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Efficiency enhancement of combined cycles by suitable working fluids and operating conditions

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ABSTRACT

Solar energy based combined cycle power plants are becoming important as an efficient option among conventional thermal power plants. However conventional thermal efficiency can be significantly improved. This research study is centred on combined cycle efficiency enhancement by researching the capacity of several working fluids such as N2, air, or He for the topping cycle which is a closed Brayton cycle (CBC) and a bottoming cycle which is a Rankine cycle (RC) operating with xenon, ethane or ammonia as working fluids. The applied strategy, which aims to increase the ideal thermal efficiency, is based on the concepts of quasi-critical condensation pressure, residual heat recovery and properly selected working fluids. The decision to propose N2, air, or He, as working fluid for the Brayton part of the CC stems from the fact that they yield high efficiency at high temperatures with acceptable power ratio. A performance study of several organic and nonorganic working fluids such as ethane, xenon and ammonia for the bottoming Rankine cycle and N2, air, or He, for topping CBC is performed. The consequences are a significant positive increment in thermal efficiency in comparison with conventional CC power plants.

1. Introduction

The objective of the research study is to determine whether a CC composed by a CBC and a RC power plant could be viable using high temperatures solar concentrated energy and/or fossil fuel (natural gas), operating with N2, air, or He at the topping cycle or CBC as working fluid while using ethane, xenon or ammonia as working fluid at the bottoming cycle or RC under acceptable thermal efficiencies. The scheme consists of a top CBC engine mounted at the focus of a solar tower type concentrator, together with a heat exchanger transferring the waste heat from the gas turbine to a working fluid (organic or not) of the bottoming cycle that would operate under a RC or an ORC depending on the operating fluid characteristics.

Gas turbines are cheaper, simpler engines and are able to support higher temperatures. Solar heat can be used to replace or complement the combustors of gas turbines [1], potentially providing a reliable and cheaper overall system. Gas turbines also permit much higher cycle temperatures, which contribute to thermal efficiency enhancement.

Some previous studies on solar CC power plants have been performed during the last two decades. In [2] an energy source based on the parabolic troughs has been investigated providing the latent-heat part of the input for a RC, with a topping CBC providing the other part. Such approach is relevant for parabolic trough systems, but as the top temperature is relatively low it is not appropriate in the case of point-focus concentrators, where the temperatures can be higher than 1000 °C. In [3] a similar configuration is mentioned, with the benefit of a bottoming cycle configuration, such as the Kalina cycle, which contributes to reducing irreversibilities in the combined-cycle heat exchanger.

A CC based on a solar-fired CBC and a steam RC has been studied by [4]. According such study, the consideration of alternative fluids in the bottoming cycle is realised mainly to resolve pinch-point issues in the steam generator, associated with irreversibility given at thermal storage.

Although the ORC is a well-known option for conversion of low grade heat to mechanical work, these systems have been used with solar ponds in [5] and low-temperature parabolic troughs in some applications including USA and Spain among others, but neither of these projects claim to have achieved satisfactory thermal-to-electric conversion efficiencies, due mostly to the low ($<$96 °C and $<$180 °C, respectively) temperatures of the heat source.

In [6] the study of organic fluids suitable for use in regenerative ORCs at temperatures up to 630 K was performed, which is a limitation imposed on many organic working fluids due to chemical decomposition associated with properties lost. They found 700 potential working fluids meeting a set of criteria oriented towards

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Nomenclature

h	specific enthalpy (kJ/kg)
s	specific entropy (kJ/kg-K)
T	temperature (K)
p	pressure (bar)
η	thermal efficiency
η_R	thermal efficiency of the RC
η_B	thermal efficiency of the CBC
η_{cc}	thermal efficiency of the CC
W	specific work (kJ/kg)
W_i	input specific work (kJ/kg)
W_o	output specific work (kJ/kg)
W_n	net specific work (kJ/kg)
Q	specific heat flow (kJ/kg-s)
Q_i	specific input heat flow (kJ/kg-s)
Q_o	specific output heat flow (kJ/kg-s)
Q_{iB}	specific input heat flow (kJ/kg-s) BC
Q_{oB}	specific input heat flow (kJ/kg-s) BC
Q_{iR}	specific output hat flow (kJ/kg-s) RC
Q_{oR}	specific output hat flow (kJ/kg-s) RC
r_P	pressure ratio of the CBC
η_{VR}	thermal efficiency of the VR
Qu	useful heat flow rate (kJ/kg-s)
Qs	supplied solar heat flow (kJ/kg-s)
$\dot{m}_{HTF}$	HTF mass flow rate (kg/s)
Cp	specific heat capacity (kJ/kg K)
To	HTF receiver output temperature (K)
Ti	HTF receiver input temperature (K)
T_A	arithmetic mean absorber temperature (K)
T_F	average HTF temperature (m^2)
A_{Ap}	VR normal aperture area (m^2)
A_{Ab}	absorber area of the VR (m^2)
A_{PD}	parabolic dish normal surface area (m^2)
U	heat transfer coefficient of the absorber
C	solar concentration factor
E_S	solar normal radiation density (800 W/ m^2)

Acronyms

CC	combine cycle
ORC	organic Rankine cycle
RC	Rankine cycle
CBC	closed Brayton cycle
ODP	zero ozone depletion potential
GWP	global warming potential
SP	state point
HP	high pressure turbine
LP	low pressure turbine
WF	working fluid
HTR	helium cooled high temperature reactor
VR	volumetric receiver

biomass cogeneration applications. The results of such study allowed for heat rejection temperatures approaching 363 K, which is not sufficiently low, since no cogeneration output is proposed in the present work.

