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Power plant driven by residual heat rejected by the bottoming low pressure steam turbines.

ABSTRACT (ARIAL, BOLD, 11 FONT, LEFT ALIGNED, CAPS)

The work deals with thermal engine structures undergoing load based expansion-contraction processes powered by the residual heat rejected by low pressure turbines of the bottoming steam Rankine cycles. The isobaric expansion-contraction based thermal cycles referred to in this paper, differs from the conventional Carnot based thermal cycles in that the conversion of heat to mechanical work is performed assuming a load reaction driven path function, in which as heat is being absorbed or rejected, mechanical work is simultaneously performed avoiding the conventional required quasi-entropic expansion, such as in conventional quadrilateral based Carnot engines. Such conversion engine is cascaded with the bottoming steam Rankine cycles at combined cycle power plants, including nuclear power plants.
An analysis of the ideal thermal cycle implemented by means of an isobaric expansion-contraction based cylinder is carried out and results compared with the Carnot cycle operating under the same ratio of temperatures. Into the range of its inherent low operating temperatures high thermal efficiency is achieved where hydrogen and helium have been used as working fluids due to its high specific heat capacity. The achieved results associated with a simple and compact machine envisage the way towards a new generation of thermal power plants.

Keywords: Carnot factor, Isobaric expansion, Isobaric compression, Residual thermal energy, Thermal efficiency.

1. INTRODUCTION

An important fraction of the supplied thermal energy to the Rankine cycle coming from the low pressure steam turbines of the Rankine cycle based power plants which may approach (50%-60%) is being rejected to the environment mainly as consequence of the latent heat of condensation. Now, most of the rejected heat may be converted into mechanical work at very high efficiency by applying thermal cycles whose thermal efficiency are not restricted by the Carnot factor. On the other hand, the known thermal cycles under use so far are derived from the Carnot engine, which means thermal cycles in which ideally heat is absorbed at constant temperature (top temperature) and work is delivered when temperature decreases from the top temperature to approach the bottom temperature under a quasi entropic transformation. The power cycles that obey to this model are classified in two main groups based on the nature of the working fluid: gas power cycles and vapour power cycle. The difference between the two groups is that in the first case the working fluid is gaseous and does not experiment any phase change, while for the second group, there is a liquid-vapour phase change process of the working fluid within the cycle. In Fig.1 a simple classification of actual thermal engine based cycles is depicted. Conventional combined cycles comprise a topping cycle operating with high temperature and a bottoming cycle operating with low or intermediate temperature. For power production with gas turbine based combined cycles, most of the bottoming cycles are steam based Rankine cycles, mainly due to its very attractive features such as good thermal coupling with the topping gas turbine cycle, high reliability and more than two decades of practical experience.

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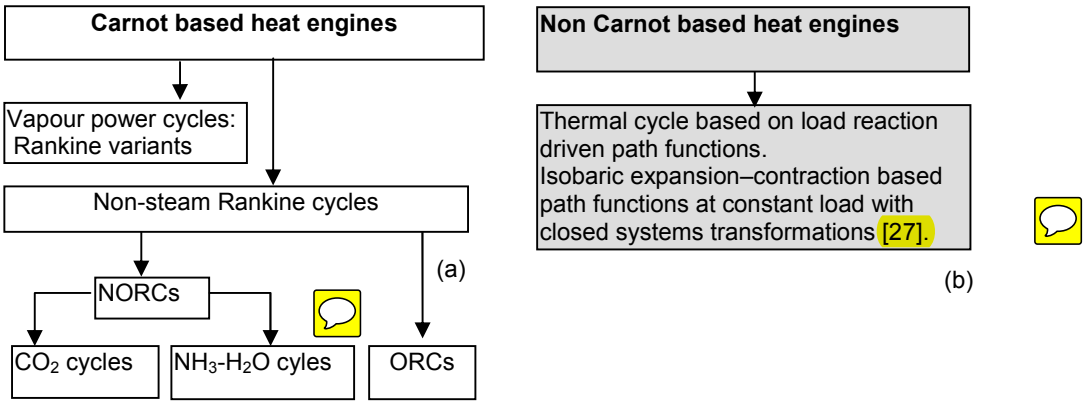

Fig. 1. Classification of Carnot and non Carnot based thermal engines suitable for applications. (a), Carnot based heat engines. (b), non Carnot based engines.

In the low temperature range, bottoming ORCs constitute another alternative, having shown good thermodynamic performance for low maximum temperature bottoming cycles, [1-3]. The interest in organic working fluids for residual heat applications with low temperature Rankine cycles (ORCs) [4-8] is an old technique that has been proposed for different applications: renewable energy and low temperature heat recovery. Furthermore, small and medium scale ORC power plants are presently commercially available Huijuan Chen et al. [9-11]. Furthermore, some of them use unconventional working fluids such dry fluids, including Hung T [12-13].

ORCs bottoming cycles in combined power plants have been proposed previously by [14], who analysed a combination of ORC fluids and cycle layouts that resulted in a global combined cycle efficiency slightly below 45.2%, also by [15] for intermediate temperature thermo-solar power plants with a carbon dioxide topping cycle, and by [16-17], and [3] for ORC's operating with micro turbine based combined cycles.

