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Experimental investigation of a scroll unit used as a compressor and as an expander in a reversible Heat Pump/ORC unit

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ABSTRACT

On one hand, the use of solar energy for heating and electricity production should increase in the future to reduce greenhouse emissions. On the other hand, the use of water to water heat pumps allows the production of heat effectively in many applications. In this context, it is appropriate to study the concept of a reversible HP/ORC unit. This innovative system includes a water to water heat pump connected to a solar roof and a geothermal heat exchanger. This heat pump is also capable of reversing its cycle and work as an Organic Rankine Cycle. This heat is used primarily to cover the annual heating needs and excess heat generated during the summer is used to produce electricity. The key feature to achieve this is the reversibility of the compressor unit, i.e. its ability to work as an expander. This paper presents the experimental investigation on a first prototype of this HP/ORC reversible unit with a scroll compressor and refrigerant R134a. A global efficiency of 5.7% is reached in ORC mode with temperature of evaporation and condensation respectively of 88°C and 25°C. In heat pump mode, a COP of 4.2 is obtained with temperature of evaporation and condensation respectively of 21°C and 61°C. The scroll unit presents encouraging performance with isentropic efficiencies of 63% in expander mode and 77% in compressor mode.

1. INTRODUCTION

The paper is organized into 4 parts. Firstly, the introduction describes the context, the concept and the three operating modes. Then, the experimental setup is detailed. Thereafter, the results from experimentation are presented and analyzed. Finally, conclusions and perspectives are listed.

1.1 Context

The European Union needs to reduce emissions of greenhouse gas emissions by 20% compared to the level of 1990 by 2020, and in this context, the European Commission plans to increase the proportion of consumed renewable energy from 9% to 20%. Also, emissions of greenhouse gases must be reduced by 20% and energy efficiency will increase by 20%. Households represent 27 % of the final consumption (European commission, 2012). In this context, buildings whose net energy consumption is zero or positive (NZEB - Net Zero Energy Building (Jagemar et al, 2011)) are of considerable importance. From 2019, all new buildings should have a renewable energy production equal or higher than the primary energy consumption (European directive 2010). On the one hand, the International Energy Agency plans to cover more than 16% of the needs of low temperature heat from a solar heating (IEA, 2012). On the other hand, the use of water to water pumps heat allows the production of heat effectively in many applications (Hepbasli and Yalinci, 2009). In this context, it is appropriate to study the concept of a reversible HP/ORC unit. This innovative system includes a water to water heat pump connected to a solar absorber integrated into a roofing solution (Innogie, 2013) and a geothermal heat exchanger (Figure 1). This heat pump is also capable of reversing its cycle and work as an Organic Rankine Cycle (Innogie, 2013; Quoilin et al., 2013; Schimpf and Span, 2013). By using the heat from the solar roof, a large amount of heat is generated throughout the year. This heat is used primarily to cover the annual heating needs and excess heat generated during the summer is used to produce electricity. The main advantage of the proposed technology is due to the reversibility of scroll compressors, which have proven their high efficiency both in compressor and in expander modes during test campaigns (Kane et al, 2003, Lemort, 2008, Lemort et al, 2009, Lemort et al, 2011, Quoilin, 2011, Saitoh et al, 2007).

