots providing cohering layers of cooling medium covering the hole blade surface. The highly stressed disc part HTTC and also the outlet of C2 blading is steam cooled keeping the whole rotor drum at temperature around 350°C thus avoiding creep and allowing to use a forging of ferritic stainless steal with its high heat conduction and low thermal expansion. Also a closed rotor design is beneficial in avoiding corrosion since in start up on cold metal surfaces condensation of the working gas could occur and the entry of acidic condensate (water with CO_2) has to be prevented.

The flow from the combustion chamber shown in Fig. 6 has been described in previous papers in more detail [6, 7]. It has been mentioned here that a strong steam vortex is created in each burner avoiding loss of oxygen or fuel from the combustor process. Reactants are delivered in close proximity from the central cone in the burner outer cylinder limiting the steam vortex flow till vortex break down at the outlet. This design has not yet been tested, but first tests on oxy-fuel burners have already been successfully conducted, e.g. [17].

<u>Material selection and stress computation</u>

Motivation in material selection has been described above. Resistance to acidity and avoidance of creep are the main properties required. Also the three-dimensional stress in disks and drums is lowered by good ductility and high internal heat conduction. Fig. 7 shows the C2-HTTC shaft with analytic solutions for three-dimensional stress indicated as a means for the reader to observe the influence of stress raisers and parameters in over all stress distribution.

Figs. 8 and 9 show design details of compressor C1. First and last stage are mounted on the central drum by means of the two centring rings of highly stress resisting material in elastic deformation. The decision to build the first C1 stage in the form of a titanium blisk reduces centrifugal load in a more costly but safer kind of manufacture. The difficult roots in fixing the 16 blades to the inner ring would be expensive in a conventional design. Also the elastic ring fixture of the blisk to the drum allows radial expansion without excentricity or deflection and is beneficial for the rotor dynamic properties of the whole shaft.

The same deliberations apply for the last C1 stage as a radial compressor stage with splitter vane design. Here the higher temperature reached can be born by the material so that no cooling has to be applied. At the outlet of the radial wheel the radial blades are bent backward 45 degrees for better flow efficiency in the radial diffuser.

