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DESIGN OPTIMISATION OF THE GRAZ CYCLE PROTOTYPE PLANT

H. Jericha, E. Göttlich, W. Sanz, F. Heitmeir

Institute for Thermal Turbomachinery and Machine Dynamics
Graz University of Technology, Austria
sanz@ttm.tu-graz.ac.at

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. The use of well established gas turbine technology enhanced by recent research results enables designers even today to present proposals for prototype plants.

Research and development work of TTM Institute of Graz University of Technology since the 90's has lead to the Graz Cycle, a zero emission power cycle of highest efficiency and with most positive features. In this work the design for a prototype plant based on current technology as well as cutting-edge turbomachinery is presented. The object of such a plant shall be the demonstration of operational capabilities and shall lead to the planning and design of much larger units of highest reliability and thermal efficiency.

INTRODUCTION

Optimisation of a thermal power plant starts with the optimisation of the cycle scheme i.e. the thermodynamic relations of cycle media in the process of power production. Like in any heat engine it involves according to Carnot´s rule introduction of fuel heat input at maximum possible temperature, compression and expansion at maximum compressor and turbine efficiency and release of non-convertible heat to ambient at minimum loss. The relations of media within the cycle have to be optimised regarding heat transfer, pressure loss, material cooling, in the many connections that have been invented and are in use today.

Turbomachinery design has to be optimised first of all in terms of flow efficiency, high temperature blade cooling methods, rotor speed and turbine-compressor driving connections in any case on the basis of sound rotor dynamics.

Surface heat exchangers such as steam generators, feed water heaters and condensers have to be carefully studied to minimize costs in general and to minimize requirements of high temperature metal for heat transfer surfaces and associated pressure losses.

Closed cycle gas turbines of zero emission with the capability of capturing or retaining combustion generated CO_2 require novel cycle solutions. A general comparison between different solutions of CO_2 retaining plants is given very detailed in [16]. Among them was the so-called Graz Cycle system, which has been presented by the authors in several papers at previous conferences (CIMAC, ASME, VDI, [1 - 9]). Any fossil fuel gas (preferable with low nitrogen content) is proposed to be combusted with oxygen so that mainly only the two combustion products CO_2 and H_2O are generated. Oxygen can be generated from air by air separation plants which are in use worldwide with great outputs in steel making industry and even in enhanced oil recovery.

Although the Graz Cycle is suited for all kinds of fossil fuels, for natural gas fuel it seems reasonable to reform CH_4 to $CO + H_2$. Hydrogen can be separated and burned in an air breathing gas turbine, a solution which reduces the oxygen requirements considerably [10]. But in using oxygen blown coal gas as a fuel a Graz Cycle plant is most effective in retaining CO_2 and in use of oxygen.

The thermodynamic details of a prototype Graz Cycle plant of 92 MW power fired with oxygen blown coal fuel gas were presented to VDI in 2000 [9]. This cycle scheme shall be used here as the basis of turbomachinery optimisation discussion. A general layout of all components, especially turbomachines, combustion chamber and burners, and general arrangement with gears and electric generators was presented at ASME IGTI conference 2002 [11].

The object of this paper is to present this kind of zero emission cycle optimised for highest thermal cycle efficiency. The deliberations which have led to the cycle scheme as well as to the special design of blading and rotors are shown. This work specially concentrates on the high temperature turbine and its first transonic stage with the associated innovative steam cooling system.

NOMENCLATURE

Latin

n	[rpm]	speed
P	[MW]	power
V	[m³/s]	volume flow
H	[kJ/kg]	enthalpy
R	[-]	degree of reaction
r	[m]	radius
D	[m]	diameter
L	[m]	blade length
z	[-]	number of stages
u	[m]	circumferential velocity
x, y, z	[m]	Cartesian coordinates
p	[bar]	pressure
c	[m/s]	velocity
C	[m/s]	absolute velocity
W	[m/s]	relative velocity

Greek

y	[-]	$= \frac{2 \cdot \Delta H}{u^2}$ ….load factor
a	[°]	nozzle exit angle
e	[°]	inclination of stream line vs. axis

Subscripts

a	tip
m	mean
i	hub
r	radial
u	circumferential
x, y, z	components in x, y, z direction
K	curvature
1,2	nozzle exit, rotor exit

Abbreviations

NEDO	New Energy Development Organization (Japan)
CRIEPI	Central Research Institute of Electric Power Industry (Japan)
TTM	Institute for Thermal Turbomachinery and Machine Dynamics – Graz Univ. of Technology
HPT	High Pressure Turbine
HTT	High Temperature Turbine
LPT	Low Pressure Turbine
C1, C2, C3	Compressor 1, 2, 3
ICS	Innovative Cooling System
HRSG	Heat Recovery Steam Generator

CYCLE OPTIMISATION

Figure 1 shows the principle flow scheme of the Graz Cycle with the main components and will be used to explain the main characteristics of this zero emission power cycle. Detailed cycle data for a 92 MW pilot plant, like mass flow, pressure, temperature, enthalpy or cycle fluid composition as well as the details of the thermodynamic simulation performed with the commercial code IPSEpro by SIMTECH Comp. can be found in the appendix (Fig. 13. from [11]).

Basically the Graz cycle consists of a high temperature Brayton cycle (compressors C2, C3, combustion chamber and HTT) and a low temperature Rankine cycle (LPT, condenser, HRSG and HPT). In the layout presented a proposed fuel with a typical composition from an oxygen blown coal gasification plant is taken (fuel gas mole fractions: 0.1 CO_2, 0.4 CO, 0.5 H_2). The fuel together with the stoichiometric mass flow of oxygen is fed to the combustion chamber, which is operated at a pressure of 40 bar. Steam as well as CO_2 is supplied to cool the burners and the liner. A mixture of about three quarters of CO_2 and one quarter of steam leaves the combustion chamber at a mean temperature of 1400 °C. The fluid is expanded to a pressure of 1 bar and 642 °C in the HTT. The hot exhaust gas is used in the following HRSG to vaporize and superheat steam for the HPT. Then it is further expanded in the LPT to a condenser pressure of 0.25 bar. In the condenser the separation of CO_2 and H_2O takes place by water condensation. The water is preheated and in the HRSG vaporized and superheated. The steam is then delivered to the HPT with 180 bar and 567 °C, after the expansion it is used to cool the burners. The CO_2 from the condenser is compressed to atmospheric pressure, the combustion CO_2 is then separated for further use or storage. The remaining CO_2 is compressed and fed to the combustion chamber to cool the liners.

