Experimentally-validated models for the off-design simulation of a medium-size solar organic Rankine cycle unit

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Abstract

Organic Rankine Cycle is an efficient and reliable technology for the thermal-to-electricity conversion of low-grade heat sources but the variability in boundary conditions often forces these systems to operate at off-design conditions. The development of reliable models for the performance prediction of organic Rankine cycle power systems under off-design conditions is therefore crucial for system-level integration and control implementation. In this paper, a mathematical model for the evaluation of the expected performance of organic Rankine cycle power units in a large range of operating conditions based on experimental data collected in a medium-size solar-organic Rankine cycle power plant is presented. Two different empirical approaches for the performance prediction of heat exchangers and machines, namely, constant-efficiency and correlated-based approaches, are proposed and compared. In addition, empirical correlations based on experimental data are proposed for the preliminary assessment of the energy demanded during the start-up phase and the corresponding duration. Results demonstrate that a good achievement in terms of accuracy of the model and reliability of the simulation performance can be obtained by using a constant-efficiency approach, with average errors lower than 5% and 2.5 K for the expected net power and outlet oil temperature respectively. The use of polynomial correlations leads to a more accurate estimation of the performance parameters used for evaporator and the turbine (in particular the evaporator heat effectiveness and the isentropic and electromechanical efficiency for the turbine), which strongly affect the main output variables of the model and, at the same time, are remarkably influenced by the operating conditions. A reduction on the average error in the prediction of the net power and outlet temperature of the heat transfer fluid to about 4% and 1.5 K respectively is therefore achieved by this approach. Average errors of 18.5% and 12.5% are achieved for the start-up time and the corresponding energy absorbed, respectively. Although the results obtained in terms of accuracy could be improved, these correlations can give an initial indication about the duration and energy required during this phase.

Keywords:

Organic Rankine cycle, off-design conditions, modelling, start-up phase, experimental data, operating condition

NOMENCLATURE

Symbols		***Subscripts***	
c	specific heat [kJ/kg K]	CD	condenser
h	specific enthalpy [kJ/kg]	EV	evaporator
$\dot{m}$	mass flow rate [kg/s]	P	pump
N	rotational speed [rpm]	REC	recuperator
p	pressure [Pa]	SH	superheating
$\dot{Q}$	thermal power [kW]	SU	start-up
Q	thermal energy [kWh]	T	turbine
t	time [h]	WF	working fluid
T	temperature [K]	i	inlet
$\dot{V}$	volumetric flow rate [m^3/s]	is	isentropic
$\dot{W}$	electrical power [kW]	ins	insulation
ΔT	temperature difference [K]	em	electro-mechanical
ε	heat exchanger effectiveness [-]	net	net power
η	efficiency [-]	nom	nominal conditions
ρ	density [kg/m^3]	o	outlet
σ	standard deviation	set	set point
Acronyms			
CSP	Concentrating Solar Power		
CTF	Cooling Transfer Fluid		
HTF	Heat Transfer Fluid		
ORC	Organic Rankine Cycle		
WHR	Waste Heat Recovery		

1. Introduction:

The transition of energy systems in ways that would be compatible with greenhouse gas reductions target for ensuring a sustainable future for subsequent generations requires the further development of renewable energy technologies (i.e. with carbon-free emission) as well as a more efficient use of energy by final customers [1]. In this context, the organic Rankine cycle (ORC) technology can fulfil a non-negligible role to valorise low-grade heat into electricity or mechanical power [2]. In fact, the current availability of different working fluids and several design alternatives for turbines, pumps and heat exchangers allows to apply ORC systems in a wide range of heat source temperatures (60-300 °C) and power outputs (from few W to 10 MW), where the use of conventional steam cycles leads to lower conversion efficiencies and higher capital costs [3].

Because they are specifically suited for small-capacity and low-grade heat sources, a common aspect of ORC systems is the versatile nature of their operating conditions [4]. The variation in ORC boundary conditions can be intentional, for instance in case the turbogenerator operates in isolated grid for the supplying of variable loads, or unintended, when undesirable fluctuations in the heat source and heat sink characteristics occur. The variability in heat source characteristics arises both in terms of fluctuations in the mass flow rate and temperature and it is typical in Waste Heat Recovery (WHR) applications or in the exploitation of not-dispatchable renewable energy sources [5]. For instance, deviations in the nominal thermal power input are usual for application of ORCs in Concentrating Solar Power (CSP) plants due to fluctuation in the available solar power. Regarding the heat sink characteristics, a variation in the cooling fluid inlet temperature is typical of condenser cooling systems based on dry air coolers or cooling towers, where a dependence on the ambient temperature occurs. Because of their unsteady environments, ORC systems are therefore commonly led to operate in conditions differing from their nominal design points. Such conditions are referred to as *off-design* and result ordinarily in a system performance alteration. Indeed, since the components geometries are not optimal to the new boundary environment, the thermodynamic state of the working fluid is different from that of design values and results (in most cases) in an efficiency degradation. A proper understanding and prediction of the ORC behaviour under off-design conditions is crucial - or even mandatory - for multiple reasons, including for on-site performance optimization, system-level integration or control development [6]. For this reason, the operability of ORC units under variable inputs has been largely studied in literature both on experimental basis and through the development of models able to assess the performance of the ORC unit under a large range of operation.

Concerning the experimental activities for investigating ORC system performance under different conditions, Park et al. [7] carried out a comprehensive analysis of the most significant, relevant and up-to-date experimental data published in scientific literature. The authors highlighted that the majority of the experimentally published works were performed for micro- and mini-scale ORC applications under 10 kWe output with a large gap between research and development for source and sink temperature differences above 150 °C. Recently, the influence of the heat and cooling conditions on the system performances were experimentally investigated in a small-scaled ORC experimental bench by Wang et al. [8]. Results indicated that the largest amplitude of the measured pressure and temperature fluctuation were at the pump outlet and evaporator outlet respectively. Similar conclusions were given by Zhang et al. [9] based on experimental data collected in a small-scale ORC system using R123 as working fluid. The results also demonstrated the importance of these input conditions also in the choice of the ORC configuration and the benefits in including or not a regeneration section.

With a particular focus on the development of reliable models for the prediction of medium scale ORC performance under off-design conditions, Manente et al. [10] proposed a theoretical 6 MWe ORC system for a geothermal application. A steady-state off-design model was developed to find the optimal operating parameters (pump speed, turbine capacity factor and air flow rate through the condenser) that maximize the electricity production in response to changes of the ambient temperatures between 0 and 30 °C and geofluid temperatures between 130 and 180 °C. The effects of the ambient temperature variation on low- and medium-temperature geothermal ORC systems, based on dry air coolers and cooling tower, installed at different geographical locations was analysed by Usman et al. [11]. The ORC design was optimized to achieve the maximum power output and an operational control strategy was developed to maximize the power output even with different ambient temperatures. Pang et al. [12] developed an ORC engineering simulator based on experimental data obtained in a ORC test rig. Three different control strategies were introduced and tested under varying heat source and environmental temperatures. Results demonstrated that the choice of the optimal control approach strongly depends on the heat source and sink characteristic during a long-time operation. The behaviour of the ORC unit under off-design conditions is also strongly influenced by the cycle architecture, as demonstrated by Hu et al. [13] by comparing the system thermodynamic performance of a serial dual-pressure ORC unit with a basic one during heat source and ambient temperature fluctuations. With reference to ORC applications for the waste heat recovery, Song et al. [14] developed a steady-state model for the design and analysis of a 534 kW ORC system for an industrial WHR application. After the selection of a suitable working fluid and the definition of cycle points under design conditions, an off-design model was used to assess the influence of the heat source and the cooling water supply temperatures on the system performance. A similar study was conducted by Kim et al. [15], in which an off-design model for the ORC driven by waste heat or residual heat from a combined cycle cogeneration plant was developed. Pili et al. [16] compared a quasi-steady approach with a dynamic approach and the trade-off between them was analysed, taking as benchmark a waste heat recovery system with ORC from a billet reheating furnace. The results demonstrated the validity of the quasi-steady state approach, for techno-economic optimization of ORC, but the validation of the modelling with experimental data for large/middle scale ORCs, in particular at off-design conditions, was declared as a required future work.

The investigation of off-design performance has been fundamental also for ORC units coupled with solar systems. In this regard, Wang et al. [17] investigated the performance of a 250 kW ORC unit coupled with compound parabolic collectors under off-design operating conditions related to the fluctuations of ambient temperature and heat transfer fluid (HTF) mass flow rate (due to variations in the solar radiation). A steady-state modelling of an ORC was built by assuming a sliding pressure control to keep a constant superheating and a variable cooling mass flow rate to ensure a same approach point in the condenser, but no experimental validation was provided. Calise et al. [18] simulated the off-design performance of an ORC system fed by medium-temperature heat sources (155–185 °C). The behaviour of the ORC unit in case of fluctuations of heat source characteristics was investigated highlighting the decrease of the ORC efficiency with the increase of heat source mass flow rate and temperature due to the reduction of the pressure difference between boiling and condensing phase by assuming a constant expansion ratio. Oyekale et al. [19] investigated the effect of off-design performance of an hybrid solar-biomass ORC plant also from and exergetic and exergoeconomic point of view. Results underscored how off-design conditions reduce the exergetic performance of the hybrid plant, due to slightly higher defects of irreversibility and losses, with the consequent increase of the exergy costs of the final products.

The main methods used for modelling the system performance are based on physical or semi-physical approaches, in which heat transfer coefficients governing the heat exchange and the pump/expander performance are analysed in detail. For instance, by referring to performance predicted models for heat exchangers, often a 1-D modelling is used for the determination of a proper overall heat transfer coefficient. This approach, from one hand, increases the reliability and the extrapolation performance of the model adopted, but relatively high computational times are usually demanded [20]. This point could become a crucial issue if a fast response from the ORC model is required or if the simulation of this model is repeated innumerable times. In this regard, some studies highlighted the need of simpler ORC model to be involved in the control strategy adopted in real plants [21], in particular during the real-time control strategy [22]. Moreover, in last years, the adoption of robust optimization for ORC units, which consider variations in the heat source and heat sink characteristics, was proposed in several papers. In this regard, a two-step optimization procedure for the design and off-design optimization of geothermal ORC units was proposed by Van Erdeweghe et al. [23]. Bufi et al.[24] proposed a robust optimization approach including the fluctuations of exhaust gas mass flow rate and temperature for the design of an ORC unit for a WHR application. Petrollese and Cocco [25] proposed a robust preliminary design method of solar ORC systems by considering the possible variations of the heat source and the heat sink conditions, which were featured as multi-scenarios with different probabilities. In all these cases, off-design modelling of the ORC unit is involved even during the design stage for a better evaluation of the actual conditions the ORC unit will operate during its lifetime. Since the robust design demands the resolution of a non-linear optimization problem, the use of simpler models could be beneficial to reduce the very high computational effort often required.

In this framework, this paper aims to investigate the reliability and accuracy of alternative and simpler approaches for the determination of the main operating parameters via an off-design simulation of a medium-size ORC unit operating under an unsteady environment. In particular, two different approaches are proposed and compared for the performance prediction of heat exchangers and machines:

- Constant-efficiency approach, in which the heat transfer effectiveness of the heat exchangers, the insulation efficiency, the efficiencies of the turbine and the pumps are assumed constant regardless of operating conditions and equal to the mean value found during the analysis of the experimental data.
- Correlation-based approach involving the variation of the mass flow rates of the organic fluid and secondary fluids (HTF or cooling fluid), in which instead of constant values, polynomial regressions by means of quadratic functions dependent on operating conditions are used for the calculation of the heat transfer effectiveness and components efficiencies.

