

An Experimental Investigation of Organic Rankine Cycle with Scroll Expander for Low-Temperature Heat Sources

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ABSTRACT

Effective utilization of low-grade waste heat remains a significant challenge in sustainable energy recovery. This study presents an experimental investigation of an organic Rankine cycle (ORC) system utilizing R134a as the working fluid, integrated with a scroll expander modified from an electric automotive compressor. The primary objective was to evaluate the system's thermodynamic performance under varying low-temperature heat source conditions (50°C to 90°C) and heat sink temperatures (8°C to 16°C). The experimental results demonstrated that the system performance is heavily dependent on the thermal gradient, which governs the pressure ratio and enthalpy drop. Under the operating condition (evaporating temperature of 90°C and condensing temperature of 8°C), the ORC system generated a maximum electrical power output of 488.55 W. At this peak point, the ORC system achieved a thermal efficiency of 2.61%, with the scroll expander exhibiting a maximum isentropic efficiency of 21.14%. Detailed analysis reveals that the efficiency limitations are primarily attributed to under-expansion losses caused by the mismatch between the system pressure ratio and the expander's fixed built-in volume ratio, compounded by internal leakage. Despite the lower efficiency compared to specifically built expanders, this study confirms the techno-economic feasibility of using low-cost, off-the-shelf automotive components for small-scale decentralized power generation, offering a robust and cost-effective solution for low-grade heat recovery.

1. Introduction

The increase in global energy consumption, coupled with the reduction in greenhouse gas emissions, has turned researchers' focus toward sustainable energy technologies and advanced waste-heat recovery systems. Currently, a substantial portion of industrial energy and renewable resources exists in the form of low-grade heat, typically at temperatures below 200°C. It is estimated that approximately 63% of this low-grade heat is released into the environment at temperatures below 100°C without being effectively utilized [1-4]. However, recovery rates remain low due to the lack of cost-effective technologies adapted for small-scale applications. Consequently, harnessing this abundant yet underutilized thermal energy represents a significant opportunity to enhance overall energy efficiency and contribute to carbon neutrality. Although the conventional steam Rankine cycle plays a crucial role in large-scale power generation, it is technically and economically infeasible for such low-temperature applications due to the specific thermophysical properties of water, such as its high specific volume at low pressures, which necessitates excessively large turbine blades and complex multi-stage expansion setups to prevent blade erosion [5].

In this regard, the Organic Rankine Cycle (ORC) has established itself as a prominent technology for valorizing low-grade heat into electricity [2, 6, 7]. Distinct from conventional steam cycles, ORC systems utilize organic working fluids possessing lower boiling points and higher molecular weights, thereby enabling efficient operation at

moderate source temperatures [8]. The selection of working fluid is a critical design parameter, as its thermodynamic properties directly influence the efficiency, operating pressures, and environmental footprint regarding Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) [9, 10]. However, the widespread commercialization of small- to medium-scale ORC systems (<10 kW) is heavily bottlenecked by a critical technical challenge: the expansion device. The expander is the core component that determines the isentropic efficiency and the net power generation of the cycle [11]. Although radial inflow turbines are dominant in high-capacity applications (>50 kW) owing to their superior efficiency, they face practical challenges in small-scale systems, including high rotational speeds (>30,000 RPM) and elevated fabrication costs. In contrast, positive displacement expanders offer a viable solution for low-power applications, offering advantageous low-speed operation and effective handling of low mass flow rates.

Among the various expander technologies, scroll expanders have gained considerable attention for small ORC applications. These devices are often modified from scroll compressors used in refrigeration systems due to their advantages, including a compact and simple construction, low noise and vibration levels, and the ability to handle two-phase fluids [12, 13]. Extensive experimental efforts have been devoted to characterizing the performance of these devices under diverse operating conditions. However, the reported isentropic

efficiencies exhibit significant variance depending on the specific device modification and design. Investigations employing specialized or compliant scroll mechanisms typically demonstrate superior performance. For instance, Wang et al. [14] achieved a remarkable isentropic efficiency of 77% using a compliant scroll compressor as an expander with R134a, while Lemort et al. [15] reported an isentropic efficiency of 68% for an open-drive scroll expander using R123. Miao et al. [16] further explored an oil-free scroll expander with R123, achieving a maximum isentropic efficiency of 58.3%, highlighting the potential of oil-free systems in eliminating separation issues. Conversely, studies focusing on cost-effective solutions by modifying stock scroll compressors often report lower efficiencies due to the lack of optimization for the expansion process. A primary factor contributing to this efficiency gap is the mismatch between the expander's built-in volume ratio and the system's operating pressure ratio. When these ratios diverge, under-expansion or over-expansion losses occur, significantly penalizing the power output. Quoilin et al. [17] investigated a modified hermetic scroll compressor with R134a and reported a maximum isentropic efficiency of approximately 42%. Similarly, Chang et al. [18] conducted experiments with an automotive-type scroll expander using R245fa and observed efficiencies peaking at 56.6%. Furthermore, investigations by Campana et al. [19], Gao et al. [20], and Kosmadakis et al. [21] have reported isentropic efficiencies ranging from 40% to 55.3% in specific setups. Collectively, the reported efficiencies range widely from 40% to nearly 80%. This underscores the sensitivity of scroll expanders to design variations, particularly regarding internal leakage and friction losses. The modification of a scroll compressor for expansion purposes introduces inherent technical complexities, which can drastically reduce efficiency when the system operates away from its design point.

More recently, research focus has increasingly shifted toward investigating the dynamic response and operational stability of ORC systems under transient conditions, reflecting the intermittent nature inherent to real-world waste heat sources. For instance, Yu et al. [22] analyzed the dynamic behavior of an ORC system, demonstrating that heat source fluctuations significantly compromise both system safety margins and efficiency. Similarly, Li et al. [23] experimentally assessed a small-scale ORC unit under variable heat source conditions, reporting that cycle efficiency is acutely sensitive to inlet temperature and flow rate variations, thereby necessitating optimized control strategies. Furthermore, Liu et al. [24] underscored the critical challenges in scroll expander operation—specifically internal leakage and structural deformation under high pressure—identifying them as the predominant factors limiting the isentropic efficiency of modified devices. These recent findings reinforce the imperative for detailed experimental mapping of modified scroll expanders to establish stable operating windows for low-grade heat recovery.

Despite the growing interest in scroll-based ORC systems, there remains a gap in comprehensive experimental data, particularly concerning performance under off-design conditions. Most existing literature focuses on theoretical modeling or steady-state experiments at fixed points. Consequently, there is a distinct lack of thorough investigations that map the scroll expander's behavior across the wide range of heat source and heat sink temperature variations typical of real-world waste heat recovery scenarios. Furthermore, theoretical models often rely on generalized assumptions regarding leakage coefficients and heat transfer, which may not accurately capture the complex physical behavior of a modified compressor. Therefore,

elucidating the expander's response to these fluctuations is crucial for validating theoretical models, defining practical operational limits, and optimizing control strategies.

To address this gap, this study presents an experimental investigation of an organic Rankine cycle integrated with a modified scroll expander using R134a as the working fluid for low-temperature heat sources. The primary objective is to characterize the ORC system's behavior under distinct operating regimes. Specifically, the power output, isentropic efficiency, and overall thermal efficiency were evaluated under varying heat source and heat sink temperatures. The findings from this study provide vital insights into component and system behavior, contributing to the development of more robust and efficient ORC systems for sustainable low-grade waste heat conversion.

2. ORC experimental setup

2.1 ORC system description

The physical arrangement of the experimental ORC test bench is depicted in Fig. 1. To provide a clear understanding of the system's architecture and address the complex spatial layout, the figure explicitly illustrates the top-down configuration of the primary components (evaporator, scroll expander, condenser, and pump). Furthermore, the continuous working fluid flow path is strategically traced using directional arrows through the thermodynamic cycle sequence. The facility is designed to simulate a complete thermal power generation cycle, comprising three interacting circuits: the working fluid loop, the heat source loop, and the heat sink loop. The core of the ORC circuit is powered by a pneumatic booster pump driven by an external air compressor. Unlike conventional centrifugal pumps, this pneumatic configuration ensures precise pressure control and facilitates a stable, pulsation-free flow. Furthermore, it allows for the decoupling of fluid pressure from the actuation speed, thereby providing a broader operating range for off-design testing.

From the pump, the pressurized fluid enters the evaporator to absorb thermal energy from the hot oil loop. A coaxial tube-in-tube heat exchanger configuration was selected for both the evaporator and condenser due to its high heat transfer coefficient and compact footprint, effectively minimizing thermal inertia during transient state changes. The resulting high-pressure vapor expands through the scroll expander to generate mechanical work, which is converted into electricity and dissipated via an electric load bank to quantify power output. Following expansion, the working fluid is condensed by an air-cooled chiller and collected in a receiver tank (green vessel) to ensure a continuous liquid supply to the pump intake.

