British Journal of Applied Science & Technology
4(26): 3840-3855, 2014

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Preliminary Study of an Efficient OTEC Using a Thermal Cycle with Closed Thermodynamic Transformations

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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.

Original Research Article

Received 29th April 2014
Accepted 1st July 2014
Published 17th July 2014

ABSTRACT

The research work is focused on thermal engine structures undergoing isobaric expansion-compression based thermal engines powered by ocean thermal energy. The isobaric expansion-compression based thermal cycles referred to in this paper, differs from the conventional quadrilateral Carnot based thermal cycles in that the conversion of heat to mechanical work is performed assuming a load reaction driven path function, where as heat is being absorbed (isobaric expansion process) and rejected (isobaric compression process), mechanical work is simultaneously performed without the conventional quasi-entropic expansion, contrary to what happens in conventional quadrilateral based Carnot engines.

An analysis of the ideal thermal cycle performed by means of an isobaric expansion-compression based cylinder is carried out and results compared with some previously achieved results of conventional technology including that of a Carnot cycle operating under the same ratio of temperatures. Into the range of its inherent low operating temperatures high ideal thermal efficiency is achieved for hydrogen and helium as working fluids. The achieved results associated with a simple and compact machine

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envisage the way towards a new generation of ocean thermal energy convertor (OTEC) power plants operating with the isobaric expansion-compression based cylinder.

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

Nomenclature

- γ *adiabatic expansion coefficient*
- η *thermal efficiency*
- η_N *net or overall OTEC efficiency*
- *Cp specific at constant pressure (kJ/kg-K)*
- *Cv specific at constant volume(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)*
- q_o *specific rejected heat (kJ/kg)*
- *R Perfect gases constant*
- *U internal energy (kJ/kg)*
- W_N *net output specific work (kJ/kg)*
- W_R *net specific work (kJ/kg) for a regenerative cycle*

Acronyms

- *CF Carnot factor, Carnot efficiency*
- *NORC Non-Organic Rankine cycles*
- *ORC Organic Rankine Cycle*
- *OTEC Ocean thermal energy convertor*
- *WF Thermal working fluid*
- *CSWP Cool deep sea water pump*
- *HSWP Hot sea water pump*
- *IECC isobaric expansion-compression cylinders*
- *RV Regenerator valve*
- *2P3W 2-positions-3 way valve*
- *CV Cooling 2P3W*
- *HV Heating 2P3W*

1. INTRODUCTION

All the known thermal cycles under use so far are derived from the Carnot engine, which means quadrilateral 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 experience 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 heat engines based cycles is depicted.

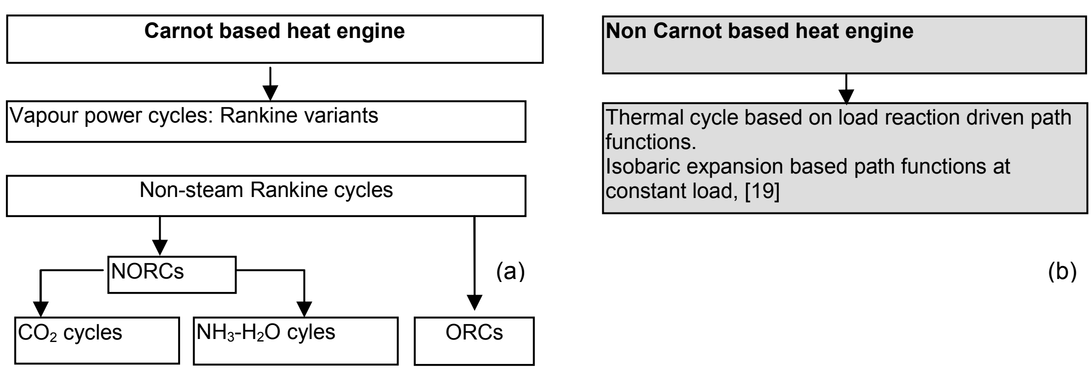

Fig. 1. Classification of Carnot and non Carnot based thermal engines suitable for OTEC applications. (a), Carnot based heat engines. (b), non Carnot based engines

