EXPERIMENTAL INVESTIGATION OF RESPONSE TIME CHARACTERISTICS OF AN ORGANIC RANKINE CYCLE EVAPORATOR FOR WASTE HEAT RECOVERY

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ABSTRACT

On this paper a test rig that focuses on the dynamic and transient behavior of an ORC system for waste heat recovery is presented. An experimental campaign is carried out in order to analyze the transient behavior of a single finned-tube evaporator under continuous and discrete fluctuations of the heat source. The experimental results show the dynamic response of a custom-made ORC evaporator to sinusoidal fluctuations and ramp changes of the heat source. Due to its large thermal inertia, the evaporator is able to dampen fluctuations of the heat source in the MHz range, in fact, effectively "filtering out" sinusoidal fluctuations with frequencies larger than at least 1.67 MHz. This large thermal inertia of the ORC evaporator is accomplished by means of a heat exchanger design where the fins have a significant mass. The study illustrates how the design of the evaporator can be an important aspect in ORC dynamic performance that is often taken for granted. This is critical for applications in which the variability of the heat source is inherent, such as waste heat recovery.

1 INTRODUCTION

Waste heat recovery from various sources is an important field of research and technology development due to its potential to increase the energy conversion efficiency of various energy-intensive processes in the industry and technological devices. Organic Rankine Cycle (ORC) is one of the leading technologies suited for the recovery of waste heat in order to produce power. Some of the waste heat sources with a significant potential for power generation include energy-intensive industries such as the steel, cement and glass (Pili *et al*., 2020), (Loni *et al*., 2021), as well as the waste heat from internal combustion engines (Shi *et al*., 2018).

However, an important technological hurdle remains in various of those waste heat applications. Many of them exhibit a fluctuating and intermittent nature because of the heat supplying processes. For instance, dynamic changes in the load of an internal combustion engine result in variations of the mass flow and temperature of the hot exhaust gases (Vaupel *et al*., 2021), or batch or intermittent production processes in the steel industry result in fluctuations of the residual thermal power available (Dal Magro *et al*., 2017). These fluctuations are more often the norm than the exception.

Therefore, it is fundamental to understand the transient behaviour of the recovering technology, in this case the ORC, in order to predict the behaviour of the system under fluctuating conditions of the input heat. Another important aspect is to rethink the design of the system (Jiménez-Arreola *et al*., 2018). If the system works the majority of the time under fluctuating conditions, it makes more sense to optimise the design of the system for the whole range of fluctuations than just for a single nominal operating point. In this context the key component of the ORC system, is the evaporator, since it is the link between the fluctuating waste heat source and the rest of the ORC components. There are not many works on the literature that show experimental data on the dynamic behaviour of ORCs and, of those, most focus on control strategies and not on the evaporator heat exchanger design. This works is intended to fill that gap.

In order to investigate the dynamics of the ORC, and in particular of the ORC evaporator, a test rig was built consisting of an ORC evaporator with a custom heat exchanger geometry. In particular the transient behaviour of the ORC evaporator under fluctuations of both the flow rate and temperatures of the hot fluid was investigated. The focus is on the impact that the geometry of the heat exchanger causes on its thermal inertia and thus the dynamic behaviour of the whole ORC system.

