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# SEMI – CLOSED HAT (SC-HAT) POWER CYCLE

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## Abstract

This new power cycle is derived from a simplified HAT cycle, with a partial recirculation of the exhaust gases added with respect to the traditional HAT configuration. The basic idea of applying recirculation to the HAT cycle stems from the interesting performance levels and general environmental advantages obtainable applying this technique to combined-cycle (SCGT/CC) and regenerative GT solutions (SCGT/RE); these power plants all share the integration with $CO_2$ chemical scrubbing of the exhaust stack in order to reduce greenhouse effects.

A relevant advantage of the proposed configuration over the original HAT solution is the possibility of complete water recovery from the separator before the recirculation node; here the temperature level is necessarily very low, allowing thus condensation of water produced by the natural-gas combustion process. This allows the self– sustainement of the HAT cycle, from the water consumption point of view, without any external supply. For the water separator, two thermodynamic models were developed (respectively simulating a single- and a multiple temperature condensation process), which have provided similar results.

The whole cycle is modeled using a modular code, thoroughly tested against the performance of a large set of existing GTs. The layout is derived from an existing HAT configuration, with suppression of the economizer section in the regenerator and the possible practice of external (non-recuperative) intercooling between the two compressors. The first choice is imposed by the presence of an additional low-temperature heat load for the $CO_2$ removal plant, while the second is sometimes necessary depending on the compressor pressure ratios and the possibility of including inside the cycle low-temperature internal cycle regeneration.

The expected performance of the plant is relatively high and close to those typical of HAT, SCGT/RE and SCGT/CC cycles: a LHV-based efficiency level exceeding 50% inclusive of $CO_2$ separation and delivery at ambient pressure and temperature; the specific work levels - in the range of 680 kJ/kg for the basic configuration - are lower than those of the HAT cycle but larger than for SCGT/CC and SCGT/RE solutions; the cycle requires relatively high overall pressure ratios (35 – 40). A notable improvement in specific work can be obtained with reheat.

## Introduction

The reduction of greenhouse effects puts stringent requirements for a substantial improvement in energy conversion and utilisation technologies. The goals prescribed by the Kyoto protocol are presently addressed in developed countries by improving the efficiency of energy conversion; however, it is very likely that this may prove to be insufficient in a very short time, so that other measures will probably be necessary and are already being experimented with pilot plants (Eliasson, 1998; Kongsjorden, 1998).

Within the possible options, $CO_2$ sequestration with disposal in the deep seas or oceans, or in exhausted gas or oil wells, is one of the most mature technologies, which is already being used in some pilot installations (DOE, 1999; Prutschek and Göttlicher, 1996, Langeland et al., 1993, Fujioka et al., 1997).

Cycle schemes developed around Semi-Closed Gas Turbines (SCGT) are the solution most close to present, open-cycle gas turbine technology (Corti et al., 1998 a, b and c; Corti et al., 1999, Facchini et al., 1996, 1997; Fiaschi and Manfrida, 1998 and 1999). In principle, the design of a suitable machine is viable adapting concepts currently employed in the normal turbomachinery practice, nevertheless that would require the physical construction of a machine that no manufacturer can at present deliver. In particular, adaptation of an existing gas turbine to this scheme of operation involves a substantial redesign of the combustor, and the addition of low-temperature equipment quite common in power plants or by chemical industries (by-pass and recirculation louvers, humidity separator, flue gas scrubbing equipment). The drawbacks over turbomachinery are typically small, and interesting solutions – such as self-produced water injection or absorption cooling - can be devised for recovering operation conditions very close to the design point. Other interesting solutions for $CO_2$ reduction plants are longer term, since their operating conditions require the design and development of specifically dedicated components, advanced with respect to current technology level (Mathieu et al., 1993, 1994 and 1995).

For all SCGT schemes, the exhaust gases are cooled down to about 40°C before recirculation; this allows the recovery of a large part of the combustion-produced water, which can eventually be reinjected in the cycle. Without water reinjection, the cycle rejects combustion-originated water to the environment, so that the well-known heat sink effect is exploited for performance improvement. On the whole, the amount of exhaust gas recirculated can be over 50%, considering typical operation of gas turbine with consistent overall excess air; this leads to a concentration of $CO_2$ in the exhaust (taking natural gas as reference fuel) larger than 12%. With such levels of concentration, scrubbing by aqueous or organic-base mixtures of amines is a well-tested technology, applied both for LPG purification and also by refineries and power plants (Corti et al., 1998 (a); Kohl and Riesenfeld, 1985).

