Analysis of optimization in an OTEC plant using organic Rankine cycle

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This study quantified the effects of evaporation temperature, condensation temperature, and the inlet-and outlet-temperature differences of deep cold seawater and warm seawater on the performance of an ocean thermal energy conversion (OTEC) plant using an organic Rankine cycle (ORC), and also investigated the optimal operations required for the performance. A finite-temperature-difference heat transfer method is developed to evaluate the objective parameter, which is the ratio of net power output to the total heat transfer area of the heat exchanger in the system, and R717, R600a, R245fa, R152a, and R134a were used as the working fluids. The optimal evaporation and condensation temperatures were obtained under various conditions for maximal objective parameters in an OTEC system.

The results show that R717 performed optimally in objective parameter evaluation among the five working fluids, and that R600a performed better than other fluids in thermal efficiency analysis. The optimal seawater temperature differences between the inlet and outlet of the evaporator and condenser are proposed. Furthermore, the influences of inlet temperatures of warm and cold seawater in the ORC are presented for an OTEC plant. The simulation results should enable the performance of an ORC system to be compared when using various organic working fluids.

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1. Introduction

Saving energy and reducing carbon dioxide emissions have become increasingly critical aspects of energy usage because of concerns regarding energy shortage, global warming, and environmental pollution. Thus, researchers have extensively investigated approaches to use renewable and sustainable energy sources effectively. The ocean thermal energy conversion (OTEC) process uses the disparity in temperature between the warm seawater on the ocean surface and the deep cold seawater to operate a Rankine cycle system for producing electrical power without consuming fuel or emitting carbon [1–3]. Although using ocean thermal energy has enormous potential and the OTEC plants have a small environment impact, the low net efficiency of OTEC resulting from the lower temperature differences between surface seawater and cold seawater restricts the implementation of this technology [4].

To improve the thermal efficiency of OTEC, suitable working fluids for use in the Rankine cycle must be identified. Ammonia, which is named R717, was used in the OTEC system widely in the past because of its excellent thermodynamic properties. Uehara et al. [5,6] investigated the major components of an OTEC plant theoretically and experimentally. In their simulation results, the temperatures of warm and cold seawater were 26 °C and 4 °C, respectively, for optimizing a 100-MW OTEC system. Uehara et al. also reported that R717 was one of the suitable working fluids for a closed-Rankine-cycle OTEC plant. Using R717 as the working fluid, Yeh et al. [7] studied theoretically the effects of temperature and flow rate of cold seawater on the net output of an OTEC plant. They concluded that the network has a maximal output at a specific cold seawater flow rate.

To improve efficiency and to reduce system costs, the organic Rankine cycle (ORC) is used with working liquids that have a low boiling temperature. To identify suitable working fluids for an ORC, Chen et al. [8] and Wang et al. [9] investigated the thermodynamic performances of ORC by using various working fluids to convert low-grade heat. Sun et al. [10] optimized numerically the design for an ORC in OTEC with R717 and R134a. The exergy-analysis mode was used to evaluate the maximal net output under various warm water mass-flow rates and evaporation temperatures. However, in the calculation, the overall heat transfer coefficient remained unchanged and the work consumed in pumping seawater was neglected. From the first law of thermodynamics, a high heat-source temperature would increase the pressure and enthalpy of the working fluid at the turbine inlet. To increase the thermal efficiency of the OTEC system, solar heat energy [11,12] and the waste...
heat of nuclear power condenser [13] have been applied to increase the temperature of warm seawater.

