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(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems

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Experimental Investigation of a Small-Scale Reversible High-Temperature Heat Pump – Organic Rankine Cycle System for Industrial Waste Heat Recovery

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ABSTRACT

A reversible system that can function as both a high-temperature heat pump (HTHP) and an organic Rankine cycle system (ORC) can enhance the efficiency of industrial waste heat recovery. The reversible system can either upgrade waste heat to process heat (HTHP mode) or generate power (ORC mode). This paper presents the experimental investigation of a small-scale reversible HTHP-ORC system using an automotive open-drive scroll machine to operate as the compressor (HTHP) and expander (ORC) in the system. The refrigerant R1233zd(E) was selected as the working fluid for both modes of operation. Based on an analysis of the experimental results, the HTHP achieved a COP_{el} of 4.6 at 30 K temperature lift, with the scroll compressor achieving a maximum overall isentropic efficiency of 73.4%. In ORC mode, at a source temperature of 75 °C, the cycle thermal efficiency was 3.8%, and the expander overall isentropic efficiency was 43%. 381

Keywords: waste heat recovery, reversible HTHP-ORC system, high-temperature heat pump, organic Rankine cycle, low GWP refrigerant.

1 INTRODUCTION

The global surface temperature has risen by 1.1 °C, and the negative impacts due to climate change will continue to intensify (IPCC, 2022). Decarbonising the industrial sector is a key strategy adopted by the European commission to achieve carbon neutrality by 2050 (European Commission, 2021). Waste heat recovery is an effective strategy in the transition towards decarbonisation. A reversible HTHP-ORC system could be an effective waste heat recovery technology in the industrial sector to upgrade heat or generate power, thus increasing energy efficiency. Due to the configuration similarities between the heat pump and organic Rankine cycle systems, it is possible to develop a reversible system with minimal modifications. Such a system can be applied in paper, chemical, and food industries to provide process heat (HTHP) when there is a heat demand and if heating is not required the same system can be used to generate electricity (ORC). The reversible system can also be implemented as part of a district heating network ,where the upgraded heat can be used for space heating in office and commercial buildings in summer season or for power generation (winter season) (Palomba *et al.*, 2020). The potential of the reversible system has also been explored in residential (Dumont *et al.*, 2015), automobile (Cuevas *et al.*, 2019) and marine (Kosmadakis *et al.*, 2022) sectors. Other research trends include a feasibility check of the reversible system coupled to a thermal storage as a Carnot battery (Eppinger *et al.*, 2021).

This paper describes a small-scale reversible HTHP – ORC system for recovering industrial waste heat within the lower temperature band (< 100 °C). The study focuses on the experimental investigation of a lab-scale prototype of the reversible system in HTHP and ORC modes, followed by a performance analysis of the whole system (HTHP and ORC) and scroll machine (compressor and expander roles).

2 REVERSIBLE SYSTEM LAYOUT

To operate a reversible HTHP-ORC system, a few equipment modifications are required. These include a compact single volumetric scroll acting as both a compressor (HTHP) and an expander (ORC), a valve arrangement to switch the refrigerant flow between the suction and discharge sides of the scroll machine, and a pump placed in parallel with the expansion valve. The schematic diagram of the reversible system is shown in Figure 1. This layout results in a compact and cost-effective system. In this system layout, the heat exchanger roles as evaporator and condenser remain unchanged when the system switches between ORC and HTHP modes.

