Proceedings of ASME Turbo Expo 2008: Power for Land, Sea and Air
GT2008
June 9-13, 2008, Berlin, Germany

GT2008-50515

DESIGN DETAILS OF A 600 MW GRAZ CYCLE THERMAL POWER PLANT FOR CO_2 CAPTURE

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ABSTRACT

The high power highest efficiency zero-emission Graz Cycle plant of 400 MW was presented at ASME IGTI conference 2006 and at CIMAC conference 2007. In continuation of these works a raise of power output to 600 MW is presented and important design details are discussed.

The cycle pressure ratio is increased from 40 to 50 bar by a half-speed stage connected via gears to the main compressor shaft allowing to keep the volume flow to the main compressors constant. The compressors are driven by the transonic compressor turbine stage. Mass flow to the compressors is increased by the factor of 1.27, density in blades of the main compressors is raised by the same factor. The turbine inlet temperature is raised to 1500°C together with the increase in the cycle pressure ratio, both are well accepted values in gas turbine technology today.

Most important development problems have to be solved in designing the oxy-fuel burners. They are presented here in the form of coaxial jets of fuel (natural gas or coal gas alternatively) held together by a steam vortex providing coherent flow and flame is ignited by its strong suction. Combustion is finalized by the mixing with a counter-rotating outer vortex flow of working gas leading to a well defined position of vortex break down.

The transonic stage of the compressor turbine is supplied with innovative steam cooling forming coherent layers outside of the blade shell of which stress deliberations will be presented.

INTRODUCTION

The recent years have shown a dramatic increase in damages all around the world with loss of properties and lives, dramatic change in geographic regions like the Artic, the Antarctic, the high seas and all mountain ranges. All this is caused by manmade changes of climate.

More and more people around the world honestly recognize their obligations to save fuel and to reduce emissions. It is now 20 years after the Toronto conference ("mankind is conducting an unintended uncontrolled experiment of possibly dire consequences"), several following conferences have exhorted in a similar way.

Today the United Nations ask for more research, the EU technology platform ZEP on Zero Emission Fossil Fuel Power Plants [1] has strongly demanded to complete research in 4 years and to have in further 10 years a major part of newly erected thermal power plants to comply with these orders. Scientists state on the basis of world wide observation that our time window is no more than 30 years from hence [2].

Solutions of permanent storing captured CO_2 are carefully assembled by the ZEP platform [1]. Geological storage in saline aquifers is considered as the most promising technology, its capacity is 10^3 – 10^4 Gt CO_2, compared to current global emissions of 30 Gt (about one third by the power generation sector) [3]. But further research is necessary before geological storage can start on a large scale. So the mapping of geological storage potential is performed in the EU project GeoCapacity, the topic of monitoring and verification of geological storage is investigated within CO2REMOVE and CASTOR [4].

In the last years many institutions, companies and universities have proposed power plants which allow CO_2 capture. In order to find the most promising solutions thermodynamic comparisons based on the same assumptions for all cycles combined with feasibility studies were performed, also within EU funded projects (e.g. [5, 6]). The Institute for Thermal Turbomachinery and Machine Dynamics at Graz

University of Technology is working since 1995 on the Graz Cycle system, an oxy-fuel cycle with internal combustion of fossil fuels with pure oxygen allowing an easy and cost-effective capture of the combustion generated CO_2 by condensation. Since then – as the authors believe – an impressive list of publications on this topic has been accomplished [7-18]. In [7, 8] first thermodynamic studies were presented on a cycle with internal combustion of methane with pure oxygen. In [9 – 13] the cycle was adopted to the firing of syngas from coal gasification and cycle modifications were proposed leading to a working fluid with two thirds CO_2 and one third steam. A layout of the turbomachinery components was presented for a pilot plant of 75 MW net power output. In 2004 the cycle scheme was rearranged similar to the original version [7, 8] with a working fluid consisting of three quarts steam and one quart CO_2 [14, 15]. In 2006 at the ASME IGTI conference the authors published a design proposal for a CO_2 retaining gas turbine of 400 MW output where condensation of the working fluid takes place at atmosphere[16]. In 2007 stress and rotor dynamic design improvements to the high speed compressor shaft where published at CIMAC conference in Vienna [17] and a design comparison to competing proposals was presented at the ASME IGTI conference in Montreal [18].

Many discussions with industry have been made. For the design of transonic stages the cooperation with GE Oil & Gas Nuovo Pignone (Erio Benvenuti, [19]) was most successful. The question of large power plant installation was discussed with Siemens Power Generation in Erlangen, Germany, Alstom Power Systems in Baden, Switzerland, and Statoil ASA in Norway. Besides the high efficiency of a Graz Cycle system, all our partners see higher costs in the manufacture and erection of a first novel design power plant and still further costs to appear for the user in the need to collect and store retained CO_2.

Only one first prototype plant in the oxy-fuel system is being built now in Germany, but as a steam plant with oxygen combustion of pulverized coal in the boiler [20].

The Graz Cycle system shows the highest efficiency values for oxy-fuel systems [e.g. 6]. In 2003 design details for a 92 MW unit were presented at the ASME IGTI conference (ASME best paper award, [11]) and in consequence the proposed power output has been raised to 400 MW [16]), which is now further enlarged.

