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Research paper

Parallel and in-series arrangements of zeotropic dual-pressure Organic Rankine Cycle (ORC) for low-grade waste heat recovery

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Cycle evaluation
Power generation

A B S T R A C T

Due to the recent interest in renewable energy and waste heat for power generation, organic Rankine cycle (ORC) has drawn considerable attention. By applying a dual-pressure evaporation strategy, the exergy loss in the evaporators can be reduced significantly, resulting in an enhanced system performance. In this study, detailed modelling analyses have been conducted to evaluate the effect of zeotropic working fluid on the performance of two dual-pressure ORC (DPORC) arrangements, namely, DPORC-P (parallel) and DPORC-S (in-series), where the working fluid pair R245fa and R152a has been selected to form a representative zeotropic mixture. It has been found that as a zeotropic mixture is used, both systems show improved cycle net power outputs. Interestingly, the highest power outputs occur at mass fractions that result in matching temperature profiles between the hot and cold fluids in the condenser. At a heat source temperature of 120 °C, a DPORC-S always has a greater net power output regardless of the mass fraction of the zeotropic mixture; as heat source temperature reduces, the difference of power output between the two systems tends to drop: for a heat source temperature of 90 °C, a DPORC-P has a better performance as the mixture mass fraction is between 0.4 and 0.9. The influence of zeotropic fluids to the heat transfer of the condenser has also been investigated and it indicates that a significantly larger heat transfer area is required for a zeotropic fluid; based on a fixed heat transfer area, the performance enhancement of a zeotropic system is only seen as the condenser heat transfer area is considerably large.

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1. Introduction

Due to the shortage of fossil fuel globally and growing concern over environmental issues, there have been increasing interests in developing technologies to achieve power generation via clean energy sources, e.g. renewable energy or waste heat from various industrial or domestic processes (Zhai et al., 2016). Compared to conventional approaches, energy in above sources is mostly stored in the form of low-grade heat which is unable to be converted into electricity effectively using a standard steam power cycle (Tchanche et al., 2011). Organic Rankine Cycle (ORC), on the other hand, has drawn much attention recently (Hung et al., 1997; Quoilin et al., 2011). Compared to traditional steam power cycles, an ORC uses working fluids with a considerably lower boiling point than water. It is a promising technology for low temperature heat recovery and offers advantages of a simple configuration, straightforward maintenance and feasibility of system scale-down (Rahbar et al., 2017).

A significant number of studies has been conducted with the focus on working fluid selection (Xu et al., 2018; Drescher and Brüggemann, 2007; Badr et al., 1985), ORC expander design and testing (Fiaschi et al., 2012; Imran et al., 2016; Da Lio et al., 2016), energy, exergy (Sun et al., 2017) and economic (Quoilin et al., 2011) analysis. An important source of system irreversibility in a standard ORC is the mismatch of temperature profiles between the working fluid and heat source/sink, or so-called the “pinch problem” (Chen et al., 2010). In the evaporator, the heat source temperature decreases continuously, however, the ORC working fluid temperature almost remains constant throughout the heat transfer process, creating a “pinch point” between the two temperature profiles. A similar effect is seen in the condenser. Mago et al. (2008) have identified that it could result in an exergy destruction rate in the evaporator as high as 77%. An effective way to minimise irreversibility in the evaporator is to apply dual-pressure evaporation configurations (Du et al., 2018; Li et al.,...
In a dual-pressure Organic Rankine Cycle (DPORC), the heat is absorbed in two evaporators: the high-pressure (HP) evaporator and low-pressure (LP) evaporator. An HP and LP pumps are required to deliver the fluid to the HP and LP evaporator, respectively. Depending on the arrangement of the pumps, two configurations are possible: the DPORC parallel (DPORC-P) and the DPORC in-series (DPORC-S), as shown in Figs. 1–2. For DPORC-P, the liquid working fluid is pumped separately to the HP and LP evaporator from the condenser; for DPORC-S, all liquid fluid from the condenser is pumped to the LP evaporator first and a portion of the liquid from the LP evaporator is then further pumped to the HP evaporator. For both configurations, the heat source flow firstly goes through the HP evaporator, causing the liquid fluid to evaporate at a high temperature; exiting from the HP evaporator, the heat source flow then enters the LP evaporator where the evaporation takes place at a relatively lower temperature. An HP and LP expanders are required to convert heat into work. DPORCs allow more heat to be absorbed and creates an improved temperature matching between the heat source and ORC working fluid.

Several relevant studies can be seen in the literature. Gnutek and Byszewska-Mazurek (2001) has briefly discussed the concept in 2001 using R123 (2,2-dichloro-1,1,1-trifluoroethane) as the working fluid. Kanoglu (2002) proposed to apply DPORC-P on a geothermal power plant. Franco and Villani (2009) have carried out modelling study to optimise DPORC systems for geothermal fields at different geothermal fluid temperatures, condensing temperatures and working fluids. DiGenova et al. (2013) considered using DPORCs to recover heat from a complex heat source for efficient energy conversion. Guzović et al. (2014) has presented a case study to compare the performance between a simple ORC and DPORC-S and found the latter had considerably higher net power generation. Stijepovic et al. (2014) examined dual- and three-pressure configurations and optimisation of different configurations was performed using an inclusive objective function. Shokati et al. (2015) compared the energy, exergy and exergy-economic performance of DPORC-S with other cycles and found it had the highest electricity generation although economically, the cost per kW power generated was high due to the use of an additional expander in the system. Recently, Li et al. (2015, 2016) compared the operation of both DPORC-P and DPORC-S and concluded that the latter had a better performance. Manente et al. (2017) developed design guidelines for the choice of simple and DPORC layouts and results showed that the benefit from a dual-pressure configuration is particularly high at low temperature heat sources (100–125 °C) and gradually diminished at higher temperature (150–200 °C). Li et al. (2018b) carried out thermo-economic performance analyses among DPORC systems with different expander layouts and found that the option with an induction expander layout achieved a higher performance. They later investigated the correlation between heat source temperature, working fluid critical temperature and system performance for DPORCs and results showed that the applicable heat source temperature range of a DPORC generally increased as the working fluid critical temperature increased (Li et al., 2018a). The possibility of DPORCs with a subcritical and supercritical heat absorption process has also been discussed (Li et al., 2019a). Du et al. (2018) developed a detailed mathematical model to study the influence of heat source temperature and flow rate to the system off-design performance.

