Research Paper

An innovative Organic Rankine Cycle (ORC) based Ocean Thermal Energy Conversion (OTEC) system with performance simulation and multi-objective optimization

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HIGHLIGHTS

- Develop an integrated OTEC model considering local resource conditions.
- Optimize and analyze trade-offs between LCOE and exergy efficiency.
- Use LINMAP to identify Pareto optimal solution for six different working fluids.
- Rank six selected working fluids according to Pareto optimal solution.

ABSTRACT

Based on multi-objective particle swarm optimization (MOPSO) algorithm, with considering levelized cost of energy (LCOE) and exergy efficiency as two different objective functions, an innovative Organic Rankine Cycle (ORC) model based Ocean Thermal Energy Conversion (OTEC) system is investigated for trade-off Pareto optimization. In the present study, six key parameters including evaporating temperature, condensing temperature, warm seawater temperature at the outlet of evaporator, cool seawater temperature at the outlet of condenser, degree of superheat, and depth of cool seawater have been selected as decision variables. R717, R152a, R134a, R227ea, R600a and R601 are chosen as working fluids. Meanwhile, Linear Programming Technique for Multi-dimensional Analysis of Preference (LINMAP) is introduced in order to make decision for Pareto frontier. The results indicate that LCOE and exergy efficiency are two conflicting objectives, which are impossible to both achieve their optimal values simultaneously. According to the non-dominated sorting of Pareto optimal solution (POS) for the six working fluids, R717 and R601 have the best performance with 0.34 $/kWh of LCOE, 28.17% of exergy efficiency and 0.52 $/kWh of LCOE, 28.47% of exergy efficiency, respectively, followed by R152a, R600a and R134a which have relatively poor performance, but better than R227ea.

1. Introduction

Ocean Thermal Energy Conversion (OTEC) is one of the most typical clean and sustainable energy technologies which can utilize the temperature difference between warm surface seawater and cool deep seawater to generate power via Organic Rankine Cycle (ORC). It is estimated that the global reserves of ocean thermal energy is approximately 4.4 × 10\(^{16}\) kWh/a. OTEC, with huge potential for application, is particularly suitable to solve the problem of power supply and refrigeration on remote islands.

1.1. Literature review

Although OTEC appears to be promising, it suffers from the problems of low thermal efficiency and high investment cost. Since the temperature difference of OTEC is as low as 20 °C, the system is relatively sensitive to the evaporating and condensing temperatures as well as other performance parameters. Therefore, researchers have been seeking for suitable working fluids and exploring the best cycle configuration to improve system performance of OTEC. In the 19th century, French scientist J d’Arsonval first suggested the OTEC concept for power generation. The first 50 kW mini-OTEC offshore plant [1] was...
constructed in Hawaii in 1979, which had the net power output of 15 kW. In 1981, a 100 kW OTEC onshore pilot plant [2] was built by TEPCO in the Republic of Nauru, in which R22 was selected as working fluid for a closed cycle system. The net power output of the system was about 31.5 kW. Later, a 50 kW Tokunoshima OTEC onshore plant [3] was built in Japan in 1982, in which seawater and ammonia (R717) were used as working fluids for closed cycle system. A 1 MW offshore floating OTEC plant [4] was built by the National Institute of Ocean Technology (NIOT) of India and Saga University of Japan, which supplied 493 kW to the power grid. In general, the OTEC technology has been well applied in tropical ocean areas such as South Pacific, Indian Ocean and Caribbean Sea. However, the system suffers from relatively low efficiency due to the high power consumption of seawater pump. Based on ORC and Kalina cycle system [5,6], Uehara et al. [7] theoretically and empirically investigated OTEC performance, drawing a conclusion that R717 with low boiling temperature performed well in the OTEC system. Consequently, Uehara [8] presented a new closed cycle system named Uehara Cycle, with a view to improving the thermal efficiency of OTEC. In addition to Uehara’s research, Yoon et al. [9] proposed a new cycle configuration of ejector pump OTEC (E-OTEC) by using R152a which as a wet working fluid has lower latent heat of vaporization than R717. Apart from studies on cycle configuration and working fluids, researchers also explored the impacts of power consumption from seawater pump and optimized key parameters to improve system performance. Yeh et al. [10] theoretically analyzed the impacts of flowrate and temperature of cold deep seawater on OTEC performance for the purpose of maximizing the net power output with R717 as working fluid. Sun et al. [11] optimized the net power output of ORC for OTEC with exergy analysis, using R717 and R134a as working fluids. Faizal et al. [12] designed and built a closed cycle OTEC demonstration plant to experimentally investigate the relationship between thermal efficiency and seawater flowrate. Yang et al. [13] conducted optimization of the OTEC system based on five different working fluids. For calculation, a new objective parameter referring to the ratio of net power to heat exchange area was introduced to explore the thermal performance of different working fluids.

1.2. Motivation

Although many studies have investigated the optimization approaches of thermal efficiency and net power output for OTEC systems, all of these previous studies are only focused on the optimization of one single objective rather than multiple objectives, such as optimization of the thermodynamic and economic performance with Pareto optimization approach by minimizing the levelized cost of energy while...
maximizing the exergy efficiency simultaneously. Moreover, most studies assumed that the temperature of deep cool seawater and the length of seawater pipeline were constant, which means that the impacts of these two key parameters were not taken into account. In addition, few studies have discussed both the optimization and comparison of working fluids for the OTEC system.

In the above context, this study aims to explore the multi-objective optimized performance of an ORC based on OTEC system by using different working fluids. An integrated model is built for the system, which the data of tropical waters resource conditions is acquired from the Argo Project [14]. Meanwhile, the multi-objective optimization in terms of levelized cost of energy (LCOE) and exergy efficiency is developed for this model. Considering the ozone depletion potential (ODP) and global warming potential (GWP), six working fluids have been selected including R717, R152a, R134a, R227ea, R600a and R601. The impacts of key parameters on LCOE and exergy efficiency will be analyzed, and the multi-objective trade-off optimization will be conducted via multi-objective particle swarm optimization (MOPSO) approach in order to find out Pareto frontiers for each working fluid. Six kinds of working fluid will be ranked by Pareto optimal solution (POS) which is obtained from Pareto frontier with a typical decision-making method named linear programming techniques for multi-dimensional analysis of preference (LINMAP). Finally, the performance of working fluids and their optimal working conditions will be determined.