Renewable energy conversion techniques contribute on reducing carbon dioxide emissions, global warming, and environmental pollution. The ocean thermal energy conversion (OTEC) procedure uses the temperature difference between the surface warm seawater and the deep cold seawater to drive Rankine cycle based power plants as power source [1-3]. OTEC techniques exhibit a very small environment impact. However the low net efficiency of OTEC as consequence of its inherent low difference of operating temperatures restricts the implementation of this technology [4]. Consequently, improving the thermal efficiency of OTEC based power plants has becoming an important objective, for instance by selecting suitable working fluids for use in the Rankine cycles. In [5,6] the major components of an OTEC plant theoretically and experimentally have been studied. In their simulation results, the temperatures of warm and cold seawater were 26ºC and 4ºC, for optimising a 100-MW OTEC system. The authors reported also that R717 was one of the suitable working fluids for a closed Rankine cycle OTEC power plant. Using R717 as the working fluid, the authors of the work referenced as [7] studied theoretically the effects of temperature and flow rate of cold seawater on the net output of an OTEC power plant. They concluded that the network has a maximal output at a specific cold seawater flow rate.

In order to enhance thermal efficiency while reducing system costs, the organic Rankine cycle (ORC) is used with working fluids that have a low boiling temperature. In this way, in [8] and [9] suitable working fluids for an ORC have been investigated to convert low-grade heat. The design for an OTEC based ORC with R717 and R134a has been optimised in [10]. Various applications aiming to increase the thermal efficiency of the OTEC system uses solar heat energy [11,12] and the waste heat of nuclear power condenser [13] to increase the temperature of warm seawater contribution on the thermal efficiency enhancement.

In [14] an experimental study measured the temperature and pressure before and after each component of a demonstration OTEC system was reported. According to the report, a maximum thermal efficiency of approximately 1.5% was obtained in the OTEC system when R134a was used as the working fluid. They also determined that both the thermal efficiency and the power output of the system increased upon increasing the operating temperature difference, which is the temperature difference between warm and cold seawater. In [15] a regenerative ORC in which R123 was used as working fluid, and reported that the performance of basic ORC system and the regenerative system were 6.15% and 7.98%, respectively, with a 130ºC geothermal heat source has been experimentally investigated. These results showed that the regenerator in the ORC system improved by 1.83% as the power output was 6kW.

From the points of view of performance and economy, finding the optimal operating temperature of the heat exchanger is critical for making the heat exchanger as small as possible when using an available temperature difference of only approximately 20ºC. The heat transfer performance in exchangers of the Rankine cycle has been considered widely to evaluate OTEC systems from an economic standpoint. An objective function, which represented the ratio of total heat transfer area to the net power output of the OTEC system, was presented in [16], who analyzed the optimization for a 1-MW OTEC power system operated in a simple closed Rankine cycle with R717 being used as the working fluid.

In [17] A Preliminary assessment of OTEC resources has been performed. Calculations indicated a long-term heating of the tropical water column if deep cold seawater were used at flow rates per unit area of the order of the average abyssal upwelling. The author concluded that this phenomenon would correspond to a transient cooling of the tropical mixed layer, and that such predictions should be further evaluated with a three-dimensional model of the oceanic circulation since a one-dimensional representation does not allow any potential adjustment from convective horizontal currents. According to the mentioned study, about 5 TW of steady-state OTEC power may be available at most. This is slightly more than a recent estimate based on conservative standardised OTEC conditions but it remains much smaller than values generally available in the technical literature. However, the present OTEC resource estimates still largely exceed today's worldwide electricity consumption. It is unlikely that a possible lack of sustainability of OTEC resources at very large scales will ever be tested in practice.

Taking into consideration the studies carried out in [17], a limited amount of OTEC based power should be installed and exploited in order to assure long time sustainability.

