Off-design performance analysis of a closed-cycle ocean thermal energy conversion system with solar thermal preheating and superheating

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ABSTRACT

This article reports the off-design performance analysis of a closed-cycle ocean thermal energy conversion (OTEC) system when a solar thermal collector is integrated as an add-on preheater or superheater. Design-point analysis of a simple OTEC system was numerically conducted to generate a gross power of 100 kW, representing a base OTEC system. In order to improve the power output of the OTEC system, two ways of utilizing solar energy are considered in this study: (1) preheating of surface seawater to increase its input temperature to the cycle and (2) direct superheating of the working fluid before it enters a turbine. Obtained results reveal that both preheating and superheating cases increase the net power generation by 20\(^\text{e}25\%)\) from the design-point. However, the preheating case demands immense heat load on the solar collector due to the huge thermal mass of the seawater, being less efficient thermodynamically. The superheating case increases the thermal efficiency of the system from 1.9% to around 3%, about a 60% improvement, suggesting that this should be a better approach in improving the OTEC system. This research provides thermodynamic insight on the potential advantages and challenges of adding a solar thermal collection component to OTEC power plants.

1. Introduction

Ocean thermal energy conversion (OTEC) is a renewable energy technology that makes use of the temperature difference between the surface and the depth of the ocean to produce electricity by running a low-pressure turbine. A closed-cycle OTEC employs a refrigerant, such as ammonia, R-134a, R-22 or R-32 as a working fluid to allow its evaporation and condensation using warm and cold seawater, respectively. OTEC has the potential to be adopted as a sustainable, base-load energy source that requires no fossil fuel or radioactive materials, while also causing many fewer environmental impacts than conventional methods of power generation. Several pilot OTEC plants in the order of 10 MW are currently under development by commercial sectors in the US, such as Ocean Thermal Energy Corporation (http://www.otecorporation.com), Makai Ocean Engineering (http://www.makai.com), and Lockheed Martin (http://www.lockheedmartin.com/us/products/otec.html). Recently, Asian countries such as China, Japan, and India have also initiated the construction of OTEC plants in their territories. However, the main technical challenge of OTEC lies in its low energy conversion efficiency due to small ocean temperature differences. Even in the tropical area, the temperature difference between surface and deep water is only 20–25 °C. The thermodynamic efficiency of OTEC is in the order of 3–5% at best, requiring large seawater flow rates for power generation (e.g., approximately 3 ton/s of deep cold seawater and as much warm seawater to generate 1 MW of electrical power [3]).

Since the 1980s, considerable research efforts have been made in two directions to improve the performance of the OTEC system. The first research direction has been targeted towards thermodynamic optimization of Rankine-based cycles for higher efficiencies. Two of the most popular cycles are the Kalina cycles and Uehara cycles, which are generally suited for large-scale OTEC plants on the order of 4 MW or higher. Another research direction is towards the increase of temperature differences...
between the surface and deep seawater by utilizing renewable energy or waste heat sources, such as solar energy [14,15], geothermal energy [16], or waste heat of a nuclear power plant [17]. Among them, solar energy has been considered to be the most appealing renewable energy source that could be integrated with OTEC due to the ever-growing solar technology and its minimal adverse impacts to the environment.

Yamada et al. [14,15] numerically investigated the feasibility of incorporating solar energy to preheat the seawater in OTEC, demonstrating that the net efficiency can increase by around 2.7 times with solar preheating. In addition, recent studies have also suggested the direct use of solar energy for the reheating of the working fluid to enhance the OTEC performance [9,14,15]. These studies have focused on the design of solar-boosted OTEC systems, suggesting the construction of a new power plant operating at a much higher pressure ratio than the conventional OTEC system. However, OTEC power plants demand huge initial construction costs (e.g., ~ $1.6B for a 100 MW OTEC power plant [18]) due to enormous seawater mass flow rates and corresponding heat exchanger and seawater piping sizes. It would be more economically feasible to consider improving OTEC plants by adding solar thermal collection on top of existing power-generating and piping components.

The research presented here aims to examine the system-level effect of integrating solar thermal collection to the power output and efficiency of an existing OTEC power plant. To this end, the study begins with the design-point analysis of a closed-cycle OTEC system with a 100 kW gross power generation capacity. The designed OTEC system is considered as an illustrative base system that allows the thermodynamic analysis of its off-design operation when solar thermal collection is integrated as an additional component. Two methods that make use of solar energy are considered in this paper. First, an add-on solar thermal collector is installed to the system to preheat the surface seawater before entering the evaporator. The second method is directly superheating the working fluid between the evaporator and the turbine with an add-on solar thermal collector. Numerical analysis is conducted to predict the performance change (i.e., net power and efficiency) in the OTEC system when solar collection is integrated as a preheater/superheater. Simulated results are presented to compare the improvement of system performances, in terms of the net power output and the efficiencies, and required collector effective areas between the two methods.

