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Experimental investigation into an ORC-based low-grade energy recovery system equipped with sliding-vane expander using hot oil from an air compressor as thermal source

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Abstract

Compressed air production is an energy–intensive sector, thus compressor manufacturers are constantly looking for enhancing the efficiency, by acting on several technological aspects. In an air compressor, about 80-90% of the input electric power used is wasted into the environment through the oil circuit, continuously cooled by ambient air blown via a fan. An interesting way to optimize the overall system efficiency is to exploit this waste heat to produce electrical power. Organic Rankine Cycles (ORCs) are a suitable solution for recovering energy from low-grade heat source. In this paper, an experimental analysis of two low-grade ORC-based recovery systems is presented. The thermal source is the hot lubricant of a mid-size air compressor, while the thermal sink is tap water. The first system is tested in a *simple* cycle configuration while the second in a *recuperative* one. An extensive experimental campaign is carried out on a test bench composed by sliding-vane expander, pump and plate heat exchangers. The expander differs in terms of geometry and aspect ratio between the two cycles. R236fa is used as working fluid in both the systems. The expander operating conditions are deeply investigated by using piezoelectric pressure transducers to determine the expansion indicated diagram and the expander mechanical efficiency. Experimental results show that the recuperative cycle has a better performance, in terms of cycle efficiency and expander mechanical efficiency, compared with the simple cycle. For this configuration, two off-design conditions are investigated, acting on the pump rotational speed. Finally, an exergy analysis is conducted, in order to evaluate the irreversible losses produced by each component.

Keywords: low-grade ORC; sliding-vane expander; compressor waste heat recovery

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Nomenclature

		Subscripts	
$\boldsymbol{c}$	specific heat, kJ/kg K	exe	exergy
$\boldsymbol{\Delta}$	difference, -	exp	expander
Ex	exergy rate, kW	hx	heat exchanger
h	specific enthalpy, kJ/kg	$HTHX$	high temperature heat exchanger
η	efficiency, -	id	ideal
$\dot{m}$	mass flow rate, kg/s	in	inlet
p	pressure, bar	ind	indicated
$\dot{Q}$	heat transfer rate, kW	$LTHX$	low temperature heat exchanger
ρ	density, kg/m^3	mc	mechanical
s	specific entropy, kJ/kg K	oil	compressor lubricant
S	entropy rate, kW/K	out	outlet
T	temperature, K	pmp	pump
$\dot{V}$	volumetric flow rate, m^3/s	p	constant pressure
$\dot{W}$	power, kW	s	isentropic
		ths	thermal source
		tot	total
		WF	working fluid
		0	reference condition

1. Introduction

The development of new sustainable technology for energy efficiency is one of the main objectives in industrial sector, because of the continuous environmental challenges and the increase of electricity demand and cost. Any time heat is wasted into the environment, the possibility of taking advantage of its energy content has to be considered. About half of the total waste thermal energy is dissipated as low-grade thermal source (30°C-200°C) in typical industrial processes such as chemical and thermal plants, exhausts from combustion, condensing and cooling systems. The most widely used solution to take advantage of these low-grade thermal sources is the Organic Rankine Cycle (ORC). In many industrial applications, low-grade waste heat can also come from equipment such as compressors, where the lubricating oil circuit is continuously cooled by ambient air blown by a fan. In compressed air applications, the energy produced by the recovery system could be used directly in the package (i.e. to feed the compressor and its electrical auxiliary) or it could be delivered to the electric grid, in compliance with the local policy. Furthermore, in either case, if a high thermal recovery is reached, the fan of the compressor cooling system can also be switched off allowing a significant energy saving.

The key elements of ORC design are the choice of the working fluid and of the expander. The choice of the working fluid is critical due to the influence on system efficiency, components sizes, stability and safety. Different studies in literature focus on the definition of a working fluid selection criterion. A review of those criteria has been made by Bao and Zhao [1], highlighting the working fluid physical properties that must be taken into account. Among them saturation curve shape, molecular complexity, critical temperature, vaporization latent heat, density are the most important. Therefore, there are several aspects to consider and different methods to evaluate them, but most of all the working fluid choice is strictly related to heat source thermal level [2].

