Steady state and dynamic modelling of a 1 MW_{el} commercial waste heat recovery ORC power plant

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Abstract:

ORC power systems have been proven to be a mature technology for low quality waste heat recovery applications. ORC units stand out for their simple structure, reliability and cost-effectiveness. The non-constant nature of the energy source requires the ORC power unit to be flexible. Dynamic modelling can be adopted to evaluate and optimize the response time of a system in case of transient conditions, to develop and test control strategies, to support the tuning of the controller and to support maintenance. In this work the dynamic model of a 1 MW_{el} commercial ORC unit is presented. The dynamic model is developed based on the ThermoCycle Modelica library. The different component model are validated in steady-state against 21 measurements points. The dynamic model of the whole power unit is then developed connecting the validated component models. Different modelling approaches of various complexity are implemented to model the heat exchangers of the power system. The performance of the developed heat exchanger (HX) models are tested by running different transient simulations. The results allow identifying benefits and limitations of the tested HX modelling approaches.

Keywords:
ORC, WHR, Steady state validation, Dynamic modelling comparison, Modelica

1. Introduction

Industrial processes absorb a third of the total energy consumed in society and are characterized by poor efficiency with 25% up to 55% of energy losses in the form of waste heat [1]. Reducing industrial energy consumption represents a fundamental and strategic process to invert the rising energy prices and to lower greenhouse gas emission. In this regard the development and promotion of waste heat recovery technologies are crucial to ensure a sustainable scenario and low-emission for future industrial processes [2]. Among waste heat recovery technologies, ORC power systems have been proven to be a mature and viable technology for low quality waste [3][4][5]. Although an ORC unit is easily operated in steady-state conditions, particular care must be given during transient conditions, for example when the load demand or the flow of the waste heat suddenly changes. In these cases the quality of the fluid should be kept within acceptable ranges especially at the inlet of the expander to avoid the formation of droplets inside it. Dynamic modelling can be used to predict such phenomena and prevent them, but also for developing an optimal control strategy for transient conditions. Dynamic modelling has attracted a lot of interest in the last years. The most common program language for developing dynamic models is Modelica which was introduced in 1997 [6]. In this work a dynamic model of a 1 MW_{el} regenerative stationary ORC unit, built by the Enertime company to recover the thermal energy from the combustion process of a foundry, is presented. The model is implemented in the Modelica language using components from the ThermoCycle Modelica library. Three different modelling approaches are selected and implemented to simulate the evaporator, in particular the finite volume, the moving boundary and a semi-empirical approach are adopted. The three evaporator models are validated in steady-state together with a turbine model against a set of 21 experimental point. The validated models are then connected together to simulate the overall unit. The paper is organized as follows: in section 2 a general description of the plant is reported together with the characteristics of the installed sensors. In section 3 the modelling approach is outlined and the models of the different components are described. In section 4 the steady-state validation for the evaporator models and the turbine is reported. In section 5 the dynamic model of the overall ORC unit is presented and the effect of the different evaporator modelling approach are assessed during transient of the unit pump rotational speed. Finally in section 6 the conclusions are reported and future work is outlined.

2. ORC power plant architecture

2.1. Plant description

The organic Rankine cycle unit presented in this study is a 1 MW_{el} commercial power plant built by Enertime, an ORC manufacturing company based in Courbevoie, France. The power unit recovers waste heat from the combustion process of a hot blast cupola furnace located in Soudan near Angers in France and owned by FMGC (Fonderie et Mécanique Générale Castelbriantaise). The high quality thermal energy of the flue gases from the combustion process are initially cooled down by pre-heating the combustion air. In a second step an oil-flue gases heat exchanger further decreases the exhaust gases temperature allowing for cleaning treatment before the exhausts are sent to the chimney. The thermal energy recovered by the oil serves as heat source for the ORC unit. This configuration eliminates the need of cooler fans to dissipate the thermal energy absorbed by the oil. Electricity generation and reduced electrical consumptions are the main benefits of the installation leading to an increase of the overall efficiency of the furnace process [7]. The cupola furnace has a production capacity of 20 tons per hour with a maximum capacity of 30 tons per hour and runs 24 hours a day, 5 days a week. The amount of recovered thermal power is 5 to 6 MW_{th}. The power generated by the ORC unit is injected into the foundry electrical grid and generates 5,000 MWh_{el} per year. The ORC unit makes the existing cooling system redundant, saving another 100 MWh a year [2]. The thermal oil of the Mobiltherm 600 series [3] enters the ORC evaporator at a temperature of around 200°C. The oil loop decouples the ORC unit from the transient behaviour of the exhaust line of the combustion process avoiding hot spot in the ORC evaporator, that could damage the organic working fluid of the power unit, and allowing the ORC system to operate close to a defined nominal point. Solkatherm (SES36) is selected as working fluid as it allows to reach a good cycle efficiency despite the low temperature of the thermal energy source. The fluid is characterized by low GWP and exhibits low toxicity and flammability, characteristics that make it suitable for application in the heavy industry sector. The front view and the process flow diagram of the ORC waste heat recovery unit are shown in figure 1. The system is a regenerative cycle equipped with an in-house built multi-stage axial turbine. The pump is a centrifugal machine allowing for high flow rate at a low pressure head. The evaporator is composed by two shell and tube heat exchanger in series. The recuperator is a shell and plate heat exchanger while a low finned tube air heat exchanger is employed as condenser. Looking at the bottom left of Figure 1b, it is possible to recognize the pump. From there (1), the working fluid is pumped through the recuperator, where it is pre-heated from (2) to (3), and then it enters the evaporator, where it undergoes a transition from liquid to vapour state (4), before expanding in the multistage axial turbine. The superheated vapour leaving the expander enters the recuperator (5) and then it flows through the air condenser (6).

