Exergo-economic assessment of OTEC power generation

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Abstract. Ocean Thermal Energy Conversion is an important renewable energy technology aimed at harvesting the large energy resources connected to the temperature gradient between shallow and deep ocean waters, mainly in the tropical region. After the first small-size demonstrators, the current technology is focused on the use of Organic Rankine Cycles, which are suitable for operating with very low temperatures of the resource. With respect to other applications of binary cycles, a large fraction of the output power is consumed for harvesting the resource – that is, in the case of OTEC, for pumping the cold and hot water resource. An exergy analysis of the process (including thermodynamic model of the power cycle as well as heat transfer and friction modelling of the primary resource circuit) was developed and applied to determine optimal conditions (for output power and for exergy efficiency). A parametric analysis examining the main design constraints (temperature range of the condenser and mass flow ratio of hot and cold resource flows) is performed. The cost of power equipment is evaluated applying equipment cost correlations, and an exergo-economic analysis is performed. The results allow to calculate the production cost of electricity and its progressive build-up across the conversion process. A sensitivity analysis with respect to the main design variables is performed.

1 Introduction

The world energy scenario is continuously experiencing a strong increase in energy demand. The simultaneous diminishing of fossil resource availability and increasing of the level of pollutant and greenhouse emissions have raised the global efforts towards the rise of the contribution of renewable energy sources in the overall energy transformation. In this framework, the energy available in the sea has a huge potential and allows different forms of exploitation. The main physical phenomena associated with sea energy transportation are waves, currents, tides, and thermal gradients. Among these sources, the Ocean Thermal Energy Conversion (OTEC) technology exploits the temperature difference between the warm surface and the deep cold sea water to produce electric energy by means of thermodynamic conversion using a suitable power cycle.

The global availability of ocean thermal energy is estimated at about $4.4 \cdot 10^{16}$ kWh per year [1]. In particular, this huge potential is mainly located in the ocean waters located around the tropics and in the

correspondence of the Gulf Stream. In these locations, the temperature difference between the warm and the cold sea water is typically in the range between 22°C and 26°C, due to the surface water temperature of about 28-30°C and the deep water temperature of approximately 4-6°C [2].

Modern OTEC systems can rely on an open or a closed-cycle. In the first case, the system uses the sea water as working fluid. A chamber is subject to high vacuum conditions and is utilised to evaporate the warm surface water. The vapour obtained is directed in a duct embedding a low-pressure turbine connected with an electric generator. The exhaust steam is condensed by refrigeration with seawater at low temperature, pumped from the sea depth. In the closed-cycle layout, a liquid with low boiling point, such as ammonia (R717) or a suitable refrigerant or hydrocarbon (ORC), is utilised as working fluid. The fluid is evaporated through heat transfer with the warm water of the sea surface. The vapour expands and drives a steam turbine. Subsequently the vapour is condensed by refrigeration with the cold water of the deep sea. Finally, in the closed cycle the condensate is pumped back to the evaporator.

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The main advantage of the closed cycle is the possibility to select the working fluid in order to limit the dimensions of the system components. Conversely, the main disadvantage is the requirement of large heat exchangers made of expensive materials such as titanium to reliably operate in a chemically aggressive environment like the sea. Furthermore, the research efforts are dedicated to increase the efficiency of the energy conversion.

Currently, small (demonstration) OTEC plants are installed in the East Coast of Asia and in the Réunion Island in the Indian Ocean. Several projects proposed the construction of this kind of systems to solve the problem of power supply of remote islands located in the tropical oceans. In general, OTEC technology is a promising solution for tropical ocean areas located in the Caribbean Sea, South Pacific Ocean and Indian Ocean. However, the system has a relatively low overall efficiency, which is hindered by the high power consumption of the pumps working with seawater.

The original concept of harvesting the thermal energy of the sea was proposed by J. A. d'Arsonval in 1881. The first demonstrative prototype of the OTEC was built by his student G. Claude in 1928 in Belgium using the water at 30°C of a steel factory as source at high temperature and water from the river Meuse as source at low temperature. Subsequently, a prototype was installed in 1930 in a rocky natural pool in Cuba. The system was based on an open-cycle ORC [3]. Many researchers made attempts to realize a large-scale OTEC plant in the following years, but only one of them was operated and for a very limited period of 11 days.

