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Thermal performance analysis of a reheating-regenerative organic Rankine cycle using different working fluids

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1. Introduction

Annual electricity generation approximately reached 9000 GWh in 2006, but demand for electricity increased more than 9% during recent years [1]. Increasing demand for electricity in addition to decreasing fossil oil fuels make us to found other renewable heat sources such as nuclear, oil shale, solar, wind, biomass, and geothermal energies, and even industrial waste heat sources for generating required electricity. Today, up to 10 GW electricity is generated from geothermal energy with various methods.

Geothermal fluids and waste heat sources have inherently low temperatures compared to the much higher combustion temperature of fossil fuels. Therefore, there is limitation on converting the heat of these sources to electricity. Form second law of thermodynamic point of view, it results lower work production and lower thermal efficiency [2].

Low temperature of the heating source limits the selection of working fluid utilized in the standard Rankine cycle. Higher thermal efficiency and optimal utilization of the heat source require a careful selection of the working fluid. Organic fluids are suitable working fluids for standard Rankine cycle. Rankine cycle using organic working fluid is known as organic Rankine cycle (ORC). However, other characteristics such as flammability, toxicity, global warming potential (GWP), and ozone depletion potential (ODP) should be considered when selecting an organic fluid for ORC [3].

Montreal Protocol seriously stresses on reduction in the use of ozone depleting substances derived from chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC) [4]. Some replacement substances such as hydrofluorocarbons (HFC) and perfluorocarbons (PFC) have high GWP. Therefore, hydrocarbon (HC) refrigerants with zero or low GWP and ODP are good alternative organic fluids [5].

From thermodynamic point of view, fluids with positive slope of saturation curve are known as dry fluids, and fluids with negative slope of saturation curve are known as wet fluids. Dry fluids show better thermal efficiency than wet fluids, because they do not condense after expanding through the turbine [6]. It means that expansion process terminates in the super heated region. Hence, it is advantageous to incorporate an internal heat exchanger (IHE) before supplying the working fluid to the condenser in order to reduce the heat supplying from heating reservoir and the heat rejection in the condenser. IHE increases the averaged higher temperature of the cycle, which results higher thermal efficiency [7]. This cycle is known as regenerative ORC.

Many researchers studied ORC from different points of views. Some studies were devoted to parametric investigation and optimization on ORC [8-11], but many studies were devoted to confirm the applicability of ORC for waste heat recovery systems [8, 9, 12, and 13]. Other low-temperature heat sources can be used for ORC. Pei et al. [14] analyzed electricity generation from a low-temperature solar thermal energy using regenerative ORC. Many researches concern with analyzing the characteristics of different working fluids applied in ORCs [15-20].

Considering previous studies show that ORC has wide application in electricity generation and hence, it should be improved for higher capacity and efficiency. The purpose of this study is to examine a new reheating-regenerative ORC. The present study evaluates that if reheating process is able to improve thermal and exergy efficiencies of regenerative ORC. Also, the optimal reheating pressure ratio will be found for any working fluid and a brief discussion will be presented in details for the best choice. Eleven working fluids are studied in this article, which their properties are listed in Table 1.

2. Thermodynamics analysis

Figs. 1 and 2 show flow diagram and corresponding *T-s* diagram of the new reheating-regenerative ORC. The cycle consists of two-stage turbine (high-pressure (*H.P.*) and low-pressure (*L.P.*) turbines), evaporator, internal heat exchanger (*I.H.E.*), condenser, and feed pump. The pump feeds the working fluid to the evaporator through IHE. IHE recovers a part of the heat that should be rejected to the surrounding in the condenser, and consequently reduces the required heating in the evaporator. High temperature reservoir heats and vaporizes the working fluid in the evaporator. High-pressure vapor enters the *H.P.T* and expands to an allocated middle pressure. Then, the same high-temperature reservoir reheats the working fluid and supplies it to the *L.P.T*. The working fluid expands to condenser pressure through *L.P.T*. Fig. 2 shows that reheating the outlet flow of the *H.P.T.* increases inlet and outlet temperatures of *L.P.T*. It increases the amount of heat recovery provided in *I.H.E.*, and consequently, evaporator inlet temperature, T_3. Therefore, the required heating in the evaporator decreases accordingly.