A relevant characteristic to be taken into account with regard to organic (and nonorganic) working fluids for use in RCs, is its thermal stability. For instance, the ethane is reported to be stable up to around 675 (K), but above this temperature, it starts to chemically break down [7]. The exact temperature limits appear not to be very well understood, and seem to be affected by oxygen and other impurities. In this study, it is assumed an upper limit of 700 K for ammonia, 675 for ethane and 750 K for Xenon, while for CO2, which is not organic, there is no practical limit. It must be considered that working fluids selection [8] and cycle modelling [8] are two relevant issues that contribute to a substantial improvement in cycle efficiency. For cycle modelling a database [8] has been used. This is a publication used for property calculations in commercial software packages (REFPROP) whose library makes use of Helmholtz fundamental equation correlations to determine fluid properties from the highest-accuracy published data available in the literature. Experience is gained in last decade with the use of carbon dioxide as working fluid in both the Brayton and RCs [9,10]. Nitrogen, air or helium are proposed as working fluids for the top cycle (BC or CBC) because of some special properties. Proposed working fluids yield a high efficiency at proposed high temperatures, even under relatively acceptable power ratios.

2. Conventional criteria to improve closed Brayton and Rankine cycles efficiencies

The main feature of both cycles is the compression phase (pumping that occurs when the working fluid, including water, is in the liquid phase for RC or gas compression in CBC). The amount of energy available for extraction by the working fluid is dependent on the operating temperature and pressure of the fluid. In the case of the RC, raising the steam temperature, the difference between top and bottom cycle temperatures determines how much energy can be extracted by the turbine, and thus the efficiency of the cycle. Typically the condenser operates at a temperature and pressure which depends on the external conditions such as the ambient temperature or the cooling fluid temperature. It is not feasible to substantially lower this line. The only technically available way to improve cycle efficiency is by pushing the upper temperature higher. In the actual state of the art technology fluid temperature is limited by available materials that can survive at elevated temperatures while maintaining their mechanical properties [11–13].

2.1. Basic ultra Rankine cycle analysis

There are four processes in the RC, each changing the state of the working fluid. These states are identified by numbers in the diagram shown in Fig. 1.

The ideal RC is inspired by the Carnot cycle. In Fig. 3 it is depicted in the T–S diagram associated with the physical components. According to Fig. 1 and equation (4), the thermodynamic efficiency of the cycle is defined as the ratio of net power output to heat input.

$$\eta_R = \frac{W_o - W_i}{Q_i} = \frac{W_n}{Q_i} = \frac{Q_i - Q_o}{Q_i} \tag{1}$$

Efficiency is bounded so that $0 \leq \eta \leq 1$ due to the irreversibility associated with the fulfillment of the second law.

2.2. Basic closed Brayton cycle analysis

The T–S diagram of an ideal CBC is shown in Fig. 2. All four processes of the CBC are executed in steady-flow devices so they should be analyzed as steady-flow processes.

When the changes in kinetic and potential energies are neglected, the thermal efficiency can be expressed as

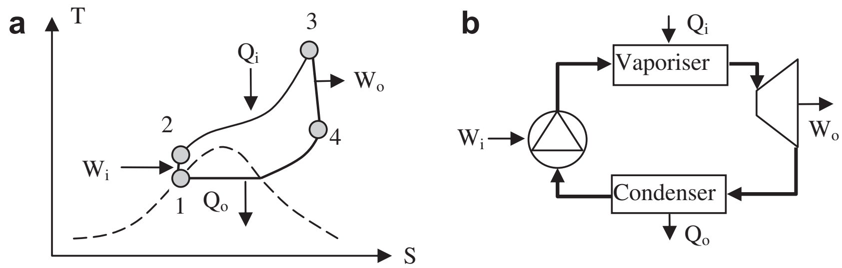

Fig. 1. Basic supercritical RC: (a) T–S diagram. (b) basic plant structure.

$$\eta_R = \frac{W_o - W_i}{W_i} = \frac{W_n}{Q_i} = 1 - \frac{Q_o}{Q_i} \tag{2}$$

The efficiency of the Brayton cycle as well as the specific power depends strongly on the top cycle temperature, isentropic efficiencies and pressure ratio.

2.3. The combined cycle efficiency

In the following analysis the advantages of implementing a combination of two cascade cycles (CBC–RC) are highlighted. The thermal efficiency according the structure of the basic CC shown in Fig. 3 is defined as

$$\eta_{CC} = \frac{Q_{iB} - r_m \cdot Q_{oR}}{Q_{iB}} \tag{3}$$

where the ratio of specific heat flow rates is $r_m = Q_{oB}/Q_{iR}$

3. Proposed viable strategy to increase the combined cycle thermal efficiency

Conventionally it is assumed [14–17], that efficiency can be improved by increasing the cycle top temperature T_H and/or lowering the bottom temperature T_C, which requires maximising W and/or minimising Q_i.

