Design enhancement of an adiabatic LiBr-H₂O absorber through a droplet flow optimization

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Design enhancement of an adiabatic LiBr-H₂O absorber through a droplet flow optimization

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School of Mechanical and Aerospace Engineering

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

2018

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.

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Alessandro Romagnoli

Authorship Attribution Statement

This thesis contains material from 3 papers published in the following peer-reviewed journal(s) where I was the first and/or corresponding author.

Chapter 2 is published as: Cola F, Romagnoli A, Hey J. An evaluation of the technologies for heat recovery to meet onsite cooling demands. *Energy Convers Manag* 2016;121:174–85. doi:10.1016/j.enconman.2016.05.021.

The contributions of the co-authors are as follows:

- Dr. Hey and Prof. Romagnoli assisted in the choice of the explored technologies in the study.
- I devised the boundary conditions of the study which with input by Prof. Romagnoli
- Prof. Romagnoli assisted in the development of the scoring logic and visual representation ideas.
- All coding and image editing was performed by me. Dr. Hey and Prof. Romagnoli reviewed the results and provided suggestions for model improvement

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- Dr. Hey and Prof. Romagnoli assisted with the design of the experimental campaign
- Prof. Romagnoli provided further assistance in the hardware sourcing
- I conducted the experiments in the Heat Transfer Lab in MAE school of NTU
- Results were collected by me and analysed through a code I developed. Dr. Hey and Prof. Romagnoli provided insight for the results interpretation and discussion
- The manuscript was drafted by and reviewed by Dr. Hey and Prof. Romagnoli who provided further input for its revision

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- Dr. Hey and Prof. Romagnoli provided further assistance in the hardware sourcing for the data logging system
- I conducted the experiments in the Heat Transfer Lab in MAE school of NTU

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interpretation and discussion

- The manuscript was drafted by and reviewed by Dr. Hey and Prof. Romagnoli who provided further input for its revision

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Summary

The increase of cooling demands is a widespread phenomenon that poses the problem of identifying new efficient ways to provide the necessary input energy. Waste heat is available in large quantities, and its recovery is a challenging problem. Additionally, small-scale applications are becoming more common in fields such as electronics and/or automotive. This work aims at providing a solution for the development of a small-scale waste *heat-to-cool* conversion system.

To achieve this, a first comparison of the potential conversion systems was performed. Parameters of efficiency, cost and size were considered both separately and combined in a comprehensive performance index. From this analysis, the H₂O-LiBr absorption chiller emerged as the most suitable technology for the development of a small-scale system.

Further investigation of the selected technology identified the absorber as the most critical component preventing the system downsizing. A critical review of the current state of the art in the absorber design and operation strategies led to the proposition of a pin-finned, adiabatic absorber, combined with a droplet flow regime.

In the next phase the proposed design was optimized. Firstly, an analytical model was developed to accurately describe the geometrical domain of a droplet forming on a solid surface, a phase which was found to be beneficial to the absorption process. The model was then validated with experiments and used to optimize the pin shape.

Secondly, the absorption process was analyzed. An analytical heat and mass transfer model of the phases of a falling droplet was developed and an experimental setup, replicating the operation of the absorber, was developed to validate the proposed

model. The validation process led to the identification of the best pin and distribution orifice sizes, completing the absorber design optimization.

Finally, the performance improvement of the optimized absorber design was evaluated with respect to a standard absorption chiller configuration.

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Nomenclature

<i>Variable name</i>	<i>Description</i>	<i>Unit</i>
A	Orifice cross section area	m^2
A_{leg}	Semiconductor (leg) section	m^2
A_{hx}	Heat transfer area	m^2
A_{norm}	Normalized area	
A_{surf}	Interface surface area	m^2
B	Curvature radius (non-dimensional)	
Bo_l	Boiling number	
$Bo = \rho g r^2 / \sigma$	Bond number	
C	Concentration	mol/m^3
C_d	Discharge coefficient	
C_{norm}	Normalizing factor	m
Co	Cost	$\text{\$}$
COP_{cool}	COP of cooling cycle	
d	Diameter	m
D	Mass diffusivity	m^2/s
EK	Kinetic Energy	J
ES	Surface energy	J
f	Friction factor	
f_m	Material factor	
f_p	Pressure factor	
Fa	Fang number	
Fr	Froude number	
g	Gravitational acceleration	m/s^2
G	Mass flux	$\text{kg}/\text{m}^2\text{s}$
$Ga = \rho \sigma_s^3 / \mu^4 g$	Galileo number	
h	Enthalpy	J/kg
H	Static head	m
h_c	Heat transfer coefficient	$\text{W}/\text{m}^2\text{K}$
H_{lv}	Latent heat of vaporization	J/kg
I	Current	A
I_{tech}	Performance index	
k	Thermal conductivity	W/mK
K	Thermal conductance	W/K
K_{form}	Mass transfer coefficient	m/s
L_{leg}	Semiconductor (leg) length	m
L	Pin length	m
$\dot{m}$	Mass flow rate	kg/s
M	Mass	kg
N_{or}	Number of orifices	
Nu	Nusselt number	

$Oh = \mu/(\rho r \sigma_s)^{1/2}$	Ohnesorge number	
P	Pressure	Pa
Pr	Prandtl number	
q	Heat flux	W/m ²
Q	Volume flow rate	m ³ /s
$\dot{Q}$	Heat exchanged	W
r	Radius	m
R	Radius (non-dimensional)	
R_{el}	Electrical resistance	Ω
Re	Reynolds number	
S	Arc length (non-dimensional)	
t	Time	s
T	Temperature	K
u	Speed	m/s
V	Volume (non-dimensional)	
w	Weighting factor	
$\dot{W}$	Power	W
W_{pin}	Pin width	m
$We = \rho u^2 / \sigma_s$	Weber number	
x	Vapor title	
X	LiBr mass fraction	
z	Height	m
Z	Height (non-dimensional)	

Greek Symbols

μ	Dynamic viscosity	Pa s
δ	Deviation	
α_{sb}	Seebeck coefficient	V/K
α	Thermal diffusivity	m ² /s
α_g	Vapor void fraction	
β	Contact point angle	rad
$\varepsilon = \sigma_s \rho g$	Capillary length	m
Γ	Linear mass flow rate	kg/ms
φ	Distance between droplets	m
$\eta_{generator}$	Generator efficiency	
η_{is}	Isentropic efficiency	
θ	Angle	rad
λ	Taylor wavelength	m
ν	Kinematic viscosity	m ² /s
ρ	Density	kg/m ³
σ	Electrical conductivity	S/m
σ_s	Surface tension	N/m
Φ_{ret}	Fraction of liquid retained	

Subscripts

<i>a</i>	Half sphere
<i>abs</i>	Absorbed
<i>amb</i>	Ambient
<i>bulk</i>	Bulk
<i>C</i>	Cold side of TEG
<i>cds</i>	Condenser
<i>comp</i>	Compressor
<i>cool</i>	Cooling at evaporator
<i>cr</i>	Critical
<i>d</i>	Droplet
<i>Dh</i>	Hydraulic diameter
<i>disc</i>	Liquid disc
<i>end</i>	End
<i>eq</i>	Equilibrium
<i>exp</i>	Experimental
<i>form</i>	Formation
<i>g</i>	Gas
<i>gen</i>	Heat source
<i>H</i>	Hot side of TEG
<i>high</i>	Discharge
<i>i</i>	Initial
<i>int</i>	Interface
<i>is</i>	Isentropic
<i>l</i>	Liquid
<i>L</i>	Surrounding fluid
<i>low</i>	Suction
<i>max</i>	Maximum
<i>mean</i>	Mean
<i>min</i>	Minimum
<i>o</i>	Boundary
<i>or</i>	Orifice
<i>orc</i>	Organic Rankine Cycle fluid
<i>sol</i>	Solution
<i>tot</i>	Total

Abbreviations

<i>ABS</i>	Absorption chiller
<i>COP</i>	Coefficient of performance
<i>CPF</i>	Cooling power flux
<i>ORC</i>	Organic Rankine cycle
<i>SIC</i>	Specific investment cost
<i>TEG</i>	Thermoelectric generator
<i>VCC</i>	Vapor compression chiller

1 Introduction

The worldwide increase of cooling demand is a phenomenon that has been occurring in the past years and will represent an important issue for the future energy scenarios. Isaac and Van Vuuren [1] estimated that the global energy demand for cooling will increase more than tenfold by 2050 and by 2100 it will surpass the total demand for heating.

This trend will be driven by developing countries. Sivak [2] showed that among the top 30 areas in terms of future potential cooling demand, 28 are located in developing countries. These are areas where the use of cooling systems (e.g. air conditioning) is not yet widespread but is expected to increase soon, as the cost of air conditioning will become more affordable for a larger part of the population.

1.1 Industrial waste heat

The growing demand for cooling carries the problem of sourcing the necessary primary energy to drive the refrigeration cycles. Church [3] proposed a ranking scheme based on the assumption that the maximum effectiveness is obtained for low-grade input energy, as is the case of polluting agents or waste. According to this ranking, pollutants like solid waste, waste oil and sawmill residue are considered of the utmost priority, followed by waste heat from commercial and industrial environments, which scored second.

According to Lu et al. [4] between 10% and 50% of total fuel consumption in industries is converted into waste heat which could be recovered. However, this potential for waste heat recovery might not be entirely exploitable; Brückner et al. [5] in fact distinguished between the theoretical potential, which is the fraction of heat that can physically be recovered, the technical potential, which is the heat that can be recovered with the current technologies, and finally the economic potential, which is the heat that can be recovered with some monetary benefit.

The result of this distinction is that only a small part of the total available waste heat can be recovered in practice. Lu et al. [4] showed that with the current technologies, only 7%-13% of the total waste heat produced can be recovered in the cement, steel and glass industry in China. However, if efficiencies were to be improved and ideally approach Carnot efficiency, the percentage could increase up to 40%-57%. The low waste heat recovery real potential is due to the low grade (i.e. temperature) at which heat is discharged. As the heat temperature increases, recovery becomes interesting and different ways of recovering it are available. A summary of the average exhaust gas temperatures and process temperatures for many industries can be found in [5].

1.2 The size problem

Recent progress made in sectors like electronics and automotive has fueled a new type of demand for cooling, where the size of the refrigerating device is a critical aspect to consider. Furthermore, as the penetration of domestic air conditioning is on the rise, more strict requirements on the system size footprint are to be expected. As a result, recent research has started focusing on the downsizing and even miniaturization of already available cooling technologies.

The easiest solution is usually to employ thermoelectric coolers, given their outstanding properties in cooling power densities. An example is given by Faraji et al. [6], who developed a small thermoelectric chiller, capable of cooling 430 ml of water with a coefficient of performance (COP) between 0.2 and 0.8. They concluded that improvement is needed not only on the thermoelectric material but also on the heat exchanger efficiency, as it affects the size of the system.

The need of miniaturization of absorption chillers is one of the limiting factors to this technology progress and market penetration as explained by Keppler [7]. This is in fact a necessary condition to step inside new markets such as the residential one. The author however

concedes that lately more effort is being spent in the direction of the chiller miniaturization. A concept of a miniaturized absorption chiller was analyzed numerically by Kim et al. [8], who concluded that it is theoretically possible to fit a 100 W chiller into a 150x150x100 *mm* envelope. A good attempt at creating a micro-scale absorption chiller was carried out by Hu and Chao [9], who used microfabrication techniques to develop the evaporator, the condenser (both with dimensions 28x30.5x40.5 *mm*), and the expansion channel of the absorption chiller. Absorber and generator were not miniaturized in the study. Results showed a measured COP ranging from 0.465 to 0.479 for an evaporation temperature of 11-19 °C and a condensation temperature of 45 °C. The device was later used by Adhami et al. [10] to model a portable micro trigeneration system by coupling the absorption chiller with a micro gas turbine. The model returned power productions below 10 W and cooling power below 5 W, for a total estimated weight of 13.21 kg.

A complete device was developed by Determan and Garimella [11], who successfully built and tested a monolithic and modular water-ammonia absorption chiller for miniaturized applications, with a size of 200x200x34 *mm* and 7 kg of weight for 300 W of cooling. Testing showed that COP was in the range between 0.25 and 0.45 for different heat inputs and cooling water temperatures at the absorber, but the authors suggested that the performance can be improved by reducing cross heat transfer between components and improving the desorber and rectifier design.

The system was later scaled up to a 7 kW unit for residential use. Testing results performed in AHRI conditions were presented by Staedter and Garimella [12]. The device, powered by natural gas, operated with a cooling COP of 0.44, although the poor operation of the evaporator and poor vapor rectification prevented further improvement in the efficiency.

Another complete system was presented by Franchini et al. [13]. The chiller was developed with a two-shell layout and spiral heat exchangers for a total area of 21 m². The system was originally designed for a 5 kW cooling power, but subsequent testing showed a measured cooling power of 3.25 kW for a COP of 0.358.

A membrane-based micro absorber was modeled by Venegas et al. [14] both in adiabatic and non-adiabatic conditions, based on a model previously developed [15]. Thanks to the solution continuous cooling the non-adiabatic condition showed little sensitivity to the inlet solution temperature. However the adiabatic condition could provide comparable high power density once the channel main dimensions were optimized. Specifically, channel width of 0.3-0.4 mm, height of 0.15 mm and length smaller than 0.5 cm could provide the highest power density in the order of 1 MW/m³.

Vapor compression chillers were also considered for miniaturization and downsizing. Yuan et al. [16] managed to miniaturize a vapor compression chiller for personal cooling. Their systems outline size is 190x190x100 mm with an overall weight of 2.75 kg. Testing of the system showed that a COP of 1.62 can be achieved, and the authors claimed that further reduction in size and weight is possible. A similar system was developed by Yu-Ting et al. [17]. An ad-hoc screw compressor was developed for the purpose; the resulting system weighted 2.85 kg and produced a COP of 2.3 when operating between 21.2 and 40 °C.

A hybrid solution is finally proposed by Gordon et al. [18], who studied a cascade system composed by a thermoelectric cooler and an adsorption chiller, which thanks to the absence of moving parts is believed to be scalable to very small sizes and thus applicable in various fields. Results of the simulations showed that COP higher than 0.9 can be achieved, although no actual device was built and tested to confirm such results.

1.3 Objectives and thesis structure

In light of the rapid increase in cooling demand forecasted for the future years, combined with a growing interest in small-scale applications, and the availability of low-grade waste heat to be recovered, this work aims at providing a solution for the development of a small-scale, waste *heat-to-cool* conversion system. To address this problem, a series of intermediate objectives were established:

- 1 The choice of the most suitable technology for the development of the small-scale conversion system is done by comparing the main *heat-to-cool* conversion systems and proposing a scoring method (Chapter 2).
- 2 The most critical component preventing the downsizing of the selected technology, the absorber of an absorption chiller, is identified, and a new design is proposed to improve its performance and reduce its footprint (Chapter 3).
- 3 Optimization of the proposed absorber design is performed and validated. This is firstly done on the geometrical and fluid domain characteristics of the proposed design (Chapter 4). Then, a heat and mass transfer model is developed to further optimize the design and maximize the absorption performance of the proposed design. (Chapter 5).
- 4 The improvement offered by the optimized design on the baseline configuration of the *heat-to-cool* conversion system is evaluated (Chapter 6).

2 Selection process for heat-to-cool conversion technology

In this Chapter, the main technologies for heat-to-cool conversion are reviewed, and their main features described. A study for a correct selection of the best candidate for a small-scale refrigeration device is then presented [19]. The constraints and boundary conditions for the selection are set, and the results are then presented into two forms. First, the results specific to each parameter investigated are presented. Then, a scoring method is introduced to synthesize the previous results into a single performance index, to finalize the technology selection.

2.1 Review of heat to cool conversion technologies

Converting waste heat into cooling power is a process that can essentially be done in two ways: directly, using thermally activated chillers, or indirectly, through a first conversion of the heat into electricity, which is then used to drive vapor compression chillers or thermoelectric coolers.

By looking at the problem from the waste heat perspective, many researchers compared different solutions to use this heat to provide upgraded heat, cooling, and power. Oluleye et al. [20] proposed a method to establish the possible increase in the efficiency of process sites (specifically petroleum refineries) when waste heat is recovered. Organic Rankine Cycles (ORC) and absorption chillers/ heat pumps were considered for this study. Results showed that 9% efficiency increase can be achieved with an ORC for a specific site, while for cooling the increase was of only 0.2%, caused by a limited demand. If though the demand were to be considered infinite, as could be the case of a site connected to a district cooling grid, the increase was calculated to be 13.5%. For what concerns instead heating and cooling only, Brückner [5] concluded that absorption chillers are profitable for the industry clients when a high utilization factor is assured (at least 6500 hours of operation per year), while in case of absorption and mechanical heat pumps a lower utilization factor is enough to make the system profitable.