2 CHARACTERISTICS OF TEST RIG AND EQUIPMENT USED

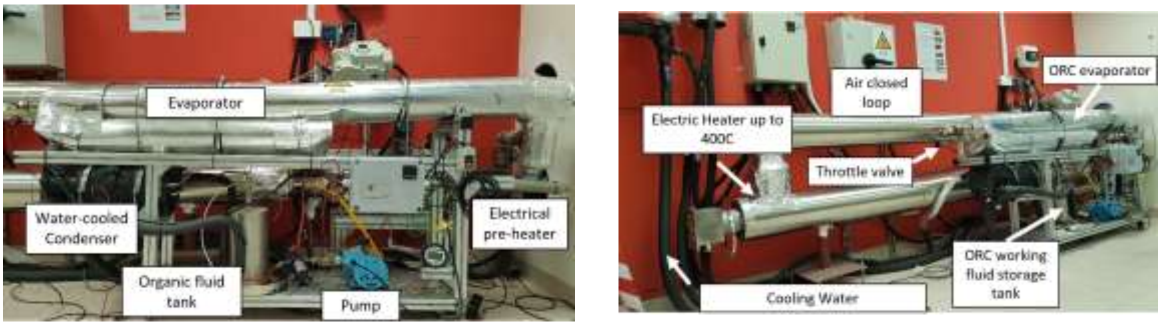


Figure 1: ORC test rig in laboratory

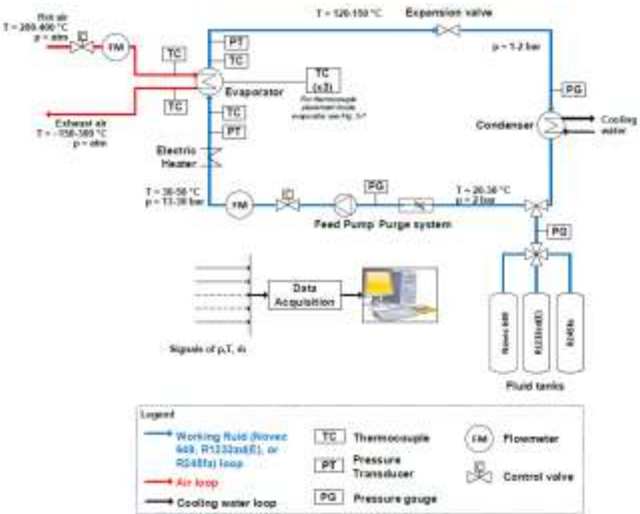
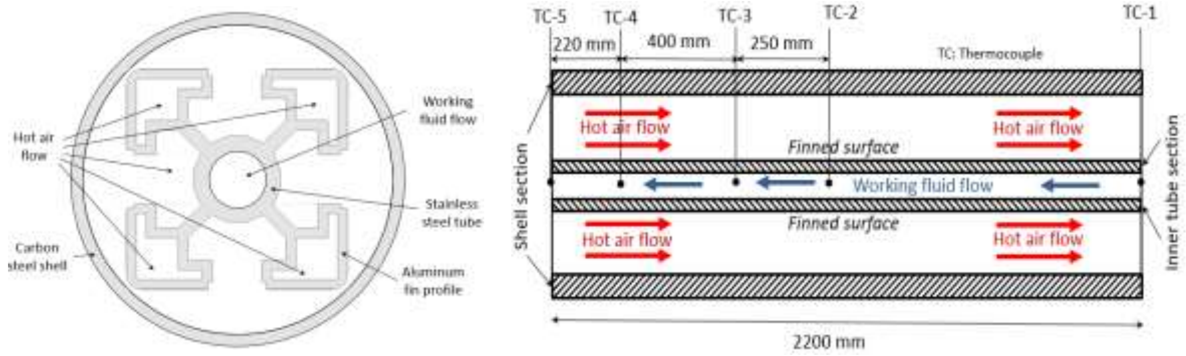


