Modelling of a thermally integrated Carnot battery using a reversible heat pump/organic Rankine cycle

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Abstract:
The growth of renewable energy requires flexible, low-cost and efficient electrical storage to balance the mismatch between energy supply and demand. The Carnot battery (or pumped thermal energy storage) converts electric energy to thermal energy with a heat pump when electricity production is greater than demand; when electricity demand outstrips production the Carnot battery generates power from two thermal storage reservoirs (Rankine mode). Classical Carnot batteries architectures do not achieve more than 60% roundtrip electric efficiency. However, innovative architectures, using waste heat recovery (thermally integrated Carnot batteries) are able to reach electrical power production of the power cycle larger than the electrical power consumption of the heat pump, increasing the value of the technology. It can be shown that the optimization of such a technology is a trade-off between the maximization of the power and the roundtrip efficiency ratio (depending on electricity prices among others). Therefore, a part-load model of reversible heat pump/organic Rankine cycle is developed. The model takes into account pressure drops in the piping, part load performance of exchangers and uses a semi empirical model for the volumetric machine. A set of simulation is performed in a wide range of inputs (waste heat temperature, storage temperature and mass flow rate) to analyse the part load performance of the system. Based on these results, several guidelines are proposed to optimally control the system in real conditions.

Keywords:

Carnot battery, Electrical storage, Reversible heat pump/organic rankine cycle, waste heat recovery, part load modelling

1. Introduction

1.1. Context

The share of electricity production needs to increase sharply in the next decades to decrease the impact of humans on the environment. However, there is a significant mismatch between renewable energy production and consumption. This means that electrical energy storages will play a very important role in the future. Among the available technologies Gravity Energy Storage, Compressed Air Energy Storage and Pumped Hydro Storage are site dependant, Fuel Cells can only achieve low efficiency up to now and electrical batteries suffers from high costs and the use of rare materials [1]. A recent alternative technology is therefore studied for several years: the Carnot battery (or Pumped Thermal Energy Storage). The principle is rather simple: a heating cycle converts electricity into thermal energy, to store it and to use a power cycle to convert it back to electrical energy when necessary. Different configurations are possible to achieve a Carnot battery: a closed Brayton cycle [2,3], an electrical heater combined with a Rankine Cycle [4], a heat pump combined with a Rankine cycle [5,6]), the Lamm-Honigmann process [7] or Liquid Air Energy Storage [8]. The performance of the system can be improved by integrating waste heat into the process [9,10]. The configuration using a heat pump and a Rankine cycle is particularly suited for this thermal integration because of its low range of temperature. Recently, it was proposed to decrease the costs of this system by having only one system acting as a heat pump or as an ORC with the same components: the reversible HP/ORC system [11,12].

1.2. Concept

The concept of Carnot battery with a reversible HP/ORC power system configuration using waste heat is presented in Figure 1. When there is an excess of electricity on the grid, the charging mode can be activated through the heat pump. It increases the waste heat temperature up to the thermal storage temperature through a classical cycle (evaporator, compressor, condenser and expansion valve). The use of waste heat allows the heat pump to work with a high Coefficient of Performance. On the contrary, when the electrical consumption is high on the electrical grid, the system is reversed and is able to work as an ORC power systems. The heat from the thermal energy storage is converted into electricity through an ORC power system. The pump creates a flow of refrigerant to the evaporator. There, the refrigerant is evaporated at thanks to the thermal energy given by the thermal energy storage. Following this, the high pressure and high temperature vapour produces electricity in the expander. Finally, the fluid is sub-cooled in the condenser to start the cycle again.

