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First and Second Analysis on the Reheat Organic Rankine Cycle

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Abstract**—**In recent years the increasing use of fossil fuels has led to various environmental problems including urban pollution, ozone layer depletion and acid rains. Moreover, with the increased number of industrial centers and higher consumption of these fuels, the end point of the fossil energy reserves has become more evident. Considering the environmental pollution caused by fossil fuels and their limited availability, renewable sources can be considered as the main substitute for non-renewable resources. One of these resources is the Organic Rankine Cycles (ORCs). These cycles while having high safety, have low maintenance requirements. Combining the ORCs with other systems, such as ejector and reheater will increase overall cycle efficiency. In this study, ejector and reheater are used to improve the thermal efficiency (ηth), exergy efficiency (η_ex) and net output power (w_net); therefore, the ORCs with reheater (RORCs) are proposed. A computational program has been developed to calculate the thermodynamic parameters required in Engineering Equations Solver (EES). In this program, the analysis of the first and second law in RORC is conducted, and a comparison is made between them and the ORCs with Ejector (EORC). R245fa is selected as the working fluid and water is chosen as low temperature heat source with a temperature of 95 °C and a mass transfer rate of 1 kg/s. The pressures of the second evaporator and reheater are optimized in terms of maximum exergy efficiency. The environment is at 298.15 k and at 101.325 kpa. The results indicate that the thermodynamic parameters in the RORC have improved compared to EORC.

Keywords**—**Organic rankine cycle, organic rankine cycle with reheater, organic rankine cycle with ejector, exergy efficiency.

I. INTRODUCTION

CONSEQUENT upon the increasing energy consumption and subsequent energy shortage, a variety of systems and mechanisms to promote exergy efficiency have been taken into consideration [1]. In order to achieve a brighter future in the field of energy, minimal environmental impact must be considered. In many industrial units, the temperature of waste heat to the ambient air is less than 400 °C, and the direct release of this heat into the atmosphere will cause environmental warming. The ORC is capable of receiving heat from sources with such temperatures. These cycles have a high level of safety and low maintenance requirements. Vapor saturation of the organic working fluid to enter the turbine occurs at lower temperatures than water and steam which reduces the efficiency of ORC which is why most studies are carried out to increase the efficiency of these cycles. Researchers now try to increase the efficiency of these cycles [1]-[5] by examining different working fluids or adding new components to the ORC. The working fluid plays an important role in the behavior and application of the cycle which is why [1]-[5] performed their research on the working fluids. Javanshir et al. [6] investigated 14 dry working fluids for ORCs and obtained thermodynamic parameters for each fluid. They provided a systematic approach to select the type of working fluid in the cycle based on the operating conditions of the cycle. Their results show that butane, isobutane and R113 deliver the highest amount of net output in cycles. Roy and Misra [7] presented the effects of R134a and R123 working fluids on an ORC. The purpose of this comparison was to select a better working fluid under the same conditions based on cycle efficiency, irreversibility rate and turbine output. Research results show that R123 is a better fluid than R134a. Wang et al. [8] proposed an ORC in which HFE7000, HFE7100 and HFE7500 hydrofluoroethers were used. The purpose of this study was to provide a better fluid based on the turbine size factor and system efficiency. The results show that the HFE7000 is a better than the other two working fluids. Ashouri et al. [9] proposed the analysis and optimization of the exergy-economics of the dual pressure ORC coupled with a solar collector. Exergy efficiency and cost of product were selected by two objective functions as well as 10 decision variable parameters for optimization with genetic algorithm. The results showed that this cycle could work at 22.7% efficiency and make an annual profit of about $2.5 million.