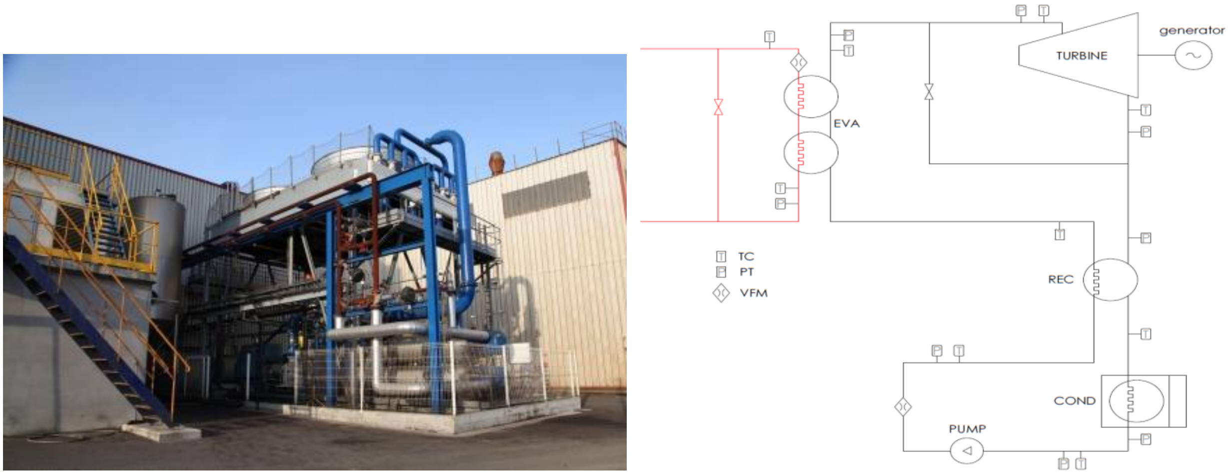


Fig. 1. (a) View of the ORC power plant. (b) Process flow diagram with the relative sensors position of the ORC test facility.

The interested reader can refer to [7] for more in depth details of ORC unit components and the project commissioning process.

2.2. Data acquisition system

Measurement devices are installed on the system to allow controlling the plant performance during operation. Resistance temperature detectors (RTD) and piezoresistive sensors (APS) measure the temperature and the pressure respectively at key points of the power unit. A vortex flow meter (VFM) is located at the outlet of the pump for the measurement of the working fluid mass flow rate. In the heat source circuit, the temperature is monitored at the inlet and at the outlet of the evaporator while pressure is measured at the inlet. A vortex flow meter (VFM) at the evaporator outlet, records the thermal oil mass flow rate. No sensors are installed on the air side of the condenser. The range and precision of the installed sensors are reported in Table 2.4.

Table 1. Range and precision of the measurement devices. k: coverage factor.

Variable	*Device type*	*Range*	*Uncertainty (k=2)*
T (ORC HP)	RTD	0 -160°C	± (0.15 + 0.002 · |t|)
T (ORC LP)	RTD	0 - 100°C	± (0.15 + 0.002 · |t|)
p (ORC LP)	APS	0.5 - 10 bar	± 0.20%
p (ORC HP)	APS	0 - 40 bar	± 0.20%
p (ORC HP)	APS	2 - 40 bar	± 0.20%
$\dot{m}_{wf}$ - $\dot{m}_{sf}$	VFM	-	± 1%

