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Design and modelling of a small scale biomass-fueled CHP system based on Rankine technology

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Abstract

The ultimate aim of the project, where the work presented in this paper is inserted, is the design of an ORC based cogeneration biomass-fired mini scale CHP system (10-100kWe). For this purpose, a medium power range (500 kWt) biomass boiler will be modified in order to use part of the energy contained in biomass combustion gases to, indirectly, evaporate the working fluid of an ORC which condenser will be responsible for pre-heating the water demanded by the users. The work presented in this paper refers to the development of a model capable of predict with accuracy the quasi-steady behavior of the ORC by affording an adequate engineering understanding of the physical phenomena affecting their performance. A good and satisfactory agreement was obtained between experimental results and computational outputs of sub-models which allow simulating with some confidence other different conditions.

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Keywords: Biomass; ORC; CHP; Energy; Model

1. Introduction

The emission of carbon dioxide from fossil fuel-fired power plants and the consequent effect on the global environment is a major concern. Given the global challenges related to climate, methods for managing and reducing these emissions must be found and implemented. Renewable sources, such as biomass, may replace the use of all or

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part of those fossil fuels and have therefore an active role in reducing CO_2 emissions [1]. Biomass, of all the renewables energies, is plentiful and prominent, unlike solar and wind energy that have the limitation of intermittent nature. The life cycle of a sustainable biomass energy system has a nearly neutral effect on the atmospheric carbon dioxide concentration. With the continuing rise of gas and electricity prices, biomass-fueled CHP systems will be more economically competitive despite of commercial available systems are very limited and still require considerable technical advances [2].

Small scale CHP systems (usually applied to combined heat and power generation systems with electrical power bellow than 100 kWe) can fulfil a number of energy and social policy targets, including the reduction in greenhouse gas emissions, improved energy security, investment saving resulted from the omission of the electricity transmission and distribution grid and the potentially reduced energy cost to consumers [3]. For this purpose a medium power range (500 kWt) biomass boiler will be modified in order to use the flow of the biomass combustion gases to, indirectly, evaporate the working fluid being the client returning hot water the cold source of the ORC. To accomplish these projects a realistic model will be developed to assist the selection of the appropriate fluid, the adequate components and operation conditions and to define control strategy of whole CHP system.

In the last years, the attention of researcher community was mainly focused on the appropriate fluid selection for ORC cycles and to evaluate the cycle thermodynamic and economic performance [4-7]. The referred analysis was supported by simplified models developed for the operation conditions retrieving the maximum efficiency of the cycle. Integral conservation equations in steady-state conditions and the components characterized by the nominal yields are the main features of those kinds of models. However, they can only get a first approximation of the ORC steady-state behavior because the components do not behave the same way in nominal and non-nominal conditions and the components do not work simultaneously at nominal conditions in the complete operation range. Nevertheless, the literature also presents a few detailed simulation models of ORC's that take into account the non-nominal characteristics of the components [8-10].

The development of a general modular modelling architecture used to simulate realistically the quasi-steady behavior of an ORC is absolutely necessary to characterize and optimize the system with our specific operation conditions. This work grants a complex, however versatile, modelling interconnecting of empirical data, or semi-empirical models, available on the various components of the ORC, very useful to optimize the configuration, to select the appropriate working fluid the adequate components and the best operating conditions and to define control strategy of ORCs.

Nomenclature

CHP	Combined Heat and Power	P2	ORC circuit Pump
G	Generator	P3	Water circuit pump
h	Enthalpy	Q_{in}	Heat power input
HE	Heat Exchanger	s	Entropy
m_f	Mass flow	T	Turbine
N_P	Pump rotational speed		
N_T	Turbine rotational speed		
ORC	Organic Rankine Cycle	ε	Error
p	Pressure	η	Efficiency
P1	Thermal oil circuit Pump	ρ	Density

2. Description of the system

A medium power range (500 kWt) fire-tube biomass boiler will be redesigned to use, at a certain point of the flue gas path within the boiler, part of the energy contained in biomass combustion gases to, indirectly, evaporate the working fluid of an ORC. The biomass-fueled ORC system presented in this work can be considered divided in three circuits: the thermal oil circuit, the ORC circuit and the client water circuit (see Figure 1). The thermal oil circuit is used to transfer part of the energy of the biomass combustion gases into the ORC working fluid at the evaporator.

