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Experimental investigations on the performance of a thermo-mechanical refrigeration system utilizing ultra-low temperature waste heat sources



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Abstract Experimental studies that investigate the utilization of ultra-low temperature (between 48 °C and 120 °C) waste heat, which forms 25 % of the global unrecovered waste heat, to drive thermal cooling systems are rare. Thus, this study fills a significant gap in the literature by (i) experimentally evaluating the performance of a new full-scale thermo-mechanical refrigeration (TMR) system at ultra-low temperatures ranging from 50 °C to 85 °C, (ii) characterizing its operation with different commercial refrigerants, and (iii) identifying optimal operating conditions and working fluids for the investigated TMR system. An organic Rankine cycle (ORC), vapor compression cycle (VCC), and expander-compressor unit (ECU) make up the TMR system. To ensure its efficient operation, flexibility, and reliability, a full-scale ECU-based TMR system with a design capacity of 1 kW at a heat source temperature of 85 °C is built and put through a series of tests. Furthermore, the TMR system is tested with different commercial refrigerants (R134a, R410A, R407C) over a wide range of operating conditions of the power loop. The results reveal that the ECU-based TMR system can work with ultra-low temperatures of 65 °C with an energy efficiency of 5.92 % and COP of 2.36, for the ORC and VCC, respectively. At a heat source temperature of 85 °C, a condenser water temperature of 15 °C, and an evaporator water temperature of 33 °C, the energy efficiency of the ORC and COP of the VCC are increased to 9.85 % and 3.99, respectively. For the cooling quality, the TMR system shows a minimum evaporator temperature of −10 °C using R134a in both the power and cooling loop, which is improved to lower than −20 °C by using R407C in the cooling loop. The results presented herein will be beneficial to the development, design, and optimization of refrigeration and power systems that utilize low-temperature waste heat.

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1. Introduction

Waste heat is produced in different industrial processes and is classified into five categories based on their temperature

Nomenclature

Symbol	Description
A_{ex}, A_{Comp}	Cross-sectional area of the expander and compressor pistons, respectively, (m ²)
$A_{h,amb}$	Heat transfer area between the heater and ambient air, (m ²)
$A_{co,amb}$	Heat transfer area between the cooler and ambient air, (m ²)
$c_{p,h,avg}$	Average specific heat of the hot water passes through the heater, (kJ/kg-°C)
$c_{p,co,avg}$	Average specific heat of the cold water passes through the cooler, (kJ/kg-°C)
h_1, h_2, \dots	Specific enthalpy of the working fluid at states 1, 2, ..., (kJ/kg)
L	Length of the expander's piston stroke, (m)
\dot{m}_{CLR}	Mass flow rate of the cooling loop refrigerant, (kg/s)
$\dot{m}_{co,in}$	Mass flow rate of the cold water passes through the cooler, (kg/s)
$\dot{m}_{h,in}$	Mass flow rate of the hot water passes through the heater, (kg/s)
\dot{m}_{PLR}	Mass flow rate of the power loop refrigerant, (kg/s)
N	Frequency of the expander-compressor unit, (Hz)
P_1, P_2, \dots	Pressure of the working fluid at state 1, 2, ..., (kJ/kg)
P_{NPSL}	Percentage of the pneumatic pump stroke length, (%)
\dot{Q}_{co}	Heat transfer between the refrigerant and cold water through the cooler, (kW)
$\dot{Q}_{co,amb}$	Heat transfer between the cold water of the cooler and ambient air, (kW)
$\dot{Q}_{co,total}$	Total cooling load of the cooler, (kW)
\dot{Q}_{ev}	Cooling load of the evaporator, (kW)
\dot{Q}_{heater}	Heat transfer between the hot water of the heater and the refrigerant, (kW)
$\dot{Q}_{h,amb}$	Heat transfer between the hot water of the heater and ambient air, (kW)
$\dot{Q}_{h,total}$	Total heating load of the heater, (kW)
T_{amb}	Ambient air temperature, (°C)
$T_{co,in}, T_{co,out}$	Inlet and outlet temperatures of the cooler cold water, respectively, (°C)
$T_{h,in}, T_{h,out}$	Inlet and outlet temperatures of the heater hot water, respectively, (°C)
T_{ev}	Evaporation temperature of the refrigerant at the inlet of the evaporator, (°C)
\dot{W}_{Exp}	Work rate of the expander, (kW)
\dot{W}_p	Work rate of the pneumatic pump, (kW)
η_{pl}	Energy efficiency of the power loop, (%)
Abbreviations	
CLR	Cooling loop refrigerant
COP	Coefficient of performance
ECU	Expander-compressor unit
HTC	Heat transfer coefficient
ORC	Organic Rankine cycle
PLR	Power loop refrigerant
TMR	Thermo-mechanical refrigeration
VCC	Vapor compression cycle

ranges, which are ultra-low temperature (less than 120 °C), low-temperature (120–230 °C), medium-temperature (230–650 °C), high-temperature (650–870 °C), and ultra-high temperature (greater than 870 °C) [1,2]. A recent report from the US department of energy [1] indicates that waste heat at low and ultra-low temperatures accounts for more than 51 % of unrecovered industrial waste heat. In particular, for a temperature range of 48 °C to 120 °C, the ultra-low temperature waste heat forms about 25 % of the total unrecovered waste heat. Thus, the recovery of the wasted heat in this temperature range is a promising step to increase energy efficiency, reduce greenhouse gas emissions, and increase profits of the industrial systems. Therefore, several technologies were developed to recover low-grade waste heat (low and ultra-low temperature) by utilizing it to preheat another process, generate electricity, or drive thermal cooling systems [3]. The choice of thermal cooling system to recover the low-grade waste heat is more feasible than other applications due to the following reasons: (1) the low efficiency of the heat-to-power systems at low-grade temperatures, which negatively affects their economic feasibility [4], (2) the high energy demand for air-conditioning and refrigeration systems, which exceeds 10 % of the worldwide energy consumption and anticipated to treble by 2050 [5], and (3) the urgent need to mitigate the global warming and ozone depletion issues which are partially caused by the electric-based cooling systems (where electricity obtained from fossil-fuel-based power plants) [6,7].

Many thermal-driven cooling systems enjoy the ability to work with low-grade waste heat including sorption-based [8] and thermal-mechanical-based [9] systems. Sorption-based systems are designed as closed-type cycles (absorption and adsorption cycles), or open-type cycles (solid and liquid desiccant systems) [10,11]. In general, sorption-based systems have high cooling capacities at heat source temperatures between 90 °C and 220 °C [12]. However, these systems suffer from limited evaporation temperatures, higher costs, and bulky size compared to other thermal-driven cooling systems [13]. The conventional thermal-mechanical-based systems include organic Rankine cycle-based (ORC) and ejector-based systems. The current status, applications, and advancements of these systems were reviewed by Sleiti et al. in [14]. Despite the low coefficient of performance (COP) of the ejector-based systems (COP between 0.1 and 0.62), they suffer from the instability raised by the inflexibility of the ejector operation. On the other hand, the ORC-based systems are a mature technology and show the most economic viability for the recovery of low-grade waste heat as reported in [1]. However, these systems lose their economic feasibility and operability at temperatures less than 100 °C [15], due to the poor performance of their conventional expanders (scroll, twin-screw, and rotary-vane expanders [16]).

Numerous studies have been conducted to improve the performance of ORC-based systems [17,18]. Some of them are specifically focusing on the selection of suitable refrigerants

as working fluids [19,20]. For example, Andreasen et al. [21] investigated the effectiveness of using pure and mixed refrigerants in an ORC system operating at two different heat source temperatures, 90 °C and 120 °C. Through their research, they found that an optimized mixture of ethane/propane resulted in a net power increase of 11.1 % at 90 °C and 12.9 % at 120 °C. Similarly, Le et al. [22] explored the use of various pure refrigerants as working fluids in an ORC system with a heat source temperature of 150 °C. Their study revealed that R152a, when used in a supercritical regenerative ORC scheme with a heater pressure above 47 bar, achieved a maximum energy efficiency of 13.1 %. However, a few experimental analyses were conducted on these systems, especially at ultra-low temperature heat sources. Over the heat source temperature range of 80 °C to 100 °C, a pumpless ORC with R1233zd(E) is examined experimentally by Lu et al. [23]. They reported an energy efficiency of 3.5 % at a water temperature (heat source) of 95 °C. Another experimental work on ORC over heat source temperature of 40 °C to 80 °C using HCFO-1233zd (E) is conducted by Araya et al. [24]. They reported a maximum energy efficiency of 5 % at a temperature of 85.7 °C. An integrated ORC-sorption system operating at a temperature range of 75 °C to 95 °C using R245fa is tested by Jiang et al. [25]. A COP of 1.61 to 1.90 is reported with an evaporation temperature of 10 °C. A similar system was tested by Chaayat et al. [26] but at a temperature of 105 °C. They reported an average COP of 0.56 at an evaporation temperature of 8 °C. At a generator temperature of 88 °C to 126 °C, an ejector-based refrigeration system was experimentally investigated by Huang et al. [27] using R32 as a working fluid. Although the generator temperatures are adequately high, the COP does not exceed 0.60, which emphasizes the need for more efficient cooling systems at ultra-low temperatures. Lower values for the COP (less than 0.40) are reported for the ejector cooling systems if operated with generator temperatures below 60 °C as examined by Gagan et al. [28]. Instead of examining the system's overall performance, other experimental studies have focused on the functionality and advancements of specific ejector and ORC cooling system parts (such as the ORC scroll expander [29], ejector-expansion freezer [30], an adjustable double-slider ejector [31], and ejector geometry [32]). While these studies aim to address the technical challenges of these systems, their performance is still limited, with a coefficient of performance (COP) of less than 2, and they are primarily suitable for air-conditioning applications rather than refrigeration.

