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DESIGN CONCEPT FOR LARGE OUTPUT GRAZ CYCLE GAS TURBINES

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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 work at Graz University of Technology since the nineties has led 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 for modern combined cycle plants.

Upon the basis of the previous work the authors present the design concept for a large power plant of 400 MW net power output making use of the latest developments in gas turbine technology. The Graz Cycle configuration is changed insofar, as condensation and separation of combustion generated CO_2 takes place at the 1 bar range in order to avoid the problems of condensation of water out of a mixture of steam and incondensable gases at very low pressure. A final economic analysis shows promising CO_2 mitigation costs in range of 20 – 30 \$/ton CO_2 avoided. The authors believe that they present here a partial solution regarding thermal power production for the most urgent problem of saving our climate.

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 from power generation. So there is a strong driving force to develop commercial solutions for the capture of CO_2 from power plants.

The authors believe that oxy-fuel cycles with internal combustion of fossil fuels with pure oxygen 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 is needed in large quantities for this kind of cycle and 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 (EOR) [1].

The basic principle of the so-called Graz Cycle has been developed by H. Jericha in 1985 [2] for solar generated oxygen-hydrogen fuel, in 1995 changed to fossil fuels [3, 4]. This was a first proposal for gas turbine oxy-fuel CO_2 capture. Improvements and further developments since then were presented at several conferences [5-9]. Any fossil fuel gas (preferable with low nitrogen content) is proposed to be combusted with oxygen so that neglecting small impurities 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 a power cycle with a thermal efficiency among the very best ever proposed, thus largely compensating the additional efforts for oxygen production.

At the ASME IGTI conference 2004 in Vienna a Graz Cycle power plant (High **S**team Content Graz Cycle, S-Graz Cycle) was presented with a thermal cycle efficiency of nearly

70 % (excluding work for oxygen supply and CO_2 compression) based on syngas firing and relatively low CO_2 retention costs [10].

The very promising data aroused interest in several institutions in Europe, among them the Norwegian oil and gas company Statoil ASA. In cooperation with Statoil the S-Graz Cycle was re-evaluated and optimised with assumptions on component losses and efficiencies that Statoil and Graz University of Technology had agreed on. At the ASME IGTI conference 2005 [11] the results were presented with a net cycle efficiency of 52.7 % for methane firing, if the efforts of oxygen supply and compression of captured CO_2 for liquefaction are considered. The CO_2 mitigation costs were evaluated to 20.7 \$/ton CO_2 avoided.

These investigations also formed the basis of a techno-economic evaluation study by the two most important and most successful gas turbine companies in Europe. The feasibility of the S-Graz Cycle was accepted and the cost structure discussed in detail. The result was on one side that the cryogenic Air Separation Unit ASU appeared to have too high investment costs. On the other side the condensation of water out of a mixture of steam and incondensable gases, a thermodynamic technical problem not yet solved in European science, had to be more clearly investigated.

Therefore the object of this paper is to present author's work on the following subjects:

- Modification of S-Graz Cycle configuration to condensation in the range of 1 bar providing for separation of combustion generated CO_2 to the delivery compressor. By slight recompression evaporation of pure steam at reasonably high pressure and efficient expansion in a large output steam turbine (LPST) is made possible.
- Increase of net plant output to 400 MW providing for the additional tasks of oxygen production, CO_2 capture and delivery for pipeline use or liquefaction. The power effort is included in the overall efficiency raising the shaft output design value to 490 MW
- Two-shaft design of the turbo set with a fast running shaft comprising the main compressors C1 and C2 and the compressor turbine and with the power output shaft from high temperature turbine and steam turbine
- Incorporation of advanced flow and cooling development throughout the gas turbine components for smaller size and cost and reduction of high temperature material by rotor steam cooling on all accessible surfaces

Deliberations on part load and cold start for the situation of the novel cycle medium are also presented.

In this work the nomination "Graz Cycle" means "S-Graz Cycle", which is the more efficient variant and will be prosecuted in the future.

GRAZ CYCLE BASIC CONFIGURATION

The Graz Cycle is suited for all kinds of fossil fuels. Best results regarding net cycle efficiency and mitigation costs can be obtained for syngas firing from coal gasification, if the syngas production effort is not considered in the thermodynamic balance (but only in the economic balance by elevated fuel costs). The higher net cycle efficiency is due to the fact that the lower oxygen demand of syngas per heat input reduces the effort of oxygen supply considerably. And finally, the higher carbon content results in more favorable mitigation costs per ton CO_2 avoided. But in this work thermodynamic data presented are for a cycle fired with methane, because it is the most likely fuel to be used in a first demonstration plant.

Figure 1 shows the principle flow scheme of the S-Graz Cycle with the main cycle data as published in [11].

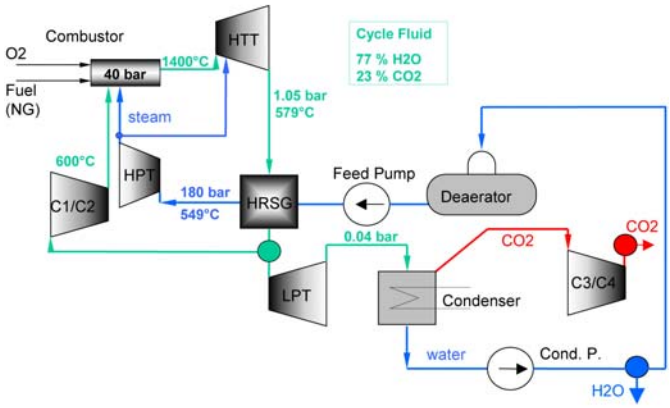

Fig. 1: Principle flow scheme of the basic Graz Cycle power plant

Basically the Graz Cycle consists of a high temperature Brayton cycle (compressors C1 and C2, 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 nearly 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 74 % steam, 25.3 % CO_2, 0.5 % O_2 and 0.2 % N_2 (mass fractions) leaves the combustion chamber at a mean temperature of 1400°C, a value achieved by G and H class turbines nowadays. 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 at about 330°C (13.7 % of the HTT inlet mass flow), increasing the steam content to 77 % at the HTT exit. 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 vaporize and superheat steam for the HPT; the pinch point of the HRSG is 25°C at the superheater exit. But after the HRSG only 45 % of the cycle mass flow are further expanded in the LPT. For a cooling water temperature of 8°C the LPT exit and thus condenser pressure would be 0.041 bar.

Gaseous and liquid phase are separated in the condenser. From there on the gaseous mass flow, which contains the combustion CO_2 and half of the combustion water, is compressed to atmosphere by C3 and C4 with intercooling and further extraction of condensed combustion water, and supplied for further use or storage. At atmosphere the CO_2 purity is 96 %; further water extraction is done during further compression for liquefaction.

After segregating the remaining combustion H_2O, the water from the condenser is preheated, vaporized and superheated in the HRSG. The steam is then delivered to the HPT at 180 bar and 549 °C. After the expansion it is used to cool the burners and the HTT stages.

The major part of the cycle medium –the return flow after the HRSG- is compressed using the main cycle compressors C1 and C2 with intercooler and is fed to the combustion chamber with a maximum temperature of 600°C.