Beyond the oil heater heat exchanger, where the oil is heated up with the energy from the combustion gases and the evaporator where the oil heats-up and vaporizes the ORC working fluid – this circuit includes a pump (P1) responsible for the fluid circulation. The oil enters in the heat exchanger at 140 °C and exits at approximately 200 °C whereas in the evaporator precisely the opposite occurs, the oil enters at 200 °C and exits at 140 °C. The flow rate is regulated by pump P1 to keep these temperatures independently of the heat load that the biomass boiler is operating.

Through the ORC circuit flows the working fluid, an organic fluid, in liquid and vapor phase. The working fluid, in the liquid phase is heated and vaporized in the evaporator after what expands and cools while flowing through and impelling the rotor of the turbine T. The turbine is coupled with the shaft of the electrical generator G ultimately responsible for the electricity output; afterwards, the vapor fluid cooldown some more and condenses back to the liquid phase in the condenser heat exchanger. While the fluid liquefies, heats the water of the external circuit; finally, the pump P2, forces the circulation of the fluid around the circuit, pressurizes the liquid fluid back to its initial thermodynamic state.

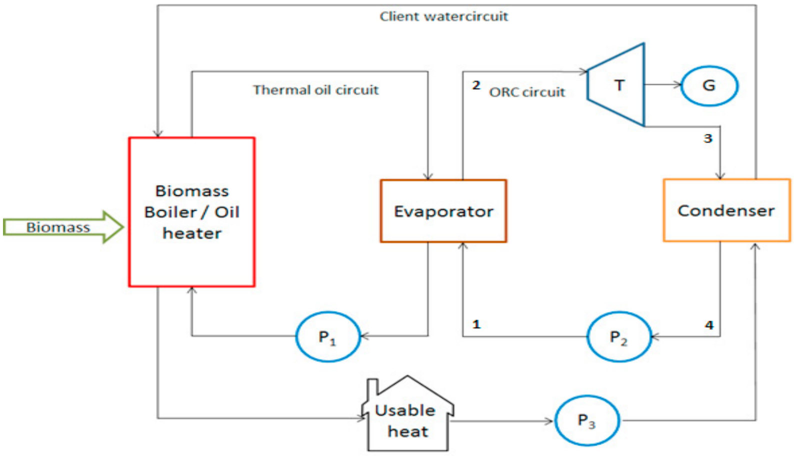

Fig. 1 - Schematics of the biomass-fuelled CHP system.

The client water circuit (external circuit) comprehends two stages of water heating, the first stage is performed in the condenser – this allow having the minimum possible temperature in the ORC condenser and a later stage in the biomass boiler, maximizing the overall system efficiency. The heat dissipation unit that is installed in the laboratory simulates a space heating system and comprehends 7 water-air heat exchangers (HE) which require a water mass flow rate of 1 kg/s for each HE, provided by the pump P3, and an inlet water temperature of 90ºC to ensure that the nominal HE power is reached. A consequence of this high mass flow rate and high HE operating temperature is the high inlet temperatures in the condenser, between 65-85ºC which may compromise the efficiency of the ORC circuit.

By transforming a simple biomass fire-tube boiler into a CHP unit, it is possible to fulfill customer heat requirements as previously and produce electricity at the same time. The production of their own power allows households and communities becoming energetically independent and the excess amount of electricity can then be fed or even sold back into the public power supply system. Along this line of thought, CHP systems leads to significantly higher efficiencies and a reasonable and sustainable use of the heat generated.