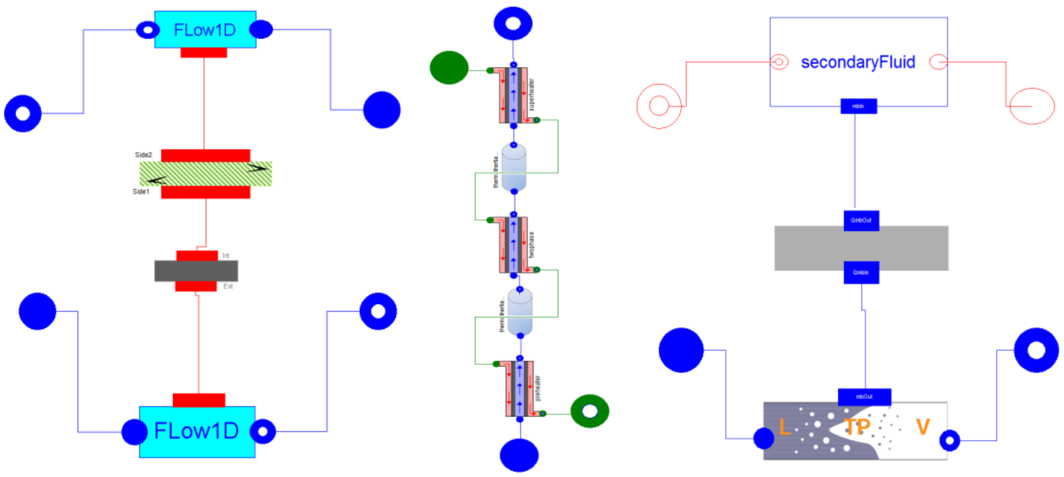
3. System modelling

A dynamic model of the ORC unit described in section 2, is built with the purpose of evaluating the performance of three different evaporator modelling approaches when applied to large system simulation. The model is developed in the Modelica language [6] with the help of existing components from the ThermoCycle library [8]. Modelica requires a modelling and simulation environment which can solve the system of equations and display the results. In this work the commercial program Dymola 2016 [9] is selected as the simulation platform. The thermo-physical properties of the involved fluids are computed using CoolProp, an open-source fluid library [10]. The ExternalMedia package [11] ensures the coupling between the Modelica standard library and CoolProp. In the next subsection the modelling approach of the different components used to simulate the commercial stationary waste heat recovery ORC power plant are described. Particular attention is paid to the description of the three different modelling approach for the evaporator.

3.1. Heat exchangers modelling

When modelling a complete power unit, it is common practice to neglect the dynamic involved in the expansion and compression process as they are orders of magnitude smaller with respect to the heat and mass transfer phenomena characterizing the heat exchanger components [12]. As a consequence the dynamic characteristic of the ORC power unit model results highly dependent on the heat exchanger models. In particular for the presented ORC power unit, the evaporator component is expected to play a major role in the dynamic of the overall system given the much bigger size compared to the other components [7]. In case of heat exchanger involving two-phase flows, the two commonly adopted modelling approaches are the finite volume (FV) and the moving boundary (MB) one [13], [14]. Recently [15] presented a novel simplified lumped-parameter approach (L-HX) based on the LMTD method [16]. The simulation results presented in the paper highlight the model robustness and the high computational efficiency while maintaining good accuracy compared to the finite volume approach.

In this work the three above mentioned approaches are selected to simulate the evaporator. All the adopted modelling approaches are implemented in an object oriented way in the ThermoCycle Modelica library, their structure being shown in Figure 2.



(a) (b) (c)

Fig. 2. Heat exchanger models based on the finite volume (a), the L-HX (b) and the moving boundary (c) modelling approach from the Dymola graphical user interface.

The finite volume model is based on the connection of three subcomponents from the ThermoCycle library. Two fluid components simulating the flow of the fluid in the two sides of the heat exchanger and one wall component accounting for the thermal energy accumulation in the metal wall. The conservation law of physics, describing the behaviour of the fluid through the heat exchanger, are derived by integrating the general 1-dimensional form of mass, energy and momentum balance over a constant volume. Dynamic energy and mass balances are considered while a static momentum balance is assumed. Thermal energy accumulation in the metal wall is taken into account. For a more in depth description of the model the interested reader can refer to [12].