The cycle arrangement of the Graz Cycle offers several advantages: On one hand, it allows heat input at very high temperature, whereas on the other hand expansion takes place to vacuum conditions, so that a high thermal efficiency according to Carnot can be achieved. But only less than half of the steam in the cycle releases its heat of vaporization by condensation. The major part is compressed in the gaseous phase and so takes its high heat content back to the combustion chamber.

LARGE POWER GRAZ CYCLE WITH WORKING FLUID CONDENSATION/ EVAPORATION IN 1 BAR RANGE

In the basic S-Graz Cycle configuration the authors have proposed to expand the portion of the working fluid which has to be segregated from the circulating flow to be expanded down to condenser pressure. This flow contains the captured CO_2 and steam from the combustion as well as the cooling steam flow. Recent research [12] shows that difficulties in condensation arise in the formation of water films on the cooling tubes and in concentration of CO_2 forming a heat transfer hindering layer so that only a low heat transfer coefficient in condensation will be achieved. This results in excessively large condenser heat transfer surface and related high costs.

Therefore it was suggested in the Austrian patent of the Graz Cycle [13] to condense this mass flow at atmosphere, separate the combustion CO_2 and re-vaporize the water at a reduced pressure level using the condensation heat. The pure steam is then fed to a Low Pressure Steam Turbine LPST, where it can be expanded to a condenser pressure lower than that for the working fluid mixture.

Thermodynamic investigations presented at the ASME 2005 conference [11] showed that best results can be obtained for a dual pressure evaporation at 0.55 and 0.3 bar. But for these low evaporation pressures, large volume flows arise and the losses of live steam pipes and valves counteract the gains of this process.

Therefore a novel configuration is proposed in this work which allows single-pressure evaporation at a reasonable pressure level. The process is now split into the high-temperature cycle and a separate low temperature condensation process as shown in the simplified scheme of Fig. 2. The high temperature part consists of HTT, HRSG, C1/C2 compressors and HPT. Again condensation of the working fluid in the 1 bar range is proposed in order to avoid the problems of a working fluid condenser at vacuum conditions as described above. 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 is mandatory. The detailed flow sheet used for the thermodynamic simulation is included in the appendix (Fig. 10) and gives mass flow, pressure, temperature and enthalpy of all streams.

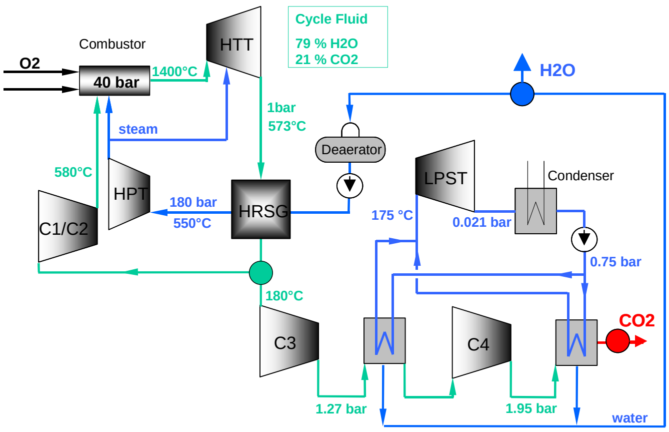

Fig. 2: Principle flow scheme of modified Graz Cycle power plant with condensation/evaporation in 1 bar range

This bottoming cycle operates by pure steam with extensively cleaned feed water and thus allows together with the very low cooling water temperatures of northern Europe to attain condenser pressures down to 0.02 bar.

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 1 bar for the condensing steam turbine.

At the first pressure level of 1.27 bar about 63 % 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 25 % 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 supplied at 1.9 bar for further compression is below 1 %. 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.

This two-step pre-compressed condensation counteracts the effect of sinking H_2O partial pressure due to condensed water extraction from working fluid and thus allows a reasonably high constant re-evaporation pressure of 0.75 bar for the bottoming steam cycle. Fig. 3 shows the heat – temperature diagram for this condensation/ evaporation process. After the start of water condensation, the working fluid temperature decreases only slightly, leading to relatively small mean temperature differences of 8 K in the first evaporator and 12 K in the second evaporator. After having condensed and separated most of the water content, the temperature of the working fluid decreases strongly in the water preheaters of the bottoming cycle (see Fig. 10).

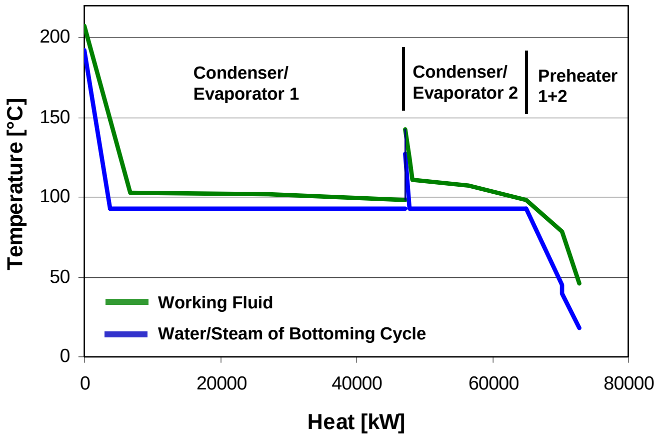

Fig. 3: Heat - temperature diagram of the condensation/ evaporation process

About three quarters of the condensation cycle mass flow is evaporated and superheated in the first condenser. It is mixed with the steam of the second condenser/evaporator unit providing steam 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 provides about 72 MW power output. A four-flow design is necessary to handle the high volume flow for a 400 MW Graz Cycle.

Steam is extracted at a pressure of 0.12 bar from the LPST and fed to the deaerator. The expansion line is to the major part in the dry steam region and crosses the Wilson line only before the last stage, so that only very fine droplets in the outer last stage sections are formed. A very high expansion efficiency hardly hindered by formation of humidity is to be expected.

Table 1 gives the power balance of the modified Graz Cycle plant of 400 MW net power output in comparison with the scaled-up basic configuration published in [11]. The heat input is the same for both cycles allowing a better comparison of the turbomachinery sizes. The C3 and C4 compressor have different tasks in both cycles. In the basic cycle they re-compress the separated CO_2 flow to 1 bar, whereas in the modified cycle they increase the working fluid pressure for a more favorable condensation/evaporation condition as described above. The modified Graz Cycle works with a smaller mass flow of the working fluid, so that both turbine and compressor total power decrease, whereas the net power output remains nearly the same. This leads to a similar thermal efficiency of about 66.5 % or an electrical net efficiency of about 64.65 %.

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 further reduces to 52.72 % for the basic cycle and 53.12 % for the modified cycle. This higher efficiency stems from a reduced CO_2 compression effort due to the higher supply pressure of 1.9 bar in the modified cycle. Thus the specific energy consumption reduces from 350 kJ/kg to 300 kJ/kg CO_2. The net efficiency of 53.12 % is higher than that of most other CO_2 capture technologies if evaluated under the same conditions, so that this new concept is worth a further feasibility investigation.