Above literatures demonstrated that both DPORC-P and DPORC-S would achieve a significant performance enhancement compared to a standard ORC, due to the reduction of irreversibility in the evaporators. However, the discussions are mainly focused on pure working fluids and it is possible that a further improvement of the cycle performance can be obtained via the use of zeotropic working fluids (Chys et al., 2012). A zeotropic fluid is a type of fluid mixtures. Unlike pure working fluid, a zeotropic working fluid exhibits non-isothermal characteristics.

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area (m²)</td>
</tr>
<tr>
<td>Bo</td>
<td>boiling number (-)</td>
</tr>
<tr>
<td>D_h</td>
<td>hydraulic diameter (m)</td>
</tr>
<tr>
<td>e</td>
<td>specific exergy (kJ kg⁻¹)</td>
</tr>
<tr>
<td>˙E</td>
<td>exergy flow (kW)</td>
</tr>
<tr>
<td>f</td>
<td>Fanning friction factor (-)</td>
</tr>
<tr>
<td>G</td>
<td>mass flux (kg s⁻¹ m⁻²)</td>
</tr>
<tr>
<td>h</td>
<td>specific enthalpy (kJ kg⁻¹)</td>
</tr>
<tr>
<td>˙i</td>
<td>exergy loss (kW)</td>
</tr>
<tr>
<td>J</td>
<td>convection heat transfer coefficient (W m⁻² K⁻¹)</td>
</tr>
<tr>
<td>˙m</td>
<td>mass flow rate (kg/s)</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number (-)</td>
</tr>
<tr>
<td>p</td>
<td>pressure (kPa)</td>
</tr>
<tr>
<td>p_0</td>
<td>ambient pressure (kPa)</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number (-)</td>
</tr>
<tr>
<td>Q_0</td>
<td>heat transfer rate (kW)</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number (-)</td>
</tr>
<tr>
<td>s</td>
<td>specific entropy (kJ kg⁻¹ K⁻¹)</td>
</tr>
<tr>
<td>T</td>
<td>temperature (°C)</td>
</tr>
<tr>
<td>T_0</td>
<td>ambient temperature (°C)</td>
</tr>
<tr>
<td>ΔT_lmtd</td>
<td>log-mean temperature difference (K)</td>
</tr>
<tr>
<td>U</td>
<td>overall heat transfer coefficient (W m⁻² K⁻¹)</td>
</tr>
<tr>
<td>W</td>
<td>work or power generation (kW)</td>
</tr>
<tr>
<td>x</td>
<td>mass fraction (-)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Greek symbols</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>η</td>
<td>efficiency (-)</td>
</tr>
<tr>
<td>λ</td>
<td>thermal conductivity (W m⁻¹ K⁻¹)</td>
</tr>
<tr>
<td>μ</td>
<td>dynamic viscosity (Pa s)</td>
</tr>
<tr>
<td>ρ</td>
<td>density (kg m⁻³)</td>
</tr>
<tr>
<td>χ</td>
<td>vapour quality (-)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Subscripts</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>c</td>
<td>condenser</td>
</tr>
<tr>
<td>cw</td>
<td>cooling water</td>
</tr>
<tr>
<td>e</td>
<td>evaporator</td>
</tr>
<tr>
<td>in</td>
<td>inlet</td>
</tr>
<tr>
<td>is</td>
<td>isentropic process</td>
</tr>
<tr>
<td>out</td>
<td>outlet</td>
</tr>
<tr>
<td>p</td>
<td>pump</td>
</tr>
<tr>
<td>pp</td>
<td>pinch-point</td>
</tr>
<tr>
<td>s</td>
<td>heat source</td>
</tr>
<tr>
<td>t</td>
<td>turbine/expander</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Abbreviations</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>DPORC-S</td>
<td>dual-pressure organic Rankine cycle in series</td>
</tr>
<tr>
<td>DPORC-P</td>
<td>dual-pressure organic Rankine cycle parallel</td>
</tr>
<tr>
<td>GWP</td>
<td>global warming potential</td>
</tr>
<tr>
<td>HP</td>
<td>high-pressure</td>
</tr>
<tr>
<td>LP</td>
<td>low-pressure</td>
</tr>
<tr>
<td>ODP</td>
<td>ozone depletion potential</td>
</tr>
<tr>
<td>ORC</td>
<td>organic Rankine cycle</td>
</tr>
</tbody>
</table>

2631
during the evaporation and condensation processes. This can potentially lead to an improved temperature matching between the hot and cold fluids for both evaporator and condenser (counter-current heat exchangers). The benefit of using zeotropic mixtures in a simple ORC has long been realised and explored extensively in the past (Modi and Haglind, 2017). Previous studies indicated that a reduction of exergy loss was achieved in both condenser and evaporator, and the impact in the condenser is more significant (Heberle et al., 2012; Liu et al., 2014). According to the above knowledge, a highly improved ORC performance is possible through the integration of DPORCs and zeotropic ORCs so that exergy losses in both evaporator and condenser are minimised. This concept has not been fully explored in the literature. Sadeghi et al. (2016) examined both DPORC-P and DPORC-S briefly using zeotropic fluids. However, their analyses were based on predefined mixtures and there was no indication of the mass fraction effect to the system performance. Luo et al. (2018) has used zeotropic mixtures containing R245fa (1,1,1,3,3-pentafluoropropane) to study an ORC system coupling liquid separation condensation and dual-pressure evaporation although in-depth discussions about the coupling effect of zeotropic fluids and dual-pressure evaporation were not provided. Li et al. (2019c) demonstrated the potential of using zeotropic mixtures to further enhance the performance of DPORC-S employing flue gas from gas turbine as the heat source; however, the study did not give sufficient explanations about the results via the variations of component exergy losses, or the changes of the temperature matching in the condenser/evaporator.