Figure 2: Schematic of ORC test rig for experimental investigation

A picture of the test rig in the lab is shown in Figure 1. The test rig has been built in the installations of the Thermal Energy Systems Lab of the Nanyang Technological University, Singapore. The objective of the test rig is to simulate fluctuations of both flowrate and temperature of a flue gas that heats up the ORC fluid in the evaporator, in order to investigate the transient behavior of the ORC fluid exiting the evaporator. For simplicity and requirements of the laboratory the "flue gas" was replaced by hot air. The choice of air instead of a "flue gas" is expected to not have a large impact on the transient behavior of the evaporator since the heat accumulation on the hot side is negligible due to the low density of the gaseous fluid. The test rig thus consists of a closed loop for hot air and a closed loop for the ORC fluid (R245fa), as well as an external cooling water supply. R245fa has been chosen as the working fluid since it is a well-known and common fluid for ORCs and is well suited for the heat source temperatures on the test rig (150- 300 °C). The hot air loop has a blower, an electronic control valve and a controllable electric heater that is able to be programmed to dynamically ramp up or down the temperature of the air. The ORC fluid loop consists of a pump to pressurize the fluid to the evaporator pressure, followed by an electrical pre-heater and a custom geometry evaporator where the liquid ORC fluid is evaporated, afterwards, a throttle valve expands the fluid back to the condenser pressure, and finally, a condenser where the ORC fluid is condensed back to liquid by transferring the residual heat to a cooling water supply. Type-k Thermocouples (TC), flowmeters (FM) and pressure transducers (PT) are installed to measure and record temperatures, flowrates and pressures in both loops. The P&ID in Figure 2 shows the locations of the sensors in the test rig. The signals from the sensors are recorded on a Data Acquisition System consisting of two modules, one for analog current signals and one for thermocouple voltages, both from the company National Instruments^{TM}.

Table 1: Relevant dimensions of ORC evaporator in test rig.

Description	Value	Unit
Inner tube inside diameter	8.25	mm
Inner tube wall thickness	2	mm
Shell inside diameter	68	mm
Evaporator length	2200	mm
Fin augmented area ratio	22	-
Fin cross-sectional area	953.3	mm^2



(a) (b)

Figure 3: ORC Evaporator geometry in test rig (a) Front view cross-section (b) Side view of cross-section and location of instrumentation in evaporator

The evaporator geometry is designed as one simple double-pipe heat exchanger with a finned external surface for the inner pipe. The pipe is made of stainless steel. The fins are made using the standard item aluminum profile 45x45, slot. The simple geometry was chosen in order to understand the characteristics of the transient behavior in ORC evaporators in the most fundamental way, as well as the effect of some general dimensioning decisions of the heat exchanger (diameter and length of the tubes) on the thermal inertia of the system. The finned surface geometry was chosen for ease of construction and to increase the fin volume in order to investigate its effects on the evaporator thermal inertia. Figure 3a shows the front view cross-section of the heat exchanger displaying the geometry of the fins. Figure 3b shows the side view of the custom heat exchanger along with the basic flow arrangement and the location of the thermocouples. Table 1 reports the main geometrical dimension of the ORC evaporator. This design is the only heat exchanger that has been used during the experimental campaign.

3 EXPERIMENTAL CAMPAIGN

The ORC test rig was subjected to two types of fluctuations from the heat source. One type of fluctuation is a sinusoidal profile of both temperature and flowrate of the hot air. The other type is a trapezoidal profile of the hot air temperature. These two types of profiles are consistent with the test rig limitations and were chosen to achieve an understanding of the ORC evaporator response to dynamic changes for different types of fluctuations. The sinusoidal profile characterizes a continuous type of fluctuation, whereas the trapezoidal profile characterizes discrete and sudden types of changes.

Figure 4 shows an example of these two profiles as measured by the flowmeter and thermocouple of the hot air at the inlet of the evaporator. Sinusoidal profiles with different frequencies where used (1.67 mHz, 1.04 mHz and 2.78 mHz) with amplitudes in temperature of about 40 °C. The trapezoidal profiles consists of a ramp-up of 20°C in 2 minutes and then a flat profile to allow the ORC fluid to reach a steady-state followed by a similar ramp-down in multiple cycles.