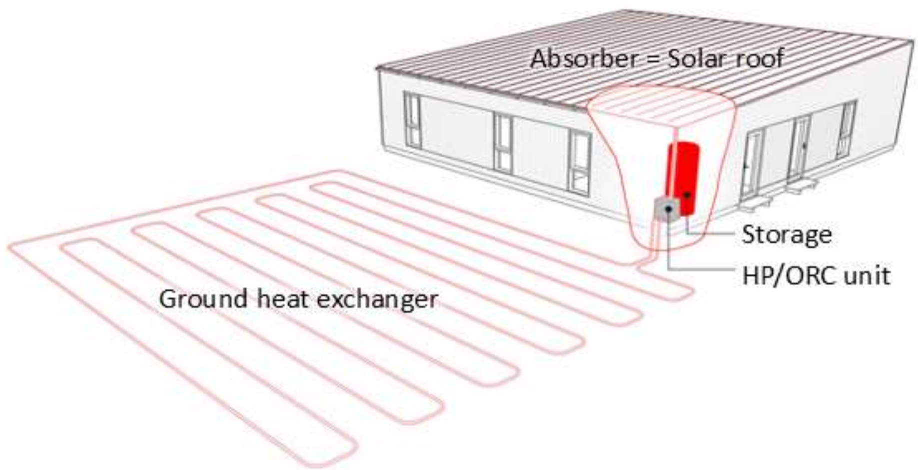

Figure 1 : HP/ORC reversible unit connected to a 120 m^2 solar roof and a horizontal ground heat exchanger (Innogie, 2012)

1.2 Operating modes

This unit is able to work following three different modes depending on the weather conditions and of the heat demand of the house:

1.2.1 Organic Rankine Cycle (ORC): When the heat demand of the building is covered by the storage and that the heat recovered by the roof is sufficient, the ORC mode is activated. The principle is detailed in Figure 2: Heat collected by the solar roof heats a glycol-water loop, which in turns evaporates the in the evaporator. Then this vapor is expanded into a scroll machine producing electricity. The refrigerant is then condensed by means of a glycol-water-cooled condenser connected to a ground heat exchanger. Finally the refrigerant is directed to the pump, which drives it to the high-pressure supply of the evaporator.

1.2.2 Direct heating (DH): If the heat demand from the house cannot be covered by the heat storage, then the direct heating mode is activated. In this mode, the heat from the roof is directly transmitted to the heat storage through an intermediate exchanger.

1.2.3 Heat pump (HP): If the direct heating mode is not sufficient to cover the heating demand of the house, the HP mode is activated. Here, the heat from the roof evaporates the working fluid in the evaporator. Then this vapor is compressed into the scroll machine. Following that, heat is released in the storage trough the condenser. Finally the refrigerant goes in the expansion valve and starts the cycle again.

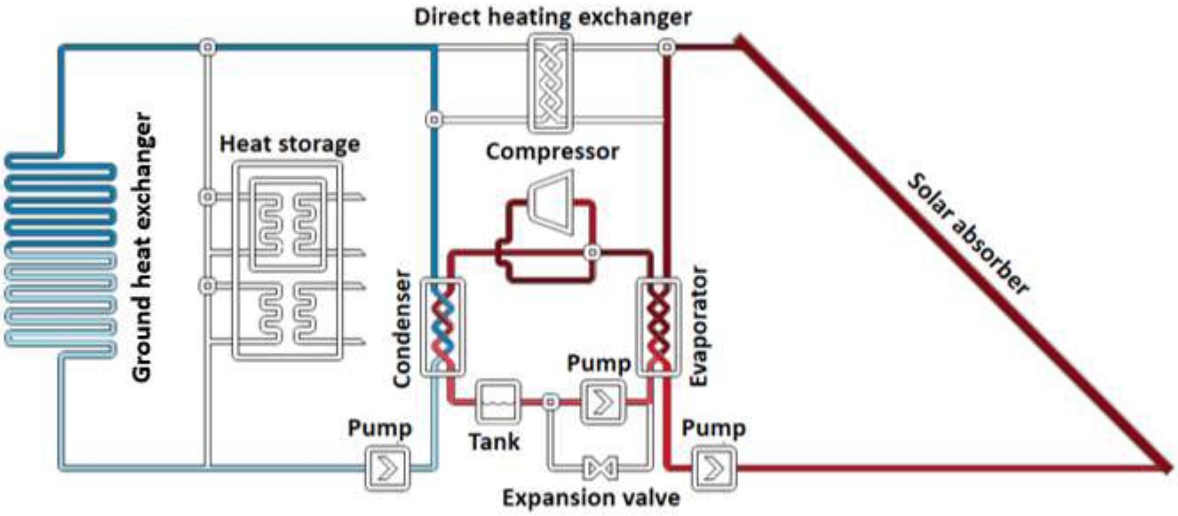

Figure 2: Simplified scheme of the reversible HP/ORC unit (ORC mode). Glycol-water is used in the ground heat exchanger, storage and solar absorber. Refrigerant R134a is used for the HP/ORC unit.