Table 1: Graz Cycle Power Balance

	Basic layout	New layout
HTT power [MW]	634.7	617.9
HPT power [MW]	48.0	49.9
LPT/LPST power [MW]	70.5	71.6
Total turbine power P_T [MW]	753.2	739.4
C1 power [MW]	137.2	131.1
C2 power [MW]	90.2	82.6
C3 power [MW]	11.5	8.9
C4 power [MW]	4.8	6.6
Pump power [MW]	5.3	5.5
Total compression power P_C [MW]	249.0	234.7
Net shaft power [MW] without mechanical losses	504.2	504.7
Total heat input Q_{zu} [MW]	758.6	758.6
Thermal cycle efficiency [%]	**66.47**	**66.52**
Electrical power output [MW] incl. mechanical, electrical & auxiliary loss	490.3	490.7
Net electrical cycle efficiency [%]	**64.63**	**64.68**
O_2 generation & compression P_{O2} [MW]	74.7	74.7
Efficiency considering O_2 supply [%]	**54.78**	**54.83**
CO_2 compression to 100 bar P_{CO2} [MW]	15.6	13.0
Net power output [MW]	**400.0**	**403.0**
Net efficiency η_{net} [%]	**52.72**	**53.12**

DESIGN CONCEPT FOR A VERY LARGE GRAZ CYCLE PLANT OF 400 MW NET OUTPUT

In this work the design concept for a Graz Cycle power plant of 400 MW electrical net output is presented. This power is derived from a 490 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

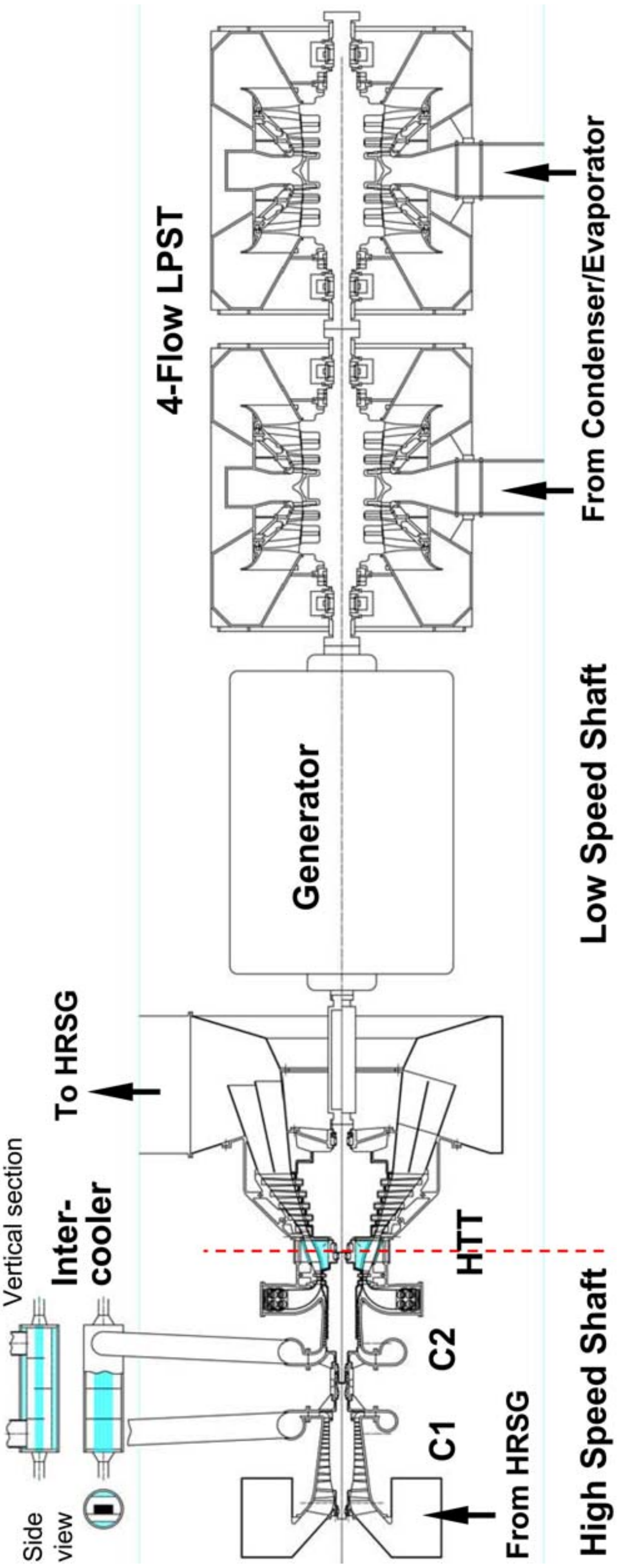

Fig. 4: Arrangement of the main turbomachinery for a 400 MW Graz Cycle plant

oxygen to the combustor at 42 bar and by the CO_2 compressor which has to deliver the captured CO_2 at a pipeline pressure of over 100 bar.

Gas turbines, compressors and combustors require the best flow development achieved up to now in gas turbine technology. In the course of this project our institute has found novel solutions for blade cooling, steam cooled combustor burner design and optimal rotor construction and rotor dynamics. The innovative cooling burner design helps to achieve the mentioned extreme high thermal efficiency (see [6] for details of the burner design), further improved by the positive change on the lower temperature end of the power cycle flow scheme as described above.

In this design proposal intensive use of steam cooling is made, not only for blades, but for all rotors in the high-speed high-temperature region. In that manner a solid and simple rotor design forged from one piece or welded from separate disks can be used with no internal friction between rotor disks as might be possible in a rotor assembled from separate disks. This type of rotor design provides for high blade load carrying capability with acceptable radial stress. The newly developed high chromium ferritic steels will be applied making use of their superior heat conduction and low thermal expansion properties. The relatively high speed selected provides for long blades in the last stages with high flow efficiency and low tip clearance loss.

The one-shaft system as in air-breathing gas turbines is not applicable since in the Graz Cycle system the amount of compressor flow volume is smaller and the number of stages required considerably higher. Therefore a much higher compressor speed as power turbine speed is an effective solution.

The main gas turbine components are arranged on two shafts, the compression shaft and the power shaft (see Fig. 4). The compression shaft consists of the cycle compressors 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. The two shafts are based on the same spring foundation. The intercooler between C1 and C2 is located on the fixed foundation.

<u>C1/C2 compressor design with intercooler:</u>

The working fluid compressor C1 is driven by the HTTC at 8500 rpm. The high speed poses a special problem for the first stage of C1 which has yet been solved by flow research and is now applied in many aircraft jet engines and also stationary compressor designs [14, 15]. The high tip Mach number on the

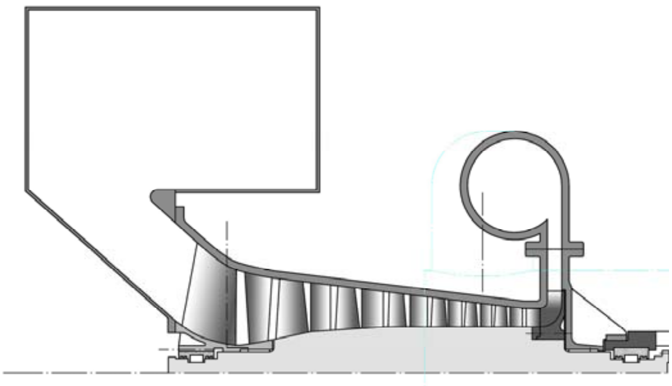

Fig. 5: C1 design with an uncooled drum rotor and an additional radial stage from nickel alloy, with radial diffuser and exit scroll to intercooler

first stage should not exceed the value of 1.4 for reasons of shock formation. With the help of a slight positive inlet swirl an inlet Mach number of 1.3 is designed.