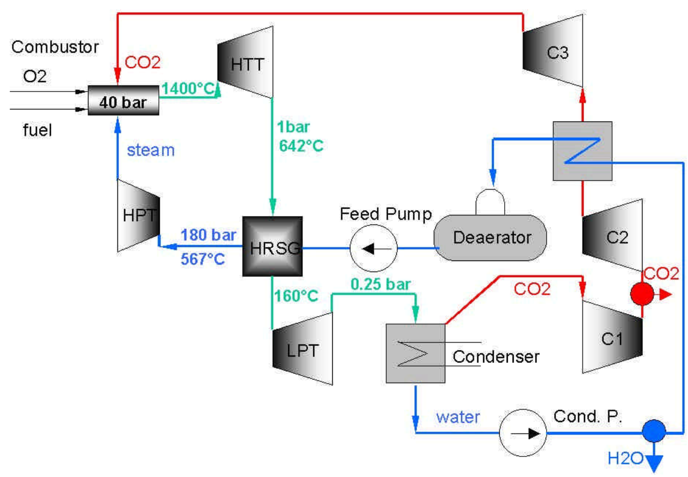

Fig. 1: Principle flow scheme of Graz Cycle power plant

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 till to vacuum conditions, so that a high thermal efficiency according to Carnot can be achieved. The dual medium CO_2 and H_2O results also in very low compression work (see appendix). With both medium components in use over the full temperature range, the cycle gives this beneficial effect since only the gas CO_2 requires turbo compressors (C1, C2, C3) whereas feed water can be pumped to form high pressure steam. High pressure steam can be expanded to generate additional power in the high pressure turbine HPT, before being united with the CO_2 flow in the combustion chamber. Further beneficial effects are the possibility to create burner vortices and to cool the hottest nozzles and blades of the HTT first stage, because the exhaust steam of the HPT is of suitable pressure and temperature and can be passed through the hollow blades in order to cool the blading. The effect of blade cooling is simulated by extracting steam after the HPT and its admixing to the cycle medium before and after the first stage of the HTT (see appendix).

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

TURBOMACHINERY OPTIMISATION

Figure 2 shows a schematic arrangement of turbomachinery shafts as presented in [11]. Mass flow, power, speed, dimensions and number of stages of the turbomachinery are given in Table 1 in the appendix. The reader is asked to take note that a priori design optimisation experience is introduced by this basic turbomachinery arrangement already. Design rules such as putting on a common shaft of turbines and driven compressors of similar optimal speed are observed as well as the rule to expand the hottest gas flow in a single annular channel without any buried bearings or casing crossovers.

From the logic of these connections and from the capabilities of the diverse components enabling them to fulfil their tasks most effectively an optimal situation is clearly visible, insofar as each specific task is made to function optional in itself. But anyhow the designer should keep in mind that care must be taken in introducing inventive features and not to contradict the rule of maximum simplicity.

In order to present a design optimisation solution the design logic of three systems is discussed: the fuel oxygen burner and combustion chamber flow, the blading layout of the high temperature turbine HTT and its associated innovative steam cooling system ICS and finally the general arrangement of compressors, turbines, gear boxes and electric generators of the prototype Graz Cycle power plant.

Combustion chamber: First the oxygen burner with stoichiometric combustion is treated. The system is being studied by several Japanese research institutions (CRIEPI, NEDO; HITACHI, MHI), recent research was done by [12]. The burners developed there are compared to the authors' proposal (see Figure 3, H_2+O_2 burner design [1], CH_4+O_2 burner[12], oxygen blown fuel gas+O_2 [11]). As detailed in [11] the choice of burner design depends on the flame speed of fuel gas in oxygen. A fuel gas containing free hydrogen has a much higher flame speed, so no kind of premix appears possible.

Careful research is done to obtain optimal combustion efficiency and to avoid loss of reaction partners during the introduction of inert cooling flow into the burner zone, a measure necessary to avoid dissociation of molecules of reaction products in the hottest region of the flame. From previous work the authors' knowledge is that the effect of dissociation becomes important above 2000 °C (see Table 1). Thus a complete combustion of the reaction partners is aimed at since dissociated molecules could be prone to be swept out of reaction zone by the surrounding inert cooling and cycle medium.

Figure 4 shows the cross section of combustion chamber and annular flame cage. The radial inflow with a strong swirl lengthens the reaction zone and gives a flatter turning angle for the HTT first stage nozzles.

Compressor layout: Before discussing the high temperature turbine the compressors have to be deliberated since they more strictly define the speed requirements. After separation of the cycle media by condensation of water content compression of CO_2 is effected in axial turbo compressors C1 3000 rpm, C2 12000 rpm and C3 20000 rpm. The compressor C3 delivers pure CO_2 to the combustion chamber whereas compressor C1 and C2 contain a small amount of water which is finally condensed out during compression. The necessity of different speeds is given by the compressibility of CO_2 because the volume flow changes from 150 m³/s at first compressor inlet to 1.17 m³/s at last compressor outlet (see [11]). The requirements for long first stage blades of C2 and C3 and a maximum admissible blade tip Mach number of 1.35 [13 - 14] lead to three different compressor speeds. This design allows to achieve at the same time a reasonable last stage blade length. Last stage of C3 is built as radial stage which allows to arrange a radial diffuser and scroll for efficient transfer to the combustion chamber.

