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Optimized cycle and turbomachinery configuration for an intercooled, recompressed sCO2 cycle

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ABSTRACT

Among the main advantages that sCO2 thermodynamic cycles can provide in power generation applications, the dramatic machinery size reduction, along with the possibility to achieve very good efficiencies with a wide range of thermal power sources, are probably the most outstanding.
The ways to enhance the cycle efficiency are essentially: leveraging on the real gas effect in the fluid compression, increase thermodynamic cycle complexity in order to reduce irreversibilities in thermal power transfers, and increase the maximum cycle temperature. Experience in power plants technologies evolution suggests that the plant complexity level can result a hard issue to overcome, and very promising thermodynamic schemes purely under the performance point of view, have been put out of games either by plant initial investment costs or huge operational complexity, that means operation higher costs due to outages or maintenance needs.
Non-condensing CO2 cycles raise the challenge of managing the huge real gas effects that compressors can experience approaching critical conditions; therefore, machinery operational flexibility can be largely compromised, resulting in major difficulties when looking at a given thermal cycle configuration ability to meet site-specific ambient conditions range of variation, and power modulation needs as well.
When designing a possible sCO2 power loop, a certain amount of the overall cycle and machinery complexity is to be attributed to the attempt of overcoming this challenge, beside to the attempt of increasing the thermal efficiency at the design point.
Up to date, it's difficult to foresee which level of cycle complexity will be assessed as acceptable by the market in sCO2 future commercial applications, but it' s almost sure that in the blend of the efficiency drivers to be chosen to hit market performance expectations, real gas effects can play an important role, as well as the possibility to manufacture industrial scale machinery with high efficiency and good reliability with respect to current state of the art, nevertheless the considerable reduction in diameters and increase in rotational speed at comparable power levels. What above mentioned implies that the optimization of turbomachinery shall be a whole with thermodynamic cycle one.
In this perspective, a non-condensing, recompressed and intercooled cycle configuration in which the re-compressor is mechanically driven by the power turbine, and the main turbocompressor is arranged with

its turbine on a unique balanced shaft, the turbine of which is fed by a main turbine bleeding in order to optimize its specific speed, is presented.

List of symbols and abbreviations

MTC	main turbo-compressor
MC	main compressor
RC	re-compressor
BPC	by-pass compressor
MT	main turbine
HHE	hot heat exchanger (cycle thermal input)
LTR	low temperature recuperator
HTR	high temperature recuperator
IC	inter-cooler
C	cooler
η	rotating machine efficiency
fr	mass flow rate fraction
DGS	Dry Gas Seal
Th Pwr fract	Thermal Power fraction
h/D_b	Blades height / base diameter ratio

INTRODUCTION

Among the various thermal cycle configuration suitable for CO2 use in closed loop (1) the non-condensing and recompressed one presents many advantages (2), allowing to achieve higher than (or comparable to) current state of the art commercial thermodynamic cycles performance, without the need of increasing the turbine inlet temperature above 700°C. Only most advanced large F and H class GTs combined cycles show an overall higher efficiency. With the exclusion of oxy-combustion Allam cycle solution (3), in the authors' thinking sCO2 will not represent a real competitor for this kind of technology in the short and medium term, since sCO2 requires a regenerative cycle configuration to achieve the highest efficiency figures, limiting the heat recovery ability from the gases exhausted from the GT of the bottoming cycle in combined cycle configuration, and the exhaust GTs temperature is currently too low to put sCO2 cycle in the condition to compete with 600°C class steam bottoming cycle (4) compensating the before mentioned limitation in GT exhaust heat recovery. On the other hand, even the sCO2 recuperative Brayton cycle (in closed loop) could result suitable in applications for which a reduction in plant footprint, weight and costs is predominant, since interesting efficiency can be achieved provided that the maximum cycle temperature approach and overcome 700°C. The non-condensing and recompressed cycle, leveraging both on the real gas effects during the compression and on rising the turbine inlet temperature at sufficiently high levels, can represent a competitive solution for different thermal sources ranging from CSP applications to fossil fuels and biomass ones, especially when cooling water is either scarcely available or not at all. The main issue in taking advantage of the fluid high density when approaching the critical point, is the abrupt change in volumetric flow at the compressor inlets that follows small changes in fluid temperature deriving from ambient conditions fluctuations.

Exergy team's work is currently aimed to design and build a demo-plant, the thermal input of which is nominally 20 MW, capable of adapting to various kind of energy source, from combustion to CSP application. The main turbine technology will be entirely coming from Exergy own know how and expertise, while solutions for the compressors design are still under evaluation. Partnerships with heat exchanger market technology leaders shall be taken into consideration.

To best fit the technology development into a realistic market scenario, addressing it from the beginning in the most possible suitable way, since now a scale up of the power block to the electrical power of 100 MW is considered; in this case, the chosen reference scenario is a CSP power plant.

THERMAL CYCLE

Although condensing cycles offer the possibility to reduce the compression work regardless to the specific working fluid, the relatively low critical temperature of CO2 along with the favorable values of the compressibility factor $z = \frac{p\ v}{R\ T}$ suggest the possibility to achieve good efficiencies with a completely supercritical recompressed cycle with air cooling. The attempt of maximizing the advantage due to the real gas effect during compression, with the minimum cycle pressure lower than 80 bar, can make this solution prone to operational troubles caused by abrupt changes in density (and other fluid properties) and, as a consequence, in volumetric flow at the compressor inlets (5). A possible variation of the recompressed non-condensing cycle, can be realized lowering the turbine exhaust pressure below the critical one, and splitting the recompression into two phases, the first of which deals with the entire turbine flow, here called RC, and addressing a fraction of it to the main compressor after intercooling, while the remaining flow is compressed by the second phase, here called BPC, to the exit of the high pressure side of the cold recuperator, according the cycle given in Fig. (1).