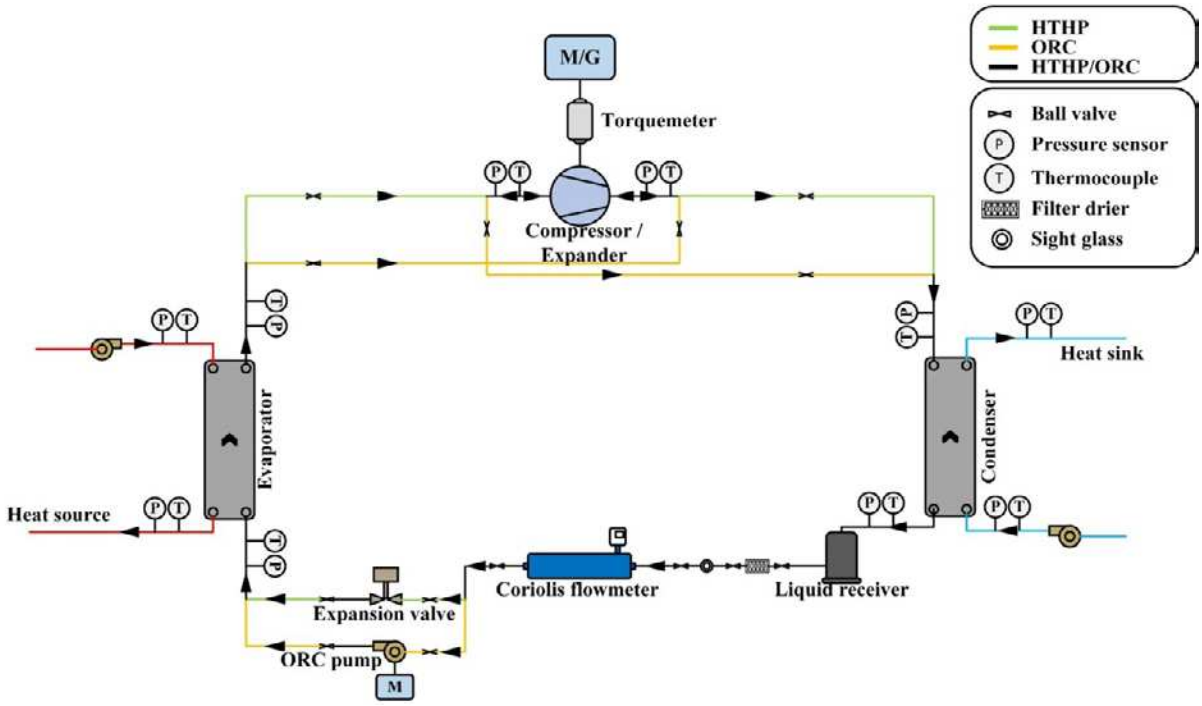

Figure 1: Schematic diagram of the reversible HTHP-ORC system

3 EXPERIMENTAL SETUP

The experimental setup of the reversible HTHP-ORC system is shown in Figure 2. A thermal balancing rig was connected to control the setpoint temperatures via secondary hydraulic circuits to the evaporator and condenser heat exchangers. A modified automotive A/C scroll compressor (Sanden TRSA09) was used for both compressor (HTHP) and expander (ORC) roles. The compressor has a swept volume of 85.7 cm^3 and a reported built-in volume ratio of 1.9. A belt drive was used for power transmission between the motor (HTHP) and the scroll machine. In ORC mode, the motor was replaced with an alternator and a rotary vane pump was chosen to pump the refrigerant. The specifications for the main system components and sensors are listed respectively in Table 1 and Table 2. The refrigerant R1233zd(E) was selected as the working fluid in both operating modes. R1233zd(E) has a very low ODP and a GWP of 3.88 (World Meteorological Organization, 2018) and is considered an effective replacement for R245fa. The selection of equipment and working fluid for the reversible system has been discussed in previous work. (Ravindran *et al.*, 2023).

Table 1: System equipment used in HTHP and ORC modes of operation.

