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Power Plant Driven by Residual Heat Rejected by the Bottoming Low Pressure Steam Turbines

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Authors' contributions

This work was carried out in collaboration between all authors. Author RFG designed the study, wrote the protocol, the first draft of the manuscript and managed literature searches. Authors BFS and CFS managed the analyses of the study and literature searches. All authors read and approved the final manuscript.

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ABSTRACT

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 at constant load referred to in this paper, is characterised by its high thermal efficiency at low temperatures, since such thermal cycle doesn't obey the Carnot statement. Such bottoming energy convertor must be implemented in cascade with the low pressure turbine of the steam Rankine cycles at combined cycle power plants, including nuclear power plants.
An analysis of the ideal isobaric expansion-contraction based thermal cycle is carried out and results are compared with the Carnot cycle operating under the same ratio of temperatures. Hydrogen and helium have been chosen as working fluids due to its high specific heat capacity and

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thermal efficiency. The satisfactory results obtained from a simple and compact installation envisage the way towards a new generation of thermal power plants.

Keywords: Carnot factor; isobaric expansion; isobaric compression; residual thermal energy; thermal efficiency.

NOMENCLATURE

γ: adiabatic expansion coefficient; η: thermal efficiency; η_N: net or overall efficiency; Cp: const. press. specific heat (kJ/kg-K); Cv: const. vol. specific heat(kJ/kg-K); p: pressure (bar); T: temperature (K); h: specific enthalpy (kJ/kg); s: specific entropy (kJ/kg-K); q_i: net specific input heat (kJ/kg); $\dot{q}_n$: net input heat flow absorbed by the IECC (kW); 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), $\dot{q}_v$: heat flow exchanged in the condenser by the exhausted steam (kW); $\dot{q}_w$: heat flow exchanged in the condenser by the cooling water (kW); C_w: const. press. specific heat of the cooling water (kJ/kg-K).

ACRONYMS

CF: Carnot factor, Carnot efficiency; NORC: Non-Organic Rankine cycles; ORC: Organic Rankine Cycle; WF: Thermal working fluid; IECC: isobaric expansion-contraction cylinder; GWP: Global warming potential

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, according to [1] and [2], 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 this model are classified into 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 undergo 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.

Conventionally, an important technique to improve the overall efficiency of thermal power plants consists of using efficiently the residual heat and/or using the heat rejected by some components of the plant. Thus, in the low temperature range, conventional bottoming ORCs constitute an interesting alternative, having exhibited acceptable thermodynamic performance [3-5]. The organic working fluids for residual heat applications with low temperature Rankine cycles (ORCs) [6-10], has been proposed for different thermodynamic applications: Renewable thermal energy and low grade residual heat from industrial uses. Furthermore, small and medium scale ORC power plants are presently commercially available [11-13]. Furthermore, some of them use unconventional working fluids such as dry fluids [14,15].

ORCs bottoming cycles in combined power plants have been proposed [16]. The study analysed a combination of ORC fluids and cycle layouts that resulted in a global combined cycle efficiency slightly below 45.2%. Also [17] examined intermediate temperature thermo-solar power plants with a carbon dioxide topping cycle, [18,19,5] assessed 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 [20] 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 [19], 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 [21]. However, ORCs and NORCs, among which are those operating with CO_2, are well suited to use medium and low temperature heat sources [22]. In this way, Table 1 gives the typical temperatures rejected by some industrial equipment [23] 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. Although in this range it is not practical to extract work from the source because of the low thermal efficiency steam Rankine cycles can provide low temperature heat that can be effectively used on bottoming energy convertors with ORCs. Low temperature waste heat may also be useful in a supplementary way for preheating purposes.

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, [24] proposed novel cycles using carbon dioxide as working fluid. The results showed a thermal efficiency of the carbon dioxide higher than those rendered by common organic working fluids. In [25] an organic Rankine cycle using R123 as working fluid and CO_2 transcritical power cycle were 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 relatively higher power output than the ORC. [26] 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 the study by [27] carbon dioxide power cycle using liquid natural gas as heat sink under a compound structure yielding an efficiency of about 55.3% at 150 bar was studied. [28] Presented a research work using CO_2, ethane and R125 as possible working fluids at temperatures of between 80 and 100ºC. The study carried out by [29] examined 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 researchers claimed that aromatic hydrocarbons are better working fluids for the ORC and OFC due to higher power output and less complex turbine designs.