Shao et al. [10] developed a micro radial inflow turbine to be applied in mini ORC. In this cycle that R123 moves as an energy carrier between the cycle components, the experimental data show that the maximum rotary speed of the micro radial inflow turbine is 53564 r/m, the maximum turbine output power is 3535 kW and the electric power is 1.884 kW. Mudasar et al. [11] used the ORCs to generate snow and heat. In this research, the ORC is integrated with a sewage plant. Flue gases from combustion of biogas are used as a heat source. The results show that this cycle has the ability to produce 156 kW of work at 345 °C and it is a suitable solution for supplying electricity to rural areas. Pang et al. [12] investigated the performance of ORC systems using working fluids R245fa, R123 and their composition at ratios of 1:0, 2:1, 1:2 and 0:1 at 110 and 120 °C. The results show that in the pure model the R245fa working fluid at 110 °C can produce the highest net output and for the mode of composition of these two fluids can produce the highest net output in the mode 2: 1 (R245fa = R123) at 120 °C. Mondejar et al. [13] investigated the use of nanofluids in the

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ORC. Although nanofluid particles increase heat transfer, problems such as a high drop in cycle components and nano-fluid concentrations keep the studies on nanofluid based ORC at their early stages. The results of mode jar research show that for a concentration of 1% of nanoparticles, 18% of the pressure drop occurs in the boiler. Mago et al. [14] found the existence of the reductants helpful and by studying the first and second laws of thermodynamics found that the existence of reductants reduced the irreversibility and thereby increased the efficiency of the second law. Gu and Sato [15] studied a supercritical power cycle with a regenerative process to reach the maximum thermal efficiency by choosing an appropriate working fluid and by optimization of the condensation temperature or pressure. Gu and Sato [15] study, the influence of working fluid properties on the thermal efficiency of the ORC was analyzed by Wang et al. [16] at various heat source temperatures, Wang also investigated the effect of working fluid properties on the optimal operation conditions and exergy destruction. It is unwise to seek a high thermal efficiency for LGHS, at all times. Mikielewicz et al. [17] studied the thermodynamic and functional features of several fluids on supercritical and subcritical modes to generate heat and power for domestic purposes. Sun et al. [18] optimally designed the ORC with exergy analysis for the ocean thermal energy appliance. An EORC and a Double Organic Rankine cycle (DORC) based on ORC were introduced by Li et al. [19] Parameters such as the cycle's power output, thermal efficiency, exergy loss, and exergy efficiency were also calculated. Their findings revealed that EORC had the highest power output followed by DORC and ORC. Also, the cycle's exergy efficiency could be rated from high to low: DORC > EORC > ORC. This study introduced a new cycle with a reheater. Lee et al. [20] presented an experimental study on the ORC with the working fluid R245fa. In this research, the effect of cooling water of condenser and R245fa turbine inlet superheat on system performance was evaluated. To ensure the safe operation of the turbine, the R245fa inlet superheat to the turbine was controlled. The results indicate that when the condenser cooling fluid temperature is 23 °C and the pressure ratio 7.3, the turbo-expander can generate 5.405 kW of electricity. Zhou et al. [21] introduced the developed ORC, which uses a wastewater from a steel plant as a low-temperature heat source, and R245fa as the working fluid. The effect of temperature changes of the low temperature heat source in this study has been investigated. The results show that when the heat source temperature increases from 70 °C to 90 °C, the electric power rises from 212 kW to 805 and the thermal efficiency increases from 2.14% to 7.76%.

II. Systems Description

A. Simple ORC

Fig. 1 shows the simple ORC, and Fig. 2 shows a relevant T–s diagram. In fact, the simple ORC is the same as the Rankine cycle, in which the boiler is replaced by an evaporator.