While some working fluids including carbon dioxide have been studied as alternative working fluid for supercritical RCs by some researchers [18–21], research results conducted to report the power plant thermal efficiency of the cycle to be too low.

In [9] and [10] new thermodynamic cycles using supercritical CO2 have been studied, in which both solar energy and carbon dioxide are used to form a combined system of power generation and water based low grade heat recovery, which approaches an efficiency of 20%.

In this research work effort was put into looking for new realizable alternatives showing that the efficiency can be improved by means of CCs in which the main contribution consists of a combination of:

- A topping CBC powered by high temperature (1300 K) solar or combustion energy, in which alternatively working fluids are N2, air or He, or alternatively high temperature reactor based energy at about 1200 K.
- An ultra-supercritical RC (bottoming cycle) in cascade with the CBC, in which the operating fluid is alternatively selected according its properties as (organic and non organic) working fluids such as xenon, ethane or ammonia.

The overall thermal efficiency will be maximised when the partial efficiencies are maximum. As consequence of such asseveration, effort must be put on the maximisation of partial efficiencies. In this way, the CBC working with N2, air or He is optimised by actuating on:

- Increasing the top temperature to approach the materials capacity limits.
- Transferring the maximum rejected energy to the bottoming cycle instead of being returned or suctioned by the compressor.
- Operating under a pressure ratio such that coupling between top and bottom cycles is realizable.

In the same way, for the bottoming cycle, the three following actions on the ultra-supercritical RC have been performed:

- Transferring the working fluid exhaust heat from the LP turbine exhaust to the feed pump discharge pre-heating phase by means of a regeneration stage instead of the condenser.
- Condensing the working fluid at quasi-critical pressure and temperature, which means to reject the minimum possible amount of heat to the heat sink or condenser.
- Selection of an appropriately available (organic or not) working fluid where xenon, ethane and ammonia are considered due to its properties.

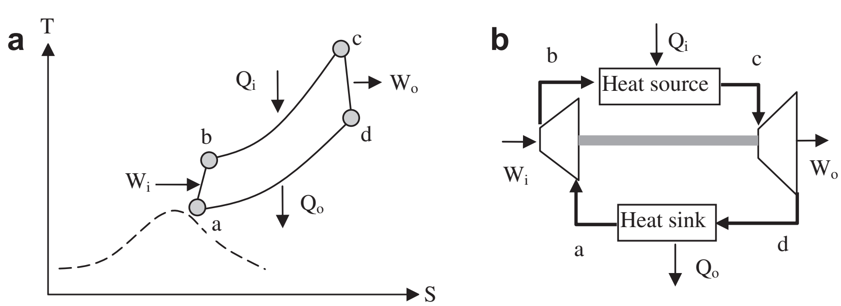

Fig. 2. Basic closed Brayton cycle (a) the T–S diagram. (b) the basic plant structure.

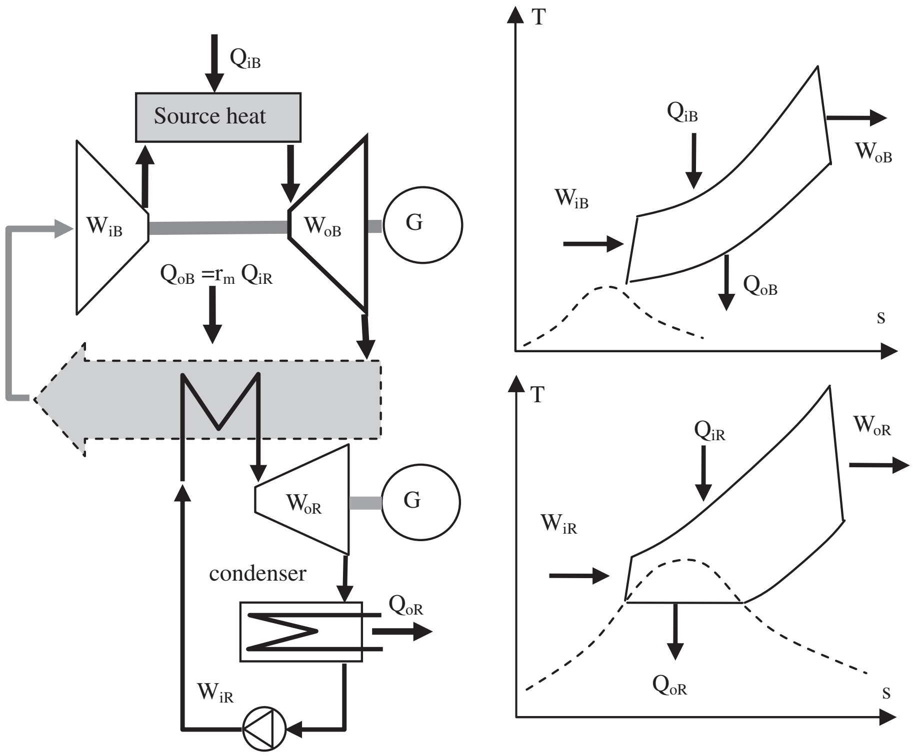

Fig. 3. Basic structure of a combined cycle.