The moving boundary approach is based on two basic models simulating the fluid flow through a variable control volume in single and two-phase state. Connecting these two models allows simulating dry, flooded or general evaporator and condenser. In the general evaporator case three cells are used one for each region of the working fluid side (sub-cooled, two-phase, super-heated). In each region the fluid enthalpy is assumed having a linear profile with respect to the region length. For each cell, the mass and energy balance are derived by integrating the general conservation laws of physics over the length of the zone. In the two phase cell, homogeneous two-phase flow condition is assumed allowing to express the mean density of the fluid in the cell as a function of the average void fraction $\bar{\gamma}$:

$$\bar{\rho} = (1 - \bar{\gamma})\bar{\rho}_l + \bar{\gamma}\bar{\rho}_v \tag{1}$$

where the average void fraction is calculated integrating the local void fraction, γ, over the length of the region. $\bar{\gamma}$ is an indicator of the fraction of the total volume of the region occupied by fluid in vapour phase [18]. Static momentum balance is assumed. Energy accumulation in the metal wall is taken into account while the secondary fluid is simulated assuming a linear temperature distribution and a static mass, energy and momentum balance. The thermal energy transfer of the secondary fluid and the working fluid model is solved either with the semi-isothermal ε-NTU method or with Newton's law of cooling. The detailed formulation of the moving boundary modelling approach is reported in [17]. As far as the lumped parameter approach (L-HX) is concerned the model is based on the connection of three HX_pT model connected in series simulating the three evaporator regions (sub-cooled, two phase, superheated). The HX_pT is a simplified lumped-parameter heat exchanger model. Static mass, energy and momentum balance are assumed in the two fluid sides and thermal energy accumulation is considered in the metal wall. The heat transfer problem is solved using a modified robust version of the log mean temperature difference (RLMTD) method which is applied twice: between the wall and the working fluid temperature gradient and between the wall and the hot fluid temperature gradient. The RLMTD method is based on a set of causal heat transfer equations which allows the model to converge even if negative pinch points occur during the simulation process.

On the working fluid side two pipe models are inserted between the three HX_pT models. The presence of these models finds a purely numerical explanations. The Pipe model is based on a static mass and momentum balance and on a dynamic energy balance, which act as a buffer smoothening the response of the HX_pT model during transient conditions and increasing the overall robustness of the L-HX model. Further details on the HX_pT model and the RLMTD method can be found in [15]. Table 2 summarizes the assumptions on the general conservation laws of physics for the two fluids side and the metal wall of the three different models.

Table 2. Assumptions on the general conservation laws of physics for the three heat exchanger modelling approaches.

	Working fluid			**Secondary fluid**			**Metal wall**
Modelling approach	$\frac{dE}{dt} \neq 0$	$\frac{dM}{dt} \neq 0$	$\frac{dMv}{dt} \neq 0$	$\frac{dE}{dt} \neq 0$	$\frac{dM}{dt} \neq 0$	$\frac{dMv}{dt} \neq 0$	$\frac{dT_w}{dt} \neq 0$
FV	✓	✓	x	✓	x	x	✓
MB	✓	✓	x	x	x	x	✓
HX-L	x	x	x	x	x	x	✓

As far as the recuperator is concerned the component is modelled with the finite volume approach. On the other hand given the low information available on the condenser, a very simplified model is used, in particular the CrossCondenser model from the ThermoCycle library is selected. This model simulates a cross-flow condenser which is assumed to be in thermodynamic equilibrium at all times. The vapour and the liquid are saturated at the condenser temperature, which is considered uniform in the whole condenser.

3.2. Turbine

The turbine model is based on a nozzle model, assuming chocked working conditions. No dynamic is considered in this component. Taking as an input the temperature and the pressure at the inlet and the pressure at the outlet and given the nozzle throat area, the model computes the mass flow by assuming isentropic chocked flow until the stator throat. The model can be modified to take as input the mass flow and calculate the area. In this study, the area is calculated, considering the mass flow at the nominal conditions. The isentropic efficiency is assumed to be a quadratic function of the ratio Φ between the turbine enthalpy isentropic difference, and the squared rotational speed of the turbine (Nrot) reported in Equation 2 [19]:

$$\Phi = \frac{h_{su} - h_{is}}{N_{rot}^2} \quad (2)$$

where h_{su} and h_{is} are the enthalpy at the inlet and the isentropic enthalpy at the outlet of the turbine respectively and N_{rot} is the turbine rotational speed.

3.3 Pump

No dynamic is considered in the pump model. As the available set of experimental data didn't provide any information on the pump consumption neither on its rotational speed, the pump is simulated as a fictitious model where the mass flow rate and the isentropic efficiency are imposed as exogenous inputs by the user.

3.4. Pressure drop

Despite the large size of the plant, negligible pressure drop where measured in the pipes with respect to the one measured in the heat exchangers.

It is common practice to compute the pressure drop directly in the heat exchanger model. However such an approach may leads to stiff problems which require small time steps and can increase the simulation time [20]. In order to avoid these problems the pressure drops are concentrated in the lowest density section of the plant: after the evaporator for the high pressure line, and after the turbine for the low pressure line. The following assumptions are considered:

- Incompressible fluid for computing the pressure drop
- No thermal energy losses to the ambient

The total pressure drop is then computed as the sum of a linear and quadratic terms as:

$$\Delta p = \left(k \cdot \dot{V}\right) + \frac{1}{A^2} \cdot \frac{M^2}{2\rho} \quad (3)$$

where k is the coefficient for linear pressure drop and A is the valve throat area for the quadratic pressure drop which are derived based on the given set of experimental data.