Some general reviews on the topic of waste heat recovery are available. Ammar et al. [21] listed and reviewed some of the main technologies that use waste heat for power generation, heating and cooling. Among these, ORC for power generation and sorption machines for cooling are the most interesting. Thermoelectric generators and thermoelectric coolers (TEG, TEC) are of interest for their small size and ease of operation, although their efficiency drops dramatically for low temperatures of the heat source. Little and Garimella [22] compared different cycles for power, heating and cooling using low-grade heat. ORC cycle proved to be the most efficient option for power generation, while absorption cooling

outperformed the alternative of a vapor compression coupled system, keeping however comparable dimensions.

Most of the studies looking into power generation from low-grade waste heat mainly focus on ORC cycles and its variations [23], with special attention given to the trilateral and Kalina cycles [24,25]. TEG systems are cited [26] too for their exceptional low size footprint.

For the case in which low-grade heat is recovered for cooling purposes, some studies provide a comparison between thermally and mechanically driven chillers. Brown and Domanski [27] reviewed the status of all the existing cooling technologies; from their research, they concluded that trans-critical CO₂ cycles and absorption chillers may have the best chances to increase their market share in the near future, while magnetic cooling is considered the most promising technology in the long term. Same conclusions were drawn by Bansal et al. [28], who included thermoelectric cooling among the most promising technologies for the future, under the condition that important breakthroughs in the material and fabrication technology can be achieved. One main downturn of both these studies is that the driving energy (e.g. renewable energy, waste heat, etc.) was not considered.

Such information was included by Kim and Infante Ferreira [29], who chose solar energy as the driver of both thermal and mechanical chillers¹. By using standard values for the efficiency, they conducted an economic comparison between different solutions and found that absorption and adsorption chillers are the most promising options. Further review studies are available on solar driven thermal chillers [30–32]. However, despite several research papers dealing with the aforementioned cycles, there is no literature available taking industrial waste heat as the driving energy of thermally driven refrigeration cycles.

¹Thermal chillers investigated are: single and double effect absorption, adsorption, desiccant and ejector chillers. The mechanical chillers considered are: vapor compression, thermoelectric, Stirling, thermoacoustic and magnetic chillers. These can be driven by the mechanical power produced by a thermal engine of the Rankine or Stirling type, but also by an electric motor running on the electricity produced by PV panels.

2.2 Technologies description

For the selection of the most suitable technology for a small-scale heat-to-cool conversion device, a comparison was made between the following technologies, which are generally the most cited: absorption chiller (ABS), ORC coupled with vapor compression chiller (ORC-VCC), TEG coupled with vapor compression chiller (TEG-VCC). Although TEG is already known to generally have lower efficiency than an ORC cycle, this option was included as well, since TEG presents unique characteristics. Firstly, the small size can make this recovery system applicable in places where ORC cycles cannot be applied. Secondly, the great potential for efficiency increase as shown in LeBlanc [33], and LeBlanc et al. [34] could make this technology competitive in the near future.

2.2.1 Absorption chiller

As already mentioned in Section 2.1, this cycle is one of the most common and widespread methods to recover low-grade heat for cooling applications. The key concept that characterizes this cycle is the chemical compression process that substitutes the standard mechanical compressor, with a resulting minimization of electricity consumption in favor of a higher consumption of heat. In Fig. 2.1 a scheme of a water-LiBr chiller is presented.

Starting from the heater (3-7), a solution relatively poor in salt (LiBr in this study) is heated up and separated into two streams: the first one consists of pure water in a superheated state. This water is condensed into a saturated liquid state in the condenser (7-8), it expands to the low-pressure side of the cycle (8-9) and enters the evaporator (9-10), where the cooling effect takes place. Finally, water exits in the saturated or superheated vapor state and proceeds towards the absorber (10-1).

The second stream consists of a hot liquid solution strong in salt, which is firstly cooled in a solution heat exchanger (4-5), and then expanded until the low-pressure level is reached (5-6), allowing it to enter the absorber (6-1).

Fig. 2.1 Scheme of a H₂O-LiBr absorption chiller

Here the water from the evaporator is reabsorbed into the liquid solution to reestablish the original salt concentration, while the generated absorption heat is discharged to facilitate the process. The resulting liquid solution is then pumped to the high-pressure side of the cycle (1-2), where it is preheated in the solution heat exchanger (2-3) and finally enters the heater (3-7), where the cycle is reiterated.

2.2.2 Organic Rankine Cycle

One of the alternatives to the absorption cycle is represented by the use of an ORC cycle for conversion of waste heat into electricity, which will then be used to drive a standard VCC. This system works the same way as a classic water-based Rankine cycle (refer to Fig. 2.2), with the only difference that water is substituted by an organic fluid having different saturation properties that are more suitable for a low-temperature heat source.

Fig. 2.2 Scheme of an Organic Rankine Cycle

Referring to the figure, the organic fluid flows into the heater where it reaches superheated vapor state (2-4). This then expands in the turbine (4-5), to produce the electric power, until the desired discharge pressure is obtained. The fluid then condensates until the saturated liquid state is obtained (5-1) and is finally pumped back to the heater so that the cycle can be reiterated (1-2).

The selection of fluid for this system varies depending on the boundary conditions, the most important ones being the temperatures of the heat source and heat sink. However, some general rules can be followed, as suggested by Chen et al. [35] who identified fluids with high density and large latent heat as the best choice for a higher power output at the turbine.

2.2.3 Thermoelectric Generator

Thermoelectric elements have been used for a long time for applications in which space available is limited and the cost is not an issue (e.g. aerospace and military applications). A thermoelectric element is sketched in Fig. 2.3. The principle of operation is based on the relationship between Seebeck, Peltier and Thomson effects inside the thermoelectric material, three important phenomena whose relation can be summarized by its figure of merit (ZT). A

voltage is created when a p-type and n-type thermoelectric material are put in contact. The generator presents two points of contact (junctions), but the net voltage of the two junctions is zero as long as they are at the same temperature. If these are brought to different temperatures, by the effect of a heat source on one junction and a heat sink on the other one, the net voltage becomes positive (Seebeck effect), and an electric current is generated. Peltier and Thomson effects work against the generated voltage, thus in the direction of reducing the temperature difference.

Fig. 2.3 Thermoelectric generator

2.2.4 Vapor compression

The vapor compression refrigeration cycle is the most widely used technology in the refrigeration industry. A scheme of a typical cycle is presented in Fig. 2.4. Vapor comes out of the evaporator (4-1) in a superheated state to avoid the presence of liquid particles that may damage the compressor. Then, electricity coming from each of the two generation cycles presented in Chapters 3 and 4 is used to drive the compressor. After being compressed (1-2), the high-temperature, high-pressure gas flows into the condenser where it is brought to a saturated or slightly sub-cooled liquid state (2-3). The refrigerant then goes through the

expansion valve (3-4) and its pressure is reduced until the desired evaporation pressure is achieved; finally, the refrigerant enters the evaporator where the cooling effect is obtained.

Fig. 2.4 Scheme of a vapor compression refrigeration cycle

2.3 Assumptions and boundary conditions for the selection process

From the review of Section 2.1, it can be concluded that previous studies followed essentially two different paths. In the first one, the choice of the driving energy was closed, for instance waste heat, while the choice of the resulting output of conversion was not univocal; electric, heating and cooling power were compared as equally viable options. One major drawback of this approach is that, when electricity is considered as a final output, no discussion is made on how electricity could be used for cooling power production.

In the other path, the approach is reversed, leaving the choice of the input energy open and the choice of the resulting product of conversion, either being electric, heating or cooling power, closed. The only exception is given by the case investigated in [29], where only solar energy is considered for the sole purpose of cooling power production.

For the scope of this research work, we followed a third path, where both input and output energy are closed choices, so that the specific problem of *heat-to-cool conversion* can be addressed.

Following the problem statement of Chapter 1, focus is specifically put on a small-scale device for low-grade industrial waste heat conversion into cooling power.

The parameters investigated for the technology comparison are then chosen to be *cycle efficiency*, *size* and *cost*. These three parameters are first investigated separately, and then the results from this analysis are used in a scoring method that will return a single *performance index* for each conversion option, allowing a quicker and more straightforward comparison and selection.

2.3.1 Definitions

The comparison between different technologies is made according to these three factors:

- Overall COP, defined as the total conversion ratio from waste heat to cool, which results from the product of the efficiency of the generation cycle (TEG, ORC) and the COP of the cooling cycle (ABS, VCC). If the same heat input were used to compare these three technologies, the overall COP value would then implicitly offer a measure of the resulting cooling power as well. Efficiencies of the generators and the COP of the cooling cycles are calculated using the models in Appendix A, based on energy and mass balances at the various components and on the standard model for the TEG, as reported by Fraisse et al. [36].

$$COP = \eta_{generator} \cdot COP_{cool} \quad (2.1)$$

- Cooling Power Flux (CPF), defined as the ratio of the resulting cooling power produced and the total area of the heat exchangers. This choice was motivated by

the need of a parameter able to provide information on the system size. Each one however adopts unique components that are not seen in the other systems (e.g. compressor, pump, thermoelectric elements). The heat exchangers are the only components that can be considered common to all technologies, while still providing a measure, although partial, of the system size. As a consequence, defining CPF as the ratio of cooling power over the total heat transfer area can give an indication of the system efficiency with regards to its footprint, which can be compared across different devices. The total heat exchanger area is calculated using the eps-NTU method and suitable correlations for the heat transfer coefficients. The assumptions used for the heat exchangers geometry are given in Section 2.3.2. From the systems models (Appendix A) it is possible to calculate the heat transferred in each heat exchanger and the heat exchanger effectiveness, from which the required NTU value is obtained. By use of the heat transfer correlations, listed in Section 2.3.4 the overall heat transfer coefficient is calculated, and finally the heat exchanger area can be calculated. It should be pointed out however that the total volume of the three systems was not investigated, and that the size of components like pumps, turbines and compressors was thus not included in the calculations.

$$CPF = \dot{Q}_{cool}/A_{hx,tot} \quad (2.2)$$

- Specific investment cost (SIC), defined as the ratio between the total cost of the equipment and the cooling power produced.

$$SIC = C_{o_{tot}}/\dot{Q}_{cool} \quad (2.3)$$

2.3.2 Geometry

The liquid-to-liquid heat exchangers at the VCC evaporators and the ORC generator considered here are of the double pipe type, copper made, with inner tube diameters of 13.84 mm (inside) and 15.87 mm (outside) and outer tube diameters of 21.41 mm and 22.23 mm. For

the ABS chiller the absorber and the desorber instead are modeled as a falling film heat exchanger with a bank of tubes (13.84/15.87 mm).

All the air-cooled condensers (for VCC, ABS and ORC and TEG) are taken as a bank of tubes (13.84/15.87 mm) with air in cross flow. Lastly the heat exchangers directly in contact with the TEG are instead of the plate type with micro-channels having a hydraulic diameter of 1 mm. The area of the heat exchangers will then be determined by the length of these pipes/channels since the diameters have already been set.

2.3.3 Heat and temperature boundary conditions

An input of 10 kW heat is considered, in the form of hot water which goes through a temperature decrease of 10 °C when it reaches the heat recovery heat exchanger. Such heat is converted directly into cooling power in the ABS case, while for the other two it is first converted into electricity, and then the output obtained is used as input for the VCC compressor as shown in Fig. 2.5.

Fig. 2.5 Heat input and cooling output calculation for the three heat-to-cool conversion technologies

The chilled water in the evaporator of the cooling cycle goes through a temperature decrease of 5 °C (12-7 °C), and the evaporator temperature on the refrigerant side is assumed constant at 5 °C. The approach temperature of the leaving fluid is set at 5 °C for air, 2 °C for water.

To better study the behavior of these systems, the temperature of the waste heat driving the conversion cycles is varied between 100 °C and 200 °C, while the condensation temperature at the condensers of both the refrigeration and the power cycles is varied between 35 °C and 50 °C, assuming a 10 °C difference with ambient air (25 – 40 °C).

2.3.4 *Models*

For all the heat exchangers considered in the model, well-known correlations available in the literature are used to estimate the overall heat transfer coefficient (refer to Appendix A). For water flowing in liquid phase, Gnielinski correlation for convection in turbulent flow in a circular tube is adopted [37]. Gungor-Winterton correlation is used for the case of forced convective boiling in horizontal channels [37], with the exceptions of the ABS generator, which uses the formula from Bakhtiari et al. for a falling film flow around the generator bank of tubes [38], and the case of refrigerant R134a in the VCC evaporators, in which case correlation from Fang [39] is chosen.

For condensing fluids, the Chato and Jaster-Kosky correlation for stratified condensation in horizontal tubes is used [37], with the exception of the ABS absorber for which the formula from Hoffmann et al. relative to a falling film absorber [40] is chosen.

For the air flowing through the air-cooled condensers, the correlation for air flow through a bare bank of tubes is used; the formula and the corresponding coefficients can be retrieved in Incropera and DeWitt [41].

Regarding the other components of the various systems, the isentropic efficiency of the compressors is calculated with the formula reported in Jain et al. [42], while for the ORC

turbine and liquid pumps a constant efficiency value of 0.7 is used, assuming the operation of the ORC to be around a constant set point with no transients.

2.3.5 Fluids and material properties

Physical properties for water, refrigerants and organic fluids are taken from the CoolProp library and were implemented in Matlab; for the VCC system R134a is chosen, while for the operating fluid of the ORC system, following the general rules given by Chen et al. [35], R141b is chosen. The properties related to the Water-LiBr solution in the ABS system instead are taken from the ASHRAE Handbook of Fundamentals [43].

For what concerns the TEG, the thermoelectric material used is nano-bulk $\text{Bi}_{0.52}\text{Sb}_{1.48}\text{Te}_3$, because of its good performance both on efficiency and cost, as reported by LeBlanc et al. [34]. Its physical properties have been extracted from Xie et al. [44] and calculated at the temperature of 90°C for the melt-spun spark plasma sintering (MS-SPS) variant, corresponding to a figure of merit of 1.45.

2.3.6 Cost

For all heat exchangers and the ORC turbine, the cost functions are taken from Walas [45]. Pumps cost is calculated with the formula reported in Gutierrez-Arriaga [46] while for the compressors formula from Jain [42] is used. Finally, for the thermoelectric elements cost function proposed by LeBlanc [34] is chosen.

2.4 Results – Effect of changing operating conditions on the selection parameters

Here the model results for the technologies under study are reported for each of the parameters considered for the selection process. The diagrams compare the effect of the condensing (T_{cds}) and generation temperature (T_{gen}) on the total coefficient of performance COP, the cooling power flux CPF and the total specific investment cost SIC. In the analysis, when the effect of varying the waste heat temperature is investigated, the condensing temperature is set by default at 40°C; on the contrary, when the effect of varying the condensing temperature is examined, the waste heat temperature is fixed at 150 °C².

2.4.1 Total COP

The first parameter investigated is the COP of the overall heat-to-cool conversion cycle. By looking at Fig. 2.6, it can be seen that changing the temperature of the waste heat source does not affect the COP of the absorption chiller and its value stabilizes at 0.74. This is due to a limitation imposed on the ABS heater temperature, which is in turn limited by the optimal concentration of salt allowed in the liquid stream to avoid crystallization. As a result, even by increasing the waste heat source temperature, the cycle still operates at a maximum heater temperature of 90 °C with a strong solution salt concentration of 60% by controlling the mass flow rate accordingly. This then leads to the unchanged value of the COP observed in Fig. 2.6.

² Condensing temperature of 40 °C was chosen as related to an ambient temperature of 30 °C typical of South-East Asia climate. Generation temperature of 150 °C is taken as mean point in the range used; values were occasionally explicitly changed to offer more insight on the discussion. In cases where no substantial difference was observed the related plots were omitted.

Fig. 2.6 Variation of total COP with waste heat temperature for a condensing temperature of 40 °C

The ORC/TEG-VCC systems instead show a continuous increase of the total COP as the waste heat source temperature increases. Since here both evaporating and condensing temperature are constant, the cause of this increase is due to the improved efficiency of both the TEG and the ORC cycles, which is greatly dependent on the temperature at the hot side of the TEG and the temperature of the organic fluid at the heater outlet as can be observed in Fig. 2.7. The efficiency of both systems is similar in the 120-160°C range, while an increasing difference in favor of the TEG system is observed as the waste heat temperature increases. By comparison with Fig. 2.6 it can be observed that the total COP for the ORC-VCC and TEG-VCC combines systems indeed follows closely the behavior of the single power generations components.