According to above studies, the concept of using zeotropic mixtures in DPORCs is worth to be noted. Nevertheless, there are still lack of knowledge about this topic, including (1) detailed analysis of the coupling effect of DPORC and zeotropic mixtures via temperature matching in the condenser/evaporator, as well as the exergy losses of each individual components; (2) influence of zeotropic mixtures to both DPORC-P and DPORC-S and their relative performance enhancement. It is crucial that above concerns to be addressed so that relevant cycle performance can be better estimated, and proper cycle arrangements are used for specific applications.

To fill the knowledge gaps, this study aims to develop a comprehensive model to analyse both DPORC-S and DPORC-P, using zeotropic working fluids. The performance variations of the cycles will be investigated from various aspects, such as the visualisation of the condenser/evaporator temperature matching and component-based exergy analysis. The two cycle arrangements will be evaluated under different heat source temperature and zeotropic mass fractions to demonstrate their advantages and limitations.

2. System description

The layouts of a dual-pressure ORC parallel (DPORC-P), dual-pressure ORC in-series (DPORC-S) and the corresponding temperature-entropy (T-s) diagrams are presented in Figs. 1–2. A basic ORC consists of an expander, a condenser, a pump and an evaporator. It absorbs heat in the evaporator, producing high-pressure vapour which is fed to the expander to generate work before condensed back to liquid in the condenser. DPORC-P has an extra pump, evaporator and expander. In this system, evaporation of the working fluid takes place in both evaporators, however, at different pressure levels. Two ORC pumps are arranged in parallel, so part of the liquid is pumped to the HP evaporator and the rest to the LP evaporator. When a zeotropic mixture is used, the mixture mass fraction is always identical in all components. During the evaporation and condensation, the temperature variation, or gliding, is evident in the T-s diagram.

DPORC-S is a relatively more complex system because it involves a liquid–vapour separation process. After the working fluid is condensed in the condenser, all liquid is pump to the LP evaporator. Only a portion of the liquid is evaporated, and the rest of the working fluid remains in a liquid form although with an increased temperature. The LP evaporator acts as a liquid–vapour separator to allow the vapour portion to be sent to the LP expander and liquid portion to be further pumped to the HP evaporator for a second heating. When a zeotropic mixture is used here, a significant different compared to a DPORC-P is that the mass fraction at different components are not identical. For example, we would expect the vapour in the LP expander has a higher ratio of the low-boiling-point substance than the HP expander.

In this study, R245fa (1,1,1,3,3-pentafluoropropane) has been used as the base working fluid. R152a (1,2-difluorotane) is selected as the second fluid to form a zeotropic mixture pair. Relevant properties of the above fluid components are listed in Table 1. The above selection has taken the consideration of the fact that the thermodynamic properties of the two pure fluids differs considerably and a temperature glide from zero up to 20 K is seen as the R245fa mass fraction varies between 0 and 1, providing an ideal representation of zeotropic mixture-based evaporation and condensing processes. Also, both fluids have ozone depletion potential of zero (ODP), relatively low global warming potential (GWP) and mild flammability, offering a high feasibility for practical applications.

3. Modelling methodologies

3.1. Cycle modelling

Parameters used in the current cycle models are presented in Table 2. The heat source flow belongs to open type without outlet temperature limit. The following assumptions are established to simplify the model (Li et al., 2019a):

- All cycles operate in a steady-state condition.
- Pressure drop and heat dissipation in all heat exchangers and pipes are negligible.
- Effects of the fluid kinetic and gravity are negligible.
- The working fluid is always at a saturated vapour state as it enters the expander and a saturated liquid state as it leaves the condenser (Hung, 2001).

Mathematic models for each component are expressed by the following equations:

**Expander:**

\[ \eta_{h,t} = \frac{h_{t,\text{in}} - h_{t,\text{out},i}}{h_{t,\text{in}} - h_{t,\text{out}}} \]  

\[ W_e = m_t (h_{t,\text{in}} - h_{t,\text{out}}) \]  

where \( \eta_{h,t} \) denotes the isentropic efficiency of the expander; \( W_e \) denotes the kW power generation of the expander; \( h \) and \( m \) are the specific enthalpy and mass flow rate, respectively; subscriptions “i”, “t”, “out” and “in” denote expander/turbine, inlet, outlet and isentropic process, respectively.

**Evaporator:**

\[ \dot{Q}_e = m_e (h_{e,\text{out}} - h_{e,\text{in}}) = m_t (h_{t,\text{in}} - h_{t,\text{out}}) \]  

where \( \dot{Q}_e \) is the heat absorption of the evaporators; subscriptions “e” and “s” denote evaporator and heat source flow, respectively.