## Nomenclature

| | |
|---|---|
| $\beta$ | Pressure Ratio |
| $\eta$ | Thermodynamic efficiency |
| $\Delta H_{dg}$ | Dry gas enthalpy drop (condensing section) |
| $\Delta H_v$ | Vapour enthalpy drop (condensing section) |
| $Cp_{dg}$ | Dry gas specific heat (condensing section) |
| $C_{pg}$ | Specic heat of the non condensing SEP (gas side) |
| $h_{cw}$ | Condensed vapour specific enthalpy |
| $h_{vout}$ | Vapour specific enthalpy at the SEP outlet |
| $h_{vsat}$ | Vapour specific enthalpy at the saturation point |
| $m_{compr}$ | Compressor inlet mass flow rate |
| $m_{dg}$ | SEP dry gas mass flow rate |
| $m_g$ | SEP gas mass flow rate |
| mstack | Stack mass flow rate |
| $m_{vin}$ | SEP inlet vapour mass flow rate |
| $m_{vout}$ | SEP outlet vapour mass flow rate |
| $m_{vsat}$ | SEP vapour mass flow rate at the saturation point |
| $m_{cw}$ | SEP condensed vapour mass flow rate |
| NTU | Number of Thermal Units |
| Pr | Prandtl Number |
| $Q_1$ | Heat exchanged in the non condensing SEP section |
| $Q_2$ | Heat exchanged in the condensing SEP section |
| SEP | Condensing heat exchanger (Separator) |
| SW | Specific Work (Relative to 1 kg/s compr. Flow rate) |
| $T_{in}$ | SEP inlet temperature (gas side) |
| $T_{max}$ | Maximum cycle temperature |
| $T_{out}$ | SEP outlet temperature (gas side) |
| $T_{sat}$ | SEP saturation temperature (gas side) |

## Basic cycle layout

The fundamental layout of a SC-HAT power plant is shown in Figure 1. The cycle includes two regenerative humidification sections, as in other HAT cycle proposals (Day and Rao, 1993; Rao and Joiner, 1990; Lindgren et al., 1992); typical components derived from the SCGT configuration are the by-pass louver system allowing partial recirculation of the exhausts and the cooler/water separator, needed before the compressor inlet to reestablish flow conditions acceptable by this component; the cooler section also allows condensation of most of the water developed by combustion of natural gas. This is the same water which is used for the humidification of the air stream, rendering thus the SC-HAT cycle completely self-sufficient from the point of view of water consumption. This is a notable advantage over all HAT cycle schemes which have been studied in the literature (Desideri and Di Maria, 1997, De Ruyck et al., 1995, De Ruyck, 1999).

Another advantage over most HAT schemes is the absence of any economizer – a component often needed in order to preheat the liquid water, recovering heat from the exhaust gas stream after the regenerator. In the SC-HAT, the economizer is effectively substituted by the heat load needed for the regeneration of the amine scrubbing solution, which allows to reach an exit temperature of the gas stream in the range of 80 °C; this also means that on the whole it is possible to add less water to the gas turbine stream (typically, an upper limit of about 20% was found), because preheating of the water before injection is only provided by the two intercoolers and the two aftercoolers (the economizer is suppressed). Moreover, as the water recovered from the cooler/separator is warmer than that which can be supplied by a conventional, open-loop reservoir, intercooling of the compression is less effective and it is necessary to add an external cooler after the two intercooler stages in order to effectively limit compression work.

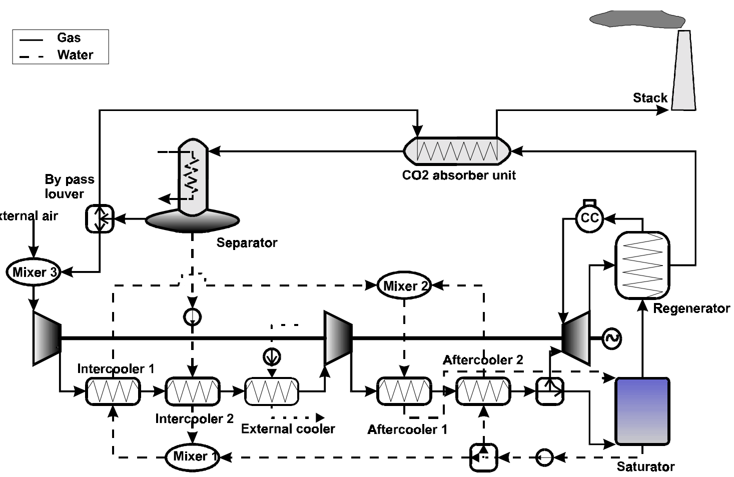


*Figure 1 – Schematic of the basic SC-HAT cycle*

 

Even if heat transfer equipment is added (external cooler; regenerators for the $CO_2$ absorber unit; cooler/separator), it should be noticed that suppression of the economizer and the fact that a large part of the heat exchangers are of the liquid/pressurized gas type guarantees a certain economy in heat transfer surface. From this point of view, the most critical piece of equipment is the cooler/separator – also because it is a non-conventional, two-phase heat exchanger with variable dewpoint conditions. This component deserves special attention and is examined in detail in the following.