The rest of the paper is organized as follows: Section 2 describes the OTEC system configuration; Section 3 provides the modelling methodology used in this study; Section 4 presents the optimization methods and parameter determination; Section 5 analyzes the research results; Section 6 summarizes the key discussions and conclusions.

2. OTEC system

2.1. Selection of working fluids

Organic Rankine Cycle is considered as the best way for OTEC to generate power. Low-boiling working fluids are usually used in the OTEC system to reduce the internal power consumption and improve the exergy efficiency. R22 used to be the most widespread refrigerant with excellent thermodynamic performance, while it is harmful to ozonosphere and has been restricted by law. Therefore, in the present study six kinds of working fluid with zero ODP and low GWP have been considered, of which three are wet working fluids (R717, R152a, R134a) and the others are dry working fluids (R227ea, R600a, R601). Table 1 shows the properties of these working fluids.

As each working fluid differs in the slope of saturation curve, dry working fluids will be superheated after turbine expansion process, while wet working fluids will experience two phases after the same process. However, when the degree of superheat of wet working fluids is higher than a certain value, the turbine exhaust for wet working fluids will also be superheated. Fig. 1 shows the T-s diagram of these working fluids.

2.2. System description

Unlike the conventional ORC, OTEC cycle consists of not only evaporator, turbine, condenser and working fluid pump but also seawater pumps, the power consumption of which have to be specifically considered. The configuration of OTEC system is shown in Fig. 2. The working fluid is pressurized and transferred by the ORC pump to the evaporator, where it is heated by warm seawater and turns to saturated or superheated steam. The saturated steam expands in the turbine to generate power until it becomes low-pressure steam, which will further release heat to the cool seawater and condenses to saturated liquid via the condenser. At last, the saturated liquid working fluid will be pumped back to the evaporator to finish an entire cycle. The thermodynamic processes for wet working fluids and dry working fluids are illustrated in the T-s diagram for OTEC system as shown in Fig. 3(a) and (b).

3. System modelling

3.1. Major specifications and modeling assumptions

In order to develop the model representing the performance characteristics of the OTEC system, a number of simplifications and assumptions have been made. As the power consumption of seawater pumps is considered, the model in this study is built on the assumption that the surface warm seawater flowrate is fixed. The power consumption is calculated based on the pressure drop in the pipeline. The deep cool seawater flowrate and the working fluid flowrate are dependent variables affected by other parameters. Other general assumptions and simplifications are listed as follows:

- The OTEC system operates under steady-state conditions.
- The potential and kinetic energy are negligible.
- The leakage of working fluid from the components are negligible.
- The heat losses (and gains in the cool water pipeline) to surrounding are negligible.
- All fluid temperatures are the average, or bulk temperature at that location.
- All fluid velocities are uniform and constant.
- The losses due to friction in heat exchangers are negligible.
- All liquids are incompressible.

3.2. Model development

3.2.1. Energy and exergy balances

The proposed OTEC model is mathematically developed within the framework of Matlab R2016a, by which the mass and energy balance equations for energy conversion processes have been determined in detail. The enthalpy and entropy of each state point has been evaluated through REFPROP 9.0 based on the National Institute of Standards and Technology (NIST) database.

The energy balance of evaporator is as follows:

\[ \dot{Q}_{E} = \dot{m}_{\text{w}}(h_{w} - h_{l1}) \]  

(1)

where \( \dot{Q}_{E} \) is the heat transfer rate of evaporator, \( \dot{m}_{\text{w}} \) is the mass flow rate of warm seawater, \( h_{w} \) and \( h_{l1} \) are specific enthalpies of warm seawater at the evaporator inlet and outlet, respectively.

The mass flow rate of working fluid is calculated as follows:

\[ \dot{m}_{l} = \frac{\dot{Q}_{E}}{h_{w} - h_{l}} \]  

(2)

where \( h_{l} \) and \( h_{w} \) are the specific enthalpies of working fluid at the evaporator inlet and outlet, respectively.

In the evaporator, the pinch point temperature difference (PPTD) is an important parameter to be calculated as constraint, which is defined as follows:

\[ \Delta T_{\text{PPTD}} = T_{l0} - T_{l1} \]  

(3)

Table 1

<table>
<thead>
<tr>
<th>Working fluid</th>
<th>Chemical formula</th>
<th>Critical temperature (K)</th>
<th>Critical pressure (kPa)</th>
<th>GWP</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>R717</td>
<td>NH₃</td>
<td>405.91</td>
<td>11,380</td>
<td>0</td>
<td>Wet</td>
</tr>
<tr>
<td>R152a</td>
<td>CHF₂CH₃</td>
<td>386.41</td>
<td>4517</td>
<td>140</td>
<td>Wet</td>
</tr>
<tr>
<td>R134a</td>
<td>CHF₂CH₃F</td>
<td>374.21</td>
<td>4059</td>
<td>1200</td>
<td>Wet</td>
</tr>
<tr>
<td>R227ea</td>
<td>CF₃CHF₂</td>
<td>374.80</td>
<td>2926</td>
<td>2050</td>
<td>Dry</td>
</tr>
<tr>
<td>R600a</td>
<td>CH₃CH₂CH₂CH₂CH₃</td>
<td>407.85</td>
<td>3629</td>
<td>20</td>
<td>Dry</td>
</tr>
<tr>
<td>R601</td>
<td>CH₃CH₂CH₂CH₂CH₃</td>
<td>469.36</td>
<td>3374</td>
<td>11</td>
<td>Dry</td>
</tr>
</tbody>
</table>
where $T_{10}$ and $T_{6}$ are the temperatures of warm seawater and working fluid, respectively.

Another important parameter of the evaporator is the degree of superheat, which is characterized as the temperature difference between the temperature at evaporator outlet $T_1$ and the saturation temperature of working fluid $T_7$. It can be calculated as:

$$\Delta T_{\text{sph}} = T_1 - T_7$$

(4)

The power output of turbine is calculated as:

$$\dot{W}_T = m_1 (h_1 - h_2) \eta_G$$

(5)

$$\eta_G$$ is the generator efficiency, $h_2$ is specific enthalpy of working fluid at turbine inlet.