Another critical aspect is the selection of the expander. In the range of small power ORCs, positive displacement expanders are more suitable then dynamic expanders due to the following features: lower speed, good off-design performance, high expansion ratio, low cost, simple manufacturing [3]. Increasingly numerous are the experimental activities on low-grade ORC using different kinds of expander. Lemort et al. [4] test an open-drive oil-free scroll expander integrated into an ORC using R123. They focus the attention on the main losses that affect the performance. Declaye et al. [5] build a performance map of an open-drive scroll expander tested in an ORC using R245fa as working fluid. Miao et al. [6] analyse the performance of an ORC system driven by a scroll expander using R123. They highlight the issues in accurate thermodynamic measures inside the expander. Small scale ORC systems equipped with screw expander are also investigated in different studies. Desideri et al [7] study the performance of a single screw expander able to reach a maximum generated power of 7.8 kW. Wang et al. [8] carry out a comparative analysis on three different prototypes of single screw expanders, using pressurized air for the purpose. Smith et al [9] focus on twin-screw expanders, discussing the power recovery optimization from low-grade heat.

Among these numerous studies, ORC experimental experiences with sliding-vane rotary expanders are few. Despite this, rotary vane expanders are suitable for such recovery systems due to their features: low speed, self-starting under load, smooth torque production, low noise and vibration, simple structure, absence of valves, low cost. Previous works [10, 11] focused on the definition of a model for sliding-vane rotary expander, the validation through tests on a prototype and the performance analysis of the overall system.

This paper presents an experimental investigation into the performance of two low-grade ORC systems characterized by different layouts and equipped with rotary vane expanders and pump. The present activity is a continuation of an industrial project that has defined and proven the integration of the sliding-vane technology in a low-grade ORC system [12]. The used sliding-vane expanders differ in geometric design (aspect ratio, eccentricity, blade thickness, etc.) and are designed to work in a wide range of operating conditions. The low-grade thermal source is the lubricant of a sliding-vane air compressor. In particular, the two systems are both coupled (alternatively) to the same compressor and they differ in the configuration: one is simple cycle, the other recuperative.

2. Experimental method

In the following, the experimental activity on the simple and recuperative ORC systems is analyzed. The two recovery systems are tested using the same thermal source. However, in order to optimize the overall performance, they show little design differences in their components in terms of expander (see Table 1) and heat exchangers sizes.

Table 1. Expanders main parameters

	Displacement [cm³]	Built-in volume ratio [-]	Rotor length [mm]	Rotor diameter [mm]
Simple cycle expander	26.50	3.34	160	80
Recuperative cycle expander	19.95	2.76	90	100

2.1. The test rig

The test bench reported in Figure 1 and Figure 2 displays the most relevant components: a sliding-vane rotary pump coupled with a brushless electric motor; a sliding-vane rotary expander coupled with an electric generator; two plate-type heat exchangers, one for evaporating (HTHX) the organic fluid and the other for condensing it (LTHX). The hot lube-oil from a sliding vane rotary compressor is used as thermal source in the HTHX, while the cold source in the LTHX is tap water.

In order to increase the system efficiency, an additional heat exchanger can be used as recuperator and economizer downstream of the expander to pre-heat the cold liquid from the pump outlet. The additional heat exchanger allows also to optimize energy recovery and to reduce the thermal load on the main heat exchangers, so that the overall dimension can be reduced. The experimental test bench for the recuperative cycle is shown in Figure 3.