Therefore, to overcome the aforementioned drawbacks of the ORC and ejector cooling systems, Sleiti et al. [33] introduced a new thermo-mechanical refrigeration (TMR) system working at low-grade heat sources. This TMR system integrates an ORC (power loop) with a vapor compression cycle (VCC, cooling loop) using an isobaric expander-compressor unit (ECU). The ECU is developed based on the work of Encontech BV [34] and consists of two rigid-connected pistons that slide inside expander and compressor cylinders. The detailed thermodynamic characteristics of the new ECU-based TMR system were investigated by Sleiti et al. in [33]. Also, the performance of this system with several refrigerants was presented in [35]. Furthermore, an experimental study was conducted by Sleiti et al. [36] to analyze the performance of the VCC within the developed pressures by the ECU using a small-size air compressor to drive the ECU. The studies demonstrate that the developed TMR system can operate

within the ultra-temperature range of up to 60 °C using various refrigerants, achieving a power efficiency of 4 % to 11 % and a coefficient of performance (COP) ranging from 1.5 to over 2.7. Based on the findings of these studies, the ECU-based TMR system is a promising technology for recovering ultra-low waste heat in various industrial processes. However, there are significant gaps in our understanding of its operability over a wide range of heat source temperatures and with mixed refrigerants in the power and cooling loops. To address these gaps, this study presents a comprehensive experimental evaluation of a full-scale ECU-based TMR system with a cooling capacity of 1 kW. The objectives of the study are as follows:

- To experimentally evaluate the performance of a full-scale ECU-based TMR system, including its efficiency and effectiveness in heat recovery.
- To characterize the performance of the ECU-based TMR system over a wide range of ultra-low temperatures, from 50 °C to 85 °C, to identify its operating limits and potential challenges.
- To experimentally evaluate the system's operation with (i) different commercial refrigerants and (ii) varying temperatures and flow rates of the heat source and sink to determine the optimal operating conditions and working fluids for the system.
- To identify the best practices for the ECU-based TMR system's design and operation to maximize its performance and efficiency in recovering ultra-low waste heat.

The rest of the manuscript is organized as follows: [Section 2](#) presents a detailed description of the experimental setup and the technical specifications of its components and measurement devices. Then, the data reduction analysis, testing procedures, and validation are introduced in [Section 3](#). [Section 4](#) discusses the results of the experimental analysis in terms of the performance indicators and major operating variables. Also, a comparison between the present TMR system and other thermal-driven systems is introduced in [Section 4](#). Finally, [Section 5](#) provides a summary of the main findings and conclusions of this study.

2. Experimental setup

[Fig. 1](#) shows the arrangement of the experimental setup. It is made up of a cooling loop (a vapor compression cycle (VCC)) and a power loop (an organic Rankine cycle (ORC)) that are integrated via the ECU. A pneumatic pump, heater, expander cylinder, and cooler make up the bulk of the power loop. Additionally, an evaporator, compressor cylinder, condenser, and expansion valve make up the majority of the cooling loop.

In the power loop, the pneumatic pump is equipped with a stroke length adjuster to control the pressure ratio through the compression of the working fluid (power loop refrigerant (PLR)) from the low-pressure side (state 1) to the high-pressure side (state 2). The heater is a heat exchanger that is used to absorb heat from external hot water to evaporate the PLR to reach saturated vapor or superheated vapor phase at state 3. The external hot water is heated using an electric heater that is equipped with a temperature controller and control

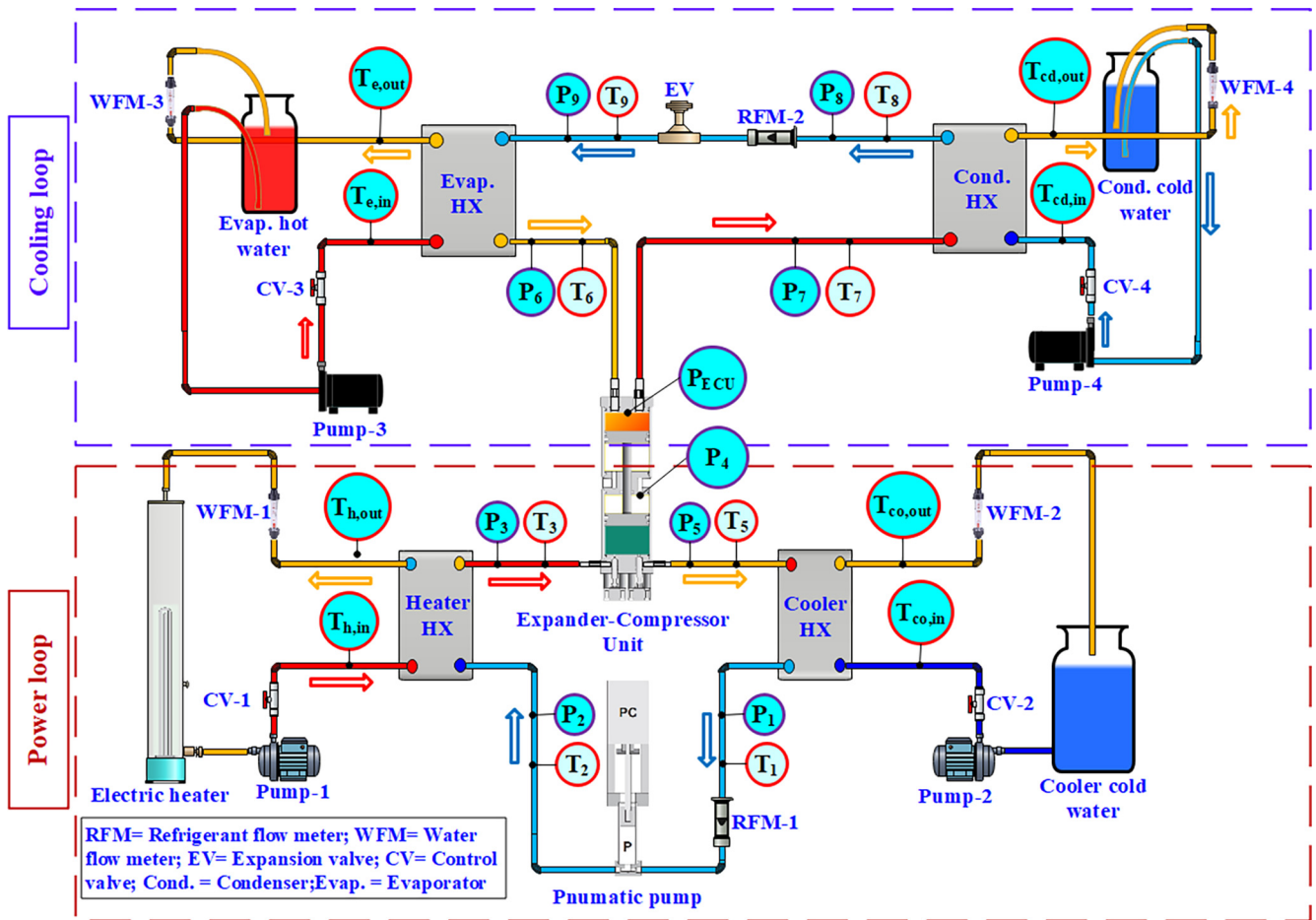


Fig. 1 Schematic diagram of the experimental layout of the TMR system.

valve to heat a specified water flow to the desired temperature. After the heating process, the heated PLR is expanded to the low-pressure side (state 5) by pushing the expander piston to compress the cooling loop refrigerant. The expander of the power loop is part of the expander-compressor unit (ECU), which is described in detail in the [Supplementary material \(SM. A\)](#). Then, the PLR is cooled to the saturated liquid phase (state 1) through the cooler by rejecting heat to the cold water of the cooler, which is circulated using pump 2. The cooler cold water temperature is controlled using ice cubes to maintain it at the desired value.

In the cooling loop, the cooling loop refrigerant (CLR) enters the compressor cylinder at low pressure (state 6) to be compressed to high pressure at state 7 using the expansion process of the PLR. Then, the pressurized CLR is cooled down to the saturated liquid phase (state 8) by rejecting the heat into the circulated cold water (using pump 4) through the condenser. After that, the CLR is expanded through an expansion valve to the low pressure at state 9 to perform the evaporation process by absorbing heat from the circulated hot water (using pump 3). A mini-dry block heater is used to control the temperature of the evaporator hot water.

In addition, the flow of the circulated cold and hot water in both loops is controlled using a control valve for each circulation and a water flow meter (WFM) is used to measure its volumetric flow rate. Furthermore, as shown in Fig. 1, both

power and cooling loops are fully instrumented by installing pressure and temperature sensors at the inlet and outlet of each component. Also, the flow rates of the PLR and CLR are measured using refrigerant volumetric flow meters RFM-1 and RFM-2, respectively.