The isentropic efficiency $\eta_{i,T}$ is used to calculate the enthalpy change of working fluid, which is expressed as:

$$\eta_{i,T} = \frac{h_{i2} - h_{i2s}}{h_{i2s}}$$

(6)

where $h_{i2s}$ is the ideal enthalpy of working fluid after expansion process in turbine.

It should be noted that the wet working fluid at state point 2 has become the two-phase gas-liquid mixture. The steam quality at point 2, a constraint for model simulation, is calculated as:

$$\chi = \frac{h_{i3} - h_4}{h_{i3} - h_{i4}}$$

(7)

where $h_3$ and $h_4$ are specific enthalpies of working fluid in saturated steam and saturated liquid, respectively.

The heat transfer rate of condenser is expressed as:

$$\dot{Q}_C = \dot{m}_c (h_4 - h_3)$$

(8)

The mass flow rate of cool seawater can thus be deduced as:

$$\dot{m}_c = \frac{\dot{Q}_C}{h_4 - h_3}$$

(9)

Similarly, PPTD of the condenser is calculated as:

$$\Delta T_{PPTD,C} = T_i - T_3$$

(10)

For the working fluid pump, its power consumption is calculated as:

$$\dot{W}_p = \frac{m_2 (h_{i2s} - h_4)}{\eta_{i,p} \eta_{i,p}} \frac{m_1 (h_3 - h_4)}{\eta_p}$$

(11)

where $\eta_{i,p}$ and $\eta_{i,p}$ represent the efficiency of pump motor and the isentropic efficiency, respectively.

In addition, the pressure drop of seawater pipeline is used to calculate the power consumption of seawater pump:

$$\Delta P_{\text{seawater}} = \frac{\rho_s D_{\text{seawater}} V_{\text{seawater}}^2}{2}$$

(12)
where $\Delta P_{\text{WW}}$ and $\Delta P_{\text{CW}}$ are the pressure drops of the warm and cool seawater pipelines respectively, $f$ represents the fanning friction factor which is a dimensionless parameter to be used in the continuum mechanics calculation.

Thereupon, the power consumption of seawater pump is obtained as follows:

$$W_{\text{wp}} = \frac{\dot{m}_{\text{sw}}(s_0-s_i)\Delta P_{\text{sw}}}{\rho_{\text{sw}} g}$$

where $W_{\text{wp}}$ and $W_{\text{cp}}$ are the power consumptions of warm seawater pump and cool seawater pump respectively.

Based on the above calculation, the net power output of OTEC can be obtained:

$$\dot{W}_{\text{net}} = W_{\text{wp}} - W_{\text{cp}}$$

The net thermal efficiency of OTEC cycle can be determined as follows:

$$\eta_{\text{net}} = \frac{\dot{W}_{\text{net}}}{\sum \dot{Q}_{\text{in}}}$$

The exergy of each state point in the OTEC is expressed as [15]:

$$\dot{E}_i = \dot{m} \left[ (h_i-h_{i-1}) - T_0 (s_i-s_{i-1}) \right]$$

where subscript $i$ represents each point for the model. $T_0$ is the environment temperature.

The exergies for main system components are calculated as follows [16]:

$$\dot{E}_{\text{loss,eva}} = \dot{E}_1 - \dot{E}_i + \dot{E}_4 - \dot{E}_6$$

$$\dot{E}_{\text{loss,con}} = \dot{E}_{i+1} - \dot{E}_{i+4} + \dot{E}_5 - \dot{E}_7$$

$$\dot{E}_{\text{loss,turb}} = \dot{E}_1 - \dot{E}_7 - W_{\text{p}}$$

$$\dot{E}_{\text{loss,pump}} = \dot{E}_4 - \dot{E}_5 + W_{\text{p}}$$

where $\dot{E}_{\text{loss}}$ represents the exergy loss of component.

The total exergy input of the system includes two parts, i.e., the exergy transferred from warm seawater to system via heat exchanger, and the exergy output from the working fluid pump. Therefore, the total exergy transferred into the OTEC system is deduced as [11]:

$$\dot{E}_{\text{in, tot}} = \dot{E}_{\text{in, WW}} + \dot{E}_{\text{in, CW}} + \dot{W}_{\text{p}}$$

where $\dot{E}_{\text{in, WW}}$ and $\dot{E}_{\text{in, CW}}$ are the exergies transferred from warm seawater and cool seawater, respectively.

Likewise, the exergy transferred out of system via heat exchanger is expressed as:

$$\dot{E}_{\text{out, tot}} = \dot{E}_{\text{out, WW}} + \dot{E}_{\text{out, CW}}$$

As a result, the exergy efficiency of the OTEC system is calculated as follows:

$$\eta_{\text{ex}} = 1 - \frac{\dot{E}_{\text{out, tot}} - \sum \dot{E}_{\text{in, tot}}}{\dot{E}_{\text{in, tot}}}$$

### 3.2.2. Heat exchanger

A plate heat exchanger with countercurrent arrangement is chosen for the OTEC system. The heat transfer area of each section in heat exchangers can be calculated as follows:

$$A_i = \frac{Q_i}{U_i \Delta T_{\text{LM}} F}$$

where $U_i$ is the heat transfer coefficient for the $i_{th}$ heat exchange section, $A_i$ is the heat exchange area, $F$ is a correction factor for heat exchange, and $\Delta T_{\text{LM}}$ represents the logarithmic mean temperature difference (LMTD) between the working fluid and seawater. Due to different phase states of the working fluid, the evaporator and condenser need to be calculated by section. Thereupon, the heat transfer coefficient of each section can be calculated separately. The LMTD of each heat exchange section is expressed as:

$$\Delta T_{\text{LM}} = \Delta T_{\text{max}} - \Delta T_{\text{min}}$$

where $\Delta T_{\text{max}}$ and $\Delta T_{\text{min}}$ denote the maximum and minimum temperature differences of each heat exchange section. It should be noted that the cooling section exists only in the condenser for dry working fluids. The heat transfer coefficient of each section is thus deduced by the following equation:

$$\frac{1}{U_i} = \frac{1}{\alpha_{i,r}} + \frac{1}{\alpha_{i,c}} + \frac{\delta}{\lambda_e} + \frac{\rho_f}{\epsilon}$$

where $\alpha_{i,r}$ is the fouling resistance of heat exchanger, $\delta$ is the thickness and $\lambda_e$ is the thermal conductivity of raw materials of the heat exchange plate, $\alpha_{i,c}$ and $\alpha_{i,c}$ are the coefficient of heat transfer for the working fluid and seawater, respectively.