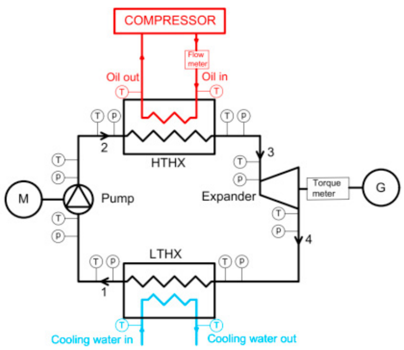

Figure 1. Diagram of the simple cycle

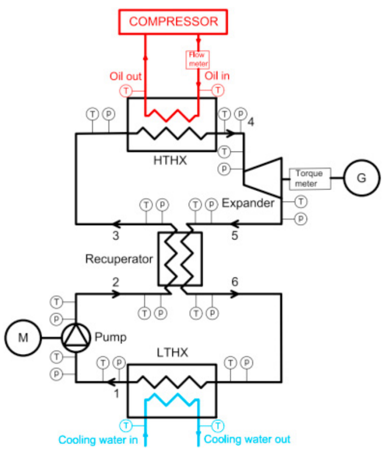

Figure 2. Diagram of the recuperative cycle

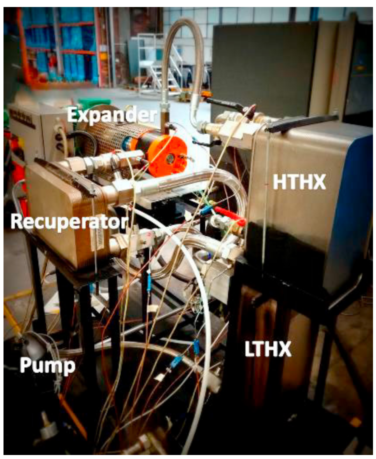

Figure 3. Recuperative cycle test rig

For both systems, the working fluid is R236fa, with the addition of POE lubricant 5% w/w (mass basis). The working fluid is selected according to specific criteria:

- high molecular complexity, in order to have a dry expansion and avoid liquid presence in the expander;
- low vaporization latent heat, to reduce temperature difference between fluids in the heat exchanger;
- high density, to have lower volume flow rate, less pressure drops and more compact heat exchangers;
- low viscosity, to reduce friction losses in heat exchangers and pipes;
- high conductivity, to increase heat transfer;
- fluid stability in cycle temperatures range, compatibility with materials in contact, availability and cost.

In order to evaluate the thermodynamic conditions of every significant point of the cycle, type T thermocouples and pressure transducers are installed on the test bench. The compressor oil flow rate is measured through a flow meter: this operating parameter allows to determine the thermal power exchanged in the HTHX. The mechanical power of the expander is measured by means of a torque meter. Four piezoelectric pressure transducers are placed on the end cover of the expander; they allow to deeply investigate the expansion process by reconstructing the indicated cycle. Instrumentation uncertainties are listed in Table 2.

Table 2. Measurement devices and uncertainties.

Instrument	Manufacturer	Model	Quantity	Uncertainty
Thermocouple	Tersid	Type T	Temperature	0.5°C
Pressure transducer	Remag	PR-100	Pressure	0.08 bar
Piezoelectric pressure transducer	Kistler	601A	Pressure	0.01 bar
Flow meter	Omega	FL-8107A	Flow rate	4 l/min
Torque meter	Kistler	4503A	Torque	0.1 Nm
			Angular speed	1 rpm

2.2. Experimental procedure

The performance of the systems are investigated in a wide range of operating conditions controlling the working fluid mass flow by acting on the rotating speed of the pump. The expander rotational speed is constrained by the electricity grid frequency.

However, some of the operating parameters are calculated indirectly by energy balances. The overall thermal power received by the ORC system, $\dot{Q}_{HTHX}$ [kW], is evaluated by temperatures and flow rate measurements on HTHX oil side, and is calculated as follows:

$$\dot{Q}_{HTHX} = \rho_{oil} \cdot \dot{V}_{oil} \cdot c_{p,oil} \cdot \left(T_{oil,HTHX,in} - T_{oil,HTHX,out}\right) \tag{1}$$

where ρ_{oil} [kg/m^3] is the oil density and $c_{p,oil}$ [kJ/kgK] is the oil specific heat. These properties are assumed constant because of the small variation of the oil temperature (maximum 20 °C). $\dot{V}_{oil}$ [m^3/s] is the oil volume flow rate while $T_{oil,HTHX,in}$ [K] and $T_{oil,HTHX,out}$ [K] are the oil temperature at the inlet and outlet of the HTHX respectively. The calculation of the working fluid mass flow rate, $\dot{m}_{WF}$ [kg/s], is based on the energy balance on the HTHX. A proper insulation is used for the purpose, in order to neglect any power loss to the environment:

$$\dot{m}_{WF} = \frac{\dot{Q}_{HTHX}}{\left(h_{WF,HTHX,out} - h_{WF,HTHX,in}\right)} \tag{2}$$

where $h_{WF,HTHX,out}$ [kJ/kg] and $h_{WF,HTHX,in}$ [kJ/kg] are the working fluid enthalpy at the inlet and outlet of the HTHX respectively.