All thermodynamic cycles, including the cycle proposed in this work, are subjected to the constraints imposed by the first law with regard to the application of conservation of energy to the system. In the same way, the second law sets limits on the possible efficiency of the thermal cycles and engines determining the direction of energy flow W.R. Wagar et al. [18] for water-ammonia cycles. Under such constraints, residual or waste heat characterised by low grade heat (moderate low temperatures) can still be very useful at industrial scale. According to [17], some of the low heat temperature heat sources from waste and residual heat sources rejected by a vast range of industrial equipment can be efficiently recovered at an effective cost by applying the available state of the art technology in ORCs and NORCs. Furthermore, some of the low temperature heat sources including ocean thermal energy can be implemented efficiently under ORCs [19]. However, ORCs and NORCs, among which are those operating with CO_2, are well suited to use medium and low temperature heat sources, [20]. In this way, Table 1 gives the typical temperatures for industrial equipment characterised by:

- Waste gases from process equipment in the medium temperature range. Most of the waste heat in this temperature range comes from the exhaust of directly fired process units.
- Some heat sources in the low temperature range. In this range, it is not usually practical to extract work from the source, though steam production may not be completely excluded if there is a need for low-pressure steam. Low temperature waste heat may be also useful in a supplementary way for preheating purposes.

Table 1. Typical waste heat sources at medium and low temperature ranges.

During the last decade carbon dioxide has been applied for solar heating based plants as heat transfer fluid and particularly for low and medium temperature NORCs as working fluid. Thus, [21] worked on a thesis in which novel cycles using carbon dioxide as working fluid were proposed. The results present a thermal efficiency of the carbon dioxide higher than those rendered by common organic working fluids. In [22] an organic Rankine cycle using R123 as working fluid and a CO_2 transcritical power cycle have been compared with respect to their abilities to convert energy from low-grade waste heat to usable work. The results show that when utilizing the low-grade heat source with equal mean thermodynamic heat rejection temperature, the carbon dioxide transcritical power cycle has a slightly higher power output than the ORC. [23] conducted an experimental study on an evacuated tube solar collector using supercritical CO_2 which yields a collector efficiency approaching 60% operating with carbon dioxide. In [24] some carbon dioxide power cycles using liquid natural gas as heat sink under a compound structure yielding an efficiency of about 55.3% at 150 bar has been studied. [25] presented a research work using CO_2, ethane and R125 as possible working fluids at temperatures of between 80 and 100 ºC. The study carried out by [26] examines 10 different aromatic hydrocarbons and siloxanes as potential working fluids, where comparisons are drawn between the organic flash cycle (OFC) and an optimized basic ORC, a zeotropic Rankine cycle using a binary ammonia-water mixture, and a transcritical CO_2 cycle. The authors claimed that aromatic hydrocarbons are better suited working fluids for the ORC and OFC due to higher power output and less complex turbine designs.

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All these mentioned power cycles obey to the Carnot cycle limitations so that its thermal efficiency is restricted to the Carnot statement.

Table 1. Low temperature heat range rejected from various industrial sources and cooling systems

Types of Devices	T (ºC)	Average (ºC)
Water circulation from steam Rankine condensers	38-40	39
Furnace doors	32-55	43
Bearings	32-88	60
Welding machines	32-88	60
Injection moulding machines	32-88	60
Annealing furnaces	66-230	148
Forming dies	27-88	57
Air compressors	27-50	38
Pumps	27-88	57
Internal combustion engines	66-120	93
Air conditioning and refrigeration condensers	32-43	37
Liquid still condensers	32-88	60
Drying, baking and curing ovens	93-230	161
Hot processed liquids	32-232	132
Hot processed solids	93-232	162

Nevertheless, a very different concept of low grade heat based power plants operating with thermal cycles based on closed systems transformations are possible as described in [27]. These power plants are characterised by the fact that in the active process of its thermal cycle, two different modes of heat energy conversion to mechanical work occur:
(a) as heat energy is being absorbed under increasing specific volume, entropy and temperature undergoing a load-dependent path function, mechanical work is simultaneously performed.
(b) as heat energy is being rejected under decreasing specific volume, entropy and temperature undergoing a load-dependent path function, mechanical work is simultaneously performed.
Summarising, mechanical work is being developed while heating or cooling a working fluid. Since the cooling processes are carried at a very low cost, this singular characteristic contributes significantly to the efficiency increasing with respect to the conventional thermal engines.
The work has been organised into 5 main sections, where section is 2 is devoted to the description of the power plant, section 3 deals with the task of modelling the closed transformation based on the isobaric expansion-contraction based cylinders (IECC). In section 4 a case study is included in which the performance analysis and discussion of results have been carried out on a feasible structure of the projected plant, and finally in section 5 some relevant conclusions are highlighted.