Table 1

Properties of some organic fluids considered in this study

Fluid	M, kg/kmol	T_c, K	P_c, MPa
n-Butane	58.12	425.1	3.796
Iso-Butane	58.12	407.8	3.640
Iso-Pentane	72.15	460.4	3.370
n-Pentane	72.15	469.7	3.364
RC-318	200	388.4	2.778
R-236fa	152	398.1	3.198
n-hexane	86.17	507.9	3.058
Cyclohexane	84.16	553.6	4.075
Trichloro-trifluoro-ethane (R-113)	187.4	487.3	3.439
Penta-fluoropropane (R-245fa)	134	427.2	3.651
Hepta-fluoropropane (R-227ea)	170	376.1	2.999

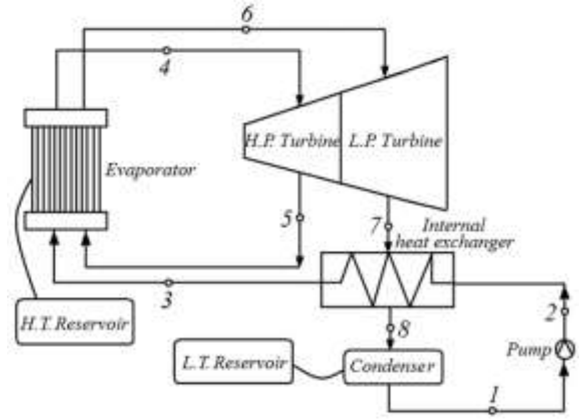

Fig. 1 Control volumes and processes in reheating-regenerative ORC

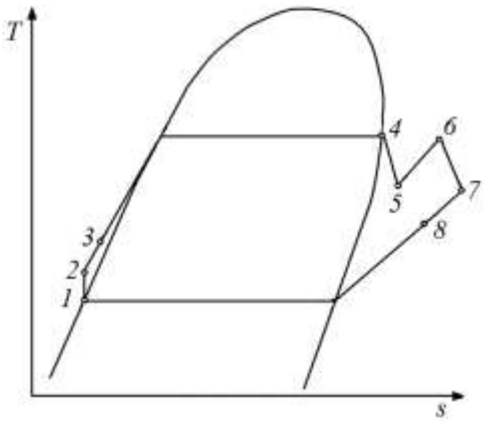

Fig. 2 T-s diagram for processes in reheating-regenerative ORC

Mass, energy, and exergy balance equations should be used for thermodynamics computations. The changes of potential and kinetic energies during a process are negligible. All processes take place steadily, so all thermodynamics states are unchangeable.

The rate of exergy within a fluid stream is defined as:

$$\dot{X} = \dot{m}[h - h_0 - T_0(s - s_0)]\,, \quad (1)$$

where $\dot{m}$ is mass flow rate, kg/s; h is specific enthalpy, kJ/kg; s is specific entropy, kJ/kg.K. Subscript "$_0$" refers to the dead state at which T_0= 298.15 K.

Power consumption and exergy destruction rate of the pump are represented in kW and computed from the following equations:

$$\dot{W}_P = \dot{m}(h_1 - h_{2s})/\eta_P\,, \quad (2)$$

$$\dot{I}_P = \dot{X}_1 - \dot{X}_2 + \dot{W}_P\,, \quad (3)$$

where subscript "$_s$" refers to the outlet state corresponding to the isentropic process between the operational inlet and outlet pressures, and η_P denotes the isentropic efficiency of the pump. η_P is assumed as 0.65 in this study [7, 20].