The ORC net mechanical power production, $\dot{W}_{net,mc}$ [kW], and the cycle efficiency η are determined as follows:

$$\dot{W}_{net,mc} = \dot{W}_{exp,mc} - \dot{W}_{pmp,mc} \tag{3}$$

$$\eta = \frac{\dot{W}_{net,mc}}{\dot{Q}_{HTHX}} \tag{4}$$

where $\dot{W}_{exp,mc}$ [kW] and $\dot{W}_{pmp,mc}$ [kW] are the mechanical power of the expander and pump respectively. Mechanical losses are taken into account evaluating the mechanical efficiency of the expander $\eta_{exp,mc}$:

$$\eta_{exp,mc} = \frac{\dot{W}_{exp,mc}}{\dot{W}_{exp,ind}} \tag{5}$$

where $\dot{W}_{exp,mc}$ [kW] is the expander mechanical power, calculated from torque and angular speed (both directly measured) and $\dot{W}_{exp,ind}$ [kW] is the indicated power, calculated from the area enclosed by the pressure-volume curve. The indicated cycle is defined by averaging several consecutive cycles, whose reconstruction is based on piezoelectric transducers angular position on expander cover plate and pressure levels at inlet and outlet ports.

Finally, an exergy analysis is carried out in order to evaluate the irreversible losses produced by each component. The overall system exergy loss, ΔEx [kW], is calculated as:

$$\Delta Ex = T_0\, \Delta S_{tot} = T_0\, \Delta S_{exp} + T_0\, \Delta S_{pmp} + \sum T_0\, \Delta S_{hx} \tag{6}$$

where T_0 [K] is the reference temperature, equal to 15 °C and ΔS [kW/K] is the entropy variation calculated over each component (expander, pump and heat exchangers) as follows:

$$\Delta S_{exp} = \dot{m}_{WF}\left(s_{exp,out} - s_{exp,in}\right) \tag{7}$$

$$\Delta S_{pmp} = \dot{m}_{WF}\left(s_{pmp,out} - s_{pmp,in}\right) \tag{8}$$

$$\Delta S_{hx} = \Delta S_{WF} + \Delta S_{ths} \tag{9}$$

Concerning the heat exchangers, the exergy loss is calculated as the sum of two terms. The first, ΔS_{WF} [kW/K], is related to the working fluid and is calculated as:

$$\Delta S_{WF} = \dot{m}_{WF}\left(s_{hx,out} - s_{hx,in}\right) \tag{10}$$

The second, ΔS_{ths} [kW/K], is related to the thermal source, depending on the heat exchanger considered. For HTHX and LTHX, the thermal sources are oil (hot source) and water (cold sink) respectively. The entropy variation is calculated as:

$$\Delta S_{ths} = \dot{m}\, c\, ln\left(\frac{T_{hx,out}}{T_{hx,in}}\right) \tag{11}$$

For the recuperator, the exergy loss is calculated according to Equation 10.

In order to evaluate the effectiveness of the thermodynamic process, an ideal cycle with finite capacity heat source is considered. The ideal efficiency η_{id} can be calculated as:

$$\eta_{id} = 1 - \frac{T_0}{\dfrac{\left(T_{oil,HTHX,in} - T_{oil,HTHX,out}\right)}{\ln\left(\dfrac{T_{oil,HTHX,in}}{T_{oil,HTHX,out}}\right)}} \tag{12}$$

Thus, the cycle exergy efficiency η_{exe} results from the comparison between the actual cycle efficiency η (Equation (4)) and the ideal one η_{id} (Equation (12)):

$$\eta_{exe} = \frac{\eta}{\eta_{id}} \tag{13}$$

3. Results and discussion

The experimental campaign consists in the performance evaluation of the cycles. Different steady state conditions are achieved by varying the pump speed, so that the measurements and the data post-processing can be carried. A comparison of the cycle main parameters at the best working conditions is reported in Table 3.

Table 3. Main parameters for simple and recuperative cycles

Cycle parameter	Symbol	Units	Simple	Recuperative
Measured				
Pump inlet pressure	p_1	bar	3.4	3.76
Pump inlet temperature	T_1	°C	19.3	14.6
Pump outlet pressure	p_2	bar	10.6	13.0
HTHX outlet temperature	T_3 , T_4	°C	85.2	81.4
Pump mechanical power	$\dot{W}_{pmp,mc}$	kW	1.10	0.65
Expander mechanical power	$\dot{W}_{exp,mc}$	kW	3.23	3.66
Calculated				
Working fluid mass flow rate	$\dot{m}_{WF}$	kg/s	0.295	0.394
HTHX thermal input	$\dot{Q}_{HTHX}$	kW	57.25	60.78
Net cycle power	$\dot{W}_{net,mc}$	kW	2.13	3.01
Expander indicated power	$\dot{W}_{exp,ind}$	kW	4.50	4.49
Expander mechanical efficiency	$\eta_{exp,mc}$	%	71.8	81.5
Net cycle efficiency	η	%	3.72	4.96
Exergy efficiency	η_{exe}	%	19.5	23.4

Both the systems operate in similar thermal input conditions and with comparable maximum cycle temperatures. The recuperative cycle displays better performance in terms of power production and cycle efficiency.