Table 1. Low temperature heat range rejected from various industrial sources and cooling systems (adapted from [23])

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

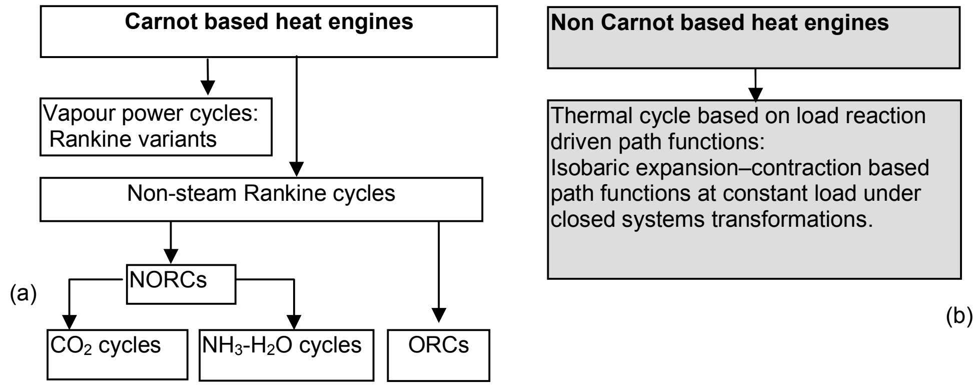

Fig. 1. Classification of Carnot and non Carnot based thermal engines suitable for applications. (a), Carnot based heat engines. (b), non Carnot based engines according to [1] and [2]

All these mentioned power cycles obey the Carnot cycle limitations and that their thermal efficiencies are restricted to the Carnot statement.

The prime objectives of this study are: (i) To demonstrate possibility of operating thermal power cycles based on low grade heat and (ii) To apply residual heat in conversion of thermal energy to mechanical energy in thermal power cycles.

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

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 in a 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.

As described in [2], the proposed power plant depicted schematically in Fig. 2, comprises 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. Furthermore, the proposed plant includes also a heat source consisting of a LP steam Rankine turbine condenser.

2.1 Description of the IECC Functioning

According to [2], 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.

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

The process of converting partially the thermal energy contained in 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 mustbe met.

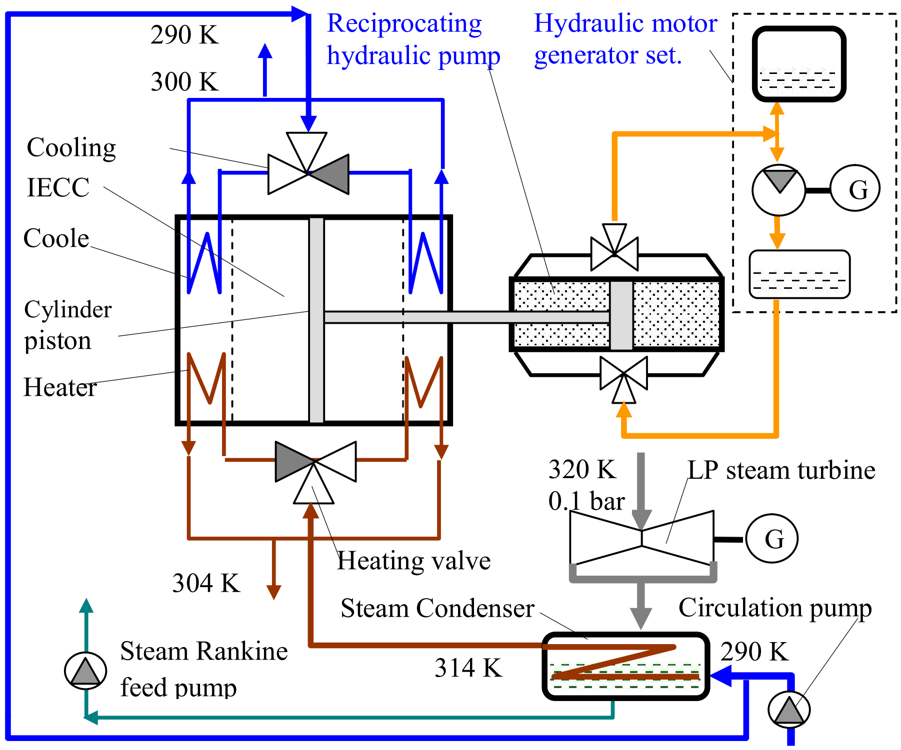

Fig. 2. Structure of the bottoming power plant using an IECC as power convertor with water as heat transfer fluid

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 in 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.

2.1.1 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

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

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

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

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

2.1.2 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

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

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.

Hence, the mechanical work delivered while heating is given 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) it follows that some mechanical work is being delivered by absorbing heat as qi.

