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CONCEPTUAL DESIGN FOR AN INDUSTRIAL PROTOTYPE GRAZ CYCLE POWER PLANT

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ABSTRACT

The gas turbine system GRAZ CYCLE has been thoroughly studied in terms of thermodynamics and turbomachinery layout. What is to be presented here is a prototype design for an industrial size plant, suited for NG-fuel and coal and heavy fuel oil gasification products, capable to retain the CO_2 from combustion and at the same time able to achieve maximum thermal efficiency.

The authors hope for an international cooperation to make such a plant available within a few years.

INTRODUCTION

The lead author and several coworkers at Graz university started as early as 1985 to develop power cycles without any emissions to atmosphere. At CIMAC conference Oslo a first proposal for a H_2-O_2 internally fired steam cycle (Jericha, 1985) was presented. An improved solution with high efficiency was shown at ASME Nice (Jericha and Ratzesberger, 1989) and detailed design of such a combustion chamber was presented to CIMAC Florence (Jericha and Starzer, 1991). On the basis of this work discussion with Japanese research institutions followed from which a cooperative proposal for a 500 MW plant by MHI resulted (Aoki, et. al., 1998).

In this stage it became clear that the fuel supply for such power plants – high output solar plants producing hydrogen and oxygen by splitting water – would not be available in time. So as a first step it seems reasonable to use fossil fuels to fire CO_2 retaining gas turbines. Essential the same cycle configuration for fossil fuels instead of hydrogen was proposed by Jericha et al. at CIMAC Interlaken 1995 and ASME, 95-CTP-79, Vienna 1995. (Here the technical term "closed cycle gas turbine" has recently gained a new meaning in so far as now a system without any emissions to atmosphere is characterized. Also the technical term "sequestration" for storing away has come into use, but the authors prefer the term "retention" in cases where useful technical application of CO_2 seems possible).

Fossil methane, CH_4 the main component of natural gas is a potential fuel as well as fuel gases from gasification processes, in particular fuel gas from oxygen blown coal or from oxygen blown heavy oil, residues from refinery processes. The change of cycle medium from almost pure steam to a mixture of varying compositions of CO_2 and H_2O requires changes in the layout of turbo machines which partly induce additional difficulties but in the case of the high temperature turbine facilitate the design task. Other scientific work mostly concerning carbon dioxide cycles has been published recently also proposing to use fuel gases from coal gasification in a more efficient way (Ulizar and Pilidis, 1997).

The authors have deliberated the design of a hydrogen and oxygen fired gas turbine combustion chamber and have done research in cooperation with DLR Lampoldshausen, already in the early 1990ies and the problem of attaining a high combustion efficiency has always been an issue together with optimal cooling of burners also avoiding dissociation of reaction partners. Recently discussion with Japanese co researchers has presented us with novel solutions for CH_4 – O_2 combustion for our common problem in the same cycle configuration. (H. Inoue, et al., 2001). According to these results the prospects for CH_4 – O_2 combustion in a GRAZ CYCLE combustion chamber are very promising. Again it should be mentioned here that this name was given by our Japanese discussion partners.

NOMENCLATURE

Machines

HPT		High Pressure Turbine
HTT		High Temperature Turbine
LPT		Low Pressure Turbine
C1		Compressor 1
C2		Compressor 2
C3		Compressor 3

Symbols

m	[kg/s]	mass flow
p	[bar]	pressure
t	[°C]	temperature
V	[m³/s]	volume flow
P	[kW]	Power
n	[rpm]	speed
z	-	number of stages
D	[m]	diameter
L	[m]	blade length

Subscripts

cool	fictional turbine, represents expansion of cooling medium
lp	low pressure
hp	high pressure
1	inlet, first stage
2	outlet, last stage
i	inner
o	outer
m	mean

FORMULATION OF DESIGN TASK

A thermal power plant prototype design shall be developed with beneficial properties.