The aim is to analyse the feasibility of implementing these empirical models in real-time control systems for the performance prediction of ORC units, in which the characteristics of the main components, the operating domain and the internal control strategy adopted are defined and known. The analysis is conducted based on an existing medium-size CSP-ORC plant presented at the Ottana solar facility (Italy). The experimental results collected during the operation of the plant are used both for the assessment of the ORC components performance and for the analysis of the error arising from the use of the proposed approaches.

A further element analysed in this study is related to the start-up phase of the ORC unit. To author's knowledge, this element has never been investigated in literature. However, the time required to complete the start-up operation and the corresponding energy absorbed in this period could be significant if frequent start-up and shut down of the ORC unit are expected (for instance, in the case the ORC unit is fed by solar-based systems). The remainder of this paper is organized as follows: in the first part, the main characteristics of the ORC unit analysed and the domain of variability of the main input parameters are presented. The modelling adopted for the off-design simulation and the various approaches used for the component simulation is subsequently introduced. The main results in terms of error analysis on the expected net power production and on the outlet temperatures are then discussed. Finally, a preliminary analysis on operating time and the energy requirement during the start-up phase is conducted.

2. Experimental investigations

In this work, a 630 kWe ORC unit (Turboden 6HR Special, see Table 1) is used as case study. The system is installed and operated in the Ottana solar power plant, an experimental facility located in Sardinia, Italy [21]. As shown in Figure 1, the ORC unit is fed by thermal oil heated by Linear Fresnel collectors in a field covering about 8600 m^2. The heat transfer fluid (HTF) is a commercial thermal oil (Therminol SP-I) and the solar field design inlet/outlet temperature are 438 K and 538 K, respectively. A two-tank direct thermal energy storage is placed between the solar field and the ORC unit. It is aimed to store about 195 tonnes of thermal oil at a maximum temperature of 538 K, which corresponds to a storage capacity of 15.2 MWh (4.9 equivalent hours of the ORC operation at nominal conditions).

Table 1 - ORC performance at reference conditions.

INPUT – Heat transfer fluid		**ORC PERFORMANCE**	
Fluid	Therminol SP-I	Organic fluid	$C_6H_{18}OSi_2$
Inlet temperature	538 K	Gross electrical power	664 kW
Outlet temperature	438 K	Gross electrical efficiency	21.4%
Mass flow rate	11.05 kg/s	Captive power consumption	35 kW
Thermal power input	3100 kW$_t$	Net electrical power	629 kW
OUTPUT – Cooling water		Net electrical efficiency	20.3%
Inlet temperature	298 K	Electric generator	50Hz/400 V
Outlet temperature	308 K		
Thermal power output	2436 kW$_t$		

(a)

(b)

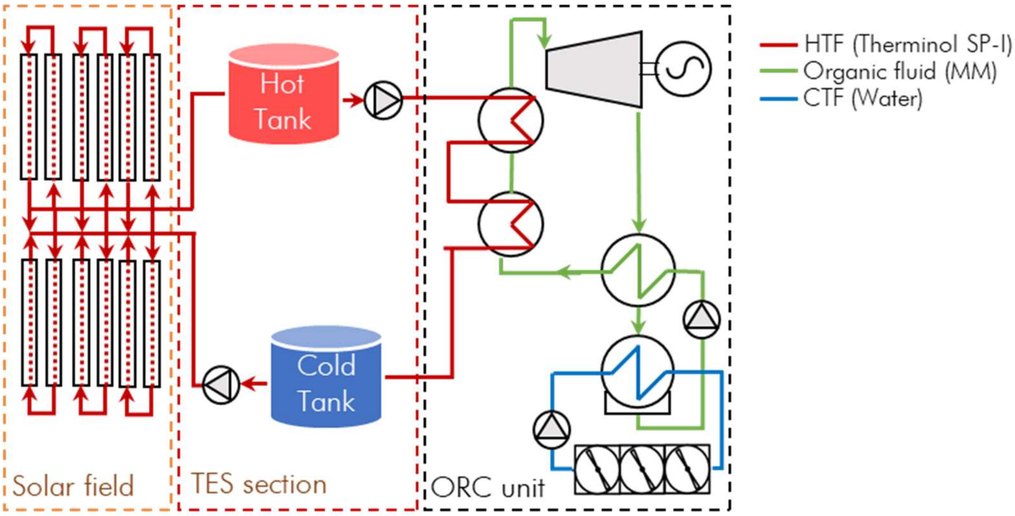

Figure 1 – (a) Aerial view of the Ottana solar facility and (b) schematic scheme of the CSP section.

Figure 2 depicts a schematic of the ORC unit together with the sensors position used for recording the system performance. A centrifugal pump is used to feed the required HTF mass flow rate from the hot tank to preheat and to vaporize the organic fluid. The latter is then expanded through a 3-stage axial turbine (the rotational speed is kept constant to about 3000 rpm) coupled with an electric generator (asynchronous type, air cooled), where the thermal energy is converted into electrical energy. Owing to the molecular complexity of the organic fluid, a small enthalpy drop occurs in the expansion phase, resulting in a large thermal power availability at the turbine discharge. A recuperator is therefore introduced between the turbine and the condenser for a regenerative preheating of the pumped liquid. The exhaust organic fluid returns to saturated liquid conditions in a water-cooled condenser and, finally, it is pressurized in a multistage centrifugal feed pump. The water from the condenser is cooled in turn by dry coolers and, in winter, used for underfloor heating of control rooms and offices. The three centrifugal pumps are coupled to inverter-controlled electric motors allowing to change their rotational speed. In particular, the thermal oil pump is controlled by a PID controller in order to arrange its rotational speed and to meet the desired electrical power imposed by a set point. The water pump is also controlled by a PID controller in order to keep the outlet water temperature from the condenser as close as possible to the imposed set-point temperature. Finally, the organic fluid pump is operated to adjust the ORC evaporating pressure accordingly to the controller logic.

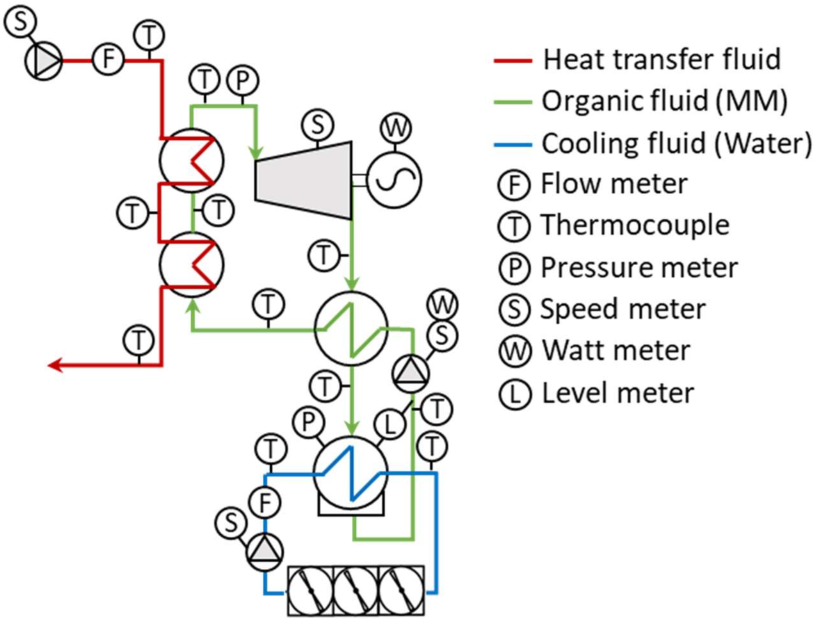

Figure 2 – Schematic view of the ORC unit with details about sensors position.

Table 2 – Main characteristics of the sensors used by the acquisition system.

Sensor type	*Range*	*Accuracy*
Flow meter	0 - 62 m^3/h	± 0.75%
Thermocouple	190 - 870 K	± 0.3 K
Pressure meter	0 - 500 mm_{H2O}	± 0.25%
Level meter	0.15 - 22 m	± 0.3%

The ORC unit is equipped with several sensors recording the system state at various key locations. Table 2 reports the main sensor characteristics in terms of range of operation and accuracy. Two volumetric flow meters are measuring the oil and the water flow rates. Several thermocouples are placed in the oil, water and organic fluid circuits and two pressure probes are monitoring the ORC maximum and minimum pressures. A level meter is introduced between the condenser and the pump intake side to measure the liquid level upstream the pump checking for a suitable net pressure suction head. Finally, both the organic fluid pump and turbine are equipped with power meters.

2.1. Experimental measurements

As mentioned, the ORC unit under examination is coupled to a concentrating solar field and TES system on the hot side, while the dry coolers are used to release the condenser cooling heat. This configuration leads to important fluctuation both in the heat source and heat sink characteristics, bringing the system to operate at off-design conditions.

The variability in heat source characteristics arises both in terms of fluctuations in the mass flow rate and temperature: the former variation is mainly due to the imposed set point for the desired electrical power and can be considered as an intentional fluctuation leading to a part-load operation of the ORC unit, instead the latter one is accidental and it is related to several phenomena, such as thermal losses occurring in the TES system, temperature tolerances introduced in the solar field control system (although the solar field is controlled to obtained a desired HTF outlet temperature by varying the volumetric flow rate, the HTF needs up to 3 minutes to heat up and a tolerance in the set point is required to avoid control system instability), temperature measurement accuracy and so on.

Similarly, the heat sink is characterized by fluctuations on the cooling fluid flow rate and inlet temperature. The former is controlled by the water feed pump and the imposed rotational speed. Because the higher the volumetric flow rate is, the lower the condenser pressure is, with the consequent benefits in terms of conversion efficiency, currently the water pump always works at its maximum rotational speed. On the other hand, fluctuations in the water inlet temperature can be considered as not intentional and it is typical of condenser cooling systems based on dry air coolers, where a dependence on the ambient temperature occurs.

Consequently, the operating domain of the analysed ORC unit is constituted by three independent variables: the HTF mass flow rate, the HTF inlet temperature and the inlet temperature of the cooling transfer fluid (CTF). Therefore, the development of a reliable model of the ORC unit for the prediction of its operating performance requires the investigation of the ORC performance under this three-variable domain.

The experimental data collected from October 2017 to September 2019 at the Ottana Solar Facility are used as main input for the calibration of a steady-state ORC modelling. Obviously, since solar energy is the sole heat source of the energy system, ORC unit mainly operates during summer. Moreover, although the plant under examination has demonstrative purpose, a control and management system is implemented, limiting the possibility of testing the ORC unit under extreme off-design conditions, but the data collected are explanatory of the real operation of the ORC unit. Since ORC operating conditions are constantly subjected to change, criteria for defining steady-state conditions are defined. In particular, three different constraints are introduced related to temperatures variations, volumetric flow rates variations and net electrical power variations. In particular, a first constraint has been related to the inlet temperatures of the heat transfer fluid ($T_{HTF,i}$) and the cooling transfer fluid ($T_{CTF,i}$): it is considered as a steady-state point when the standard deviations (σ) of these two measured temperatures during a time interval of 10 minutes are lower than given threshold values:

$$\sigma\left(T_{HTF,i}\right) < 0.5\,K \quad \& \quad \sigma\left(T_{CTF,i}\right) < 0.5\,K \tag{1}$$

A similar constraint is introduced related to the relative variation (compared to the nominal one) of the volumetric flow rate of the HTF ($\dot{V}_{HTF}$) and CTF ($\dot{V}_{CTF}$):

$$\sigma\left(\frac{\dot{V}_{HTF}}{\dot{V}_{HTF,nom}}\right) < 1.5\% \quad \& \quad \sigma\left(\frac{\dot{V}_{CTF}}{\dot{V}_{CTF,nom}}\right) < 1.5\% \tag{2}$$

Finally, a third constraint is related to the variation of the net electrical production ($\dot{W}_{net}$):

$$\sigma(\dot{W}_{net}) < 3\ kW\ \&\ \max(|\Delta\dot{W}_{net}|) < 10\ kW \tag{3}$$

Where $\sigma(\dot{W}_{net})$ is the standard deviation obtained during the considerd time interval, while $\max(|\Delta\dot{W}_{net}|)$ is the variation of the maximum absolute power difference observed during the considered time interval.
Among the various points that satisfied these constraints simultaneously, 100 points are identified. As depicted in Figure 3, the experimental database is then randomly split in two groups to permit some model cross-validations, namely, a training dataset, used to calibrate the ORC model parameters, and a test set, used to quantify the model fitting and extrapolation capabilities.