2. DESCRIPTION OF THE PROPOSED POWER PLANT OPERATING WITH CLOSED ISOBARIC EXPANSION-CONTRACTION BASED TRANSFORMATIONS.

The core of the proposed power cycle convertor consists of isobaric expansion-contraction based cylinders (IECC) which is schematically depicted in Fig. 2. The IECC system shown in Fig. 2 uses the following operating fluids:
-the thermal working fluid (WF) confined into the cylinder, which should exhibit high specific heat capacity such as helium or hydrogen,
-the cooling fluid consisting of cool air or water at environment temperature, acting as heat rejection device transferring heat from cylinder to the environment-the heating fluid consisting of low pressure steam exhausted from the low pressure turbine, and
-the hydraulic fluid responsible for transferring hydraulic energy to the hydraulic motor-generation set.

The proposed power plant depicted schematically in Fig. 2, is composed of the following components: the IECC consisting of an actuator cylinder, coolers, heaters, I/O cooler valves, I/O heater valves, cooling water pump/air blower, reciprocating hydraulic pump equipped with inlet and outlet three way-two position valves, and a hydraulic motor-generator set.

2.1. Description of the IECC functioning

The IECC is an active part of the power plant responsible for converting the thermal energy of a gaseous working fluid into mechanical work. According to the structure of the IECC, two general modes of converting thermal energy to mechanical work are feasible:
- Conversion of thermal energy to mechanical work by cooling a gaseous working fluid and,
- Conversion of thermal energy to mechanical work by heating a gaseous working fluid.

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The processes of cooling and heating a working fluid, which are responsible for converting heat to mechanical work, are depicted in Fig. 3.

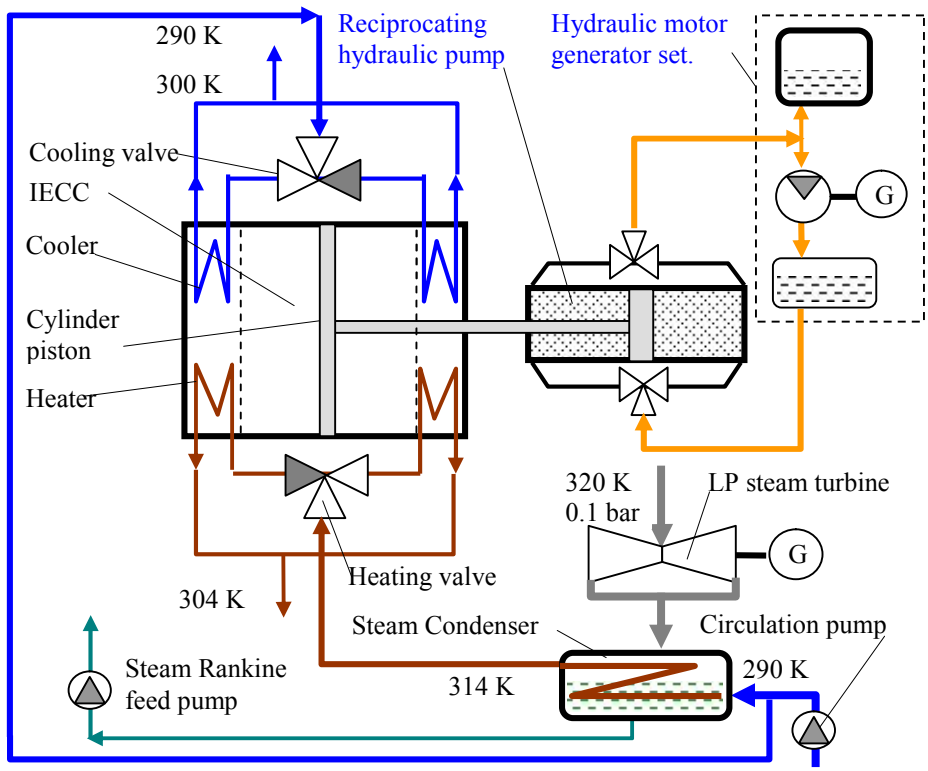

Fig. 2. Structure of the bottoming power plant using an IECC as power convertor with water as heat transfer fluid.

The process of converting partially the thermal energy contained into a gaseous working fluid to mechanical work can be carried out by rejecting heat to the environment for which the condition that the cooling environment temperature is colder than the temperature of the working fluid must be met.

To show such paradigm, a double effect cylinder equipped with two coolers and two heaters is depicted in Fig. 3(a). The closed thermodynamic transformations carried out during the cooling process of the cylinder chamber X is referred to the T-s diagram shown in the Fig. 3(b), where the state changes (4)-(5)-(6)-(1) are performed while cooling the cylinder chamber X.

In the same way, the process of converting partially the thermal energy contained into a gaseous working fluid to mechanical work can be carried out by absorbing heat from a heat source for which the condition that the heating environment temperature is hotter than the temperature of the working fluid must be met. Thus, the closed thermodynamic transformation carried out during the heating process of the cylinder chamber X is referred to the T-s diagram shown in the Fig. 3(c), where the state changes (1)-(2)-(3)-(4) shown in Fig. 3(d) are performed.