Equipment	Model	Value
Scroll compressor/expander	Sanden TRSA09 (Model No: 3663)	Displacement (85.7 cm^3)
		Power (1.5 kW)
Motor	ETEC 3-phase induction motor (2 pole)	Power (3 kW)
	ETEC 3-phase induction motor (6 pole)	Power (0.25 kW)
Alternator	RTX alternator	12V, 65A
Inverter	Tec drive inverter	4 kW
	Tec drive inverter	0.37 kW
Evaporator	SWEP B85H (20 plates)	Heat exchange area (1.08 m^2)
Condenser	SWEP B26T (26 plates)	Heat exchange area (1.34 m^2)
Expansion valve	Danfoss (ETS 12.5)	-
Pump	Fluid-O-Tech (FPO 501)	Displacement (500 l/h)

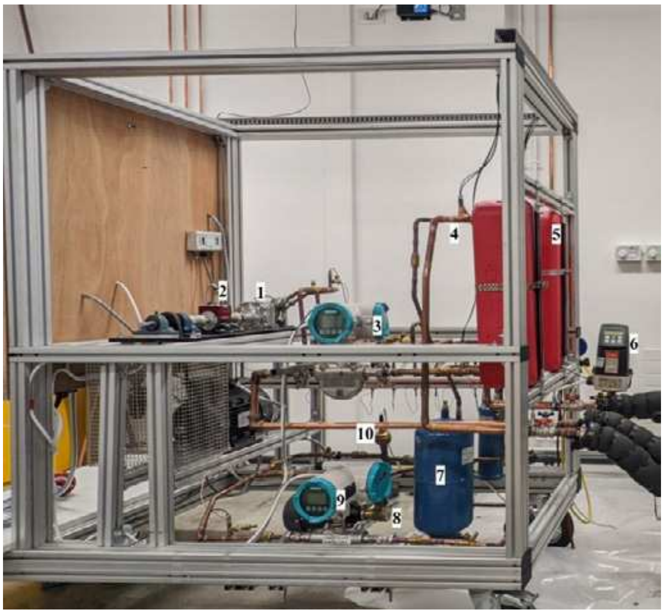

Figure 2: Experimental setup for reversible system (1). Scroll machine (2). Torque meter (3). Coriolis flow meter (sink side) (4). Condenser (5). Evaporator (6). Flow meter (source side) (7). Liquid receiver (8). ORC vane pump (9). Coriolis flow meter (refrigerant) (10). Expansion valve.

Table 2: Sensors used for measurement and data acquisition.

Sensor	Model	Variable	Range
PT100	RS PRO PT100 RTD Sensor	Temperature (˚C)	-50 – 250 °C
Pressure	Emerson PT3-30A (5 Nos)	Pressure (bar)(gauge)	0-30 bar
	ALCO PT4-50S (2 Nos)		0-50 bar
Torque meter	Sensor Technology SGR511-EB	Torque (Nm) & RPM	0-30 Nm
Flow meter	Siemens SITRANS FCS400	Mass flow rate (kg/s)	-
	Danfoss MAG 6000		-
Energy	Siemens 7KT 1665	Power (kW)	-
Data acquisition	Datalogger DT85	-	-

4 METHODOLOGY

The operating conditions for experiments in HTHP and ORC modes are presented in Table 3. The compressor was modified to operate as an expander by removing the check valve on the discharge side. In HTHP mode, motor speed controlled the refrigerant mass flow rate, whereas, in ORC mode, it was set by the rotary vane pump. Water and Therminol 66 were used as the secondary fluids for the source and sink sides, respectively, in HTHP mode; in ORC mode, normal tap water replaced Therminol 66 as the secondary fluid on the sink side. The refrigerant and water properties were taken from the REFPROP 10 database, while the Therminol 66 properties were provided by SWEP SSPG8 software. A DT85 datalogger was used for data acquisition.

Table 3: Operating conditions.