## Features of the modular code

Given the complexity of the SC-HAT plant configuration, the present study requires the use of adequate calculation tools for plant simulation and performance predictions; it is also important that the calculation tools are validated against the performance of actual plants based on commercial GT engines. These conditions are fulfilled by the modular code developed by Carcasci and Facchini (1996); the reader is referred to previous papers (Carcasci and Facchini, 1996; Carcasci et al., 1996; Carcasci et al., 1997) for a complete presentation of the modular approach used.

A modular simulation code must be able to create a new power plant configuration, without creating a new source program. The code must also be able to handle any combination of input data. The modular code easily allows addition of new components (Carcasci and Facchini, 1996); in the present case, only the separation unit model was added.

The power plant configuration is defined by connecting a number of elementary components representing different unit operations, such as compressors, pumps, combustion chambers, splitters, mixers, etc. Thus each component is defined as a black box capable of simulating a given chemical and thermodynamic transformation. All equations defining the power plant are linearized (the coefficients are however updated in the course of the calculation), so the code reduces the non-linear equation system to a linear system with variable coefficients. All equations are solved simultaneously using a classic matrix method applied to an implicit linear system. With this approach, none of the data describing the different components of the system is considered essential, unlike a number of other semi-parallel or sequential methods.

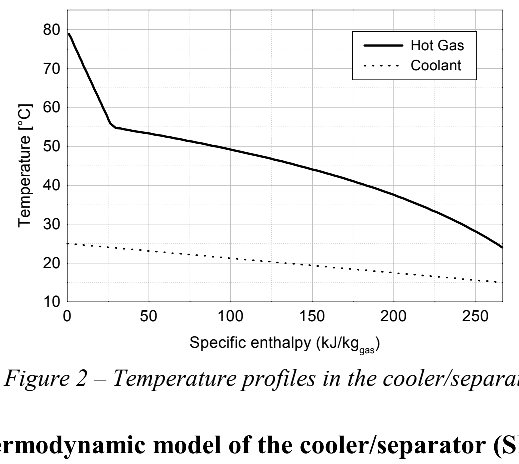


*Figure 2 – Temperature profiles in the cooler/separator*

## Thermodynamic model of the cooler/separator (SEP)

The cooler/separator is a key component in all SCGT solutions, whose purpose is to effectively cool the recirculated stream before the compressor inlet, and to separate by condensation most of the water vapor present in the stream. In the SC-HAT case, its importance is still increased by the fact that all the recuperated water has to be reinjected within the cycle.

The temperature profile of the cooler/separator is of the type shown in figure 2: in the entry section no condensation is present, while at a given section dewpoint conditions are reached ($T_{sat}$) and there is a substantial variation of the heat capacity of the humid gas stream, due to startup of condensation.

Several models of air/water mixers are possible, ranging from thermodynamic to thermo-fluid-dynamic with droplet evaporation/condensation modeling (Gallo et al., 1995; Desideri and Di Maria, 1997). A thermodynamic model was considered sufficient for use in the modular code in the present case: however, two different models were developed:

a) a simple, one-step one, developed from those used in other SCGT simulations (Facchini et al. 1996, 1997); this model assumes that all condensation takes place at $T_{sat}$.
b) a higher-level, multiple-step model, similar to that used by Desideri and Di Maria (1997), which divides the cooling/condensation process in a number of cascaded heat transfer sections (up to 80), with variable dew point temperature

The following equations describe the operation of the one-step model:

Energy balance of non-condensing section:

$$Q_1 = C_{pg} \cdot (T_{in} - T_{sat}) \cdot m_g$$

Mass and energy balances of condensing section:

$$m_g = m_{dg} + m_{vin}$$

$$Q_2 = \Delta H_{dg} + \Delta H_v$$

$$\Delta H_{dg} = Cp_{dg}(T_{sat} - T_{out}) m_{dg}$$

$$m_{vin} = m_{vsat} = m_{cw} + m_{vout}$$

$$\Delta H_v = (h_{vsat} - h_{vout}) m_{vout} + (h_{vsat} - h_{cw}) m_{cw}$$

As most of the heat subtracted to the wet fraction of the gas stream is due to phase transition, the assumption that the $h_{cw}$ conditions are at $T_{out}$ introduces a very small error and simplifies the solution.