- Very high thermal efficiency reaching top values of combined cycle gas and steam industrial plants.
- Avoidance of any emissions to atmosphere, retention of CO_2 for further technical application. Small amounts of nitrogen contained in the fuel gas as well as in the oxygen supply capable of forming nitrogen oxides in the combustion chamber shall be continuously sequestered from the cycle medium together with combustion generated CO_2.
- Avoidance of solvents, membranes etc., use of progressively developed gas turbine components only.
- Use of fossil fuels such as natural gas or gasification products of coal or heavy fuel residues.

ADVANTAGES OF THE PROPOSED PROTOTYPE DESIGN

- The highest cycle pressure only slightly surpasses the maximum pressure used in aircraft engines manufactured in large numbers.
- Highest temperature is in the frame of high power stationary gas turbine practice.
- The cycle medium a variable mixture of CO_2 and H_2O allows to build the high temperature turbine with a low number of stages.
  The first stages can be effectively cooled by steam in an innovative cooling system developed by our institute (Woisetschläger, et al., 1995, Jericha, et al., 1997, Jericha and Neumayer, 2000, Moser, et al., 1998 and 2000).
- The heat exchangers incorporated within the cycle work in closed cycle flow with high temperature differences. Thus economic low cost heat exchangers with limited amount of high temperature tube alloy can be built. These can be kept clean on both sides and operate with high heat transfer.
- The high efficiency of the cycle follows mainly from the low compression work. Only the CO_2 component in this temperature range being a non condensable gas has to be compressed with turbo compressors, whereas the steam part of the cycle medium is generated from feed water – extensively cleaned – that can be pumped in order to generate high temperature steam.
- An other thermodynamic advantage in comparison to conventional combined cycle plant is the avoidance of temperature differences, and the relatively low volume flow in the low temperature region.

DESCRIPTION OF CYCLE

Fig.1a shows the design data of the proposed heat cycle in digital form printed within the cycle scheme. The values of pressure, enthalpy, mass flow and temperature are given at the specific locations in the cycle scheme, an arrangement which has been published before. (Jericha et al., 2000). The T-s diagram given in Fig. 1b visualizes the same thermodynamic data, temperature differences and flow relations in compression, expansion and steam generation.

As an example of the proposed fuels a typical composition of an oxygen blown coal gasification fuel gas is taken (Fuel gas mole fractions: 0.1 CO_2, 0.4 CO, 0.5 H_2). The use of natural gas as fuel induces only small differences in the data of the cycle scheme.

If the required pressurized cycle oxygen is made available at the burner inlet and for the cycle data as shown in Fig. 1a, the thermal efficiency can be computed to a value of 63.9 %. The effort for air separation in modern plants is in the range of 0.3 kWh/kg oxygen. Taking into account that the layout of the associated air separation plant has often to be much larger than necessary to provide oxygen for the power cycle alone, most advanced performance can be assumed. Pressurized oxygen is needed for the gasification and pressurized nitrogen obtained simultaneously is needed for gasifier sealing purposes. Also nitrogen is a valuable agent in enhanced oil recovery. (The world's largest air separation plant provides a high amount of pressurized nitrogen for enhanced oil recovery off shore, as reported by "Gas Turbine World" May 1998, p. 51).

This situation justifies to allocate the air separation power effort half to half to the produce of oxygen and nitrogen. This as shown in the result in Table 1, as 4506.4 kW as power requirement for cycle oxygen generation.

Thus with mechanical and generator efficiency a cycle efficiency of 59.93 % is obtained.

In case the cycle oxygen compression effort, atmospheric to burner inlet of 3640 kW is also included, the final cycle efficiency turns out to be 57.51 %, quite near the optimum conventional combined cycle plants have recently achieved.

Due to the composition of the cycle medium – a mixture of CO_2 and H_2O at varying ratio and due the high compressibility of the gaseous component a high number of turbo machinery casings at several speeds is necessary. As will be demonstrated later on all turbomachines are thought to have a layout that provides optimum flow properties and thus highest stage efficiencies.