The higher pressure in the recuperative cycle is due to a slightly higher temperature of the heat source. Consequently, the working fluid temperature and pressure raises in the expander inlet of the recuperative cycle, which implicates a greater density in comparison to the simple cycle. The greater density leads to a higher mass flow rate, despite the expander displacement in recuperative cycle is lower than the one in simple cycle (see Table 1).

Focusing on the two expanders, they both have similar indicated powers. The recuperative cycle expander operates with higher inlet pressure and mass flow rate leading to a greater mechanical power. Moreover, the recuperative cycle pump operates in conditions closer to the design ones, allowing for a lower power consumption. Consequently, the overall cycle efficiency is better for the recuperative cycle.

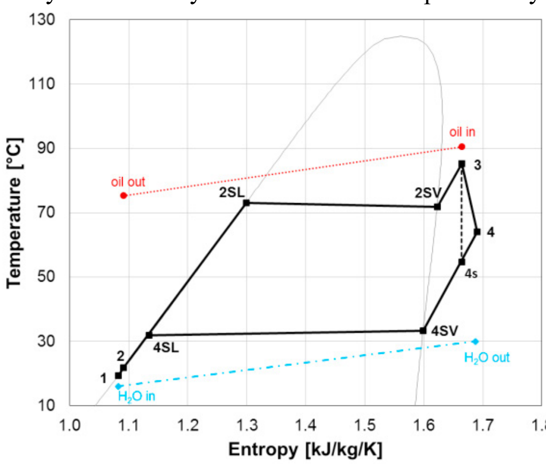

Figure 4. Simple cycle T-s diagram

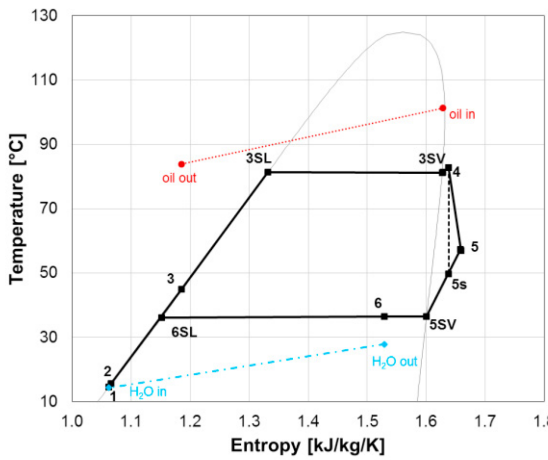

Figure 5. Recuperative cycle T-s diagram

In Figure 4 and Figure 5 the temperature-entropy diagrams are shown. The simple cycle presents a greater superheating at the expander inlet, while the recuperative is about in saturated vapor condition. Moreover, the expansion curve slope is lower in the recuperative cycle resulting in a better expansion process. It can be noted that a large sub-cooling occurs in the recuperative cycle. This phenomenon is probably due to an excessive quantity of working fluid, so the condenser turns to be flooded in an appreciable portion.

Figure 6 shows the pressure evolution inside the expander, considering the trailing vane angular position as reference. The diagrams are plotted using the piezoelectric sensors measurements that cover the entire expansion process as well as part of the suction and discharge (the vertical lines individuates the suction port closing and the discharge port opening). The pressure oscillations are likely due to pressure waves within the expansion chamber. By means of the volume evolution inside the expanders, determined from the geometrical features of the machines, it is possible to calculate the expansion indicated power $\dot{W}_{ind}$ [kW] as the area enclosed by the pressure-volume curve and consequently the mechanical efficiency of the expanders $\eta_{exp,mc}$ via Equation (5). Results of this analysis are shown in Table 3, where a higher expander mechanical efficiency can be noted in recuperative cycle (81.5%) compared with the simple cycle (71.8%). The Figure 6 shows also that an over-expansion happens inside the simple cycle expander. In fact, in the last part of the process, the expander outlet pressure is lower than the system pressure, affecting negatively the expander performance and leading to a mechanical efficiency reduction.

