Designing the dynamic response of Organic Rankine Cycle evaporators in waste heat recovery applications

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**DESIGNING THE DYNAMIC RESPONSE OF ORGANIC
RANKINE CYCLE EVAPORATORS IN WASTE HEAT
RECOVERY APPLICATIONS**

MANUEL JIMÉNEZ ARREOLA

**Interdisciplinary Graduate School
Energy Research Institute @ NTU**

2019

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RANKINE CYCLE EVAPORATORS IN WASTE HEAT
RECOVERY APPLICATIONS**

MANUEL JIMÉNEZ ARREOLA

INTERDISCIPLINARY GRADUATE SCHOOL

A thesis submitted to the Nanyang Technological University in
partial fulfilment of the requirement for the degree of Doctor of
Philosophy

2019

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

02 August 2019

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I have reviewed the content and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

02 August 2019

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Date

A handwritten signature in blue ink, consisting of stylized initials 'AR' followed by a long, sweeping horizontal stroke.

.....

Asst. Prof. Alessandro Romagnoli

Authorship Attribution Statement

This thesis contains material from 4 papers published in the following peer-reviewed journals and conference proceeding where I was the first author.

Chapter 2 is published partially as M. Jiménez-Arreola, R. Pili, F. Dal Magro, C. Wieland, S. Rajoo and A. Romagnoli. Thermal power fluctuations in waste heat to power systems: an overview on the challenges and current solutions. *Applied Thermal Engineering* **134**, 576–584 (2018). DOI: 10.1016/j.applthermaleng.2018.02.033.

The contributions of the co-authors are as follows:

- I prepared the manuscript drafts. The manuscript was revised by Prof. Alessandro Romagnoli, Dr. Christoph Wieland and Prof. Srithar Rajoo
- I compounded the literature review, performed the technical assessments, designed the sections layout and prepared and formatted all figures.
- Mr. Roberto Pili provided the methods and calculations of the economics considerations section.
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The contributions of the co-authors are as follows:

- I wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli and Dr. Christoph Wieland
- I performed all the dynamic simulations, built the response time maps and provided the discussion and interpretation of results.
- Dr. Christoph Wieland and Alessandro Romagnoli assisted with ideas for the development of the response time maps.

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- I wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli, Dr. Christoph Wieland and Mr. Roberto Pili
- I performed all the dynamic simulations, built the response time maps, performed the application case study and provided the discussion and interpretation of results.
- Mr. Roberto Pili assisted on the interpretation of the results.
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The contributions of the co-authors are as follows:

- I wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli and Dr. Christoph Wieland
- I performed all the dynamic simulations, built the response time maps, performed the application case study, developed the concepts of the amplitude ratio and provided the discussion and interpretation of results.
- Dr. Christoph Wieland and Alessandro Romagnoli assisted with ideas for the development of the methodology

02 August 2019

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Manuel Jiménez Arreola

Abstract

This dissertation investigates an alternative method for Organic Rankine Cycle (ORC) systems to manage thermal power fluctuations in waste heat recovery (WHR) applications.

Organic Rankine Cycle is one of the most prominent technologies for power generation from waste heat sources. However, due to their nature as residual energy from an upstream process waste heat sources typically present a fluctuating behavior that makes the recovery of the heat for power generation a challenging task. On ORC systems in particular, the high variability of the waste heat thermal power can lead to system inefficiencies due to off-design conditions and in extreme cases to chemical decomposition of the ORC fluid or to expander damage due to liquid droplets.

Because of this thermal power fluctuations, an adequate control system may be required to maintain reliable operation of the ORC system. However, that may not be sufficient and additional measures are often put in place to ensure operation within safe boundaries. The most common is the implementation of heat transfer fluid as an intermediary for the heat transfer process of the waste heat to the ORC effectively damping the fluctuations. Another option is the addition of an external thermal energy storage unit. However, intermediary heat transfer fluids or external energy storages increase the complexity of the system, reduce its potential for high thermal efficiency and increase the weight and volume of the system, which is limiting factor in some applications such as the mobile.

This dissertation explores a different approach. It proposes that the thermal inertia of the heat exchanger that is used as the evaporator in the Organic Rankine Cycle can be customized by design in order to obtain a dynamic behavior that provides a more robust system to the changes in thermal power and enables the possibility of a potentially more efficient system with lower footprint and complexity. For these purposes, the evaporator design is reimaged in order to include its thermal inertia as an essential factor to be considered.

In order to investigate the dynamic behavior and performance of different ORC evaporators and their thermal inertia, a full dynamic model must be used. This model is successfully validated against experimental data to increase the confidence on the results. The model is used then to simulate the dynamic behavior of different candidate evaporators.

Based on extensive simulation results of different types and geometries of heat exchangers, a methodology for the evaporator design, with an emphasis on dynamic behavior, is progressively developed and finally integrated into a cohesive procedure. The novel methodology incorporates new tools and concepts such as the response time maps, dynamic regimes and amplitude ratios.

Notably, the results and the methodology developed in this dissertation are not bound by any specific case and can be applied to any situation of ORC systems recovering waste heat.

Lay Summary

The objective of this thesis is to aid in the development of more sustainable energy systems. One way to increase the sustainability of energy systems and processes is by increasing their energy conversion efficiency and to reduce the waste of energy resources. A method to reduce waste of energy resources is by recuperating residual heat -from sources such as industrial processes or engines- which is normally discarded to the ambient and left unused. This residual heat is called *waste heat*. The energy of the recuperated *waste heat* can be transformed to produce electrical power which is an energy form that is more versatile and easier to utilize.

One particular technology that is used to transform unused *waste heat* into electrical power is called *Organic Rankine Cycle (ORC)* and is studied in this thesis. Although this technology is very well established there are still some limiting factors that hinder its applicability. One of those limiting factors is the fact that the *waste heat* content is typically intermittent and has a fluctuating nature. Since *ORCs* work better when the supply of *waste heat* is of a regular nature, different ways to adapt the *ORC* to a fluctuating *waste heat* supply have been researched previously. This include the integration of an external unit to store momentarily the energy of the *waste heat* and deliver a more constant supply or the implementation of complex control schemes.

What this thesis proposes is that the fluctuations of the *waste heat* can be managed by the unconventional design of one of the *ORC* components which is a heat exchanger called *evaporator*. In this way, there is no need to add complex arrangements or additional equipment to the basic unit of *ORC*. For this purpose, new concepts are introduced and a novel methodology for the design of the *evaporator* is developed and presented. The results from the application of the methodology prove promising results for the stable operation and improvement of the energy conversion efficiency of *ORCs* while keeping the complexity simple and the size small.

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Nomenclature

Symbols

M	Mass, kg
V	Volume, m ³
t	Time, s
ρ	Density, kg/m ³
h	Specific enthalpy, J/kg
p	Pressure, Pa
$\dot{m}$	Mass flow rate, kg/s
$\dot{Q}$	Heat transfer rate, W
$\dot{q}$	Heat flux, W/m ²
T	Temperature, K
c	Specific heat capacity, J/(kg·K)
R	Thermal resistance, K/W
A	Heat transfer area, m ²
θ	Film heat transfer coefficient, W/(m ² ·K)
U	Overall heat transfer coefficient, W/(m ² ·K)
Δp	Pressure drop, Pa
f_D	Friction factor, -
L	Tube length, m
D	Diameter, m
FL	Filling level, -
ϵ	Tube roughness, m
τ_{ev}	Evaporator response time, s
τ_w	Wall conduction time constant, s
th	Thickness, m
α	Thermal diffusivity, m ² /s
k	Thermal conductivity, W/(m·K)
ξ	Statistical error, -

ΔH_{vap}	Enthalpy of vaporization of the fluid, J/kg
λ	Air to fuel ratio, -
Γ	Dynamic regime number, -
T_{step}	Source step time, s
f_{sin}	Frequency, sinusoidal source, Hz
$\dot{H}$	Enthalpy gain, W
AR	Amplitude ratio, -
ΔT_{max}	Maximum allowable temperature fluctuation, K
Cp_{wf}	Average heat capacity of working fluid, J/(kg·K)
AR_{max}	Maximum allowable amplitude ratio, -
D_h	Hydraulic diameter
N	Number of (e.g. tubes), -
cl	Clearance, m
κ_{plate}	Plate wave number, -
EF	Expansion factor (plate heat exchanger), -
u	Velocity of fluid, m/s
μ	Dynamic viscosity, Pa·s
G	Mass flow per unit area per unit time, kg/(m ² ·s)
g	Gravitational constant, m ² /s
ζ	Darcy friction factor, -
x	Vapor mass fraction, -
j_c	Collburn factor, -
f_D	Darcy friction factor, -

Dimensionless numbers

Ja_{lv}	Jakob number, -	$Ja_{lv} = \frac{C_{p,v}(T_v - T_{sat}) + C_{p,l}(T_{sat} - T_l)}{\Delta H_{vap}}$
Nu	Nusselt number, -	$Nu = \frac{\theta \cdot D_h}{k}$
Re	Reynolds number, -	$Re = \frac{\rho \cdot u \cdot D_h}{\mu}$

Pr	Prandtl number, -	$Pr = \frac{c_p \cdot \mu}{k}$
Fr	Froude number, -	$Fr = \frac{G}{\rho_l^2 g D}$
Bo	Boiling coefficient, -	$Bo = \frac{\bar{q}}{G \cdot \Delta H_{vap}}$
Hg	Hagen number, -	$Hg = \frac{\rho \Delta p d_h^3}{\mu^2 L_p}$

Subscripts

w	Metal wall (heat exchanger)
wf	Working fluid
exh	Exhaust
oil	Thermal oil
in	Inlet condition
out	Outlet condition
int	Internal side
ext	External side
exp	Expander
pump	Pump
tank	Tank
liquid	Liquid
is	Isentropic
sh	Super-heating
tv	Throttle valve
eff	Effective
solid	Solid material (heat exchanger)
hx	Heat exchanger
tube	Tube
banks	Tube banks
tubes/bank	Tubes per banks

fin	Fin(s)
ports	Ports
louver	Louver(s)
lam	Laminar
turb	Turbulent

Acronyms/Abbreviations

ORC	Organic Rankine Cycle
WHR	Waste heat recovery
IC	Internal combustion (engine)
WHP	Waste heat to power
TEG	Thermo-electric generator
WHTC	World Harmonized Transient Cycle
PID	Proportional-integral-derivative (controller)
MPC	Model predictive control
TES	Thermal energy storage
SHS	Sensible heat storage
LHS	Latent heat storage
PCM	Phase change material

Chapter 1

Introduction

This chapter presents the main thesis of this dissertation. A brief description of the problem that is trying to be solved provides the rationale. In a concise way the objectives, scope and original contributions envisioned for this work are described. An overview of the thesis structure and each Chapter's contents is also provided.

1.1 Thesis Statement

This dissertation investigates an alternative method for Organic Rankine Cycle (ORC) systems to manage thermal power fluctuations in waste heat recovery (WHR) applications.

The main thesis of this work is that the thermal inertia of the ORC evaporator can be customized at the design stage in order to improve the dynamic performance and control of ORC systems and subsequently a methodology for this purpose is proposed and proven.

1.2 Background

One of the most important concerns in the contemporary world is the development of sustainable energy systems to ensure that current and future energy needs are met and reduce harmful consequences such as climate change. According to a typical energy hierarchy such as the one shown in Figure 1-1, the reduction of energy use and improvements in the energy efficiency sit in the top of priorities to build a more sustainable future.

Figure 1-1 Energy hierarchy for sustainability, adapted from [1], [2]

According to these priorities, waste heat recovery is an effective method to reduce consumption of energy resources and increase the energy efficiency of current energy conversion technologies. Furthermore, there is a huge energetic and economic potential on the utilization of waste heat [3], [4]. Among the technologies available for power generation from waste heat the Organic Rankine Cycle (ORC) is the most well-established due to its superior maturity, reliability and simplicity [5].

However, one of the most important technical and economic barriers that limit the implementation of waste heat to power (WHP) systems is the fluctuating and/or intermittent nature of the waste heat source. These fluctuations occur inherently in most waste heat sources such as industrial processes or engines, due to non-uniform production, batch processes or irregular loads. Waste heat recovery from Internal Combustion (IC) engines is particularly challenging, especially the mobile applications, due to the highly dynamic conditions of the waste heat during varying driving conditions.

Figure 1-2 illustrates some of the problems and challenges that an ORC system faces when dealing with fluctuations of the available waste heat source thermal power. For this, a conceptual profile of a waste heat source with fluctuations of thermal power over time is used. Four exemplary points of operation of the ORC system are also shown represented by a red circle, a yellow star, a green triangle and a blue square.

ORC systems are normally designed for a nominal operating point, called design-point (represented by the yellow star in Figure 1-2). At design-point, the conversion efficiency is maximum because all components work at rated conditions. However, if the thermal power of the source is lower than the design point, the system operates at part-load (represented by the green triangle in Figure 1-2), leading to a less efficient conversion efficiency. Furthermore, too low thermal power (represented by the blue square in Figure 1-2) can lead to the risk of liquid droplets in the ORC expander that can damage irreversibly the system. If the thermal power or the temperature of the source become abnormally high (represented by the red circle in Figure 1-2), the ORC working fluid suffers the risk of chemical decomposition, which leads to degradation of performance and eventually system

failure. Furthermore, thermal fluctuations lead the ORC system to be operating at transients most of the time. These transients must be recognized when proposing solutions to improve the performance of the ORC system.

Figure 1-2 Effect of thermal power fluctuations in performance of an ORC system (a) Conceptual thermal power profile and different points of operation (b) Typical efficiency curve of ORC system and unsafe areas of operation.

It has to be noted that in Figure 1-2 the thermal power available in the evaporator for a given sized heat exchanger will vary with fluctuations of both or any of the mass flow rate or temperature of the heat source.

Because of the high variability of the source, a control system is required to maintain reliable operation of the system. However, that may not be sufficient and additional measures are often put in place to ensure operation within safe boundaries. The most common is the implementation of heat transfer fluid as an intermediary for the heat transfer process of the waste heat to the ORC, which helps to effectively dampen the fluctuations. This type of layout is called indirect evaporation. However, indirect evaporation increases the complexity of the system on the assumption that an additional piece of equipment and fluid does so, reduces its potential for high thermal efficiency and may increase the weight and volume of the system, the latter being a limiting factor in some applications such as the mobile. Direct evaporation on the other hand, offers technical and thermodynamic

advantages but increases the chance that the system may operate outside the safe boundaries.

1.3 Objectives and Scope

This dissertation investigates an alternative for ORC systems to manage thermal power fluctuations of the waste heat. It focuses in one of the components of the ORC cycle that is key for this alternative: the evaporator. Direct evaporation is identified as the desired ORC layout due to its simplicity, potential for superior energy conversion and the system's smaller size.

The objective is then to find a solution to implement direct evaporation under highly dynamic varying boundary conditions. A compromise is sought between safe operation, increased thermodynamic performance, and reduction of weight and volume of the system for size-sensitive applications. A methodology to design the heat exchanger for improved dynamic behavior under direct evaporation is developed.

In order to investigate the behavior and performance of the ORC evaporator under fluctuating heat, a full dynamic model must be used. This model must be validated against experimental data to increase the confidence on the results. The model can be used then to simulate different candidate evaporators for different fluctuating characteristics of the waste heat source. Because of its more challenging nature, waste heat recovery from the exhaust mobile IC engines is used as the benchmark case to showcase the methodology developed by this thesis. The results and the methodology can then be generalized to other fields of waste heat recovery with ORC.

1.4 Dissertation Overview

The dissertation follows a familiar structure of scientific works in order to present the research question, the methods used to answer the question, a comprehensive description and discussion of the results and a closing chapter where the results are confronted to the

original thesis statement.

Chapter 1 introduces the technical thesis that is intended to be proven as well a general background of the problem that it tries to solve. The motivations, objectives and intended contribution of the thesis are stated.

Chapter 2 presents a comprehensive literature review about the topics relevant to the thesis following a logical progression of the different areas and with an emphasis on the most recent state of the art. The literature review is intended to highlight the gaps and opportunities for improvement on each area presented. At the end of the chapter, the identified research gaps are recapped showing how the objectives and scope of this thesis follow naturally from them.

Chapter 3 presents, in detail, the methods used to answer the research questions. This includes a through description of the mathematical model used in the simulation as well as the specifications of the laboratory test rig used to validate the model. The focus is to report all the information required to duplicate satisfactorily any result found on this thesis.

Chapter 4 presents the validation of the models with experimental results and introduces the concepts and the methodology for the analysis of evaporator response times. This is done for a basic geometry of ORC evaporators, highlighting qualitatively, quantitatively and in the most general way, the contributions of the main factors to the response times.

Chapter 5 expands the concepts of *Chapter 4* to more complex geometries belonging to the real types of heat exchangers that can be used for direct evaporation. The potential of the methodology to customize the thermal inertia of the ORC system is shown on the case of the profile of an IC engine exhaust during a standard driving cycle.

Chapter 6 applies the methodology for the highly relevant case of replacing an indirect evaporation with a direct evaporation layout. It analyzes qualitatively and quantitatively the dynamics of both layouts according to frequencies and amplitudes of fluctuation of the

source and proposes a geometry of a direct evaporator that can better handle the fluctuations of an IC engine exhaust during a driving cycle.

Chapter 7 concludes the thesis by contrasting the results presented throughout with the original thesis and providing a summary of the concepts and methodology introduced and its possible impact, as well as the limitations and further improvements that can be developed in the future.

1.5 Original contribution of this work

The most important concepts introduced by this work as well as novel research outcomes can be summarized in the following list:

1. The re-thinking of evaporator design in ORC, not just from a standard heat exchanger optimization but also considering the thermal inertia as an important aspect for the ORC system dynamic performance.
2. A fundamental analysis of the ORC evaporators' response time and the relative impact of several factors on it.
3. A proposed methodology to customize the thermal inertia of ORC evaporators to better match the expected dynamic characteristics of the waste heat profile.
4. The possibility of replacing the more convenient indirect evaporation layout with the more efficient and size-reducing direct evaporation layout while minimizing the difficulties related to it.

Chapter 2 *

Literature review and research gap

This Chapter presents a comprehensive literature review on the topics relevant to the thesis. Starting from the general field of ORCs in waste heat recovery applications, the areas of opportunities are identified and the literature review narrows down into the state of the art of the methods to manage thermal power fluctuations and the research that focuses on ORC evaporators and dynamic modelling of ORCs. The research gaps are identified for each area reviewed and finally assembled together and examined in the last part of the Chapter. This gap provides the rationale behind the thesis direction in the following Chapters.

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2.1 Waste Heat Recovery using ORC systems

There is an enormous potential on Waste Heat Recovery worldwide. Forman et al. [6] has estimated the waste heat potential at global scale to be around 72% of the world's primary energy consumption, with 38% of it available at temperatures above 100 °C. The U.S. department of technology [3] estimated that 20 to 50% of the energy annually consumed by the industry is lost as waste heat.

Power generation from waste heat is usually economically and technically feasible when the temperature of the heat source is higher than 150 °C [7]. Some sources that have been identified as the most suitable for waste heat to power (WHP) systems include energy intensive industries with heat loads in the MW range such as the steel industry (waste heat temperatures in the range of 300 to 100 °C), or the cement industry (waste heat temperatures in the range of 200 to 400 °C) along with Internal Combustion (IC) engines in the hundreds of kW to the MW ranges (waste heat temperatures in the range of 200 to 900 °C).

Table 2-1 shows a summarized comparison of the most well-known technologies for WHP. Among the available technologies, those based on Rankine cycles are the most widespread. Other thermodynamic cycles include the Kalina cycle which are 15 to 25% more efficient than ORCs at the same temperature level[7]. However, they are scarcely used due to their complexity and non-mature state [8]. Thermo-electric generator (TEG) represents an alternative to thermodynamic cycles [9], but the adoption of this technology is still hindered by its high capital cost and low thermal efficiencies in the range of 5%.

Comparing traditional steam Rankine cycles to ORCs, steam Rankine cycles have a superior economical and technical feasibility in the high power (MWe to GWe range) and/or high heat source temperature (above 400 °C) than ORCs. Furthermore, thermal efficiencies of steam Rankine cycles can be as high as 40% due to the larger temperatures of the heat sources. ORCs on the other hand, are more economical and efficient at low power ranges (kWe to few MWe) [5] and are better technically suited for temperatures

below 400 °C. They also have superior flexibility in terms of heat temperature matching and better part-load behavior [10]. Therefore, ORCs are the most suitable technology for WHP in the low and medium power ranges and for any size at temperatures below 400 °C. Due to the lower temperature of the heat source, typical thermal efficiencies of ORCs are between 5% to 20%.

Table 2-1 Comparison of waste heat to power technologies. [5], [7-10] Heat source temperatures and power output values are ranges for technical and economic feasibility.

	Heat source temperature	Power output (electrical)	System complexity	Costs	Maturity
Steam Rankine Cycle	300-600 °C	1 MW-1 GW	Lower	Lower	Higher
Organic Rankine Cycle	100– 400 °C	1kW–10 MW	Lower	Lower	Higher
Kalina Cycle	100-500 °C	1kW–10 MW	Higher	Higher	Lower
Thermoelectric generator	250-850 °C	1W to 10 kW	Lower	Higher	Lower

The Organic Rankine Cycle (ORC) is a closed thermodynamic power cycle that is based on the conventional Rankine cycle, which is used in steam power plants. The difference lies in that the ORC uses an organic compound as the working fluid circulating inside the cycle instead of water/steam. This difference allows the ORC system to recover heat from sources at lower temperatures compared to a steam cycle. Furthermore, the choice of the organic compound represents an additional degree of freedom that allows to thermally match the waste heat source adequately. Good review papers that cover most of the aspects on ORC technology, components, different applications and market outlooks include the works by Hung et al. [11], Quoilin et al. [12] and Colonna et al. [5] among many others.

The basic configuration of an ORC system is shown in Figure 2-1, along with an exemplary T-S diagram of the thermodynamic cycle. The working fluid starts the cycle as saturated liquid or slightly sub-cooled (1) at the lower pressure level and then it enters a pump where is compressed to the higher pressure level (2). Afterwards the fluid is heated up in the

evaporator until it reaches the saturated or super-heated vapor state (3). The vaporized fluid then expands to the lower pressure level in an expander that can be of the turbo-machine or positive-displacement type producing mechanical work. The work output may be used to drive a generator and produce electrical power. A tank is often included between condenser and pump to store the fluid, but it does not have any effect on the thermodynamic cycle.

Figure 2-1 (a) Basic configuration of ORC system and (b) T-S diagram.

2.1.1 Waste heat sources and profiles

The most suitable waste heat sources for power generation with ORCs are found in energy-intensive industrial processes as well as IC engines from the transport sector. This is due to the temperature level and economic potential [6]. As it has been mentioned, the relevant waste heat sources for ORCs very often experience fluctuations of the available thermal power. These fluctuations can be classified on whether they are due to variations of the mass flow rate, temperature or both simultaneously. Another quantity of interest is how fast these variations take place. Figure 2-2 shows some examples of profiles from the literature with different fluctuations characteristics. Table 2-2 provides a summary of the fluctuation characteristics of some noteworthy waste heat sources.

Figure 2-2 Fluctuation in waste heat sources (a) Steel billet reheating furnace: mass flow fluctuations [13], (b) Clinker cooling: temperature fluctuations [14], (c) Electric arc furnace (EAF) after water cooling system: fluctuations of both mass flow and temperature [15], (d) Diesel engine exhaust: fast fluctuations [16].

Table 2-2 Selected waste heat sources relevant for ORC systems with the temperature range and fluctuation characteristics of the waste heat stream [13]–[18]

Waste heat source	Waste heat temperature (°C)	Significant type of fluctuation	Typical range of periods of fluctuation (frequency)
Steel – Coke dry quenching	650-1000	Temperature	Minutes – hours
Steel – Electric arc furnace	1370-1650	Mass flow and temperature	Minutes
Steel – Billet reheating furnace	700-1200 (no preheater) 300-600 (with preheater)	Mass flow	Minutes
Cement – clinker cooling	200-400	Temperature	Minutes – hours
IC engine exhaust	200-900	Mass flow and temperature	Seconds - minutes

Waste heat from the steel industry can be harnessed from the electric arc furnace (EAF) and billet reheating furnaces. In EAF the waste flue gas typically experiences large fluctuations of both temperature and flow rate due to its batch nature [15], [19], [20]. In billet reheating furnaces, flue gas temperature variation is minor due to a fixed temperature profile inside the furnace in order to meet the required properties of the slabs [21], whereas the flow rate fluctuations can vary due to irregular or discontinuous production rates. In the cement industry, waste heat from clinker cooling is particularly suitable for power conversion [22]. The mass flow rate of the clinker cooling air typically stays relatively constant while its temperature presents large fluctuations [14], [23] due to intermittent production or limited control of the cooling carrier. Mobile IC engines exhaust on the other hand present simultaneous fluctuations of both the flow rates and temperatures [16] depending on the driving conditions. The fluctuations from industrial sources have in common that the typical range of periods of fluctuations of the source is in the minute or even hours of time scales. On the other hand, IC engines fluctuations is in the seconds to minutes time scales. This shows that the handling of fluctuations on IC engines is more challenging but also represents a bigger opportunity for improvement.

This dissertation is based on results focusing on IC engines WHR implementation due to the fact that it is more demanding. In this way, the methodology presented can also be applied to the less challenging field of industrial waste heat. The next sub-section focuses on the literature of ORC systems for waste heat recovery of IC engines.

2.1.2 ORC for IC engine WHR

It is estimated that around 60% of the primary energy in IC engines is lost as waste heat [7]. Approximately half of that energy is lost through the exhaust. Waste heat recovery of IC engines by means of ORC can be done for stationary or mobile engines. Furthermore, the waste heat available in IC engines includes the higher temperature (200-900 °C) thermal power present in the engine exhaust or engine gas recirculation (EGR) system [24] and the lower temperature (80–100 °C) thermal power present in the engine coolant [25]. Because of the temperatures, most of the exergy of IC engines waste heat is present in the

exhaust.

Due to the existence of different sources of waste heat at different temperature levels, different architectures of ORC have been proposed in the literature. These include using engine cooling heat as a preheating source of a single-loop ORC cycle [25] as well as double-loop configurations and cascaded cycles. In the dual-loop configuration a high temperature ORC loop is used to recover heat from the engine exhaust and a second low temperature ORC loop is used to recover heat from the engine coolant as well as residual heat from the high temperature loop [26]–[30]. Other cascaded configurations with more complex layouts and multiple heat exchangers [31] or dual stream expanders [32] have also been proposed. Other modifications include using mixtures as working fluids [33]–[35] allowing to a more flexible heat exchange that can better match the different waste heat sources in the engine [36].

However, an important aspect to consider of the architectures that take advantage of the different waste heat temperature levels is the added complexity and the significant increase in the footprint and weight of the overall system. This makes such modifications interesting in stationary engines, but unpractical in mobile IC engines. For this reason, this dissertation will focus on the waste heat from the exhaust only where the higher exergy is present.

In terms of stationary engines, there is an important potential on WHR of Diesel generators. Baidya et al. [37] described the implementation of ORC for waste heat recovery of Diesel generators in off-the-grid locations. Chatzopoulou et al. [38] performed a whole system optimization of an integrated Diesel generator and ORC for combined heat and power, and then investigated the off-design conditions with varying conditions of the engine [39].

