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Operation of an organic Rankine cycle dependent on pumping flow rates and expander torques

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ABSTRACT

An ORC (organic Rankine cycle) was developed with R123 as the working fluid. The heat capacity is in ~100 kW. The match between pump and expander is investigated. Lower pump frequencies ($f < 10$ Hz) adapt the whole range of expander torques, yielding stable flow. Higher pump frequencies ($f > 10$ Hz) adapt low expander torques only, and cause unstable flow and pump cavitation for larger expander torques. Ultra-low expander torques generate sufficiently high vapor superheatings to decrease expander efficiencies. Ultra-high expander torques achieve saturation vapor at the expander inlet, causing liquid droplets induced shock wave to worsen expander performance. An optimal range of expander torques exists to have better expander performance. A liquid subcooling of 20 °C is necessary to avoid pump cavitation. Expander powers and efficiencies show parabola shapes versus expander torques, or vapor superheatings at the expander inlet. The optimal vapor superheating is 13 °C. The cavitation mechanisms and measures to avoid cavitation are analyzed. This paper notes the overestimation of ORC performance by equilibrium thermodynamic analysis. Assumptions should be dependent on experiments. Future studies are suggested on organic fluid flow, heat transfer and energy conversion in various components.

1. Introduction

The energy shortage and environment pollution encourage us to develop new and clean energy technologies. The low grade thermal energy with the temperature of less than 250 °C has received great attention worldwide [1]. ORC (organic Rankine cycle) is an important system to recover low grade thermal energy [2,3], including geothermal energy [4–7], solar energy [8–11], waste heat in various industry sectors [12–19] and biomass thermal energy [20–22].

Many investigations have been performed for ORCs. The work focused on equilibrium thermodynamic analysis and fluid selection. ORCs can be classified as subcritical pressure ORC, transcritical pressure ORC and ORC with mixture working fluid [23]. Based on heat source temperatures and computation conditions, the recommended organic fluids are quite different [16,24–26]. Generally, due to better thermal match of organic fluid and heat carrier fluid of heat source in the evaporator, transcritical pressure ORC and ORC with mixture fluid have better performance than subcritical pressure ORCs [16,27,28]. For subcritical pressure ORCs, isothermal evaporation heat transfer takes place in the evaporator to increase exergy destruction. However, subcritical pressure ORCs attract industries due to low pressure operation and easy fluid selection.

From thermodynamic cycle point of view, saturation vapor at the expander inlet achieves higher thermal efficiency. Fewer authors investigated ORCs at superheating vapor cases at the expander inlet [19,29,30]. Saturation vapor at the expander inlet may cause following issues: (1) liquid droplets are entrained in vapor to worsen the expander performance; (2) wet expansion happens in the expander, which should be avoided; (3) the fluid state (either two-phase mixture or saturation vapor) cannot be judged by pressure and temperature only.

Besides, thermodynamic analysis has several ideal assumptions which should be further verified by experiments. For instance, isentropic efficiency of expanders was assumed to be 75–87% [25,31,32], which is higher than practical values. Many theoretical studies assume higher pressures in the range of 2.5–32.67 MPa [10,16,29,31], but experimental studies operate ORCs with the pressure of ~1 MPa (see Table 1). The temperature difference at the pinch point, vapor superheating at the expander inlet and liquid subcooling at the pump inlet are assumed to be not changed during operation. All these assumptions should be verified by experiments.

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Table 1
The literature survey on ORCs.