Heat rejection and exergy destruction rates within condenser are:

$$\dot{Q}_{cond} = \dot{m}(h_1 - h_8)\,, \quad (4)$$

$$\dot{I}_{cond} = T_0\dot{m}\left[s_1 - s_8 - \frac{h_1 - h_8}{T_L}\right], \quad (5)$$

where T_L is the temperature of the low-temperature reservoir in Kelvin scale.

Internal heat exchanger is ideal with 100% efficiency [7, 20]. Therefore:

$$h_7 - h_8 = h_3 - h_2\,. \quad (6)$$

The rate of exergy destruction through the internal heat exchanger is:

$$\dot{I}_{IHE} = \dot{X}_7 + \dot{X}_2 - \dot{X}_8 - \dot{X}_3\,. \quad (7)$$

Heat absorption and exergy destruction rates through the evaporator are:

$$\dot{Q}_{evap} = \dot{m}(h_4 + h_6 - h_3 - h_5)\,, \quad (8)$$

$$\dot{I}_{evap} = T_0\dot{m}\left[s_4 + s_6 - s_3 - s_5 - \frac{h_4 + h_6 - h_3 - h_5}{T_H}\right], \quad (9)$$

where T_H is the temperature of the high-temperature reservoir in Kelvin scale.

Power outputs of L.P. and H.P. turbines are computed from the following equations:

$$\dot{W}_{HPT} = \dot{m}(h_4 - h_{5s})\eta_{tur}\,, \tag{10}$$

$$\dot{W}_{LPT} = \dot{m}(h_6 - h_{7s})\eta_{tur}\,, \tag{11}$$

where η_{tur} is the turbine isentropic efficiency, and is selected as 0.85 in the present study [7, 20]. Exergy destruction rates through the turbine stages are:

$$\dot{I}_{HPT} = \dot{X}_4 - \dot{X}_5 - \dot{W}_{HPT}\,, \tag{12}$$

$$\dot{I}_{LPT} = \dot{X}_6 - \dot{X}_7 - \dot{W}_{LPT}\,. \tag{13}$$

There are two overall thermodynamics equations for computing the net power output and exergy destruction rate of the cycle:

$$\dot{W}_{net} = \dot{W}_{tur} + \dot{W}_{pump}\,, \tag{14}$$

$$\dot{I}_{cycle} = \sum_{all\,components} \dot{I}_i\,. \tag{15}$$

The first law (thermal) and the second law (exergy) efficiencies of the cycle are computed with the following equations:

$$\eta_I = \frac{\dot{W}_{net}}{\dot{Q}_{evap}}\,, \tag{16}$$

$$\eta_{II} = \frac{\dot{W}_{net}}{\dot{Q}_{evap}\left(1 - \dfrac{T_L}{T_H}\right)}\,. \tag{17}$$

Based on this analysis, steady state equations are programmed in Engineering Equation Solver (EES) software, version 8.379, for reheating-regenerative and conventional regenerative ORC cycles.

3. Modeling validation

In order to validate the thermodynamics modeling, a conventional regenerative ORC with different working fluids, which was previously studied by Saleh et al. [7] and Aljundi [20], was simulated. According to Fig. 1, thermodynamics states at different points of the cycle were $T_1 = 303.15$ K, $T_4 = 373.15$ K, $T_8 = 313.15$ K. Isentropic efficiencies of IHE, pump, and turbine were 1, 0.65, and 0.85, respectively, and output power was 1 MW. The present computed results, in terms of thermal efficiency and mass flow rate, are compared with referenced data [7, 20] in Table 2. Good agreement between the present and referenced results confirms the validity of the present thermodynamics computations.

Table 2

Thermal efficiency and mass flow rate for different working fluids at specified conditions

	Thermal efficiency, %			Mass flow rate, kg/s		
Working fluid	This work	Ref. [7]	Ref. [20]	This work	Ref. [7]	Ref. [20]
Iso-Butane	12.44	12.45	12.43	20.58	18.840	20.423
n-Butane	13.02	13.01	13.04	17.84	16.825	17.746
R-245fa	13.04	13.01	13.07	33.85	32.541	33.424
RC-318	11.82	11.82	12.09	67.64	60.473	66.828

4. Results and discussion

Thermal efficiencies of reheating-regenerative ORC and conventional regenerative ORC are shown in Table 3 for different working fluids at specified conditions. Results totally show that reheating process improves thermal efficiency because of increasing the specific work. The improvement in thermal efficiency is ranged from 3.18% for n-hexane to 7.93% for R-236fa.