Mobile IC engines are more challenging due to the fast dynamic conditions present and the volume and weight restrictions in order to make the implementation of ORC economically viable. This is because additional weight requires additional fuel demand on the engine that may offset the surplus provided by the WHR system. Additional volume also competes with the volume available for transport capacity. A detailed analysis on the cost vs revenues

of ORC implementation on the transport sector is provided on the work by Pili et al. [40].

The application of ORC to recover waste heat from mobile IC engines has been considered for maritime engines [41]–[43], trains [44], heavy duty long-haul trucks [45] and even light-duty passenger cars [46]. Long-haul trucks and cars are the fields that are more volume and weight sensitive and with the faster dynamics due to the less steady journey conditions compared to ships and trains, although all of them present those challenges to some degree.

Wang et al. [47] compared diesel and gasoline IC engines, and concluded that the application of ORCs on light-duty gasoline cars is not economically and practically viable in the current state of the art whereas heavy-duty diesel engine is a promising field.

ORC implementation in WHR from Diesel engine trucks has been proposed for many years including installations and laboratory tests as early as the 1970s [48]–[51]. More recent studies have been focused on the optimization, economic feasibility and tackling the challenges of the dynamic conditions of the exhaust waste heat. Espinosa et al. [52] reviewed the approaches, constraints and modelling techniques for the implementation of ORCs in commercial trucks.

Dolz et. al [53] analyzed the incorporation of Rankine cycles to a Diesel engine at system-level in order to identify irreversibilities and the best configuration. It recommended ORCs over steam Rankine Cycle due to the variable engine operating conditions. Serrano et al. [54] expanded the work to more complex layouts but still concluded that the gain in efficiency was not enough to justify the added complexity and volume of the system. Hountalas et al. [55] estimated a 11.3% in brake-specific fuel consumption (BSFC) when incorporating an ORC to a heavy-duty Diesel engine for heat recovery of exhaust only. Macián et al [56] presented an iterative methodology for the optimization the ORC as a bottoming cycle to a vehicle Diesel engine stablishing reduction of BSFC while minimizing space requirements and cost. Yue et al. [57] recommended to perform the optimization of the IC engine with ORC as bottoming cycle in an integrated way rather

than the IC engine being optimized first. This in order to achieve a better performance overall. Yang et al. [58] proposed a genetic algorithm in order to optimize an ORC for Diesel engine for a wide range of operating points of the engine and evaporation pressure, super-heating degree and condensation pressure as key parameters.

All these studies, only investigated steady-state performances at system-level. In terms of dynamic performances Xie et al. [59] utilized a simple dynamic model of the ORC with single pipe heat exchangers to study the dynamic characteristic during a driving cycle. Dynamics models of the ORC cycle have also been used for studies incorporating control strategies during dynamic operation of truck engines [44], [60], [61]. A more detailed literature review on dynamic models of ORCs is presented in Section 2.3.1.

Most of the experimental data of ORCs incorporated into Diesel engines is limited to laboratory scale and research based test rigs, since the concept has not reached commercial maturity. Battista et. al [62] investigated in a test bench the effect that exhaust WHR with ORC has on the pressure losses of IC engines, as well as its contribution in mechanical power and the effect of the extra weight. It concluded that at severe off-design conditions the ORC presents important challenges to keep the fluid vaporized or below the decomposition temperature. Shu et al. [63] used a test-rig to investigate the performance of an ORC recovering heat from a Diesel engine exhaust with an indirect evaporation arrangement including an intermediary thermal oil loop. Huster et al. [64] validated a dynamic model of an ORC system with the use of data from an IC engine under the World Harmonized Transient Cycle (WHTC).

Because the exhaust heat profile depends on the particular driving conditions in mobile heavy-duty diesel engines, standard driving cycles have been developed to study the dynamic conditions of the engines in common road driving conditions. Of this, the most widely used in recent times is the World Harmonized Transient Cycle [65]. Figure 2-2d shows the exhaust profile of a Diesel engine of an omnibus at operation during the WHTC [16].

2.1.3 Summary and assessment

From the literature review, it has been determined that waste heat recovery with ORC is technically and economically promising for the energy-intensive fields with waste heats with temperature ranges of 200 to 600 °C. This includes industrial sources and stationary or mobile IC engines exhaust. Regarding mobile IC engines, heavy-duty long haul trucks is the applications that presents the more challenges due to the dynamic conditions while still being promising in the economic feasibility.

However, ORC application in mobile heavy-duty diesel engines has remained mostly in research and laboratory study stage, and more research and breakthroughs are needed to ensure its commercial application. This is therefore, an area with great necessities and opportunities for research.

2.2 Managing thermal power fluctuations in ORC systems

As it has been mentioned previously, waste heat to power systems in general work better at the design point because all the components are chosen and optimized for operation at that condition. Mass flow rate and temperature fluctuations of the source each and both present a challenge to all waste heat to power systems including ORCs. First, there is a drop in efficiency due to part-load behavior. Second, extreme fluctuations can lead to unfeasible or unsafe operation.

ORC systems provide more flexibility than the regular Steam Rankine Cycle, as it can work at part-load condition down to 10% of the maximum power output according to Erhart et al. [10]. However, the average output of an ORC system under thermal power fluctuations has been investigated by Kim et al. [66] to be 20 to 40% lower compared to a constant design-point.

Furthermore, large thermal power fluctuations can lead to irreversible damage of the system. Too high temperatures or thermal power can lead to chemical decomposition of the

working fluid. Too low temperatures or thermal power can lead to the fluid not being fully vaporized and therefore to damage of the expander due to liquid droplets.

Figure 2-3 shows an overview of the current state of the art solutions to manage the thermal power fluctuations of the waste heat in ORC systems according to current commercial technologies and the literature. They can be classified in two main categories, those which solely focus on stream control (either the waste heat stream or working fluid or both) and those which aim to buffer the fluctuations via an intermediary thermal energy storage.

Figure 2-3 Principal solutions in commercial applications and literature to manage waste heat thermal power fluctuations in waste heat to power systems.

2.2.1 Stream control

Stream control is needed in ORC systems subjected to thermal power fluctuations when no intermediary thermal energy storage is present. This is to ensure that the working fluid is fully vaporized before the expander so that it will not get damaged by liquid droplets and that the working fluid does not get overheated which can lead to chemical decomposition in the case of organic fluids.

Not until recently the issue of control stream of ORC on WHR applications due to the variability of the thermal power has been fully investigated. While this also attains to industrial applications, the issue becomes even more imperative on the IC engines mobile

applications due to the high transients related to the driving conditions.

Stream control can be performed by means of:

1. By-pass valves for either or both of waste heat stream and working fluid
2. Adjusting the flow of the working fluid from a tank by means of pump or expander speed.

Figure 2-4 shows different ways in which control valves can be implemented for stream control in ORC systems for WHR.

Figure 2-4 Examples of different waste heat to power stream control configurations. (a) Intermediary thermal oil stream control of flow entering different sections of waste heat boiler [67] (b) By-pass valve controlling amount of waste heat stream entering the waste heat boiler [68] (c) Dilution of waste heat stream with fresh air (d) Working fluid by-pass to protect the expander and mass flow control with variable speed pump.

Regarding by-pass valves, Mazzi et al. [67] analyzed the case of an ORC operating in the glass industry. In their proposed solution, the ORC evaporator is separated into three separate heat exchangers: an economizer, an evaporator proper and a super-heater. The thermal state of the system was controlled by means of controlling the flow of the hot stream (a thermal oil) to the separated heat exchangers by means of variable flow valves and by-pass flows in between each of these sections (see Figure 2-4a).

Grelet et al. [60] proposed a system controlled by two stream control valves in an ORC system recovering heat from an IC engine exhaust. One by-pass valve located in the exhaust stream to control the amount of waste heat available to the ORC and a second one located in the working fluid stream just before the expander, to by-pass the fluid if it is not fully vaporized and protect the expander from any liquid. Other researchers have also made use of by-pass valves for the waste heat stream. These include the research by Peralez et al. [68] with a by-pass valve to regulate the waste heat stream from an IC engine and Shi et. al [69] that used the recirculation of some of the waste heat stream back into the evaporator and having a controllable T-valve on the ORC side before the expander. Feru et. al [70] also used valves to control the exhaust flow as well as controlling the pressure level of the working fluid with the expander speed.

Another solution to reduce temperature fluctuations in the waste heat stream is to mix the waste heat stream with a fresh cool stream by means of a mixing valve as it has been implemented by the company Turboden and described by Dal Magro et al. [13] and Pili et al. [71]. This is shown in Figure 2-4c

As it has been mentioned, apart or together with by-pass valves, the working fluid flow can also be controlled by varying the pump or expander speed, thus changing the pressure mass flow conditions of the working fluid in the evaporator.

Manente et al. [72] used the pump rotational speed and a variable nozzle valve opening on the expander to control the mass flow and pressure in the evaporator according to the characteristic curves of both the pump and expander. Hu et al. [73] set up a sliding pressure

control strategy using the movable inlet guide vanes of the turbine to control the evaporator pressure while setting a constant super-heating temperature in the model. Fu et al. [74] investigated the required evaporator pressure needed in order to have a constant fluid mass flow optimized for the turbine with varying conditions of the waste heat. They however did not specify the means to achieve this pressure.

Regardless of the type of stream control, the controlled variables need to be manipulated via a suitable controller. The simplest type of controller is the PID or PI controller. Quoilin et al. [75] used traditional PID controllers to test stream control strategies, using the pump speed to control super-heating and the expander speed to control evaporator pressure. Wang et al. [76] used a neuro-PID controller to control the outlet temperature of the evaporator, achieving a better step response than the traditional PID controller.

Other more sophisticated types of controllers include modern type's architectures. For instance, Peralez et al. [44] used a low order dynamic model of an ORC for recovery of a diesel-electric engine on board a railcar to solve a real-time optimization problem by an Adaptive-Grid Dynamic Programming (DP) algorithm. The study found a 7% increase in the amount of recovered energy compared to a simpler state of the art controller based on static optimization. Hou et al. [77] proposed a minimum variance controller algorithm for a generic 100 kW ORC for WHR applications, showing that this controller could rely simply on real-time measured data of the system performance, without the need for a precise mathematical model of the system while achieving satisfactory set-point tracking.

Another modern control architecture that has gained attention in the last years is Model Predictive Control (MPC) [78] which uses a model of the system to predict its behavior and calculate the optimum input signal by an optimization algorithm. Zhang et al. [79], [80] used a controller auto-regressive integrated moving average (CARIMA) model to identify the dynamics of the ORC and develop a multi variable constrained MPC algorithm. Hernandez et al. [81], [82] also developed a constrained MPC algorithm and compared it to a PID scheme. The research shows that MPC can lead to operating points closer to the constraints, maximizing the output of the system. Esposito et al. [83] also developed a non-

linear MPC for automotive application and applied it to an overtaking driving profile based on experimental data allowing for a theoretical 6.2% reduction in fuel consumption of the vehicle.

Some limitations that the control systems can encounter include the finite time constant for variables such as the pump speed or valve opening, the actuators maximum displacement, as well as physical constraints such as the achievable working fluid mass flow dictated by the storage tank size or the need for cavitation prevention.

One thing lacking in the reviewed literature in stream control is that for the most part they make use of simplified models for the ORC components, specially the heat exchangers. Another important aspect that most of the literature on control applied to ORC systems have in common is that they focus on the control algorithm for a system whose components have been designed already. However, the component design has a direct impact on the control of the system due to, for instance, the thermal inertia of the components. The component design with control in mind is then, one area of opportunity that can be identified and has not been addressed to the best of the author's knowledge.

2.2.2 Thermal energy storage (TES)

Another option to mitigate the detrimental effect of thermal power fluctuations is through thermal energy storage (TES). Thermal energy of the waste stream can be stored during peak times and reused when the thermal power available is lower effectively leveling off the fluctuations of the source. In this way the operation of the ORC is shifted to a narrower range closer to the design-point. TES options are installed as an intermediary system between the waste heat stream and the ORC working fluid. Depending on the type of material used for the TES, they can be classified into Sensible Heat Storage (SHS) or Latent Heat Storage (LHS).

Traditional options of SHS such as molten salt tanks are used in ORCs driven by solar thermal heat such as it is the case in the arrangement presented by Pantaleo et al. [84].

However, variations in solar-thermal application is on a seasonal time scale, unlike waste heat recovery applications where the dynamic characteristic is in the order of minutes or even seconds for IC engines waste heat (see Figure 2-2d in section 2.1).

Pressurized water tanks of up to 21 bar have been proposed as SHS in waste heat recovery with ORC in the food industry [85] as a cheaper alternative to LHS, in a quasi-steady model.

The introduction of an intermediary thermal oil loop between the heat stream and the ORC power cycle is widely used in ORC systems for WHR as a buffer in order to lower the amplitude of fluctuations of the source [5]. This thermal oil loop can also be considered as a low term SHS since the fluctuations are dampened due to the sensible heat of the thermal oil. This type of arrangement is called “Indirect Evaporation”.

Latent heat storage (LHS) employs the latent heat of fusion or vaporization of a medium in order to store energy and it offers many advantages for WHR applications compared to the SHS option. The energy density for LHS is higher than SHS, meaning that it can be used in volume-restricted applications such as the transport sector. Moreover, because of the phase change these devices can provide an almost constant temperature and heat rate output. In this way the ORC can theoretically operate under steady state condition close to the design-point. LHS options include steam accumulator [86], [87], as well as technologies based on Phase Change Materials (PCMs). PCMs store energy that is above their temperature of phase change and release energy when the waste heat stream temperature is below it. PCMs selection is based on several criteria [88]. Materials such as paraffins [89], [90] are common options for PCMs on the melting temperature range of -30 to 120 °C. For the temperatures encountered in WHR applications (200 to 800 °C) the materials that are considered generally include metal alloys [91], [92] or fused salts [93].

Examples of LHS use in WHR include a PCM-based storage device for buffering thermal power fluctuations of EAF flue gas. The device consists of concentric stainless steel pipes with aluminum encapsulated as PCM [94], [95]. In the case of WHR from IC engines, there is a big opportunity and necessity for thermal buffering of the exhaust waste heat [96].

However, due to volume and weight constraints [40], the actual implementation is challenging and has remained mostly at research stage [97].

Figure 2-5 Conceptual schematic of the differences on the effect of TES in thermal power fluctuations. (a) SHS or thermal oil loop – attenuation of fluctuations, (b) LHS – near constant output (optimum case).

Comparing the different options of TES, from the literature and commercial applications it can be concluded that SHS can be significantly cheaper than LHS. SHS, however, has a larger density compared to LHS and delivers a non-isothermal temperature output compared to LHS [98]. This is shown in Figure 2-5. A specific type of SHS, the intermediary thermal oil loop is the most widely used in ORC systems, to the point that it is the most widely used option for commercial WHR. This option is commonly referred to as Indirect Evaporation.

2.2.3 Summary and assessment

Table 2-3 shows an assessment and comparison of generalized strengths and weaknesses for the different technical options to manage thermal power fluctuations in ORC systems

based on a critical analysis of the literature reviewed. It is to be noted that this assessment is very general and purely qualitative, and for a given system with some particular conditions some strengths or weaknesses may differ slightly.

Table 2-3 Comparison of technical options of thermal power fluctuation management according to their strengths (+) and weaknesses (-). A neutral assessment is indicated by (o).

	Stream control Heat source by-pass	Stream control Heat source dilution	Stream control Working fluid flow control	Sensible heat storage	Latent heat storage
Fluctuation removal	+	+	-	++	+++
Additional volume/weight	-	-	o	---	---
Implementation effort	-	-	---	-	---
Control complexity	-	-	---	-	---
Capital cost	-	-	-	---	---
Efficient energy use potential	-	---	+	++	+++

As a general trend of the assessment in Table 2-3, two general areas of trade-off can be identified between TES and stream control. One area can be grouped as that of efficiency and fluctuation removal. Another area is that of the additional volume/weight on the system that is also somewhat related to the capital cost. TES can potentially offer a more efficient use of the energy due to the better performance on fluctuation removal. However, this comes at the cost of a higher economical expenditure and additional volume and weight to the system.

In volume and weight restricted applications as is the case in mobile IC engines, the additional volume and weight of the system makes TES system a non-feasible solution. This reduces the options in these applications to that of stream control. From the stream control options, it can be seen that working fluid stream control is the most attractive option in terms of efficient use of the energy. It seems that these options offer the best trade-off in

terms of no additional volume/weight and energy efficiency for mobile applications. However, this comes at the cost of a more complex control requirement and higher implementation effort. An area of opportunity arises then: how to implement working fluid stream control keeping the efficiency and size advantages while reducing the control complexity and implementation effort.

2.3 Dynamic behavior of ORC systems

In order to study and propose solutions for the issue of the thermal power fluctuations in ORC systems, it is necessary to understand and model the dynamics of the system. For this reason, full dynamic models of the ORC systems are required.

Dynamic models are significantly more complex than steady-state models because they use systems of several differential equations instead of algebraic ones. Therefore, it is important to study the literature on dynamic modelling to be able to choose a correct modelling approach and find ways to simplify the models to make the models solvable and computationally efficient.

2.3.1 Dynamic modelling

One of the first publications to address the dynamic modelling of ORC systems is that by Wei et. al [99]. In the paper, full dynamic models of the evaporator and condenser were used while the pump and expander used quasi-steady models based on performance data and points. The use of only quasi-steady models of the pump and expander is a standard practice in dynamic modelling of ORC systems. The reason is that the time scales of the heat exchangers are considerably larger than the rest of the components of the ORC system. This fact, allows for the simplification of the ORC systems and focuses the effort of the dynamic modelling to the heat exchangers.

Quoilin et. al [75] developed dynamic models of a small scale ORC for WHR using the Modelica language for the purpose of implementing different control strategies for

optimizing the performance of the system under fluctuations of the heat source obtaining the best result with an strategy based on controlling the evaporating temperature based on an optimized value based on the waste heat conditions. It used a finite volumes approach for the dynamic models of the expander and condenser. Pump and expander used quasi-steady models.

Different approaches exist for the dynamic modelling of heat exchangers with phase change in one of the fluids. The main methods can be classified in two: the finite volumes approach and the moving boundary approach [100].

The finite volumes approach, discretizes the heat exchanger flows and wall into N number of volume elements with a defined dimension. Each element is a control volume where the thermodynamic variables are treated as homogeneous but time dependent, and energy and mass balances are performed. Mass and energy cross the boundaries of each volume cell into or from one of the other cells.

The moving boundary approach is similar, but instead of having N cells, there are only three cells with time dependent dimensions. Each cell corresponds to one of the phases present in the working fluid side of the heat exchanger: liquid, two-phase and vapor. Similarly to the finite volumes, mass and energy balances are performed for each cell, but in this case the length (or volume) of the cell is also time dependent and appears on the governing equations as one more variable.

Figure 2-6 shows a diagram of both discretization approaches for dynamic modelling of heat exchangers.

Desideri et. al [101] compared and validated both types of modelling approaches for ORC evaporators and condensers. The models were developed in the Modelica language (source) as part of the ThermoCycle library [102]. The research concluded that both approaches are well suited for modelling the dynamics with the moving boundary approach being considerably faster. The moving boundary approach however is unsuitable to small

capacities heat exchanger due to the assumption of constant void fraction overestimating the dynamics. More recently Chu et al. [103] proposed an interesting algorithm coupling both finite volumes and moving boundary approaches.

Figure 2-6 Different approaches for discretization in dynamic modelling of heat exchangers. (a) Finite volumes approach (b) Moving boundary approach.

2.3.2 Importance of dynamic response as design criteria

The dynamic response of the ORC system is an important aspect that should be considered at design stage for WHR applications. Sun et al. [104] recognized this, and studied the performance and dynamic behavior of ORC systems under fluctuations of an exhaust gas as the heat source. The paper studied individually the performance under sinusoidal variations of the exhaust temperature as well as the performance under sinusoidal variations of exhaust mass flow rate. The objective was to find the optimum design point of the ORC system under the range of fluctuations. It found that the optimum point to

design the ORC is the upper boundary of the fluctuations ranges and that, when there are temperature fluctuations, the optimal turbine inlet pressure obtained using a dynamic model is 7.9% lower than the one obtained by a purely static analysis, suggesting that it is important to consider the dynamic response at design point.

Pierobon et al. [105], went one step forward and proposed a methodology to design energy conversion systems, highlighting the importance of taking into account the dynamic requirements at design stage. The methodology used a parametrization of the dynamic behavior from models based on components from the Modelica-based ThermoPower library [106]. The study stated that systems conceived for flexible operation under fluctuations of the source must take into account the full dynamic and transient performance as early as possible in the design stage in order to avoid costly changes afterwards. It presented a case study where the methodology was applied for an ORC system with direct evaporation and a control system recovering heat from a gas turbine. It found that the ORC evaporator volume played a key role on keeping the ORC fluid below the chemical decomposition temperature. Under sudden changes of the heat load, systems with small volumes of the evaporator below 45 m^3 had an overshoot and undershoot of the performance that was unacceptable. On the other hand, a very large volume was more difficult to control due to its higher inertia. It found an optimum volume from the control point of view of 65 m^3 . This work highlights, how the design of the ORC evaporator clearly affects the performance and the operation within safety margins of an ORC under dynamic conditions of the heat source.

Tong et al. [107] studied the evaporator dynamic response on an IC engine-ORC system for different changing conditions of the IC engine exhaust. It used step changes of engine load to study the response time of the ORC components. It found that the ORC system had a delay to the changes in the heat load and reported response times of the evaporator temperature of 16 seconds for the ORC system considered. The evaporator however was designed based mainly on steady-state performance without considering its dynamic behavior during the design stage. A different design of the evaporator for the same input and output conditions would have led to a different response time.

These are some examples of the literature that has identified the dynamic behavior as an important design criterion. However, it must be noted that in general the literature that focuses on this idea is very limited for the moment, so there is a huge area of opportunity to develop this concept.

2.3.3 Summary and assessment

Dynamic models are required to study the behavior of ORC systems under fluctuating thermal power. Only the heat exchangers (evaporator and condenser) need to be modelled as fully dynamic because of their much slower dynamics compared to the rest of the components. The other components of the ORC system (pump, expander) can be modelled as quasi-steady.

For the dynamic modelling of the evaporator, two main approaches can be identified: finite volumes and moving boundary. Moving boundary allows for slightly faster computational time but finite volumes can produce more accurate results depending on the number of discretization cells selected. The modelling language most widely used in the ORC scientific community is the Modelica language which can be compiled in commercial simulations environments such as Dymola.

Because of the importance of the evaporator as the key component to understand the dynamic behavior of the ORC system it is pertinent to focus on that component in order to comprehend the challenges and to suggest a different way to handle thermal power fluctuations that has been proposed before. Furthermore, the importance of dynamic response as design criteria has been highlighted but not sufficient research effort has been done in this topic.

2.4 ORC evaporators

From the previous literature review, it is clear that when studying the dynamics of an ORC

system with thermal power fluctuations of the heat source, the evaporator is the key component due to its slower dynamics and it being the link with the heat source.

2.4.1 Direct vs indirect evaporation

Figure 2-7 Different ORC layouts for working fluid evaporation (a) Indirect evaporation (b) Direct evaporation.

For simple single-loop ORCs, the evaporation of the working fluid can be performed in one evaporator unit, where heat is exchanged directly from the exhaust. This is called direct evaporation. Alternatively, the exhaust can transfer the heat to a different heat transfer fluid (usually a thermal oil or cooling water [108]) in one heat exchanger and subsequently the heat transfer fluid flows into a second heat exchanger where the ORC working fluid is evaporated. This is called indirect evaporation. Figure 2-7 shows a diagram of both of these configurations.

Indirect evaporation is often a preferred option in laboratory or real-world equipment [109] for WHR from IC engine or other variable heat sources. Indirect evaporation can be

regarded as the implementation of a particular type of sensible heat storage (the intermediary fluid), thus damping considerably the fluctuations of the thermal power and the problems that arise from them. Furthermore, the intermediary fluid reduces the supply temperature compared to that of the source, with a lower risk of chemical decomposition of the fluid, especially when using halogenated hydrocarbons (low-temperature) fluids [110].

The ease of implementation of indirect evaporation compared to direct evaporation is confirmed by the literature available on experimental ORC test rigs. For instance, Yu et al. [111] built a test rig of an indirect evaporation cascaded ORC with a thermal oil intermediary circuit finding a 5.6% increment. Alshamari et al. [112] experimentally studied the performance of a pilot ORC at different Diesel engine operating conditions. The ORC recovered heat from the exhaust by using a thermal oil intermediary loop. Likewise Li et al. [113] used an experimental setup utilizing a thermal oil as the heat transfer fluid from the waste heat of a CHP unit. Ntavou et al. [114] designed and tested an indirect evaporation two-stage ORC for variable heat source conditions including a heat transfer fluid circuit using Monoethylene Glycol.

Direct evaporation, on the other hand, has a lower capability for damping of a variable heat source, thus leading to an increased risk of damage to the expander or working fluid. ORC direct evaporation from exhaust gases is certainly conceptually much simpler. Therefore, it is very often considered as the option in the literature studies at system level [38], [58], [115] and component level [116].

Many of the simulation researches found in the literature that utilize a direct evaporation layout for IC engine WHR disregard the technical difficulties that such a configuration has during transients. Song et al. [43] proposed the design optimization of a system for IC engine heat recovery. However, no dynamic considerations were presented. Koppauer et al. [117] proposed a model-based optimization to find the best steady state operating points of an ORC recovering heat from a Diesel engine. Shu et al. [118] proposed a system evaluation with a multi-approach strategy that includes 1st and 2nd law analysis as well as

economic considerations, without any consideration of the dynamics.

Most of the works that do take into consideration the technical challenges of direct evaporation on WHR with variable thermal power include papers focused mainly on control design [31], [119], [120]. These works, for the most part, consider the ORC components as already optimized and focus only on implementation of control during operation. Furthermore, they very often use simplified geometries of heat exchangers (i.e. a single tube) that do not completely capture the thermal inertia of real heat exchangers with more complex geometries.

In terms of costs, for indirect evaporation the capital cost of the additional piece of equipment and fluid must be considered, however the total costs may be offset if the implementation of a heat transfer fluid loop results in a reduction of the ORC size and capital cost of the ORC-alone [71]. However, for instance for mobile applications, the revenues due to a reduction in fuel consumption are tied to the weight of the ORC system [40]. The increase in weight in indirect evaporation may lead to lower or no reduction in fuel consumption.

2.4.2 Heat exchangers types and geometries

There has been considerable research regarding heat exchanger options for ORC evaporators. One concern in the integration of ORCs in engine exhaust heat recovery is that of the additional back pressure on the engine due to the presence of the ORC. It has been found that the additional back pressure from a shell and tube heat exchanger on the exhaust has a minimal effect on the acceleration performance of the engine [121]. Nevertheless, direct evaporation geometries usually use finned surfaces in the exhaust side. This can incur in higher back pressure. In this sense, the evaporator design must ensure that the back pressure is kept within acceptable limits.

Shell and tube is the most common type of heat exchanger in the industry and its optimized design as ORC evaporator has been the subject of several studies [122]–[124]. However,

they are usually more suitable for large scale ORC systems, such as geothermal applications in which the hot fluid is a liquid and flows through the tubes (e.g. the kettle boiler configuration [125]).

In small scale applications, compact heat exchangers as evaporators are a preferred option [126]. The most widely researched option for compact ORC evaporators are brazed plate heat exchangers [127]–[129]. Brazed plate evaporators, however, are better suited when the hot fluid is a liquid [130] as it is the case only for indirect evaporation from IC engines exhaust.