In the evaporator, the heat transfer coefficient of the single-phase working fluid for preheating and superheating processes can be calculated as [17]:

$$\alpha_{\text{eva.r}} = \frac{d}{4} \left( \frac{1}{\epsilon R_{\text{e}}^{0.8}} + \frac{1}{\epsilon R_{\text{e}}^{0.8}} + \frac{1}{\epsilon R_{\text{e}}^{0.8}} + \frac{1}{\epsilon R_{\text{e}}^{0.8}} \right)$$

where $R_{\text{e}}$ is the Reynolds Number and $Pr$ is the Prandtl Number (applicable for $6000 < Re < 10^5$ and $0.5 < Pr < 1.2$), $\lambda_e$ is the thermal conductivity of working fluid, $d$ is the equivalent diameter of heat exchanger, $n$ is an empirical constant which is 0.4 for endothermic process and 0.3 for exothermic process.

For the evaporation process, the heat transfer coefficient of the working fluid in the evaporation section is calculated with the boiling two-phase flow correlation [18]:

$$\alpha_{\text{eva.b}} = S_{\text{eva.b}} \alpha_{\text{eva.b}} + \alpha_{\text{eva.b}}$$

where $S$ is the impact factor of nucleate boiling, $\alpha_{\text{eva.b}}$ and $\alpha_{\text{eva.b}}$ are the heat transfer film coefficients of pool nucleate boiling and forced-convection two-phase flow, respectively. The pool nucleate boiling coefficient is calculated based on the Forster-Zuber's formulation [19]:

$$\alpha_{\text{eva.b}} = 0.00122 \left( \frac{C_i}{\rho_i} \right) \text{Pr}^{0.4} \text{Re}^{0.1}$$

where $C_i$ is the specific heat of working fluid in liquid state, $\rho_i$ is the density of working fluid in liquid state, $g$ is the gravitational constant, $\sigma$ is the surface tension, $\mu$ is the dynamic viscosity, $\gamma$ is the heat of vaporization, $\Delta T_e$ is the difference between boiling temperature and heat exchange plate temperature, $\Delta P_e$ is the pressure difference corresponding to the temperature difference of $\Delta T_e$.

Another heat transfer film coefficient modified from the Dittus-Boelter equation [18] is expressed as follow, which is applicable for $Re > 10^4$ and $0.6 < Pr < 160$:

$$\alpha_{\text{eva.b}} = 0.023 \frac{\alpha_{\text{eva.b}}}{\alpha_{\text{eva.b}}} \text{Re}^{0.8} \text{Pr}^{0.4} \left( 1 + X_{\text{eva.b}} \right)$$

where $X_{\text{eva.b}}$ is the Lockhart-Martinelli parameter, and $\chi$ denotes the steam quality. To simplify this model, the vapor quality variation is assumed to be linear with the evaporation section [16]. The average value of $\alpha_{\text{eva.b}}$ is calculated as:

$$\bar{\alpha}_{\text{eva.b}} = \int \alpha_{\text{eva.b}} \text{d}X$$

On the other hand, LMTD can be used to calculate the temperature difference for each heat exchange section of the condenser. The heat
transfer film coefficient of condensation section is expressed as [20]:

\[
\alpha_{\text{cond}} = 0.943 \times \left( \frac{\lambda_{\text{water}} (T_{\text{con}} - T_{\text{ev}})}{d \mu T_{\text{con}} \rho_{\text{water}} (T_{\text{con}} - T_{\text{ev}})} \right)^{0.25}
\]

(33)

where \( T_{\text{con}} \) is the condensing temperature, and \( T_{\text{ev}} \) is the plate temperature of the condenser.

As for cooling process of the dry working fluid, the single phase heat transfer coefficient of the working fluid is calculated as follows [21], applicable for \( Re > 650 \).

\[
\alpha_{\text{cool}} = 0.5 \left( \frac{\lambda_{\text{water}}}{d} \right)^{0.5} Re^{0.8} Pr^{0.33}
\]

(34)

Unlike the heat transfer coefficient of working fluid, for all the heat exchangers in the present model, the heat transfer coefficient of seawater is calculated as [22]:

\[
\alpha_{\text{sw}} = 0.159 \frac{\lambda_{\text{water}}}{d} Re^{0.64} Pr^{0.58}
\]

(35)

where \( \lambda_{\text{water}} \) is the thermal conductivity of seawater, \( n \) is an empirical constant which is 0.4 for the endothermic process and 0.3 for the exothermic process.

The total heat transfer area of evaporator is summed up as follows:

\[
A_{\text{tot.e}} = A_{\text{pre}} + A_{\text{ev}} + A_{\text{qph}}
\]

(36)

where \( A_{\text{pre}} \) is the heat transfer area for preheating section; \( A_{\text{ev}} \) is the heat transfer area for evaporating section; \( A_{\text{qph}} \) is the heat transfer area for superheating section.

What should be noted is that the calculation of the total heat transfer area of condenser differs for dry and wet working fluids. For dry working fluid, the total heat transfer area of condenser is calculated as:

\[
A_{\text{tot.c}} = A_{\text{cool}} + A_{\text{con}}
\]

(37)

where \( A_{\text{cool}} \) and \( A_{\text{con}} \) is the heat transfer area for cooling section and condensing section, respectively. For wet working fluid, the calculation is:

\[
A_{\text{tot.c}} = A_{\text{con}}
\]

(38)

3.2.3. Economic analysis

According to the OTEC configuration, the total cost of the system involves the costs of evaporator, condenser, turbine, circulating pump of the working fluid, seawater pump and pipes. The levelized cost of electricity (LCOE) [23,24] is defined as the ratio of the total system cost to the total net power output, which is used to characterize the thermal-economic performance.