The step-by-step process is much more correct from the point of view of thermodynamics of humid gas mixtures; the model starts after the dewpoint conditions are reached ($T_{sat}$) and applies energy and mass balances similar to the previous ones, but with $T_{sat}$ conditions recalculated after each step of condensation; the condensate is produced at $T_{sat}$ for each step. A number of up to 80 steps was experimented: however, as the value of the latent heat of vaporization is practically constant, and most heat after condensation is triggered is transferred from the $H_2O$ component of the mixture by condensation, the final result in terms of energy balance and amount of condensed water is very close for the one-step and multiple - step processes, even if a discrepancy of the average temperature of the condensation process of about 3 °C was found in the worst case (20% water vapor in the flue gases).

## Cycle performance

The performance of the SC-HAT cycle was analysed using the modular code, previously described and referenced. The cycle basic data are shown in Table 1.

| | |
|---|---|
| Compressor politropic efficiency | 0,89 |
| Gas turbine politropic efficiency | 0,91 |
| Water coolant temperature for the separator | 15°C |
| Pinch point temperature difference of the SEP | 15°C |
| Thermal load of the $CO_2$ removal plant | $3MJ/kg_{CO2}$ |
| Maximum cycle temperature ($T_{max}$) | 1623 K |
| Pressure ratio $\beta$ | 35 |

*Table 1 – Basic SC-HAT configuration data*

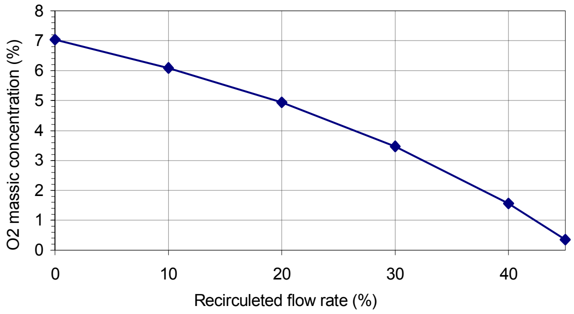


*Figure 3 – Concentration of $O_2$ in the recirculating flow versus recirculation percentage*

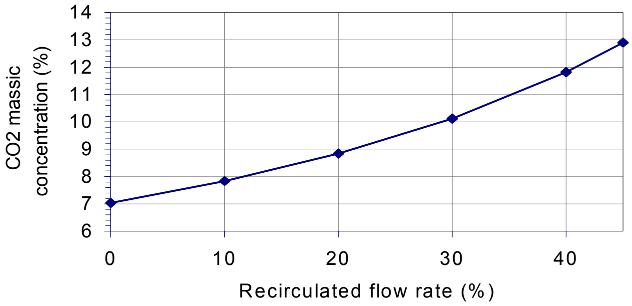


*Figure 4 – Concentration of $CO_2$ in the recirculating flow versus recirculation percentage*

The first step of the cycle analysis has been the evaluation of the main effects of recirculation, in order to determine the operative range for the amount of recirculated exhaust gases. The recirculated flow rate is limited by two parameters: the concentration of the oxygen at the combustor outlet and the concentration of the $CO_2$ at the $CO_2$ removal plant inlet. In Figure 3, the values of the $O_2$ concentration for different recirculated flow rate conditions are shown (percentage of SEP outlet mass flow rate, figure 1). This parameter fixes the upper theoretical limit for recirculation, reached when the $O_2$ concentration approaches zero and combustion is no longer possible.

The $CO_2$ concentration versus recirculated exhausts mass flow rate is shown in figure 4. The features of the removal plant and the associated economic considerations lead to fix the minimal recirculated flow rate at about the 30% of the SEP outlet, in order to achieve at least a 10% $CO_2$ mass concentration in the gas to be chemically scrubbed.

Off-design operation of turbomachinery is not included in this work, but the working fluid properties are kept under close control so that the relevance of this effect can be estimated. The analysis has shown two important features:

1) The cycle performance is not very sensitive to the amount of recirculation, if only thermodynamic effects are accounted.
2) Increasing the recirculated flow rate, the temperature and the composition of the gas at the compressor inlet change. In figures 5 and 6, the increase in temperature and humidity of the gas entering the compressor versus recirculated amount are shown. The first effect is partly compensated by the change in gas composition, which tends to reduce the inlet gas constant R, so a little recovery in gas density occurs; this, however, is not enough to compensate the effect of the temperature rise.

This last could put some difficulties for the technical feasibility of the plant, because compressors taken from commercial open-cycle GTs are designed to work with standard atmospheric air and are very sensitive to any variation of the inlet conditions (with special reference to aero-derivative units). In practice, the complete matching of compressor and gas turbine should be accurately checked, considering both the variations of conditions at compressor inlet and the overpressure which can be introduced by massive water injection on account of limited changes to turbine geometry. The amount of variation of compressor inlet conditions (considering temperature and density) is not large, so that usually pre-conditioning of compressor inlet can be avoided. However, this problem - coupled with statistical data on weather conditions on the power plant site - can sometimes suggest to apply preconditioning of the compressor flowrate (e.g. by absorption cooling) before recirculation.