In the case of direct evaporation from gaseous fluids, due to the lower heat transfer coefficient of gases compared to the evaporative heat transfer coefficient, a good way to ensure the compactness of the heat exchanger is to have extended surfaces on the gas side [125]. Options for direct evaporation include fin and tube [58] and louver fin multi-port flat tubes [131] geometries. Studies considering fin and tube heat exchanger as evaporator from gaseous fluid typically aim at the optimization only at design point [132].

Comparisons of the performance of different types of heat exchangers as ORC evaporators, including shell and tube and compact geometries have also been undertaken based on system optimization [124] or on cost-effectiveness and return of investment [133].

However, all of the studies mentioned early only focus on heat exchanger design optimization in isolation of the system, and do not take into consideration the dynamic response of the heat exchanger.

2.4.3 Summary and assessment

For a simple ORC configuration there are two types of evaporation layouts usually considered in the literature. Direct evaporation where the heat from the main heat carrier is transferred directly to the ORC fluid and indirect evaporation that uses an intermediary heat transfer fluid. Compact types of heat exchangers for indirect evaporation include

brazed plate heat exchangers. For direct evaporation, suitable heat exchangers have extended surfaces on the gas side. These include the fin and tube and the louver fin geometries.

Most of the effort has been focused on standard heat exchanger optimization at design point. However, the selection and geometry of the evaporator at design stage is a significant factor that ultimately will affect the dynamics of the system and thus the effectiveness of the control. An area that has not been considered in previous research is the selection of the type and geometry of the heat exchanger based also on its thermal inertia and the response to the fluctuations of the heat source. This is ultimately important for direct evaporation where the damping is considerably lower compared to indirect evaporation.

2.5 Research gap

From the comprehensive review of previous and contemporary researches, areas of opportunity are identified as well as the research gap that this dissertation proposes to address.

As per the literature review, it is decided to use as case studies automotive heavy-duty diesel engines exhaust heat recovery. This is because mobile engines present the more challenging dynamics and a methodology that can address these challenges can be easily extrapolated, if needed, to less dynamic conditions. It also represents the opportunity to advance the knowledge on a field that has not reached commercial application stage due to the lack of technical feasibility.

An assessment of the literature review on the options to manage thermal power fluctuations in ORC systems reveals that for volume and weight-restricted applications, working fluid stream control with direct evaporation offers the best trade-off between the minimization of system size and thermal efficiency. However direct evaporation is a more challenging layout to implement due to its lower capability to protect the fluid and components from extreme changes in thermal power.

It is clear that, especially in automotive applications, the dynamic response of the system is an important criterion for the design of the system that it is, more often than not, overlooked. It is also clear that the evaporator is the key component to understand the dynamic response of the system, due to its direct link to the fluctuations and its lower dynamics compared to the pump and expander.

The design of the dynamics of the evaporator is, therefore, a vital aspect for ORC systems in automotive applications and, in fact, in all WHR applications. However, no work to date, satisfactorily utilizes this fact to propose a way to mitigate the challenges due to thermal power fluctuations of the source.

This dissertation proposes a new way to manage that challenge. One way that does not use an external thermal energy storage or heat transfer fluid loop, but proposes the design or dimensioning of the evaporator in unconventional ways in order to use its damping capabilities due to its thermal inertia. In this way the complexity of using ORC with direct evaporation can be reduced.

Chapter 3

Modelling and experimental methods

In this Chapter, the methods used to obtain the data to prove the thesis propositions are described in detail. They include the mathematical models for the results based on simulations as well as the experimental methods used to validate the model. The dynamic models are elaborated showing the constitutive differential and algebraic equations, the discretization arrangements, the modelling language and the simulation software utilized. Prominent emphasis is dedicated to the dynamic model of the evaporator which is the focus of this thesis. The specifications of the laboratory test rig for the model validation are documented including information on the equipment, heat exchanger geometry and instrumentation employed. The information presented should be adequate to be able to reproduce the results in this thesis by replicating the methods used.

3.1 Introduction to modelling of ORC systems

Dynamic models are very important to investigate the transient behavior of ORC systems under fluctuating thermal power sources. It is crucial to have some sort of prediction of the performance of a system before committing any resources for the building of any prototype or final product. Furthermore, if a screening of many different types and geometries is desired, as it is the case in this work, it is only practical to utilize dynamic models than build hundreds of different prototypes.

Although steady-state models of ORC systems can be good enough in some applications, in waste heat recovery applications, the highly dynamic conditions may require the use of accurate enough dynamic models. Dynamic models are considerably more complex than simple steady state since they make use of non-linear time dependent differential equations.

The dynamic model of an ORC system can be simplified by comparing the relative dynamics of each of the main components of the system. It is a well-known fact that the heat exchangers (i.e. the evaporator, condenser and recuperator if any) exhibit considerably slower dynamics than the expander and the pump [75], [99]. From this knowledge, the expander and pump models can be simplified with a quasi-steady state treatment, while the full dynamic models are reserved to the heat exchangers.

In this thesis, the focus is on the behavior of the ORC evaporator. Full detailed dynamic models are used for the evaporator, while the other components are incorporated into the models as boundary conditions.

3.2 Modelling language and simulation environment

The models in this thesis are constructed in the Modelica language. The Modelica language is an equation-based, object-oriented programming language that is especially useful for simulation of dynamic systems based on differential equations dependent on time. The choice of this language for the models is based on the fact that it is the most widely used

language for dynamic models of Organic Rankine Cycles and other low grade heat thermal energy systems in the research community (see the literature review in Section 2.3) Therefore, the models and simulation results of this thesis can be easily replicated and compared to other works by the scientific community on this research field.

Furthermore, all the models in this thesis use as a basis the models available in the TIL library developed by TLK Thermo [134] and written in the Modelica language. This library contains detailed models of heat exchangers and other ORC components that have been validated in house. Therefore, the use of this library components allows to have a very good confidence on the results of the simulations of this work. The TIL library models have been modified and further expanded for the purposes of this thesis investigation.

The models are simulated in the commercial software Dymola [135] which is a simulation environment for Modelica-based models. Dymola is also the most widely used software for dynamic simulations of ORC systems and components in the scientific research community as it has been determined from the literature review.

3.3 Dynamic models of heat exchangers

The dynamic models of the heat exchangers in this thesis are based on the finite volume approach. This is a 1-D discretization approach in which each cell is assumed as a volume with homogenous thermo-physical properties inside. Two types of cells are considered: fluid cells and wall cells. The fluid cells are implemented to discretize the flow of both fluids through the heat exchanger. It is assumed that changes on the properties of the fluid do not vary on the radial direction and only on the direction of the flow. Therefore, the only governing equations assumed for each fluid cell are linear differential equations for mass, momentum and energy balances. It is assumed that mass and momentum are only transferred through the boundaries of the cell into or from other fluid cells in the direction of the flow, whereas energy in form of heat is only transferred through the cell boundary into or from a wall cell. The wall cells discretize the metal wall that separates both fluids in the heat exchanger. For this type of cell, the only governing equations assumed is a linear

differential equation for the energy balance. It is assumed that there is no heat conduction to other wall cells, and that energy in form of heat is only transferred through the cell boundary into or from a fluid cell. Figure 3-1 illustrates and summarizes the two types of cells considered in this finite volume approach along with the assumptions made for each cell.

Figure 3-1 Concept and assumption of the two types of discretization cells for the heat exchangers
(a) Fluid flow cell (b) Metal wall cell

Although other modelling approaches exist, such as the moving boundary model [99], [101], the finite volume method can achieve more accurate results depending on the number of volume cells [99]. A drawback however is the higher computational effort. However, since the models in this work focus on the evaporator, the computational effort is still manageable, and the accuracy is preferred.

The number of discretization volumes, or cells, for the heat exchanger is chosen as a compromise between accuracy and computational effort. It has been found throughout the simulation campaigns that exceeding 40 cells in ORC heat exchangers does not significantly change the results, while it increases the computational time. Therefore, in this work 40 cells are considered for the heat exchangers unless otherwise stated.

3.3.1 Conservation equations

The physics of the heat exchangers can be described by the conservation of three properties: mass, energy and momentum. The internal fluid in the heat exchangers in this work is, unless stated otherwise, the working fluid of the ORC. Selecting the enthalpy h and pressure p as the fundamental thermodynamic variables, the corresponding governing differential equations for the internal fluid (working fluid) are as follows:

Mass balance in each cell (working fluid):

$$\frac{dM}{dt} = \frac{d(V \cdot \rho)}{dt} = V \cdot \left(\frac{\partial \rho}{\partial h} \frac{dh}{dt} + \frac{\partial \rho}{\partial p} \frac{dp}{dt} \right) = \dot{m}_{in} - \dot{m}_{out} \quad (3-1)$$

Energy balance in each cell (working fluid):

$$V \cdot \rho \cdot \frac{dh}{dt} - V \cdot \frac{dp}{dt} = \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} + \dot{Q}_{int} \quad (3-2)$$

Momentum balance in each cell (working fluid):

$$p_{in} - p_{out} = \Delta p \quad (3-3)$$

In the case of the momentum equation, since the focus of this work is focused on the thermal response of the fluids, a quasi-steady equation for momentum is considered instead of a full differential equation. This is in order to simplify the convergence of the dynamic model. The momentum balance is expressed in engineering terms, through the pressure drop.

The heat transfer from the external fluid to the internal fluid is achieved through the metal wall of the heat exchanger. Both flows of the heat exchanger are connected in the model

thermally through a discretized wall component. This finite volume wall component accounts for the thermal resistance and heat capacity of each section of the metal wall. Conduction in the wall is only considered in the mean heat transfer direction. The metal wall heat balance in each cell is governed by the differential equation as follows:

$$c_w \cdot M_w \cdot \frac{dT_w}{dt} = \frac{(T_{w,ext} - T_{w,int})}{R_w} \quad (3-4)$$

Where R_w is the thermal resistance across the thickness of the wall material in units of thermal power per temperature difference. The thermal resistance depends on the material and geometry of the wall. The heat transfer from the wall interfaces to the internal fluid is calculated as follows:

$$\dot{Q}_{int} = \theta_{int} \cdot A_{int} \cdot (T_{w,int} - T_{int}) \quad (3-5)$$

Where θ_{int} is the internal fluid heat transfer coefficient, calculated with a suitable heat transfer correlation.

The heat transfer from the external fluid to the wall interface is calculated differently depending if the external fluid properties are taken into consideration (Section 4.4 of Chapter 4, Chapter 5 and Chapter 6) or omitted (Section 4.5 of Chapter 4). In the conceptual case of Section 4.5 of Chapter 4 the thermal energy from the external fluid is taken as the known homogenous heat flux $\dot{q}_{ext}$, and the heat transfer is calculated as:

$$\dot{Q}_{ext} = \dot{q}_{ext} \cdot A_{ext} \quad (3-6)$$

When the external fluid properties are taken into consideration, such as in the real heat exchangers case of Chapter 5 and Chapter 6, the heat transfer from the external flow is calculated as:

$$\dot{Q}_{ext} = \theta_{ext} \cdot A_{ext} \cdot (T_{ext} - T_{w,ext}) \quad (3-7)$$

Where θ_{ext} is the internal fluid heat transfer coefficient, calculated with a suitable heat transfer correlation.

The external fluid (the hot source fluid), considered in this work is, unless stated otherwise, a gaseous exhaust. When the external fluid properties are taken into consideration, the external flow is also discretized in finite volumes using the same number of cells as the working fluid. However, only steady-state mass and energy balances are considered in order to avoid unnecessary complexity of the model, reduce computational effort, and help with convergence of the models. The justification for this simplification is that the focus of this work is in the transients in the working fluid side and the heat accumulation in the gaseous side is negligible due to its low density. The conservation equations for the secondary fluid are:

Mass balance in each cell (external fluid):

$$\dot{m}_{in} = \dot{m}_{out} \quad (3-8)$$

Energy balance in each cell (external fluid):

$$\dot{m}_{in} \cdot h_{in} = \dot{m}_{out} \cdot h_{out} + \dot{Q}_{ext} \quad (3-9)$$

Momentum balance in each cell (external fluid):

$$p_{in} - p_{out} = \Delta p \quad (3-10)$$

3.3.2 Heat transfer correlations

The convective heat transfer of both fluids in the heat exchangers (i.e. $\dot{Q}_{int}$ and $\dot{Q}_{ext}$ in equations (3-5) and (3-7) are calculated by a replaceable heat transfer model each. The heat transfer model utilizes the heat transfer area calculated by the particular geometry of the heat exchanger (see section 3.3.5 and Appendix A). To calculate the heat rates, information on the temperatures provided by the balance equations are required as well as values for heat transfer coefficients. The heat transfer coefficients are calculated with suitable heat transfer correlations depending on the type of flow.

In the case of the working fluid (internal side), the fluid flows inside a tubular geometry for the most part. Furthermore, the flow undergoes a phase change process, therefore different heat transfer correlations are used for the convective heat transfer coefficient depending if the fluid is in the one-phase region (i.e. sub-cooled liquid or super-heated gas) or in the two-phase region. For the one-phase regions, the Gnielinski correlation [136] is used in its range of validity of Reynolds numbers between 2300 and 100,000. For Reynolds numbers higher than 100,000 the Dittus-Boelter correlation, valid in such a range, is employed [137]. For the two-phase region, the heat transfer coefficient is calculated based on the correlation by Shah [138]. All the correlations are chosen because they are non-specific for any kind of fluid, suitable for cylindrical tubular flow and are the correlations most often used in the literature for ORC systems. When the working fluid flows in non-cylindrical channels, such as in the plate heat exchanger, the hydraulic diameter is used for the calculation of the Reynolds number.

Since the external fluid in the heat exchanger, does not flow inside tubular geometries but rather in more complex flow channels, different correlations suitable for external flow are required. The convective heat transfer coefficient of the external fluid in the direct evaporation heat exchangers depends on the particular geometry of the heat exchangers. For the fin and tube heat exchanger, the correlation provided by VDI Heat Atlas for flue gases in this type of heat exchangers is applied [139]. In the case of the louver fin multi-port heat exchanger, the correlation used is the one proposed by Chang and Wang [140]

which was developed for this type of geometry.

For the indirect evaporation case, a correlation of VDI Heat atlas for the shell side shell side [141] is used for gaseous fluid of the shell and tube heat exchanger. For the thermal oil, the Gnielinski/Dittus Boelter correlations are used in the case of the shell and tube heat exchanger since the oil flows inside a tubular geometry. A specific correlation for one-phase flow in plate heat exchangers [142] is used for the thermal oil as the hot-side of the plate heat exchanger.

The heat transfer correlations used in each heat exchanger are summarized in Table 3-1, details and equations for each correlation can be found in Appendix B.

Table 3-1 Heat transfer correlations summary.

Heat exchanger type	Heat transfer correlation	
	Internal fluid	External fluid
Single-tube (isolated)	Gnielinski/Dittus-Boelter [136], [137] (single phase) Shah [138] (two-phase)	N/A
Fin and tube	Gnielinski/Dittus-Boelter [136], [137] (single phase) Shah [138] (two-phase)	VDI Heat Atlas for cross flow around finned tubes [139]
Louver fin multi-port	Gnielinski/Dittus-Boelter [136], [137] (single phase) Shah [138] (two-phase)	Chang and Wang [140]
Shell and tube	Gnielinski/Dittus-Boelter [136], [137] (single phase)	VDI Heat Atlas for shell side [141]
Plate	Gnielinski/Dittus-Boelter [136], [137] (single phase) Shah [138] (two-phase)	VDI Heat Atlas for one-phase flow in plate hex [142]

3.3.3 Pressure drop correlations

In Chapter 5, the pressure drop is calculated for the direct evaporation heat exchangers.

In the working fluid side, the pressure drop is calculated with the standard equations from the theory of friction of flows inside rough pipes. Using the Darcy-Weisbach friction factor [143] the pressure drop is calculated with the equation from Poiseuille's law [144] for laminar flow and with the Swamme Jain correlation for rough pipes [145] for turbulent flow.

In the external gas side, the pressure drop equation is different according to the type of heat exchanger. For the fin and tube heat exchanger the Haaf correlation [146] for cross flow over finned tubes is used. Whereas for louver fin multi-port heat exchangers the model proposed by Kim and Bullard [147] is employed.

The details and equations of the pressure drop correlations are presented in Appendix C.

3.3.4 Cells interconnections

The interconnection of the different finite volume cells of the internal fluid, metal wall, and external fluid in the model is done depending on the geometry of the heat exchanger considered. In this work, different types of heat exchangers are studied depending on their suitability for the type of layout of ORC system.

Figure 3-2 Volume cells interconnection for external heat flux (baseline case) heat exchanger.

In Chapter 4, a base geometry case is considered, in which the effect of the external fluid is omitted in order to study a fundamental conceptual case and isolate the factors concerning only the working fluid and metal wall. For this case, Figure 3-2 summarizes the discretization approach and interconnection of the cells. The internal fluid is discretized

in N number of cells and connected thermally to an element of metal wall each. Each element of metal wall gets heat transferred from an exterior homogenous heat flux $\dot{q}_{ext}$.

In Chapter 5 and Chapter 6 more complex and real-life geometry heat exchangers are studied. For direct evaporation layouts, the type of heat exchangers that are studied due to their suitability for gas to working fluid heat exchange are fin and tube and louver fin multi-port heat exchangers. Both of these types of heat exchangers exhibit cross-flow. The interconnection of the finite volume cells of internal and external flows for this type of heat exchangers are shown in Figure 3-3.

Figure 3-3 Volume cells interconnection for cross-flow heat exchanger.

For indirect evaporation, two heat exchangers are required, one to transfer the heat from the gaseous side to a thermal oil in liquid phase and a second one to transfer the heat from the intermediary thermal oil to the working fluid. In this case, the suitable types of heat exchanger are shell and tube heat exchanger for gas to oil, and a plate heat exchanger for oil to working fluid. These types of heat exchanger can be considered conceptually as a counter-flow arrangement. Figure 3-4 shows the interconnection of the finite volume cells for this case.

Figure 3-4 Volume cells interconnection for counter-flow heat exchanger.

Table 3-2 summarizes the cells interconnection arrangements and the type of heat exchangers they correspond to.

Table 3-2 Heat exchangers' geometries and the layouts where they are used.

Model type	Heat exchanger geometry				
	Base geometry	Direct evaporation		Indirect evaporation	
	Single Tube	Fin and tube	Louver fin multi-port	Shell and tube (gas to oil)	Plate (oil to working fluid)
External heat flux	X				
Counter-flow				X	X
Cross-flow		X	X		

3.3.5 Geometric parameters

For each type of heat exchanger, some user-defined geometric dimensions must be provided in order to define the geometry of the heat exchanger. Based on said provided dimensions, the rest of the geometry of the heat exchanger relevant for the model can be calculated.

The user-defined geometric dimensions depend on the type of heat exchanger. These include the internal tube/port diameter, wall thickness, tube length, number of tubes, fin

augmentation area, number of plates, etc.

The most important calculated geometric parameters for the model are the hydraulic diameters, the heat transfer areas, the flow volumes and the wall volume and geometry. The hydraulic diameters are required to calculate the Reynolds number and subsequently the heat transfer coefficients and pressure drops based on the relevant correlations. The heat transfer areas are required in order to calculate the heat rate based on the heat transfer coefficients. The flow volumes are required to calculate extensive fluid properties based on intensive or volume specific thermodynamic properties. The metal wall volume and geometry are needed to calculate the total heat resistance and heat capacity of the metal wall. The geometric parameters calculations based on the user-defined dimensions for each type of heat exchanger are presented in Appendix A.

3.3.6 Summary of heat exchangers models

Figure 3-5 shows a diagram of the heat exchanger model structure. The information required to define the models are the type of heat exchangers and flow arrangement, the user-defined geometry, as well as the heat transfer and pressure drop correlations. The models calculate the heat transfer and fluid properties in each cell based on this information.

Figure 3-5 Heat exchanger dynamic model structure.

3.4 Models of other components

Apart from the heat exchangers some other models of the ORC system are used in some parts of this work. The pump and expander are modeled in order to provide inlet and outlet boundary conditions to the evaporator.

3.4.1 Pump

In the pump model, the mass flow, inlet pressure $p_{pump,in}$ and enthalpy $h_{pump,in}$ are the preset or input variables. A constant pump efficiency η_{pump} is also preset. The following are the equations of the pump model:

Mass balance:

$$\dot{m}_{pump,in} - \dot{m}_{pump,out} = 0 \quad (3-11)$$

Energy balance:

$$h_{pump,out} - h_{pump,in} = \frac{(p_{pump,in} - p_{pump,out})}{\bar{\rho} \cdot \eta_{pump}} \quad (3-12)$$

The pressure increase in the pump is calculated based on the outlet pressure value. The outlet pressure $p_{pump,out}$, which is equal to the evaporator pressure, is either imposed as a boundary condition, or defined by the expander model in the ORC system model. When the pump model is used in this work, the flow through it is always connected to the evaporator and subsequently the expander.

3.4.2 Expander

The expander model assumes a constant isentropic efficiency $\eta_{exp,is}$, and a volumetric flow

intake based on a constant volume intake V_{intake} , a given expander speed n_{exp} and a constant volumetric efficiency $\eta_{exp,vol}$. The outlet pressure is predetermined as a boundary condition (the condenser pressure). When the expander is present in the ORC model, it defines the pressure in the evaporator upstream (the inlet pressure of the expander).

The governing equations of the expander are as follows:

$$\dot{m}_{exp,out} = \frac{\rho_{exp,in} \cdot n_{exp} \cdot V_{intake}}{\eta_{exp,vol}} \quad (3-13)$$

$$h_{exp,out} = h_{exp,in} + \frac{(h_{exp,out,is} - h_{exp,in})}{\eta_{exp,is}} \quad (3-14)$$

where $h_{exp,out,is}$ is the isentropic discharge enthalpy from the thermodynamic state at the outlet pressure and inlet entropy.

3.4.3 Tank

A model of an expansion tank is required for the closed loop of thermal oil in the indirect evaporation layout model. This model is used to define and buffer the absolute pressure in the closed loop. The tank has a fixed volume and a variable filling level FL defined as.

$$FL = \frac{M_{liquid}}{V_{tank}\rho_{liquid}} \quad (3-15)$$

The mass balance in the tank is

$$\frac{dM_{liquid}}{dt} = \dot{m}_{tank,in} - \dot{m}_{tank,out} \quad (3-16)$$

Initial values of pressure and enthalpy of the tank as well as initial filling level are required to initiate the model.

3.4.4 Throttle valve

In the experimental setup for the validation of the evaporator model, a throttle valve replaces the expander. Therefore, the throttle valve is situated after the evaporator, allowing the fluid to expand to the condenser pressure. The valve is modeled as an orifice valve where the pressure drop $p_{tv,in} - p_{tv,out}$ is dependent on the mass flow $\dot{m}_{tv}$ using the Bernoulli equation:

$$\dot{m}_{tv} = A_{tv,eff} \cdot \sqrt{(p_{tv,in} - p_{tv,out}) \cdot 2\rho_{tv,in}} \quad (3-17)$$

The throttle valve effective area $A_{tv,eff}$ refers to the smallest constriction in the valve.

3.5 Thermodynamic and physical properties

The main thermodynamic properties of the fluids calculated in the model are the enthalpy and pressure. However, other thermodynamic properties are also of interest and/or required for calculations such as the heat transfer correlations. Among the relevant thermodynamic properties of the fluids are the temperature, density, entropy, specific heat capacity, thermal conductivity, and the dynamic viscosity.

In order to calculate the rest of the thermodynamic properties from the knowledge of enthalpy and pressure, equations of state or thermodynamic tables for the specific fluid are required. In this work, different media libraries are used for the calculation of the thermodynamic properties. These include the proprietary TIL media library [148] developed by TLK Thermo ©, Refprop © [149], and the open-source library CoolProp [150]. Table 3-3 summarizes the types of fluid considered throughout the thesis and the corresponding media library from which the thermodynamic properties have been calculated in the models.

Table 3-3 Thermodynamic properties libraries used for each fluid.

Fluid name	Thermodynamic properties library
R245fa	TIL media
Novec649	Coolprop
R1233zd(E)	TIL media
Hexamethyldisiloxane	Coolprop
Therminol66	TIL media
Exhaust Gas, $\lambda=1$	TIL media
Air	TIL media

Furthermore, besides the fluid's properties, the thermodynamic properties of the solid metal wall such as heat resistance and heat capacity are taken from the TIL library based on the material selected for the metal wall in the model.

3.6 Issues with discretized two-phase flow models

The modelling strategy described before has been identified as the most accurate and simple enough way to simulate the systems described in this thesis. However, as in any mathematical and numerical model, it is not exempt of some problems. In particular, the finite volume method utilized in the modelling of the heat exchangers is prone to some numerical stability problems, especially when two-phase flow is present as it is the case in the ORC evaporators.

One of the main issues with this type of discretization is the problem of chattering. This has been well described in the literature [151]. Chattering consists of the instabilities due to the discontinuity of fluid thermodynamic properties during phase change, for instance, the density. These problems can cause artificial phenomena such as flow reversal that are caused by the discretization and not by the physical laws. The issues such as chattering affect the robustness of simulation of the finite volume models and can lead to extremely slow simulations or to simulation failures. For more details and a formal mathematical explanation, the reader can refer to the following sources [151], [152].

Some solutions have been proposed in the literature to reduce the problems due to chattering [153]–[155]. Some of them, such as the smoothening of the density at phase boundaries, and smooth transitions in discontinuous functions have been implemented in the models of this thesis.

3.7 Test-rig for model validation

The baseline ORC evaporator dynamic model is validated by means of an in-house test rig built for the purpose. The test bench aims to replicate an ORC evaporator recovering heat from a gaseous waste heat source with fluctuating mass flow or temperature. Figure 3-6 shows a diagram of the test rig. The test rig has been designed and specified by the PhD candidate. The rig has been built and assembled in collaboration with an external contractor under the supervision and assistance of the PhD candidate.

Figure 3-6 ORC test rig for model validation.

The test rig consists of two closed loops as well as external cooling water supply provided by the laboratory. The heat source of the ORC consists of a closed loop of air propelled by a blower, and heated by means of a 20 kW electric heater. The blower and heater are able to supply air at atmospheric pressures, mass flow rates of 0.01 to 0.025 kg/s and temperatures at the inlet of the evaporator of 150 to 400 °C. This is the range of temperatures found in the exhaust of heavy duty diesel engines among other waste heat sources. The mass flow is downsized in order to keep the system compact. The electric heater has a programmable closed loop controller for the supply temperature of the air and is able to be programmed to dynamically ramp up and down the heat supplied. The ramping down of the air temperature is limited by the thermal inertia of the air since it flows in a closed loop. For this reason, a cooling water coil is installed in the air flow after the evaporator in order to allow a faster temperature ramping down. The loop also contains a servo-electric control valve to control the air mass flow that is able to ramp up and down the mass flow. By this means fluctuations of the thermal power of the heat source can be determined by the user.

The main loop of the test rig is the ORC loop. The loop consists of a pump that pressurizes the working fluid to the evaporator pressure. The 0.25 kW pump is capable of providing 20 bar and 1.5 liter per minute flow. A 3 kW electric heater is installed afterwards in order to control the inlet temperature of the working fluid if needed. Afterwards the working fluid enters the main component that is tested, the evaporator. The evaporator is a heat exchanger with simplified geometry where the heat is transferred from the hot air loop to the working fluid in counter-flow configuration. After the evaporator a throttle valve is installed to depressurize the fluid to the condenser pressure. This replaces the expansion of the working fluid in the case of a real ORC, where it would be performed in the expander. After the throttle valve the working fluid enters the condenser, a heat exchanger where cooling water supplied by the laboratory condenses the fluid back to liquid form. The working fluid flows then back to the storage tank and subsequently to the pump, completing the loop. The system is designed for evaporator pressures of 4 to 20 bar and working fluid mass flow rates of 0.01 to 0.03 kg/s.