3.5. Liquid receiver

In small ORC unit a liquid receiver is generally placed at the outlet of the condenser to absorb the working fluid fluctuations and ensure saturated liquid condition at the condenser outlet. In the Enertime ORC power unit this role is played by the pipe connecting the outlet of the condenser to the inlet of the pump (Figure 4.6).

Fig. 3. View of the pipe connecting the condenser outlet to the pump inlet.

Furthermore the liquid level in the pipe needs to be above a certain threshold to ensure the static pressure head required by the centrifugal pump and avoid cavitation. The vertical pipe is modelled as a liquid receiver assumed to be in thermodynamic equilibrium at all times, i.e. the vapour and liquid are saturated at the given pressure. The Tank_pL model from the Thermocycle library is used. It is a lumped model where energy and mass balance accumulation are taken into account. The supply flow rate can be either sub-cooled in which case the pressure is going to decrease, saturated in which case the pressure remains constant or two phase in which case the pressure increases. The exhaust flow rate is always defined as saturated liquid. The Tank_pL model has been slightly modified to account for the static pressure of the liquid column inside the pipe. Furthermore the partial pressure of non-condensable gases is taken into account into the model, as non-condensable gases were detected based on the experimental data set. The volume of the tank is defined as the volume of the pipe plus the internal volume of the condenser as the CrossCondenser volume does not consider mass and energy accumulation in the working fluid side.

4. Steady-state experimental validation

The model performances are compared against a set of experimental data recorded during 9 consecutive working days. A total of 21 experimental points are collected. Each point is obtained by averaging the measurements over a period of 10 minutes while the system is in steady-state condition. The system is considered stable when the variable oscillations are within 3%. During the tests the unit operates between a maximum thermal oil temperature of 172°C and a minimum cooling air temperature of 4.7°C. In figure 4 the evaporators model prediction of the working fluid outlet temperature and the oil outlet temperature is reported. The developed models are able to reproduce the real system operating conditions with a deviation of less than 5% for the working fluid outlet temperature and less than 10% for the oil outlet temperature. It is possible to see that in steady-state the variations between the three different evaporator model are within 1%.

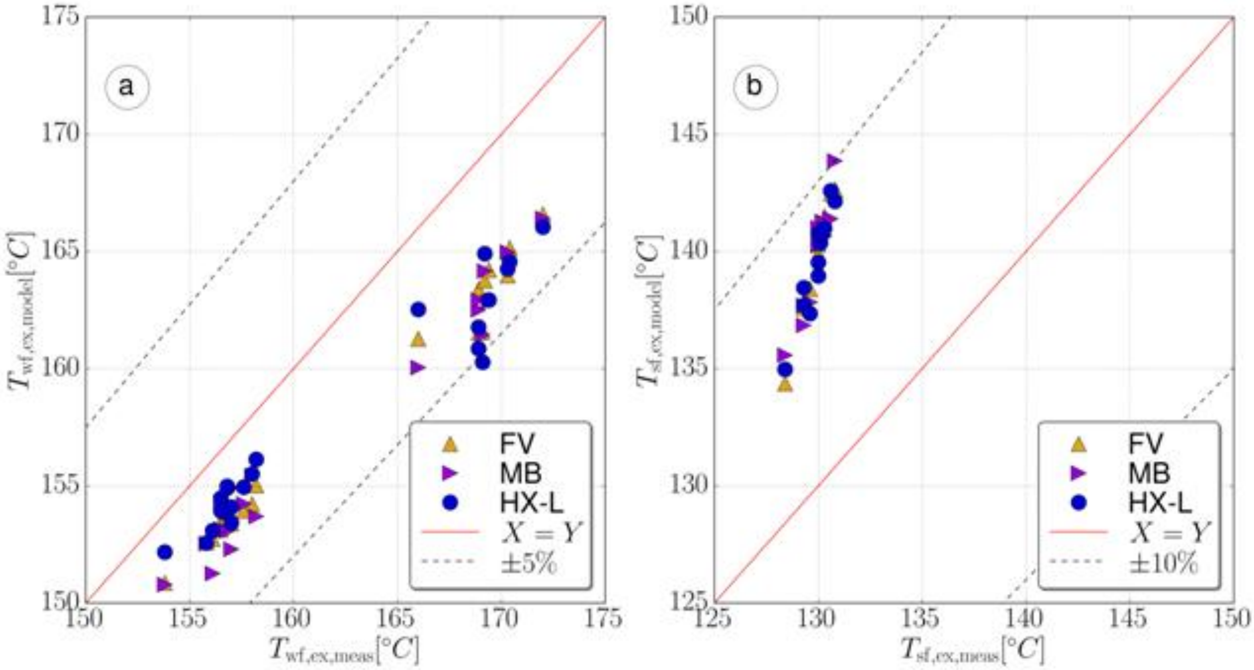


Fig. 4. Parity plot of the evaporator working fluid outlet temperature (a) and the evaporator secondary fluid outlet temperature (b) for the three heat exchanger models.

