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Dynamic modeling and optimization of an ORC unit equipped with plate heat exchangers and turbomachines

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Abstract

Nowadays environmental concerns call for a transition towards an economy based on fossil fuels to a low carbon one. In order to achieve this goal, efficiency optimization of existing energy systems through waste heat to power conversion units based on bottoming Organic Rankine Cycles (ORC) is one of the actions that appears to be suitable and effective both from cost and environmental perspectives. Indeed, these units are able to increase the overall efficiency of production processes, existing facilities and renewable power plants with a limited payback time. However, despite the increasing number of ORC installations at megawatt scale, the waste heat rejected by industrial processes has rather a widespread nature. Hence, ORC units with a power output in the range of kilowatts should be developed to address this opportunity for heat recovery and for business.

In the current research activity, a dynamic model of an ORC system was developed in a commercial 1D Computer Aided Engineering software platform. Sub-models of the two plate heat exchangers and of the multi-stage centrifugal pump were developed and calibrated using performance data of industrial components at design and off-design conditions. On the other hand, the R245fa radial turbine design was accomplished using a design procedure that provided geometrical and performance data for the mapping of the device by means of a 1D tool. A steady-state off-design analysis at different operating conditions at the evaporator was further carried out optimizing pump and turbine speeds to maximize the net power output. Furthermore, the thermal inertial effects at the evaporator were assessed with reference to a sample heat load profile of the water hot source and at different time scales.

Keywords: Waste heat recovery; Organic Rankine Cycle; Dynamic modelling; Efficiency optimization; turboexpander

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1. Introduction

The growing energy demand in the world, the scarcity of natural resources and the increasing environmental concerns due to the excessive pollutant emissions led governments, national and international organization to more sustainable and eco-friendly energetic and industrial policies. As a consequence, many research activities have been carried out in order to investigate reliable solutions capable to effectively exploit renewable energy sources and to improve the overall efficiency of existing power generation and industrial facilities.

Among these, since the heat rejected by industrial processes is still a remarkable share of the world energy consumption, waste heat to power conversion systems represent one of the most promising and effective ways to contribute to the achievement of these targets. In particular, the organic Rankine cycle (ORC) technology has proven to be reliable, cost-effective and easy to maintain [1] both for low grade and ultra-low grade waste heat recovery applications [2, 3]. For large scale heat sources between few hundreds of kW_{th} up to few MW_{th}, an ORC power system is a mature and economic-optimized technology to convert heat into electricity from geothermal reservoirs [4-7], concentrated solar power plants [8, 9] and in biomass combined heat and power installations [10].

However, considering smaller units, whose thermal power input range is between few kilowatts to 100 kW, there are aspects which have not been fully addressed yet and the research is still undergoing. Among them, one of the most relevant topics is undoubtedly the accurate modeling of the system dynamics, especially of the heat exchangers and the expander side, for design, optimization and control purposes [11]. In fact, in applications as the automotive ones, ORC systems are asked to be flexible and reliable for several working conditions, since the heat load supplied to the system can change rapidly in a periodic or random way, as it occurs in engine ORC systems under transient driving cycle conditions. For these reasons, the proper modeling and analysis of transients, is strongly needed, also to allow the optimization of the power plant in several operating points.

Among the research works that have been carried out in this area, in [12] a dynamic model of a small scale ORC power plant for WHR applications using a scroll expander as power conversion unit has been proposed. The aim of the work was the validation of a control strategy able to optimize the system overall efficiency varying the heat load supplied. In [13] a 150 kW_{el} ORC system using an innovative turbogenerator has been presented with the same purpose. In [14, 15] the effects of the introduction of an 11 kW_{el} screw expander in a ORC system have been analyzed, while in [16, 17] the authors presented the dynamic model of the same system and designed a control strategy acting on the pump frequency in order to regulate the working fluid superheating. In [18, 19], a dynamic model able to predict the behavior of a few MW_{el} scale ORC WHR systems has been proposed, with the aim of reproducing and analyzing its performance and behavior at off-design conditions without having the control actions affecting the system dynamics. The tools used in these aforementioned studies were Modelica and Matlab/Simulink.