Performing mechanical work by cooling a fluid

Considering the piston located at position Y and the two position three way valves positioned so that cooler X and heater Y are active, with the rest of heat exchangers inactive, the energy balance for the cooling transformation yields

$$q_0 = \Delta u_{65} - W_{65} = u_6 - u_5 - p_5 \cdot (v_6 - v_5) \qquad (1)$$

where the expression $q_0 = \Delta u_{65} - W_{56} = Cv_{65}(T_6 - T_5) - R(T_6 - T_5)$ has been assumed. Therefore the mechanical work delivered during the cooling process is defined as

$$W_{56} = \int_{V5}^{V6} p \cdot dv = p_5(V_6 - V_5) = p_6(V_6 - V_5) = \Delta u_{65} - q_o \qquad (2)$$

From equations (1) and (2) follows that mechanical work is being developed by rejecting heat as qo.

Performing mechanical work by heating a fluid

Considering the piston located at position X and the two position three way valves positioned so that the heater X and the cooler Y are active, with the rest of heat exchangers inactive, the energy balance for the heating transformation yields

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$$q_i = \Delta u_{23} + W_{23} = u_3 - u_2 - p_2 \cdot (v_3 - v_2) \quad (3)$$

where the relation $q_i = \Delta u_{32} + W_{23} = Cv_{23}(T_3 - T_2) + R(T_3 - T_2) = Cp_2(T_3 - T_2)$ has been assumed. As consequence, the mechanical work delivered while heating process is defined as

$$W_{23} = \int_{V2}^{V3} p \cdot dv = p_2(V_3 - V_2) = p_3(V_3 - V_2) = q_i - \Delta u_{23} \quad (4)$$

From equations (3) and (4) follows that some mechanical work is being delivered by absorbing heat as qi.

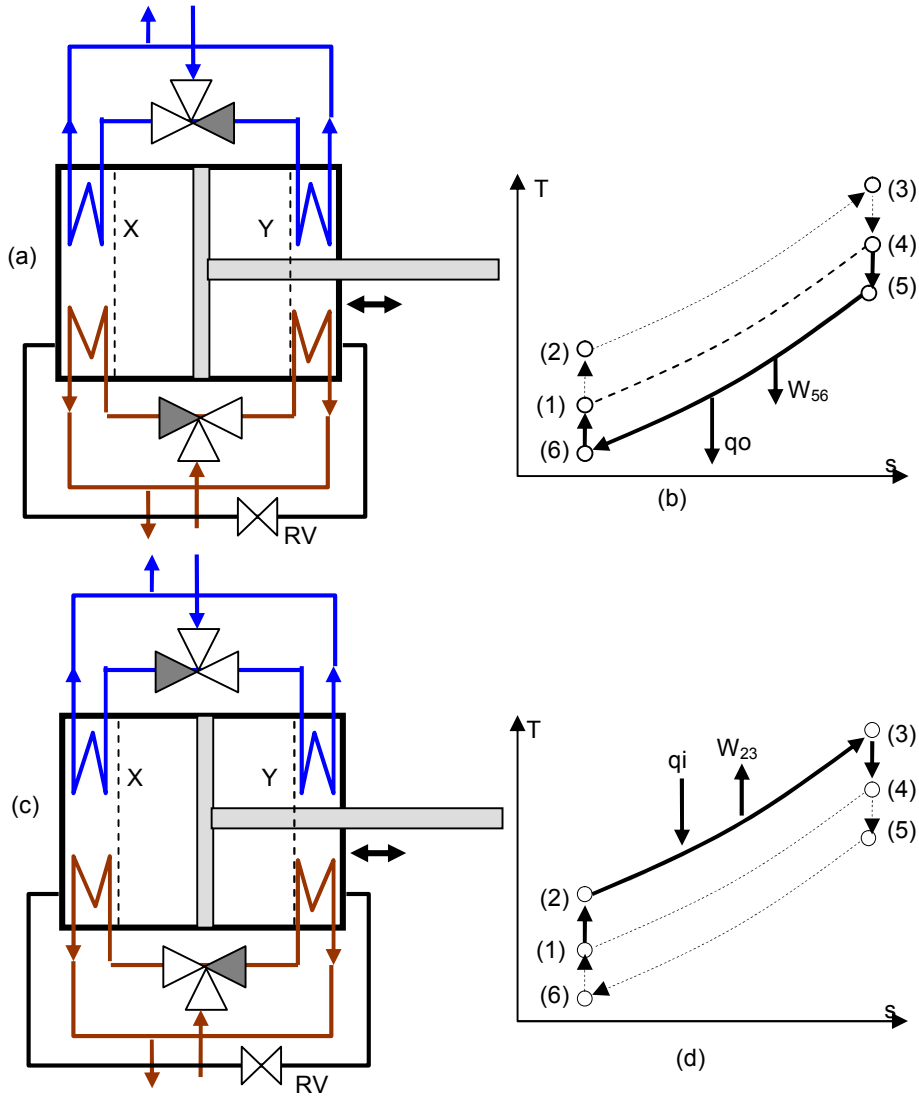

Fig. 3. The simultaneous processes of isobaric cooling and heating while developing mechanical work.