3.1. The basic combined cycle structure

The proposed CC plant structure is shown in Fig. 4. This scheme depicts a CBC which could be powered by solar concentrated energy, methane as fossil fuel or a high temperature reactor cooled by helium. The top cycle is a CBC which works with argon, helium or hydrogen as working fluid where the efficiency depends strongly on the temperature of the working fluid at the turbine inlet and the temperature at compressor suction (as low as technically possible). The last known published advances with regard to CC plants working fluids deals with the optimal selection process of working fluids as described in [22]

This scheme depicts a CBC powered by solar concentrated energy and/or fossil fuels where with the exception of solar power, natural gas is a preferable fossil energy source among other fossil fuels. Nevertheless, helium cooled HTRs could be applied as a topping cycle since the helium low-temperature side satisfy the requirements of a power source to supply heat to the bottoming cycle. Except for the HTR as topping cycle, the top cycle is a CBC which works with N2, air, He or air as alternative working fluids where the efficiency depends strongly on the top temperature of the working fluid at the turbine input (1300 K) as well as the temperature at compressor suction (as low as technically possible). Expression (11) gives us a solution for the efficiency. Once fixed the minimum suction pressure and pressure ratio, then the compressor discharge pressure is a constant parameter as well as the discharge temperature. The exhaust temperature from the turbine is such that an acceptable energy flow rate between both cycles ensures the bottoming cycle efficiency. This temperature is restricted to a value higher by at least 50 °C over the maximum temperature of the working fluids of the bottoming cycle (ammonia, 700 K, xenon 750 K and ethane 675 K).

The bottoming cycle composed by a RC is depicted with Fig. 5. Optionally, depending on the working fluid, it can deliver more efficiency with a single turbine. For both options (single and double turbine), a regenerator is applied at the feed pump discharge. Low pressure turbine exhaust at quasi-critical pressure towards the condenser implies delivering the working fluid at a temperature higher than its critical temperature. Consequently, the amount of heat carried out with the working fluid to the condenser must be recovered by means of a heat exchanger or regenerator. As mentioned, such amount of heat is transferred to the working fluid after feed pump discharge.

In the scheme shown in Fig. 6, the heat rejected by the working fluid at the LP turbine exhaust side ($h_4 - h_{4x}$) is recovered by the working fluid at feed pump discharge as ($h_{2x} - h_2$).

Although regeneration is a conventional means to increase the RC efficiency, in this case regeneration is placed between the LP turbine and the condenser, which is not typical in RCs. This contribution is relevant to efficiency enhancement when associated to the fact of applying quasi-critical condensation in combination with the proposed (organic or not) working fluids.

The efficiency of the proposed quasi-critical condensation based RC is obtained according to (1) as the following:

The supplied specific input heat flow Q_i for a unique turbine stage is defined as

$$Q_{i1} = (h_3 - h_{2x}) \tag{4}$$

The supplied specific input heat flow Q_i for two turbine stages is defined as

$$Q_{i2} = (h_3 - h_{2x}) - (h_{3b} - h_{3a}) \tag{5}$$

The specific output heat flow Q_o is defined as

$$Q_o = (h_{4x} - h_1) \tag{6}$$

The specific net work for a unique turbine stage is defined as

$$W_1 = (h_3 - h_4) - (h_2 - h_1) \tag{7}$$

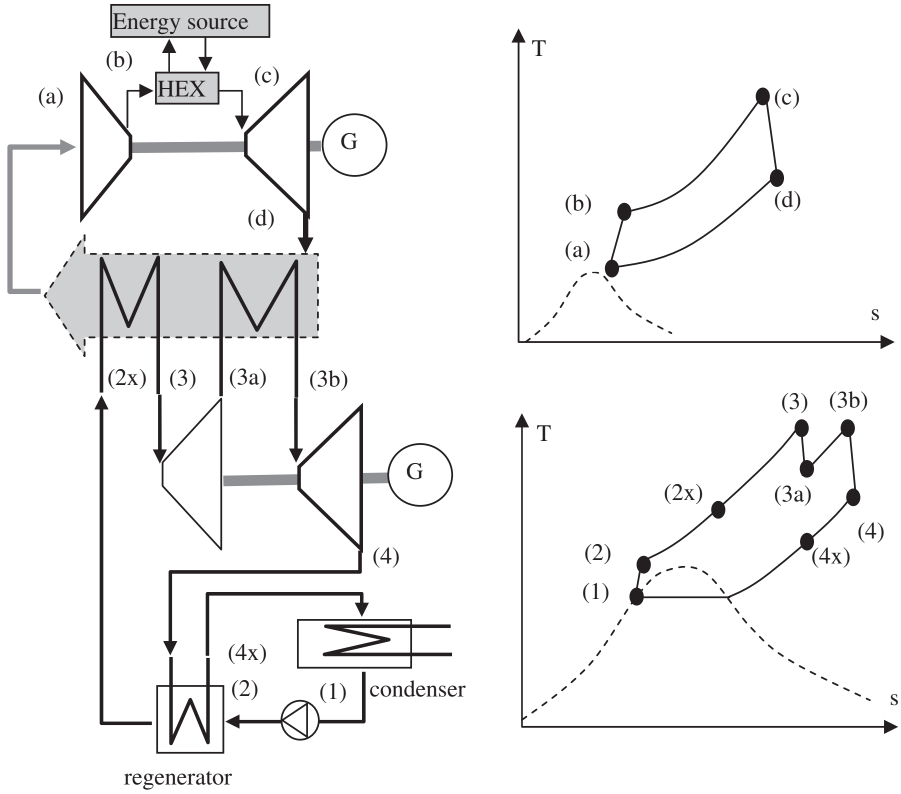

Fig. 4. The proposed combined cycle plant structure.