The work has been organised into 5 main sections, where section is 2 is devoted to the description of the combined OTEC power plant, section 3 describes the new thermodynamic concepts regarding the thermal cycle options, 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 OTEC POWER PLANT OPERATING WITH CLOSED ISOBARIC EXPANSION-COMPRESSION BASED TRANSFORMATIONS

The core of the proposed OTEC power convertor is the isobaric expansion-compression based cylinders (IECC), which is schematically depicted in Fig. 2. The OTEC-IECC system shown in Fig. 2 uses four operating fluids:

- the thermal working fluid (WF) confined into the cooler and heater heat exchangers and the cylinder,
- the cooling working fluid consisting of cool deep seawater necessary to cool the WF acting as heat rejection device which transfers heat from cylinder to the environment, and
- the heating working fluid pumped from the hot seawater surface by means of the HSW necessary to heat the WF, and
- the hydraulic fluid responsible for transferring hydraulic energy to the hydraulic motor-generation set.

The proposed OTEC-IECC depicted schematically in Fig. 2, is composed of the following components: actuator cylinder, coolers, heaters, I/O cooler valves, I/O heater valves, regenerator valve, hot sea water pump, cold sea water pump, 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 OTEC 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.

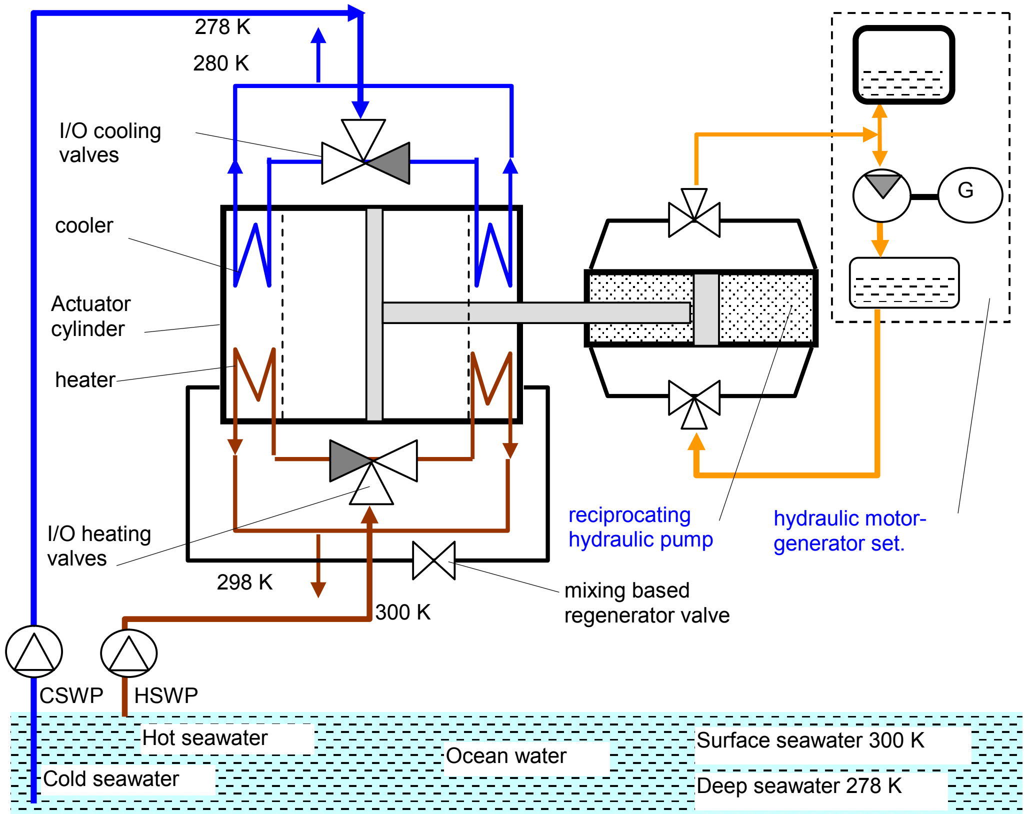

Fig. 2. Structure of the OTEC implemented with an actuator cylinder as power power convertor

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

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.