Compression at C1 starts at 106°C and reaches 442°C at the outlet to the intercooler. For reasons of rotor dynamics the shafts of C1 and C2 are separated with intermediate bearings and a solid coupling. This makes the transfer of cooling flow difficult, so that cooling of the drum rotor of C1 will not be applied. This is possible by a combination of rotor materials.

The first part with seven axial stages is a ferritic steel drum, which reaches only 390°C. This material can be highly stressed without creep at temperatures below 400°C. By the application of a final radial wheel which has to be milled separately from nickel alloy and which is mounted to the main drum by elastic centering completes the rotor construction of C1, as shown in Fig. 5. The radial wheel with a wide vaneless diffuser and scroll improves the flow transfer to the intercooler.

The inlet temperature to C2 is somewhat lowered by the intercooler but still reaches 380°C. During course of compression the working fluid reaches an outlet temperature of 580° C, so that from the second stage onwards cooling has to be applied on the rotor surface of the bladed annular flow channel. Seven axial stages with a stepwise decrease of blade length from 90 to 40 mm are supported on a drum rotor with disk extensions of constant diameter. Fig. 6 shows the C2 rotor with the counter flow of cooling steam on the drum surface. It is guided by means of openings under the bladed disk extensions and is prevented by sealing strips from flowing into the main flow. These strips are carried on both sides of the stationary blades. By proper selection of the feed pressure this flow can be optimized at a small penalty in dilution of the main flow.

Excellent flow properties of this compressor can be expected due to its blade mounting on a stiff rotor with very small radial tip clearances and flow losses together with an aspect ratio of outer to inner flow radius of 440 to 400 mm.

The intercooler requires some development work. The fluids on both sides are unconventional insofar as the working fluid on one side is to be cooled by high pressure steam on the other side. Heat transfer from compression work to steam superheat is thus achieved. The authors can point at previous development work at their institute in a similar problem, i.e. the design of an 850°C steam plant in double loop configuration as published by Perz et al. [16], where many boiler heat transfer problems had been treated.

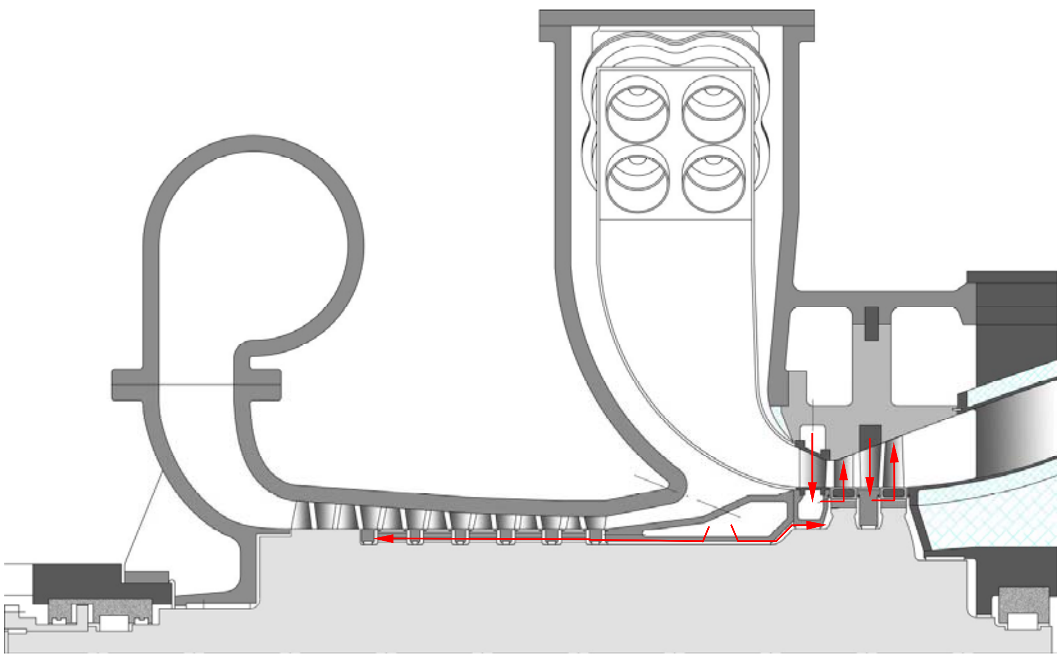

Fig. 6: Design of C2 drum rotor with cooling steam flow arrangement, combuster and HTTC

In the case given the intercooler is thermodynamically part of the HRSG superheater and is thus arranged close to the HRSG. Its cooling flow is steam of 196 bar pressure. It is designed as a solid tube which should be supported on solid ground foundation. The heat transfer surface is realized by 180 tubes of 3.1 m length held in support plates which guide the working gas flow from C1 outlet to C2 inlet. The outer shell of the intercooler is internally insulated and is connected by ample flow areas in flexible scroll and tube arrangements to both compressors (see Fig. 4).

HTT compressor turbine (HTTC)

The same drum rotor either forged in one piece or welded up from separated disks carries not only the C2 but also the compressor turbine HTTC. The flow design of the HTTC will be a two-stage reaction turbine with 50 % reaction at the mean section of both blade rows. The high rotor speed of 8500 rpm as mentioned before provides for long blade lengths, i.e. a first stage blade of 100 mm and a second stage blade of 164 mm with an inner radius of 533 mm (see Fig. 7). This results in excellent flow properties in subsonic condition and together with the high reaction of the blade on all radii (55 % at mean section and at least 25 % at hub) a high blade flow efficiency is expected. Low tip clearances are applied also contributing to this goal and can be achieved by the excellent rotor dynamics of the stiff drum rotors and very careful blade cooling.

The high speed and power of this turbine is made possible by ample steam cooling. Nozzles and blades are cooled in conventional serpentine passage design with holes as well as the rotor inlet edge as shown in the cooling arrangement of Fig. 7. Rotor cooling steam is supplied along the whole drum surface. It is fed into a labyrinth seal in the inner range of the combustion chamber allowing the steam to flow to both sides. One flow is directed backwards under the dump diffuser into the outer surface of the C2 providing cooling steam as described above. The main amount of cooling steam flows along the rotor drum at the inner radius of the combustor casing towards the first disk of the HTTC.

The first nozzles are hollow with proper cooling passages and are cooled by steam fed from the casing outside in radial inwards direction. The steam is collected in a chamber of the diaphragm just opposite the first blade root. Via nozzles, blowing in direction of rotor speed, the cooling steam is then fed to the lower part of the blade fir-tree root. From there it flows along the serpentine passages under pumping action of the rotor wheel and is delivered to the blade surface via laser drilled holes to form the conventional cooling films at the appropriate locations of the blade. The second guide vanes are supplied with cooling steam which is fed into the outer rim of the diaphragm. There it flows radially inwards also to supply the rotor surface in-between stages and the inlet to the second stage blades which are also built with serpentine passages and the appropriate cooling holes. In principle this design was applied for the well-known gas turbine model GT10, originally designed by F. Zerlauth [17].

