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Energy Procedia 101 (2016) 814 – 821

Energy Procedia

71st Conference of the Italian Thermal Machines Engineering Association, ATI2016, 14-16 September 2016, Turin, Italy

A Micro-ORC Energy System: Preliminary Performance and Test Bench Development

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Abstract

A large market potential for small electricity and heat generators can be identified in the domestic sector. Among the under development micro-scale power generation technologies the ORC (Organic Rankine Cycle) concept is a promising solution, already proven in the MW-range of power. There is still a prospective for smaller units for domestic users, with low temperature thermal demand.

A test bench for a micro-CHP unit, currently run with a prototypal prime mover, is under development at University of Bologna. In particular, the system in study in the test facility is a micro-ORC system, rated for up to 3 kW. The ORC input heat is provided from an external source, which can be an external combustion system (a 46 kW biomass boiler will be connected to the thermal cycle) or an electric heater. The heat source delivers hot water to the bottoming ORC, currently operated with R134a as working fluid, which evolves in a recuperated cycle, with a 3-piston reciprocating expander, producing mechanical/electric power. The residual low-value heat is discharged to the environment with a water cooled condenser. The hot and cold water circuits have been realized in the lab to test the ORC performance.

The micro-ORC internal layout and the external hot and cold water lines have been instrumented, implementing an acquisition and control software by means of LabVIEW software. A preliminary test campaign has been performed on the micro-ORC system, obtaining information on the actual thermodynamic cycle and the real performance under different operating conditions.

Peer-review under responsibility of the Scientific Committee of ATI 2016.

Keywords: ORC; test bench; thermodynamic measurements; system performance.

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Peer-review under responsibility of the Scientific Committee of ATI 2016.
doi:10.1016/j.egypro.2016.11.103

1. Introduction

In the residential energy sector, a large market potential can be identified for external combustion energy systems [1], able to exploit renewable fuels (biomass) for producing heat and also residual electricity, as secondary valuable product. Among the external combustion electricity generation technologies in small scale applications, the ORC technology is expected to provide higher values of energy saving and environmental performance [2 – 4].

A recent review on the ORC technology for low-grade heat conversion is presented by Tchanche et al. [5], showing benefits of ORC for biomass and other applications. In the medium power size (100 – 1500 kW) the ORC technology has been demonstrated with several plants in operation around the world [5], while in the micro size range only few machines (e.g., by manufactures such as GMK, Electratherm, Infinity Turbine, Verdicorp, Zuccato Energia, etc.) and prototypes are under development/experimentation in different research institutes (see for example [6 – 11]). The key component of the micro-ORC is the expander, which is often suggested for these prototypes with a volumetric machine design: piston, scroll or screw expanders have been proposed, while axial and radial turbines are more common for mid-large size ORC systems. Micro/nano size ORC, with simple expander design, with common organic fluids and with simple or recuperated thermodynamic cycle, could be used to recover residual low-grade heat, such as hot water or stack gases, available at temperature below 100 °C. The possibility to recover small amount of low-grade waste heat in order to produce residual electricity is here investigated with an experimental test bench of a micro-ORC, based on a volumetric expander.

Nomenclature

FS	full scale	Q	exchanged heat
h	enthalpy	R	Reading
ORC	Organic Rankine Cycle	T	Temperature
P	power		
p	pressure	η_{th}	efficiency

2. Aim of the study

The general aim of the study is a preliminary investigation of a micro-ORC energy system, showing data collected in the experimental test campaign. In particular, this paper presents both the test bench developed in the laboratories of the University of Bologna for the characterization of micro-ORC and the first obtained results in terms of thermodynamic performance and main components characteristics.

3. Experimental test facility

The layout of the test facility developed is shown in Figs 1 and 2. More in details, Fig. 1 presents the micro-ORC internal layout, while Fig. 2 shows the external circuits, realized in the laboratory in order to test the ORC. The ORC main internal components have been originally provided by Newcomen Company, while the instrumentation and the ORC external circuits of the test facility have been implemented for this study.