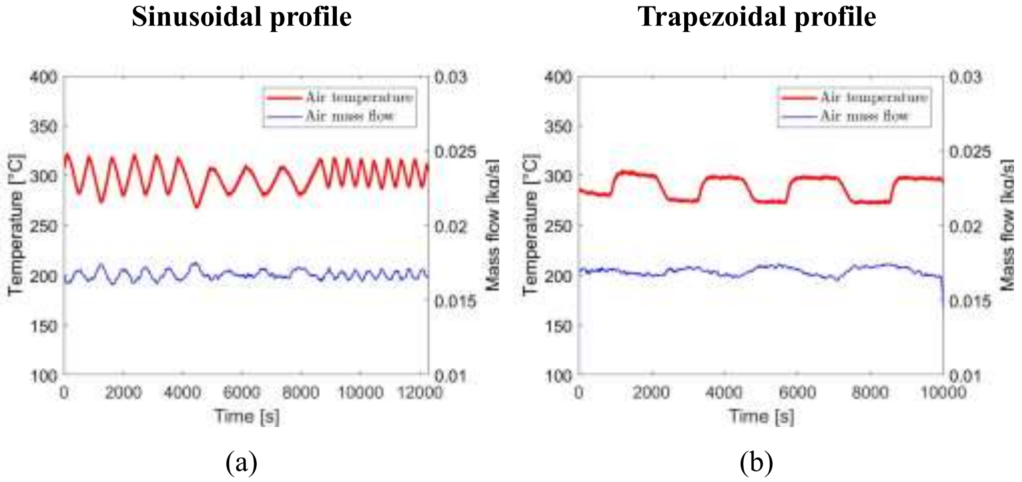


Figure 4: Examples of two profiles of inlet flowrate and temperature of the hot air in the ORC evaporator. (a) Sinusoidal profile. (b) Trapezoidal (ramp-up and down) profile.

The tests were carried out for several different ORC fluid evaporation pressures (8, 10, 12 and 15 bar) with their corresponding saturation temperatures. One constraint imposed on the tests is that the thermal power available from the hot air was sufficient to fully evaporate and slightly superheat the organic fluid at the outlet of the evaporator. Some parameters such as the expansion valve opening as well as the mass flow and temperature level of the hot air were adjusted before each test to comply with these constraints. However, no active control actions were implemented during the course of each tests, because the aim is to investigate the natural response of the system to the changes.

4 RESULTS FROM EXPERIMENTS

Selected results from the experimental campaigns are shown in Figure 5, for two tests with a sinusoidal profile of the heat source as well as two tests with a trapezoidal profile of the heat source, showing the behavior of the thermal response of the fluid to changes in the hot source. A close-up of the response to the changes for a selected ramp-up transient is shown in Figure 6. The black line in Figure 5 represents the heat transferred from the hot air on the evaporator each second Q_{in}, and is defined as:

$$\dot{Q}_{in} = \dot{m}_{air}(h_{air,in} - h_{air,out}) \tag{1}$$

The blue line in Figure 5 represents the actual thermal energy gained by the working fluid in the evaporator each second (enthalphy gained H_{gain}) and is defined as:

$$\dot{H}_{gain} = \dot{m}_{wf}(h_{wf,in} - h_{wf,out}) \tag{2}$$

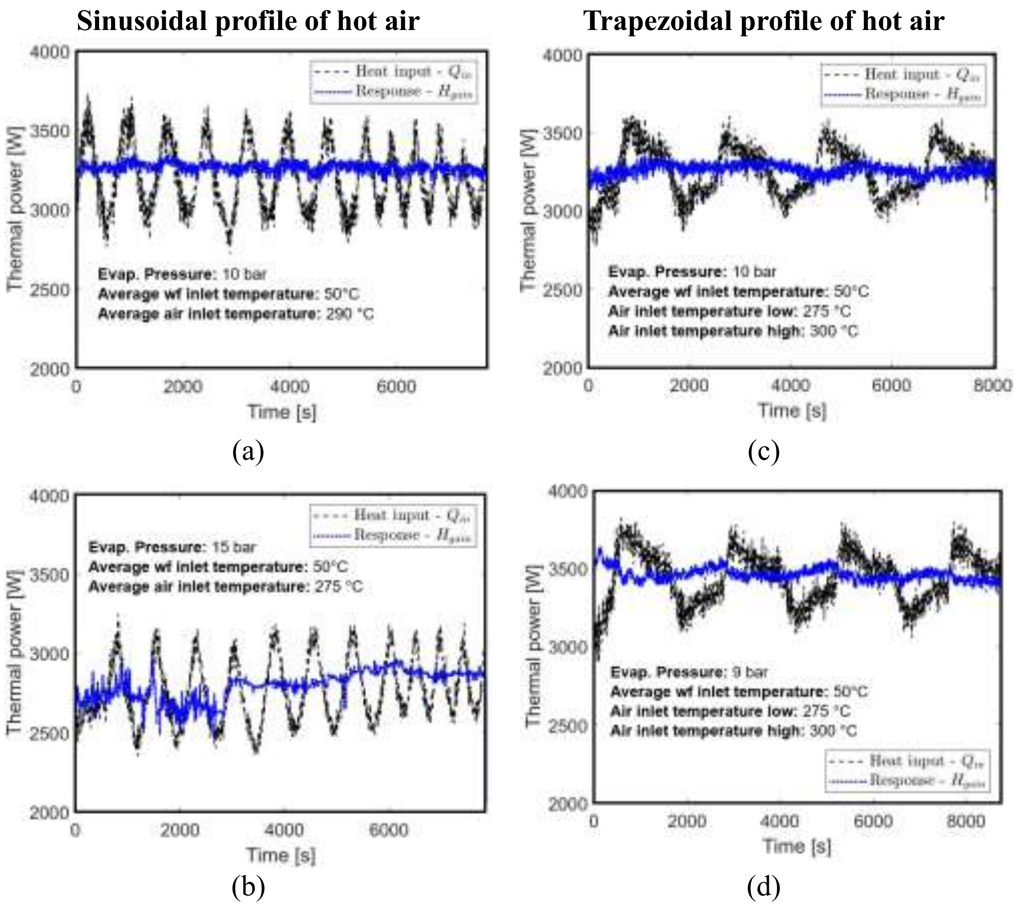


Figure 5: Measurements of thermal response of ORC evaporator subjected to different profiles of the hot air. (a) and (b) heat input and response under sinusoidal fluctuations. (c) and (d) heat input and response under trapezoidal fluctuations.

The flow rates are obtained from the measurements of the flowmeters. The enthalpy of the air and of the working fluid R245fa depends on the temperature and pressure of the thermal state and is obtained from the open source library Coolprop using the measured temperatures and pressures on the test rig. By using Q_{in} and H_{gain} the plots can show graphically how the thermal inertia of the evaporator "dampens" the response of the system to fluctuations of the heat source. It is to be noted that the "trapezoidal profiles" of the heat input are no longer completely trapezoidal, this is due to the thermal inertia of the system that also affects the hot air outlet temperature and thus the heat transferred in the evaporator.

5 DISCUSSION

5.1 General characteristics of the dynamic response

It can be observed from Figure 5 that the ORC evaporator on the test rig has an important "damping" effect to the fluctuations of the heat source. This thermal inertia comes from many factors but in the case of this particular evaporator the high mass of the fins plays an important role.

In the case of continuous, sinusoidal fluctuations the speed of change of the heat source also plays a role on how much the evaporator can "dampen" these fluctuations. For instance, in Figure 5b for the first part of the test, the heat input frequency of fluctuation is the slowest (1.04 mHz), and in that case the response of the system exhibits some mild fluctuations, whereas for the second part of the test with faster frequencies of fluctuation (1.67 mHz and 2.78 mHz) the fluctuations are basically "filtered-out" by the high thermal inertia of the evaporator. For the data shown in Figure 5a only the faster frequencies are present and thus the system basically filters-out completely the fluctuations.

In the case of the trapezoidal profiles of hot air (Figure 5c and d) it also shows how the system reacts very slowly to the ramp-ups and ramp-downs, due to the large thermal inertia of the evaporator. In fact, the working fluid does not really have sufficient time to reach a new steady-state. There is also some asymmetry for the response in case of ramp-down compared to ramp-up. This is, in part, credited to the asymmetry of the heat input profile, which was inevitable due to hardware constraints of the test rig equipment.

It must be stated that, there was not a significant difference on the response times for different evaporator pressures, indicating that the most important aspect that affects the dynamic response is the mass of the evaporator and the mass of the working fluid inside the evaporator.