In order to improve the investment situation the authors demonstrate here that the same turbine type of 400 MW can be raised in power to an output of 600 MW. This can be accomplished by a single pre-compressor stage driven from the main compressor shaft and quite reasonable increases in thermodynamic performance. So the combustor pressure is increased to 50 bar and turbine inlet temperature to 1500°C. These values appear in standard gas turbine development well acceptable [21]. Further progress in the design of important components complete this work.

THERMODYNAMIC LAYOUT

All thermodynamic simulations were performed using the commercial software IPSEpro by SIMTECH Simulation Technology [22]. This software allows to implement user-defined fluid properties to simulate the real gas properties of the cycle medium.

The oxy-fuel system is suited for all kinds of fossil fuels, e.g. methane or syngas from coal or biomass gasification. In this work thermodynamic data are presented for a cycle fired with methane with a lower heating value of 50015 kJ/kg.

The component efficiencies and losses were agreed with the Norwegian oil and gas company Statoil ASA in the course of a thermodynamic evaluation of the Graz Cycle and can be found in [15]. Some important assumptions are listed here again: 1) The isentropic efficiency of the cooled gas turbines is 90.3 % and includes the flow losses due to cooling. It corresponds to a polytropic efficiency of 85.5 %. The demand of cooling flow is calculated as described below. 2) Oxygen production is considered with an effort of 900 kJ/kg (0.25 kWh/kg), the compression needs 325 kJ/kg. 3) The compression of combustion generated CO_2 from 1.7 bar to 100 bar is considered in the power balance with a value of 310 kJ/kg CO_2.

Figure 1 shows the principle flow scheme of the Graz Cycle plant for a net output of 600 MW. Basically the Graz Cycle consists of a high-temperature cycle (compressors C0, C1 and C2, combustion chamber, High-Temperature Turbine HTT, Heat Recovery Steam Generator HRSG and High Pressure Turbine HPT) and a low temperature cycle (Low Pressure Turbine LPST, condenser and compressors C3 and C4). The fuel together with the nearly stoichiometric mass flow of oxygen is fed to the combustion chamber, which is operated at an increased pressure of 50 bar (compared to 40 bar in previous layouts). Steam drives the burner vortex core bringing together the reaction components. The high flame temperature is further reduced by the inflow of the working gas (CO_2/steam) around the burners and into the combustion chamber liner.

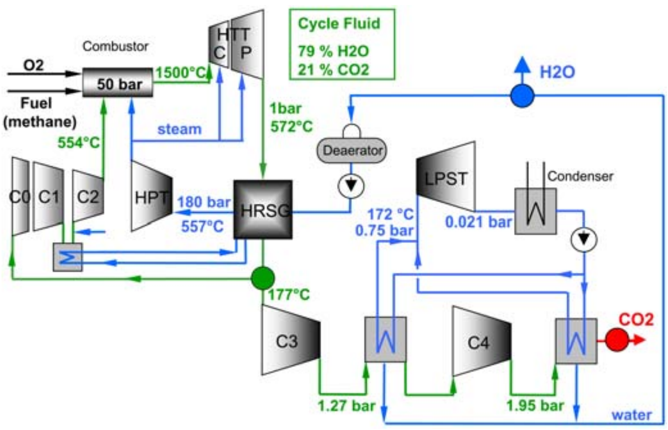

Fig. 1: Principle flow scheme of the 600 MW Graz Cycle plant

A mixture of about 74.1 % steam, 25.4 % CO_2, 0.4 % O_2 and 0.1 % N_2 (mass fractions) leaves the combustion chamber at a mean temperature of 1500°C (compared to 1400°C in previous layouts), a value achieved by G and H class turbines nowadays. The fluid is expanded to a pressure of 1.06 bar and 572°C in the HTT. The HTT consists of a free-running compressor turbine HTTC and a power turbine HTTP running at 3000 rpm. Cooling which is performed with steam coming from the HPT at about 366°C is increased to 18.4 % of the HTT inlet mass flow, increasing the steam content to 78.1 % at the HTT exit. The hot exhaust gas is cooled in the following HRSG to vaporize and superheat steam for the HPT; the pinch point of the HRSG is 5°C (an aggressive value but used for both cycles), the approach point at the superheater exit is 10°C. After the HRSG about 50 % of the cycle mass flow is re-compressed using the main cycle compressors C0, C1 and C2. Between C1 and C2 an intercooler is arranged superheating steam from the HRSG. Additionally feed water fogging immediately after the cooler ahead of C2 blading is done to keep the compressor exit temperature at about 554°C. The compressed working fluid is then fed to the combustion chamber. The pre-compressor C0 is arranged in order to keep the same volume flow at the C1 and C2 inlet at increased mass flow. This allows to use the same compressors for a 400 MW and 600 MW Graz Cycle plant. All compressors are driven by the HTT compressor turbine.

The remaining mass flow which contains the combustion generated CO_2 is fed to a condensation process in the 1 bar range in order to avoid the problems described above. The heat content in the flow is still quite high so re-evaporation and expansion in a bottoming cycle is mandatory. For proper re-evaporation two sections of working fluid condensations are provided, each following a compressor stage with reasonable increase of flow pressure resulting in a higher partial condensation pressure of the water content. The two compressor stages can be regarded as pre-runners of the CO_2 delivery compressor and will be helpful in cleaning the turbomachinery, piping and HRSG interior from air in preparation of a cold start. The heat exchangers are well developed modern boiler elements providing steam just below atmosphere (0.75 bar) for the condensing steam turbine LPST.