Although these simulation platforms allow to model and analyze all the ORC components, they also require an extensive effort and time to implement customized or mere literature models that are not anymore innovative. Therefore, when the goal is to look into the interactions between conventional components, commercial software platforms become equally attractive and timewise effective. For these reasons, in this work a modeling methodology to predict the steady-state and transient behavior of a small scale ORC system is proposed. The approach adopted permits to consistently reduce the model implementation time and it has the additional benefit to be highly replicable both in research and in industry. The approach refers to a radial turbine, which presents several advantages as compactness, economic convenience and ease of maintenance. Besides the modeling methodology and implementation, this study shows the ORC performance at off-design conditions due to different hot source mass flow rates and temperatures and achieved through an optimization of turbine and pump speeds. The transient response of the water-R245fa plate heat exchanger evaporator and the effects on the turbine inlet temperature have been eventually assessed with reference to a series of heat loads at different time scales.

2. System description

The model presented in this work describes an ORC power plant for stationary waste heat recovery applications. With reference to the modeling scheme in Figure 1, the heat recovery takes place through a plate heat exchanger having water on the hot side and the working fluid of the system, which is R245fa, on the cold side. After being pressurized in a multi-stage centrifugal pump, the working fluid undergoes to a full vaporization during the heat recovery process and expands in the turbine, where the useful energy conversion process takes place, from a slightly superheated state. After the expansion, the working fluid is eventually condensed in a second plate heat exchanger using water as heat sink. A refrigerant receiver is eventually positioned between the condenser and the pump such that thermal expansion phenomena during start-up or large transient can be absorbed.

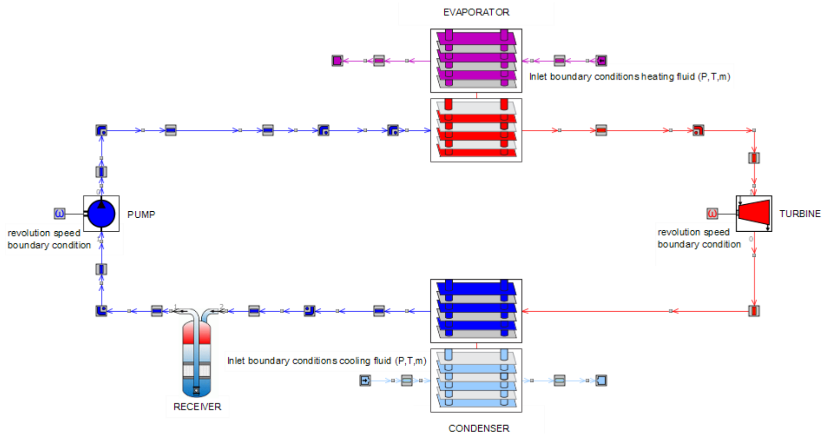

Fig. 1. GT-SUITE^{TM} scheme of ORC model

3. Dynamic model

The dynamic model of the ORC system above described has been developed in GT-SUITE^{TM}, a 0D/1D/3D multi-physics Computer Aided Engineering software. This commercial tool offers templates to model all the components employed in the design configuration considered. Nevertheless, accurate input data are of paramount relevance for the overall development and success of the models. These inputs can result either from an experimental campaign or, as in the current case, from more detailed or complex models. Finally, connections between these devices are made through piping sub-models. In the following paragraphs, a more detailed description of each sub-model is provided.

The full ORC model is schematically illustrated in Figure 1. The boundary conditions used in the simulations are revolution speeds of pump and turbine as well as inlet temperatures and flow rates of hot and cold sources. The equations are solved with an implicit numerical method that approximates the system of algebraic differential equations to a system of nonlinear algebraic equations and eventually solves them iteratively. Hence, the solution values at the next time step are simultaneously provided to all the sub-volumes of a given model (e.g. pipes divisions, heat exchangers channels etc.) [20]. Thermodynamic properties of the working fluid are interfaced with the solver through a dynamic-link library of the NIST Refprop database [21].