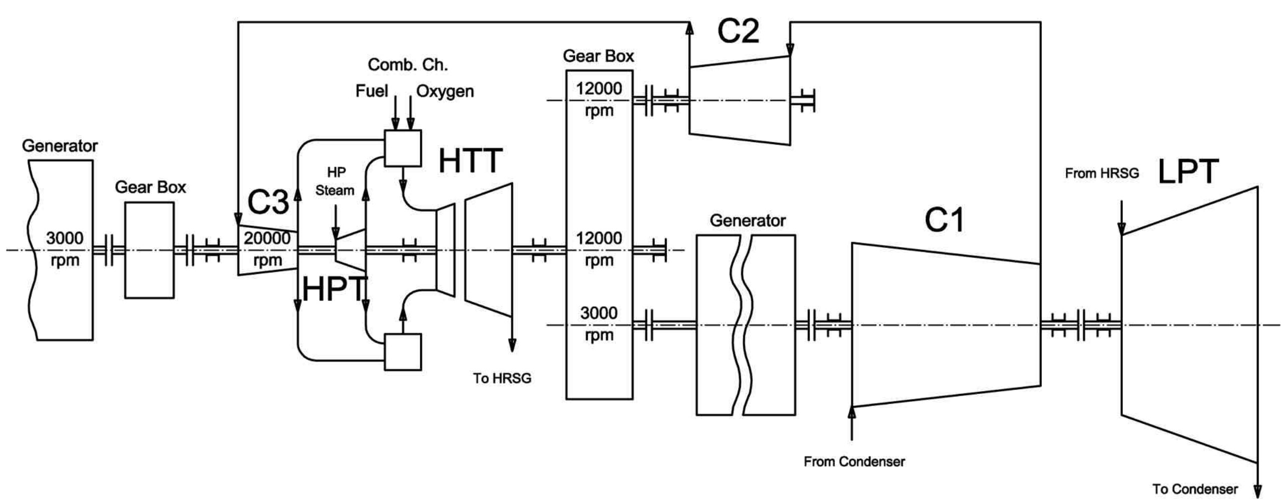

Fig. 2: Schematic arrangement of turbomachine shafts

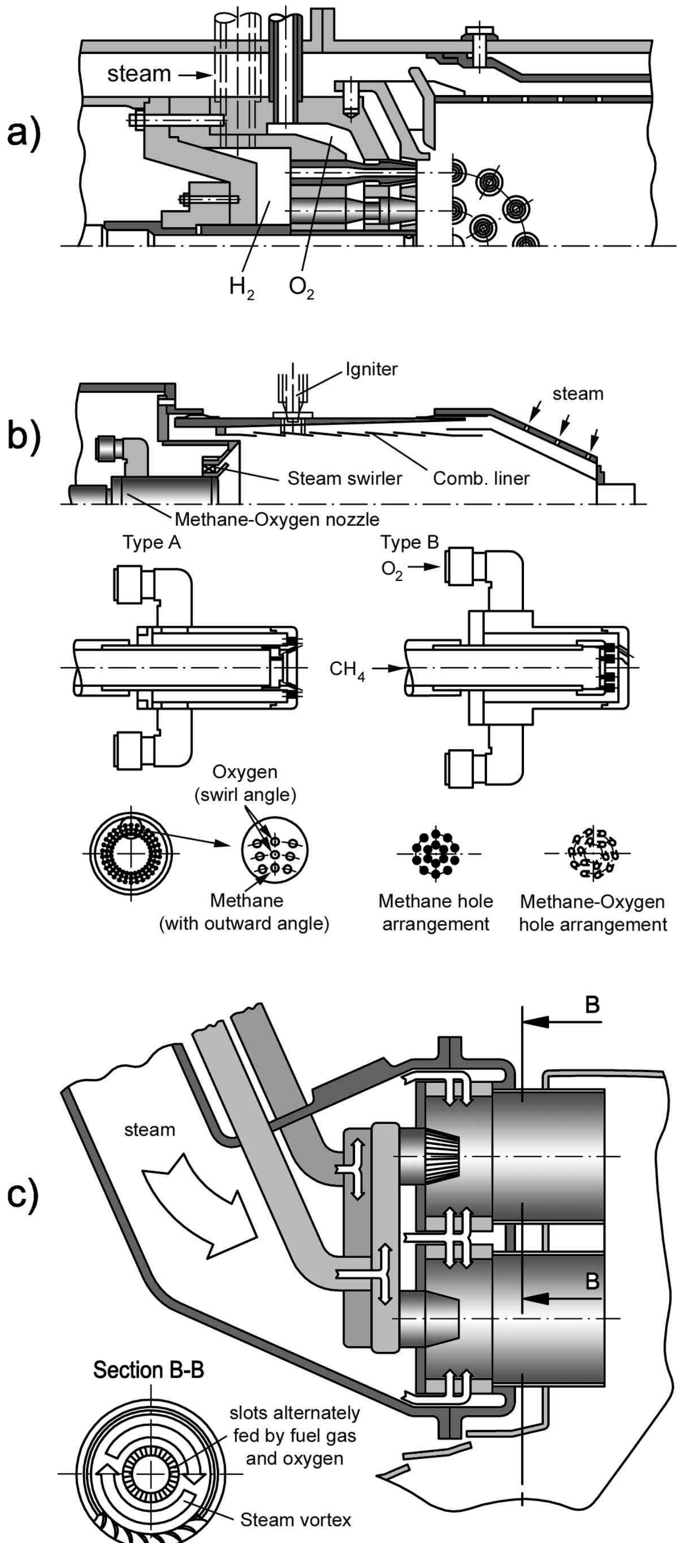

Fig. 3: Comparison of O_2-Burner design solutions a): H_2+O_2, b):CH_4+O_2, c): oxygen blown fuel gas+O_2

High Temperature Turbine: The most advanced turbo machinery proposal is that of the high temperature gas turbine. Compared to an air-breathing gas turbine the cycle fluid has a gas constant R which is 11 % smaller and a heat capacity c_p which is 23 % larger. This results in nearly the same enthalpy drop for a given pressure ratio, but in higher temperatures, so that cooling is more important. The design features of the HTT make full use of the possibilities offered by the cycle. Even after part of the steam is mixed with the CO_2 flow in the combustion chamber already there is sufficient steam of low temperature and suitable pressure available to effect the cooling of the high temperature blading.