In terms of rotor dynamics the drum rotor of C2 and HTTC will be designed for the high stress considering the effect of steam cooling on all surfaces. Stiff bearing shaft extensions and solid double-lobe oil bearings provide for high shaft and high bearing stiffness in order to have all critical speeds sufficiently high above running speed.

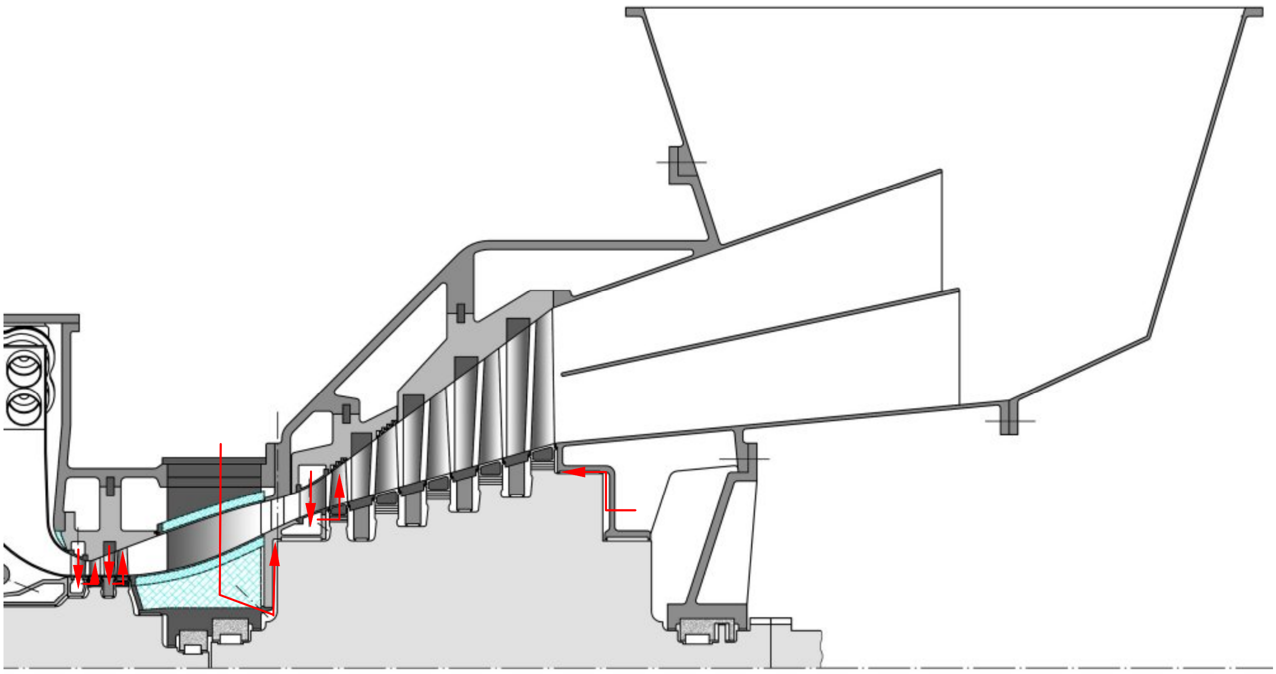

Fig. 7: Design of two-stage HTTC and 50 Hz HTTP

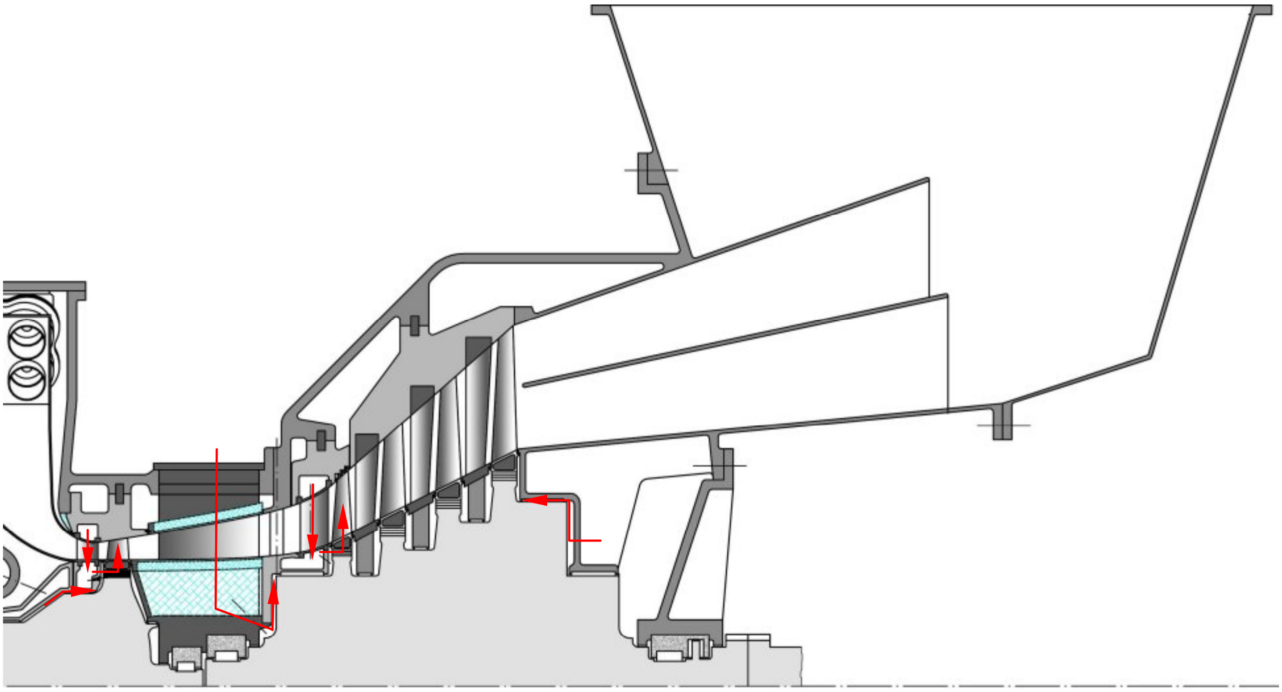

Fig. 8: Design of transonic one-stage HTTC and 60 Hz HTTP

HTTC alternative transonic stage design

The authors' institute has done extensive development work for the design of transonic turbine stages. Not only several computer programs have been developed for investigating three-dimensional transonic flow, but a unique test installation for transonic stages was built where many effects of unsteady viscous transonic flow were investigated (e.g. [18, 19]). The test installation has aided the development of industrial gas turbines and is now in use for several EU projects.

A novel innovative cooling system has also been developed and could be applied here in order to save cooling medium, high temperature material and cost of manufacture at the same time providing most effective blade cooling at the blade leading edge in transonic flow [20, 21]. The design could follow the development path of General Electric in providing thermal barrier coating on the rotating blades since these are free of the multitude of cooling holes and are supplied only by low number of slots creating cooling steam films covering the whole surface.

Therefore, alternatively the HTTC expansion could also be done with one transonic stage as shown in Fig. 8. This can be achieved by a higher radius and stage loading at a somewhat reduced degree of reaction. Such a stage would sit on the same rotor as described before and it would have a mean radius of 750 mm at a blade length of 120 mm. A further advantage of a transonic stage would be the much smaller radius difference from compressor turbine outlet to power turbine inlet, depending also on the speed of the power turbine for which design proposals for 50 and 60 Hz are presented here.