3.1 Evaporator and condenser

To model evaporator and condenser, a simulation tool provided by a well-known OEM has been used to retrieve several operating points of the two plate heat exchangers manufactured by the same company [22]. These operating points have been obtained varying model inputs as the refrigerant mass flow rate, the inlet temperature of the refrigerant and the inlet and outlet temperatures of cold and hot sources at the condenser and evaporator respectively. The output data were the outlet and the condensing or evaporating temperatures, the quality of the refrigerant, the cold and hot source mass flow rates, and the pressure losses through the heat exchangers. These quantities have been eventually inserted into the plate heat exchanger template of GT-SUITE^{TM} to calculate the best fitting coefficients of Nusselt-Reynolds correlations that are used in the heat transfer calculation along the equivalent one-dimensional passages with whom the heat exchanger channels are approximated [23]. In particular, the software predicts the vapor formation inside each sub-volume of the heat exchanger following the Rayleigh-Plesset equation [24] and calculates the extension of the two-phase region, if any. Then, based on these predictions, it applies different correlations according to the fluid phase. In particular, Dittus-Boelter correlation is used in single phase heat transfer [23], while in the two-phase region, the correlation from Yao et al. has been considered for the condenser [25] and the one from Donowsky and Kandlikar for the evaporator [26].

The heat exchangers' inertia, affected by the metallic mass and the capacity of the device, is eventually taken in account by giving in input to the software the geometrical data and the heat exchanger material.

3.2 Pump

The pump has been modeled by calculating its performance and isentropic efficiency map by means of a regression analysis based on a set of real operating points (pressure rise, volumetric flow rate and power consumption) available on the website of the manufacturer. Regarding the performance map, since the real data were available only for a certain range of revolution speeds, in particular between 1800 and 3000 RPM, the performance curves for speeds lower than 1800 RPM have been computed using a linear extrapolation method. The isentropic efficiency map instead, has been calculated based on the interpolated power consumption data. Figure 2 reports performance and isentropic efficiency maps of the pump.

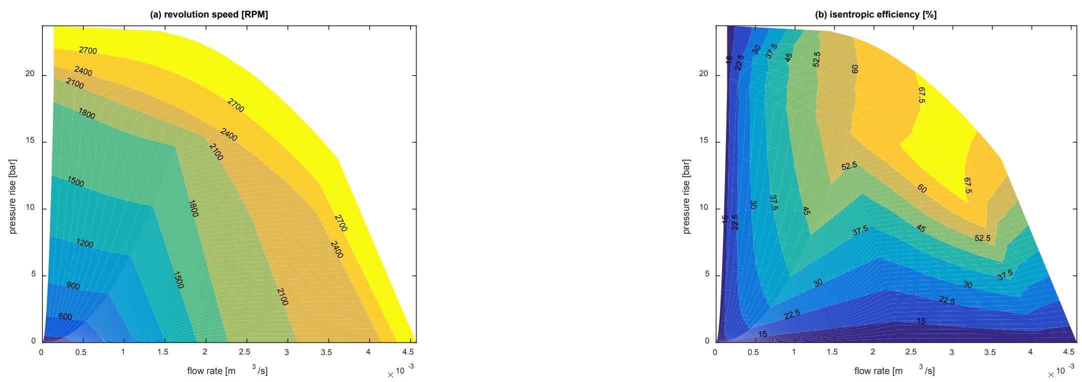

Fig. 2: Pump performance (a) and efficiency (b) maps

3.3 Radial Turbine

The radial turbine has been modeled following the approach developed and described in [27]. The model considers the rotor, the stator and the volute of the turbine, assumed for simplicity as a circular cross section, in order to accurately estimate the fluid dynamic losses. To start the procedure, first, a set of geometrical and thermodynamics parameters are required [28-31], chosen respectively by manufacturability considerations and according to the design operating point of the ORC power plant object of this work. Given these input data, an isentropic efficiency at the design operating point of the expander is estimated, then the enthalpy drops in the rotor, stator and in the volute are computed. Consequently, a new isentropic efficiency value is calculated and compared with the previous estimated one. The procedure is thus repeated until a convergence is reached. The above expander design loop has been coupled with the MATLAB™ optimization toolbox in order to optimize the inlet parameters, using as objective function the isentropic efficiency of the expander.

After this design stage, the commercial 0D/1D commercial software RITAL™ has been used to calculate the turbine performance and isentropic efficiency map, in order to have a model able to describe accurately the expander behavior under off-design conditions. Firstly RITAL was calibrated in order to predict the expander design conditions. Then it was used to calculate the off-design performance of the expander. Last but not least, these operating maps obtained with the aforementioned approach have been implemented in GT-SUITE™ and are reported in Figure 3.