Ref.	Heat source & working fluid	Expander & pump types	Major parameters	Comments
[13]	Flue gas, 90–220 °C; R123	Scroll expander; multistage centrifugal pump	P_{eva} = 0.56–1.08 MPa W_{exp} = 0.16–0.65 kW $\eta_{exp,s}$ = N/A η_{th} = 2–8.5%	Both expander shaft power and pump consumption power were computed by the enthalpy difference. The output power was below 1 kW.
[18]	Diesel engine exhaust gas, 417–485 °C; R123	Single screw expander; multistage centrifugal pump	P_{eva} = 1.34 MPa W_{exp} = 10.38 kW $\eta_{exp,s}$ = 73.25% η_{th} = 6.48%	The expander shaft power was determined by dynamometer. The heat source temperature was extremely high. The thermal efficiency was lower than 5%.
[35]	Hot air, 105 °C; R245fa	Scroll expander; metering pump	P_{eva} = 0.5–1.0 MPa W_{exp} = 0.151 kW $\eta_{exp,s}$ = 41–72% η_{th} = 1.7–3.2%	Both expander shaft power and pump consumption power were computed by the enthalpy difference. The output power was below 1 kW, and the thermal efficiency was lower than 5%.
[37]	Hot water, 77.3 °C; R134a	Scroll expander; piston-diaphragm pump	P_{eva} = 22 bar W_{exp} = 2.05 kW $\eta_{exp,s}$ = N/A η_{th} = 4%	The expander shaft power was determined by dynamometer. The evaporating pressure was relatively high. The thermal efficiency was lower than 5%.
[38]	Solar energy, 41–75 °C; R134a	Scroll expander; positive displacement pump	P_{eva} = 3.5–9.5 bar W_{exp} = 0.25–0.92 kW $\eta_{exp,s}$ = N/A η_{th} = 0.73–1.17%	The expander shaft power was determined by dynamometer. The solar energy was unstable, and the actual thermal efficiency was very low.
[39]	Steam, 198 °C; R123	Gerotor expander, scroll expander; gear pump, piston pump	P_{eva} = 1.878–2.041 MPa W_{exp} = 2.07–2.96 kW $\eta_{exp,s}$ = 83–85% η_{th} = N/A	The way to compute the power was not given in the reference. The heat source temperature and evaporating pressure were very high. The piston pump had better performance than gear pump for ORC system.
[40]	Hot air, 101.7–165.2 °C; R123	Scroll expander; diaphragm pump	P_{eva} = 5.45–11.12 bar W_{exp} = 0.382–1.820 kW $\eta_{exp,s}$ = 68% η_{th} = N/A	The way to compute the power was not given. A semi-empirical model of scroll expander was established.
[41]	Conductive oil, 75–130 °C; R245fa, R245fa/R601a (0.72/0.28)	Scroll expander; diaphragm pump	P_{eva} = 3.6–10 bar W_{ele} = 0.2–0.55 kWe $\eta_{exp,s}$ = 71–83% η_{th} = 2.9–4.45%	The expander output power was calculated by the measured voltage and current. Two types of working fluids were compared. The output power was below 1 kW. The thermal efficiency was lower than 5%.
[42]	About 126 °C; HFE 7000	Vane-type expander; pump: N/A	P_{eva} = 6.66–6.724 bar W_{shaf} = 1.69–1.72 kW W_{ele} = 0.8248–0.8607 kWe $\eta_{exp,s}$ = 52.38–55.45% η_{th} = 3.73–3.89% η_{ele} = 1.38–1.40%	Both expander shaft power and pump consumption power were computed by the enthalpy difference. At the same time, the electric power was calculated by the measured voltage and current of the power generation machine. The result showed that the shaft power was overestimated by the enthalpy difference.
[43]	Electric boiler, 120–150 °C; R245fa	Scroll expander; diaphragm pump	P_{eva} = 13–18 bar $W_{exp,ele}$ = 1.5 kWe $\eta_{exp,s}$ = 60–74% η_{th} = 8%	The expander shaft power was calculated by the measured voltage and current of the power generation machine. The thermal efficiency was relatively high.
[44,45]	Pressurized hot water, 115–125 °C; R245fa	Twin screw expander; multistage centrifugal pump	P_{eva} = 1.2–1.4 MPa W_{exp} = 17–50 kW $\eta_{exp,s}$ = N/A η_{th} = 8.05%	The way to compute the power was not given. The output power reached 50 kW, and the thermal efficiency was relatively high.
[46]	Hot water, 85–116 °C; R245fa	Twin screw expander; pump: N/A	P_{eva} = 0.581–0.911 MPa W_{exp} = 15.5–39.9 kW $\eta_{exp,s}$ = N/A η_{th} = 5.6–8.3%	Both expander shaft power and pump consumption power were computed by the enthalpy difference, and the thermal efficiency was relatively high.
[47]	Steam and hot water; R245fa	Radial turbine; centrifugal pump	P_{eva} = 8.65 bar W_{exp} = 32.7 kW $\eta_{exp,s}$ = 78.7% η_{th} = 5.22%	Power output of the turbine was measured by power meter, and the pump consumption power was computed by the enthalpy difference. The thermal efficiency was more than 5%.
[56]	Hot water, 90 °C; R245fa	Scroll expander, trochoidal expander; DC diaphragm pump	P_{eva} = N/A W_{exp} = 7.2–8.4 W W_p = 4.9–5.6 W $\eta_{exp,s}$ = 4.55% η_{th} = 0.16–0.20%	Micro ORC generator was packaged without external power supply. The expander shaft power was determined by torque and speed meter. The thermal efficiency was very low.
[57]	Conductive oil, 140–160 °C; R123	Scroll expander; piston pump	P_{eva} = 0.55–1.5 MPa W_{exp} = 2.35–3.25 kW W_p = 0.2–1 kW $\eta_{exp,s}$ = 45–82% η_{th} = 5.12–6.39%	The expander shaft power was determined by AC dynamometer. The study identified that the measured shaft power was about 15–20% lower than the enthalpy determined value. The result indicated that the shaft power was overestimated by the enthalpy difference. The thermal efficiency was more than 5%.
[58]	Pressurized hot water, 120 °C; R245fa	Scroll expander; plunger pump	P_{eva} = 0.55–1.5 MPa W_{exp} = 1.7–3.4 kW W_p = N/A $\eta_{exp,s}$ = 60.9–61.2% η_{th} = 7.5%	ORC with multiple expanders used in parallel (PE-ORC) for large variation waste heat source. The expander shaft power was determined by torque and speed. The thermal efficiency was relatively high.
[59]	Exhaust gas for capstone diesel turbine, 302.7 °C; R245fa/R365mfc (0.485/0.515)	Scroll expander; plunger pump	P_{eva} = 1.4052 MPa W_{ele} = 0.7 kWe W_p = N/A $\eta_{exp,s}$ = N/A η_{th} = 3.9%	The actual electrical power was measured with the light bulb loading. The pump inlet subcooling temperature of 13 °C was reported. The thermal efficiency was lower than 5%.

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