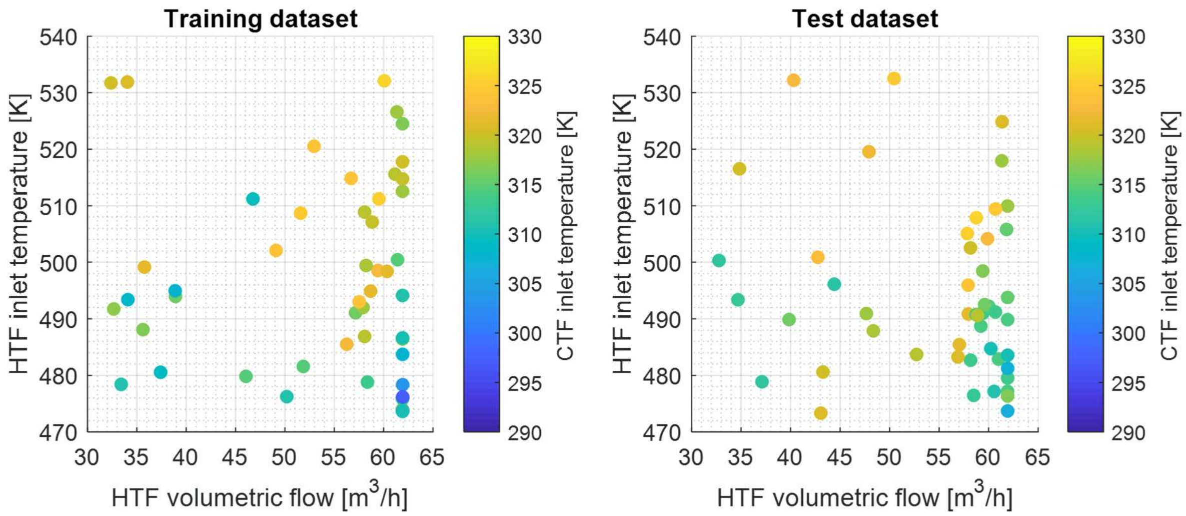

Figure 3 - Operating domain of the experimental points used for the calibration (left side) and the testing (right side) of the ORC model.

3. Mathematical model

A mathematical model of the ORC unit for the evaluation of the expected net electrical power production and outlet HTF temperature based on the system boundary conditions (mass flow rates and inlet temperatures of the heat source and heat sink) is implemented in MATLAB. The goal intended is not to achieve highly accurate simulations of the turbogenerator but to obtain a versatile simulation tool able to predict the ORC performance under a certain range of uncertainty and low computational effort.
The following modeling assumptions have been made in this study:

i. The system operates in quasi-steady state regimes.
ii. Pressure drops in all heat exchangers are neglected.
iii. A perfect insulation of the turbomachines is considered.
iv. The first stage nozzle of the turbine is choked
v. A constant degree of subcooling (equal to 3 K) at the outlet condenser side is imposed.

3.1. Components performance

The simulation of heat exchangers and turbomachines with deterministic models is the most reliable way for accurate predictions and reliable extrapolations of their performances. The temperature dependence of the correlations used for the determination of proper heat exchange coefficients, the requirement of a discretization of the heat exchange area, the need of accurate models for the performance assessment of the turbomachine under off-design conditions are typical issues occurring during the implementation of physical models, which required high computational times and proper model validations, but they are mandatory during the development of novel ORC units or for the assessment of the ORC performance during extreme conditions.

On the other hand, other simplified approaches can be followed if not high model accuracy is required. In this case, a typical way is to characterize the components behaviour through proper performance indicators. For instance, as followed in this study, the heat exchanger performance can be characterized by two parameters, that are the global heat transfer effectiveness (ε) and the thermal insulation efficiency (η_{ins}). The former performance parameter is defined as a ratio of the actual heat transfer rate from the hot fluid to the cold fluid to the maximum possible heat transfer rate thermodynamically permitted (i.e. corresponding to a pinch point equal to zero). The insulation efficiency is defined as the ratio of the heat transfer rate transferred to the cold fluid to the one introduced from the hot fluid, mainly due to a not perfect insulation of the component (see Eqs A.1, A.3 and A.5). These two parameters can be defined for each heat exchanger, that is, evaporator ($\varepsilon_{EV}, \eta_{EV,ins}$), recuperator ($\varepsilon_{REC}, \eta_{REC,ins}$) and condenser ($\varepsilon_{CD}, \eta_{CD,ins}$).

The turbine can be characterized by two different efficiencies, the isentropic efficiency ($\eta_{T,is}$), which is related to the irreversibility generated in the interaction fluid-machine, and the electro-mechanical efficiency ($\eta_{T,em}$), defined as the ratio of the electrical power and the power transferred from the fluid to the shaft. Finally, one parameter is used for predicting the pump performance, i.e. the pump efficiency (η_P) defined as the ratio of the hydraulic power transferred to the fluid and the electrical power absorbed by the electrical motor.

Overall, the definition of nine different parameters is required for the characterization of the main ORC components. In this study, the experimental training dataset is used for predicting the heat exchanger and turbomachines behaviour. Two different approaches are adopted and compared:

- A constant-efficiency approach, in which the key performance parameters are assumed independent from the fluid conditions and equal to the average value assumed during the experimental tests
- A correlation-based approach, in which the components efficiencies are modelled with polynomial correlations.

Table 3 reports the main values identified for the two different approaches together with the mean absolute percentage error (MAPE), defined as:

$$\text{MAPE} = \frac{1}{N}\sum_{i=1}^{N}\frac{|x_{model} - x_{exp}|}{x_{exp}} \tag{4}$$

where x_{model} is the expected performance values found by following the two aforementioned approaches and x_{exp} is the actual values obtained during the experimental activity by considering the test dataset and N is the number of points included in the test dataset.

Regarding the correlation-based approach, polynomial correlations introduced for the evaporator and condenser effectiveness, the evaporator efficiency and the parameters characterizing the turbine and pump performance are reported in Appendix A.2. Conversely, the recuperator effectiveness and efficiency as well as the condenser efficiency is assumed constant because limited variability with the operating conditions was detected during the experimental tests (this fact is also proved by the very low MAPE achieved for these parameters by using a constant-efficiency approach). The independent variables are selected based on a backward elimination method, in which the operating variables generating the most significant effect on the given parameter are chosen.

Table 3 – Performance parameters for the main ORC components.

	Constant–efficiency approach		**Correlation-based approach**	
	Value	**MAPE**	**Value**	**MAPE**
ε_{EV}	0.907	3.7%	$f(\dot{m}_{HTF}, \dot{m}_{WF})$	1.6%
ε_{CD}	0.914	1.2%	$f(\dot{m}_{CTF}, \dot{m}_{WF})$	0.6%
ε_{REC}	0.895	0.5%	0.895	0.5%
$\eta_{EV,ins}$	0.978	5.4%	$f(\dot{m}_{HTF}, \dot{m}_{WF})$	3.6%
$\eta_{CD,ins}$	0.99	0.1%	0.99	0.1%
$\eta_{REC,ins}$	0.99	0.7%	0.99	0.7%
$\eta_{T,is}$	0.783	3.3%	$f(\dot{m}_{WF}, \Delta h)$	1.5%
$\eta_{T,em}$	0.965	4.6%	$f(\dot{W}_T)$	3.8%
η_P	0.429	14.7%	$f(N_P)$	6.9%

3.2. System model

The methods proposed for the prediction of the component performance is included in an overall model of the ORC unit. Figure 4 shows the model architecture used for the prediction of the ORC performance. The main input parameters refer to the heat source and heat sink supply conditions in terms of mass flow rate ($\dot{m}_{HTF}, \dot{m}_{CTF}$) and inlet temperature ($T_{HTF,i}, T_{CTF,i}$). The main variable used in the iterative based model is the rotational speed of the pump (N_P) while the degree of superheating (ΔT_{SH}) at the evaporator outlet side is used as main control variable. In fact, as implemented in the real ORC unit, the pump rotational speed is controlled to keep the degree of superheating as low as possible (in the order of few degrees). The calculation of the volumetric flow rate of the working fluid at the pump location and the maximum pressure of the cycle is directly related to the assumed pump rotational speed, according to pump characteristic curves and the assumption of choked conditions in the turbine first stage nozzle. This assumption has been validated by the collected experimental data, as shown in Figure 5. The following step is related to the determination of the thermodynamic cycle points, based on basic ORC equations (reported in Appendix A.1) and components performance. The latter can be assumed constant (and thus independent from the cycle conditions) or proper empirical correlations can be used (in this case there is a dependency with the working fluid conditions), as reported in the previous section. Subsequently, the obtained degree of superheating with the imposed pump rotational speed is compared with the value imposed as set-point (5 K) and the iterative loop is closed when a difference lower than 0.5 K has been found. The main performance parameters, that are the net electrical production ($\dot{W}_{net}$) and the HTF outlet temperature ($T_{HTF,o}$) are calculated.

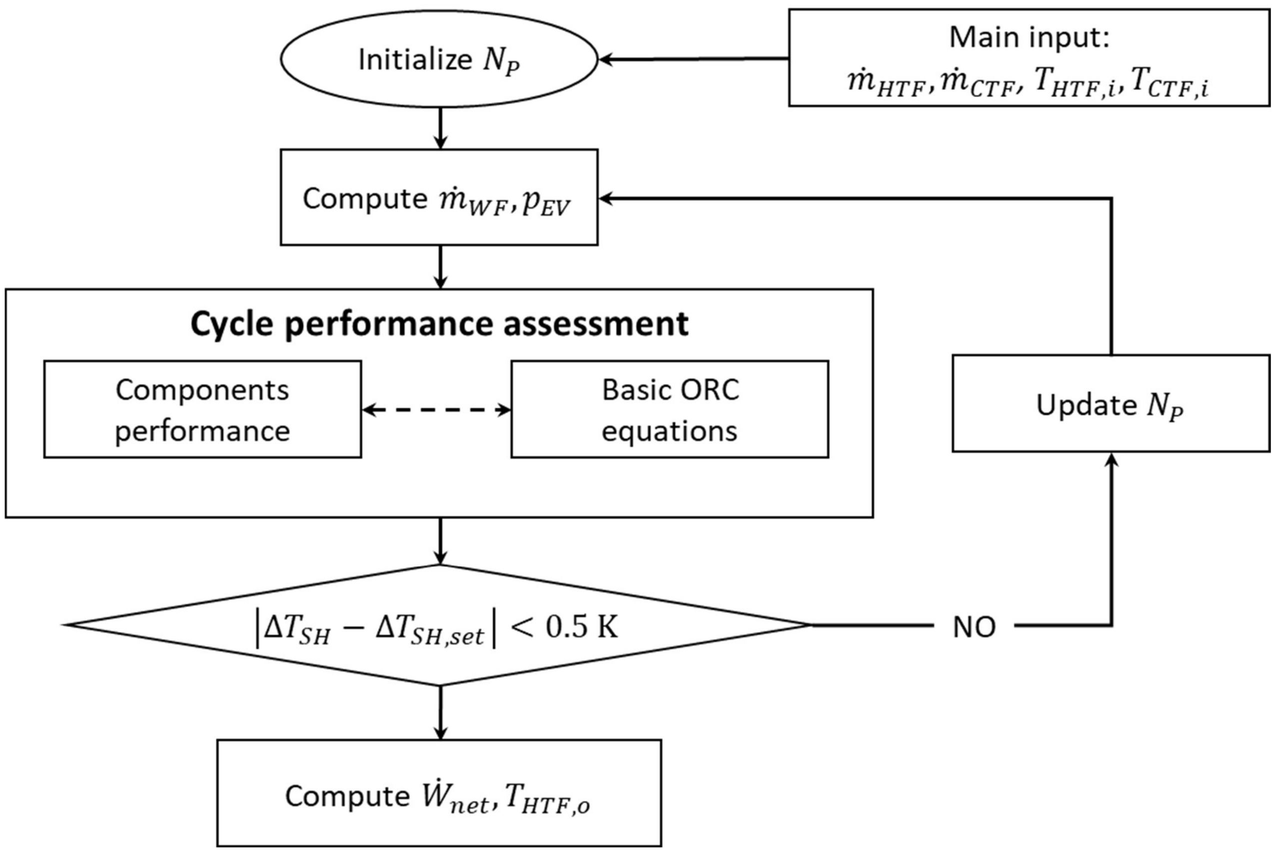

Figure 4 – Model architecture for the prediction of ORC unit performance.

