Study of a Combined Power and Ejector Refrigeration Cycle with Low-temperature Heat Sources by Applying Various Working Fluids

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Abstract. A power and cooling cycle which combines the organic Rankine cycle and the ejector refrigeration cycle supplied by waste heat energy sources is discussed in this paper. Thirteen working fluids including one wet, eight dry and four isentropic fluids are studied in order to find their performances on the combined cycle. First and second law analysis has been performed by using a computer program in order to investigate various operating conditions’ effects on the proposed cycle by fixing power/refrigeration ratio and varying waste heat source and evaporator temperature. According to the results, in general, dry and isentropic ORC fluids have better performance compared with wet fluids. The increase in evaporator temperature leads to the decrease in exergy efficiency. On the other hand, exergy efficiency rises with the turbine inlet temperature decrease and an increase of heat source temperature. Rising expansion ratio and inlet temperature of the turbine causes an increase in the thermal efficiency of the cycle.

1. Introduction
In recent years, scientist and engineers have tried to find more efficient power systems to reduce environmental problems such as atmospheric pollution, acid precipitation, ozone depletion and global warming. Low-temperature heat sources, such as waste heat and renewable energies (geothermal energy and solar energy) exist in the considerable quantities. Due to these reasons, exploring combined power and refrigeration cycles which use such low-grade heat sources, has attracted more and more attention.

A novel combined power and ejector refrigeration cycle was proposed by Dai et al. \cite{1}. The cycle combined the Rankine cycle and the ejector refrigeration cycle by adding a turbine between the boiler and the ejector. The vapor from the boiler could be expanded through the turbine to generate power, and the turbine exhaust can drive the ejector.

Wang et al. \cite{2} studied this cycle with R123. According to the results, the biggest exergy destruction occurs in the heat recovery vapor generator; it can be reduced by increasing the area of heat transfer and the coefficient of heat transfer in the HRVG. For more performed investigation related to ORCs and combined power and ejector, refrigeration cycles see \cite{3}-\cite{7}.

In the present study, a combined power and refrigeration cycle is proposed to produce both power and refrigeration by utilizing different working fluids and a low-grade heat source. This cycle combines the ORC and the ejector refrigeration cycle. First and second law analysis is conducted to compare different working fluids and different working conditions.

2. Cycle operation and assumptions
In this study, the waste heat is used as the heat source to simulate the combined power and ejector refrigeration cycle shown in Figure 1. To simplify the modeling of the combined cycle, the following assumptions are made:
- The system runs in a steady state.
- The kinetic and potential energies, as well as friction losses, are neglected.
- Vapor generator, evaporator, turbine, ejector, and condenser are assumed adiabatic.
- The expansion valve process is at constant enthalpy (isenthalpic).
- The working fluid at the evaporator outlet is saturated vapor.
- The outlet state from the condenser is saturated liquid.
- A temperature difference of 10 K is assumed between state 2 and state 9.
- A temperature difference of 10 K is assumed between state 4 and state 16.
- A temperature difference of 10 K is assumed between state 13 and state 14.

The base case conditions for the simulation of the combined cycle are summarized in Table 1.

![Figure 1. Schematic diagram of the combined power and ejector refrigeration cycle.](image)

### Table 1. Physical, safety and environmental data for analyzed fluids

| Parameter                           | Value   |
|-------------------------------------|---------|
| Environment temperature (K)         | 298.15  |
| Environment pressure (kPa)          | 101.325 |
| Turbine inlet pressure (Mpa)        | 0.6     |
| Turbine inlet temperature (K)       | 373.15  |
| Turbine extraction pressure (MPa)   | 0.2     |
| Extraction ratio                    | 0.35    |
| Turbine isentropic efficiency (%)   | 85      |
| Pump inlet temperature (K)          | 293.15  |
| Pump isentropic efficiency (%)      | 80      |
| Evaporator temperature (K)          | 263.15  |
| Heat source mass rate (kg/s)        | 75      |
| Power refrigeration ratio            | 2.5     |
| Cooling water mass rate (kg/s)      | 20      |
| Cooling water inlet temperature (K) | 288.15  |

3. Choice of working fluids

One of the main concerns for choosing a working fluid is its environmental effects. Ozone depletion potential (ODP), global warming potential (GWP) and the atmospheric lifetime (ALT) are three important factors that should be regarded. Fortunately, most working fluids used in the ORC cycle can be used in the ejector refrigeration cycle.