The system accommodates for the use of different working fluids. One is R245fa, a hydrofluorocarbon (HFC) that is a widely used and benchmark ORC fluid in the low to medium temperature range. Another one is its future replacement R1233zd(E), a hydrofluoroolefin (HFO) which has a lower global warming potential. The third one is Novec649, which is a relatively modern fluid with interesting characteristics for heat recovery at the medium temperature range and environmentally friendly. All these fluids are non-flammable and non-toxic.

Figure 3-7 ORC test rig in laboratory.

Figure 3-7 shows some photographs of the actual test-rig in the laboratory, labelling the visible components accordingly.

As for the instrumentation, thermocouples, pressure transducers and flowmeters are installed to measure, monitor and record temperatures, pressures and flows in both fluids of the ORC evaporator.

For both the hot air and ORC working fluid loops volumetric flowmeters are installed. For the hot air loop the flowmeter is positioned after the blower and before the electrical heater. In the case of the ORC loop, the flowmeter is mounted immediately after the pump and before the pre-heater and evaporator. Both flowmeters are capable of producing a 4-20 mA analog output signal proportional to the flow that is sent to the Data Acquisition system.

Two pressure transducers are installed in the ORC fluid loop. One at the entrance of the evaporator and another directly at the outlet. They monitor the pressure level at the evaporator as well as the pressure loss in the heat exchanger. The sensors also produce a 4-20 mA analog output current signal proportional to the gauge pressure in the flow that is sent to the Data Acquisition system. A third pressure sensor, an analog pressure gauge with needle display is installed in the condenser to monitor the pressure after the throttle valve.

In total, seven k-type thermocouples are installed in the system. Two of them installed in the hot air loop, one at the inlet of the evaporator and the other one at the outlet. The other five thermocouples are installed in the evaporator in the ORC working fluid side. One thermocouple at the inlet, one at the outlet, and three more positioned at different points inside the evaporator. Their exact locations can be found in Figure 3-8. Their function is to record the temperatures of the fluids at all times, during the testing. The thermocouples produce a voltage related to the temperature that is sent to the Data Acquisition system.

Table 3-4 shows the measurement ranges and accuracies of all the instrumentation in the test-rig.

Table 3-4 Measurement ranges and accuracy of sensors in test-rig.

Description	Measuring range	Accuracy
Thermocouples	(-200)-1250 °C	greater of ± 2.2 °C or $\pm 0.75\%$
Flowmeter – air	0-140 m ³ /h	$\pm 0.75\%$
Flowmeter – working fluid	0.025-0.5 m ³ /h	$\pm 0.50\%$
Pressure transducers – evaporator	0-25 bar	$\pm 0.25\%$
Pressure gauge – condenser	0-16 bar	$\pm 1.00\%$

The Data Acquisition system consists of two modules of the company National InstrumentsTM. One receives analog current signals (pressures and volumetric flows) and the other receives thermocouple voltages. The Data Acquisition system interfaces with the software LabVIEWTM in order to monitor the measurements at real time during testing and log the data.

The evaporator geometry is designed as a simple counter flow double pipe heat exchanger. The working fluid flows in the inner tube while the air flows in the annular side. Because the heat transfer coefficient in the gas side is considerably lower in comparison, extended fin surfaces are added on the exterior of the working fluid tube in contact with the gas flow. This basic geometry serves as a baseline to study the dynamic heat transfer characteristics of a direct evaporator in its more primary form, isolating the main factors of the phase change and heat transfer phenomena of a fluid in a tubular straight tube. Figure 3-8 shows the flow arrangement in the evaporator as well as the location of the thermocouples installed to monitor the temperatures at different points.

For ease of construction, a standard aluminum profile is used as the extended fin geometry outside the stainless steel inner tube. The profile is the standard 45x45, slot 10 profile by GRIEGER Automation [156]. Figure 3-9 shows a cross section of the evaporator, showing the fin profile shape. The relevant geometric dimensions of the evaporator can be found in

Table 3-5. Note that the fin augmented area ratio A_o/A_i is defined as the ratio between the outside heat transfer area to the inside heat transfer area.

Figure 3-8 Flow arrangement in ORC evaporator test rig and location of thermocouples.

Figure 3-9 Cross-section schematic of ORC evaporator in test rig.

Table 3-5 Relevant dimensions of ORC evaporator in test rig.

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

Chapter 4 *

Dynamic response of basic geometry and experimental validation

This Chapter presents the validation of the dynamic model with experimental results measured on the laboratory test rig and introduces the concepts and the first steps of a methodology for the analysis of evaporator response times. The validation and preliminary analysis is performed for a basic geometry of ORC evaporators, highlighting qualitatively, quantitatively and in the most general way, the contributions of the main factors that affect the dynamic response times. The response time maps are first introduced in this Chapter as a tool to customize the thermal inertia of ORC evaporators.

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4.1 Introduction

The objective of this chapter is to get the first insight into the dynamic response of ORC evaporators. For this purpose, a basic geometry that highlights the fundamental phenomena of most types of heat exchanger is used. This insight forms the basis to introduce a methodology to find and analyze the response time of ORC evaporators and its dependence on component and system design factors.

Because of the challenges ORCs face during transient conditions, it is important to understand and characterize the dynamic behavior of the system in a systematic and simple way that gives insight into the main phenomena and factors contributing to the dynamics of the evaporator.

As it has been mentioned previously, when working under a fluctuating heat source, ORCs will respond to the heat changes in a certain amount of time according to their thermal inertia. As transients are significantly slower in heat exchanger than the other ORC components [75], [99], the system thermal inertia can be well represented by the evaporator thermal inertia.

Before presenting an analysis of response time on more realistic types of heat exchangers with all their complex geometries, it is important to show the fundamentals of the proposed response time methodology with a simple, fundamental case as an illustration.

In this Chapter, the dynamic response of ORC evaporator is analyzed for a fundamental and simplified case. The basic geometry in focus is a double pipe heat exchanger. The simple configuration is analyzed in terms of the theoretical governing equations and a numerical discretized model. Furthermore, some of the results from the simulations are validated with data obtained in-house from a test rig, in order to gain confidence of the model and heat transfer correlations accuracy.

These results are the validated fundamental foundation for the next chapters, in which the

methodology for the analysis is expanded and used with more complex geometries and layout configurations for the ORC evaporator that are used in real world applications.

4.2 A basic geometry of ORC evaporators

A good analysis practice is to start from the general to the particular. A starting point for a general dynamic response analysis of ORC evaporators is, therefore, a basic type of heat exchanger. A typical basic geometry that is used to investigate the fundamental behavior of heat exchangers is the double pipe type. This geometry is simple enough to isolate the chief fundamental behavior yet still contains relevant parameters present in many heat exchangers. This is because most of the heat exchangers utilize tubular or circular conduits configurations for the working fluid side. Other geometries, such as plate heat exchangers, contain ducts of different shapes that can still be handled in a similar way with the use of the equivalent hydraulic diameter.

Therefore, the basic geometry of a double pipe heat exchanger can give already a good insight into the effect of the main variables (i.e. geometry, the materials and the fluid properties) on the dynamic behavior of ORC evaporators.

The basic geometry has the added value that is easy to build and therefore the model of the basic geometry can be verified against measured data provided by experiments performed in a test-rig. In this way, the basic general model can be validated to a certain degree and give more certainty to the results when the model is applied to heat exchangers with more complex geometries.

A schematic of the basic geometry ORC evaporator is shown in Figure 4-1. From the point of view of the working fluid, the geometric parameters can be reduced to two: the tube diameter D (or hydraulic diameter) and the tube (or conduit) length L .

Figure 4-1. Basic geometry for ORC evaporators.

The particular dimensions of the tube and diameter define the internal heat transfer area and total working fluid volume in the evaporator. The heat transfer area is normally defined by the thermodynamic requirements at system-level (e.g. a given degree of super-heating at the outlet for a certain mass flow and inlet conditions). However, different combinations of diameter and tube can still produce the same heat transfer area (or heat exchanger thermal capacity) while providing different working fluid volumes. This is relevant, as the working fluid volume in the evaporator ultimately is an important factor affecting the speed of dynamic response of the evaporator, as it will be presented later.

The study of the case of the basic geometry heat exchanger is not only simpler but also significant. This is because, although heat exchangers can have complex geometries with various parameters required to fully define the geometrical dimensions, for most heat exchangers these two dimensions (conduit diameter and path length in a general terminology) are the most important that define the physical phenomena.

4.3 Methods to evaluate the dynamic behavior of the ORC evaporator

Once the basic general type of ORC evaporator has been presented, the next step is to specify the method with which to evaluate the dynamic response of ORC evaporators in general. An obvious way is to look at the governing equations for answers about the characteristic times. However, as it is the case, it proves difficult due to the complex phenomena present and it is ultimately only practical to use numerical simulations for this purpose.

4.3.1 Characteristic time scales from model equations

The response time of the working fluid in the evaporator can be theoretically predicted based on an analysis of the governing differential equations that define the transient behavior of the ORC evaporator. When changes of the heat carrier happen, there is a particular response time on the working fluid (cold) side due to a certain thermal inertia. This thermal inertia can be separated into two main components: that due to the inertia from the metal wall between both fluids, and that corresponding to the working fluid itself. The general differential equations governing the phenomena in the ORC evaporator include the constitutive (heat transfer) and conservation (fluid mechanics) equations. These equations are a system of second order differential equations. In order to simplify the analysis, the continuous volume of the ORC evaporator can be divided into a number of discrete cells inside which the fluid and material properties are assumed constant and first-order differential equations can be used. This is the strategy of the finite volume model described in Section 3.3 which is the model used throughout this thesis.

Based on the finite volume model, the metal wall characteristic time can be found from the energy balance equation in the wall. For this 1-D discretized model that assumes negligible longitudinal heat conduction the energy balance (equation (3-4) in Section 3.3) for one discretization cell is:

$$c_w \rho_w V_w \frac{dT_w}{dt} = \frac{(T_{w,ext} - T_{w,int})}{R_w} \quad (4-1)$$

This is an example of a simple first-order linear differential equation, and thus, the characteristic time τ_w of the metal wall during heat transfer transients can be easily identified as:

$$\tau_w = \frac{\rho_w \cdot c_w \cdot V_w}{R_w} \quad (4-2)$$

The thermal resistance and volume of the wall, depend on the geometry of the wall. In case of a simple rectangular plate, the characteristic time of the metal wall is

$$\tau_w = \frac{\rho_w \cdot c_w \cdot t h_w^2}{k_w} = \frac{t h_w^2}{\alpha_w} \quad (4-3)$$

Where the thermal diffusivity α_w of the material, which includes all the relevant properties of the wall material for the transient analysis, is defined as:

$$\alpha_w = \frac{k_w}{\rho_w \cdot c_w} \quad (4-4)$$

Metal walls in heat exchangers are, however, more often, cylindrical, or with more complex geometries such as finned surfaces. In this case the characteristic time from equation (4-3) will only give a first-order approximation.

In case of the working fluid response time, the energy balance equation on the working fluid is more complex, being a non-linear differential equation. For the case of the 1-D discretized model, the energy balance in each cell (equation (3-2) in Section 3.3) is:

$$\begin{aligned} V \cdot \rho(T_{wf}, p) \cdot \frac{dh(T_{wf}, p)}{dt} - V \cdot \frac{dp}{dt} \\ = \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} + \theta_{wf} \cdot A_{int} \cdot (T_{w,int} - T_{wf}) \end{aligned} \quad (4-5)$$

As it can be seen, this equation is more complicated since it a non-linear differential equation. Since there is phase change happening as well, the equations of state relating the different thermodynamic properties are discontinuous which leads to further complications. Furthermore, the heat transfer correlation term is calculated based on semi-empirical or fully empirical heat transfer correlations. This shows that in order to find a practical solution for the calculation of the response times of the evaporator, it is more convenient to use numerical approximations based on simulations of discretized models,

4.3.2 Dynamic response from numerical simulations

Although a theoretical analysis of the governing equations of the phenomena present in the ORC evaporator is an obvious starting point, it is not practical or even possible to obtain exact results from an analytical solution since the mathematical model of the physics in the ORC evaporator consists of non-linear equations. This is because the phenomena is a complex, multi-variable, non-linear phenomena, and furthermore some semi-empirical equations are required for the heat transfer correlations. A more pragmatic approach is required for the investigation of the transient response. This involves the solving of the equations numerically in a computational environment.

A method to identify the dynamics of any generic system is the one used in control theory called system identification [157]. In that approach, the transient behavior of a dynamic system is characterized by the output of a system to a certain input signal. This input-output behavior is observed by means of an off-line or on-line test. In the on-line test, a change in a certain variable (input) is applied to the physical system and the behavior over time of one or a system of measured variables (output) is observed and identified as the dynamic response of the system. In the off-line test, an accurate model of the system is used to simulate the input-output behavior of the real system instead.

Based on standard practice, the most common types of input signals for the identification of dynamic systems include the step input, impulse input, ramp input and sinusoidal input [158]. The step input signal is typically the basic type of signal used to characterize the time domain response of the system. Another important signal is the sinusoidal input, and the response to this type of signal is called the frequency response of the system.

An on-line system identification on a real physical ORC evaporator gives tentatively more certainty over that one particular ORC evaporator due to the incapacity of any model to completely capture with absolute precision all the parameters of the system. However, it is purely based on empirical test and gives little insight into the physical phenomena and main

influencing factors affecting the dynamic response. Furthermore, if general conclusions are to be drawn for different sizes, materials, boundary conditions or other parameters of ORC evaporators, it might be impractical to build several physical systems to obtain general trends of the dynamic response behavior.

Therefore, an ORC evaporator model based on the governing equations can provide insight into the physical phenomena happening and how it affects the dynamic response. Furthermore, it has the flexibility of being able to change at will certain parameters, while keeping others fixed, and investigate the sensitivity that the dynamic response has on that parameter. The model can be solved numerically with the help of a computer environment.

A drawback of using only simulation models is that there is an uncertainty on its accuracy. Therefore, to increase the confidence on the results based on the model, a certain degree of verification based on experimental results is required. This validation is presented in the next section for the model used in this work.

4.4 Experimental validation of basic geometry model

The test rig that has been described in Section 3.7 is used to validate the model of the basic ORC geometry. It consists of a closed loop of an organic fluid, with an emphasis on the evaporator, where the organic working fluid enters at sub-cooled liquid state and gets vaporized and slightly super-heated by heat transferred from an external hot air flow. The organic working fluid utilized in the experiments is R245fa which is a well-known fluid for ORC applications in the lower and medium temperature range. Furthermore, R245fa is the main organic fluid considered in the case studies presented in this thesis.

An experimental campaign has been performed by applying different varying temperature profiles of the air at the inlet of the evaporator. According to what has been described in Section 4.3.2, step changes and sinusoidal profiles are very useful to extract the behavior of dynamic systems. Sinusoidal profiles of the inlet air temperature were achieved by continuous ramp-up and ramp-down of the air heater. However, due to hardware limitations

(such as the hot air flowing in a closed loop), infinitely small changes of air inlet temperatures are not physically possible and thus, air inlet temperature ramps were used to approximate the step response behavior. The blower providing the air flow rate was kept at constant speed during all experiments, however some variations of the flow rate were inevitable due to the changing properties of the air with temperature such as the density.

Figure 4-2 shows an example of the measured hot air temperature and mass flow at the inlet of the evaporator for the two types of experiment: the sinusoidal profile and the trapezoidal (ramps) profile. The trapezoidal profile approximates the step response by providing a fast ramp-up followed by a plateau and a fast ramp-down followed by an extended valley.

Figure 4-2 Examples of different air flow and temperature inputs for experimental campaign (a) Sinusoidal temperature profile (b) Trapezoidal temperature profile.

The sinusoidal input profiles provide a temperature fluctuation with an amplitude of approximately 40 °C and three different frequencies. From the start to approximately 5000 s the fluctuation frequency is 1.67 mHz (a cycle period of 10 minutes), from ~5000s to ~8500 s, the fluctuation frequency is 1.04 mHz (a cycle period of 16 minutes) and from ~8500s to the end the frequency is 2.78 mHz (a cycle period of 6 minutes)

The trapezoidal profiles consists of periodic temperature ramps (up and down) with periods

of constant temperature in between. The ramp-up involves the increase of the air temperature by 20 °C in 2 minutes time. Due to cooling of air limitations in the test rig, the ramp-down to decrease the air temperature by 20 °C could only be performed in 4 minutes.

One constraint imposed on the profiles of the temperature and mass flow of the air is that the thermal power available is able to ensure a slightly super-heated outlet state of the working fluid (between 1 to 10 °C super-heating).

Both sinusoidal and trapezoidal profiles of the hot air tests were performed in the lab for several pressures (and saturation temperatures) of the working fluid. The evaporator pressure is manipulated by manually opening or closing the throttle valve at the outlet of the evaporator. Tests at approximate evaporator pressures of 8, 10, 12 and 15 bars have been performed. In this way the model can be validated for different sets of conditions. The average inlet temperature of the working fluid for all test is 55 °C with minor fluctuations. The volumetric flow provided by the pump for all tests is around 0.8 to 1 L/min.

The measurements from all the experiments on the test rig are used to validate the dynamic model of the basic geometry. The model is described in detail in Section 3.3 with the discretized cells interconnected in the way shown in Figure 3-4. As mentioned previously, the model uses the modelling language Modelica and the simulation interface software Dymola. The exterior heat transfer area is calculated with the inclusion of the fin augmented area ratio A_o/A_i . The model considers the thermal inertia of both the metal wall and the fins. It has to be noted that the thermal inertia of the fins in the test-rig evaporator is considerable and larger than the metal wall due to the large volume of the fin profile (see Figure 3-9).

Figure 4-3 shows a schematic of the graphical interface of the evaporator model for validation in Dymola. The inputs to the model consists of the measured inlet conditions (flow rates and temperatures) of both the air and the organic working fluid. The detailed geometric dimensions of the heat exchanger are also provided and are equal to the ones presented in Table 3-5. A simple model of a throttle valve is described in Section 3.4.4 is

included after the evaporator and the measured pressure after the throttle valve is provided. The only parameter that is estimated is the throttle valve effective area which is adjusted in order to provide the measured evaporator pressure values. The model considers the fluid properties of both the organic working fluid R245fa and the external fluid air according to the TIL Media library properties.

Figure 4-3 Graphical interface of evaporator model validation in Dymola.

Figure 4-4 shows some of the results of the Dymola model validation for one of the sinusoidal profile experiments, the one performed with the profile presented in Figure 4-2a. The evaporator pressure for this experiment was set around 12 bar. The results show the main variables of interest from the parameters that were measured in the test and are not inputs to the model. These include the temperatures of the working fluid and air at the outlet of the evaporator as well as the working fluid pressure and an intermediate temperature of the working fluid inside the evaporator (for the location of the thermocouples refer to Figure 3-8).

As it can be seen there is a general agreement of the measured variables to the ones provided by the simulation. There are only very small quantitative mismatches, particularly

for the working fluid temperatures and pressure. Importantly, qualitatively the general shape and amplitude of the model results are nearly identical compared to the experiments, which shows that the model predicts accurately the transient behavior of the heat exchanger. The air outlet temperature is the measurement that shows a higher quantitative discrepancy for this test. This can be attributed to the fact that the thermocouple at the air side outlet is located after a bent in the piping causing a slight variation in the dynamic pressure of the flow which is not accounted for in the model. Nevertheless, it has to be noted that the error is of around 10 °C, which in the absolute temperature scale is less than 3%. Qualitatively, the general shape and amplitude of the fluctuations shows a very accurate match with the experimental one.

Figure 4-4 Comparison of variables measured in the experiments to simulation results for the sinusoidal input profile of Figure 4-2a with an evaporator pressure around 12 bar.

Table 4-1 summarizes the statistical errors of the simulation model values to the measured values in all the experiments with input sinusoidal profiles. The errors are defined as the deviation for each point in time k between the measured value of the variable $\tilde{\xi}_k$ and the value of the variable from the simulation ξ_k . The four statistical errors consist of the maximal absolute deviation $\Delta\xi_{max}$, defined as:

$$\Delta\xi_{max} = \max \left[\sqrt{(\tilde{\xi}_k - \xi_k)^2} \right] \quad (4-6)$$

the maximal relative deviation $\Delta\xi_{max,rel}$ defined as:

$$\Delta\xi_{max,rel} = \max \left[\frac{\sqrt{(\tilde{\xi}_k - \xi_k)^2}}{\tilde{\xi}_k} \right] \quad (4-7)$$

the average absolute deviation $\Delta\bar{\xi}$ for an n number of time points defined as:

$$\Delta\bar{\xi} = \frac{\sum_k^n \left[\sqrt{(\tilde{\xi}_k - \xi_k)^2} \right]}{n} \quad (4-8)$$

and the average relative deviation $\Delta\bar{\xi}_{rel}$ defined as:

$$\Delta\bar{\xi}_{rel} = \frac{\sum_k^n \left[\frac{\sqrt{(\tilde{\xi}_k - \xi_k)^2}}{\tilde{\xi}_k} \right]}{n} \quad (4-9)$$

The errors are included for all the measured variables that are not inputs to the model. These are the outlet temperature of the working fluid in the evaporator $T_{out,wf}$, the outlet

pressure of the working fluid in the evaporator $p_{out,wf}$, the outlet temperature of the air in the evaporator $T_{out,gas}$, and the intermediate temperatures of the working fluid inside the evaporator $T_{2,wf}$, $T_{3,wf}$, $T_{4,wf}$ (for the location of the thermocouples refer to Figure 3-8).

Table 4-1 Statistical errors between simulation and experimental results for inputs of sinusoidal profiles.

Experiments identifier	Variable	$\Delta\xi_{max}$	$\Delta\xi_{max,rel}$	$\Delta\bar{\xi}$	$\Delta\bar{\xi}_{rel}$
Experiments A Evaporator Pressure ~8 bar	$T_{out,wf}$	20.7441	5.76%	7.811	2.14%
	$p_{out,wf}$	0.7803	9.43%	0.142	1.61%
	$T_{out,gas}$	9.3903	2.45%	5.683	1.48%
	$T_{2,wf}$	19.9936	5.65%	11.324	3.17%
	$T_{3,wf}$	22.6831	6.39%	13.215	3.70%
	$T_{4,wf}$	26.5420	7.51%	16.180	4.54%
Experiments B Evaporator Pressure ~10 bar	$T_{out,wf}$	15.2272	3.95%	2.288	0.61%
	$p_{out,wf}$	1.0766	9.97%	0.154	1.45%
	$T_{out,gas}$	14.8220	3.85%	10.391	2.69%
	$T_{2,wf}$	13.1245	3.63%	5.235	1.44%
	$T_{3,wf}$	16.5963	4.61%	6.628	1.82%
	$T_{4,wf}$	18.1709	5.02%	7.842	2.16%
Experiments C Evaporator Pressure ~12 bar	$T_{out,wf}$	10.3944	2.66%	2.114	0.56%
	$p_{out,wf}$	2.4172	15.88%	0.300	2.24%
	$T_{out,gas}$	12.6498	3.25%	9.678	2.49%
	$T_{2,wf}$	52.2705	14.92%	8.586	2.34%
	$T_{3,wf}$	54.7556	15.56%	9.615	2.61%
	$T_{4,wf}$	5.6193	1.50%	3.004	0.80%
Experiments D Evaporator Pressure ~15 bar	$T_{out,wf}$	11.0151	2.81%	2.257	0.57%
	$p_{out,wf}$	3.5812	22.01%	0.532	3.47%
	$T_{out,gas}$	22.2110	5.78%	7.755	1.99%
	$T_{2,wf}$	21.5322	5.72%	11.819	3.13%
	$T_{3,wf}$	23.1156	6.14%	13.626	3.60%
	$T_{4,wf}$	22.3915	5.91%	12.541	3.30%

It can be observed from Table 4-1 that the average relative errors of the model for all tests performed under sinusoidal profiles is less than 5% and most are below 3.5%. The average relative error of the outlet temperature of the working fluid is less than 1% except for the tests at 8 bar where it is less than 2.5%. This small error gives confidence on the analysis of the following chapters where the transient thermal condition on the working fluid side is prominent.

Please note that the magnitude maximum absolute deviations are at times much larger than the average values. This is because the maximum absolute deviation represents the single measurement that deviates the most from the experimental data, a single measurement that at times is an outlier that differs significantly from most of the measurements. This can be due to faulty data, local momentary overheating/ pressure drop, instrument accuracy error, among other factors. Outliers are expected in any experimental with a large sample, but nevertheless must be reported.

Figure 4-5 shows the validation of the Dymola model for one of the experiments with the air inlet trapezoidal temperature, in this case for the one performed with the profile presented in Figure 4-2b. The evaporator pressure for this experiment was set around 15 bar. In order to observe, in detail, the different fast ramp changes (approximating step changes), Figure 4-6 shows a close-up of the measured and simulated working fluid outlet temperature after the four ramp-ups of Figure 4-2b have been applied.

Figure 4-5 Comparison of variables measured in the experiments to simulation results for the trapezoidal input profile of Figure 4-2b with an evaporator pressure around 15 bar.

It can be seen on both Figures that again there is both a good quantitative and qualitative agreement on the results of the model compared to the measured values. The quantitative agreement can be seen in Table 4-2, that summarizes the four more important statistical errors on the model values compared to the experimental value for all the experiments with trapezoidal input profiles. There is overall an average relative error of less than 3.5% for all measured variables and less than 1% for the outlet temperature of the working fluid. In Figure 4-5d it can be observed that there is a larger systematic deviation for the intermediary temperature at the location of thermocouple number 2. In general, there are

larger errors in all the intermediary temperatures in the heat exchanger compared to the outlet temperature. These larger errors in the intermediary temperatures can be explained due to the fact that the model considers a number of discrete cells, inside which the fluid properties are considered constant. Therefore, the model considers a series of discrete values of temperature instead of the real continuous variation. If the location of the thermocouple is, for instance, near to a border between two cells, a small error of the measured temperature compared to the model value is expected. Nevertheless, quantitatively, all the average relative errors of the intermediary temperatures are less than 5% which can be considered acceptable.

Figure 4-6 Four different expanded details of Figure 4-5a highlighting the comparison of measured to simulation values for the four different temperature ramp-ups in the experiment.

Table 4-2 Statistical errors between simulation and experimental results for inputs of trapezoidal profiles.