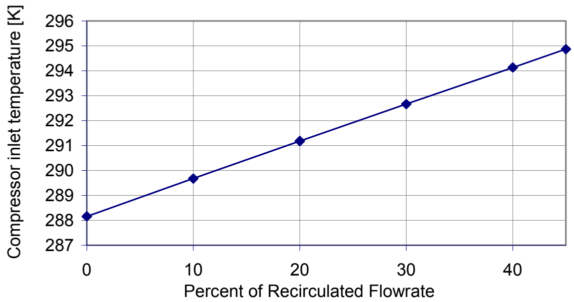


*Figure 5 – Compressor inlet temperature versus recirculation percentage*

sensitive to the thermal load of the $CO_2$ removal plant, (Table 2) because the degree of regeneration of the cycle is conditioned by the heat load of the removal plant.

- The thermodynamic efficiency and the specific power of the plant are close to those of a typical HAT cycle (Table 3; data for the HAT cycle are taken from Corti et al., 1998 (a)). The reduction of the stack flow rate, typical of semi-closed cycle plants, is emphasised by the separation of the water and of the $CO_2$ contained in the exhaust gas. At the maximum recirculation conditions, the mass of the exhaust gas sent to the stack is less than half of the gas flow rate entering the compressor (Table 3). This feature allows a remarkable reduction of the exhaust gas treatment costs.
- Assuming a $CO_2$ removal efficiency of 80%, the SC-HAT plant releases in the atmosphere 76,9 kg of $CO_2$ per MWh produced, while an "open" high efficiency cycle achieves values of 300-500 kg of $CO_2$/MWh. For this reason, the SC-HAT could represent an interesting short-time answer to the Kyoto conference goals.
- Cooling the exhaust gas at the separator to 30°C the cycle becomes self sufficient for the water needed for humidification of the gas stream.

| | Tmax [°C] | β | SW [kJ/kg] | η [%] | $m_{stack}/m_{compr}$ |
|---|---|---|---|---|---|
| HAT | 1377 | 35 | 744 | 53.2 | 1.19 |
| SC-HAT | 1350 | 35 | 659 | 51.8 | 0.48 |

*Table 3 – Comparison between standard HAT and SC-HAT cycle operation*

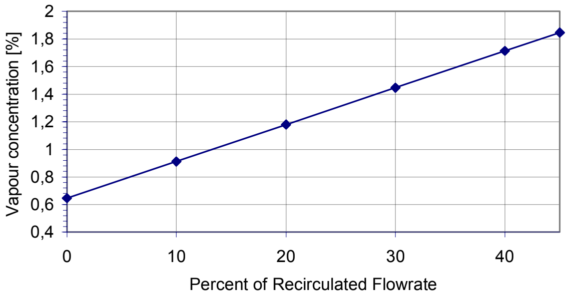


*Figure 6 – Compressor inlet vapour concentration versus recirculation percentage*

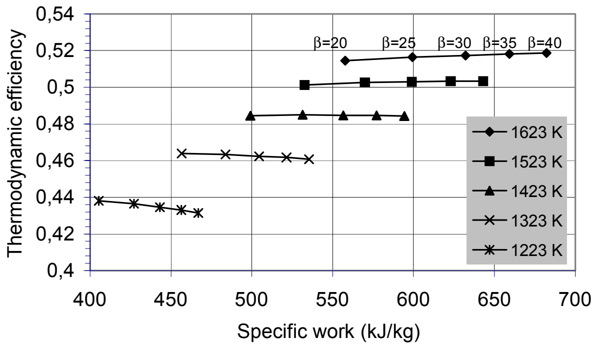


*Figure 7 –Performance map of the simple SC-HAT with variable β and $T_{max}$ (at maximum recirculation level, Thermal Load=3MJ/kg)*

| $CO_2$ Removal plant thermal load | SC-HAT Thermodynamic efficiency |
|---|---|
| kJ/kg ($CO_2$) | - |
| 2800 | 0,525 |
| 3000 | 0,518 |
| 3200 | 0,512 |
| 3400 | 0,505 |
| 3600 | 0,499 |

*Table 2 – Efficiency of SC-HAT cycle versus $CO_2$ removal thermal load ($T_{max}$=1623 K, β=35, max recirculation level)*

The cycle performance was analysed for the maximum possible recirculation, because of the attractiveness that these conditions offer for the treatment of the exhaust gases. The most significant results of the cycle performance analysis can be resumed as follows:

- In figure 7, the values of these parameters at different pressure ratios and turbine inlet temperature are shown (at the maximum 45% recirculation allowable). The thermodynamic efficiency depends mainly on the maximum cycle temperature, while the pressure ratio has a much greater influence on the specific power. The thermodynamic efficiency of the cycle is also very

## Preliminary heat transfer sizing of the cooler/separator (SEP)

Powerplants are not usually equipped with two-component/two-phase heat transfer equipment, even if this is common in chemical industry. The possible difficulties that can be encountered in sizing the cooler/separator (SEP) - referring to power plant or chemical plant practice – suggest that at least a preliminary analysis in this direction is needed. This can also be the starting point for a subsequent economical study of a topic component of the new proposed SC-HAT cycle.