**Condenser:**

\[ \dot{Q}_c = m_c (h_{c,\text{in}} - h_{c,\text{out}}) = m_w (h_{w,\text{out}} - h_{w,\text{in}}) \]  

where the subscripts “c” and “cw” denote the condenser and cooling water, respectively.
Table 1
Properties of the selected working fluid components.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Molecular formula</th>
<th>Safety group&lt;sup&gt;a&lt;/sup&gt;</th>
<th>ODP</th>
<th>GWP</th>
<th>Normal boiling point (°C)</th>
<th>Critical temperature (°C)</th>
<th>Critical pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R245fa</td>
<td>C&lt;sub&gt;3&lt;/sub&gt;H&lt;sub&gt;5&lt;/sub&gt;F&lt;sub&gt;5&lt;/sub&gt;</td>
<td>B1</td>
<td>0</td>
<td>858</td>
<td>15.14</td>
<td>154.01</td>
<td>3651.0</td>
</tr>
<tr>
<td>R152a</td>
<td>C&lt;sub&gt;2&lt;/sub&gt;H&lt;sub&gt;4&lt;/sub&gt;F&lt;sub&gt;2&lt;/sub&gt;</td>
<td>A2</td>
<td>0</td>
<td>138</td>
<td>24.02</td>
<td>113.26</td>
<td>4516.8</td>
</tr>
</tbody>
</table>

*ASHRAE 34 safety group classification: 1: no flame propagation; 2: lower flammability; 3: higher flammability; A: lower toxicity; B: higher toxicity.

Fig. 1. Cycle layout (a) and T-s diagram (b) of a DPORC-P.

Fig. 2. Cycle layout (a) and T-s diagram (b) of a DPORC-S.

Table 2
Parameters used in the current modelling.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat source flow inlet temperature, T&lt;sub&gt;x,in&lt;/sub&gt; (°C)</td>
<td>90 to 120</td>
<td>–</td>
</tr>
<tr>
<td>Heat source flow rate, m&lt;sub&gt;x&lt;/sub&gt; (kg/s)</td>
<td>1</td>
<td>–</td>
</tr>
<tr>
<td>Expander isentropic efficiency, η&lt;sub&gt;ex&lt;/sub&gt; (%)</td>
<td>70%</td>
<td>Zhao and Bao (2014)</td>
</tr>
<tr>
<td>Pump isentropic efficiency, η&lt;sub&gt;p&lt;/sub&gt; (%)</td>
<td>70%</td>
<td>Manente et al. (2017)</td>
</tr>
<tr>
<td>Cooling water supply temperature, T&lt;sub&gt;c,w,in&lt;/sub&gt; (°C)</td>
<td>20</td>
<td>Li et al. (2019b)</td>
</tr>
<tr>
<td>Cooling water return temperature, T&lt;sub&gt;c,w,out&lt;/sub&gt; (°C)</td>
<td>25</td>
<td>Li et al. (2019b)</td>
</tr>
<tr>
<td>Pinch point temperature, evaporator, ΔT&lt;sub&gt;pp,e&lt;/sub&gt; (K)</td>
<td>5</td>
<td>Li et al. (2019b)</td>
</tr>
<tr>
<td>Pinch point temperature, condenser, ΔT&lt;sub&gt;pp,c&lt;/sub&gt; (K)</td>
<td>5</td>
<td>Li et al. (2019b)</td>
</tr>
<tr>
<td>Ambient pressure, p&lt;sub&gt;a&lt;/sub&gt; (atm)</td>
<td>1</td>
<td>Yari et al. (2015)</td>
</tr>
<tr>
<td>Ambient temperature, T&lt;sub&gt;a&lt;/sub&gt; (°C)</td>
<td>20</td>
<td>Dai et al. (2009)</td>
</tr>
</tbody>
</table>
Where \( \eta_{\text{in}},p \) and \( W_p \) denote the isentropic efficiency and power draw of the pump, respectively; the subscript “p” denotes pump.

Apart from above equations, the conservation of mass has been applied as additional boundary conditions to fully solve the problem.

A thermal efficiency of a cycle, or a first-law efficiency, is the most commonly used performance indicator for a cycle, which is defined as the ratio of the cycle net power output to the total heat absorption, as expressed below:

\[
\eta_i = \frac{W_{\text{net}}}{Q_c}
\]  

However, for heat recovery applications, it is not as suitable as an exergy efficiency, or a second-law efficiency, which evaluates precisely the ability of a cycle to convert available energy into usable work. The definition of the second-law efficiency can be expressed as

\[
\eta_{\text{II}} = \frac{W_{\text{net}}}{E_{\text{s,in}}}
\]

where \( W_{\text{net}} \) is the net power output of a cycle, which is the total of expander power generation subtracting the total power draw from pumps; \( E_{\text{s,in}} \) is the total exergy flow entering the system. Matlab 2018b (The MathWorks Inc., 0000) has been used as the mathematical platform for the modelling and all properties of pure fluids and mixtures have been obtained from REFPROP 9.0 (Lemmon et al., 2007).

### 3.2. Exergy analysis

Exergy analysis has been carried out in this study to gain further understandings to the cycle characteristics. Exergy loss from the heat source, as well as those from different cycle components, have been evaluated and relevant terms are listed below.

**Heat source:**

\[
i_s = \dot{m}_s e_{s,\text{out}}
\]

where \( i_s \) is the exergy loss from the heat source, or the remaining exergy in the heat source flow to be released to ambient; \( e_{s,\text{out}} \) is the specific exergy of the heat source flow to be released to ambient.

**Expander:**

\[
i_e = \dot{m}_e (s_{e,\text{out}} - s_{e,\text{in}})
\]

where \( i_e \) is the exergy loss from an expander and \( s \) is the specific entropy of the fluid.

**Evaporator:**

\[
i_e = T_0 [\dot{m}_c (s_{c,\text{in}} - s_{c,\text{out}}) + \dot{m}_e (s_{e,\text{in}} - s_{e,\text{out}})]
\]

where \( i_e \) is the exergy loss from an evaporator and \( T_0 \) is the ambient temperature in Kelvin (293.15 K in this study).