HTT power turbine (HTTP)

A gas turbine system with two shafts at highly different speed as it has to be built here, requires an intermediate bearing to be arranged right between the stages of compressor turbine outlet and power turbine inlet. The flow of gas transmitted is at very high velocity, at temperatures of 1075°C and at a pressure of 14 bar. A conventional design would provide an outlet diffuser, an outlet casing, a transition duct and an inlet casing in between this gas turbine parts. Frictional loss, heat loss, even with internal and external insulation, would be unavoidable. In previous design solutions for industrial size turbines with almost the same cycle conditions the authors have proposed a single overhung disk with a transonic stage for the compressor turbine and one or two overhung disks for the power turbine directly opposite to take over the gas flow in a common casing [10]. This solution requires a high speed power turbine which is only possible to build relying on gears of high speed and high power. The power output of 92 MW in that case made it possible since gas turbines transferring around 100 MW from gas turbine speeds at 5400 rpm to 3000 rpm are in use in industry.

The large power output in the case presented here forbids the use of gears for such high-speed power transfer. Therefore the possible electrical frequencies of 50 Hz in Europe and 60 Hz in USA and western hemisphere were investigated. The power turbine is proposed with a strong change of inner radius on a solid shaft. Five stages are necessary for the 50 Hz design of Fig. 7 and four stages for the 60 Hz of Fig. 8. So the axial outlet speed should be kept at medium value in order to reduce the exhaust loss, to reduce axial diffuser exit length and to facilitate the flow transfer to the HRSG inlet.

The design proposed provides last blade lengths of 750 mm at 50 Hz and of 600 mm at 60 Hz, both at 1300 mm inner radius. At the inlet the inner radius at 50 Hz is somewhat larger than the HTTC outlet, but at 60 Hz, together with a transonic HTTC, it provides a flow path at almost the same radius as shown in Fig. 8.

The intermediate bearing casing in its hot environment has to be insulated on its outer surface in a mode of insulation withstanding the friction of the hot outer flow. The same holds true for the three supporting ribs, which have to provide ample inner space for transfer of oil, cooling air and steam leakage outlet from labyrinths on both shaft sides as well as for monitoring equipment. At the same time the bearing should be as short as possible and the ribs should provide only a minimum of flow resistance. Certainly this is an object that deserves intensive flow, stress and heat transfer deliberations.

The thrust equalization of both types of power turbines cannot be made in the conventional manner of steam turbines. A balance piston requires a diameter of about the mean blade mean diameter to give proper balance of axial forces. In this case such a design would require an unacceptable flow turn and deviation of the hot gas flow. (In a one-shaft gas turbine the problem does not exist, since compressor thrust and turbine thrust balance each other.) On the other hand, to carry the axial thrust of a large power turbine especially in the conical form is impossible for oil thrust bearings. Size and oil friction power loss would be too high. Therefore a stepped labyrinth on the exhaust side of the rotor drum is proposed as shown in Figs. 7 and 8, which is supplied with internal steam pressure to provide for the necessary thrust equalization. The steam supply feeds also the cooling flow which is led along the rotor drum surface under the root sealing plates for the last and the penultimate stage, whereas cooling flow to the first and second stage is supplied via the hollow nozzle blades to an inner diaphragm cavity from which the inflow to the hollow rotor blades is effected. Power turbine thrust bearing is arranged outboard of casing in vicinity of steam operated balance piston (see Figs. 7 and 8).

Low Pressure Steam Turbine (LPST)

The LPST is fed with steam of 0.75 bar and 175°C. Expanding the steam to a condensation pressure of 0.021 bar leads to a high volume flow. At 50 Hz a four-flow design with three stages, as shown in Fig. 4, is able to handle the high steam flow with excellent efficiency. The last stage is transonic with a blade length of 970 mm. In the shaft arrangement this steam turbine is coupled to the far side of the main generator.

HPT

The HPT is a standard high-speed back-pressure steam turbine of 50 MW power output for which many designs are in the market. A geared type seems to be a superior solution since better flow efficiency and operability due to nozzle boxes and low number of stages with long blades and low leakage loss can be achieved. It can be coupled to the far end of the LPST or can be used to drive a separate smaller electric generator.

Compressors C3 and C4

The delivery compressors C3 and C4, which increase the pressure of the working fluid prior to condensation in order to obtain better evaporation conditions for the bottoming steam cycle, are also needed to vent the internal volume before start up. They are driven by two separate speed-controlled motors.

Combustion

The combustion chamber and burner design proposed has been thoroughly tested in science of combustion. Research partners have run CH_4/oxygen burners in a steam environment successfully [22, 23]. The authors' proposal [7] of setting a separate oxygen and fuel inlet in the center of a strong steam vortex in a large number of separate burners within the combustion chamber provides for easy control of amount and ratio of oxygen and fuel together with ignition and flame observation. The steam vortex keeps together both reactants. The independent supply of both reactants together with the high flame speed caused by pure oxygen lets expect improvements compared to the otherwise acoustic vibration prone conventional low-NOx combustion chamber flows.

PART LOAD AND START-UP

In part load the maximum gas turbine temperature can be lowered by reduction of heat input. With the free running compressor shaft adjustment of flow and temperature can be effected precisely in operating IGVs and turbine valves accordingly.

To keep the working fluid and the CO_2 delivery line free of nitrogen in each cold start careful scavenging of all internal volumes in turbomachinery, HRSG and piping has to be done. Since fuel and oxygen input can be governed for each burner quadruple ignition and safe operation of flames is secured. See further details of the start-up process in the appendix.

ECONOMIC EVALUATION

Despite the high efficiency and the positive impact on the environment by a Graz Cycle power plant, a future application of this technology and an erection of a power plant mainly depends on the economical balance. The main indicator characterizing the economical performance of a power plant for CO_2 capture are the mitigation costs. They represent the increased capital and operational costs incurred by new and additional equipment and lower cycle efficiencies in relation to the CO_2 mass flow avoided. The CO_2 captured has an economic value of about 10 $/ton, if it can be used for enhanced oil recovery (EOR) or of about 30 $/ton in the future CO_2 emission trading scenario. These prices show the current threshold for the economic operation of zero emission power plants.

In order to estimate the mitigation costs for a Graz Cycle plant, an economic comparison with a state-of-the-art combined cycle power plant of 58% efficiency is performed. The economic balance is based on following assumptions: 1) the yearly operating hours is assumed at 8500 hrs/yr; 2) the capital charge rate is 12%/yr, which corresponds to an interest rate of 8 % over a depreciation period of 15 years; 3) methane fuel costs are 1.3 ¢/kWh_{th}; 4) the investment costs per kW are the same for the reference plant of about 400 MW net power output and the Graz Cycle plant (see below); 5) additional investment costs are assumed for the air separation unit (ASU), for additional equipment and CO_2 compression to 100 bar (see Table 2 [24]); 6) the costs of CO_2 transport and storage are not considered because they depend largely on the site of a power plant.