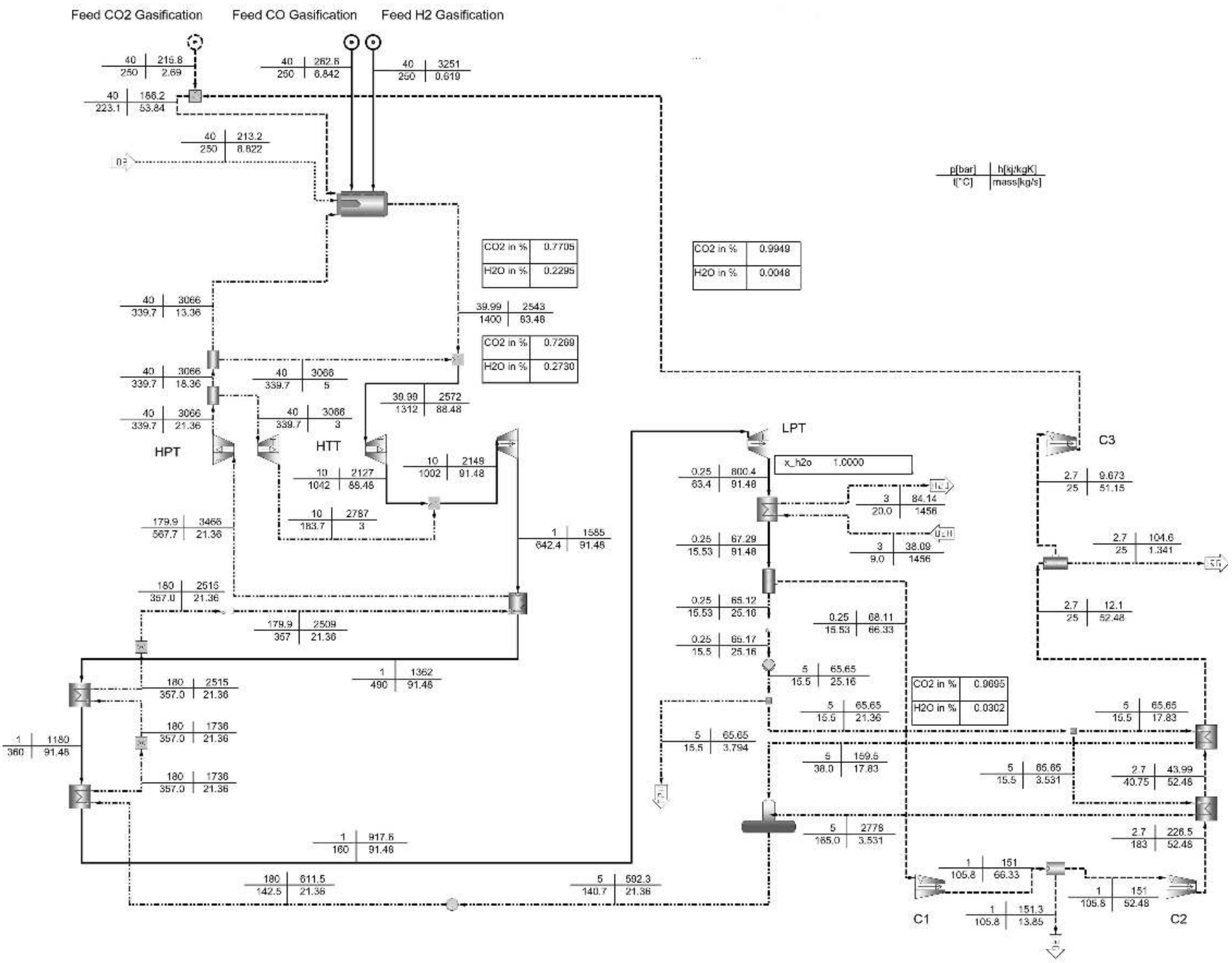

Fig. 1a: Details of proposed heat cycle

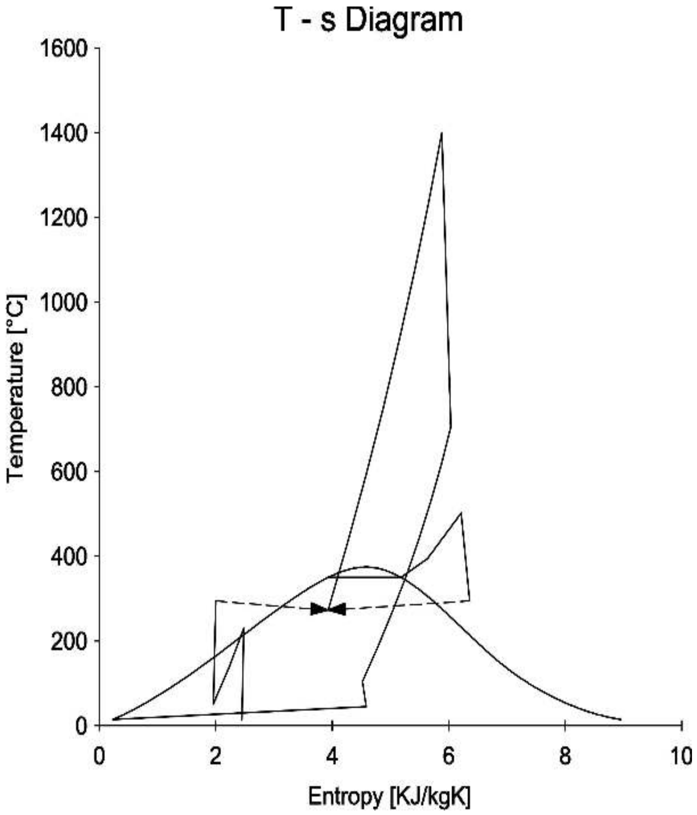

Fig. 1b: T–s Diagram

From the cycle scheme pressures, temperatures and volume flow at inlet and outlet of each turbomachine is shown in Table 1 which gives also rotor and blading dimensions, speed and power input and output respectively.

See Fig. 2 for general arrangement of a 92 MW turbo set and Fig. 3 for details of the high speed shaft.

An industrial size unit below 100 MW has been selected in order to keep within available size of power supply for industrial plant as well as to stay within the range available for gasification and air separation installations.

DESCRIPTION OF DESIGN DETAILS

Combustion Chamber:

In a conventional air breathing machine the oxidant and the flame cooling medium are identical. In combusting fuel care has to be taken to avoid formation of nitrogen oxides and to attain high combustion efficiency at the same time. But within ignition limits any fuel particle that might have strayed from the main flame has still the chance to burn in the high amount of oxygen flow in the cooling medium.

The task here is different, very low content of nitrogen together with operation in closed cycle mode alleviates the NO_X problem. A fuel with a high hydrogen content insures ignition in a wide temperature range and a high flame speed. (Döbbeling, et alia, 1996). Flame speed further increases in burning with pure oxygen.

Table 1:

Turbines Total Turbine Power 111081 kW						