3.4 Piping and receiver

Straight pipes have been considered as one dimensional ducts with a circular cross section and with a roughness value typical for stainless steel. Additional bends have been modeled using the software library that already contains the localized pressure drops due to such components. For sake of simplicity, assuming that the pipes are insulated, adiabatic conditions were set to prevent any heat loss to the environment. The receiver size was chosen to be the 25% of the system capacity as suggested in the software guidelines [20].

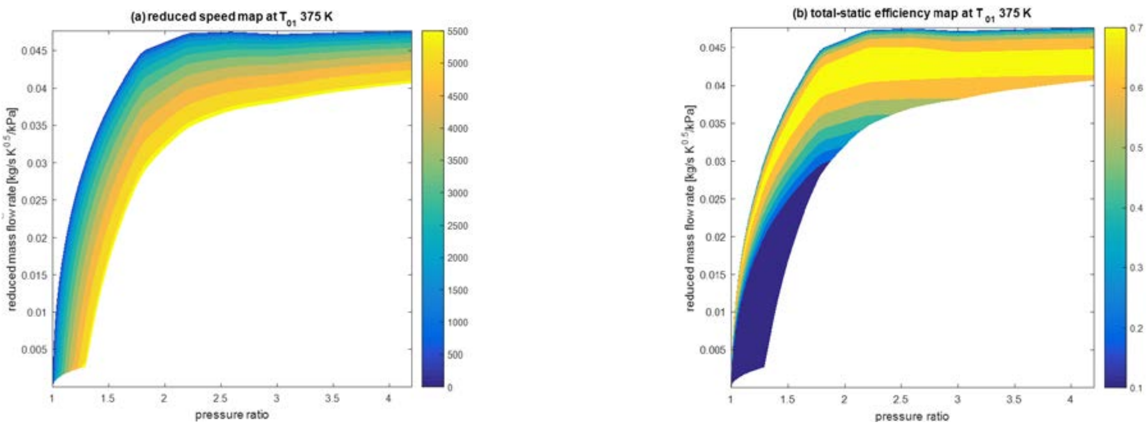

Fig. 3. Non dimensional turbine performance (a) and total-static efficiency (b) maps

4. Results and discussion

4.1. Off-design performance

After the modeling stage, a series of simulations were carried out in GT-SUITE^{TM} to characterize the steady-state performance of the ORC system varying the heat load supplied by the stream rejected by the topping facility. Since the heat load of this stream is completely defined by its mass flow rate and inlet temperature, these conditions were changed during the simulations with respect to the design point at 10 kg/s and 110°C. On the other hand, the thermodynamic conditions of the heat sink remained unchanged to the design values of 14.3 kg/s and 35°C.

For each case, the net power output was maximized with respect to pump and turbine speeds as independent variables. The optimization algorithm that was employed is the Nelder Mead SIMPLEX one, which is suitable for finding a local minimum and at a low computational cost since no derivative calculations are involved. For two independent variables as in the current case, the method is a pattern search that compares function values at the three vertices of a triangle. The worst vertex, is rejected and replaced with a new vertex. A new triangle is formed and the search is continued. The process generates a sequence of triangles for which the values of the optimization function at the vertices get smaller and smaller. The size of the triangles is reduced and the coordinates of the minimum point are found. Furthermore, this algorithm allows to perform also constrained optimizations by penalizing the regions which violate the constraints imposed [32, 20]. In this case the constraint was the fulfilment of at least saturated conditions at the turbine inlet.

The results of the analysis are reported in Figure 4. In particular, Figure 4.a shows the vapor quality of the working fluid at the inlet of the turbine and, in turn, the operating range of the system. In fact, a durable and reliable usage of turboexpander requires the inlet conditions of the working fluid to be slightly superheated. To face this well-known issue, a future control strategy to be implemented in this ORC unit will need to shut down the system when the waste heat temperature is below 100°C or 110°C for mass flow rates lower than 7.2 kg/s. Alternatively, the turbine could be by-passed using an additional branch in parallel with the expander and where an expansion valve will simulate the pressure drop due to the turbine. This isenthalpic transformation might however require an oversizing of the condenser. Figure 4.b shows the maximized net power output in the operating range identified by the quality constraint. This quantity ranges from 34.5 kW to 55.5 kW and increases with both hot source inlet temperature and mass flow rate. In particular, for a given hot source temperature, beyond a given threshold (e.g. 9 kg/s) the inlet quality at the turbine (e.g. 1.05) is not affected by the increase of hot source mass flow rate. Therefore, the increase in power output is not due to a greater specific work but rather to a greater working fluid mass flow rate that balances the heat load at the evaporator and that is due to higher revolution speeds of the turbomachines. Below that threshold, a lower hot source mass flow rate leads to a lower inlet quality and working fluid mass flow rate that both contribute to decrease the net power output. However, these considerations are fluid dependent and may highly differ in other design configurations (e.g. with gaseous heat sources.)