2.2. Description of the bottoming thermal cycle

The structure of the thermal cycle composed by IECCs (isobaric expansion-contraction based cylinders) which exhibits certain similarity with the plant structure described in [9] is shown in Fig. 2 and Fig. 4 respectively. The cycle functioning consists of four main operating steps described according to the plant structure depicted in Fig. 4 (a) and the T-s diagram of the Fig. 4 (b). The status of the valves is described in Table 2 for non regenerative and regenerative cycles.

The non regenerative cycle is composed by the following steps according to the information depicted in Fig. 4(a) and Fig. 4(b):

-step (6)-(1)-(2)-(3) where hot WF expands at constant pressure p_3=p_2 into the side (X) of the cylinder coming through the valve (A) at high temperature, and simultaneously, cooled WF is being compressed at constant pressure p_5= p_6 into the side (Y), leaving the cylinder through the valve (D) at low temperature.

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-step (3)-(4)-(5)-(6) where hot WF expands at constant pressure $p_2 = p_3$ into the cylinder through the valve (B), and simultaneously cooled WF is being compressed at constant pressure $p_5 = p_6$ into the side (X) leaving the cylinder through the valve (C), completing a cycle.

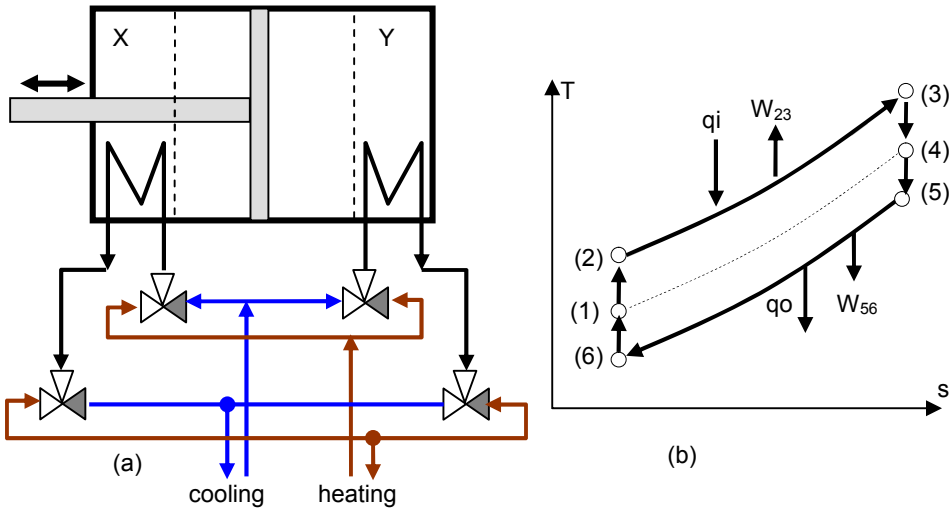

Fig. 4. The process of heating and cooling a fluid at constant pressure undergoing closed transformations: (a), the engine structure showing the heat exchangers actuating alternatively as heater and cooler. (b), the corresponding non regenerative T-s diagram.

Table 2. The status of the inlet, outlet and regeneration valves during the cycle evolution.

Cycle	**Non regenerative cycle**	
Process	(3)-(4)-(1)	(1)-(2)-(3)
CV-X	Open	Closed
CV-Y	Closed	Open
HV-X	Closed	Open
HV-Y	Open	Closed
RV	--------	-------

3. MODELLING THE IECC.

This section introduces a class of thermal engines characterised by its ability to develop mechanical work simultaneously during the heat absorbing and heat rejection processes. In order to show the behaviour of a generic IECC to which the Carnot statement does not apply, the case of a double acting cylinder designed to develop mechanical work under a variable load-based path function (isobaric path function at constant load) along its active strokes is described.

3. 1. The non regenerative IECC

In order to apply a technique on closed system based transformations in which the cooling phase (4)-(1) responsible for heat rejection is carried out by performing simultaneously useful mechanical work, the thermal cycle depicted in Fig. 4(a) and 4(b) is analysed under ideal assumptions.

The supplied heat from an external power source is

$$q_i = \Delta u_{32} + W_{23} = u_3 - u_2 + p_2 \cdot (v_3 - v_2) = Cv(T_3 - T_2) - R(T_3 - T_2) = Cp_2(T_3 - T_2) \qquad (5)$$

The rejected heat to the heat sink is

$$q_0 = \Delta u_{41} - W_{41} = u_4 - u_1 - p_1 \cdot (v_4 - v_1) = Cv(T_4 - T_1) - R(T_4 - T_1) = \\ = (2Cv_4 - Cp_4)(T_4 - T_1) \qquad (6)$$

The specific work is then given as

$$Wn = q_i - q_o = Cp_2(T_3 - T_2) - (2Cv_4 - Cp_4)(T_4 - T_1) \qquad (7)$$

Thus the ideal thermal efficiency is

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$$\eta = \frac{q_i - q_o}{q_i} = \frac{Cp_2(T_3 - T_2) - (2Cv_4 - Cp_4)(T_4 - T_1)}{Cp_2(T_3 - T_2)} = \\ = 1 - \frac{(2Cv_4 - Cp_4)(T_4 - T_1))}{Cp_2(T_3 - T_2)} \quad (8)$$

Taking into account that the term $\frac{(T_4 - T_1)}{(T_3 - T_2)}$ in (8) is approaching almost the unity, in practical applications follows that equation (8) tends to

$$\eta = \frac{q_i - q_o}{q_i} = \frac{Cp_2(T_3 - T_2) - (2Cv_4 - Cp_4)(T_4 - T_1)}{Cp_2(T_3 - T_2)} \approx 1 - \frac{(2Cv_4 - Cp_4)}{Cp_2} \quad (9)$$

which means that the cycle temperatures exerts little influence on the ideal thermal efficiency.