A top view of the experimental setup is presented in Fig. 2 (a). The circulating pumps of the power loop have a flow capacity of 600 – 1800 L/h and are connected with water volumetric flow meters (300–3000 L/h Plastic Tube Water rotameter, model: LZS-25). Similar pumps and volumetric flow meters are used for the circulated water in the cooling loop with a capacity of 100 – 270 L/h for each pump and 100 – 1000 L/h for each rotameter (model: LZS-15). For the refrigerant volumetric flow, calibrated LZJ-10F Glass tube flow meters are used. The hot water of the heater is heated by a 12 kW 3-phase electric heater, which is installed inside a PVC pipe with an inner diameter of 101.6 mm and length of 1.60 m (Fig. 2(b)). The PVC pipe is partially filled with water to be heated to the desired temperature using a thermostat controller and the hot water is circulated using Pump-1. A plate heat exchanger (Model: B25THx30/1P-SC-M 2x22U + 2 x1" & 22U) is used for the heater and another heat exchanger (Model: B25THx40/1P-SC-M 2x22U + 2 x1" & 22U) is used for the cooler. These heat exchangers are arranged together with the pneumatic pump and the ECU as shown in Fig. 2(d). Also, as shown in Fig. 2(c), two identical

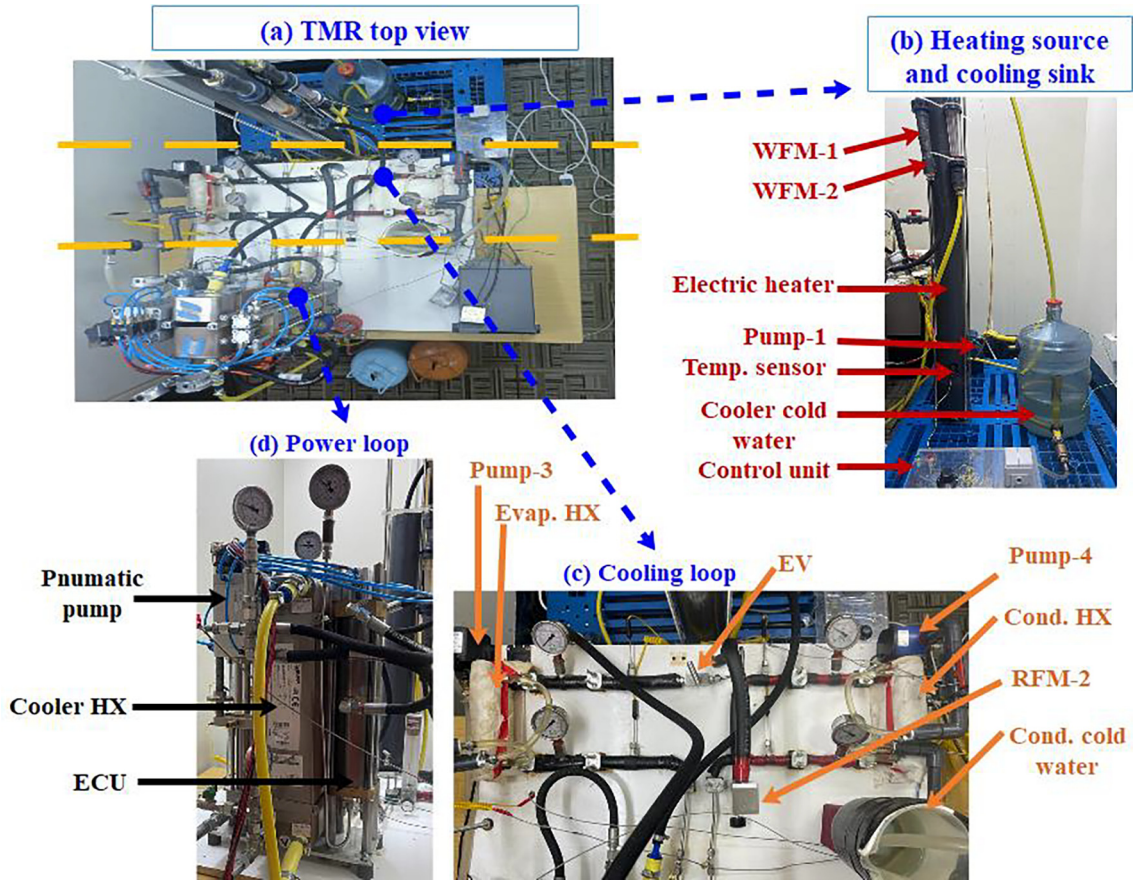


Fig. 2 Experimental setup.

plate heat exchangers are used for the evaporator and condenser of the cooling loop (Model: B3-014-12D-3.0). These heat exchangers are selected due to their compactness, high performance, and compatibility with several refrigerants including R134a, R407C, and R410 that are used in this study. Finally, for the expansion process of the CLR, a needle-based expansion valve is used (Model: WINFLOW 1/4" NPT [F], stainless steel). The technical details of the measuring instruments are provided in the [Supplementary material](#) (SM. B, Table B.1).

3. Methodology

This section outlines the methods for the data reduction of the experimental setup, as well as the testing procedures, validation, and uncertainty analysis.

3.1. Data reduction

The energetic performance of the investigated TMR system is analyzed based on the collected experimental data in this work. These data are used to obtain the thermodynamic properties of refrigerants using the real fluid properties available in the library of Engineering Equation Solver (EES) software. Also, EES is used for the performance evaluation of the tested TMR system of this study. The performance of the TMR sys-

tem is mainly evaluated based on the energy efficiency of the power loop (η_{pl}), and the COP of the cooling loop. Other performance indicators could be used to assess the system such as the expander work (\dot{W}_{Exp}) and the cooling capacity of the evaporator (\dot{Q}_{ev}). Thus, a thermodynamic model for the TMR system is developed as shown below with the assumptions of (1) steady-state analysis, (2) the variation in the kinetic and potential energies are neglected, and (3) the pressure drops through connecting pipes are neglected.

Referring to the T-s diagram of the TMR system shown in [Fig. 3](#), the power of the pneumatic pump is given as [\[37\]](#):

$$\dot{W}_p = \dot{m}_{PLR}(h_2 - h_1) \quad (1)$$

where \dot{m}_{PLR} , h_2 , and h_1 are the flow rate of the power loop refrigerant, and specific enthalpies of the refrigerant at state 2 and state 1, respectively. The heat absorbed by the PLR from the hot water as it passes through the heater (\dot{Q}_{heater}) is given as [\[37\]](#):

$$\dot{Q}_{heater} = \dot{m}_{PLR}(h_3 - h_2) \quad (2)$$

Part of the heater's hot water energy is lost to the surrounding air, thus, it must be calculated to obtain the total amount of heat provided by the electric heater. Thus, the heat loss from the hot water to the ambient through the heater is given as [\[38\]](#):

$$\dot{Q}_{h,amb} = U_{h,amb} \times A_{h,amb} \times (T_{h,avg} - T_{amb}) \quad (3)$$

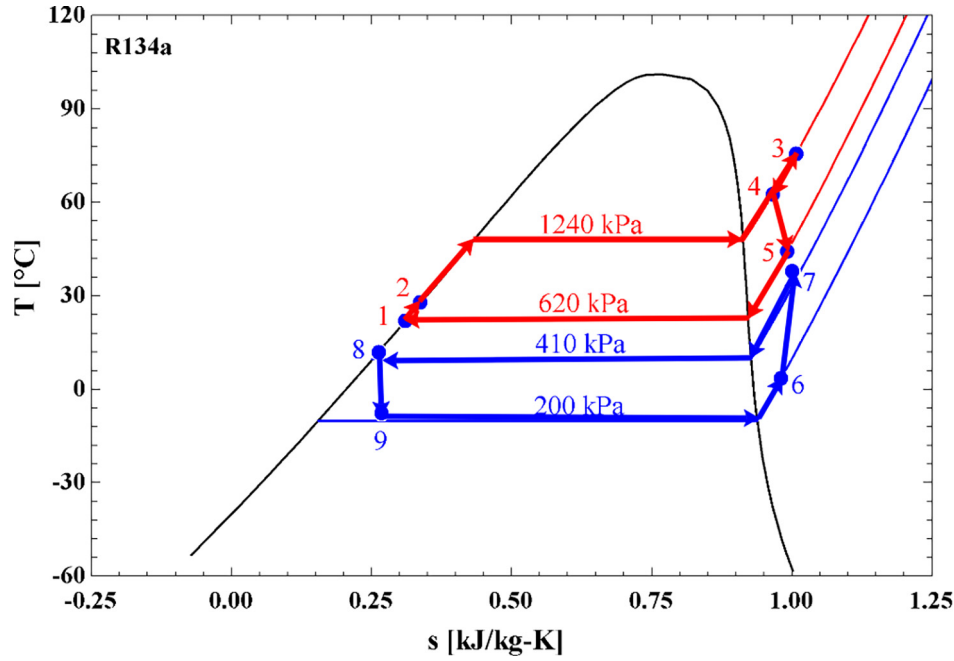


Fig. 3 T-s diagram of the TMR system.

where $U_{h,amb}$, and $A_{h,amb}$ are the heat transfer coefficient (HTC), and surface area of the heater. $T_{ev,avg}$, and T_{amb} are the average hot water and air temperatures, respectively. The detailed procedures for the calculation of the HTC of the plate heat exchanger are given in [36]. From Equations (2) and (3), the total heat provided by the hot water of the heater ($\dot{Q}_{h,total}$) is given as [38]:

$$\dot{Q}_{h,total} = \dot{Q}_{heater} + \dot{Q}_{h,amb} \quad (4)$$

$$\dot{Q}_{h,total} = \dot{m}_{h,in} \times c_{p,h,avg} \times (T_{h,in} - T_{h,out}) \quad (5)$$

where $\dot{m}_{h,in}$, $c_{p,h,avg}$, $T_{h,in}$, and $T_{h,out}$ are the mass flow rate, average specific heat, inlet temperature, and outlet temperature of the hot water that passes through the heater, respectively. Part of the expansion process will be at an isobaric pressure (from state 3 to state 4), then expand to the low pressure of the cycle at state 5. Thus, the expander power produced through the expansion of the PLR through the expander cylinder is given as [33]:

$$\dot{W}_{Exp} = \dot{m}_{PLR}(h_3 - h_4) \quad (6)$$

$$\dot{W}_{Exp} = NLA_{ex}(P_3 - P_5) \quad (7)$$

where N is the frequency of the ECU, L is the length of the stroke, A_{ex} is the cross-sectional area of the expander piston, and P_3 and P_5 are the pressures of the CLR at states 3 and 5, respectively. Similar to the heater model, the heat rejected by the PLR to the cold water of the cooler is given as [37]:

$$\dot{Q}_{co} = \dot{m}_{PLR} \times (h_5 - h_1) \quad (8)$$

And the heat transferred from the ambient air to the cold water of the cooler is given as [38]:

$$\dot{Q}_{co,amb} = U_{co,amb} \times A_{co,amb} \times (T_{amb} - T_{co,avg}) \quad (9)$$

where $U_{co,amb}$ and $A_{co,amb}$ are the HTC, and surface area of the cooler. $T_{co,avg}$ is the average temperature of the cooler cold water. The total heat absorbed by the cold water of the cooler is given as [37]:

$$\dot{Q}_{co,total} = \dot{Q}_{co} + \dot{Q}_{co,amb} \quad (10)$$

$$\dot{Q}_{co,total} = \dot{m}_{co,in} \times c_{p,co,avg} \times (T_{co,out} - T_{co,in}) \quad (11)$$

where $\dot{m}_{co,in}$, $c_{p,co,avg}$, $T_{co,out}$, and $T_{co,in}$ are the mass flow rate, average specific heat, outlet temperature, and inlet temperature of the cold water passes through the cooler, respectively.

