Thermal efficiency analysis of Organic Rankine Cycle (ORC) System from low-grade heat resources using various working fluids based on simulation

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Abstract. Thermal Efficiency of Organic Rankine Cycle (ORC) Power Plant System from low-grade heat resources using various working fluids has been analyzed based on the simulation. Four working fluids, namely R-134a, R-32, R-407A, and R-422C were selected on the simulated ORC system to determine its thermal efficiency in some temperature set up of evaporator and condenser. The working fluids are simulated with mass flow rate of 0.15 kg/s at evaporator exit temperature of 75°C, 80°C, and 85°C and at condenser exit temperature of 20°C to 50°C at each 5°C temperature difference. Fluid properties in these conditions are analyzed with REFPROP software then become data input for Cycle Tempo simulation. The thermal efficiency values of each temperature and refrigerant variation are then analyzed to obtain optimum value and variation for the simulated ORC system. The efficiency was obtained at the evaporator exit temperature of 75°C and condenser exit temperature of 45°C. ORC simulation revealed that the optimum and realistic working fluid was R-32 with thermal efficiency of 7.03 %.

1. Introduction
Organic Rankine Cycle (ORC) is a modified rankine cycle. In the ORC, the rankine cycle is modified by replacing the working fluid used from water into refrigerant or hydrocarbons [1]. The working fluid replacement is done so that the ORC system can be operated in lower temperatures and pressures compared to water. The ORC system can therefore be applied by utilizing low-grade heat resources such as recovered waste heat. According to Drescher and Brüggemann [2], despite working at lower temperatures and pressures than water, organic fluids are capable of producing higher thermal efficiencies.

The thermal efficiency of an ORC system is influenced by the property of the working fluid, the mass flow rate of the working fluid, the heat needed by the system, the power needed by the system, and the power generated by the system. In this study, simulation was done to get feasible thermal efficiency on several pure working fluids (R-134a and R-32) and mixed working fluids (R-407A and R-422C) of ORC System.

The benefit of this research is to provide information and guidelines for the simulation of working fluid R-134a, R-32, R-407A and R-422C on low power ORC models with variations in the temperature of the evaporator and condenser outlets using REFPROP and Cycle Tempo software. In addition, recommendations for the type of working fluid are obtained through the analysis of the
temperature and pressure of each component to achieve thermal efficiency, which is useful in further development of ORC.

2. Methods

2.1. Study Approach and Scope
The theoretical ORC system performance can be determined by simulation. Quoilin et al. [3] stated that detailed model simulations from ORC including separate models of each component have been developed. Currently there are several ORC simulation software including EES (Engineering Equation Solver), ACMODEL, and Cycle Tempo (Ziviani et al. [4]). In this research, Cycle Tempo was chosen because it was able to solve problems in various types of working fluids and apparatus. In addition, the software openly provides the simulation code. Thermal efficiency of ORC System operated using several pure working fluids (R-134a and R-32) and mixed working fluids (R-407A and R-422C) were simulated and analysed.

R-32 is a pure refrigerant with a composition of 100% difluoromethane (CH2F2) which is commonly used as refrigerant in air conditioners. Despite its flammability, R-32 has a COP value that is 5% higher in heating mode and 6% higher in cooling mode than R-410A (Yıldırım et al. [5]).

R-407A is a mixture of R-32 (20%), R-125 (40%), and R-134a (40%) refrigerant. As explained earlier, CFC and HCFC refrigerants are replaced with HFC refrigerants because they are environmentally damaging. In that case, R-407A refrigerant can replace the use of R-22. This refrigerant is also commonly used for commercial and industrial refrigeration.

R-422C is a mixed HFC refrigerant with a composition of R-125 (82%), R-134a (15%), and R-600a (3%). This refrigerant is able to replace refrigerant R-502, R-404a, R-507, and R-22 because it has properties that are similar but environmentally friendly. This refrigerant application is for commercial and industrial cooling.

R-134a is used because it is one of the ORC refrigerants that is friendly to the environment and able to work in relatively low temperatures. R-134a includes a Hydro Floro Carbon (HFC) type refrigerant which is very commonly used for refrigerators, air conditioners, etc. (Willis [6]).

2.2. Simulation Procedure
In this study the simulation was run with some condition that the pressure increase at the pump is assumed to be 3 bar and the pressure drop at the turbine is also assumed to be 3 bar because the pressure increase and the pressure drop at the pump and turbine in the ideal rankine cycle are the same (Nusiaputra et al. [7]). The condenser outlet temperature is varied from 20°C-50°C with a 5°C rise interval to simulate the work of the actual ORC system with the condenser system using a heat sink. The outflow temperature of the condenser will be higher as time goes by and the longer the condensation process the less optimal. Steam that is not condensed to the maximum makes the pump unable to suck and drain the refrigerant properly. As a result, the turbine will return the vapor pressure so that the turbine has difficulty rotating. The less optimal the condensation process, the faster the turbine stops spinning and the power generation process will stop too.