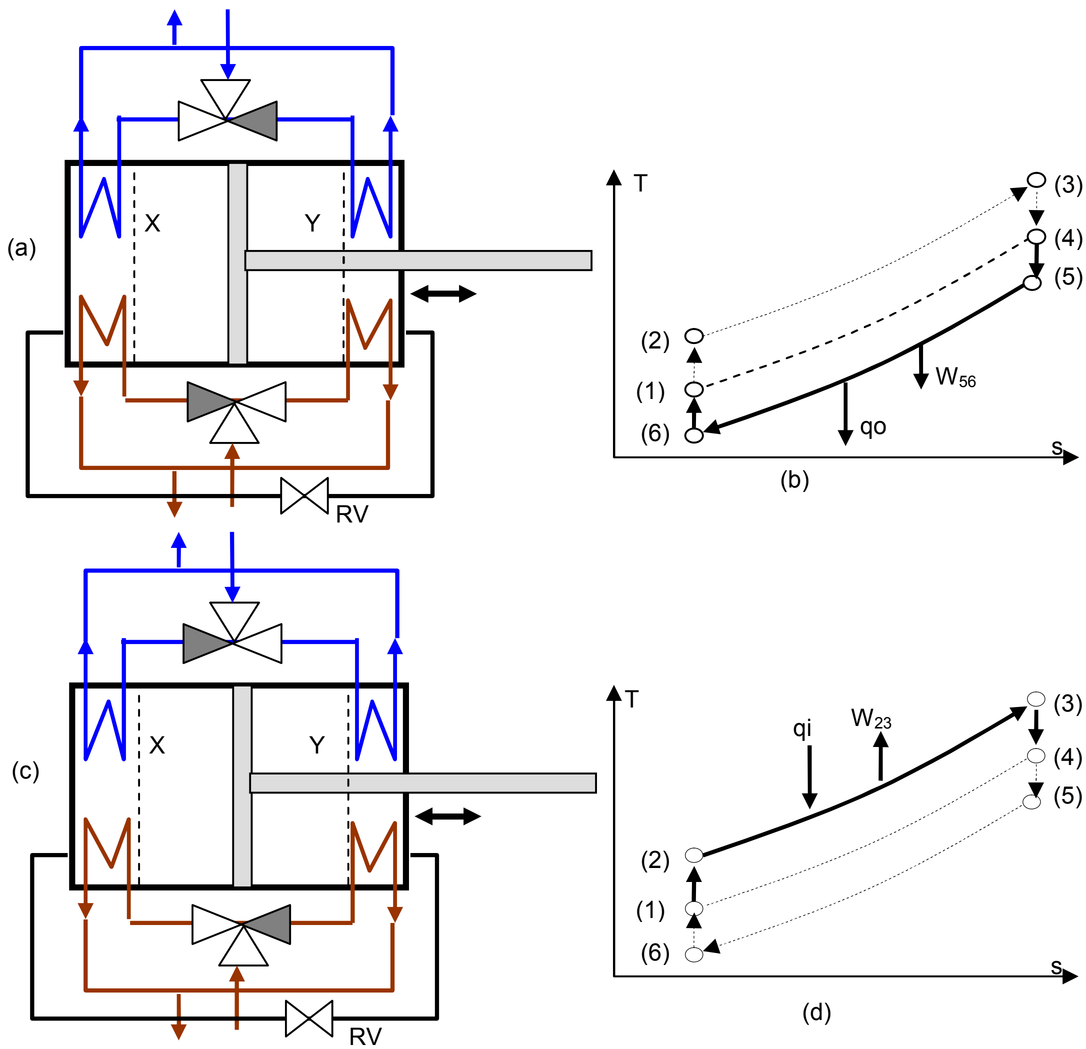

Fig. 3. The simultaneous processes of isobaric cooling and heating while performing mechanical work

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

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

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

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

From equations (1) and (2) 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

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

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

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

From equations (3) and (4) follows that mechanical work is being developed by absorbing heat as q_i.

2.2 Description of the OTEC Thermal Cycle

The structure of the thermal cycle composed by IECCs (isobaric expansion-compression based cylinders) which exhibits certain similarity with the plant structure described in [18] 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 1 for non regenerative and regenerative cycles.

2.2.1 Non-regenerative IECC

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

-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 valve (HV) 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 valve (CV) at low temperature.

-step (3)-(4)-(1)-(2) where hot WF expands at constant pressure p_2 = p_3 into the cylinder through the valve (HV), and simultaneously cooled WF is being compressed at constant pressure p_4= p_1 into the side (X) leaving the cylinder through the valve (CV), completing a cycle.