Parametric analyses are presented in the following paragraphs with fixed and specified operational conditions as follows. Condenser temperature, $T_{1,}$ is 303.15 K, which is 5 K greater than the temperature of the low-temperature reservoir. The temperature of the fluid leaving IHE and directing to the condenser, T_8, is 313.15 K. Outlets evaporator temperatures, T_4 and T_6, are 5 K less than that of the high-temperature reservoir, while keeping T_4 at saturated vapor state. The net output power is fixed at 100 kW.

Table 3

Thermal efficiencies of ordinary and reheating ORCs for different working fluids

Working fluid	T_4,T_6(K)	T_5(K)	T_7(K)	Ordinary ORC	Reheating ORC	$\Delta\eta$%
R227ea	357.03	337.5	337.5	9.646	10.22	5.95
R236fa	373.15	348.8	346.7	11.73	12.66	7.93
n-Hexane	373.15	353.4	353.4	14.14	14.59	3.18
Cyclohexane	373.15	350.9	347.7	14.21	14.75	3.8
R113	373.15	349.8	351.1	13.87	14.41	3.89
Iso-Pentane	373.15	351.4	352.8	13.74	14.31	4.15
RC318	372.08	351.2	350.6	11.82	12.6	6.6
Iso-Butane	373.15	345.6	348	12.44	13.42	7.88
n-Butane	373.15	348.1	347.4	13.02	13.84	6.3
R245fa	373.15	347.3	344.2	13.04	13.86	6.29
n-Pentane	373.15	352.9	351.1	13.78	14.32	3.92

Fig. 3 shows the heat load variation of the evaporator as a function of turbine inlet temperature. Results show that the heat rate needed for fixed output power decreases with increasing turbine inlet temperature. It is due to the decrease in the mass flow rate of the working fluid. Figure shows that cyclohexane requires the minimum energy among the other working fluids. On the other hand, Fig. 4 shows the required mass flow rate of the working fluids as a function of turbine inlet temperature. It shows that the required mass flow rate decreases with increasing the turbine inlet temperature. Since the output power is fixed, the higher enthalpy input to the turbine requires lower mass flow rate of the working fluid. These conclusions are qualitatively consistent with Aljundi's conclusions for conventional regenerative ORC [20].