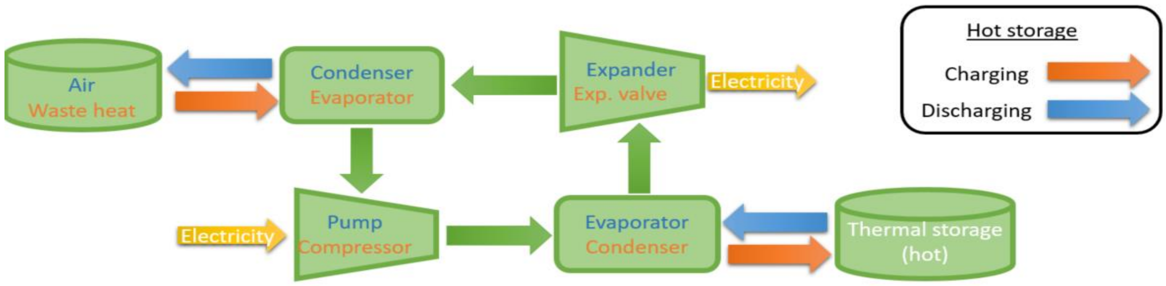

Fig. 1. Concept of Carnot battery with a reversible HP/ORC power system using waste heat

After a brief context and description of the concept of the Carnot battery using a reversible HP/ORC power system (section 1 - Introduction), this paper describes the part load modelling of such a system (section 2 - Methodology). Following this, the results section provides the influence of each parameter on the results. From there, an optimal control is developed and a case study is given with classical performance indicators. Finally, the last section concludes and gives opportunities for further research.

2. Methodology and modelling

2.1. Performance criteria

Three main parameters (Coefficient of Performance of the heat pump, ORC efficiency and power to power ratio) are defined to characterize the system (Eq. 1-3).

$$COP = \frac{\dot{Q}_{cd}}{\dot{W}_{cmp,el}+\dot{W}_{pp,aux,el}} \tag{1}$$

$$\eta = \frac{\dot{W}_{exp,el}+\dot{W}_{pp,el}+\dot{W}_{pp,aux,el}}{\dot{Q}_{ev}} \tag{2}$$

$$P2P = COP.\eta \tag{3}$$

2.2. Case study

This papers proposes to model an existing prototype extensively described in Dumont et al. (2020). A summary of the technical information is proposed in this section. Figure 2 shows an hydraulic scheme with all the components and sensors. The dark blue loop is the refrigerant loop composed of a high pressure (hp) heat exchanger, a scroll volumetric machine able to work as a compressor or as an expander, a low pressure heat exchanger (lp), and two parallel branches with an expansion valve for the heat pump and a pump for the ORC power system. The refrigerant is going clockwise for the ORC mode and counter clockwise for the heat pump mode. The red loop (water) corresponds to the thermal energy storage loop with two storages in order to operate with a perfect stratification between the hot zone and cold zone. A single storage could be used but a mixing between the high temperature and low temperature part could occur. A circulator provides the necessary flow using a four way valve to provide a counter flow in the high pressure heat exchanger (hp). The light blue loop (water) is able to regulate both the flow and the temperature entering in the low pressure heat exchanger (lp) with a four way valve for the same reason as mentioned before. This water flow simulates the waste heat in HP mode and the air cooled condenser in ORC mode.