Experiments identifier	Variable	$\Delta\xi_{max}$	$\Delta\xi_{max,rel}$	$\Delta\bar{\xi}$	$\Delta\bar{\xi}_{rel}$
Experiments A Evaporator Pressure ~8 bar	$T_{out,wf}$	16.45	4.59%	2.82	0.77%
	$p_{out,wf}$	1.36	16.46%	0.17	1.99%
	$T_{out,gas}$	21.91	5.72%	15.16	3.95%
	$T_{2,wf}$	15.39	4.35%	4.65	1.30%
	$T_{3,wf}$	17.77	5.01%	5.60	1.56%
	$T_{4,wf}$	21.00	5.94%	8.10	2.27%
Experiments B Evaporator Pressure ~10 bar	$T_{out,wf}$	7.06	1.93%	1.55	0.41%
	$p_{out,wf}$	0.94	9.55%	0.16	1.52%
	$T_{out,gas}$	10.66	2.77%	6.78	1.76%
	$T_{2,wf}$	11.84	3.27%	7.37	2.03%
	$T_{3,wf}$	13.56	3.74%	9.21	2.53%
	$T_{4,wf}$	15.02	4.14%	10.87	2.99%
Experiments C Evaporator Pressure ~12 bar	$T_{out,wf}$	15.20	3.99%	2.31	0.61%
	$p_{out,wf}$	1.64	12.95%	0.28	2.18%
	$T_{out,gas}$	16.86	4.38%	5.71	1.47%
	$T_{2,wf}$	21.92	5.94%	7.90	2.14%
	$T_{3,wf}$	25.35	6.89%	10.81	2.93%
	$T_{4,wf}$	23.83	6.42%	9.08	2.44%
Experiments D Evaporator Pressure ~15 bar	$T_{out,wf}$	21.81	5.62%	2.41	0.62%
	$p_{out,wf}$	3.38	22.93%	0.38	2.43%
	$T_{out,gas}$	32.11	8.36%	7.50	1.92%
	$T_{2,wf}$	29.02	7.74%	11.66	3.08%
	$T_{3,wf}$	31.37	8.34%	13.03	3.43%
	$T_{4,wf}$	31.53	8.34%	12.65	3.31%

Qualitatively, the match is also very satisfactory and gives a good confidence on the capacity of the model to accurately predict transients. As it can be seen in all the comparisons of Figure 4-6 the transient curve shape is almost identical for both the model

result and the experimental measurements with slight differences in the magnitude of the values.

It has to be noted the large thermal inertia of the test-rig evaporator. There is a relatively large duration of the transient after the 2-minute ramp change in air temperature. In fact, it takes around 600 s (6 minutes) for the temperature to reach within 95% of the final outlet temperature (~ 123 °C) from an initial temperature of around 118-120 °C. One important factor for the large thermal inertia is the large volume of the fins, as it has been observed that the response time of the model is significantly faster if the thermal inertia of the fins is not considered. However, the large volume of the fins is not standard practice in heat exchangers, and it is only the case for the test rig due to construction ease. Therefore, in other geometries of real-case evaporators the thermal inertia is expected to be noticeably lower.

One important consideration to take into account is that there is a limit in the accuracy of all the test rig measurement devices (see Table 3-4). For instance, in general the average errors of the outlet temperature of the working fluid in the validation are, in fact, within the 2.2 °C accuracy of the thermocouple.

Furthermore, any inaccuracy on any measurement of pressure, temperature or flow not only affects directly the error of the aforementioned variable, but also affects indirectly all the other variables, since the system is a highly-coupled multi-variable system. For instance, an inaccuracy on the organic fluid value affects the evaporator pressure in the model, and in turn the evaporator pressure on the model affects the saturation temperature and ultimately the intermediate and outlet temperatures of the evaporator.

Because of all these reasons, the fact that the great majority of all the relative average errors on the validation are below 3.5% is consider considerably good, and it is substantially within the values that have been accepted as satisfactory in the validation of dynamic models in the literature [64], [159].

All the models throughout this thesis share the same basic structures as this validated model, the only difference being the way the cells are interconnected and the calculation of the geometric dimensions of interest. This gives a high level of confidence on the results based on simulations for all types of heat exchangers in throughout this thesis.

4.5 A systematic characterization of response times of ORC evaporators

In the previous sections of this chapter, the basic geometry for ORC evaporators has been described as well as the possible methods to characterize the dynamic behavior of the evaporators. It has been concluded that numerical simulations based on discretized mathematical models is the most convenient way to study the dynamics of ORC evaporators. The discretized dynamic model of the evaporator has been validated against experimental data showing good agreement at steady-state and during transients both quantitatively and qualitatively.

Once the confidence on the dynamic model has been established, it can be used to provide a more general analysis not bounded by the limitations of the evaporator on the test rig which can only provide information on the dynamic behavior of only one group of particular dimensions of a heat exchanger.

In this section, a systematic methodology to characterize and present the results of the dynamic response of ORC evaporators in general is introduced. The main purpose of this methodology is to be able to characterize the dynamics of ORC evaporators and to show in the most general yet compact of ways the impact that the main variables have on the dynamic response.

4.5.1 Characterization method

Based on the previously recommended approaches to characterize the response time of the ORC evaporator, the method presented in Figure 4-7 is employed. Using the dynamic model in the Modelica language described in Section 3.3, with the cell's interconnection

as in Figure 3-2 the dynamic response of the ORC evaporator to a step change of the heat is investigated by simulating the dynamic model in the software Dymola. In order to reduce the number of variables to a minimum, isolate the factors concerning only the working fluid and metal wall and give more insight to the parameters that matter most, for this fundamental case the heat carrier is replaced by a homogeneous heat flux in the exterior of the tube wall.

The simulation of the model starts at a certain steady-state condition with a prescribed degree of super-heating at the outlet of the working fluid side. Then, a step change of the homogeneous heat flux is applied in the outer part of the tube wall while keeping all the other boundary conditions fixed. The free response of the evaporator is then observed and recorded in the simulation environment.

Since the outlet enthalpy is the property more representative of the thermal response of the evaporator, it is chosen as the parameter that represents the dynamic response of the system. The outlet enthalpy will gradually increase from the initial steady state and slowly converge into a new steady state after the transient is over. This is represented in the right-hand graph of Figure 4-7. The time when the outlet enthalpy reaches 95% of the new steady state is defined as the response time τ_{ev} . This response time, in control terminology, is equivalent to a rise time from 0 to 95% [158].

Figure 4-7 Dynamic response characterization schematic for simplified geometry.

For a formal definition of the evaporator response time τ_{ev} , a dimensionless outlet enthalpy can be defined as:

$$\tilde{h} = \frac{h_{out}(t) - h_{out}(0)}{h_{out}(\infty) - h_{out}(0)} \quad (4-10)$$

The time zero is the time when the step change in the heat flux is applied. Using this dimensionless enthalpy, the evaporator response time is formally defined as:

$$\tau_{ev} = t(\tilde{h} = 0.95) \quad (4-11)$$

From the validation in Section 4.4, it has been proven that the model can predict with accuracy this response time for a real physical system,

4.5.2 Main factors affecting the response time

The evaporator response time τ_{ev} depends mainly on the boundary conditions as well as the thermal inertia of the metal wall and of the working fluid inside the tube. This thermal inertia depend on the intensive properties, as well as the volume of wall material and working fluid present. The intensive properties depend exclusively on the properties of the selected wall material as well as the properties of the working fluid on the system, whereas the volume depends on the geometry of the heat exchanger. The factors affecting the thermal inertia of the evaporator can be then classified in the following three areas:

- 1) Geometry of heat exchanger. This affects the volume of both the wall material and the working fluid. The geometry of the heat exchanger is chosen as to achieve a desired heat transfer at design point. However, a different set of geometric dimensions can still provide the same heat capacity UA of the heat exchanger, and thus the same heat transferred in the heat exchanger at design point. For the case of the simplified geometry considered in this chapter, and assuming the mass inertia of the fins as

negligible there are three parameters required to define the geometry. The thickness of the wall is usually defined by mechanical considerations. The other two are the tube diameter D and its length L . Both can be lumped into a geometric ratio D/L .

- 2) Wall material thermal properties. The properties of the wall material that affect the thermal inertia are the heat capacity c_w , the density ρ_w , and the thermal conductivity k_w . All these parameters can be lumped into the thermal diffusivity α_w , defined as:

$$\alpha_w = \frac{k_w}{\rho_w \cdot c_w} \quad (4-12)$$

- 3) Working fluid thermal properties. The selection of the working fluid often will be done beforehand and obeys the suitability for the temperature of the heat source as well as other considerations such as safety, environmental and other practicalities. The heat capacity of the working fluid is different depending if it is in liquid, vapor or two-phase state. One parameter that is of suitable interest that relates the heat capacities to the latent heat of vaporization is the Jakob number Ja , defined as:

$$Ja_{lv} = \frac{C_{p,v}(T_v - T_{sat}) + C_{p,l}(T_{sat} - T_l)}{\Delta H_{vap}} \quad (4-13)$$

Where $C_{p,v}$ and $C_{p,l}$ are the specific heat capacities of the fluid at vapor and liquid phases respectively, T_v and T are the initial outlet (super-heated vapor) and inlet (sub-cooled liquid) temperatures, T_{sat} is the saturation temperature and ΔH_{vap} is the enthalpy of vaporization of the fluid.

- 4) Boundary conditions. The boundary conditions are the heat flux from the heat source, the inlet thermal condition of the working fluid as well as the initial outlet condition of the working fluid. The boundary conditions depend on the characteristic of the heat source and the steady-state design of the system.

4.5.3 Response time maps

In order to illustrate the dependence of the aforementioned factors on the response time of ORC evaporators in a generalized way, a graphical method is proposed. The response time of the basic geometry ORC evaporator is found by simulating the dynamic model with varying values of the factors of interest. The results from several simulations are interpolated and charts are built indicating curves of constant response time for different varying factors in the x and y axis.

It has to be noted that certain factors and boundary conditions of an ORC evaporators are defined and fixed by considerations such as the available thermal power and temperature of the heat source and the thermodynamic cycle optimization. Therefore, some of them cannot be changed, once the boundary conditions and the thermodynamic cycle are defined. This can be called *fixed parameters*. However, there are others factors that can be considered as degrees of freedom in order to modify the response time of the evaporator. This can be called *dynamic response parameters*. For instance, the heat transfer area is *fixed parameter* defined by the available thermal power of the source and the cycle requirements. However, the geometric diameter to length ratio D/L of the heat exchanger is not, and can be varied while still keeping the same heat transfer area.

Figure 4-8 shows the response time maps for the basic geometry as a function of the *dynamic response parameters*. This include the geometric ratio D/L and the wall material thermal diffusivity α_w for two different values of the Jakob number. The maps are built for the *fixed parameters* presented in Table 4-3 with R245fa as the working fluid.

Regarding the *fixed parameters*, in order to showcase in a general way qualitative trends, in the maps of Figure 4-8 the heat transfer area is set as unitary, with a nominal heat flux (heat transferred per area) observed from external gaseous flows (i.e. the heat flux observed in the test rig), the wall thickness has a standard value of 0.02 mm. The evaporator pressure is set at 15 bar with an initial super-heating value of 1 °C.

It must be noted also that each of the two maps represent two different working fluid inlet temperatures, which define the two different values of Jakob number. Since the inlet temperature depends on the conditions of the condenser or on the design of a potential recuperator, this is also shown as a potential decision factor or *dynamic response parameter* that affects the dynamic behavior. The mass flow of the working fluid is adjusted for each inlet temperature condition in order to achieve the same initial super-heating with the same heat transfer area.

The two Jakob numbers selected for the response time maps, are exemplary of two different inlet working fluid temperature conditions in the evaporator representing two different configurations. The smaller Jakob number ($Ja_{lv} = 0.593$) represents an inlet temperature (55 °C) that is probable for an ORC that includes a recuperator, while the larger Jakob number ($Ja_{lv} = 0.923$) represents an inlet temperature (25° C) that is probable for an ORC without a recuperator.

Table 4-3 *Fixed parameters* for response time maps of Figure 4-8

Ja_{lv}	Working fluid	Heat transfer area	Wall thickness	Saturation pressure	Heat flux	Initial super-heating	Inlet temperature working fluid	Mass flow rate
0.593	R245fa	1 m ²	2 mm	15 bar	7000 W/m ²	1 °C	55 °C	0.34 kg/s
0.923	R245fa	1 m ²	2 mm	15 bar	7000 W/m ²	1 °C	25 °C	0.28 kg/s

Figure 4-8 Response time maps for basic geometry with unitary heat transfer area for two different Jakob numbers. *Fixed parameters* as in Table 4-3

From the response time maps of Figure 4-8 some general insight can be drawn on the influence of each *dynamic response parameter* and their interactions:

- Geometric ratio D/L . The response time is slower with an increasing value of D/L . This is because larger diameters with shorter lengths represent a higher volume of the working fluid while still keeping the same heat transfer area. A larger volume of the working fluid requires more time to reach a new steady state. There is an enhanced variation of the response time with D/L for lower values of the wall material α_w
- Wall material thermal diffusivity α_w . As expected a larger thermal diffusivity means shorter response times. A material with higher values of α_w allows for a faster heat transfer and stores less thermal power during transients. A low value of α_w , makes the response time more sensitive to changes in geometric dimensions.
- Fluid properties / thermal state represented by the Jakob number Ja_{lv} . The Jakob number Ja_{lv} is related in this case to the inlet temperature of the working fluid. For

the two Ja_{lv} considered the response time values are similar but differ specially on the left top and right bottom corner. For low D/L and high α_w the response time is slower for the larger Ja_{lv} and the opposite is true for high D/L and low α_w . This is confirmed by highlighting the deviation of the response time with the Jakob number at is shown in Figure 4-9. It is clearly seen in Figure 4-9b that for a large thermal diffusivity of the wall the response time is slower for the larger Ja_{lv} specially at larger values of D/L . The opposite is true in Figure 4-9a, representing a smaller thermal diffusivity of the wall. In this case the response time is slower for the smaller Jakob number. The latter case can be explained by the fact that the mass flow of working fluid in the evaporator is less (see last column of Table 4-3). This lower mass of fluid plays a major role when the volume of the heat exchanger is larger as it is the case with higher D/L . For low D/L the quantity of working fluid has less of an impact on the thermal inertia and the dominant factor is the less effective heat transfer for one-phase fluid which slows down slightly the dynamics.

Figure 4-9 Deviation of response time for the two Jakob numbers considered for two different values of the thermal diffusivity of the wall material α_w , corresponding to common construction materials (a) Steel and (b) Aluminium

It can be observed from the maps that the response time is most sensitive to the geometric dimensions than to the other *dynamic response parameters*. In other words, this is the parameter that can be changed to most effectively influence the dynamic response of the system. The material of the wall comes in second place.

The response time maps also serve as a tool to customize the dynamic response of the system. A faster response time may be required in order to achieve fast control, or rather a large response time may be preferred in order to have a more robust system to sudden changes of the heat source, and one that can dampen certain fluctuations.

The potential of the response time maps will be further highlighted in the following Chapters where they will be expanded and used in practical cases to improve the dynamic performance of ORC systems.

4.6 Summary

In this chapter a general analysis of the factors affecting the ORC evaporator response time has been carried out for the case of a basic general geometry. Although an examination of the formal governing equations is an important starting point to find the characteristic time scales of the system, the phenomena is complex, multi-variable and non-linear and ultimately simulation of models that utilize discretization methods prove more practical to obtain meaningful results.

The level of confidence on the results based on these models is higher when some validation based on experimental data is available. Because of that, this chapter has shown a validation of the dynamic model used in Dymola by comparing simulation results to measured data obtained during an extensive experimental campaign on a test rig in the laboratory. The validation has shown that the model can accurately predict both quantitatively and qualitatively the behavior of the ORC evaporator in the lab during transients.

With a high level of confidence on the dynamic model, a first glimpse into a methodology to systematically characterize the dynamic response of ORC evaporator has been presented. The concept of response time maps has been introduced, presenting its potential for customization of the evaporator dynamic response. These concepts will be expanded and applied to relevant situations in the next two chapters.

Chapter 5 *

Dynamic behavior of different types of heat exchangers for direct evaporation

This Chapter expands the concepts of response time maps introduced in the previous Chapter to more complex geometries belonging to the real types of heat exchangers that can be used for direct evaporation. The potential of the methodology to customize the thermal inertia of the ORC system is shown on the case of the profile of an IC engine exhaust during a standard driving cycle. The concept of dynamic regimes is introduced and the dynamic regime number is used as a tool to relate the response time of the evaporator with the fluctuation characteristics of the heat source in order to obtain a certain dynamic behavior.

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5.1 Introduction

There are several different designs and types of heat exchangers that could be utilized as ORC evaporators. Furthermore, more complex layouts including more than one heat exchanger are usually chosen for practical considerations. That is the case of indirect evaporation layouts, in which an intermediary heat transfer fluid is used to decouple the heat transfer from the main heat source to the ORC working fluid.

Direct evaporation is the ORC layout in which only one heat exchanger is used to transfer the heat directly from the main heat carrier of the heat source to the ORC fluid. Direct evaporation is the most simple and efficient way to transfer heat to the ORC fluid, yet in real practical applications that include high variability of the heat source is not often the preferred choice. This is because indirect evaporation can more effectively protect the fluid from extreme changes in the boundary conditions by damping the variability of the heat source. The working fluid must be above saturation temperature and below chemical decomposition temperature at the outlet of the evaporator at all times. As has been explained previously in Chapter 2, the intermediary heat transfer loop in indirect evaporation layouts can be considered a type of sensible heat thermal energy storage.

In this chapter, some relevant types of heat exchanger for direct evaporation are analyzed based on their dynamic response to changes of the heat source. This analysis is done based on the concepts introduced in Chapter 4. However, unlike Chapter 4, in this Chapter more detailed models of heat exchangers with real-life complex geometries are analyzed. The waste heat carrier properties are also taken into account.

In waste heat recovery applications, the heat carrier is most often a gaseous fluid (i.e. an engine exhaust). Due to the lower heat transfer coefficient of gases compared to the evaporative heat transfer coefficient, a good way to ensure the compactness and effectiveness of the direct evaporation heat exchanger is to have extended surfaces on the gas side. Therefore, in this Chapter, the analysis is restricted to the types of heat exchangers that are suitable for direct evaporation in most practical cases. Two different types are

considered: a multiple fin and tube heat exchanger and a louver fin multi-port heat exchanger.

5.2 System assumptions and characterization approach

As an illustrative and relevant case, the evaporators analyzed in this work are intended to be a part of an ORC system recovering waste heat from the exhaust of a diesel engine. This is a situation where direct evaporation can be an attractive option due to volume and weight restrictions in mobile applications.

The study in this Chapter is intended for a high efficiency system. A schematic of the ORC system is shown in Figure 5-1 along with a representative T-S diagram. An optional small recuperator can be installed to increase the thermal efficiency and decrease the size of the evaporator.

Figure 5-1 (a) Schematic of ORC under investigation (b) Qualitative T-S diagram of the process.

The working fluid considered is the benchmark R245fa, which is suitable for the temperatures and mass flows available in this type of applications [160]. Furthermore, R245fa is a well-known and widely used fluid in ORC systems and it's the fluid used in the research papers in which the ORC evaporator geometries of this work is based [58], [131].

The exhaust gas boundary conditions are assumed based on the range of values reported by Yang et. al [58] for a six-cylinder in line Diesel engine with rated power of 240 kW. They represent values for the exhaust on the medium to high torque. The composition of the exhaust is taken as the combustion gas of a diesel fuel with an air-fuel equivalence ratio λ of equal to 1.2. The inlet conditions of the exhaust for the base case are reported in Table 5-1.

As per the working fluid, the conditions at the inlet of the evaporator will depend on the dynamic conditions in the condenser and, if it is the case, the recuperator. The working fluid inlet temperature is treated as one of the parameters to be studied. For the base case it is assumed a fixed value of the inlet conditions in the evaporator of 10 below the saturation temperature. Furthermore, it is assumed that the working fluid is completely evaporated and super-heated to 1 °C above the saturation temperature. This is because low-super-heating allows for higher efficiencies when using organic fluids with high molecular weights such as R245fa [75]. The working fluid boundary conditions at base case are also reported in Table 5-1.

As it has been done in Chapter 4 the dynamic response of the evaporators is characterized by its transient response to a step change on the heat conditions. As defined in Chapter 4 the response time τ_{ev} of the evaporator is the time it takes to rise to 95% of the new steady state when subjected to a step change of either the mass flow or the temperature of the engine exhaust.

Figure 5-2 shows the schematics of the characterization approach. The model in Dymola starts at a steady state with given initial parameters. At a certain time, a step change on

mass flow or temperature of the exhaust is imposed. The simulation is carried out during the subsequent transient until a new steady state is reached. The response time τ_{ev} is recorded.

Table 5-1 Boundary conditions and fluid descriptions in ORC evaporator for the base case.

Description	Value
Exhaust inlet temperature	350 °C
Exhaust inlet pressure	1 bar
Exhaust mass flow	0.35 kg/s
Working fluid inlet temperature	133 °C
Working fluid inlet pressure	30 bar
Working fluid initial degree of super-heating at outlet	1 °C
Working fluid	R245fa
Exhaust properties	Combustion gas $\lambda = 1.2$

Figure 5-2 Dynamic response characterization approach.

5.3 Heat exchanger geometries

Figure 5-3 shows the geometry and main dimensional parameters of fin and tube heat exchangers. The base case dimensions for this study are based on the real heat exchanger dimensions reported by Yang et al. [58] and reported in Table 5-2. The referred heat exchanger has been dimensioned considering the base boundary conditions of Table 5-1.

Figure 5-4 shows the geometry and main dimensional parameters of the louver fin multi-port flat tubes heat exchanger. For this type of heat exchanger, the base geometric dimensions are based on the one reported by Mastrullo et al. [131]. The base case dimensions are reported in Table 5-3.

Figure 5-3 Geometry of fin and tube heat exchanger.

Table 5-2 Dimensions of fin and tube heat exchanger at base case.

Description	Value
Number of tube banks	20
Number of tubes per bank	9
Tubes inner diameter	20 mm
Tube length	800 mm
Tube thickness	2.5 mm
Clearance between tube banks	35 mm
Clearance between tubes per bank	75mm
Fin height	12 mm
Fin thickness	0.1 mm
Fin pitch	2 mm
Tube banks arrangement	Staggered
Tube and fin material	Stainless steel

Figure 5-4 Geometry of louver fin multi-port heat exchanger.

Table 5-3 Dimensions of louver fin multi-port heat exchanger at base case.

Description	Value
Number of ports per tube	20
Number of flat tubes	7
Port inner diameter	2 mm
Tube length	840 mm
Port wall thickness	2.5 mm
Clearance between ports	35 mm
Clearance between tubes	75 mm
Fin thickness	0.5 mm
Fin pitch	2 mm
Louver angle	25 °
Louver pitch	0.8 mm
Tube and fin material	Stainless steel

5.4 Parameters of interest

As explained in Chapter 4, the main aspects that affect the dynamic response of the evaporator can be grouped in the following areas: wall material, boundary conditions and geometry. Of these, the first two can be manipulated at the design stage, whereas the third aspect also depends on the operational characteristics at a given time.

In order to study the response time dependence on these different areas for each of the heat exchanger types, a parametric study is carried out where the dynamic characterization is performed for various series of combinations of the studied parameters. A description of each of these parameters is given below:

5.4.1 Wall material

The simulations are carried out for different relevant engineering materials for the heat exchanger walls. These include, stainless-steel, aluminum and copper. To draw general

conclusions, the thermal diffusivity α_w of the material is used as the parameter of interest. As it has been shown in Chapter 4 it is defined as:

$$\alpha_w = \frac{k_w}{\rho_w \cdot c_w} \quad (5-1)$$

As it has been explained in Chapter 4, this parameter includes all the relevant material properties for the transient analysis in solids.

5.4.2 Boundary conditions

The boundary conditions are varied reflecting the changing conditions of the heat source and working fluid according to a range of values expected during operation. Three different varying conditions are varied: exhaust inlet temperature, exhaust mass flow, and working fluid inlet temperature. It is to be noted that in these simulations the working fluid mass flow is automatically determined by the requirement that the outlet conditions is 1 °C of super-heating. This is equivalent to assume there is a perfect controller that changes the working fluid mass flow to reach a given super-heating set point.

5.4.3 Geometric dimensions

The geometry of the heat exchanger has been found in Chapter 4 to have a higher impact on the dynamic response as compared to the wall material. Therefore, it is the most important design aspect that should be the focus on tailoring the dynamic response of the system.

Since both types of heat exchangers have complex geometries, only the main geometric dimensions are studied. In this analysis, the tubes and ports clearances and the fins geometries are kept fixed at the base design values. The dimensions that are varied include the tube or port diameter and the parameters that affect the three main volume dimensions.

For fin and tube heat exchangers the varied dimensions are the diameter of tubes, the

number of tubes banks (number of tubes in the direction parallel to the exhaust flow), the number of tubes per bank (number of tubes in the direction perpendicular to the exhaust flow) and the length of the tubes.

For louver fin multi-port heat exchangers, the corresponding geometric dimensions are the port diameters, the number of ports, the number of flat tubes and the length of flat tubes.

However, for each type of heat exchanger, the four dimensions are not independent of each other. In order to properly and independently assess how the different aspects affect the dynamic response, the thermal capacity UA of the heat exchanger must be fixed. In this way, the same heat transferred and the same output properties the fluid at steady state are achieved for a given set of boundary conditions. Therefore, if one of the geometric parameters is varied while keeping the others fixed, one other dimension must change in order to keep the same thermal capacity of the heat exchanger and the same outlet thermal condition of the working fluid. Since the diameter is the geometrical dimension that has a larger effect on the heat transfer and the response time, it is decided to vary each of the other dimensions independently and always adjust the tube or port diameter accordingly. This is summarized in Table 5-4.

Table 5-4 Different cases of geometric dimensions varied in the simulations for each type of heat exchanger.

Heat exchanger type	Case	Parameter varied	Parameter adjusted to keep same UA
Fin and tube	1	No. of tube banks	Tube inner diameter
	2	No. of tubes per bank	Tube inner diameter
	3	Tube length	Tube inner diameter
Louver fin multi-port	1	No. of ports per tube	Port diameter
	2	No. of tubes	Port diameter
	3	Tube length	Port diameter

5.5 Response time maps

The results of the parametric simulations are compiled and interpolated in order to develop general the maps of the response time for both heat exchangers in terms of evaporator design and boundaries parameters. As it has been introduced in Chapter 4, the response time maps represent a guide showing what it takes, in terms of design of the heat exchanger, to achieve the desired thermal inertia of the evaporator.

It has been found that the response time of the evaporator to step changes in temperature is very similar to the case of step changes in exhaust mass flow rate. The results of the simulations show a variation of less than 5% on the response time in the case of 10 °C step change in comparison to the case of 10% step change of mass flow rate for all simulation points considered. Because of this, the response time maps for the case of step changes in exhaust temperature are excluded from this analysis since step changes in mass flow rate are sufficient enough to characterize the dynamic response times of the heat exchangers in a general way.

Therefore, the results shown correspond to 10% step change of mass flow rate. Response time maps have been built for a different combination of the main parameters of interest described in Section 5.4. Below a detail description and discussion of each case.

5.5.1 Geometry and wall material

Figure 5-5 presents the response time maps showing the dependence that the evaporator response time has on the different heat exchanger design parameters for each type of heat exchangers. The x-axis corresponds to a variation in the geometry while the y-axis to a variation in the wall material represented by its thermal diffusivity α_w . Two x-axes are shown, the bottom x-axis shows the geometric parameters varied (number of tubes, length, etc.) while the main x-axis shows the corresponding adjusted value of tube/port diameter (to keep same thermal capacity). In order to correlate the y-axis values to some relevant engineering materials, Table 5-5 summarizes the thermal diffusivity of some materials that

can be used to build the heat exchangers along with their main thermal properties.

The geometry and material of the heat exchanger not only affect the thermal response time of the evaporator but also the volume and mass of heat exchanger. This is especially relevant for mobile applications and others with size restrictions. The material mass also represents some of the material costs implication when changing the geometry. Figure 5-6 shows how the volume and mass of the evaporator is affected by the corresponding changes in geometry for three wall materials of interest.

From the maps it can be seen that the variation of the diameter of the working fluid conduit is the dominant parameter that modifies the response time of the evaporator, whereas the thermal properties of the wall material play a smaller role. As expected, a higher thermal diffusivity of the material reduces the response time due to a faster heat conduction through the wall. Larger diameters increase the quantity of the working fluid present in the evaporator at a given time and therefore, because a larger mass of fluid needs to be heated up to the next steady state, the thermal inertia and response time of the evaporator is increased.

