

Preliminary study of an efficient OTEC using a thermal cycle with closed thermodynamic transformations.

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

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 thermal efficiency is achieved for hydrogen and helium as working fluids. The achieved results associated with a simple and compact machine envisage the way towards a new generation of OTEC power plants operating with the isobaric expansion-compression based cylinder.

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

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 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 heat engines based cycles is depicted.

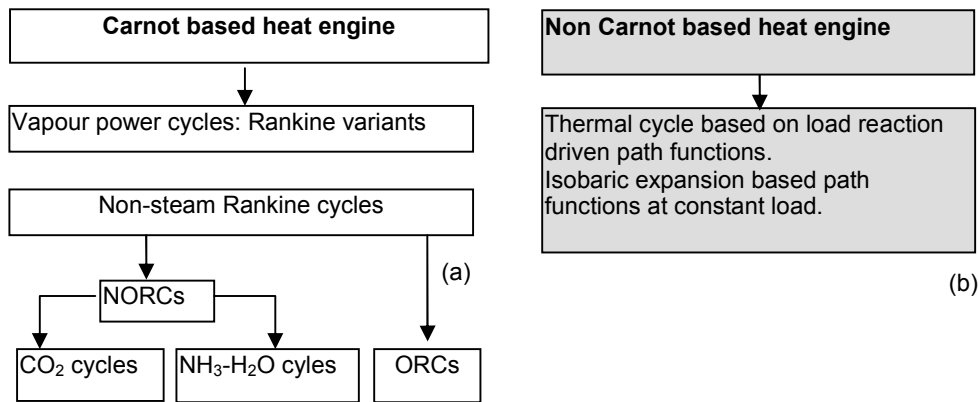


Fig. 1. Classification of Carnot and non Carnot based thermal engines suitable for OTEC applications. (a), Carnot based heat engines. (b), non Carnot based engines.

Interesting scientific works dealing with OTEC have been recently published. Thus, in this way a review dealing with such topic emphasising the potential of OTEC to produce not only clean energy but also potable water, refrigeration, and aquaculture products has been reported in [1]. The study concluded that successful implementation of OTEC at scale will require the application of insights and analytical methods from economics, technology, materials engineering, marine ecology, and other disciplines as well as a subsidized demonstration plant to provide operational data at near-commercial scales.

In the same line reported in [2] the authors reviewed the most recent advances and challenges for the exploitation of low grade thermal energy resources with particular emphasis on ORC based systems, and developed a simulation tool for the exploitation of low grade thermal energy by means of ORCs which is useful in choosing an appropriate ORC particularly designed to operate with low grade heat sources such in OTEC industry. In [3] an ammonia-water based OTEC system was experimentally investigated, finding an approach the maximum thermal energy that could be delivered without affecting significantly the surface ocean temperature, where an annual OTEC net power production of about (7 TW, for example), could be achieved with little effect on the oceanic temperature field.

A hybrid plant based on the principle of OTEC coupled to a 740 MW coal-fired power plant project located in an ocean place where the surface to deepwater temperature difference would not suffice for regular OTEC plants has been reported in [4]. They presented the thermodynamic model to assess the overall efficiency gained by adopting an ammonia Rankine cycle plus a desalinating unit, heated by the power plant condenser discharge and refrigerated by cold deep seawater, where the authors concluded that this approach would improve the acceptability of thermal and nuclear power plant projects regardless of the plant location.

In [5] it is reported an interesting research focused on attaining very low levelised electricity costs by designing a system with very low material cost, while maintaining appreciable conversion efficiency and achieving low maintenance cost, where all investigated designs were dimensioned at a 50 MW scale production. They investigated the levelised electricity costs showing that a new proposed hybrid of ocean thermal energy conversion with an offshore solar pond (OTEC-OSP) may have the lowest levelised electricity costs of 0.04 €/kWh., and concluded that the addition of a floating offshore solar pond (OSP) to an OTEC system increases the temperature difference in the Rankine cycle, which leads to an improved efficiency of 12%, while typical OTEC efficiencies are 3%. However in the proposed study such achieved efficiency can be duplicated at reduced costs, since the required installation is simpler. Furthermore, cycles that can efficiently use low grade heat are being investigated to fulfil the requirements of OTEC based power plants. Some studies

dealing with trilateral thermal cycles are being considered for possible publication. Thus, for instance, a comparison of trilateral cycles and organic Rankine cycles has been carried out by [6] in which a clear contribution to the thermal efficiency enhancement has been reported. The objective of the study is to efficiently convert low temperature heat mainly from ocean thermal and solar thermal sources to be converted by means of the proposed thermal cycle and plant structure into electric energy. The proposed conversion technology is based on a non conventional thermodynamic cycle: the IECC with or without regeneration. The proposed IECC is not based on the quadrilateral Carnot cycle but on the trilateral cycle which absorbs low grade heat and develops mechanical work simultaneously along a closed system based transformation of the IECC. As a consequence of this contribution, this cycle is not restricted by the Carnot factor constraints. Its maximum thermal efficiency is influenced not only by the ratio of the top to bottom temperatures like for instance in the cycles' analysis described in [7] and [8], but also by some factors such as the characteristics of the working fluids and regeneration capabilities when regeneration is considered. The work has been organised into 5 main sections, where section 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 PROPOSED OTEC POWER PLANT OPERATING WITH CLOSED ISOBARIC EXPANSION-COMPRESSION BASED TRANSFORMATIONS.