Table 2: Estimated investment costs

Component	Scale parameter		Specific costs
Reference Plant			
Investment costs	Electric power	$/$kW_{el}$	414
Graz Cycle Plant			
Plant investment costs	Electric power	$/$kW_{el}$	414
Air separation unit [24]	O_2 mass flow	$/(kg O_2/s)	1 500 000
Other costs (Piping, CO_2-Recirc.) [24]	CO_2 mass flow	$/(kg CO_2/s)	100 000
CO_2-Compression system [24]	CO_2 mass flow	$/(kg CO_2/s)	450 000

The assumption of similar investment costs for a conventional and a Graz Cycle power plant is based on a comparison with typical turbomachinery sizes for a 400 MW combined cycle plant as given in Table 3. It shows that the turbine power and the HRSG is of similar size, whereas the compressor power is remarkably smaller. On the other hand the Graz Cycle needs a larger generator due to the additional power consumption for ASU and CO_2 compression. Development efforts needed especially for HTT and combustor are not considered in the investment costs.

Table 3: Comparison of equipment size for a 400 MW plant in terms of power

	Conventional CC plant	Graz Cycle plant
turbine of "gas turbine"/ HTT	667 MW	618 MW
compressor of "gas turbine"/ C1+C2+C3+C4	400 MW	232 MW
steam turbine/ HPT+LSPT	133 MW	120 MW
HRSG	380 MW	360 MW
Generator	400 MW	490 MW

Table 4 shows the result of the economic evaluation. Compared to the reference plant, the capital costs are about 70 % higher only by considering the additional components for O_2 generation and CO_2 compression. So they contribute mostly to the difference in COE. The fuel costs have the major influence on the COE, especially for syngas firing, but they do not differ largely between reference and Graz Cycle plant. The O&M costs are assumed 15 % higher for a Graz Cycle plant due to the operation of additional equipment.

Based on these assumptions, the COE of a methane fired Graz Cycle plant of 53.1 % net efficiency is 0.72 ¢/kWh_{el} higher than for the reference plant. The mitigation costs are 21.0 $/ton of CO_2 avoided, if CO_2 liquefaction is considered. This value is clearly below the threshold value of 30 $/ton showing the economic potential of the Graz Cycle.

Table 4: Economic data for a 400 MW Graz Cycle plant

	Reference plant	S-GC base version
Reference Plant		
Plant capital costs [$/$kW_{el}$]	414	414
Addit. capital costs [$/$kW_{el}$]		288
CO_2 emitted [kg/kWh_{el}]	0.342	0.0
Net plant efficiency [%]	58.0	53.1
COE for plant amort. [¢/kWh_{el}]	0.58	0.99
COE due to fuel [¢/kWh_{el}]	2.24	2.45
COE due to O&M [¢/kWh_{el}]	0.7	0.8
Total COE [¢/kWh_{el}]	**3.52**	**4.24**
Comparison		
Differential COE [¢/kWh_{el}]		**0.72**
Mitigation costs [$/ton CO_2 avoided]		**21.0**

The results of the economic study depend mainly on the assumptions about investment costs, fuel costs and capital charge rate. A cost sensitivity analysis performed in [11] showed that a variation of the capital costs has the main influence on the economics, since they contribute most to the mitigation costs. Unfortunately, there is a large uncertainty of these costs. A survey of the ASU costs vary in the range of 230 to 400 $/$kW_{el}$ (the same price as for a complete power plant). These costs are for a cryogenic ASU as used e.g. in steel industry for half a century. There is certainly a potential for effectivity increase. Oxygen from membranes which are under intensive development now are not yet available for plants of the output in discussion. The ASU appears to be a decisive cost factor. Only considering its cost variation, the mitigation costs vary between 21.0 and 27.9 $/ton CO_2 for the methane fired plant (see Fig. 9).

This high sensitivity to the capital costs shows the dilemma in performing an exact economic evaluation, since their estimation for a Graz Cycle power plant is very difficult because of new turbomachinery components. But the authors

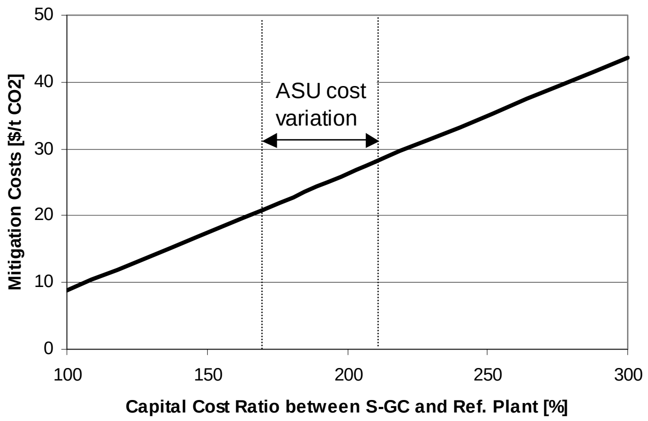

Fig. 9: Influence of capital costs on the mitigation costs (CO_2 provided at 100 bar)

claim that their design of high-speed transonic stages with innovative steam cooling allows a cost-effective manufacture. In these considerations about the height of additional investment costs, a further advantage of the Graz Cycle, the almost NOx-free combustion was not evaluated. According to [25] 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.

CONCLUSIONS

The Graz Cycle is an oxy-fuel power cycle with the capability of retaining all the combustion generated CO_2 for further use. In order to avoid the difficulties of condensation of water out of a mixture of steam and incondensable gases at very low pressures, a modified cycle configuration was presented with condensation in the range of 1 bar. It allows a separate bottoming steam cycle with reasonably high pressures and efficiencies, so that a high net cycle efficiency above 53 % can be expected.

The output of the Graz Cycle plant is raised from industrial size to 400 MW net output. A design concept for this size is presented with two shafts. A fast running compression shaft is driven by the compressor turbine HTTC, whereas the power shaft comprises the power turbine HPT and the LPST.

In an economical analysis the Graz Cycle power plant is compared with a reference plant. The resulting mitigation costs are in the range of 20 – 30 $/ton CO_2 avoided depending on the costs of the ASU and thus are below a threshold value of 30 $/ton CO_2 (assumed for future CO_2 emission trading).

The authors have thus presented a design solution for an oxy-fuel CO_2 retaining gas turbine system which can by acceptance of international gas turbine industry be put into operation within a few years. The authors believe, that this system is equal in thermodynamic performance to any other proposal in the field of CO_2 reduction and is superior in applying gas turbine experience and research accumulated to our day.

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APPENDIX

The CO_2 retaining gas turbine power plant requires auxiliary equipment for start up and part load which have not yet been described fully in previous publications.

Details of installed equipment

The flow is to be governed by inlet valves to high and low pressure steam turbines and by inlet guide vanes to the first stages of each compressor. In particular these are:

- Governing valves to HPT inlet
- Governing valves to LPST inlet
- IGVs to compressors C1, C2, C3 and C4
- Mechanical turning gear to all shafts
- Electric intermediate drive for compressor C3 and geared C4
- Starter motor for high speed shaft consisting of compressors C1, C2 and HTTC
- Ignition and flame watch for each burner assembly in combustion chamber.
- Control of cooling steam flow to HTT and combustion chamber
- Gas temperature control at outlet of HTTP and HTT last stage
- Control for fuel and oxygen supply.
- Steam sealing for all external labyrinths with external steam suction to condenser

For the start up the authors assume the plant to be erected, all components ready for operation. In particular these are:

- Fuel gas pressure available at main valve.
- Cooling water supply ready to be started
- All internal volumes in turbomachinery, boiler, tubes and drums, condenser and feed pumps are cleaned and ready to be filled with the relevant flow medium
- Lubrication system filled with lubricating oil, pumps and piping ready to operate.
- Connection to ASU ready and oxygen compressors ready to start
- Connection to CO_2 delivery pipeline ready, CO_2 recycling supplied with main shut-off valve closed till proper delivery pressure is attained.
- Bypass to stack after C3 and C4 compressor for vent of internal air to atmosphere , valves closed till start
- External auxiliary steam supply (20 bar, 300°C approx.) operable to supply steam in a quantity of flow volume equivalent to HTT and combustion chamber and burner steam cooling flow
- Feed water tanks filled with sufficient amount of clean feed water for HSRG and low pressure steam plant.