2. Design-point analysis

As shown in Fig. 1, the closed OTEC cycle consists of two heat exchangers (evaporator and condenser), a turbine connected to a generator, and a pump for the working fluid. The heat source for the evaporator is warm seawater at the surface level of the ocean and the heat sink for the condenser is cold seawater (typically pumped out of ~1000 m or deeper in the ocean.) In this study, the temperature of the warm seawater is assumed to be constant at 26 °C, and that of the cold seawater is 5 °C, which are close to the average ocean temperatures in tropical areas [2]. As for the working fluid, difluoromethane (R-32) was chosen over pure ammonia (NH3) owing to its non-corrosive, lower toxic characteristics and better suitability for superheated cycles [19]. Previous research has also shown that R-32 has a smaller vapor specific volume and thus requires a smaller turbine size than when ammonia is used [17]. The pinch point temperature difference is defined as the minimum temperature difference between the working fluid and seawater and set to 2 °C for the evaporator and 1.8 °C for the condenser, respectively, which are similar to Ref. [14]. The vapor quality of the working fluid is assumed to be unity at the exit of the evaporator and zero at the exit of the condenser; neither subcooling nor superheating is allowed during the design-point operation. Table 1 summarizes the design conditions for an OTEC system with a 100 kW gross power output. The proceeding section describes the thermodynamic modeling of each component of the OTEC cycle in detail.

2.1. Heat exchangers (evaporator and condenser)

Two critical parameters in the design-point analysis of the heat exchanger are the overall heat transfer coefficient and surface area. Among potential heat exchanger configurations, the present study has selected a titanium (Ti) shell-and-plate type heat exchanger due to its favorable heat transfer and compact size [20]. In the evaporator, a working fluid is evaporated to saturated vapor by receiving heat from the warm seawater. The energy balance equation at each side of the evaporator can be written as:

\[ Q_E = m_{wf}(h_1 - h_4) = m_{ws}f_{P}(T_{wsi} - T_{wso}) \]  

(1)

under the assumption that seawater is an ideal incompressible fluid. Enthalpy and entropy of the working fluid, which are in general a function of pressure and vapor quality during phase change, were determined from REFPROP — NIST Reference Fluid Thermodynamic and Transport Properties Database [21,22]. It is...
also assumed that the working fluid maintains at the saturation pressure without experiencing pressure loss at the evaporator. Overall heat transfer coefficient and effective surface area of the evaporator is correlated with the heat addition rate as shown in the following equation:

\[ \dot{Q}_E = U_E A_E \Delta T_{lm,E} \]

where \( \Delta T_{lm,E} \) is the logarithmic mean temperature difference across the evaporator expressed as \( \Delta T_{lm,E} = (T_{wsi} - T_{wso}) / \ln((T_{wsi} - T_E)/(T_{wso} - T_E)) \), and the effective thermal conductance \( U_E A_E \) can be approximated as

\[ \frac{1}{U_E A_E} = \frac{1}{h_{wsi} A_E} + \frac{1}{h_{wso} A_E} \]

(3)

It should be noted that the thermal resistance of the Ti plate is ignored since it is extremely small compared to other thermal resistances. The present study implemented the following empirical correlations of the Nusselt number, or correspondingly the convection heat transfer coefficient, for the phase-changing working fluid [23]:

\[ N_{uwf} = 0.023 \Re_l^{0.8} \Pr_l^{0.4} \left[ 1 + 4.863 \left( -\ln \left( \frac{P_{sat}}{P_f} \right) \right) \frac{x}{1-x} \right]^{0.836} \]

(4)

where \( x \) is the mean vapor quality, \( \Re_l \) is the Reynolds number, \( \Pr_l \) is the Prandtl number, \( P_{sat} \) is the saturation pressure, and \( P_f \) is the critical pressure. The heat transfer coefficient of the seawater side is also calculated using the Dittus-Boelter equation for single-phase heat transfer [24]:

\[ N_{uw(s)} = 0.023 \Re_l^{4/5} \Pr_l^{1/3} \]

(5)

For the calculation of the Reynolds number, the equivalent hydraulic diameter is defined as the ratio of four times the cross-sectional channel flow area to the wetted perimeter of the duct. For a shell-and-plate type heat exchanger, the channel flow area is a function of mean channel spacing inside the heat exchanger and horizontal length of the plates [25].