Regarding the heat source, a thermal oil circuit—driven by an oil pump and regulated by an electric heater—was utilized to maintain the desired evaporating temperatures. Thermal oil was specifically selected over water to prevent localized boiling and cavitation within the heating circuit at temperatures approaching 100°C, ensuring uniform heat flux delivery to the evaporator. The entire operation was monitored and governed via a centralized control cabinet, which housed the Programmable Logic Controller (PLC) and provided a real-time Human-Machine Interface (HMI) for data visualization and system safety management. Detailed specifications of the primary experimental components, alongside the operating ranges and uncertainties of the measurement instrumentation, are comprehensively summarized in Table 1.

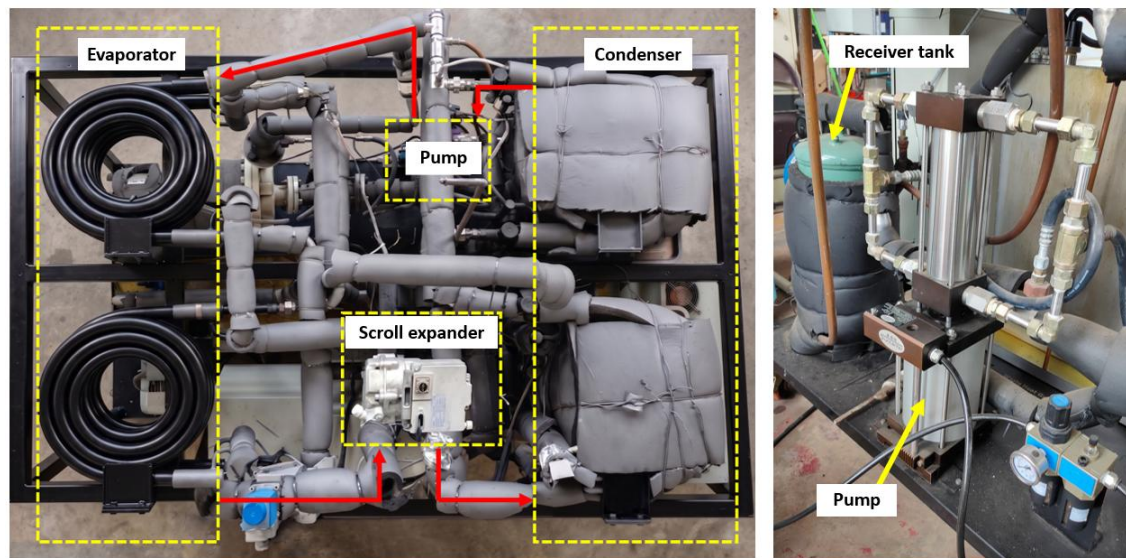


Fig. 1 Top-down view of the ORC test bench illustrating the physical arrangement of primary components and the working fluid path.

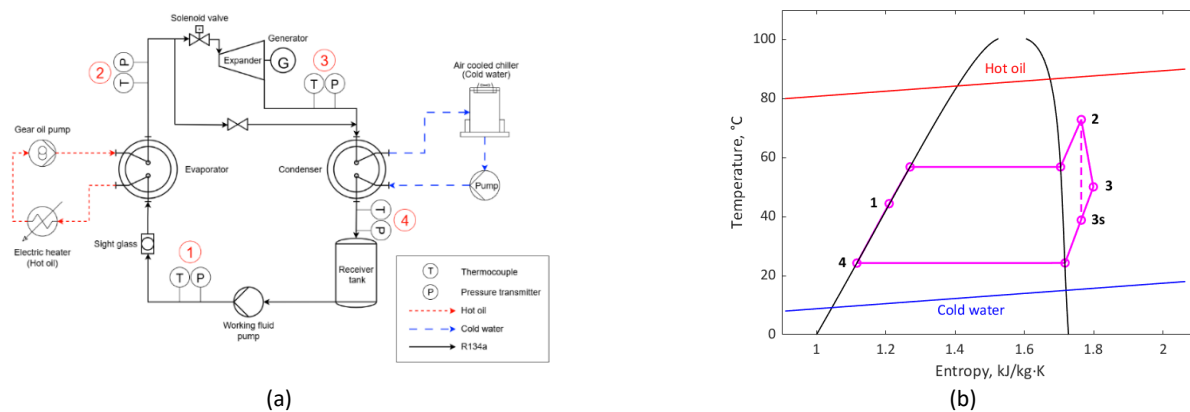


Fig. 2 (a) ORC test bench diagram with sensors position and (b) Thermodynamic process on temperature-entropy diagram.

Table 1 Specifications of the main ORC components and measurement instrumentation.

Category / Component	Model / Type	Key Specifications & Characteristics
Main System Components		
Evaporator	Coaxial heat exchanger	Coupled with gear oil pump (KCB-83.3)
Condenser	Coaxial heat exchanger	Coupled with air-cooled chiller (MIC-5A)
Expander	Modified scroll compressor (DQ34A350AB)	Displacement: 34 cm ³ /rev Speed range: 1,000 - 6,000 rpm
Working fluid pump	Booster pump (2SLM06)	Max pressure: 48 bar Output flow: 30 LPM Boost ratio: 6:1 Stroke: 120 mm Diameter: 63 mm
Liquid receiver tank	Standard vessel	Capacity: 19 L
Measurement Instrumentation		
Pressure transmitter	ZGPT9200W	Range: 0–40 bar Uncertainty: $\pm 0.5\%$ Full scale
Temperature sensor	Type-K Thermocouple	Range: -25 - 200 °C Uncertainty: ± 1 °C
Power meter	Multifunction meter (KM-07)	Range: N/A Uncertainty: $\pm 0.5\%$ Full scale

Table 2 Thermophysical properties of working fluid.

Refrigerant	Critical temperature (°C)	Critical pressure (MPa)	Boiling point (°C)	ODP	GWP	ASHRE standard 34 Safety group
R134a	101.06	4.06	-26.3	0	1,430	A1

2.2 Heat source and heat sink loops

To simulate realistic low-grade waste heat conditions, the system was driven by a heating loop comprising a hot oil tank integrated with electric heaters and a gear oil pump (Model KCB-83.3). This setup allowed for the precise simulation of a heat source with temperatures ranging from 50°C to 90°C. The thermal load was regulated by adjusting the power input to the electric heaters, thereby ensuring stable and uniform evaporation temperatures within the coaxial evaporator during steady-state data acquisition.

Simultaneously, on the heat rejection side, the cooling loop consisted of an industrial air-cooled chiller (Model MIC-5A) coupled with a buffer water reservoir and a circulation pump. This loop served as the heat sink, maintaining nominal cooling water setpoints between 8°C, 12°C and 16°C. The flexibility of the air-cooled chiller allowed for the investigation of system performance under varying ambient conditions, which directly influenced the condensation pressure within the coaxial condenser. By actively controlling the cooling water temperature, the experimental setup can effectively simulated diverse seasonal operational scenarios, thereby providing a comprehensive assessment of the ORC's adaptability and robustness in response to environmental thermal gradients.

2.3 Data acquisition and measurement instrumentation

For the purpose of a comprehensive evaluation of the ORC system's thermodynamic performance, a complete set of measurement instrumentation was strategically installed at critical state points, as illustrated in the schematic diagram in Fig. 2(a). Specifically, temperature and pressure variations were continuously monitored at four distinct nodes—corresponding to the inlet and outlet of the major components—to accurately characterize the thermodynamic cycle and determine the working fluid properties at each stage:

- The pump outlet (State 1), measuring the high-pressure liquid entering the evaporator
- The evaporator outlet (State 2), monitoring the superheated vapor entering the expander
- The expander outlet (State 3), characterizing the low-pressure vapor entering the condenser
- The condenser outlet (State 4), tracking the subcooled liquid returning to the receiver tank and pump.

Regarding the energy conversion measurement, the gross electrical power output generated by the scroll expander was quantified using a precision multifunction power meter (Model KM-07) electrically coupled to the generator terminals. All analog signals collected from the field instruments were transmitted to a centralized Programmable Logic Controller (PLC) for digital processing and subsequently logged via a dedicated supervisory interface on a laptop computer to facilitate real-time visualization and historical recording. The data acquisition was executed at a sampling frequency of 0.2 Hz. This sampling rate was selected as it was sufficient to capture the system's thermal dynamic behavior—which is governed by the relatively high thermal inertia of the heat exchangers—while effectively filtering out high-frequency noise from the dataset.