3. Model

The development of the modelling architecture intend to obtain a model capable of predict with accuracy the quasi-steady behavior of the ORC by affording an adequate engineering understanding of the physical phenomena affecting their performance. The development of the kernel of the architecture, encompassing the physical and analytical problem formulation, and the numerical solver (detection of non-existence of solution, ignition of the iterations, control of the robustness and speed of convergence of the iteration process, and possible discrimination of multiple solutions) was the first step of this development process. The second step was to cast into canonical forms the empirical data or semi-empirical sub models of the various components of prototype, and to assemble them in the architecture's library.

The condenser is compact plate heat exchanger. The inputs of this component are the fixed design parameters such as length, width, number of plates, chevron angle and corrugation depth; the fixed operation inputs, cold fluid (water), hot fluid (working fluid) and foiling factors of these fluids; and variable operation inputs, namely, the inlet temperature, outlet temperature and pressure of the cold fluid and inlet temperature and pressure of the hot fluid. The main outputs are the mass flow rate, the outlet temperature and outlet pressure of the hot fluid. The condenser sub model contains a thermal part based on a three zone (liquid, two-phase and vapour) Logarithmic Mean Difference Temperature method [11], as shown in Figure 2, and a hydraulic model for the head losses. For each zone a different correlation was used in order to obtain the convection coefficients. The following correlations: Maslov and Kovalenko [12], Han et al. [13], and Thonon et al. [14] were adopted for liquid, two-phase and vapour zones respectively.

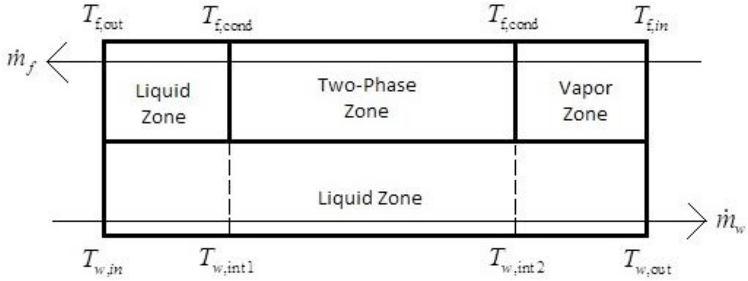

Fig. 2 - Three-zone modelling of condenser

The pump P2 component is composed by a rotary vane pump, with hermetic magnetic coupling and an electrical motor with rotational speed controller. The submodel inputs of this component can be subdivided into constant characteristics of the component, such as, the pump displacement and suction duct diameter and operating characteristics such as, working fluid, the rotational speed, the fluid mass flow rate, pressure and temperature at the entrance of the pump. The outputs are the pressure and temperature of the working fluid at the pump exit and the fluid pressure rise. This sub model was developed extrapolating, based on the usual similarity hypothesis in turbomachinery, the manufacturer given characteristic curve for water to a characteristic curve for the organic fluid.

The evaporator, as the condenser, is also a compact plate heat exchanger. The fixed design parameters used in the condenser maintain as inputs of this component but the variable operation inputs are different. Inlet temperature, outlet temperature, pressure and heat-transfer power are the inputs for the hot fluid (thermal oil) while for the cold fluid (working fluid), the inputs are the inlet temperature and pressure. The main outputs of this submodel are the mass flow rate of the cold and hot fluids and the outlet temperature and pressure of the cold fluid. The structure of the evaporator sub model follows the one of the condenser but, in this case, is the cold fluid that undergoes a change in the thermodynamic state. The correlations used in the sub model are the same used for the condenser with a slightly difference in the Han et al. correlation if the fluid is condensing or boiling [15].