A 100-kW OTEC pilot plant, in which R22 was used the working fluid, was constructed on-land for demonstration in the Republic of Nauru in 1981 [14]. The system operated between the warm surface water and a cold seawater source of 5–8 °C at a depth of 500–700 m, with a temperature difference of 20 °C. An experimental study measured the temperature and pressure before and after each component of a demonstration OTEC system was reported by Faizal and Ahmed [15]. According to the report, a maximum thermal efficiency of approximately 1.5% was obtained in the OTEC system when R134a was used as the working fluid. They also determined that both the thermal efficiency and the power output of the system increased upon increasing the operating temperature difference, which is the temperature difference between warm and cold seawater. Wang et al. [16] investigated a regenerative ORC experimentally in which R123 was used as working fluid, and reported that the performance of basic ORC system and the regenerative system were 6.15% and 7.98%, respectively, with a 130 °C geothermal heat source. These results showed that the regenerator in the ORC system improved by 1.83% as the power output was 6 kW.

From the points of view of performance and economy, finding the optimal operating temperature of the heat exchanger is critical for making the heat exchanger as small as possible when using an available temperature difference of only approximately 20 °C. The heat transfer performance in exchangers of the Rankine cycle has been considered widely to evaluate OTEC systems from an economic standpoint. An objective function, which represented the ratio of total heat transfer area to the net power output of the OTEC system, was presented by Ganic and Moeller [17], who analyzed the optimization for a 1-MW OTEC power system operated in a simple closed Rankine cycle with R717 being used as the working fluid.

To evaluate the influence of the heat exchanger on the power output, Nakaoka and Uehara [18] tested the performance of a shell-and-plate-type evaporator for OTEC plants. Moore and Martin [19] proposed a method to determine the minimal heat transfer area for
an OTEC system that involves generating hydrogen from a pure-ammonia working fluid. Baik et al. [20] simulated the power optimization in ORC by using R125, R134a, R245fa, and R152a; in their analysis, the heat-source and heat-sink temperatures were assumed to be 100 °C and 20 °C, respectively. The total overall conductance, which is the product of the overall heat transfer coefficient and the heat transfer area, of the evaporator and condenser ranged from 20 to 80 kW/K. The results revealed that the R125 trans-critical cycle performed better than the other working fluids, because the total overall conductance was more than 35 kW/K. However, the variations of the heat transfer coefficient and the area of the heat exchangers were not described further.

Few studies have analyzed the optimal operational conditions with respect to working fluids while considering both maximal net power output and minimal heat transfer area. The aim of this study is to investigate the performance of the objective parameters that represent the ratio of net power output to heat transfer area for an ORC system with various evaporating and condensing temperatures. In this study, the preliminary principles for selecting working fluids are the environmental considerations of zero ozone depletion potential and low global warming potential. From the first law of thermodynamics and the heat transfer theory, the maximal objective parameters and their corresponding optimal condensing and evaporating temperatures are obtained for using R134a, R152a, R245fa, R600a, and R717 as working fluids.

2. Cycle description and analysis

The ORC in an OTEC plant consists mainly of a pump, an evaporator, a turbine, and a condenser, as shown in Fig. 1(a). In the evaporator, the working fluid absorbs heat that is transferred from the warm seawater and reaches saturation temperature, after which the working fluid continues to be heated and thereby becomes a saturated vapor at the outlet of the evaporator. The vapor expands while passing through the turbine and produces power because of the pressure difference of evaporation and condensation. The low-pressure vapor then enters the condenser cooled by cold deep seawater. After condensation, the liquid working fluid is pumped back into the evaporator to complete the cycle. In the OTEC

![Fig. 3. Comparisons of \( W_{\text{net}} \) and previous work [9] under various warm seawater flow rates for R717 in an OTEC system.](image)

![Fig. 4. Dependence of (a) \( W_{\text{net}} \), (b) \( A_t \), and (c) \( \gamma \) on evaporation temperatures at \( T_{\text{con}} = 10.7 \degree C \) and \( \Delta T_w = 2.1 \degree C \).](image)
system, cold and warm seawater pumps are installed to supply heating and cooling sources to maintain the power cycle. Fig. 1(b) presents the temperature and entropy relationship of the ORC system in an OTEC plant. The three wet fluids analyzed in this study, which have negative slopes of saturation curve in the T–s diagram, are R134a, R152a, and R717. In the turbine outlet, the wet fluids saturate with little liquid mixing. In this study, R245fa and R600a are dry fluids with a positive slope of the saturation vapor fluids. In this study, R245fa and R600a are dry fluids with a positive slope of the saturation vapor fluids. In this study, R245fa and R600a are dry fluids with a positive slope of the saturation vapor fluids. In this study, R245fa and R600a are dry fluids with a positive slope of the saturation vapor fluids.