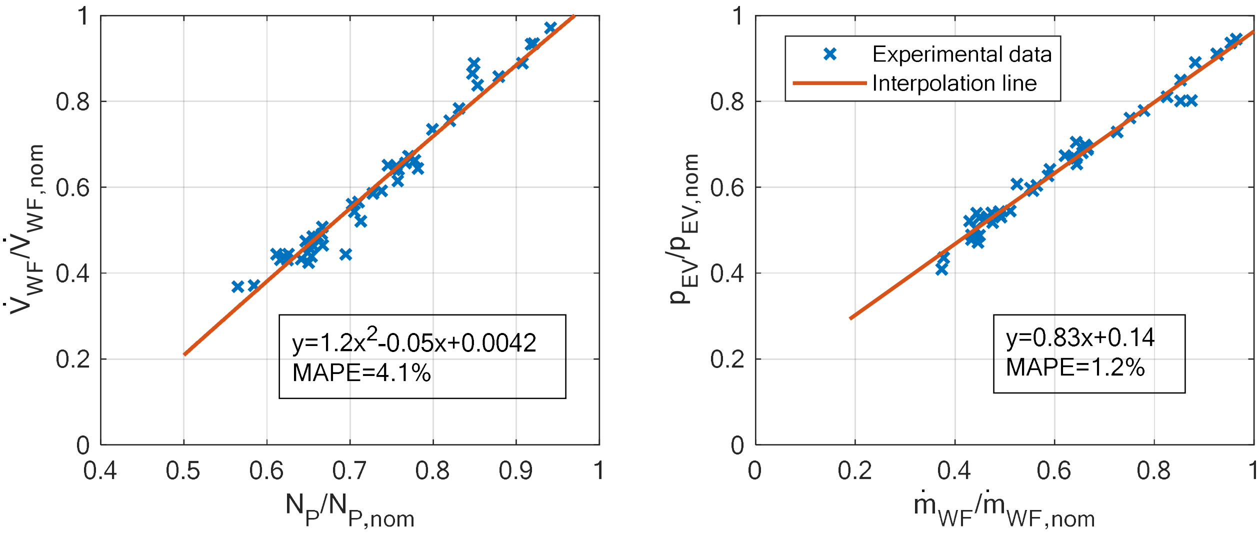

Figure 5 -Correlations adopted in mathematical model for the calculation of organic fluid flow rate and operative evaporating pressure.

3.3. Model pre-validation

A preliminary validation of the mathematical model has been carried out by using the operating conditions (in terms of HTF and CTF mass flow rate and inlet temperature) adopted as test dataset and the values of the component performance parameters obtained during the experimental campaign. In this way, the reliability of the resolution scheme and the impact of the model assumptions (i.e. negligible pressure losses, constant sub-cooling etc.) are investigated. Figure 6 shows the main model outputs (i.e. net electrical power and HTF outlet temperature) in comparison with the reference data obtained during the experimental activity by means of parity plots. The MAPE is used to illustrate comprehensively the precision of the model in terms of net power (a MAPE of about 1% have been found for this output), while the Mean Absolute Error (MAE) is proposed as figure of merit for the HTF outlet temperature:

$$\mathrm{MAE} = \frac{1}{N}\sum_{i=1}^{N}\left|x_{model} - x_{exp}\right| \tag{5}$$

obtaining an average error of about 1 K. In addition, the coefficient of determination (R^2) is also determined to show the accuracy of the simulation model based on the data variability:

$$R^2 = \frac{\sum_{i=1}^{N}(x_{model} - \bar{x})^2}{\sum_{i=1}^{N}(x_{exp} - \bar{x})^2} \tag{6}$$

where $\bar{x}$ is the mean value of the data obtained during the experimental activity.

Since in this validation test, the component behaviour is perfectly forecasted, the variance obtained between simulation results and experimental data is due to the simplified assumptions done in terms of constant degree of superheating at the evaporator outlet, constant degree of subcooling at the condenser outlet and negligible pressure losses, the residuals observed in using empirical correlations for the determination of the organic fluid mass flow rate and evaporating pressure as well as in the accuracy errors related to the sensors used for the monitoring of the ORC performance.

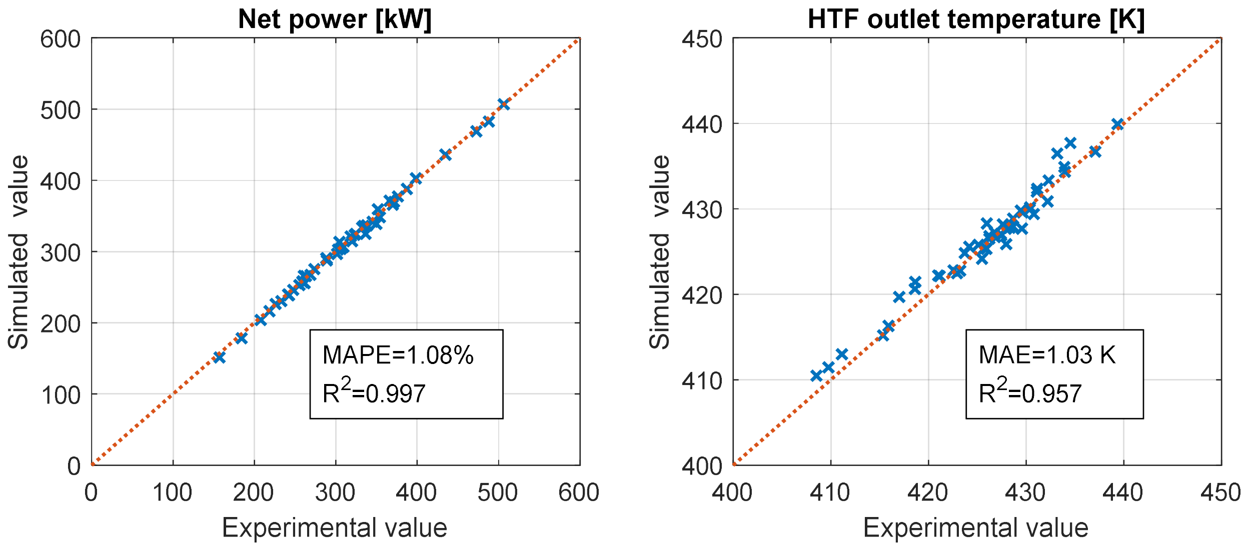

Figure 6 – Parity plots of the net power production (left side) and HTF outlet temperature by imposing the actual values of the component performance parameters.

4. Results and discussion

The results in terms of accuracy of the mathematical model developed for the performance prediction of the ORC unit by following two different approaches for the characterization of the ORC components behaviour under off-design conditions is explained and discussed in this section. The components performance parameters have been tuned according to the training dataset while the ORC mathematical model has been tested by considering a different dataset, as mentioned in the previous section. Figure 7 shows the comparison between the performance obtained in terms of net power and HTF outlet temperature via simulation and the corresponding values obtained during experimental activities by adopting a constant-efficiency approach.

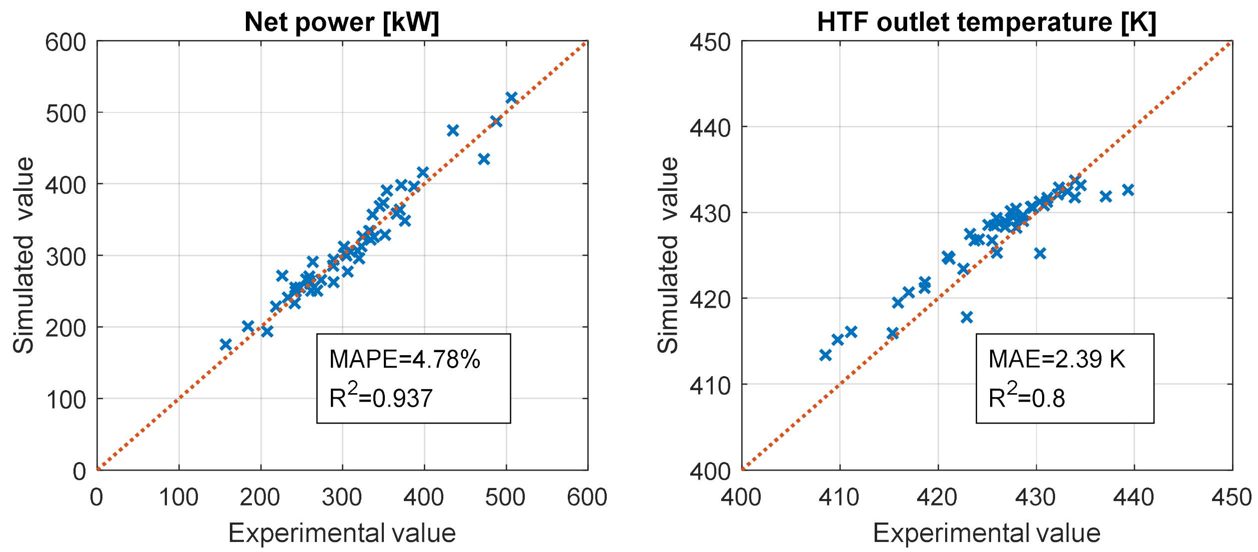

Figure 7 - Parity plots of the net power production (left side) and HTF outlet temperature obtained with the constant-efficiency approach.

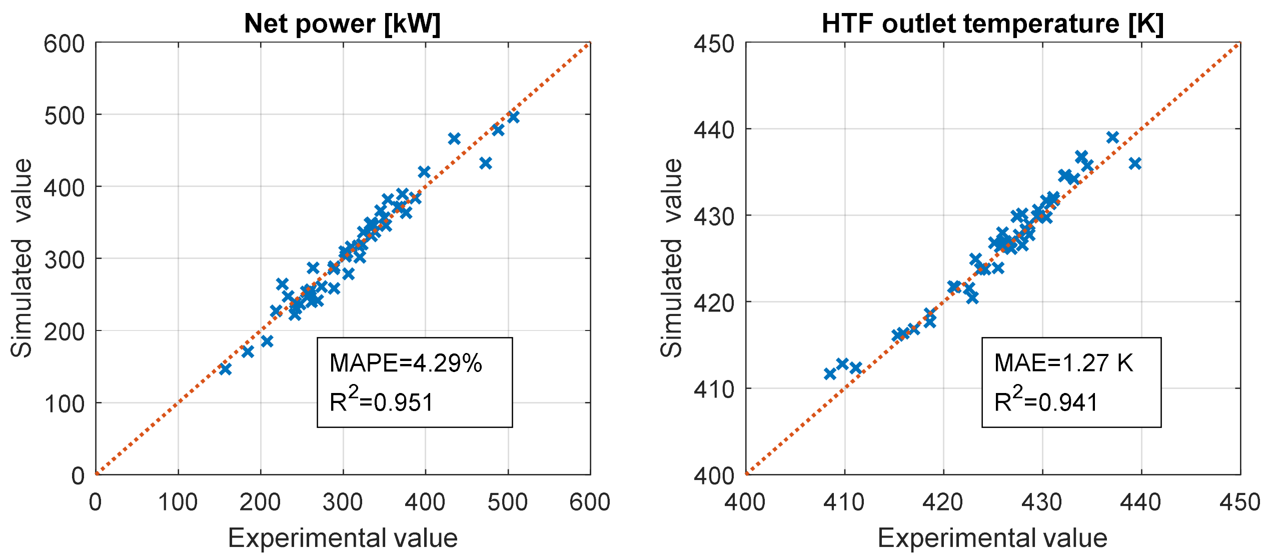

Figure 8 - Parity plots of the net power production (left side) and HTF outlet temperature obtained with the correlation-based approach.