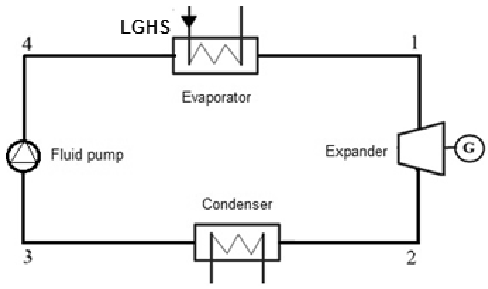

Fig. 1 The ORC configuration and processes

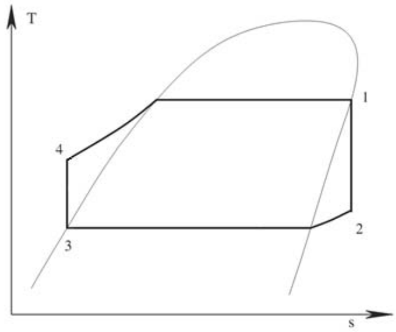

Fig. 2 The ORC T-s diagram

B. Description of the Organic Rankine Cycle with Ejector (EORC)

In EORC, an ejector and a secondary evaporator are added to the ORC (Fig. 3). The output vapor from the second evaporator is used as the primary fluid (pf), and the ejector reduces the output pressure of the turbine to increase its work.

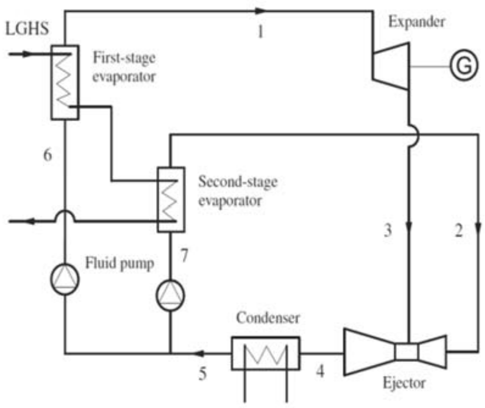

Fig. 3 Configuration and processes of the EORC

C. The Description of the RORC

Fig. 4 shows the simple RORC, and Fig. 5 shows a relevant T–s diagram, the inlet steam temperature to the turbine is increased by adding a reheater. In the first cycle a part of the working fluid in saturated liquid state is transferred from the condenser pump to the first stage evaporator (7-9) and removed from it in saturated vapor (9-1). The saturated vapor enters the turbine and loses its energy by turning the turbine blades (1-3). The turbine outlet vapor enters the reheater to recover its energy

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and returns to its initial temperature (3-5) and reaches the condenser pressure by passing through the turbine (5-6). The superheated vapor at the stage is converted into a saturated fluid by losing its heat in the condenser (6-7).

In the second cycle another part of the saturated fluid is transferred to the second stage evaporator (7-8) and it is removed from it in saturated vapor state (8-2). The saturated vapor generates power through the turbine and its pressure decreases to the condenser pressure (2-4) and it is converted into saturated fluid in the condenser (4-7).