Fig. 2.7 Variation of TEG and ORC power generation efficiency with waste heat temperature, for a condensing temperature of 40 °C

When compared with the two coupled systems ORC/TEG-VCC, it's apparent that for low temperatures the absorption chiller efficiency is considerably higher. This difference is gradually reduced until a break-even temperature is reached, after which the performance of the coupled systems becomes higher. In this case, the breakeven temperature is found to be between 175 °C and 185 °C.

A change in the condensing temperature can affect the breakeven point. For example, if the condensing temperature is decreased down to 35 °C, corresponding to a lower ambient temperature of 25°C, the breakeven point is found at approximately 145 °C, as shown in Fig. 2.8.

Another important aspect of the comparison is that the ORC/TEG-VCC have a very similar performance for most of the range of temperature used, but as the waste heat temperature approaches 200 °C the TEG coupled system performs better. The ORC-VCC system instead is shown to have a slower increase of the COP towards higher values of the waste heat temperature. This might be due to the type of operating fluid used and in particular to its latent heat of vaporization. Indeed if this value is too low an early onset of the fluid

superheating is possible; this would lead the fluid reaching the maximum temperature allowed (limited by the waste heat temperature), thus causing a lower enthalpy at the heater outlet and consequently a lower efficiency of the generator.

Fig. 2.8 Variation of total COP with waste heat temperature for a condensing temperature set at 35 °C

Fig. 2.9 Variation of total COP with condensing temperature for a waste heat temperature of 150 °C

The effect of the condensing temperature is shown by the curves in Fig. 2.9. These show a clear advantage of the absorption chiller (ABS) over the other two options. In all three cases, however, the overall COP decreases as the condensing temperature increases. This is because when each cooling cycle (ABS, VCC) provides cooling across a larger temperature difference, cooling COP decreases. This can be directly observed in Fig. 2.9 for the ABS chiller, while the COP behavior of the VCC with a varying condensing temperature can be observed specifically in Fig. 2.10.

Fig. 2.10 Variation of VCC cycle COP with condensing temperature. Value is not affected by the waste heat source temperature.

Furthermore, generation cycles would be producing power across a smaller temperature difference, thus leading to an efficiency drop. As a result, the overall COP for the coupled ORC/TEG-VCC systems is shown to decrease faster than the COP of the ABS system, which requires no generation cycle and thus is not subjected to further efficiency loss. It is again interesting to modify the default value of the waste heat source temperature and study

the consequent changes; in Fig. 2.11 the results obtained for waste heat temperature equal to 200 °C are reported.

Fig. 2.11 Variation of total COP with condensing temperature for a waste heat temperature of 200 °C

It can be observed that for waste heat temperature of 200 °C, the COP curves for the ORC/TEG-VCC are shifted upwards; in this way a better performance than the ABS system can be obtained over a large range of condensing temperatures, even though their COP still drops at a faster rate with respect to the ABS system. Like the case of the variation of the waste heat temperature, a break-even point can be identified, above which the two coupled systems perform better. In Fig. 2.9 this point is found at 36 °C and it's common for the two coupled systems. In the case of higher waste heat temperature represented in Fig. 2.11 the point corresponds to a condensing temperature of approximately 42 °C for the ORC-VCC and 44 °C for the TEG-VCC system. This shift of the break-even point towards higher condensing temperature is a direct consequence of the higher COP obtained for the coupled systems as the waste heat temperature increases.

2.4.2 Cooling power flux - CPF

For what concerns the cooling power flux, the ABS chiller outperforms the two coupled systems independently of the waste heat and condensing temperatures used (Figs. 2.12 and 2.13).

For the case of the ABS (Fig. 2.12), a constant value for the CPF is obtained. This can be attributed to the fact that the air-cooled condenser area is considerably larger than that of the other liquid-to-liquid heat exchangers. As a result, since the ABS has a relatively constant COP (refer to Fig. 2.6) the heat discharged at the condenser will be constant as well, thus resulting in a constant behavior of the CPF parameter. In the case of the ORC/TEG-VCC systems instead, there are two condensers: one relative to the generator cycle and one relative to the VCC system. For the generators, as the waste heat temperature increases the efficiency increases, and this results in a reduction of the discharged heat at the condenser and therefore of the condenser area as well. On the other hand, higher generator efficiency means more energy input for the VCC cycle. This results in a larger condenser area but more importantly in a higher cooling power output, which causes an overall increase in the CPF.

Fig. 2.12 Variation of CPF with waste heat temperature for a condensing temperature of 40 °C

It needs to be reminded to the reader that the size of components like pumps, turbines and compressors was not included in the study. If these were to be accounted for, CPF value for the ORC-VCC and ABS systems would certainly decrease with respect to the TEG-VCC system, which doesn't require any of those components.

Similarly to the COP, for all the systems under study, the CPF decreases as the condensing temperature increases (Fig. 2.13). This can be attributed to the decrease in total COP which is caused by the decrease of both the generation and the cooling cycle efficiencies. The consequence is a lower cooling power output and a higher condensing heat to be discharged, resulting in a lower CPF for all cases.

In particular, ORC/TEG-VCC systems show a worse performance for this parameter. This result is due to the more complex nature of the two coupled systems, which have to employ two stages for the energy conversion and thus cause an increase in the number of heat exchangers required.

Fig. 2.13 Variation of CPF with condensing temperature for a waste heat temperature of 150 °C

It is interesting to note that although thermoelectric elements are known to have very high specific power outputs, the overall system still requires a large area for the heat exchangers on the hot and cold side, which causes a low final value of CPF.

2.4.3 *Total specific investment cost*

From Fig. 2.14 it is apparent that the absorption chiller has the lowest cost per Watt of cooling, independent of the temperature of the waste heat source. This is again a consequence of the constant behavior of the COP as the waste heat temperature is changed, as observed previously in Section 2.4.1. The constant COP leads to a constant value of the heat discharged at the condenser and of the corresponding heat transfer area required. The condenser is also the largest heat exchanger in the system, being the only air-cooled one, so the total investment is mainly influenced by its cost. This cost is again constant, because it's a function of the heat transfer area only, and this explains the behavior of the SIC shown in Fig. 2.14. The ORC/TEG-VCC systems show a much higher SIC for low waste heat source temperature, which drops significantly as waste heat source temperature increases. This is mainly due to the larger cooling output of the VCC, resulting from the better efficiency of the two generation cycles obtained at higher temperatures of the waste heat.

It is worth noting that the cost of the TEG-VCC system decreases in the same way as the ORC-VCC even though the former exhibits on average a 20% higher cost due to the thermoelectric material.

The effects of the condensing temperature can be seen in Fig. 2.15. As it was already seen in Figs. 2.9 & 2.11, lower COP values are obtained for higher condensing temperatures, and as a result less cooling power can be generated.

The consequence of this is that the specific cost increases for the same high condensing temperatures.

Fig. 2.14 Variation of SIC with waste heat temperature for a condensing temperature of 40 °C

Fig. 2.15 Variation of SIC with condensing temperature for a waste heat temperature of 150 °C

Here the ABS system shows a slower growth of the cost as the condensing temperature increases. The ORC/TEG-VCC systems instead show a steeper growth of the cost which follows the steeper decrease in COP observed previously in Fig. 2.9, with TEG-VCC being again the most expensive solution.

2.4.4 *Summary*

From this first analysis it can be concluded that absorption chillers have, on average, a higher COP than the coupled systems. However, as the temperature of the waste heat increases and in colder environments, the coupled systems can become more efficient, with the TEG being the most efficient one. It should be noted that research on innovative thermoelectric materials is still ongoing, thus further improvements in efficiency are expected in the future. Indeed, some more efficient materials already exist in the market [34]; however, the prohibitive cost makes them still not applicable at the commercial level. For what concerns size, the ABS chiller proved to be better than the other solutions, although the assumptions made for this study give only a partial view on this aspect. Size of compressors, pumps and turbines was omitted for lack of reliable data; thus, a complete analysis of the size of the systems was not possible. This created a bias in favor of ORC-VCC systems, because the TEG-VCC only requires a compressor and the ABS a pump only, besides the heat exchangers. As a result, if a more complete analysis of the size of the systems could be carried out, TEG and ABS system would probably increase their advantage over the ORC system.

Finally, absorption system performed best in terms of cost too, although for higher waste heat source temperatures its advantage over the two coupled systems is strongly reduced, thanks to the higher cooling output they're able to provide. The reason for the difference between ABS and the ORC/TEG-VCC systems could be identified in the high cost of the key element of the two generators, the turbine and the thermoelectric material respectively.

2.5 Scoring method

2.5.1 Definitions

In the previous sections, each parameter has been investigated separately. However, in the assessment and selection process of suitable technologies (e.g. feasibility studies, purchasing phase, maintenance, retrofitting, etc.), it is not practical to work with several parameters at once. Hence, a scoring method which combines all the three parameters (COP, CPF and SIC) into a single *performance index* was proposed. The scoring method is meant to give a better insight into the way a technology can be chosen over another. In real applications, in fact, an industrial site boundary conditions may require special attention to be put on the size of the system rather than on the other parameters, and the same could be said for cost or efficiency. It is clear then that the relative importance of these parameters should be accounted for in the selection process.

To do so, the scoring algorithm was developed in two steps: firstly, values of COP, CPF and SIC corresponding to a specific pair of waste heat and condensing temperatures were normalized according to the following rules:

- if minimization of the value of a generic parameter A (be that COP , CPF or SIC) is required to have a higher score, such as in the case of the SIC parameter, the values calculated for each system (A_i) are normalized based on the lowest value recorded ($A_{\min}$), using

$$A_{\text{norm}} = \frac{A_{\min}}{A_i} \quad (2.4)$$

- if maximization of the values is required for a higher score, like for COP and CPF , the values calculated for each system (A_i) are normalized based on the highest value recorded ($A_{\max}$), using

$$A_{\text{norm}} = \frac{A_i}{A_{\text{max}}} \quad (2.5)$$

The second step requires the use of weighting factors W that indicate the relative importance of each parameter. By multiplying the weights for their respective normalized parameter and by summation of the results, an overall performance index of each technology, I_{tech} can be obtained:

$$I_{\text{tech}} = COP_{\text{norm}} \cdot w_{COP} + SCP_{\text{norm}} \cdot w_{CPF} + SIC_{\text{norm}} \cdot w_{SIC} \quad (2.6)$$

2.5.2 Scoring results – Case scenario: textile drying process

The algorithm was implemented in Matlab and applied to the case of a drying process in the textile industry. The waste heat temperature was taken at 170 °C, within the temperature range reported by Brückner [5], while condensing temperature was chosen to be 35 °C, corresponding to an ambient temperature of 25 °C. Values for the three parameters of each technology are summarized in Table 2-1. Values after normalization are then reported in Table 2-2.

Table 2-1 Summary of the simulations results for the case of a textile industry

	COP	CPF (W/m ²)	SIC (\$/W)
ABS	0.769	0.856	2.563
ORC-VCC	0.882	0.456	2.859
TEG-VCC	0.905	0.358	3.481

Table 2-2 Summary of the simulations normalized results for the case of a textile industry

	COP	CPF	SIC
ABS	0.850	1	1
ORC-VCC	0.975	0.533	0.897
TEG-VCC	1	0.418	0.736

In the second step, the algorithm takes in all possible combinations of weighting factors to produce a map for each technology, in the form of a contour plot. Two of the three weighting factors are clearly showed in the two axes, while the third is automatically determined since the sum of the three must be equal to 1. The maps will then have a triangular shape and it can be inferred that low values of the third weighting factor correspond to points closer to the diagonal.

The maps so developed can return a more general picture of the performance but also give specific information according to the different relative importance of the parameters.

To better explain the use of these maps, two possible scenarios are considered. In the first one, COP is assumed as the most important parameter (Case A), while in the second one, cost is taken as the most critical one (Case B). Table 2-3 summarizes the weights used for both cases.

Table 2-3 Weights for two possible choice scenarios

Weights (%)	COP	CPF	SIC
Case A	80	10	10
Case B	30	20	50

These weights can then be used as coordinates in the following maps to identify the area of performance in which each system can operate, so that finally an optimal choice can be made by a simple visual comparison.

Fig. 2.16 Index map for the ABS chiller

Fig. 2.17 Index map for the ORC-VCC coupled system

Fig. 2.18 Index map for the TEG-VCC coupled system

Figs. 2.16-2.18 show the three maps for the case scenarios under study. The map corresponding to the ABS system shows a very simple layout where the performance index takes relatively high values as long as the weight related to COP stays below a threshold value of 68%. When this value is surpassed the overall performance is shown to decay and a new region with lower index can be observed.

This is the result of the COP of the ABS being lower than the other two systems, so as the importance of COP increases this system becomes less desirable with respect to the ORC/TEG-VCC ones. The border line between the high and low performance zones is horizontal because the other two normalized parameters have value 1, thus leaving only the COP to influence more decisively the overall index.

The case of the ORC-VCC is different: here five different zones of performance can be distinguished, each zone determined by an index range of 0.1 width. This means that the effectiveness of this solution can be more influenced by the boundary conditions in an industrial site. Here the highest performance is reached for very high values of the COP weight, meaning

that whenever efficiency is the most important parameter this technology proves a better option than ABS. It is interesting to note that high scores can be obtained also at lower values of the COP weight but higher values in the SIC weight. Just by looking separately at the values of the three parameters in Section 2.4 this result would have been overlooked, since it was shown that ABS performed best in terms of SIC. Here instead it can be concluded that overall, even when cost is considered important relatively to the other parameters such as size, the ORC-VCC system is a valid alternative to the ABS one.

A similar behavior is observed in the case of the TEG-VCC system, although the high performance region is reduced in size and corresponds to very high values of the COP weight. Here the large size of the system, in terms of total heat exchange area, and in particular the high cost of the system make this solution comparable to the others only when high efficiency is the key requirement. Also, contrary to common belief regarding thermoelectricity, when size reduction is required this system performs overall worse than the other two solutions, as it can be concluded by the larger low performance region observed.

By examining the two choice scenarios, for Case A the ORC-VCC coupled system resulted to be the best option, closely followed by the TEG-VCC system whose score is on the border between two different regions. Both however are a better choice than ABS system, which is placed in a lower performance region.

In Case B instead the higher importance of cost makes the ABS system the best option available, while the other two systems fall in lower performance regions, ORC-VCC system being this time clearly better than the TEG-VCC.

These are just two possible cases for a system choice, but many others can be analyzed in the same way, according to each industrial site-specific requirements.

2.6 Summary and conclusions

This Chapter aimed at exploring the different solutions that can be adopted to convert industrial waste heat into cooling power, and to develop an objective tool to properly select the most suitable technology for the development of a small-scale refrigeration system.

Here we analyzed three systems, namely an absorption chiller and a vapor compression chiller coupled with an ORC and a TEG generator. Analysis was carried out by considering three parameters, those being total conversion COP, cooling power flux, and specific investment cost.

These were firstly considered separately, and their behavior with changes in waste heat and condenser temperature was analyzed. For low waste heat temperatures, absorption chillers were the best choice available since efficiency of the generators drops excessively in this range. However, at temperatures as low as 150°C the two coupled systems showed a better performance in terms of pure COP.

In the second part, a scoring method was used to produce a single performance index for each solution, which considers all the three parameters together and the relative importance they may have in a realistic decisional process. Results were presented in the form of an index map that can give a general overview of the technology performance and provide a quick selection tool.

The absorption chiller, thanks to its relatively reduced complexity when compared to the two coupled systems, showed better results in terms of cooling power flux and cost, whose effect could be better observed in the index maps. Here the advantage in COP of the coupled systems is determinant only when efficiency is the most critical requirement. In the other cases, absorption chiller still performed better overall, despite having a lower COP at higher waste heat temperatures.

The ORC-VCC systems can in some cases be comparable to the former, but boundary conditions can greatly influence the range in which this system could be a valid alternative.

Similar considerations can be made for the TEG-VCC system, with the high cost of the TEG making this technology competitive only for high waste heat temperatures.

From these results then it was concluded that the absorption chiller is the best candidate for the development of a small-scale device for heat-to-cool conversion. This is a consequence of the higher cooling power flux observed in Section 2.4.2, and from the generally higher performance obtained from the scoring index maps in Section 2.5.