The energy balance equation at the condenser is basically the same as the evaporator and can be written as

\[ \dot{Q}_C = \dot{m}_{uwf} (h_2 - h_3) = m_{cs} c_p (T_{cso} - T_{csl}) \]

(6)

Likewise, the effective thermal conductance of the condenser is correlated with the heat transfer rate as

\[ \dot{Q}_C = U_C A_C \Delta T_{lm,C} \]

(7)

where \( \Delta T_{lm,C} \) is the logarithmic mean temperature difference across the condenser, i.e., \( \Delta T_{lm,C} = (T_{cso} - T_{csl}) / \ln((T_C - T_{csl})/(T_C - T_{cso})) \). The effective thermal conductance of the condenser can be determined using the same equations at the evaporator, i.e., Eqs. (4) and (5).

2.2. Pumps

After condensed, the working fluid is pressurized and pumped through the inlet of the evaporator. The energy balance equation for the working fluid pump can be written as

\[ W_{P,uw} = \dot{m}_{uwf} v_4 (P_4 - P_3) / \eta_{uw} \]

(8)

where \( \eta_{uw} \) is the efficiency of the working fluid pump and is assumed to be 75%. Some of the power generated by the OTEC cycle is consumed to pump the warm and cold ocean water. The power required to run the seawater pumps can be simply calculated using the following equation [7]:

\[ W_{P,uw(s)} = \dot{m}_{uw(s)} g \Delta H / \eta_{uw(s)} \]

(9)

where \( g \) is the gravitational acceleration, and \( \eta_{uw(s)} \) is the efficiency of the seawater pump and is assumed to be 80%. The head difference \( \Delta H \) of each seawater pump is obtained from the previous work [7], which estimated the head difference from the friction and bending losses in the pipes.

2.3. Turbine

The vaporized working fluid rotates the blades of a low-pressure turbine while expanding adiabatically. Vapor pressure at the exit of the turbine is set equal to the saturation pressure at condensation temperature of the condenser, i.e., \( P_2 = P_{sat}(T_c) \). The power output from the turbine connected with the generator, or the turbine-generator power, can be written as

\[ W_{T-G} = \dot{m}_{uwf} \eta_T \eta_G (h_1 - h_{2s}) \]

(11)

Here, \( h_{2s} \) is the isentropic enthalpy at the exit of the turbine and can be calculated by \( h_{2s} = h_{sf} + x_{2s} h_{fg} \), where \( h_{sf} \) and \( h_{fg} \) are the saturated liquid enthalpy and the enthalpy of vaporization at \( P_2 \), respectively. The isentropic quality \( x_{2s} \) can be expressed as

Table 1

<table>
<thead>
<tr>
<th>Conditions and assumptions for the design of an OTEC system with a 100-kW gross power capacity.</th>
<th>Symbol</th>
<th>This study</th>
<th>Yamada [14]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seawater inlet temperature (°C)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Surface seawater</td>
<td>( T_{wsi} )</td>
<td>26</td>
<td>25.7</td>
</tr>
<tr>
<td>Deep seawater</td>
<td>( T_{csi} )</td>
<td>5</td>
<td>4.4</td>
</tr>
<tr>
<td>Pinch point temperature difference (°C)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>@ Evaporator</td>
<td>( \Delta T_{\text{E}}^{\text{pp}} )</td>
<td>20</td>
<td>1.2</td>
</tr>
<tr>
<td>@ Condenser</td>
<td>( \Delta T_{\text{C}}^{\text{pp}} )</td>
<td>1.8</td>
<td>1.3</td>
</tr>
<tr>
<td>Vapor quality</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>@ Evaporator exit</td>
<td>( x_1 )</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>@ Condenser exit</td>
<td>( x_3 )</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Component efficiency (%)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine</td>
<td>( \eta_T )</td>
<td>–</td>
<td>80</td>
</tr>
<tr>
<td>Generator</td>
<td>( \eta_G )</td>
<td>95</td>
<td>90</td>
</tr>
<tr>
<td>Working fluid pump</td>
<td>( \eta_{uw} )</td>
<td>75</td>
<td>75</td>
</tr>
<tr>
<td>Seawater pumps</td>
<td>( \eta_{uw(s)} )</td>
<td>80</td>
<td>80</td>
</tr>
<tr>
<td>Overall heat transfer coefficient (kW/m² K)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporator</td>
<td>( U_E )</td>
<td>–</td>
<td>4.0</td>
</tr>
<tr>
<td>Condenser</td>
<td>( U_C )</td>
<td>–</td>
<td>3.5</td>
</tr>
<tr>
<td>Seawater specific heat capacity (kJ/kg K)</td>
<td>( c_p )</td>
<td>4.025</td>
<td>–</td>
</tr>
<tr>
<td>Seawater density (kg/m³)</td>
<td>( \rho_s )</td>
<td>1025</td>
<td>–</td>
</tr>
</tbody>
</table>
\[ x_2 = (s_1 - s_2)/s_2 \text{ by considering that the entropy at point 2 is the same as point 1.} \]