Before being simulated with the Cycle Tempo, the property of the working fluid will be analyzed first with REFPROP. With the simulation it is expected to obtain a working fluid with an evaporator and condenser outlet temperature which produces the most optimal efficiency for the ORC system. In addition, the relationship between evaporator outlet temperature and condenser outlet temperature will be observed with ORC thermal efficiency. The working fluid used must meet several conditions and restrictions, namely

(1) The condition of the refrigerant exiting the evaporator and turbine is superheated steam,
(2) The condition of the refrigerant out of the condenser is saturated liquid,
(3) The condition of the refrigerant out of the pump is compressed liquid,
The process that occurs in the condenser and evaporator is isobaric,

The pump is assumed to have an increase in pressure of 3 bar and an increase in temperature by 5°C.

The turbine is assumed to have a pressure drop of 3 bar and a temperature drop of 20°C,

The working fluid is evaporated at 75°C to 85°C at each 5°C rise,

The working fluid is condensed at a temperature of 20°C to 50°C at every 5°C increase,

The mass flow rate of the fluid is 0.15 kg/s, and the system is in a steady state and in a closed system.

Equations for Simulation

Some equations used in energy calculations in the Cycle Tempo simulation are as follow (Asimptote [8]):

1. Conditioning in pump and turbine apparatus:
   \[
   \sum_{j=1}^{n} \Phi_{m,in}(j)h_{in}(j) - \sum_{i=1}^{n} \Phi_{m,out}(i)h_{out}(i) = Q + W
   \]  
   (1)

2. Conditioning the condenser and the heat exchanger:
   \[
   \sum (\Phi_{m,in} \times h_{in}) - \sum (\Phi_{m,out} \times h_{out}) = \Delta E
   \] 
   (2)

3. Calculation of the enthalpy value:
   \[
   h_{tot} = h_{stat} + \frac{1}{2} \rho^2 \text{ and } \\
   \Delta h = h_{in} - h_{out}
   \] 
   (3)

To calculate the energy, work, and efficiency of ORC (Dai et al. 2008), used several equations:

\[
\dot{W}_{in} = \dot{m} (h_{out} - h_{in}) \\
\dot{W}_{out} = \dot{m} (h_{in} - h_{out}) \\
\dot{Q}_{in} = \dot{m} (h_{out} - h_{in}) \\
\dot{Q}_{out} = \dot{m} (h_{in} - h_{out})
\] 

\[
\eta = \frac{\dot{W}_{out} - \dot{W}_{in}}{\dot{Q}_{in}} \times 100\% \\
BWR = \frac{\dot{W}_{in}}{\dot{W}_{out}}
\] 

(9) \hspace{1cm} (10)

Where

\( Q \) = heat taken from the system
\( W \) = work done by the system
\( \eta \) = thermal efficiency of the system (%)
\( \dot{m}, \Phi \) = refrigerant mass flow rate (kg/s)
\[
\begin{align*}
Q_{in} &= \text{heat rate in at evaporator (kJ/s)} \\
Q_{out} &= \text{heat rate out at condenser (kJ/s)} \\
W_{in} &= \text{pump power (inlet power) (kJ/s)} \\
W_{out} &= \text{turbine power (output power) (kJ/s)} \\
h_{in} &= \text{inlet enthalpy of the component (kJ/kg)} \\
h_{out} &= \text{outlet enthalpy of the component (kJ/kg)} \\
\Delta h &= \text{enthalpy change of the component (kJ/kg)} \\
h_{tot} &= \text{total enthalpy (kJ/kg)} \\
h_{stat} &= \text{stationary enthalpy (kJ/kg)} \\
v &= \text{fluid velocity (m/s)} \\
\Delta E &= \text{energy change}
\end{align*}
\]

3. Result and Discussion

The simulation was done by first analyzing the working fluid properties with REFPROP to ensure that the phases of the refrigerants are subcooled liquid phase at the condenser and pump outlet and superheated steam at the evaporator and turbine outlet. After reaching the appropriate phase, all temperature and pressure data from each component was entered into the Cycle Tempo simulation to determine the performance of R-134a, R-22, R-407A, and R-422C refrigerants in an ORC system. The results of working fluid properties analysis is shown in table 1 to table 4.