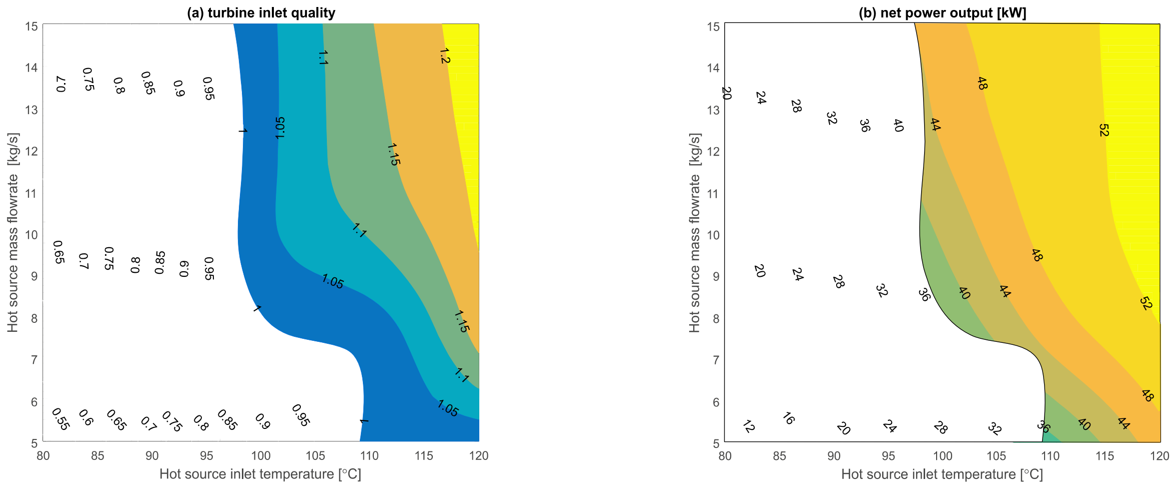

Fig. 4. ORC system performance at off-design conditions with optimized pump and turbine speeds: effects on turbine inlet quality (at least equal to 1) (a) and net power output (b)

4.2. Transient thermal input

In the simplest architecture, such as the one investigated in the current study, an ORC system is at least composed of two heat exchangers and one receiver. These components are not only filled with large quantities of working fluid but also have metallic surfaces whose thicknesses need to withstand high pressures. The thermal inertia due to these coexisting effects acts as a low pass filter for the ORC system and needs to be taken into account when developing a control system for an ORC unit, especially if the final application is a transportation one.

In order to stress and demonstrate this concept, transient simulations were carried out on the modeled ORC system. In particular, the simulations imposed a variable mass flow rate at the hot side of the evaporator and observed the effects on a fundamental parameter of the ORC operation, namely the turbine inlet temperature. Figure 5 summarizes the transient analysis with respect to a steady state value marked with SS. The grey line refers to the input while the black one to the output. The mass flow rate profile is composed of two ramps and two plateaus. Durations of the transient inputs are 200, 350, 600 and 1100s. Other operating parameters in the ORC system were not varied, neither optimized.

Figure 5.a shows that the response to the transient input is delayed and it is not able to follow the load variation. In fact, the intermediate plateaus are not reproduced in the temperature trace. In turn, the overall effect on the turbine inlet temperature is a variation of 2.7K. In Figure 5.b the duration of the transient and of the single phases was nearly doubled compared to the previous one. In turn, although the temperature overshooting after the first ramp is still noticeable and delayed, the temperature trend starts to resemble the mass flow rate profile. Moreover, the second plateau is still not well resolved and the overall temperature decrease from the initial value is 4.1K. This new steady state is reproduced in the cases shown in Figures 5.c and 5.d that are further characterized by a confident reproduction of the mass flow rate input.