The ORC system is based on a recuperative configuration and it currently operates with R134a as working fluid. The ORC main components shown in Fig. 1 are described below:

- the evaporator: a brazed plate heat exchanger with 64 plates (model ONDA S 202); it recovers heat from a hot water source to produce superheated organic fluid vapor;
- the expander: a volumetric three pistons radial expander. The pistons are located at 120° and they work alternately to each other. This configuration allows to avoid mechanical block problems;
- the recuperator: a plate heat exchanger with 16 elements. It recovers residual heat from the expander outlet stream to preheat the liquid prior to the evaporator inlet, in order to improve the overall cycle efficiency;
- the condenser: a shell and tube heat exchanger (model ONDA CT 292); it uses cold water extracted from a well;

- the organic fluid recirculation pump: a volumetric gear pump, normally operated with variable speed, by means of an inverter, in order to control the mass flow of R134a flowing through the cycle.

Figure 1 also shows the presence of several manual valves (VM) installed for inspection and in case of breakage; a normally closed valve (VNC1) and normally open valves (VNO2, VNO2') are also installed for the ORC startup, when the fluid flows through the external casing of the expander (branch 1-2'), by-passing the expander cylinders (branch 1-2). This procedure allows to increase the temperature of the heat exchangers surfaces, avoiding thermal energy dissipation trough the expander, before the ORC is activated. When the desired temperature conditions are reached at the expander inlet, the VNC1 valve can be opened, in order to admit the fluid inside the expander. Moreover, a safety valve is installed at the outlet of the evaporator to avoid unexpected overpressure (max pressure is set at 26 bar). Finally, a puffer tank is placed in the cycle (between point 6 and 7 in Fig. 1).

The expander is directly connected with an electric AC generator. The considered system is rated for an electrical power output up to 3 kW, assuming an ORC efficiency close to 10 %, as declared by the manufacturer, and a nominal input thermal power around 30 kW.

The electric load implemented in this study is a variable resistive load, able to dissipate the produced electric power, with different set points up to 3 kW, with steps of 600 W.

The hot and cold water supply lines are shown in Fig. 2. The heat input is provided to the ORC by a biomass boiler producing hot water, supplied to the ORC evaporator via an intermediate hot water storage tank (whose capacity is equal to 500 l). This tank is also equipped with four electric resistances (R1-R4 in Fig. 2, each one absorbing an electric power equal to 8 kW), useful to compensate the boiler load variations or, if necessary, to