Variable	HTHP	ORC
Source inlet temperature (°C)	40 - 85	65 - 75
Sink outlet temperature (°C)	55 - 105	10
Evaporator thermal load (kW)	2.70 – 8.53	5.78 – 12.00
Condenser thermal load (kW)	3.00 - 8.82	5.69 – 11.70
Mass flow rate (kg/s)	0.0186 – 0.056	0.024 - 0.047

4.1 HTHP mode

The experiments in HTHP mode were conducted by varying the sink outlet temperature and compressor speed and maintaining the source inlet temperature setpoint constant. A total of 55 steady-state points were obtained. The compressor power can be calculated from the compressor speed and torque. The thermal power consumption of the compressor was determined from the suction and discharge enthalpies and the refrigerant mass flow rate. 384

$$\dot{W}_{cp,mech} = \frac{2\pi N_{cp}\tau}{60} \tag{1}$$

$$\dot{W}_{cp,therm} = \dot{m}_r * (h_{dis} - h_{su}) \tag{2}$$

The mechanical COP is defined as the ratio of heat supplied to the sink and mechanical power consumption, $\dot{W}_{mech}$.

$$COP_{mech} = \frac{\dot{Q}_{cd}}{\dot{W}_{mech}} \tag{3}$$

The COP_{el} is defined as the ratio of heat supplied to the sink and electrical power consumption, $\dot{W}_{el}$.

$$COP_{el} = \frac{\dot{Q}_{cd}}{\dot{W}_{el}} \tag{4}$$

The internal isentropic efficiency of the compressor was calculated as follows.

$$\eta_{cp,is,int} = \frac{(h_{dis,is} - h_{su})}{(h_{dis} - h_{su})} \tag{5}$$

Compressor overall isentropic efficiency was determined from refrigerant mass flow rate and suction and discharge enthalpies.

$$\eta_{cp,is,overall} = \frac{\dot{m}_r(h_{dis,is} - h_{su})}{\dot{W}_{mech}} \tag{6}$$

The volumetric efficiency of the compressor is defined as the ratio of the actual volume flow rate and the theoretical volume flow rate. The actual volume flow rate was determined from the refrigerant mass flow rate and specific volume at the suction side of the compressor.

$$\eta_{cp,vol} = \frac{\dot{m}_r v_{su}}{(\dot{V}_s N_{cp}/60)} \tag{7}$$

4.2 ORC mode

In ORC mode, the experiments were conducted by varying the source inlet temperature and the pump speed and maintaining the sink inlet temperature setpoint constant. The refrigerant mass flow rate was set by controlling the speed of the rotary vane pump. The superheat at the evaporator outlet was controlled by varying the mass flow rate.

The mechanical power produced by the expander was calculated using the expander RPM and torque. The thermal power generated was determined from the suction and discharge enthalpies and mass flow rate.

$$\dot{W}_{exp,mech} = \frac{2\pi N_{exp}\tau}{60} \tag{8}$$

$$\dot{W}_{exp,therm} = \dot{m}_r * (h_{su} - h_{dis}) \tag{9}$$

The thermal efficiency of the ORC was determined using the thermal power of the expander and pump, as well as the thermal load of the evaporator.

$$\eta_{orc,therm} = \frac{\dot{W}_{exp,therm} - \dot{W}_{pp,therm}}{\dot{Q}_{ev}} \tag{10}$$

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The system overall efficiency was calculated as the ratio of net power generated to heat supplied, $\dot{Q}_{ev}$.

$$\eta_{orc,overall} = \frac{\dot{W}_{mech} - \dot{W}_{pp}}{\dot{Q}_{ev}} \tag{11}$$

The performance of the scroll expander was characterized by determining the internal isentropic and overall isentropic efficiency and filling factor.

$$\eta_{exp,is,int} = \frac{(h_{su} - h_{dis})}{(h_{su} - h_{dis,is})} \tag{12}$$

$$\eta_{exp,is,overall} = \frac{\dot{W}_{mech}}{\dot{m}_r(h_{su} - h_{dis,is})} \tag{13}$$

$$\phi_{ff} = \frac{\dot{m}_r v_{su}}{(\dot{V}_s N_{cp}/(60 * r_v))} \tag{14}$$

5 RESULTS AND DISCUSSION

5.1 HTHP mode

The scroll compressor performance was analysed through isentropic and volumetric efficiencies. The internal isentropic and overall isentropic efficiencies of the compressor at different pressure ratios is shown in Figure 3. Overall isentropic efficiency at T_{source} = 60 °C and 1500 RPM was 73.4%. Higher isentropic efficiencies were obtained between 1500 and 2000 rpm, with a slightly decreasing trend as the speed increased.