In the case of fin and tube evaporators, for Figure 5-5c and e, the diameter similarly affects the response time value regardless of the other geometric parameter that is varied. However, in the case of Figure 5-5a, when the number of tube banks is varied, the response time is slower at larger diameters and faster at smaller diameters in comparison. This is because of the disparity in the trend of mass variation that the evaporator has when the number of tube banks is varied as compared to the other two cases. This can be observed in Figure 5-6a.

For louver fin multi-port evaporators, Figure 5-5b and d have similar trends of the response time variation when the diameter is varied, while Figure 5-6f, which corresponds to a variation of tube length, has a different trend. This is also because of the variation of mass when the tube length changes is different than the two other cases as observed in Figure 5-6b.

From this analysis, it is concluded that both a larger amount of fluid and a larger amount of material of the heat exchanger increases the response time. However, the dominant effect is the volume of the working fluid.

5.5.2 Geometry and exhaust boundary conditions

The exhaust boundary conditions are constantly varying according to the operational point of the engine. Apart from the thermal power fluctuations, the mean value of the mass flow or temperature of the exhaust around which the fluctuations happen can also vary. Therefore, it is important to consider also how the mean value affects the response.

Figure 5-7 shows the variation of the evaporator response time maps for different exhaust mass flow rates corresponding to the different geometric dimensions. Figure 5-8 shows the response time maps in case of different temperature of exhaust and the different geometric dimensions. The maps also include the corresponding pressure drops for the working fluid and the exhaust in the evaporator, highlighting to the heat exchanger designer the penalties and benefits in pressure drops related to a certain design with a given response time.

In the model, the mass flow of the working fluid is adjusted according to the different boundary conditions to achieve the requirement of 1 °C of initial super-heating at the outlet of the evaporator. The values of the working fluid mass flow for the different values of boundary condition are summarized in Table 5-6. The parametrization is done around the base case values of Table 5-1.

As seen in Figure 5-7 and Figure 5-8, higher flow or temperature of the heat source reduces the response time of the evaporator. This is because with higher mass flow or temperatures of the exhaust, the heat available is higher and the mass flow of the working fluid required to achieve 1 °C super-heating at the outlet is higher. The higher the mass of the working fluid at a given time results in higher thermal inertia of the system.

For fin and tube evaporators, the response time for a tube diameter of 0.02 m reduces from 88 s with an exhaust mass flow of 0.2 kg/s to 45 s with an exhaust mass flow of 0.4 kg/s. In the case of different exhaust temperature, the response time for a tube diameter of 0.02 m reduces from about 97 s at an exhaust temperature of 250 °C to 43 s at an exhaust temperature of 400 °C. For both cases, the reduction gradient in the maps is increased with larger diameter, resulting in similar percentage reduction. In general, the response time reduces approximately 30% per 0.1 kg/s increase in exhaust mass flow and reduces approximately 30% for 50 °C increase in exhaust temperature.

For louver fin multi-port evaporator with a port diameter of 2 mm, the response time decreases from 30 s at an exhaust mass flow rate of 0.2 kg/s to 20 s at 0.4 kg/s and decreases from 35 s at an exhaust temperature of 250 °C to 18 s at 400 °C. In general, the response time reduces approximately 20% per 0.1 kg/s increase in exhaust mass flow and reduces approximately 20% per 50 °C increase in exhaust temperature.

5.5.3 Geometry and working fluid inlet condition

The inlet conditions of the working fluid in the evaporator also can vary due to changes in the boundary conditions corresponding to a different operating point in the condenser on in the optional recuperator. These variations are also considered and their effect on the evaporator response time for different geometries is shown in the maps of Figure 5-9 for the case of fin and tube heat exchangers.

In Figure 5-9 it is shown that the effect on the response time is more significant and increases with a higher degree of sub-cooling of the fluid at the inlet. For instance, for a tube diameter of 0.02 m the response time increases from about 40 s at 5 °C inlet sub-cooling to 90 s at 30 °C inlet sub-cooling. This slower response is due to the fact that the heat transfer in the one-phase region is slower than in the evaporation process. The gradient of response time increase is higher at smaller degrees of sub-cooling and is similar to all values of tube diameter.

Figure 5-5 Effect of wall material thermal diffusivity α_w and heat exchanger geometry on evaporator response time for different varying conditions and a 10% step increase in exhaust mass flow rate.

Fin and tube evaporator

(a)

Louver fin multi-port evaporator

(b)

(c)

(d)

Figure 5-6 Weight and volume of evaporator for different varying geometric parameters – No. of banks/ports, No. of tubes, Tube length- as function of their corresponding tube diameters. Wall material: stainless steel (SS), aluminium (Al) and Copper (Cop). (a) Fin and tube heat exchanger weight. (b) Louver fin multi-port heat exchanger weight. (c) Fin and tube heat exchanger volume. (d) Louver fin multi-port heat exchanger volume.

Figure 5-7 Effect of exhaust mass flow and heat exchanger geometry on evaporator response time and pressure drops for different varying geometric dimensions and a 10% step increase in exhaust mass flow rate.

Figure 5-8 Effect of exhaust inlet temperature and heat exchanger geometry on evaporator response time and pressure drops for different varying geometric dimensions and a 10% step increase in exhaust mass flow rate.

Table 5-5 Relevant properties of wall materials considered, according to values of the TIL media library [148]

Material	Thermal conductivity k [W/(m·K)]	Density ρ [kg/m ³]	Specific heat capacity c [J/(kg·K)]	Thermal diffusivity α [mm ² /s]
Steel	14.6	7900	450	4.1
Aluminium	215	2700	920	86.5
Copper	398	8960	380	116.9

Table 5-6 Required working fluid mass flow rate as function of boundary conditions in order to achieve 1 °C of initial super-heating at the outlet of the evaporator in the case of the base geometry of fin and tube evaporator.

Exhaust inlet mass flow [kg/s]	Exhaust inlet temperature [°C]	Working fluid inlet degree of sub-cooling [°C]	Required working fluid mass flow [kg/s]
0.2	350	10	0.51
0.25	350	10	0.63
0.3	350	10	0.76
0.35	250	10	0.45
0.35	300	10	0.66
0.35	350	10	0.88
0.35	350	20	0.75
0.35	350	30	0.66
0.35	400	10	1.11
0.4	350	10	1.01

Figure 5-9 Effect of working fluid inlet temperature and heat exchanger geometry on evaporator response time and pressure drops for different varying geometric dimensions and a 10% step increase in exhaust mass flow rate, for the case of fin and tube evaporator.

5.5.4 Implications for fin and tube heat exchangers

An evaporator with a larger thermal inertia may be desired in order to dampen fluctuations of the thermal power. From the response time maps, it is clear that in order to increase the thermal inertia of the evaporator, the most effective way is to increase the tube diameter.

For the fin and tube heat exchangers in this study, in order to keep the same thermal capacity UA of the heat exchangers an increase in tube diameter requires a reduction of number of tubes or of tube length. Large tube diameter reduce the Reynolds number and heat transfer coefficient but increase significantly the heat transfer area per tube. The reduction of number of tubes or tube length balances the increase in heat transfer area.

As seen in Figure 5-6a, the overall weight of the heat exchanger increases with a larger diameter. However, when the number of banks is reduced, the weight increases almost linearly as the tube diameter increases (e.g. 120 kg at a diameter of 0.01 m to 280 kg at a diameter of 0.04 m for stainless steel). However, for a reduction on the tube length and/or tubes per bank, the weight increases initially, but after about 0.02 m tube diameter the curves flatten and the weight stays approximately the same. The volume (Figure 5-6c) also increases with larger diameters while reducing tube banks. However, the volume reduces for larger diameters when tubes per bank or tube length is reduced.

As seen in Figure 5-7 and Figure 5-8, the working fluid pressure drop increases with a smaller diameter. This is expected because the velocities of the fluid increase with the inverse of the diameter square. The corresponding larger flow path because of the larger number of tubes or tube length also contributes. In general, the pressure drops decrease from about 0.3 bar at tube diameters of 0.01 m to about 0.035 at diameters of 0.04 m; this corresponds to 1% and 0.12% of the absolute pressure respectively. In the case of the exhaust, the number of tube banks (number of tubes parallel to the exhaust flow direction) does not change significantly the pressure drop as seen in Figure 5-7a and Figure 5-8a, but the number of tubes per bank (number of tubes perpendicular to the exhaust flow) or the tube length does affect the pressure drop because the cross-sectional area of the flow is

altered. The maximum pressure drops in the exhaust side is around 7 mbar which is 0.7% of the absolute pressure. It is to be noted that is better to keep the pressure drop in the exhaust side as low as possible to not affect the engine performance.

From this analysis, it can be concluded that if a fin and tube evaporator with a large response time is desired, a better design strategy in terms of weight and volume of the system is to increase the diameter of the tubes while reducing the number of tubes per bank (tubes perpendicular to exhaust flow) or the length. This strategy will also reduce the pressure drops in both the working fluid and exhaust sides. On the other hand, a reduction of the number of tubes parallel to the exhaust flow would increase the volume and weight of the evaporator, while not influencing significantly the pressure drop in the exhaust side.

5.5.5 Implications for louver fin multi-port heat exchangers

The response time of louver fin multi-port heat exchangers also increases with an increase in port diameter. However, contrarily from fin and tubes heat exchangers, in order to keep the same heat capacity UA of the heat exchanger, larger port diameters need to be compensated with yet larger number of ports, larger number of tubes or larger tube length. This is because of the smaller scale of flow conduit diameters. Larger port diameters decrease the heat transfer coefficient, but does not increase significantly the heat transfer area per tube, therefore a larger increase in heat transfer area is required by increasing the number of tubes or tube length.

Because of this, the weight and volume of the heat exchanger always increases with the port diameter size as seen in Figure 5-6b and d. The mass increases in an exponential trend when the number of ports or number of tubes is increased, while the variation is more linear for an increase in tube length.

In terms of pressure drops, the working fluid pressure drop increases, as expected, with smaller diameters as well as longer flow path. In general, the pressure drops decrease from about 1.5 bar at diameters of 1.8 mm to about 0.02 bar at diameters of 2.6 mm; this

corresponds to 5% and 0.06% of the absolute pressure respectively. On the exhaust side, the geometric dimension that affects the most the pressure drop is the number of ports (dimension parallel to exhaust flow). This is because in this type of heat exchanger there are louvered fins in the flow path and this additional surface friction has a more important effect than the change in cross-sectional area. The maximum pressure drop in the exhaust side is around 300 mbar which is 30% of the absolute pressure. This is considerably higher than the values for the fin and tube heat exchanger, meaning that higher back-pressure in the exhaust needs to be carefully considered for this type of heat exchanger.

From this discussion it can be concluded that if a louver fin multi-port heat evaporator with larger response time is desired, a more effective design strategy, in terms of weight and volume of the system, is to increase the port diameters increasing the length of the tubes accordingly. Furthermore, this will not have much effect on the pressure drop in the exhaust side, but will reduce the pressure drops of the working fluid.

5.5.6 Comparison

The results show that for reasonable ranges of geometries, louver fin multi-port heat exchangers have approximately half the response time than fin and tube heat exchangers. Thus, if an evaporator that can dampen some of the variability of the heat source is required, fin and tube heat exchanger is a better choice. On the other hand, louver fin multi-port heat exchangers are the preferred choice for fast control operation.

While louver fin multi-port heat exchangers are significantly more compact in terms of volume than fin and tube heat exchangers, it has a larger mass for a given construction material due to the extra material in the louver and flat tubes. This can be different if the materials for fins and louvers is replaced by a lighter one.

The pressure drops are also substantially higher in louver fin multi-port heat exchangers, an order of magnitude higher for the exhaust side. This will lead to higher back pressure

which can reduce engine performance.

5.6 Dynamic regimes for frequency response

The thermal inertia of a heat exchanger is usually not considered at design stage but rather until the operation and control phases once the geometry and materials are defined. However, as it is recommended throughout this dissertation, it is an important aspect that should be addressed as well when designing systems intended to operate under dynamic conditions. The “design” response time of the evaporator will have an important effect on the operation and performance of the waste heat recovery system.

Depending on the magnitude of the response time of the evaporator when compared to the rate of variation of the heat input, the evaporator will be able to follow, to a higher or lesser extent, the fluctuations of the heat profile. As an illustration, let's take the example of a perfectly sinusoidal waste profile. The response of the evaporator can be classified roughly into one of the three *dynamic regimes* depicted in Figure 5-10. The dynamic regime I is a quasi-steady case where the response of the evaporator follows the input in the whole amplitude of variation. The dynamic regime III is a quasi-constant case in which the evaporator does not have time to react to the changes and the fluctuations are essentially filtered-out. The dynamic regime II is a case in between in which the inertia of the evaporator provides some damping of the amplitude of fluctuation of the source. A dynamic regime number Γ can be defined as:

$$\Gamma = \frac{\tau_{ev}}{T_{step}} = \frac{\tau_{ev}}{1/(2 \cdot f_{sin})} \quad (5-2)$$

This is the ratio of the evaporator response time τ_{ev} to a characteristic periodic time of the source T_{step} , which can be a half oscillation time in a sinusoidal profile with frequency f_{sin} . The magnitude of the dynamic regime number Γ provides an indication of the type of *dynamic regime* the ORC evaporator will be operating at. Approximate ranges of Γ for each dynamic regime are included also in Figure 5-10.

Figure 5-10 *Dynamic regimes* according to evaporator response time and period of fluctuation of the heat source.

For waste heat profiles that are not sinusoidal, Fourier analysis can be performed in order to identify the main equivalent periods of fluctuations at different sections of the profile. The ORC evaporator can then be designed in order to remove some of the variability of the waste heat source according to typical periods of fluctuation of the source by shifting the *dynamic regime* operation to one where there is a damping or filtering out of the fluctuations. The response time maps that have been presented show what it will take in terms of design of the heat exchanger to do so.

Typical periods of fluctuations for exhaust from Diesel engines of long-haul trucks are expected as low as the seconds range based on standard driving cycles such as the World Harmonized Transient Cycle (WHTC) [65]. Figure 5-11a reproduces Figure 2-2d showing the exhaust properties of the turbocharged Diesel engine of an omnibus subjected to the

WHTC based on data presented in [16]. From this profile, Fourier analysis can be performed to extract the different components of fluctuation in order to make more methodical the dynamic analysis. In particular, for the discrete data available of the exhaust properties, the Discrete Fourier Transform can convert the time domain data into the frequency domain to get the main frequency components. The Discrete Fourier Transform is computed with an algorithm called Fast Fourier Transform.

It is to be noted that the Discrete Fourier Transform, and thus the Fast Fourier Transform is applicable to any sequence of n samples of data $f(n)$ that satisfies:

$$\sum_{-\infty}^{\infty} |f(n)| < \infty \quad (5-3)$$

In other words, if all the values of the data are absolutely summable, the Fast Fourier Transform can be performed. This is the case for most sets of data, including the data of the exhaust profile of Figure 5-11a.

The Fourier analysis of the exhaust profile is shown in Figure 5-11b where the power spectral density of the different frequency components of the mass flow are shown by using the Fast Fourier Transform. The peaks represent the main frequency components present in the profile. For illustration purposes three of the peaks are highlighted corresponding approximately to 0.03 Hz, 0.01 Hz and 0.002 Hz.

The response time maps of Figure 5-5, Figure 5-7, Figure 5-8 and Figure 5-9 can be used to select a possible evaporator geometry that provides a desired dynamic response number Γ and the subsequent filtering or damping of these frequency components of the source. Table 5-7 presents, as an example, three possible evaporator geometries from the response time maps and the amount of thermal damping that these three evaporators will provide to each of the three selected frequency components of the source.

Figure 5-11 (a) Mass flow and temperature profile of the IC engine exhaust under the World Harmonized Transient Cycle from [16] (b) Spectral density – frequency components of exhaust profile using Fast Fourier Transform.

According to Table 5-7 and the dynamic regime number Γ values, evaporator A can filter out fast components of fluctuation in the range of 0.03 Hz or faster, and significantly dampen the fluctuation components in the range of 0.01 Hz. However, evaporator B can only dampen moderately the fluctuations in the range of 0.03 Hz and is not able to provide damping for fluctuations in the range of 0.01 Hz. This is confirmed, as shown in **Figure 5-12**, by the simulation of these three evaporators under regular sinusoidal mass flow fluctuations with frequencies of 0.03 Hz and 0.01 Hz. In the graphs the input fluctuation is defined in terms of the heat transferred by the exhaust at each instantaneous moment of time as:

$$\dot{Q}_{exh}(t) = \dot{m}_{exh}(h_{exh,out} - h_{exh,in}) \quad (5-4)$$

And the response of the evaporator is defined as the enthalpy gained by the working fluid in the evaporator by means of the following equation:

$$\dot{H}_{wf}(t) = \dot{m}_{wf}(h_{wf,out} - h_{wf,in}) \quad (5-5)$$

By using $\dot{Q}_{exh}$ and $\dot{H}_{wf}$ the input and the response can be plotted in the same graph and the damping of the fluctuations can be easily observed.

Table 5-7 Dynamic regime number Γ for different evaporator types and geometric dimensions given a characteristic period of fluctuation of the source. Response times read from figures Figure 5-7a and b. Average values of the source: flow rate of 0.3 kg/s, temperature of 350 °C.

Frequency component of oscillation of the source [Hz]	Evap. ID	Type	No. of banks/ No. of ports	No. of tubes	Length of tubes [m]	Resp. time τ_{ev}	Γ	Fluctuation dampening
0.03	A	Fin and tube	16	9	0.80	118	7.1	Nearly filtered out
0.03	B	Fin and tube	24	9	0.80	24	1.4	Moderate
0.03	C	Louver fin multi-port	20	6	0.84	18	1.1	Moderate
0.01	A	Fin and tube	16	9	0.80	118	2.4	High
0.01	B	Fin and tube	24	9	0.80	24	0.5	Minimal
0.01	C	Louver fin multi-port	20	6	0.84	18	0.4	Minimal
0.002	A	Fin and tube	16	9	0.80	118	0.5	Minimal
0.002	B	Fin and tube	24	9	0.80	24	<0.1	No damping
0.002	C	Louver fin multi-port	20	6	0.84	18	<0.1	No damping

This analysis shows that an ORC evaporator can be designed with a certain thermal inertia to protect the working fluid from fluctuations of the source on the order of a certain frequency of variation, reducing in the process deviations from a design point. In the case of this example, the geometry of the evaporator can be selected in order to dampen frequency components of 0.01 Hz or faster (evaporator A). The response time maps provide the requirements in terms of geometry and materials that will allow the system to do so.

Figure 5-12 Dampening of sinusoidal heat source for two different Evaporators as in Table 5-7. (a) Sinusoidal mass flow profile with frequency of 0.03 Hz and amplitudes of 0.01 kg/s. (b) Heat power input for profile with frequency of 0.03 Hz and enthalpy gained in the evaporator by the working fluid for evaporator A and B of Table 5-7. (c) Sinusoidal mass flow profile with a frequency of 0.01 Hz and amplitude of 0.01 kg/s. (d) Heat power input for profile with frequency of 0.01 Hz and enthalpy gained in the evaporator by the working fluid for evaporators A and B of Table 5-7.

In the case of the high inertia evaporator A, the dampening of slower periods of fluctuation such as 0.002 Hz is low, meaning that for these slow components of fluctuation with high amplitude changes, the control system needs to safeguard the integrity of the fluid. However, for evaporator A, the control does not need to respond extremely fast because the design of the evaporator itself protects the system in the case of fast changes. This is not the case for evaporator B, in which even fast periods of fluctuation of 0.01 Hz cannot be dampened due to the fast thermal inertia of the heat exchanger.

5.7 Summary

In this chapter a systematic characterization of the dynamic thermal response of two different types of heat exchangers for direct evaporation in ORC has been carried out. Response time maps have been built to show in a compact and general way the dependence that the response time has on different construction parameters as well as the changing boundary conditions. The implications in weight, volume and pressure drops have also been included.

The results show that fin and tube heat evaporators have double the response time in comparison to louver fin multi-port evaporators and are a preferred choice when some of the variability of the heat source needs to be dampened. It has also been concluded that the best way to increase the thermal inertia of the fin and tube evaporator is to increase the tube diameter while reducing either the number of tubes per bank or the length of the tubes accordingly. This represents the lowest penalty in weight, volume and pressure drops. On the other hand, if an evaporator with a faster response is desired, the option is to use a louver fin multi-port evaporator reducing the port sizes. This, however, comes with the penalty of high pressure drops of the exhaust and working fluid.

As it has been shown with the introduction of the *dynamic regime* analysis in the last part of the chapter, the response time maps represent guidelines for heat exchanger design that incorporates a customized thermal inertia in the optimization phase. The inclusion of this aspect is fundamental for direct evaporation so that the integrity of the system and fluid can be protected from fast extreme changes in boundary conditions. This is because it must be ensured that the working fluid at the outlet of the evaporator is below the chemical decomposition temperature and above the saturation temperature. Damping of fluctuations can also limit the inefficiencies related to deviations from a potential design point. The methodology presented can be easily extended to other heat exchangers and applications and serve as a systematic tool for heat exchanger screening and design under highly dynamic conditions.

In the next chapter, the recommendations and conclusions drawn from this chapter are applied to a scenario in which a real-case indirect evaporation layout is proposed to be replaced by a direct evaporation layout that still satisfies, within a degree, the working fluid integrity criteria.

Chapter 6 *

Replacing an indirect evaporation layout with direct evaporation

This Chapter applies the methodology of the response time maps from the previous Chapters for the highly relevant case of replacing an indirect evaporation with a direct evaporation layout. It analyzes qualitatively and quantitatively the dynamics of both layouts according to frequencies and amplitudes of fluctuation of the source and proposes a geometry of a direct evaporator that can better handle the fluctuations of an IC engine exhaust during a driving cycle. The methodology is further expanded with the concept of the maximum allowable amplitude ratio to identify the ranges of frequencies and amplitudes of thermal power fluctuation in which the design of the ORC evaporator ensures operation within safe boundaries even if the control system fails to act in a timely manner.

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6.1 Introduction

In the previous chapter, a methodology for the incorporation of the thermal inertia of the evaporator in the design phase has been proposed. The geometry of different types of heat exchangers has been shown to have an important impact in the damping capabilities when using direct evaporation. In this chapter the knowledge and recommendations from the previous chapter are applied in a case study. It is proposed to replace with direct evaporation an ORC that has an indirect evaporation layout. In Figure 6-1 a schematic of an ORC system with an indirect evaporation layout can be found along with the proposed direct evaporation layout intended to replace it. The direct evaporator must still, to a certain degree, protect the system from fast changes in the boundary conditions, while aiming to significantly reduce the weight and volume of the ORC. As it has been discussed, protecting the system means that the working fluid at the outlet of the evaporator stays below the chemical decomposition temperature and above the saturation temperature.

From the literature review, it is clear that to this date direct evaporation is not yet a very feasible practical option when dealing with high variability of the heat source. In this chapter it is proposed, then, a methodological quantification of the challenges compared to indirect evaporation and the proposal of an option that represents the best compromise in terms of performance, footprint, implementation and safe operation.

This is done by comparing two suitable systems with same fluctuating boundary conditions based on the relevant fluctuations from the World Harmonized Transient Cycle (WHTC). The aim is that the results enable to establish the range of frequencies and amplitudes of fluctuations of the source in which the system control using direct evaporation is more critical compared to indirect evaporation. It also must indicate how the direct evaporation heat exchanger methodology with dynamic behavior in mind can assist on reducing those challenges.

Because direct evaporation allows for benefits that are crucial for the successful widespread implementation of ORC system in mobile applications, it is pertinent that a

study such as this quantifies the challenges and propose solutions to mitigate those challenges.

Figure 6-1 (a) Layout of ORC-WHR system with indirect evaporation structure. (b) Layout of ORC-WHR system with direct evaporation structure.

6.2 Indirect evaporation reference system

A benchmark ORC system for exhaust WHR of an IC engine is considered. The benchmark system is a built and tested system from the literature [63] that utilizes an indirect evaporation structure with thermal oil as the intermediary heat transfer fluid. The exhaust comes from a 6-cylinder, 8.4L, heavy-duty Diesel engine with a rated power of 240 kW.

The benchmark ORC system schematic is shown in Figure 6-1a. The exhaust transfers its heat to the thermal oil in a conventional shell and tube heat exchanger. The heated thermal oil then enters the ORC evaporator proper, which is a plate heat exchanger, transferring the heat to the ORC working fluid. The ORC cycle is a simple non-regenerative cycle. The working fluid is R245fa, the benchmark fluid that has been used throughout this

dissertation. It has to be noted that R245fa has a decomposition temperature of around 250 °C [161]. The composition of the exhaust is based on the combustion products of a Diesel fuel with air using an equivalent fuel ratio λ equal to 1.2. The boundary conditions of the benchmark system are presented in Table 6-1. They represent an operating condition of the engine related to a medium-high torque and rotational speed.

Figure 6-2 shows a schematic of the main geometric parameters of the heat exchangers used in the indirect evaporation layout. The dimensions of the two heat exchangers are reported in Table 6-2. The dimensions are consistent with the reported geometry and effectiveness of the experimental setup in the literature [63]. The weight and volume of the heat exchangers are also calculated and shown in Table 6-3 and Table 6-4 respectively. All contributions to the weight of the construction material, oil and working fluid are considered. The exhaust weight is disregarded as it is negligible in comparison.

Table 6-1 Boundary condition and fluid descriptions of ORC system for a representative engine operating point.

Description	Value
Exhaust mass flow	0.25 kg/s
Exhaust inlet temperature exhaust	380 °C
Working fluid mass flow	0.17 kg/s
Working fluid inlet temperature	20 °C
Working fluid inlet pressure	14 bar
Working fluid initial degree of super-heating at outlet	35 °C
Working fluid	R245fa
Exhaust properties	Combustion gas $\lambda = 1.2$

(a)

(b)

Figure 6-2 Geometries of heat exchangers of indirect evaporation layout. (a) Shell and tube heat exchanger (exhaust to oil) (b) Plate heat exchanger (oil to working fluid).

6.3 Proposed direct evaporation heat exchangers

The benchmark ORC is compared to a proposed layout consisting of a direct evaporation structure subjected to the same engine exhaust conditions. The layout is shown in Figure 6-1b. In order to make the comparison meaningful, the system is designed so that given the engine operating point of Table 6-1, the same super-heating of the working fluid at the outlet of the evaporation is achieved given the same working fluid type, evaporator pressure and mass flow.

Based on the recommendations of the previous chapter, a fin and tube type is considered for the direct evaporator. This is because a fin and tube heat exchanger has a higher

damping capability while keeping a smaller back pressure to the engine compared to, for instance, a louver fin multi-port heat exchanger.

As has been presented in the previous chapter, different combinations of the geometric parameter values of the fin and tube heat exchanger – such as the diameter and length of the tubes, as well as the number and arrangement of the tubes – can achieve the same inlet and outlet conditions of the fluids (i.e. same thermal heat capacity UA of heat exchanger). It has also been shown in the previous chapter that the particular combination of geometric parametric values of the heat exchanger will play an important role in the dynamics of the system.