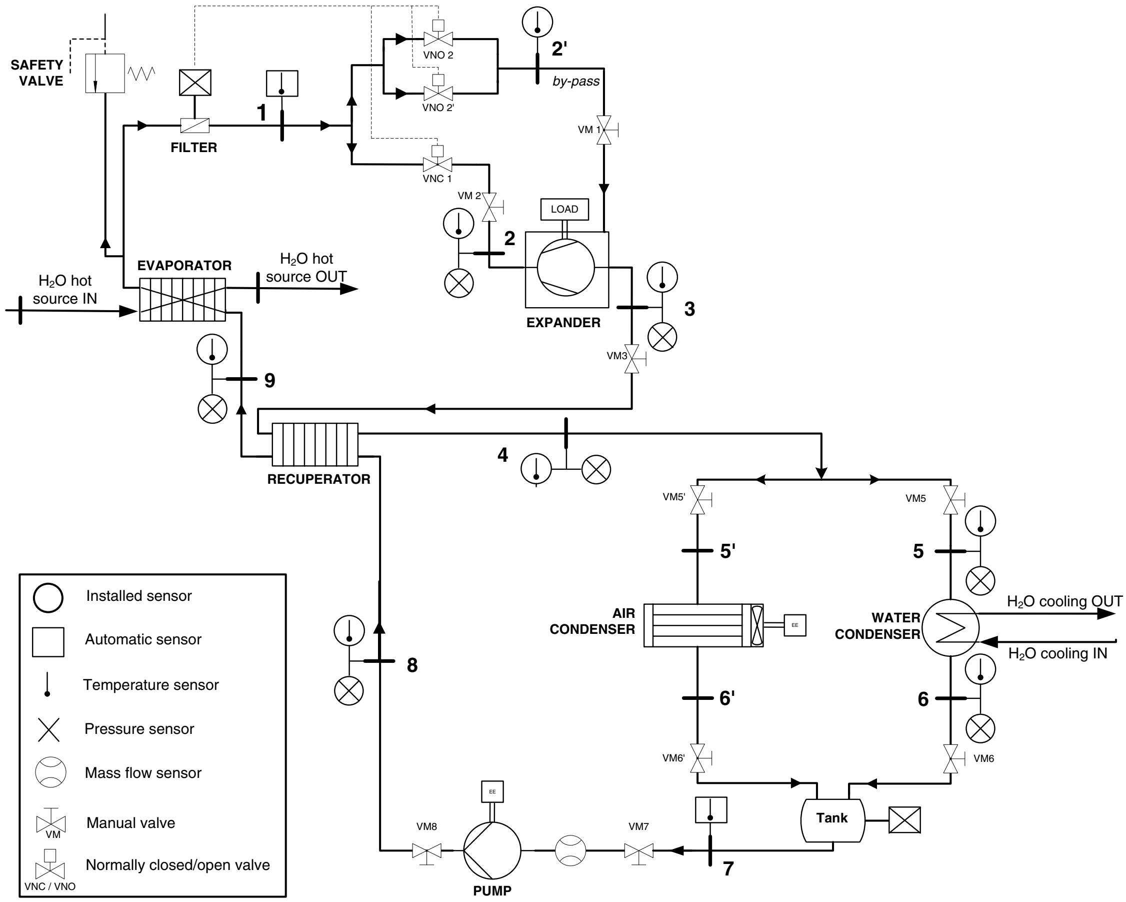


Fig. 1. The micro-ORC internal layout

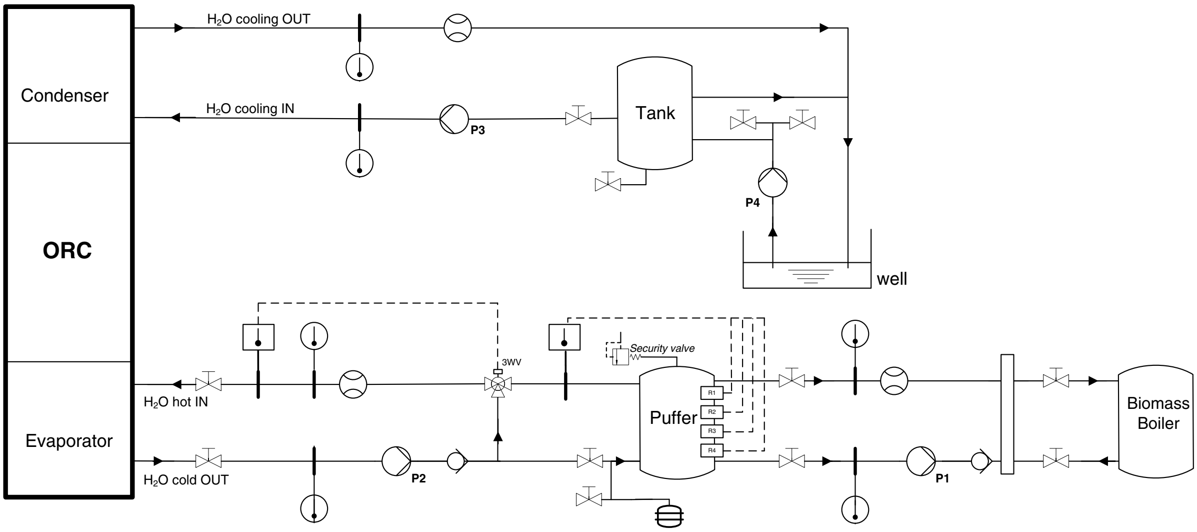


Fig. 2. Hot and cold water supply lines

completely replace the boiler thermal output. The water temperature at the ORC evaporator inlet can be controlled through a motor-driven 3-way valve (3WV). The valve control loop allows to decrease the evaporator inlet water temperature, by mixing the evaporator inlet with the return water. A circulation pump (P2) is installed in the return line. The cold water is provided by a well available in the lab; a tank is installed in the line to compensate the well water variations. Two pumps (P4 and P3) are placed in the cold water circuit. All the pumps (Grundfos Magna1 40-180F model) are operated at constant speed.

4. The test bench instrumentation

In order to collect data on the operation of the system, the test bench has been instrumented with temperature and pressure sensors. The measurements of temperature are made through 8 T-type and 6 K-type thermocouples, respectively placed in the ORC and in the water cycle. Thermocouples are individually calibrated in a thermostatic bath against a reference temperature sensor at the laboratories of the University of Bologna. Thus, a first order linear calibration curve is obtained in the range (288 – 350) K. The accuracy of these sensors is estimated as equal to ± 0.5 K.