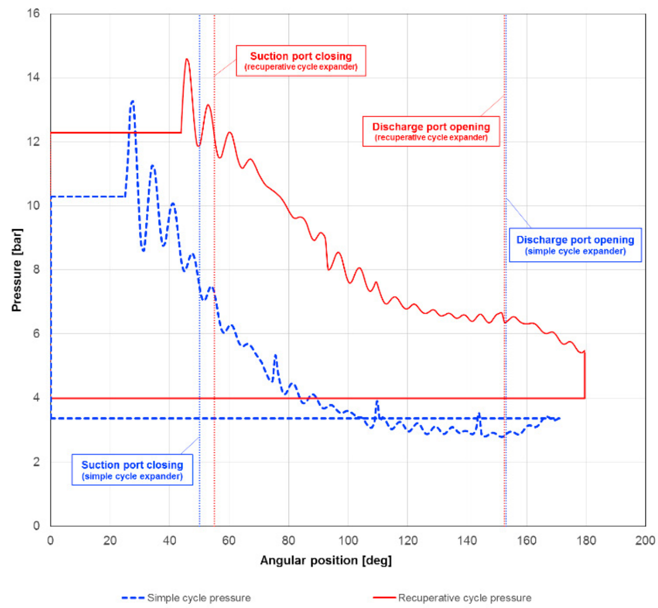

Figure 6. Pressure-angle diagram for simple and recuperative cycles

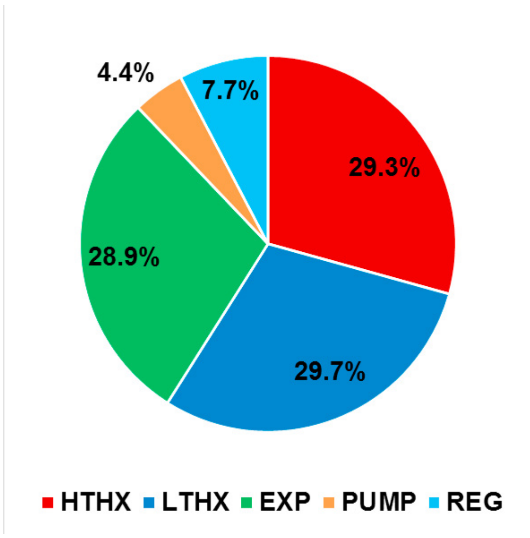

Figure 7. Rate of each component to the total exergy loss for recuperative cycle

The exergy analysis lets to appreciate the energy recovery effectiveness while using such a low-grade thermal source. Whenever the heat source has a low temperature, the possibility of exploiting it is limited. It is possible to quantify it, using the second law of thermodynamics, through the exergy efficiency. For the simple cycle the exergy efficiency is 19.5% and for the recuperative one results 23.4%. A deeper analysis is carried out on the recuperative cycle due to its highest performance. Figure 7 highlights the impact of each ORC component on the total exergy loss. The main contributions are given by the LTHX, the HTHX and the expander. This approach gives a hint on the components that could be optimized to enhance the energy recovery.

4. Conclusions

An experimental study is carried out on two ORC recovery systems equipped with rotary vane expanders. They are respectively in simple and recuperative configurations and are coupled to the same thermal source consisting in the hot lubricant of a mid-size air compressor. They both use R236fa as working fluid. The evaluation of the systems overall performance is made on a wide range of operating conditions. This work draws conclusions as follows:

- the recuperative cycle allows for higher power production (3.01 kW) and net cycle efficiency (4.96%) in comparison to the simple cycle (2.13 kW and 3.72%);
- the greater performance is due to the better expansion process, as confirmed by the higher expander mechanical efficiency: +9.7 percentage points for the recuperative cycle expander (81.5%) in respect to the simple cycle expander (71.8%);
- the exergy analysis gives indications on the components that major affect the performance, which are the expander (28.9%), and the heat exchangers: LTHX (29.7%) and HTHX (29.3%).

In brief, despite the low-temperature source, both the cycles appear to be promising solutions in a wide range of operating conditions. The recuperative cycle seems to be better in terms of overall performance and expansion process. Future works will focus on system components optimization with particular attention on the heat exchangers size.

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