The equipment module costing technique is a extensively-used method to estimate and calculate the complete cost of new chemical plant [25], which is also suitable for ORC plant. With this approach, all direct and indirect costs are considered and represented by \( C_p \) as the purchased cost for “base conditions” [26]. The capital cost of each unit of the OTEC system can be calculated accordingly. The capital cost of each heat exchanger is expressed as follows [27,28]:

\[
\lg C_p = 4.6656 - 0.1557 \lg A_i + 0.1547 (\lg A_i)^2
\]

(39)

where \( A_i \) is the heat exchange area for each component.

For pumps and turbine [29,30], the correlation is as follows:

\[
\lg C_p = K_1 + K_2 \lg W + K_3 (\lg W)^2
\]

(40)

where \( K_1, K_2 \) and \( K_3 \) are the empirical coefficients between cost evaluation and installed power capacity \( W \), which are considered as constant as 3.8696, 0.3161 and 0.122 for the circulating pump of working fluid, as 3.3892, 0.0536 and 0.1538 for the seawater pump, and as 2.6259, 1.4398 and \(-0.1776\) for the turbine [26].

In addition to the purchased cost for “base conditions”, specific material of constructions, equipment type and operating pressure of the...
heat exchanger are taken into account as well. The “bare model cost” $C_{BM}$ is used to represent a more accurate capital cost estimation of each component, with considering both direct and indirect costs. It is calculated as follows:

$$C_{BM} = C_{B} + B_2 F_M F_P$$  \hspace{1cm} (41)

where $B_1$ and $B_2$ are constants for bare module cost factor which vary by different types of equipment. $B_1$ and $B_2$ are set as 0.96 and 1.21 for heat exchanger, as 1.89 and 1.35 for working fluid pump and seawater pump, respectively. $F_M$ is the material correction factor which has the value of 4.6 for heat exchangers and 2.36 for pumps. $F_P$ is the pressure factor and set as 1.0 [26].

The actual cost can be calculated from the “bare model cost” of year 2001 by referring to the Chemical Engineering Plant Cost Index (CEPCI) [26]. The cost of year 2017 can be estimated as [31]:

$$C_{BM,2017} = \frac{C_{BM,2001} \cdot CEPCI_{2017}}{CEPCI_{2001}}$$  \hspace{1cm} (42)

where CEPCI$_{2017}$ is 623.5, and CEPCI$_{2001}$ is 397 [32]. The total capital cost is calculated as:

$$C_{tot} = C_{BM,eva} + C_{BM,con} + C_{BM,turb} + C_{BM,pump} + C_{BM,swpump} + C_{BM,swpipe}$$  \hspace{1cm} (43)

The capital recovery factor (CRF) is expressed as:

$$CRF = \frac{i(1+i)^LT}{(1+i)^LT-1}$$  \hspace{1cm} (44)

where $i$ is annual loan interest rate which is set as 4.9% [33], and $LT$ denotes the life cycle time which is set as 20 [28].

Consequently, the LCOE, which is defined as an evaluation criterion of thermal-economic performance, can be calculated as follows [33,34]:

$$LCOE = \frac{C_{tot} \cdot CRF + OMC}{t_{op} \cdot W_{net}}$$  \hspace{1cm} (45)

where OMC (operation & maintenance cost) is set as 1.5% of the investment cost [28], $t_{op}$ is the operating time which is set as 7000 h, considering the impact of natural disasters such as typhoon.

4. Multi-objective optimization

4.1. Methods of optimization and decision-making

Compared with the single-objective optimization, multi-objective optimization can be adopted for multiple criteria decision making and to analyze the integrated performance based on chosen objective functions. When there are optimal decisions need to be taken under the condition of trade-offs between two or more conflicting objectives, multi-objective optimization is able to quantify the trade-offs in satisfying the different objectives, and figuring out a single solution that meets the subjective preferences or objective information.

A multi-objective optimization problem can be expressed as:

$$\min (\max) F(X) = [f_1(X), f_2(X), f_3(X), \ldots, f_n(X)]^T$$  \hspace{1cm} (46)

which must be subject to constraint equations as follows:

$$g_i(X) \leq 0, \ i = 1, 2, \ldots, m$$  \hspace{1cm} (47)

where $X$ is decision vector, $F(X)$ represents objective function, and $m$ is the number of constraint equations.

Among various multi-objective optimization methods, the multi-objective particle swarm optimization (MOPSO) [35–37], which is developed by particle swarm optimization (PSO), is a typical evolutionary algorithm for optimization problems. As there are some disadvantages in the original MOPSO due to high computational complexity and slow convergence speed, the crowding-distance and elite strategy of Non-dominated Sorting Genetic Algorithm II (NSGA-II) is introduced to improve the algorithm performance of MOPSO, by which satisfactory solutions can be uniformly distributed on the set of Pareto frontier.

There is a suitable decision-making method for Pareto frontier, i.e., the Linear Programming Techniques for Multi-dimensional Analysis of Preference (LINMAP), which is the classical method for addressing Multiple-criteria decision-making (MCDM) problems with preference information over given alternatives. Based on pairwise comparisons of alternatives, LINMAP [38,39] produces the best compromise alternatives as the solution nearest to the ideal solution. The ideal solution can be decided as a reference point with the best value of objective functions simultaneously. Generally, since the ideal solution cannot be obtained, Pareto optimal solution (POS) is deemed as desired solution, which is the nearest solution in Pareto frontier set based on geometric distance after normalization of objective function value. LINMAP is introduced for decision-making in order to find out the best solution for each working fluid. The schematic diagram of Pareto frontier combined with LINMAP is shown in Fig. 4.

The geometric distances between each solution in Pareto frontier set and ideal point are expressed as follow, and the solution with the minimum value of geometric distance is desired and marked by POS.

$$i_{POS} = \arg \min_{i} \left( \sum_{j=1}^{n} (f_j - f_{j,\text{Ideal}})^2 \right)$$  \hspace{1cm} (48)

where $f_{j,\text{Ideal}}$ is the optimum solution in a $j$th single-objective optimization; subscript $i$ is the number of each solution in Pareto frontier set.