The organic fluid pressure is measured by ceramics pressure transducers (Honeywell FP2000 model) with a total error band equal to ± 0.25 % FS. Pressure transducers have different characteristics according to the high and low pressure branches (see Table 1). The pressure taps are flush mounted (thus measuring the static pressure) at the inlet and outlet of each ORC component: this arrangement allows to characterize the components, evaluating for instance the physical states, the pressure drops and the thermal exchanges.

Regarding flow sensors, on the water side it is measured by an electromagnetic volumetric flow sensor model Endress+Hauser Promag 50P, placed in the evaporator water inlet branch. The output signal is (4 – 20) mA. The measurable flow values range is set between (0 – 10) l/s. Meanwhile, the ORC mass flow rate measurer is a Coriolis unit (Endress+Hauser Promass 80E model), located downstream the working fluid tank and upstream the gear pump. This placement allows (i) to fill always the tube, avoiding measurement problems, (ii) the reduction of the induced-vibration due to the gear pump operation and (iii) the correct placement in terms of straight pipe lengths. Also in this case, the signal output is a (4 – 20) mA current output, corresponding to (0 – 1000) g/s. This sensor can measure three parameters at the same time. The other available parameters are the density and the temperature. Off-the-shelf sensor accuracy is equal to ± 0.2 % R for the mass flow rate, while it is equal to ± 0.02 g/cm^3 and ± 0.5 K for the density and temperature respectively. The specific data of the sensors installed in the ORC system are summarized in Table 1.

The test bench is also endowed with a data acquisition system. A real-time microcontroller, namely a National

Table 1. ORC sensors main data

	Pressure transducers		T-type thermocouples	ORC mass flow sensor
	High Pressure	Low Pressure		
Layout point (Fig.1)	2, 8, 9	3, 4, 5, 6	2, 2', 8, 9, 3, 4, 5, 6	7
Output signal	0 – 5 Vdc		± 80 mV	4 – 20 mA
Measurement range	(0 – 30) bar	(0 – 10) bar	(-270°C – +400) °C	(0 – 1000) g/s

Instrument Compact RIO system [12], has been used to acquire signals from the installed sensors. In particular, several NI 9211 input modules read the voltage thermocouples signals, while a NI 9207, an analog voltage and current input module elaborates the signals from the flow and pressure sensors. An acquisition software and a display panel have been developed in the LabVIEW environment. Data have been acquired with a time step equal to 1 second and, the fluid library properties CoolProp [13] has been integrated in the developed acquisition software, in order to obtain the full thermodynamic state and evaluate the system performances. More in details, the temperature and pressure values allow to identify the physical state of each ORC point. Therefore, the evaluated state are used to instantly display the ORC thermodynamic cycle on temperature/entropy and pressure/enthalpy diagrams.

5. Performed tests

The aim of the experimental test carried out at our laboratory is a preliminary investigation on the ORC thermodynamic performances in different operating conditions. The test campaign has been executed varying the pump rotational speed with an invariable thermal power from the hot sources. More in details, the 3WV position in the hot water line is controlled to maintain the average temperature value on the set value.

The thermodynamic performances of the ORC are evaluated using the equations listed in Table 2. P_{EXP} and P_{PUMP} are respectively the thermodynamic power produced by the expander and the hydraulic power of the pump; P_{NET} is the difference between the expander and pump power; Q_{EVAP} is the heat recovered by the ORC fluid from the hot source in the evaporator, while Q_{COND} represents the heat exchanged with the cold water in the condenser. The ORC thermodynamic gross and net efficiency have been calculated in a simplified way according to Eqs 6 and 7. The thermodynamic enthalpy values have been obtained by means of the CoolProp library [13].

An experimental campaign of approximately 4 hours is considered in this paper, in which different conditions have been tested. For each operating condition, a limited time window is considered, characterized by: (i) constant organic fluid mass flow and (ii) a quite constant P_{EXP} value. The resulting average values of the measured quantities and the corresponding calculated performance are shown and analyzed in the following section.