4. Thermodynamic analysis

#### 4.1. Energy analysis

\[
\dot{Q}_{\text{conv}} = \dot{m}_5 (h_5 - h_{10})
\]

\[
\dot{Q}_{\text{ph}} = \dot{m}_1 (h_3 - h_2)
\]

\[
\eta_p = \frac{h_{2s} - h_1}{h_2 - h_1}
\]

\[
\dot{m}_5 h_5 + \dot{m}_{13} h_{13} = \dot{m}_7 h_7
\]

\[
R_{\text{extr}} = \frac{\dot{m}_5}{\dot{m}_4}
\]

\[
\dot{W}_p = \dot{m}_1 (h_2 - h_1)
\]

\[
W_t = \dot{m}_1 (h_4 - h_5) + (\dot{m}_4 - \dot{m}_5) (h_5 - h_6)
\]

\[
\eta_t = \frac{h_4 - h_6}{h_4 - h_{6s}}
\]

\[
\beta = \frac{P_4}{P_5}
\]
\[ \dot{Q}_{vg} = \dot{m}_3 (h_4 - h_3) \]
\[ \dot{W}_{net} = \dot{W}_r + \dot{W}_p \]

4.2. Exergy analysis

Energy efficiencies provide neither information of how nearly the performance of a system approaches ideality nor the reversibility aspects of the thermodynamic processes. To determine more meaningful efficiencies, a quantity which provides a measure of an approach to an ideal is required. Thus, exergy efficiency must be introduced. Exergy destruction equations for condenser, ejector, evaporator, preheater, pump, expansion valve, turbine, and vapor generator are as follows:

\[ I_{con} = E_9 + E_{10} - E_9 \]
\[ I_{eva} = E_{12} + E_{ref} - E_13 \]
\[ I_p = -\dot{W}_p + \dot{E}_1 - \dot{E}_2 \]
\[ I_r = \dot{E}_4 - \dot{E}_3 - \dot{E}_6 - \dot{W}_r \]
\[ I_{net} = I_{con} + I_{eva} + I_{ph} + I_p + I_{ev} + I_r + I_{vg} \]

4.3. Efficiency

\[ \eta_{th} = \frac{\dot{W}_{net} + \dot{Q}_{eva}}{Q_{vg}} \]
\[ \eta_{ex} = \frac{\dot{W}_{net} + \dot{E}_{ref}}{E_{in}} \]

4.4. Entrainment ratio

The performance of an ejector is evaluated by its entrainment ratio, which is defined as the mass flow rate ratio of the secondary fluid to that of the primary fluid:

\[ \mu = \frac{\dot{m}_{sf}}{\dot{m}_{pf}} \]

5. Validation

Based on the above analysis, a simulation program using EES software [8] for the combined ORC and ejector refrigeration cycle was developed. Obtained solution is validated with the results of Dai et al. [1] in Table 2 in which R123 was selected as the working fluid which shows a very good agreement.

| Parameter                  | This work  | Dai et al. |
|----------------------------|------------|------------|
| Generating temperature (K) | 413        | 413        |
| Condensing temperature (K) | 293        | 293        |
| Evaporating temperature (K)| 263        | 263        |
| Heat input (kJ/kg)         | 1263       | 1246.96    |
| Entrainment ratio          | 0.396      | 0.389      |
| Pump work (kJ/kg)          | 3.45       | 3.45       |
| Turbine work (kJ/kg)       | 115.8      | 114.14     |
| Net work (kJ/kg)           | 112.35     | 110.69     |
| Refrigeration capacity (kJ/kg)| 61.61      | 60.44      |
| Thermal efficiency (%)     | 13.77      | 13.72      |
| Exergy efficiency (%)      | 22.53      | 22.2       |
6. Results and discussion

The detailed data of the analyzed cycles for 13 different working fluids are listed in Table 3.

Table 3. Comparison of the combined power and refrigeration cycle with 13 different working fluids