2. EXPERIMENTAL SETUP

2.1 Scheme

In order to verify the promising simulation results described in (Quoilin, 2013), an experimental study was performed on the HP/ORC prototype with refrigerant R134a . A scheme of the test rig is given in Figure 3 with the main elements (compressor, evaporator, condenser, pump and valves). The refrigerant loop (dark blue) also includes a reservoir and a subcooler used to provide a sufficient degree of subcooling at the inlet of the pump. This loop also includes a four way valve that allows switching between ORC and HP modes and a bypass valve necessary to start the ORC. The evaporator is supplied by a water loop (red) connected to an electrical boiler (150 kW). The condenser is cooled by tap water (light blue). An oil loop (black) is installed, but it is never used during the presented tests.

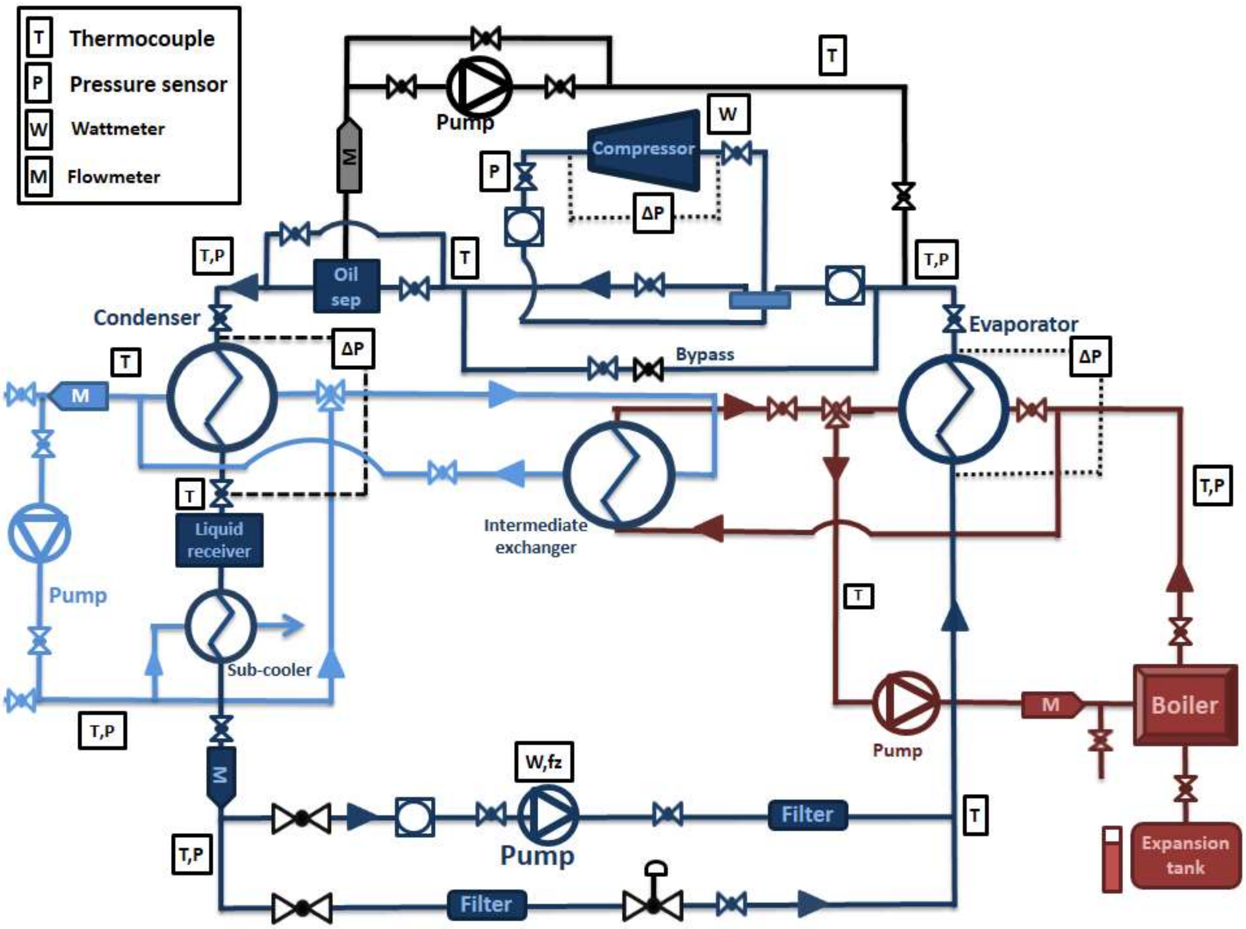

Figure 3: Detailed scheme of the experimental setup (dark blue = refrigerant loop, red = heating loop, light blue = cooling loop and black = oil loop)

2.2 Components

The design, modelling and technological choices have been presented in (Quoilin et al, 2013). Here is a summary of the components used in the unit.

A scroll unit is used to play the role of the expander (ORC) and of the compressor (HP) (Table 1). The selection of the compressor has been done by comparing the net electrical production (ORC production minus heat pump consumption) over a whole year (Quoilin et al, 2013). The optimum Scroll unit results in a trade-off between winter and summer conditions. Too small of a compressor leads to a maximum electrical production that limits the performance when large amounts of heat are generated in summer. Too large of a compressor leads to low electrical production at part load because it is working mainly at low pressure ratio (low isentropic efficiency).