Turbine Name		HPT	HPT cool	HTT hp	HTT lp	LPT
m	kg/s	21.36	3.0	88.48	91.48	91.48
p_1	bar	179.9	40.0	39.99	10.0	1.0
p_2	bar	40.0	10.0	10.0	1.0	0.25
t_1	°C	567.7	339.7	1312	1002	160
t_2	°C	339.7	183.7	1042	642.4	63.4
V_1	m³/s	0.4125	0.195	9.264	31.50	106.67
V_2	m³/s	1.387	0.590	30.65	226.05	331.71
P	kW	8544	837	39374	51595	10731
n	rpm	20000	20000	20000	12000	3000
z	number	1rad+ 2axi	1part. adm.	1	2	2
D_{m1}	m	0.468	-	0.496	0.800	1.640
L_1	m	0.01	-	0.064	0.170	0.338
D_{m2}	m	0.227	-	0.510	0.880	1.640
L_2	m	0.027	-	0.070	0.250	0.513
Compressors and Pumps Total Compr. Power 18830 kW						
Compressor Name		C1	C2	C3	Cond. Pump	Feed Pump
m	kg/s	66.33	52.48	51.15	25.16	21.36
p_1	bar	0.25	1	2.7	0.25	5
p_2	bar	1	2.7	40	5	180
V_1	m³/s	150.83	39.13	10.6o	0.0252	0.0233
V_2	m³/s	49.46	17.43	1.169	0.0252	0.022
P	kW	5498	3957	8953	12	410
n	rpm	3000	12000	20000	3000	3000
z	number	7	5	7+1rad		
D_{o1}	m	1.47	0.528	0.274		
L_1	m	0.304	0.137	0.068		
D_i/D_o	-	0.586	0.481	0.504		
Mach Nr.	rel.at tip	1.00	1.31	1.39		
D_{o2}	m	1.47	0.462	0.280		
L_2	m	0.084	0.071	0.014		
Efficiency						
Net Power: 111081 – 18830 = 92251 kW						
Fuel gas mole fractions:						
0.1 CO_2, 0.4 CO, 0.5 H_2, LHV: 14.121 MJ/kg						
Fuel mass flow: 10.1510 kg/s						
Total Combustion Heat Input: 143342 kW						
Effort O_2 Generation: 4506.4 kW						
Effort O_2 Compression (atm./burner): 3640 kW						
Mech. and Generator Efficiency: 0.98						
Thermodynamic Efficiency Fuel and O_2 (Generation and Compression) to el. Bus Bar: 57.51 %						