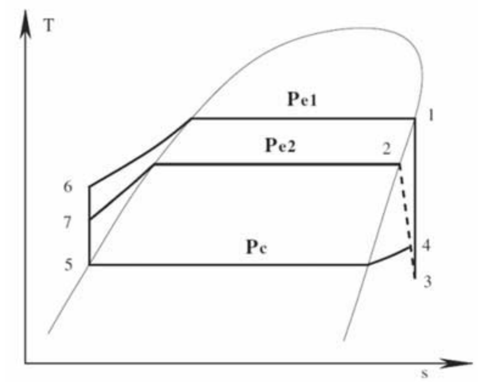

Fig. 4 T–s diagram of the EORC

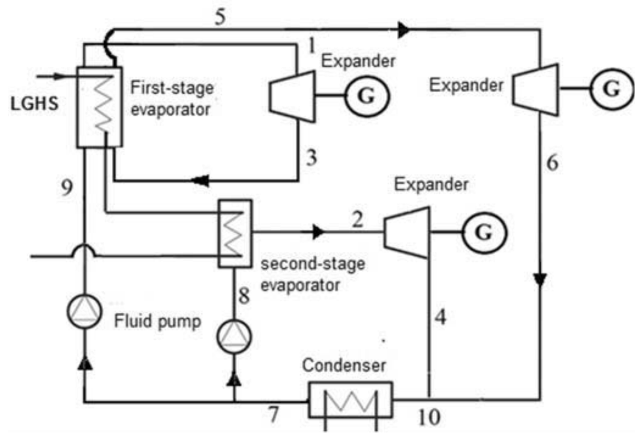

Fig. 5 Configuration and Processes of the RORC

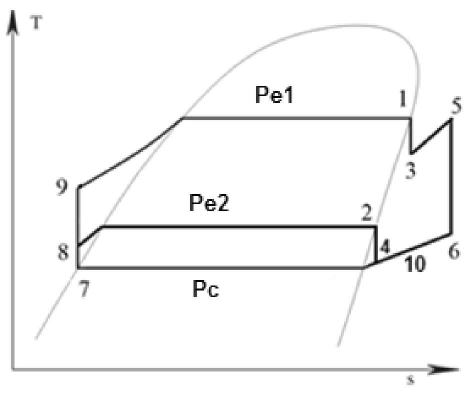

Fig. 6 T – s diagram of the RORC

III. Thermodynamic Analysis of the ORC, EORC, and RORC

In this study, a thermodynamic analysis was performed among ORC, EORC, and RORC. The processes were assumed to be ideal cycles, which means that there was no pressure drop and no heat loss for the expender and the pump. A calculating program was prepared to determine the performance of the cycle in EES. The following parameters were calculated: Expander's output power (W_{out}), pump power (W_{pump}), net output power (W_{net}), thermal efficiency (η_{th}), exergy loss (E_{loss}), and exergy efficiency (η_{ex}).

A. The Terms and Conditions for Computing

1. R245fa is chosen as the working fluid.
2. Water is chosen as LGHS with a temperature of 95 °C, having a mass flow rate of 1 kg/s.
3. The working fluid released from the evaporator is considered as the saturated vapor.
4. The working fluid released from the condenser is considered as a saturated liquid, and the boiling point of the working fluid in the condenser is 35 °C.
5. The working fluid outlet temperature from the reheat is equal to the working outlet fluid temperature from the evaporator.
6. The pinch point temperature difference between the working fluid and heat source water is set to 5 °C.
7. Environmental conditions: P_0 = 101.325 kPa, T_0 = 298.1 k

B. Analysis of the First Law

The evaporator's evaporation heat (Q_{evap}) is:

$$Q_{evap} = \sum m_{wf} * (h_{evap-out} - h_{evap-in}) \quad (1)$$

The condenser's condensation heat (Q_{cond}) is:

$$Q_{cond} = \sum m_{wf} * (h_{cond-in} - h_{cond-out}) \quad (2)$$

The expanders' power output (W_{out}) is:

$$W_{out} = \sum m_{wf} * (h_{exp-in} - h_{exp-out}) \quad (3)$$

The pump's pumping power (W_{pump}) is given by:

$$W_{pump} = \sum m_{wf} * (h_{pump-out} - h_{pump-in}) \quad (4)$$

The cycle's net power output (W_{net}) is:

$$W_{net} = \sum W_{out} - \sum W_{pump} \quad (5)$$

The cycle's thermal efficiency (η_{th}) is:

$$\eta_{th} = W_{net} / Q_{evap} \quad (6)$$

C. Analysis of the Second Law

The heat source water inlet to the evaporator has an exergy as follows:

$$E_{w-in} = m_w * (h_{w-in} - h_0 - T_0 * (s_{w-in} - s_0)) \quad (7)$$

The heat source water outlet from the evaporator has an exergy as follows:

$$E_{w-out} = m_w * (h_{w-out} - h_0 - T_0 * (s_{w-out} - s_0)) \quad (8)$$

The loss of exergy in the evaporator, as well as the exergy loss in the working fluid and heat source water, is:

$$E_{loss-evap} = \sum T_0 * (m_{wf} * (s_{evap-out} - s_{evap-in}) + m_w * (s_{w-out} - s_{w-in})) \quad (9)$$