The specific net work for a two turbine stages is defined as

$$W_2 = (h_3 - h_{3a}) + (h_{3b} - h_4) - (h_2 - h_1) \tag{8}$$

The thermal cycle efficiency for a unique turbine stage η_R is defined as

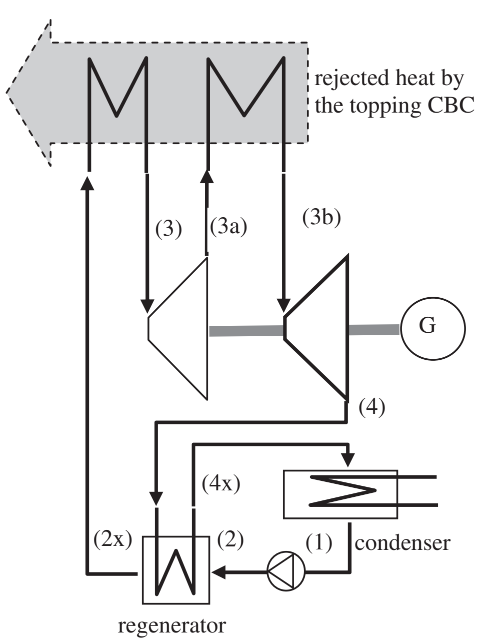

Fig. 5. Bottoming cycle with a heat recovery by means of the bottom regenerator.

$$\eta_{R1} = \frac{W_o - W_i}{Q_{i1}} = \frac{W_n}{Q_{i1}} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_{2x})} \tag{9}$$

The thermal cycle efficiency for a two turbine stages η_{R2} is defined as

$$\eta_{R2} = \frac{W_o - W_i}{Q_{i2}} = \frac{W_n}{Q_{i2}} = \frac{(h_3 - h_{3a}) + (h_{3b} - h_4) - (h_2 - h_1)}{(h_3 - h_{2x}) - (h_{3b} - h_{3a})} \tag{10}$$

According the T–S diagram shown in Fig. 3, in order to satisfy the 1st and 2nd thermodynamic laws, the following conditions must be fulfilled.

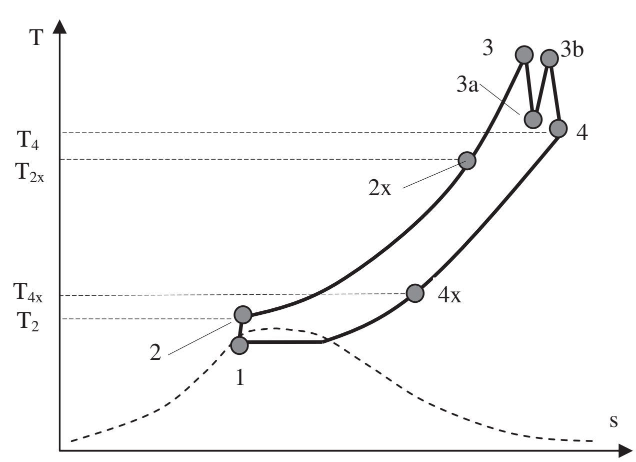

Fig. 6. The T–S diagram of a quasi-critical Rankine cycle with regeneration.

1st principle:

$$(h_4 - h_{4x}) = (h_{2x} - h_2) \tag{11}$$

2nd principle:

$$T_{4x} \geq T_2 \tag{12}$$

$$T_4 \geq T_{2x} \tag{13}$$

$$T_4 \geq T_{4x} \tag{14}$$

$$T_{2x} \geq T_2 \tag{15}$$

In an endeavour to find the maximum achievable thermal efficiency of the cycle the following considerations must be taken into account: according to expressions (9) and (10), maximum thermal cycle efficiency implies minimum Q_i, which means maximum h_{2x} and minimum h_{4x}. According to expression (7) in agreement with the information provided by Fig. 6, the cycle must satisfy the condition ($T_{4x} \geq T_2$). Consequently, the minimum value for h_{4x} (the enthalpy at temperature T_{4x}) is h_2, (the enthalpy at temperature T_2). Nevertheless, requirements for practical realisation demand a temperature difference ($T_{4x} - T_2$) of at least 20 K in order to permit an acceptable heat transfer flow. As consequence of such restriction, a compromise must exist between the aforementioned temperature difference and the heat transfer flow which definitely affects the size of the plant.

3.2. The selection of a proper available working fluid

While for the CBC the working fluid is selected according the mentioned criteria: high efficiency at high temperatures and acceptable pressure ratio and power ratio, the criterion to select a working fluid to be applied on the proposed bottoming RC and/or ORC deals mostly with its physical condensation properties. The objective for the bottoming cycle is that the selected working fluid will be condensed at quasi-critical temperature and pressure, and such temperature coincides with the ambient temperature in order to be cooled by the available cooling fluid generally at ambient temperature. Only several fluids fulfill the required characteristics and among them only the three shown in Table 1 are able to operate as working fluids at ultra-high pressure and temperatures in comparison with the conventional organic working fluids.