At the first pressure level of 1.27 bar about 62 % of the water content can be segregated, so that the power demand of the second compression stage is considerably reduced. It compresses up to 1.95 bar, which allows the segregation of further 26 % of the contained water. Further cooling of the working fluid, also for water preheating, leads to the separation of additional 11 %, so that the water content of the CO_2 stream which is supplied after pressure losses at 1.7 bar for further compression, is below 1 %. More details of the condensation/ evaporation process can be found in [16]. After segregation of the water stemming from the combustion process, the water flow is degassed in the deaerator with steam extracted after the HPT and fed to the HRSG for vaporization and superheating. The steam is then delivered to the HPT at 180 bar and 557 °C. After the expansion it is used to cool the burners and the HTT stages.

The two-step condensation/evaporation counteracts the effect of sinking H_2O partial pressure due to condensed water extraction from working fluid and thus allows reasonable steam inlet conditions of 0.75 bar and 175°C at the LPST inlet. Expanding the steam to a condensation pressure of 0.021 bar for a cooling water temperature of 8°C (Northern Europe) provides about 101 MW power output.

The detailed flow sheet used for the thermodynamic simulation can be found in the appendix (Fig. 8) and gives mass flow, pressure, temperature and enthalpy of all streams.

Power Balance

Table 1 gives the power balance for a Graz Cycle plant of 400 MW and 600 MW net power output. The heat input, the mass flow and thus the power of turbomachinery are increased for higher power output. The mass flow is increased by about 27 %, whereas the total heat input and most turbomachinery power is increased by 40 to 45 %. So the total turbine power increases by 45 % to 1071 MW, whereas the total compression power is only increased by 38 %. The shaft power of the 600 MW cycle is 724.6 MW. The higher cycle peak temperature of 1500°C leads to a thermal and electrical net efficiency about 1 %-point higher than for the 400 MW cycle. The net electrical efficiency is 65.71%.

If considering the efforts for oxygen production and compression as well as the efforts of CO_2 compression to 100 bar for liquefaction, the net efficiency reduces to 54.14 %,

Table 1: Graz Cycle Power Balance

	400 MW [16]	600 MW
HTT power [MW]	617.9	908
HPT power [MW]	49.9	62
LPST power [MW]	71.6	101
Total turbine power P_T [MW]	739.4	1071
C0 power [MW]	-	8.8
C1 power [MW]	131.1	178
C2 power [MW]	82.6	108
C3 power [MW]	8.9	13
C4 power [MW]	6.6	10
Pump power [MW]	5.5	7.2
Total compression power P_C [MW]	234.7	325
Net shaft power [MW] without mechanical losses	504.7	746
Total heat input Q_{zu} [MW]	758.6	1100
Thermal cycle efficiency [%]	**66.52**	**67.6**
Electrical power output [MW] incl. mechanical, electrical & auxiliary loss	490.7	724.6
Net electrical cycle efficiency [%]	**64.68**	**65.71**
O_2 generation & compression P_{O2} [MW]	74.7	109
Efficiency considering O_2 supply [%]	**54.83**	**55.83**
CO_2 compression to 100 bar P_{CO2} [MW]	13.0	18.6
Net power output [MW]	**403.0**	**597**
Net efficiency η_{net} [%]	**53.12**	**54.14**

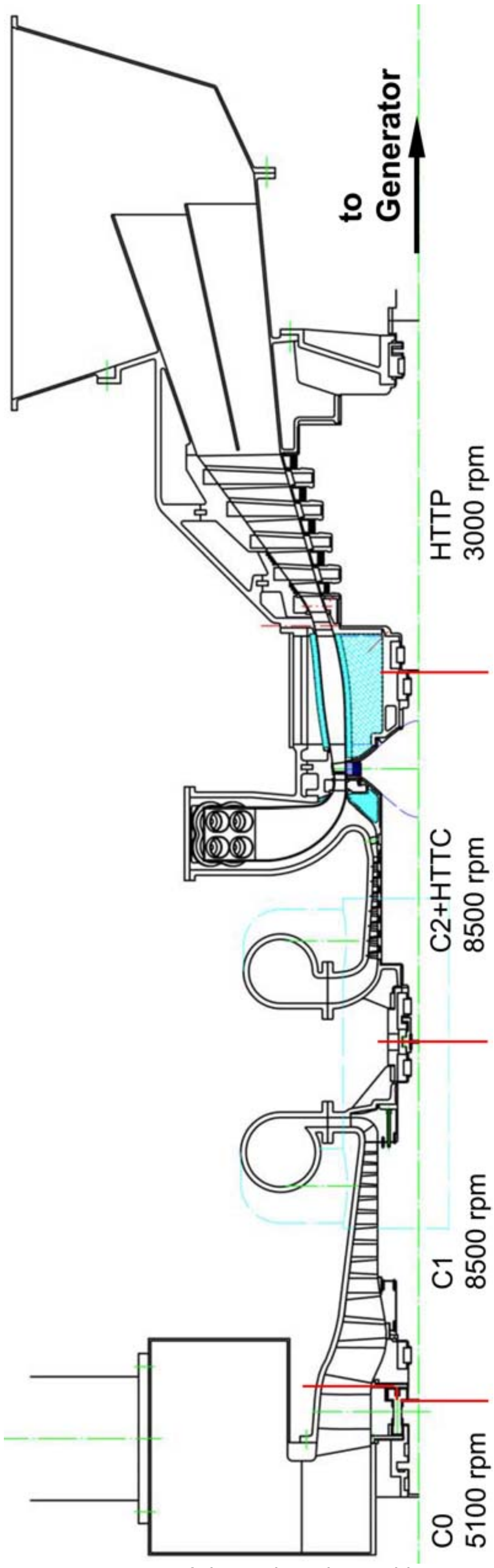

Fig. 2: Arrangement of the main turbomachinery conmponents for a 600 MW Graz Cycle plant

compared to 53.12 % for the 400 MW plant. This high net efficiency shows that this new concept is worth further investigation.