The loss of exergy in the condenser, as well as the difference in exergy between the inlet and the outlet, is:

$$E_{loss-cond} = m_{wf} * ((h_{cond-in} - h_{cond-out}) - T_0 * (s_{cond-in} - s_{cond-out})) \quad (10)$$

The cycle has a total exergy loss as follows:

$$E_{loss-total} = E_{loss-evap} + E_{loss-cond} + E_{w-out} \quad (11)$$

Each component of the cycle has an exergy loss ratio represented as:

$$r_i = E_{loss-i}/E_{loss-total} \quad (12)$$

The cycle has an exergy efficiency loss as follows:

$$\eta_{ex} = 1 - E_{loss-total}/E_{w-in} \quad (13)$$

IV. Results and Discussion

Figs. 7-15 present the net power output (W_{net}), thermal efficiency (η_{th}), exergy efficiency (η_{ex}) and exergy loss ratio of each component of the RORC and EORC versus the same water inlet temperature of 95°C.

A. Heating Load

The EORC heat in the HP section is slightly higher than the HP section at RORC. Therefore, the heat of the heat source at the entrance to the LP evaporator in RORC is slightly higher, thus its received heat is slightly higher.

B. Thermal Efficiency (η_{th})

As shown in Fig. 8, the thermal efficiency (η_{th}), of the ORC linearly increased with the increasingT_{fe}. The EORC and RORC efficiency first increased and then decreased. The maximum thermal efficiency for EORC is 6.675%, and it is 8.054% for RORC. The corresponding T_{fe} are 69.65°C and 82.8°C, respectively. The rising ORC efficiency which results from increasing the temperature of the evaporator can be attributed to the decrease in absorbed heat. Thus, despite the reduction in power generation, its efficiency is going to be better.

In the ORC given that heat recovery is low, despite the lower power generation, the efficiency of the first law is higher. In the RORC the efficiency of the EORC has increased due to the higher power and lower recycled heat.

C. Net Power Output (W_{net})

The net power output W_{net} of the cycles first increased and then decreased with the increasing evaporator temperature.As shown in Fig. 9, the maximum output power levels are 10.63kW, 11.94kW, and 12.93kW for ORC, EORC, and RORC respectively, with the corresponding working temperatures being 63.3°C, 65.6°C, and 71.56°C, respectively. According to Fig. 9 it is indicated that this relationship is established for the net output power:

$$RORC > EORC > ORC$$

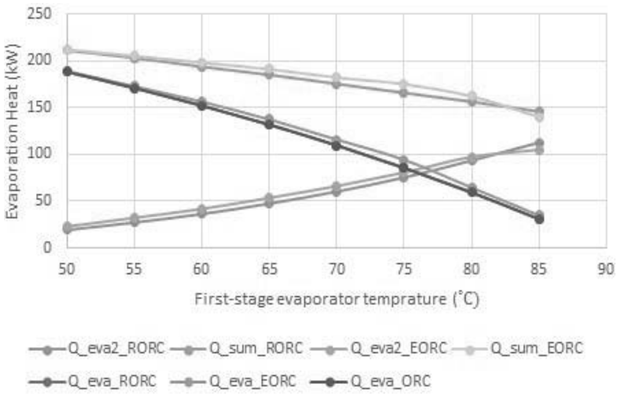

Fig. 7 Evaporation heat of RORC, EORC and ORC with T_{fe}

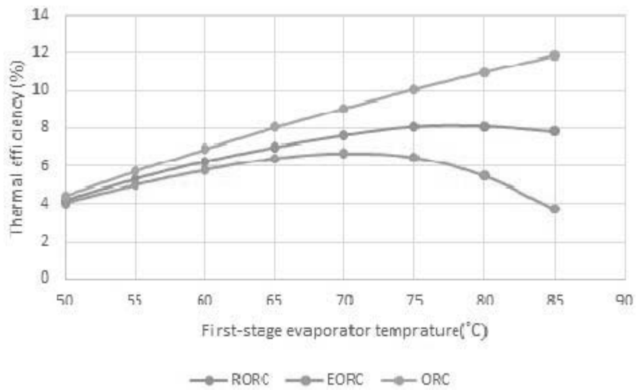

Fig. 8 Thermal efficiency of RORC, EORC and ORC with T_{fe}

Given that LP does not exist in ORC and less heat is recycled, it has less productive power. But although in the RORC the heat recovery is a bit less than the EORC, its potential to work is higher due to reheating the working fluid and more power is produced. Meanwhile by increasing the working fluid temperature in EORC, the working fluid energy in the LP section increases and this excess energy is wasted in the ejector and condenser and it is not recovered well. It should be noted that by increasing the working fluid temperature of the HP section in the EORC, its flow rate is reduced and LP section needs less energy in the ejector but the LP evaporator produces more energy.

D. Exergy Efficiency (η_{ex})

Fig. 10 shows the exergy efficiencies of ORC, EORC, and RORC versus the evaporator temperature (T_{fe}). It can be seen that the changes are consistent with the net power output. The maximum exergy efficiencies for ORC, EORC, and RORC cycle are 33.83, 39.33, and 42.18% respectively, while their corresponding temperatures are 64.65°C, 66.2°C, and 72.7° C respectively. In the sameT_{fe}, the exergy efficiency of cycles can be rated in accordance with the following relationship:

$$RORC > EORC > ORC$$

which happens to be the same as the net output power. In the ORC since the heat recovery in the evaporator is low, the exergy of the received power is low as well but the exergy destruction with the output water (heat source) is high. Therefore, its exergy efficiency is lower than the EORC and RORC. However, in RORC the destruction related to ejector does not exist. On the other hand, by increasing the working fluid temperature of the fluid, the working fluid temperature of the second evaporator increases and the difference with HP turbine is increased in the place of being mixed with the input working fluid into the ejector which increases the exergy destruction in the EORC which reduces exergy efficiency compared to RORC.

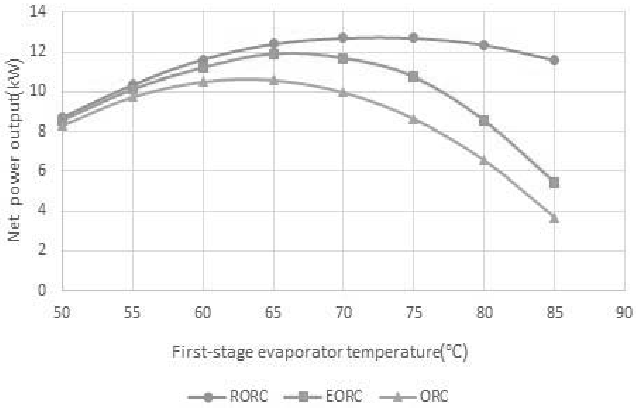

Fig. 9 Net power output of RORC, EORC and ORC with T_{fe}

E. Exergy Loss

Exergy loss in different components of ORC, EORC, and RORC can be defined as follows:

1. Exergy Loss in ORC

As can be seen in Figs. 7 and 11, the recovered heat in the evaporator can be greatly reduced by increasing the evaporator temperature. Therefore, most exergy losses occur in the exiting water more than 90% of energy losses occur at 85°C.

2. Exergy Loss Rate in First-Stage Evaporator

As the temperature increases, the amount of heat transfer in the evaporator decreases, therefore, exergy destruction is reduced. However, in the RORC, due to the fact that heat transfer is used to reheat the working fluid, the working fluid flow rate is reduced as a result of which the received heat is slightly less than EORC and reheating the working fluid reduces its entropy changes is less than the first evaporator of the EORC. But since the total exergy destruction is less than EORC, its partial exergy destruction is slightly higher.