A radial inflow turbine is typically employed for the OTEC cycle due to its high isentropic expansion efficiency and good moisture erosion resistance [26]. The turbine efficiency for a radial turbine is defined as [27]

\[ \eta_T = 0.87 - 1.07(n_s - 0.55)^2 - 0.5(n_s - 0.55)^3 \]  \hspace{1cm} (12)

where \( n_s \) is the specific speed, a nondimensional parameter that characterizes the turbine performance. For the radial-inflow turbine, \( n_s \) is defined as [27,28]:

\[ n_s = \frac{2\pi N m_{\text{infl}}^{1/2}}{60 \rho_{\text{fluid}} \Delta h_T^{3/4}} \]  \hspace{1cm} (13)

where \( N \) is the rotational speed (rpm), \( \rho_{\text{fluid}} \) is the density of the working fluid, and \( \Delta h_T \) is the enthalpy drop (J/kg) between the turbine inlet and outlet. Another design parameter that defines the rotor tip speed is the total-to-static velocity ratio, defined as [28]:

\[ \nu_s = \frac{V_{\text{tip}}}{\sqrt{2 \Delta h_T}} \]  \hspace{1cm} (14)

where \( V_{\text{tip}} \) is the rotor tip speed. Once the rotor tip speed is determined, the rotor tip radius can be calculated using

\[ r_{\text{tip}} = \frac{V_{\text{tip}}}{(2\pi/60)N} \]  \hspace{1cm} (15)

### 2.4. Results of design-point analysis

The design-point analysis of the OTEC system producing a turbine-generator power of 100 kW was numerically conducted using MATLAB. Since the governing equations at each component are strongly coupled, they were solved iteratively with an initial guess of the outlet seawater temperatures (i.e., 23 °C at the evaporator and 8 °C at the condenser exits, respectively). From the preset pinch point temperature differences, the saturation temperature and pressure of the working fluid at the evaporator and condenser are determined by using REFPROP, which also provides the enthalpy at each point (i.e., \( h_1, h_2, h_3, \) and \( h_4 \)) as well. Once the enthalpy values are determined, the energy balance equations at the evaporator, condenser, and turbine, i.e., Eqs. (1), (6) and (11) allow the calculation of the mass flow rates of warm seawater, cold seawater, and working fluid, respectively, along with the heat transfer rates at the evaporator and condenser. It should be noted that in the design of the OTEC system, the most stringent design condition is the mass flow rate of the deep seawater, as a tremendous cost is required to construct a pipeline reaching a ~1000 m depth in ocean. Thus the present study identified a design point as the operation condition requiring the minimum mass flow rate of the deep seawater to generate 100 kW of \( W_{\text{T,G}} \), although this design point may compromise the system efficiency. The previous OTEC studies suggested that the mass flow ratio of the deep seawater to the surface seawater \( m_{\text{CS}}/m_{\text{WS}} \) should be between 0.5 and 1 for optimal performance [1,6], which was used as a criterion for the validation of the obtained results. After the cold seawater mass flow rate is specified, the net power output is obtained by calculating the turbine and pump powers:

\[ W_N = W_{\text{T,G}} - W_{\text{P,WS}} - W_{\text{P,CS}} \]  \hspace{1cm} (16)

which allows the calculation of the net thermal efficiency, i.e., \( \eta_{\text{th}} = W_N/Q_E \). Design parameters for the evaporator and condenser, such as the effective heat transfer coefficient and surface area can also be obtained.