DESIGN CONCEPT FOR A VERY LARGE GRAZ CYCLE PLANT OF 600 MW NET OUTPUT

For the 600 MW Graz Cycle plant thermodynamic properties have improved as shown by the power balance. Gas turbine entry temperature is raised and combustor pressure also in a way to keep the outlet temperature of the power turbine in the same range. Accordingly the steam cooling supply is enlarged.

In this work design details of the components for a Graz Cycle power plant of 600 MW electrical net output are presented. This high power is derived from a 725 MW turbo shaft configuration. The difference is caused by the power demand of the ASU and by the driving power for the oxygen compressor in order to deliver oxygen to the combustor at 52 bar and by the CO_2 compressor which has to deliver the captured CO_2 at a pipeline pressure of over 100 bar.

The main gas turbine components are arranged on two shafts, the compression shaft and the power shaft (see Fig. 2). The compression shaft consists of the working fluid compressors C0, C1 and C2, which are driven by the first part of the high temperature turbine HTT, the compressor turbine HTTC. It runs free on its optimal speed of 8500 rpm. This relatively high speed is selected for reason of obtaining sufficient blade length at outlet of C2 and to reduce the number of stages in both compressors. The second part of the HTT, the power turbine HTTP, delivers the main output to the generator. A further elongation of the shaft is done by coupling the four-flow LPST at the opposite side of the generator. The HPT can be coupled to the far end of the LPST or can drive a separate generator. In order to use the same compressors C1 and C2 as for the 400 MW plant, the pre-compressor C0 is arranged ahead of C1. It increases the pressure up to 1.22 bar, which leads to the same volume flow to C1. It runs at 5100 rpm.

Pre-compressor C0

In order to maintain the design of the working fluid compressors C1 and C2, as presented at the CIMAC conference in Vienna [17] for increased power and mass flow, a pre-compressor is arranged ahead of the main compressors. Fig. 3 shows compressors C0 and C1 connected via a planetary gear. It has a pressure ratio of 1.2 and a driving power of 8.8 MW.

The first stage of C1 is a development of the University of Darmstadt [23] having developed a transonic test compressor. Here it is scaled up in size, tip radius 0.7 m, and is driven at 8500 rpm. Due to temperature rise of the working gas (CO_2/steam) the tip Mach number can be kept well below admissible limits.

C0 is moderately loaded. The tip Mach number is 1.3. Four pairs of planetary shafts supported in the disc of C0 fan, geared to the main compressor shaft on one end and engaged with a