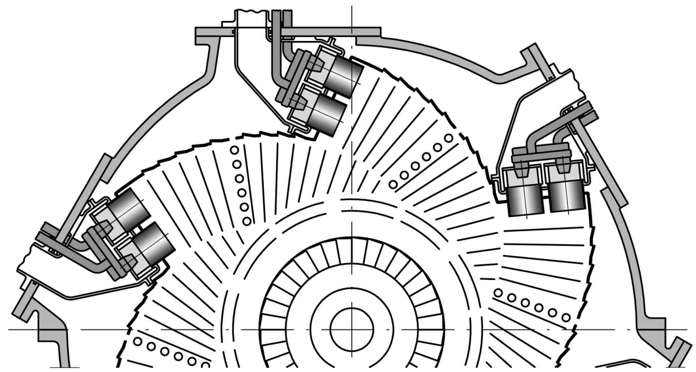

Fig. 4: Comb. camber, O_2-burner and annular flame cage cross section at entry to HTT

The high pressure ratio of 40:1 in the HTT together with the specific strong volume change of the media results in a very high ratio of outlet volume flow to inlet volume flow. So it is optimal to split the HTT in a first stage of 20.000 rpm directly connected to HPT and C3 thus giving also optimal speed for the last CO_2 axial compressor with a final radial stage (see Fig. 5). Two more stages also overhang on a common shaft with the main gear pinion with a side drive to C2 (this for physical arrangement reason only). This arrangement also allows a direct introduction of cooling steam from the HPT into the combustion chamber burners and into the HTT blade and disk cooling as described below (see Fig. 2 or 5).

As an example of design optimisation the speed variation of the rotor of a standard gas turbine unit is shown in Fig. 6a. Given is the enthalpy head and volume flow in and out of the 3 stage blading. The rotational speed is varied in the range of plus and minus 20 % keeping the load factor ψ and degree of reaction constant at mean diameter. At higher speed a smaller mean diameter and thus higher blade lengths are obtained and accordingly a lower degree of reaction at the inner radius and a higher degree of reaction at the outer radius. On the other hand lower speed at unchanged volume flow and enthalpy head leads to higher degree of reaction at the inner radius and lower degree of reaction at the outer radius.

Comparing the three cases shown in Fig. 6a at lower speed an improvement in blade isentropic flow efficiency (i.e. stage flow efficiency covering losses by friction on blade surfaces and side walls, tip leakage loss etc., due to higher degree of reaction R at the root) can be expected but an increase in leakage loss (larger clearance and outer radius, shorter last blade). On the other hand an increase in speed lengthens the blade, decreases the rotor outer radius but at the inner radius leads to an unacceptable negative degree of reaction, a indication of possible hub and blade flow separation.

We observe a situation quite general in flow theory, that going close to separation gives some improvement in efficiency, but bears risks, so that a safe distance should be kept. So it can be concluded that for this turbine 5252 rpm is close to the optimal speed.

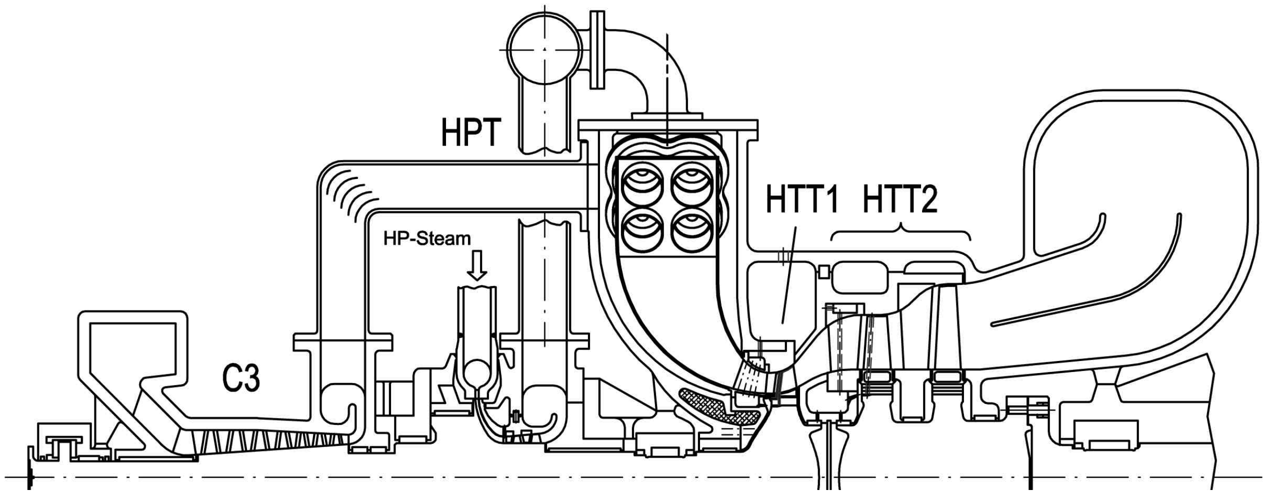

Fig. 5: High speed shaft, HTT and HPT driving C3 and power gear to el. generator Nr. 2