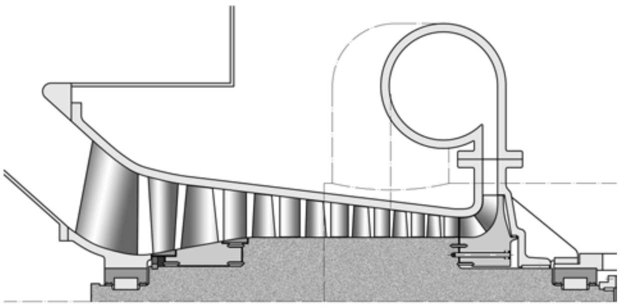

Figure 8 - Detailed view of compressor C1

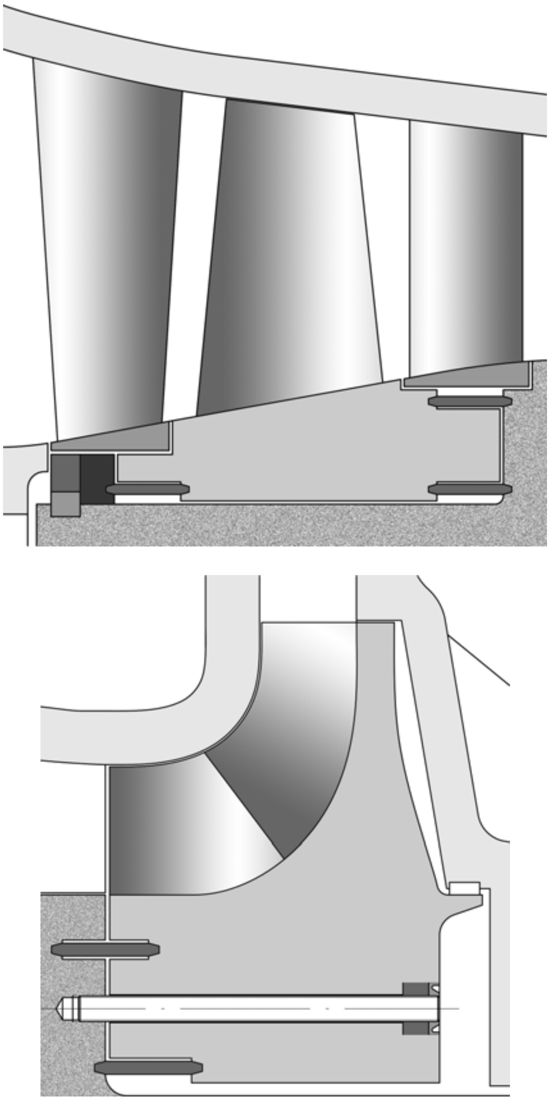

Figure 9 - Details of the first and last stage of compressor C1

<u>Rotor dynamic of C1</u>

All shafts require extensive rotor dynamic calculations and design optimisation, especially the most advanced design of the high speed compressor shaft presented here. As an example the details of compressor C1 are given. Fig. 10 shows Double Wedge Bearing design as patented by Jericha in 1963 and described in [18]. This type of bearing ensures stability against oil whip and resistance against high bearing lateral force as might arise from high rotor unbalance. Also they allow passing through critical speeds at run up of a shaft with very little increase of vibration amplitude.

Both rotors C1 and C2-HTTC are designed to run at operating speed in the range between first and second critical bending vibration speed (see Fig. 11). The second critical speed is high above operating, the first critical speed is passed between 50 % and 65 % of operating speed. The four bearing arrangement with its high damping and the low unbalance forces – at half speed – cause a very small increase of vibration amplitude. At operating speed the rotors are in so called inverted position with the centre of gravity inward on an only slightly bent shaft, which is an optimal rotor dynamic situation for continuous running.

In the cycle design it is shown that the flow to the first stage of C1 is sucked in from the cold side outlet of the HRSG. In boiler operation tube ruptures are very unlikely - providing careful design and manufacture – but may occur and should not cause sequential damage.

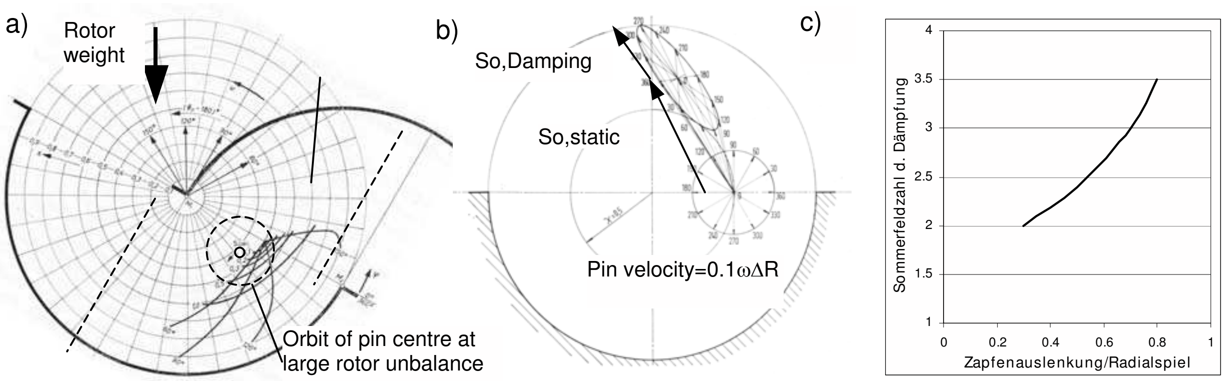

Figure 10 - Double Wedge Bearing:

a) Space of rotor movement inside the two fixed circular bearing halves (rotor reduced to its centre point in this diagram);
b) Bearing properties: close centring of rotor allow to withstand even very high lateral forces;
c) Diagram of Sommerfeld number vs. pin movement shows high damping and high elasticity of oil films.