As expected, an increase of the mean absolute error both for the ORC net power and HTF outlet temperature occurs with the introduction of performance parameters that foresee the performance of heat exchangers and turbomachines compared to the results obtained during the pre-validation model step. This increment is particularly evident for the calculation of the net power, but the residuals are uniformly distributed along the range of variation. On the other hand, the MAPE obtained for the HTF temperature is limited, but the mathematical model tends to overestimate this variable for low values and to underestimate it if high temperature values are expected.

As shown in Figure 8, the introduction of a polynomial correlations mitigates this effect in addition to reduce on average the error. An average reduction of the errors is also observed for the ORC net power thanks to the better estimation of the components performances by varying the operating conditions.

To better understand the uncertainty introduced with the use of non-physical approaches and the consequent effects on the overall accuracy of the simulation model, Figure 9 depicts the error distribution committed by these two approaches while predicting the components efficiencies. By referring to the heat exchangers, the evaporator model (characterized by effectiveness and insulation efficiency) is the component introducing the main uncertainty in the model, with a percentage deviation from the actual value in the range of 10% and 20% for ε_{EV} and $\eta_{EV,ins}$ respectively, leading to the main benefits in using polynomial correlations instead of constant values.

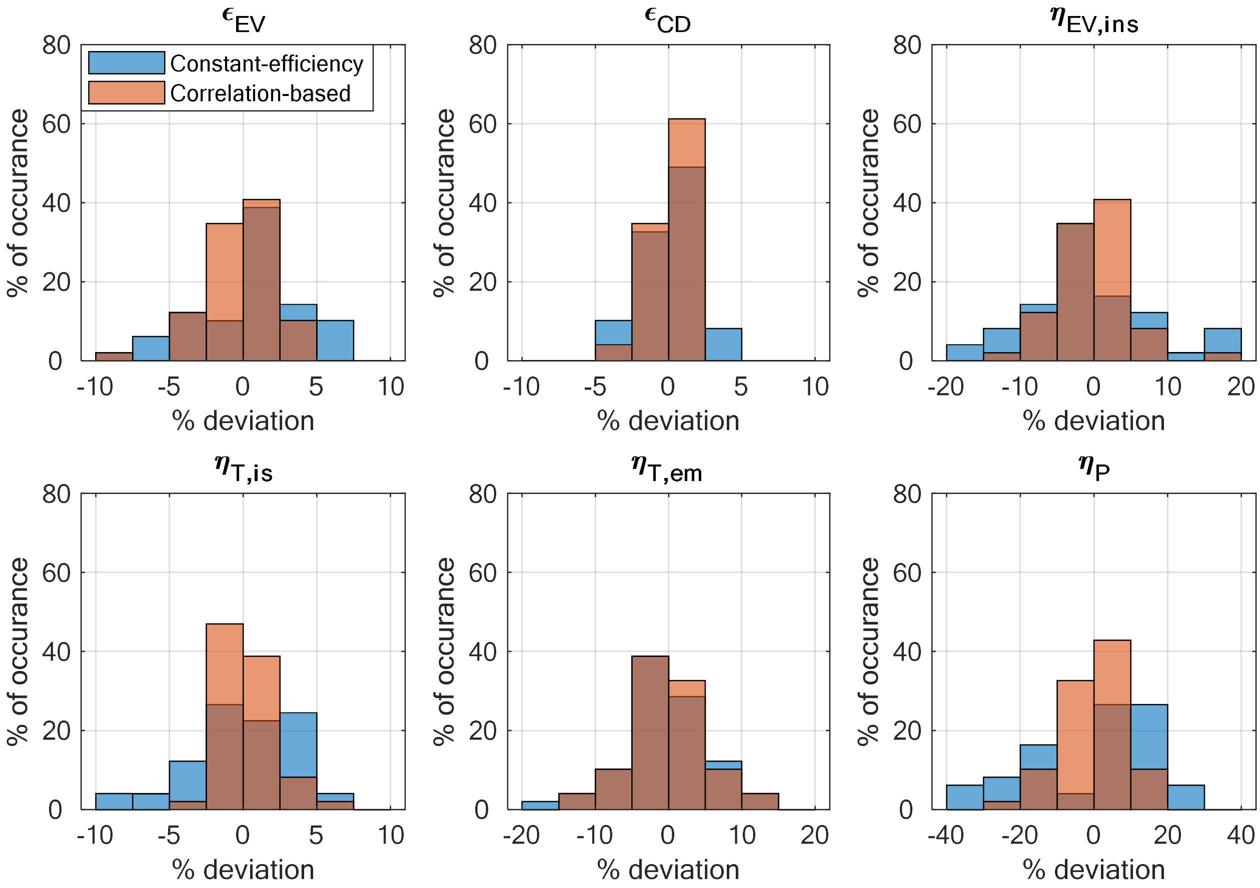

Figure 9 – residual distribution for the main six component performance parameters by using a constant-efficiency approach and a correlation-based approach.

In this regard, a relationship between the residual distribution and the thermal power input is observed: the lower the thermal power input is, the higher the evaporator effectiveness is. Consequently, the positive deviations observed for this parameter applying the constant-efficiency approach occurs for low thermal power input and vice versa, explaining the behaviour observed for $T_{HTF,o}$ in the previous figure. As already reported Section 3.1, the use of constant values for the characterization of the recuperator performance results in a very small inaccuracy as well as for the insulation efficiency of the condenser (due to the lower temperature difference between the CTF and environment). Furthermore, the residuals in the condenser effectiveness forecast value ranges between ±5% and the use of a correlations-based approach for the determination of this parameter leads to marginal benefits.

By referring to the turbomachinery performances, because of the influence of the off-design conditions on the turbine isentropic efficiency, important benefits arise from the use of proper correlations that take into account the fluctuation on the working fluid flow rates and enthalpy drops. On the other hand, marginal improvements are observed in using polynomial correlations for the calculation of the turbine electro-mechanical efficiency, while more significant deviations are found in comparison with the previous parameter. Finally, the pump efficiency shows the most significant fluctuations from its mean value (±40% by using a constant-efficiency approach) and the use of polynomial correlations with the rotational speed can only mitigate this effect. However, this uncertainty has a marginal effect on the main outputs.

This point is clearly evidenced in Figure 10, where the mean absolute percentage error found for each component efficiency is correlated with the variable importance in the determination of the two main outputs. This second parameter is computed by the decomposition of the ORC parameter variances by means of ANOVA [26], providing the main contributions to the total variance of the ORC performance (net power and HTF outlet temperature), expressed in terms of first order Sobol' indices.

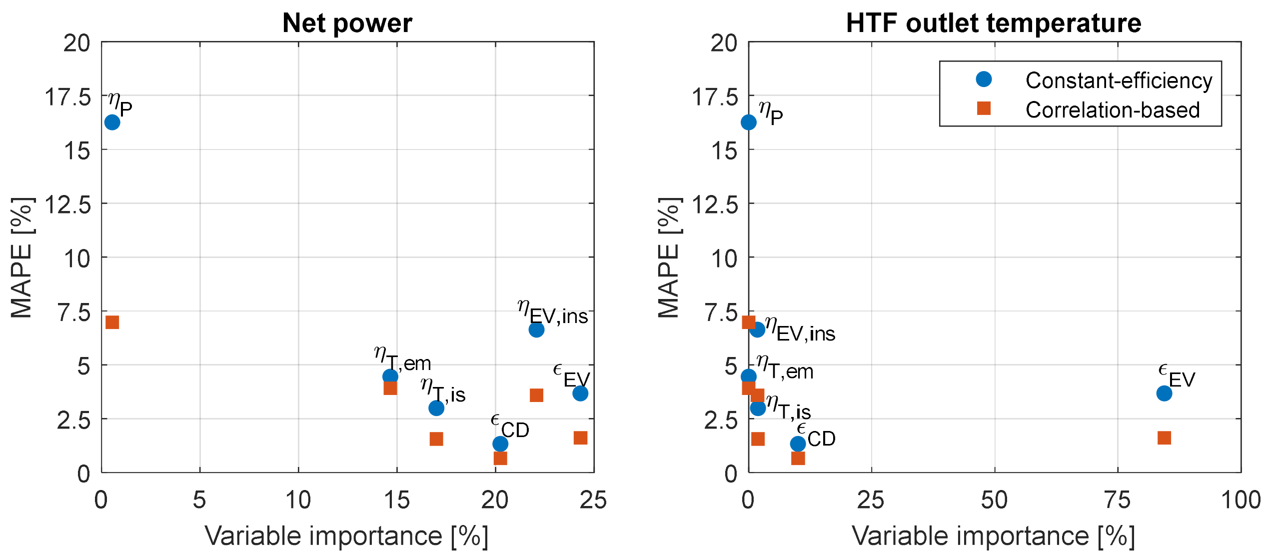

Figure 10 - Evaluation of the performance parameters contribution to the total variance for the ORC net power and HTF outlet temperature vs observed MAPE.

As can be observed, although the pump efficiency is the parameter with the highest error, its effect on the net power production and HTF outlet temperature is minimal and its contribution on the overall uncertainty of the mathematical model is marginal. The performance parameters proposed for the turbine simulations ($\eta_{T,is}$ and $\eta_{T,em}$) have a greater importance for the calculation of the net power and for the increase of accuracy of the correlation-based model while they are also marginal for the HTF outlet temperature. The parameters with the highest impact on the system performances are those correlated to the two-phase heat exchangers, especially the evaporator. In this case, the introduction of polynomial correlations for determining the expected performance of this device based on the operating conditions leads to the main benefits in using this approach, due to the high sensitivity of the main outputs to these variables. This fact is particularly significant for the evaporator effectiveness, which has the highest Sobol index both by referring to the net power production and the HTF outlet temperature.

Overall, the effect of using the two different approaches for the estimation of the net power produced by the ORC unit under different operating conditions is shown in Figure 11. In particular, Figure 11(a) shows the variation in the net power with HTF inlet temperature and flow rate (CTF inlet temperature set equal to 300 K) while Figure 11(b) shows the variation of the net power with the CTF inlet temperature by assuming a heat source characterized by an inlet temperature of 500 K and a volumetric flow rate of 60 m^3/h. By adopting a constant-efficiency approach, the net power varies almost linearly with the operating conditions with an obvious benefit in rising the HTF inlet temperature and flow rate and a drawback in rising the CTF inlet temperature. Conversely, the adoption of a correlation-based approach introduces non-linear effects on the expected trend, particularly evident for the higher HTF temperatures and flow rates. Furthermore, the use of polynomial correlations emphasizes the effect of the CTF inlet temperature variation on the system performance: more significant degradations in the ORC performance are observed compared to those obtained with the other approach, mainly due to the increase of the saturation pressure in the condenser.