The core of the proposed OTEC power converter 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 (CSW) 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 (HWF) 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.

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.

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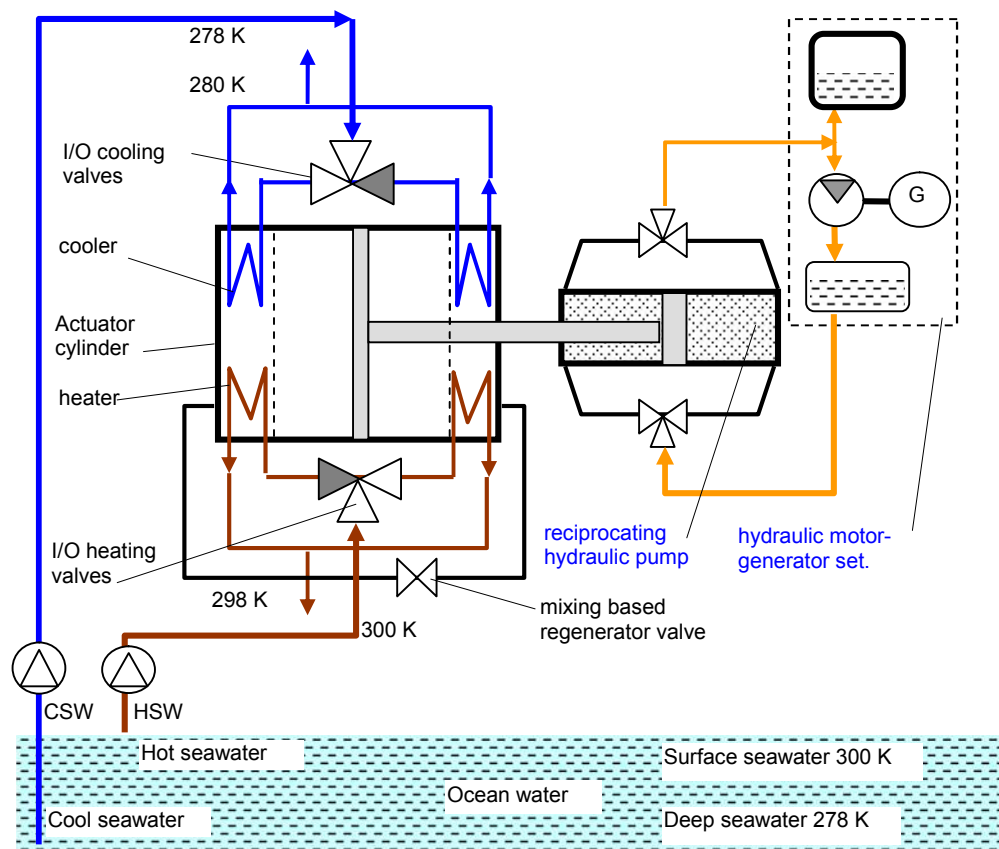


Fig. 2. Structure of the OTEC implemented with an actuator cylinder as power power convertor.