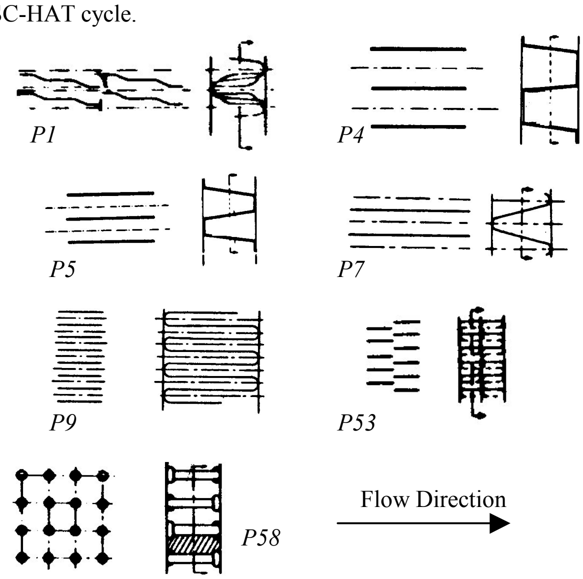


*Figure 8 – Schematic of CHED plates surfaces*

The reference compressor flow rate for the power plant is 100 kg/s, making reference to a power output of about 70 MW. The use of commonly available compact heat exchanger (Kakac and Liu, 1998) is here considered; calculations were performed by the means of the commercial CHED Software (Kays and London, 1987). Several kinds of surface couples have been analyzed and compared (figure 8). The disposition of all the couples of surfaces here analyzed is horizontal counter – flow.

Two operating conditions can typically occur, with respect to the coolant available: open-loop cooling (river, sea etc.) or use of a cooling tower. In the first case, a limited coolant side temperature differences ($\Delta T$ = 10 °C) must be adopted for environmental constraints, which causes high coolant flow rates; in the second case, the coolant $\Delta T$ can approach 40 °C, with a strong reduction of coolant flowrate with respect to the preceding case. Table 4 summarizes the main operating conditions in the two cases.

<table>
<tr><th></th><th></th><th>m [kg/s]</th><th>p [kPa]</th><th>T [K]</th><th>Cp<sub>m</sub> [kJ/kgK]</th><th>Pr</th></tr>
<tr><td rowspan="2">Open Loop</td><td>Water (in)</td><td>1234</td><td>-</td><td>288</td><td rowspan="4">4.181</td><td rowspan="4">7.091</td></tr>
<tr><td>Water (out)</td><td>1234</td><td>101,3</td><td>298</td></tr>
<tr><td rowspan="2">Cooling Tower</td><td>Water (in)</td><td>308</td><td>-</td><td>288</td></tr>
<tr><td>Water (out)</td><td>308</td><td>101,3</td><td>328</td></tr>
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*Table 4 - Cold side SEP operating conditions*