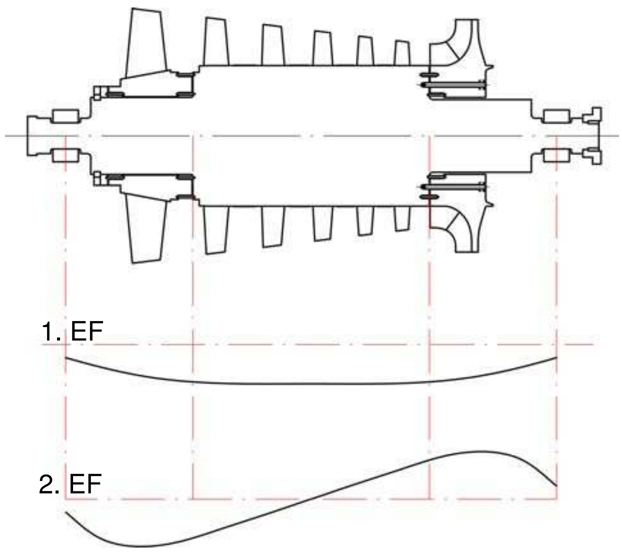

Figure 11 - Undamped eigenforms of rotor C1: First eigenform 88 Hz, second eigenform 280 Hz, operating speed 142 Hz.

Besides in discussion with industry we have been warned about erosion of the first stage of C1 at part load. Although there is sufficient superheat of the steam component of the working gas and temperature is rising quickly at the compressor inlet we have deliberated potential damage to the titanium blisk. We assumed a foreign damage on blade tip of first stage (0.7 m radius) with an unbalance in the form of metal loss of 0.5 kg at design speed of 8500 rpm. The bearing diameter was 144 mm and the relative radial clearance 0.002. Relevant Sommerfeld numbers are given in Fig. 10.

As Fig. 12 shows, even at this very high unbalance assumed – the unbalance force is equal to 2,5 times the total rotor weight – bearings and rotor are not harmed. No further damage will be induced in the other stages of C1 as well as rotor blading of C2. The tip clearances provided in manufacture are safe to take up the higher movements in the C1 bearings which are in the range of +/- 30 microns off the point of equilibrium between rotor weight and bearing oil film pressure.

So the design makes sure that even in the case of such harsh foreign damage only the casing of C1 has to be opened and the rotor blisk be repaired and no further damage from excessive rotor movement has to be feared.

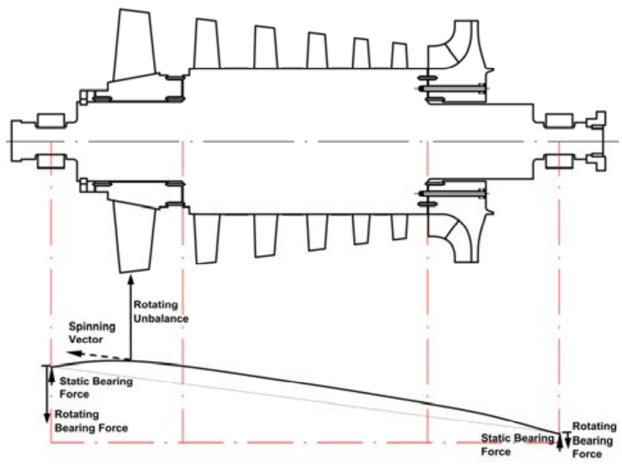

Figure 12 - Bearing amplitudes and rotor shaft bending of C1-HTTC rotor with foreign damage on blade tip of titanium blisk

CONCLUSIONS

Turbomachinery design for the oxy-fuel Graz Cycle system of 400 MW net power has been presented. The authors have given detailed deliberations on all aspects and on safety measures for the up-to-now largest Graz Cycle power plant presented in international conferences. We hope that our work will help industry to achieve a successful result in manufacture and operation in the short time limits indicated by European ZEP initiative.