Taking this into consideration, and to highlight the importance of this aspect, two different geometries of a fin and tube direct evaporator will be considered as candidates to be compared with the indirect evaporation structure. Using the methodology proposed in Chapter 5, response time maps are developed based on dynamic simulations for the particular boundary conditions of this case study (Table 6-1). It is to be noted that in this case, the evaporation temperature (and evaporation pressure) is considerably lower because indirect evaporation has inherently lower temperatures compared to the potentiality of direct evaporation. From the previous chapter it has been concluded and recommended that to increase the thermal inertia the better strategy is to modify the tube length or number of tubes perpendicular to the exhaust flow.¹ Therefore, Figure 6-3 shows a compact version of the response time maps of the fin and tube evaporator. The x-axis shows the diameter and the corresponding total flow length (total number of tubes times the tube length). Since it has been found, in the previous chapter, that the wall material does not have a significant effect on the response time, here it is only shown the case where the y-axis represent different values of mass flow and temperature of the exhaust.

¹ Another option is to increase the volume of the metal wall or the fins however this would incur in considerably weight increase which is not acceptable in mobile applications.

Two evaporators are selected from the response time maps in Figure 6-3. A direct evaporator labeled as direct evaporator A is selected on the fast response time range. It consists of a conventional tube diameter size within the usual range for tube-based heat exchangers [162]. The second, direct evaporator B, is selected in the slow range of response time in order to increase the thermal inertia of the heat exchanger while still keeping dimensions to reasonable values. The location of both evaporators in the response time maps of Figure 6-3 is also shown. The geometric dimensions of both evaporators are reported in Table 6-2.

The weight and volume of both direct evaporation options are shown in Table 6-3 and Table 6-4 respectively. In the last row of the tables, the percentage of mass and volume of the evaporation system is compared to the benchmark indirect evaporation structure.

Figure 6-3 Response time maps of fin and tube heat exchanger with working fluid boundary conditions as in Table 6-1 for different values of exhaust properties and heat exchanger geometric dimensions. (a) Geometry vs exhaust mass flow (b) Geometry vs exhaust temperature.

Table 6-2 Geometry and properties of heat exchangers considered in this Chapter. Direct evaporator B corresponds to a high thermal inertia evaporator.

Geometric parameter	Indirect Gas to Oil HEX - Shell and tube	Indirect evaporator - Plate	Direct evaporator A – Fin and tube	Direct evaporator B – Fin and tube
Number of tubes/plates [-]	400	40	16	18
Length of tubes/plates [-]	0.85	0.7	0.63	0.68
Tube inner diameter [-]	0.02	N/A	0.02	0.06
Plate width [m]	N/A	0.1	N/A	N/A
Wall thickness [mm]	2.5	1	2.5	2.5
Tube/plate/fin material	Stainless steel	Stainless steel	Stainless steel	Stainless steel

Table 6-3 Mass of heat exchangers for indirect and direct evaporation structures including solid materials and fluids inside.

Component	Indirect evaporation	Direct evaporation A	Direct evaporation B
Gas to oil heat exchanger	468.6 kg		
Evaporator	38.1 kg	14.1 kg	47.1 kg
Thermal oil	100.3 kg		
Working fluid	4.5 kg	1.9 kg	23.9 kg
Total	611.5 kg	16.0 kg	71.0 kg
Mass % of indirect evaporation option	100%	3%	12%

Table 6-4 Volume of heat exchangers for indirect and direct evaporation structures.

Component	Indirect evaporation	Direct evaporation A	Direct evaporation B
Gas to oil heat exchanger	0.54 m ³		
Evaporator	0.01 m ³	0.06 m ³	0.17 m ³
Total	0.55 m ³	0.06 m ³	0.17 m ³
Mass % of indirect evaporation option	100%	11%	31%

It is to be noted the dramatic reduction in terms of weight and volume compared to the benchmark indirect evaporation case. Direct evaporator A has only 3% of the mass and 11% of the volume of the indirect evaporation structure. Direct evaporator B has 12% of the mass and 31% of the volume of the indirect evaporation structure. The higher mass and volume of direct evaporator B is expected because it has been selected in order to increase the volume of the working fluid, thus increasing the thermal inertia. However, the reduction of mass and volume compared to the indirect case is still highly significant. It showcases one of the advantages of direct evaporation as lighter systems with less footprint are particularly crucial in mobile applications.

6.4 Dynamic response comparison for representative fluctuations

As it has been done in Chapter 5, the World Harmonized Transient Cycle is used as a way to study the IC engine variations in a diverse range of operating conditions. For easier reference the mass flow and temperature profile of the exhaust under the WHTC from [16] is reproduced again in Figure 6-4a. It is to be noted that the values are consistent with the range of variation of the IC engine reported in the benchmark system [63]. Also following the same procedure of Chapter 5, the profile is decomposed into its constituent sinusoids using Discrete Fourier Analysis. Figure 6-4b reproduces again the power spectral density for the mass flow of the profile along with some of the main frequencies identified in Chapter 5. The temperature profile follows closely the mass variation and thus it is understood that these relevant frequencies also apply to the temperature profile. In terms of amplitude of fluctuations of the exhaust, from the profiles it is seen that roughly a variation of 0.05 kg/s of the mass flow corresponds to a temperature variation of 100 °C.

For the purpose of a methodological dynamic response comparison between the different ORC evaporation structures, sinusoids with the relevant frequencies and amplitudes of fluctuation can be used. The strategy for this dynamic comparison is illustrated in Figure 6-5. A sinusoidal variation of both mass flow and temperature of the exhaust is placed at the inlet boundary condition of the model. The fluctuations are centered around the exhaust base boundary conditions of Table 6-1. The model is simulated and the outlet condition of the working fluid recorded.

Figure 6-4 (a) Mass flow and temperature profile of the IC engine exhaust under the World Harmonized Transient Cycle from [16]. (b) Spectral density – frequency components of exhaust profile using Fast Fourier Transform.

Figure 6-5 Strategy for dynamic response comparison of evaporation structures. (a) Indirect evaporation (b) Direct evaporation.

Figure 6-6 Heat transferred from exhaust $\dot{Q}_{exh}$ and response of oil $\dot{H}_{wf}(t)$ and working fluid $\dot{H}_{wf}(t)$ (enthalpy gain) for two different frequencies and amplitudes of sinusoidal variation of the exhaust mass flow and temperature.

Figure 6-6 shows the dynamic response at the evaporator side for different frequencies and amplitudes of the mass flow and temperature of the exhaust, conforming to the range in Figure 6-4. Figure 6-6a and c for a frequency of fluctuation of 0.002 Hz of both mass flow and temperature and Figure 6-6b and d for a frequency of 0.03 Hz. On the other hand,

Figure 6-6a and b correspond to an amplitude of fluctuation of 0.5 kg/s and 100 °C, while Figure 6-6c and d correspond to amplitude of fluctuation of 0.25 kg/s and 50 °C. In this way, large and small amplitudes as well as slower and faster frequencies within the range of variation of the exhaust in the driving cycle are represented. A more general analysis of the results for the whole range of fluctuations is presented in Section 6.5.

As it has been done in Chapter 5, in the graphs the input fluctuation is defined in terms of the heat transferred by the exhaust at each instantaneous moment of time as:

$$\dot{Q}_{exh}(t) = \dot{m}_{exh}(h_{exh,out} - h_{exh,in}) \quad (6-1)$$

And the response of the evaporator is defined as the enthalpy gained by the working fluid in the evaporator by means of the following equation:

$$\dot{H}_{wf}(t) = \dot{m}_{wf}(h_{wf,out} - h_{wf,in}) \quad (6-2)$$

Similarly, the instantaneous enthalpy gained by the thermal oil in the indirect evaporation structure is defined as:

$$\dot{H}_{oil}(t) = \dot{m}_{oil}(h_{oil,out} - h_{oil,in}) \quad (6-3)$$

By using $\dot{Q}_{exh}$ and $\dot{H}_{wf}$ the input and the response can be plotted in the same graph and the damping of the fluctuations can be easily observed.

It is shown that, as expected, the indirect evaporation option has a much higher capability of damping the fluctuations of the source and that, direct evaporator A has the lower damping capability. For instance, in Figure 6-6a the response of indirect evaporation shows a very slight variation in comparison to the heat input while the direct evaporator A follows closely the heat input profile. Direct evaporator B shows somewhat higher value of damping compared to direct evaporator A owing to its higher thermal inertia. It is also

observed that the relative damping of fluctuation depends on the frequency of variation of the source. In the case of fluctuations with a faster frequency, as in Figure 6-6b and d, the damping is higher for all systems, since the evaporation options have less time to “sense” the changes. This is in line with the concept of *dynamic regimes* of Chapter 5. In terms of different amplitudes, the relative damping stays approximately the same for a given frequency; however, the absolute change of variation is, obviously, less with a smaller amplitude, as it is observed when comparing Figure 6-6a and c.

Figure 6-7 shows the result of the simulation in terms of the temperature of the working fluid at the outlet of the evaporator.

In the case of Figure 6-7a it is seen that because of the negligible thermal damping of evaporator A, for this particular frequency and amplitude of fluctuation the temperature of the working fluid at the outlet of the evaporator falls below the saturation line, which is unacceptable for the ORC operation. This means that for this scenario, direct evaporator A would require the action of a robust controller to adjust the parameters such as the working fluid mass flow in order to assure the safe operation of the system. This contrasts with the indirect evaporation option which leaves the working fluid close to the target super-heating value without the need of control. Direct evaporator B, however, due to its higher thermal inertia, still protects the fluid from falling below the saturation line. This shows that for direct evaporation, the evaporator can still be designed in order to dampen the fluctuations to a certain degree. For smaller amplitudes such as the case of Figure 6-7c, the outlet temperature stays within acceptable limits. This is also the case for faster frequencies such as in Figure 6-7b and d, due to the higher relative damping.

Figure 6-7 Response of outlet temperature of working fluid $T_{wf,out}$ to fluctuations of exhaust mass flow and temperature for two different frequencies and amplitudes of sinusoidal variation of the exhaust mass flow and temperature.

6.5 Amplitude ratio and thermal power damping

In the previous Section 6.4 some results have been shown comparing the dynamic response of indirect evaporation structure and two different designs of direct evaporators for a case study. This has been done for some specific frequencies and amplitudes of fluctuation within the range found in a standard driving cycle. Now, the analysis is generalized and

quantified for all frequencies and amplitudes as well as different target super-heating values at the outlet of the evaporator. The case study remains the same, so the other boundary conditions are still the same as before.

From the previous results, it has been shown that there is a difference on the *thermal damping* of the exhaust heat fluctuations depending on the type of evaporator structures and heat exchanger design. The *thermal damping* refers to the lower amplitude of fluctuations of the instantaneous heat absorbed by the working fluid compared to the instantaneous heat transferred by the exhaust. In order to quantify this *thermal damping*, the relative amplitudes of heat fluctuation are compared with the *amplitude ratio* AR defined as:

$$AR \leq \frac{\max(\dot{H}_{wf}) - \min(\dot{H}_{wf})}{\max(\dot{Q}_{exh}) - \min(\dot{Q}_{exh})} \quad (6-4)$$

Figure 6-8 shows the *amplitude ratio* AR of the different evaporator structures considered as a function of the frequency of exhaust heat fluctuation. The curves are trend lines resulting from the best fit of different simulations at different frequencies and amplitudes. It is to be noted that Figure 6-8 only shows the relationship with the frequency of exhaust fluctuation and not with the amplitude. This is because it has been observed through an extensive simulation campaign that, although the amplitude of exhaust heat fluctuation evidently affects the resulting *absolute amplitude* of fluctuation of the working fluid, it does not have a major effect on the *amplitude ratio*, which is a normalized term that quantifies the *thermal damping*. The reason for this is that, as can be observed from Figure 6-3, for moderate exhaust flow or temperature changes, the response time of the evaporator is affected mainly by the geometry and less by the amplitude of change of exhaust conditions. Therefore, only one fitting curve of the AR , independent on the amplitude but dependent on the frequency of exhaust heat fluctuation is considered for each evaporation structure.

Figure 6-8 Amplitude ratio AR of different evaporator structures according to different frequencies of exhaust fluctuation.

A low amplitude ratio closer to the minimum value of zero represents a system with almost complete *thermal damping*, a system where the working fluid state at the outlet of the evaporator stays constant under sinusoidal variation. A higher amplitude ratio closer to the maximum value of unity represents a system with almost no *thermal damping*, a system that follows exactly the fluctuations of the exhaust heat.

For safe operation of the system the working fluid state at the outlet of the evaporator must fall below the fluid chemical decomposition temperature as well as above the vapor saturation state. Therefore, there is a maximum *amplitude ratio* (or minimal *thermal damping*) required for the safe operation of the system. The maximum *amplitude ratio* can be defined as:

$$AR_{max} \leq \frac{\dot{m}_{wf} \cdot C_{p_{wf}} \cdot \Delta T_{max}}{\max(\dot{Q}_{exh}) - \min(\dot{Q}_{exh})} \quad (6-5)$$

ΔT_{max} is the maximum allowable temperature fluctuation. If it is assumed that the working

fluid in the vapor phase has a constant, average heat capacity Cp_{wf} , ΔT_{max} can be defined as:

$$\Delta T_{max} = \min[(T_{wf,out} - T_{wf,sat}), (T_{wf,decomp} - T_{wf,out})] \quad (6-6)$$

The maximum allowable temperature fluctuation is the smallest fluctuation at which the working fluid at the outlet of the evaporator will be at an unacceptable state. That is, either below the lower limit saturation temperature or above the upper limit chemical decomposition temperature.

From equation (6-5) it can be seen that the maximum *amplitude ratio* AR_{max} depends on the magnitude of the amplitude of fluctuation of the exhaust heat: $\max(\dot{Q}_{exh}) - \min(\dot{Q}_{exh})$. As it was shown in the exemplary simulations of section 6.4, a smaller amplitude is less problematic than a larger one since the smaller magnitude of variation means that the fluid can still stay within acceptable boundaries.

In Figure 6-8, two horizontal bold black dashed lines are also included. They represent the maximum allowed *amplitude ratio* AR_{max} for two different amplitudes of fluctuation of the exhaust heat. That means that for such an amplitude, a system that provides AR smaller than the AR_{max} line would be enough to keep the system operating at safe conditions even if no control is present.

From Figure 6-8, it can be seen that for a small amplitude of exhaust heat of 8 kW, the evaporation systems are able to dampen the fluctuations to a safe state unless the fluctuations are too slow (i.e. around 0.003 Hz for direct evaporator A). For fluctuations of the exhaust heat of 20 kW of amplitude, the direct evaporator A is able to dampen fluctuations to the safe range only for frequencies of the source faster than around 0.024 Hz. That means that frequencies slower than that absolutely require a control system that reacts fast enough. This unsafe range however is reduced for the direct evaporator B, who can dampen safely the fluctuations of the source that are faster than around 0.003 Hz. In

summary, direct evaporator B keeps the advantages of direct evaporation, but also possesses a higher thermal inertia that enables the system to be more robust to fast changes of the heat source. In this way the control measures in the unsafe area of fluctuation don't need to be as fast as direct evaporator A.

The maximum allowable *amplitude ratio* AR_{max} also depend on the target super-heating at the outlet of the evaporator. A smaller super-heating value closer to the saturation point increases the thermal damping requirement if the system is to be in safe operation without any control measure. Figure 6-9 shows the dependence of AR_{max} on the target super-heating value at the outlet of the evaporator as well as the amplitude of fluctuation. From Figure 6-8 and Figure 6-9, it is seen that a robust control system is required when a very small target super-heating is pursued, or when the ORC evaporator structure has a poor thermal damping capability. From this analysis it can be concluded that a more robust system to exhaust fluctuations, that does not require an extremely fast control response, can be achieved by relaxing the super-heating requirement or by designing the evaporation structure with the dynamic response in mind, increasing its thermal inertia.

Figure 6-9 Maximum amplitude ratio AR_{max} required as function of the amplitude of thermal power fluctuation for different values of initial super-heating for a thermal power sinusoid of 20 kW of amplitude.

6.6 Implications of results

As it can be seen from the previous results, direct evaporation in ORCs can have a considerably lower capability of thermal damping compared to indirect evaporation. This can lead to a lower safety margin in terms of fluid chemical decomposition or liquid droplets in the expander when there are fluctuations of the heat source. However, the results also show that the direct evaporator can be designed, with these drawbacks in mind, with a large thermal inertia in order to increase the safety margin. Because of the indisputable benefits that direct evaporation has over indirect evaporation, a design such as direct evaporator B of this case study represents a good compromise between the direct evaporation benefits and safety margin during fast dynamic conditions.

The principal benefits of direct evaporation can be classified in two areas. One is the reduction of the footprint, weight and number of components of the system. In particular the results of this case study show that direct evaporator B's mass is only 12% of the combined weights of the indirect evaporation heat exchangers. This is shown in Table 6-3. In Table 6-4, it is shown that direct evaporator B has a volume of only 31% of the combined volume of the indirect evaporation heat exchangers.

The other benefit of direct evaporation stems from the potential of a system with less exergy losses and a higher thermal efficiency. With direct evaporation there is no longer a need for an intermediary heat exchange process, and therefore the saturation temperature of the working fluid in the evaporator can be increased in order to better match the heat source temperature.

In Figure 6-10a, the heat exchange process in the indirect evaporation structure considered in this study is presented in a Q-T diagram. Figure 6-10b presents the same heat exchange process for the direct evaporators A and B. It is to be noted that the exhaust and working fluid profiles are exactly the same for the two graphs since the same boundary conditions were imposed for both indirect and direct evaporation in this case study in order to make

the comparisons consistent.

However, it can be seen from these figures, that in the case of direct evaporation, there is an exergy potential unused, since there is no longer the intermediary thermal oil. Direct evaporation can allow increasing the saturation temperature in the evaporator. In Figure 6-10c the heat exchange process is shown for an evaporator with R245fa at 30 bar corresponding to a saturation temperature of 143 °C. These values of saturation correspond to the evaporators studied in Chapter 5.

Table 6-5 presents the thermal efficiency of the system studied and the potential higher thermal efficiency of a system using direct evaporation with a higher evaporation temperature based on the thermodynamic cycle. The ORC gross thermal efficiency $\eta_{t,ORC}$ reported in the table is defined as:

$$\eta_{t,ORC} = \frac{\dot{W}_{net,out}}{\dot{Q}_{in}} \quad (6-7)$$

$\dot{W}_{net,out}$ is the net power output of the thermodynamic cycle and $\dot{Q}_{in}$ the heat supplied by the source. The gross thermal efficiencies of both systems are calculated at a design point condition, considering the same heat source and working fluid conditions as in Table 6-1. For the thermodynamic cycle of both systems also the same expander and pump isentropic efficiencies and the same condenser pressure are considered. The relevant values for the calculation are reported in Table 6-5.

The system of the case study in this Chapter has an evaporator pressure of 14 bar that corresponds to a thermal efficiency of 11.5%. This efficiency can be increased to 14.9% if direct evaporation is used with 30 bar as the evaporation pressure for R245fa as was considered in Chapter 5. This example shows the potential for efficiency increase by using direct evaporation instead of indirect. Furthermore, other different working fluids that better match the heat profile while maintaining lower pressures can also be used (e.g. Novec 649). The use of this fluids with higher saturation temperature can further increase

the thermal efficiency.

Figure 6-10 Q-T diagram of ORC evaporation heat exchange process (a) Indirect evaporation structure (b) Direct evaporators A and B (c) Direct evaporation with higher evaporation pressure.

Table 6-5 Thermal efficiencies of ORC systems

	ORC system in case study of this Chapter	ORC system with high evaporation pressure
ORC fluid	R245fa	R245fa
Evaporator pressure	14 bar	30 bar
Condenser pressure	2 bar	2 bar
Expander isentropic efficiency	80%	80%
ORC gross thermal efficiency	11.5%	14.9%

6.7 Summary

In this chapter a comparison of the dynamic behavior of different types of ORC evaporation structures recovering heat from the same engine exhaust has been presented. A real case indirect evaporation option from the literature has been compared to two different direct evaporation geometries. The results have been presented in a qualitative and quantitative way for different frequencies and amplitudes of heat fluctuations based on ranges present in a typical driving cycle for a particular engine size.

The results show that indirect evaporation has a much higher capability of damping the heat fluctuations, thus protecting the system from fluid chemical decomposition or liquid droplets in the expander even when control measures are not present. However, direct evaporation has important advantages over indirect evaporation, mainly because of its considerably lower footprint and potential for higher thermal efficiency. A conventional geometry selection of a direct evaporator based on fin and tube heat exchanger has shown to dampen the exhaust heat fluctuations of 20 kW of amplitude only for frequencies that are faster than 0.024 Hz. However, the geometry of the heat exchanger can be chosen in a way as to increase its thermal inertia. With this in mind, a different fin and tube geometry has been also chosen and it proves to increase the range of frequencies faster than 0.003 Hz for a 20 kW amplitude of fluctuation. Such a design only requires a control system that does not need to react as fast, and it is a good compromise between the benefits of direct evaporation and the safety provided by indirect evaporation.

This further supports the main statement of this dissertation that postulates that the thermal inertia of the evaporator can be effectively customized at design stage in order to achieve a desired dynamic behavior that supports the control scheme and makes an overall more robust system to fluctuations of the source.

Direct evaporation can dramatically reduce the weight and volume of the ORC system. In this case, the mass of the direct evaporator with high thermal inertia is only 12% of the indirect evaporation arrangement, and has a volume of only 31% of the indirect evaporation

volume. Furthermore, if the challenges of direct evaporation can be mitigated using this method, the large temperature difference between heat source and ORC working fluid, inherent of indirect evaporation, can be reduced. This allows for a system with less exergy destruction in the heat transfer process, and one with a potential increase in thermal efficiency due to the evaporation temperature being increased. Such a reduction in footprint and increase in efficiency for a system that still operates within the acceptable integrity boundaries are crucial requirements for the widespread adoption of ORC on mobile engine waste heat recovery.

Chapter 7

Conclusions and future perspectives

This Chapter concludes the dissertation by contrasting the results presented throughout with the original thesis statement of Chapter 1. The concepts and methods presented in the previous Chapters are summarized and drawn together into a condensed cohesive methodology. The impact and implications of the thesis contributions are stated, as well as the limitations and further improvements that can be developed in the future.

7.1 Recapitulation of this work and its contribution.

This thesis proposes an alternative method to manage thermal power fluctuations of the source when using ORC for waste heat recovery. As it has been presented in the literature review, currently the efforts are focused on either stream control or thermal energy storage (including indirect evaporation with a thermal oil loop) as methods to efficiently handle the fluctuations and allow the system to operate within acceptable boundaries and reduce inefficiencies.

However, thermal energy storage or indirect evaporation involves additional complexity as well as weight and volume that may not be acceptable in size-restricted applications. Stream control, on the other hand, may be difficult to implement in practice specially when dealing with a system with direct evaporation.

The proposal is then to incorporate to a certain degree the benefits of thermal energy storage / indirect evaporation while keeping the advantages of a simple and potentially more efficient direct evaporation layout. This can be done by reimagining the heat exchanger design method when dealing with ORC evaporators, and design a counterintuitive heat exchanger with higher thermal inertia that can dampen and reduce the variability on operation by matching and damping certain components of fluctuations inherent to the waste heat source. For this proposal a methodology is presented and applied throughout this work.

In this Section, the rationale is recapitulated and the methodology is drawn together and presented in condensed form including the implications it can have on the future of ORC systems for waste heat recovery.

7.1.1 Rethinking the design of ORC evaporators for WHR

At the design stage, an ORC system is usually thermodynamically optimized at a system level and the components are selected based on the boundary conditions found by the

optimization. The selection of the ORC evaporator in particular follows the standard practice of heat exchanger design. This standard method can be seen in on the left hand side of Figure 7-1 on the light blue boxes. It involves the selection of the type of heat exchanger based on the application and then a series of iterative steps to find the required heat transfer area within a certain permissible pressure drop. These steps can be done manually but are more often performed with the aid of a computational tool.

This standard methodology has been tested and proven for many years and is used regularly on the process industries. However, one thing that it ignores is a unique characteristic of waste heat to power systems: that in such systems the heat exchanger (the evaporator in the case of ORCs) operates at transients most of the time due to the constant fluctuations of many waste heat sources. The behavior of the system during those transients will be dictated by the thermal inertia of the heat exchanger that links the heat source to the power system, that is, for the case of ORCs, the evaporator.

This is the novel concept that this work is addressing. The standard methodology for heat exchanger design must be adapted to the case of ORC evaporators for waste heat recovery. In other words, the thermal inertia of the evaporator must be taken into consideration, not only at the operation stage when the heat exchanger has been already defined, but rather necessarily at the design stage when the thermal inertia can be customized to better match the requirements of the system and the particular type of profile of the waste heat. For instance, an evaporator with fast response if cold start-ups are often the norm, or an evaporator with high inertia for a system that operates for extended periods of time but under a heat profile that fluctuates often and at high frequencies around average values.

This additional information provided by this thesis influence the type of heat exchanger selected as well as the geometric dimensions and materials of the heat exchanger. Figure 7-1 shows the additional input provided by this work on the right hand side on dark red boxes and where it fits into the standard methodology,

Figure 7-1 Modification of heat exchanger design methodology for ORC evaporators proposed by this work

7.1.2 Proposed methodology for evaporator dynamic response customization

This thesis not only has identified the estimation and customization of the evaporator thermal inertia as an important asset for ORCs in WHR applications, but it has also presented methods to achieve these goals. Throughout this thesis different method steps have been explored. They have been used to analyze the impact of different design factors on the evaporator thermal inertia for different types of heat exchanger and the application of this knowledge to modify the evaporator design in order to better match the dynamics of the heat source.

In Chapter 4, the response time of the ORC evaporator has been defined and the concept of response time maps has been introduced as a tool to quantify the effect of different heat exchanger design parameters on the response time.

Chapter 5 has expanded the use of response time maps for complex geometries of heat exchangers for direct evaporation and introduced the dynamic regime number Γ as a

method to use the maps to select the heat exchanger type, materials and dimensions in order to customize the dynamic behavior of the evaporator based on the dynamic characteristics of the waste heat profile.

In Chapter 6, these tools have been applied for the case of replacing an indirect evaporation layout with a direct evaporation one. The methodology has been further expanded to include the considerations of the weight and volume (an advantage of direct evaporation) and introducing the concepts of the amplitude ratio AR and the maximum allowable fluctuation defined by the maximum amplitude ratio AR_{max} .

All this different concepts and methods have been introduced gradually at different points throughout this thesis to showcase the potentiality and compare the unconventional dimensioning of heat exchangers to the conventional evaporator options. Now, finally all these method steps can be drawn together into a cohesive methodology. Figure 7-2 presents the methodology proposed in this thesis as a block diagram. The methodology follows a series of steps (dark blue boxes) with some decision blocks (green diamonds). Notes (yellow circles) have been included besides the relevant steps referencing to the sections that contain more information or to table or figures that show an example.

The methodology starts by identifying the waste heat profile. In the case of profiles with very stable conditions or with dynamics that are not significant, the methodology is not pertinent and standard methods of heat exchanger design can be used. However, if the waste heat profile exhibits important fluctuations or dynamics, the methodology is applicable.

The next step is to use standard methods to specify the ORC at a system level and obtain the main requirements for the evaporator based on the thermodynamic cycle requirements. These include the heat exchanger thermal capacity UA required to transfer a certain amount of heat from the heat source and obtain the necessary outlet conditions of the evaporator. Once the constrained requirements are defined, response time maps can be used to quantify the response time as a function of geometrical dimensions, materials and/or boundary

conditions. Response time maps for one or more types of heat exchangers can be used. For more information on the response time maps refer to Sections 4.5 and 5.5

In parallel, it is necessary to identify the main frequency components of the waste heat profile by means of Fourier analysis or other method. For an example, refer to Figure 5-11.