It is worth mentioning that the detailed model of the cooling loop was provided by Sleiti et al. in [36]. Thus, as the main focus of this study is on the performance of the TMR system at ultra-low heat source temperatures, only the definitions of the compressor power (\dot{W}_{Comp}) and the cooling capacity of the evaporator are adapted here for the complement of the TMR system model. The power consumed through the compression stroke is given as [37]:

$$\dot{W}_{Comp} = \dot{m}_{CLR}(h_7 - h_6) \quad (12)$$

where \dot{m}_{CLR} is the mass flow rate of the cooling loop refrigerant. In terms of the frequency, length stroke, and surface area of the compressor piston alongside the pressure gradient between the condenser and evaporator, the compression power is given as [33]:

$$\dot{W}_{Comp} = NLA_{comp}(P_7 - P_6) \quad (13)$$

The evaporator cooling capacity is given as.

$$\dot{Q}_{ev} = \dot{m}_{CLR}(h_6 - h_9) \quad (14)$$

Now, the energy efficiency of the power loop is given as [33]:

$$\eta_{pl} = 100 \times \frac{(\dot{W}_{Exp} - \dot{W}_p)}{\dot{Q}_{h,total}} \quad (15)$$

And, the COP of the cooling loop is expressed as [33]:

$$COP = \frac{\dot{Q}_{ev}}{\dot{W}_{Comp}} \quad (16)$$

The next subsection introduces the validation of the above model that is used for the analysis of the TMR system with a detailed description of the systematic procedures that were employed through the experimental testing of the TMR in this study.

3.2. Testing procedures and validation

To start the experimental testing of the TMR setup, the test rig is installed and tested against the leakage of the refrigerants and circulating water through its heat exchanger. Then, based on the results of the experimental investigation of the cooling loop that was performed by Sleiti et al. in [36], 85 g of R134a is charged to the VCC as recommended for optimal operation of the cooling loop. To provide a cooling load to the evaporator of the VCC, water is heated to a temperature of 33 °C and circulated at a rate of 275 L/h through it. The water is heated to the 33 °C to achieve a high cooling load with the compact size of the evaporator by increasing the temperature difference

between the hot water and the working fluid of the VCC. Similarly, to achieve a fast condensation process within the compact size of the condenser of the VCC and the cooler of the ORC, cold water is circulated in both of them with an inlet temperature of 15 °C. The cold water of the condenser (in the VCC) is circulated at a rate of 275 L/h and a rate of 1200 L/h in the ORC. To drive the ORC, hot water is circulated at a flow rate of 1750 L/h, and an inlet temperature of 85 °C with 200 g of R134a is charged as the working fluid of the power loop. This set of values for the circulated water is referred to as the *Base Case* of the experimental test of this study. Then, the pneumatic pump and the pneumatic control valves of the ECU are actuated by compressed air provided by a small air compressor to start the operation of the TMR system frequency of 0.30 Hz. It was noted that the developed pressure by the ECU is not sufficient to provide a remarkable pressure gradient through the compressor of the cooling loop. Thus, the charged mass to the power loop was increased gradually, it was noted that a higher charged mass caused higher pressure by the ECU at state 7 until the charged mass reached 600 g. As the charged mass increased to 700–800 g, no increase was noted in the pressure developed by the ECU at state 7 with higher temperatures at the inlet of the cooler which increase the cooling load of the cooler. Thus, the test rig was evacuated and charged again with 600 g of R134a (the optimal mass of the first tested refrigerant). At this mass with the aforemen-

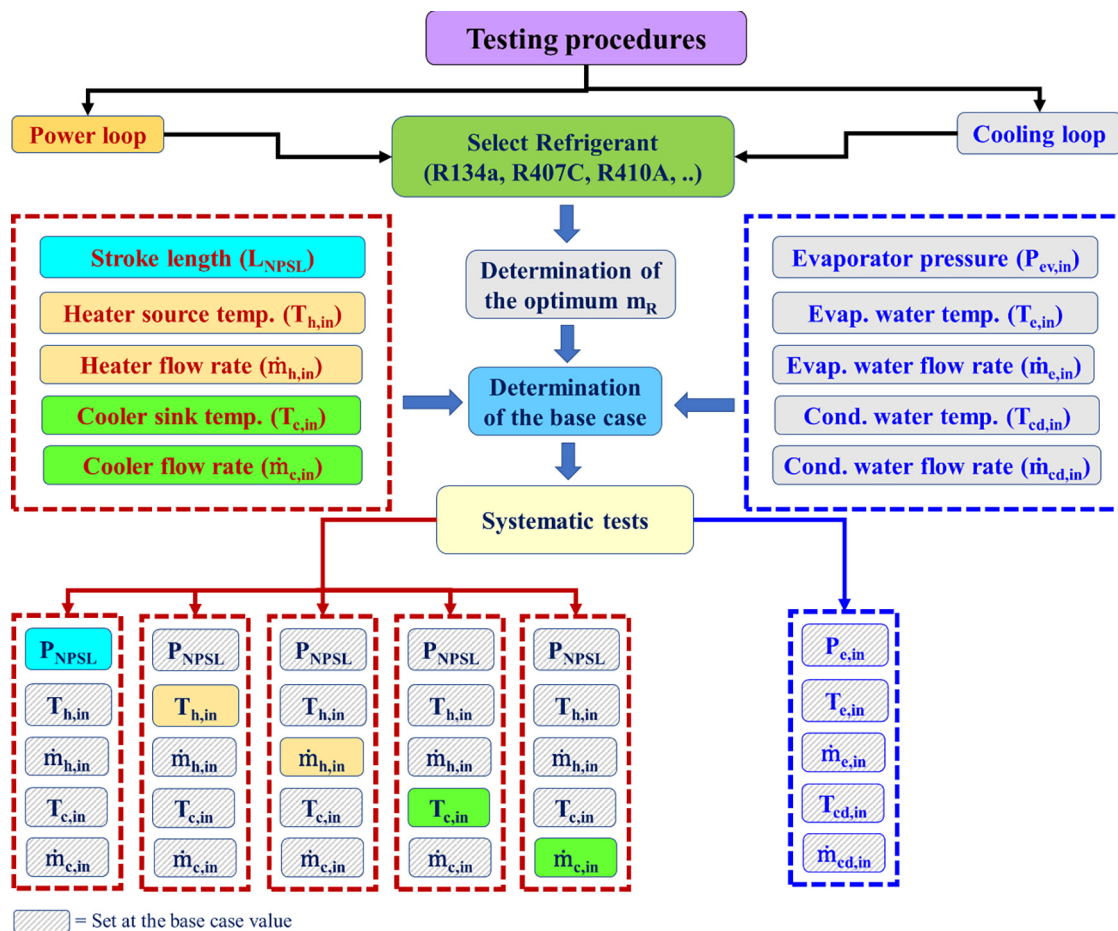


Fig. 4 Flow chart of the testing procedures.

Table 1 Validation of the measured parameters compared to their calculated values*.

Parameter Unit	T_2 [°C]	T_3 [°C]	T_5 [°C]	\dot{m}_{PLR} [g/s]	\dot{m}_{CLR} [g/s]	P_{ECU} [kPa]	T_7 [°C]	T_9 [°C]
Measured Value	15.20	64.10	39.80	4.20	1.10	540	44.0	-8.10
Calculated value	14.90	63.40	38.98	4.16	1.13	544	42.79	-7.88
Abs. relative error*, [%]	2.01	1.10	2.10	0.96	2.65	0.74	2.83	2.79

*Abs. relative error [%] = $100 \times |(\text{Calculated value} - \text{Measured value})| / |\text{Calculated value}|$.

tioned set for circulated water through all heat exchangers, the frequency of the pneumatic pump and ECU expander is increased to 0.50 Hz, but an unstable operation for the cycle was noted as the developed pressure by the ECU fluctuated for a longer time than at 0.30 Hz. Thus, the experimental test of the TMR system was performed at a frequency of 0.30 Hz in this study.

At the base case of the circulated water and frequency of 0.30 Hz, the test rig was operated several times before the beginning of the systematic testing of the controlled variables. This is done to ensure the repeatability of the measured data. Then, as shown in Fig. 4, five controlled parameters were selected for the systematic experimental assessment of the TMR system, which are: (1) stroke length percentage of the pneumatic pump (P_{NPSL}), inlet temperature ($T_{h,in}$) and (3) flow rate ($\dot{m}_{h,in}$) of the heater hot water, (4) inlet temperature ($T_{co,in}$), and (5) flow rate ($\dot{m}_{co,in}$) of the cooler cold water. The performance of the TMR system was examined by varying each of these parameters over a predetermined range while maintaining the other parameters at their base case values, as discussed and detailed in Section 4.

To ensure the accuracy of the measured data, several parameters of the TMR systems were calculated using the above thermodynamic model (Section 3.1) and compared to their measured values by the pressure, temperature, and flow rate sensors of the test rig. These parameters with their calculated and measured values are presented in Table 1. It is noted that the measured values show a good agreement with the calculated values with an absolute relative error of less than 2.90 %. In addition, the accuracy of the measurement devices is used to perform the uncertainty analysis of the performance indicators as presented in Supplementary material (SM. B).