Detailed parameters and specifications of the measuring instrumentation are summarized in Table 1. The system utilized Type-K thermocouples for temperature measurements with an accuracy of $\pm 1^\circ\text{C}$, and ZGPT9200W pressure transmitters with an operating range of 0–40 bar and an uncertainty of $\pm 0.5\%$ of the full scale. To ensure the reliability of the experimental results, an uncertainty analysis was conducted to estimate the propagation of measurement

errors into the calculated performance indicators. The uncertainty of the direct measurements (temperature, pressure and power) depends on the accuracy of the sensors listed in Table 1. For the derived parameters (enthalpy, power output, and efficiency), the uncertainty was calculated using the root-sum-square method based on previous research [9, 18, 25]. Suppose a calculated result R is a function of independent variables $x_1, x_2, \dots, x_n$. In that case, the total uncertainty (ΔR) is determined by combining the individual uncertainties (Δx_i) of each variable, as expressed in Eq. (1):

$$\Delta R = \sqrt{\sum_{i=1}^n \left(\frac{\partial R}{\partial x_i} \right)^2 \Delta x_i^2} \quad (1)$$

Where $\partial R / \partial x_i$ represents the sensitivity coefficient of the result R with respect to the variable x_i .

2.4 Working fluid selection

The selection of an appropriate working fluid constitutes a fundamental determinant in the thermodynamic design of ORC systems, particularly when targeting low-grade heat recovery applications ($<100^\circ\text{C}$). From a thermodynamic perspective, the ideal working fluid should exhibit a low critical pressure and a critical temperature that marginally exceeds the target evaporation temperature to ensure subcritical operation while maximizing cycle efficiency [12, 26]. Furthermore, environmental and safety parameters, such as the Ozone Depletion Potential (ODP), Global Warming Potential (GWP), and toxicity/flammability levels, act as critical constraints for experimental setups. While various pure fluids and mixtures have been extensively investigated across different ORC applications [27, 28], R134a was selected as the optimal working fluid for this specific investigation. As detailed in Table 2, R134a exhibits a critical temperature of 101.06°C , which aligns perfectly with the target heat source temperature range (50°C – 90°C) utilized in this study, thereby guaranteeing stable subcritical cycle execution. Crucially, from an operational safety perspective, R134a is designated under the A1 safety group (ASHRAE Standard 34), indicating non-flammability and low toxicity [29]. This classification facilitates safe handling and implementation within an indoor laboratory environment without the need for complex mitigation systems. Additionally, a significant technical advantage of utilizing R134a is its excellent compatibility with standard sealing materials and lubricating oils inherent to modified automotive scroll compressors. This intrinsic compatibility was paramount for mitigating the risk of seal degradation and minimizing internal leakage during the long-term off-design experimental mapping.

2.5 Experimental procedure and operating conditions

The experimental investigation was structured to evaluate the thermodynamic performance of the ORC system under a wide range of off-design scenarios, utilizing R134a as the working fluid. The study focused on two primary independent variables: the heat source temperature (evaporating temperature) and the heat sink temperature (condensing temperature). As detailed in Table 3, the experimental design involved varying the evaporating temperature across four specific levels: 50°C , 60°C , 80°C , and 90°C , achieved through regulation of the hot oil loop. Concurrently, the nominal cooling water setpoints were modulated to 8°C , 12°C , and 16°C using the industrial chiller unit.

The specific heat source temperatures were dictated by the discrete operational setpoints of the heating loop, which was driven by four electric heaters. These heaters were activated

sequentially to create a stable thermal equilibrium at temperatures of 50, 60, 80 and 90°C, effectively simulating the ultra-low-grade industrial waste heat. The intermediate setpoint of 70°C was excluded, as the selected intervals were deemed adequate to characterize the critical transition from the parasitic-dominant regime (<80 °C) to the net positive power generation regime (≥80 °C).

To ensure the reliability of the data, all experiments were conducted in a temperature-controlled laboratory. During each test, the system was operated until a strict steady-state condition was achieved. This steady state was quantitatively defined as the operational period where fluctuations at all measurement nodes were confined entirely within the instrument uncertainty limits: specifically, ±1°C for temperature and ±0.2 bar for pressure. Once these specific stability criteria were verified, the operating condition was maintained for an initial stabilization period of 1,800 seconds to overcome system thermal inertia. Subsequently, the actual experimental data were continuously logged for a duration of 3,600 seconds to accurately capture the long-term stable performance characteristics of the cycle [12, 13].

Table 3 Operating conditions of the experiment.

Parameters	Value
Working fluid	R134a
Evaporating Temperature (Hot oil), T_H	50, 60, 80, 90 °C
Condensing Temperature (Cold water), T_C	8, 12, 16 °C

3. Thermodynamic analysis method

The thermodynamic cycle of the experimental ORC system is graphically depicted in the Temperature-Entropy (T-s) diagram presented in Fig. 2(b). Rather than assuming ideal thermodynamic processes, the state points (1–4) were determined explicitly using experimentally measured temperatures and pressures, thereby enabling accurate representation of real-fluid effects, including pressure drops and thermodynamic non-idealities, across the heat exchangers and piping network. The thermophysical properties of the R134a working fluid were retrieved utilizing the NIST REFPROP database.

To formulate the governing energy balance equations based on the First Law of Thermodynamics, the following assumptions were applied:

- (1) The system operates strictly under steady-state conditions.
- (2) Heat dissipation to the ambient surroundings and variations in kinetic and potential energies are negligible.
- (3) The expansion process encompasses all internal mechanical and leakage irreversibilities.

Instead of detailing the standard macroscopic energy balance for each individual component, a generalized approach was utilized. The heat transfer rates in the evaporator ($\dot{Q}_{in}$) and condenser ($\dot{Q}_{out}$), as well as the mechanical power of the pump ($\dot{W}_{pump}$) and the gross indicated power of the expander ($\dot{W}_{exp}$), were computed using the general steady-state energy equation: $\dot{Q}_{in}$ or $\dot{W}_{exp} = \dot{m}_r \Delta h$. Here, $\dot{m}_r$ represents the working fluid mass flow rate, and Δh denotes the specific enthalpy difference between the outlet and inlet of the respective component.

To evaluate the energy conversion quality of the modified scroll expander, the isentropic efficiency (η_{is}) was assessed. Since direct shaft torque measurement was practically constrained, the measured gross electrical power ($\dot{W}_{elec}$) was utilized to evaluate the overall expander performance, inherently accounting for electromechanical losses. The isentropic efficiency is expressed as shown in Eq. (2).

$$\eta_{is} = \frac{\dot{W}_{elec}}{\dot{m}_r(h_{in,exp} - h_{out,is})} \quad (2)$$

where $h_{in,exp}$ is the specific enthalpy of the superheated vapor at the expander inlet, and $h_{out,is}$ is the ideal specific enthalpy at the expander discharge following an isentropic expansion to the measured condensing pressure.

To quantify the detrimental impact of the pump energy consumption on the small-scale system, the Back-Work Ratio (BWR) is defined as the ratio of the parasitic pump power to the gross electrical power generated as shown in Eq. (3).

$$BWR = \frac{\dot{W}_{pump}}{\dot{W}_{elec}} \quad (3)$$

Finally, the thermal efficiency (η_{th}) of the ORC system is defined as the ratio of the net power output to the total heat input—reflects the First Law efficiency, quantifying the system's capability to convert low-grade thermal energy into useful electrical work, as shown in Eq. (4).

$$\eta_{th} = \frac{\dot{W}_{exp} - \dot{W}_{pump}}{\dot{Q}_{in}} \quad (4)$$

4. Result and discussion

4.1 Working characteristic

This section details the working characteristics of the ORC system integrated with the scroll expander using R134a as the working fluid. Understanding the transient behavior and the transition to a steady state is fundamental to ensuring the validity of the thermodynamic performance data.

Fig. 3(a) illustrates the typical temperature and pressure profiles of the working fluid at key locations under a high-temperature heat source scenario ($T_H = 90^\circ\text{C}$ and $T_C = 8^\circ\text{C}$). Upon system startup, a distinct transient phase is observed where both pressure and temperature rise sharply due to the thermal energy input from the heating loop. This initial ramp-up phase is characterized by non-linear fluctuations as the two-phase fluid distribution stabilizes within the heat exchangers. As the system overcomes the thermal inertia of the components and the heat transfer rates stabilize, the thermodynamic parameters exhibit an asymptotic approach toward equilibrium. Importantly, under such high thermal load conditions, this equilibrium for the heat sink is dictated by the equipment's maximum cooling capacity rather than its nominal setpoint. Because the massive heat rejection demand exceeded the chiller's limits, the cooling water inlet temperature naturally stabilized at an elevated equilibrium of $21.1 \pm 0.46^\circ\text{C}$, rather than the 8°C setpoint. The ORC system typically requires approximately 1,800 seconds to reach a fully developed steady-state condition. In this study, the steady-state criterion was defined as the period where fluctuations in temperature and pressure at all measurement nodes remained within a narrow band corresponding to the measurement uncertainty limits relative to the mean value over a rolling time window. This high degree of stability indicates a well-balanced operation between the heat input from the heat source and the heat rejection processes.

