Thermodynamic Performance of Regenerative Organic Rankine Cycles

Kyoung Hoon Kim

Abstract—ORC (Organic Rankine Cycle) has potential of reducing consumption of fossil fuels and has many favorable characteristics to exploit low-temperature heat sources. In this work thermodynamic performance of ORC with regeneration is comparatively assessed for various working fluids. Special attention is paid to the effects of system parameters such as the turbine inlet pressure on the characteristics of the system such as net work production, heat input, volumetric flow rate per 1 MW of net work and quality of the working fluid at turbine exit as well as thermal efficiency. Results show that for a given source the thermal efficiency generally increases with increasing of the turbine inlet pressure however has optimal condition for working fluids of low critical pressure such as iso-pentane or n-pentane.

Keywords—low-grade energy source, organic Rankine cycle (ORC), regeneration, Patel-Teja equation.

I. INTRODUCTION

SINCE it is impossible to converse low grade energy to electricity efficiently by conventional methods, most of the low grade energy are just discarded. Therefore the research is important how to generate electricity efficiently from low grade energy sources. In recent years, organic Rankine cycle has become a field of intense research and appears as a promising technology for conversion of heat into useful work of electricity. In an ORC the saturation vapor curve is the most crucial characteristics of a working fluid. This characteristic affects the fluid applicability, cycle efficiency, and arrangement of associated equipment in a power generation system [1-2].

Schuster et al [3] mention numerous running applications, such as geothermal power plant, biomass fired combined heat and power plants, solar desalination plants, waste heat recovery or micro CHP. Drescher and Brueggemann [4] investigate the ORC in solid biomass power and heat plants. They propose a method to find suitable thermodynamic fluids for ORCs in biomass plants and found that the family of alkylbenzenes showed the highest efficiency. Dai et al [5] use a generic optimization algorithm, identified isobutane and R236ea as efficient working fluids. Heberle and Brueggemann [6] investigate the combined heat and power generation for geothermal re-sources with series and parallel circuits of an ORC. Tranche et al. [7] investigate comparatively the performance of solar organic Rankine cycle using various working fluids. Volume flow rate, mass flow rate, power ratio as well as thermal efficiency are used for comparison. Hung et al. [8] examine Rankine cycles using organic fluids which are categorized into three groups of wet, dry and isentropic fluids. They point out that dry fluids have disadvantages of reduction of net work due to superheated vapor at turbine exit, and wet fluids of the moisture content at turbine inlet, so isentropic fluids are to be preferred. Kim [9] investigated comparatively the thermodynamic performance of ORC with regeneration for various working fluids. The various thermodynamic characteristics of the ORC such as enthalpy ratio, net work production, heat input and volume flow rate as well as thermal efficiency are investigated in terms of the parameters such as turbine inlet pressure.

II. SYSTEM ANALYSIS

The system considered in this work consists of condenser, pump, turbine, regenerator, pre-heater, boiler, and super-heater and its schematic diagram is shown in Fig. 1. The working fluids considered in this work are nine fluids of NH₃ (ammonia), R134a, R22, iC₅H₁₁ (iso-butane), R152, R143a, C₃H₈ (butane), iC₅H₁₂ (iso-pentane), nC₅H₁₂ (normal pentane). In this work the thermodynamic properties of the working fluids are calculated by Patel-Teja equation of state [10-11]. The basic data of the fluids which are needed to calculate Patel-Teja equation are shown in TABLE 1, where M, T_c, P_c and ω are molecular weight, critical temperature, critical pressure, andacentric factor, respectively [12]. The molecular weights of NH₃ and...
TABLE I

<table>
<thead>
<tr>
<th>Substance</th>
<th>M (kg/kmol)</th>
<th>Tc (K)</th>
<th>Pc (bar)</th>
<th>ω</th>
</tr>
</thead>
<tbody>
<tr>
<td>NH3</td>
<td>17.031</td>
<td>405.65</td>
<td>112.78</td>
<td>0.252</td>
</tr>
<tr>
<td>R134a</td>
<td>102.031</td>
<td>380.00</td>
<td>36.90</td>
<td>0.239</td>
</tr>
<tr>
<td>R22</td>
<td>86.468</td>
<td>369.30</td>
<td>49.71</td>
<td>0.219</td>
</tr>
<tr>
<td>iC6H13</td>
<td>58.123</td>
<td>408.14</td>
<td>36.48</td>
<td>0.177</td>
</tr>
<tr>
<td>R152a</td>
<td>66.051</td>
<td>386.60</td>
<td>44.99</td>
<td>0.263</td>
</tr>
<tr>
<td>R143a</td>
<td>84.041</td>
<td>346.25</td>
<td>37.58</td>
<td>0.253</td>
</tr>
<tr>
<td>C6H10</td>
<td>58.123</td>
<td>425.18</td>
<td>37.97</td>
<td>0.199</td>
</tr>
<tr>
<td>iC7H12</td>
<td>72.150</td>
<td>462.43</td>
<td>33.81</td>
<td>0.228</td>
</tr>
<tr>
<td>nC7H12</td>
<td>72.150</td>
<td>469.65</td>
<td>33.69</td>
<td>0.249</td>
</tr>
</tbody>
</table>