Table 1: Technical data of the compressor

Parameter	Value
Displacement (cp mode)	82.6 cm^3
Built-in volume ratio	~ 2.8
Max pressure	32 bar
Motor power at max current	4 kW

Plate heat exchangers are used for the evaporator and compressor (Table 2).

Table 2: Technical data of the evaporator

Parameter	Evaporator	Condenser
Exchange area	4.08 m^2	3.12 m^2
Max. pressure	27 bar	27 bar
Max temperature	225°C	225°C

The pump is a volumetric one (plunger pump) and its rotational speed is controlled by an inverter (Table 3).

Table 3: Technical data of the pump

Parameter	Value
Maximum flow	15.1 l/min (1725 RPM)
Max. pressure	206 bar
Max temperature	121°C
Motor	1.5 kW

An electronic expansion valve is chosen because of its precise control allowing low over-heating (Table 4).

Table 4: Technical data of the expansion valve

Parameter	Value
Flow	Uniflow
Max. pressure	45 bar
Max temperature	100°C
Nominal power	13 kW

2.3 Sensors

Here are listed the different sensors and their technical data. Pressure sensor are described in Table 5.

Table 5: Pressure sensors technical data

Location	Range [bar]	Accuracy FS [%]
Condenser in	[0;25]	0.5
Condenser out	[0;25]	0.5
Evaporator out	[0;40]	0.5
Cold water	[0;6]	0.5
Hot water	[0;6]	0.5
Scroll su	[0;20]	0.05
Scroll ex	[0;40]	0.3
Difference condenser	[0;0,3]	0.5
Difference evaporator	[0;0,15]	0.5
Difference scroll	[0:20]	1

Flow sensors are described in Table 6. A Coriolis flowmeter is used to measure the refrigerant mass flow rate. Cold loop and hot loop water volume flow rates are measured manually with water counter meter.