Table 2 compiles the determined design parameters of the OTEC system that generates a 100 kW turbine-generator power output. While our results are in overall good agreement with Ref. [14] that designed the same-scale OTEC system, there are noticeable differences in some design parameters. It should be noted that the present study chose R-32 as a working fluid while Ref. [14] used NH3. Since the latent heat of vaporization for R-32 (218.59 kJ/kg-K at 290 K) is almost five times smaller than that for NH3 (1064.38 kJ/kg-K at 290 K), more mass flow rate is required to generate the same power when R-32 is used as a working fluid. The determined \( m_{\text{CS}}/m_{\text{WS}} \) is 0.85, falling into the acceptable range. It should be noted that the overall heat transfer coefficients computed in the present study agree reasonably well with those in Refs. [14], which are experimentally obtained values from Ref. [20]. This suggests that Eqs. (4) and (5) are valid correlations and can be used for different flow conditions of seawater and working fluid at off-design operations. The estimated net power generation is 68 kW, indicating that 32% of the turbine-generator power is consumed by pumps. The corresponding net thermal efficiency is estimated at 1.9%, which is slightly lower than Ref. [14] mainly due to the bigger heat transfer rate at the evaporator. Fig. 2 shows the isentropic efficiency of the turbine as a function of the rotational speed when the turbine is designed to meet the system requirements. For the working fluid mass flow rate of 12.3 kg/s and the enthalpy drop of 10.6 kJ/kg, the turbine efficiency curve demonstrates a polynomial trend with a maximum of ~87% at 8800 rpm. However, the turbine efficiency is determined to be 80.6% at the design-point operation, corresponding to 12,500 rpm, at which the mass flow rate of the deep seawater meets the design requirement. As mentioned above, the mass flow rate of the deep seawater is a more stringent design condition than the turbine efficiency for the construction and operation of OTEC plants. Moreover, a turbine designed at higher rotational speeds is more compact and guarantees a better

### Table 2

Design-point analysis results for the 100-kW gross power OTEC system. The obtained results are favorably compared to the results of Yamada et al. [14].

<table>
<thead>
<tr>
<th>Symbol</th>
<th>This study</th>
<th>Yamada [14]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seawater outlet temperature (°C)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Warm seawater</td>
<td>( T_{\text{WS}} )</td>
<td>22.83</td>
</tr>
<tr>
<td>Cold seawater</td>
<td>( T_{\text{CS}} )</td>
<td>8.61</td>
</tr>
<tr>
<td>Mass flow rate (kg/s)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Warm seawater</td>
<td>( m_{\text{WS}} )</td>
<td>288.6</td>
</tr>
<tr>
<td>Cold seawater</td>
<td>( m_{\text{CS}} )</td>
<td>246.6</td>
</tr>
<tr>
<td>Working fluid</td>
<td>( m_{\text{ref}} )</td>
<td>12.3</td>
</tr>
<tr>
<td>(R-32)</td>
<td>(NH3)</td>
<td></td>
</tr>
<tr>
<td>Evaporator</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporation temperature (°C)</td>
<td>( T_E )</td>
<td>20.83</td>
</tr>
<tr>
<td>Evaporation pressure (kPa)</td>
<td>( P_{\text{ref}} )</td>
<td>1509</td>
</tr>
<tr>
<td>Heat transfer rate (kW)</td>
<td>( Q_{\text{E}} )</td>
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</tr>
<tr>
<td>Overall heat transfer coefficient (kW/m² K)</td>
<td>( U_{\text{E}} )</td>
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<td>Surface area (m²)</td>
<td>( A_E )</td>
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<tr>
<td>Condenser</td>
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<td>Condensation temperature (°C)</td>
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<td>Heat transfer rate (kW)</td>
<td>( Q_C )</td>
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<td>Overall heat transfer coefficient (kW/m² K)</td>
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<tr>
<td>Surface area (m²)</td>
<td>( A_C )</td>
<td>334</td>
</tr>
<tr>
<td>Power output/consumption (kW)</td>
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<td></td>
</tr>
<tr>
<td>Turbine-generator power output</td>
<td>( W_{\text{T,G}} )</td>
<td>100.0</td>
</tr>
<tr>
<td>Working fluid pump power consumption</td>
<td>( W_{\text{P,WS}} )</td>
<td>(6.2)</td>
</tr>
<tr>
<td>Warm seawater pump power consumption</td>
<td>( W_{\text{P,CS}} )</td>
<td>(8.9)</td>
</tr>
<tr>
<td>Cold seawater pump power consumption</td>
<td>( W_{\text{P,CS}} )</td>
<td>(16.9)</td>
</tr>
<tr>
<td>Net power output</td>
<td>( W_E )</td>
<td>68.9</td>
</tr>
<tr>
<td>Turbine isentropic efficiency (%)</td>
<td>( \eta_T )</td>
<td>80.6</td>
</tr>
<tr>
<td>Net thermal efficiency (%)</td>
<td>( \eta_{\text{th}} )</td>
<td>1.9</td>
</tr>
</tbody>
</table>
performance when the enthalpy drop across the turbine is demanding [28]. The rotor tip speed and the rotor tip radius are determined to be 102.3 m/s and 15.6 cm, respectively.