**Condenser:**

\[
i_e = \dot{m}_c (e_{c,\text{in}} - e_{c,\text{out}})
\]

where \( i_e \) is the exergy loss from a condenser.

**Pump:**

\[
i_p = \dot{m}_p T_0 (S_{p,\text{out}} - S_{p,\text{in}})
\]

where \( i_p \) is the exergy loss from a pump.
3.3.1. Single-phase flow

For single-phase flow, the following correlations of Focke et al. (1985) for plate heat exchangers are used:

\[ N_u = C_1 Re^m Pr^{0.5} \]

\[ C_1 = 0.77, \ m = 0.54 \text{ for } 120 < Re < 1000 \]

\[ C_1 = 0.44, \ m = 0.64 \text{ for } 1000 < Re < 42000 \]

\[ f = \frac{C_2}{Re^m} + C_3 \]

\[ C_2 = 57.5, \ C_3 = 0.093, \ n = 1.0 \text{ for } 0 < Re < 3000 \]

\[ C_2 = 0.8975, \ C_3 = 0, \ n = 0.263 \text{ for } 3000 < Re < 50000 \]

where \( N_u, Re \) and \( Pr \) are the Nusselt number, Reynolds number and Prandtl number of the flow.

Above expressions are applicable for both pure and zeotropic fluids.

3.3.2. Two-phase flow

Pure fluid:

The following equations are employed to calculate the Nusselt number and Fanning friction factor of the ORC flow during condensation (Kuo et al., 2005):

\[ Nu = 4.118 Re_{eq}^{0.4} Pr^{1/3} \]

\[ f = 21500Re_{eq}^{-1.14}Bo^{-0.085} \]

where \( Bo \) is the boiling number of the flow (\( Bo = Pb/(\phi PI) \)) and \( Re_{eq} \) is the equivalent Reynolds number which is calculated by the following equations:

\[ Re_{eq} = \frac{G_{eq} D_p}{\mu_{liq}} \]

\[ G_{eq} = G \left[ (1 - \chi_m) + \chi_m \left( \frac{\rho_{eq}}{\rho_{vap}} \right)^{0.5} \right] \]

where \( \mu \) and \( \rho \) are the dynamic viscosity and density of the fluid, respectively, with the subscripts "liq" and "vap" stand for liquid or vapour portion of the fluid, respectively; \( \chi_m \) is the mean vapour quality of the flow.

Zeotropic mixture:

For mixtures, Eq. (19) is still eligible and the thermodynamic properties for mixtures shall be used in this calculation. To calculate the heat transfer coefficient of the mixture, the following equations are used (Bivens and Yokozeki, 1994):

\[ J = \left( \sum_{i=1}^{n} \frac{X_i}{J_i} \right)^{-1} \]

\[ c = 0.85 - 0.01454 (T_d - T_b) \text{ for } G > 160 \]

\[ c = (0.10676 + 0.12483 \ln (G)) \left( 1.25 - 0.04545 (T_d - T_b) \right) \text{ for } G \leq 160 \]

where \( T_d \) and \( T_b \) is the dew point and bubble point temperature of the mixture; \( X_i \) is the mass fraction of component \( i \) in the mixture; \( J_i \) is the calculated convective heat transfer coefficient of assuming a pure fluid of component \( i \).

3.3.3. Calculation process

An iterative process is applied to determine the total heat transfer area of the heat exchanger, as indicated in the flow chart in Fig. 4. Firstly, an initial number of plates is assumed and Eqs. (16), (18) and (20)–(22) are used together with the given plate geometries in Table 2 to calculate the total pressure drop of the PHE; by comparing the calculated pressure drop with a 3% of the flow pressure (Hu et al., 2015), if the relative difference is within 0.1%, the iteration is converged; otherwise, an updated number of plates is used for the next iteration until the pressure drop target is achieved. A total heat transfer area is then calculated according to Eqs. (14), (15), (17), (19) and (21)–(23).

3.4. Iteration and optimisation processes

For both DPORC-P and DPORC-S, all cycle parameters can be calculated once the HP and LP evaporation temperatures \( T_{e,HP} \) and \( T_{e,LP} \) are given. An iterative approach is employed for both systems: for DPORC-P, an initial condensing temperature \( T_c \) is assumed and updated after each iteration based on the calculated \( T_c \), and the iteration does not terminate until the difference between the calculated and assumed \( T_c \) is sufficiently small; for DPORC-S, the vapour quality of the LP evaporation \( X_{e,LP} \) are also assumed, and the entire iteration process terminates after both \( T_c \) and \( X_{e,LP} \) are converged. \( T_{e,HP} \) and \( T_{e,LP} \) can be optimised for a maximised cycle performance at any given heat source temperature. The optimisation has been carried out using the genetic algorithm in Matlab with the cycle net power output being the objective function.
Details of the above iteration and optimisation processes are described in Appendix. For zeotropic mixtures, the bubble point temperature is used as the evaporation temperature.

3.5. Validation

The system models for both DPORC-P and DPORC-S have been validated using the results listed in Li et al. (2015). The same model assumptions have been applied and pure R245fa has been validated using the results listed in Li et al. (2015). The same model assumptionshave been applied and pure R245fa has been validated using the results listed in Li et al. (2015). The same model assumptions have been applied and pure R245fa has been validated using the results listed in Li et al. (2015). The same model assumptions have been applied and pure R245fa has been validated using the results listed in Li et al. (2015).

Validation of the calculation results based on the present models with data published in Li et al. (2015).