In the evaporator, the heat flow rate is calculated as

\[ Q_{\text{eva}} = m_i(i_2 - i_1) \]  

where \( i_1, i_2 \) are the enthalpies of the working fluid at the inlets of the evaporator and the turbine, respectively. In the turbine, the power output from the working fluid can be shown as

\[ W_{\text{out}} = m_t(i_3 - i_2) / \eta_t \]  

where \( i_3 \) is the enthalpy of the working fluid at the inlet of the condenser. In the condenser, the heat flow rate is expressed as

\[ Q_{\text{con}} = m_t(i_3 - i_4) \]  

The power consumed by the working fluid pump can be calculated as

\[ W_p = m_{\text{vis}}(p_1 - p_4) / \eta_p \]  

### 3. Solution method

A shell-and-tube heat exchanger is designed for the evaporator and condenser. In the evaporator, the heat transfer rate between the working fluid and warm seawater can be written as

\[ Q_e = U_e A_e F \Delta T_{\text{mean},e} \]  

where \( F \) is a correction factor for the heat exchangers, and \( \Delta T_{\text{mean},e} \) is the logarithmic mean temperature difference (LMTD) between warm seawater and working fluids in the evaporator and is given by

\[ \Delta T_{\text{mean},e} = \frac{(T_{\text{wsi}} - T_{\text{i0}}) - (T_{\text{wso}} - T_{\text{f1}})}{\ln[(T_{\text{wsi}} - T_{\text{i0}})/(T_{\text{wso}} - T_{\text{f1}})]} \]  

where \( T_{\text{wsi}} \) and \( T_{\text{wso}} \) are the inlet and outlet temperatures of warm seawater, respectively, and \( T_{\text{i0}} \) and \( T_{\text{f1}} \) are the working fluid inlet and outlet temperatures. The overall heat transfer coefficient of the heat exchanger of ORC is defined by

\[ U = \frac{1}{(1/h_o) + (A_o/h_1) + (A_0/h_2)(1/h_1)} \]  

where \( h_o \) and \( h_1 \) are the heat transfer coefficients of working fluid and seawater, respectively, \( A_o \) represents the total outside surface area of the tubes. In the evaporator, the dimensional empirical expression for nucleate boiling heat transfer coefficient of the working-fluid side [21].

\[ h_o = 55 \Pr^{0.12} \ln(R_p)^{-0.4343} \ln \Pr^{-0.55} M^{-0.5} (q/0.67)^{0.67} \]  

where \( M \) is the molecular weight of the working fluid, \( q \) is the heat flux of the tube, and \( R_p \) is set to 10 \( \mu \)m for surface roughness of the tube. The heat transfer coefficient of the water side of the heat exchanger can be calculated by the correlation of Dittus–Boelter for 6000 \( < \Re < 10^5 \) and 0.5 \( < \Pr < 120 \) [22].

\[ h_t = 0.023 \Re^{0.8} \Pr^{0.4} \left( \frac{k_w}{D_h} \right) \]  

where \( n = 0.4 \) is for the condenser and \( n = 0.3 \) is for the evaporator. The corresponding heat transfer coefficient of the tube wall is calculated as

\[ h_t = \frac{2 \pi k_t L_t}{\ln(D_o/D_i)} \]  

Similarly, the LMTD of condenser can be calculated from Eq. (6). For the working-fluid side around the horizontal tubes, the correlation of the average heat transfer coefficient for the film condensation is applied [23].