In 1979, the first power plant rated 50 kW, based on the OTEC technology was installed off-shore in Hawaii, and was capable of delivering a net power output of about 15 kW. In 1981, an on-shore pilot plant rated 100 kW was constructed by TEPCO in the Republic of Nauru and provided a power of 31.5 kW; the system worked with a closed-cycle ORC and R22 was selected as working fluid [4]. In 1982, a 50 kW on-shore system was built in Japan and worked with seawater using ammonia in a closed-cycle [5]. An off-shore floating OTEC plant rated 1 MW was installed by the National Institute of Ocean Technology of India and the Saga University of Japan; the system was connected to the grid, delivering 493 kW [6].

Apart from these pilot demonstration successful projects, the lack of funding retarded the implementation of further projects to build OTEC systems until the beginning of the new millennium. Indeed, in the last decade, the raise of the interest towards renewable energy systems led to a renovated attention to the ocean thermal energy conversion.

In 2013, a 100 kW OTEC plant working with a closed-cycle was built on the island of Kumejima in Japan and is still operational [7]. In 2015, an OTEC system rated 100 kW was built on the Makai island in Hawaii and was connected to the electric grid [7].

Nowadays, many projects based on the OTEC technology are underdevelopment worldwide.

2 Power plant configuration

The OTEC configuration assessed in this work exploits a typical binary cycle power plant for the conversion of the ocean energy. The binary cycle architecture can vary from the basic layout to very complex ones, such as double pressure, flash or double-fluid configurations.

The studied OTEC uses the basic configuration architecture, which includes a pump, an evaporator, an expander and a condenser, in order to reduce the capital cost of the power plant, which is the only contribution here accounted for the evaluation of the build-up of the electricity production cost (specifically, the cost of the floating barge and of its propulsion system are not considered, as well as the costs associated with storage or delivery of electricity). The working fluid assessed in this analysis is Ammonia, which is recognized at present to be the most performing fluid for this range of temperature, as reported in [1].

Figure 1 displays the schematic of the OTEC power plant, highlighting, not only the binary power plant, but also the hot and cold fluid pipelines that feed the evaporator and the condenser.

The OTEC configuration allows to extract water in the proximity of the surface for heating the fluid in the evaporator, while it requires to extract cold water at approximately 1000 m below the surface level in order to cool the working fluid in the condenser. With this configuration, the binary cycle is allowed to work between 29.5°C and 4.5°C. The Carnot efficiency (which represents the limit value for thermodynamic conversion assuming infinite capacities of the hot and cold streams) for this range of temperatures is of 8%.

Figure 2 displays the temperature profile of the ocean water considered in this work.

Fig. 1. Schematic of OTEC power plant

Fig. 2. Oceanic water temperature profile against depth

3 Power plant model

The power plant calculations were developed through standard steady-state mass and energy balances for open systems. The working fluid and the sea water properties were taken from a reliable source [8]. The input data are summarized in Table 1 and the results of the calculations are collected in Table 2. The pumping losses of the pumps were calculated through a specifically developed model taking into account the friction losses in the pipes, as well as in the heat exchangers and the efficiency of the pumps.

It is necessary to introduce a performance parameter which is fundamental for this kind of power plant, as remarked in [1]. The parameter is the water mass flow rate ratio and is defined in Eq. (1). The value chosen for the base case study has been set at 0.4, as it is within the suggested value range by [1].

$$\gamma = \frac{\dot{m}_c}{\dot{m}_H} \quad (1)$$

Table 1. Assumed design input data for the OTEC power plant

Parameter	Unit	Value
Reference temperature	°C	15
Turbine isentropic efficiency	%	90
Pump isentropic efficiency	%	90
Hot water inlet temperature	°C	29.56
Cold water inlet pressure	°C	4.5
Net Power Output	kW	1000
Mass flow rates ratio	-	0.4
Condenser Range	°C	8

Table 2. Main calculated performance parameters for the OTEC Power Plant

Parameter	Unit	Value
Hot water mass flow rate	kg/s	2798
Cold water mass flow rate	kg/s	1119
Binary cycle efficiency	%	2.9%
Power plant efficiency	%	2.6%
Heat input from hot water flow	kW	38743

4 Exergy analysis

The exergy analysis allows to evaluate not only the efficiency of the assessed energy systems, but also to individuate clearly the sources of irreversibilities (exergy destructions) of each component, which hinder the performance of the global power plant.