In the following chapter, the absorption chiller is investigated more in detail, to identify the critical elements in the system that can prevent its downsizing and investigate the potential solutions to overcome them. This will lead to the definition of the more specific problem statement, whose solution will be the main focus of this work.

3 Current work on absorbers

As the comparison study presented in Chapter 2 revealed that the ABS chiller is the most promising candidate for a comprehensively efficient heat-to-cool conversion, the subsequent step is to identify which is the component that more than any other prevents the chiller downsizing, and to propose a potential solution to address the issue.

As anticipated in the Introduction, the absorber has been identified as the critical component. In this Chapter the motivations for this choice are explained more in detail, followed by a review of the work previously done on the subject, which led to the proposed new absorber design that is the main focus of this work.

3.1 Critical constraints of the absorber design and operation

As reported by Yuan and Herold [47], the absorber is commonly regarded as the largest component and its design affects the rest of the system. Because of the large specific volume of water at low operating pressure (normally ~ 1 kPa), for a desired mass flow rate the velocity of the vapor moving from the evaporator to the absorber tends to be large. This would result in a large pressure drop that can cause a dramatic change in saturation temperature and increase the risk of crystallization. Because of this, the frontal area for the vapor flow needs to be large to reduce the velocity, and this affects the overall size of the absorber.

Furthermore, the poor heat and mass transfer especially occurring on the solution side of the absorber, caused by poor mixing, requires higher heat and mass transfer areas. This also translates in a higher exergy destruction, which is common also for other working fluids, such as the cases of Aman et al. [48], who investigated the exergy destruction in each component of a water/ammonia absorption chiller. They showed that absorbers are responsible for 63% of the total exergy destruction, followed by the generator (13%) and the condenser (11%). Izquierdo et al. [49] showed that the cooling medium adopted at the absorber and condenser can influence the exergy destruction occurring during the heat exchange process. Results show that an air-cooled system can increase the exergy destruction in a double stage chiller by 14%, with respect to a water-cooled system. These results however only give a partial picture, how later Gong and Goni Boulama [50] proved. Although they confirmed that large part of the exergy destruction occurs in the absorber, they discovered that part of this destruction is, in fact, unavoidable, while instead the exergy destruction in the condenser and evaporator, although minor in comparison, is for the most part avoidable.

Designing an absorber that allows a low vapor pressure drop can be easily achieved by placing the evaporator and the absorber in the same shell, as it is now a common practice in commercial systems. As this would constrain two of the three dimensions of the absorber (i.e. the frontal area between evaporator and absorber), the reduction of the total absorber volume can still be achieved by reducing the size along the third dimension. This is directly depending

on the total absorber area, which in turn is affected by the quality of the heat and mass transfer. If the heat and mass transfer occurring in the absorber can be maximized, for the same rated cooling power a smaller component can be developed. This will directly affect the overall size of the chiller, which could then be downsized to fit more strict space requirements, as could be the case of a domestic application. Furthermore, a reduction in size can reduce the overall cost of the component and the time required for the regular maintenance of the system, which can translate into further savings.

For these reasons, the absorber was chosen to be the first and main component to study and improve. In the next Section, a review of the various absorbers, their characteristics and the possibilities for improvement of the heat and mass transfer are presented.

3.2 Type of absorbers

After establishing that the absorber is the critical component of the absorption chiller, an extensive literature review on the subject revealed that great effort has been spent in finding the best way to improve its performance. As explained in Section 2.2, the absorber is the part of the absorption chiller where a liquid solution with high salt concentration re-absorbs the water vapor coming from the evaporator. The output of this process is a low salt concentration solution that is then pumped back to the generator to close the cooling cycle. It is clear that good mass transfer is of the utmost importance to achieve a very efficient operation, so that size can be kept small while maintaining similar COP. Furthermore, when water vapor is absorbed, condensation heat is generated; this raises the solution temperature and consequently reduces its absorption capabilities. To avoid the worsening of the performance, the absorber must be cooled down, usually either with water or air [49].

Previous research offers a range of different absorber layouts. The most relevant ones have been summarized by Ibarra-Bahena and Moreno [51] and are:

- Falling film type

- Adiabatic type
- Bubble type
- Membrane type

In the following sections, these are described in detail, along with a critical analysis of their advantages and limitations.

3.2.1 *Falling film absorber*

Until now falling film absorbers have been the most used ones, thus extensive review exists on this type of absorber. The typical configuration is that of a *helical coil* where cooling water flows inside the tube coil, while the strong solution film flows on the external surface. Here the solution is exposed to the refrigerant water vapor so that absorption can occur (Fig. 3.1).

Fig. 3.1 Falling film absorber [51]

Part of the research done on this type of absorber deals with the accurate modeling of the heat and mass transfer. The complexity of the models is observed to vary considerably, starting from purely analytical ones like the one developed by Islam [52], proceeding then with nonlinear models of the coupled heat and mass transfer phenomena (Nabil et al. [53]) and finally with more sophisticated numerical models that use a volume of fluid (VOF) approach

and 3D geometries to study the transient behavior of the absorption process. A review of the different modeling techniques is provided by Subramaniam and Garimella [54]. From the experimental validation of these models, it was concluded that high operating pressure and low cooling medium temperature are beneficial to the optimal operation of the absorber, results also confirmed by Seol and Lee [55].

Apart from these characterization works, great effort has been spent in finding better configurations of the absorber to improve the mass transfer. Among the various solutions studied, reduction of the tube diameter was found to increase the absorption rate (Yoon et al. [56]). In addition, surface treatment of the tubes, by artificially creating rough surfaces, increased the heat and mass transfer by 40% with respect to a bare tube (Yoon et al. [57]). Miller and Perez-Blanco [58] also proposed the surface roughness modification to improve the absorption process and obtained an increase of 225% in the mass of water vapor absorbed. Islam et al. [59] proposed a new design consisting of flow guiding fins capable of inverting the solution film and thus exposing a larger part of the solution to the water vapor to be absorbed. Results showed a maximum increase in the vapor absorption rate of 100% with respect to bare tubes.

Fig. 3.2 Film inverting technique, from Islam et al. [59]

Apart from the more standard coil configuration, other layouts of the falling film absorber have been investigated. Vertical tubes absorbers are one of the simplest modification of the original layout and the modeling can be done in similar ways as before, which obtained good agreement with the experimental results (Bo et al. [60]).

A simple absorber layout is the plate type, where the solution is distributed on a cooled or adiabatic plate which is exposed to the refrigerant vapor. A simple numerical model of this type of absorber has been developed by Karami and Farhanieh [61], using different tilt angles of the plate. In all cases investigated, good agreement with the experimental data was obtained, and an optimal tilt degree of 85-90° was found to return the highest mass transfer.

The modification of the plate surface with the addition of fins was studied by Mortazavi et al. [62]. Thanks to a more uniform distribution of the film on the plate surface the average film thickness could be reduced, and consequently, the measured absorption rate was double the one of a plain plate.

Fig. 3.3 Flow distribution comparison before a) and after b) the application of finned surfaces - Mortazavi et al. [62]

Bigham et al. [63] and Isfahani and Moghaddam [64] then brought this concept further and they included structures to obtain also micro-mixing of the liquid solution flowing on the

plate of a membrane-type absorber. The absorption rates measured were improved by a factor 2.5 with respect to an undisturbed flow on a flat plate.

Finally, an active technique to improve the process was tested by Perez-Blanco and Tsuda [65]. The system consists of a vibrating screen that perturbs the falling film enhancing the solution mixing and consequently the absorption. Results showed that at increasing frequencies the predicted mass fluxes can be as high as double the ones obtainable with the highly performant membrane absorbers (See Section 3.2.4)

3.2.2 *Adiabatic absorber*

Adiabatic absorbers separate the cooling of the solution from the actual absorption process, so that in the absorber no cooling medium is present.

Fig. 3.4 Adiabatic absorber [51]

In the usual and most simple configuration, the solution is distributed in the absorber chamber in the form of droplets or columns (Fig. 3.4). This layout has been investigated experimentally by Li et al. [66] who stated the advantages of this solution. These are the possibility to cool the solutions separately and thus in a more efficient way, and the lack of

active components in the chamber, since the whole process is activated by gravity alone. In this study, the flow was distributed by means of an orifice plate, a device whose properties have been explored in a precedent study by Wang et al. [67].

Alternatively, a promising alternative to the falling column solution is provided by Palacios et al. [68]. It consists of flat fan sheets of the solution created by a Vee-jet type nozzle, which considerably increases the liquid-vapor interface. In this study, the measured absorption coefficient is one order of magnitude larger than coefficients for the basic falling film case. Bell [69] explored different distribution strategies for the adiabatic spray absorber using different nozzle types. The vee-jet nozzle returned, on average, the highest absorption ratio, thanks to the larger width of the flat fan sheet this nozzle could create. Later the same configuration was found to perform better than the droplet configuration too, making it possible to reduce the size of the absorber of about 50% with respect to the droplet absorber (Gutierrez-Urueta et al. [70]).

3.2.3 *Bubble absorber*

Bubble absorbers can increase the area of the liquid-vapor interface by injecting vapor bubbles directly into the liquid solution stream in a countercurrent flow (Fig. 3.5). This layout of the absorber has been found to require an area for the heat exchange about 50% lower than in the case of falling film absorption (Kang et al. [71]), but in a later study the estimation of the reduction was reduced to 10%, especially in the region of low flow rates, i.e. low refrigeration capacities.

However, some good examples of high-performance bubble absorbers exist. One of these was developed and tested by Cardenas and Narayanan [72] and consists of a constrained liquid film of an ammonia/water solution where the refrigerant vapor bubbles are injected from a porous plate. A test revealed that this configuration led to an effectiveness of the absorber

measure between 0.75 and 0.9. This came at the expense of a large pressure drop between the evaporator and the absorber, measured to be between 0.08 and 0.29.

Fig. 3.5 Bubble absorber [51]

3.2.4 Membrane absorber

Following a similar idea to the one of Cardenas and Narayanan [72], applied this time to the LiBr-H₂O solution, membrane absorbers realize a similar distribution of the vapor. Here a hydrophobic membrane is used so that vapor can distribute through the small pores of the membrane, while backflow of liquid water can be avoided. In this way, the area of the liquid-vapor interface can be maximized, and the absorption process improved.

Ali and Schwerdt [73] tested various types of membranes and concluded that high porosity above 80%, maximum pore size of 0.45 μm , and a membrane active layer thickness of 60 μm are some of the desirable characteristics of a well-performing membrane. Ali [74] then further investigated the effect of various properties of the membrane on the absorber operation and found that reducing the liquid solution film thickness can greatly improve the performance and lead to a more compact size of the absorber.

Fig. 3.6 Membrane absorber with the detail of the temperature and water vapor partial pressure behavior as it flows through the membrane [73]

Similar results were obtained later by Isfahani and Moghaddam [64], who also stated that, other than decreasing the film thickness, increasing its entry velocity in the absorber can be beneficial to the overall performance.

3.2.5 Comments

By comparing the solutions presented in this literature review on the absorbers it is possible to draw some conclusions about which of them is best suited for integration in a small-scale chiller.

Bubble absorbers were found to have a limited performance, except for the case in [72], which proved to be remarkably good. However, the test of the porous plate in that study required a pressure differential up to 0.29 bar to drive the ammonia vapor through the porous plate. In the case of a realistic application in a water-LiBr chiller, that differential would be

unacceptable, considering that the absorber and evaporator both work at around 1 kPa. For this reason, the bubble absorber was discarded.

The membrane absorber works on a similar principle, and although the high porosity reduces the necessary driving pressure differential between evaporator and absorber, the high material and manufacturing cost would make this option not desirable for a small-scale unit.

Still, membrane absorbers are very effective. The principle behind the membrane efficacy shown in [59–61] is the increased contact surface between refrigerant and solution that it can provide. By using the same principle, adiabatic absorbers can offer some advantages; in fact, by cooling the solution beforehand, the fluid distribution can be optimized more freely to increase the contact area between the refrigerant and the solution.

In a standard absorber this would not be possible, as the presence of cooling tubes requires maximization of the solution heat transfer as well. This creates a constraint in the absorber design and the solution distribution, as shown by Hu and Jacobi [75], who showed for example that a jetting regime offers an advantage on the heat transfer coefficient, and by and Olbricht [76], who identified an improvement in the heat transfer coefficient even during the initial stages of transition from dripping to jetting regime.

Some adiabatic absorber designs have been presented, such as that based on solution falling column flow by Li et al. [66] or the vee-jet nozzle introduced by Ryan [69] and further investigated by Palacios et al. [68]. For all these adiabatic absorbers the diameter of the droplets or the falling column and the thickness of the fan sheets are the key values to be reduced to obtain a performance comparable with the membrane absorber type, without incurring in additional costs.

However, in these cases the solution internal mixing is prevented by the absence of any physical obstacle to interrupt the flow. This can cause a limitation to the amount of solution that actively participates in the absorption process, and thus hinders further volume reduction for the absorber.

The benefits deriving from the presence of an obstacle were remarked by Jeong and Garimella [77]. They proved that in a horizontal tube absorber, the most effective flow regimes

in terms of mass transfer are the falling film regime around the tube, thanks to the solution cooling occurring, but most importantly the droplet formation regime. This regime returned a mass transfer coefficient up to one order of magnitude larger than the falling film regime. Similar results were obtained later by Ben Hafsia et al. [78], although droplet formation and fall were not considered separately; nevertheless, they concluded that between 74% and 77% of the total mass absorption occurred during the combined droplet formation and fall regime.

3.3 Conclusions and problem statement

Following these considerations, here a solution is proposed to improve the design of an adiabatic absorber by promoting the internal mixing of the H_2O -LiBr solution and by exploiting the benefits of a droplet flow regime, especially during the droplet formation phase. This is done by including pins in the absorber chamber, which can interrupt the free fall of droplets that was instead used for example in [66]. The pins can create a separation of the droplet flow regime in more phases, namely droplet free fall, impingement on the pin, film flow around the pin, and droplet formation at the pin bottom side. The mixing induced by the presence of the pin can normalize the concentration and temperature profile and allow a higher fraction of solution to participate in the absorption process.

Since the use of an adiabatic absorber eliminates the requirement of a circular tube, the geometry of the absorber can now be optimized more freely, to find a solution that minimizes both the droplet size the absorber can work with and the risk of droplet coalescence. This would result in the maximization of the vapor-liquid interface area which, along with the additional solution mixing, can allow the reduction of the absorber footprint.

In Fig. 3.7 it is reported a simple scheme describing the absorber chamber under design. The bank of tubes is substituted here by a bank of solid pins, with a distribution reservoir placed on top of it and with the side walls having a purely structural function.

Fig. 3.7 Proposed absorber design. (Left) Frontal view of the bank of pins. (Right) Side view of the absorber

Starting from this design concept, the following work is then organized to follow the necessary steps taken to identify the optimal absorber geometry and validate the beneficial effects on the absorber operation, which can enable the chiller downsizing. The problem can be subdivided into two parts:

1. In the first part, the proposed design is investigated, focusing only on the fluid geometrical domain involving the droplet flow regime. The main objectives are:
 - a. To develop an analytical model of the droplet formation regime that can accurately reproduce the droplet profile at different stages of the droplet formation.
 - b. To use this model to optimize the absorber geometry to minimize the risk of droplet coalescence and maximize the interface surface area during the absorption process.
2. In the second part, the optimized pin geometry is used, and the analytical model of the droplet formation regime is extended to describe the absorption process. Here the main objectives are:
 - a. To develop an analytical model of the coupled heat and mass transfer during the droplet formation regime.

- b. To utilize the model developed to estimate the absorption performance of the proposed optimized absorber design.

4 Heat and Mass transfer model - Part I

In this Chapter the flow arrangement of the absorber design proposed in the previous Section is analyzed, and a geometrical description of the droplet formation phase is provided. The model of the droplet formation regimes is here presented and subsequently used to optimize the absorber design parameters such as the pin geometry, the overall length defined as the total length of the first row of pins, the number of orifices and their diameter as presented in Fig. 3.7 [79]. The correct operating conditions are also defined. These are to ensure that no droplet coalescence occurs, so that the vapor-liquid interface surface area can be maximized.

4.1 Modeling

4.1.1 Droplet regime and distribution

To ensure a stable droplet flow throughout the absorber, a correct design of the orifice plate for the solution distribution is needed. Extensive literature is available on the differences between dripping and jetting flow regimes generating from an orifice, and the conditions occurring during the transition between the two regimes.