4. Results and discussion

Based on the measurement results, the specifics of the testing procedure and the performance indicators are provided and discussed in this section. Sections 4.1 to 4.3 present the results of the tested controlled variables of the power loop including the stroke length of the pneumatic pump (Section 4.1), heating source temperature and flow rate (Section 4.2), and cooling sink temperature and flow rate (Section 4.3). It should be noted that during the test of a specified parameter, the other parameters of the system were maintained at their base case values as mentioned in Section 3.2. Section 4.4 present the experimental results of the TMR system using different refrigerants in the power and cooling loops and compares them to using the same refrigerant in both loops. Then, a comparison between the present TMR system with other cooling technologies was introduced in Section 4.5.

4.1. Effect of stroke length (full versus partial length)

As shown in Fig. 5, the full stroke length of the pneumatic pump piston is 110 mm. Using the stroke length adjuster, the stroke length can be reduced from 110 mm (100 %) to 11 mm (10 % of the full stroke length). This is referred to as the stroke length percentage of the pneumatic pump (P_{NPSL}). This means that $P_{NPSL} = 100\%$ at a stroke length of 110 mm and $P_{NPSL} = 10\%$ at a stroke length of 11 mm. For the test of the stroke length effect, the test rig first operated at the full stroke length ($P_{NPSL} = 100\%$), then tested at $P_{NPSL} = 75\%$ (82.5 mm) and $P_{NPSL} = 50\%$ (55 mm).

Fig. 6(a) shows the performance indicators of the TMR system with the variation of the stroke length percentage. It is noted that with the decrease of the stroke length percentage (from 100 % to 50 %), the energy efficiency (η_{pt}) decreased from 9.85 % to 7.18 % (27 % reduction). This is explained by the fact that the decrease of P_{NPSL} yields lower pressure for the PLR at the inlet of the heater (state 2) as shown in Fig. 6(b). Thus, the amount of heat absorbed from the hot water of the heater (from 0.88 kW to 0.43 kW, Fig. 6(a)). In addition, the expander work rate is reduced (from 0.087 kW to 0.031 kW) as the pressure ratio through the expansion process depends on the developed pressure by the pneumatic pump.

Although the expander power is decreased with the reduction of the P_{NPSL} , the developed pressure by the ECU at the outlet of the cooling loop compressor (state 7) is slightly reduced from 424 kPa to 413 kPa (Fig. 6(b)). However, the

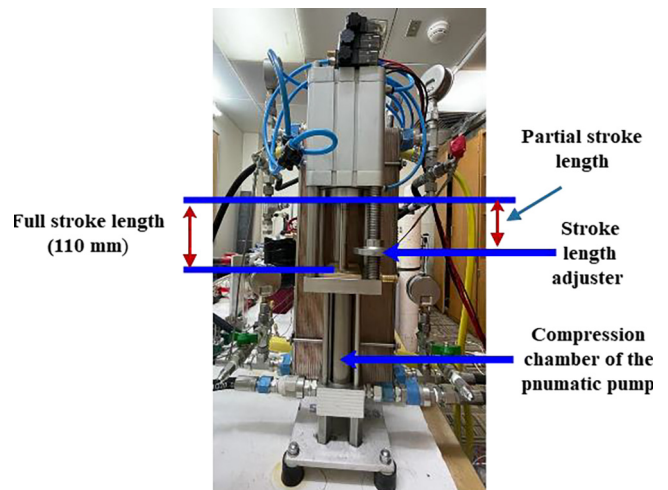


Fig. 5 View of the pneumatic pump mechanism.

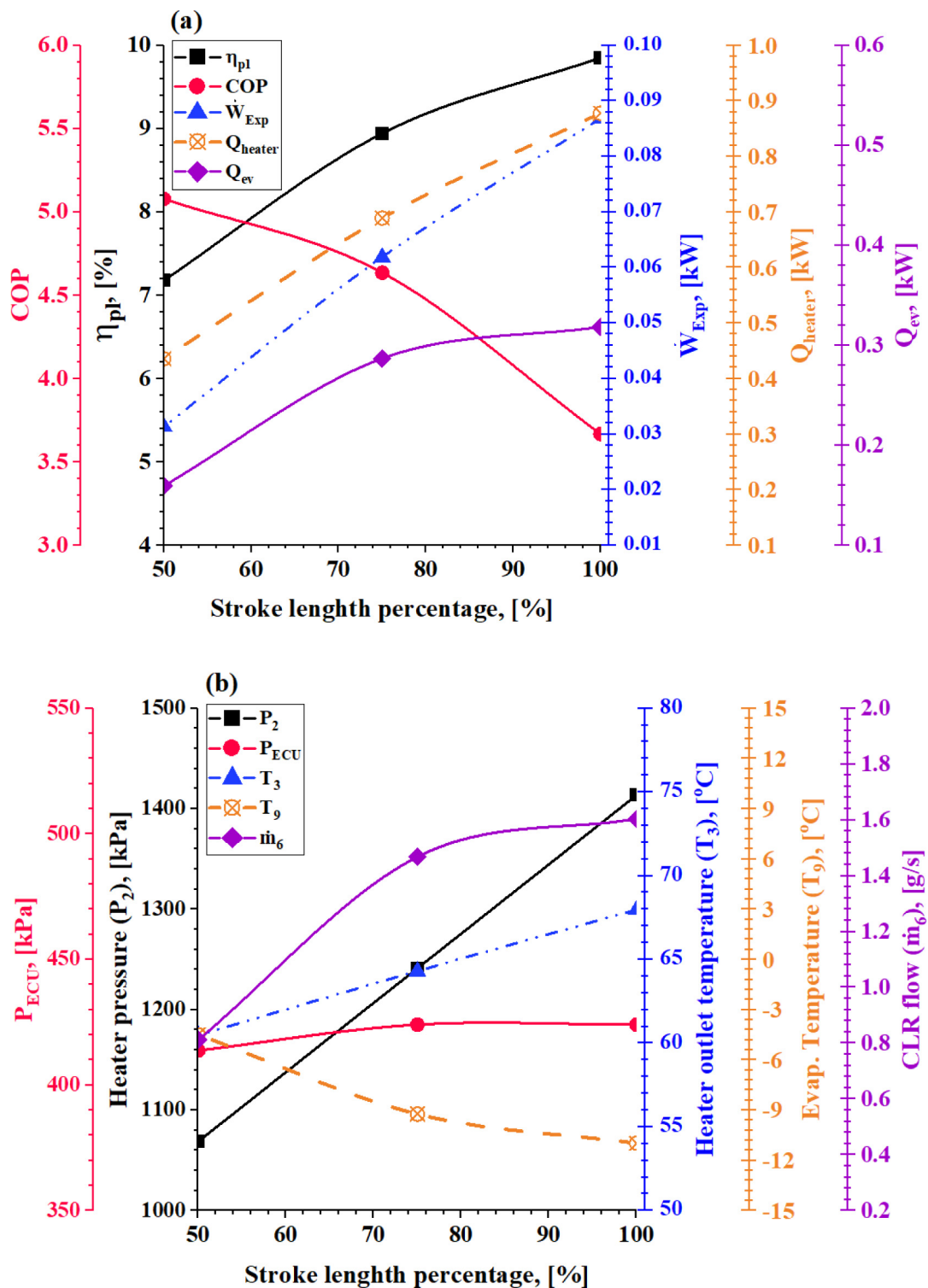


Fig. 6 Relationship between the stroke length percentage and TMR (a) performance indicators, and (b) major operating parameters.

flow rate of the CLR is reduced from 1.6 g/s to 0.81 g/s and the pressure at the inlet of the evaporator is increased. This in turn reduces the cooling capacity of the evaporator (Q_{ev}) from 0.32 kW to 0.15 kW and increases the evaporation temperature from $-10.98\text{ }^\circ\text{C}$ to $-4.5\text{ }^\circ\text{C}$. Thus, even though the cooling

capacity is reduced by 46.87 %, the cooling quality is acceptable at $P_{NPSL} = 50\text{ \%}$ as the evaporation temperature is considerably below $0\text{ }^\circ\text{C}$.

From the above results, it is noted that the expander power is reduced by 64.36 % compared to a 46.87 % reduction in the

cooling capacity, thus the COP of the cooling loop is increased from 3.66 to 5.08 with the decrease of the P_{NPSL} from 100 % to 50 %.

In conclusion, the decrease of P_{NPSL} negatively affects the efficiency of the power loop and the cooling capacity of the evaporator. However, the cooling quality is acceptable and the COP is improved even with the reduction of the cooling capacity.

4.2. Effect of heating source temperature and hot water flow rate

The effect of the inlet temperature of the hot water of the heater ($T_{h,in}$) is evaluated by decreasing it gradually from the base case value (85 °C) to less than 50 °C to mimic the available energy sources with ultra-low temperatures. Through this test, the flow rate of the hot water was maintained at 1850 L/h. It is noted that as the $T_{h,in}$ is less than 65 °C, the evaporation temperature increases to values higher than 0 °C, which is considered an unacceptable cooling quality in this study. Thus, the performance curves were plotted over the range of $T_{h,in}$ from 65 °C to 85 °C as shown in Fig. 7.

As shown in Fig. 7(a), the energy efficiency of the power loop is increased from 5.92 % at $T_{h,in}$ of 65 °C to 9.52 % at $T_{h,in}$ of 85 °C. Similarly, the COP of the cooling loop is improved from 2.36 to 4.40 over the range of $T_{h,in}$. The improvement of the power loop efficiency is explained by the increase of the temperature (T_3) and pressure of the PLR at the inlet of the expander cylinder with the increase of $T_{h,in}$ as shown in Fig. 7(b). The increase of the pressure occurs due to the increase of the pressure developed by the pneumatic pump (P_2). This is because the higher heater temperature reduces the viscosity of the PLR, thus the pressure loss through the back-stroke of the expander piston and the flow through the cooler is minimized which allows higher pressure at both the inlet and outlet of the pneumatic pump.