2.2 Description of the Non Regenerative Bottoming Thermal Cycle

As described in [2], the structure of the thermal cycle composed by IECCs (isobaric expansion-contraction based cylinders), which exhibits certain similarity also with the plant structure described in [21] 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) according [1] and [2]. The status of the valves is described in Table 2 for non regenerative cycle.

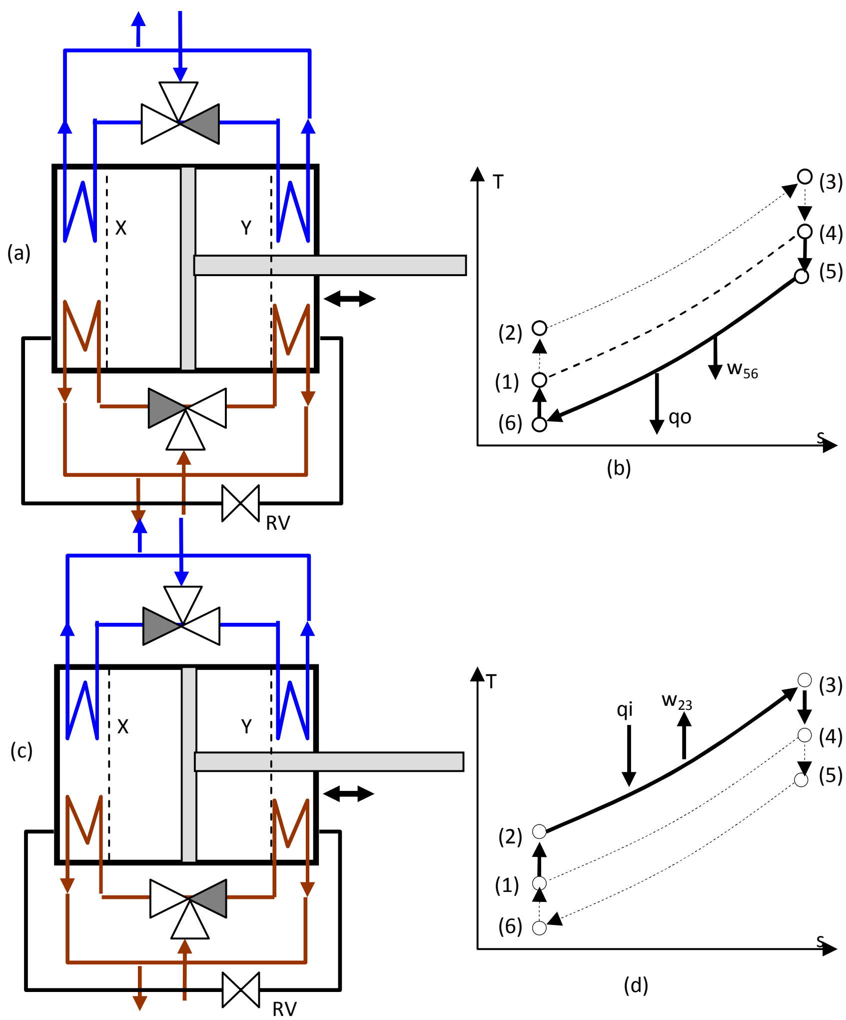

Fig. 3. The simultaneous processes of isobaric cooling and heating while developing mechanical work

- Step (4)-(1)-(2)-(3) where hot WF expands at constant pressure $p_3=p_2$ into the side (X) of the cylinder coming through the heating circuit (HC) at high temperature, and simultaneously, cooled WF is being compressed at constant pressure $p_4= p_1$ into the side (Y), leaving the cylinder through the cooling circuit (CC) at low temperature.
- Step (3)-(4)-(1)-(2) where hot WF expands at constant pressure $p_2 = p_3$ into the cylinder through the heating circuit (HC), and simultaneously cooled WF is being compressed at constant pressure $p_4= p_1$ into the side (X) leaving the cylinder through the cooling circuit (CC), completing a cycle.

Table 2. The status of the inlet and outlet cooling and heating valves during the cycle evolution

Cycle	Non regenerative cycle	
Process	(3)-(4)-(1)	(1)-(2)-(3)
CC-X	Open	Closed
CC-Y	Closed	Open
HC-X	Closed	Open
HC-Y	Open	Closed

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.

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 cycles depicted in Figs. 4(a) and 4(b) is analysed under ideal assumptions.