To verify the ORC system's robustness under varying operating envelopes, the dynamic response was also evaluated at a lower thermal gradient condition ($T_H = 50^\circ\text{C}$ and $T_C = 16^\circ\text{C}$), as depicted in Fig. 3(b).

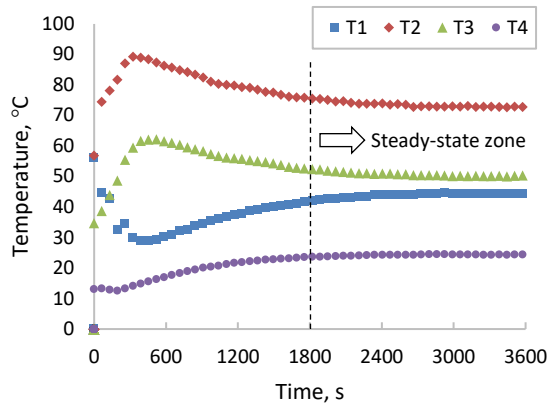
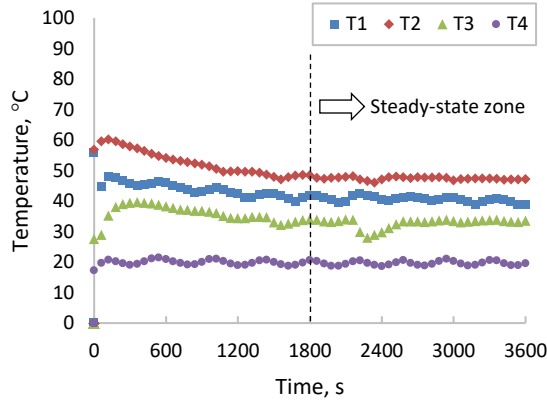
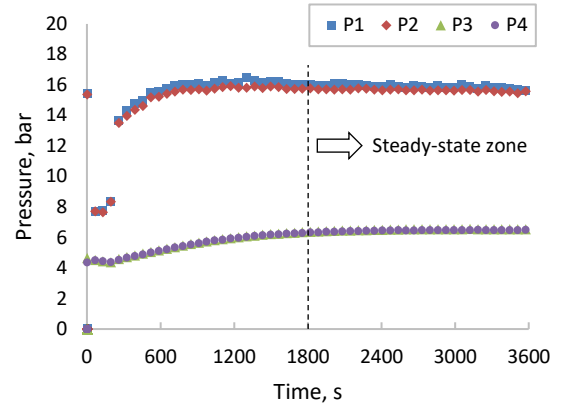
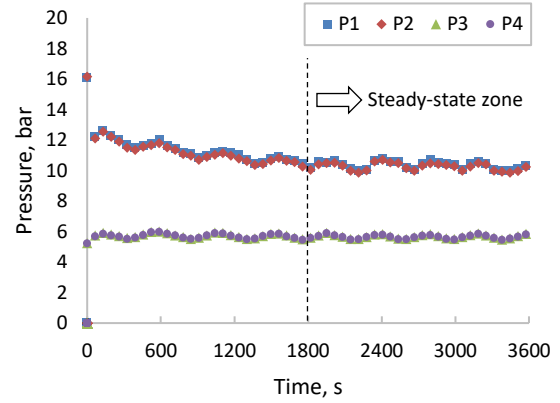
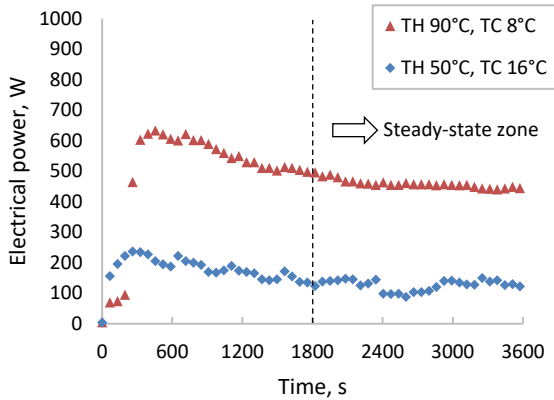
(a) At $T_H = 90^\circ\text{C}$, $T_C = 8^\circ\text{C}$ (b) At $T_H = 50^\circ\text{C}$, $T_C = 16^\circ\text{C}$ Fig. 3 Temperature and pressure profiles of the ORC: (a) At $T_H = 90^\circ\text{C}$, $T_C = 8^\circ\text{C}$, (b) At $T_H = 50^\circ\text{C}$, $T_C = 16^\circ\text{C}$.

Fig. 4 Electrical power of the ORC in various operation.

Unlike the high-temperature scenario, the reduced thermal load in this case allowed the chiller to effectively manage the heat rejection, stabilizing the cooling water temperature of $18.6 \pm 0.81^\circ\text{C}$ closer to its designated setpoint. Despite the difference in absolute operating pressures and temperatures, the system exhibited a comparable dynamic behavior and stabilization time. This consistency confirms that the control strategy and the component sizing are adequate for handling a wide range of heat source conditions without inducing instability or significant oscillations, such as hunting phenomena often observed in poorly tuned thermal systems.

Fig. 4 presents the time-dependent profile of the electrical power generated by the scroll expander. A strong correlation is observed between the thermodynamic parameters (pressure and temperature) and the mechanical output. The power generation curve follows the same trend as the pressure difference across the expander. During the transient phase, the power output ramps up in correspondence with the establishment of the pressure ratio. Once the thermodynamic steady state is achieved (after 1,800 seconds), the electrical power output stabilizes at a constant value with negligible deviation. This synchronization confirms that the variations in power output are directly governed by the thermodynamic state of the working fluid.

Consequently, the high stability observed in both the thermodynamic properties and the power output validates the reliability of the experimental setup. Based on these observations, all performance analyses presented in the subsequent sections are derived exclusively from time-averaged data collected during these verified steady-state periods to ensure accuracy and reproducibility.

4.2 Performance analysis

4.2.1 Heat transfer characteristics and pinch point analysis

The thermal performance of the ORC system was evaluated using the pinch point temperature difference (PPTD), which directly measures thermal matching and irreversibility within the heat exchangers. In the evaporator pinch point temperature difference (PPTD_H) exhibited strong dependence on operating conditions.

While low heat source temperatures (50–60°C) yielded a relatively narrow $PPTD_H$ (6.84°C to 14.66°C), increasing the heat source to 90°C escalated the $PPTD_H$ significantly, reaching up to 48.21°C (at $T_C = 16^\circ\text{C}$). This substantial widening reflects a pronounced thermal mismatch and considerable exergy destruction. This behavior is explicitly attributed to system constraints; the operational evaporating pressure limited by the fixed capacity of the pneumatic feed pump could not scale proportionally with the available heat input, thereby restricting the effective utilization of thermal energy.

Furthermore, the system exhibited a strong thermal coupling effect, where the heat rejection process directly influenced evaporator performance. For instance, at $T_H = 90^\circ\text{C}$, reducing the cooling setpoint from 16°C to 8°C decreased the $PPTD_H$ from 48.21°C to 20.30°C. This physically indicates that higher condensing pressures impose a back-pressure penalty on the pump, altering mass flow dynamics and exacerbating the thermal mismatch. In contrast, the condenser demonstrated near ideal behavior, with the condenser pinch point temperature difference ($PPTD_C$) remaining within an exceptionally narrow range of -0.36°C to 0.68°C. The minute negative values fall entirely within the $\pm 1^\circ\text{C}$ measurement uncertainty of the sensors, confirming that the condenser operated at its maximum theoretical effectiveness. However, at the lowest heat sink condition ($T_C = 8^\circ\text{C}$), extreme heat rejection loads exceeded the chiller's capacity, which increased the actual cooling water temperature and kept the $PPTD_C$ slightly positive.

Overall, these findings explicitly establish that the evaporator acts as the primary source of thermal irreversibility, while the condenser operates near its thermodynamic limit. The severe thermal mismatch (wide $PPTD_H$) observed at elevated heat source temperatures directly restricts the achievable evaporating pressure. This constraint subsequently limits the pressure ratio across the expander, ultimately penalizing the system's overall power output. These results underscore the critical importance of proper thermal matching and auxiliary component capacities in defining the performance boundaries of small-scale ORC systems operating under severe off-design conditions.