The expander is a volumetric scroll type and unlike scroll compressors is still a developing technology and is incompletely characterized by the manufacturer. For that reason, the sub model of the expander T was developed based on the characteristic curves obtained on an in-house appropriate turbine test facility. Illustratively, the curve of volumetric efficiency depends exclusively from the expander rotational speed $\eta_V(N_T)$ and the curve of mechanic-magnetic efficiency depends of expander rotational speed and the pressure ratio $\eta_m \eta_{mag}(N_T, r_p)$. The inputs of this sub model are the constant characteristics of the expander such as volume of the admission chamber, diameter of the entrance duct and volumetric efficiency curve; the variable operation conditions of the expander are: the inlet temperature and pressure, the outlet pressure and the mass flow rate. The outputs are the outlet temperature of the fluid, the rotational speed and the mechanical power.

Each submodel of the library corresponds to a functional component of the ORC (e.g. pump, evaporator, expander and condenser). The overall model inputs are composed by the natural input parameters of the various submodels of each component, together with the client water circuit variables (water inlet and outlet temperature and mass flow rate) and the control variables of thermal oil heat power (Q_{oil}) and rotational speed of the pump (N_P) and expander (N_T). In fact, the thermal oil heat power is not the control variable but the oil mass flow rate imposed by the pump P1 because with the oil temperatures remaining constant, the heat power is given by the mass flow rate. The main outputs resulting from this model are the working fluid thermo-physical properties along the organic Rankine cycle of the system, the mass flow rate and some outputs specifically related with each component (e.g., various efficiencies, hot water temperature, power of the CHP), from which all the main performance indicators of the biomass-fueled CHP system can be computed e.g., thermal and electrical efficiencies, heat-to-power ratio).

Figure 3 shows the representation of the architecture of the general modular model applied to a CHP system. The 17 main equations of the overall ORC model encompasses the component submodels equations with closure relationships that express the conservation of mechanical and thermal energy along the ducts linking the various components. A second closure relationship and final equation, expresses the mass conservation of the working fluid along the circuit and must be equal to the initial working fluid system charge. At any time instant, the mass of working fluid initially fed into the circuit remain constant.

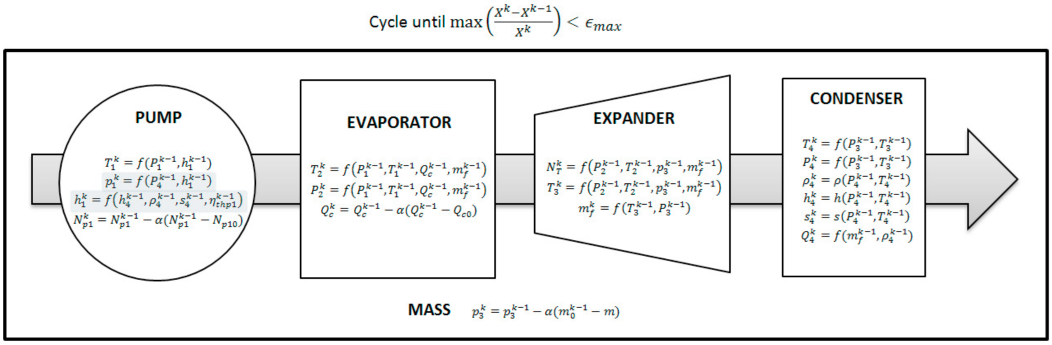

Fig. 3 - General modular model summary

Summing up, given the machine control inputs: thermal oil heat power (Q_{oil}), the pump (N_P) and expander (N_T) rotation speed, the model will solve simultaneously all the equations that describe the behavior of the different components of the system, until converge to a stable solution, presenting all the outputs necessary to control and characterize the ORC system.

4. Results

The working fluid can be a variable in this model but the working fluid R-245fa was previously selected from the literature as suitable for this kind of applications and tested with a simplified model in previous work. A set of experimental tests were performed in order to verify and calibrate the submodels that compose the main model. The submodels were calibrated through the fixed parameters of each component with several experimental points. Subsequently they were validated with new points, different from previous ones, in and out of the range of calibration points.