The standard definition of exergy is “*the maximum work attainable from a system or a process through the interaction with the surrounding environment*”. This definition clearly indicates the exergy analysis as the right tool to evaluate the maximum achievable possible exploitation of a renewable resources of thermal nature, like the OTEC application.

The total flow exergy can be calculated at every point in the thermodynamic cycle, as displayed in Eq. (2).

$$Ex_t = \dot{m} \cdot [(h - h_0) - T_0 \cdot (s - s_0)] \quad (2)$$

Moreover, defining the exergy destruction (D) and losses (L) of a system, it is possible to set up the exergy balance of every part of the power plant, identifying

correctly the fuels (F) and products (P), as expressed in Eq. (3).

$$\dot{Ex}_{F,k} = \dot{Ex}_{P,k} + \sum \dot{Ex}_{D,k} + \sum \dot{Ex}_{L,k} \quad (3)$$

The conceptual difference between exergy destruction and losses is that the exergy destruction derives from friction or irreversibility of heat transfer within a defined control volume, while an exergy loss is associated with exergy transfer (waste) to the surroundings (stream of matter or heat flux).

Finally, the exergy efficiency of a component can be defined as the ratio between the product exergy and the fuel exergy. The same is true for the whole system, in which case the indirect formulation (obtainable through the subtraction of exergy destructions and losses) allows to compare among them the different sources of irreversibility. The exergy efficiency at system level can be calculated thus (direct or indirect form) as from Eq. (4).

$$\eta_k = \frac{\dot{Ex}_{P,k}}{\dot{Ex}_{F,k}} = 1 - \frac{\sum \dot{Ex}_{D,k} + \sum \dot{Ex}_{L,k}}{\dot{Ex}_{F,k}} \quad (4)$$

5 Exergo-economic analysis

The exergo-economic analysis couples the exergy and economic analyses in order to explain the cost build-up throughout the power plant process and to ultimately obtain the levelized cost of electricity (understanding the reason which lead to its construction).

Particularly, through the combination of the exergy an economic models, it is possible to assess the cost effectiveness of each component, allocating to each exergy stream a specific cost [9].

The applied economic model considers the instantaneous cost of each component in €/s. For each component, the annual investment cost is defined as shown in Eq. (5):

$$Z_k^{an} = \frac{i \cdot (1 + i)^n}{(1 + i)^n - 1} Z_k \quad (5)$$

Where:

i is the interest rate, which was assumed at 10% [10].

n is the year lifetime here assumed at 25 years.

$\dot{Z}_k$ is the sum of cost rates associated with investments for the k -th component (including direct and indirect costs). Standard rules [10, 11, 13] are applied in process engineering to evaluate $\dot{Z}_k$ starting from the Purchase Equipment Cost (PEC).

The PEC of each component was determined through the correlations reported in [11]. Costs were actualized to 2019 values, by using the CEPCI indexes [12]. The applied currency exchange rate applied was of 0.85 €/\$. The investment cost of the whole power plant also includes installation and maintenance costs, following a consolidated approach [10].

After the costs of each component are determined (from the water pipelines to each element of the power cycle), the exergo-economic approach outlined in [13, 14] can be deployed, providing for each component k a cost balance equation, as resumed in Eq. (6).

$$\begin{aligned}\dot{C}_{P,k} &= \dot{C}_{F,k} + \dot{Z}_k \\ c_{P,k} \dot{E}x_{P,k} &= c_{F,k} \dot{E}x_{F,k} + \dot{Z}_k\end{aligned}\quad (6)$$

Where:

$\dot{C}_{P,k}$ and $\dot{C}_{F,k}$ are the cost rates associated with exergy product and fuel, respectively in €/s.

$c_{P,k}$ and $c_{F,k}$ are the costs per unit of exergy of product or fuel, respectively, in [€/kWh].

In order to solve the system, the cost balances are not sufficient, therefore, auxiliary (“functional”) equations taken from [13, 14] need to be applied.

Finally, the exergo-economic indicators, namely the exergy destruction cost rate and the exergo-economic factor, can be calculated for each component as displayed in Eqns. (7) and (8).

$$\dot{C}_{D,k} = c_{F,k} \cdot \dot{E}x_{D,k} \quad (7)$$

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{D,k}} \quad (8)$$

A high value of f_k means that there is a high relevance of the capital (including operation and maintenance) costs over the cost related to the exergy destruction of the component.