4.2.2 Effect of heat source temperature on power output

The gross electrical power generated by the scroll expander serves as a definitive performance metric indicating the system's capability for low-grade waste heat recovery. Fig. 5 illustrates the variation in electrical power output as a function of the heat source temperature (T_H). The experimental results reveal a correlation between the evaporating temperature and the power output. Quantitatively, as the heat source temperature was elevated from 50°C to 90°C, the generated electrical power exhibited a substantial amplification, escalating from a baseline of 224.20 W to a peak of 488.55 W. This performance enhancement is thermodynamically attributed to the increased saturation pressure at the expander inlet, which creates a larger pressure differential across the machine. This elevated driving force results in a significantly greater specific enthalpy drop during the expansion process, thereby maximizing the shaft work converted into electricity.

From a thermodynamic perspective, this enhancement is attributed to two simultaneous mechanisms. First, increasing the evaporating temperature leads to a higher saturation pressure at the expander inlet. Since the condensing pressure remains relatively constant (governed by the cooling loop), the pressure differential across the expander widens. This increased driving force results in a larger specific enthalpy drop across the expansion process.

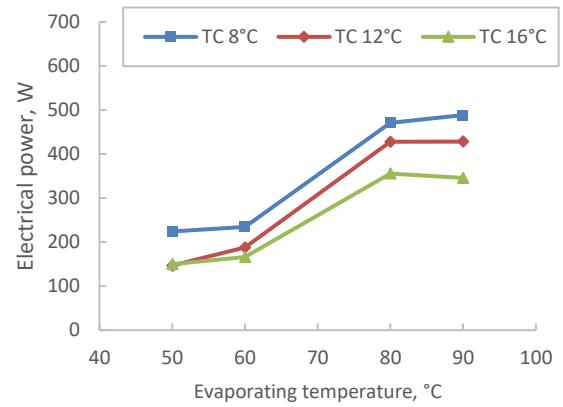


Fig. 5 Electrical power with difference evaporating and condensing temperature.

Second, the density of the vapor entering the expander increases with pressure, which typically contributes to a higher mass flow rate through the fixed-displacement volume of the scroll machine, assuming the volumetric efficiency remains relatively stable. Consequently, the combination of a larger enthalpy drops and increased mass throughput amplifies the total work output. Conversely, for a fixed evaporating temperature, an increase in condensing temperature reduces the pressure differential, thereby penalizing the power generation capability.

4.2.3 Interrelation between pressure ratio and power generation

To provide a deeper insight into the thermodynamic characteristics of the expansion process, the intrinsic relationship between the system operating pressure ratio and the resulting electrical power output is analyzed in Fig. 6. The experimental data reveals a direct correlation driven by the fluid's saturation properties: as the evaporating temperature is elevated from 50°C to 90°C, the inlet saturation pressure rises significantly while the condensing pressure remains relatively stable. Consequently, the operational pressure ratio of the system expands from 1.81 to 2.50. This observed range is particularly critical as it dictates the expansion regime (under- or over-expansion) and is fully consistent with characteristic findings reported in previous literature for similar scroll machines [13].

A critical observation from Fig. 6 is the non-linear dependency of electrical power on the pressure ratio when comparing different temperature regimes. Specifically, in the pressure ratio range of 2.0–2.2, the system shows distinct power output levels depending on the heat source temperature.

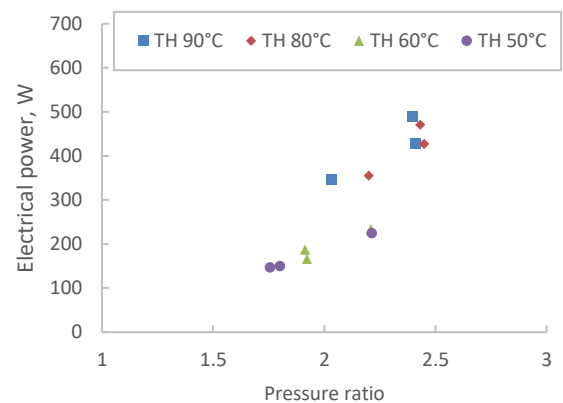


Fig. 6 Electrical power with pressure ratio of the ORC.

For instance, at high-evaporating temperatures (80°C and 90°C), the power output ranges between 345.59 W and 355.63 W. In contrast, at lower evaporating temperatures (50°C and 60°C) operating under the exact same pressure ratio, the power output is constrained to a much lower range of 224.20–234.24 W. This substantial discrepancy highlights that the pressure ratio, while a governing parameter, is not the sole determinant of performance in volumetric expanders. The fundamental driver for this phenomenon lies in the thermodynamic state of the working fluid at the suction port—specifically, its specific volume (density) and specific enthalpy. Since the scroll expander operates as a fixed-displacement machine, the mass flow rate processed is directly proportional to the suction vapor density. At higher evaporating temperatures, the vapor density increases significantly due to the elevated saturation pressure. Consequently, the expander ingests a greater mass of fluid per revolution. This increased mass throughput, combined with the higher internal energy possessed by the high-temperature vapor, enables significantly greater work extraction and power generation, even when the pressure ratios are comparable to those in low-temperature cases.

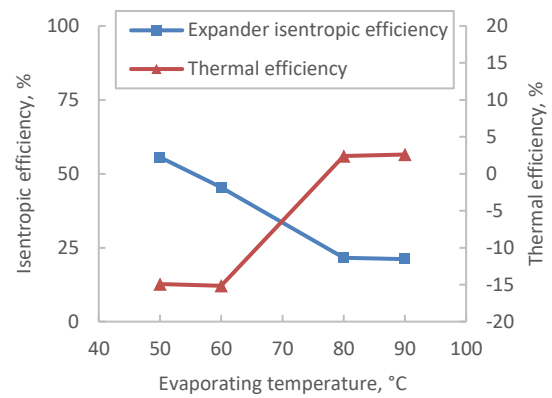
4.2.4 Isentropic and thermal efficiency characteristics

The thermodynamic quality of the expansion process and the overall system energy conversion performance were evaluated through the isentropic efficiency and thermal efficiency, as presented in Fig. 7.

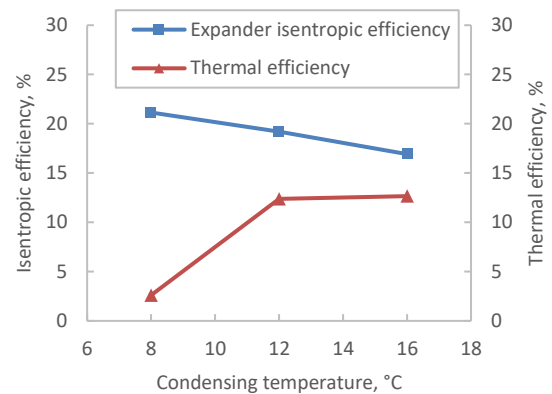
Fig. 7(a) depicts the evolution of the isentropic efficiency profile under varying operating conditions. A counter-intuitive but physically explicable trend is observed: the isentropic efficiency exhibits a declining trajectory as the evaporating temperature increases. This phenomenon, which aligns with the theoretical constraints highlighted by Couvreur et al. [30], can be primarily attributed to the inherent geometric limitation of the fixed built-in volume ratio of the scroll expander.

In a fixed-displacement machine, the optimal efficiency occurs only when the system's operating pressure ratio matches the device's built-in volume ratio. However, as the operating pressure ratio deviates further from this design point—driven by the elevated pressures associated with higher evaporating temperatures—the expansion process suffers from severe under-expansion losses. In this regime, the working fluid does not fully expand within the scroll pockets before the discharge port opens, leading to a sudden, irreversible throttle-like expansion at the exhaust. Furthermore, the higher operating pressures exacerbate internal leakage across the scroll tip seals and flank gaps. The increased pressure differential across these clearances drives a higher mass flow of refrigerant to bypass the expansion chambers without performing useful mechanical work, thereby degrading the isentropic efficiency. A similar degrading trend is observed with increasing condensing temperature in Fig. 7(b).

In the scope of this experimental investigation, the maximum isentropic efficiency recorded was 21.14%, achieved at the operating condition of $T_H = 90^\circ\text{C}$ and $T_C = 8^\circ\text{C}$. To contextualize this performance, it is crucial to benchmark these results against existing literature. The observed peak efficiency is notably lower than values reported for specialized or purpose-built scroll expanders. For instance, Wang et al. [14] achieved a remarkable isentropic efficiency of 77% using a scroll device equipped with a compliant mechanism specifically designed to minimize radial and axial leakage. Similarly, Lemort et al. [15] reported an efficiency of 68% for an open-drive scroll expander for ORC applications.