The expander characteristic curves were obtained in a laboratory test bench specially design for testing scroll volumetric expanders due to a lack/ incomplete information of the manufacturer. The pump was also tested but in this case, minor changes should be done to the manufacturer characteristic curves, essentially regarding to the working fluid change. The plate heat exchanger submodels (evaporator and condenser) were obtained using the semi-empirical equations referred in section 3 and calibrated by reverse engineering with the manufacturer data.

Submodel outputs and experimental values in stationary mode were compared. For each variable tested and calibrated, the measured versus predicted values were compiled and as can be easily verified from the analysis of the Figures 4 and 5 good correlations were obtained for the different models.

For the plate heat exchanger, both outlet temperature and pressure (Figure 4a) are predicted with a medium relative error below 2%. Pump efficiency was proven to be approximately 1 by a maximum relative error predicted to outlet temperature of 0,1% (Figure 4b).

Expander and evaporator submodels were the ones presenting the worst agreement between experimental and computed values. Medium relative error for predicted outlet temperature (Figure 5a) of refrigerant and gas flow in evaporator are under 1,5% and rotation, exhaust temperature, generated power (Figure 5b) and pressure are under 1,3%.

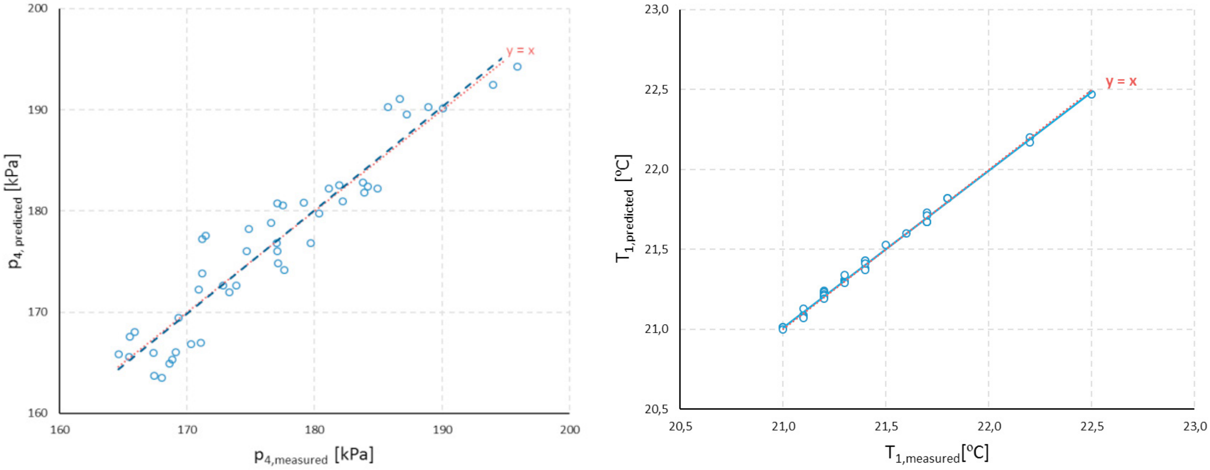

Fig. 4 - Measured versus Predicted values for a) outlet pressure of condenser (P4) and b) for outlet temperature of the pump (T1).

