Working fluid selection for a façade integrated solar organic Rankine cycle

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Abstract

A façade integrated solar power generation system has the potential to reduce fossil fuel energy use, and to make the building envelope better thermally perform. The façade integrated system proposed in this study consists of evacuated tube solar collectors (ETSC) installed in the cavity of double skin façades (DSF) to collect solar energy to be used in an organic Rankine cycle (ORC). A subcritical ORC with regenerator configuration is adopted. To maximise the overall system performance, the working fluid used in the ORC needs to be selected carefully. Various working fluids (R134a, R290, R227ea and R152a) were compared to be employed in the system. The selection is based on thermal efficiency, exergy efficiency, pressure ratio, volume flow rate at the expander of the Rankine cycle, as well as considering safety and environmental issues of working fluids. Engineering Equation Solver (EES) was used to evaluate the system performance by applying the first and second law of thermodynamics. R134a and R152a with superheating appear as most suitable working fluids for the ORC under the predefined operating conditions of the façade integrated system.

1. Introduction

Due to concern about decreasing consumption of traditional energy resources and reducing environmental pollutants, more attention has been paid to renewable energy and energy efficiency. Quadrelli and Peterson (2007) have stated that 84% of greenhouse gas (GHG) emissions were attributed to the energy sector, mainly in the form of CO₂ emissions. More than 40% of global energy usage and one third of global GHG emissions are contributed by commercial buildings (Fong et al. 2010). To stabilise and reduce the greenhouse gas emission, various renewable energy technologies have been investigated. Buildings should have good thermal performance and they should produce renewable energy onsite. Solar energy is one of the renewable energy sources and more abundant than other renewables (Goldemberg & Johansson 2004). Building integrated solar thermal power generation, which uses a non-depletable energy source, has great potential to reduce fossil fuel use and to improve the thermal performance of the building envelope.

Steam Rankine cycles have been widely used to produce power from thermal energy on a large scale. However, these high temperature thermal power plants are not financially viable in small scale applications. The organic Rankine cycle (ORC) is a state-of-the-art technology which is applicable for small scale power generation from a variety of different heat sources. ORCs driven by various renewable heat sources have been investigated, including solar (Wang et al. 2010), biomass
(Drescher & Brüggemann 2007), geothermal (Madhawa Hettiarachchi et al. 2007) and waste heat from industrial processes (Hung 2001). In ORCs, organic compounds, which have lower boiling point than water are used as working fluid. The reason is that a lower driving temperature is needed to produce vapour to operate the expander. A summary of fluids properties comparing in steam and organic Rankine cycles can be found in (Tchanche et al. 2011).

Solar thermal ORCs are less cost competitive than conventional steam Rankine cycles (Quoilin 2011). The life cycle cost of an ORC depends on the system performance, which in turn depends heavily on thermodynamic properties of working fluid. Many investigations were carried out to provide guidelines for working fluid selection for particular ORC applications. Stijepovic et al. (2011) explored the relationship between working fluid properties and ORC economic and thermodynamic performance from both a theoretical and analytical point of view. Wang et al (2011) analysed nine pure organic working fluids under a specific region by using a thermodynamic model. Chen, Goswami & Stefanakos (2010) summarised working fluid selection criteria and influence of fluid properties on cycle performance for ORCs. Vélez et al. (2012) conducted a comparison between different working fluids by modifying turbine inlet pressure and temperature to achieve maximum thermal efficiency. Heberle & Brüggemann (2010) examined four working fluids with a temperature below 450 K for both series and parallel circuits of combined ORC and heat generation. Tchanche et al. (2009) found R134a was the most suitable working fluid for a small scale solar driven ORC based on thermodynamic and environmental properties.

The brief review shows that the selection of working fluid has significant effects on the energy efficiency, system cost and impact on environment. Thermodynamic and environmental properties, thermal stability, environmental impacts, safety, material compatibility, availability and cost are the main criteria that need to be considered. However, there is very limited previous research on the working fluid selection for solar ORCs. In particular, no analysis exists to take into account the performance of façade integrated solar thermal energy collection systems. This paper aims to select the most appropriate working fluid for the designed façade integrated solar ORC by conducting a performance analysis. Thermal efficiency, exergy efficiency, pressure in the heat exchanger and volume flow rate at the turbine expander of each working fluid were analysed.