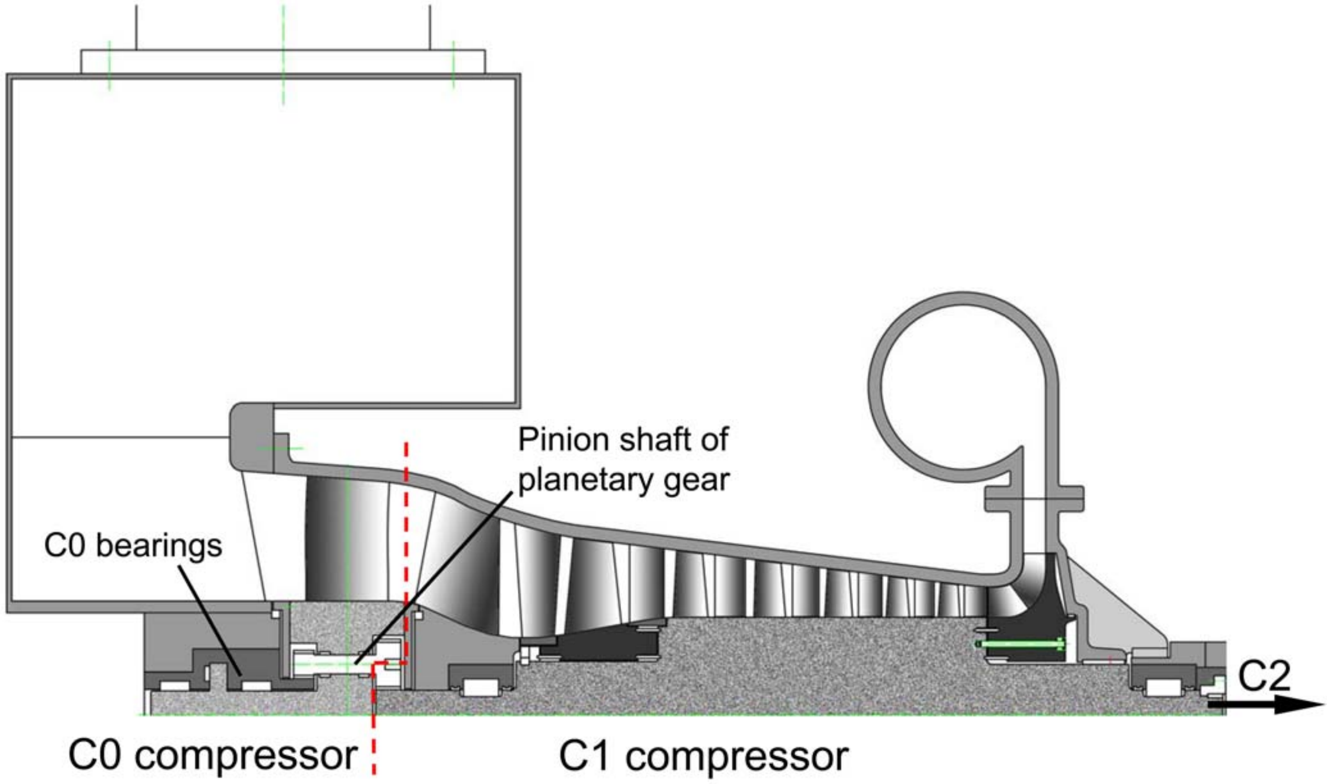

Fig. 3: Design of pre-compressor C0 and compressor C1 connected via a planetary gear

static gear on the opposite side serve to reduce the speed of the main compressor to 5100 rpm (counter rotating). This gearing is compact and allows lubricant sealing on a small diameter surface. Bearing struts and the inlet guide vanes to C1 serve to equalize the flow and to avoid vibration excitation of the transonic blades of the first stage of C1.

Combustor

Oxyfuel combustion leads to very high temperatures compared to combustion in air. It is proposed to use steam to enter the burners in a vortex wrapping around the reactants fuel and oxygen in its core, thus avoiding the loss of a reaction partner. Cooling the flame below dissociation temperature by steam is necessary and serves for complete combustion [24].

Fig. 4 shows the proposed design of burner quadruples. An inner burner zone and two concentric cylindrical shells with guide vanes create first the burner vortex from fuel gas, oxygen and steam and at the outlet of each burner the vanes of the outer passage create a counter vortex of working gas.
So this design provides an additional cylindrical sheet outside of the steam vortex through which about half of the working gas flow is introduced at each burner in counter rotation so that after combustion and mixing with steam by strong vortex break down counter-rotating vortices are introduced in an axially fixed position. A strong axial velocity generated by the volume increase in the flame serves for proper mixing with the rest of the working gas introduced in the flame cage.

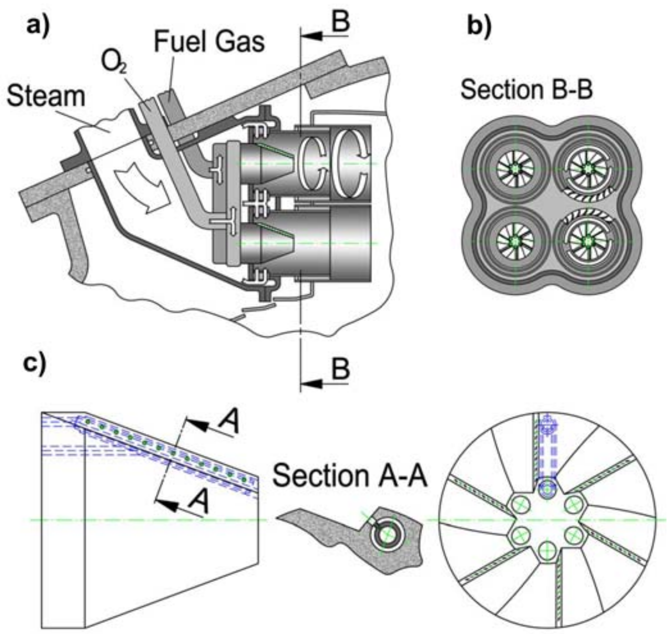

Fig. 4: Details of a) burner design, b) burner quadruple arrangement, and c) detail of burner cone with nozzles for concentric flames of fuel and oxygen

The low pressure in the inner core of the burner vortex is created not only by steam but also by the coherent jets of fuel and oxygen forming thus a large number of flames driving the circumferential velocity further on. This arrangement serves to suck in hot combustion gas from the burner outlet. Thus the gas and oxygen jets on the inner burner cone are continuously ignited. The counter rotating vortices of working gas at the outlet care for an ensuing vortex breakdown resulting in a flame temperature of 1800°C. Mixing in the flame cage with the rest of the working gas flow provides an exact mixture with constant temperature and a certain increase in pressure ahead of the first stage nozzles.

An arrangement of six burner quadruples around the outer circumference of the flame cage provides an even distribution of flame temperature and the mixing with the full amount of working gas creates optimal inflow conditions to the first turbine stage.

High speed shaft (C1, C2 and Compressor Turbine HTTC)

The high speed shaft consisting of the compressors C1 and C2 driven by the HTTC is basically identical with the design solution presented at previous conferences [17, 18]. The presentation here offers a very strong power upgrade from 400 to 600 MW output with basically the same compressor rotors driven by a still stronger HTTC transonic stage. The general arrangement of the high power gas turbine cycle concepts is in the form of a high speed shaft and a gas flow transmission to a co-axial standard speed power shaft (see Fig. 2). The relatively low volume flow -compared to the outflow of the power turbine- requires high-speed compressors which in this power range cannot be driven via gears. So a high-speed first turbine part has to run as the same speed as compressors C1 and C2. The power enlargement mentioned above is obtained here by the arrangement of a pre-compressor – raising inlet pressure from 1.02 to 1.22 bar. Thus mass flow to compressors increases by 26% and also combustor pressure rises from 40 to 50 bar. This changes are made in comparison to the previous mentioned design solutions for 400 MW Graz Cycle units [16-18]. Driving all compressors, now C0, C1 and C2, from a single stage transonic turbine with innovative steam cooling is a development of our institute which has been presented before. The somewhat more stringent conditions due to the higher turbine inlet temperature can be coped with and are to be shown later on.

