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 superheater for various working fluids including wet, dry and isentropic fluids.

In this paper, the thermodynamic performance of ORC with regeneration is comparatively investigated 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, iC4H10 (iso-butane), R152, R143a, C5H12 (butane), iC3H6 (iso-pentane), nC3H8 (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, Tc, Pc, ω are molecular weight, critical temperature, critical pressure, andacentric factor, respectively [12]. The molecular weights of NH3 and

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The working fluid leaves the condenser as saturated liquid at temperature of \( T_L \). The evaporating temperature, \( T_E \) is lower than the critical temperature of the fluid and the turbine inlet temperature becomes \( T_E - \Delta T_{PP} \) by the superheater. The energy source is air at temperature of \( T_S \). As the heat exchange area of regenerator increases, the temperature difference between the hot and cold streams, \( T_{hot} - T_{cold} \), decreases and finally reaches the prescribed limit of the pinch point, \( \Delta T_{PP} \). Then the thermodynamic properties at 2 and 5 can be determined from the following conditions.

\[
\begin{align*}
\frac{P_2}{P_H} &= P_T = \rho_2 = \rho_T \\
(h_5 - h_4) &= h_3 - h_2 \\
\min(T_{hot} - T_{cold}) &= \Delta T_{PP}
\end{align*}
\]

Then heat addition and net work per unit mass of a working fluid \( q_{in} \) and \( w_{net} \), and thermal efficiency \( \eta_{th} \) be obtained as

\[
\begin{align*}
q_{in} &= h_2 - h_3 \\
w_{net} &= \eta_{th} = (h_5 - h_3) - (h_2 - h_1) \\
\eta_{th} &= w_{net} / q_{in}
\end{align*}
\]

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

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

5) Pressure drop and heat loss of the systems are negligible.
where $h_f$ 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 $T_s = 200^\circ C$ and $\Delta T_H = 15^\circ C$, so the turbine inlet temperature in this work is fixed at $T_H = T_s - \Delta T_H = 185^\circ C$. 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 to 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

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**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>$^\circ C$</td>
</tr>
<tr>
<td>$T_L$</td>
<td>condensing temperature</td>
<td>20</td>
<td>$^\circ C$</td>
</tr>
<tr>
<td>$\Delta T_H$</td>
<td>temperature difference at source inlet</td>
<td>15</td>
<td>$^\circ C$</td>
</tr>
<tr>
<td>$\Delta T_P$</td>
<td>pinch point</td>
<td>10</td>
<td>$^\circ C$</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>0.80</td>
<td></td>
</tr>
</tbody>
</table>

The basic calculation conditions are given in Table II.

\[
\frac{h_f - h_g}{h_g - h_f} = x
\]  

(6)

where $h_f$ 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.
Heat input of NH$_3$, 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 NH$_3$ is much higher than those of other fluids in the range of high turbine inlet pressure, and net work of nC$_5$H$_{12}$ or iC$_5$H$_{12}$ is relatively high, and R22 or R143a is relatively low. In the case of nC$_5$H$_{12}$ or iC$_5$H$_{12}$, 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 nC$_5$H$_{12}$ or iC$_5$H$_{12}$ is relatively high, whereas that of R143a or NH$_3$ 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 NH$_3$, 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 NH$_3$, 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.
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REFERENCES