\[ h_o = 0.729 \left( \frac{g \eta (\rho_l - \rho_g) k_f^3 \mu^4}{\mu_l (T_{\text{sat}} - T_{\text{w}}) D_o} \right)^{1/4} \]  

where \( \rho_l \) and \( \rho_g \) are the liquid and vapor densities of the working fluid, \( T_{\text{sat}} \) represents the condensation temperature in condenser, and \( k_f \) is the modified latent heat of the working fluid. The total heat-exchanger area of the condenser and evaporator in ORC can be obtained by

\[ A_t = A_c + A_e \]  

The seawater pressure drop in the pipe is calculated by

\[ \Delta P_W = f \frac{L_W \rho_W V_t^2}{D_w^2} \]  

where \( f \) is a dimensionless friction factor, and \( L_W \) and \( D_w \) are the length and inner diameter of the seawater pipe. The power consumption of the cold and warm seawater pumps can be defined as

\[ W_{wp} = m_{wp} \Delta P_W / \rho_w \eta_p \]  

The net power output of ORC can be determined by

\[ W_{\text{net}} = W_{\text{out}} - W_p - W_{wp,c} - W_{wp,w} \]  

The net thermal efficiency of ORC is calculated by

\[ \eta_{\text{net}} = \frac{W_{\text{net}}}{Q_e} \]
Finally, the objective parameter that represents the ratio of net power output, \( W_{\text{net}} \), to total heat transfer area, \( A_t \), in the system is defined as

\[
\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{eva}}} \tag{16}
\]

4. Results and discussion

4.1. Problem description

A finite-temperature-difference heat transfer method, which means heat transfer occurred only with temperature difference between seawater and working fluid, is applied to evaluate the heat transfer area of the condenser and evaporator in an OTEC system with an ORC. R134a, R152a, R245fa, R600a, and R717 are used as the working fluids. In this study, the warm seawater flow rate remains fixed, the mass flow rate of working fluid varies to obtained the optimal temperature conditions. This section presents the results of parametric studies on the various working fluids in the system. The following general assumptions were made in thermodynamic analysis:

1. Steady-state conditions are applied to all components.
2. Warm seawater flow rate: 2000 kg/s.
3. Evaporation temperature: 18–24 °C.
4. Condensation temperature: 8–14 °C.
5. Cold seawater temperature: 5 °C.
6. Warm seawater temperature: 28 °C.
7. Seawater temperature differences between inlet and outlet are the same in the evaporator and the condenser.
8. Efficiencies of cold and warm seawater pumps and working fluid pump: 0.8.
9. Turbine efficiency: 0.9.
10. Condenser and evaporator are shell-and-tube heat exchangers and the correction factor \( F \) is 0.9.

The thermodynamic data used in this analysis are obtained from the National Institute of Standards and Technology (NIST) database REFPROP 9.0 [24].

4.2. Verification

To evaluate the accuracy of the power output calculation for the OTEC system, the numerical solution of net power output in the ORC system is verified using the results of Sun et al. [10]. The relationships between net power output and various mass flow rates of warm seawater are compared using R717 as the working fluid with \( T_{\text{eva}} = 22.2 °C \) and \( T_{\text{con}} = 10.8 °C \). The cold and warm seawater inlet temperatures are assumed to be 5 °C and 28 °C, respectively, for the simulation. However, the power consumptions of cold and warm seawater pumps were not included in evaluating system power output, \( W_{\text{sys}} \), for verification. Overall comparison shows that the numerical solutions of this study agree well with those of Sun et al. [10], as shown in Fig. 3.