3. Off-design performance with solar preheating/superheating

Since the closed-cycle OTEC system is based on the Rankine thermodynamic cycle, its net power generation and thermal efficiency can be improved by increasing the temperature difference between the heat source and heat sink [15]. This study considers two different ways to improve the performance of the OTEC system with solar energy, i.e., preheating of the warm seawater and superheating of the working fluid using solar energy: see Fig. 3. When the solar preheater/superheater is integrated with the OTEC system, the system operation shifts from its design point to find a new state of balance. For the off-design point calculation, an iterative algorithm was developed to revisit the energy balance equations at each component and to find out a converged solution. For the heat exchangers, the heat transfer coefficients of each fluid and the resultant heat transfer rate were calculated for different operational conditions (i.e., different mass flow rates and inlet conditions) during off-design operations. However, the geometrical parameters of the OTEC system, such as the effective surface areas of the heat exchangers and the rotor tip radius of the turbine, remained as the designed values.

The net thermal efficiency for the solar preheating/superheating OTEC system is determined by considering the additional solar energy input, i.e., \( \eta_{th} = W_N/(Q_S + Q_S) \), where \( Q_S \) is the absorbed solar energy. However, since solar preheating/superheating does not consume exhaustible energy sources, such as fossil fuels, the conventional net thermal efficiency may underestimate the OTEC efficiency under off-design operation conditions. Instead of simply comparing the net power generation to the total heat input, more attention should be given to the increase of useful net power generation out of the total power increase when consuming additional solar energy. To address this issue, Wang et al. [9] suggested the net cycle efficiency defined as

\[
\eta_{NC} = \frac{W_N}{\dot{W}_{T\rightarrow C}}
\]  

which compares the net power generation of the system to the turbine-generator power output. However, it should be noted that since the net cycle efficiency compares the off-design performance of the system to its design-point; and it should not be used to compare between different energy conversion systems.

Since the solar collector for the OTEC system does not need a high concentration of solar irradiation, we chose a CPC (compound parabolic concentrator) type solar collector as a solar thermal preheater/superheater in this study. CPC-type solar collectors provide economical solar power concentration for low- to medium-pressure steam systems, allowing high collector efficiency in the moderate temperature range (i.e., 80–150 °C) [29,30]. They also can effectively collect diffuse radiation, especially at lower concentration ratios, producing satisfactory performance even in cloudy weather [30,31]. The efficiency of the CPC solar collector can be written as [32]

\[
\eta_S = F_i \left[ \eta_0 - \frac{U_i \Delta T}{G_i R} \right]
\]  

where \( F_i \) is the generalized heat removal factor, \( \eta_0 \) is the optical efficiency and assumed to be 80%, \( U_i \) is overall thermal loss coefficient, \( \Delta T \) is the temperature difference between the inlet heat transfer fluid temperature and the ambient temperature, \( G_i \) is total solar irradiation and \( R \) is the concentration ratio. \( F_i \) is a function of boiling status and concentration ratio and is taken from the data available in literature [32]. \( U_i \) is a variable that correlates with many factors led by temperature and is taken from the measured data for
a similar CPC type solar collector [33]. Fig. 4 shows the collector thermal efficiency of a typical CPC solar collector as a function of \( \Delta T/G_r \) (m²·K/W) when \( n_0 \) is set to 80% and the concentration ratio \( R \) to 3, a typical value that would provide a high energy gain [33]. The solar irradiation is assumed to be 500 W/m², which is the approximated daytime average in Honolulu, Hawaii during the summer [34]. From these given conditions, the solar collector efficiency was determined to be 65%, which is used to estimate the required collector effective area from the following energy balance equation:

\[
A_S = \frac{\dot{m} \cdot \Delta h_{\text{ws}(wf)}}{q_S + G_r}
\]  

(19)

Here, \( \dot{m} \) is the mass flow rate and \( \Delta h \) is the enthalpy change at the preheater/superheater. The subscript ws(wf) indicates the warm seawater for preheating and the working fluid for superheating.