6. Results

Table 3 reports the mass flow, temperature and pressure values of 20 ORC operating points, named A-T. The data have been obtained as average of the punctual values of mass flow, temperature and pressures in several time intervals. The time periods, with time window span not less than one minute, have been chosen based on constantly

Table 2. Performance equations

Expander thermodynamic power [kW]	$P_{EXP} = \dot{m}_{ORC} \cdot \Delta h_{EXP} = \dot{m}_{ORC} \cdot (h_3 - h_2)$	(1)
Pump hydraulic power [kW]	$P_{PUMP} = \frac{\dot{m}_{ORC} \cdot \Delta p_{PUMP}}{\rho_{IN}} = \frac{\dot{m}_{ORC} \cdot (p_8 - p_6)}{\rho_7}$	(2)
Net power [kW]	$P_{NET} = P_{EXP} - P_{PUMP}$	(3)
Thermal power exchanged in the evaporator [kW]	$Q_{EVAP} = \dot{m}_{ORC} \cdot \Delta h_{EVAP} = \dot{m}_{ORC} \cdot (h_2 - h_9)$	(4)
Thermal power exchanged in the condenser [kW]	$Q_{COND} = \dot{m}_{ORC} \cdot \Delta h_{COND} = \dot{m}_{ORC} \cdot (h_5 - h_6)$	(5)
ORC gross efficiency [%]	$\eta_{thEXP} = \frac{P_{EXP}}{Q_{EVAP}}$	(6)
ORC net efficiency [%]	$\eta_{thORC} = \frac{P_{EXP} - P_{PUMP}}{Q_{EVAP}}$	(7)

Table 3. Temperature and pressure average values in the ORC system sections for the considered time intervals

	$\dot{m}_{ORC}$ [g/s]	T_2 [°C]	p_2 [bar]	T_3 [°C]	p_3 [bar]	T_4 [°C]	p_4 [bar]	T_5 [°C]	p_5 [bar]	T_6 [°C]	p_6 [bar]	T_8 [°C]	p_8 [bar]	T_9 [°C]	p_9 [bar]
A	55.97	63.5	11.96	54.1	8.68	34.1	8.67	34.1	8.66	28.1	8.65	28.1	12.06	28.1	12.04
B	59.55	62.0	12.46	53.5	8.80	34.5	8.80	34.6	8.80	29.1	8.80	30.7	12.52	31.6	12.51
C	60.25	64.1	11.83	54.1	9.03	35.6	9.02	35.5	9.00	29.9	9.00	29.2	11.93	29.2	11.90
D	64.99	62.7	11.97	49.5	8.80	34.8	8.79	34.6	8.77	29.0	8.76	27.9	12.07	27.9	12.04
E	68.53	64.8	12.15	51.3	8.85	34.9	8.83	34.8	8.81	29.3	8.81	28.0	12.25	28.0	12.22
F	72.41	63.14	12.40	52.7	9.00	35.6	8.98	35.5	8.96	30.0	8.96	28.6	12.50	28.6	12.48
G	76.67	65.2	12.67	53.5	9.09	35.8	9.06	35.8	9.05	30.4	9.04	29.1	12.77	29.1	12.75
H	77.27	62.5	13.19	52.9	8.88	34.8	8.85	34.7	8.84	29.2	8.83	28.0	13.25	28.0	13.24
I	77.39	63.2	13.85	51.8	8.95	35.1	8.92	35.0	8.91	29.4	8.91	29.0	13.90	29.8	13.90
J	80.53	64.6	13.62	52.6	9.11	35.8	9.08	35.7	9.07	30.3	9.07	29.0	13.68	29.0	13.68
K	84.67	64.7	13.94	52.8	9.28	36.5	9.25	36.4	9.24	31.2	9.24	30.0	14.0	30.0	14.00
L	85.24	63.1	13.13	53.7	9.18	36.0	9.15	35.9	9.14	30.6	9.13	29.5	13.23	29.5	13.20
M	87.87	62.9	14.17	51.9	9.33	36.6	9.29	36.5	9.28	31.3	9.28	30.3	14.23	30.3	14.23
N	89.00	65.4	14.70	51.4	9.18	36.0	9.14	35.9	9.13	30.5	9.12	29.4	14.75	30.0	14.75
O	92.23	64.8	14.41	51.9	9.37	36.7	9.33	36.6	9.32	31.4	9.31	30.4	14.47	30.5	14.47
P	93.10	64.0	15.06	51.1	9.47	37.1	9.43	37.0	9.42	32.0	9.41	30.7	15.12	31.1	15.12
Q	96.38	63.7	15.36	50.2	9.56	37.5	9.52	37.4	9.51	32.3	9.50	31.2	15.41	31.8	15.41
R	97.83	65.3	14.70	52.2	9.42	36.8	9.37	36.7	9.36	31.6	9.36	30.7	14.75	31.0	14.75
S	100.1	65.4	15.60	50.2	9.60	37.6	9.55	37.5	9.54	32.4	9.54	31.5	15.66	32.0	15.67
T	113.0	66.1	16.27	50.1	9.71	37.9	9.64	37.8	9.63	32.8	9.62	31.8	16.33	32.2	16.33

of mass flow and expander thermodynamic power. The operating pointe A-T in Table 3 are listed for increasing values of $\dot{m}_{ORC}$, obtained by varying the ORC pump rotational speed.