But most important is the task of bringing reaction partners – fuel and oxygen – in close contact ahead and towards the end of the flame. Fuel and oxygen should be introduced in stoichiometric ratio since any deviation from this equivalence value would result either in reduced combustion efficiency or a surplus in power for oxygen generation.

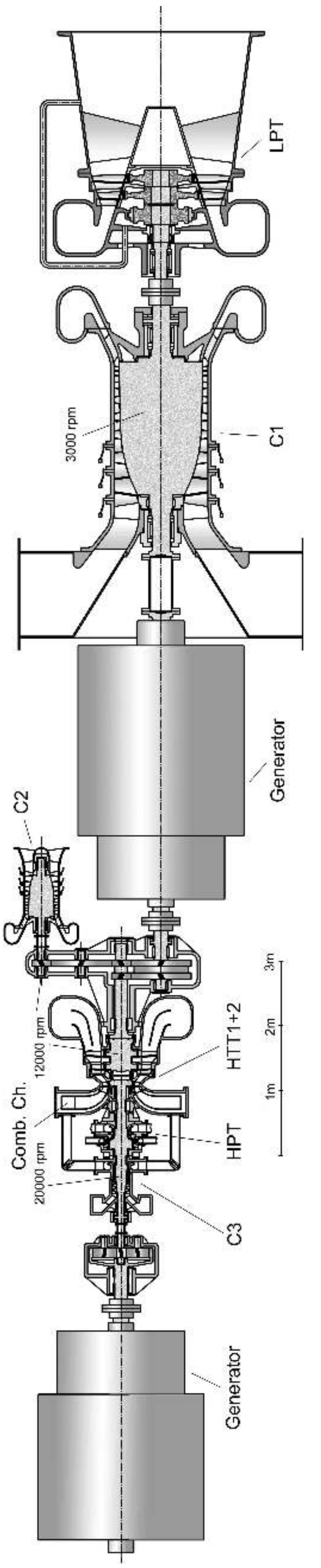

Fig. 2: General arrangement of turboset

The cooling medium being an inert gas must be mixed into the flame, otherwise the flame temperature would be too high with the danger of dissociation of the reaction products to be formed – i. e. hydrogen oxide H_2O the combustion water and CO_2 from carbon content of the fuel.

But this mixing and flame cooling process should not allow even small flow volumes or mass fractions of reaction partners – fuel gas and oxygen – to be lost and swept away out of ignition range by the inert cooling medium. In the case given CO_2 delivered from compressors and steam out of the high pressure turbine shall serve as cooling medium.

Fig. 4 shows the proposed design of burners and combustion chamber annular flame casing. At the burner inlet a head is providing slots ejecting sheet like jets of fuel gas and oxygen in close proximity. Steam is fed tangentially into the burner tube forming a strong steam vortex which is wrapped around the burner head. Thus providing high vorticity in the core for intimate mixing and ignition at the same time cooling the burner tube at the inside and holding the flame together at the exit into the free space after the end of it. The low pressure in the center induces a strong backflow from the hot flame gases so that continuous ignition and complete combustion in the following vortex break down can be achieved.