Table 1. Property of R1-34a using REFPROP analysis

| Temp (°C) | Fluid Properties |
|----------|-----------------|
|          | Pressure (bar)  | Entalpy (kJ/kg) | Quality (kg/kg) |
| Condenser outlet |                  |                 |                  |
| 50       | 12              | 426.41          | Superheated      |
| 50       | 13              | 423.91          | Superheated      |
| 20       | 14              | 227.54          | Subcooled        |
| 25       | 14              | 234.57          | Subcooled        |
| 30       | 14              | 241.71          | Subcooled        |
| 35       | 14              | 248.96          | Subcooled        |
| 40       | 14              | 256.34          | Subcooled        |
| 45       | 14              | 263.87          | Subcooled        |
| 50       | 14              | 271.59          | Subcooled        |
| Evaporator outlet |                |                 |                  |
| 75       | 17              | 445.48          | Superheated      |
| 80       | 17              | 451.48          | Superheated      |
| 85       | 17              | 457.37          | Superheated      |
Table 2. Property of R-32 using REFPROP analysis

| Temp (°C) | Fluid Properties | Pressure (bar) | Entalpi (kJ/kg) | Quality (kg/kg) |
|-----------|------------------|----------------|-----------------|-----------------|
| Condenser outlet |                  |                |                 |                 |
| 50        |                  | 12             | 561.28          | Superheated     |
| 50        |                  | 31             | 508.89          | Superheated     |
| 20        |                  | 32             | 235.79          | Subcooled       |
| 25        |                  | 32             | 245.07          | Subcooled       |
| 30        |                  | 32             | 254.67          | Subcooled       |
| 35        |                  | 32             | 264.57          | Subcooled       |
| 40        |                  | 32             | 274.89          | Subcooled       |
| 45        |                  | 32             | 285.74          | Subcooled       |
| 50        |                  | 32             | 297.37          | Subcooled       |
| Evaporator outlet |              |                |                 |                 |
| 75        |                  | 35             | 544.73          | Superheated     |
| 80        |                  | 35             | 552.79          | Superheated     |
| 85        |                  | 35             | 560.45          | Superheated     |

Table 3. Property of R-407A using REFPROP analysis

| Temp (°C) | Fluid Properties | Pressure (bar) | Entalpi (kJ/kg) | Quality (kg/kg) |
|-----------|------------------|----------------|-----------------|-----------------|
| Condenser outlet |                  |                |                 |                 |
| 50        |                  | 12             | 431.25          | Superheated     |
| 50        |                  | 23             | 293.28          | 0.12623         |
| 20        |                  | 24             | 226.60          | Subcooled       |
| 25        |                  | 24             | 256.00          | Subcooled       |
| 30        |                  | 24             | 263.05          | Subcooled       |
| 35        |                  | 24             | 251.31          | Subcooled       |
| 40        |                  | 24             | 259.30          | Subcooled       |
| 45        |                  | 24             | 267.60          | Subcooled       |
| 50        |                  | 24             | 276.30          | Subcooled       |
| Evaporator outlet |              |                |                 |                 |
| 75        |                  | 27             | 434.31          | Superheated     |
| 80        |                  | 27             | 441.05          | Superheated     |
| 85        |                  | 27             | 447.52          | Superheated     |

Table 4. Property of R-422C using REFPROP analysis

| Temp (°C) | Fluid Properties | Pressure (bar) | Entalpi (kJ/kg) | Quality (kg/kg) |
|-----------|------------------|----------------|-----------------|-----------------|
| Condenser outlet |                  |                |                 |                 |
| 50        |                  | 12             | 383.77          | Superheated     |
| 50        |                  | 23             | 275.69          | 0.037383        |
| 20        |                  | 24             | 226.62          | Subcooled       |
| 25        |                  | 24             | 233.56          | Subcooled       |
| 30        |                  | 24             | 240.68          | Subcooled       |
| 35        |                  | 24             | 248.01          | Subcooled       |
| 40        |                  | 24             | 255.62          | Subcooled       |
| 45        |                  | 24             | 263.61          | Subcooled       |
| 50        |                  | 24             | 272.16          | Subcooled       |
| Evaporator outlet |              |                |                 |                 |
| 75        |                  | 27             | 389.47          | Superheated     |
| 80        |                  | 27             | 395.94          | Superheated     |
| 85        |                  | 27             | 402.16          | Superheated     |
At the highest temperature of 50°C, R-134a refrigerant enters sub-cooled at 14 bar pressure, R-32 refrigerant at 32 bar, R407A and R-422C at 24 bar. In R-134a and R-32, at pressures of 13 bar and 31 bar, the fluid is still in the superheated phase. In the R-407A and R-422C, at a pressure of 23 bar, the fluid is in the liquid-gas mixture phase with a value of less than 0.15. Therefore, the pressure needs to be raised again by 1 bar to reach sub-cooled. This is done so that exits from the pump (when the fluid is raised in temperature and pressure by 5°C and 3 bar), the working fluid is still in the sub-cooled phase. At the condenser exit temperature below 50°C and the same pressure, the working fluid will be in the sub-cooled phase as well. Evaporator outlet temperatures vary at 75°C, 80°C and 85°C. At this temperature, the pressure is 3 bar above the specified condenser outlet pressure because the turbine outlet pressure is the same as the condenser, whereas the pressure drop at the turbine is 3 bar.