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

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

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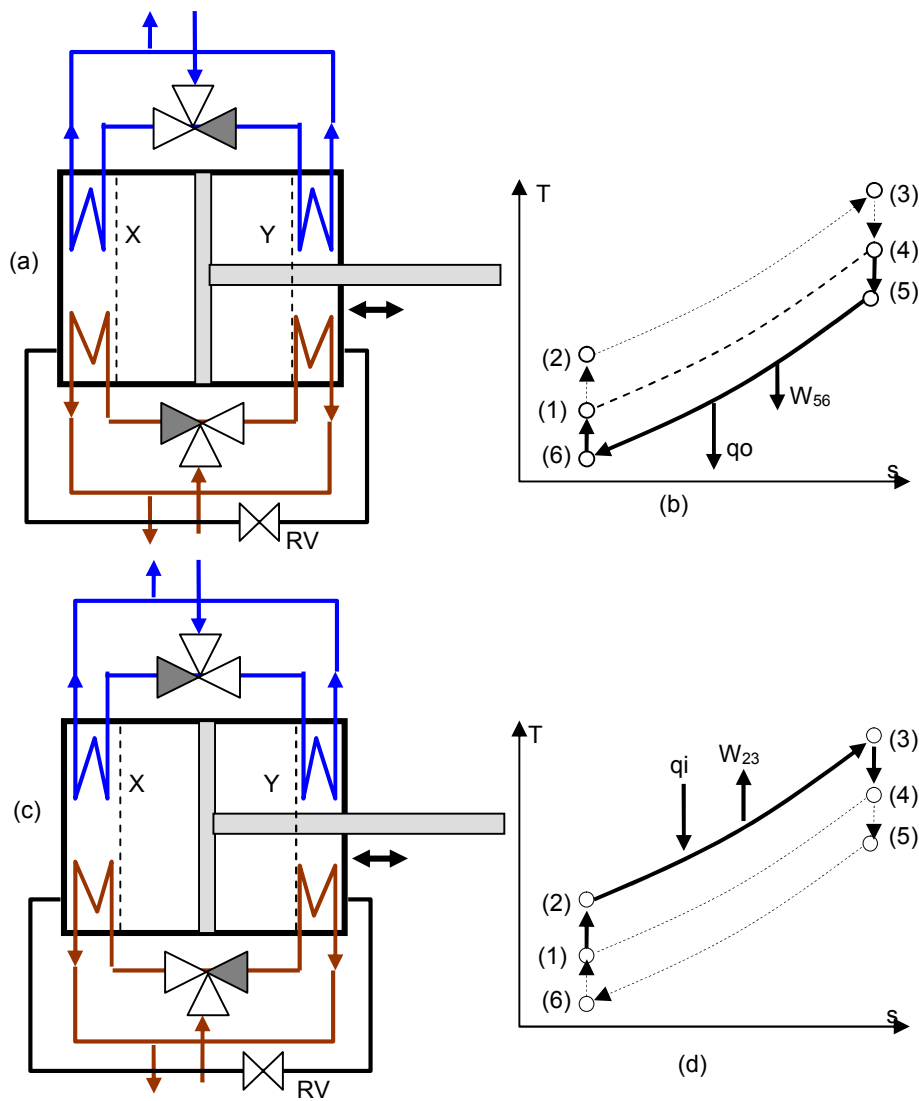


Fig. 3. The simultaneous processes of isobaric cooling and heating while performing mechanical work.

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

$$W_{56} = \int_{V_5}^{V_6} p \cdot dv = p_5(V_6 - V_5) = p_6(V_6 - V_5) = \Delta u_{65} - q_0 \quad (2)$$

From equations (1) and (2) follows that mechanical work is being developed by rejecting heat as q_0 .

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 $q_i = \Delta u_{32} + W_{23} = C v_{23}(T_3 - T_2) + R(T_3 - T_2) = C p_2(T_3 - T_2)$ has been assumed. As consequence, the mechanical work delivered while heating process is defined as

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

From equations (3) and (4) follows that 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 [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 1 for non regenerative and regenerative cycles.

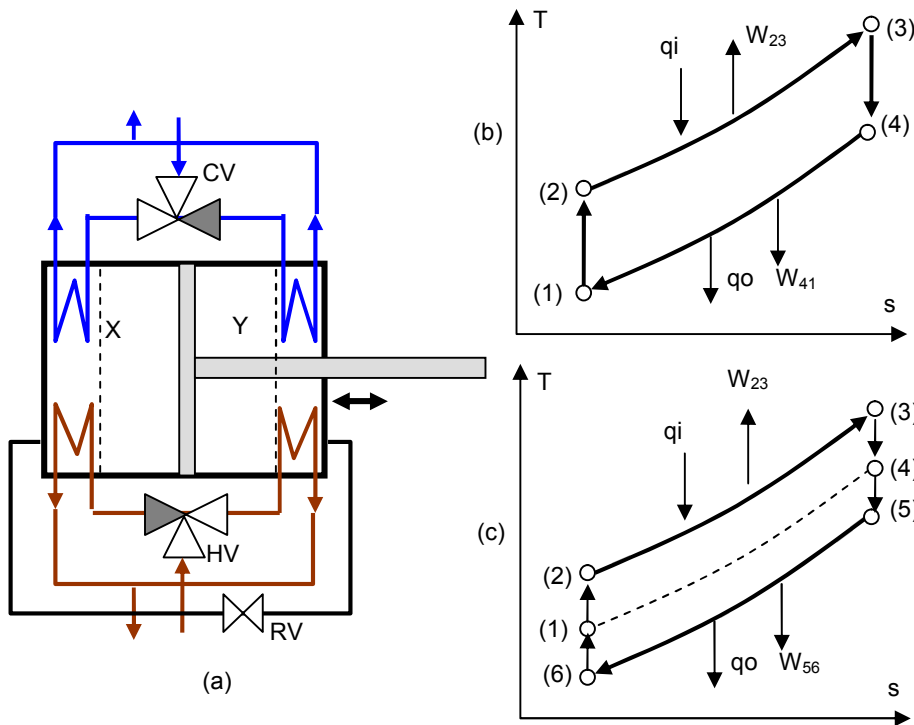


Fig. 4. The process of heating and cooling a fluid at constant pressure undergoing closed transformations: (a), the engine structure showing the heat supply and the heat sink heat exchangers. (b), the corresponding non regenerative T-s diagram. (c), the mixing based regenerative T-s diagram.