An estimation of typical measurement uncertainty for the reported parameters values are the follows:

- the Coriolis mass flow meter was individually calibrated and achieved measurement uncertainty is ± 0.15 %;
- the temperature uncertainty is equal to ± 0.9 °C;
- while a ± 0.4 % estimated uncertainly is considered for the pressure data.

Figure 3 shows the calculated thermodynamic power produced by the expander and required by the pump (evaluated according to Eq. 1 and Eq. 2 in Table 2) plotted versus the ORC mass flow (a) and versus the expander

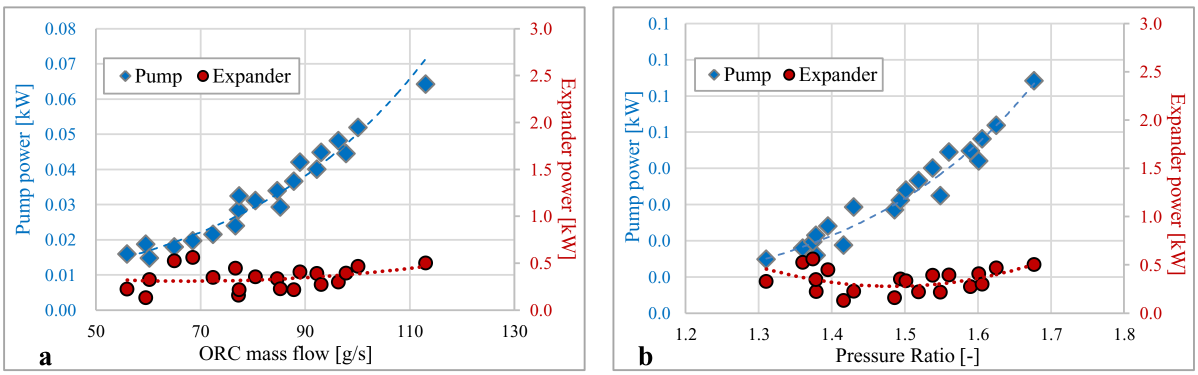


Fig. 3. Pump and expander thermodynamic power trend as function of ORC mass flow (a) and pressure ratio (b).

pressure ratio (b): increasing trends are obtained in both cases. The maximum obtained power at the expander is equal to 0.56 kW with an expander pressure ratio close to 1.4 and an ORC mass flow rate equal to around 68.53 g/s (interval E in Table 3).

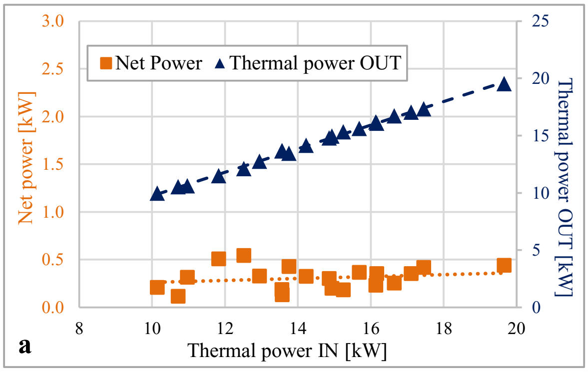
Figure 4(a) shows the net thermodynamic output power P_{NET} and the thermal power output at the condenser Q_{COND} versus the thermal power exchanged in the evaporator Q_{EVAP}; Fig. 4(b) shows the hot and cold source temperature values and the ORC evaporation (T_2) and condensation (T_5) averaged temperature values in the considered time intervals. The hot water temperature during the test campaign was constantly close to 70 °C, while the condenser cold water temperature was close to (25 − 27) °C (a slight increase occurred during the test).

Figure 5 shows the actual temperature-entropy diagram of the analysed cases A and T. In case the pressure ratio results higher than in case A, but the organic fluid superheating at the expander inlet is lower. However, the maximum temperature of the cycle (T_2) is quite similar in both cases, as shown also in Fig. 4(b). Finally, Fig. 6 presents the gross and net efficiency values obtained in the experimental campaign, evaluated as shown in Table 2. The resulting thermodynamic cycle efficiency values are in the range 1 % − 4.5 %. The quite limited performance are due to a relatively high condensation pressure (and temperature), due to the available cold source, which has been used in this test campaign.