The supplied heat from an external power source is:

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

The rejected heat to the heat sink is:

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

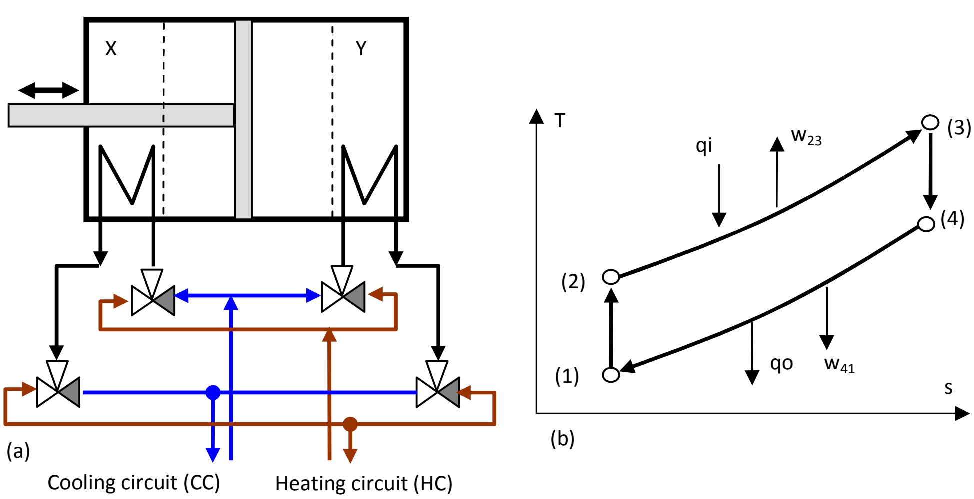

Fig. 4. The processes of heating and cooling a fluid at constant pressure undergoing closed transformations: (a) The non regenerative engine structure showing the heat exchangers actuating alternatively as heater and cooler. (b) The corresponding non regenerative T-s diagram

The specific work is then given as:

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

Thus the ideal thermal efficiency according [27] and [28] is:

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

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 was carried out according to [2], with data extracted from [30] (REFPROP) and the Engineering Equation Solver (EES) tool. 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.

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_w) exchanged in the condenser, 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 (between cooling water $\dot{q}_w$ and exhausted steam $\dot{q}_v$), with reference to Fig. 5 is defined as:

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

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

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

and the net heat flow ($\dot{q}_n$) absorbed by the IECC is then expressed as

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

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

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

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

5. RESULTS AND DISCUSSION

The results of the study using the EES tool are presented in Table 3 and Fig. 5 and have been carried out following the methodology described in [2]. 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 [30]. Cycle computation for the non regenerative IECC is referred to the T-s diagram of Figs. 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.

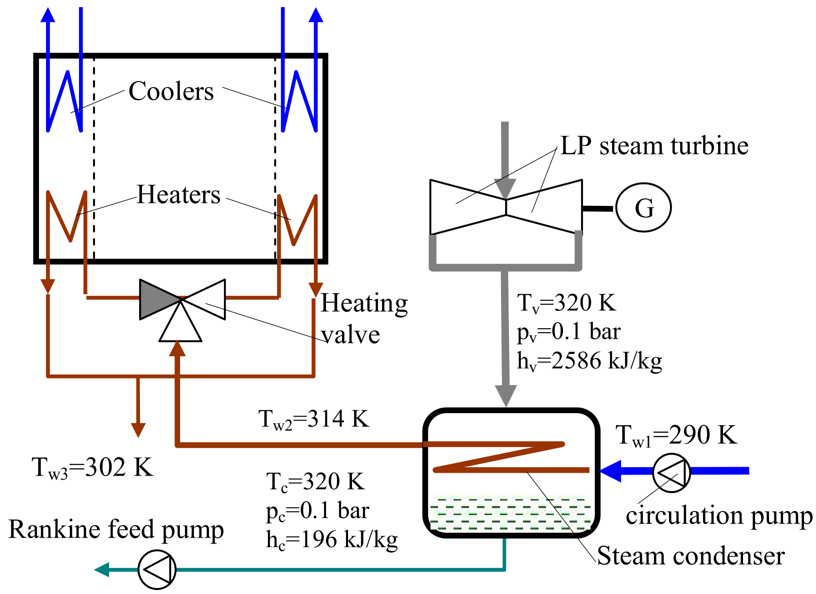

Fig. 5. Structure of the bottoming power plant of the case study using an IECC as power convertor with water as heat transfer fluid

Table 3 shows the performance results of 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. The heat sink temperature and top temperature are respectively 288 K and 312 K, with heat energy from the exhausted steam of the low pressure steam turbine of a bottoming steam Rankine power plant. Based on the results shown in Table 3, 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.