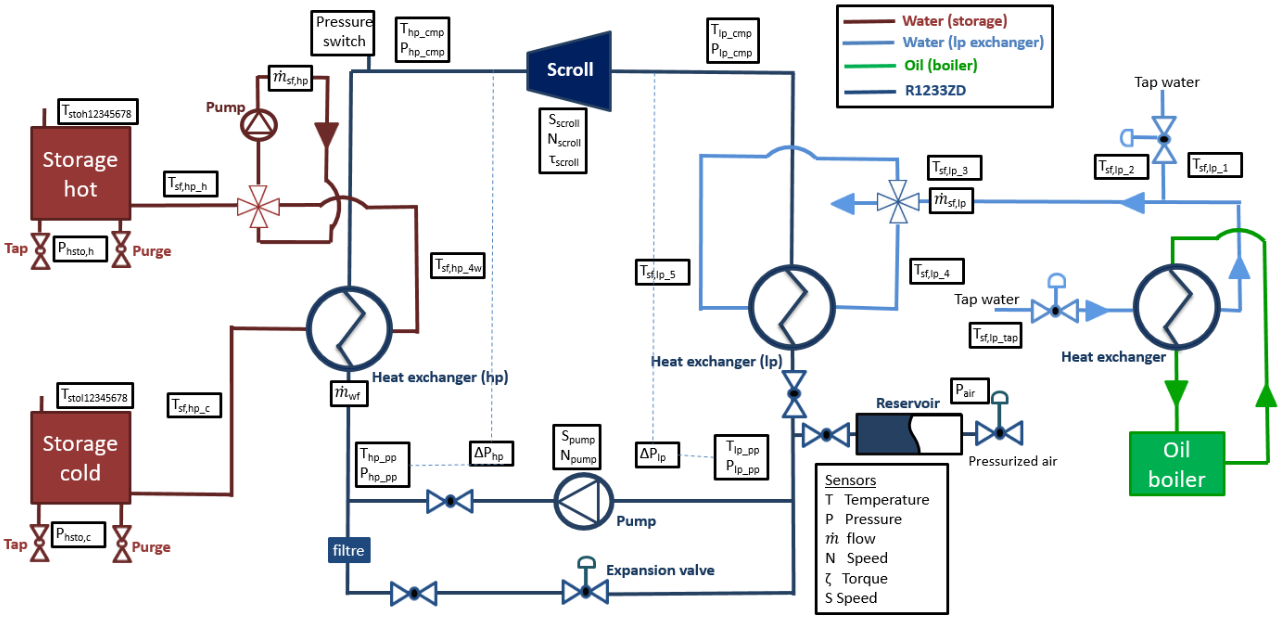

Fig. 2. Hydraulic scheme of the prototype of reversible HP/ORC power system using waste heat.

The sizing and modelling of the prototype was achieved according to the following choices:

- The air temperature is set to 15°C and the waste heat temperature is 75°C. These conditions are representative of many industrial application and should lead to an electrical production equal to the electrical consumption (Dumont et al., 2019).
- The architecture is selected to simplify the layout and the cost. Only one volumetric machine with a reversed cycle (Figure 2) is selected (Dumont, 2017)
- A constant efficiency model is used for both HP and ORC mode. The isentropic efficiency of the compressor and of the expander are set to 75% maximum (possible decrease due to under or over expansion). The volume ratio is equal to 2.2. The isentropic efficiency of the pump is 50%, the pinch points are equal to 2 K and the sub-cooling and superheating are set to 5 K. The Reynolds ratio is characterizing the ratio between the highest Reynolds number of the ORC and the highest Reynolds number in the ORC (Dumont et al., 2019). It is set to one in order to obtain similar working conditions and therefore good performance in HP and ORC mode (Dumont et al., 2019). The pressure drop is assumed to be equal to 200 mbars both on hp and lp side. The nominal electrical power of the volumetric machine is 2 kW.
- The storage capacity should be around 10 kWh. The glide (i.e. difference of temperature between hot and cold storage) is set to 10 K.

Based on these considerations, it was possible to size the unit and obtain the nominal conditions summarized in Table 1. The refrigerant R1233zd reached the best performances and was therefore selected (Dumont et al., 2019).

Table 1. Nominal point of the prototype

		ORC	HP
Parameters	Sub-cooling [K]	5	5
	Exchangers pinch points [K]	2	2
	Evaporator secondary fluid glide [K]	10	10
	Condenser secondary fluid glide [K]	2	10
	Compressor volume ratio [-]	2.2	
	Compressor maximum efficiency [-]	0.75	
	Condenser secondary fluid temperature [°C]	15	85
	Evaporator secondary fluid temperature [°C]	80	75
Outputs	COP/η [-]	0.076	12.9
	Condenser thermal power [W]	20985	24110
	Evaporator thermal power [W]	22844	21720
	Compressor power [W]	1942	1865
	Condenser pressure [bar]	1.25	8.27
	Evaporator pressure [bar]	5.45	5.17
	Working fluid flow rate [kg/s]	0.096	0.14
	Compressor speed [RPM]	5000	8000
	Auxiliary pump consumption	129	29
	Compressor efficiency [-]	0.70	0.72
	Volume ratio [-]	2.26	0.57
	Compressor swept volume [m^3]	$3e^{-5}$	
	P2P [-]	1.01	
	Reynolds ratio [-]	0.76	
	Electrical ratio [-]	1.04	

2.3. Modelling and Assumptions

The idea of this paper is to propose a part load model in order to understand the influence of the different parameters on the performance of the aforementioned prototype.