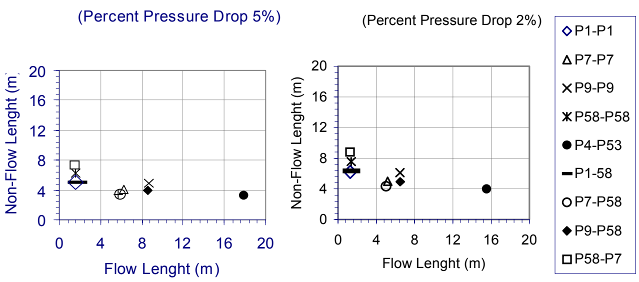



*Figure 9 - SEP sizing with Open-Loop cooling (NTU = 10.5)*

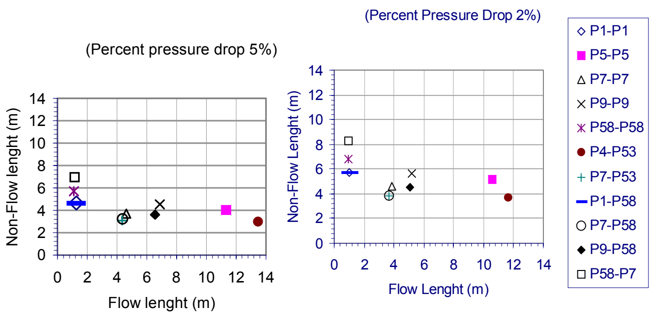



*Figure 10 - SEP sizing with with Cooling Tower (NTU = 14.7)*

For each case, two different percent pressure drops were considered due to the friction (gas side): 2% and 5% (the cycle performance analysis was carried out on 2% pressure drops basis). Figures 9 and 10 summarize the results of the SEP sizing, for several couples of surfaces (labeled as indicated on CHED Software, notice that the two orthogonal non – flow dimensions are the same). It is clear from figure 9 (open loop cooling) that allowing higher pressure drops, the heat exchanger is slightly reduced in the non – flow direction, and increased in the flow direction. This is due to the increase of flow velocity and, consequently, in Reynolds number. In this way, an improvement of the overall heat transfer coefficient is obtained (so that a an economy in heat transfer surface is achieved), as well as an obvious reduction of the cross – sectional area needed for the same fixed flow rate. This implies the adoption of relatively long (in the flow direction) heat exchangers; on the whole, the SEP volume is reduced. The increase in pressure losses is convenient for this heat exchanger only considering surfaces with a low elemental cross sectional area, having non-flow dimensions slightly higher with respect to flow dimensions; in cases with elemental large size in the flow direction, the increase in pressure drop leads to less compact configurations. This behavior is somewhat different from that found in air – air heat exchanger typical of gas turbine regenerators, where the non flow dimensions are dominant: it is mainly due to the different cooling media, water instead of air (Corti et al., 1998, (b)). One of the best choices is the couple P1 – P58 (louvered fin gas side and pin – fin water side), but also P7 – P7 and P7 – P58 show very interesting compactness characteristics.

Similar conclusions can be drawn for the case with evaporative cooling towers (figure 10): in this case, the heat exchanger size is on the whole larger on account of the higher mean coolant-side temperature, which is closer to the gas side value; the result is a considerably larger NTU value. In this case, the best couples of surfaces are the P1 – P1 (louvered fins) and P1 – P58, which achieve also a good compactness with respect to the case without cooling tower. Other intermediate couples still remain attractive both for size and compactness.

On the whole, the preliminary analysis of the SEP size has shown the possibility of realizing this heat exchanger if adequate kinds of enhanced heat transfer surfaces are chosen, without incurring in economically and practically prohibitive sizes.

## Possible improvements to the cycle (Absorption chilling of the recirculated stream; reheating of the exhausts)

The analysis of the basic SC-HAT cycle has raised two technical questions that suggest the introduction of some modifications to improve the feasibility of the plant:

1. The conditions of the gas at the compressor inlet can lead to some degree of off – design operation. Moreover, depending on the power plant location and prevailing climate conditions, pre-conditioning of the flowrate at compressor inlet can be considered in order to allow operation close to ISO-rated conditions.
2. The exhaust gas temperature at the outlet of the removal plant is too low (about 50 °C) to allow the dispersion in the atmosphere by a simple draft stack.

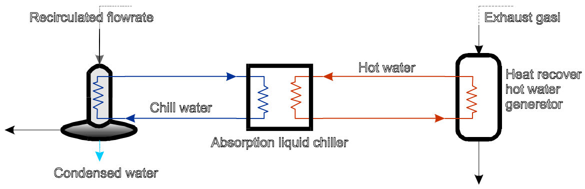



*Figure 11 – Schematic of the absorption chiller*

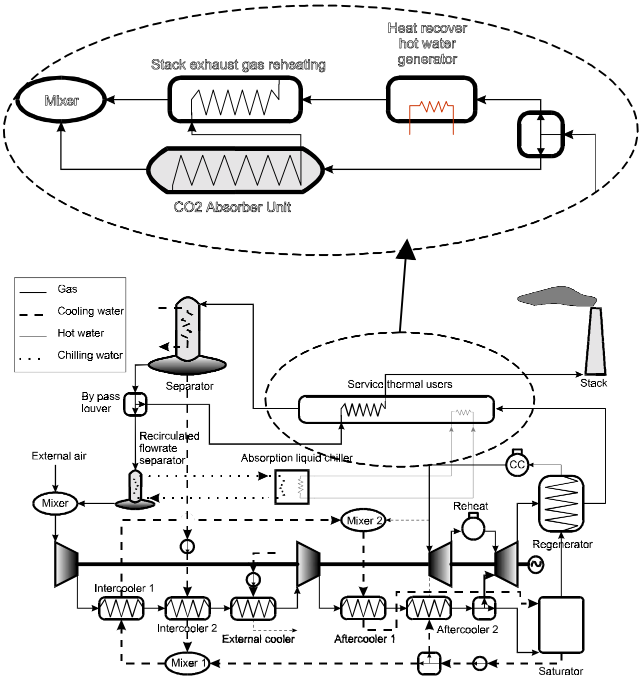
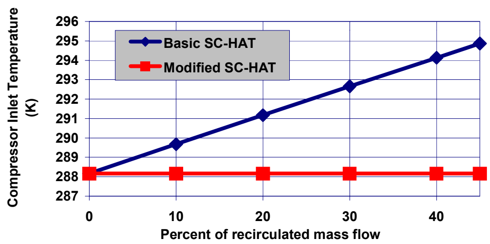
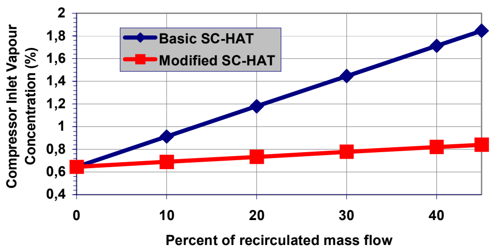