7. Conclusions

An experimental test facility has been developed to fully characterize the energy performance of a micro-ORC unit. All the ORC internal temperature, pressure and mass flow values are measured. Moreover, the hot and cold source external circuits operating parameters are measured.

Preliminary experimental results on thermodynamic performance are shown in this paper related to different



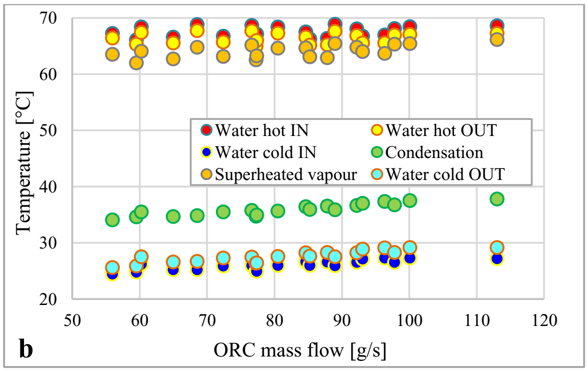


Fig. 4. (a) ORC net power and thermal power exchanged at the condenser as function of the input heat at the evaporator; (b) evaporation and condensation temperature and hot and cold sources temperatures versus ORC mass flow rate.

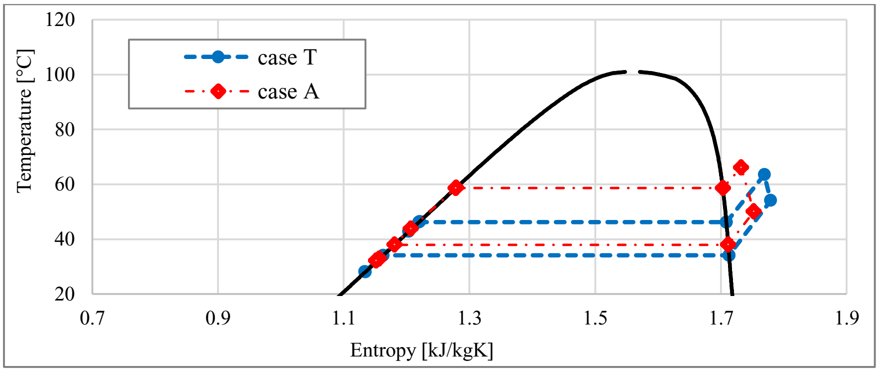


Fig. 5. The obtained ORC cycle in case A and T plotted on the temperature/entropy thermodynamic diagram.

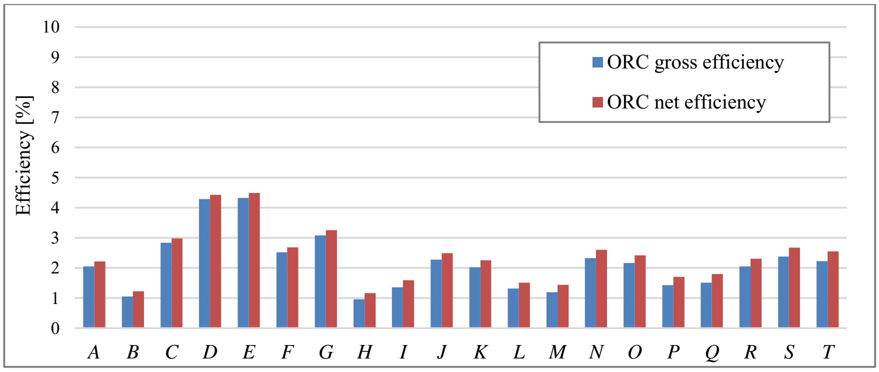


Fig. 6. ORC gross and net average efficiency values for the analysed time intervals (A-T).

operating conditions in terms of ORC mass flow rate, while quite constant hot and cold source conditions are considered. These preliminary results show positive thermodynamic performance. Nevertheless, the system has been tested in part load conditions, with the cold source at relatively high temperature conditions and with reduced net power output: the maximum measured values of the output power are less than 1 kW and efficiency is close to 4 %.

Future tests of the ORC expander will be carried out with increasing thermal power input and by increasing the turbine inlet vapor thermodynamic conditions.

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