4.3. Performance optimization

Fig. 4(a)–(c) plot the influence of evaporation temperature on net power output, total heat transfer area, and the objective
parameter in the ORC system with evaporating temperature of $T_{\text{ev}} = 10.7 \degree C$. Cold and warm seawater temperature differences between the inlet and outlet, $\Delta T_{\text{sw}}$, are assumed to be $2.1 \degree C$ for the evaporator and condenser. Fig. 4(a) shows that with an increase in evaporating temperature, the pressure difference across the turbine becomes large and hence the net power output increases. Among the working fluids tested, R600a generates the highest net power output at various $T_{\text{ev}}$. Under the same heat-source conditions, the ORC system operated with R600a also achieves excellent thermal efficiency. As the evaporation temperature increases, the total heat transfer area, $A_t$, rises gradually initially and later increases steeply, which is because the temperature difference between warm seawater and the working fluid decreases in the evaporator. Fig. 4(b) shows that the $A_t$ curve of R245fa performs highest value among the working fluids because R245fa has the poorest heat transfer properties as shown in Table 1. Notably, the $A_t$ curve of R717 is the smallest because R717 has the largest latent heat and thermal conductivity. Fig. 4(c) demonstrates the effects of $T_{\text{ev}}$ on $g$, which is the ratio of net power output to total heat transfer area of the OTEC system. For all working fluids, the values

![Fig. 6. Contours of $\gamma$ for (a) R717, (b) R134a, (c) R245fa, (d) R152a and (e) R600a with $\Delta T_{\text{sw}} = 2.1 \degree C$.](image-url)
of $\gamma$ can be seen to increase with evaporation temperature, reach a maximum, and finally decrease. This result indicates that an optimal objective parameter exists with which maximal net power output per unit area of heat exchangers can be obtained from the system. For convenience, the optimal condensing and evaporating temperatures, which have the corresponding maximal objective parameter $\gamma_{\text{max}}$ in the OTEC system, are denoted as $T_{\text{con,o}}$ and $T_{\text{eva,o}}$ in this work. Note that $\gamma_{\text{max}} = 0.203$ is obtained at the corresponding optimal evaporating temperature $T_{\text{eva,o}} = 22$ °C for R717 at $T_{\text{con}} = 10.7$ °C.

Conversely, high condensation temperature decreases the net power output because the enthalpy of the turbine outlet increases, and the enthalpy difference between the turbine inlet and outlet decreases, as shown in Fig. 5(a). The figure also shows that R600a has largest net power output at $T_{\text{con}} = 8 - 14$ °C and $T_{\text{eva}} = 22.3$ °C. Furthermore, R134a and R245fa have small $W_{\text{net}}$ values that differed only slightly. Similarly, in Fig. 5(b), the $A_{\gamma}$ curves show a tendency to decline with condensation temperature. This is because as the condensation temperature rises, the temperature difference between cold seawater and the working fluid in the condenser increases, resulting in a reduction in the total heat transfer area. The $A_{\gamma}$ of R245fa is the largest among the working fluids, as mentioned earlier. This is because of the weak thermal conductivity of R245fa. Fig. 5(c) demonstrates the influence of $T_{\text{con}}$ on $\gamma$ at $T_{\text{eva}} = 22.3$ °C for each working fluid. As expected, with increasing condensation temperatures, the values of $\gamma$ increase first, then peak, and finally decrease. The maximal value, $\gamma_{\text{max}} = 0.203$, occurs at $T_{\text{con,o}} = 11.1$ °C for R717. In objective parameter evaluation, R717 performs clearly better than the other working fluids tested.

To indicate the optimal operating temperatures of the ORC system, the distributions of $\gamma$ for various evaporation and condensation temperatures at $\Delta T_w = 2.1$ °C are presented. In Fig. 6(a)–(e) for each working fluid, it is expected, a series of optimal evaporation temperatures, $T_{\text{eva,o}}$, and optimal condensation temperatures, $T_{\text{con,o}}$, can be obtained for the maximal ratio of $W_{\text{net}}$ to $A_\gamma$. These contours of $\gamma$ show the variations of optimal operating temperatures for the working fluids. These figures show that the $\gamma_{\text{max}} = 0.204$ of R717 is the largest among the five working fluids for $\Delta T_w = 2.1$ °C.