3.1. Solar preheating of seawater

As shown in Fig. 3(a), an add-on solar thermal preheater is installed next to the evaporator of the pre-designed OTEC system. The solar preheater has its own heat transfer fluid (typically synthetic/hydrocarbon oils or water [35]) that indirectly delivers solar energy to the seawater via the auxiliary heat exchanger. The preheated surface seawater will alter the operation condition of the turbine, allowing more energy extraction from the working fluid. The off-design operation of the turbine should be fully characterized to understand the off-design performance of the OTEC system. Fig. 5 shows the isentropic efficiency change of the turbine as a function of \( \Delta T_{\text{wsi}} \), the offset of the warm seawater inlet temperature from its design point, for several turbine rotational speeds. Generally at high rotational speeds, the isentropic efficiency of the turbine reaches a maximum and gradually decreases as \( \Delta T_{\text{wsi}} \) increases. At lower rotational speeds, on the other hand, the turbine efficiency monotonically decreases without having a maximum. It should be noted that preheating the warm seawater increases the turbine-generator power output although the turbine efficiency decreases. In the present study, we fixed the turbine performance at the design point (i.e., 80.6% at 12,500 rpm) during the off-design operation of the OTEC system.

Fig. 6 shows the simulation results of the OTEC system when preheating the ocean water. In Fig. 6(a), the net power output slightly increases as the solar power absorption at the preheater increases up to 3000 kW, and substantially increases with the further increase of the solar power absorption. The existence of these two regimes is mainly due to the control algorithm selected in this study. The priority in the control algorithm is to maintain the turbine efficiency at the design point. Thus, as can be seen in Eq. (12), \( n_s \) should be constant under the solar preheating operation. Since the solar preheating enhances the enthalpy drop across the turbine, \( \Delta h_T \) in Eq. (13), the mass flow rate of the working fluid should also increase accordingly to keep \( n_s \) constant. On the other hand, the mass flow rate of the warm seawater should be reduced to make a good balance between the energy demand of the working fluid and the energy input from the solar preheater. Any excessive solar power absorption at the solar preheater would lead to the waste heat instead of being used to evaporate the working fluid at the evaporator, which is not desirable for the cost-effective and eco-friendly operation of the OTEC system. Fig. 6(b) clearly shows the continuous decrease of the warm seawater mass flow rate until the solar power absorption reaches 3000 kW. This decrease of the warm seawater mass flow balances with the temperature increase of the preheated seawater to provide almost the same energy to the turbine as the design point. The slight increase of the net power generation is mainly attributed to the slight increase of the working fluid mass flow rate, which is required to run the turbine at its design point. In this region, as shown in Fig. 7(a), the net thermal efficiency remains almost the same as the design point, indicating that the absorbed solar energy is effectively used to generate power in the turbine.

When the solar thermal absorption reaches 3000 kW, the preheated seawater temperature reaches a point at which the net power generation cannot increase any further unless the working fluid is superheated. At this point, the surface seawater mass flow rate has been reduced to 48.5 kg/s, which is then fixed to allow the superheating of the working fluid. In order to run the turbine at the design point, i.e., constant \( n_s \) in Eq. (13), the mass flow rate of the
working fluid should increase accordingly to match the increase of $\Delta h_f$ due to the superheating of the working fluid; see Fig. 6(b). These increases in both the mass flow rate of the working fluid and its enthalpy drop across the turbine drastically enhance the net power generation in the second regime, which increases up to 83 kW, or ~25% compared to the design point, as the solar absorption reaches 8500 kW. The net thermal efficiency shown in Fig. 7(a) decreases to ~1%, indicating that the absorbed solar energy is not as efficiently used in the OTEC system. However, the net cycle efficiency shows an improvement from 71% to 76%. Although the working fluid pump consumes more power according to the increasing mass flow rate of the working fluid, solar preheating produces more useful net power out of the gross power generation.

The partial use of solar energy from the excessive preheating is manifested by the increasing outlet temperature of the seawater at the evaporator shown in Fig. 7(b). In the first region, the warm seawater outlet temperature is lower than the design point, indicating that more thermal energy is transferred from the seawater and converted to the power generation. However, further preheating drastically increases the exit temperature of warm seawater, which reaches up to ~50 °C when the solar power absorption increases to 8500 kW. This inefficient use of absorbed solar energy is because of the limited surface area of the evaporator and condenser, which are originally optimized at the 100 kW turbine-generator power capacity. Unless this hot seawater is used somewhere else, returning it back to the ocean will cause adverse environmental and ecological impacts. The hot seawater could be used to reheat the working fluid by installing the second turbine, which can extract more work out of the working fluid vapor before it enters the condenser. However, this study focuses on the effect of solar preheating on the existing OTEC system and thus did not
consider further modification of the system beyond the installation of the solar preheater. Fig. 7(b) also shows the temperature difference between the incoming seawater and the exiting working fluid vapor, i.e., $T_{\text{wsi}} - T_1$ at the evaporator. In the first region, this temperature difference keeps increasing because the control algorithm does not allow the superheating of the working fluid; thus $T_1$ in this range is the evaporation temperature of the working fluid. However, in the second regime, the superheating of the working fluid reduces $T_{\text{wsi}} - T_1$ below the preset pinch-point temperature difference. Thus the practical limit of the solar preheating comes from the evaporator unless the evaporator is replaced with a bigger one.