At 6 locations around the annular combustion chamber 4 burner tubes in parallel are mounted injecting into the inner part of the flame casing. The direction of rotation of the steam vortex is the same so that the 4 flames at the outlet create high counter rotation velocities which shall fix each individual vortex break down near the end of its burner tube. (The flow mechanism of vortex break down has been described by Keller et alia 1991). The high mixing involved should result in the desired high combustion efficiency. For each quadruple of burners a ignition flare and suitable crossovers are installed.

The rest of the cooling medium is introduced to the hot flame gases in conventional manner through slots and holes in the annular flow guiding insert. A strong rotation around the turbine axis is provided in order to offer additional flow path length for better mixture. The first stage nozzles have less flow turning angle to provide and can thus save blade cooling medium.

High Temperature Turbine HTT:

For the extreme high temperature region an axial flow path of minimum length is provided, consisting of one overhang stage of 20.000 rpm at the entry side followed by two overhang stages at 12.000 rpm towards the outlet, each supported on the bearing of the respecting rotors which at the entry side are those of the high pressure turbine HPT and the final CO_2 compressor C3 whereas the connection of the second rotor is made to the pinion of the main gear box. The first stage is a transonic stage cooled by steam which is introduced at the inner side of the high pressure labyrinth to the pressure side of the disk and from there to the hollow blades in an innovating cooling scheme ICS developed and patented for TU Graz (Jericha, et al., 1997). For the second rotor cooling steam is introduced through the diaphragm and the guide vanes of the second stage towards stage two disk and blading and to stage three disk and blade root. At the inner side cooling of stage 1 and stage 2 disks is provided in the form of a hydrostatic bearing at each disk rim which serves as a vibration damper for both rotors. This design provides for minimum heat loss, minimum cooling effort and minimum pressure loss in the flow from stage to stage. Both outer labyrinth seals are provided with CO_2 suction connection and an outer steam sealing supply.

All other turbomachines are sealed in the same manner, leaking CO_2 is recompressed to the main cycle and steam sealing is necessary in start up in order to raise vacuum.

High Pressure Turbine HPT:

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

The radial first stage shown here is proposed with the intention to keep the highest speed shaft short and with a minimum number of bearings (see Fig. 2, Fig. 3).

Low Pressure Turbine LPT:

Proposed is a conventional low pressure turbine with two overhang stages and with axial outflow. Thrust piston and labyrinths are sealed as described above. The rotor is built together in one set of bearings with the first compressor C1 (see Fig. 2).

CO2 Compressors C1, C2, C3:

For the compression of CO_2 from condenser to combustion chamber pressure level three compressor casings are required. This follows from the given flow volume and the thermodynamic properties of the gas to be compressed. CO_2 has a very high compressibility – a strong change of volume with pressure ratio and compared to air a relatively low sonic speed. The first stage of C1 can be built subsonic whereas compressors C2 and C3 require in their first stages a relative inlet tip Mach number in the range of 1,3 to 1,4 at highest possible stage diameter ratio. The last stages in all compressors can be kept at reasonable blade length ensuring high stage efficiency even there. Compressors C1 and C2 carry together with CO_2 still some steam – from incomplete separation in the condenser – which is condensed out in the intermediate cooler. Compressor C3 compresses only noncondensible gas, the decreasing volume flow of which requires the speed increase from C1 3000 rpm to C2 12000 rpm and further to C3 20000 rpm. See Fig. 2, Fig 3.

For comparison the result of these design deliberations is collocated in Table 1 which contains actual rotor dimensions, blade length, speed and output for all turbomachines.