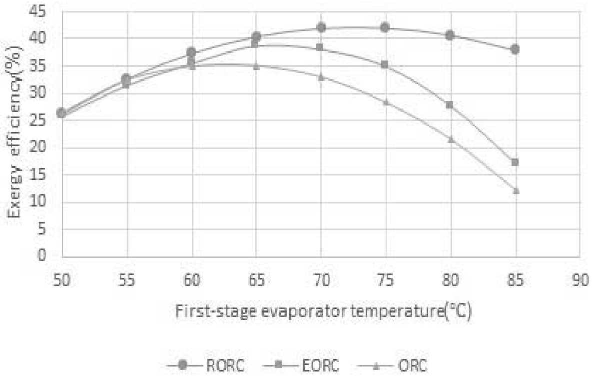

Fig. 10 Exergy efficiency of ORC, EORC and RORC with T_{fe}

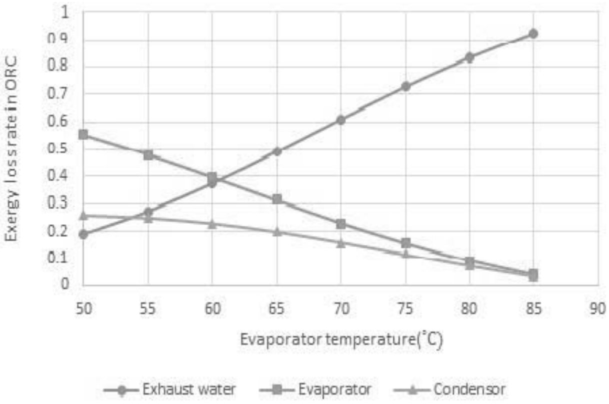

Fig. 11 Ratio of exergy loss in each ORC component with T_{fe}.

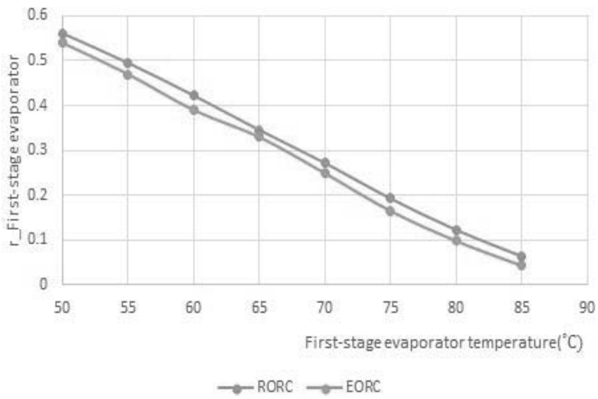

Fig. 12 Exergy loss ratio in the first evaporator with T_{fe}

3. Exergy Loss Rate in Second-Stage Evaporator

The heat recovered in the second evaporator does not differ much in both cycles. Therefore, exergy destruction in them is also not much different. By increasing the vapor temperature of the HP unit its flow rate decreases and less heat is recycled. Hence, more heat reaches LP section, therefore the exergy destruction of the second evaporator increases.

4. Exergy Loss Rate in Condenser

In RORC, due to the fact that part of the heat is spent on reheating, the fluid flow rate of the working fluid is reduced, so its exergy destruction in the condenser is less than the EORC.

However, due to the total exergy destruction, the relative exergy destruction is according to Fig. 14.