5.2 Effect of fins thermal inertia on thermal response of evaporator

The data from the sensors has been compared to simulation data from a dynamic model of the ORC evaporator built in the Modelica language and simulated in the commercial software Dymola. The dynamic model is a finite-volume discretized model that considers the geometry of the heat exchanger including the mass and material properties of the tube and fins. The dynamic model is the same as the one used by Jiménez-Arreola *et al.* (2018). The details of the model can be found in the same reference.

Model considering thermal inertia of fins | **Model NOT considering thermal inertia of fins**

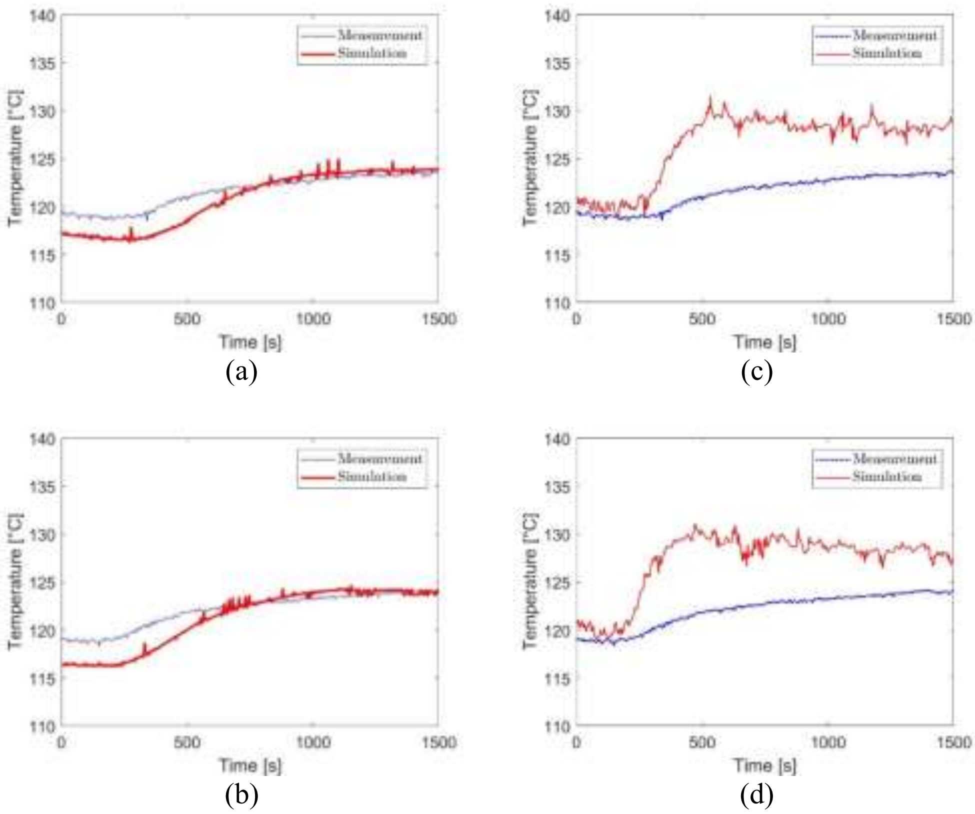


Figure 6: Comparison of experimental results with simulations results from dynamic model for ramp-up changes of the heat source. (a) and (b) comparison with a dynamic model that considers the thermal inertia due to the fins, (c) and (d) comparison with a dynamic model that neglects the thermal inertia due to the fins.

Figure 6 shows the comparison of the measured data (blue line) to the data from the simulations of the model (red line) for a ramp-up change on the hot air temperature. The data corresponds this time to the outlet temperature of the working fluid in the evaporator. It is relevant to note that the only way the dynamic model can have a good agreement with the experimental data is if the model considers comprehensibly the contribution of the fins' mass to the thermal inertia of the evaporator. The evaporator geometry on the test-rig as shown in Figure 3 was built with a fin geometry whose mass cannot be neglected. On the left-hand side of Figure 6, data from the model that considers the mass, thermal conductivity and thermal resistance of the fins material according to its geometry is shown. Whereas in the right-hand side of the figure, data from a model that neglects the thermal inertia of the fins is shown. In the Modelica model, the thermal inertia of the fins is accounted for by means of the mass of the fins that is added to the mass of the heat exchanger wall for each discretized cell.