Teng et al. [80] proposed a correlation to predict the diameter of droplets d_m generated after the jet breakup as a function of the orifice diameter d and the fluid properties, expressed by the Ohnesorge number, Oh . The equation is the result of an explicit dispersion equation that considers the instability wavelength in a low velocity falling jet to be the key factor in determining the final droplet size after breakup. Eq. (4.1) represents the solution of this explicit equation, with the additional simplification deriving from the large difference in viscosity between the liquid solution in the jet and the gas (air, water vapor) surrounding it.

$$d_d/d_{or} = (3\pi/\sqrt{2})^{1/3} (1 + 3Oh/\sqrt{2})^{1/6} \quad (4.1)$$

A criterion to identify the conditions for the jetting-dripping transition was not provided here. This issue was addressed firstly with the theory proposed by Weber and it was later improved by Grant and Middleman [81], who identified a critical jet breakup length as a function of the orifice diameter, and the Weber and Reynolds numbers. Ambravaneswaran et al. [82] confirmed the importance of the Ohnesorge Oh number in the definition of the transition point, identifying a corresponding critical We number, We_{cr} for the different regimes. In case gravity is taken into consideration, the jetting-dripping transition is also dependent on the Bond number Bo . For very small values of the Oh number, Clanet and Lasheras [83] proposed a correlation for the critical transition We number for a water flow, as a function of

the Bo number only. Eggers and Villermaux [84] finally reported a more general correlation for the critical We_{cr} :

$$We_{cr} = 0.034 (Bo Oh)^{-6/15} \quad (4.2)$$

By use of Eq. (4.2) it is then possible to identify the maximum We number value above which it's not possible to maintain the droplet flow, and Eq. (4.1) returns the expected droplet size. From the value of We , the corresponding flow rate per orifice can be obtained, and the static liquid head to be maintained can be found by means of Eqs. (4.3-4.4)

$$Q = C_d A \sqrt{2\Delta P / \rho} \quad (4.3)$$

$$\Delta P = \rho g H \quad (4.4)$$

where Q is the volumetric flow rate, depending on the discharge coefficient C_d , the orifice cross-section area A , the fluid density ρ and the pressure differential ΔP . From the latter, the static head H can be calculated with Eq. (4.4).

The value for the discharge coefficient is set to an average value of 0.7, taken from [85]. In addition to this, in order to maintain a droplet flow after the falling droplets impact on the pins, a minimum distance between the orifices, to prevent coalescence of the droplets onto the pins in the horizontal direction, is determined according to the work done by Wang and Jacobi [86]. Here the authors assumed the distance between the droplet or jet generation sites on a circular tube to be equal to the fastest-growing Taylor wavelength λ , calculated with Eq. (4.5) as a function of the density and the surface tension σ_s .

$$\lambda = 2\pi \sqrt{3\sigma_s / g\rho} \quad (4.5)$$

By setting the orifices distance equal to this value, the number of orifices on the distribution plate will be equal to the number of droplet generating sites on the pins. Therefore, the probability that two or more droplets belonging to different generating sites will merge on the pins is reduced, and the total liquid-vapor interface area can be maximized.

4.1.2 Droplet formation description

When considering just a single orifice and its corresponding generating site on the pin, droplet coalescence can still occur between two subsequent falling droplets. To minimize this risk as well, the pin geometry can be optimized to find the design that retains the least amount of liquid during the droplet formation phase. By doing this, it can be ensured that most part of a droplet volume will detach from the pin before the subsequent droplet can merge with it.

Fig. 4.1 Geometrical variables for a droplet formation process

An analytical model for the droplet formation phase is used, based on the variational approach [87,88].

The model considers an axisymmetric droplet, described in Fig. 4.1. A balance between the droplet surface energy, the potential gravitational energy and the liquid-surface interfacial energy is performed on a thin slice of the droplet. The droplet profile is then calculated as the one that minimizes the droplet energy at constant volume. This translates into the Euler-Lagrange equation, expressed in the following system of differential equations:

$$d\theta/dS + \sin\theta/X = 2/B - Z \quad (4.6)$$

$$dX/dS = \cos\theta \quad (4.7)$$

$$dZ/dS = \sin\theta \quad (4.8)$$

$$dV/dS = \pi X^2 \sin\theta \quad (4.9)$$

In Eqs. (4.6-4.9) X and Z are the geometrical coordinates, S is the arc length and B the curvature radius at the droplet axes, after non-dimensionalization by multiplication of the respective dimensional variables by the factor C_{norm} of Eq. (4.10).

$$C_{norm} = \sqrt{\frac{(\rho_d - \rho_L)g}{\sigma_s}} \quad (4.10)$$

In Eq. (4.10) ρ_d and ρ_L are the droplet and the surrounding fluid density respectively. By use of the scaling factor C_{norm} of Eq. (4.10) it is possible to include all the fluid properties in this number, so that the non-dimensional model obtained is generally valid. The fluid-specific, dimensional results can be then obtained by dividing the non-dimensional values obtained by the C_{norm} factor corresponding to the fluid being used. Because of the general character of the model, for the validation purposes in Section 4.3, water is used. For the absorber design process of Section 4.4 instead, a 60%_{weight} H2O-LiBr solution is used. Density

and viscosity values for the solution are obtained from the Fchart EES properties library, based on the work of Yuan and Herold [89], assuming the solution at 25°C. Surface tension value is taken from [47]. The values are summarized in Table 4-1.

Table 4-1 Fluid properties

Fluid	Density [kg/m ³]	Viscosity [Pa s]	Surface tension [N/m]
Water	1000	8.9E-04	0.072
H2O-LiBr (60% w)	1688	7.3E-03	0.095

The system of Eqs. (4.6-4.9) is then solved using the following boundary conditions and assuming fully wetted condition of the pins.

$$[X = 0, V = 0]_{z=0} \quad (4.11)$$

$$[\sin \theta / X = 1/B]_{z=0} \quad (4.12)$$

$$[dX/dZ = \cot(90 - \beta)]_{z=z_0} \quad (4.13)$$

Here β is the angle between the vertical axis of symmetry and the tangent to the pin at the point of contact between the droplet and the pin. For the circular and elliptical cases, this angle varies depending on the point of contact by effect of the pin curvature, while for the rectangular and rhomboidal cases this value is constant.

The fourth order Runge-Kutta method is used in Matlab to solve the system of ordinary differential equations for different values of the curvature radius B . The computation of the solution starts at the droplet apex, placed in the axes origin. The calculation of the subsequent steps proceeds until the boundary condition is satisfied. This condition is set by Eq. (4.13), which imposes that the droplet profile needs to be tangent to the pin surface at the point of contact, identified by the point of coordinates (R_0, Z_0) , in Fig. 4.1. The droplet volume is calculated as the volume swept by the revolution around the z-axis of the droplet profile, as given in Eq. (4.9). The volume occupied by the pin is then subtracted from the total. This is

calculated by multiplying a section of the frontal area of the pin by $2 \cdot R_0$, where the section area depends on the location of the contact point between the droplet profile and the pin.

4.2 Experimental setup and procedure for droplet geometry analysis

Experiments have been carried out to test the validity of the analytical model previously described. A pin-finned plate was manufactured in 3D printing using a high detail resin with an average roughness of $50 \mu\text{m}$ (Fig. 4.2).

Fig. 4.2 (Left) Test plate in 3D printed resin. (Right) Details on tested pin geometries

Seven different pin geometries were chosen for the analysis, as seen in Fig. 4.2. The choice has been made by considering two categories. The first includes pins with a rounder surface, such as the circular and the two ellipticals. The second includes pins with sharp angles separating the upper and lower half of the pin, as is the case of the three rhomboidal and the rectangular pins. Different axes ratios are then chosen for each group to vary the angle at which the droplet profile maintains contact with the solid surface during the formation phase. The pin width is set at 5 mm for all the pins, so that the projected area on the horizontal plane is the same for all the pins. In this way the space occupied by the pin, as seen from the perspective

of the droplet, is the same. Moreover, given a fixed constraint on the absorber dimensions, this choice will result in having the same number of pins for all the geometries studied and consequently the same number of droplet generating sites on the pins.

Fig. 4.3 Experimental setup for the capture of droplet profiles

The experimental setup is presented Fig. 4.3. A support is used to hold a syringe with a 2 mm orifice diameter and position it on top of the different pins columns. During the experiments, the syringe is manually operated to produce the droplets and initiate their fall. A high-speed camera (Photron SA5) is used to record the droplet formation at the pins bottom side. An Argon light source is used, and its output regulated to highlight the area of interest. For the high-speed camera a low resolution of 768x768 is used, which allows for the camera to record the droplet fall dynamics with at least 2000 fps.

During post-processing, a set of the highest quality images captured are processed with *imageJ*. For each of the images recorded, the droplet profile is manually marked by a series of points by choosing the pixel observed to be the nearest to the profile, whose cartesian coordinates are then exported. As the error in the profile marking is equal to one pixel, with the 768x768 resolution the accuracy is estimated to be 0.13%. Of these exported coordinates, few points (those closer to the bottom tip of the droplet) are used in a Matlab script to calculate the

corresponding experimental radius of curvature B needed by the main theoretical model. If three points only are used, the radius is univocally calculated. If more than three points are used, a least square method is used to find the best fit circle that can approximate the data. The calculated experimental radius B is then exported back to the system of ODE described in Section 4.1 and used as input. With this value the model returns the theoretical droplet profile having the same radius of curvature of the experimental one, making the comparison between the two profiles and hence the validation possible.³ In the model validation, for each point in the experimental profile, the coordinate z is used to find the corresponding point on the theoretical profile. For each droplet profile, between 100 and 120 points, depending on the profile observed and limited by the image resolution, are taken on the experimental profile and compared with the corresponding theoretical point.

The deviation of the predicted theoretical droplet profile from the experiments is calculated with Eq. (4.14) and described in Section 4.3.

$$\delta = |r_{th} - r_{exp}|/r_{exp} \quad (4.14)$$

4.3 Experimental results and validation

As explained in Section 4.2, the comparison with the experimental data requires the calculation of the droplet radius of curvature B . This calculation is influenced by the number of experimental points chosen for the calculation, as it will affect the results of the least square method used to find the radius of the best fit circle. To assess the sensitivity of the model to a change in the number of points, five different ranges have been chosen; their length is

³ Since the deviation is only calculated on the radius, it is not required to measure the volume of the droplet. The calculation of the volume depends on the contact point between the droplet and the pin, with coordinates R_0 and Z_0 ; this is found by using the condition of Eq. (4.12). Therefore, Eq. (4.8) along with the condition in Eq. (4.12) are not used for the validation. This is an exception made for the experimental validation only, later the condition set by Eq. (4.12) will be reinstated in the second part of the study, where the droplet volume calculation is required, and the knowledge of the contact point becomes a necessary requirement for its accurate estimation.

determined by the X coordinate of the profile calculated from the droplet tip (the origin in the scheme of Fig. 4.1, set at five different limits from 0.2 mm to 1 mm.

Each of these returns a radius of curvature, which is then used to run the model described in Section 4.1. The results for the sensitivity analysis and the processed images from the high-speed camera are presented here for the case of the 120° rhomboidal pin in Fig. 4.4; the results for the other pins are included in the Appendix B. Three images were chosen to represent the three stages of the droplet formation phase (namely *Start*, *Middle* & *End*) before detachment from the pin occurs.

Fig. 4.4 Droplet profile analysis for the circular pin case⁴.

⁴ The different scales observed on the y-axis of the charts in Fig. 4.4 are due to the different droplet heights measured, as each of them corresponds to a different stage of the droplet formation phase

In the first column, it is shown the experimental droplet profile captured during the experiments. Then, in the second column, the experimental droplet profile has been magnified and post-processed to make it clearer and more visible for the model validation. In the third column, the comparison between the droplet profile obtained by the model and the experiments (black dashed line) is given. Observation of the experimental results shows that the assumption of cylindrical symmetry of the droplet is not always met. This is due to the nature of the experiment, in which the residual kinetic energy after the impact of the droplet onto the pin caused some oscillations during the droplet formation. Results from the profiles comparison show that the experimental curves fall, in most cases, within the range of profiles obtained from the analytical model, for different ranges used. In particular, the profiles all seem to match with higher accuracy the experimental data close to the droplet bottom, i.e. close to the origin. Farther from the origin instead, different ranges may result in considerably different profiles in some cases, as it can be observed for example in the “Rhomboidal-end” case of Fig. 4.4.

This might be due to the imprecision in collecting the points of the experimental profile. Automatic, more precise methods for shape recognition would improve the accuracy. However, this would require for neater images in input, a condition that couldn't be met due to the poor lighting created by the plate material being used and the constraints on the maximum resolution and framerate that the camera can provide.

Nevertheless, analysis of the different cases shows that in many instances the analytical profiles tend to fall in a narrow range. It is also noted that as the range width increases, there is a general tendency of the profiles to collapse toward the same curve, with the “0.2 mm” range curves being the only ones that maintain a marked distance in some of the cases investigated. This might be due to the low accuracy that comes with using fewer points for the radius of curvature calculation.

Observation of the overlapped profiles shows that the main cause of the discrepancy between the analytical and experimental one can be explained by the finite dimension of the pins. The analytical model used assumes that the droplets are forming on an infinite surface, or a surface large enough to be approximated as infinite. Experiments instead are showing that

this condition is not always met. Cases like the elliptical or the rhomboidal show a neat droplet forming at the bottom half of the pin. In the other cases instead, such as the circular (see Appendix B), the droplet surrounds completely the pin, and this creates a distortion in the profile that the current model cannot compensate for.

Table 4-2 summarizes the results of the sensitivity analysis by reporting the deviation of the best matching theoretical profile, with respect to the experimental data. The deviation is calculated on the X variable, for same values of the Z variable, using Eq. (4.14).

Table 4-2 Average deviation between the experimental curves and the best fit curves derived from the sensitivity analysis

Pin	Phase	Deviation (%)	Pin type	Phase	Deviation (%)
Circular	Start	14	Rhomb 60°	Start	16.3
Circular	Middle	35.1	Rhomb 60°	Middle	14.5
Circular	End	19.6	Rhomb 60°	End	28.7
Elliptic -1x1.25	Start	6.4	Rhomb 90°	Start	4.6
Elliptic -1x1.25	Middle	31.4	Rhomb 90°	Middle	28.2
Elliptic -1x1.25	End	26.5	Rhomb 90°	End	27.4
Elliptic -1x1.5	Start	15.7	Rhomb 120°	Start	17.3
Elliptic -1x1.5	Middle	28.9	Rhomb 120°	Middle	18.9
Elliptic -1x1.5	End	24.3	Rhomb 120°	End	10.4
Rectangular	Start	8.1			
Rectangular	Middle	8.7			
Rectangular	End	15.7			

Table 4-2 shows that the deviations for all the cases are distributed in a band from 4% to 35%. This variation can also be seen in each case investigated, with the rectangular and the 120° rhomboidal case having a sensibly narrower deviation band and returning, on average, a smaller deviation. The fact that these two cases refer to pins with a flat and quasi-flat bottom surface might explain the similar behavior. The sharp angle at the pins sides might also be the

cause for the higher accuracy measured. As previously explained, droplet distortion caused by the presence of water in the upper half of the pins is one of the main cause of the low accuracy of the model, far from the droplet base. In the case of the rectangular and 120° rhomboidal pins, these sharp angles might create a good separation between the upper and lower part of the pin, reducing the distortion effect during droplet formation.

4.4 Design process for the LiBr-H₂O absorber

After validation of the droplet formation model, this can now be used along with the orifice distribution model of Section 4.1.1 to provide a comprehensive description of the absorber design and operation. The condition for an optimal droplet flow to be maintained is that the droplet coalescence must be avoided both between two droplets generated by the same orifice and between two droplets generated by two different orifices. The models described in Section 4.1 can then be used together to ensure that both conditions are met and estimate the optimal absorber size required to meet a desired design value for the solution flow rate. The procedure is explained in Fig. 4.5.

The droplet formation model of Eqs. (4.6-4.9) and the orifice droplet model of Eq. (4.1) are independently solved to return respectively the maximum droplet volume the pin can retain, and the droplet volume generated by the orifices. For the droplet formation model, the boundary condition expressed by Eq. (4.13) is used to identify the contact point between droplet and pin. The numerical model is then solved until convergence is reached on the dX/dZ of Eq. (4.13), within an error of 0.5%. These two values are then compared (Fig. 4.6) to find which is the best combination of pin geometry and orifice size that can allow the absorber to operate with smaller droplet without coalescence risk.