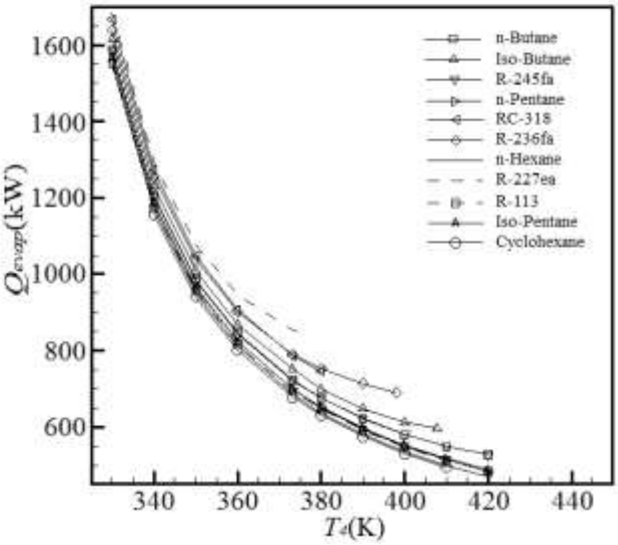

Fig. 3 Evaporator heat load as a function of turbine inlet temperature

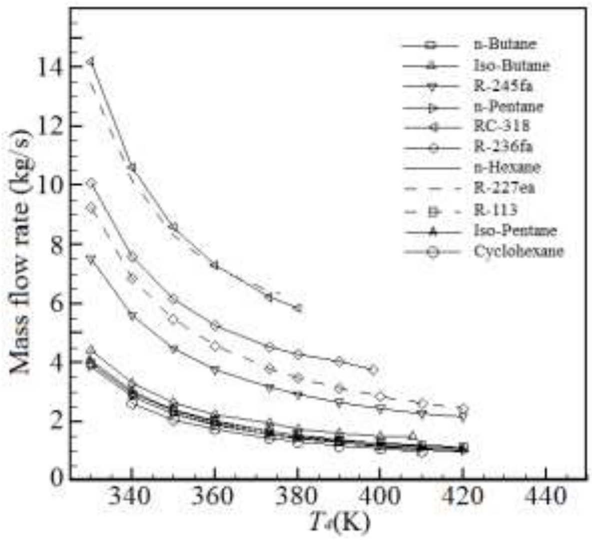

Fig. 4 Mass flow rate of the working fluid as a function of turbine inlet temperature

The ratio of turbine input pressure to reheating pressure, P_4/P_5, is an important parameter in reheating cycles. It changes the thermal and exergy efficiencies of the power cycle. Therefore, it should be adjusted for optimal operation. Figs. 5 and 6 show variations of thermal and exergy efficiencies against the reheating pressure ratio for a fixed turbine inlet temperature, respectively. There are several noticeable results in these figures. Firstly, they show that cyclohexane and R-227ea have always the highest and lowest efficiencies, respectively, among the other examined working fluids. Secondly, RC-318 and R-236fa have slightly the same efficiency curves, and consequently, they are good replacements for each other in ORC applications. The same behavior is noticeable for n-pentane and Iso-pentane; they are good replacements for each other. According to the thermal (or exergy) efficiency, eleven examined working fluids can be sorted from high to low performances as: cyclohexane, n-hexane, R-113, n-pentane and Iso-pentane, R-245fa, n-butane, Iso-butane, RC-318 and R-236fa, and at last R-227ea. Optimum reheating pressure ratios corresponding to optimum thermal efficiencies for different working fluids are listed in Table 4. Results show that the optimum reheating pressure ratio generally increases with critical temperature of the working fluid. The optimum reheating pressure ratio ranges from 1.5, for R-227ea, to 3.1, for cyclohexane.

Table 4

Reheating pressure ratio corresponding to the optimal thermal and exergy efficiencies for different working fluids

Working fluid	P_4/P_5	T_C, K
cyclohexane	3.1	553.6
n-hexane	2.9	507.9
R-113	2.6	478.3
n-pentane	2.3	469.7
Iso-pentane	2.3	460.4
R-236fa	2.0	398.1
R-245fa	2.0	427.2
n-butane	2.0	425.1
Iso-butane	1.8	407.8
RC-318	1.6	388.4
R-227ea	1.5	376.1