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APPENDIX

See Fig. 13:

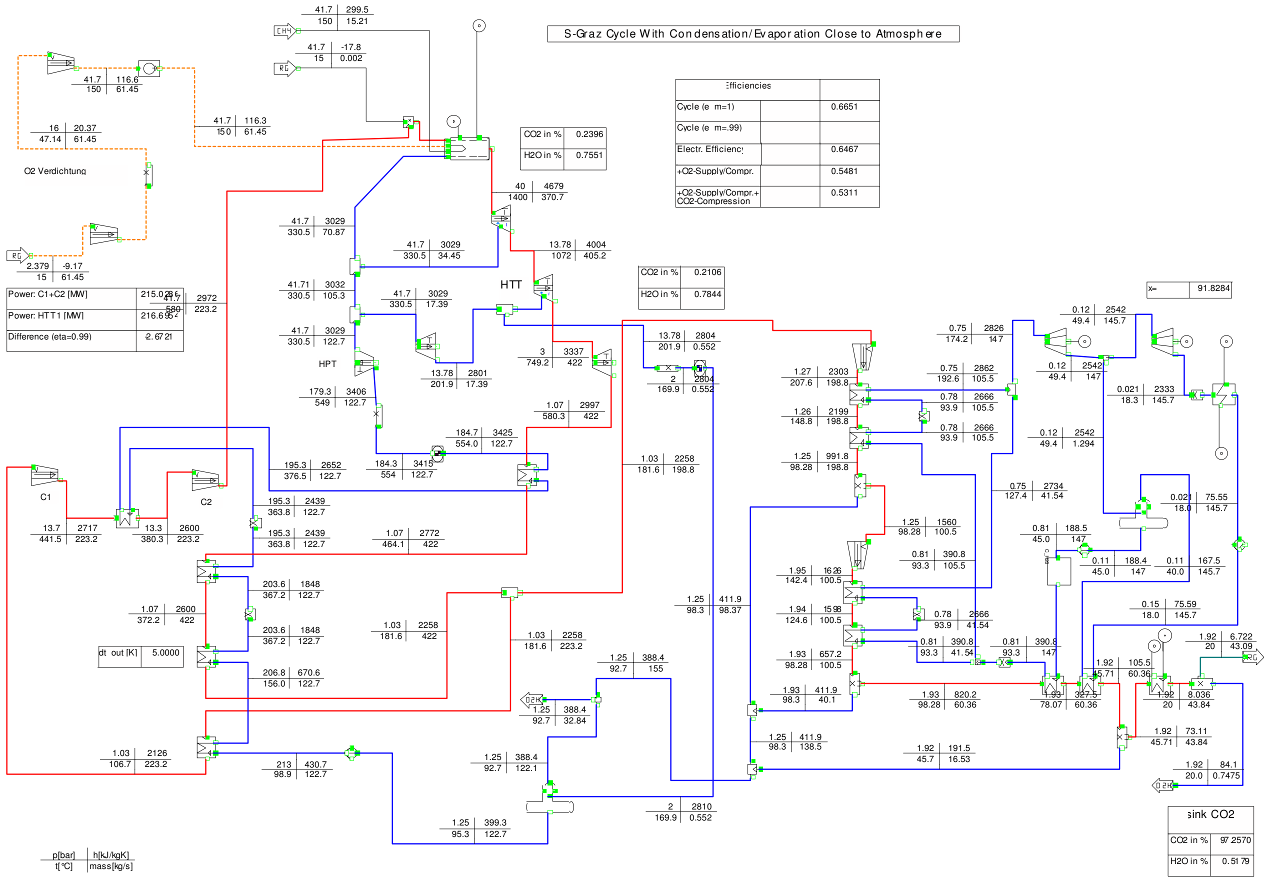

Figure 10 - Detailed thermodynamic cycle data of a 400 MW S-Graz Cycle Power Plant fired with methane