Once the best combination is found, this is then used in the following phase, where Eqs. (4.2-4.5) are used to estimate the overall size of the absorber, in terms of total pin length needed to meet a desired value of solution mass flow rate. In particular, Eq. (4.2) is used to

calculate the critical flowrate that each orifice can process without transition to jet flow. This critical flowrate is then used in Eq. (4.3-4.4) to derive the corresponding critical static head value for the distribution reservoir.

By dividing a design value of total pin length by the orifice distance, obtained using Eq. (4.5), the total number of orifices available can be calculated. This value, multiplied by the critical flowrate per orifice, returns the overall mass flow rate that the absorber with chosen length can process. Additionally, the correct solution static head to be maintained at the distribution reservoir is determined, to ensure a stable droplet flow.

Fig. 4.5 Flowchart for the absorber design process

4.4.1 Optimization of orifice size and pin geometry

To find the best layout, firstly the model was solved iteratively for each pin shape with different pin widths, ranging from 1 *mm* to 10 *mm*, using the LiBr-H₂O solution as working fluid⁵.

For each configuration of the pin shape and width, the input radius of curvature B is varied to find the maximum volume of liquid that the configuration can retain. The maximum volumes obtained for all the configurations are plotted in Fig. 4.6, where the volume of liquid generated by the orifices of different sizes is also shown.

The continuous lines represent the liquid volume retained by the different pin shapes for different pin widths. From the figure it is apparent that the rhomboidal pins show a constant retained volume for any width. This can be explained by considering that the range of pin widths is relatively small and that the angle of the rhomboidal surface, with respect to the horizontal axis, is constant. Because of this, only one profile satisfies Eq. (4.13) for all the widths, leading to the same calculated volume.

The two elliptical pins show an increasing retained volume for pin widths up to 3 *mm* after which the value decreases constantly.

This can be explained by considering that the condition set by Eq. (4.13) translates, in geometrical terms, into finding the tangent point between droplet profile and pin surface. For the case of the ellipse, the angle of a tangent to its surface varies faster near the bottom, slenderer region. Because of this, the probability for the droplet-pin contact points to fall in this region is higher. Hence the retained volume shows an initial rise (for smaller pin widths), thanks to the abundance of angles of contact available in the narrow region. However, as the pin size increases more volume is subtracted to the liquid, which leads to a decrease in the retained volume.

⁵ The rectangular pin can retain fluid not only on the lower side but also on the upper surface, where the model is not applicable; hence the comparison would create an unfair advantage for the rectangular pin. For this reason, the rectangular pin, although used for the model validation purposes, is here neglected.

Finally, the circular pin shows an increasing retained volume, with a gradient that slows down as the pin size increases. This can be explained by considering that the angle of a tangent to a circle varies at a constant rate. Because of this, the possible droplet-pin contact points are more evenly distributed along the pin surface. As the pin size increases then, the contact point moves farther as well, and the resulting increase in droplet volume compensates for the volume reduction given by the larger size of the pin.

Fig. 4.6 Comparison between maximum droplet diameter at the reservoir orifices and at the pins bottom side

The retained volumes of Fig. 4.6 are then compared with those of the droplet generated by the distributor. Three different orifice diameters are used; for each of them, the diameter of the droplet generated is calculated using Eq. (4.1) (reported in dashed lines). If the volume of liquid that a pin can retain (continuous lines) is smaller than the volume of the droplet generated by the distributor, the risk of coalescence of two or more droplets is reduced, since the pin cannot retain enough liquid. Conversely, if the volume of liquid retained by the pin is larger than that generated by the orifice, the risk of coalescence is higher. For all the cases investigated, using

an orifice diameter larger than 1.5 mm reduces the risk of coalescence for all the pins used, as none of them is able to retain a larger volume of liquid. In particular, both the 120° and the 90° rhomboidal pins can be coupled with a 1 mm orifice with the same reduced risk of coalescence. However, the 120° still returns a lower retained volume for all the pin widths used, and thus represents a better option for the absorber design.

4.4.2 *Optimal absorber size and flow rate*

With the chosen pin geometry and orifice sizes, the overall design of the absorber and the optimal operating flow rate can be determined. By use of Eq. (4.2-4.4) it is possible to find the critical mass flow rate per orifice allowed, above which it will be impossible to generate a stable droplet flow. At the same time, Eq. (4.5) gives the minimum distance to be kept between the orifices. Consequently, from the total length of the first row of pins, the total number of orifices can be calculated, along with the maximum allowed mass flow rate. Fig. 4.7 maps the relationships between length, orifice diameter and mass flow rate to give a quick tool for the absorber sizing.

Given the 1 mm orifice diameter found in Section 4.4.1 of the design process, the map of Fig. 4.7 can be readily used to get an estimation for the absorber size.

For instance, if a target mass flow rate of 0.05 kg/s is assumed, the map returns a total pin length of the first row of approximately 4 m , corresponding to a linear mass flow rate of 0.0125 kg/m s . This length can then be distributed in the absorber in multiple shorter pins or a few longer ones, depending on the geometrical constraints of the application, as long as the total pin length is the same. The overall length found, along with the orifice size and the pin shape obtained from Section 4.4.1 constitute the main geometrical features needed for the absorber design.

To ensure a droplet flow is maintained at the distribution plate, the critical We number found with Eq. (4.2) needs to be translated to a more easily measurable quantity. Using Eqs.

(4.3-4.4) the corresponding critical static head, to be maintained on the distribution plate, can be found.

Fig. 4.7 Flow rate map for different orifices sizes and total pin length of first row

Fig. 4.8 Critical fluid static head for different orifice diameters

Fig. 4.8 shows that the static head decreases rapidly with increasing orifice diameters. For the case of the 1 *mm* orifice diameter found in Section 4.4.1, a liquid level of 1 *mm* in the distribution reservoir is then needed to maintain a stable droplet flow.

A higher level would lead to *We* numbers above the critical limit, meaning that the droplet flow would become progressively unstable until transition to jet flow would occur.

5 Heat and Mass transfer model - Part II

In this Chapter, the solution for the species and energy conservation equations are given along with the assumptions and boundary conditions used for the solution process. Particular focus is given to the droplet formation phase and an analysis of the solution behavior is provided. The work regarding the development of the experimental test bench is then described in detail, before presenting the validation procedure and results [90].

5.1 Modeling

The energy and species conservation equations have been solved with respect to the 120° rhomboidal pin, which is the pin shape that was found to be the best option for a good absorber design in the first part of the model development, presented in Chapter 4. The droplet flow in the absorber is subdivided into three main stages:

1. Droplet free fall occurring in between two subsequent pins in the same pin column
2. Droplet impact onto the pin and rapid film flow around the same
3. Droplet formation occurring at the bottom side of the pin

The physical properties of the LiBr-H₂O have been calculated with the correlations by Yuan and Herold [89] and via the EES software, except for the mass diffusivity value, which is taken from Bo et al. [60]. Nominal values are presented in Table 5-1.

Table 5-1 Main physical properties for a 60% LiBr solution at 303 K

Density [kg/ m ³]	1685	Specific heat [J/kg K]	1888
Thermal conductivity [W/m K]	0.4139	Mass diffusivity [m ² /s]	1.6e-9
Viscosity [Pa s]	6.5e-3	Surface tension [N/m]	0.0945

5.1.1 Droplet free fall

The free fall of a droplet is usually simulated assuming a spherical geometry. However, as the space separating two subsequent pins in the same column is expected to be small, the free fall time is small enough to use the semi-infinite medium approximation. For instance, if a distance between two subsequent pins is taken as 2 cm, the free fall time would correspond to approximately 60 ms. The characteristic length of the problem is then calculated by $\sqrt{D * t} \sim 1\mu\text{m}$, where D is the mass diffusion coefficient. This length is three orders of magnitude smaller than the droplet radius. If the heat transfer problem is considered, the

thermal diffusivity of a 60%w LiBr-H₂O solution is $\alpha = 1.28e - 7 \text{ m}^2/\text{s}$, and the corresponding characteristic length is approximately $\sim 87 \mu\text{m}$, two orders of magnitude smaller than the droplet diameter. As a result, the semi-infinite medium approximation can be applied.

With this assumption, the species and energy conservation equations reduce to the Eqs. (5.1-5.2).

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial r^2} \quad (5.1)$$

$$\frac{\partial C}{\partial t} = D \frac{\partial^2 C}{\partial r^2} \quad (5.2)$$

Here T is the temperature and C is the LiBr concentration in the solution expressed in mol/m³.

The boundary conditions are given in Eqs. (5.3-5.8) and visualized in Fig. 5.1.

Fig. 5.1 Boundary conditions for the droplet fall absorption model

In Fig. 5.1, H_{lv} is the latent heat of condensation, P the pressure. Towards the center of the droplet, which corresponds to an infinite distance from the point of view of the absorption process, the solution temperature and concentration are unchanged with respect to the initial values (Eqs. 5.3-5.4, 5.7-5.8). A heat flux is imposed for the energy equation at the disc interface, which is equal to the mass absorbed at the interface multiplied by the latent heat of condensation (Eq. 5.5). At the interface, thermodynamic equilibrium of the water vapor surrounding the droplet and the water inside the LiBr-H₂O solution is assumed. This translates to an equilibrium concentration (Eq. 5.6) imposed at the interface, which is a function of the

operating pressure and the temperature of the vapor and the solution. The formula for the equilibrium concentration calculation is given by Yuan and Herold [89].

$$C_{r \rightarrow \infty} = C_i \quad (5.3)$$

$$T_{r \rightarrow \infty} = T_i \quad (5.4)$$

$$q = H_{lv} D \rho \left(\frac{\partial C}{\partial t} \right)_{r=0} \quad (5.5)$$

$$C_{r=0} = C_{eq}(T_{r=0}, P) \quad (5.6)$$

$$C_{t=0} = C_i \quad (5.7)$$

$$T_{t=0} = T_i \quad (5.8)$$

The solutions of Eqs. (5.1-5.2) derive from the application of a similarity variable given by $\eta = r/\sqrt{4\alpha t}$ to reduce the partial differential equation in space and time to an ordinary differential equation in the similarity variable η [91]. The solutions are given in Eqs. (5.9-5.10).

$$T(r, t) = T_i + \frac{q}{k} \left[\sqrt{\frac{4\alpha t}{\pi}} \exp\left(-\frac{r^2}{4\alpha t}\right) - r \operatorname{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) \right] \quad (5.9)$$

$$C(r, t) = C_i + (C_{eq} - C_i) \operatorname{erfc}\left(\frac{r}{2\sqrt{Dt}}\right) \quad (5.10)$$

As this is a conjugate heat and mass transfer problem the two equations need to be evaluated iteratively for each timestep to return the correct profiles. Subdividing the time into smaller timesteps also allows the iterative update of the constant heat flux q , increasing the accuracy of the prediction.

Each iteration starts with an assumption on the absorbed mass flux of water vapor at the solution-vapor interface, defined by Eq. (5.11).

$$G(t) = \frac{\sqrt{D} (C_{eq} - C_i)}{\sqrt{\pi t}} \quad (5.11)$$

From this, the heat flux can be obtained with Eq. (5.5). This is used to solve Eq. (5.9) and obtain the temperature profile. From the temperature value at the solution-vapor interface, the new value of C_{eq} can be calculated, which leads to a new assumption of the absorbed mass flux G via Eq. (5.11). The iteration is repeated until the error of convergence of G falls below $1e-5$.

Once the convergence is reached, the mass of water vapor absorbed for the current time step is given by Eq. (5.12), and the calculation for the next time step can begin.

$$M_w(t) = G(t) * A_{surf} * (\Delta t_j) \quad (5.12)$$

At the end of the simulation time, the intermediate values of water vapor absorbed can be summed up to return the total absorption for the droplet fall phase.

5.1.2 Droplet impact and flow around the pin

When the solution droplet impacts on the subsequent pin, a full-mixing phenomenon is assumed, as a result of the high turbulence generated [92–94]. This translates into a normalization of the mass fraction (and the concentration with it) and temperature profiles, so that in the phase of droplet formation, a uniform initial condition can be used. The resulting uniform value of the temperature is calculated in Eq. (5.13) as the mass averaged value over the entire droplet, based on the profile obtained at the last time step of the droplet fall phase.

$$T_i' = \frac{\sum_{j=1}^N T_j * \Delta m_j}{\sum_{j=1}^N \Delta m_j} \quad (5.13)$$

The new value for the solution mass fraction is obtained by applying the principle of mass conservation of the LiBr fraction in the droplet, with Eq. (5.14).

$$X_i' = \frac{X_{i*} M_d}{M_d + \sum M_w} \quad (5.14)$$

5.1.3 Droplet formation

The modeling of the droplet formation phase is based on the droplet profile model of Chapter 4. In there, the input of the radius of curvature B was originally required by the user for the model to return the correct droplet profile. To develop a heat and mass transfer model for a forming droplet that is applicable a priori (i.e. without the need of an experimentally-based input) a new condition is set.

The coordinate Z of the droplet tip is used as an input constraint for the correct calculation of the droplet profile. To obtain the transient development of the vertical coordinate of this point, an equation from the balance of surface tension and gravity forces was made. An acceleration term resulting from the opposite action of gravity and surface tension $a = (M_d * g - \pi \sigma_s d) / M_d$ is calculated.

Given a chosen instant in time t_0 then, the corresponding z_0 of the droplet (Fig. 5.2) is calculated with $z_0 = 0.5 a t_0^2$. The droplet profile is then calculated using the system of Eqs. (5.5-5.8), by iteratively using different values of B until a matching z_0 is found. Additionally, a condition is set to check that the resulting droplet volume is equal to or smaller than the volume of the droplet generated by the orifice. This is calculated with Eq. (5.8), using the boundary condition set by Eq. (5.13), so that the accurate contact point between the droplet and the pin can be identified.

In the next step, the droplet profile obtained is sliced into multiple cylindrical discs; the thickness of each disc is obtained by dividing the total height z_0 by a chosen slice number N , $\Delta z = z_0 / N$, while the disc radius R_0 is given by the r coordinate corresponding to the $i \Delta z$ ($i = 1, \dots, N$) vertical coordinate, i being the slice index. For each slice, the transient

equations for energy and species conservation are used for the cylindrical geometry under study, according to Eqs. (5.14-5.15).

Fig. 5.2 Droplet slicing for the heat and mass transfer model, with boundary conditions used for each slice

To solve these equations, the Fourier Series method is used. The method has been used in the literature to solve the heat and mass transfer problem for the falling film case, as reported by Grigor'eva and Nakoryakov [95,96] and Grossman [97]. Later works by Mortazavi et al. [98] and Meyer [99–101] have explained that the model is not effective throughout the entire liquid domain because of differences in the boundary conditions at the inlet and at the film interface. However, these differences are relative to the solution falling film phase and are not present in the current problem, as the focus is restricted only to the formation of the droplet.

$$\frac{\partial T}{\partial t} = \alpha \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) \quad (5.14)$$

$$\frac{\partial C}{\partial t} = D \left(\frac{\partial^2 C}{\partial r^2} + \frac{1}{r} \frac{\partial C}{\partial r} \right) \quad (5.15)$$

$$\left(\frac{\partial T}{\partial t} \right)_{r=0}, \left(\frac{\partial C}{\partial t} \right)_{r=0} = 0 \quad (5.16)$$

$$k \left(\frac{\partial T}{\partial t} \right)_{r=r_0} = D \rho \left(\frac{\partial C}{\partial t} \right)_{r=r_0} H_{lv} = q \quad (5.17)$$

$$C_{r=r_0} = C_{eq}(T_{r=r_0}, P = P_{\infty}) \quad (5.18)$$

$$T_{t=0} = T_i, C_{t=0} = C_i \quad (5.19)$$

The boundary and the initial conditions are given by Eqs. (5.16-5.19) and reported in Fig. 5.2 as well. The solutions must be bounded at the center of the cylinder disc, which returns the conditions in Eq. (5.16). A heat flux is imposed for the energy equation at the disc interface, which is equal to the mass absorbed at the interface multiplied by the latent heat of condensation (Eq. (5.17)). At the interface, thermodynamic equilibrium of the water vapor surrounding the droplet and the water inside the LiBr- H₂O solution is assumed. This translates in the equilibrium concentration (Eq. (5.18)) imposed at the interface, like previously done in Section 5.1.1 for the spherical droplet free fall absorption model. The absorber is assumed to be adiabatic, so no cooling of the solution is provided through the pin.