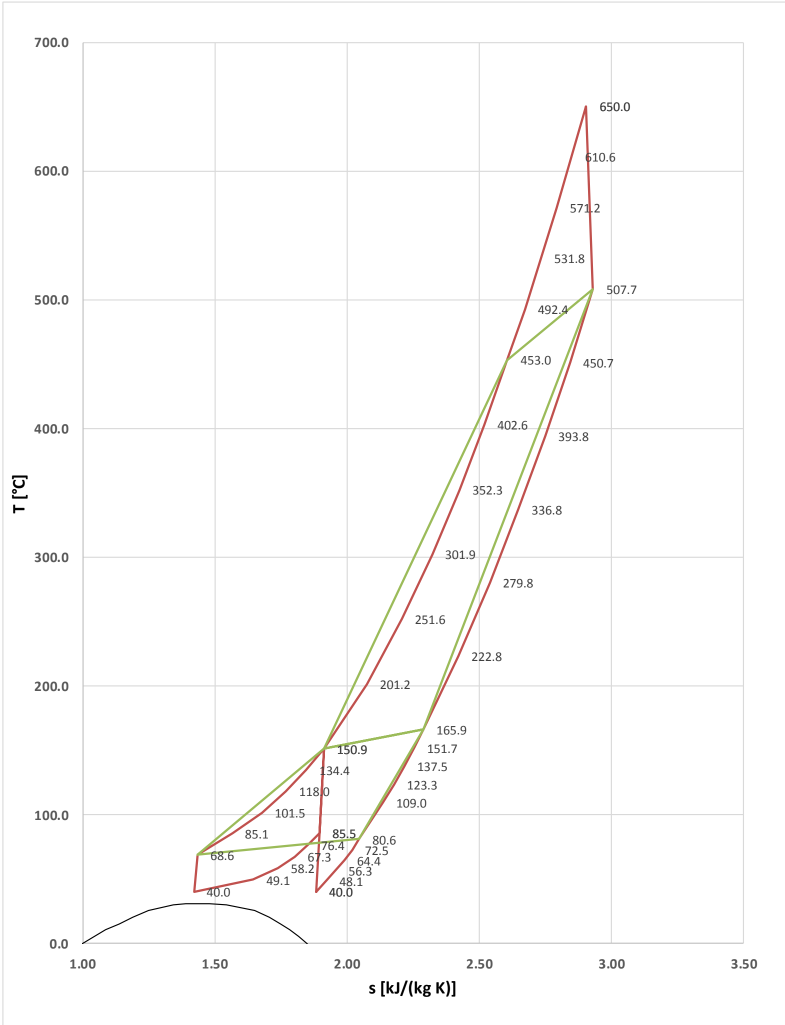

Fig. (1) – Recompressed and intercooled non-condensing cycle – Demo plant

In (1) the possible advantages in terms of efficiency of this scheme are clearly enunciated. This kind of cycle has been chosen as a good compromise between cycle efficiency and industrially acceptable level of complexity, potentially allowing to manage also off design conditions in a flexible way.

Setting at first the nominal maximum cycle temperature at 650°C, the minimum ones at RC and MC inlets at 40°C, with the main other operating and machinery parameters given in Table (1), the electric efficiency for a 20 MW thermal input power plant (demo plant) is expected to overcome 40%, while more than 45% can be reasonably got due to the improvements in machinery efficiencies and design refinements applicable when scaling-up the power cycle up to a thermal input of 240 MW, corresponding roughly to an electric power output of more than 100 MW.

Pth in [kW]	Tmax [°C]	pmax [bar]	pmin [bar]	Tmin 1 [°C]	Tmin 2 [°C]	η MT [%]	η RC [%]	ηTC [%]	RC fr [%]	P MT [kW]	P MC [kW]	P RC [kW]	P el [kW]	η el [%]
20000	*650*	*185.5*	*55.5*	*40*	*40*	*90*	*84*	*64*	*46*	*12543*	*898*	*3980*	*8207*	*41*

Tab. (1)

Following Fig. (2) shows the power loop schematic.