2.2.2 Regenerative IECC

The regenerative IECC is composed by the following steps according to the information depicted with Fig. 4(a) and Fig. 4(c):

- step (6)-(1) corresponding to the regeneration phase, where the regeneration valve (RV) remains open and the pressure in the side (X) of the cylinder increases from p_6 to $p_1=p_4$ and in the side (Y) decreases from p_3 to $p_4 = p_1$, so that the pressure in both sides is equal $p_4 = p_1$.

-step (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 (HV) at high temperature, and simultaneously, cooled WF is being compressed at constant pressure $p_5= p_6$ into the side (Y) at low temperature.

- step (3)-(4) corresponding to the regeneration phase, where the regeneration valve remains open and the pressure in the side (X) of the cylinder deceases from p_3 to $p_4 = p_1$ and increases in the side (Y) from p_6 to $p_1=p_4$.

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

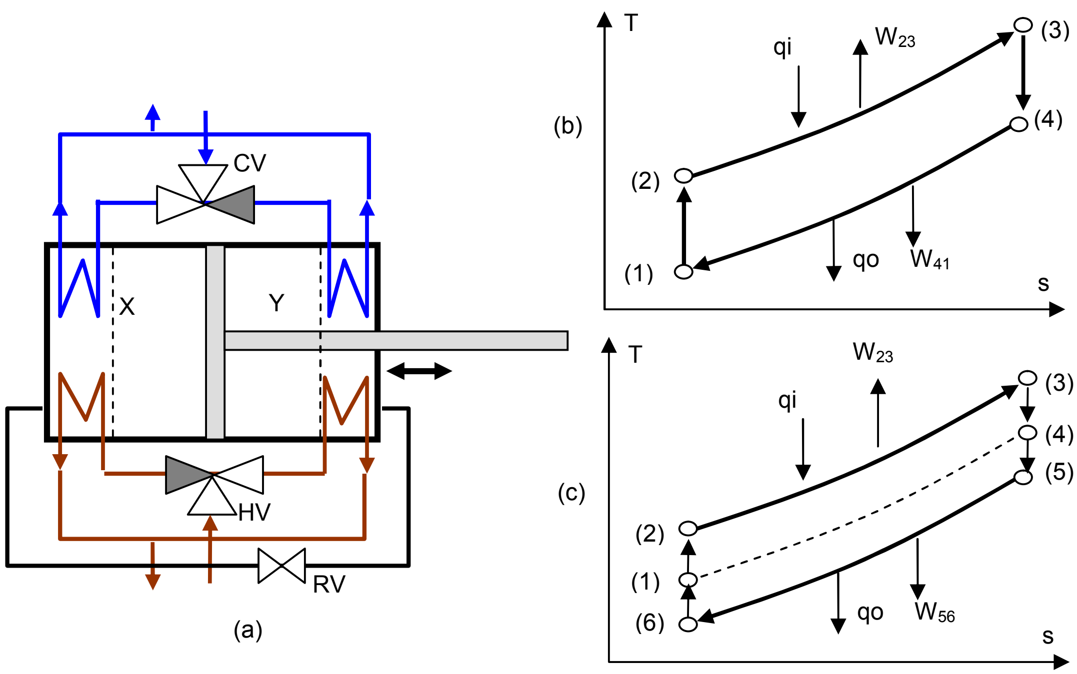

Fig. 4. The process of heating and cooling a fluid at constant pressure undergoing closed transformations: (a), the engine structure showing the heat supply and the heat sink heat exchangers. (b), the corresponding non regenerative T-s diagram. (c), the mixing based regenerative T-s diagram

Table 1. The status of the inlet, outlet and regeneration valves during the cycle evolution

Cycle	Non-regenerative cycle		Regenerative cycle			
Process	(3)-(4)-(1)	(1)-(2)-(3)	(6)-(1)	(1)-(2)-(3)	(3)-(4)	(4)-(5)-(6)
CV-X	Open	Closed	Closed	Open	Closed	Closed
CV-Y	Closed	Open	Closed	Closed	Closed	Open
HV-X	Closed	Open	Closed	Closed	Closed	Open
HV-Y	Open	Closed	Closed	Open	Closed	Closed
RV	--------	-------	Open	Closed	Open	Closed

3. MODELLING THE CLOSED TRANSFORMATION BASED NON REGENERATIVE 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 constant 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 [19].