2. System Description

The basic concept of the proposed low grade heat solar ORC system is shown in Figure 1. This integrated system consists of façade integrated ETSCs, a buffer tank, pumps and an ORC subsystem. ETSCs cover the entire façade to maximise useful energy gain. The water is pumped into the tubes of the ETSCs to transfer the heat. The spacing between each tube in glazing area allows visible light to pass through the façade. The integrated façade consists of three glass layers; one is at the front and two at the back of the cavity, where the ETSCs are located. The cavity with ETSCs was assumed to be fully ventilated for the simplified analysis. The back panel is a sealed double-glazed window. The proposed façade integration maximises the conversion of the incoming solar radiation into useful heat as well as minimising the indoor heat gain, without significantly reducing the view.
The ORC subsystem consists of an evaporator, an expander, a recuperator, a condenser and a pump. The expander is a scroll type which is similar to the scroll expander investigated by Quoilin et al. (2011). A recuperator is added to the ORC to improve the cycle efficiency. It has been shown that a regenerative ORC not only has higher first and second law efficiency than a basic ORC, but it also has lower irreversibility and lower heat requirement to produce the same output (Mago et al. 2008). The heated water in the ETSCs transfers solar energy to heat and evaporates the working fluids of the ORC. A buffer tank is included to ascertain continuous flow for the ETSC and ORC. The vapour is typically saturated or slightly superheated after passing through the evaporator. The generated high pressure vapour flows into the expander which produces shaft power. The exhausted working fluid from the expander is in the form of either superheated or wet vapour. The vapour exhausted from the expander is transported to the inlet of the low pressure side of recuperator and the low temperature liquid exported from the pump is conveyed to the inlet of the high pressure side of the recuperator to allow heat transfer in the recuperator. After passing through the recuperator, the vapour enters the condenser to be condensed to saturated fluid and enters the pump.

![Figure 1 Schematic diagram of façade integrated solar ORC](image)

The operating conditions of the ORC and characteristics of the expander, the recuperator and the pump are given in Table 1. The condenser is cooled by cooling water. The heat source temperature could vary between 55°C and 95°C depending on the flow rate provided by façade integrated ETSCs. The initial temperature of water in the buffer tank is assumed to be 70°C (i.e. the state-of-charge is 100%). The maximum temperature of the heat source is limited by the boiling point of water.

<table>
<thead>
<tr>
<th>Table 1 ORC analysis input data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporating temperature</td>
</tr>
<tr>
<td>Condensing temperature</td>
</tr>
<tr>
<td>Cooling source temperature</td>
</tr>
<tr>
<td>Superheating/subcooling</td>
</tr>
<tr>
<td>Reference temperature</td>
</tr>
<tr>
<td>ORC net output</td>
</tr>
<tr>
<td>Isentropic efficiency of the expander</td>
</tr>
<tr>
<td>Effectiveness of the recuperator</td>
</tr>
<tr>
<td>Isentropic efficiency of the pump</td>
</tr>
</tbody>
</table>
3. Preliminary Selection

The working fluids for the ORC can be categorised in three groups based on the slope of the saturation vapour curve in a T-s diagram. A dry fluid has a positive slope, while a wet fluid has a negative slope and an isentropic fluid has a zero slope. Wet fluids with very steep saturated vapour curves in the T-s diagram have a better overall performance than that of dry fluids (Hung et al. 2010), but they form liquid droplets in the expander during expansion, causing pitting and erosion. To avoid this problem, superheating is required for wet fluids. However, superheating is not required for dry fluids unless more power needs to be gained (Rayegan & Tao 2011). Isentropic fluids are proven to be the most suitable for a low temperature heat source (Hung, Shai & Wang 1997). A recuperator helps dry fluids to achieve higher thermal efficiency and minimum irreversibility (Mago et al. 2008). Therefore, dry, wet and isentropic fluids were all considered in this study.