4.4. Effects of seawater temperature difference in ORC

To illustrate the influence of seawater temperatures different on operating temperatures and $\gamma$, the variations of optimal condensation and evaporator temperatures from $\Delta T_w = 1 - 4$ °C are revealed in Fig. 7(a) at $T_{\text{wwi}} = 28$ °C and $T_{\text{wwi}} = 5$ °C. In this study, seawater temperature difference between the inlet and outlet, $\Delta T_w$, is assumed to be the same in the evaporator and condenser for warm and cold seawater. As the seawater temperature difference increases in the heat exchangers, the optimal evaporation temperatures rise and the optimal condensation temperatures decline. Notably, optimal evaporation and condensation temperatures of R600a in all cases are larger than those of others working fluids, and the optimal operational temperatures of R134a and R245fa are smaller than those of other fluids. Thus, R600a is suitable for higher operational temperatures of the ORC system. Reducing condensation temperature and enhancing evaporation temperature augment the power output in the ORC system, but the changes also increase the heat transfer rate in the condenser and evaporator. These lead to an increase in the sizes of the heat exchangers of an OTEC system and in the cost of the apparatus. Fig. 7(b) shows the $\gamma_{\text{max}}$ variation of an OTEC system in relation to $\Delta T_w$ under optimal operational temperatures; with increasing $\Delta T_w$, the $\gamma_{\text{max}}$ values of all working fluids rise to their peak values and then decline. The variation of $\gamma_{\text{max}}$ of R152a and R600a are close to various $T_{\text{eva,o}}$ and $T_{\text{con,o}}$.

The maximal objective parameter, $\gamma_{\text{max}}$, and its corresponding optimal operating temperatures, $T_{\text{eva,o}}$ and $T_{\text{con,o}}$, and $\Delta T_w$ of the OTEC system at $T_{\text{wwi}} = 5$ °C and $T_{\text{wwi}} = 28$ °C are obtained numerically and displayed in Table 2. With various $T_{\text{eva,o}}$ and $T_{\text{con,o}}$, the $\Delta T_w$ values from high to low are for R717, R152a, R245fa, R134a, and R600a. Although the optimal evaporation temperatures, $T_{\text{eva,o}}$, are similar for all the working fluids, both $T_{\text{eva,o}}$ and $T_{\text{con,o}}$ of R600a are higher than corresponding values of other fluids.

![Graph showing the influence of $\Delta T_w$ on $T_{\text{eva,o}}$ and $T_{\text{con,o}}$.](image1)

![Graph showing the variation of $\gamma_{\text{max}}$ with $\Delta T_w$.](image2)

**Table 2**

The $\gamma_{\text{max}}$ and its corresponding to $T_{\text{eva,o}}$, $T_{\text{con,o}}$, and $\Delta T_w$ for an OTEC system with $T_{\text{wwi}} = 5$ °C and $T_{\text{wwi}} = 28$ °C.

<table>
<thead>
<tr>
<th></th>
<th>R717</th>
<th>R134a</th>
<th>R245fa</th>
<th>R152a</th>
<th>R600a</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{eva,o}}$ (°C)</td>
<td>22.32</td>
<td>22.6</td>
<td>22.41</td>
<td>22.62</td>
<td>22.75</td>
</tr>
<tr>
<td>$T_{\text{con,o}}$ (°C)</td>
<td>10.78</td>
<td>11.02</td>
<td>10.69</td>
<td>10.98</td>
<td>11.13</td>
</tr>
<tr>
<td>$\Delta T_w$ (°C)</td>
<td>1.61</td>
<td>1.56</td>
<td>1.58</td>
<td>1.591.5</td>
<td>1.5</td>
</tr>
<tr>
<td>$\gamma_{\text{max}}$ (kW/m²)</td>
<td>0.2068</td>
<td>0.1157</td>
<td>0.1011</td>
<td>0.1282</td>
<td>0.1283</td>
</tr>
</tbody>
</table>
4.5. Effects of seawater temperature in ORC