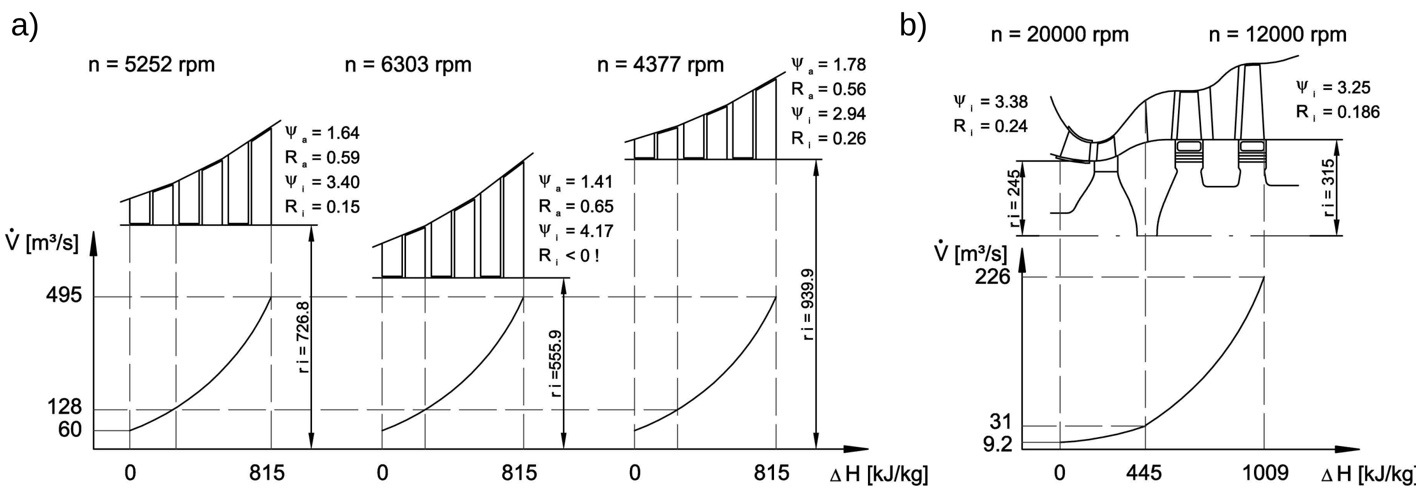

Fig. 6: a) Standard GT rotor speed optimisation at given volume flow and enthalpy head; b) HTT shaft built in two overhung disks (single and double) with optimal speed selected for compressor and gear drive

Figure 6b gives the comparable data of the Graz Cycle HTT. The enthalpy head is still higher and the volume flow change is still more pronounced, so there is good reason to split into two shafts with the HTT first stage running much faster than the second and third stages.

The speeds are optimally suited to drive compressors and gear boxes for power transfer. The difference in diameter of stage one to stage two requires an annular flow connection and an inward curved channel from combustion chamber inlet via first stage outward to the larger radius of the second stage. This inward curved channel has the beneficial effect of reducing the otherwise high value of change of reaction of the first stage blade (see HTT first stage radial equilibrium of flow in the appendix). A velocity triangle (see Fig. 7) obtained in this way shows the beneficial property of low twist and an almost straight trailing edge. The inlet edge follows a cone oblique to the blade radial axis and allows to keep the inlet edge radius almost constant. This facilitates manufacture of the high temperature material blade.

HTT Cooling: The cooling is performed with an innovative cooling system ICS. The innovative feature is the use of the property of an underexpanded jet which means a jet from a choked convergent nozzle where the flow subsequently expands supersonically because the external pressure is below the critical value. This jet has the strong tendency to bend towards a convex surface like the leading edge of a turbine blade. So air or steam of high total pressure is ejected from slits of small height, partly against the main flow direction (see Figs. 7 and 8).

Experimental investigations showed that the ICS film is able to cover the thermally heavily loaded leading edge safely and is very resistant against the trailing edge shocks of the previous vanes [4 - 8]. Applying the ICS no internal serpentine passages are required, only two almost radial holes near the center providing cooling steam for the slits are needed (ICS is patented to the authors [5]). Slit inlets and supply holes can be manufactured on a blade cast in one piece by modern electro-erosive machining.

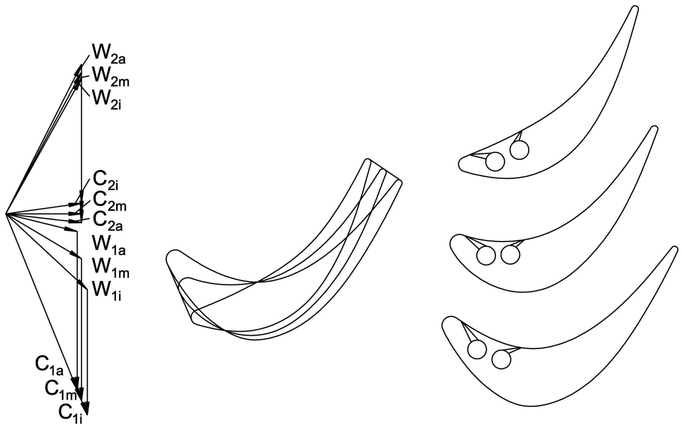

Fig. 7: HTT1 velocity triangles and blade profiles over blade height, for tip, mean and root section in curved annular blade channel

The high speed and centrifugal load involves high stress of the disk which has to be built as constant stress disk with a bell shaped radial thickness variation. This disk surface has to be and can be cooled on both sides all over to a temperature of 350 °C where full cold stress properties can be maintained.

The cooling flow path in the first stage is shown in more detail in Figs. 9 - 11. Cooling flow introduction to HTT first stage nozzles and blades is done by introduction of 40 bar steam exhausted from the HPT into the outer annular cavity of the HTT first stage nozzle ring. From here steam flows radially inward through circular holes, the large ones with swirl inducers into the inner cavity ring where the cooling steam is collected at conditions of 36 bar and 380 °C. The last radial hole supplies ten axial holes of smallest diameter ejecting steam into the main flow thus effectively cooling the trailing edge (see Fig. 8).