The pump semi-empirical model has been proposed (Eq. 4) and validated [13]. Values of the coefficient are given in Table 2.

$$\dot{W}_{pp} = K_1 + K_2 N + K_3 \dot{V} \Delta P + \dot{W}_{mot,n}\left(\frac{1}{\eta_{mot,n}} - 1\right)\left(K_4 \frac{(K_2 N + K_3 \dot{V} \Delta P)^2}{\dot{W}_{mot,n}{}^2} + (1 - K_4)\frac{N^2}{N_n^2}\right) \qquad (4)$$

Table 2. Model parameters

Component	Parameter	Value
Pump	K_1 [W]	20
	K_2 [Ws]	0.07

	K_3 [-]	1.17
	K_4 [-]	0.6
Compressor	Ambient heat losses [W/(m^2.K)]	5
	Supply heat losses [W/(m^2.K)]	10
	Exhaust heat losses [W/(m^2.K)]	10
	Leakage area [m^2]	1e-8
	Mechanical proportional losses [-]	0.15
	Mechanical constant losses [W]	10
	Swept volume[m^3]	8.57e-5
	Exhaust pressure drop diameter [m]	0.098
Expander	Ambient heat losses [W/(m^2.K)]	1
	Supply heat losses [W/(m^2.K)]	50
	Exhaust heat losses [W/(m^2.K)]	94
	Leakage area [m^2]	2.6e-6
	Mechanical proportional losses [-]	0.02
	Mechanical constant losses [W]	8
	Swept volume[m^3]	3.8e-5
	Supply pressure drop diameter [m]	0.098
Evaporator	PP [K]	2
	k (pressure drop) [Pa.s^2/kg^2]	2e6
Condenser	PP [K]	2
	k (pressure drop) [Pa.s^2/kg^2]	4e6
Auxiliary pumps	Isentropic efficiency [-]	70 %

The compressor and expander model are the one proposed by Lemort [14]. This semi-empirical model accounts for the most influent physical phenomena in the process with a limited number of parameters. This model demonstrates a good ability to extrapolate the performance while maintaining low computational times. Besides of under- and over-expansion losses (due to the fixed built-in volume ratio of the machine), the model can account for pressure drops and heat transfers at the inlet and outlet ports of the machine, internal leakages, mechanical losses, heat losses to the environment and recompression phenomena. The parameters of the model are given in Table 2 and are derived from Dumont et al. (2015).

As already proven (Dumont, 2019), the performance of the system is very sensitive to the heat exchanger performance. Therefore, a maximal pinch point of 2 K is considered in this model as a conservative hypothesis since both the condenser and evaporator should present a lower value. The pressure drop are evaluated with Eq. 5.

$$\Delta P = k.\dot{m}_{wf}{}^2 \qquad (5)$$

This part load model is able to evaluate every output based on the parameters of Table 2 and on the 5 inputs: the glide of the condenser (g_{cd}), the glide of the evaporator (g_{ev}), the shaft speed of the volumetric machine (N) – for convergence reasons the mass flow rate is used for the ORC model instead of the shaft speed- , the temperature of the secondary fluid at the supply of condenser ($T_{cd,sf,su}$) and the temperature of the secondary fluid at the supply of evaporator ($T_{ev,sf,su}$). The next section shows the performance of the machine in function of these 5 inputs.