In the results of the analysis conducted with REFPROP, the highest system pressure is obtained by the working fluid R-32 at 32-35 bar pressure. According to Douvartzides and Karmalis [9] at a work fluid flow rate of 19.68 kg/s, and turbine inlet pressure 50 bar, and turbine inlet temperature 162 °C, the ORC efficiency obtained by R-32 working fluid is 25.1%. Therefore, this pressure is still within reasonable limits for the R-32 working fluid. R-32 refrigerant has environmentally friendly properties, however, the refrigerant has flammable properties. Therefore, the use of R-32 with high pressure must be accompanied by tight work safety.

The running set up of the Cycle Tempo simulation is shown in figure 1. Cycle Tempo simulation uses assumptions made on pumps and turbines, at a mass flow rate of 0.15, the required pump work is about 1 kW and the resulting turbine power is 2.5-4.3 kW. The thermal efficiency resulted from the simulation ranging from 6.19 to 13.72 %. According to Gaos et al. [10], in the working fluid R-134a with turbine inlet temperature 90°C, the obtained efficiency value was 8.42%. In this study, the working fluid R-134a at the evaporator exit temperature 75°C-85°C has a thermal efficiency value ranging from 4.85% -6.19% at the varied condenser outlet temperature. According to previous research conducted by Yu et al. [11], an increase in evaporator exit temperature causes an increase in the value of thermal efficiency. Therefore, the results of this study on the working fluid R-134a was not significantly different from the previous results.

Table 5 shows that the highest thermal efficiency of 13.72% is obtained by R-407A refrigerant at evaporator temperature of 75°C and condenser temperature of 50°C. However, at the highest efficiency by refrigerant R-407A, there was a mismatch in the turbine exit conditions. For this refrigerant, a temperature drops of 20°C and a 3 bar pressure drop in the turbine are able to change the fluid phase. The temperature drops to 19°C will make the proper fluid phase, but the thermal efficiency drops to become 8.32%, lower than that of R-422C (12.61%). This condition of that difference fluid phase can be seen clearly from the P-h graphics in figure 2 and 3. However, the setting temperature seems to be unconfirmed since its higher that that of Carnot efficiency. The optimum and realistic result was using R-32 with thermal efficiency of 7.03 %. The efficiency was obtained at the evaporator exit temperature of 75°C and condenser exit temperature of 45°C.
Figure 1. The running set up of the Cycle Tempo simulation
Table 5. Optimum thermal efficiency of ORC System and its temperature conditions

| Working Fluids | Temp. of Evaporator Outlet (°C) | Temp. of Condenser Outlet (°C) | Optimum Efficiency (%) |
|----------------|---------------------------------|---------------------------------|------------------------|
| R-134a         | 75                              | 50                              | 6.19                   |
| R-32           | 75                              | 45                              | 7.03                   |
| R-407A         | 75                              | 50                              | 13.72*                 |
| R-422C         | 75                              | 50                              | 12.61                  |

*Not the best efficiency since the working fluid still in mix steam faze of 95%

Figure 2. P-h Graphic refrigeran R-407A on the thermal efficiency 13.72%) with temperature drop in turbin of 20°C

Figure 3. P-h Graphic refrigeran R-407A on the thermal efficiency 8.32%) with temperature drop in turbin of 19°C.
4. Conclusion

Four working fluids, namely R-134a, R-32, R-407A, and R-422C were selected on the simulated ORC system to determine its thermal efficiency in some temperature set up of evaporator and condenser. The working fluids are simulated with mass flow rate of 0.15 kg/s at evaporator exit temperature of 75°C, 80°C, and 85°C and at condenser exit temperature of 20°C to 50°C at each 5°C temperature difference. The optimum and realistic result was using R-32 with thermal efficiency of 7.03 %. The efficiency was obtained at the evaporator exit temperature of 75°C and condenser exit temperature of 45°C. The higher efficiency obtained on the setting of R-407A and R-422C seems to be uncompromised since its higher that that of its Carnot efficiency.

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