| Element | Unit | R123 | R124 | R134 a | R141 b | R142 b | R152 a | R227 ea | R236 fa | R245 fa | R600 | R600 a | R601 a | RC3 18 |
|---------|------|------|------|--------|--------|--------|--------|--------|--------|--------|------|-------|-------|--------|
| μ       |       | 0.13 | 0.1  | 0.15   | 0.11   | 0.1    | 0.1    | 0.25   | 0.19   | 0.15   | 0.1  | 0.1   | 0.1   | 0.28   |
| m       | kgs⁻¹| 2.04 | 2.0  | 4.0    | 6.50   | 1.67   | 3.1    | 4.5    | 6.41   | 4.04   | 2.71  | 3.1   | 4.0   | 5.47   |
| i_con   | kW   | 6.50 | 10.6 | 14.5   | 6.72   | 2.9    | 12.2   | 3.6    | 6.9    | 3.7    | 2.8  | 1.5   | 12.9  | 1.3    |
| i_eje   | kW   | 8.49 | 5.0  | 19.3   | 10.8   | 12.1   | 18.8   | 23.2   | 5.81   | 5.37   | 8.0  | 10.1  | 0.41  | 1.3    |
| i_dev   | kW   | 0.23 | 0.3  | 0.68   | 0.07   | 0.4    | 0.6    | 0.63   | 0.92   | 0.46   | 0.2  | 0.8   | 0.8   | 0.61   |
| i_exv   | kW   | 0.26 | 0.48 | 0.3    | 0.22   | 0.3    | 0.4    | 0.57   | 0.47   | 0.37   | 0.7  | 0.9   | 0.7   | 0.57   |
| i_p     | kW   | 0.18 | 0.2  | 0.03   | 0.17   | 0.2    | 0.1    | 0.24   | 0.27   | 0.23   | 0.5  | 0.5   | 0.5   | 0.3    |
| i_ph    | kW   | 1.41 | 5.0  | 4.19   | 1.35   | 5.8    | 28.6   | 23.3   | 8.2    | 3.87   | 10.0 | 18.5  | 4.9   | 12.3   |
| i_ev    | kW   | 9.85 | 9.8  | 11.2   | 10.3   | 11.4   | 14.4   | 10.7   | 11.2   | 11.9   | 24.0 | 24.4  | 24.4  | 10.6   |
| i_vg    | kW   | 17.5 | 38.1 | 94.8   | 14.4   | 76.5   | 97.4   | 89.1   | 62.2   | 37.0   | 51.0 | 77.7  | 46.3  | 62.8   |
| Q_ev    | kW   | 21.2 | 2.2  | 25.5   | 23.4   | 25.5   | 32.7   | 24.4   | 25.4   | 26.9   | 55.5 | 55.5  | 54.5  | 24.1   |
| Q_in    | kW   | 426.1| 64.8 | 131.1  | 458.7  | 740.1  | 144.8  | 855.7  | 707.1  | 622.2  | 135.7| 155.0 | 105.0 | 694.9  |
| W_f     | kW   | 53.0 | 55.0 | 63.9   | 58.6   | 64.0   | 80.6   | 61.1   | 64.7   | 7.0    | 138.0| 137.0 | 136.0 | 60.4   |
| W_p     | kW   | 0.91 | 0.1  | 0.18   | 0.90   | 1.0    | 0.54   | 1.22   | 1.36   | 1.6    | 2.7  | 2.6   | 2.6   | 1.50   |
| W_net   | kW   | 52.1 | 3.8  | 63.8   | 57.7   | 63.0   | 80.0   | 59.9   | 62.6   | 66.2   | 135.0| 135.0 | 133.0 | 58.9   |
| Refrigeration exergy | kW | 1.94 | 2.3  | 2.37   | 2.14   | 2.3    | 2.9    | 2.34   | 2.33   | 2.46   | 5.0  | 5.0   | 5.0   | 2.21   |
| η_sh    | %    | 17.1 | 11.6 | 6.78   | 17.6   | 12.0   | 7.7    | 9.86   | 12.4   | 14.9   | 14.0 | 12.0  | 17.0  | 11.9   |
| η_exv   | %    | 14.2 | 14.1 | 17.4   | 15.7   | 17.5   | 21.7   | 16.3   | 17.1   | 18.0   | 36.0 | 36.0  | 36.0  | 16.0   |

6.1. Effects of evaporator temperature \( T_{ev} = T_{13} \)
Figure 2 shows that the exergy efficiency decreases with the increase in the evaporator temperature. The reduction of the refrigeration output exergy and the entrainment ratio of the ejector are the main reasons for the exergy decrease.

The results shown in Figure 3 indicate that the entrainment ratio of the cycle for all of the working fluids decreases with increasing evaporator temperature.

6.2. Effects of turbine inlet temperature ($T_4$)

It is found from Figure 4 that the exergy efficiency of the cycle increases with increasing turbine inlet temperature. As the power/refrigeration ratio is kept constant, turbine inlet temperature will affect the ejector entrainment ratio which leads to the increase of the refrigeration output exergy.

As shown in Figure 5 the total exergy destruction in the cycle increases monotonically with the turbine inlet temperature.

According to Figure 6, when the turbine inlet temperature goes up, the thermal efficiency rises, too. As the inlet temperature increases, the turbine power, the net power output and the entrainment ratio rise correspondingly.

**Figure 2.** Effect of the evaporator temperature on the exergy efficiency.

**Figure 3.** Effect of the evaporator temperature on the entrainment ratio of the ejector.
Figure 4. Effect of the turbine inlet temperature on the exergy efficiency.

Figure 5. Effect of the turbine inlet temperature on the total exergy destruction.

Figure 6. Effect of the turbine inlet temperature on the thermal efficiency.

6.3. Effects of heat source temperature \( (T_{16}) \)

Figure 7 shows that increase in the heat source fluid temperature leads to the increase in the exergy efficiency. Since the mass flow rate of the turbine and the entrainment ratio of the ejector rise as the vapor generator temperature goes up, the turbine work output and the cooling capacity increase similarly.

6.4. Effects of expansion ratio

From Figure 8 it is apparent that when the expansion ratio of the turbine increases from 2 to 6, the thermal efficiency rises for all of the working fluids. The reason for this is that increasing expansion ratio leads to a decrease in the temperature and pressure of the primary flow entering the ejector.
7. Conclusions

The main conclusions from this study are as follows:

1. The results confirm the thermodynamic superiority of dry and isentropic ORC fluids over the wet fluids.
2. Exergy efficiency decreases with increasing evaporator temperature but increases with decreasing turbine inlet temperature and increasing heat source temperature.
3. Thermal efficiency increases with the increase in the turbine inlet temperature and expansion ratio of the turbine.
4. Entrainment ratio of the ejector decreases as the evaporator temperature rises.
5. From the exergy efficiency and environmental friendly point of view, R600 and R 600a are the most suitable working fluids for the proposed combined cycle among the different working fluids studied in this paper.

8. References

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