In figure 5 the steady-state validation results for the turbine model are shown in terms of the measured mass flow rate and the measured electrical output power. The model is able to replicate with an accuracy within 5% both variables. Based on the reported results the models can be considered validated in steady-state conditions

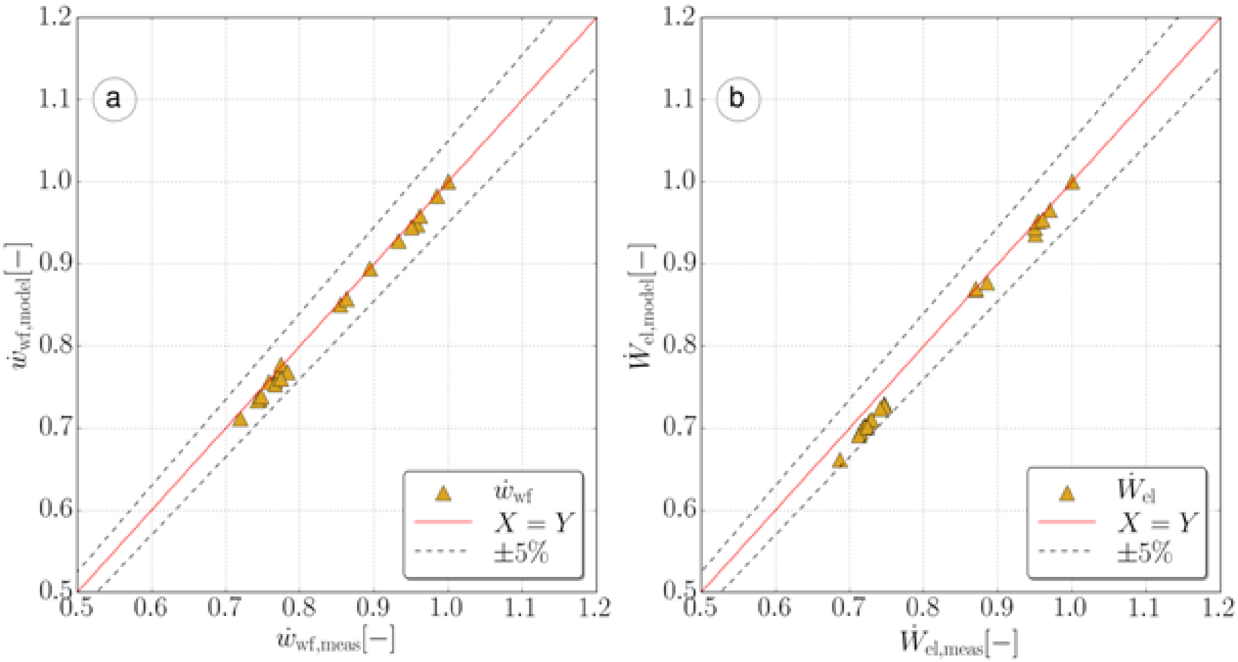


Fig. 5. Parity plot of the working fluid mass flow (a) and the turbine outlet power (b) computed with the turbine model. The data have been normalized with respect to the maximum measured variable.

5. Heat exchanger dynamic comparison

In this section the effects of modelling the evaporator with the selected three different approaches are analysed during transient condition. The Differential Algebraic System solver (DASSL) is selected as numerical solver, setting the relative tolerance to 10^{-4}.