The analytical solution for Eqs. (5.14-5.15) was obtained by first solving the respective homogeneous problem. From this, the solution was rewritten to identify the corresponding Green function. This can then be used to calculate the complete solution given by the actual non-homogeneous boundary condition, (Neumann type in the case of the energy equation, and Dirichlet type for the species equation) by treating them as source terms [102].

$$T(r, t) = \frac{2}{r_0^2} \sum_{n=0}^{\infty} \exp(-\alpha \lambda_n^2 t) \frac{J_0(\lambda_n r)}{J_0^2(\lambda_n r_0)} \int_{r'=0}^{r_0} J_0(\lambda_n r') T_i r' dr' + \frac{2\alpha q t}{r_0 k} + \frac{2q}{r_0 k} \sum_{n=0}^{\infty} \frac{1}{\lambda_n^2} \frac{J_0(\lambda_n r)}{J_0(\lambda_n r_0)} [1 - \exp(-\alpha \lambda_n^2 t)], \quad J_1(\lambda_n r_0) = 0 \quad (5.20)$$

$$C(r, t) = \frac{2}{r_0^2} \sum_{n=0}^{\infty} \exp(-D\lambda_n^2 t) \frac{J_0(\lambda_n r)}{J_1^2(\lambda_n r_0)} \int_{r'=0}^{r_0} J_0(\lambda_n r') C_i r' dr' + C_{eq} - C_{eq} \frac{2}{r_0} \sum_{n=0}^{\infty} \frac{1}{\lambda_n} \frac{J_0(\lambda_n r)}{J_1(\lambda_n r_0)} [1 - \exp(-D\lambda_n^2 t)], \quad J_0(\lambda_n r_0) = 0 \quad (5.21)$$

The analytical expressions of the solutions were finally implemented in a Matlab model that couples the two solutions with the droplet geometry following the procedure in Fig. 5.3.

As for the case of the droplet free fall absorption model of Section 5.1.1, the mass of water vapor absorbed at each time step of the simulation is calculated with Eq. (5.12) for every disc composing the droplet. A summation of all the intermediate values in space and time of water vapor absorbed by each disc of liquid solution will return the total mass absorbed.

Fig. 5.3 Solution procedure for the heat and mass transfer problem

5.2 Focus on droplet formation model behavior

To better understand the behavior of the droplet formation model, an analysis of the solution is given for the transient behavior of a single droplet forming at the bottom of the 5 mm wide 120° rhomboidal pin.

The initial LiBr mass fraction is taken in the range used by previous studies [103,104]. The initial temperature is chosen so that its value is smaller than the equilibrium value corresponding to the initial concentration and the working pressure $T_i < T_{eq}(C_i, P)$; the initial chosen values are reported in Table 5-1.

Table 5-2 Initial values for the heat and mass transfer model

T_i [K]	X_i	C_i [mol/m ³]	P [kPa]
308	0.6	3.801e+4	1

The implicit assumption made by using uniform values in Table 5-1 is that the droplet, once it impacts on the pin, becomes fully mixed, as described in Section 5.2.2. At the starting point of the droplet formation phase then, uniform temperature and concentration distributions can be assumed.

5.2.1 Temperature and Concentration behavior for a single, forming droplet

The solution procedure of Fig. 5.3 has been applied to a single droplet at three different times t_0 equivalent to 15, 20 and 25 ms respectively. The profiles obtained can be seen in the Fig. 5.4.

Fig. 5.4 Droplet profiles for the three cases under study

In Fig. 5.5 the temperature and concentration profiles for the values of Table 5-1 are given for the three droplet profiles. The effect of the large difference in the thermal and mass diffusivity values can be observed by a different distribution in all the three profiles.

Heat can penetrate deeper in the droplet than the absorbed water vapor, which is shown to deposit in a very narrow region near the droplet-vapor interface. In both cases, however, a large bulk of liquid is unaffected by the heat and mass transfer phenomena and its properties are left unchanged. This shows that, even with an optimized geometry and droplet flow, large part of the solution used still does not take part in the absorption process, causing a low liquid utilization factor that might prevent the absorber downsizing.

Even though full mixing was assumed as the falling droplet impacts the pin [92–94], during the droplet formation phase no internal droplet circulation was considered for the heat and mass transfer calculations. This can be seen in Eqs. (5.14-5.15), which express the heat and mass transfer only as pure diffusion, without including the advective terms that would indicate an additional mixing effect. The experimental work done by Skelland and Minhas [105] reported that the measured mass absorption was higher than that predicted by previous theoretical models which did not include the internal circulation.

Fig. 5.5 Temperature and concentration (with detail) distributions for the three droplets. Initial uniform $T=308\text{K}$ and LiBr mass fraction $X=0.6$

They argued that this circulation can be attributed to several factors, such as droplet growth speed, turbulence created by the impinging jet from above the droplet, interfacial turbulence and liquid residues from previous droplets formed and detached. In this study the orifices generating the droplet are sized so that no jetting regime can be achieved. This was ensured by Eq. (4.2) which sets the critical flow rate value for the dripping-jetting transition.

The effect of the residual liquid on the pin has been minimized by choosing the optimized rhomboidal pin shape with low liquid retention, but it would still be present. As for the other circulation-inducing factors (growth speed, turbulence), their effect may vary from case to case (geometry, size, droplet volume, etc.) and could only be captured by complex numerical modelling which would make it harder for such a model to be later applied as a tool for absorber preliminary design optimization. For these reasons they were not considered in the model, so to keep a simpler and more fundamental approach, which can be more readily and easily implemented with any programming language.

Fig. 5.6 Mass flux profiles for the three droplets. Initial uniform temperature $T=308\text{K}$ and LiBr mass fraction $X=0.6$

Fig. 5.6 shows the absorbed mass flux measured at different droplet height, z . The droplet-pin contact point is identified by the coordinate $z=0$ for all the three cases considered, the same criterion used for Fig. 5.5. A trend can be observed in all three cases, where the mass flux seems to be higher in the middle region of z values, i.e. in the region where the droplet radius is higher. This might be caused by the faster heat diffusion that, in the regions with lower radius, such as the smaller slices seen in Fig. 5.2 closer to the droplet tip and the droplet-pin contact point, might cause the equilibrium temperature to be reached faster than in the other

regions of the droplet. As the solution concentration at the vapor-liquid interface gets closer to the equilibrium value, the driving gradient for the mass absorption becomes smaller, causing a decrease in the mass flux.

5.2.2 *Comments*

Although the results have shown that the penetration depth of heat and mass is not large, the results of Chapter 4 have shown that smaller droplets, generated by smaller orifices, can be used if a correct choice of the pin geometry is made. In particular, it was shown that the 120° rhomboidal geometry was the best one for this purpose. This allows a larger vapor-liquid interface area to be available for the same mass of solution during absorption.

Even with low heat/mass flux values then, the net mass of water vapor absorbed with an optimized pin geometry can still be higher than a non-optimal configuration that can only operate with larger droplets, be that one with inefficient pins or no pins at all.

To better explore this, the effect of the pin size on the total mass of water vapor absorbed has been assessed. Nine pins of the 120° rhomboidal type with different widths (ranging from 2 mm to 10 mm) were considered. The three phases of the droplet flow (fall, impact, formation) were modeled for each one of them, and a series of experiments has been performed to validate the model results.

5.3 **Experimental setup and procedure for absorption process analysis**

The validation of the model presented in Section 5.1 requires the development of an experimental setup capable of replicating the proposed absorber design and measure the corresponding water absorbed, whose value would be compared with the predicted values of the proposed model. Here the steps taken to develop such system are presented.

Two iterations of the system are presented. The first one aimed at replicating a complete absorption chiller following the concept of an open layout, where all the components of the cycle were separated, contrary to the common layout with two vessels, one for the generator/condenser side and one for the absorber/evaporator. This was done to allow measurement of the intermediate values of temperature, pressure and mass flow rate.

Testing revealed a critical problem concerning the vacuum tightness of the system. Stable vacuum pressure could not be achieved due to the many fittings and connectors used to develop the system, and it thus prevented from obtaining a full operational setup.

This led to the second system iteration, which replicates the low-pressure evaporator/absorber compound. This simplification led to a successful, stable operation of the system, and was the one used for the experimental validation process.

5.3.1 First setup iteration: complete absorption chiller

The first iteration of the experimental setup follows the scheme indicated in Fig. 5.7 and the end results is shown in Fig. 5.8.

Fig. 5.7 Absorption chiller scheme. Three pipelines can be distinguished, namely, from the top: solution return line, water vapor line, solution supply line

The system is designed to work at 1 kPa absolute pressure, corresponding to a saturation temperature of 7°C, in the low-pressure side of the system which comprises absorber and evaporator. The desorber and condenser nominal operating pressure is set at 5.6 kPa, corresponding to a saturation temperature of 35°C.

Fig. 5.8 Developed first iteration of the experimental setup

The estimation of the nominal cooling power of the system depends on the absorber operation and it was thus impossible to accurately set it a priori. The value was then estimated by assuming an optimistic value for the absorption flux of 0.01 kg/m²s higher than the highest values reported by Mortazavi et al. [62]. Considering an active area for the absorption process of 0.01 m², calculated as the product of the absorber cylinder height and diameter, the water vapor mass flow obtained is 0.1 g/s, equal to 250 W of cooling power.

Following this indication, the operating pressures and the cooling power were used as input in the absorption model developed for the cooling technologies scoring described in Chapter 2. The model finally returned the rating powers of the condenser, desorber and absorber heat exchangers, along with the nominal solution mass flow rate required. Table 5-2 summarizes the nominal operating conditions of the system.

Table 5-3 Nominal operating conditions for the experimental absorption chiller

Name	Value	Unit
Evaporator-Absorber Pressure	1	[kPa]
Condenser-Desorber Pressure	5.6	[kPa]
Refrigerant mass flow rate	0.1	[g/s]
Weak solution mass flow rate	7	[g/s]
Weak solution concentration	55	[%]
Strong solution concentration	65	[%]
Evaporator heat	250	[W]
Condenser heat	262	[W]
Absorber heat	316	[W]
Desorber heat	328	[W]

5.3.1.1 Absorber and Desorber chambers

The chambers (Fig. 5.9) were custom manufactured; they are constituted by a stainless-steel cylindrical vessel, with a removable vacuum-tight lid on the top to facilitate maintenance.

Fig. 5.9 Chamber for the absorber and desorber

The vessels present a lateral viewport, a top and a bottom KF16 port, a KF25 and a KF16 lateral port. The K25 port is used to connect the chambers to the vacuum pump via a ball valve. The remaining KF16 ports are used to connect the water line and the solution supply and return lines.

5.3.1.2 Heat exchangers and hot/cold water supply

The pre-cooling of the solution entering the absorber and the preheating of the solution entering the desorber is performed by two dedicated copper double pipe heat. Same devices have been used for the evaporator and the condenser heat exchangers. The H₂O-LiBr solution and water vapor are distributed in the outer side of the double pipe.

Water at ambient temperature is used as cooling water for the absorber and condenser; given the low operating pressure of the evaporator, corresponding to a saturation temperature of 7°C, the same water source was used for the heat load of the evaporator. An external pump and manifold system are connected to a water reservoir and supplies the water to the inner pipe of the heat exchangers.

The heat supplied to the desorber heat exchanger is instead provided by a hot water bath with supply water temperature and flow rate control.

5.3.1.3 Problems encountered and subsequent modifications

During the system testing and troubleshooting, several problems related to the system vacuum tightness were identified. Vacuum tests were performed on each of the three lines of the system (water, solution supply and return). Each of the lines was initially completely vacuumed; after the measured pressure reached a near zero value, the vacuum pump was shut down and the line isolated by closing the corresponding ball valves. From this point onwards, the pressure inside the line was monitored to measure the infiltration rate in terms of Pa/hr.

The results of these tests are presented in Fig. 5.10, and they highlight the presence of high infiltration rates that reach up to 7 kPa/hr. The cause of this is the high number of

connections, fittings and adapters. The net effect of all these sources of infiltration constituted a critical bottleneck for the correct operation of the system.

Fig. 5.10 Leak-up rate test for the three main lines of the system

It needs to be reminded that the low-pressure side of the system must be kept at a pressure of 1 kPa to work in nominal conditions. The absorption process relies on the difference between the saturation water vapor pressure of the solution and the current water vapor pressure in the absorber chamber. Any air infiltration would lower this differential and could eventually prevent it altogether. The values seen in Fig. 5.12 were not compatible with a proper operation of the absorber. Some actions were taken to address the problem:

- ❖ The fittings were changed from the KF type to the compression type. This included substitution of all ball valves (except for the KF25 connected to the vacuum pump) and tubing (stainless steel, OD 6 mm).

- ❖ The custom-made double pipe heat exchangers presented damaged threads at the outlet ports that generated additional infiltrations. They were substituted with factory made plate heat exchangers.
- ❖ The solution pump connections presented very high infiltration rate. It was then substituted with a new one mounting compression fittings (TCS MGD-1000)

Additional vacuum tests performed after the above changes produced apparent improvements in the infiltration rate, but the desired condition still wasn't ultimately met. As a result, a new, simplified iteration of the experimental system was developed.

5.3.2 Second setup iteration: Absorber-Evaporator compound

As the focus of the experiment is the falling droplet in the absorber, this setup constitutes the simplest reduction of the absorption chiller that can still allow the validation of the absorption model presented in Section 5.1. A scheme is given in (Fig. 5.11). The real system is pictured in Fig.5.12.

Fig. 5.11 Scheme of the second iteration of the experimental setup.

Instead of measuring the water absorption rate by means of the two flowmeters, as described in Section 5.3.1, in this system the absorption is measured by weighing a sample of $\text{H}_2\text{O-LiBr}$ solution before and after passing through the absorber section, using an electronic precision scale (see Section 5.3.2.3 for details).

The difference between the two would provide the net water absorbed, and this would be the quantity used to validate the model.

Following the scheme of Fig. 5.11 and the real system in Fig. 5.12, water at ambient temperature is directly injected into the evaporator vacuumed vessel through a valve connected to the vessel top lid, causing water flashing which results in a partial evaporation of the water injected. This fraction of water vapor constitutes the refrigerant that would normally be generated in the evaporator by the application of a heat load.

On the absorber side, the $\text{H}_2\text{O-LiBr}$ solution is supplied from the top of the column and its flow regulated to ensure a droplet regime is maintained. Droplets are generated by a smaller reservoir located in the transparent section of the absorber column. Droplets the impact on the tested pin sample, positioned so that its horizontal axis of symmetry is placed at 10 *mm* distance from the orifice.

After impact, droplets form at the bottom side of the pin sample and detach. The subsequent fall ends at the bottom of the absorber column, where the solution is stored, and a solution sample is finally extracted for measurement.

A consequence of the water injection procedure is that the operating conditions of this system are now revised, with respect to the previous iteration. Specifically, the system is now designed to work at the saturation pressure corresponding to the temperature of the water injected into the evaporator chamber.

This choice is a consequence of the available electronic precision scale used for the measurement process. As explained earlier, the validation of the model relies on the measure of the change in solution density, which in turn relies on a correct measurement of the weight of a solution sample before and after passing through the absorber.

Fig. 5.12 Developed experimental setup

Operation at a higher pressure was necessary to obtain a higher absorption rate, thanks to the larger pressure differential between the water vapor and the solution saturation value. This would then provide a larger difference in mass of the measured sample, which is well above the resolution of the electronic precision scale used.

5.3.2.1 *Evaporator*

The cylindrical vessel previously used as the absorber chamber (Section 5.3.1.1) is here used to store the refrigerant water.

To keep the pressure to a stable level throughout the experiments, the chamber is enveloped by an electrical heating strip which is manually controlled to increase the evaporation rate of the water inside the evaporator and compensate for the vapor loss caused by the absorption process. Because of this balance, large changes in the operating pressure can be avoided.

The bottom outlet of the vessel is used to discharge excess water and to connect a thermocouple, while the lateral KF16 port is used to directly connect the evaporator the absorber.

5.3.2.2 *The absorber*

The absorber is constituted by a column subdivided into two main components (Fig. 5.13).

Fig. 5.13 Main components of absorber column (left) metering valve (right) absorber test section

At the top, a small reservoir of 20 ml capacity is used to store the $\text{H}_2\text{O-LiBr}$ used during the experiment. The reservoir is connected to the manual metering valve (Swagelok SS-4L) and to the absorber region of interest, constituted by a borosilicate glass KF16 fitting (Fig. 5.13). An additional glass KF25 fitting was later used to accommodate the wider pins and avoid interference of the pins with the side wall of the glass fittings, so to ensure more accurate results (Refer to Section 5.4 for details).