Once the response time maps of the candidate evaporators and the main frequencies of the waste heat profile have been identified, the dynamic regime number Γ for the different frequencies and evaporator candidates can be calculated. The dynamic regime number Γ represents the damping capabilities of an evaporator to certain frequencies of fluctuation. Refer to Section 5.6 for a more complete explanation. The evaporator type, construction and dimensions can then be selected from the response time maps based on the desired dynamic behavior represented by Γ . For an example refer to Table 5-7 on Section 5.6.

At this stage the volume and weight of the candidate evaporator can be calculated and compared to a reference in order to decide if they lie within acceptable values for the application (e.g. mobile). If that is the case, the amplitude ratio AR can be calculated for the selected evaporator based on simulations at different frequencies as it has been done in Figure 6-8. See Section 6.5 for further details.

In the next step the amplitude ratio can be compared to the maximum allowable amplitude ratio in a diagram similar to that of Figure 6-8 to identify the frequencies and amplitudes where the evaporator is effective in filtering out or damping the fluctuations. If this is satisfactory within the weight and size restrictions and thermodynamic requirements, the evaporator is accepted and the control of the system can be designed with its thermal inertia on mind. If it is not, the process must restart from the step involving the selection of the evaporator from the response time maps.

Figure 7-2 Summary of methodology proposed for dynamic behaviour design of ORC evaporators.

7.1.3 Impact

The methodology and concepts presented in this thesis originate from the research gaps and opportunities identified in the literature review of Chapter 2. The main motivation for the usage of the proposed methodology is the possibility to use direct evaporation even with highly dynamic heat sources and potentially increase in that way the thermal efficiency while reducing the complexity, weight and volume of the system.

As explained in Section 6.6 and exemplified in Table 6-5, the potential increase in thermal efficiency of a direct evaporation system compared to an indirect evaporation layout is significant (14.9% compared to 11.5% in that case). The weight and volume can also be considerably reduced as exemplified in Table 6-3 and Table 6-4 (only 12% weight of the indirect evaporation arrangement, and 31% volume for that case). However direct evaporation, has the disadvantage of a lower capability of damping of the heat source fluctuations.

These disadvantages are trying to be solved with the methodology provided by this thesis. The unconventional dimensioning of a heat exchanger with a certain thermal inertia is proposed as a compromise between the advantageous more stable operation of indirect evaporation and the reduction of weight and volume of direct evaporation. And it enables the full potential of higher thermal efficiency of a system with lower exergy destruction in the heat exchange process that direct evaporation provides.

Furthermore, the customization of the thermal inertia of the ORC evaporator allows for a more effective optimization of the control system design. If the thermal inertia can be customized to a set value, it opens the possibility of being able to design the hardware that best works for the controller and not the other way around. In this way there is more freedom to design a control system and strategy that works in a more robust and efficient way.

7.2 Limitations

Any technology has limitations and a constrained field of applicability. It is important to analyze and understand these restrictions in order to better utilize the concepts and methodology proposed in this thesis and be able to propose congruent recommendations for further improvement in future work.

The methodology presents the possibility of the customization of the thermal inertia of ORC evaporators and has been applied to standard types of heat exchangers. Although the methodology can potentially be used for any type, even novel ones, there will always be a limit on how much the customization of the thermal inertia can be achieved. The response times cannot be selected arbitrarily large or small, since there are constraints imposed by the geometry. For instance, for fin and tube heat exchangers, the response time can be increased by increasing the diameter size and reducing the number of tubes or tube length, however the diameter cannot be increased infinitely because a certain length of the tubes is needed in order to provide a fully developed flow allowing for effective heat transfer. Likewise, very large diameters can reduce flow velocities and Reynolds numbers to impractical values. Furthermore, pumps with unconventional flow-head curves may be required. For very small diameters, on the other hand, the pressure drops may become impermissible.

It has been proposed that an evaporator with a high thermal inertia may be desired because it has the potential of damping many frequencies of fluctuation and allow for a safe operation even if the control system fails or does not respond fast enough. However, it must also be noted that such an evaporator will carry other disadvantages due to its high thermal inertia. For instance, the process of start-up of the system could be considerably slow which may become a bigger disadvantage in systems that very often require cold start-ups. Furthermore, if for some reason the evaporator reaches an unsafe level (e.g. a high temperature of the fluid risking chemical decomposition), the action to move the system back to the safe operational area will also take more time. Therefore, care must be taken to ensure that the robustness of the evaporator and its control helps against such situations

and not the other way around. It is important to notice, however, that the methodology helps customize the thermal inertia for whatever value, even if a large one or a shorter one is preferred.

It is to be noted, also, that the numerical results in this thesis depend on the assumption that the dynamic model has been validated for a simple geometry and certain conditions and that this validation has been extrapolated to other scales and conditions. The main assumption is that, at least for heat exchangers with common geometries (e.g. tubular or cylindrical conduits) with a common range of diameter sizes, the fluid properties do not vary much in the radial direction (and assumption of the model and experimental measurements) and the phenomena can be described purely with balances in one direction (the flow direction), and momentum and heat transfer phenomena described by correlations that depend on dimensionless numbers such as the Reynolds and Prandtl numbers. For very unconventional geometries or scales that depart considerably from the ones considered, the dynamic models must be re-adjusted (e.g. consider the radial direction balances). However, it must be also noted that the main contribution of this thesis is the methodology, and it does not depend on the dynamic models being perfect, or on the particular numerical results.

7.3 Recommendations for future work

Based on the work done, its potential impact, but also its current limitations, some recommendations for further refinement and improvement can be proposed. This include the integration of controller design in parallel with the thermal inertia customization and a multi-objective optimization of the evaporator design that takes into account the thermal inertia but also considerations such as the weight and cost of the component.

7.3.1 Integration of controller design with evaporator design methodology

As it has been shown in the literature review, the control design of ORC systems is most often designed as a reactive measure to system design and components that have already been defined based on other considerations. The methods presented in this work however

allow for the simultaneous optimization of the control system with the component design based on its thermal inertia.

Control strategies and control algorithms can be integrated into the models in order to try the best geometry or type of heat exchanger for candidate control strategies. This can be integrated in parallel or as a final step in the methodology as proposed in Figure 7-2.

For instance, models of by-pass valves in the hot side of the evaporator can be incorporated and its impact on the response time measured. Another example is the integration of a detailed model of the expander with control measures such as variable inlet guide vanes in order to obtain an adjusted dynamic response time accounting for such actions.

Furthermore, the customization of the dynamics opens a lot of potential for advanced control architectures. For instance, the customized thermal inertia of the evaporator based on the response time maps can be integrated as a variable into the predictive model of a Model Predictive Control architecture and its value optimized before-hand in order to obtain the best performance.

7.3.2 Multi-objective optimization

The objective of this thesis has been to show in the most general and inclusive approach a different method to handle the challenges imposed in ORCs by thermal power fluctuations and allow for improvement performance (e.g. the possibility in practice of implementing direct evaporation). Evaporators have been dimensioned for certain case studies. However, it must be said that there is no geometry or dimensioning of an ORC evaporator that works the best for any situation. That is why the methodology does not impose a single design and allows for the freedom of customization based on the preferences of the designer.

In the future work, this customization can be fully enhanced by the implementation of a multi-objective optimization in the methodology in order to get the one single evaporator design that best matches certain requirements. In this context the response time maps can

be used as one of the inputs of the optimization.

The objectives for the optimization can be, among others, a certain range of evaporator response time, the minimization of the weight and volume of the component and the reduction of the cost. Other more complex objectives that relate to the thermodynamic cycle can be further included. For example, the evaporator pressure which has an impact on the thermal efficiency. This is because, for instance, the optimal evaporator pressure regarding thermal efficiency might not be optimal in terms of evaporator cost and volume for a given range of response times.

As it is standard in multi-objective optimization methods, different weights should be given to the different optimization criteria depending on the relative importance of each other. The optimization problem can be solved by genetic algorithms or other methods.

7.4 Final assessment

This dissertation has proven the original thesis statement regarding the possibility to effectively customize the thermal inertia of ORC evaporators for improved dynamic performance within certain boundaries.

This has been done by applying a novel methodology of ORC evaporator design developed and proposed by this thesis and summarized in Figure 7-2. Results based on simulations of a validated dynamic model have shown, for instance in the case study of Chapter 6, that a direct heat exchange evaporator can be sized in an unconventional way to dampen to a safe range fluctuations faster than 0.003 Hz and 20 kW of amplitude, and that such a design is a compromise of safe dynamic performance while having the potential of increasing the thermal efficiency to a 3.4% higher value and decreasing the heat exchangers volume and weight by 88% and 70% respectively due to the use of direct evaporation instead of indirect.

There is a limit in how much the dynamic response can be customized due to physical constraints, and the improvements can vary case by case. Nevertheless, the methodology

developed and contributed by this thesis is not bound by any specific case and can be applied to any situation of ORC systems recovering waste heat.

APPENDIX A

Calculation of geometry of heat exchangers

Below the calculation for each type of heat exchangers considered in this work. Please note that the heat transfer areas and volumes for each discretization cell in the finite volume models are equal to the total values divided by the number of discretization cells. The main output parameters of the model for all heat exchanger types are:

Main output parameters (all heat exchangers):

Hydraulic diameter, inside, $D_{h,int}$

Hydraulic diameter, outside $D_{h,ext}$

Heat transfer area, inside fluid, $A_{ht,int}$

Heat transfer area, outside fluid, $A_{ht,ext}$

Material volume, V_{solid}

Internal fluid volume, V_{int}

External fluid volume, V_{ext}

Heat exchanger volume, V_{hx}

Fin and tube heat exchanger

Please refer to Figure 5-3 for a diagram of the heat exchanger main geometric parameters. The notation for the input dimensions to the model, as well as additional calculated parameters are presented below:

Input parameters:

Tube inner diameter, D

Tube length, L

Number of tubes banks, N_{banks}

Number of tubes per bank, $N_{tubes/bank}$

Tube thickness th_w

Clearance between tube banks cl_{banks}

Clearance between tubes per bank $cl_{tube/banks}$

Fin height ht_{fin}

Fin thickness th_{fin}

Fin pitch pt_{fin}

Additional calculated parameters:

Tube outer diameter, D_o

Total number of tubes, N_{tubes}

Number of fins, N_{fins}

External surface area of tubes, $A_{tube,ext}$

Surface area of fins, A_{fins}

Volume of tube wall, V_w

Volume of fins, V_{fins}

The calculations are presented below:

Additional calculated parameters:

$$D_o = D + 2 \cdot th_w \quad (A-1)$$

$$N_{tubes} = N_{banks} \cdot N_{tubes/bank} \quad (A-2)$$

$$N_{fins} = integer\{(L \cdot N_{tubes})/pt_{fin}\}+1 \quad (A-3)$$

$$A_{tube,ext} = (\pi \cdot D_0 \cdot L - \pi \cdot D_0 \cdot th_{fin} \cdot N_{fins}) \cdot N_{tubes} \quad (A-4)$$

$$A_{fins} = 2 \cdot \left[\frac{\pi \cdot (2 \cdot ht_{fin} + D_o)^2}{4} - \frac{\pi \cdot D_o^2}{4} \right] \cdot N_{fins} \quad (A-5)$$

$$V_w = \left(\frac{\pi \cdot D_o^2}{4} - \frac{\pi \cdot D^2}{4} \right) \cdot L \cdot N_{tubes} \quad (A-6)$$

$$V_{fins} = \left[\frac{\pi \cdot (2 \cdot ht_{fin} + D_o)^2}{4} - \frac{\pi \cdot D_o^2}{4} \right] \cdot th_{fin} \cdot N_{fins} \quad (A-7)$$

Hydraulic diameters:

$$D_{h,int} = D \quad (A-8)$$

$$D_{h,ext} = \frac{4 \cdot V_{ext}}{A_{ht,ext}} \quad (A-9)$$

Heat transfer areas:

$$A_{ht,int} = \pi \cdot D \cdot L \cdot N_{tubes} \quad (A-10)$$

$$A_{ht,ext} = A_{tube,ext} + A_{fins} \quad (A-11)$$

Volumes:

$$V_{solid} = V_w + V_{fins} \quad (A-12)$$

$$V_{int} = \frac{\pi \cdot D^2}{4} \cdot L \cdot N_{tubes} \quad (A-13)$$

$$V_{ext} = V_{hx} - V_{fins} - V_w - V_{int} \quad (A-14)$$

$$V_{hx} = [L \cdot N_{tubes}] \cdot [N_{banks} \cdot (cl_{banks} + D_0)] \cdot [N_{btubes/bank} \cdot (cl_{tube/banks} + D_0)] \quad (A-15)$$

Louver fin multi-port heat exchanger

Please refer to Figure 5-4 for a diagram of the heat exchanger main geometric parameters. The notation for the input dimensions to the model, as well as additional calculated parameters are presented below:

Input parameters:

Port inner diameter, D

Tube length, L

Number of flat tubes, N_{tubes}

Number of ports per tube, $N_{ports/tube}$

Port wall thickness th_w

Clearance between ports cl_{ports}

Clearance between tubes cl_{tubes}

Fin thickness th_{fin}

Fin pitch pt_{fin}

Louver angle, β_{louver}

Louver pitch, pt_{louver}

Additional calculated parameters:

Tube thickness, th_{tube}

Tube width, w_{tube}

Total number of ports, N_{ports}

Number of fins, N_{fins}

Number of louvers, $N_{louvers}$

Fin length, L_{fins}

Louver length, L_{louver}

External surface area of tubes, $A_{tube,ext}$

Surface area of fins, A_{fins}

Louver cutting area, A_{louver}

Volume of tube wall, V_w

Volume of fins, V_{fins}

The calculations are presented below:

Additional calculated parameters:

$$th_{tube} = D + 2 \cdot th_w \quad (A-16)$$

$$wd_{tube} = (D + cl_{ports}) \cdot N_{ports/tube} \quad (A-17)$$

$$N_{ports} = N_{tubes} \cdot N_{port/tube} \quad (A-18)$$

$$N_{fins} = integer\{L/pt_{fin}\} + 1 \quad (A-19)$$

$$N_{louvers} = \text{integer}\{(wd_{tube})/pt_{louver}\} + 1 \quad (\text{A-20})$$

$$L_{fins} = \sqrt{cl_{tubes}^2 + pt_{fin}^2} \quad (\text{A-21})$$

$$L_{louver} = 0.8 \cdot L_{fins} \quad (\text{A-22})$$

$$A_{tube,ext} = 2 \cdot \left[\left((L - th_{fin} \cdot N_{fins}) \cdot wd_{tube} \right) + (L \cdot th_{tube}) \right] \cdot N_{tubes} \quad (\text{A-23})$$

$$A_{fins} = 2 \cdot (wd_{tube} + th_{fin}) \cdot L_{fins} \cdot N_{fins} \cdot N_{tubes} \quad (\text{A-24})$$

$$A_{louver} = 2 \cdot (L_{louver} \cdot N_{fins} \cdot N_{louvers} \cdot th_{fin}) \cdot N_{tubes} \quad (\text{A-25})$$

$$V_w = N_{tubes} \cdot th_{tube} \cdot wd_{tube} \cdot L - V_{int} \quad (\text{A-26})$$

$$V_{fins} = (N_{tubes} + 1) \cdot (wd_{tube} \cdot L_{fins} \cdot N_{fins}) \quad (\text{A-27})$$

Hydraulic diameters:

$$D_{h,int} = D \quad (\text{A-28})$$

$$D_{h,ext} = \frac{4 \cdot V_{ext}}{A_{ht,ext}} \quad (\text{A-29})$$

Heat transfer areas:

$$A_{ht,int} = \pi \cdot D \cdot L \cdot N_{tubes} \quad (A-30)$$

$$A_{ht,ext} = A_{tube,ext} + A_{fins} + A_{louver} \quad (A-31)$$

Volumes:

$$V_{solid} = V_w + V_{fins} \quad (A-32)$$

$$V_{int} = \frac{\pi \cdot D^2}{4} \cdot L \cdot N_{ports} \quad (A-33)$$

$$V_{ext} = V_{hx} - V_{fins} - V_w - V_{int} \quad (A-34)$$

$$V_{hx} = L \cdot wd_{tube} \cdot [cl_{tubes} \cdot (N_{tubes} + 1) + (N_{tubes} \cdot th_{tube})] \quad (A-35)$$

Shell and tube heat exchanger

Please refer to Figure 6-2 for a diagram of the heat exchanger main geometric parameters. The notation for the input dimensions to the model, as well as additional calculated parameters are presented below:

Input parameters:

Tube inner diameter, D

Tube length, L

Number of tubes banks, N_{tubes}

Tube thickness th_w

Shell inner diameter, D_{shell}

Clearance between tubes cl_{tubes}

Additional calculated parameters:

Tube outer diameter, D_o

External surface area of tubes, $A_{tube,ext}$

Volume of tube wall, V_w

The calculations are presented below:

Additional calculated parameters:

$$D_o = D + 2 \cdot th_w \quad (A-36)$$

$$A_{tube,ext} = (\pi \cdot D_o \cdot L) \cdot N_{tubes} \quad (A-37)$$

$$V_w = \left(\frac{\pi \cdot D_o^2}{4} \right) \cdot L \cdot N_{tubes} \quad (A-38)$$

Hydraulic diameters:

$$D_{h,int} = D \quad (A-39)$$

$$D_{h,ext} = \frac{4 \cdot V_{ext}}{A_{ht,ext}} \quad (A-40)$$

Heat transfer areas:

$$A_{ht,int} = \pi \cdot D \cdot L \cdot N_{tubes} \quad (A-41)$$

$$A_{ht,ext} = A_{tube,ext} \quad (A-42)$$

Volumes:

$$V_{solid} = V_w \quad (A-43)$$

$$V_{int} = \frac{\pi \cdot D^2}{4} \cdot L \cdot N_{tubes} \quad (A-44)$$

$$V_{ext} = V_{hx} - V_w - V_{int} \quad (A-45)$$

$$V_{hx} = \frac{\pi \cdot D_{shell}^2}{4} \cdot L \quad (A-46)$$

Plate heat exchanger

Please refer to Figure 6-2 for a diagram of the heat exchanger main geometric parameters. The notation for the input dimensions to the model, as well as additional calculated parameters are presented below:

Input parameters:

Plate length, L

Number of plates, N_{plates}

Plate width, wd_{plate}

Wall thickness th_w

Plate pattern angle, β_{plate}

Plate pattern amplitude, a_{plate}

Plate pattern wave length, λ_{plate}

Additional calculated parameters:

Plate wave number, κ_{plate}

Area expansion factor, EF

The calculations are presented below:

Additional calculated parameters:

$$\kappa_{plate} = (2 \cdot \pi \cdot a_{plate}) / \lambda_{plate} \quad (A-47)$$

$$EF = \frac{1}{6} \cdot \left[1 + \sqrt{1 + \kappa_{plate}^2} + 4 \cdot \sqrt{1 + \frac{\kappa_{plate}^2}{2}} \right] \quad (A-48)$$

Hydraulic diameters:

$$D_{h,int} = \frac{4 \cdot a_{plate}}{EF} \quad (A-49)$$

$$D_{h,ext} = \frac{4 \cdot a_{plate}}{EF} \quad (A-50)$$

Heat transfer areas:

$$A_{ht,int} = (N_{plates} - 2) \cdot wd_{plate} \cdot L \cdot EF \quad (A-51)$$

$$A_{ht,ext} = (N_{plates} - 2) \cdot wd_{plate} \cdot L \cdot EF \quad (A-52)$$

Volumes:

$$V_{solid} = wd_{plate} \cdot L \cdot th_w \cdot N_{plates} \cdot EF \quad (A-53)$$

$$V_{int} = wd_{plate} \cdot 2 \cdot a_{plate} \quad (A-54)$$

$$V_{ext} = wd_{plate} \cdot 2 \cdot a_{plate} \quad (A-55)$$

$$V_{hx} = V_{int} + V_{ext} + V_{solid} \quad (A-56)$$

APPENDIX B

Heat transfer correlations

The heat transfer correlations are expressed in terms of the Nusselt number Nu defined as:

$$Nu = \frac{\theta \cdot D_h}{k} \quad (B-1)$$

k refers to the thermal conductivity of the respective fluid, D_h is the characteristic length such as a hydraulic diameter and θ is the heat transfer coefficient.

The correlations are usually computed in terms of Re and Pr , which are the Reynolds and Prandtl numbers respectively.

The Reynolds number is defined as:

$$Re = \frac{\rho \cdot u \cdot D_h}{\mu} \quad (B-2)$$

ρ is the density of the fluid, u is the velocity of the fluid, D_h is the characteristic length such as a hydraulic diameter and μ is the dynamic viscosity of the fluid.

The Prandtl number is defined as:

$$Pr = \frac{c_p \cdot \mu}{k} \quad (B-3)$$

c_p is the specific heat of the fluid.

Other relevant quantities are the Darcy friction factor ζ calculated as:

$$\zeta = (0.79 \cdot \ln(Re) - 1.64)^{-2} \quad (\text{B-4})$$

The Boiling coefficient Bo calculated as:

$$Bo = \frac{\bar{q}}{G \cdot \Delta H_{vap}} \quad (\text{B-5})$$

And the Froude number Fr defined as:

$$Fr = \frac{G}{\rho_l^2 g D} \quad (\text{B-6})$$

Where $\bar{q}$ is the mean heat flux, ΔH_{vap} is the enthalpy of vaporization of the fluid, G is the mass flow per unit area per unit time, ρ_l is the density of saturated liquid, g is the gravitational constant and D is the tube diameter

(a) Working fluid heat transfer correlations

One-phase

Gnielinski correlation ($2300 < Re < 100,000$) [136]

$$Nu = \frac{\left(\frac{\zeta}{8}\right) (Re - 1000) Pr}{1 + 12.7 \sqrt{\frac{\zeta}{8}} Pr^{2/3}} \quad (\text{B-7})$$

Dittus-Boelter correlation ($Re > 100,000$) [137]

$$Nu = 0.023 Re^{4/5} Pr^{1/3} \quad (B-8)$$

Two-phase

The Shah correlation [138] depends on various dimensionless parameters including the Froude number Fr and the Boiling number Bo . For a low Froude number and large Boiling coefficient, the Nusselt number is calculated as:

$$Nu = \max(Nu_b, Nu_k) \quad (B-9)$$

$$Nu_b = 230 Bo^{1/2} 0.023 Re^{4/5} Pr^{0.4} \quad (B-10)$$

$$Nu_k = \left(\frac{1.8}{N^{0.8}} \right) 0.023 Re^{4/5} Pr^{0.4} \quad (B-11)$$

The coefficient N is calculated as

$$N = 0.38 Fr^{-0.3} \left(\frac{1}{x} - 1 \right)^{0.8} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \quad (B-12)$$

x is the vapor mass fraction and ρ_g and ρ_l are the saturation densities of gas and liquid respectively.

(b) Exhaust heat transfer correlations

Shell and tube heat exchanger – Shell side of shell and tube heat exchanger according to VDI Heat Atlas [141]

$$Nu = 0.3 + \sqrt{Nu_{lam}^2 + Nu_{turb}^2} \quad (B-13)$$

Nu_{lam} and Nu_{turb} are calculated as:

$$Nu_{lam} = 0.664 Re^{1/2} Pr^{1/3} \quad (B-14)$$

$$Nu_{turb} = \frac{0.037 Re^{0.8} Pr}{1 + 2.443 Re^{-0.1} (Pr^{2/3} - 1)} \quad (B-15)$$

Fin and tube heat exchanger – Perpendicular flow outside a bank of finned tubes according to VDI Heat Atlas [139]

$$Nu = 0.22 Re^{0.6} Pr^{1/3} \left(\frac{A}{A_{to}} \right)^{-0.15} \quad (B-16)$$

A is the total outer surface and A_{to} is the surface of bare tube without fins

Louver fin multi-port heat exchanger – Flow through louver fin geometry by Chang and Wang [140]

$$Nu = j_c \cdot Re_{LP} \cdot Pr^{0.33} \quad (B-17)$$

Re_{LP} is the Reynolds number calculated with the louver pitch as the characteristic length. The louver pitch is also used as the characteristic length of the Nusselt number. The Colburn factor j_c is calculated as:

$$j_c = Re_{LP}^{-0.49} \cdot \left(\frac{L_\theta}{90} \right)^{0.27} \cdot \left(\frac{F_p}{L_p} \right)^{-0.14} \cdot \left(\frac{F_l}{L_p} \right)^{-0.29} \cdot \left(\frac{T_d}{L_p} \right)^{-0.23} \cdot \left(\frac{L_l}{L_p} \right)^{0.68} \cdot \left(\frac{T_p}{L_p} \right)^{-0.28} \cdot \left(\frac{\delta_f}{L_p} \right)^{-0.05} \quad (B-18)$$

L_θ is the louver angle, L_p is the louver pitch, L_l is the louver length, F_p is the fin pitch, F_l is the fin length, T_d is the tube depth, T_p is the tube pitch and δ_f is the fin thickness.

(c) Thermal oil heat transfer correlations

Plate heat exchanger – One phase flow in plate heat exchanger according to VDI Heat Atlas [142]

$$Nu = 0.122Pr^{1/3} \left(\frac{\mu}{\mu_w} \right)^{1/6} [2Hg \sin(2\varphi)]^{0.374} \quad (\text{B-19})$$

μ and μ_w are the viscosity of the fluid at the bulk and wall positions respectively. φ is the plate angle and Hg is the Hagen number calculated as:

$$Hg = \frac{\rho \Delta p d_h^3}{\mu^2 L_p} \quad (\text{B-20})$$

APPENDIX C

Pressure drop correlations

The pressure drops Δp are calculated with the standard equation that uses the Darcy friction factor f_D [143]:

$$\frac{\Delta p}{L} = f_D \frac{\rho \cdot v^2}{2D} \quad (C-1)$$

L is the flow path length, v is the mean flow velocity, D is the path hydraulic diameter and ρ is the fluid density. The Darcy friction factor depends on the geometry and flow path. Below the specific correlations:

(a) Working fluid pressure drop

Laminar flow - Poiseuille's law [144]:

$$f_D = \frac{64}{Re} \quad (C-2)$$

Turbulent flow - Swamee Jain correlation [145]:

$$\frac{1}{\sqrt{f_D}} = -2 \cdot \log \left(\frac{\epsilon}{3.7D} + \frac{5.74}{Re^{0.9}} \right) \quad (C-3)$$

The tube roughness ϵ is assumed to be 0.015 mm throughout.

(b) Exhaust gas pressure drop

Fin and tube heat exchanger – Haaf correlation [146]

$$f_D = 10.5 \cdot Re^{-\frac{1}{3}} \cdot \left(\frac{D_h}{\rho}\right)^{0.6} \quad (C-4)$$

D_h is the hydraulic diameter of the flow

Louver fin multi-port heat exchanger – Kim and Bullard [147]:

$$j_c = Re_{LP}^{-0.781} \cdot \left(\frac{L_\theta}{90}\right)^{0.444} \cdot \left(\frac{F_p}{L_p}\right)^{-1.682} \cdot \left(\frac{F_l}{L_p}\right)^{-1.22} \cdot \left(\frac{T_d}{L_p}\right)^{0.818} \cdot \left(\frac{L_l}{L_p}\right)^{1.97} \quad (C-5)$$

Re_{LP} is the Reynolds number calculated with the louver pitch as the characteristic length, L_θ is the louver angle, L_p is the louver pitch, L_l is the louver length, F_p is the fin pitch, F_l is the fin length and T_d is the tube depth.

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