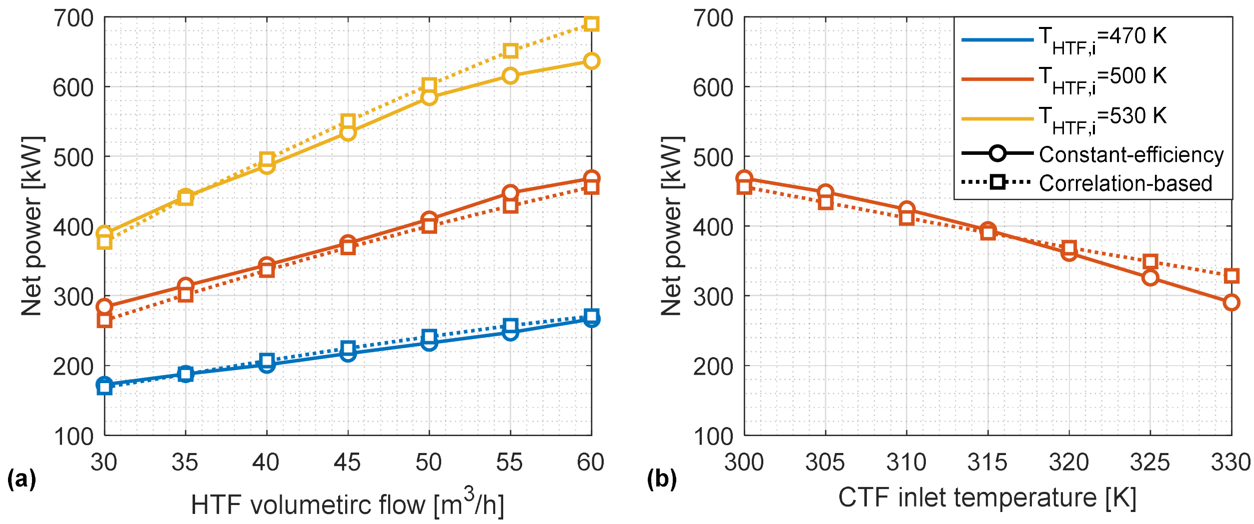

Figure 11 – Effects of (a) HTF volumetric flow rate, HTF inlet temperature and (b) CTF inlet temperature on the ORC net power by using a constant-efficiency approach and a correlation-based approach.

It is therefore evident the better performance obtained by the adoption of polynomial correlations for the prediction of ORC performances. On the other hand, higher computational effort is required for the resolution of the mathematical problem in this case, due to the introduction of further correlations and the increase of number of iterations needed to find stationary values. In this regard, Figure 12 shows the frequency distribution of the computational times required for the resolution of the ORC model by referring to the two approaches proposed (simulations performed with a pc Dell XPS8900, CPU Intel Core i7-6700 3.4 GHz, 16 GB RAM). As expected, the average computational time required for simulating the ORC model using a correlation-based approach increases by about 60% compared to that obtained with the simpler approach. A higher variance is also observed by adopting this approach, since the model becomes more sensitive in terms of iterations required to find the state-point to the imposed initial values.

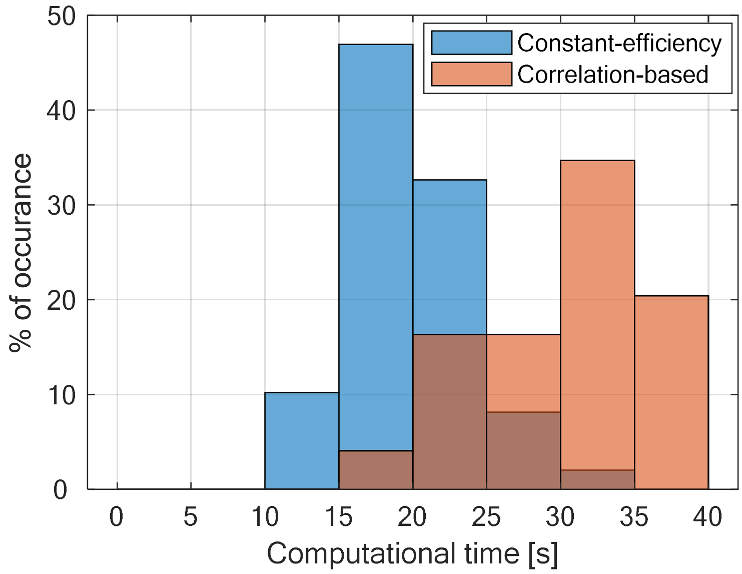

Figure 12 – Probability distribution of the computational time required for the resolution of the mathematical model by following the two proposed approaches.

4.1. Start-up phase analysis

Finally, an analysis on the start-up phase of ORC units and the operating time and energy required during this phase has been investigated. In general, the start-up phase of an ORC unit means all the operations required to bring the ORC unit from a state of thermal equilibrium with the environment and turbomachinery not in operation to a fully operating state, with the working fluid following a cycle able to produce useful work and the electrical generator operating in parallel with the grid. Obviously, several steps are followed during this phase, which usually involves a first pre-heating of the working fluid and all the circuit, the gradual starting-up of the turbine up to the design rotational speed, the consequent parallel with the grid of the electrical generator and the regular increase of the rotational speed of the thermal oil pump (with the consequent rise in the HTF mass flow rate) up to reach the desired power output. Obviously, the supplying of a certain amount of thermal power input is required during this phase to face the thermal inertia of the working fluid and the various components, to face the mechanical inertia of turbine and so on. Furthermore, to avoid important thermal stress and to guarantee a proper charging of the working fluid in all the main components, a progressive preheating is imposed, with the consequent release to the heat sink of part of thermal power input. As example, Figure 13 shows the main experimental data collected during a start-up phase (duration of about 35 min) of the ORC unit at the Ottana solar facility. By referring to Figure 13(a), it is evident the unbalance between the inlet and outlet thermal power recorded in the start-up phase, since a share of the energy input is used to heat up the organic fluid and face the thermal inertia of all the components. The increase of the rotational speed of the turbine begins about 20 min later (Figure 13(b)) and 10 min are required to reach the design rotational speed. Finally, in the last step the gradual increase of the evaporator pressure up to the operative values is observed (Figure 13(c))

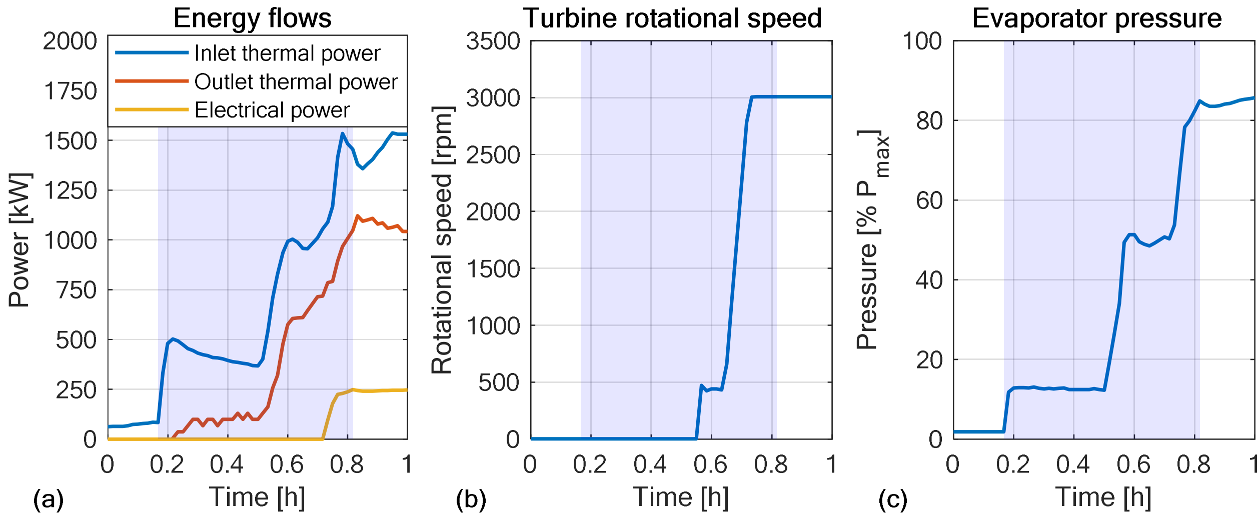

Figure 13 – Trend of the (a) main energy flows (b) turbine rotational speed and (c) evaporator pressure recorded during a typical start-up phase of the ORC unit at the Ottana solar facility (in evidence the so-called start-up time).

The energy and time requirements of the start-up phase are not influential if the ORC unit is continuously fed by a dispatchable source, such as biomass, but it becomes significant if daily start-up/shut down of the turbogenerator are expected, as in the case of CSP-ORC power plants. In this case, a proper characterization of the energy demanded during this phase as well as the time required to complete all the operations should be conducted during both a design-stage (in particular during feasibility analysis) and during an operating phase (in particular during the definition of the control strategy to be followed).

For this reason, a preliminary analysis of the energy demanded during the start-up phase and the corresponding time required is carried out based on the experimental data collected during 75 different operating days at the Ottana solar facility. Figure 14 shows the frequency distribution of the two main variables investigated, that are, the periods needed to the ORC unit to complete the start-up operation and the thermal energy absorbed during this phase.

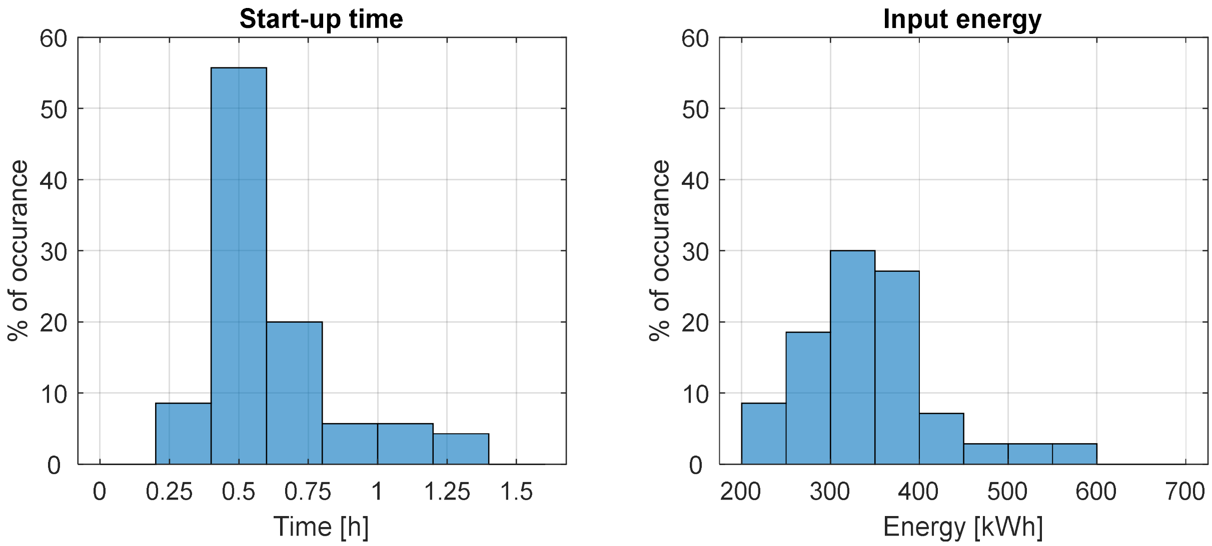

Figure 14 – Frequency distribution of the operating time required during the ORC starting phase and the corresponding demanded energy (based on measurements).