3. Results and discussion

3.1. Mapping of performance

The aim of this section is to illustrate the performance of the machine by varying the parameters. Due to their large number, it is not possible to draw the influence of the combination of all the parameters. This section is only illustrative and the behaviour of the system is not optimized. For the heat pump, the COP, the compressor electrical consumption, the charging time, the compressor efficiency, the compressor consumption, the auxiliary pumps consumption are depicted in Figure 3 for an evaporator glide of 10 K.

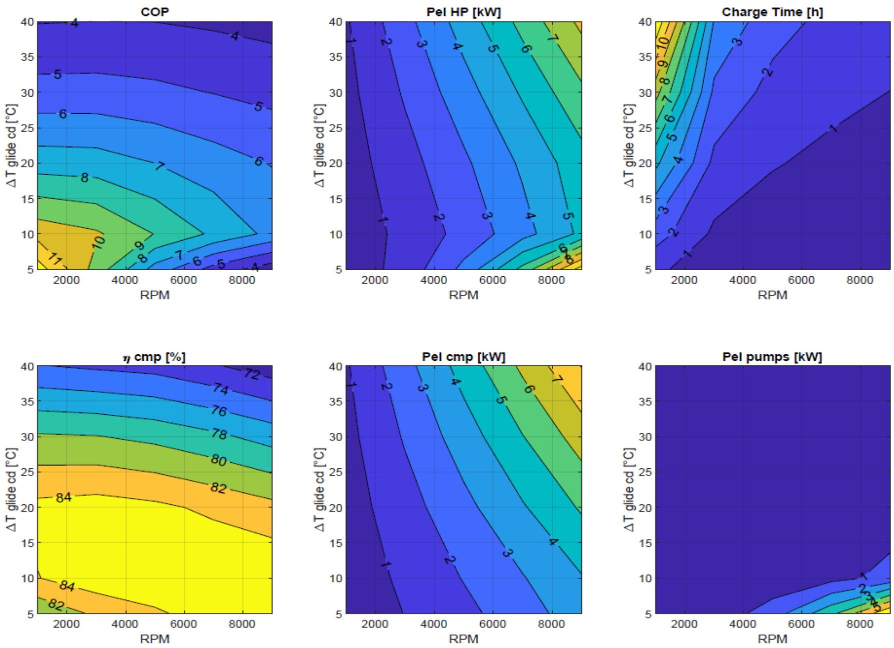

Fig. 3. COP, compressor electrical consumption, charging time, compressor efficiency, compressor consumption, auxiliary pumps consumption for the heat pump. In this example the evaporator glide is set to 10 K.

Multiple aspects can be emphasized from these graphs:

- The influence of the auxiliary pumps consumption has to be taken into account and becomes important at high thermal power (i.e. high RPM) and low glide temperatures.
- In most of the cases, the thermal power increases with the RPM and with the glide while the COP follows the opposite trend. It leads to one of the most important conclusion, optimizing the behaviour of such a system is a trade-off between maximizing the power to power ratio or maximizing the electrical power to absorb for the heat pump. Indeed, the charging time is low when considering low compressor speed that maximizes the COP.
- The compressor is not working in optimal conditions. For glide temperatures of the condenser below 15 K, under-compression decreases the efficiency while over-compression is present for glides above 15 K. At high compressor shaft speeds, the mechanical losses and the pressure drops have a significant impact on the performance. It shows that the compressor shaft speed should be optimized differently for every combination of inputs and parameters.

For the ORC, the global efficiency, the compressor electrical consumption, the charging time, the expander efficiency, the expander consumption, the auxiliary pumps consumption are depicted in Figure 4 for an evaporator glide of 10 K.