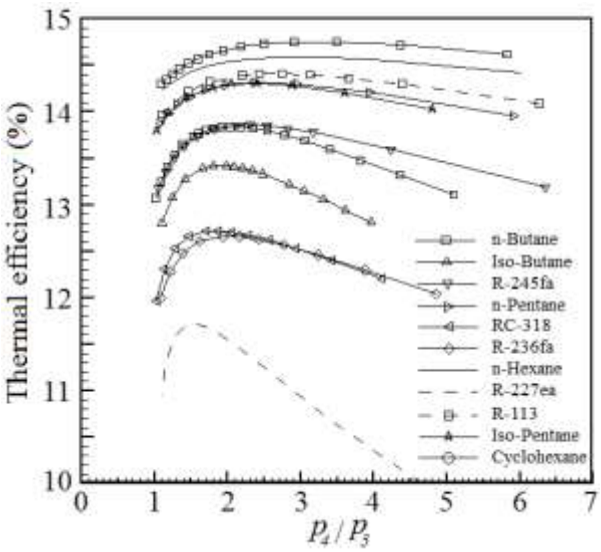

Fig. 5 Influence of reheating pressure ratio on thermal efficiency

Choosing cyclohexane as the working fluid (the best working fluid according to the previous results), the specific exergy destructions in the main components of the reheating-regenerative ORC are analyzed in terms of the reheating pressure, while the inlet turbine temperature is fixed at 373.15 K and saturated vapor state. It is worth to mention that the output power remains 100 kW. Fig. 7 shows specific exergy destructions in the turbine, IHE, condenser, evaporator, and pump. This figure shows that exergy destruction in the pump is inherently low and is not affected by varying the reheating pressure. Also, turbine and evaporator have higher exergy destructions compared to the condenser and IHE. Results show that the reheating pressure does not affect the condenser exergy destruction. Turbine exergy destruction very slightly increases with increasing the reheating pressure. Exergy destructions in the evaporator and IHE significantly vary with the reheating pressure. Increasing the reheat pressure ratio decreases the turbine outlet temperature and also evaporator inlet temperature, while IHE inlet temperature decreases. It increases temperature difference between the working fluid flowing through evaporator and heating reservoir, and decreases temperature difference between the hot and cold streams flowing through IHE.

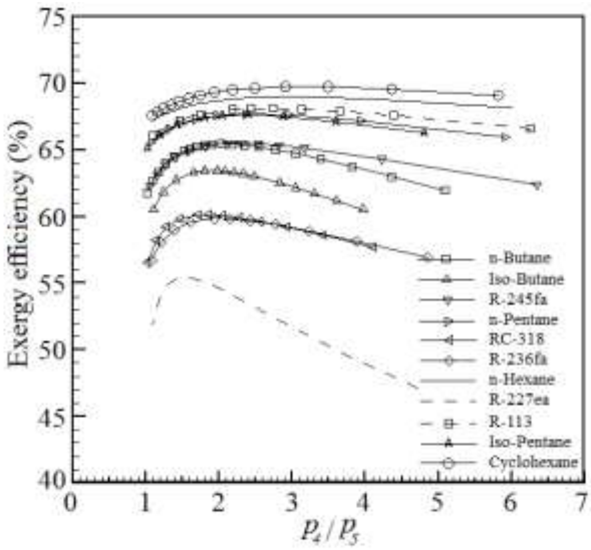

Fig. 6 Influence of reheating pressure ratio on exergy efficiency

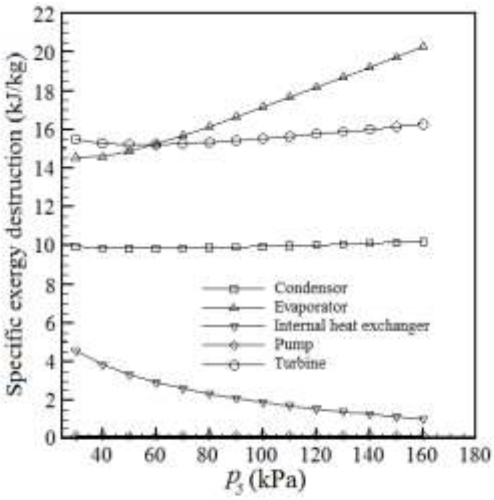

Fig. 7 Specific exergy destruction in ORC components

Figs. 8 and 9 show the influence of turbine inlet temperature on thermal and exergy efficiencies, respectively, as a function of the reheating pressure ratio for cyclohexane cycle at the other already described thermodynamics states. These figures show that thermal and exergy efficiencies increase with increasing turbine inlet temperature, while the optimal reheating pressure ratio remains unchanged.