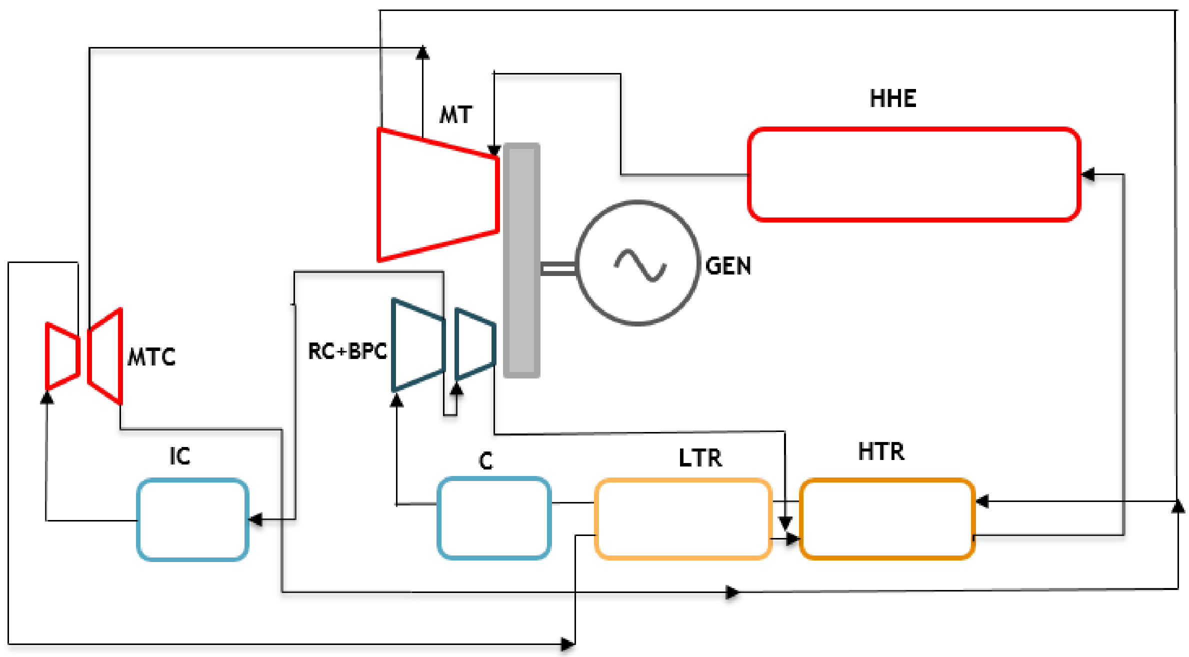

Fig. (2) – Power loop schematic – Demo plant

In case of use of combustion (fossil fuel, flare gas, biomass) as thermal source, the regenerative configuration of the power cycle causes the need of using a combustion air pre-heater in order to avoid huge detrimental effects on the overall power plant efficiency. As put in evidence by the cycle (T, s) diagram, the air inlet temperature to the "combustion chamber" would be higher than 450°C: an industrial design of an adequate air preheater has demonstrated that such a heat exchanger is currently available on the market, and it doesn't represent any particular technology challenge, even when the target is reaching very good heat exchange efficiencies in order to exhaust the combustion gases with temperatures lower than 120°C. The main issue in using such kind of devices is related to their footprint and weight: their cost is less critical with respect to the overall plant investment, provided that the combustion process of interest doesn't require high corrosion resistant materials.

Assuming constant the rotating machinery efficiencies, keeping constant as well the minimum cycle pressure and temperatures, the following figure shows the variation of the 100 MWel cycle efficiency at 650°C and 700°C of maximum temperature. In Fig. (3) the dashed curves are referred to calculation keeping equally fractioned the total pressure ratio on the RC and on the BPC, while the continuous curves are referred to optimum fractioning of the pressure ratio on the two compressors. Grey lines, the same for the two temperature levels, gives the pressure ratio for the BPC. The continuous line is referred to optimum fractioning, while the dashed one gives the pressure ratio when equally fractioned on the two compressors. The higher efficiencies correspond to Tmax = 700°C.

It's interesting to note that, for the optimized cycles, the unbalancing of the pressure ratio on the RC and the BPC gets more and more evident as the maximum pressure of the thermal cycle increases; since this could lead to mechanical design difficulties, it has to be taken into account.

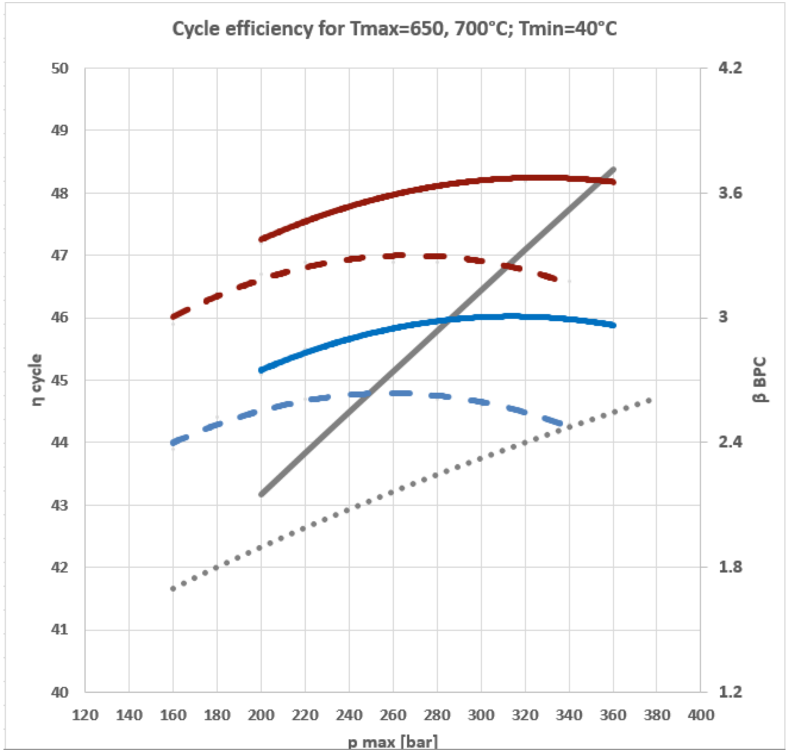

Fig. (3) – Thermal cycle efficiency with 650, 700°C of maximum temperature and corresponding BPC pressure ratio

For all calculations, CO_2 properties are always taken from REFPROP database and correlations.