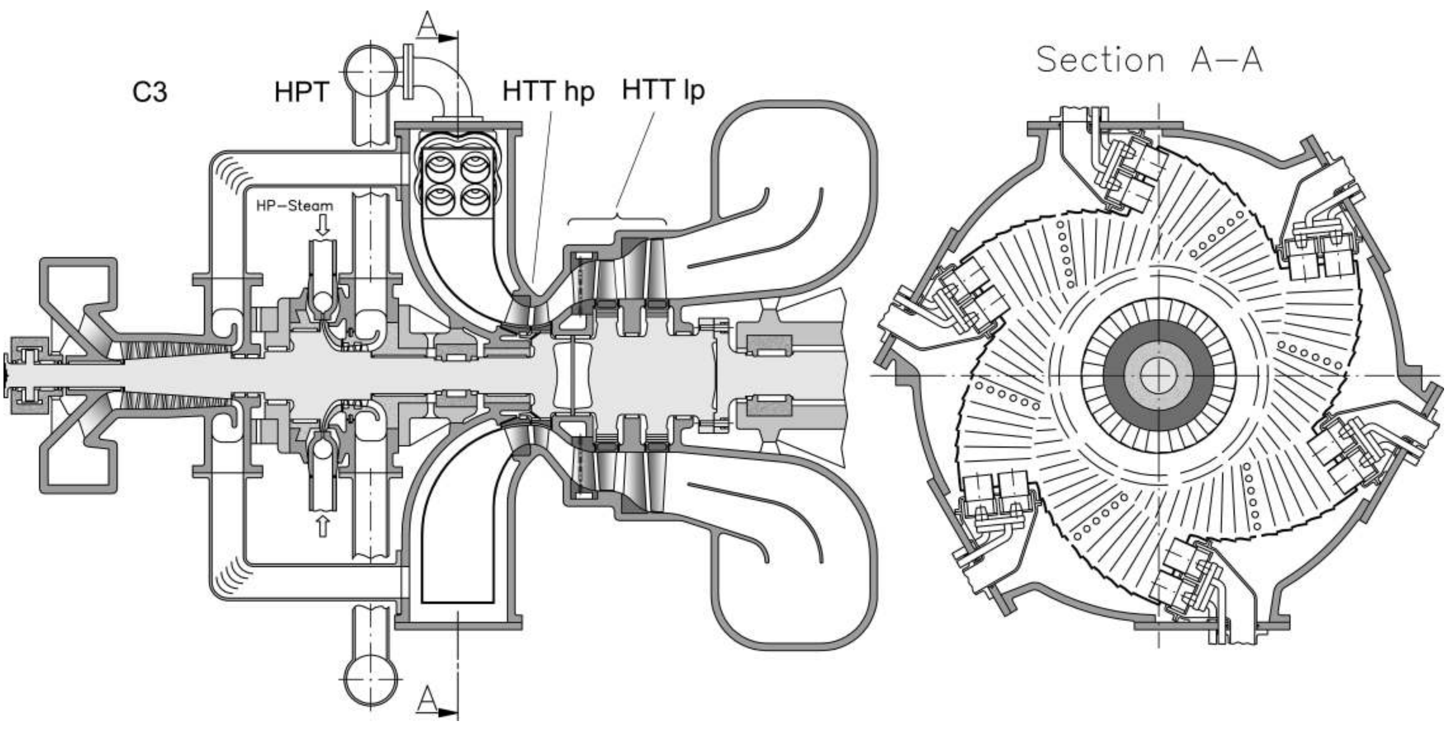

Fig. 3: Details of hp-shaft and combustion chamber

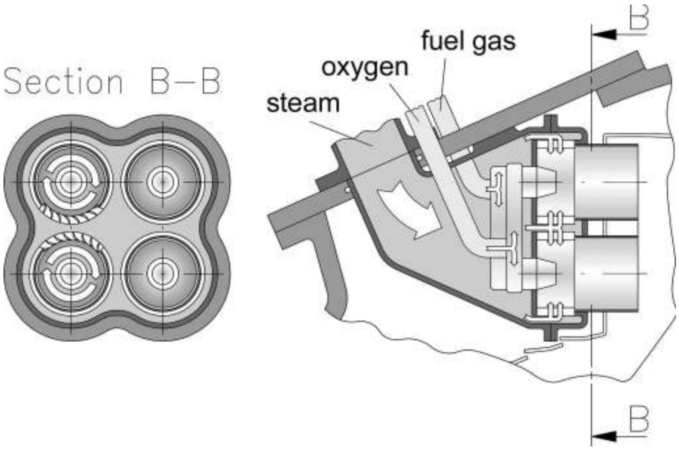

Fig. 4: Combustion chamber and burner details
(flares and crossovers not shown)