In a control perspective, these results suggest the adoption of different control strategies for variable heat loads at different time scales. For instance, when the hot source presents slow variations in time, as in Figure 5.c, 5.d and typically in stationary applications, the time constants of the heat exchanger and the one of the variation are comparable. In turn, the temperature of the working fluid at the turbine inlet could be set acting on the mass flow rates of the evaporator. Instead, when more rapid changes of the heat load are considered, as in Figure 5.a and 5.b and generally in automotive applications, then an action on the mass flows can result trivial, and the regulation of a variable with a faster dynamics is required. In this sense, acting on the turbine speed might be a viable solution since this parameter affects the evaporating pressure inside the evaporator. However, this is in part a feedforward action since it requires the correlation between the evaporation pressure inside the evaporator and its outlet temperature. In addition, the problem is even more complex considering that the turbine speed affects also the expander isentropic efficiency and thus the net electrical power generated, which should be the main target of the control system. Then it appears that further analysis is required on this topic, which will be addressed in the authors' future works.

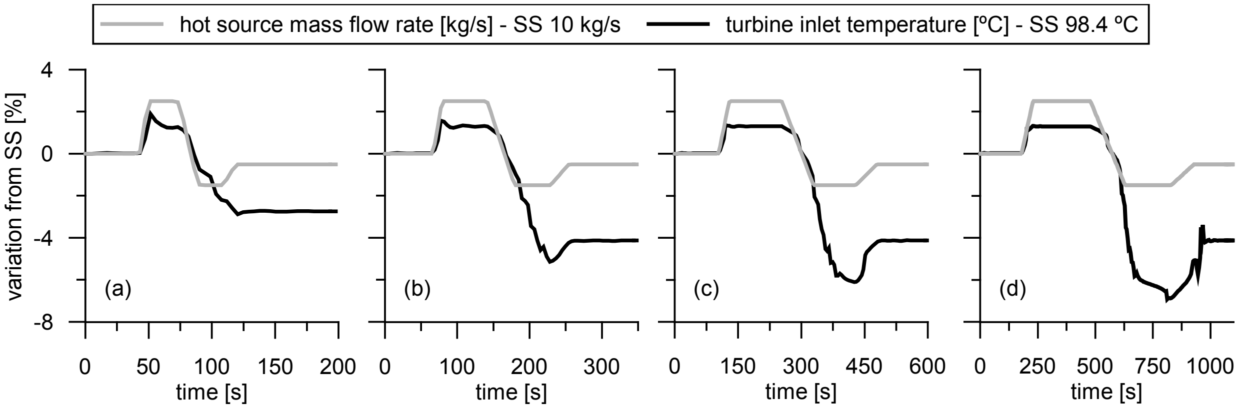

Fig. 5. Transient response of the ORC system at different heat load profiles:
variation of hot source mass flow rate and turbine inlet temperature with respect to steady state value (10 kg/s, 98.4°C at 0s)

6. Conclusions

This paper presented a modeling approach to analyze and design waste heat to power conversion units based on an Organic Rankine cycle. The modeling methodology and its implementation in the commercial software platform GT-SUITE™ allows high replicability and further exploitation of the current study both at research and industrial levels. If on one hand the availability of templates for most of the equipment required in an ORC system shortens the model development, a calibration of these models is strictly required either via experiments using advanced models. In the current study, the latter approach has been employed for heat exchangers and the radial turbine while for the pump, performance data from the manufacturer have been used.

The model has been tested to assess the performance of the ORC unit at off-design conditions. In particular, variable hot source flow rates and inlet temperatures at the evaporator produced a grid of points in which the ORC net power output was optimized using the SIMPLEX algorithm and with respect to pump and turbine speeds. In this analysis, it resulted that for low heat duties at the evaporator, vaporization of the working fluid was not fully accomplished. Hence, in those conditions the usage of the turboexpander is discouraged. In the acceptable range of operation, power output, which varied from 34.5 kW to 55.5 kW, increased with hot source mass flow rate and mostly with inlet temperature of the waste stream at the evaporator.

The response to a transient heat load on the system and particularly on the evaporator was eventually assessed at different time scales. Even though the magnitude of the variations imposed to the hot source mass flow rate did not vary, the duration of the transient highly affected the key operating parameter of the inlet temperature at the turbine. Indeed, due to the thermal inertia of the fluids and of the metallic parts of the heat exchanger, the temperature response was delayed with respect to the mass flow rate input, especially at fast time scales. Furthermore, it was noted that the fastest transient led to a different steady state point with respect to the slower ones.

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