4. THE NON REGENERATIVE CASE STUDY

Cycle modelling is carried out with data extracted from [28] (REFPROP) and the tool Engineering Equation Solver (EES). The REFPROP consists of an up-to date data base used for property calculations and is available in commercial software packages whose library makes use of Helmholtz fundamental equation correlations to determine fluid properties from the highest-accuracy published data available in literature. Furthermore, the database can be linked with software based tools such Visual Basic, Fortran, C, and even spread sheets to help the automation of data processing. In the proposed case study the data necessary to carry out the performance analysis is presented in the Table A1 shown in the appendix A.

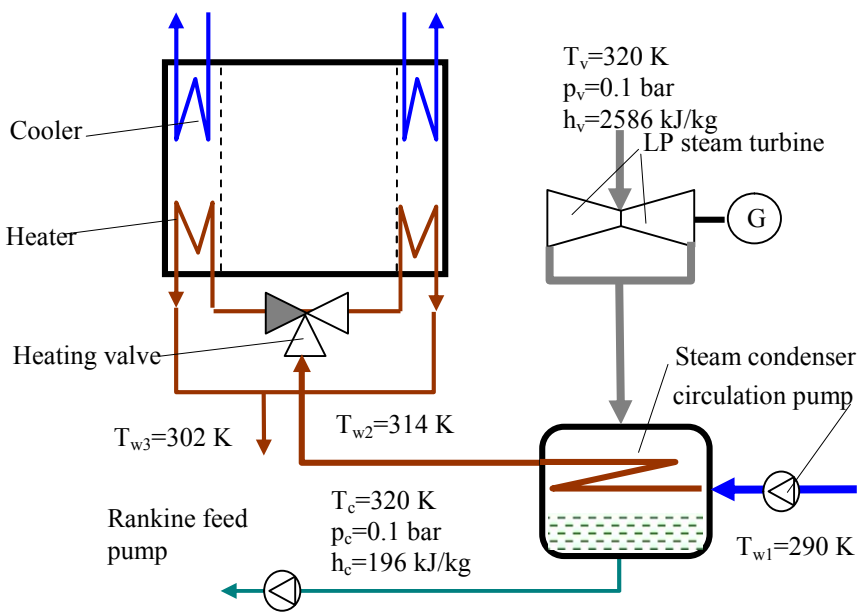

Fig. 5. Structure of the bottoming power plant of the case study using an IECC as power convertor with water as heat transfer fluid.

For the analysis of the cycle with hydrogen and helium as optional working fluids, several tests have been carried out to study the cycle performance and behaviour into a range of top temperatures of (300-312K). The plant structure chosen for the case study is shown in Fig. 5. A heat carrier fluid (water) is used due to its high specific heat capacity, absorbing the condensation latent heat being transferred to the IECC. The same heat transfer fluid is also used to extract the heat from the IECC, as the heat sink at environment temperature.

The total amount of heat supplied to the IECC (q_c), cannot be completely absorbed by the IECC, so that an important amount of this supplied heat must be rejected. As consequence it is interesting to know the amount of heat (q_i) absorbed by the IECC in order to be able to estimate the power of the IECC as function of the net efficiency.

The amount of specific heat exchanged by the condenser with reference to the Fig. 5 is defined as

$$q_c = h_v - h_c = \dot{m}_w \cdot C_w \cdot (T_{w2} - T_{w1}) \quad (10)$$

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From (10) the mass flow rate of cooling water through the condenser and the IECC, yields

$$\dot{m}_w = \frac{q_c}{C_w \cdot (T_{w2} - T_{w1})} = \frac{h_v - h_c}{C_w \cdot (T_{w2} - T_{w1})} \quad (11)$$

and the amount of heat (q_i) absorbed by the IECC is then expressed as

$$q_i = \dot{m}_w \cdot C_w \cdot (T_{w2} - T_{w3}) \quad (12)$$

Consequently, the ratio of the heat rate absorbed by the IECC to the heat rate supplied by the condenser from the low pressure steam Rankine turbine is defined as

$$\frac{q_i}{q_c} = \frac{\dot{m}_w \cdot C_w \cdot (T_{w2} - T_{w3})}{\dot{m}_w \cdot C_w \cdot (T_{w2} - T_{w1})} = \frac{T_{w2} - T_{w3}}{T_{w2} - T_{w1}} \approx 0.5 \quad (13)$$

Equation (13) suggests us that approximately only half of the amount of rejected heat by the low pressure steam condenser can be absorbed by the IECC.