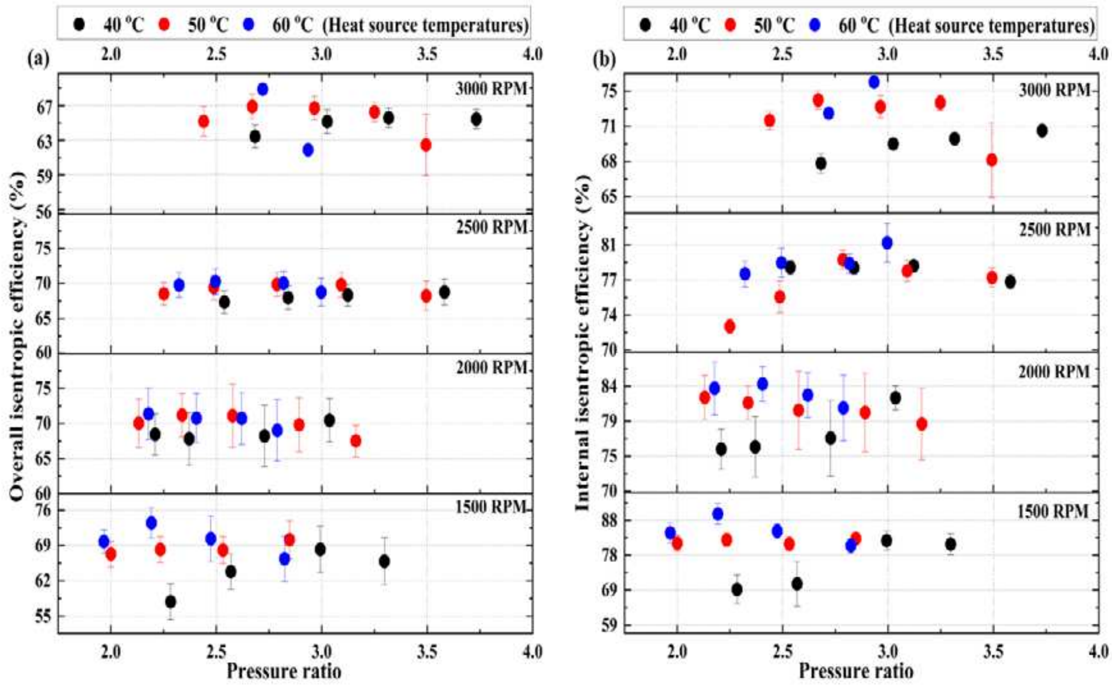

Figure 3: (a) Overall isentropic and (b) internal isentropic efficiencies of the scroll compressor with respect to operating pressure ratio.