iC4H10 are small, and those of R123 and C6H12 are large among the fluids. The critical temperatures of R143a and R22 are low and those of nC6H12 and iC7H12 are high. The critical pressures of nC6H12 and iC7H12 are low and those of NH3 and R22 are high. The temperature-entropy diagrams for the fluids are shown in Fig. 2. It can be seen from the figure that iC4H10, C6H10, iC7H12 and nC7H12 belong to dry fluids, R134a and R143a to isentropic fluids, and NH3, R22 and R152a to wet fluids. Especially, latent heat of vaporization of NH3 is much greater than the others so the whole temperature-entropy diagram for NH3 is not shown in the figure. The temperature-volume diagrams of the fluids are shown in Fig.3. The volume of saturated vapor at a given temperature is large for nC6H12 or iC7H12, and small for R143a or R22.

A low-grade sensible energy is supplied to the system and important assumptions used in this work are as follows.

1) The energy source is air at temperature of T0.
2) The working fluid leaves the condenser as saturated liquid at temperature of T1.
3) The evaporating temperature, T2 is lower than the critical temperature of the fluid and the turbine inlet temperature becomes T1 - ∆TH by the superheater.
4) The minimum temperature difference between the hot and cold streams in the regenerator is operated at a prescribed pinch point, ∆TPP.

5) Pressure drop and heat loss of the systems are negligible.

At point 1, the fluid is saturated liquid at T1, and the corresponding saturated pressure P1 is the low pressure of the system. When the evaporating temperature is T2, the corresponding saturation pressure P2 is the high pressure of the system. The thermodynamic properties at point 4 are determined with the temperature T4 and the pressure P4. The thermodynamic properties at points 2 and 5 are determined with the isentropic efficiencies of pump and turbine, ηp and ηt, respectively.

As the heat exchange area of regenerator increases, the temperature difference between the hot and cold streams, Thot - Tcold, decreases and finally reaches the prescribed limit of the pinch point, ∆TPP. Then the thermodynamic properties at 2 and 5 can be determined from following conditions.

\[ P_2 = P_H \]  
\[ P_5 = P_t \]  
\[ h_5 - h_2 = h_3 - h_4 \]  
\[ \min (T_{hot} - T_{cold}) = \Delta T_{PP} \]

Then heat addition and net work per unit mass of a working fluid \( q_m \) and \( w_{net} \), and thermal efficiency \( \eta_a \) are be obtained as

\[ q_m = h_2 - h_3 \]  
\[ w_{net} = \dot{w}_t - \dot{w}_p = (h_5 - h_2) - (h_4 - h_3) \]  
\[ \eta_a = w_{net} / q_m \]

where \( h \) denotes specific enthalpy and subscripts \( t \) and \( p \) denote turbine and pump, respectively.

In this work enthalpy ratio, \( x \), is defined as

**Fig. 2** Temperature-entropy diagrams for the working fluids

**Fig. 3** Temperature-specific volume diagrams for the working fluids
where $h_l$ and $h_g$ denote the specific enthalpy of saturated liquid and vapor of the working fluid, respectively. So when $0 \leq x \leq 1$, $x$ is same as the quality of the fluid and the fluid is the mixture of saturated liquid and vapor. When $x < 0$, the fluid is a compressed liquid, and when $x > 1$, the fluid is a superheated vapor.

III. RESULTS AND DISCUSSIONS

The system parameters used in this work are summarized in Table II. In this work the basic data for analysis are $T_S = 200 \degree C$ and $\Delta T_H = 15 \degree C$, so the turbine inlet temperature in this work is fixed at $T_H = T_S - \Delta T_H = 185 \degree C$. Fig. 4 shows the effects of the turbine inlet pressure on the enthalpy ratio for various working fluids. It can be seen from the figure that the enthalpy ratio decreases as the turbine inlet pressure increases. Enthalpy ratio of R143a or R22 is relatively high, and that of NH$_3$ or nC$_5$H$_{12}$ is relatively low. On the other hand, since this work is limited to the subcritical range, the turbine inlet pressure is lower than the critical pressure of the fluid so that the phase transition from liquid of vapor exists.