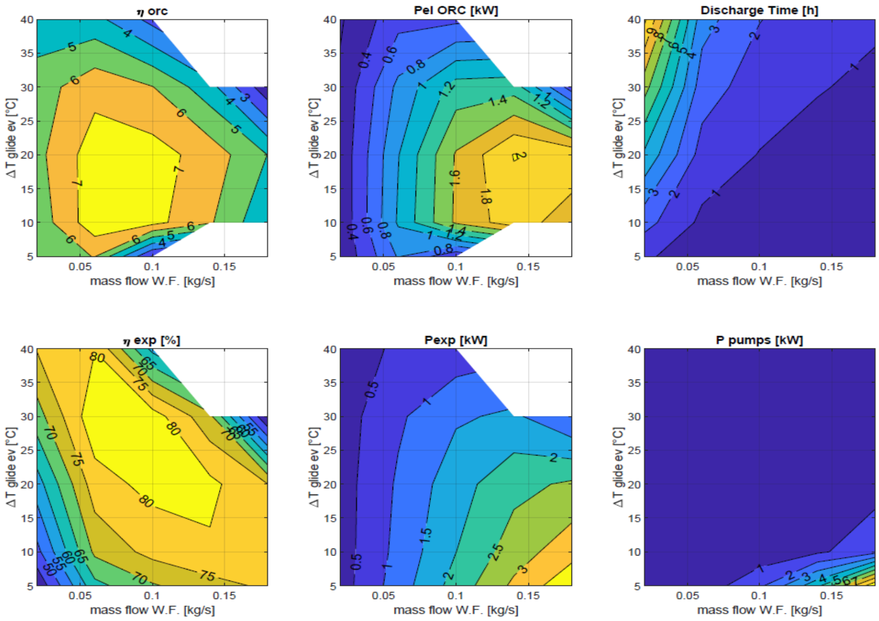

Fig. 4. ORC efficiency, compressor electrical consumption, charging time, expander efficiency, compressor consumption, auxiliary pumps consumption for the heat pump.

From these graphs, some additional observations can be made:

- In most of the cases, the electrical power increases with the mass flow rate and with the glide while the ORC efficiency follows the opposite trend. It leads to one of the most important conclusion, optimizing the behaviour of such a system is a trade-off between maximizing the power to power ratio or maximizing the electrical power to reject for the ORC. Indeed, the discharging time is low when considering low mass flow rates that maximizes the efficiency.
- The expander is not working in optimal conditions. For low glide temperatures of the condenser, over-expansion decreases the efficiency while under-expansion is present for higher glides. At high expander shaft speeds, the mechanical losses and the pressure drops have a significant impact on the performance. It shows that the expander shaft speed should be optimized differently for every combination of inputs and parameters.

3.2. Optimization of performance

The optimization of such a system is not trivial and requires the optimization of different parameters. As mentioned in the mapping performance section, the control parameters can be optimized to control either the electrical (thermal for HP) power output or the efficiency (COP for the HP). In real case studies, the optimal operation of such a system will be a compromise between power output and efficiency depending on air temperature, waste heat temperature, electricity prices… In this section two extreme cases are shown. The system is firstly optimized ($glide_{ev}$, $glide_{cd}$ and RPM) for every combination of secondary fluid temperature to maximize the efficiency (or COP) and secondarly to maximize the power to power ratio. For the ORC, the optimization in terms of efficiency (Fig. 5) and power (Fig. 6) is given. Globally, the efficiency is decreased by roughly 1% when optimizing the power instead of the efficiency (see Figs. 5-6) On the contrary, the power produced is decreased significantly (roughly 50% on average) when optimizing the efficiency (see Figs. 5-6). It means that optimizing the power is generally more interesting for low charging and discharging time despite of a small decrease of efficiency. By comparing Fig. 5 and Fig. 6, it appears that optimizing the efficiency leads to lower glides and lower mass flow rates than the power optimization for the ORC system.