<table>
<thead>
<tr>
<th>T_{in} (°C)</th>
<th>DPORC-P</th>
<th></th>
<th>DPORC-S</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>90</td>
<td>77</td>
<td>67</td>
<td>399.6</td>
<td>400.0</td>
</tr>
<tr>
<td>100</td>
<td>82</td>
<td>67</td>
<td>657.6</td>
<td>647.7</td>
</tr>
<tr>
<td>110</td>
<td>88</td>
<td>68</td>
<td>919.1</td>
<td>922.9</td>
</tr>
<tr>
<td>120</td>
<td>94</td>
<td>69</td>
<td>1224.2</td>
<td>1231.4</td>
</tr>
</tbody>
</table>

The state points for both systems at a heat source temperature of 105 °C are listed in Table 5. An exergy analysis on different system components has been carried out for the same operation condition, as shown in Fig. 7. Generally, there are comparable exergy losses between the two for all components except for the evaporators. The effect of liquid preheating in DPORC-S greatly reduces the exergy destruction in the HP evaporator. Although the higher heat absorption rate in the LP evaporator has brought a relatively higher exergy loss for DPORC-S, which offsets its performance enhancement, there is still an overall gain against the DPORC-P counterpart.

According to above results, a DPORC-S is preferable over DPORC-P for a pure fluid scenario, regardless of the heat source temperature. However, with a decreasing heat source temperature, the benefit of DPORC-S tends to be less significant. Whilst additional complexities are involved in a DPORC-S, due to the liquid–vapour separation process in the LP evaporator, a DPORC-P is possibly a more feasible option for ultra-low temperature applications.

4. Results and discussion

4.1. Base case (pure fluid)

An R245fa mass fraction of 1 (pure fluid) is used in this section to gain understand of the characteristics of DPORCs in a simplified scenario. HP and LP evaporating temperatures have been optimised at a given heat source temperature to achieve a maximum net power output.

Fig. 5(a) shows the net power output of both DPORC-P and DPORC-S at different heat source temperatures. The results for a single stage ORC are also included for comparison. It indicates that both DPORC arrangements can generate more net power outputs than a standard ORC; compared to DPORC-P, a DPORC-S achieves a higher performance and the trend is more obvious with an increasing heat source temperature. The second-law efficiency of the three systems at different heat source temperatures has been examined, as shown in Fig. 5(b). It is correlated directly with the net power output, implying that DPORC-S has a higher system performance among others. It is worth noting that the second-law efficiency increases with an increasing heat source temperature for all three systems, although the absolute value for the standard ORC is always below those for DPORCs.

Compared to the results in Li et al. (2015), where a minimum heat source flow outlet temperature of 70 °C was applied, a higher performance enhancement is seen here: the increase of W_{net} is up to 27% in this study, compared to an approximately 9% in Li et al. (2015).

For both DPORC-S and DPORC-P, the variations of optimised evaporation temperatures, absorbed heats, mass flow rates and expander power generations are shown in Fig. 6. It is found that DPORC-S has a higher LP evaporation temperature and both systems have a similar HP evaporation temperature. The absorbed heat of the HP evaporator for DPORC-P, however, is considerably higher and grows rapidly with the increase of heat source temperature. In a DPORC-P, a significant amount of heat is required in the HP evaporator to lift the temperature of the liquid working fluid before the evaporation takes place; in a DPORC-S, however, the required heat for temperature lifting is less, since the liquid working fluid has already been preheated in the LP evaporator. The mass flow rates of the two systems are comparable and the DPORC-S has a slightly higher LP and a lower HP mass flow. This, together with the trend of evaporation temperature, has resulted in a similar HP power generation between the two systems and a higher LP power generation for DPORC-S, as shown in Fig. 6(d).

4.2. Zeotropic mixture

Calculations for both systems operating on zeotropic mixtures have been carried out at a heat source temperature of 90–120 °C. Fig. 8(a) shows the results of net power generation at different R245fa mass fractions, x_{R245fa}, for DPORC-P at T_{in} = 105 °C with a fixed T_{LP} at 45 °C and T_{HP} at 70 °C, 75 °C and 80 °C, respectively. All three curves have two peaks with one located at x_{R245fa} = 0.25 and the other at x_{R245fa} = 0.95, approximately. The shape of the curves is the result of an overall effect from all system components, which will be later illustrated using exergy analyses. It is seen that there is an optimised T_{c,HP} (approximately 75 °C in this case) which results in a maximum W_{net} below or above the optimum, W_{net} tends to drop.

Fig. 8(b) shows the corresponding results with a fixed T_{c,HP} at 75 °C and T_{LP} at 40 °C, 45 °C and 50 °C, respectively. Similarities can be found, and it indicates an optimised T_{LP} is approximately 45 °C at the given conditions.

Other parameters, such as the heat absorption, the mass flow rate, the isentropic enthalpy change and the expander power generation, have been further studied at T_{in} = 105 °C, T_{c,HP} = 75 °C and T_{LP} = 45 °C as shown in Fig. 9(a)–(d). For both HP and LP evaporators, the heat absorption rates are nearly constant as x_{R245fa} varies from 0 to 1; Fig. 9(d) shows that the expander power generations have two peaks for both HP and LP expanders, which is well correlated with the system net power output results. Since the expander power generation is directly related to both mass flow rate and the isentropic enthalpy change across the expander, both parameters have been examined, as shown in Fig. 9(b) and (c). The mass flow rate shows an increasing trend as x_{R245fa} increases and it is mainly due to the different thermodynamic
properties of the two zeotropic components: R245fa has a smaller specific vaporisation heat therefore needs a higher mass flow rate to carry the same amount of heat. The variation of the isentropic enthalpy change resembles that for the power generation, which has two local peaks at approximately $x_{R245fa} = 0.25$ and 0.95, respectively.