*Figure 12 – Modified SC-HAT configuration with reheat and amount compressor cooler*




*(a)*




*(b)*

*Figure 13 – Behaviour of Compressor Inlet Temperature and vapour concentration versus percent of recirculated flow*

To restore the standard conditions at the compressor inlet, cooling of the recirculating flow by an absorption refrigeration plant was considered (figure 11). This kind of solution is already being used in the energy conversion field, to tune the compressor inlet conditions, with special reference to power plants based on modern aero-derivative gas turbines (such as the LM6000). In SC–HAT cycle, only the recirculated stream (which at the separator outlet is at the temperature of 30°C ), is cooled down to ISO conditions. The attractiveness of the absorption refrigerating cycle is the possibility to use low temperature heat extracted from the exhaust gas, in parallel to the heat extraction of the scrubbing solution: this recovered heat provides the hot water stream for the absorption unit. For this analysis, the authors referred to commercial water- Lithium Bromide absorption machines of different sizes like those built by Mitsubishi Heavy Industries. The heat load of the absorption unit is typically 1/6 of that of the amine scrubber, so that the overall thermal balance of the cycle is only slightly affected.
To reheat the stack exhaust gas at the Carbon dioxide removal plant, a counter current gas-gas heat exchanger is introduced in the exhaust gas circuit. The stack exhaust stream is heated up to 90°C, with a minimum temperature difference of 25°.
These two modifications increase the overall heat extraction from the gas stream at turbine exit. In order to be able to provide this enhanced thermal load, reheat has been introduced in the gas turbine. This allows to raise appreciably the turbine exhaust temperature, in order to obtain the heat required for all the downstream thermal loads. The solution is similar to that practised in some large size industrial gas turbines (ABB GT24 and GT26), in combined cycle configuration.
In order to optimise heat transfer for all thermal users, the low temperature part of the plant was re-configured. In figure 12, the proposed configuration is shown.
The main results of the modified cycle analysis can be summarised as follows:

1) The exhaust gas temperature at the stack is adequate to allow natural dispersion in the atmosphere.
2) The temperature at the compressor inlet does not change varying the recirculated flow rate. In figure 13 (a) and (b), the basic and the modified cycle behaviour are shown. Also the gas humidity variation is very limited, as is shown in figure 13 (b). On the whole, compressor inlet conditions very close to design open-cycle operation can be restored.
3) Under optimised operating conditions, the modified cycle inclusive of absorption cooling, reheat turbine and stack reheat produces a loss in efficiency of about 1.2 percentage points, and an increase of specific power to about 710 kJ/kg from the original 659 kJ/kg (Table 3). The optimal reheat temperature is about 1400 K, but the thermodynamic efficiency and the specific power decay is limited (about 3% of the optimal values) if different reheat temperatures are considered in the range between 1300 K to 1450 K.

## Conclusions

A new gas turbine based power cycle, named SC – HAT, has been proposed, starting from two previously studied power cycles: the HAT and the SCGT. The basic idea of the SC-HAT is to combine the interesting features of both cycles:

a strong reduction of $CO_2$ emissions, by means of partial recirculation of the exhaust gases and application of $CO_2$ sequestration by chemical scrubbing, and high performance levels without a combined-cycle arrangement (especially in terms of specific work, typically high in the HAT solutions).
The coupling of the two cycles is interesting also to resolve the water consumption problem of the HAT cycle; the SCGT is, in fact, a water-producing cycle.
Possible improvements to the basic cycle layout, which should facilitate its feasibility, were also identified and discussed (absorption cooling of the recirculated stream; raise of stack temperature; introduction of GT reheat).
A preliminary sizing of the recirculating heat exchanger (SEP) has been carried out, showing the existence of interesting compact solutions.
On the whole, the SC-HAT cycle appears to be an appealing candidate for the development of fossil-fuel power cycles of new generation, inclusive of $CO_2$ sequestration.

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