6 Results

6.1 Performance analysis

The first analysis that was performed is the evaluation of the contribution that the pumping power due to the friction losses of the water pipelines has to the net power output of the power plant. Fig. 3 displays (in function of the cold fluid resource flow rate) the net power of the power plant, the gross power produced by the turbine and the pumping work required to overcome the pressure losses in the pipes. From Fig.3, it is possible to notice that there is a maximum of the obtainable power from the power plant, which is given by the combination of trends of the turbine gross power (linear behaviour) and the required pumping power (quadratic behaviour) as function of the water mass flow rate in the cold pipeline.

Fig. 3. Net power output against cold water mass flow rate (velocity within the pipes of 1 m/s).

The next analysis carried out is aimed to assess the optimal range of specific power output and global efficiency of the power plant. The specific power output is defined as the ratio between the net power output and the mass flow rate of the cold water (Eq. 9). The global efficiency of the power plant is defined as the ratio between the net power output and the heat rate entering the evaporator (Eq. 10).

$$W_{sp} = \frac{\dot{W}_{net}}{\dot{m}_c} \quad (9)$$

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{Eva}} \quad (10)$$

$$\text{Where } \dot{W}_{net} = \dot{W}_T - \dot{W}_{P1} - \dot{W}_{P2} - \dot{W}_{P3}$$

A sensitivity analysis was carried out, looking for optimal specific work and efficiency, as function of the two most relevant design parameters of the power plant, namely the water mass flow rates ratio (γ) and the range at the condenser.

Figures 4 and 5 display the variation of specific power output and global efficiency in function of γ and of the condenser range. It is interesting to remark the trend of specific work, which diminishes decreasing the water mass flow rates ratio. Indeed, the utilization of a higher mass flow rate of hot water (with respect to the cold-water flow) is beneficial because the pumping losses are mainly associated with the cold pipeline, which is much longer.

Moreover, the range at the condenser is another fundamental parameter for the design selection of the power plant. Indeed, when small values of the range are utilized, the water mass flow rate increases, increasing the friction losses in the pipes.

The present sensitivity analysis shows that the optimal range of the condenser should be between 5 and 8 °C to achieve the highest specific power.

On the other hand, the global efficiency is mostly influenced by the range at the condenser. The lower the range, the higher the efficiency. This is mainly due to increase in the thermodynamic efficiency of the power cycle, as it allows to reach lower condensing temperature of the cycle.

Fig. 4. Specific power output against water flow rate ratio and condenser range.

Fig. 5. Global efficiency against water flow rate ratio and condenser range

Fig. 7. Exergy efficiency of OTEC power plant as function of γ and Range_c

6.2 Exergy analysis

In the present case, no exergy losses were considered, as the discharge of the heated cold flow rate and that of the cooled but still warm stream exiting the evaporator represent conditions imposed by plant operation. Only destructions are consequently considered for the evaluation of the exergy efficiency. The exergy performance of the assessed OTEC power plant is of 88.32% when a water mass flow rate ratio $\gamma = 0.4$ and a condenser range of 8°C are considered.

As displayed in Fig. 6, the highest contribution to the inefficiency of the system is the exergy destruction in the condenser. This is due to the non-optimal thermal matching between the condensing ammonia and the cooling water. A possible solution for reducing this destruction would be to substitute the ammonia working fluid with a dedicated zeotropic mixture with a suitable temperature glide.

After the exergy performance of the design case study, the sensitivity analysis considering the exergy efficiency of the system, varying the water mass flow rates ratio and the condenser range was performed. As shown in Fig. 7, it was found that the exergy efficiency presents a similar trend to the global efficiency. Indeed, the highest exergy efficiencies are reached for low values of the condenser range. This is due to a reduction of exergy destruction losses at the condenser, as the match between the thermal curves improves.

Fig. 6. Exergy destruction and losses of the design case study ($\gamma = 0.4$, $\text{Range}_c = 8$)

6.3 Exergo-Economic analysis

Finally, following the thermodynamic and exergy analyses, the exergo-economic calculations were carried out.