4.2. Objective functions and decision variables

The difficulties in promotion of OTEC technology are mainly due to
two aspects: the cost and exergy efficiency of the OTEC system. The former problem lies in the large heat transfer area of heat exchanger and relatively low net power output of system. Moreover, the heat exchanger is made of titanium to prevent seawater corrosion, which in turn leads to an additional investment. On the other hand, OTEC system has a relatively low temperature difference for waste heat recovery, resulting in large heat exchange area. As a result, present studies are focused on how to increase exergy efficiency while decrease LCOE simultaneously.

In terms of decision variables, six key parameters are chosen as decision variables for OTEC optimization, including evaporating temperature $T_{eva}$, condensing temperature $T_{con}$, warm seawater outlet temperature of evaporator $T_{wwo}$, cool seawater outlet temperature of condenser $T_{cwo}$, degree of superheat $T_{sph}$, and depth of cool seawater $Depth$. The vector of decision variables is thus expressed as follows:

$$X = [T_{eva}, T_{con}, T_{wwo}, T_{cwo}, T_{sph}, Depth]^T$$

Note that these decision variables are mutually independent.

4.3. Constraints

In the optimization, following constraints should be taken into account:

1. The terminal temperature difference of plate heat exchangers should be no less than design requirement.
2. The pinch point temperature difference (PPTD) is subject to the heat exchange requirements of plate heat exchangers.
3. The steam quality of turbine exhaust should be larger than 0.92.
4. Each decision variable is subject to their lower and upper boundaries.

5. Results and discussion

5.1. Correlation of ocean temperature and depth

The temperature and depth of seawater in this object area (14°48′N, 110°54′E) of the present study are obtained from the Global Marine
The correlation between temperature and depth can be obtained by means of cubic spline integration. Fig. 5 illustrates a negative correlation between temperature and depth. It is seen that below 600 m of ocean depth, the temperature sharply decreases with depth. When the depth is over 600 m, the temperature is reduced to below 7 °C at a slower pace. When the depth is 800 m, the temperature is below 5 °C. When the depth is beyond 1200 m, the temperature is about 3 °C almost without change. The correlation between depth and temperature is obtained and presented in Fig. 5 by the Cubic Spline Interpolation.

The depth of seawater not only relates to the inlet temperature of cool seawater, but also obviously influences the length of pipelines used for pumping the deep cool seawater. In order to simplify the present model, a piecewise linear correlation between pipeline length and seawater depth has been considered [40,41]. When the seawater depth is shallower than 600 m, the pipeline length is almost without change. The correlation between depth and temperature is obtained and presented in Fig. 5 by the Cubic Spline Interpolation.

The temperature of warm surface seawater (intake warm seawater) is 28 °C and the flowrate of warm seawater is constant. R717, R152a, R134a, R227ea, R600a and R601 are selected as working fluids of the OTEC system. For the purpose of validation, the performance of OTEC is calculated with the present model based on validated data given by previous literatures [10,11,13,41]. The parameters adopted are summarized in Table 2(a). The boundaries of decision variables are shown in Table 2(b).

5.3. Influence of decision variables on system performance

The variations of LCOE & exergy efficiency with evaporating temperature (T_{evw}) and condensing temperature (T_{cw}) for six different working fluids are illustrated in Fig. 6. It can be observed that the trend of variations for six working fluids is similar to each other. As evaporating temperature rises or condensing temperature declines, the exergy efficiency will continuously improve, while LCOE first decreases and then increases. This is because LCOE is significantly affected by both the net power output and heat exchange area. With the increase of evaporating temperature or the decrease of condensing temperature, these two characteristics increase at different paces. Hence, LCOE tends to first decrease and then increase. Moreover, the LCOE of R717 is significantly less than other five working fluids due to the smallest heat exchange area caused by the highest latent heat of phase change and thermal conductivity of R717. In addition, the exergy loss keeps reducing with the increase of evaporating temperature (or condensing temperature declines), which benefits the exergy efficiency. As described in Fig. 6, the exergy efficiencies of six working fluids do not differ from each other obviously (about 1%). For better illustration, the partially enlarged views of six working fluids’ exergy efficiencies present in Figs. 6 and 7.

The variations of LCOE & exergy efficiency with warm seawater temperature at evaporator outlet (T_{wwo}) and cool seawater temperature at the outlet of condenser (T_{cwo}) are described in Fig. 7. It can be found that the exergy efficiency increases with the decrease of T_{wwo} or the increase of T_{cwo} the variation of both results in a decrease at first and an increase afterwards in LCOE. The increase of T_{wwo} (decrease of T_{cwo}) leads to more exergy losses in the evaporator (condenser), resulting in a reduction of exergy efficiency. Another reason is that when T_{wwo} increases, the enthalpy change of warm seawater becomes smaller, resulting in a declined turbine power output. However, the reducing amount of heat exchange in the evaporator will cause a decrease in heat exchange area and cool seawater flowrate, leading to less power consumption of seawater pump. As a result, LCOE descends initially, and when the turbine power output decreases at a faster pace than that of the pump power consumption, the net power output starts to reduce.
The accumulated effects of these aforementioned factors contributes to the LCOE trend of change, i.e., decreases at first and increases later. Note that LCOE has relatively less change when $T_{\text{wwo}}$ exceeds 25°C. Moreover, the variation of LCOE with $T_{\text{sph}}$ is relatively small compared with the variation with $T_{\text{wwo}}$, which is caused by the change of LMTD resulting in the significant change of heat transfer area in condenser. Comparing the performance of six working fluids in Fig. 7, the LCOE of R717 is significantly better than that of others. However, the exergy efficiencies of six working fluids do not differ from each other obviously.

The variations of LCOE and exergy efficiency with degree of superheat ($T_{\text{sph}}$) are illustrated in Fig. 8. It can be observed that, with an increasing degree of superheat, LCOE tends to first decrease and then increase. The trends of change for exergy efficiency between wet and dry working fluids are complete opposite, showing a positive correlation for the wet working fluids and negative correlation for the dry ones. The reason of $T_{\text{sph}}$'s effect on LCOE is similar to four decision variables. Furthermore, with increasing $T_{\text{sph}}$, the exergy loss in cooling process increases, but the losses in turbine and evaporator reduces. However, the former loss has greater effects than the latter two losses, resulting in different trends of change of exergy efficiency for wet and dry working fluids. Note that the influence of $T_{\text{sph}}$ on LCOE and exergy efficiency are extremely smaller than the influence of other decision variables.