An exergy analysis has been carried out at $T_{s, in} = 105 \, ^\circ{C}$, $T_{e, HP} = 75 \, ^\circ{C}$ and $T_{e, LP} = 45 \, ^\circ{C}$ for a DPORC-P, as shown in Fig. 10. It is evident that the most significant variation of the exergy loss occurs at the condenser, which records a highest and lowest exergy losses of 2.2 kW and 1.3 kW, respectively. Other components, such as the evaporators and turbines, have noticeable variations on their exergy losses, however, not comparable to the condenser.

Fig. 11 shows the variations of the temperature glides for both condenser and evaporators, compared to the second-law efficiency of the system. The glides increase gradually as $x_{R245fa}$ increases, reach their highest at around $x_{R245fa} = 0.75$ and then start to drop. Also, a condenser temperature glide of 5 K coincides with the two local peaks of the system second-law efficiency,
indicating that the system performance is the highest as the hot and cold temperature profiles in the condenser become parallel. For a more direct visualisation of the above behaviour, temperature profiles for all condenser and evaporators have been plotted at three different $x_{R245fa}$, as shown in Fig. 12(a)–(f). At $x_{R245fa} = 1$, the ORC fluid shows flat temperature profiles during both condensation and evaporation processes and there are significant exergy losses in both evaporators and condenser due to the mismatch between the cold and hot temperature profiles, as seen in Fig. 12(a) and (d). At $x_{R245fa} = 0.75$, where the temperature glides are among the highest, the hot and cold temperature profiles in both HP and LP evaporators match closer, which is reflected by the improved exergy losses of both evaporators compared to a pure fluid scenario as seen in Fig. 10; however, the exergy loss of the condenser does not improve since a large temperature glide on the ORC fluid creates an over-inclined temperature profile and it moves further away from the temperature profile of the cooling medium. At $x_{R245fa} = 0.25$, a high-level temperature match between the hot and cold fluids is seen in the condenser, resulting in the lowest exergy loss; for the evaporators, the exergy losses are not the lowest although still less than a pure fluid scenario. Above results indicate that it is the temperature matching in the condenser that largely dominates the overall performance of the cycle, and the highest system performance is expected as the hot and cold fluid temperature profiles parallel each other. The temperature profiles in the evaporators also play a role, however, is less significant than the condenser.

The operation of a DPORC-S has also been examined under the same conditions and similar behaviours have been observed.

4.3. Optimised operation

Using the approach described in Section 3.4, optimisations have been carried out for both DPORC-P and DPORC-S at a heat source temperature ranging from 90 °C to 120 °C. Fig. 13(a) shows the second-law efficiencies of the cycles at $T_{s,in} = 90$ °C, indicating that for both DPORC-S and DPORC-P, the second-law efficiencies improve due to the use of zeotropic mixtures and the highest efficiency appears at approximately $x_{R245fa} = 0.25$ where the exergy loss from the condenser is the lowest. At $0.4 < x_{R245fa} < 0.9$, it is found that the second-law efficiency of DPORC-P is higher than DPORC-S. As the heat source temperature increases to 120 °C, as shown in Fig. 13(b), the second-law efficiency of the DPORC-S is always greater, regardless of the mass fraction of the mixture.

To gain further understandings to the above results, temperature profiles of the HP and LP evaporators have been plotted, as shown in Fig. 14(a)–(d). At $T_{s,in} = 90$ °C and an $x_{R245fa}$ of 0.2 (Fig. 14(a)), it is evident that the DPORC-S have a better HP evaporator temperature matching than the DPORC-P counterpart, due to its lifted temperature at the evaporator inlet; for the LP evaporator, the temperature matching of the DPORC-S is not as high as the DPORC-P, since it requires a larger evaporator section for low-temperature liquid heating. At $T_{s,in} = 90$ °C and $x_{R245fa} = 0.8$ (Fig. 14(b)), the benefit of DPORC-S at the HP evaporator enhances, due to the further lifted evaporator inlet temperature, whilst the LP evaporating temperature increases, resulting in a larger sensible liquid heating section and therefore a worse LP temperature matching. This effect leads to an overall reduction for the DPORC-S evaporator exergy loss. However, it is worth noting that at $x_{R245fa} = 0.8$, a DPORC-S corresponds to a higher heat source outlet temperature, meaning a higher exergy destruction in the heat source and this ultimately causes its lower second-law efficiency at $0.4 < x_{R245fa} < 0.9$.

The cycle operations at a higher $T_{s,in}$ has also been evaluated, as shown in Fig. 14(c)–(d). The most significant change is that the benefit of a DPORC-S to the evaporator irreversibility is further enhanced, mainly due to the increase of the cycle evaporation temperatures, making the preheating of the liquid working fluid more effective. This has caused a higher overall performance of the DPORC-S despite the worse exergy loss in the heat source for a large range of the mass fraction.

Figs. 15 and 16 shows the variations of the exergy losses for the heat source and evaporators, respectively. The results indicate that the heat source exergy loss difference between the DPORC-S and DPORC-P is almost independent of the heat source temperature. However, the differences of exergy losses for the evaporators are considerably larger at a higher heat source temperature.

Major cycle operating parameters of both DPORC-P and DPORC-S at heat source temperatures of 90 °C and 120 °C have been listed in Table 6. A DPORC-S tends to have a higher LP...
evaporation temperature than a DPORC-P and this trend is more obvious at a higher heat source temperature. Also, there are only small differences between a DPORC-P and DPORC-S for both LP and HP mass flow rates, which all increase with an increasing $x_{R245fa}$. Table 7 lists the details of the state point parameters for both DPORC-P and DPORC-S at $T_{s,in} = 120 ^\circ C$ and $x_{R245fa} = 0.8$ as an example.