The transonic turbine stage drives the compressors C0, C1 and C2 with a total power of 294 MW. The blading in these compressors C1 and C2 remains unchanged insofar as the same flow volume is to be tackled. Mass flow is increased requiring some minor changes in blade strength and root fastening. The only difference in C2 compared to [16] is a reduction of final compression temperature effected by feed water fogging immediately after the cooler ahead of C2 blading (see Fig. 1). The arrangement of two compressors in sequence on different shafts allows intercooling and fogging and results in a relatively low number of stages on each shaft. The well known meridional flow profile deterioration stemming from tip leakage flow is thus kept quite low. In the second half of the stages of C2 steam jets at the outer radius could be introduced to further equalize the flow profile and to improve flow efficiency (see [17]).

The temperature of the drum near the transonic stage -the disc of which is arranged as a common piece of forging- has to be reduced by steam cooling. The drum part of compressor C2 can thus be kept below 400°C, even at working gas temperatures of around 540°C. Metal heat conduction to the inlet side serves the same purpose. The amount of cooling steam is relatively low and a secure material position avoiding creep is given..

The two rotors of compressors C1 and C2 run above their first critical speeds, these being different by about 20% and near half the running speed of 8500 rpm. In start running-up they pass through their first critical speed at the given difference. With the damping of four large area double wedge bearings strong damping is given. This feature makes the vibration peak in actual operation hardly noticeable. A more detailed discussion on rotordynamics can be found in [17].

Transonic stage of compressor turbine HTTC

Disc material is selected as a properly alloyed steal forging. The symmetric disc profile is designed according to Traupel`s formula for a disc of constant stress. Analytical solution of the compressor rotor drum stress is shown in Fig. 5 as presented in [17].

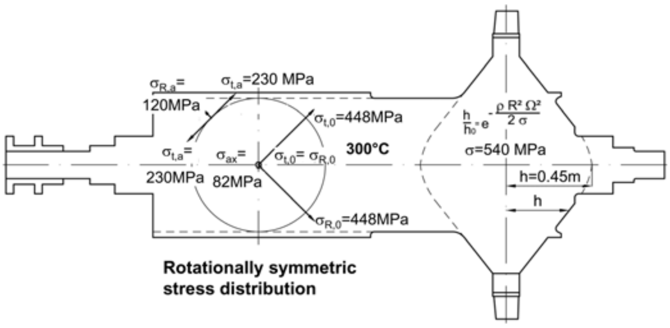

Fig. 5: C2-HTTC shaft with analytic solutions for three-dimensional stress indicated [17]

It is cooled on all accessible surfaces by cooling steam of about 300°C keeping the metal below creep range and also inaccessible to carbonic acid that might form in start up in condensation of the H2O part of the working gas in which CO_2 could go into solution (see Fig. 6). Still this is a mild acid and all drum and disc surfaces are thus resistant against corrosion and could be shielded by additional surface treatment as well. There is no danger of accumulation of carbonic acid inside the rotor as might be possible in a multi-disc rotor design.

HTTC Blade design

One of the main advantages of the Graz Cycle is the arrangement of the high pressure turbine HPT expanding the high pressure steam from the boiler for direct injection into the burners and into the high temperature turbine stages. We follow here the development of the EU research grant "DITTUS"(development of industrial transonic turbine stages) conducted by Graz University of Technology together with GE Nuovo Pignone [25]. Together with supporting computer calculations by the University of Florence [26] our innovative cooling system was tested repeatedly in our laboratory. The results were the proof for applicability and reliability of our steam cooling system which is applied also in this case.

Steam cooling supply in general is conducted from the outflow of the HPT, usually a separately arranged steam turbine, into the combustion chamber, and into the blading of the the transonic HTTC stage, and into the first part of blading of the HTT power turbine. Here also blade cooling, blade root cooling and thrust balancing is achieved by steam supply.

The cooling system of the HTTC transonic stage consists of an annular chamber outside the first nozzle ring, hollow vanes providing steam paths inside which cool the vanes and allow the assembly in an inner annular chamber (see Fig. 6). From there steam is directed by small nozzles arranged all around the circumference in an angle to turbine axis, so that in relative velocity a more or less axial inlet into disc openings below the blade fir-tree roots is achieved.

In this blade of the first stage exposed to highest temperature all profiles leave areas open for the radial flow of steam inside the blade to supply the cooling slots. The cooling steam expands to sound velocity and further on sideways covering all the outside of the blade creating an actually coherent layer of cold steam inside the hot working gas (see Fig. 7) [26].

In selection of material for this proposal the experience from partners in the metallurgy department at Graz University of Technology was a great help giving prescriptions for stress and temperature operation keeping all machine parts free of creep.