Higher warm seawater temperatures and lower cold seawater temperatures increase the net output power and efficiency of an OTEC system. To investigate the influence of cold seawater temperature at the heat-exchanger inlet, the variation of $T_{\text{eva,o}}, T_{\text{con,o}}, g_{\text{max}}$, and thermal efficiency for an OTEC system are shown in Fig. 8(a)–(d). To collect colder seawater, the cold seawater extracting inlet has to reach deeper and therefore the pump will consume more power. The lengths of the cold seawater pipe are assumed to be 750 m, 900 m, 1050 m, and 1200 m to correspond to $T_{\text{cwi}} = 5/14^\circ\text{C}$, $6/14^\circ\text{C}$, $7/14^\circ\text{C}$, and $8/14^\circ\text{C}$, respectively. Fig. 8(a) and (b) show that the values of both $T_{\text{eva,o}}$ and $T_{\text{con,o}}$ increase with $T_{\text{cwi}}$ from 5/14 to 8/14°C at $T_{\text{wwi}} = 28^\circ\text{C}$. Furthermore, the influence of $T_{\text{cwi}}$ on $T_{\text{con,o}}$ is stronger than on $T_{\text{eva,o}}$ in an OTEC system. Interestingly, the $T_{\text{con,o}}$ values from high to low are for R600a, R134a, R152a, R717, and R245fa, respectively. The $T_{\text{eva,o}}$ values varied similarly, but the $T_{\text{eva,o}}$ distribution of R245fa is slightly higher than that of R717. The maximal objective parameter $g_{\text{max}}$ and the net thermal efficiency $\eta_{\text{th}}$, which correspond to $T_{\text{con,o}}$ and $T_{\text{eva,o}}$, both decreased with $T_{\text{wwi}}$ from 5°C to 8°C as shown in Fig. 8(c) and (d). Although R717 shows the best results in objective parameter analysis, R600a has the highest net thermal efficiency among the five working fluids.

Fig. 9(a)–(d) show the influence of warm seawater inlet temperature of the evaporator on optimal operating temperatures and their corresponding maximal objective parameter and thermal efficiency. Fig. 9(a) and (b) reveal that the increments of $T_{\text{eva,o}}$ are larger than those of $T_{\text{con,o}}$ from $T_{\text{wwi}} = 25^\circ\text{C}$–28°C for each working fluid. The results in these figures show that the $g_{\text{max}}$ values from high to low are for R717, R152a, R600a, R134a, and R245fa. The values of $g_{\text{max}}$ for R152a and R600a are close both in variation of $T_{\text{cwi}}$ and $T_{\text{wwi}}$. By contrast, the values $\eta_{\text{th}}$, which correspond to $g_{\text{max}}$, from high to low are for R600a, R717 R152a, R245fa, and R134a, respectively. From these results, one can deduce that R245fa performs worst in object parameter analysis, and R134a has the lowest net thermal efficiency for an OTEC system. These results also indicate that R600a is most suitable for operating at high evaporating and condensing temperatures in an OTEC system.

To analyze the application of seawater thermal energy in the OTEC system, the variation of optimal seawater temperature difference $\Delta T_{\text{w,o}}$ in relation to $T_{\text{cwi}}$ and $T_{\text{wwi}}$ are shown in Fig. 10(a) and (b). It should be noted that in this study, the temperature differences, $\Delta T_{\text{w}}$, between the inlet and outlet for warm seawater of the evaporator, and for cold seawater of the condenser, are assumed to the same. The $\Delta T_{\text{w}}$ of the OTEC system with R717 and R152 are both the highest among corresponding values for all the working fluids at various $T_{\text{cwi}}$ and $T_{\text{wwi}}$. This phenomenon also reveals that the thermal energy absorbed from warm seawater and discharged to cold seawater is the lowest for R600a even though R600a has the highest thermal efficiency. Furthermore, $\Delta T_{\text{w}}$ increased slightly with $T_{\text{wwi}}$ but decreased markedly with $T_{\text{cwi}}$, because lower cold seawater temperature increases pump power consumption obviously. This result also indicates that the cold-seawater inlet temperature, $T_{\text{cwi}}$, has a stronger effect than the
5. Conclusions