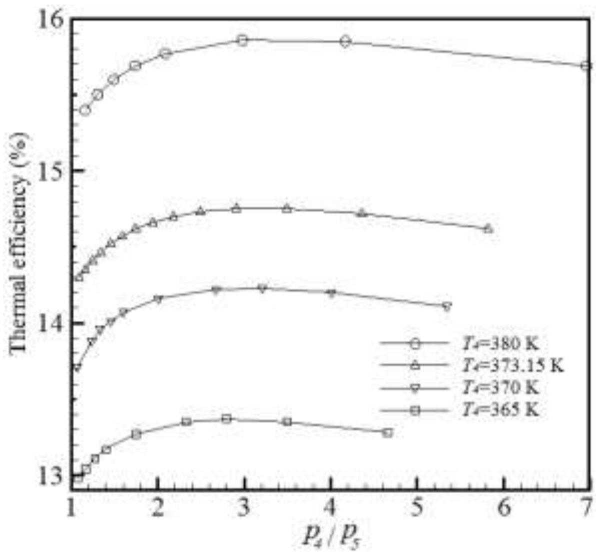

Fig. 8 Influence of turbine inlet temperature on thermal efficiency

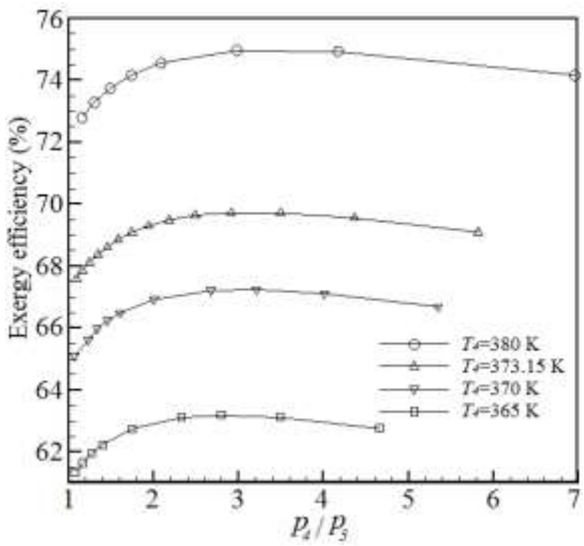

Fig. 9 Influence of turbine inlet temperature on exergy efficiency

5. Conclusions

A reheating-regenerative ORC was studied in this article. Analyses were presented in terms of the first and second laws efficiencies. Eleven organic working fluids were examined at the first stage to choose the best working fluid. Then the influence of the reheating pressure ratio on the thermal performance of ORC was investigated to find the optimal condition. Based on the obtained results, the following conclusion can be inferred:

Including a reheating process in the conventional regenerative ORC improves the thermal performance of the cycle.

Eleven examined working fluids can be sorted according to their thermal and exergy efficiencies from high to low performances as cyclohexane, n-hexane, R-113, n-pentane and Iso-pentane, R-245fa, n-butane, Iso-butane, RC-318 and R-236fa, and at last R-227ea.

The type of working fluid slightly affects the optimum reheating pressure ratio, such that working fluid with higher critical temperature requires a greater reheating pressure ratio.

Generally, thermal and exergy efficiencies increase with increasing turbine inlet temperature.

Turbine inlet temperature does not affect the optimum reheating pressure ratio.

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THERMAL PERFORMANCE ANALYSIS OF A REHEATING-REGENERATIVE ORGANIC RANKINE CYCLE USING DIFFERENT WORKING FLUIDS

S u m m a r y

Higher efficiency and optimal utilization of low-temperature heat sources, such as geothermal energy, requires a careful selection of working fluid and improving organic Rankine cycle (ORC). This article concerns with analyzing a new reheating-regenerative ORC. Present study considers selection of optimal working fluid and evaluation of the optimal reheating pressure ratio. It investigates the influence of the turbine inlet temperature on thermal and exergy efficiencies. Results show that reheating process can improve thermal performance of ORC. Derived curves for thermal and exergy efficiencies show that optimal reheating pressure ratio does not depend on turbine inlet temperature, but slightly increases with the critical temperature of the working fluid.

Keywords: Organic Rankine cycle, reheating pressure, Thermal efficiency, exergy efficiency, dry hydrocarbon.

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