Fig. 9 demonstrates the variations of LCOE and exergy efficiency with the depth of cool seawater. It can be seen that the increase of this decision variable generates a positive influence on exergy efficiency, and contributes to a decreasing trend of LCOE for working fluids except R717 for which LCOE decreases initially and increases afterwards. As the depth increases, the temperature of cool seawater keeps decreasing, causing a reduction in exergy loss in condenser. Beyond the thermocline (about 600 m), temperature declines marginally with depth, while pipeline becomes longer, resulting in higher investment cost. Meanwhile, the pressure drop in pipeline increases, resulting in a lower net power output of system. As a result, LCOE first decreases and then keeps stable or slightly increases.

### 5.4. Bi-objective optimization for OTEC based on Pareto frontier solution

Based on MOPSO, the bi-objective optimization is conducted via maximizing exergy efficiency and minimizing LCOE simultaneously.

### Table 3

<table>
<thead>
<tr>
<th>Working fluid</th>
<th>Optimal value</th>
<th>Design variables</th>
<th>Pareto optimization</th>
<th>$W_{\text{net}}$ (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R717</td>
<td>$LCOE_{\text{mn}}$</td>
<td>22.46 10.61 25.61 7.87 1.11 945 18.50 0.251</td>
<td>–</td>
<td>385.53</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{ex \text{mn}}} %$</td>
<td>23.82 11.90 24.16 10.88 1.98 949 32.05 0.554</td>
<td>–</td>
<td>664.02</td>
</tr>
<tr>
<td></td>
<td>POS</td>
<td>23.17 11.05 24.74 9.63 1.54 999 28.17 0.341</td>
<td>–</td>
<td>564.58</td>
</tr>
<tr>
<td>R152a</td>
<td>$LCOE_{\text{mn}}$</td>
<td>22.08 11.23 25.82 7.96 1.52 953 16.51 0.355</td>
<td>–</td>
<td>318.94</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{ex \text{mn}}} %$</td>
<td>23.77 11.46 24.00 10.95 2.19 959 33.14 1.144</td>
<td>–</td>
<td>700.55</td>
</tr>
<tr>
<td></td>
<td>POS</td>
<td>22.96 11.39 24.67 10.25 2.50 960 27.49 0.526</td>
<td>–</td>
<td>547.84</td>
</tr>
<tr>
<td>R134a</td>
<td>$LCOE_{\text{mn}}$</td>
<td>21.95 10.69 25.89 7.62 1.76 948 15.58 0.375</td>
<td>–</td>
<td>310.46</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{ex \text{mn}}} %$</td>
<td>23.57 11.36 23.71 10.95 2.39 949 33.10 1.284</td>
<td>–</td>
<td>730.03</td>
</tr>
<tr>
<td></td>
<td>POS</td>
<td>22.90 11.30 24.72 10.21 2.26 953 26.88 0.549</td>
<td>–</td>
<td>532.42</td>
</tr>
<tr>
<td>R227ea</td>
<td>$LCOE_{\text{mn}}$</td>
<td>21.72 11.52 25.88 8.50 1.66 958 16.34 0.438</td>
<td>–</td>
<td>290.27</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{ex \text{mn}}} %$</td>
<td>22.85 11.07 23.90 10.92 2.88 977 33.33 1.816</td>
<td>–</td>
<td>716.83</td>
</tr>
<tr>
<td></td>
<td>POS</td>
<td>22.79 11.21 24.65 10.49 2.46 958 26.75 0.666</td>
<td>–</td>
<td>543.52</td>
</tr>
<tr>
<td>R600a</td>
<td>$LCOE_{\text{mn}}$</td>
<td>21.85 10.81 25.96 7.73 0.98 941 15.69 0.378</td>
<td>–</td>
<td>300.60</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{ex \text{mn}}} %$</td>
<td>23.92 11.31 24.12 10.94 1.23 961 33.50 1.392</td>
<td>–</td>
<td>695.83</td>
</tr>
<tr>
<td></td>
<td>POS</td>
<td>22.89 11.45 24.61 10.24 1.84 980 27.24 0.571</td>
<td>–</td>
<td>548.75</td>
</tr>
<tr>
<td>R601</td>
<td>$LCOE_{\text{mn}}$</td>
<td>21.93 11.22 25.67 7.98 0.10 941 16.90 0.333</td>
<td>–</td>
<td>339.52</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{ex \text{mn}}} %$</td>
<td>23.45 11.30 23.67 10.98 2.52 983 34.18 1.342</td>
<td>–</td>
<td>757.65</td>
</tr>
<tr>
<td></td>
<td>POS</td>
<td>23.08 11.25 24.75 10.31 2.34 978 28.47 0.523</td>
<td>–</td>
<td>556.63</td>
</tr>
</tbody>
</table>
Considering the speed of optimization and the diversity of calculated results, the population size is set as 100 and the number of generations is set as 200. Moreover, crowding-distance and elitist strategy of Non-dominated Sorting Genetic Algorithms 2 (NSGA-2) are introduced to improve the algorithm performance of MOPSO. The six parameters shown in Eq. (49) act as decision variables. The entire solution process must be subject to constraint conditions.

A Pareto frontier is shown in Fig. 10, which is a set of non-dominated solutions for each working fluid. The Pareto frontier for OTEC system shows trade-offs between two objective functions. The relationship between LCOE and exergy efficiency is illustrated as a curved line convex to ideal point (IPD), which means that the cost of improvement of one objective function will cause deterioration of the other objective function.

As plotted in Fig. 10, there are various non-dominated solutions, which assemble the Pareto frontier set. To help searching the Pareto frontier, Multi-dimensional Analysis of Preference (LINMAP) has been introduced as a standard of performance to compare six different working fluids. The nearest point to the ideal point (based on geometric distance) in Pareto frontier set is defined as Pareto optimal solution (POS), which is selected as a desired solution. Note that LCOE and exergy efficiency have different dimensions and units, so that both of them need to be normalized, based on which the position of POS can be therefore determined. The optimization results of POS, maximum exergy efficiency and minimum LCOE for six working fluids are listed in Table 3.