4.4. Heat transfer calculation

The required heat transfer area of the condenser has been investigated since it is crucial for the evaluation of the overall benefit of zeotropic DPORCs. Fig. 17 compares the condenser heat transfer areas of both DPORC-S and DPORC-P at different heat source temperatures. It indicates that they are strongly correlated to the R245fa mass...
fraction and the maximums appear at $x_{R245fa} = 0.5$, approximately. As $x_{R245fa}$ increases or decreases, the required heat transfer areas reduce, and pure fluids result in the smallest heat transfer areas. The use of zeotropic mixtures improves the cycle performance although a condenser with a much larger heat transfer area is expected.

The curves for DPORC-P and DPORC-S overlap each other, mainly attributed to the similarity of the mass flow rate between...
Fig. 12. Temperature profiles of DPORC-P condenser (a)–(c) and evaporators (d)–(f) at different mixture mass fractions at $T_{s,\text{in}} = 105^\circ$C, $T_{e,\text{HP}} = 75^\circ$C and $T_{e,\text{LP}} = 45^\circ$C.

The two. As the heat source temperature increases from 90$^\circ$C to 120$^\circ$C, there are only insignificant changes to the heat transfer areas. Above behaviours are closely related to the variation of the fluid heat transfer coefficient in the condenser which is found to be a strong function of the R245fa mass fraction.

Due to the combination of an enhanced system efficiency and increased heat transfer area, a quantification of the overall gain from the zeotropic DPORCs has become crucial. A possible way is to compare the net power outputs of the systems based on a certain fixed heat transfer area, as shown in Fig. 18. The curves
are plotted by reducing the condenser pinch point temperature gradually from 8 K to close to 0. It is seen that with the increase of the heat transfer area, the net power outputs for both pure fluid and zeotropic mixture increase. At a low heat transfer area, the power output for the pure fluid ($x_{R245fa} = 1$) is found to be higher; however, there are upper bounds for both curves and that for a zeotropic mixture is higher, resulting in a greater power output once the heat transfer area is over 17 m$^2$, approximately. Based on above discussions, the benefit of using zeotropic mixtures would only be more obvious under the assumption that the heat transfer area of the condenser is largely abundant.

5. Conclusions

In this study, the performances of both DPORC-P and DPORC-S have been examined, considering the use of zeotropic mixtures as ORC working fluids. Modelling has been developed for both systems at a heat source temperature ranging from 90 °C to 120 °C. Cycle optimisations have been conducted at different heat source temperatures. Exergy analyses have been carried out to gain further understandings to various cycle characteristics. A heat transfer model has been developed to study the effect of zeotropic mixtures to the required condenser heat transfer area. For both
systems, the use of zeotropic mixtures improves the cycle performances, and the levels of the performance enhancements depend on the heat source temperature and zeotropic mixture mass fraction. The following conclusions can be obtained:

1. For pure fluid scenarios, a DPORC-S is always preferable over a DPORC-P. However, the difference between the two tends to be less significant with the decrease of the heat source temperature. Whilst additional complexities are introduced for a DPORC-S, a DPORC-P may be a more feasible option for ultra-low temperature applications.

2. As a zeotropic mixture is used, both DPORC-P and DPORC-S show improved cycle net power outputs at given evaporation temperatures. For a DPORC-P at \( T_{s,in} = 105 \, ^\circ C \), the enhancement of the cycle performance is approximately 15%. The variation of irreversibility of the condenser is found to be crucial, and the highest power outputs occur at a mixture mass fraction which results in parallel temperature profiles between the hot and cold fluids in the condenser.

3. The cycle performances of optimised DPORC-S and DPORC-P depends on both R245fa mass fraction and heat source temperature. At a heat source temperature of 120 \( ^\circ C \), the system performance of a DPORC-S, in terms of its second-law efficiency, is always higher than a DPORC-P, regardless of the mass fraction of the R245fa. However, at a reduced heat source temperature of 90 \( ^\circ C \), the performance of a DPORC-P tends to be higher at an R245fa mass fraction ranging from 0.4 to 0.9.
Although the use of zeotropic mixtures improves the cycle performances of both DPORC-P and DPORC-S, it requires a much larger heat transfer area for the condenser. Considering the above effect, the benefit of using zeotropic mixtures is only obvious as the condenser heat transfer area is largely abundant.

CRediT authorship contribution statement

Zhanying Zheng: Methodology, Investigation, Validation, Software, Writing – original draft, Funding acquisition. Jingyu Cao: Methodology, Writing – review & editing. Wei Wu: Conceptualization, Writing – review & editing. Michael K.H. Leung: Conceptualization, Methodology, Resources, Writing – review & editing, Supervision, Funding acquisition.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix. Details of the iteration and optimisation processes

An iterative approach, together with the genetic algorithm, has been used to determine the optimised system operating parameters for maximum net power output. For DPORC-P, the input parameters $T_{s,in}$, $m_s$, $\eta_{el,t}$, $\eta_{el,p}$, $T_{cw,in}$, $T_{cw,out}$, $\Delta T_{pp,e}$, $\Delta T_{pp,c}$ are selected. An initial $T_{c,HP}$ and $T_{c,LP}$ are then assumed, so is $T_c$ before an updated condensing temperature $T_{c,iter}$ is calculated based on above parameters. If the relative residual between the assumed and calculated condensing temperatures in within 0.01%, the iteration terminates and the net power output is calculated; otherwise, an updated condensing temperature is assumed, and the iteration continues. Once the net power output is obtained, the genetic algorithm is used to determine the new $T_{c,HP}$ and $T_{c,LP}$ until the maximum $W_{net}$ is obtained. A detailed flow chart of the above procedure is shown in Fig. A.1. For DPORC-S, a similar approach has been used except that not only $T_c$ but also $\chi_{c,LP}$ are assumed, as shown in Fig. A.2.