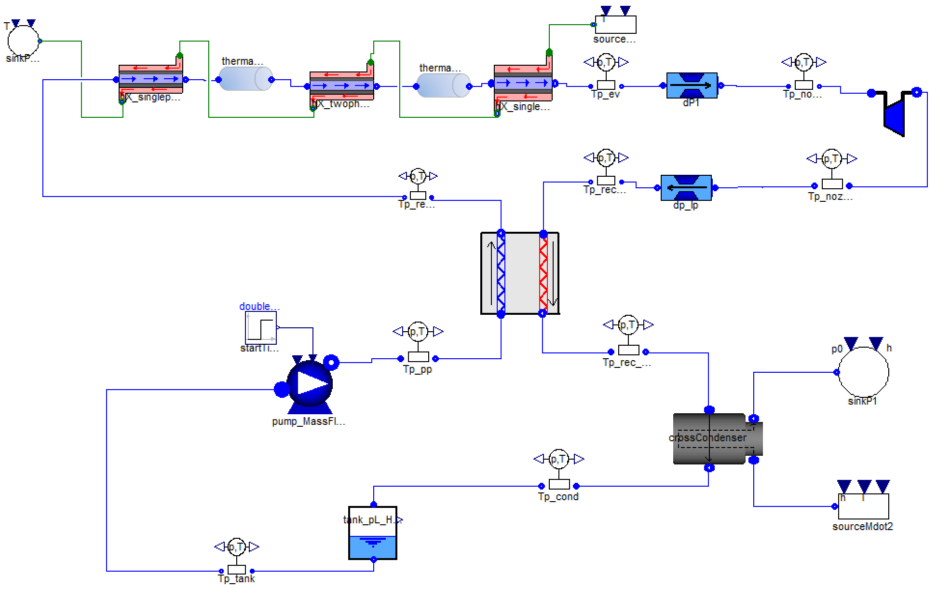


Fig. 6. ORC unit model from the Modelica-Dymola GUI using the HX-L as evaporator.

5.1. Model inputs and parameters

Each model usually contains variables, values that change with respect to time in a continuous or discrete manner and parameters which stay constant during the simulation. The variables can be further distinguished in inputs and outputs. The first are provided by the user while the latter are computed by the model. The solving process of a dynamic system in Modelica is composed by two steps: the initialization and the simulation phase. During initialization, Modelica language allows introducing start values as an input i.e. pressures and temperatures so as the first iteration to start. The initialization process is one of the most challenging aspects of dynamic modelling. In order to initialize the dynamic model variables with meaningful and consistent values, an experimental steady-state point is used to derive the initial values. The heat exchangers and the liquid receiver are parameterized based on the technical data sheet provided by Enertime.

5.2. Simulation results

The dynamic trend of the ORC unit is investigated when the pump rotational speed is subjected to a step change. As the pump speed is increased/decreased, the velocity and pressure of the fluid in the high pressure line decreases/increases. This results in a decrease/increase of density and consequently of mass flow rate. Since the pump is a fictitious model, the pump speed change is simulated by imposing a step change to the mass flow rate of the pump via an exogenous input. In particular a rectangular signal to the mass flow rate of the pump of 10% downward and upward is defined. During the simulations the hot source and heat sink mass flow and temperature are constant and imposed to the ORC unit model as exogenous inputs. In figure 7 the simulation results for the three evaporator modelling approaches are reported. The step down in the mass flow is imposed at t=1500 seconds while the step up happens at t=4000 seconds. The total simulation time lasts for 6000 seconds. The dynamic trends of all of the reported variables are characterized by a much bigger time constant compared to the FV and the HX-L approach. The MB evaporator model simulates a much slower mass flow at the outlet of the expander as is shown in figure 7a. As a consequence the same smoothed trend characterized the turbine inlet pressure (Figure 7c) and the electrical output power (Figure 7b). On the other hand these two variables are characterized by fast overshoot for the HX-L and the FV cases. The higher overshoot amplitude registered when using the HX-L model are related to the fact that no energy neither mass accumulation is considered on the working fluid side (see Table 2). Once the second steady-state condition is reached at t=3000, results show a small deviation between the three models. The relative deviation do not exceed 1.5%. As far as the computational speed of the model is concerned, the required CPU time to run the presented simulations are reported in Table 3 for the three models. As expected the HX-L model is the fastest given the simplified modelling approach. On the other hand, the MB approach results to be the slowest, contrary to what is normally found in the literature [12], [17]. A clear and logic explanation of this behaviour has not been identified yet.

Table 3. CPU time for integration for the three ORC unit model.