In the preliminary selection process, critical temperature, operating pressure, environment and safety issue were considered as selection criteria. The upper heat source temperature and the temperature difference between the hot water inlet and the evaporating temperature were fixed. The critical temperature of the working fluid has to be above 80°C and 85°C for dry/isentropic and wet fluids respectively. Delgado-Torres & García-Rodríguez (2007) reported that the highest temperature of the cycle needs to be around 10-15°C lower than critical temperature. For comparison Table 2 lists the pre-selected fluids with their main physical, environmental and safety properties at 80°C evaporating temperature and 35°C condensing temperature. According to Badr, Probert & O’callaghan (1985), a suitable moderate pressure range in an ORC is 0.1-2.5 MPa and a pressure ratio is about 3.5. The ozone depletion potential (ODP) and global warming potential (GWP) assess the substance’s potential to contribute to ozone degradation and global warming (Chen, Goswami & Stefanakos 2010). ODP is the ratio of ozone destruction potential of working fluid reference to R11 which has an ODP of 1. GWP estimates how much the working fluid will contribute to global warming during 100 years reference to carbon dioxide which has a GWP of 1 (Facão & Oliveira 2009). The atmospheric lifetime (ALT) is the length of time a working fluid will remain in the atmosphere based on its decay rate and its likeliness to bond with other gases (Facão & Oliveira 2009). ASHRAE 34 provides a safety classification for fluids according to flammability and toxicity (see Table 3). It should be noted that non-flammable and non-toxic are generally expected, but they are not always critically necessary. A flammable working fluid is not a problem if there is no ignition source.

In considering environmental effects, R12, RC318 and R500 are not suitable due to either high ODP, high GWP or long ALT. R22 and R1270 have pressures above the recommended 3.5 MPa limits in the evaporator. R1270 also has a very low pressure ratio. The rest of the working fluids listed suit the predefined operating conditions from pressure related aspects. R22 will be phased out completely in 2030. R227ea is non-flammable but toxic. R1270 and R290 are non-toxic but flammable, hence require safety devices. R152a classified as lower flammability and non-toxic. The others are non-toxic and non-flammable. Therefore, R227ea, R134a, R152a and R290 were selected for further comparison.
Table 2 Physical, safety and environmental data of the pre-selected working fluids at 80°C evaporating temperature and 35°C condensing temperature

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Tc (°C)</th>
<th>Pc (MPa)</th>
<th>Pmax (MPa)</th>
<th>Pmin (MPa)</th>
<th>PR</th>
<th>Safety Group</th>
<th>ODP</th>
<th>GWP</th>
<th>ALT (years)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R12</td>
<td>112.0</td>
<td>4.11</td>
<td>2.30</td>
<td>0.85</td>
<td>2.71</td>
<td>A1</td>
<td>100.00</td>
<td>1</td>
<td>10890.0</td>
</tr>
<tr>
<td>RC318</td>
<td>115.2</td>
<td>2.78</td>
<td>1.34</td>
<td>0.43</td>
<td>3.12</td>
<td>A1</td>
<td>0.00</td>
<td>10300</td>
<td>3200.0</td>
</tr>
<tr>
<td>R500</td>
<td>102.1</td>
<td>4.17</td>
<td>2.74</td>
<td>1.00</td>
<td>2.74</td>
<td>A1</td>
<td>0.74</td>
<td>8100</td>
<td>N/A</td>
</tr>
<tr>
<td>R227ea</td>
<td>102.8</td>
<td>2.95</td>
<td>1.86</td>
<td>0.61</td>
<td>3.05</td>
<td>B1</td>
<td>0.00</td>
<td>3220</td>
<td>34.0</td>
</tr>
<tr>
<td>R22</td>
<td>96.0</td>
<td>4.97</td>
<td>3.66</td>
<td>1.36</td>
<td>2.69</td>
<td>A1</td>
<td>0.06</td>
<td>1700</td>
<td>12.0</td>
</tr>
<tr>
<td>R134a</td>
<td>101.0</td>
<td>4.06</td>
<td>2.64</td>
<td>0.89</td>
<td>2.97</td>
<td>A1</td>
<td>0.00</td>
<td>1430</td>
<td>14.0</td>
</tr>
<tr>
<td>R152a</td>
<td>113.3</td>
<td>4.52</td>
<td>2.35</td>
<td>0.79</td>
<td>2.97</td>
<td>A2</td>
<td>0.00</td>
<td>124</td>
<td>1.4</td>
</tr>
<tr>
<td>R1270</td>
<td>92.4</td>
<td>4.67</td>
<td>3.50</td>
<td>1.75</td>
<td>2.00</td>
<td>A3</td>
<td>0.00</td>
<td>~20</td>
<td>0.0</td>
</tr>
<tr>
<td>R290</td>
<td>96.7</td>
<td>4.25</td>
<td>3.13</td>
<td>1.22</td>
<td>2.57</td>
<td>A3</td>
<td>0.00</td>
<td>~20</td>
<td>0.4</td>
</tr>
</tbody>
</table>