3.3. Concentrated solar energy based thermal source

During the last two decades, concentrated solar energy has become an objective of intensive research study in order to maximise converted solar energy flux into electric power. Three types of concentrators are available: central tower, parabolic dish and parabolic trough. In Fig. 7 a parabolic dish based concentrator is coupled to a volumetric receiver followed by a thermal power plant. This type of concentrator converts solar energy flux into useful thermal energy by means of a volumetric receiver. A heat transfer fluid (i.e., helium) is responsible for transferring the thermal solar heat to the thermal cycle source energy by means of a heat exchanger (HEX).

Table 1
The selected working fluids for the bottoming cycle.

WF	Critical point		Range of applicable temperatures (K)		Condensation temperature
	Pressure (bar)	Temperature (K)	Minimum	Maximum	Ambient (K)
Ethane	48.5	305.3	90.36	675	>290
Xenon	58.42	289.73	161.41	750	<285
Ammonia	113.3	405.4		700	>290

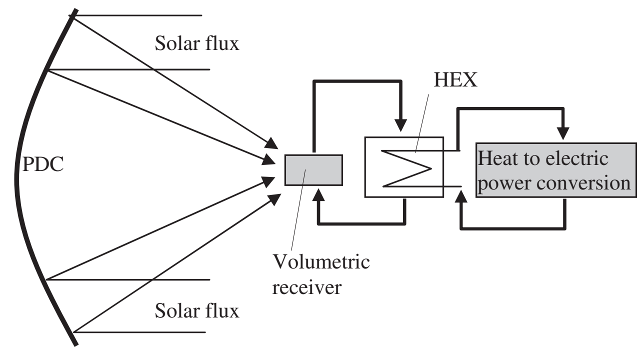

Fig. 7. Basic solar based power plant structure powered by a parabolic dish concentrator.

According to the scheme of Fig. 7, the device that is used for the conversion of concentrated solar radiation to high temperature heat is a volumetric receiver. It is designed so that concentrated solar radiation is absorbed and transferred to the used heat transfer fluid which renders an energy conversion rate defined by the volumetric receiver efficiency. Inherent unavoidable heat losses exists from the fact that the heat absorbing surface cannot be completely black, that it emits thermal radiation to the environment, because it has an elevated temperature, and that convection as well as conduction occur. Assuming that the receiver is affected by inherent heat losses including irreversibilities, the useful heat flow collected by the receiver can be estimated by applying the energy balances as

$$Qu = \dot{m} \cdot Cp \cdot (To - Ti) = A_{Ab} \cdot U \cdot (T_A - T_F) \tag{16}$$

The incident solar energy on the parabolic dish concentrator is defined as

$$Qs = A_{Ap} \cdot C \cdot E_S = A_{PD} \cdot E_S \tag{17}$$

Hence the thermal efficiency of the volumetric receiver defined as the solar energy to heat power conversion composed by the parabolic dish concentrator and a volumetric receiver is defined by the ratio of the useful heat at the heat exchanger to the incoming solar radiation on the parabolic dish concentrator

$$\begin{aligned} \eta_{VR} &= \frac{Qu}{Qs} = \frac{\dot{m}_{HTF} \cdot C_P \cdot (To - Ti)}{A_{Ap} \cdot C \cdot E_S} = \frac{A_{Ab} \cdot U \cdot (T_A - T_F)}{A_{Ap} \cdot C \cdot E_S} \\ &= \frac{\dot{m}_{HTF} \cdot C_P \cdot (To - Ti)}{A_{PD} \cdot E_S} = \frac{A_{Ab} \cdot U \cdot (T_A - T_F)}{A_{PD} \cdot E_S} \end{aligned} \tag{18}$$

The real efficiency scenario depends strongly on the receiver structure in such a way that a lot of concentrating process variables are involved. Considering achieved experience [23] with regard to the solar to heat conversion process by means of parabolic dish concentrators, it must be taken into account that

- maximum efficiency corresponds to a particular temperature for a given receiver instead of its maximum temperature.
- higher concentration ratios lead to higher efficiencies.
- obviously convection and conduction losses are of minor importance at high concentration ratios.

Fig. 8 shows the total efficiency of a high temperature solar concentrating system (parabolic dish concentrator) as the function

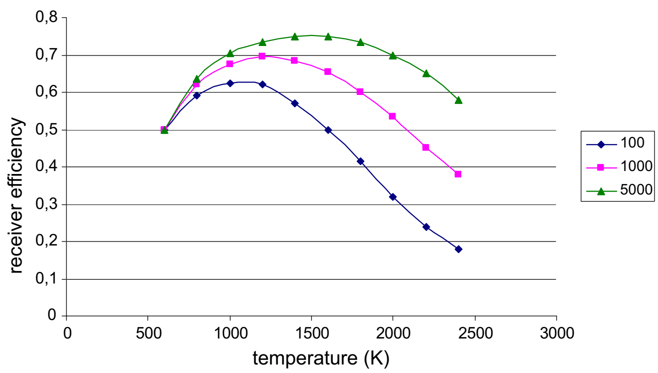

Fig. 8. The total efficiency of a receiver corresponding to a high temperature solar concentrating system.

of the upper receiver temperature for different concentration ratios and a selective characteristic of the absorber [23]. According to the information provided by this figure, once the desired concentration ratio is chosen, as well as the working absorber temperature of the receiver, then the total efficiency of the volumetric receiver is estimated. Thus, using expressions (16–18), the design top cycle temperature is proposed.