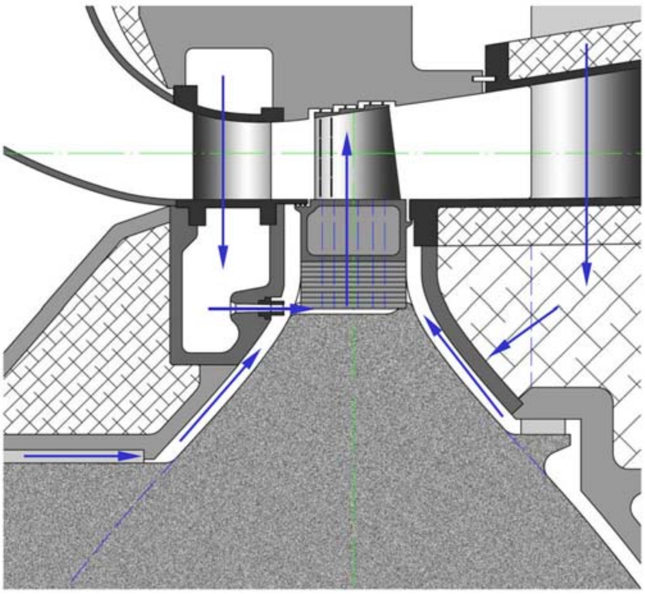

Fig. 6: Detailed meridional profile of HTTC with first nozzle ring, first stage blade and cooling steam admission (blue vectors).

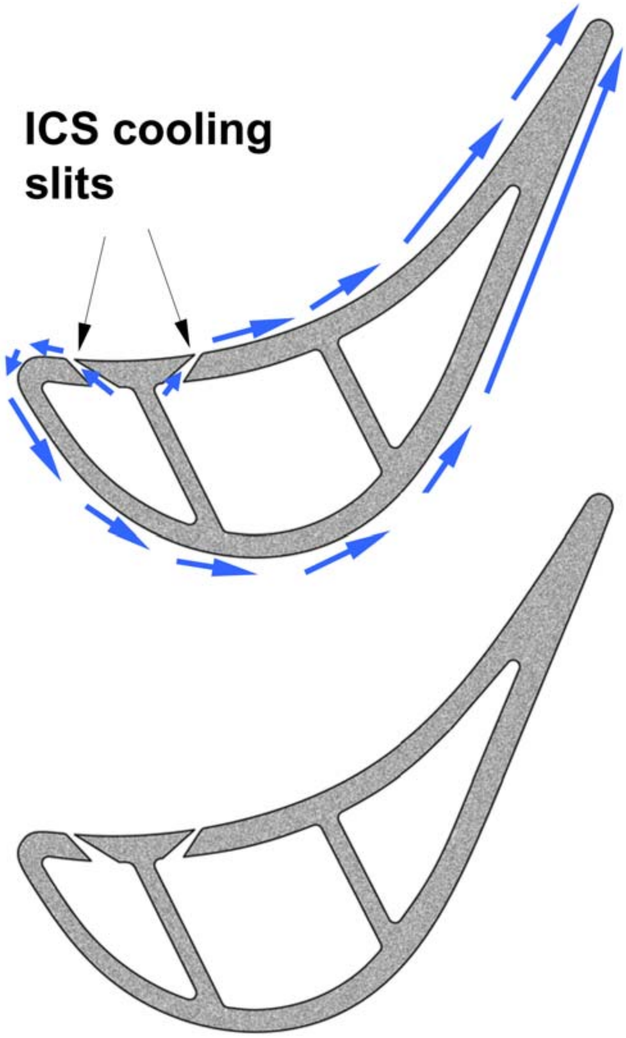

Fig. 7: Blade profile section made up from a shell and internal ribs with slots for the formation of underexpanded transonic steam layers covering the whole blade surface.

In guaranteeing high stage efficiency of the HTTC transonic stage our laboratory tests on transonic flow with advanced laser-optical measurement methods were prerogative to present this design advance. To demonstrate this we show in the following Fig. 7 just the mean section blade profile and flow channel. The selection of circumferential speed and heat drop and thus flow velocity and Mach number development in through-flow gives the following positive results. The turn of flows through the blading is relatively low, the width of the flow channel is decreasing towards the outlet showing a strong acceleration and thus minimal profile loss, this at a degree of reaction of 38 %. Leakage loss is reduced by cooling steam inlet at the inner diameter and by a optimal stress-designed shrouding at top.

Power turbine

The cross over from the transonic stage to the 3000 rpm power turbine is in the same design as presented before [16, 17]. Also power turbine drum and cooling on the low pressure side have the same dimensions. Only the last stage requires a lengthening from 600 to 700 mm. The higher centrifugal load can be accommodated due to low drum temperature and steam cooling throughout the blading roots.

CONCLUSIONS

On the basis of 12 years' work with many congenial partners who have given help and support we present here a novel type of power station. By design deliberations and intensive calculations we were able to raise the output of our proposed gas turbine from 400 to 600 MW in one year.

This increase of plant output is achieved by increase of the turbine inlet temperature from 1400° to 1500°C and the cycle pressure from 40 to 50 bar. The simultaneous increase of pressure and temperature allows to keep the High Temperature Turbine (HTT) exit temperature at the same level. The increase in pressure is realized by a an additional stage ahead of the the C1 working fluid compressor. This arrangement allows to maintain the inlet volume flow at an increased mass flow of 27 %. So the same design of the C1 and C2 compressors can be used for the 400 and 600 MW plant. The increased peak values of temperature and pressure lead to an increased plant efficiency of 54.14 % compared to 53.12 % for the 400 MW plant.