The loop configuration chosen for the 240 MW thermal input plant is based on a relatively low maximum pressure if compared with the optimum one related to the thermal level; i.e., for 650°C, the pressure at main turbine valves is approximately 230 bar, that means less than 240 of max pressure at the exit of the HHE. This choice has been done to keep a conservative approach with respect to the loop hot parts cost and feasibility, with specific reference to a CSP receiver with CO_2 as primary fluid circulating into the receiver passages. Up to now, no detailed analysis of this aspect has been performed.

ROTATING MACHINERY

Main Turbine

According to what above exposed, the optimization of the cycle pressure has not been applied, especially for the 20 MW thermal input plant, in order to reduce overall plant weight and costs. In this case, the turbine inlet pressure will not exceed 180 bar. This limitation shouldn't affect the significance of the demo plant machinery running experience as a reference when the maximum pressure should be increased to 200 bar and more (up to 250) to gain efficiency for scaled up plants, neither under the material corrosion/scaling effects by operating fluid, nor under the general rotating machinery / heat exchangers configuration, provided the maximum cycle temperature will not be significantly increased (6). Currently, the selected material for rotor forgings and other hot critical parts up to 650 °C is the Fe-Ni alloy 1.4980 (or A 638 Grade 660). In any case, an extensive selected critical materials test campaign has been already planned and started.

The maximum cycle temperature will be 650°C, very close (difference < 1°C) to the turbine inlet one assuming that reasonable pressure losses are considered between the hot heat exchanger and the turbine stop valves.

Such a high fluid density combined to high inlet temperatures and pressures is resulting in very high power density, that, associated to high thermal gradients mainly due sCO2 high convective heat transfer coefficients, makes the development of a sCO2 turbine a challenging design process (7); this is why it's necessary to manufacture and test machinery similar to future industrial realizations to overcome the technical gap between early sCO2 studies phase laboratory power loop machinery and actual industrial scale future ones. In fact, the ability of CO2 to minimize the size of machinery, requires a minimum power to be able to use scaling up criteria widely used in turbomachinery design with an acceptable degree of confidence.

The overall configuration of the axial MT is similar to a HP steam turbine section; to better deal with high thermal gradients generated by the CO2 with respect to steam at similar conditions (up to 4 times higher), a barrel configuration with a horizontally split inner cylinder has been chosen. It's important to underline that also this configuration has been selected in order to test from the very early stage of the development the largest possible number of design choices suitable for a typical large industrial scale power plant. This approach has caused, for the demo plant main turbine, two major consequences: first, the turbine isn't as small as it could be, second, the rotational speed has been limited to achieve similar bladings $\frac{h}{D_b}$ ratio to future larger turbines, with a blades base diameter of approximately 270 mm. This second choice has been made taking into account that the rotational speed of the large size turbines shall be limited to synchronous speed (3000/3600 rpm). It's important to underline that sCO2 density cause the average tangential and axial forces acting on the blades to be huge with respect to steam or gas applications of comparable dimensions; for this reason, the blade slenderness and overall heights have to be limited.

Other features of the 11000 rpm, five 50% reaction stages main turbine currently under detailed design, are:

- Between pedestals rotor configuration
- Rotor (thrust bearing) and stators fixed points on the same machine side (HP side corresponding to combined thrust+journal bearing one)
- Tilting pads bearings
- Casings supported by pedestals with ambient conditions interposition between shaft ends seals and bearing casings
- Two symmetrical HP inlets, each of them equipped with an assembly of one 90° stop valve plus one 90° throttling valve, both of plug type
- Dry gas seals (DGS) at shaft ends (with CO2 separation and recovery system)
- Extensive use of internal flow path brush type seals
- Side entry roots, integral shroud, reaction advanced aerodynamic design tapered and twisted bladings

Mainly due to high power density, the rotordynamic stability of the turbine rotor can be an issue; accurate seals type and arrangement, along with rotor stiffness and bearing damping ranges selection have been performed to stay off the stability limit; preliminary calculations have confirmed the expectations. The span between the two journal bearings of the MT rotor is less than 1700 mm.

In Fig. (4) a plot of the frequency response to an unbalance of the MT rotor with actual bearings stiffness and damping figures is shown. The only critical speed of interest is evidenced at about 5500 rpm.