However, as described in [2], since the most interesting data concerns to the net efficiency, it is convenient to take into consideration the plants studied in [21] and [1] where a ratio of the net to the ideal efficiencies of 0.63 has been assumed. This value for the overall efficiency has been achieved on the basis of previous experimental works [21]. Thus, in [19] and [1] it is reported a thermal efficiency of 1.9% at a top temperature of 299 K, while the overall efficiency approaches 1.2%. This means that the overall losses due to internal and external irreversibilities in the plant described in [21,2] approaches a net efficiency (η_N) given as 1.2/1.9= 0.63.

Consequently, the net efficiency for the non regenerative IECC can be approached under the same assumptions and methodology. Thus, assuming the losses for the IECC similar to the losses of the plant described in [1,2], 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

$$\eta_N = \frac{1.2}{1.9}\cdot\eta = 0.63\cdot\eta \tag{14}$$

Table 4 shows the performance results of 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.

Comparing the results shown in Table 3 (ideal thermal efficiency) with the results shown in Table 4 (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 3.

Table 3. The performance results showing the ideal thermal efficiency and specific work versus the top cycle temperatures with a pressure ratio of 1:1

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

Table 5 shows the performance results of 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.

Considering the heat rejected by the steam condenser then, according to equation (3), it follows that the overall efficiency is reduced by 50% with respect to the efficiency reported in Table 4, as shown in Table 5, where the overall efficiency is estimated as about 18.5% for hydrogen and 25.3% for helium as working fluids. It was also observed that Carnot efficiency is violated without violating first and second laws.

Fig. 6 depicts the performance results of 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. As presented below, the overall efficiencies are nearly constant as function of the top temperatures for both working fluids. In any case greatly exceeds the Carnot factor, although helium is more efficient than hydrogen.

Table 4. The net efficiency (η_N) versus the top cycle temperatures

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

Table 5. The net efficiency versus the top cycle temperatures with respect to the amount of available heat rejected by the condenser

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

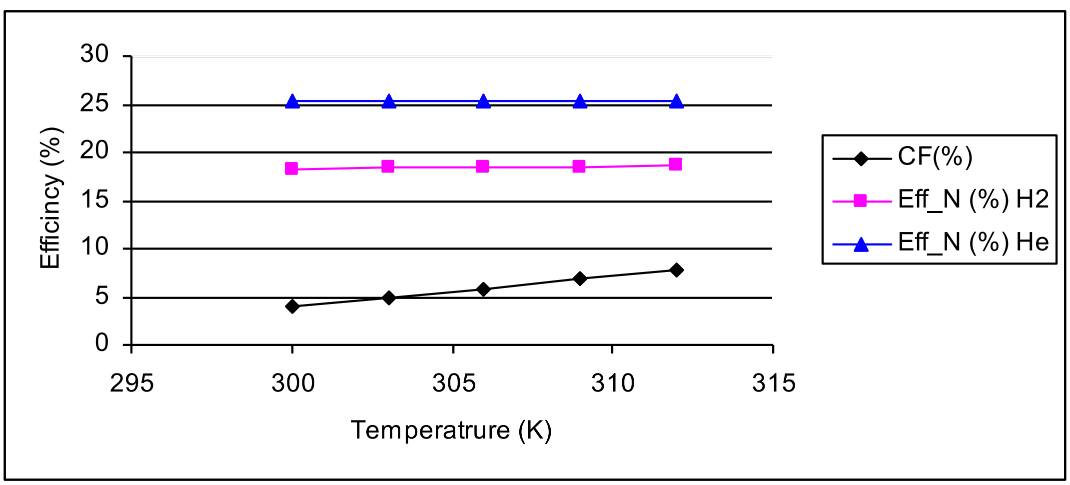

Fig. 6. The net efficiency versus the top cycle temperatures with the data shown in Table 5

6. CONCLUSION

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 to 65% is being rejected to the environment mainly as a waste energy. 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 described previously in [1] and [2], which is characterised by its ability for 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 under the same range and ratio of temperatures, and compared also with that of reference [2].

The most important conclusion is related to the IECC thermal efficiency which largely exceeds the Carnot efficiency under appropriated operating conditions. As in the previously studied case [2], the reason is due to the fact that since the IECC consists of a double effect cylinder:

- Both sides of the IECC cylinder performs mechanical wok simultaneously by absorbing and rejecting heat,
- Avoids the conventional degradation of heat due to the isentropic efficiency because of the isobaric contraction expansion processes, and.
- The selected working fluids contribute to the enhancement of the thermal efficiency due to its inherent thermodynamic characteristics and behaviour.

As shown in [2], on the basis of the proposed IECC structure, a strategy that rejects heat to the environment while performing mechanical work is possible. As consequence, the estimated net efficiency is accordingly high. Furthermore, the widespread use of steam Rankine condensers as potential power sources contribute on reducing the massive use of fossil fuels as well as the GWP.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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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

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