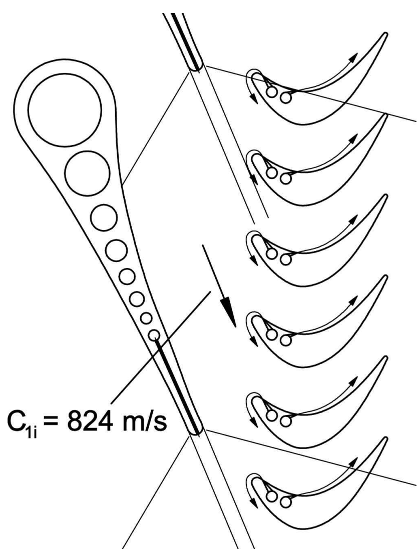

Fig. 8: HTT1 transonic expansion in nozzle profile near blade root and tail shocks of nozzle trailing edges reaching into blade profile inlet creating a high frequency shock passing flow situation there

From the inner nozzle ring cavity steam nozzles are fed accelerating the cooling flow into the wheel space in circumferential direction below the radius of blade root. Under an extension of the root plate the cooling steam enters first axially into the blade fir tree root part, then flows radially outward to supply the first row of slits of the innovative cooling system ICS (see Fig. 9). The second more central hole is supplied from a slot in the disk groove under the fir tree. This second radial hole feeds in the same manner the ICS slits pointing downwards on the blade pressure side.

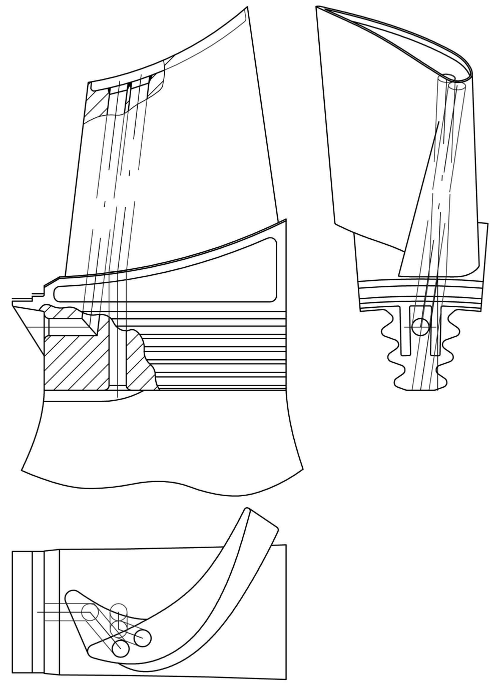

Fig. 9: HTT blade design drawing, steam cooling ICS-slits and supply hole arrangement

In Fig. 10 the pressure distribution of steam flow inside HTT first stage nozzles, during expansion into the wheel space and supply to radial holes in the blades is shown proving that for all radii the necessary transonic pressure ratio is provided.

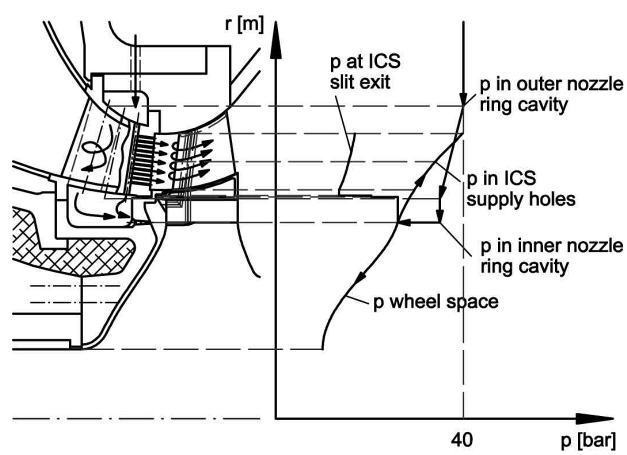

Fig. 10: HTT 1 steam cooling flow; pressure distribution radial, through nozzle cooling holes inward, through tangential acceleration nozzles into wheel space and radially outwards in ICS supply holes is shown, transonic pr. ratio for ICS slits is secured

Cooling of HTT first stage disk low pressure side is done by introduction of further cooling steam into the annular space between first and second stage (see Fig. 11). Passing through the radial holes in the second stage nozzles, an ICS for the rotor as well as for two hydrostatic bearings on both first and second stage inner disks is supplied. These are intended to operate as seals for the steam flow and to act as mild dampers improving the rotor dynamics of both shafts involved. The second stage disk is provided with internal flow channels to lead the cooling flow to the fir tree roots of the blades cooled in a similar way with ICS slits.

At part load and even during start up (auxiliary steam supply) the ICS steam supply is sufficient high to form transonic underexpanded cooling layers ensuring proper cooling function.

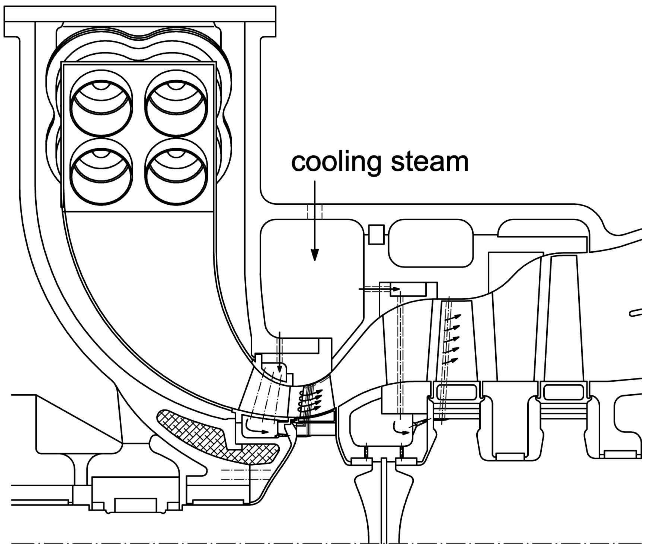

Fig. 11: HTT cooling flow in both rotors and intermediate shaft vibration damper

GENERAL ARRANGEMENT

As the last issue to be treated the most far reaching optimisation deliberation is presented. According to the design decision of having the high temperature flow channel with minimum surface and minimum heat loss and also with minimum cooling flow supply the general arrangement of turbomachines is given (see Fig. 12). Two overhang disks of different speed provide the shortest possible high temperature annular flow channel. So power end drive has to be on opposite sides. At 20000 rpm HPT and C3 can be optimally connected. At the 12000 rpm side main power is delivered via gears to the main generator. On the other side of the generator C1 and low pressure turbine LPT are arranged.