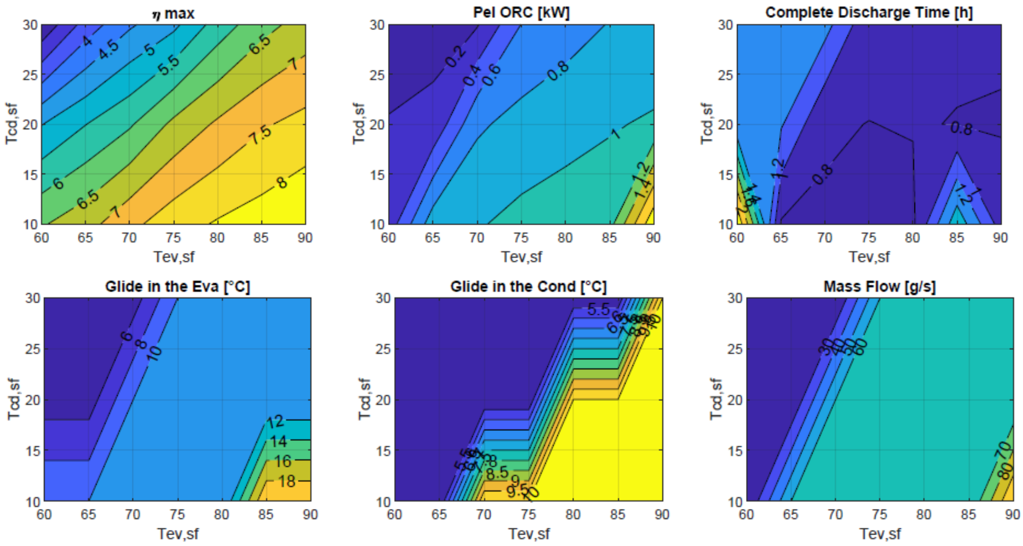

Fig. 5: Efficiency optimization results: ORC performance in terms of efficiency, electrical power production and discharge time with their optimized parameters (glides and mass flow rate).

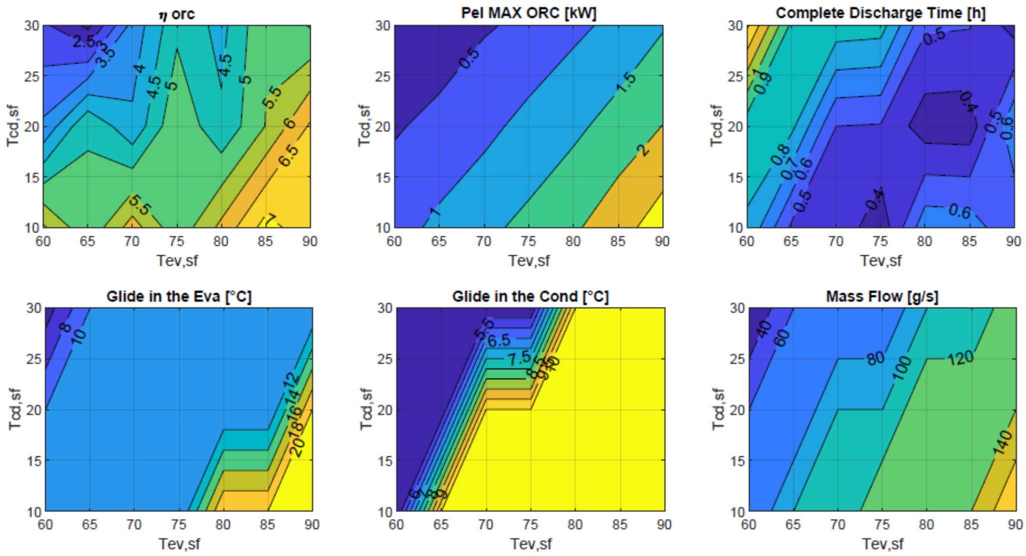

Fig. 6: Power optimization results: ORC performance in terms of efficiency, electrical power production and discharge time with their optimized parameters (glides and mass flow rate).

In the case of the heat pump, the maximization of the COP always leads to the minimal compressor speed (1000 RPM) while the optimization of the thermal power at the condenser leads to the largest value (10,000 RPM). Contrarily to the ORC case, a significant differences in terms of COP and thermal power are observed between the two types of optimization. This means that the optimization of the system will be more sensitive to the inputs at a specific time for the heat pump than for the ORC system.