The non-dominated ranking method can be used not only in optimization algorithm, but also for performance comparison among different working fluids. As can be seen in Table 3, based on the values of POS for six working fluids, R601 (Pentane) has the highest exergy efficiency. The LCOE of R601 is lower than R717 at the lower but higher than the others, while exergy efficiency of R717 is lower than R601 but higher than the rest. Hence, R717 and R601 tie for first place. In addition, R152a ranks second with better LCOE and exergy efficiency than rest three working fluids. Furthermore, the LCOE of R134a is lower than that of R600a, but its exergy efficiency is worse than that of R600a. Hence, both of them tie for third place. Among all six working fluids, R227ea performs the worst in both terms of exergy efficiency and LCOE. Therefore, R717 and R601 perform best, followed by R152a, while R134a and R600a have relatively poor performance, and R227ea shows the worst performance.

Note that R717 (Ammonia) has the highest latent heat of phase change and thermal conductivity, resulting in smallest heat exchange area (much smaller than that of other working fluids), which in turn leads to lowest LCOE. It is worth mentioning that wet working fluids such as R717, R152a and R134a generally have higher risks of turbine failure. Therefore, the quality of turbine exhaust must be strictly controlled in the practical engineering design of OTEC system. Moreover, based on the net power of each optimization result, the OTEC system is sensitive to working temperature, which means small temperature variation may cause big difference between net powers.

In order to analyze the correlation between LCOE and exergy efficiency among these working fluids, six polynomial curves have been fitted accordingly. The ranges of two objective functions and fitting curve equations are available in Table 4.

## Table 4
Fitted curves of Pareto solution set for six working fluids.

<table>
<thead>
<tr>
<th>Working fluid</th>
<th>Fitting equation</th>
<th>Range of $\eta_{ex}$</th>
<th>Range of LCOE</th>
</tr>
</thead>
<tbody>
<tr>
<td>R717</td>
<td>$\text{LCOE} = 268.77\eta_{ex}^3 - 184.32\eta_{ex}^2 + 42.51\eta_{ex} - 3.02$</td>
<td>15.30–18.00%</td>
<td>0.251–0.554</td>
</tr>
<tr>
<td>R152a</td>
<td>$\text{LCOE} = 437.04\eta_{ex}^3 - 592.92\eta_{ex}^2 + 66.12\eta_{ex} - 4.61$</td>
<td>16.31–16.82%</td>
<td>0.305–1.144</td>
</tr>
<tr>
<td>R134a</td>
<td>$\text{LCOE} = 433.18\eta_{ex}^3 - 289.60\eta_{ex}^2 + 63.60\eta_{ex} - 4.26$</td>
<td>15.58–33.10%</td>
<td>0.375–1.284</td>
</tr>
<tr>
<td>R227ea</td>
<td>$\text{LCOE} = 900.32\eta_{ex}^3 - 619.98\eta_{ex}^2 + 142.28\eta_{ex} - 10.35$</td>
<td>16.43–33.33%</td>
<td>0.438–1.816</td>
</tr>
<tr>
<td>R600a</td>
<td>$\text{LCOE} = 528.62\eta_{ex}^3 - 353.71\eta_{ex}^2 + 79.39\eta_{ex} - 5.50$</td>
<td>15.69–33.50%</td>
<td>0.378–1.392</td>
</tr>
<tr>
<td>R601</td>
<td>$\text{LCOE} = 702.45\eta_{ex}^3 - 503.62\eta_{ex}^2 + 120.30\eta_{ex} - 9.14$</td>
<td>16.90–34.18%</td>
<td>0.333–1.342</td>
</tr>
</tbody>
</table>

### 6. Conclusions

This study has established a multi-objective optimization model for Organic Rankine Cycle (ORC) of Ocean Thermal Energy Conversion (OTEC) system. The Multi-Objective Particle Swarm Optimization (MOPSO) algorithm has been used to solve the model equations with considering six decision variables, i.e., evaporating temperature, condensing temperature, warm seawater temperature at evaporator outlet, cool seawater temperature at condenser outlet, degree of superheat, and depth of cool seawater. The trade-offs between levelized cost of energy (LCOE) and exergy efficiency have been analyzed by plotting the Pareto frontier. Furthermore, Linear Programming Techniques for Multi-dimensional Analysis of Preference (LINMAP) has been introduced to identify Pareto-optimal solution (POS). The influence of six decision variables on two objectives have been analyzed and compared with each other. At last, the performance of six working fluids, i.e., R717, R152a, R134a, R227ea, R600a and R601, have been compared and ranked by using the non-dominated ranking method. The major conclusions that have been drawn are as follows:

1. The Pareto frontiers for six working fluids have been presented, and the trade-off optimization has been conducted for two contradictory objectives, i.e., LCOE and exergy efficiency. The improvement of one objective function leads to the deterioration of the other objective function. Combining MOPSO algorithm and LINMAP decision making method, the Pareto-optimal solution (POS) has been obtained from Pareto frontier set.

2. With the increase of cool seawater depth, LCOEs of six working fluids will all first decrease and then keep stable or slightly increase. As other five decision variables change individually, LCOE has a corresponding optimal value. In general, the evaporating temperature, cool seawater temperature at condenser outlet, and depth of cool seawater have positive influence on exergy efficiency. On the contrary, the condensing temperature and warm seawater temperature at evaporator outlet both have negative influence on exergy efficiency. Meanwhile, it is observed that the degree of superheat has positive correlation with exergy efficiency for wet working fluids and negative correlation with exergy efficiency for dry working fluids.

3. The ranking of six working fluids in terms of LCOE and exergy efficiency is as follows: R717 (28.17% and 0.333 $/kWh) and R601 (28.47% and 0.523 $/kWh) have the best performance, followed by R152a (27.49% and 0.526 $/kWh), while R134a (27.68% and 0.549 $/kWh) and R600a (27.24% and 0.571 $/kWh) have relatively poor performance, and R227ea (26.75% and 0.666 $/kWh) is the least desirable.

4. The restrictive correlation between LCOE and exergy efficiency have been obtained based on curve fitting to Pareto frontier set, which can provide technical guidance for the practical design of OTEC systems.
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