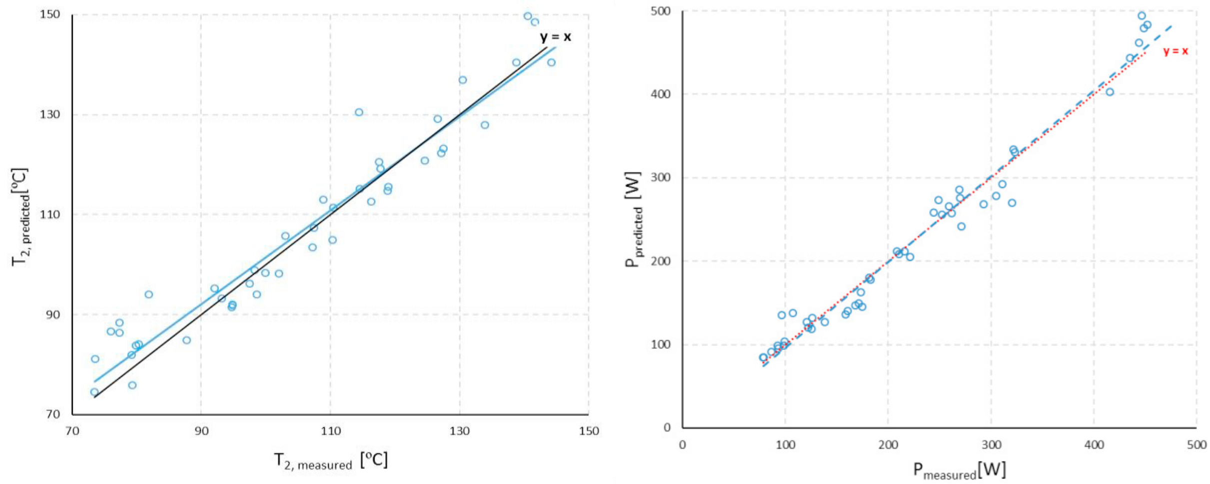

Fig. 5 - Measured versus predicted values for a) evaporator outlet temperature (T2) and b) expander generated power (P).

A parametric analysis was performed using the overall model with the purpose of describe and examine the relationships between different parameters. Thermal oil heat power (Q_{oil}), the rotational speed of the pump (N_P) and expander (N_T) are the variables in this parametric analysis since they are the operation controls of CHP system. The remain inputs such as client water circuit variables or component input parameters were kept constant. Two different turbine rotational speeds (N_{T1}=2500rpm < N_{T2}=2750rpm) and two different thermal oil heat power ($Q_{oil,1}$=140kW < $Q_{oil,2}$=160 kW) were selected and a systematic variation of the pump rotational speed was carried out keeping the temperature of the working fluid at the evaporator exit between admissible values (from 5ºC above the saturation temperature to a maximum of 150ºC). The variation of the pump rotational speed directly alter the working fluid flow rate since it is a volumetric type pump. The influence of the turbine rotational speed (N_T) in the ORC efficiency and in the turbine pressure ratio is shown in Figure 6a and Figure 6b, respectively. The influence of thermal oil heat power (Q_{oil}) in the ORC efficiency and in the turbine pressure ratio is shown in Fig 7a and Fig. 7b, respectively.

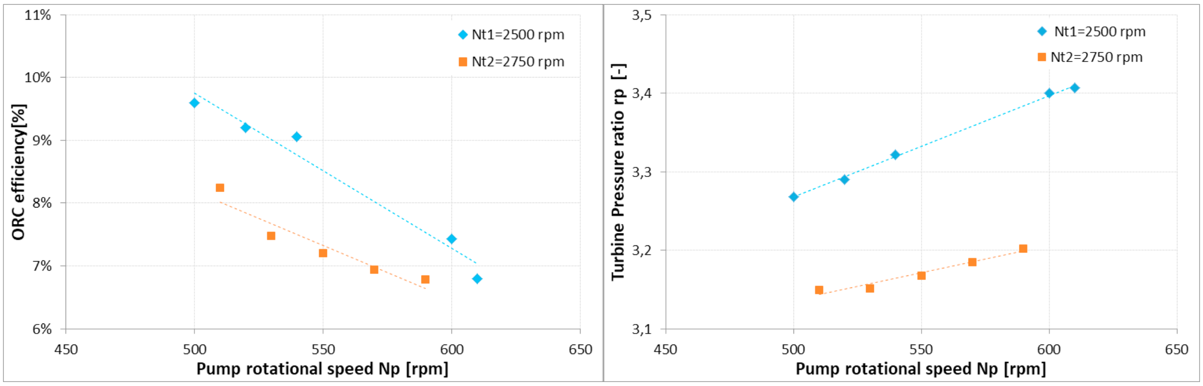

Fig. 6 – Parametrical analysis of a) Overall ORC efficiency and b) turbine pressure ratio for different N_T inputs