Table 6: Flow sensors technical data

Location	Range	Accuracy [%]
Refrigerant	[0;300] g/s	0.2
Cold water	[0;3] m3/h	1
Hot water	[0;10] m3/h	1

All temperatures are measured by thermocouples type T. The measuring range is −200°C to 350 °C. Accuracy inferior to 1K is reached regarding the temperature range of this application.

Wattmeters used for pump and Compressor electrical consumption are presented in Table 7.

Table 7: Wattmeter's technical data

Location	Range [W]	Accuracy [%]
Pump	[-2000;2000]	0.5
Scroll	[-6000;6000]	0.5

3. EXPERIMENTAL RESULTS

3.1 Nominal points in HP and ORC mode

Performance of the unit and of its components is characterized by the following efficiencies (global efficiency of the ORC, Coefficient of performance for the heat pump, isentropic efficiency of the expander, isentropic efficiency of the compressor, filling factor and volumetric efficiency:

$$\eta_{global} = \frac{\dot{W}_{exp,el} - \dot{W}_{pp,el}}{\dot{Q}_{ev}} \tag{1}$$

$$COP = \frac{\dot{Q}_{cd}}{\dot{W}_{comp,el}} \tag{2}$$

$$\varepsilon_{exp,is} = \frac{\dot{W}_{exp,el}}{\dot{m}_r(h_{exp,su} - h_{exp,ex})} \tag{3}$$

$$\varepsilon_{comp,is} = \frac{\dot{m}_r(h_{comp,ex,s} - h_{comp,su})}{\dot{W}_{comp,el}} \tag{4}$$

$$\Phi_{exp} = \frac{\dot{V}_{exp,su}}{\dot{V}_{exp,th}} \tag{5}$$

$$\varepsilon_{comp,vol} = \frac{\dot{V}_{comp,su}}{\dot{V}_{comp,th}} \tag{6}$$

Table 8 presents a comparison between the theoretical point and the closest experimental points in HP and ORC mode. It shows the feasibility of the reversible HP/ORC unit with encouraging results in both modes. For the heat pump mode, experimental results are close to theory with decent compressor efficiency (Figure 4). The obtained results in ORC mode are slightly lower than expected. This is explained by low efficiency of the expander (Figure 5), high pressure drop (up to 4 bar) on the four way valve, non-thermally insulated pipes and additional sub-cooling needed for the non-cavitation of the pump in ORC mode. Higher efficiencies (up to 5.7%) are reached but can only be measured at part load where the refrigerant flow and four way valve pressure drop are lower.

Tableau 8: Comparison between Theory and experimentation on the nominal points

Mode	Parameter	Nomenclature	Theoretical nominal	Experimentation
ORC	Evaporator thermal power [kW]	$\dot{Q}_{ev}$	62	62
	Evaporation pressure [bar]	P_{ev}	33	32
	Condensation pressure [bar]	P_{cd}	7	10.3
	Mass flow rate [g/s]	$\dot{m}_r$	300	266
	Electrical production [kW]	$\dot{W}_{exp,el}$	4,7	3.7
	Global efficiency [%]	η_{global}	7.5	4.2
	Expander isentropic efficiency [%]	$\varepsilon_{exp,is}$	68	58
	Expander filling factor [-]	Φ_{exp}	1.019	1.12
HP	Condenser thermal power [kW]	$\dot{Q}_{cd}$	13	13.6
	Evaporation pressure [bar]	P_{ev}	5	5.7
	Condensation pressure [bar]	P_{cd}	17	17.3
	Mass flow rate [g/s]	$\dot{m}_r$	100	102
	Power consumption [kW]	$\dot{W}_{comp,el}$	4	3.8
	Coefficient of performance [-]	COP	4.2	4.21
	Compressor isentropic efficiency [%]	$\varepsilon_{comp,is}$	60	76
	Compressor volumetric efficiency [%]	$\varepsilon_{comp,vol}$	91	97