It is clear from the figure that the model that neglects the extra material contribution to the thermal inertia does not even have a qualitative agreement with the measured data. Such a model reacts faster to the changes on the heat source conditions including a slight over-shoot, whereas the real ORC evaporator reacts much slower reaching a new steady-state after a long transient.

5.3 Importance of ORC evaporator design and sizing for a custom dynamic response.

As it can be observed from the previous discussion, the geometry and sizing of the ORC evaporator plays a key role on the dynamic response of the system. For instance, how fast or how slowly the ORC evaporator reacts to sudden changes of the hot source. In the case presented, the fin mass is the difference between a damped and slow response or a fast response but with an overshoot.

In this study the fin mass and geometry is the one who changes the dynamic response of the system, but other factors can influence this response time. For instance, the diameter to length ratio of the tubes, or the number and configuration of the tubes or conduits in heat exchanger with multiple passages.

Although the heat exchanger thermal capacity (UA) is defined by the requirements of the system (i.e. that the organic fluid is completely evaporated or slightly super-heated at the outlet of the evaporator), multiple geometry configurations and sizes can be selected that correspond to the same heat exchanger thermal capacity. In this way, the behavior of an ORC system under dynamic circumstances can be customized at the design stage by means of the geometry, material selection and sizing of the ORC evaporator. This *design-based* dynamic behavior approach such as the one proposed by Jiménez-Arreola *et al.* (2018), contrasts with a conventional *reactive-based* method of designing the ORC evaporator, on which the system is sized and specified based on purely steady-state nominal conditions and only afterwards a control system is selected to deal with the dynamic behavior of the system.

Recommendations, guidelines and a methodology to choose the right geometry of heat exchanger for a desired dynamic behavior of the system has been discussed by Jiménez-Arreola *et al.* (2018) for systems with direct evaporation, as well as a comparison with the case of indirect evaporation (Jiménez-Arreola *et al.*, 2019). This research has been done on a system with a 2.5-3.5 kW_{th} capacity, for a larger system the response time would be scalable to the larger mass of the heat exchanger and of the working fluid, thus a larger system with the same type of evaporator geometry would be slower to respond while a smaller system would react faster.

The dynamic behavior of the ORC system is of critical importance when there are heat source fluctuations. The correct transient performance along with a suitable control strategy is necessary to ensure that the ORC system works at an optimal operating point at most times and within permissible limits that will not compromise the integrity of the system – for instance, due to chemical decomposition of the working fluid.

6 CONCLUSIONS

The experimental campaign performed on the test rig has provided some relevant data on the dynamic response of an ORC evaporator to continuous and discrete changes of the heat source. It is also shown how one particular aspect of the heat exchanger design, i.e. the fin mass, can dramatically change the thermal inertia of the ORC evaporator and thus the dynamic behavior of the whole system. This is an important aspect to consider when designing and sizing components for ORC systems in which the heat source have thermal power fluctuations at most times. For a practical application, the response time of the evaporator must be compared to the expected or known variability or fluctuations of the relevant heat source. This information can be fed to an optimization tool to obtain an optimum mass of the evaporator that could, for instance, remove some of the frequencies of fluctuation of the heat source, subject to any constraint on the size of the system that the specific application imposes. In the future, further tests can be performed with other geometries of the ORC evaporator to showcase how different geometry designs and dimensioning can alter the dynamic response of the system.

NOMENCLATURE

				Subscript	
$\dot{Q}_{in}$	heat transferred from hot air	(kW)		wf	working fluid
$\dot{H}_{gain}$	enthalpy gained by working fluid	(kW)		air	air (hot source)
$\dot{m}$	mass flow rate	(kg/s)		in	inlet
h	specific enthalpy	(kJ/kg)		out	outlet

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