Also, as $T_{h,in}$ is increased, the developed pressure by the ECU is increased from 393 kPa to 448 kPa and the flow rate of the CLR is increased from 0.67 g/s to 1.35 g/s, which enhances the COP. Although the efficiency and COP of the TMR system at $T_{h,in}$ of 65 °C are considerably lower than at $T_{h,in}$ of 85 °C, the cooling quality is still appreciable as an evaporation temperature of -1.66 °C is achieved at $T_{h,in}$ of 65 °C. This demonstrates the ability of the TMR system to perform an efficient cooling process even at ultra-low temperatures of the heat source compared to those required by other thermal cooling technologies such as absorption-based and ejector-based systems. These systems need heating source temperatures higher than 90 °C to perform an effective cooling process as mentioned in [14].

Fig. 8 shows the performance of the TMR system if the hot water temperature of the heater is maintained at 85 °C and its flow rate is changed from 1000 L/h (16.66 L/min) to 2000 L/h (33.33 L/min). Note that the flow rate of the heater hot water ($\dot{m}_{h,in}$) is presented in L/min instead of L/h for the convenience of the figure's presentation. It is noted that the efficiency of the power loop is slightly improved (from 8.89 % to 9.47 %, Fig. 8(a)) with the increase of the $\dot{m}_{h,in}$ comparison to that of the $T_{h,in}$ increase. This is due to the increase of the flow rate which improves the overall heat transfer coefficient of the heater with a minor effect on the pressure and temperature at state 3 up to a flow rate of 25 L/min. However, at

$\dot{m}_{h,in}$ higher than 30 L/min, the temperature of the PLR at state 3 (T_3) arises from 63.21 °C (at 25 L/min) to 71.97 °C (at 30 L/min). This enhances the developed work on the refrigerant of the cooling loop and increases the developed pressure by the ECU (from 492 kPa to 523 kPa) and the flow rate of the CLR (from 1.15 g/s to 1.95 g/s) as shown in Fig. 8(b). This is explained by the increase of the cooling capacity Q_{ev} (from 0.24 kW to 0.35 kW) and the COP (from 2.91 to 3.99) of the cooling loop as presented in Fig. 8(a).

In terms of the cooling quality, it is noted that evaporation temperature is slightly decreased from -6.0 °C to -7.5 °C with the increase of $\dot{m}_{h,in}$ as the increase of the ECU pressure is lower than that obtained with the increase of $T_{h,in}$. Therefore, it can be concluded that the performance of the TMR system is greatly affected by the heat source temperature more than the flow rate available at that source. Thus, if the heat source temperature is higher than 65 °C, then the TMR system can be worked with appreciable cooling quality and efficiency higher than 5 % even at low flow rates for the heating source fluid (water in this study).

4.3. Effect of cooling sink temperature and cold water flow rate

The effect of the water cooler temperature ($T_{c,in}$, cooling sink temperature) on the performance of the TMR system is evaluated by increasing it gradually from 10 °C to 30 °C as shown in Fig. 9. At the base case value of $T_{c,in}$ (15 °C), the power loop efficiency and COP of the cooling loop are 8.62 % and 2.97, respectively. The increase of $T_{c,in}$ yields a higher temperature of the PLR at the inlet of the pneumatic pump which negatively affects its performance and reduces its developed pressure at the heater (P_2) (Fig. 9(b)). Thus, an even higher temperature is obtained at state 3 with the increase of $T_{c,in}$, the developed pressure by the ECU, and the flow rate of the CLR are reduced (Fig. 9(b)). Therefore, both the efficiency and the COP of the TMR system are reduced (from 9.6 % to 5.51 %, and 3.15 to 2.52, respectively) with the increase of $T_{c,in}$. However, the cooling capacity is slightly affected by the increase of $T_{c,in}$ and levelized at an average value of 0.41 kW up $T_{c,in}$ of 25 °C and sharply reduced to 0.28 kW at $T_{c,in}$ of 30 °C. Furthermore, the evaporation temperature is levelized at -8.80 °C and negligibly changed with the variation of $T_{c,in}$. This reveals that the cooling process through the cooler can maintain a stable cooling capacity for the TMR system as the cooling sink temperature is less or close to the ambient temperature (25 °C).

For the effect of the cold water flow rate ($\dot{m}_{co,in}$) at the base case inlet temperature of 15 °C, $\dot{m}_{co,in}$ is gradually increased from 11 L/min to 22 L/min as shown in Fig. 10. It is noted that the effect of $\dot{m}_{co,in}$ is the opposite for the $T_{c,in}$ effect. That means the increase of the $\dot{m}_{co,in}$ enhances the efficiency of the power loop (Fig. 10(a)) even with the decrease of the temperature at the inlet of the expander (T_3 , Fig. 10(b)). This is explained by that at higher $\dot{m}_{co,in}$, the PLR enters the pneumatic pump at a lower temperature (denser phase), which enhances the pressure through the heater. However, the temperature of the PLR is reduced at the inlet of the heater, which minimizes T_3 (from 82.39 °C to 56.67 °C) and levelized the absorbed heat from the hot water at an average value of $Q_{\text{heater}} = 0.88$ kW. Furthermore, the developed pressure by the ECU and the flow rate of the CLR are increased from

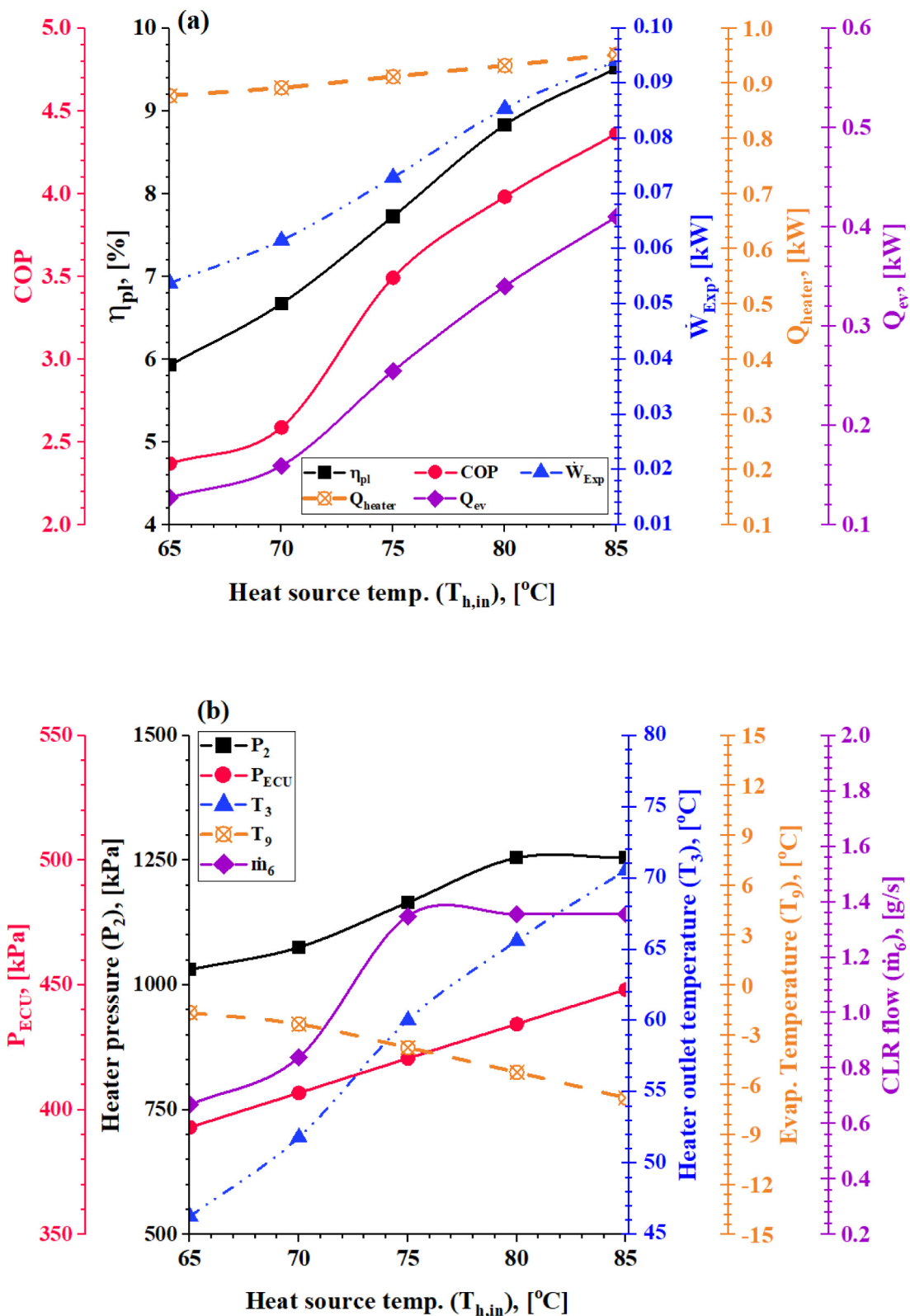


Fig. 7 Relationship between the heat source temperature and TMR (a) performance indicators, and (b) major operating parameters.

413 kPa to 448 kPa, and from 0.66 g/s to 1.38 g/s, respectively. This is returned to the increase of the COP with the increase of $\dot{m}_{co,in}$. Also, it is noticed that the evaporation temperature is not affected by the variation of $\dot{m}_{co,in}$ and is maintained at -8.79 °C (Fig. 10(b)).

In conclusion, it can be noted that the TMR system can work with energy efficiency higher than 8.5 % and COP higher than 2.40 even at low flow and high temperature for the cold water of the cooler.