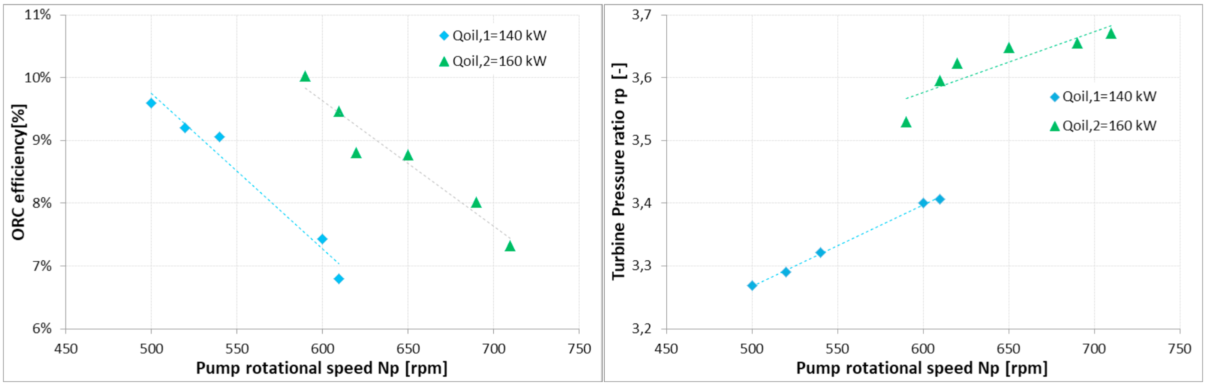

Fig. 7 – Parametrical analysis of a) Overall ORC efficiency and b) turbine pressure ratio for different Q_{in}, N_T and N_P inputs

Through this figures it is possible to verify that lowers turbine speed rotations increases significantly the maximum efficiency achieved (Fig. 6a) due to an increase of the pressure ratio (Fig. 6b). Increasing the heating power at the same turbine speed rotation causes an increase in the pressure ratio (Fig. 7b) but only a slight increase on the cycle maximum efficiency (Fig. 7a). For the same conditions, in each case of the three presented, is denoted a direct influence of the pump rotational speed in the overall cycle efficiency and in the turbine pressure ratio. Lower pump speed cause a decrease of pressure ratio, but, in contrast, cause an increase of the cycle efficiency, probably due to the increase of the overheating temperature at the evaporator exit.

5. Conclusions

The modular open architecture proposed allows to model in a realistic and fast way the steady behavior of an ORC cycle. A library with the sub models of the main components of the ORC (evaporator, turbine, pump and condenser) was created and is ready to use in this type of modelling. The three control variables are the heat power of the thermal oil and the speeds of rotation of the pump P_1 and the turbine T. The calibrated submodels simulate with some confidence the steady-state behavior of the components that are part of the ORC system implemented. Just illustratively, the expander submodel exhibited a maximum of 7% error in the outlet temperature and 5 % in the evaporator outlet temperature.

The possibility to model the behavior of the system in different operation conditions, carrying out parametric analysis of the variation of the speed of rotation of the pump and turbine and power transmitted to the fluid in the evaporator allow us built control maps and to devise the safe and efficient start up procedures. Initial simulations with the global model were already obtained and their data analyzed. We are currently finishing the control maps that will allow defining the control strategy of the CHP. After that, with the complete prototype, the model will be calibrated and validated and according to these good sub-models results, a good prediction in the global model is expected. Such a modelling tool can be employed to optimize the design, selection of components and control of this kind of CHP systems based on ORC.

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