(a) With $T_C = 8^\circ\text{C}$



(b) With $T_H = 90^\circ\text{C}$

Fig. 7 Expander isentropic efficiency and thermal efficiency with difference temperatures, (a) Evaporating temperature; (b) Condensing temperature.

This significant discrepancy underscores the fundamental design differences between purpose-built expanders and the modified automotive compressor used in this investigation. Specialized expanders often feature tighter machining tolerances and floating seal designs that are optimized for low-speed expansion processes, features that are absent in mass-produced automotive compressors.

However, the results of this study align more closely with investigations utilizing similar low-cost, off-the-shelf compressor modifications. Quoilin et al. [17] reported efficiencies around 42% for a modified hermetic scroll, and Chang et al. [18] observed similar performance limitations in automotive-type expanders. It is worth noting that the efficiency obtained in this work (21.14%) is lower even compared to Quoilin et al., which can be attributed to two primary factors. First is the volume ratio mismatch: the operating pressure ratios in this study (1.81–2.50) likely deviated significantly from the built-in volume ratio of the specific automotive compressor model used, leading to substantial under-expansion or over-expansion losses. Second is the issue of internal leakage and friction. Automotive compressors are designed for high-speed operation (typically $>2,000$ RPM) where hydrodynamic oil sealing is effective. When operated as expanders at lower speeds or with different lubrication regimes in an ORC system, the sealing efficiency of the tip seals diminishes, and mechanical friction losses relative to the power output become more pronounced. Despite the lower efficiency, this study confirms that a standard automotive scroll compressor can be successfully converted to generate net positive power (up to 488 W), demonstrating a trade-off between high efficiency (which requires

expensive, custom hardware) and economic feasibility for small-scale applications.

Regarding the overall ORC system performance, the maximum thermal efficiency achieved was 2.61% at the optimal temperature condition ($T_H = 90^\circ\text{C}$). At this stage, the ORC system showed a BWR of 0.65, indicating that the pump consumed 65% of the gross generated power. As the heat source temperature decreased to 80°C , the BWR increased to 0.72, signaling a narrowing margin of net power generation.

However, a critical operational threshold was identified during the experimental campaign: specifically, when the evaporating temperature dropped below 80°C , the thermal efficiency became negative. This observation is fundamentally substantiated by the BWR analysis. Under these low-temperature conditions, the BWR surged significantly beyond unity, reaching 2.34 at $T_H = 60^\circ\text{C}$ and 1.87 at $T_H = 50^\circ\text{C}$. A BWR exceeding 1.0 dictates that the parasitic power consumed by the feed pump exceeded the gross electrical power generated by the expander.

Although the expander continued to sustain rotation and produce measurable electricity, the net energy balance of the system was thermodynamically negative. This finding underscores the significant challenge of utilizing ultra-low-grade heat sources ($<80^\circ\text{C}$) with this specific configuration, as the parasitic losses in the auxiliary components—particularly the pneumatic booster pump—become dominant relative to the meager power output generated by the low-efficiency expander.

To explicitly contextualize this limitation, the performance characteristics of this specific working fluid pump must be addressed. The double-acting pneumatic refrigerant booster pump (Model 2SLM06) utilized in this test bench operates with a high-pressure boost ratio of 6:1. Consequently, its specific pneumatic energy consumption is inherently massive, resulting in an exceptionally low overall efficiency estimated at 12–15%. Despite this significant parasitic energy penalty, this pneumatic configuration was deliberately selected. The double-acting mechanism ensures a continuous, strictly pulsation-free mass flow while entirely decoupling fluid pressure generation from actuation speed. These characteristics were absolutely necessary to maintain the highly stable boundary conditions required for the rigorous off-design mapping of the scroll expander. Therefore, the negative net energy balance observed at ultra-low temperatures is explicitly an artifact of this specific experimental hardware selection, rather than an inherent failure of the organic Rankine cycle itself.

4.2.5 Under- and over-expansion analysis

To provide a physical interpretation of the efficiency limitations observed in Fig. 7, it is essential to consider the geometric characteristics of the scroll expander. Unlike variable-geometry expansion devices (such as turbines with variable inlet guide vanes), the scroll expander utilized in this study operates with a fixed built-in volume ratio. This parameter is determined by the involute geometry of the scroll wraps and the ratio between the suction and discharge pocket volumes. This fixed geometry imposes a specific internal expansion profile that cannot adapt to varying operating conditions.

However, the experimental ORC system operated under varying off-design conditions driven by fluctuating heat source temperatures (50°C – 90°C). As previously shown in Fig. 6, the system pressure ratio varied between 1.81 and 2.50. Based on the measured inlet and outlet thermodynamic states, the corresponding effective

volume ratio was found to range from 1.73 to 2.28. To provide a rigorous physical interpretation of the efficiency limitations, these operational requirements must be correlated with the specific geometry of the selected expander. The geometric built-in volume ratio of the modified automotive scroll expander utilized in this study was exactly 1.5.

By comparing the demanded operational expansion against this inherent geometric constraint, a distinct mismatch regime becomes clearly evident. The results indicate that the lowest effective volume ratio (≈ 1.73), which occurred at the lower pressure ratios ($\text{PR} = 1.81$), provided the closest geometric matching to the expander's built-in capability, thereby relatively minimizing volumetric losses. Conversely, at higher operating pressure ratios ($\text{PR} = 2.50$), the demanded effective volume ratio surged to 2.28, representing a severe deviation from the optimal built-in expansion capacity. Because the demanded expansion ratio consistently exceeded the geometric limits of the scroll wraps across all test cases, the system was forced to operate in a non-ideal state explicitly defined by the mechanism of under-expansion, as described by Lemort et al. [15].

In particular, this predominance of under-expansion behavior occurs when the expansion process terminates prematurely due to the discharge port opening before the working fluid has fully expanded to the condensing pressure. As a result, the fluid remains at a pressure higher than the discharge line pressure at the moment of port opening, as noted by Wang et al. [14]. The subsequent rapid pressure equalization (blow-down) takes place at nearly constant volume, leading to a direct, irreversible loss of recoverable work and increased entropy generation.

Conversely, over-expansion would occur if the working fluid expands beyond the condenser pressure prior to discharge, resulting in a pressure lower than the outlet pressure and inducing backflow from the discharge line into the expansion chamber. This phenomenon creates a recompression process that consumes part of the generated work, thereby reducing the net power output, as reported by Quoilin et al. [17]. However, because the demanded effective volume ratio never dropped below the 1.5 geometric limit, such behavior is not evident in the present dataset.

These volumetric losses are further compounded by internal leakage effects. As highlighted by Mendoza et al. [31] and Olmedo et al. [32], off-design pressure distributions significantly alter the radial and axial forces acting on the scroll elements. In modified automotive compressors, where compliant sealing mechanisms are typically absent, these unbalanced forces can induce transient separation between the orbiting and fixed scrolls. This increases leakage pathways, allowing high-pressure vapor to bypass the expansion process and further degrading performance.

Overall, the mismatch between the operating pressure ratio and the inherent geometric characteristics of the scroll expander emerges as a dominant source of thermodynamic inefficiency. This is consistent with the findings of Dumont et al. [33] concluded that the volumetric mismatch as a primary contributor to exergy destruction in positive displacement expanders. Therefore, the relatively low peak efficiency (21.14%) reported in this study is physically consistent with the expected behavior of a fixed-geometry expander operating under persistent under-expansion conditions across a wide range of off-design scenarios.

4.2.6 Effect of heat sink temperature variations

Fig. 7(b) and Fig. 8 show that the thermal efficiency appeared to decrease when the condensing temperature was reduced. Theoretically, lowering the condensing temperature should increase the pressure ratio and cycle efficiency according to the Carnot principle. However, the experimental results showed a contrasting trend. This degradation in performance can be attributed to the combined effects of thermodynamic losses and system coupling:

Firstly, as established in Section 4.2.5, the scroll expander operates with a low fixed geometric built-in volume ratio. Reducing the condensing temperature significantly elevates the overall system pressure ratio. This forces the demanded effective volume ratio even further beyond the 1.5 geometric limit, thereby severely exacerbating the under-expansion losses as the highly pressurized fluid is throttled at the discharge port.

Secondly, a strong thermal coupling effect and capacity limitation were observed between the heat sink and heat source loops during high-gradient operation. Attempting to aggressively reduce the condensing temperature via the chiller induced a massive heat transfer rate that simultaneously exceeded the maximum capacities of both auxiliary thermal loops. On the heat sink side, the chiller became overloaded, causing the actual cooling water temperature to deviate upward from the 8°C setpoint. Concurrently on the heat source side, this massive heat extraction imposed an excessive thermal load on the finite-capacity electric heating loop, which struggled to maintain the targeted TH setpoint. Consequently, the actual specific enthalpy of the vapor at the expander inlet inadvertently dropped. This simultaneous degradation of boundary conditions at both the heat source and heatsink directly penalized the network output, effectively overriding any theoretical thermodynamic benefits expected from lowering the heat sink setpoint.