3.2 Compressor efficiency (HP)

Figure 4 shows the volumetric and the isentropic efficiencies of the compressor for different pressure ratios. These efficiencies show classical trends: volumetric efficiency is quite constant, decreasing slightly with the pressure ratio (more internal leakages) and isentropic efficiency raises rapidly up to its maximum value occurring at the optimum pressure ratio (around 3) and then decreasing slowly for higher pressure ratios. This performance are close to manufacturer prediction (difference<5%). Volumetric efficiency higher than unity is obtained because the oil fraction is not taken into account (Eq. 6). Uncertainty error bars are evaluated with a 10% oil fraction to show its influence on the performance. Further investigation needs to be done to determine the oil fraction accurately but this 10% oil fraction is evaluated thanks to the density measured by the Coriolis flowmeter . Uncertainty error propagation due to the sensor accuracy is negligible compared to the uncertainty due to of the oil fraction.

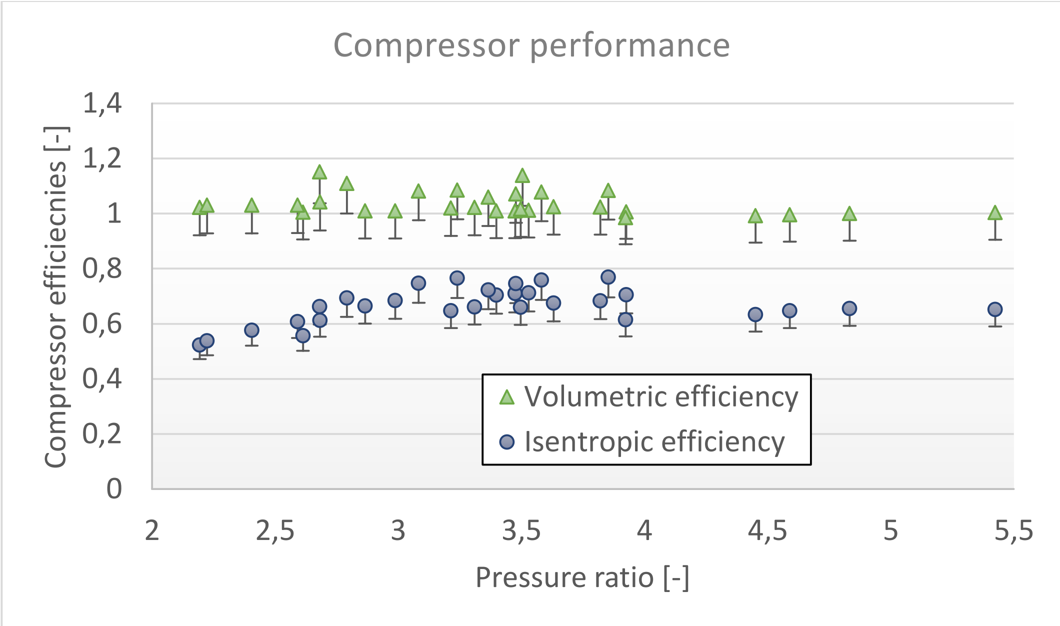

Figure 4: Compressor efficiencies versus pressure ratio

3.3 Expander efficiency (ORC)

Figures 5 presents the evolution of the expander efficiency with the pressure. Just like the heat pump, the trends are as expected but the efficiency is slightly lower than literature (Kane et al, 2003, Lemort, 2008, Lemort et al, 2009, Lemort et al, 2011, Quoilin, 2011, Saitoh et al, 2007). This is because the design is not optimized to work in both compressor and expander modes. The filling factor decreases with the pressure ratio, which is not intuitive because leakages are larger at higher pressure ratios. Oil fraction is not taken into account to evaluate the filling factor. A 10% oil fraction is considered to evaluate the uncertainty propagation.