The supplied heat from an external power source is

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

The rejected heat to the heat sink is

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

The specific work is then given as

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

Thus the ideal thermal efficiency is

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

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

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

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

4. THE NON-REGENERATIVE CASE STUDY

Cycle modelling is carried out with data extracted from [20] (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 Tables A1 and A2 shown in appendix A.

For the analysis of the cycle with hydrogen and helium as working fluids, several tests have been carried out to study it's the cycle performance and behaviour into a range of top temperatures of (294-300 K).

4.1 Results and Discussion

The results of the study achieved using the EES yield the values of Table 2 and Fig. 5 using the data shown in Table A1 of the appendix A. The data used in the case study for the study of the hydrogen and helium as working fluids is taken from reference [20]. Cycle computation is referred to the T-s diagram of Fig. 4(a) and 4(b) for the non regenerative IECC. The values of the state variables corresponding to each cycle state point for the two working fluids are shown in appendix A, where for every working fluid a range of top temperatures from 294K to 300K has been considered. The seawater temperatures are assumed into a range of 279 K for the cold seawater near to the seabed and 293-299 K for the warm seawater at the sea surface.

According to the results shown in Table 2, as the top temperatures increases from 294 K to 300 K it is observed that the thermal efficiencies remain almost constant approaching 58.5% for hydrogen and 80.5% for helium.

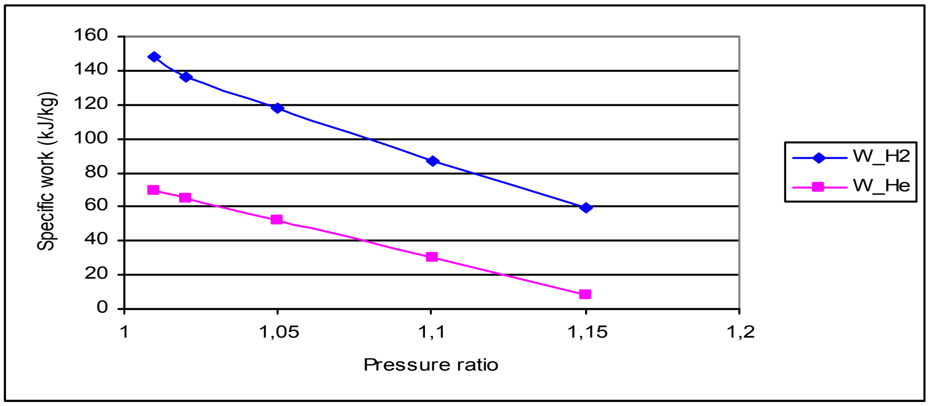

Fig. 5. Specific work as function of the pressure ratio for p_1 = 100 bar, T_1=280 K and T_3 = 298 K

It is observed also that while the ideal thermal efficiency exhibits almost no temperature dependence, specific work depends strongly on temperature according to the results shown in Table 2. However, the pressure ratio (defined as the ratio of the highest (top) pressure to the lowest (bottom) pressure into the cylinder) exerts a strong influence on the specific work as depicted in Fig. 5. Consequently, the specific work depends on the top temperature and pressure ratio. From the commented premises follows that high operating pressure with low pressure ratios yield good performance on the basis of a low specific volume, which means low structural masses and sizes while keeping high constant thermal efficiencies.

Table 2. The performance results (ideal thermal efficiency), showing the thermal efficiency, specific work, the Carnot specific work and the Carnot efficiency 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

T(K)	294	296	298	300
CF (%)	4.76	5.40	6,04	6.66
		H2		
η(%)	58.54	58.59	58.78	58.55
Carnot W(kJ/kg)	4.17	6.32	8.82	11.80
W(kJ/kg)	50.68	67.63	84.7	101.14
		He		
η (%)	80.53	80.66	80.68	80.63
Carnot W(kJ/kg)	0.77	1.44	2.25	3.2
W(kJ/kg)	12.94	21.35	29.73	38.08

Restrictions on the thermal efficiency imposed by the Carnot factor cannot be applied to the IECC because of its very low dependence on the operating temperatures.