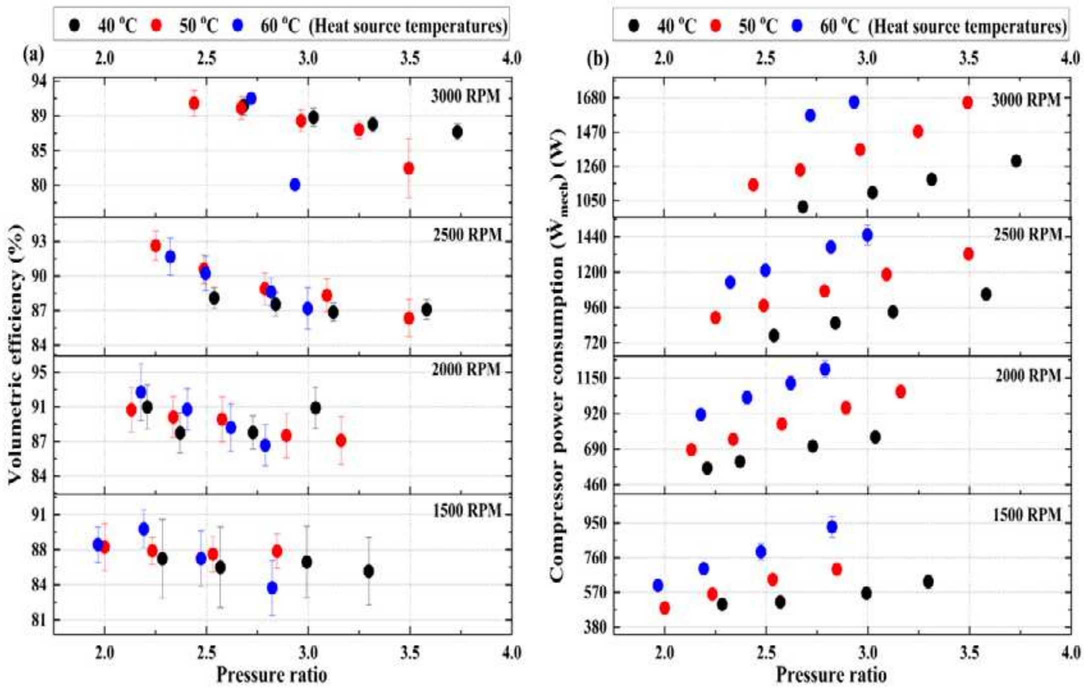

Figure 4: (a) Volumetric efficiency of the scroll compressor and (b) power consumption with respect to operating pressure ratio.

As shown in Figure 4, The volumetric efficiency of the compressor was close to unity (less internal leakages) at lower pressure ratios and decreases with increasing pressure ratio (indicating internal leakages in the compressor) across the range of compressor speeds and source temperatures. The compressor speed also have an impact on the volumetric efficiency. Higher volumetric efficiencies were observed in the 2000 – 2500 RPM range. Volumetric efficiencies showed a slightly decreasing trend in the 2500 – 3000 RPM range. The mechanical power consumption ($\dot{W}_{mech}$) of the compressor varied between 0.48 kW (Ncp: 1500 RPM, rp: 2) and 1.65 kW (Ncp: 3000 RPM, rp: 2.93). The compressor power increased with respect to increase in pressure ratio, compressor speed as well as with respect to rise in source temperature.

The COPs of the system for different temperature lifts (absolute temperature difference between source inlet and sink outlet temperatures) are shown in Figure 5. COP decreases with increase in temperature lift. For a temperature lift of 30 K, maximum COP_{el} and COP_{mech} values of 4.6 (T_{source} = 50 °C, at 1500 RPM) and 5.5 (T_{source} = 60 °C, at 1500 RPM) respectively were obtained.