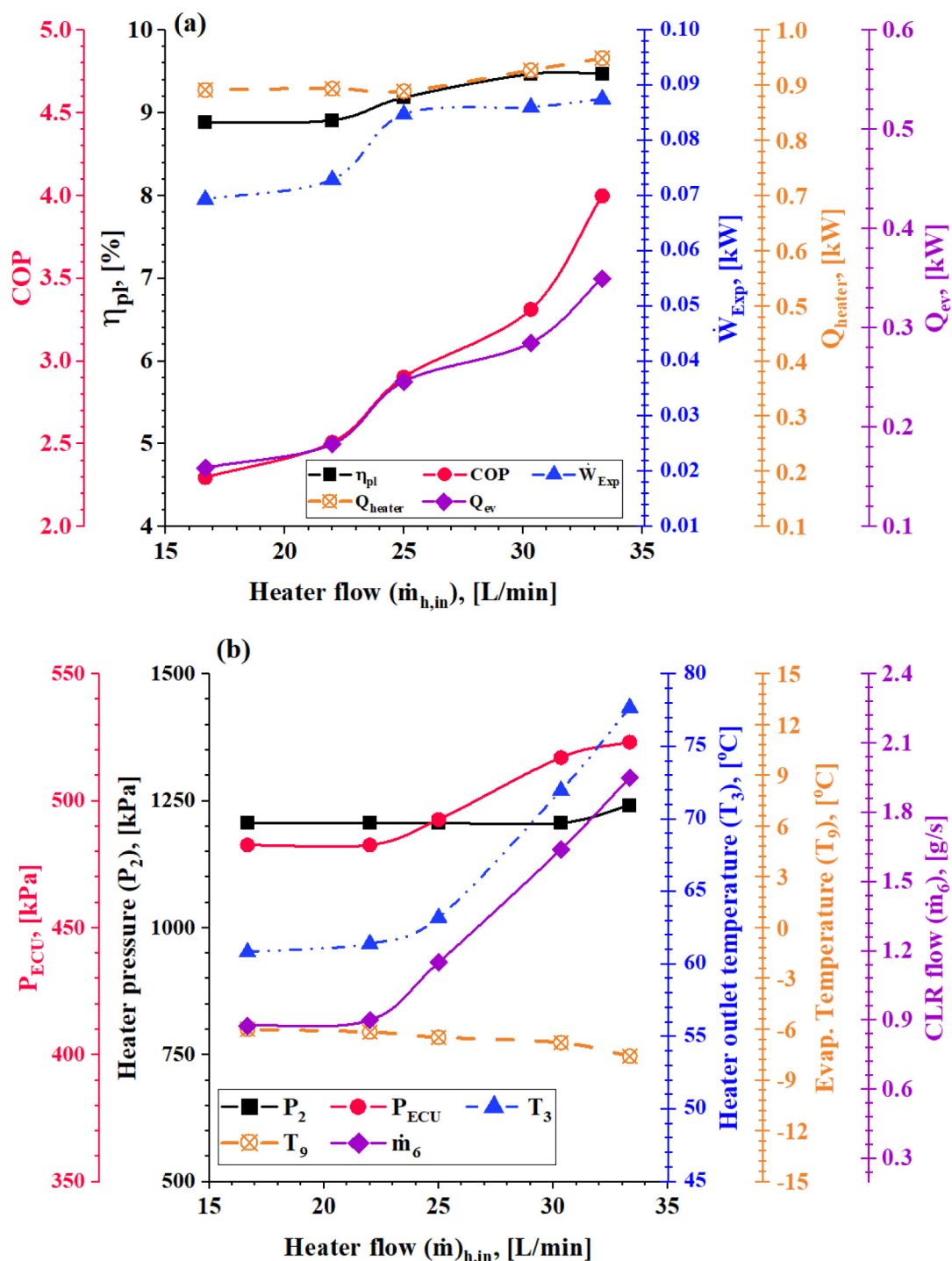


Fig. 8 Relationship between the water flow rate through the heater and TMR (a) performance indicators, and (b) major operating parameters.

4.4. Comparison between using the same refrigerant versus different refrigerants in each loop

The results of the previous sections (Sections 4.1 to 4.3) were obtained for the refrigerant R134a in both power and cooling loops. This is referred to as scenario 1 (S1). After the examination of the controlled parameter of the power loop, the effect

of the refrigerant type was tested by replacing R134a with R407C and R410A.

It is noticed that the TMR system has a poor performance in case of replacing R134a with R407C or R410A in the power loop or replacing R134a with R410A in the cooling loop. Thus, their results were not discussed in this study as the evaporation temperature does not reach below 0 °C, which is con-

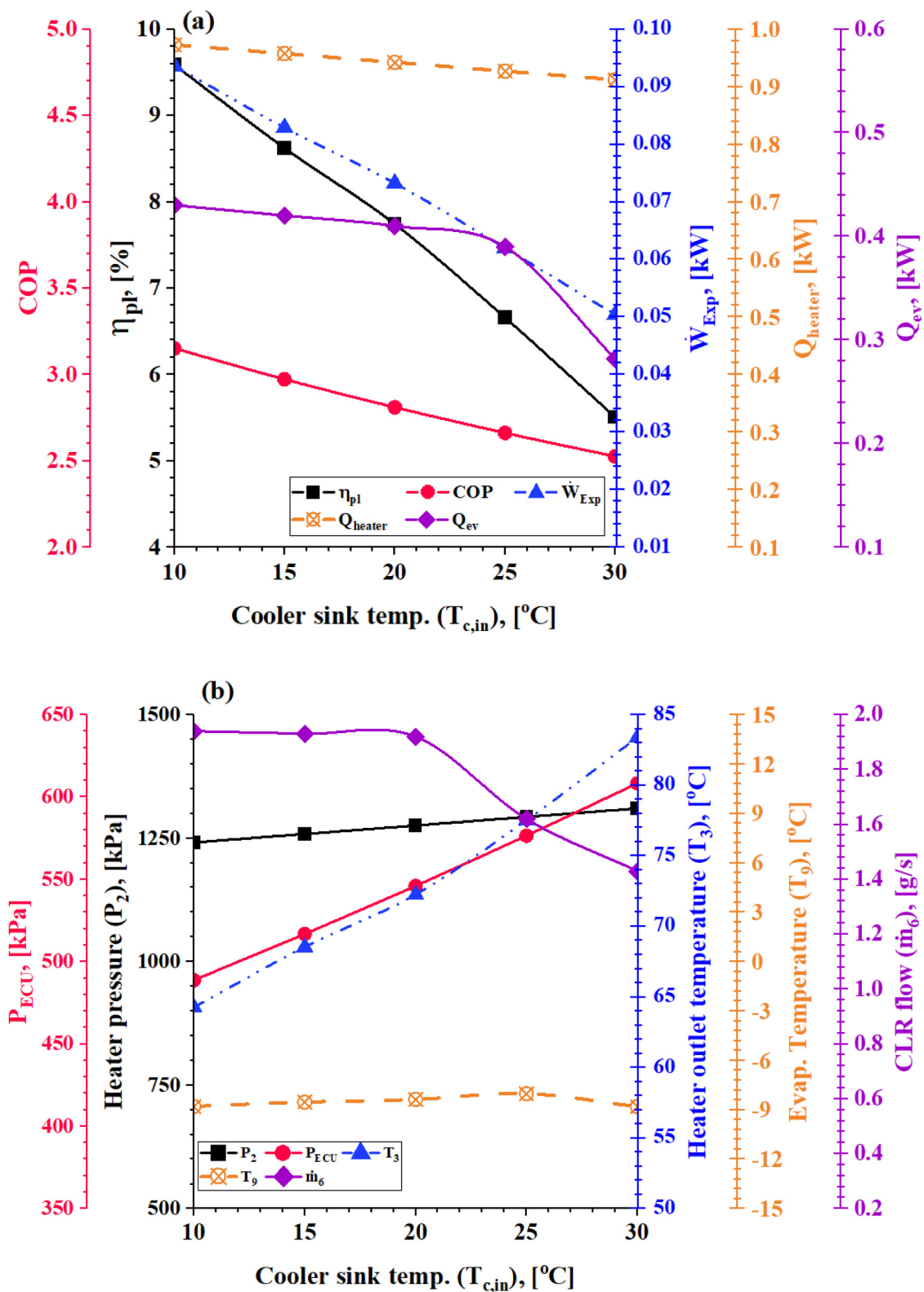


Fig. 9 Relationship between the cooler sink temperature and TMR (a) performance indicators, and (b) major operating parameters.

sidered poor cooling quality. However, at the base case values of the controlled parameters (discussed in Section 3.2), the TMR system works at comparable efficacy and better cooling quality for the case of using R134a in the power loop (PL) and

R407C in the cooling loop (CL). This is referred to as scenario 2 (S2). The performance indicators for S1 and S2 are depicted in Fig. 11 with the variation of the pneumatic pump stroke length.