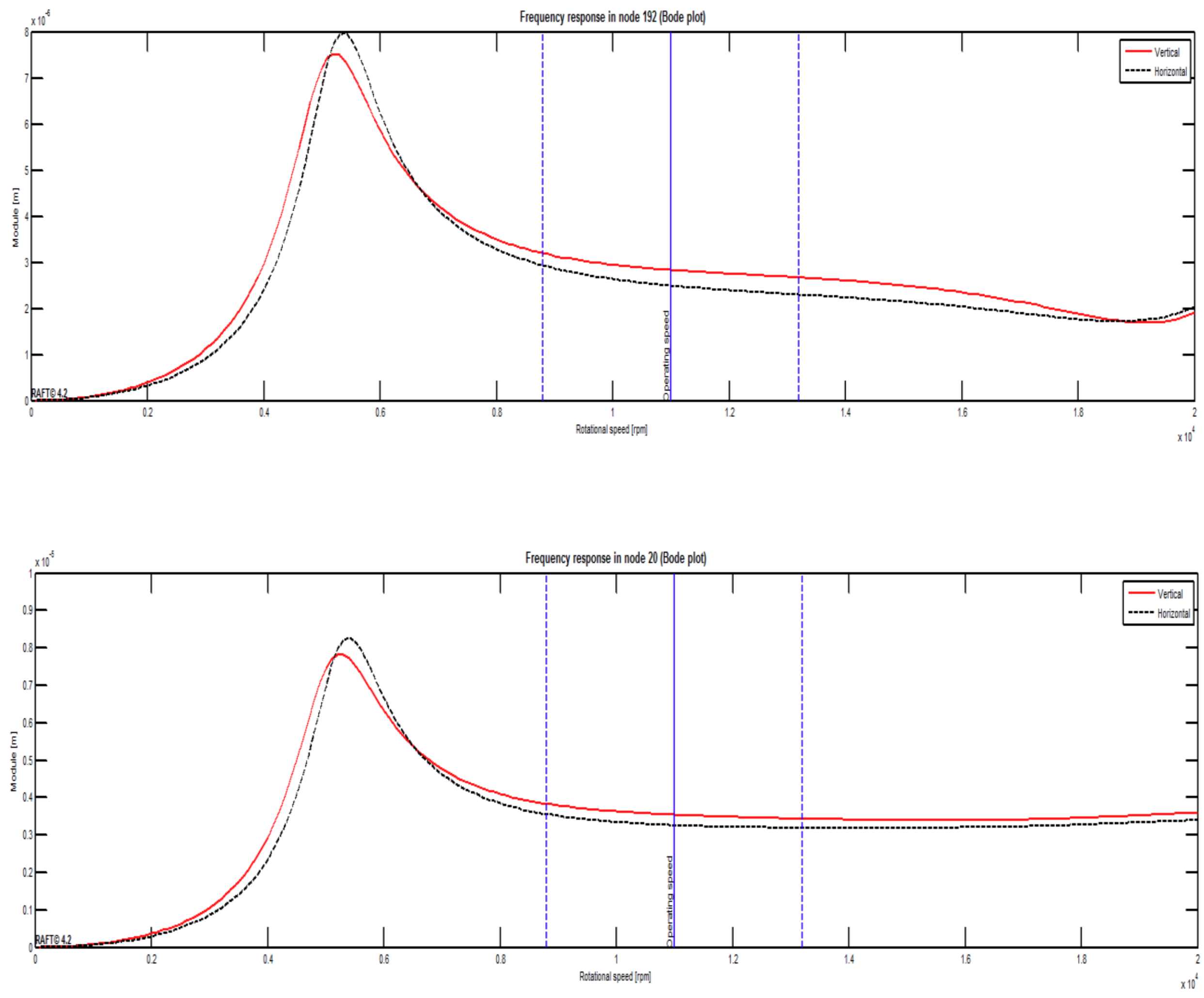

Fig. (4) – Frequency response @: exhaust side guide bearing (upper), inlet side guide bearing (lower)

In the nominal conditions of the demo plant, the MT power, with BTC feed bleeding in operation according to what below explained, is expected to be 12540 kW, roughly corresponding to 90% efficiency, pretty good for a turbine of this size. Good bladings performances, better than initially expected, have been calculated either through a proprietary 1D code and by ANSYS CFX on the same blades configuration, very likely due to favorable CO2 Mach number range at this flow conditions. In addition, the leakages should be minimized thanks to the advanced internal seals configuration. The proprietary 1D code tool is capable of operate both in direct approach (design) and inverse one (verification), taking into account seal leakages and all internal bladed path losses, including friction ones. The blading performances curves are based on specific profiles model. The MT efficiency shall be verified during the turbine performance test.

The turbine is also equipped with connections from the inner casing bleeding collecting chamber, to the external, axially arranged through the exhaust side cover in order to manage the thermal differential expansion. The bleeding, positioned before the last two turbine stages, has the purpose of feeding the MTC turbine with an enthalpy drop tailored to allow a proper design of this last machine. Fig. (5) shows a sectional view of the MT.

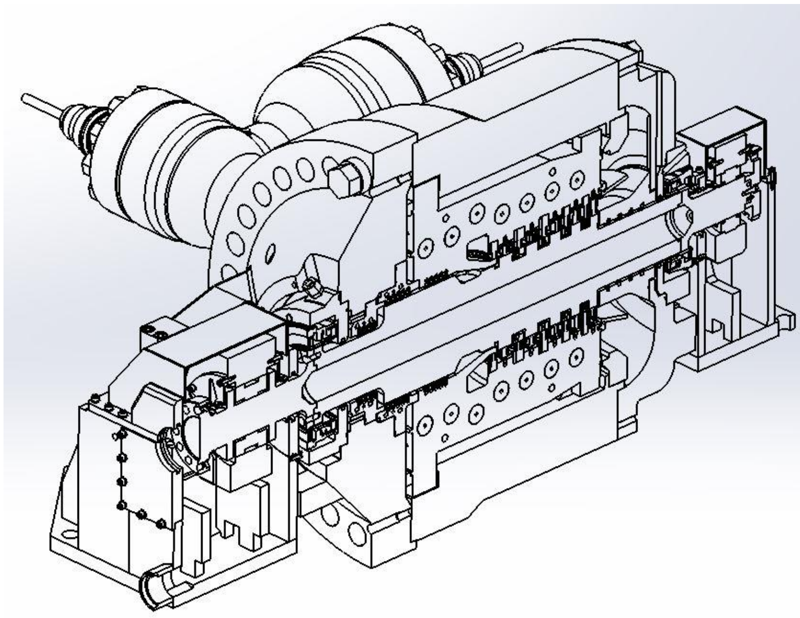

Fig. (5) – Demo plant main turbine sectional view

Main Turbocompressor (MTC)

What is here called MTC, it's the power balanced machines assembly composed by the single stage MC operating after the intercooling and the turbine driving it. The MC deals with a fraction of the flow exhausted by the MT and compressed by the RC, once the entire flow coming from the MT exhaust has been cooled in the hot side of the two recuperators. The actual amount of this fraction flow is selected on the base of the thermal cycle efficiency optimization at the desired operating point, and it's possible to avoid moving its value too much even when the plant is operated at partial load (as low as 60-70% of the nominal power). The peculiarity of the MC is the very low inlet volumetric flow when compared to any of other machines, coupled with a very low enthalpy rise as well.