As consequence of the analysis of the achieved results shown in Table 2 regarding the Carnot factor, follows that it isn't comparable to the results rendered by the IECC based OTEC because of the inherent differences of the proposed cycle. However the OTEC power plant investigated in [18] will be taken as reference to compare performance results. Thus, in [18] it is reported a thermal efficiency of η[18] = 1.9% at a top temperature of 299 K, while the overall efficiency approaches η_N [18]= 1.2%. This means that the overall losses due to internal and external irreversibilities in the plant described in [18] approaches a net efficiency (η_N) given as $\eta_N[18]/\eta[18]$= (1.2/1.9)= 0.63.

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

$$\eta_N = \frac{\eta_N[18]}{\eta[18]} \cdot \eta \tag{10}$$

Assuming the losses for the IECC based OTEC similar to the losses of the OTEC described in [18], because of its structural similarities with respect to its heat exchangers, the overall efficiency for a non regenerative IECC can be estimated as

$$\eta_N = \frac{\eta_N[18]}{\eta[18]} \cdot \eta = \frac{1.2}{1.9} \cdot \eta = 0.63 \cdot \eta \tag{11}$$

Therefore, the net IECC thermal efficiency has been computed for the same top temperatures assumed for the Table 1, which are within the range of 294 K and 300 K. As noted from Table 3, net efficiencies are yet substantially higher than both, the Carnot factor and the CAPILI engine studied in [18].

Table 3. Estimated net efficiency for the non regenerative IECC based OTEC as function of the top operating temperatures using information from the reference [18]

	T(K)	294	296	298	300
	CF (%)	4.76	5.40	6,04	6.66
H_2	**η_N (%)**	36.9	36.9	37.0	36.9
He	**η_N (%)**	50.73	50.81	50.82	50.8

With the aim of scaling up the proposed model, the results achieved as consequence of the studied case have been extrapolated to a 10 MW OTEC. Thus, using hydrogen and helium as working fluids and using the results depicted in the Table A2 of the appendix A, the main operating parameters for the 10 MW OTEC are shown in Table 4. In this way, Table 4 shows the amount of required working fluid (WF mass kg), the volume at 11 bar (WF vol. m^3), including the volume of the heat exchangers located into the cylinder, the total heat flow (heat flow kJ/s), and the total seawater flow (seawater flow kg/s), which includes the cold seawater and the warm seawater.

Table 4. Approached data to implement a 10 MW OTEC based power plant operating with hydrogen or helium as working fluids

WF	**T (K)**	**p (bar)**	**v (m^3/kg)**	**Power (MW)**	**Work (kJ/kg)**	**WF mass (kg)**	**WF vol. (m^3)**	**net Eff (%)**	**heat flow (kJ/s)**	**water flow (kg/s)**
H_2	300	11	1.1321	10.00	101.14	98.9	111.9	37	27027	3233
He	300	11	0.5694	10.00	38.08	262.6	149.5	51	19608	2345

5. CONCLUSIONS

A preliminary design study of an OTEC based power plant using the IECC characterised by its ability to delivering mechanical work while rejecting and absorbing heat under load reaction based path functions has been proposed, and the IECC operating optionally with hydrogen or helium as working fluids has been analysed. According to the characteristics of the new 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 [18].

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 a IECC, the following conditions are met:

- The fact of performing mechanical work by rejecting heat to the environment means a relevant contribution to the thermal efficiency of the IECC.
- The conventional quasi-isentropic expansion process is avoided in the IECC, so that the degradation of heat energy due to the isentropic efficiency is neglected.

Consequently, the main reasons for the efficiency enhancement with respect to conventional OTECs operating under quadrilateral 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.
- The selected working fluid due to its inherent characteristics and behaviour.