An objective parameter $\gamma$, which is the ratio of net power output to total heat-exchanger area, was used to analyze the performance of an OTEC system with an ORC. The optimal operating temperatures of the system, $T_{\text{con,o}}$, $T_{\text{eva,o}}$, and $\Delta T_{\text{w,o}}$, are obtained numerically to evaluate the maximal objective parameter, $\gamma_{\text{max}}$, with $T_{\text{con}} = 5^\circ\text{C}$ and $T_{\text{eva}} = 18^\circ\text{C}$ to 24°C for R134a, R152a, R245fa, R600a, and R717. The results obtained support these conclusions:

1. In objective parameter evaluation, R717 performs best, followed by R152a and R600a, and R134a and R245fa perform the worst, when $T_{\text{eva}} = 18^\circ\text{C}$ to 24°C and $T_{\text{con}} = 5^\circ\text{C}$ to 8°C. Conversely, R600a has the highest thermal efficiency.
2. Cold seawater temperature affects optimal condensing temperature more strongly than optimal evaporating temperature in an OTEC system with an ORC. By contrast, warm seawater temperature affects optimal evaporating temperature more strongly than optimal condensing temperature.

3. The optimal temperature difference of seawater for obtaining the maximal objective parameter for an OTEC system is affected strongly by cold-seawater inlet temperature but weakly by warm-seawater inlet temperature.

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References


Glossary

- \( A_c \): total heat transfer area of heat exchangers (\( m^2 \))
- \( A_i \): heat transfer area of condenser (\( m^2 \))
- \( A_p \): heat transfer area of evaporator (\( m^2 \))
- \( D \): diameter (m)
- \( D_h \): hydraulic diameter (m)
- \( f \): dimensionless friction factor
- \( g \): acceleration due to gravity, \( m/s^2 \)
- \( h \): heat transfer coefficient (W/m\(^2\)-C)
- \( i \): enthalpy of working fluid (kJ/kg)
- \( k \): thermal conductivity (W/m-C)
- \( L \): length of tube or pipe (m)
- \( L_t \): thickness of tube wall (m)
- \( M \): molecular weight of working fluid (g/mol)
- \( m \): mass flow rate (kg/s)
- \( P \): pressure (kPa)
- \( Pr \): Prandtl number
- \( Q \): heat transfer rate (kW)
- \( q \): heat flux (W/m\(^2\))
- \( Re \): Reynolds number
- \( T \): temperature (°C)
- \( T_{ci} \): cold seawater inlet temperature of heat exchanger (°C)
- \( T_{cw} \): warm seawater inlet temperature of heat exchanger (°C)
- \( T_{e} \): warm seawater outlet temperature of heat exchanger (°C)
- \( T_{p} \): working fluid inlet temperature of heat exchanger (°C)
- \( T_{op} \): working fluid outlet temperature of heat exchanger (°C)
- \( \Delta T \): temperature difference between inlet and outlet of heat exchanger (°C)
- \( \Delta T_{mean} \): logarithmic mean temperature difference of heat exchanger (°C)
- \( U \): overall heat transfer coefficient of heat exchanger (W/m\(^2\)-C)
- \( W \): power of turbine or pump (W)

Greek symbols

- \( \gamma \): ratio of \( W_{net} \) to \( A_i \)
- \( \eta \): efficiency
- \( \mu \): dynamic viscosity (Pa-s)
- \( \rho \): density (kg/m\(^3\))
- \( \nu \): kinematic viscosity (m\(^2\)/s)

Subscripts

- \( \text{con} \): condensation, condenser
- \( \text{cwc} \): cold seawater
- \( \text{e} \): evaporation, evaporator
- \( \text{f} \): fluid
- \( \text{g} \): vapor
- \( i \): inside, inlet
- \( \text{max} \): maximal
- \( \text{net} \): net
- \( \text{out} \): outside, optimization
- \( \text{p} \): pump
- \( \text{r} \): working fluid
tube
- \( \text{th} \): thermal
- \( \text{ver} \): verification
- \( \text{w} \): seawater
- \( \text{ww} \): warm seawater