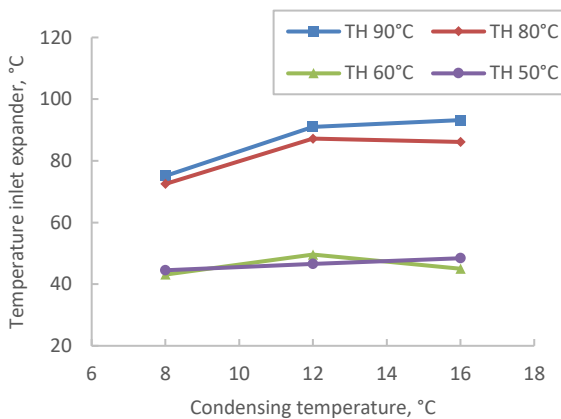


Fig. 8 Temperature inlet expander of the ORC with difference evaporating and condensing temperature.

4.3 Economic viability and practical implementation perspectives

While the thermodynamic analysis reveals that the maximum isentropic efficiency (21.14%) is lower than that of specifically built expanders (which can exceed 70%), the practical viability of the proposed system must be evaluated from a techno-economic perspective. High-efficiency expanders typically require precision machining and custom fabrication, leading to exorbitant capital costs that often render small-scale ORC projects (<10 kW) financially unfeasible.

In contrast, the scroll expander investigated in this study is a modified automotive compressor, which benefits from mass

production economies of scale. Quoilin et al. [34] demonstrated through a techno-economic survey that utilizing off-the-shelf HVAC components can reduce the specific cost of the expander unit by an order of magnitude compared to specialized prototypes. For low-grade waste heat recovery applications where the thermal energy input is essentially free (e.g., industrial exhaust or solar thermal), Lecompte et al. [35] argued that the primary economic driver is not the absolute thermal efficiency, but rather the minimization of the initial capital expenditure (CAPEX) to achieve a reasonable Payback Period (PB).

From a practical implementation perspective, the utilization of automotive components ensures global availability of spare parts and simplified maintenance procedures, addressing a major logistical challenge for ORC deployment in remote or developing regions. Clemente et al. [36] explicitly recommended the use of modified air-conditioning compressors for small-scale ORC systems in remote areas, citing their robustness and tolerance to two-phase flow as key advantages over sensitive micro-turbines. The results of this study demonstrate that a low-cost, modified scroll expander can successfully generate stable power output (up to 488.55 W) from low-temperature sources. Therefore, despite the modest efficiency, this configuration presents a promising pathway for sustainable power generation where initial capital expenditure is the main constraint. Future work focusing on simple, low-cost modifications to reduce leakage—such as optimizing oil circulation—could further enhance the cost-benefit ratio of this technology.

5. Conclusion

This study presented a comprehensive experimental investigation of an Organic Rankine Cycle (ORC) system utilizing R134a as the working fluid and integrated with a modified scroll expander. The system's thermodynamic performance was mapped under low-temperature heat source (50–90°C) and heat sink (8–16°C) conditions. The core findings are summarized as follows:

(1) Power Generation: The system achieved a maximum gross electrical power of 488.55 W at the peak thermal gradient ($T_H = 90^\circ\text{C}$, $T_C = 8^\circ\text{C}$). Despite the inherently lower efficiency of the modified scroll compressor, stable mechanical work can be extracted successfully if the thermal driving force is sufficient to overcome mechanical friction.

(2) Thermal Efficiency and Operational Limits: The maximum thermal efficiency was 2.61%. Crucially, a critical threshold was identified below an 80°C heat source, where the net system efficiency became negative. In such ultra-low-grade regimes, the parasitic pump work (resulting in a high Back-Work Ratio) completely outweighs the expander's output, highlighting the absolute necessity for highly efficient pumps in small-scale ORCs.

(3) Expander Characteristics: The expander's isentropic efficiency peaked at 21.14%. Efficiency degraded significantly at off-design pressure ratios due to under-expansion losses and internal leakage. This highlights an unavoidable trade-off in low-cost implementations: maximizing power output via high pressure introduces severe geometric limitations when utilizing a device with a fixed built-in volume ratio.

(4) Thermal Coupling: While steady-state operation was robust, a significant thermal coupling effect was observed. Aggressive cooling at the heat sink inadvertently destabilized the heat source temperature due to excessive heat extraction rates. This implies that practical applications necessitate dynamic control strategies to balance thermal loads and maintain a stable thermodynamic state at the expander inlet.

6. Further study

To enhance the commercial and technical viability of scroll-based ORC systems for low-grade waste heat recovery, future research efforts should be directed toward the following strategic areas:

(1) Geometry optimization: While completely redesigning the scroll wrap is costly, developing scroll geometries specifically tailored for expansion processes is crucial to minimize internal leakage and mechanical friction, which are prevalent in modified compressors.

(2) Working fluid investigation: Exploring zeotropic mixtures could offer superior temperature glide matching with the heat source and sink. This would reduce the irreversibility in the heat exchangers, potentially improving the overall cycle thermal efficiency.

(3) Advanced control strategies: Implementing adaptive control algorithms is necessary to optimize the net power output by balancing pump consumption and expander performance under fluctuating operating conditions. Specifically, optimizing the pump speed to maintain the system pressure ratio near the expander's built-in volume ratio could significantly reduce off-design losses.

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Author Contributions:

Suppachai Chumnumwat: Conceptualization, Methodology, Investigation, Formal analysis, Writing – original draft. **Sorawit Somsaree:** Supervision, Writing – review & editing. **Chontida Thongdonsa and Supansa Plangklang:** Investigation. **Kwanchai Kraitung:** Supervision, Resources, Writing – review & editing.