The power produced by HTT first stage and HPT greatly surpasses the power demand of C3, so a second electric generator is necessary. The only alternative would have been an outside gear shaft connecting to the main gear box and the main generator. The electric shaft proposed here seems to be a superior solution since for much larger units the same type of arrangement can be kept. Since gears with a power of about 100 MW are in successful operation in standard gas turbine units, the output of this Graz Cycle prototype unit could be doubled in a range to 200 MW.

Deliberations about the magnitude of cost of manufacture of such a plant in comparison to a standard combined cycle plant have already been made in [11] on the bases of the number of stages assuming that there is a very little variation of cost with size of blading. There it was shown that the total number of stages for a Graz Cycle power plant is 28, compared to about 40 for a standard combined cycle power plant.

CONCLUSIONS

The Graz Cycle, a novel type of closed cycle gas turbine plant with the capability of retaining all the combustion generated CO_2 for further technical use, has been presented. It is based on the internal stoichiometric combustion of fossil fuels with oxygen and offers very high efficiency. The cycle uses only turbomachinery components. A general layout of all components has been performed for a pilot power plant to verify the feasibility of the components.

The deliberations of the design of the most difficult components, the combustion chamber and the HTT, as well as the general arrangement of the turbomachinery are discussed in detail. Especially for the HTT an innovative design is presented where the first stage has a higher speed than the succeeding two stages. An innovative cooling system for the rotor blades using steam is applied.

The next step is the investigation of the economics of such a zero emission power plant, which depends largely on the amount of a future CO_2 tax. But if it proves economically reasonable, a Graz Cycle power plant could be put into operation within a few years based on the vast experience of successful gas turbine operation and research now well under way.

ACKNOWLEDGMENTS

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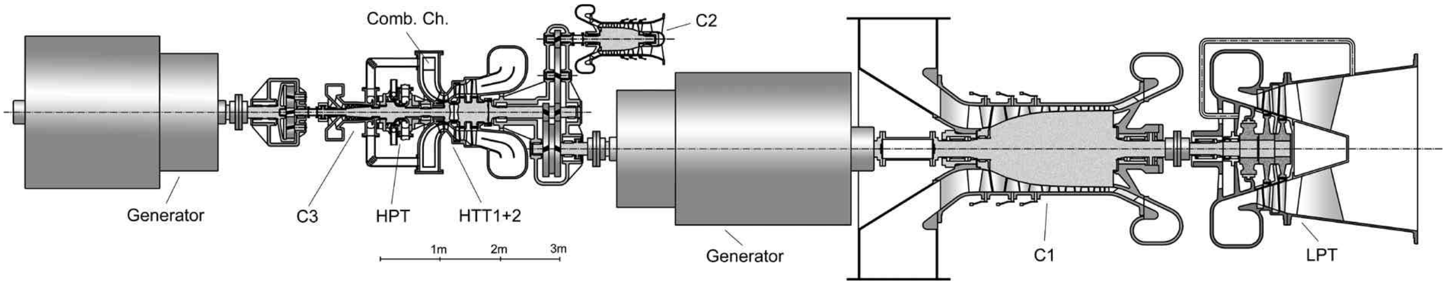


Fig. 12: General optimised arrangement of turbomachines for a 92 MW prototype unit