The duty of this compressor requires radial or mixed flow machinery for the demo plant application size and, changing the rotational speed, the same configuration remains good even for much bigger plant power size. Defining the compressor specific speed and the specific diameter respectively as:

$$\omega_s = \omega \frac{\sqrt{\dot{Q}_{in}}}{{\Delta h_{is}}^{0.75}}$$

$$D_s = D \frac{{\Delta h_{is}}^{0.25}}{\sqrt{\dot{Q}_{in}}}$$

Good radial or mixed flow single stage compressors efficiency and operational flexibility can be achieved in the following range (8):

$$0.2 < \omega_s < 2$$

In Fig. (6) a diagram gives D_s and efficiency vs. ω_s according (8).

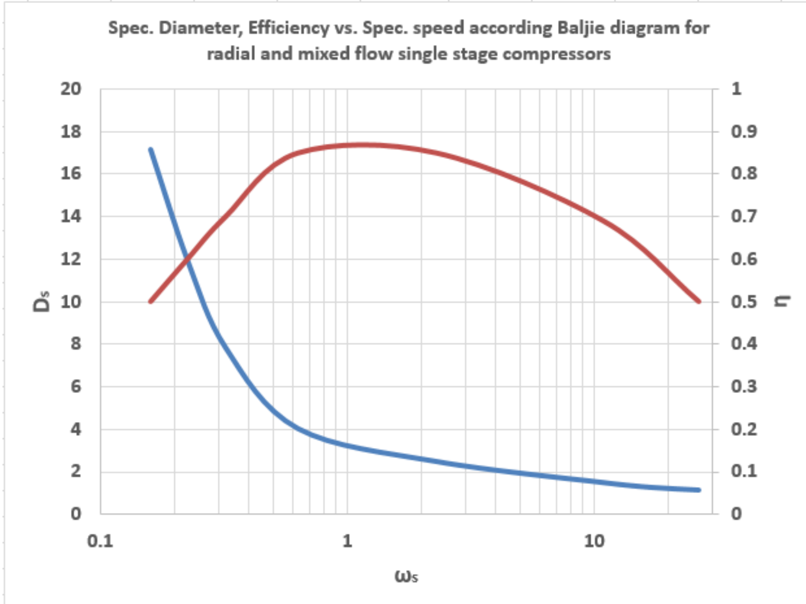

Fig. (6) – Specific diameter vs. specific speed extracted from Balje diagram for radial and mixed flow single stage compressors

The demo plant MC at nominal operational point has: $\Delta h_{is} = 15.3\ kJ/kg$ and $\dot{Q}_{in} = 0.081\ {}^{m^3}/_{s}$ (mass flow rate of 44.1 kg/s, density of 540 kg/m^3).

The choice for the demo plant is 35000 rpm for a single stage compressor driven by a single stage turbine; as a consequence, we get:

$$\omega_s = 0.76$$

for the MC.

According Fig. (6), selecting $D_s = 3.8$ allows to achieve good compressor stage efficiency; this assumption suggests a preliminary choice for the impeller diameter of 97 mm applying the D_s definition. Preliminary 1D MC calculation has indicated a larger diameter, 110 mm, as the best choice. The entire rotor 2D general drawing at full scale can be roughly contained by an A4 sheet.

The advantage of keeping the MC mechanically separate from the MT shaft become evident considering that in addition to the possibility to avoid the power losses associated to (a more complicated) gear box or an electrical drive, a high degree of operational flexibility along with good efficiency can be achieved adjusting the rotational speed of the booster shaft according the actual plant load and operational conditions, with main reference to ambient temperature directly affecting the volumetric flow at compressor inlet in the CO2 most critical region under this point of view. For this reason, the margin from surge at any transient condition is probably the main issue the opportunity to change MTC speed allows to overcome. According Exergy experience and performance maps for this kind of machinery, the MC efficiency for the demo plant has been prudentially assumed 0.76, including a correction factor of 0.96 due to the small size of the impeller, and attributing to the compressor the overall mechanical losses of the MTC, by a further efficiency detrimental factor of 0.92. The corresponding power absorption is 898 kW.

The turbine driving the MC needs to be fitted to the requested range of operational speed, staying within good or reasonable range of specific speed as well. The demo plant size as well as larger size plant can be served by a radial booster turbine. The definition of the specific speed is identical to the compressor one, once the volumetric flow at inlet is substituted by the outlet one. This reveals that using the whole main turbine isentropic enthalpy drop would result in design difficulties and, as consequence, in reduced operational flexibility of the MTC. Under this point of view, although the small power input of the MTC makes its efficiency less important with respect to other loop rotating machinery ones, it's useful to put in evidence that the BTC power request, i.e. the flow diverted from main turbine expansion with the associated reduction in electrical power, depends on the product of compressor and turbine efficiencies:

$$\dot{m}_{MTCt} = \frac{\dot{m}_{Mc}\Delta h_{is\ Mc}}{\eta_t\,\eta_c\eta_{m\ MTC}\Delta h_{is\ MTCt}}$$