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 (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.

-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.

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 decreases 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.

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

Cycle Process	Non regenerative cycle			Regenerative cycle		
	(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.

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

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

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 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 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 has very little influence on the ideal thermal efficiency.

4. THE NON REGENERATIVE CASE STUDY

Cycle modelling is carried out with data extracted from [10] (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. The data used in the case study for the study of the hydrogen and helium as working fluids is taken from reference [10]. 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 and for the three working fluids are shown in appendix A, where for every working fluid a range of top temperatures from 294 K to 300 K 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.

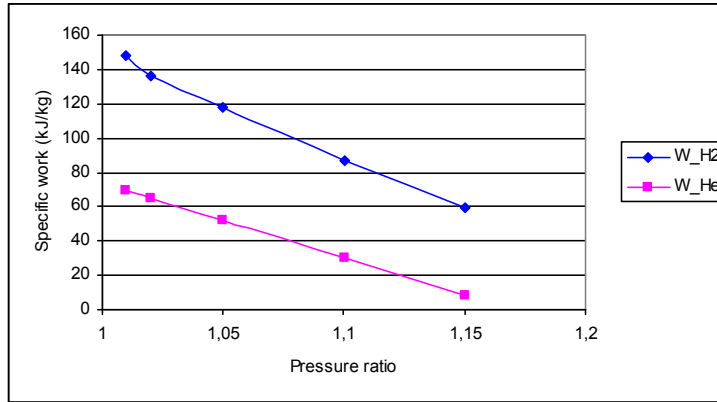


Fig. 5. Specific work as function of the pressure ratio for $p_1 = 100$ bar, $T_1=280$ K and $T_3 = 298$ K.

It is observed also that while the 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 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 showing the 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.

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
W(kJ/kg)	50.68	67.63	84.7	101.14
He				
η (%)	80.53	80.66	80.68	80.63
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 [9] will be taken as reference to compare performance results. Thus, in [9] it is reported a thermal efficiency of $\eta[9] = 1.9$ % at a top temperature of 299 K, while the overall efficiency approaches $\eta_N[9] = 1.2$ %. This means that the overall losses due to internal and external irreversibilities in the plant described in [9] approaches a net efficiency (η_N) given as $\eta_N[9]/\eta[9] = (1.2/1.9) = 0.63$.

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

$$\eta_N = \frac{\eta_N[9]}{\eta[9]} \cdot \eta \quad (10)$$

Assuming the losses for the IECC based OTEC similar to the losses of the OTEC described in [9], because of its structural similarities with respect to its heat exchangers, the overall efficiency for a non regenerative IECC can be estimated as

$$\eta_N = \frac{\eta_N[9]}{\eta[9]} \cdot \eta = \frac{1.2}{1.9} \cdot \eta = 0.63 \cdot \eta \quad (11)$$

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Table 3. Estimated net efficiency for the non regenerative IECC based OTEC as function of the top operating temperatures using information from the reference [9].

	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

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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 [9].

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5. CONCLUSIONS

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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 [9].

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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:

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- The fact of performing mechanical work by rejecting heat to the environment means a relevant contribution to the thermal efficiency of the IECC.

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- 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.

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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:

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- 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.

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- The selected working fluid due to its inherent characteristics and behaviour.

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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 GWP, including contaminant fossil fuels based emissions, supposes a relevant fact that needs to be 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 OTEC efficiency	HWF	Heating working fluid
C_p	specific at constant pressure (kJ/kg-K)	CWF	Cooling working fluid
C_v	specific at constant volume(kJ/kg-K)	NORC	Non-Organic Rankine cycles
p	pressure (bar)	ORC	Organic Rankine Cycle
T	temperature (K)	OTEC	Ocean thermal energy convertor
h	specific enthalpy (kJ/kg)	WF	Thermal working fluid
s	specific entropy (kJ/kg-K)	CSW	Cool deep sea water pump
q_i	net specific input heat (kJ/kg)	HSW	Hot sea water pump
q_o	specific rejected heat (kJ/kg)	IECC	isobaric expansion-compression cylinders
W_N	net output specific work (kJ/kg)	RV	Regenerator valve
W_R	net specific work (kJ/kg) for a regenerative cycle	2P3W	2-positions-3 way valve
		CV	Cooling 2P3W
		HV	Heating 2P3W

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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

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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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