APPENDIX

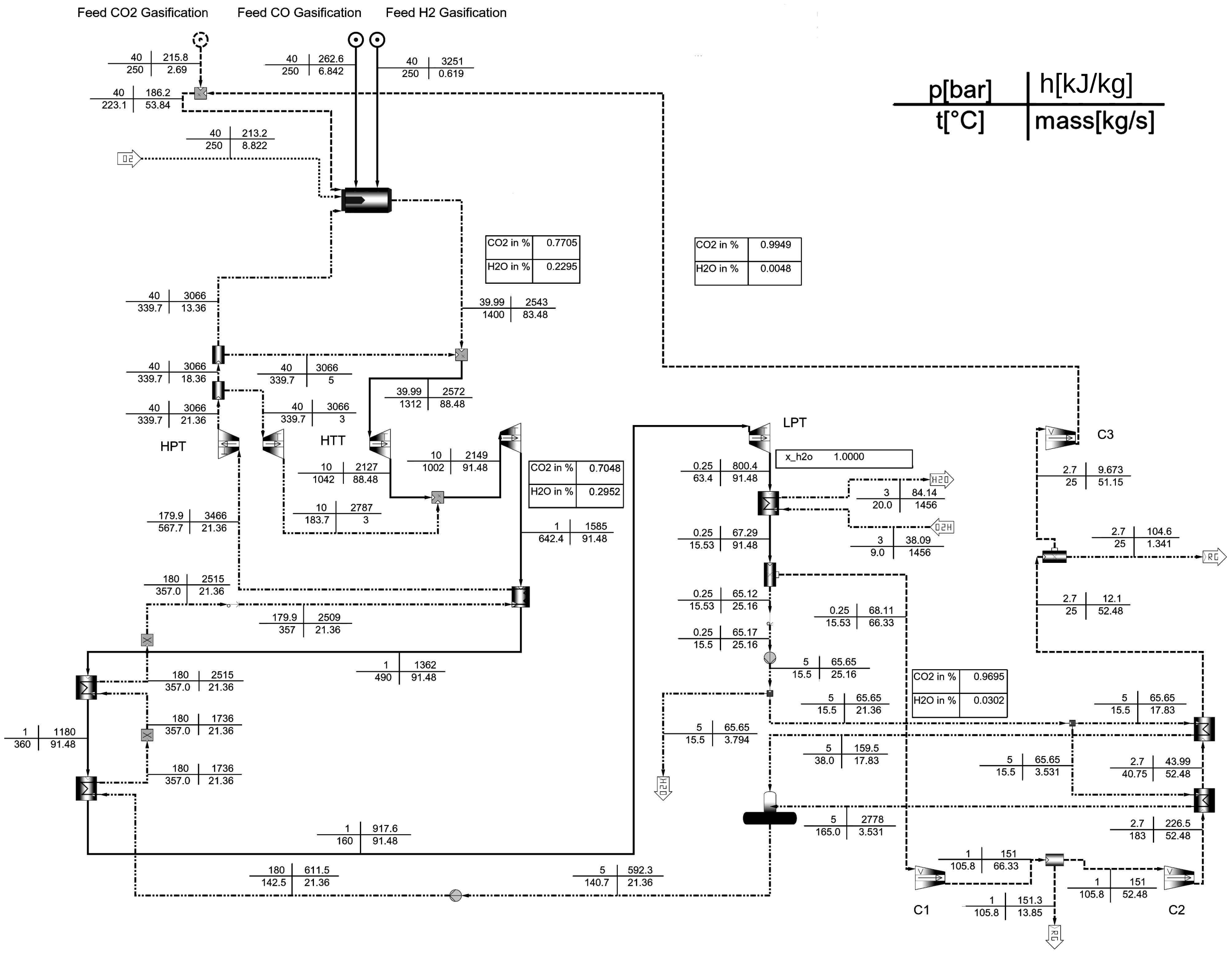


Fig. 13: Detailed thermodynamic cycle data of 92 MW Graz Cycle plant [11]

Table 1: Main turbomachinery data and Graz cycle power balance

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
V_{inlet}	m³/s	0.4125	0.195	9.264	31.50	106.67
V_{exit}	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	-	1rad+ 2axi	1part. adm.	1	2	2
$D_{m,inlet}$	m	0.468	-	0.496	0.800	1.640
L_{inlet}	m	0.01	-	0.064	0.170	0.338
$D_{m,exit}$	m	0.227	-	0.510	0.880	1.640
L_{exit}	m	0.027	-	0.070	0.250	0.513

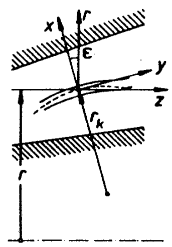
Compressors and Pumps Total Compr. Power 18830 kW						
Compressor Name		C1	C2	C3	Cond. Pump	Feed Pump
m	kg/s	66.33	52.48	51.15	25.16	21.36
V_{inlet}	m³/s	150.83	39.13	10.6o	0.0252	0.0233
V_{exit}	m³/s	49.46	17.43	1.169	0.0252	0.022
P	kW	5498	3957	8953	12	410
n	rpm	3000	12000	20000	3000	3000
z	-	7	5	7+1rad		
$D_{a,inlet}$	m	1.47	0.528	0.274		
L_{inlet}	m	0.304	0.137	0.068		
D_i/D_a	-	0.586	0.481	0.504		
M	rel.at tip	1.00	1.31	1.39		
$D_{a,exit}$	m	1.47	0.462	0.280		
L_{exit}	m	0.084	0.071	0.014		

Power balance
Net power: 92251 kW
Total combustion heat input: 143342 kW
Mech. and generator efficiency: 0.98
Thermal cycle efficiency: 63.0 %
O_2 generation by air separation (0.25 kWh/kg): 7940 kW
O_2 compression (atmosphere to burner): 3440 kW
Net cycle efficiency: 55.0 %

HTT first stage radial equilibrium of flow:

According to [15] the radial equilibrium of flow in the annular space between nozzle exit and blade entry can be calculated with the equation and associated sketch, estimation of 3d- flow by streamline curvature method:

$$\underbrace{-\frac{c_u^2}{r}\cdot\cos e}_{a}-\underbrace{\frac{c_y^2}{r_K}}_{b}=\underbrace{-\frac{1}{r}\cdot\frac{\partial p}{\partial x}}_{c} \qquad (1)$$



a … x component of the centripetal acceleration evoked by the movement in circumferential direction

b … x component of the centripetal acceleration evoked by the movement in meridional direction

c … x component of the force evoked by the pressure distribution

With angle of deviation from axis zero and constant head at all radii the equation modifies to:

$$-c_u\cdot\frac{\partial c_u}{\partial r}-c_z\cdot\frac{\partial c_z}{\partial r}=\frac{c_u^2}{r}+\frac{c_z^2}{r_K}=\frac{c_u^2}{r}+\frac{c_u^2\cdot\tan^2 a}{r_K} \qquad (2)$$

$$c_z=c_u\cdot\tan a \qquad (3)$$

With the assumption of constant nozzle angle alpha follows:

$$-\frac{\partial c_u}{c_u}\cdot\left(1+\tan^2 a\right)=\frac{\partial r}{r}\cdot\left(1-\frac{r\cdot\tan^2 a}{r_K}\right) \qquad (4)$$

$$\int\frac{dc_u}{c_u}=-\cos^2 a\cdot\int\left(1-\frac{r\cdot\tan^2 a}{r_k}\right)dr \qquad (5)$$

numerical integration:

$\ln c_u=-\cos^2 a\cdot\int f(r)dr$ with $f(r)=f_0+f_1\cdot\Delta r+f_2\cdot\Delta r^2$,

$\Delta r=r-r_m$ and $dr=d(\Delta r)$ follows:

$$\ln c_u=-\cos^2 a\cdot\left(f_0\cdot\Delta r+f_1\cdot\frac{\Delta r^2}{2}+f_2\cdot\frac{\Delta r^3}{3}\right) \qquad (6)$$

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