The obtained overall specific investment cost of the system is of 21983 €/kW. Since the cost of the power output from the cycle is dependent on several input variables, a base configuration was assumed with the following values:

- $\gamma = 0.4$
- $\text{Range}_c = 8$
- The cost of streams 10 and 20 are assumed as 0;
- The cost of electricity required by the pump is the same as that produced by the turbine.

Considering the above mentioned assumptions, table 3 presents the results obtained from the exergo-economic analysis of the OTEC power plant for the base case study.

In terms of economic analysis, the condenser is the most expensive component, followed by the evaporator. From the exergo-economic point of view, if $\dot{Z}_k$ and $\dot{C}_{D,k}$ are summed, again the same components are the largest contributors, followed by the turbine, which has a higher exergy destruction rate than the evaporator.

The specific cost of the exergy product c_p associated with the turbine of the power cycle represents the cost of the output of the OTEC power plant, i.e., the production cost of electricity.

Table 3. Results of exergo-economic analysis

Component	PEC (€)	$\dot{Z}_k$ (€/s)	$\dot{C}_{D,k}$ (€/s)
Pump 1	23039	0.0001588	0.000124
Pump 2	39294	0.0002709	0.0006275
Pump 3	29708	0.0002048	0.0003963
Turbine	1.744E+06	0.01202	0.008076
Condenser	8.237E+06	0.05678	0.0526
Evaporator	4.141E+06	0.02855	0.0008991
Cold circuit	754410	0.005201	1.690E-13
Hot circuit	44969	0.00031	1.008E-13
Component	$C_{F,k}$ (€/kJ)	$C_{P,k}$ (€/kJ)	f_k (%)
Pump 1	0.00008073	0.0001012	56.16
Pump 2	0.00008073	0.00009298	30.15
Pump 3	0.00008073	0.00009367	34.07
Turbine	0.00006322	0.00008073	59.82
Condenser	0.00006322	0.00006833	51.91
Evaporator	0.00001681	0.00002627	96.95
Cold circuit	2.301E-15	0.00000575	100
Hot circuit	4.232E-15	7.441E-08	100

In the base case study, the production cost is of 29.06€/kWh, which is a quite high value compared to other RES, mainly because of the relatively low power output and the low conversion efficiency of the system.

7 Conclusions

Energy, exergy and exergo-economic analyses were carried out in this work, highlighting the performance behaviour, the inefficiency sources of the system, as well as confirming the validity of the assessed technology.

The thermo-fluid dynamics analysis allowed to conclude that in order to obtain relatively high efficiency of the power plant, the work of the water pumps should be kept low. Furthermore, it was possible to define the proper range values of the water mass flow rates ratio and of the condenser range (0.4-0.5 and 6-8, respectively), in order to maximize the specific power output.

The exergy analysis allows to understand the sources of inefficiency of the system. Particularly, it was found that the main exergy destruction was at the condenser, due to the inefficient matching of the heat transfer curves.

Finally, the exergo-economic analysis enabled to calculate the production cost of electricity in the design configuration. The obtained value (29.06 €/kWh) is within the range expected, as confirmed by the literature [1]. The high value is mainly due to the low conversion efficiency of the whole power plant (2.5%), which is thermodynamically hindered by the very low temperature application (evaporator $T=29.5^{\circ}\text{C}$).

The developed analysis confirmed that OTEC power plants could be a feasible solution for the production of electricity, as they are exploiting a green “free” resource, which is potentially inexhaustible. However, significant improvements in performance are necessary to be competitive with other RES. The possible application of zeotropic mixtures, which would allow to design better-matched heat exchangers, looks as a promising development to further reduce the production cost of electricity.

Nomenclature

Ex_t	Exergy rate, (kW)
f_k	Exergo-economic factor
h	Specific enthalpy, (kJ/kg)
LCOE	Levelized cost of electricity (€/kWh)
$\dot{m}$	Mass flow rate, (kg/s)
OTEC	Ocean thermal energy conversion
$\dot{Q}$	Heat rate, (kW)
s	Entropy, (kJ/kgK)
$\dot{W}$	Power, (kW)
$\dot{Z}_k$	Economic impact rate, (€/s)
Greek	
η	Efficiency
γ	Mass flow rates ratio
Subscripts	
C	Condenser
L	Loss
D	Destruction
P	Product
F	Fuel
sp	Specific

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