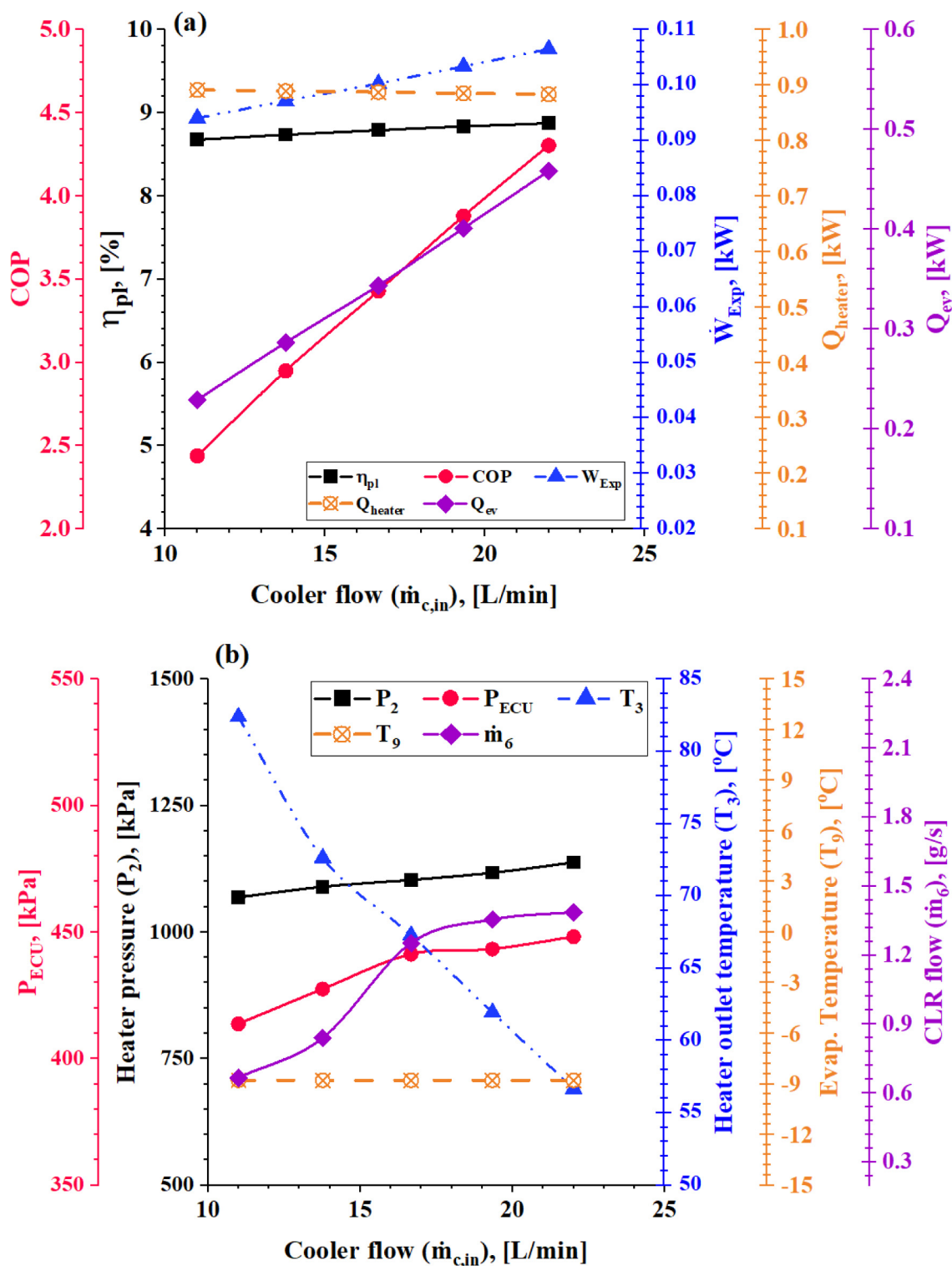


Fig. 10 Relationship between the cooler flow and TMR (a) performance indicators, and (b) major operating parameters.

It is noted that the power loop efficiency of S2 is close to that of S1 with an efficiency of 9.95 % and 7.01 % at 100 % and 50 % of the full stroke length, respectively. However, the COP of S1 is about 11 % higher than that of S2 as the cooling capacity of R134a is much higher than that of R407C at evaporation temperatures between -5°C to -10°C . But, with

R407C, the developed pressure by the ECU is much higher than that of R134a, which increases the pressure gradient through the cooling loop. Thus, the evaporation temperature for S2 varies from -18.82°C to -20.43°C , which is two to three times lower than that of S1. This implies that the cooling quality of S2 is better than that of S1 as lower evaporation

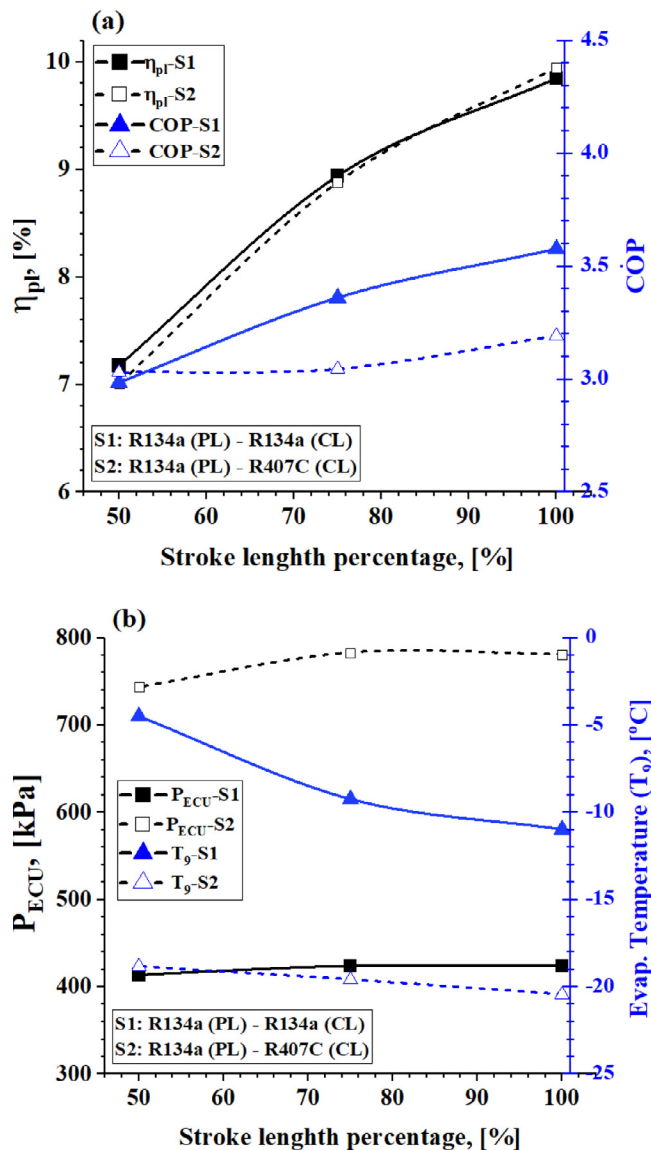


Fig. 11 Comparison between using the same refrigerant in the power loop (PL) and cooling loop (CL) (S1: R134a (PL) – R134a (CL)), versus different refrigerant (R134a (PL) – R407C (CL)).

temperatures are achieved. Thus, if the demand from the cooling loop is to produce a higher cooling capacity rather than the value of the evaporation temperature, then S1 is recommended. But, if low evaporation temperatures are needed as applied for some food treatment applications, then S2 is recommended.

4.5. Comparison

From the previous results, it can be stated that the TMR system can work efficiently with power loop efficiency between 5.92 % and 9.85 %, COP between 2.36 and 3.99, and evaporation temperature from -1.6 °C to -20.43 °C over a wide range of operating conditions for the heat source and heat sink of the power loop. Furthermore, the present TMR system shows its workability and acceptable cooling quality at ultra-low temperatures of the heat source up to 65 °C at the minimal stroke

length of the pneumatic pump. Also, the cooling quality could be improved by alternating the refrigerant of the cooling loop (R134a) with a more suitable refrigerant to reach a lower evaporation temperature such as R407C (as discussed in Section 4.4).

The present experimental analysis reveals that the COP (2.36–5.08) of the tested ECU-based TMR system is higher (by four times) than that of the ORC-based and Ejector-based systems which have COP of 0.1–0.75, and 0.1–0.62, respectively [39]. The high COP of the ECU-based system is mainly due to the direct use of the expander power to compress the refrigerants, with minor losses due the friction. However, the ECU of the TMR system has some technical limitations such as fluctuated operation at a high operating frequency and the need for lubrication and sealing to prevent leakage from the ECU’s cylinders.

5. Conclusions

This study presents a systematic experimental analysis of the performance of a thermo-mechanical refrigeration (TMR) system that utilizes an expander-compressor unit (ECU) to integrate an organic Rankine cycle (ORC) cycle (power loop) with a vapor compression cycle (VCC, cooling loop). The experimental investigation is conducted on a new full-scale ECU-based TMR system with a design cooling capacity of 1 kW at a heat source temperature of 85 °C, which includes the novelty points of (i) experimentally evaluating the performance of a new full-scale TMR system at ultra-low temperatures, (ii) characterizing its operation with different commercial refrigerants, and (iii) identifying optimal operating conditions and working fluids for the investigated TMR system. The TMR system is tested with a heat source temperature range of 50 °C to 85 °C (using circulated hot water) to match the range of ultra-low temperature waste heat sources, and the performance of the system is analyzed for a wide range of operating conditions of the flow rates and temperatures of the heat source and heat sink of the power loop. Moreover, the conducted tests are performed at different stroke lengths for the pneumatic pump of the power loop with three commercial refrigerants (R134a, R407C, and R410A). The main findings of the present study are:

- With R134a in both the power and cooling loops, the TMR system shows efficient operability up to an ultra-low temperature of 65 °C with power loop efficiency of 5.92 % and COP of 2.36 at 50 % of the pneumatic pump stroke length.
- The TMR system shows a power efficiency of 9.85 % and COP of 5.08 at an ultra-low temperature of 85 °C and 100 % of the pneumatic stroke length.
- At high temperatures of the power loop heat sink (up to 30 °C), the TMR system shows stable operation with a power loop efficiency of 8.5 % and COP of 2.40.
- The cooling quality of the TMR cooling loop is enhanced by replacing R134a with R407C which provides an evaporation temperature of less than -20 °C.

The current experimental study also demonstrates that the ECU-based TMR system can overcome the ORC ejector cooling systems’ limitations when used with ultra-low temperature

waste heat sources. Furthermore, the presented results serve as guidelines for designing and building experimental setups for thermo-mechanical refrigeration systems and for developing more efficient ECU-based TMR future systems. Finally, it is worth noting that the choice of refrigerants as working fluids can significantly impact the performance of the TMR system. Moreover, many commercial refrigerants have high global warming potential, making it important to explore the use of more eco-friendly mixed refrigerants in future studies. To this end, we recommend conducting experimental evaluations of the TMR system under different conditions with environmentally sustainable mixed refrigerants, building upon the optimal conditions reported in this study.

CRedit authorship contribution statement

Ahmad K. Sleiti: Conceptualization, Investigation, Writing – original draft, Resources, Formal analysis, Project administration, Funding acquisition, Supervision. **Wahib A. Al-Ammari:** Conceptualization, Writing – original draft, Investigation, Software, Data curation, Validation, Formal analysis. **Mohammed Al-Khawaja:** Conceptualization, Project administration, Funding acquisition, Supervision.

Declaration of Competing Interest

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.

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Appendix A. Supplementary material

Supplementary data to this article can be found online at <https://doi.org/10.1016/j.aej.2023.03.083>.

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