The information provided by the curves shown in Fig. 8 is being used for design purposes since once chosen a concentration factor and the top temperature of the receiver absorber thermal cycle operating conditions are proposed by applying expressions (16–18).

4. Case study: CC (CBC–RC)

The proposed case study is based on a combined cycle operating under different conditions. The fluids (N2, air, or He, for the topping CBC and xenon, ethane or ammonia for the bottoming RC) were selected as potentially working fluids for the proposed CC power plant.

The topping CBC is studied for the top temperatures of 1300 K and 1000 K operating with nitrogen, air and helium under several pressure ratios so that once given some performance criteria, the availability of the analysed results permits us to choose the most convenient operating conditions.

The topping cycle of the CC or CBC behaviour is analysed using data relative to N2, air and He from a database [8] being represented in Tables 1, 2 and 3 of appendix 1. The assumed isentropic efficiencies are 0.92 for gas turbines, 0.88 for gas compressors and 0.85 for feed pumps.

In Tables 4, 5 and 6 shown in appendix 1 the RC state points operating with xenon, ethane and ammonia as working fluids are presented. Although the RC analysis has been performed for single and double expansion turbine cycles, only the results of two (HP-LP turbine) based cycles are presented since thermal efficiency for a unique turbine cycle is significantly lower.

In Table 2 , the input specific heat flow, output specific heat flow, specific work and efficiency for the studied working fluids is shown in Table 2.

Table 2
Summary of RC operating characteristics.

Working fluid	Xe	C2H6	NH3
Q_i (kJ/kg)	121.17	623.51	2095.2
Q_o (kJ/kg)	68.67	380.6	1258.2
Efficiency	43.33	27.34	39.95
W_{nR} (kJ/kg)	52.5	242.91	837

Table 3
CC performance for a top temperature of 1300 K and nitrogen as working fluid.

r_p	5	10	20	30
N2–Xe	0.598	0.644	0.673	0.679
N2–C2H6	0.567	0.616	0.647	0.654
N2–NH3	0.574	0.623	0.653	0.660

Table 4
CC performance for a top temperature of 1300 K and helium as working fluid.

r_p	3	5	8	10
He–Xe	0.600	0.649	0.675	0.679
He–C2H6	0.570	0.621	0.650	0.655
He–NH3	0.577	0.628	0.656	0.660

Table 5
CC performance for a top temperature of 1300 K and air as working fluid.

r_p	5	10	20	30
Air–Xe	0.597	0.642	0.672	0.679
Air–C2H6	0.566	0.614	0.647	0.654
Air–NH3	0.573	0.620	0.652	0.659

The CC (CBC–RC) performance resulting from the combination of described CBC and RC respectively operating under a top temperature of 1300 K is presented in Tables 3–5 for some pressure ratios.

The CC (CBC–RC) performance resulting from the combination of described CBC and RC respectively operating under a top temperature of 1000 K is presented in Tables 6–8 for some pressure ratios.

5. Discussion of results

Since the physical properties of air and nitrogen are relatively close, as expected, by comparing the results of Tables 1 and 2, which belongs to nitrogen and air as working fluids, very similar results in terms of efficiencies are achieved. Pressure ratios of 25 operating with a top temperature of 1300 K, renders a thermal efficiency which approaches 43% in both cases. This value is relevant since such performance is achieved by satisfying the conditions to be coupled to a bottoming Rankine cycle, that is, gas turbine exhaust temperature is sufficiently high to operate a bottoming Rankine cycle with the proposed working fluids. With regard to helium as working fluid a pressure ration of 9 for a top temperature of 1300 K renders a thermal efficiency of 43%. However the specific work is much higher than the cycles of nitrogen and air.

The Rankine cycles shown in Tables 4, 5 and 6 of Appendix I deliver the performences specified in Table 2. In spite of the fact that the available temperatures for Rankine cycle implementation is relatively low, (700 K), the achieved performance is satisfactory since the amount of rejected heat is very low for ethane and xenon

Table 6
CC performance for a top temperature of 1000 K and nitrogen as working fluid.

r_p	5	10	20	30
N2–Xe	0.591	0.624	0.620	0.569
N2–C2H6	0.560	0.595	0.590	0.536
N2–NH3	0.567	0.602	0.597	0.543

Table 7
CC performance for a top temperature of 1000 K and helium as working fluid.

r_p	3	5	8
He–Xe	0.590	0.625	0.619
He–C2H6	0.558	0.596	0.590
He–NH3	0.565	0.603	0.596

Table 8
CC performance for a top temperature of 1000 K and air as working fluid.

r_p	5	10	20	30
Air–Xe	0.589	0.621	0.621	0.574
Air–C2H6	0.557	0.592	0.591	0.541
Air–NH3	0.565	0.599	0.598	0.549

due to the quasi-critical condensation condition. In the case of ammonia it is condensed at its ambient temperature and corresponding pressure according the data shown in Table 6 of Appendix 1. In any case the efficiency approaches 40%. For the case of xenon, efficiency approaches 43% which is due to the fact that the condensation temperature is chosen as 285 K to satisfy the quasi-critical point condensation condition.