4.1. Results and discussion

The results of the study achieved using the EES yield the values depicted in Table 2 and Fig 5. The data used in the case study for the analysis of the thermal cycle operating with hydrogen and helium as optional working fluids is taken from reference [28]. Cycle computation for the non regenerative IECC is referred to the T-s diagram of Fig. 4(a) and 4(b). The values of the state variables corresponding to each cycle state point and for the chosen working fluids are shown in appendix A, where for every working fluid a range of top temperatures from 300 K to 312 K has been considered. The cooling air or water temperatures are assumed as of 288 K.

According to the results shown in Table 2, as the top temperatures increases from 300 K to 312 K it is observed that the ideal thermal efficiencies remain almost constant approaching 58.5 % for hydrogen and 80.5 % for helium.

Table 2. The performance results showing the ideal thermal efficiency and specific work versus the top cycle temperatures for a non regenerative IECC operating with hydrogen and helium as working fluids with a pressure ratio of 1.1, for a heat sink temperature of 288 K and top temperature of 312 K, with heat energy from the exhausted steam of the low pressure steam turbine of a bottoming steam Rankine power plant.

T(K)	300	303	306	309	312
CF(%)	4.00	4.95	5.88	6.79	7.69
		H_2			
η (%)	58.19	58.7	58.74	58.95	58.87
W(kJ/kg)	32.57	83.42	126.66	109.15	134.88
		He			
η (%)	80.25	80.55	80.66	80.74	80.67
W(kJ/kg)	3.33	15.9	28.48	41.1	53.62

However, since the most interesting data concerns to the net efficiency, it is convenient to take into consideration the plants studied in [19] and [27] a ratio of the net to the ideal efficacies of 0.63 has been assumed. This value for the overall efficiency has been achieved on the basis of previous experimental works [19]. Thus, in [19] and [27] it is reported a thermal efficiency of η[27] = 1.9 % at a top temperature of 299 K, while the overall efficiency approaches η_N[27]= 1.2 %. This means that the overall losses due to internal and external irreversibilities in the plant described in [19] and [27] approaches a net efficiency (η_N) given as η_N[27]/η[27]= (1.2/1.9)= 0.63.

Consequently, the net efficiency for the non regenerative IECC can be approached as

$$\eta_N = \frac{\eta_N[27]}{\eta[27]} \cdot \eta \quad (10)$$

Assuming the losses for the IECC similar to the losses of the plant described in [27], because of its structural similarities with respect to its cycle and heat exchangers, the overall efficiency for a non regenerative IECC can be estimated as

$$\eta_N = \frac{\eta_N[27]}{\eta[27]} \cdot \eta = \frac{1.2}{1.9} \cdot \eta = 0.63 \cdot \eta \quad (11)$$

Table 3. The performance results showing the net efficiency (η_N) and specific work (W) versus the top cycle temperatures for a non regenerative IECC operating with hydrogen and helium as working fluids with a pressure ratio of 1.1, for a heat sink temperature of 288 K and top temperature of 312 K, with heat energy from the exhausted steam of the low pressure steam turbine of a bottoming steam Rankine power plant.

T(K)	300	303	306	309	312
CF(%)	4.00	4.95	5.88	6.79	7.69

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		H_2			
η_N (%)	36.65	37.0	37.0	37.1	37.2
		He			
η_N (%)	50.55	50.5	50.6	50.74	50.6

Comparing the results shown in Table 2 (ideal thermal efficiency) with the results shown in Table 3 (net efficiency) follows that an important decrease in the useful energy conversion efficiency is observed.
It is observed also that while the thermal efficiency exhibits almost no temperature dependence, specific work depends strongly on the top temperature according to the results shown in Table 2.
Considering the heat rejected by the steam condenser then, according to the equation (3) follows that the overall efficiency is reduced 50% with respect to the efficiency reported in Table 3, as shown in Table 4, where the overall efficiency is estimated in about 18.5 % for hydrogen and in 25.3 % for helium as working fluids. It is also observed that Carnot efficiency is violated without violating first and second laws.

Table 4. The performance results showing the net efficiency with respect to the amount of available heat rejected by the condenser versus the top cycle temperatures for a non regenerative IECC operating with hydrogen and helium as working fluids with a pressure ratio of 1.1, for a heat sink temperature of 288 K and top temperature of 312 K, with heat energy from the exhausted steam of the low pressure steam turbine of a bottoming steam Rankine power plant.

T(K)	300	303	306	309	312
CF(%)	4.00	4.95	5.88	6.79	7.69
		H_2			
η_N (%)	18.3	18.5	18.5	18.55	18.6
		He			
η_N (%)	25.27	25.27	25.3	25.32	25.3

5. CONCLUSIONS

An important fraction of the supplied thermal energy to the Rankine cycle coming from the low pressure steam turbines of the Rankine cycle based power plants which may approach (55%-65%) is being rejected to the environment mainly as consequence of the latent heat of condensation. In order to mitigate such losses a preliminary design study of a bottoming power plant driven by the residual heat rejected by the steam condenser of the bottoming Rankine cycles has been proposed.
This plant has been implemented and analysed using the IECC characterised by its ability to delivering mechanical work while rejecting and absorbing heat under load reaction based path functions, where the IECC is operating optionally with hydrogen or helium as working fluids. According to the characteristics of the studied IECC, expressions for the ideal thermal efficiency have been achieved. The performance results have been compared with the results obtained for the Carnot cycle operating with the same range and ratio of temperatures, and compared also with that of reference [27].
The most important conclusion is related to the IECC thermal efficiency which largely exceeds the Carnot efficiency under appropriated operating conditions. The reason obeys to the fact that in an IECC, the following conditions are met:

- the IECC performs mechanical wok by rejecting heat to the environment, and
- avoids the conventional quasi-isentropic expansion process in the IECC, so that the degradation of heat due to the isentropic efficiency is neglected.