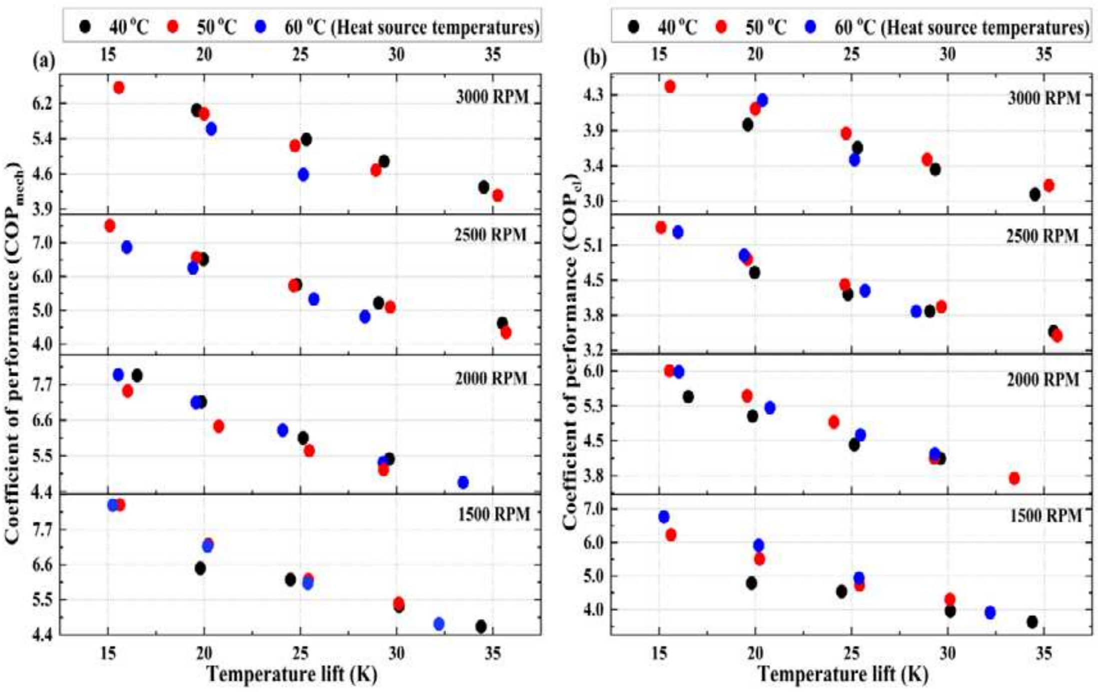

Figure 5: HTHP system performance (a) (COP_{mech}) and (b) (COP_{el}) for various temperature lifts.

5.2 Organic Rankine cycle mode

The internal and overall isentropic efficiencies of the scroll expander is shown in Figure 6. For the operating conditions, the overall isentropic efficiency increases as the mass flow rate increases (filling factor (ϕ_{ff}) reduces at higher mass flow rates indicating less internal leakage and better overall performance of the expander). Maximum isentropic efficiency of 43 % was obtained at a source temperature of 75 °C. The highest value for internal isentropic efficiency was found to be 69 %.

The thermal and overall cycle efficiencies of the system in ORC mode are shown in Figure 7. At a source temperature of 75 °C (pressure ratio, r_p: 2.13), maximum thermal and overall cycle efficiencies of 3.8% and 2.1%, respectively, were achieved. The expander performance in terms of filling factor is shown in Figure 8(a). The filling factor (ϕ_{ff}) decreased as the refrigerant mass flow rate increased. Higher filling factor values up to 2 at T_{source} = 65 °C were observed at lower mass flow rates indicating higher internal leakages in the scroll expander as the working chamber is overfilled. Throughout the experimental campaign under the mentioned operating conditions, filling factor was always found to be above unity. The gross mechanical power generated by the expander increased with mass flow rate and source temperature (Figure 8(b)) and reached a maximum of 307 W (T_{source}: 75 °C).

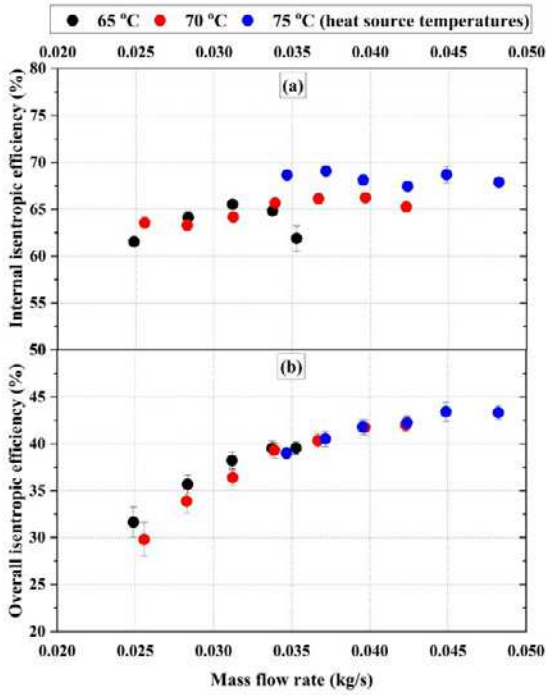

Figure 6: (a) Internal and (b) overall isentropic efficiencies of the expander with respect to mass flow rate.