On the basis of the feasible structure proposed for the IECC as well as the ideal thermal efficiency which is significantly increased in comparison with the conventional OTECs under realizable and viable conditions a new family of OTEC convertors is expected. Furthermore, the estimated net efficiency is accordingly high. Finally the widespread use of ocean thermal energy as 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 global warming potential, including contaminant fossil fuels based emissions, supposes a relevant fact that needs to be experimentally researched.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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APPENDIX A: The data for the Case study

Table A1. Data used to analyse the behaviour of the pressure ratio on the specific work for hydrogen and helium as working fluids taking the bottom operating pressure as 100 bar

point	T(K)	h(kJ/kg)	u(kJ/kg)	s(kJ/kg.K)	p(bar)	v(m^3/kg)
			H_2			
1	280	3714.60	2487.30	33.44	100	0.12272
2	281.68	3730.01	2496.10	33.44	101	0.12195
3	298	3976.50	2672.80	34.30	101	0.12908
4	297.15	3963.60	2664.20	34.30	101	0.12994
2	281.68	3744.36	2504.10	33.44	102	0.12126
3	298	3977.10	2672.60	34.26	102	0.12789
4	296.33	3951.80	2655.80	34.26	100	0.12960
2	281.68	3786.72	2527.80	33.44	105	0.11887
3	298	3978.60	2671.80	34.14	105	0.12445
4	293.9	3916.30	2630.60	34.14	100	0.12857
2	287.91	3855.54	2566.30	33.44	110	0.11528
3	298	3981.20	2670.60	33.94	110	0.11914
4	289.88	3858.00	2589.10	33.94	100	0.12688
2	291.64	3922.36	2603.60	33.44	115	0.11197
3	298	3983.80	2669.50	33.75	115	0.11429
4	286.11	3803.20	2550.30	33.75	100	0.12530
			He			
1	280	1491.70	881.13	18.115	100	0.06105
2	282.22	1498.88	884.65	18.115	101	0.06070
3	298	1585.40	937.41	18.418	101	0.06416
4	296.85	1579.00	933.73	18.418	100	0.06453
2	282.22	1505.94	888.13	18.115	102	0.06036
3	298	1585.70	937.41	18.397	102	0.06356
4	295.65	1572.80	929.99	18.397	100	0.06428
2	282.22	1527.11	898.46	18.115	105	0.05936
3	298	1586.60	937.51	18.337	105	0.06182
4	292.25	1555.20	919.37	18.337	100	0.06358
2	290.83	1561.58	915.28	18.115	110	0.05780
3	298	1588.30	937.68	18.241	110	0.05914
4	286.89	1527.40	902.64	18.241	100	0.06248
2	296.02	1595.11	931.66	18.115	115	0.05634
3	298	1589.90	937.84	18.150	115	0.05670
4	281.9	1501.50	887.07	18.150	100	0.06144

Table A2. 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 respectively

point	T(K)	h(kJ/kg)	u(kJ/kg)	s(kJ/kg.K)	p(bar)	v(m^3/kg)
			H_2			
1	280	3676.10	2514.20	43.030	10	1.16190
2	287.93	3809.75	2594.40	43.030	11	1.08670
3	300	3962.50	2717.20	43.620	11	1.13210
4	291.9	3846.10	2635.00	43.620	10	1.21110
3	298	3933.90	2696.80	43.520	11	1.12460
4	289.87	3817.00	2614.40	43.520	10	1.20270
3	296	3905.20	2676.50	43.425	11	1.11700
4	287.95	3789.60	2594.80	43.425	10	1.19470
3	294	3876.60	2656.10	43.328	11	1.10950
4	286	3761.70	2575.10	43.328	10	1.18670
			He			
1	280	1462.50	878.02	22.887	10	0.58450
2	290.9	1529.44	912.03	22.887	11	0.55220
3	300	1566.70	940.38	23.047	11	0.56936
4	288.79	1508.20	905.41	23.047	10	0.60274
3	298	1556.30	934.15	23.012	11	0.56559
4	286.85	1498.10	899.36	23.012	10	0.59872
3	296	1545.90	927.92	22.977	11	0.56182
4	284.92	1488.10	893.36	22.977	10	0.59472
3	294	1535.50	921.68	22.942	11	0.55804
4	283.01	1478.10	887.39	22.942	10	0.59074

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