Apart from some sporadic case, the time required by the ORC unit to complete the starting up is around 30-45 min. This data is important during the operating phase if the ORC unit wants to follow a given power output profile with constrained delivery time. The average energy demanded (which represents the difference between the thermal energy inlet and thermal energy output occurring during the start-up phase, as shown in Figure 12(a)) during this phase is about 350 kWh, supplied in the investigated case through and HTF heated up in a solar field. However, a higher variance of this parameter is observed, compared to the previous one, ranging from 200 to 600 kWh.

Several parameters influence these two main outputs: heat source and heat sink characteristics during the starting up, ambient temperature, the period occurred between the last shutting down and the followed starting up and so on. Consequently, a specific physical model should be implemented for a reliable prediction of the expected time and the demanded energy during a start-up phase. On the other hand, a simpler empirical correlation approach is here proposed based on two parameters, that are the HTF inlet temperature (usually, the HTF mass flow rate is automatically controlled) and the average working fluid temperature at the beginning of the start-up phase. The correlations proposed are reported in Appendix B. In general, the HTF inlet temperature strongly affects the operations of starting up of the ORC unit, in particular the operating time required to complete this phase. However, this parameter influences the start-up time and the input energy in a not univocal way. In fact, on the one hand the increase of the HTF inlet temperature accelerates the start-up phase, on the other hand, the need of a controlled and gradual warming up of the organic fluid and main components, which leads to the need of partially realise the thermal power input, together with the increase in the insulation losses results in an increase of the overall input energy required during the start-up phase with the rise in the HTF inlet temperature. The average working fluid temperature is the second parameter used in this analysis. The state of the working fluid before the beginning of the start-up phase mainly depends on the final conditions of the fluid at the last shutting down of the ORC unit, the time elapsed between the last shutting-down and following starting up and the environment conditions occurred in this period. The variation in the average working fluid temperature has a minor effect on the start-up energy demand compared to the previous one, since it influences only the first starting-up operations, aimed at preheating the fluid and the mechanical components. Obviously, the lower the average working fluid temperature is, the higher the external energy required and the duration of the preheating are. Figure 15 shows the comparison of the simulation results obtained with the proposed correlations with the actual duration and input energy measured during different 75 starting up of the ORC unit at the Ottana solar facility. The accuracy obtained in this case is lower than that obtained for the ORC predictive model, as confirmed by the mean absolute percentage error observed. As mentioned, the start-up phase demands a series of operations in which various aspects are involved and the use of only two parameters is restricted in a certain way for the prediction of the overall duration and energy demand. The subdivision of the overall start-up phase in various steps and the use of multiple correlations could improve the accuracy of the predictions. However, the proposed approach could give a general indication of the duration and energy required during this phase, which could be useful for both a techno-economic analysis and the definition of daily operations of the ORC unit.

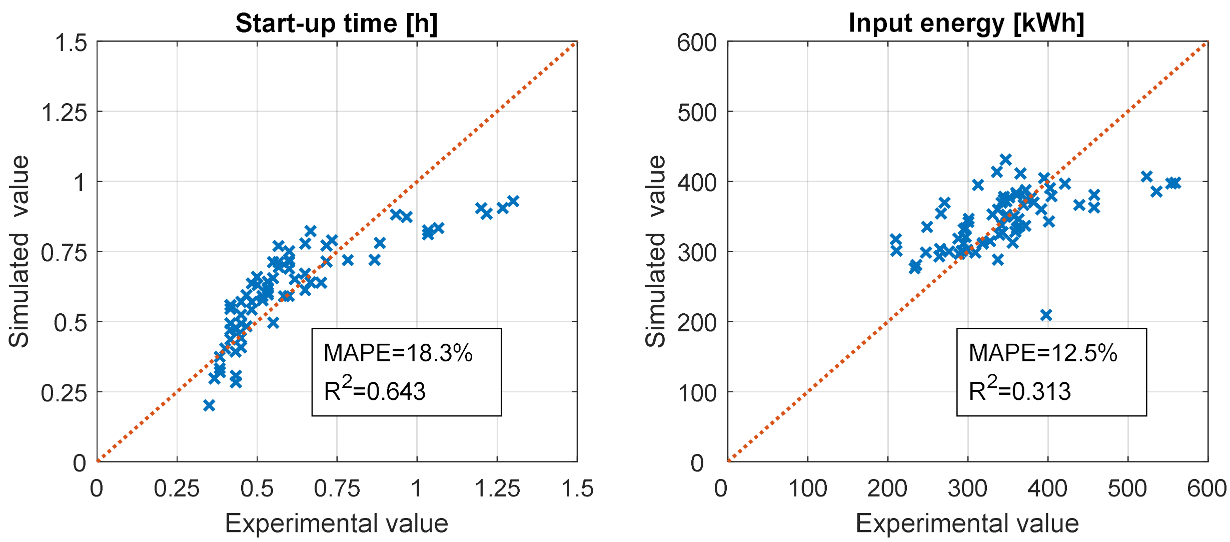

Figure 15 - Parity plots of the time required to complete the start-up phase (left side) and the corresponding energy absorbed during this phase.

5. Conclusions

Organic Rankine Cycle is an interesting technology for valorising the thermal-to-electricity conversion of medium- and low-grade heat sources, but often ORC units operate at off-design conditions due to the variable conditions of the heat sources and heat sinks coupled with these energy systems. The development of robust and reliable models able to predict the ORC performance in a large range of operating conditions is therefore crucial both for optimal system design and for a proper control strategy during the operating time.

In this paper, a numerical tool able to predict the performance of an ORC unit for a given operating domain and based on experimental data was proposed. Unlike a physical approach for the implementation of performance prediction models for the main system components, simpler approaches aimed at reducing the computational time needed for the performance prediction were investigated. The simulation of heat exchangers and turbomachines were based on nine performance parameters calculated following two different approaches: in the first approach, the performance parameters were imposed constant and equal to the mean values observed during an experimental campaign; in the other approach, the variations of these performance parameters with the operating state were considered via polynomial correlations.

The mathematical model was developed and tested based on a medium-size CSP-ORC plant installed at the Ottana solar facility (Italy). The experimental results collected onsite were used both as test set for calibrating the ORC model parameters and as training set for the model validation.

The results demonstrated that the numerical tool proposed is characterized by low computational time (on average, about 30 seconds were required for the resolution of the mathematical model) while keeping a good achievement in terms of accuracy of the model and reliability of the simulation performance, with average errors lower than 5% and 2.5 K for the expected net power and outlet oil temperature respectively. The use of polynomial correlations led to better results in terms of accuracy thanks to a better approximation of the components efficiencies. In particular, it was demonstrated that the main benefits arise from the more accurate prediction of the evaporator effectiveness, which is characterized by the highest variance importance (about 25% and 80% in terms of first order Sobol' indices by referring to net power and the HTF outlet temperature, respectively) and the adoption of correlation-based approach reduces the mean percentage error from 3.6% to 2.6%. The accuracy achieved is in line with other models proposed in literature for small scale ORC units (see [8,12,20,27]). The proposed mathematical model could therefore be suitable for its implementation if a fast response from the ORC model is required or if the simulation of this model is repeated innumerable times.

Although the mathematical model was experimentally validated for medium-size solar applications, it can be also adopted for the simulation of the expected ORC performance coupled with a different heat source (such as biomass, WHR etc). However, it must be stressed that the design of the ORC unit was optimized for a solar application with given heat sources and heat sink characteristics and the experimental dataset adopted for the validation is dependent from the plant management system. For this reason, it is important to previously define the operating domain in terms of boundary conditions in which the ORC unit should operate, since the extrapolability of the results could be unreliable.

Finally, a preliminary assessment about the energy requirement and the duration of the start-up phase of medium-size ORC unit was carried out based on the experimental data collected during 75 different operating days. An energy demanded during this phase ranging from 200 to 600 kWh with a corresponding duration of 30-45 min was observed. Empirical correlations based on the HTF inlet temperature and the average working fluid temperature at the beginning of the start-up phase were therefore proposed for the prediction of the time required and energy demanded during this phase. Although the results obtained in terms of accuracy could be improved (a mean percentage error of 18.5% and 12.5% was achieved for the start-up time and the corresponding energy absorbed, respectively), these correlations can give an initial indication about the duration and energy required during this phase, which could be useful for both a techno-economic analysis and the definition of daily operations of the ORC unit.

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Appendix A Steady-state model of the organic Rankine cycle unit

This appendix provides the constitutive equations introduced in the methods presented in section 3. For the identification of the thermodynamic state points, the working fluid condition at the inlet or outlet side of the main ORC component are used as reference. For a better understanding of the sub-model interaction, the architecture of the ORC model is depicted in Figure A.1.

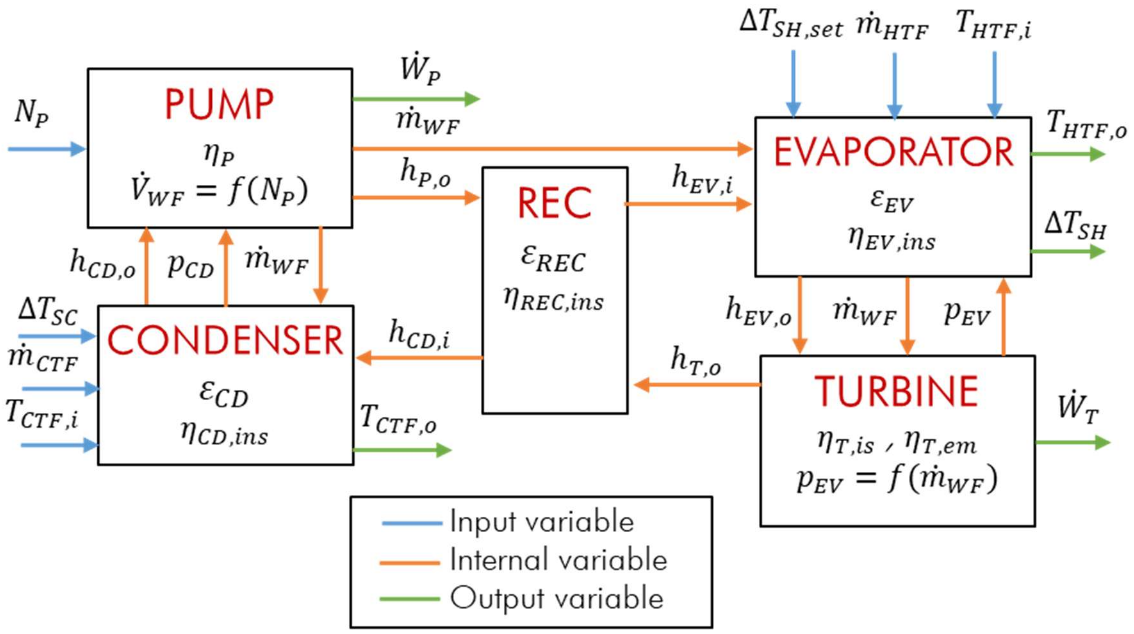

Figure A.1 - Solver architecture of the ORC model.