Fig. 7(c) shows the required effective area of the solar collector for preheating. When plotted as a function of the net power output, the collector effective area has an abrupt jump to ~2000 m² when the net power increases from 68 kW to 70 kW, or only a ~3% increase from the design point. A larger collector effective area should be installed in order to take further advantage of the solar preheating; for example, nearly 6000 m² of the collector area is needed to increase the net power of the system by ~20%. The collector area could be significantly reduced by improving the design of the collector or using a different fluid that has a higher solar absorption coefficient. Previous studies revealed that mixing nanoparticles in the fluid can enhance the light absorptance to almost 100% above a certain nanoparticle concentration [36–39]. This enhancement of the light absorbance directly affects the solar collector efficiency, e.g., ~10% increase of the efficiency when aluminum nanoparticles are suspended in water [37].

3.2. Superheating of working fluid

Evidently from the simulation results, preheating the ocean water requires a huge amount of solar energy due to its massive flow rate and high specific heat capacity. This demands a large effective area of the solar collector. On the other hand, difluoromethane (R-32) has a significantly lower specific heat, and its mass flow rate is much smaller than that of the warm seawater. Therefore, direct superheating of the working fluid by solar energy may improve the OTEC cycle with much less effective solar collector area. As shown in Fig. 3(b), an add-on solar thermal collector is installed in between the evaporator and the turbine of the OTEC system to superheat the working fluid. The solar superheater, same as in the preheating case, has its own heat transfer fluid and provides the heating to the working fluid via the auxiliary heat exchanger.

Fig. 8 shows the simulation results for solar superheating. The net power generation increases from 68 kW to 85 kW, enhanced by 25% from design-point, as the solar power absorption increases. This enhancement is mainly attributed to the improvement of the net thermal efficiency from 1.9% to 3%, indicating that solar superheating generates more useful net power. The net cycle efficiency also increases from 71% to 76%. It should be noted that the net thermal efficiency could be further increased until the critical temperature of R-32 is reached at around 78 °C. However, this extreme superheating will cause the working fluid vapor to remain superheated at the exit of the turbine, undesirably requiring more mass flow rate of the deep seawater at the condenser. The present study simulates only the sub-critical superheating case, where the vapor quality of the working fluid at the exit of the turbine remains around unity.

Fig. 9(a) shows the mass flow rate and the turbine inlet temperature of the working fluid as a function of the solar power absorption. The increase of the solar power absorption yields a higher temperature of the working fluid at the inlet of the turbine, resulting in a greater enthalpy drop across the turbine. As discussed in the preheating case, the mass flow rate of the working fluid and the enthalpy drop are correlated in Eq. (13), suggesting that the system should have a larger mass flow rate of the working fluid to maintain the turbine operation at its design point. Fig. 9(b) shows the required collector effective area as a function of net power generation for the superheating case. When compared to the preheating case, much less collector effective area is required for the superheater to obtain the almost same amount of net power enhancement. For example, around 1100 m² of effective collector area is required to enhance the net power by 20% with solar superheating, while more than 6000 m² is required to generate the same amount of power using solar preheating: more than five times of the solar collector area should be constructed. This result strongly suggests that the solar superheater may be more beneficial in improving the thermodynamic performance of the OTEC system although it requires greater care to prevent a leakage of the working fluid during long-term operations [15].

4. Conclusions

The present study reports the effects of the solar thermal preheating/superheating to the performance of a closed-cycle OTEC system. To this end, a closed-cycle OTEC system generating a 100 kW turbine-generator power was designed and used as a basic system. The designed system, when using R-32 as its working fluid, produces 68 kW of net power with a net thermal efficiency of 1.9%
and the net cycle efficiency of 71%. Off-design performance analysis of the designed OTEC system was conducted when a CPC-type solar collector was added as the solar preheater of warm seawater or the superheater of the working fluid. Simulation results demonstrate that both preheating/superheating cases increase the net power generation up to 20–25% from the design-point. However, superheating of the working fluid requires up to 5 times less solar collector area compared to solar preheating. The superheating case also increases the thermal efficiency of the system from 1.9% to ~3%, about a 50% improvement, suggesting that it should be a better approach in improving the OTEC system. The obtained results will provide insights on the thermodynamic perspective when combining sustainable energy conversion technologies — ocean thermal energy conversion and solar thermal energy conversion — to improve the system performance.

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References