Heat Recovery Steam Generator HRSG and Feed System:

The HRSG is laid out for low cost with high temperature differentials at the top temperature region in order to save cost for high temperature tube alloy. A certain difficulty is the solubility of CO_2 in the feed water which will require better material in the low temperature region. A clear advantage is the possibility of intensive cleaning or change of feed water in case of combustor defects which might have caused soot formation. In general the low pumping power for the feed water supply improves efficiency since compression power is greatly reduced in comparison to conventional plants.

Start up and Part Load:

The combustion chamber and burner design as well as the blade cooling requires the availability of steam for start up. So it seems necessary to provide an auxiliary boiler or a steam connection to a neighboring conventional set in an industrial plant. Part load is achieved solely by lowering fuel and oxygen input. Temperature reduction results in a lower flow volume to the HTT and leads to a reduction of combustion chamber pressure. Condenser pressure will fall equivalently so that flow similarity for compressors and turbines is maintained. All turbomachines remain on constant speed. Reduction of medium density and pressure results in the desired reduced output. Thus cycle efficiency is maintained at high levels also at part load. A more detailed investigation of this thermodynamic situation will follow.

COST OF MANUFACTURE AND INSTALLATION

A prediction of manufacture and installation cost can only be done in cooperation with gas turbine manufacturers when all designs have been worked out to drawing design level. The lead author has carried out such procedures already. (Perz, et al., 1988). A rough estimate can be given by comparing the number of stages in turbomachines. It is well known that the cost of a single blade is almost independent of its size. Cost of gears seems to be power related.

So when a standard combined cycle plant can be provided with 16 stages for the air compressor, 4 stages for the gas turbine and 20 stages for the steam turbine the total number of stages is 40 plus a main power gear box for the gas turbine.

In the design presented here we can sum up the number of stages as follows. HTT 3, LPT 2, HPT 3, C1 7, C2 5, C3 8 with a total of 28 stages, a main gear box plus 2 smaller gears and a splitted electro generator.

Further detailed design work in cooperation with gas turbine industry will give more accurate figures.

RECENT RESEARCH WORK OF TTM

At the Institute of Thermal Turbomachinery and Machine Dynamics a transonic stage similar to the stage 1 in HTT was designed and tested in the institutes transonic test turbine facility (see Fig. 5).

Also an innovative cooling system with transonic cooling films was developed which is easy to apply to this prototype GRAZ CYCLE power plant (see Fig. 6) since the cooling is to be effected by high pressure steam available from the HPT.

Fig. 5: typical transonic stage

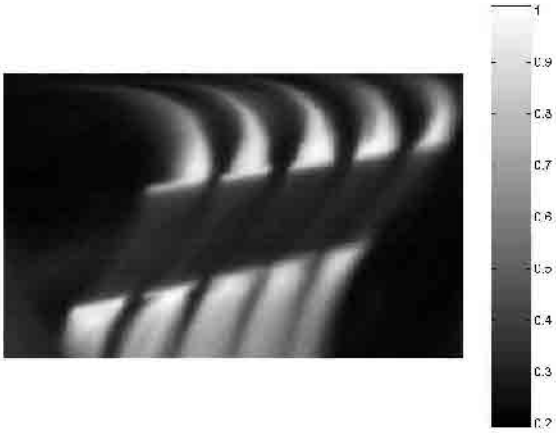

Fig. 6: Innovative cooling system (ICS) with transonic cooling films (film cooling effectiveness)

CONCLUSION

Detailed design work concerning a closed cycle system capable of retaining CO_2 for further technical application and avoiding any other emissions to atmosphere has been presented. High efficiency on a broad fuel basis will make the development work and the innovation to power industry a hopeful asset for environmentally friendly power production.

ACKNOWLEDGMENTS

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