Fig. 5 shows the effects of the turbine inlet pressure on the temperature difference of the regenerator which is the difference between the temperature at the exit of turbine and the temperature at the exit of pump. The figure shows that the temperature difference decreases as the turbine inlet pressure increases when the turbine inlet temperature is held at a constant value, since as the turbine inlet pressure increases, the enthalpy ratio at turbine inlet or exit decreases and consequently the temperature at the turbine exit decreases. For a fixed value of turbine inlet pressure, the temperature of R143a R22 is relatively high, whereas that of nC$_5$H$_{12}$ or iC$_5$H$_{12}$ is relatively low.

Fig. 6 shows the effects of the turbine inlet pressure on the heat input per unit mass of working fluid for various working fluids. The figure shows that for a fixed value of turbine inlet pressure the heat input generally increases with increasing of the turbine inlet pressure. However, it has an optimal value with respect to the turbine pressure for working fluids of low critical

Table II

<table>
<thead>
<tr>
<th>symbol</th>
<th>Parameter</th>
<th>data</th>
<th>unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_S$</td>
<td>source temperature</td>
<td>200</td>
<td>$\degree C$</td>
</tr>
<tr>
<td>$T_L$</td>
<td>condensing temperature</td>
<td>20</td>
<td>$\degree C$</td>
</tr>
<tr>
<td>$\Delta T_H$</td>
<td>temperature difference at source inlet</td>
<td>15</td>
<td>$\degree C$</td>
</tr>
<tr>
<td>$\Delta T_P$</td>
<td>pinch point</td>
<td>10</td>
<td>$\degree C$</td>
</tr>
<tr>
<td>$\eta_p$</td>
<td>isentropic efficiency of pump</td>
<td>0.80</td>
<td></td>
</tr>
<tr>
<td>$\eta_t$</td>
<td>isentropic efficiency of turbine</td>
<td></td>
<td></td>
</tr>
<tr>
<td>source</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

$$ x = \frac{h_l - h_g}{h_g - h_l} \quad (6) $$
pressure such as iC₅H₁₂ or nC₅H₁₂. Heat input of NH₃ is much higher than that of other working fluids, whereas heat input of R143a or R22 is relatively low.

Fig. 7 shows the effects of the turbine inlet pressure on the net work per unit mass of fluid for various working fluids. The figure shows that the net work per unit mass of fluid generally increases as the turbine inlet pressure increases. Net work of NH₃ is much higher than those of other fluids in the range of high turbine inlet pressure, and net work of nC₅H₁₂ or iC₅H₁₂ is relatively high, and R22 or R143a is relatively low. In the case of nC₅H₁₂ or iC₅H₁₂, there exists an optimal value of the net work with respect to the turbine inlet pressure.

Fig. 8 shows the effects of the turbine inlet pressure on the thermal efficiency for various working fluids. The figure shows that the thermal efficiency generally increases with the turbine inlet. For a fixed value of turbine inlet pressure, thermal efficiency of nC₅H₁₀ or iC₅H₁₂ is relatively high, whereas that of R143a or NH₃ is relatively low.

Fig. 9 shows the effects of the turbine inlet pressure on the volume flow rate at the turbine exit to produce 1 MW of net work for various working fluids. It can be seen from the figure that the volume flow rate of a working fluid to produce the same amount of net work decreases as the turbine inlet pressure increases, and the decreasing rate of the flow rate are relatively high for NH₃, R22 and R143a. Since the volume flow rate to produce a same amount of net work relates directly with the size or cost of the turbine, the volume flow rate may be an important fact for selection of a working fluid for a system.

IV. CONCLUSION

In this paper, the performance of organic Rankine cycle with regeneration has been thermodynamically analyzed. The main results are as follows:

1) As turbine inlet pressure increases for a fixed source temperature, net work per unit mass of working fluid or thermal efficiency generally increases, however, has an optimal value for working fluids which have low critical pressures.

2) The volume flow rate per 1 MW of net work would be a criterion for selection of working fluid, and its decreasing rate are relatively high for NH₃, R22 and R143a.

3) There is no working fluid which is the best for every aspect of thermodynamic performance. Therefore, in order to select an optimal working fluid for a system, various thermodynamic properties should be synthetically and comparatively considered.
ACKNOWLEDGMENT

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REFERENCES