Evaporator model	CPU-time [sec]
FV	214
MB	265
HX-L	116

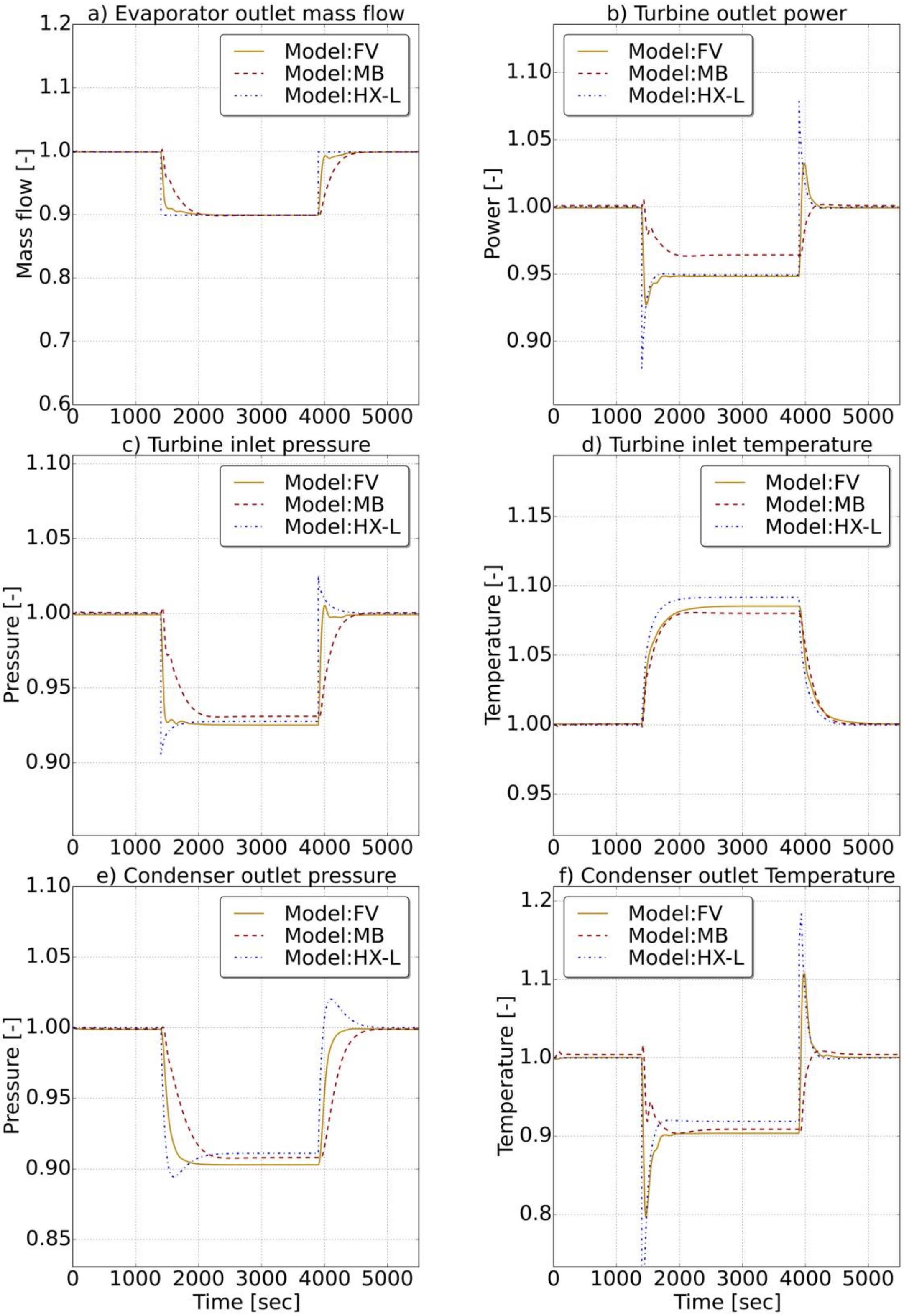


Fig. 7. Downward-Upward step change to the pump mass flow rate. The variables have been normalized with respect to the initial steady-state condition.

6. Conclusions and future work

In this work a dynamic model of a 1 MW_{el} stationary WHR ORC power unit is presented. The unit is based on a simple regenerative ORC cycle equipped with an in-house build axial multistage turbine using Solkatherm36 as a working fluid. The plant is modelled using three different evaporator modelling approaches the finite volume, the moving boundary and a simplified lumped-parameter approach. The three evaporator models and the turbine model are compared against a set of 21 steady-state experimental points. The models show good accuracy with respect to the experiments and deviation lower than 10% and 5% are obtained for the evaporator models and the turbine model respectively. The validated models are then used to simulate the whole ORC system. Three ORC models are developed, one for each evaporator modelling approaches. The power plant dynamic models are then compared during transient simulations to assess the performance of the different evaporating modelling approaches. The main conclusions are reported hereunder:

- The three evaporator models lead to similar performance in off design steady-state conditions.
- The MB model shows much slower dynamic with respect to the FV and the HX-L approach.
- The HX-L model results the fastest while the MB model is the slowest.

Dynamic data are deemed necessary in order to better understand the behaviour of the different evaporator approaches and to validate the developed dynamic model. An experimental campaign is planned at the ORC unit facility to record the required dynamic data. The dynamic validation will be also extremely useful in order to have a better picture of the whole power plant. Furthermore a plant model to simulate start-up and shutdown processes is under development.

Acknowledgments

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Nomenclature

$\dot{V}$ volume flow rate, $m^3/(s)$

h specific enthalpy, J/(kg)

$\dot{w}$ mass flow rate, kg/s

$\dot{W}$ power, W

T temperature, °C

Greek symbols

ρ density, kg/m^3

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