Declaration of Interest Statement

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

References

- [1] Daniarta, S., Sowa, D., Błasiak, P., Imre, A. R. and Kolasinski, P., Techno-economic survey of enhancing Power-to-Methane efficiency via waste heat recovery from electrolysis and biomethanation. *Renewable and Sustainable Energy Reviews*. 194 (2024) 114301, doi: <https://doi.org/10.1016/j.rser.2024.114301>.
- [2] Fagioli, F., Manieri, M., Agostini, G., Salvestroni, M., Taddei, F., Cottone, F. and Lucia, M. D., Experimental Characterization of Commercial Scroll Expander for Micro-Scale Solar Organic Rankine Cycle Application: Part 2. *Energies*. 18, 2875 (2025), 2875, doi: <https://doi.org/10.3390/en18112875>.
- [3] Forman, C., Muritala, I. K., Pardemann, R. and Meyer, B., Estimating the global waste heat potential. *Renewable and Sustainable Energy Reviews*. 57 (2016) 1568-1579, doi: <https://doi.org/10.1016/j.rser.2015.12.192>.
- [4] Lecompte, S., Huisseune, H., van den Broek, M. and De Paepe, M., Methodical thermodynamic analysis and regression models of organic Rankine cycle architectures for waste heat recovery. *Energy*. 87 (2015) 60-76, doi: <https://doi.org/10.1016/j.energy.2015.04.094>.
- [5] Löffler, M., Batch Processes in Heat Engines. *Energy*. 125 (2017) 788-794, doi: <https://doi.org/10.1016/j.energy.2017.02.105>.
- [6] De Lucia, M., Pierucci, G., Manieri, M., Agostini, G., Giusti, E., Salvestroni, M., Taddei, F., Cottone, F. and Fagioli, F., Experimental Characterization of Commercial Scroll Expander for Micro-Scale Solar ORC Application: Part 1. *Energies*. 17, 2205 (2024), 2205, doi: <https://doi.org/10.3390/en17092205>.
- [7] Gu, Z., Shi, Y. and Wu, P., Experimental Study on the Organic Rankine Cycle for the Recovery of the Periodic Waste Heat Source. *Energies*. 18, 1458 (2025), 1458, doi: <https://doi.org/10.3390/en18061458>.
- [8] Twomey, B., Jacobs, P. A. and Gurgenci, H., Dynamic performance estimation of small-scale solar cogeneration with an organic Rankine cycle using a scroll expander. *Applied Thermal Engineering*. 51 (2013) 1307-1316, doi: <https://doi.org/10.1016/j.applthermaleng.2012.06.054>.
- [9] Yang, S.-C., Hung, T.-C., Feng, Y.-Q., Wu, C.-J., Wong, K.-W. and Huang, K.-C., Experimental investigation on a 3kW organic Rankine cycle for low-grade waste heat under different operation parameters. *Applied Thermal Engineering*. 113 (2017) 756-764, doi: <https://doi.org/10.1016/j.applthermaleng.2016.11.032>.
- [10] Wieland, C., Schiffelechner, C., Dawo, F. and Astolfi, M., The organic Rankine cycle power systems market: Recent developments and future perspectives. *Applied Thermal Engineering*. 224 (2023) 119980, doi: <https://doi.org/10.1016/j.applthermaleng.2023.119980>.
- [11] Jiménez-García, J. C., Ruiz, A., Pacheco-Reyes, A. and Rivera, W., A Comprehensive Review of Organic Rankine Cycles. *Processes*. 11, 1982 (2023), 1982, doi: <https://doi.org/10.3390/pr11071982>.
- [12] Hijriawan, M., Pambudi, N. A., Wijayanto, D. S., Biddinika, M. K. and Saw, L. H., Experimental analysis of R134a working fluid on Organic Rankine Cycle (ORC) systems with scroll-expander. *Engineering Science and Technology, an International Journal*. 29 (2022) 101036, doi: <https://doi.org/10.1016/j.jestch.2021.06.016>.
- [13] Zhang, H.-H., Xi, H., He, Y.-L., Zhang, Y.-W. and Ning, B., Experimental study of the organic rankine cycle under different heat and cooling conditions. *Energy*. 180 (2019) 678-688, doi: <https://doi.org/10.1016/j.energy.2019.05.072>.
- [14] Wang, H., Peterson, R. B. and Herron, T., Experimental performance of a compliant scroll expander for an organic Rankine cycle. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*. 223 (2009) 863-872, doi: <https://doi.org/10.1243/09576509JPE741>.
- [15] Lemort, V., Quoilin, S., Cuevas, C. and Lebrun, J., Testing and modeling a scroll expander integrated into an Organic Rankine Cycle. *Applied Thermal Engineering*. 29 (2009) 3094-3102, doi: <https://doi.org/10.1016/j.applthermaleng.2009.04.013>.
- [16] Miao, Z., Xu, J. and Zhang, K., Experimental and modeling investigation of an organic Rankine cycle system based on the scroll expander. *Energy*. 134 (2017) 35-49, doi: <https://doi.org/10.1016/j.energy.2017.06.001>.
- [17] Quoilin, S., Lemort, V. and Lebrun, J., Experimental study and modeling of an Organic Rankine Cycle using scroll expander. *Applied Energy*. 87 (2010) 1260-1268, doi: <https://doi.org/10.1016/j.apenergy.2009.06.026>.
- [18] Chang, J.-C., Hung, T.-C., He, Y.-L. and Zhang, W., Experimental study on low-temperature organic Rankine cycle utilizing scroll type expander. *Applied Energy*. 155

- (2015) 150-159, doi: <https://doi.org/10.1016/j.apenergy.2015.05.118>.
- [19] Campana, C., Cioccolanti, L., Renzi, M. and Caresana, F., Experimental analysis of a small-scale scroll expander for low-temperature waste heat recovery in Organic Rankine Cycle. *Energy*. 187 (2019) 115929, doi: <https://doi.org/10.1016/j.energy.2019.115929>.
- [20] Gao, P., Jiang, L., Wang, L. W., Wang, R. Z. and Song, F. P., Simulation and experiments on an ORC system with different scroll expanders based on energy and exergy analysis. *Applied Thermal Engineering*. 75 (2015) 880-888, doi: <https://doi.org/10.1016/j.applthermaleng.2014.10.044>.
- [21] Kosmadakis, G., Mousmoulis, G., Manolakis, D., Anagnostopoulos, I., Papadakis, G. and Papantonis, D., Development of Open-Drive Scroll Expander for an Organic Rankine Cycle (ORC) Engine and First Test Results. *Energy Procedia*. 129 (2017) 371-378, doi: <https://doi.org/10.1016/j.egypro.2017.09.236>.
- [22] Yu, X., Huang, Y., Li, Z., Huang, R., Chang, J. and Wang, L., Characterization analysis of dynamic behavior of basic ORC under fluctuating heat source. *Applied Thermal Engineering*. 189 (2021) 116695, doi: <https://doi.org/10.1016/j.applthermaleng.2021.116695>.
- [23] Li, Y.-M., Hung, T.-C., Wu, C.-J., Su, T.-Y., Xi, H. and Wang, C.-C., Experimental investigation of 3-kW organic Rankine cycle (ORC) system subject to heat source conditions: A new appraisal for assessment. *Energy*. 217 (2021) 119342, doi: <https://doi.org/10.1016/j.energy.2020.119342>.
- [24] Liu, Z., Xie, D., Li, Z., Huang, T. and Liu, Z., Effect of tooth head modification on the performance of a scroll expander for ORC waste heat recovery system. *Frontiers in Energy Research*. 10 (2023) doi: <https://doi.org/10.3389/fenrg.2022.1088719>.
- [25] Moffat, R. J., Describing the uncertainties in experimental results. *Experimental Thermal and Fluid Science*. 1 (1988) 3-17, doi: [https://doi.org/10.1016/0894-1777\(88\)90043-X](https://doi.org/10.1016/0894-1777(88)90043-X).
- [26] Hung, T. C., Wang, S. K., Kuo, C. H., Pei, B. S. and Tsai, K. F., A study of organic working fluids on system efficiency of an ORC using low-grade energy sources. *Energy*. 35 (2010) 1403-1411, doi: <https://doi.org/10.1016/j.energy.2009.11.025>.
- [27] Bamorovat Abadi, G., Yun, E. and Kim, K. C., Experimental study of a 1 kw organic Rankine cycle with a zeotropic mixture of R245fa/R134a. *Energy*. 93 (2015) 2363-2373, doi: <https://doi.org/10.1016/j.energy.2015.10.092>.
- [28] Heberle, F. and Brüggemann, D., Exergy based fluid selection for a geothermal Organic Rankine Cycle for combined heat and power generation. *Applied Thermal Engineering*. 30 (2010) 1326-1332, doi: <https://doi.org/10.1016/j.applthermaleng.2010.02.012>.
- [29] ASHRAE. *Designation and Safety Classification of Refrigerants*, <https://www.ashrae.org/file%20library/technical%20resources/standards%20and%20guidelines/standards%20addenda/34_2024_k_20251031.pdf> (2025).
- [30] Couvreur, K., Tassenoy, R., van Heule, X., De Paepe, M. and Lecompte, S., Experimental and numerical analysis of variable volume ratio as additional optimization parameter in organic Rankine cycle expanders. *Applied Thermal Engineering*. 216 (2022) 119007, doi: <https://doi.org/10.1016/j.applthermaleng.2022.119007>.
- [31] Mendoza, L. C., Navarro-Esbrí, J., Bruno, J. C., Lemort, V. and Coronas, A., Characterization and modeling of a scroll expander with air and ammonia as working fluid. *Applied Thermal Engineering*. 70 (2014) 630-640, doi: <https://doi.org/10.1016/j.applthermaleng.2014.05.069>.
- [32] Olmedo, L. E., Mounier, V., Mendoza, L. C. and Schiffmann, J., Dimensionless correlations and performance maps of scroll expanders for micro-scale Organic Rankine Cycles. *Energy*. 156 (2018) 520-533, doi: <https://doi.org/10.1016/j.energy.2018.05.001>.
- [33] Dumont, O., Parthoens, A., Dickes, R. and Lemort, V., Experimental investigation and optimal performance assessment of four volumetric expanders (scroll, screw, piston and roots) tested in a small-scale organic Rankine cycle system. *Energy*. 165 (2018) 1119-1127, doi: <https://doi.org/10.1016/j.energy.2018.06.182>.
- [34] Quoilin, S., Broek, M. V. D., Declaye, S., Dewallef, P. and Lemort, V., Techno-economic survey of Organic Rankine Cycle (ORC) systems. *Renewable and Sustainable Energy Reviews*. 22 (2013) 168-186, doi: <https://doi.org/10.1016/j.rser.2013.01.028>.
- [35] Lecompte, S., Huisseune, H., van den Broek, M., Vanslambrouck, B. and De Paepe, M., Review of organic Rankine cycle (ORC) architectures for waste heat recovery. *Renewable and Sustainable Energy Reviews*. 47 (2015) 448-461, doi: <https://doi.org/10.1016/j.rser.2015.03.089>.
- [36] Clemente, S., Micheli, D., Reini, M. and Tacconi, R., Energy efficiency analysis of Organic Rankine Cycles with scroll expanders for cogenerative applications. *Applied Energy*. 97 (2012) 792-801, doi: <https://doi.org/10.1016/j.apenergy.2012.01.029>.