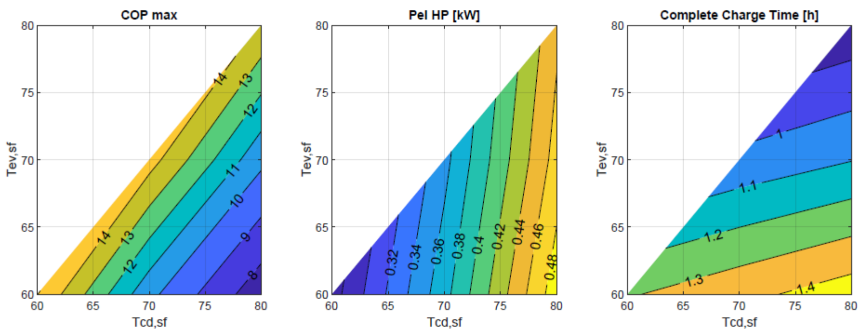

Fig. 7: COP optimization results: HP performance in terms of COP, electrical consumption and charging time

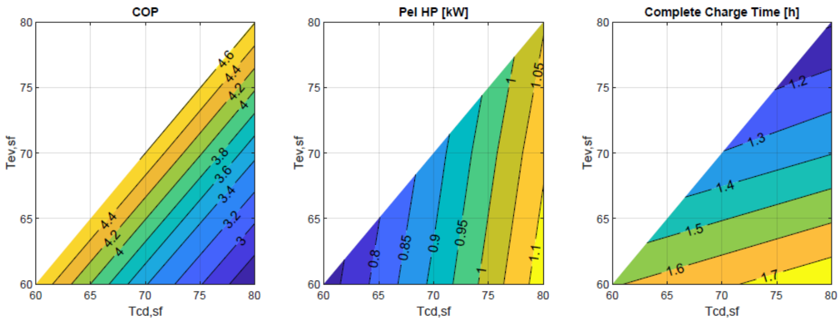

Fig. 8: Power optimization results: HP performance in terms of COP, electrical power consumption and charging time.

From these graphs, using the nominal conditions of air and waste heat temperature (Table 2), the charging-discharging ratio can be as high as 100% (7% ORC efficiency, COP equals to 14) when optimizing the efficiency while it can decrease up to 20% when the power is maximized (i.e. the charging and discharging time are minimized).

4. Conclusion

This paper proposes a part load model for a Carnot battery based on a reversible heat pump/organic Rankine cycle system and shows the influence of the control variables on the performance. Numerous simulations are performed in a way to understand how to optimize each control variable depending on the working conditions. The main control variable is the shaft speed of the volumetric machine. Large values of shaft speed induces lower performance of the system while low values lead to lower power and therefore larger charge and discharge time. It means that an optimum needs to be found depending on the nominal conditions and on the electricity costs. In this case study (air temperature of 15°C and waste heat temperature of 75°C), the COP of the heat pump can reach values up to 14, the ORC efficiency reaches 8% when optimized. This lead to an achievable roundtrip efficiency larger than one. Further studies should investigate how to optimize the profitability of such a system with different case studies.

Nomenclature

COP	*Coefficient Of Performance [-]*
g	*Glide [K]*
hp	*high pressure*
HP	*Heat Pump*
K	*Empirical coefficient [-]*
lp	*low pressure*
ORC	*Organic Rankine Cycle*
N	*Shaft speed [RPM]*
P	*Pressure [pa]*
PP	*Pinch point [K]*
PTES	*Pumped Thermal Energy Storage*
P2P	*Power to power ratio [-]*
$\dot{Q}$	*Thermal power [W]*

T	*Temperature [°C]*
$\dot{V}$	*Volumetric flow [m³/s]*
$\dot{W}$	*Power [W]*

Greek symbols

η	*efficiency*

Subscripts and superscripts

cd	*Condenser*
cmp	*Compressor*
el	*Electrical*
ev	*Evaporator*
exp	*Expander*
mot	*Engine*
n	*Nominal*
pp	Pump
sf	Secondary fluid
su	Supply

References