The flow design of the working fluid compressors is presented in detail. The additional stage is a transonic stage driven by the C1 compressor via an innovative gearing design at 60% speed. For the oxygen burners a design with layers of counter-rotating vortices is proposed caring for proper mixing of the reactants kept together by a steam vortex. The drum stresses of the high-speed compressor/compressor turbine shaft C2-HTTC are demonstrated by analytical solutions, showing acceptable values well below the creep range. Steam is applied for shaft cooling and also for the first HTT compressor turbine stage with an innovative cooling system. Also the shaft of the power turbine and blade roots are cooled with steam. Further design details presented indicate the maturity of the proposed design.

The authors ask international gas turbine industry especially the world wide active large gas turbine and power station manufacturers to discuss with them and control their proposal and to proceed towards a successful world-wide carbon capture and storage system. Please make haste, time for mankind is running out ([2]).

Personal Remark by the lead author, Herbert Jericha, Professor of thermal turbomachinery and rotordynamics, Fellow ASME

From CIMAC Vienna 2007 to ASME Berlin 2008 I have been able to work out and hope to be soon able to present this result to the world of thermal energy production. This may be my last chance to ask you all to co-operate.

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APPENDIX

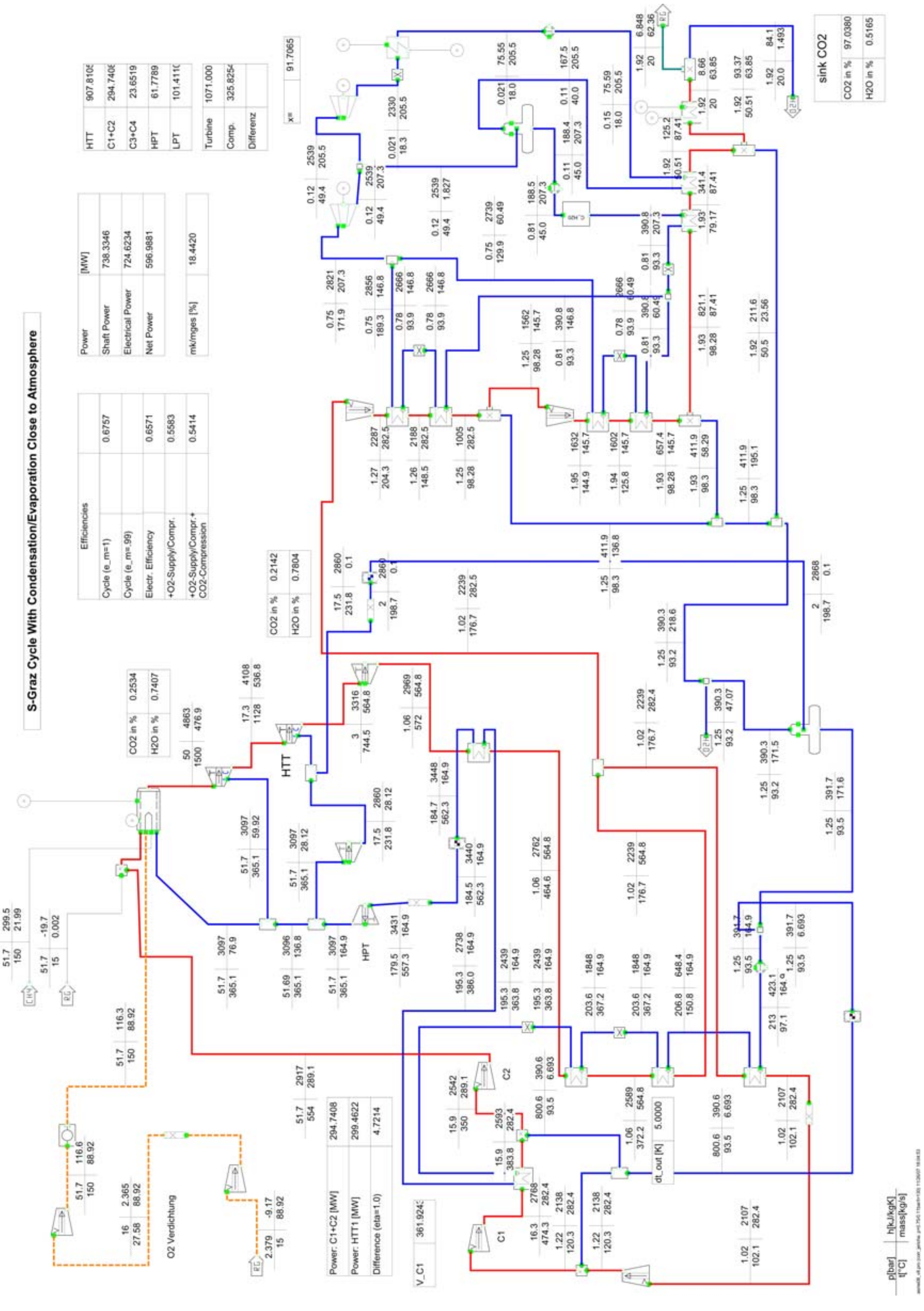

Fig. 8: Detailed thermodynamic cycle data of a 600 MW Graz Cycle power plant fired with methane