N/A: not available, Tc: critical temperature, Pc: critical pressure, PR: pressure ratio, * see Table 3

Table 3 ASHRAE safety classification groups (Fação & Oliveira 2009)

<table>
<thead>
<tr>
<th>Lower toxicity</th>
<th>Higher toxicity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Higher flammability</td>
<td>A3</td>
</tr>
<tr>
<td>Lower flammability</td>
<td>A2</td>
</tr>
<tr>
<td>No flame propagation</td>
<td>A1</td>
</tr>
</tbody>
</table>

4. System Modelling

4.1 Façade integrated ETSC

The proposed solar thermal system was modelled using the TRNSYS 16 simulation studio (Klien et al. 2010) with both standard and TESS Types to represent the major components of the system. Typical Meteorological Year (TMY) weather data was used (Morrison & Litvak 1999), as it has been proven to give a reliable estimate of the long term average annual performance of a wide range of solar thermal systems. The main characteristics of the ETSC applied in the TRNSYS model are summarised in Table 4. The outlet temperature of the ETSC is fixed at a certain value to provide a constant temperature heat source to the ORC. The ETSC will only be operated when water reaches the target temperature.

Table 4 Input data of the ETSC in TRNSYS

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intercept efficiency based on gross area</td>
<td>0.4980</td>
</tr>
<tr>
<td>Negative of first order efficiency coefficient (WK(^{-1})m(^{-2}))</td>
<td>1.6100</td>
</tr>
<tr>
<td>Negative of second order efficiency coefficient (WK(^{-1})m(^{-2}))</td>
<td>0.0027</td>
</tr>
</tbody>
</table>

4.2 Thermodynamic analysis for the ORC

In order to study the impact of thermodynamic properties of working fluids on the performance of an ORC under certain operating conditions, the thermodynamic cycle of the ORC was modelled in Engineering Equation Solver (EES). The model was developed for subcritical operating conditions and azeotropic working fluids. The analysis assumed steady state conditions and there is no pressure drop in heat exchangers and connecting pipes. A typical T-s diagram for a regenerative ORC with wet working fluids is shown in Figure 2 (R134a). At state point 1, the working fluid is at saturated liquid state. The working fluid is then pumped from state 1 to 2. Process 1-2s represents an isentropic compression. The state point at the liquid outlet of the
recuperator is 2a. The heating and evaporation of the working fluid occurs between state point 2a and 3. For a wet fluid, it is superheated at constant pressure. The expansion process is from state point 3 to 4. Again, 4s shows an isentropic process. Process 4-4a describes the vapour side recuperation. State point 4a to 1 is the condensation process.

The mathematical model of the ORC is expressed by equations (1)-(8) (Wang et al. 2011). The first law of thermodynamics was applied to individual components of the cycle and the second law of thermodynamics was applied to the whole cycle.

\[
\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{ev}} \quad \ldots \quad (1)
\]

\[
\dot{W}_{net} = \dot{W}_{ex} - \dot{W}_p \quad \ldots \quad (2)
\]

\[
\dot{W}_{ex} = \dot{m}(h_3 - h_4) = \dot{m}\pi_{ex}(h_3 - h_{4s}) \quad \ldots \quad (3)
\]

\[
\dot{W}_p = \dot{m}(h_1 - h_2) = \frac{\dot{m}(h_1 - h_{2s})}{\eta_p} \quad \ldots \quad (4)
\]

\[
\dot{Q}_{ev} = \dot{m}(h_3 - h_{2a}) \quad \ldots \quad (5)
\]

Where \(\eta_{th}\) is the thermal efficiency of the cycle, \(\dot{W}_{net}\) is the net power output and \(\dot{Q}_{ev}\) is the heat absorption rate from evaporator. \(\dot{W}_p\) and \(\dot{W}_{ex}\) are the power input to the pump and power output from the expander. \(\dot{m}\) is the mass flow rate of working fluid. \(h\) is the enthalpy.