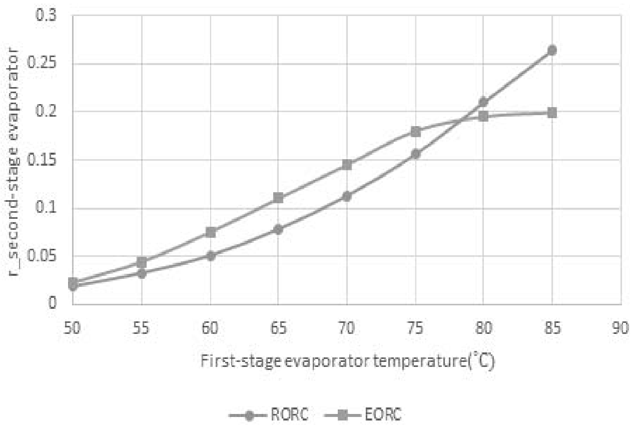

Fig. 13 Exergy loss ratio in the second evaporator withT_{fe}

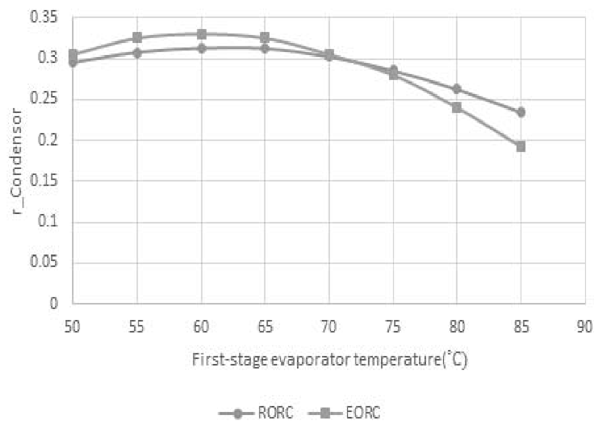

Fig. 14 Exergy loss ratio in the condenser withT_{fe}

5. Exergy Loss Rate in Exhaust Water

Given the higher heat recovery in EORC than RORC, the water output temperature in the EORC is slightly lower than that of RORC and exergy destruction is lower. However, since the total exergy destruction in the RORC is lower, the relative destruction in RORC will be higher.

V. Conclusions

1. RORC and EORC have higher power output than ORC. RORC has a higher power output and thermal efficiency than EORC. RORCs due to having additional components such as generator and expander are more costly than EORCs and it is necessary to choose one of them according to the type and duration of operation before their application
2. The exergy efficiency of the cycle can be rate as follows:

RORC > EORC > ORC

3. The minimum exergy loss occurs at the maximum exergy efficiency which takes place in the evaporator, condenser, and exhaust water, but it is smallest in the EORC cycle for T_{fe} < 65°C.But for higher T_{fe}, the ejector's exergy loss increases continuously and is eventually considered.

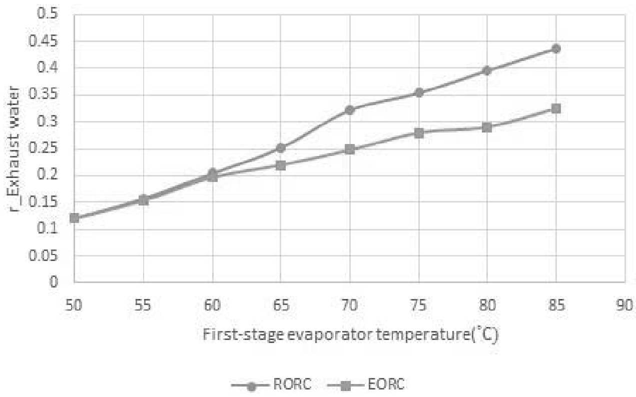

Fig. 15 Exergy loss ratio in the Exhaust water withT_{fe}

Nomenclature

E	Exergy (kW)
h	Enthalpy (KJ.Kg^{-1})
m	Mass flow rate (kg.s^{-1})
Q	Heat transfer (Kw)
r	Exergy loss ratio
s	Entropy (KJ.Kg^{-1}.s^{-1})
T	Temperature (˚C)
W	Power output (kW)
Greek symbol	
η	Efficiency (%)
Subscripts	
cond	Condensor
eje	Ejector
evap	Evaporator
ex	Exergy
fe	First-stage evaporator
in	Inlet
loss	Exergy loss
max	Maximum
net	Net power
out	Outlet
p	Pump
pf	Primary fluid in ejector
sf	Secondary fluid in ejector
sum	Summation
t	Turbine
th	Thermal
w	Water
wf	Working fluid

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