Performance of individual cycles is not interesting since the cycles are designed to operate in a combined cycle structure. As shown in Tables 3–8, thermal efficiencies depend strongly on the cycle top temperatures and pressure ratios. Temperatures of 1300 K are common with the actual state of the art technology. Consequently taking into consideration the temperature of 1300 K, efficiencies of 66%–68% are achieved for the most favourable cases.

In agreement with expected results, thermal efficiency has been improved by considering the association of several specific technical contributions to the CC where N2, air, or He in the topping cycle and a suitable optional working fluid such as xenon, ethane or ammonia for the bottoming cycle have been proposed. According to the results achieved, the CC structure that delivers the highest thermal efficiency corresponds with a CBC operating with helium, nitrogen or air (for air CBC or BC) coupled to a RC for the bottoming cycle with xenon ethane or ammonia. Consequently selection criteria depends more on the plant compactness which is a consequence of the specific volume of each working fluid at CBC gas turbine exhaust as well as at the RC turbine exhaust sides, aside from the availability and cost of working fluids.

6. Conclusions

Total CC efficiency has been improved with respect to conventional power plants due to the association of the following actions to the CC composed by a topping CBC or a BC and a bottoming RC or ORC:

For the topping cycle,

- The selected working fluid due to its inherent characteristics
- The operating parameters

For the bottoming cycle,

- Inherent regeneration.
- Quasi-critical pressure and temperature condensation in the cases of xenon and ethane, which means to reject the minimum possible amount of heat to the heat sink.
- Selection of a proper organic and nonorganic working fluid.

As revealed in the results, the ideal thermal efficiency is significantly increased (66%–68%) in comparison with the conventional CCs under realizable e conditions.

Some of the drawbacks to implement the proposed techniques to increase thermal efficiency require the design of high pressure condensers and heat exchangers, and back-pressure turbines, as well as high temperature solar receivers, since the existing high temperature blade cooled gas turbines technology that is mature.

Furthermore, the use of xenon as working fluid to be applied on the bottoming cycle has an important disadvantage related with the condensation temperature which must be lower than 285 K (ambient temperature). However ethane hasn't such a disadvantage, almost rendering a similar efficiency in a CC structure.

Finally the widespread use of solar and/or nuclear potentials as power sources apart from the fact of contributing to reducing the massive use of fossil fuels is strongly associated with the reduction of ODP and GWPs.

Appendix 1

Table 1
Brayton cycle efficiency operating with nitrogen as working fluid.

r_p/T(K)	1300	1000
5	29.06	27.89
10	37.15	33.65
15	40.55	34.45
20	42.25	32.87
25	43.06	29.41
30	43.31	23.98
40	42.20	10.35

Table 2
Brayton cycle efficiency operating with air as working fluid.

r_p/T(K)	1300	1000
5	28.83	27.53
10	36.78	33.33
15	40.32	34.43
20	42.05	33.03
25	42.93	29.87
30	43.22	24.87
40		

Table 3
Brayton cycle efficiency operating with helium as working fluid.

r_p/T(K)	1300	1000
3	29.49	27.64
5	37.98	33.83
8	42.67	32.77
10	43.42	
12	42.99	

Table 4
The RC state points for xenon as working fluid.

Xe	T (K)/T'(K)	h(kJ/kg)	h'(kJ/kg)	s(kJ/kg-K)/S'	p(bar)	v(m^3/kg)
1	282	51.33	51.33	0.21845	50	0.0006
2	361.1/355	71.73	69.69	0.21845	400	0.0005
3	750	174.3	174.3	0.42303	400	0.0013
3a	542.7	142.52	139.76	0.42929/0.423	170	0.0035
3b	750	181.14	181.14	0.48794	170	0.0027
4	485/461	140.02	136.44	0.49738/0.488	50	0.0053
4x	380	120	120		50	0.0042
2x	405	86	86		400	0.0005

Table 5
The RC state points for ethane as working fluid.

C_2H_6	T(K)/T'(K)	h(kJ/kg)	h'(kJ/kg)	s(kJ/kg-K)/S'	p(bar)	v(m^3/kg)
1	300	359.8	359.8	1.4334	45	0.003
2	350/346.3	464.47	454	1.4636/1.4334	400	0.002
3	675	1521	1521	3.574	400	0.005
4	543.7/531	1197.27	1161.3	3.641/3.574	45	0.031
4x	450	942	942	3.1264	45	0.018
2x	435	719.5	719.5	2.5	400	0.004

Table 6
The RC state points for ammonia as working fluid.

NH_3	T(K)/T'(K)	h(kJ/kg)	h'(kJ/kg)	s(kJ/kg-K)/S'	p(bar)	v(m^3/kg)
1	295	445.8	445.8	1.823	10	0.0016
2	305/303.5	513.80	507	1.8456/1.823	400	0.0016
3	700	2462	2462	5.9726	400	0.0077
3a	490.35/474	2028.96	1991.3	6.0745/5.9726	60	0.0341
3b	700	2658	2658	7.1255	60	0.0557
4	521.78/502	2186.04	2145	7.2256/7.1255	10	0.2413
4x	325	1704	1734		10	0.1637
2x	397.6	948	948		400	0.0020

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