Exergy is the maximum amount of work that can be done by a subsystem as it approaches thermodynamic equilibrium with its surroundings by a sequence of reversible processes (Wang et al. 2011). Exergy destruction rate (irreversibility rate) represents the loss of exergy during the process, expressed as:

\[
\dot{i}_{tot} = T_0\dot{m}\left(\frac{h_3 - h_{2a}}{T_H} - \frac{h_1 - h_{4a}}{T_L}\right) \quad \ldots \quad (6)
\]

Where \(\dot{i}_{tot}\) is total irreversibility rate. \(T_0\), \(T_H\) and \(T_L\) are the temperatures of the reference state, heat source and cold sink.

Figure 2 A typical T-s process diagram of ORC with superheating condition (R134a)
The second law efficiency ($\eta_{II}$) of the ORC is based on the exergy destruction rate in processes within the cycle. It captures the interaction between the ORC and secondary fluids.

$$\eta_{II} = \frac{\dot{W}_{ex}}{\dot{W}_{ex} + I_{tot}} \quad \ldots \quad (7)$$

The turbine outlet volume flow rate determines the size of the turbine that influences system cost. Therefore, working fluids with low volume flow rate are preferred for economic reasons.

$$\dot{V}_4 = \frac{\dot{m}}{\rho_4} \quad \ldots \quad (8)$$

Where $\dot{V}$ and $\rho$ is volume flow rate and density at the state point.

The model was verified with the data sets provided in Wang et al. (2011) and Tchanche et al. (2009). Therefore the validity of the model developed was confirmed.

5. Results and Discussion

5.1 Influence of the heat source inlet temperature to the evaporator

To select an optimal evaporating temperature for the ORC, both the solar collector efficiency and ORC efficiency need to be considered since the ETSC and ORC are major components which determine the overall efficiency of the system. According to the pre-selection of working fluids, four working fluids were simulated under different evaporating temperatures. The corresponding ETSC outlet temperatures were applied in the ETSC model. Other parameters were kept constant.

As shown in Figure 3, increasing the heat source inlet temperature leads to higher thermal efficiency in the ORC with all working fluids but lower ETSC efficiency. The overall efficiency for each working fluid is also shown in this diagram. The maximum overall efficiency is located at 95°C that is the highest allowable temperature. To achieve higher overall efficiency, higher outlet temperature from the ETSC needs to be obtained by using other fluids with higher boiling temperature. Higher overall efficiency can also be achieved by decreasing heat loss of ETSCs. A detailed investigation of efficiency improvement is beyond the scope of this paper.

5.2 Overall system performance analysis

According the analysis presented at the previous section, the maximum temperature of the heat source is taken as 95°C that corresponds to 80°C evaporating temperature for the pre-selected working fluids. Table 5 displays the results from thermodynamic analysis with and without superheating. The difference ranging between the thermal efficiencies of the various working fluids are small. It ranges from 6.6% to 7.8%. It was found that with 5°C superheating R152a has the maximum thermal efficiency, followed by R227ea, R134a and R290. For the ORC without superheating, the order of the thermal efficiency is the same but with values lower than that of superheating cycles. Having superheating in ORC slightly improves the thermal efficiency, second law efficiency, irreversibility rate and expander outlet flow rate for each working fluid investigated. R227ea has relatively high expander outlet volume flow rate, which would lead to a large size of expander.