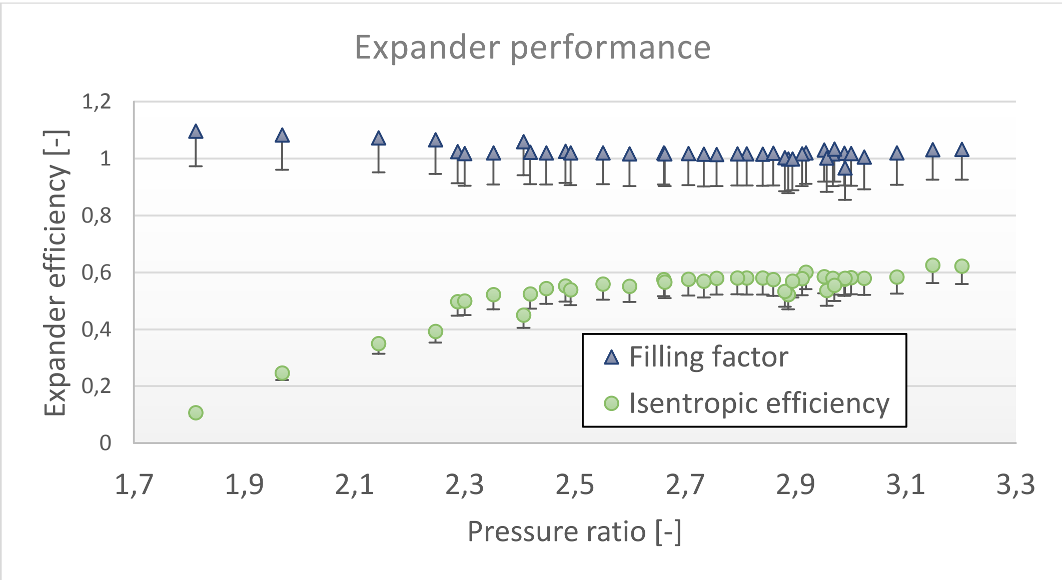

Figure 5: Expander efficiencies versus pressure ratio

4. CONCLUSION

The experimental investigation of a reversible HP/ORC reversible unit is presented. A global efficiency of 5.7% is reached in ORC mode with temperature of evaporation and condensation respectively of 88°C and 25°C. In heat pump mode, a COP of 4.2 is obtained with temperature of evaporation and condensation respectively of 21°C and 61°C. The scroll unit presents encouraging performance with isentropic efficiencies of 63% in expander mode and 77% in compressor mode. This demonstrates the feasibility of using a unique scroll unit as a compressor and as an expander with encouraging performance. Next steps are as follows:

- Validation of semi-empirical models based on measurements
- Utilization of these models to simulate the performance of the unit without its weaknesses and optimize its control variables (pump frequency, glycol-water flow…)
- Dynamic simulation of the unit coupled to the house, the storage, the solar roof and the ground heat exchanger.
- Building of a new unit closer to a commercial one (more efficient, more compact and cheaper). This will be done by (among others) using an optimal expander geometry and connecting the pump directly on the expander shaft.

NOMENCLATURE

COP	Coefficient of performance	(-)
h	Specific enthalpy	(J/(kg.K))
P	Pressure	(bar)
$\dot{Q}$	Thermal power	(kW)
$\dot{m}$	Mass flow rate	(g/s)
$\dot{V}$	Volumetric flow rate	(m^3/s)
$\dot{W}$	Power	(kW)
E	Efficiency	(-)
η	Global ORC efficiency	(-)
Φ	Filling factor	(-)

Subscript

cd	Condenser
comp	Compressor
is	Isentropic
el	Electrical
ev	Evaporator
ex	Exhaust
exp	Expander
r	Refrigerant
su	Supply
th	Theoretical
vol	Volumetric

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