Start up

The first start up procedure is envisaged as follows:

1. Supply lubricating oil to all bearings and gears
2. Start turning gears on all shafts
3. Control all blade clearances
4. Seal all labyrinths with auxiliary steam
5. Open bypass steam supply and inject auxiliary steam to combustion chamber burners and HTT inlet cooling flow passages
6. Start C3 and C4, and vent internal volume of turbomachinery and HSRG via bypass valve to stack
7. Continue so with HPT valves open and steam flow through boiler tubes till all air is expelled from internal volumes.
8. Close IGVs and turn C1 and C2 to higher speed by increased auxiliary steam input to HTT compressor turbine to drive the compressor shaft up to 1/3 speed .
9. Close HPT valves, supply feed water to Benson boiler tubing, check for flow of return bottle and establish continuous feed water flow through it.
10. Check auxiliary steam flow through HTT cooling passages
11. Check for minimum nitrogen content (air freedom)
12. Internal surface temperature is around 300°C as given by auxiliary steam input.
13. Start ignition on two opposite burner quadruples. Check security of fuel and oxygen supply, maximum temperature and burner driving steam flow. Ignite all other burners. Contain flame temperature below admissible maximum.
14. Main shaft starts turning, heat is supplied to boiler tubes, evaporation and minimum superheat is to be reached.
15. Open HPT valves to take over steam flow from HSRG
16. Increase fuel and oxygen supply and run compressor shaft up to three quarter speed.
17. Main power shaft attains synchronizing speed , synchronization of main generator to grid.
18. In the meantime low pressure steam plant is to be made operable. Cooling water pumps started, air suction by vacuum pump established, steam feed water pump operated to start evaporation. LPST valves are partly closed .
19. All internal volumes filled with auxiliary steam allowing to close the C3 and C4 bypass and to start one bar condensation with feed water flow to HRSG
20. Check all temperatures and increase heat input towards continuous operation with careful observation of temperature limits in rotor blades and disks. Check boiler feed water flow and steam conditions
21. Change cooling steam supply to combustion chamber, HTT blading and rotors from auxiliary steam to HRSG steam supply via HPT outlet flow.
22. All other systems are to be controlled accordingly and delivery of electricity and CO_2 to the grid and pipeline can be started.

Restart after short intermission (hot start) is to be done taking into account the availability of the auxiliary systems mentioned above with speed of loading adjusted to the measured temperature distribution in the critical machine parts, bladings rotors and casings. In such a hot start the initial temperature difference between inlet flow and metal surface should be kept low and temperatures could and should be raised quickly to the situation of continuous operation.

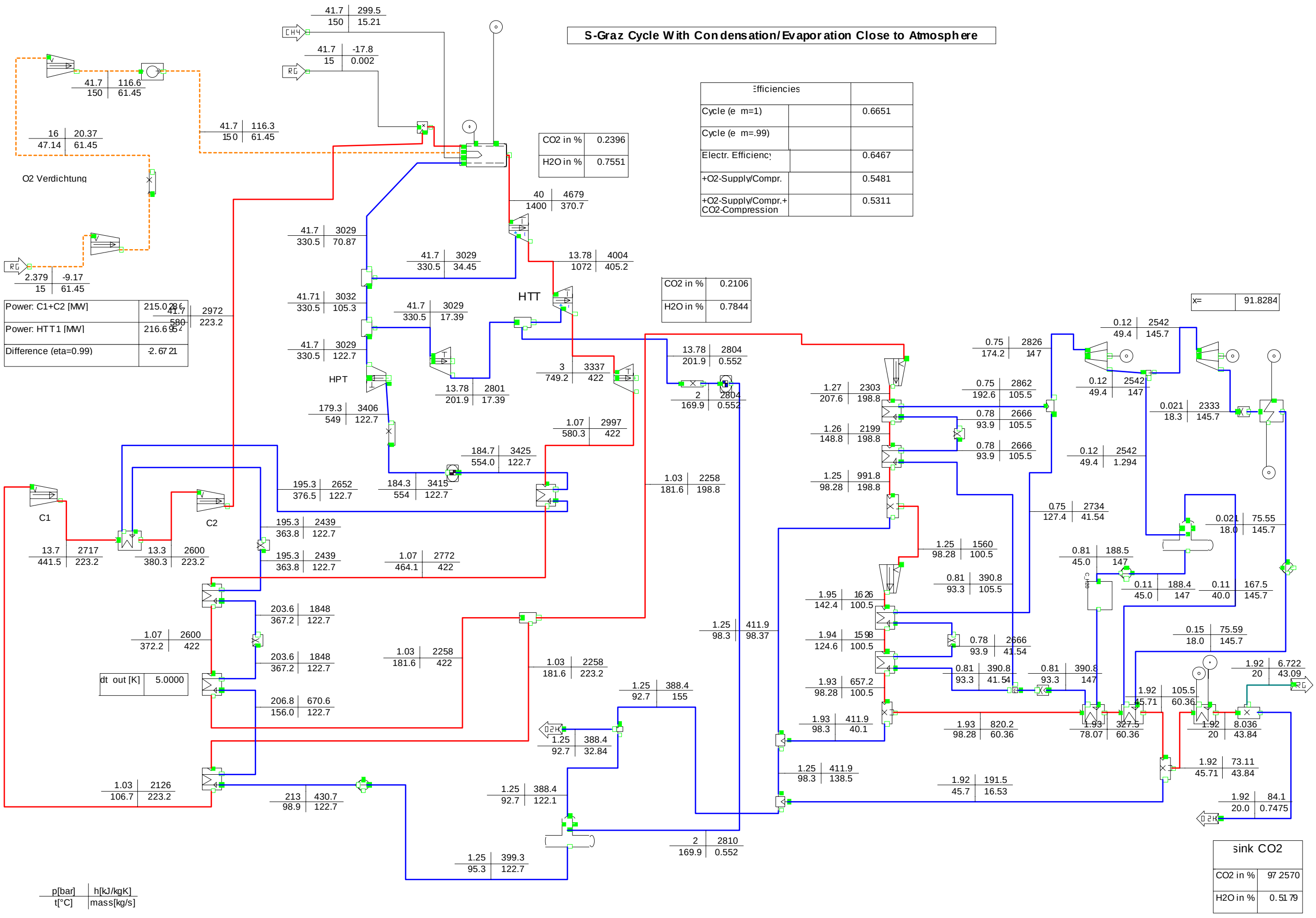

Fig. 10: Detailed thermodynamic cycle data of a 400 MW S-Graz Cycle Power Plant fired with methane