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and high initial cost. Meanwhile R152a has the lowest exergy loss. From the analysis conducted in the previous section, none of the fluids satisfy all the requirements. All the parameters mentioned previously are important for ORC design, and none of these working fluids satisfied all criteria. The limitations of each working fluid are listed below:

- R227ea (relative low efficiency and high expander outlet flow rate)
- RC318 (High GWP and long ALT)
- R1270 (high evaporator pressure and small pressure ratio)
- R22 (not available in the market)
- R290 (low efficiency and high exergy loss)
- R500 (high GWP)
- R152a (relative high expander outlet volume flow rate and low flammability issue)
- R134a (relative low efficiency)
- R12 (high ODP and long ALT, not available in the market)

Compared to R134a, R152a has about an order of magnitude (i.e. ~10 times) lower GWP and shorter ALT which makes R152a more environmentally friendly. Although R152a has about 5% higher volume flow rate at the turbine outlet, it would increase cost only marginally compared to R134a. The flammability issue of R152a can be avoided by carefully selecting operating conditions. However, R134a is more commonly available in the market. Finally, both R152a and R134a were selected as the most appropriate working fluid for this system.

Figure 3 Influence of the heat source inlet temperature on the overall performance

Table 5 Comparison of the cycle performance of different working fluids

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Superheating (°C)</th>
<th>( \eta_{th} ) (%)</th>
<th>( \eta_{II} ) (%)</th>
<th>( \dot{i}_{tol} ) (kW)</th>
<th>( \dot{V}_4 ) (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R134a</td>
<td>0</td>
<td>6.85</td>
<td>41.6</td>
<td>1.61</td>
<td>6.66</td>
</tr>
<tr>
<td>R290</td>
<td>0</td>
<td>6.59</td>
<td>41.7</td>
<td>1.71</td>
<td>6.00</td>
</tr>
<tr>
<td>R227ea</td>
<td>0</td>
<td>7.15</td>
<td>43.1</td>
<td>1.50</td>
<td>9.55</td>
</tr>
<tr>
<td>R152a</td>
<td>0</td>
<td>7.27</td>
<td>43.0</td>
<td>1.46</td>
<td>6.97</td>
</tr>
<tr>
<td>R134a</td>
<td>5</td>
<td>7.29</td>
<td>43.8</td>
<td>1.45</td>
<td>6.09</td>
</tr>
<tr>
<td>R290</td>
<td>5</td>
<td>7.05</td>
<td>43.9</td>
<td>1.54</td>
<td>5.47</td>
</tr>
<tr>
<td>R227ea</td>
<td>5</td>
<td>7.59</td>
<td>45.2</td>
<td>1.36</td>
<td>8.76</td>
</tr>
<tr>
<td>R152a</td>
<td>5</td>
<td>7.78</td>
<td>45.7</td>
<td>1.30</td>
<td>6.36</td>
</tr>
</tbody>
</table>
A similar study has been conducted by Tchanche et al. (2009) for a low temperature (below 90°C) solar ORC without superheating and a regenerator. They considered flammability also as a critical selection criterion. R134a was recommended to be the most suitable working fluid for small scale applications. Tchanche et al. (2009) assumed the evaporating temperature to be 75°C. R134a, R290 and R152a were the working fluids they investigated. Their results are similar to the results presented in Table 5. The reason to reach different selection outcome is that 5°C superheating was applied for wet fluids (R152a and others) in this present work. Application of superheating avoids wet expansion (i.e. change of phase of working fluid to liquid).

6. Conclusions

Working fluid selection for a façade integrated solar collector driven ORC was investigated in this study. A façade integrated ETSC and an ORC were modelled by TRNSYS and EES respectively. To achieve the highest overall efficiency, the optimal evaporating temperature is recommended to be 80°C which is limited by the maximum allowable temperature of the heat source fluid (water) from the ETSC.

Thermodynamic properties and cycle performances of several working fluids were analysed for the working fluid selection. Several criteria were used for the selection, including pressures (maximum and minimum), pressure ratio in the expander, volume flow rate at the expander outlet, thermal efficiency of the cycle, exergy efficiency, irreversibility rate, safety and environment performance of the working fluid. Different criteria favour different working fluids. It is difficult to find the best working fluid that simultaneously has high efficiency, low volume flow rate at the expander, moderate pressures, minor environment impacts and low safety risk. In terms of pressure, R134a and R500 are the best fluids. RC318 has the most moderate pressure ratio. Expander outlet volume flow rate favours R290. From an efficiency point of view, R152a with superheating is the most efficient. RC318, R12 and R500 have been phased out due to their environmental impacts. R152a and R134a with superheating are recommended as most appropriate working fluids for an ORC under the operating conditions specified in this study.

References


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