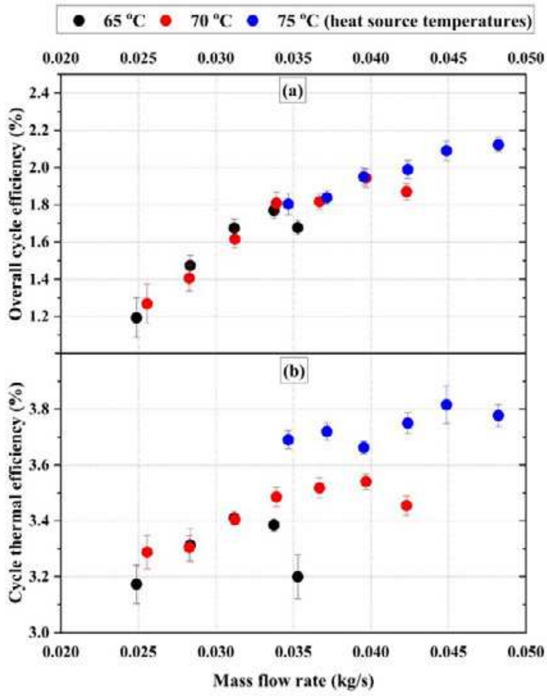

Figure 7: (a) Thermal and (b) overall efficiencies of the system in ORC mode.

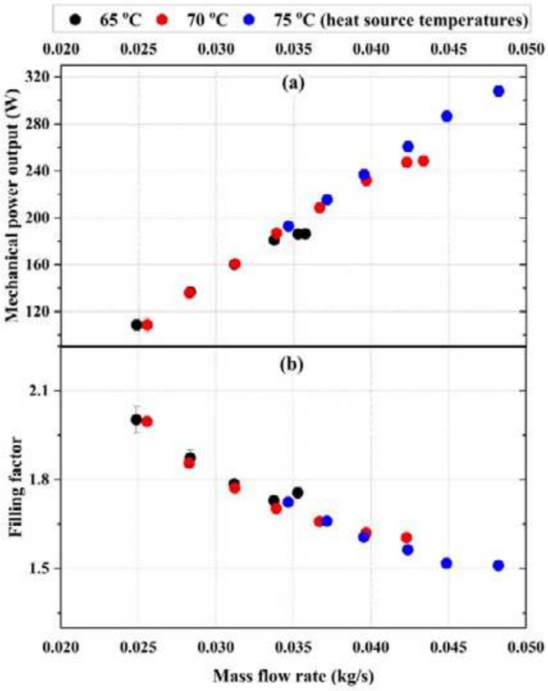

Figure 8: (a) Mechanical power generated and (b) filling factor of the expander with respect to mass flow rate.

6 CONCLUSION

The paper presents the experimental study of a reversible HTHP-ORC system. The performance of the system and the scroll machine in both operating modes were investigated. The scroll machine had better isentropic efficiency values in the compressor role when compared to that of the expander. Maximum overall isentropic efficiencies were 73.4% and 43% respectively in compressor and expander roles. In terms of system performance, a maximum COP_{el} of 4.6 in HTHP mode and a maximum overall cycle efficiency of 2.1% (T_{source}: 75 °C, r_p: 2.13) was obtained in ORC mode. Future experiments in ORC mode will focus on heat source temperatures of 80, 85 and 90 °C.

NOMENCLATURE

$\dot{m}$	Mass flow rate (kg/s)	ff	filling factor
N	Speed (RPM)	int	internal
h	specific enthalpy (J/kg)	is	isentropic
$\dot{Q}$	heat transfer (kW)	lift	lift
$\dot{W}$	power (kW)	mech	mechanical
		overall	overall
Subscript		pp	pump
cd	condenser	r	refrigerant
cp	compressor	sf	secondary fluid
dis	discharge	su	suction
ev	evaporator	therm	thermal
exp	expander	v	volume
el	electrical	vol	volumetric

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The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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