The demo plant MT isentropic enthalpy drop at nominal condition is 183 kJ/kg, while the corresponding rise one for the MC is 15.3 kJ/kg, to which is associated a compressor flow of 44.1 kg/s. The MTC turbine flow, operating under the same enthalpy drop of the main one and rotating at same speed selected for the booster compressor, 35000 rpm, with a turbine efficiency of 0.84, would result in:

$$\dot{m}_{MTCt} = 5.77\ kg/s$$

calculating the associated volumetric flow at MTC turbine exhaust, positioned at the same pressure of the main one, trough the fluid density (about 38 kg/m^3), the turbine specific speed results

$$\omega_s = 0.161$$

value that is well away from the typical applications for this kind of machinery, usually ranging

$$0.4 < \omega_s < 1.2$$

In Fig. (7) a diagram with the typical total to static efficiency range vs. specific speed for radial turbines is given (9).

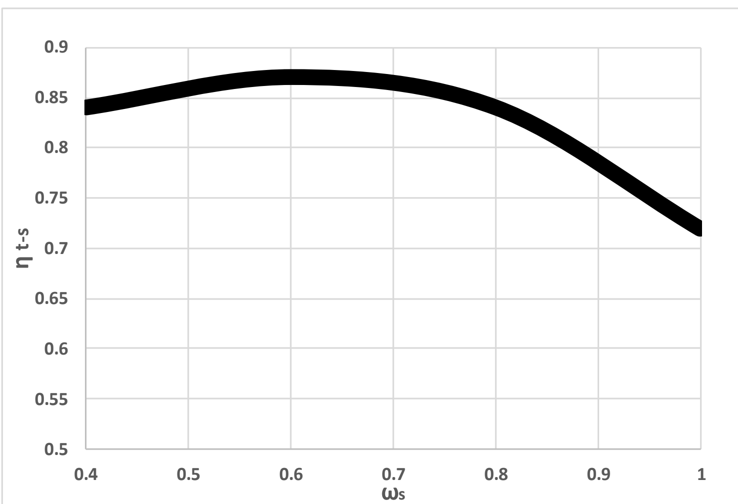

Fig. (7) – Typical total to static efficiency vs. specific speed for radial single stage turbines

The opportunity to select the right isentropic enthalpy drop through a bleeding from the main turbine (patent pending), allows a proper design of the MTC turbine. Assuming this makes possible to set the specific speed of the MTC turbine in the central area of ω_s range, a turbine efficiency of about 0.84 can be assumed, and positioning the bleeding from the main turbine before its fourth stage, the isentropic enthalpy drop becomes 68.5 kJ/kg once realistic circuit head losses have been taken into account.

Correspondingly, the MTC turbine mass flow results:

$$\dot{m}_{MTCt} = 15.43\ kg/s$$

with a corresponding

$$\omega_s = 0.55$$

allowing a proper design of the radial MTC single stage turbine at the same speed of the booster compressor. Moreover, a reasonable margin is allowable for changing the speed of the MTC during plant transients.

The MTC will be very likely supported and axially constrained by magnetic bearings.

Re-compressor (RC+BPC)

While the MC has been arranged to be directly driven by its dedicated turbine, the re-compressor of the demo-plant shall be driven directly by the MT through a gear box, on the same side of the MT, on the opposite one of the 1500 rpm synchronous generator.

The RC and BPC preliminary selected rotational speed is 18000 rpm, and the machine configuration is radial multistage, with the two phases, the first of them compressing the entire flow (RC) and the second one being the actual BPC, addressing the fraction that bypasses the LTR at the discharge of this, both arranged in a unique machine casing (10). This arrangement allows to have the MT and the RC+BPC joined together in one mechanical assembly, becoming for the larger industrial plant size a one shaft-same speed building block.

Both RC and BPC shall be equipped with its dedicated anti-surge bypass loop, as well the MC bypass will allow to manage specific plant transients.

The total power input of RC and BPC at nominal condition is 3980 kW, with an expected efficiency from Balje diagram (in agreement to preliminary 1D calculation) of 84%. As consequence, the expected gross power at generator flange, once taken into account the gear box losses calculated in 1.5 % of input power, is 8375 kW. With typical synchronous generator losses for this size, assumed as high as 2 %, it allows to achieve roughly 41% of gross electrical efficiency for the demo plant. Depending on the plant auxiliaries, the requested power of which can change related to type of thermal source and site conditions, the final plant net electrical efficiency can realistically reach or overcome 40%, a very interesting value for an 8 MWel size power facility.

Although an extensive analysis of the plant off-design and transient conditions is currently in progress, it can be interesting to point out that, depending on plant size, the thermal source, combustion or solar/nuclear kind, and on the related possibility to modulate quickly the thermal power input, different strategies are allowed by this loop configuration to manage transient plant conditions: they are essentially based on the possibility to choose operational modes in which either only the RC and BPC or only the MTC sustains the power loop, even with partial compressors and MT bypass and with the MT valves in throttling mode, while in normal operation the main turbine is operated in sliding pressure.

100 MWel power loop (CSP application)

The MT of the 100 MWel plant, keeps the same configuration of the demo one, with a synchronous rotational speed (3000 rpm), a blades base diameter of about 690 mm and 12 axial reaction stages. The calculated internal bladed path efficiency is as high as 93%, again essentially due favorable Mach number regimes and to the enhanced effectiveness in leakages reduction achieved by advanced seals configuration (brush type).