Appendix A.1. Basic ORC equations

The main equations related to energy balances and effectiveness of the heat exchangers are listed below:

- *Evaporator*

$$\dot{m}_{HTF}c_{HTF}\left(T_{HTF,i} - T_{HTF,o}\right)\eta_{EV,ins} = \dot{m}_{WF}\left(h_{EV,o} - h_{EV,i}\right) \tag{A.1}$$

$$\varepsilon_{EV} = \dot{m}_{WF}\left(h_{EV,o} - h_{EV,i}\right)/\dot{Q}_{MAX,EV} \tag{A.2}$$

- *Condenser*

$$\dot{m}_{CTF}c_{CTF}\left(T_{CTF,o} - T_{CTF,i}\right) = \dot{m}_{WF}\left(h_{CD,i} - h_{CD,o}\right)\eta_{CD,ins} \tag{A.3}$$

$$\varepsilon_{CD} = \dot{m}_{CTF} c_{CTF} \left(T_{CTF,o} - T_{CTF,i}\right) / \dot{Q}_{MAX,CD} \quad (A.4)$$

- *Recuperator*

$$(h_{T,o} - h_{CD,i}) \eta_{REC,ins} = (h_{EV,i} - h_{P,o}) \quad (A.5)$$

$$\varepsilon_{REC} = \dot{m}_{WF} (h_{EV,i} - h_{P,o}) / \dot{Q}_{MAX,REC} \quad (A.6)$$

- *Pump*

$$\dot{W}_P = \dot{m}_{WF} \frac{\left(h_{P,o} - h_{CD,o}\right)}{\eta_P} \quad (A.7)$$

- *Turbine*

$$\dot{W}_T = \dot{m}_{WF} \left(h_{EV,o} - h_{T,o}\right) \eta_{T,em} \quad (A.8)$$

$$\eta_{T,is} = \left(h_{EV,o} - h_{T,o}\right) / \left(h_{EV,o} - h_{T,o,is}\right) \quad (A.9)$$

The maximum possible heat transfer $\dot{Q}_{MAX}$ is calculated according to [28] as the heat transfer leading to a temperature difference at pinch point equal to 0 K (without any assumption on the pinch point location). The mass flow rates, pressures and temperatures/enthalpies of the hot and cold fluids are used as main inputs.

Appendix A.2. Correlations used during the correlation-based approach

The effectiveness of the evaporator and condenser as well as the evaporator efficiency are estimated by quadratic polynomials based on the HTF mass flow rate ($\dot{m}_{HTF}$) and organic fluid mass flow rate ($\dot{m}_{WF}$):

$$\varepsilon_{EV} = 0.692 + 0.0671 \dot{m}_{HTF} - 0.0105 \dot{m}_{WF} - 0.00279 \dot{m}_{HTF}^2 - 0.0018 \dot{m}_{HTF} \dot{m}_{WF} + 0.000719 \dot{m}_{WF}^2 \quad (A.10)$$

$$\varepsilon_{CD} = 8.782 - 0.2456 \dot{m}_{CTF} + 0.0262 \dot{m}_{WF} + 0.00193 \dot{m}_{HTF}^2 - 0.00052 \dot{m}_{HTF} \dot{m}_{WF} - 0.00013 \dot{m}_{WF}^2 \quad (A.11)$$

$$\eta_{EV,ins} = 1.54 - 0.1558 \dot{m}_{HTF} + 0.016 \dot{m}_{WF} + 0.00513 \dot{m}_{HTF}^2 + 0.00893 \dot{m}_{HTF} \dot{m}_{WF} - 0.00546 \dot{m}_{WF}^2 \quad (A.12)$$

According to Ghasemi et al. [29], the isentropic efficiency of the turbine is calculated as a function of ratio of enthalpy drop (r_H) and ratio of volumetric flow rate (r_V):

$$\eta_{T,is} = -10.02 + 18.35\, r_H + 3.552\, r_V - 7.654\, r_H^2 - 3.006\, r_H r_V - 0.4168 r_V^2 \quad (A.13)$$

Where

$$r_H = \sqrt{\frac{\left(h_{EV,o} - h_{T,o,is}\right)}{max\left(h_{EV,o} - h_{T,o,is}\right)}} \quad (A.14)$$

$$r_V = \sqrt{\frac{\dot{m}_{WF} / \rho_{EV,o}}{max\left(\dot{m}_{WF} / \rho_{EV,o}\right)}} \quad (A.15)$$

Finally, the turbine electro-mechanical efficiency and the pump efficiency are characterized by quadratic polynomials based on a single independent variable, namely, the net electrical power and the pump rotational speed respectively.

$$\eta_{T,em} = 0.967 + 0.0002675\, \dot{W}_T \quad (A.16)$$

$$\eta_P = -0.8959 + 0.002901 N_P - 1.47 \cdot 10^{-6} N_P^2 \tag{A.17}$$

Appendix B Correlations used for the start-up phase analysis

The correlations proposed for the prediction of the time required for the conclusion of the start-up phase (t_{SU}, expressed in hours) and the correspondent thermal energy demanded (Q_{SU}, expressed in kWh) are based on HTF inlet temperature ($T_{HTF,i}$) and the average temperature of the organic fluid at the beginning of the start-up phase ($T_{WF,SU}$).

$$t_{SU} = -0.415 + \frac{597.8}{T_{HTF,i}} + \frac{1090}{T_{WF,SU}} \tag{B.1}$$

$$Q_{SU} = \left(-0.3105 - \frac{161.1}{T_{HTF,i}} + \frac{315.9}{T_{WF,SU}}\right) \cdot 10^3 \tag{B.2}$$

where

$$T_{WF,SU} = \frac{T_{CD,i} + T_{CD,o} + T_{EV,i} + T_{EV,o} + T_{T,o}}{5} \tag{B.3}$$

References

[1] European Commission. Energy Roadmap 2050. Brussels: 2011.

[2] Quoilin S, Broek M Van Den, Declaye S, Dewallef P, Lemort V. Techno-economic survey of Organic Rankine Cycle (ORC) systems. Renew Sustain Energy Rev 2013;22:168–86. doi:10.1016/j.rser.2013.01.028.

[3] Tchanche BF, Lambrinos G, Frangoudakis A, Papadakis G. Low-grade heat conversion into power using organic Rankine cycles - A review of various applications. Renew Sustain Energy Rev 2011;15:3963–79. doi:10.1016/j.rser.2011.07.024.

[4] Colonna P, Casati E, Trapp C, Mathijssen T, Larjola J, Turunen-Saaresti T, et al. Organic Rankine Cycle Power Systems: From the Concept to Current Technology, Applications, and an Outlook to the Future. J Eng Gas Turbines Power 2015;137. doi:10.1115/1.4029884.

[5] Pethurajan V, Sivan S, Joy GC. Issues, comparisons, turbine selections and applications – An overview in organic Rankine cycle. Energy Convers Manag 2018;166:474–88. doi:10.1016/J.ENCONMAN.2018.04.058.

[6] Dickes R. Charge-sensitive methods for the off-design performance characterization of organic Rankine cycle (ORC) power systems. University of Liege, 2019.

[7] Park BS, Usman M, Imran M, Pesyridis A. Review of Organic Rankine Cycle experimental data trends. Energy Convers Manag 2018;173:679–91. doi:10.1016/j.enconman.2018.07.097.

[8] Wang T, Liu L, Zhu T, Gao N. Experimental investigation of a small-scale Organic Rankine Cycle under off-design conditions: From the perspective of data fluctuation. Energy Convers Manag 2019;198:111826. doi:10.1016/j.enconman.2019.111826.

[9] Zhang H-H, Xi H, He Y-L, Zhang Y-W, Ning B. Experimental study of the organic rankine cycle under different heat and cooling conditions. Energy 2019;180:678–88. doi:10.1016/j.energy.2019.05.072.

[10] Manente G, Toffolo A, Lazzaretto A, Paci M. An Organic Rankine Cycle off-design model for the

search of the optimal control strategy. Energy 2013;58:97–106. doi:10.1016/j.energy.2012.12.035.

[11] Usman M, Imran M, Yang Y, Lee DH, Park B-S. Thermo-economic comparison of air-cooled and cooling tower based Organic Rankine Cycle (ORC) with R245fa and R1233zde as candidate working fluids for different geographical climate conditions. Energy 2017;123:353–66. doi:10.1016/j.energy.2017.01.134.

[12] Pang KC, Hung TC, He YL, Feng YQ, Lin CH, Wong KW. Developing ORC engineering simulator (ORCES) to investigate the working fluid mass flow rate control strategy and simulate long-time operation. Energy Convers Manag 2020;203:112206. doi:10.1016/j.enconman.2019.112206.

[13] Hu S, Li J, Yang F, Yang Z, Duan Y. Thermodynamic analysis of serial dual-pressure organic Rankine cycle under off-design conditions. Energy Convers Manag 2020;213. doi:10.1016/j.enconman.2020.112837.

[14] Song J, Gu C, Ren X. Parametric design and off-design analysis of organic Rankine cycle (ORC) system. Energy Convers Manag 2016;112:157–65. doi:10.1016/j.enconman.2015.12.085.

[15] Kim IS, Kim TS, Lee JJ. Off-design performance analysis of organic Rankine cycle using real operation data from a heat source plant. Energy Convers Manag 2017;133:284–91. doi:10.1016/j.enconman.2016.12.016.

[16] Pili R, Romagnoli A, Jiménez-Arreola M, Spliethoff H, Wieland C. Simulation of Organic Rankine Cycle – Quasi-steady state vs dynamic approach for optimal economic performance. Energy 2019;167:619–40. doi:10.1016/j.energy.2018.10.166.

[17] Wang J, Yan Z, Zhao P, Dai Y, Woodruff GW. Off-design performance analysis of a solar-powered organic Rankine cycle. Energy Convers Manag 2014;80:150–7. doi:10.1016/j.enconman.2014.01.032.

[18] Calise F, Capuozzo C, Carotenuto A, Vanoli L. Thermoeconomic analysis and off-design performance of an organic Rankine cycle powered by medium-temperature heat sources. Sol Energy 2014;103:595–609. doi:10.1016/j.solener.2013.09.031.

[19] Oyekale J, Petrollese M, Heberle F, Brüggemann D, Cau G. Exergetic and integrated exergoeconomic assessments of a hybrid solar-biomass organic Rankine cycle cogeneration plant. Energy Convers Manag 2020;215:112905. doi:10.1016/j.enconman.2020.112905.

[20] Dickes R, Dumont O, Daccord R, Quoilin S, Lemort V. Modelling of organic Rankine cycle power systems in off-design conditions: An experimentally-validated comparative study. Energy 2017;123:710–27. doi:10.1016/j.energy.2017.01.130.

[21] Petrollese M, Cau G, Cocco D. The Ottana solar facility: dispatchable power from small-scale CSP plants based on ORC systems. Renew Energy 2020;147:2932–43. doi:10.1016/j.renene.2018.07.013.

[22] Massimiani A, Palagi L, Sciubba E, Tocci L. Neural networks for small scale ORC optimization. Energy Procedia 2017;129:34–41. doi:10.1016/j.egypro.2017.09.174.

[23] Van Erdeweghe S, Van Bael J, Laenen B, D'haeseleer W. Design and off-design optimization procedure for low-temperature geothermal organic Rankine cycles. Appl Energy 2019;242:716–31. doi:10.1016/J.APENERGY.2019.03.142.

[24] Bufi EA, Camporeale SM, Cinnella P. Robust optimization of an Organic Rankine Cycle for heavy duty engine waste heat recovery. Energy Procedia 2017;129:66–73. doi:10.1016/j.egypro.2017.09.190.

[25] Petrollese M, Cocco D. Robust optimization for the preliminary design of solar organic Rankine cycle (ORC) systems. Energy Convers Manag 2019;184:338–49. doi:10.1016/j.enconman.2019.01.060.

[26] Chatfield C, Zidek J V. Beyond ANOVA: basics of applied statistics. 1997th ed. n.d.

[27] Alshammari F, Pesyridis A. Experimental study of organic Rankine cycle system and expander performance for heavy-duty diesel engine. Energy Convers Manag 2019;199:111998. doi:10.1016/j.enconman.2019.111998.

[28] Bell IH, Quoilin S, Georges E, Braun JE, Groll EA, Horton WT, et al. A generalized moving-boundary algorithm to predict the heat transfer rate of counterflow heat exchangers for any phase configuration. Appl Therm Eng 2015;79:192–201. doi:10.1016/j.applthermaleng.2014.12.028.

[29] Ghasemi H, Paci M, Tizzanini A, Mitsos A. Modeling and optimization of a binary geothermal power plant. Energy 2013;50:412–28. doi:10.1016/j.energy.2012.10.039.