Consequently, the main reasons for the efficiency enhancement with respect to conventional thermal cycles operating under Carnot based cycles (conventional ORCs) are due to the association of the following contributions:

-The cycle absorbs heat and simultaneously converts a fraction of the absorbed heat into mechanical work avoiding the conventional quasi-isentropic expansions which contributes on the heat degradation.
-Simultaneously, the IECC performs mechanical wok by rejecting heat to the environment,
-The working fluids selected contributes the thermal efficiency due to its inherent thermodynamic characteristics and behaviour.

On the basis of the feasible structure proposed for the IECC, a new strategy to avoid rejecting so much useful heat to the environment considered as the heat sink is avoided. Consequently, the estimated net efficiency is accordingly high. Furthermore, the widespread use of steam Rankine condensers as potential power sources as well as the fact of increasing the thermal efficiency, apart from the fact of contributing on reducing the massive use of fossil fuels and consequently the GWP, including contaminant fossil fuels based emissions, supposes a relevant fact that needs to be considered and experimentally researched.

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Nomenclature		**acronyms**	
γ	adiabatic expansion coefficient	CF	Carnot factor, Carnot efficiency
η	thermal efficiency	GWP	Global warming potential
η_N	net or overall efficiency	NORC	Non-Organic Rankine cycles
Cp	const. press. specific heat (kJ/kg-K)	ORC	Organic Rankine Cycle
Cv	const. press. specific volume (kJ/kg-K)	WF	Thermal working fluid
p	pressure (bar)	IECC	isobaric expansion-contraction cylinder
T	temperature (K)		
h	specific enthalpy (kJ/kg)		
s	specific entropy (kJ/kg-K)		
q_i	net specific input heat (kJ/kg)		
q_o	specific rejected heat (kJ/kg)		
W_N	net output specific work (kJ/kg)		
T_v	exhausted steam temperature (K)		
p_v	exhausted steam pressure (bar)		
h_v	exhausted steam enthalpy (kJ/kg)		
T_c	condensate steam temperature (K)		
p_c	condensate steam pressure (bar)		
h_c	condensate steam enthalpy (kJ/kg)		
T_{w1}	water inlet temp. to the condenser (K)		
T_{w2}	water inlet temp. to the IECC (K)		
T_{w3}	water outlet temp. from the IECC (K)		
q_c	specific heat exchanged in the condenser (kJ/kg)		
C_w	const. press. specific heat of the cooling water (kJ/kg-K)		

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APPENDIX A. THE DATA USED ON THE CASE STUDY

Table A1. The data to apply on the performance analysis for hydrogen and helium as working fluids operating with bottom and top pressures of 10 and 11 bar and bottom temperatures of 288 (K).

point	T(K)	h(kJ/kg)	u(kJ/kg)	s(kJ/kg.K)	p(bar)	v(m^3/kg)
			H_2			
1	288	3790.30	2595.40	43.43	10	1.19490
2	296.09	3927.01	2677.40	43.43	11	1.11740
3	312	4134.80	2839.90	44.18	11	1.17720
4	303.54	4012.80	2753.60	44.18	10	1.25920
3	309	4091.70	2809.20	44.04	11	1.16590
4	300.59	3970.50	2723.50	44.04	10	1.24700
3	306	4048.60	2778.50	43.90	11	1.15460
4	297.71	3929.20	2694.20	43.90	10	1.23510
3	303	4005.60	2747.90	43.76	11	1.14330
4	300.79	3887.50	2664.40	43.76	10	1.22300
3	300	3962.50	2717.20	43.62	11	1.13210
4	291.9	3846.10	2635.00	43.62	10	1.21110
3	298	3933.90	2696.80	43.52	11	1.12460
4	289.87	3817.00	2614.40	43.52	10	1.20270
			He			
1	288	1504.10	902.95	23.03	10	0.60110
2	299.2	1572.81	937.89	23.03	11	0.56785
3	312	1629.00	977.78	23.25	11	0.59201
4	300.36	1568.20	941.47	23.25	10	0.62677
3	309	1613.40	968.43	23.20	11	0.58635
4	297.43	1553.00	932.32	23.20	10	0.62067
3	306	1597.80	959.08	23.15	11	0.58069
4	300.58	1538.20	923.44	23.15	10	0.61476
3	303	1582.30	949.73	23.09	11	0.57503
4	291.7	1523.20	914.47	23.10	10	0.60878
3	300	1566.70	940.38	23.05	11	0.56936
4	288.79	1508.20	905.41	23.05	10	0.60274