The expected turbine gross power is higher than 150 MW, with inlet conditions of 230 bar, 650°C and an exhaust pressure of 58.1 bar; the overall isentropic head of the main turbine is 216.7 kJ/kg. This corresponds to a main turbine efficiency as high as 92%. The minimum cycle temperatures (40°C) in nominal conditions is the same of the Demo plant. The generator efficiency is assumed to be 99%. With an expected efficiency of the compressors of 88% and 84% respectively for the RC+BPC and the MC, an overall performance of the thermal cycle can be calculated in 45.6%, referred to an electrical power of about 109500 kW.

The RC and the BPC are in this case arranged on the same shaft line of the MT, and consequently rotate at 3000 rpm. Both the two compressors are centrifugal and multistage. Depending on the final design of the compressor shafts, the shaft configuration could be either with three bearings, one interposed between turbine and compressors, or with only two. It's important to underline that a constraint on the rotordynamics will be very likely represented, on the larger size machinery, by the maximum diameter limiting the DGS slipping speed within the allowable value.

As before reported, simulations have been done for 700°C as maximum temperature to get an estimation of the possible gain in plant efficiency, even if the design process is currently focused on the 650°C solution. A preliminary evaluation of the possible stationary partial load operating conditions has been performed; reduction in maximum cycle temperature has been foreseen below 80% of nominal load. In Table 2 a resume of the power loop main stationary parameters is presented; thanks to the possibility of varying the speed of the MTC, and by means of IGV on the RC and on the BPC as well, the operating points of this machinery fall always in a typical operational range for this kind of machinery. So, no special issues are expected for partial loads as low as 70% of nominal power, provided that the loop transients are properly managed. Specific operational strategies are currently under definition: they will be tested by simulation codes like ASPEN or similar.

The conditions of minimum pressure and temperature have been kept constant; this is for sure an approximation implying the possibility to manage the capabilities of the heat exchangers even by partial by-pass. On the other hand, big changes in the thermodynamic conditions, and especially into the heat exchangers which provide a big fraction of the whole mass of CO2 contained into the closed loop, can force to move fluid in and out of the loop itself to preserve the overall mass balance and to allow to get the desired stationary operating condition, trough on or more capacities connected to the closed loop. These aspects have still to be deeply investigated, and the operational experience on the demo plant will be fundamental to confirm the validity of the foreseen behavior of the power loop during transients (11).

Table (2) gives the main expected operational figures for the CSP 100 MW el cycle, at different stationary operating conditions.

Th Pwr fract [%]	El Pwr [kW]	MT flow [kg/s]	p min [bar]	T in RC [°C]	RC pout [bar]	Tin MC [°C]	MT p in [bar]	MT T in [°C]	HT Rec out T [°C]	HT Rec eff. [%]	LT Rec eff. [%]
100%	109535	837.8	55.5	40	94	40	224	649.9	420.6	95.1%	91.8%
90%	97681	784.3	55.5	40	93.6	40	210.1	649.9	428.8	95.3%	91.5%
80%	85397	735.5	55.5	40	93.1	40	195.4	639.9	429.7	95.4%	91.1%
70%	72466	687.7	55.5	40	92.6	40	182.3	629.9	432.2	95.5%	90.7%

			Comp.s power			Vol. flow fract.					
Th Pwr fract [%]	Th. Cycle eff [%]	BP fract [%]	RC+BPC [kW]	MC [kW]	MT eff. [%]	MC [%]	RC [%]	BPC [%]	RC eff. [%]	BPC eff. [%]	MC eff. [%]
100%	45.6%	41%	42042	13104	92.1%	100%	100%	100%	88.0%	88.0%	84.0%
90%	45.2%	42%	40603	11153	92.1%	93%	94%	96%	87.3%	86.4%	83.0%
80%	44.5%	43%	35078	9234	92.1%	87%	88%	94%	87.9%	86.3%	83.0%
70%	43.1%	44%	31182	7605	90.5%	81%	82%	90%	87.8%	85.9%	83.0%

Table (2) – Main 100 MW el CSP stationary parameters at nominal and partial loads

Even if a CSP application very likely doesn't require operational stable conditions below 70% of the nominal load, at very partial loads during start-ups, the volumetric flow can be provided to the compressors even through the increase of the temperatures at their inlets, as well as by-passing the heat exchangers and having the antisurge loop of each compressor in continuous operation, along with the speed control of the MC and the position one of the IGVs of the RC and BPC. The MT can be partially by-passed as well as the bleed feeding the MTC turbine.

Conclusions

A specific sCO2 cycle configuration, with recompression and intercooling of the main compressor flow has been investigated, with the goal of achieving good cycle efficiency avoiding the condensation of CO2 and keeping sufficiently quite far away from the critical temperature, in order to reduce the critical issues related to compressors operation as the density change abruptly for small cooling medium temperature changes.

One of the solution used to achieve the purpose is to have the main compressor driven by a dedicated turbine on a power balanced shaft. To get a proper design of the two machines on the same shaft, the turbine driving the compressor is fed by a bleeding taken from the main turbine of the power loop.

Even limiting the maximum thermal cycle temperature at 650°C, very interesting efficiency levels have been calculated, nevertheless the conservative assumptions made for compressors efficiencies, both for the demo plant power size and for the large scale one.

Detailed design of the main turbine has been developed for a demonstrator, adopting all the main features that shall be applied on a large industrial scale plant main turbine.

After this, the basic design of a 100 MWel CSP power block has been developed, with preliminary calculation of partial load stationary operating parameters. The results appear to confirm the initial assumptions, showing the possibility to manage partial load operations and start up and shut down transients in a reasonably easy way.

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