Two elements are fitted inside this section. First, a smaller, 3D-printed reservoir (approximately 10 ml) orifice is placed in the upper section of the transparent fitting. Two sizes of the orifice diameter have been tested, namely 1 mm and 1.5 mm.

Secondly, the 3D-printed sample of the pin under test is placed at 10 mm distance from the orifice outlet. The transparency of this section is necessary to visualize the solution flow during the experiment. The metering valve can then be used to regulate the flow and ensure a droplet regime is established.

Nine 3D-printed samples of the rhomboidal 120° pin have been manufactured, representing the nine pin widths ranging from 2 mm to 9 mm (Fig.5.14).

Fig. 5.14 Smaller reservoirs for the 1 mm and 1.5 mm orifices and pins tested

The bottom part of the column is constituted by another 20 *ml* reservoir isolated by a top and a bottom ball valve to extract the solution at the end of each experiment.

5.3.2.3 *Sensors*

The new setup iteration mounts one pressure sensor (Inficon CDG-020) mounted on the connecting fitting between evaporator and absorber. The sensor has an accuracy of 1%, with a resolution of 0.05% of the full scale value, equal to 1100 bar. The temperature of the solution at the top reservoir and the temperature of the water at the bottom of the evaporator chamber is measured by two K-type thermocouples.

Fig. 5.15 Thermocouple assembly (left) and pressure sensor with controller (right)

Finally, an electronic precision scale (Sartorius TE46) is used to measure the weight of the solution sample collected in each experiment. The measuring range of the scale is 0 to 60

grams, with a resolution of 0.1 mg; reproducibility of the scale is equal to 0.1 mg, with an accuracy of 0.02 mg.

5.3.2.4 *Data logging and control*

Data from the pressure and temperature sensors was collected by an NI cDAQ-9174 mounting the NI 9401, 9213 and 9219 modules.

A Labview interface was used to record the data at steps of 1 second and save it in csv format. The same interface was also used to control the solution pump operation through a PWM control of the speed.

Finally, a DC power supply was used to power the pressure sensor.

Fig. 5.16 Data collection system cDAQ with 9401/9213/9219 modules

5.3.2.5 *Experimental procedure*

For the experiments all possible combinations of pins and orifice sizes have been tested. A summary of the experimental combinations used is given in Table 5-3.

Table 5-4 Summary of experimental combinations

Pin size	Orifice size
2~10 mm	1 mm
2~10 mm	1.5 mm

The experiments are executed according a consistent schedule to ensure no bias is created for any pin sample. Each experiment has a duration of approximately 5 minutes, since only a small amount of solution is used. The preparation of each experiment however requires a longer time, and is constituted by the following steps:

1. The fittings of the absorber column are disassembled and cleaned with dry paper. Additionally, the transparent section is thoroughly washed with water to avoid stains that can worsen visibility.
2. The absorber column is re-assembled, and a vacuum test is performed:
 - a. The vacuum pump is connected to the evaporator chamber and all the valves are opened to allow complete vacuuming, aside from the top metering valve and the bottom extraction valve in the absorber column.
 - b. After the monitored pressure reaches a value < 200 Pa, the vacuum pump is shut down and the system isolated.
 - c. Pressure is constantly monitored along with the water temperature measurement until the pressure value reaches the saturation value corresponding to the temperature reading. If the pressure increases from the saturation value, the infiltration rate, is above 50 Pa/min, additional cleaning and tightening is done. The vacuum test is then repeated until the infiltration rate falls below 50 Pa/min. This value is deemed sufficient to prevent a negative effect of the infiltrated air on the results, given the short duration of the experiment and the relatively high vapor pressure.

3. An amount equal to 20 ml of H₂O-LiBr solution is taken from the solution bottle after agitating it to ensure well mixing. This is loaded in the upper reservoir of the absorber column from the solution bottle. The reservoir is closed with a lid to prevent ambient pressure to affect the solution flow through the metering valve.
4. The experiment begins when the metering valve is gradually opened, and the solution flow is visually monitored. The valve opening is then constantly adjusted to ensure a stable droplet flow throughout the experiment. A heating strip is powered if the pressure falls by more than approximately 200 Pa from the saturation value, by effect of the absorption. The power is cut off when the pressure falls more than 100 Pa above the saturation pressure.
5. When the solution in the upper reservoir is exhausted, the upper lid is opened to bring the system pressure back to ambient.
6. The discharge valve at the bottom of the absorber column is opened and a sample of approximately 10 *ml* is extracted for measurement. The remaining solution is extracted and stored in a discharge bottle.

The procedure is repeated with all the other samples. At the end of each experimental session, the solution samples for each experiment performed are collected, along with a solution sample taken directly from the solution bottle (the “baseline”).

The solution samples are then taken to the measuring station. Here an empty test bottle is firstly weighted on the electronic scale. The latter is then zeroed, and a fixed volume of 5 *ml* fluid is taken with a syringe from the tested solution sample and weighted. Ten measurements are taken for each sample following the same steps.

The entire procedure is repeated for the other solution samples and for the baseline, not before cleaning the test bottle in between two sets of the ten measurements, to ensure no cross contamination occurs.

Weights recorded are finally exported to a Microsoft Excel file for further analysis.

5.4 Validation procedure

5.4.1 Mass of water absorbed – by experiment

The experimental value of water absorbed for each solution sample extracted is calculated with the following procedure:

1. The recorded weights of the solution samples are divided by the corresponding volume to obtain the solution density.
2. A droplet volume equal to the one calculated with Eq. (4.1) is considered.
3. Droplet volume is multiplied by the density of the “baseline” sample to obtain the initial droplet weight. The same volume is then multiplied by the density of the solution sample extracted after the experiment (i.e. after the absorption process), to obtain the final droplet mass.
4. The mass of water absorbed is obtained by difference of the two droplet masses previously calculated. This value is then compared to the one provided by the model for the final validation.

5.4.2 Mass of water absorbed – by model

5.4.2.1 Additional assumption for the validation procedure

For the validation procedure, the model presented in Section 5.1 has been used with the inclusion of three additional assumptions.

1. The fall time is calculated as the sum of the free fall time to cover the distance from the orifice to the pin, and an additional term that accounts for the initial droplet growth at the orifice that has been observed during the experiments. The first is calculated with the standard formula for the free fall time given by $t_{fall} = \sqrt{2 * h/a}$, where a is an acceleration term resulting from the opposite action of gravity and surface tension $a = (M_d * g - \pi\sigma d)/M_d$. The second is obtained

by dividing the droplet volume obtained from Eq. (4.1) by the critical volume flow rate obtained from Eq. (4.3).

2. The modeling of the droplet formation phase presented in Section 5.1.3 is done by assuming that a single droplet profile is maintained for the entire timespan. This is the profile corresponding to the necking point, the critical condition occurring before droplet detachment in which the radial coordinate of the droplet is smallest. This profile is calculated with the model presented in Chapter 4. As for the timespan of the droplet formation phase, this is again corresponding to the time leading to necking. This is calculated with Eq. (5.22), provided by Clanet and Lasheras [83].

$$t_{form} = \frac{1}{0.326} \sqrt{\frac{\rho R_d^3}{\sigma}} \quad (5.22)$$

3. An additional phase is introduced, which corresponds to the absorption occurring in the residual solution being retained by the pin. The fraction of volume retained for the 120° rhomboidal pins can be estimated with the experimental correlation in Eq. (5.23) derived by Babu [88].

$$\varphi_{ret} = 2.96935 - 4.71385 * B + 2.32845 * B^2 \quad (5.23)$$

where the value of the normalized radius of curvature B corresponds to the value used to derive the droplet profile at the necking point. The retained volume is finally calculated by multiplying φ_{ret} by the original droplet volume generated by the orifice.

This volume is then used to evaluate the additional term of absorption. A half sphere geometry is assumed, and the model of Section 5.1.1 is applied. The timespan of this phase is taken to be equal to the droplet growth time: since there is no wait time between the growth onset of two subsequent droplets.

5.4.2.2 *Modeling procedure*

To summarize the modeling approach for the validation, the sequence of steps taken is here presented.

1. The droplet volume generated by the orifice used during the test is calculated with Eq. (4.1);
2. The absorption during the first droplet fall, from the orifice to the pin, is solved with the Eqs. (5.9-5.10);
3. Droplet impact and film flow is reduced, according to the assumption made in Section 5.1.2, to a full mixing phase during which the absorption is considered negligible. The droplet temperature and concentration are then normalized using Eqs. (5.13-5.14);
4. The absorption problem during the droplet formation regime is solved with the use of Eqs. (5.20-5.21), and the time span of the phase is calculated with Eq. (5.22). The mass of water absorbed in this phase is calculated again with Eq. (5.12);
5. The residual fluid retained by the pin is calculated with Eq. (5.23), and the equations for the absorption problem during the droplet formation regime are used again for this case;
6. The second droplet fall, from the pin to the bottom of the absorber column, is calculated again with Eqs. (5.9-5.10). The time span of this phase corresponds to the free fall time from the pin to the bottom of the absorber column. The mass of water absorbed is then calculated with Eq. (5.12);
7. The total mass of water absorbed by the droplet is finally calculated by summation of the intermediate phases values.

5.5 Validation results

The results of the experiments performed on the pins and the corresponding operating conditions have been summarized in Appendix C. Here only the results functional to the validation process are reported, to streamline the analysis of the results.

The experimental results have been compared with the model presented in Section 5.1, used according to the procedure summarized in Section 5.4.2.2, and also with the simplified model of Jeong and Garimella [77].

They proposed an alternative model for the absorption of water vapor during the droplet formation phase, where the droplet is idealized as a half sphere, independently of the shape onto which the droplet formation occurs. Adiabatic condition was assumed for this phase. The formula of the mass of water vapor absorbed is given by Eq. (5.24).

$$M_w = \rho K_{form} \frac{\pi}{2} d_a^2 t_{form} (X_i - X_{eq}) \quad (5.24)$$

Where d_a is the diameter of the half sphere having the same volume of the original droplet and K_{form} is the mass transfer coefficient taken from Skelland and Minhas [105]. This considers a short droplet formation time, below 1.5 seconds, which induces internal circulation in the droplet; this leads also to the assumption of uniform temperature inside the droplet and the expression of K_{form} given in Eq. (5.25).

$$K_{form} = \frac{24}{7} \left[\frac{D}{\pi t_{form}} \right]^{1/2} \quad (5.25)$$

In Eqs. (5.24-5.25) t_{form} is the droplet formation time, a function of the jetting-to-dripping transition Reynolds number, the Galileo number and the tube diameter [77] as seen in Eqs. (5.26-5.27).

$$t_{form} = \frac{M_d}{2m\dot{\phi}} \quad (5.26)$$

$$\varphi = \left(0.836 \frac{2\pi\sqrt{3}}{\sqrt{1 + 2(\varepsilon/d)^2}} - 0.863 \frac{Re}{Ga^{1/4}} \right) \varepsilon \quad (5.27)$$

Where ε is the capillary length and Ga is the Galileo number.

5.5.1 Results for the 1 mm orifice

The experimental results for the 120° rhomboidal pins coupled with the 1 mm orifice are plotted in Fig. 5.17 and compared with the proposed model results and the results of the simplified model of Jeong and Garimella [77].

Fig. 5.17 Experimental results and comparison with the model from Jeong and Garimella [77] and proposed model – 1 mm orifice

5.5.1.1 Experimental data

The experimental results of the mass of water absorbed show an initial increase with the pin width, with a faster rate of increase seen in the 3-5 mm range.

Fig. 5.18 Experimental observations for the interaction between droplet and pin for the 1 mm orifice case.
Sequence starting from the 2 mm wide pin (a) ending at 10 mm (i)

Between the 7 mm and 8 mm width, a decrease in mass absorbed seems to be observable, although the error on the measurement given by the error bars indicates that no precise confirmation of this can be obtained. During the experiments it was in fact observed

that, for the wider pin, the droplet would form only from half portion of the pin, leaving the other part unwetted, as can be observed in Fig. 5.18 (e)-(i). This poor wetting would then translate in a different droplet profile, more similar to the ones observed for the smaller pins, causing the lower performance.

Values for the 9 mm and 10 mm pins have not been reported. This is because the large width of the pins interfered with the absorber column structure during the experiments. As the droplet impacted on the pins, the solution would often flow against the column walls, since little space was left between pin and walls. Because of this, a film flow was generated, which prevented a consistent operation of the system and invalidated the absorption results by providing a much larger value of mass absorbed (see Fig. 5.19).

Fig. 5.19 Film flow onset due to narrow space for the droplet formation on larger pins

This is a result of the additional phase of film flow that was not present during testing of the smaller pins. Details on the actual values obtained can be found in Appendix C.

5.5.1.2 Results of Jeong and Garimella model for the 1 mm orifice case

The model from Jeong and Garimella [77] overestimates the results for all the pins considered. Furthermore, opposite relation between pin width and mass absorbed is observed. Higher values of mass absorbed are obtained for small pins widths, and the values then decrease for wider pins.

The discrepancy in the values calculated can be attributed to the mass transfer coefficient used, which is taken from Skelland and Minhas [105]. They used a coefficient fitted to their experimental results, during which many sources of internal mixing were identified, as previously described in Section 5.2.1. These possible sources were the droplet growth speed, the turbulence created by the impinging jet from above the droplet, the interfacial turbulence and the liquid residues from previous droplets formed and detached.

Because of the controlled conditions of the present experiments, the influence of these phenomena was reduced. In particular, droplet growth speed was controlled via manual actuation of the metering valve to avoid a faster rate that would create risk of droplet coalescence. Moreover, the valve was controlled to impose a constant droplet flow, so the effect of the impinging jet was avoided.

The opposite relationship between mass absorbed and pin width instead can be attributed at the modeling equations used to evaluate the droplet formation time. The formula provided in [77] presents an inverse correlation between formation time and tube diameter, here substituted with the pin width, and this causes the behavior seen in Fig. 5.17.

5.5.1.3 Results of the proposed model for the 1 mm orifice case

The proposed model instead follows the experimental data in Fig. 5.17 in a similar way, and for all the pins considered the model estimation falls within the error band calculated for the experimental data. An initial increase is obtained for the 3-5 mm pins although the values

modeled are higher than the measured ones, and the rate of increase does not change visibly with respect to the experimental data.

Values of the mass absorbed then stabilize between 5-7 *mm* pins to a lower value of ~195 μg (Fig. 5.17). In this phase a shift is observed, with the model underpredicting the experimental data.

Lastly, instead of a decrease in mass absorbed between 7-8 *mm* pins a small increase is observed. This can be explained by the poor wetting mentioned earlier in Section 5.4.1.1, which cannot be modeled with the current model but was indeed measured during the experiments.

5.5.1.4 Discussion on findings

From this analysis, it can be concluded that the proposed model is able to reproduce the experimental data with higher accuracy. For smaller pins the model is more effective in capturing the poor performance due to the fast formation than the Jeong and Garimella [77] model. For larger pins instead, the model still offers a good accuracy in the value estimation but is not capable of capturing the change in direction observed. This suggests that for larger pins the discrepancy between model and experimental data would likely increase.

However, an important conclusion of this analysis is that an optimal range of pin widths exists, within which a high and stable absorption can be achieved. If the pin size is below this range, poor performance is obtained, caused by short time available for the droplet formation. If a pin is above the optimal range, poor wetting of the surface causes a partial surface availability for the droplet formation which leads to a sub-optimal droplet profile and thus to a performance loss.

5.5.2 Results for the 1.5 mm orifice

The experimental results for the pins coupled with the 1.5 *mm* orifice are plotted in Fig. 5.20 and compared again with the Jeong and Garimella [77] model.

Fig. 5.20 Experimental results and comparison with the model from Jeong and Garimella [77] and proposed model – 1.5 mm orifice

5.5.2.1 Experimental data

The behavior of the experimental data is similar to the one observed for the 1 mm orifice. An initial increase in the mass of water absorbed is observed, with a faster rate of increase in between 5-6 mm width. This steeper increase is shifted towards higher pin width values with respect to the 1 mm case, where the increase was instead observed in between 4-5 mm.

This might be a direct consequence of the larger droplet generated by the 1.5 mm orifice. Before, pins smaller than 4 mm were not able to sufficiently slow down the droplet and thus caused short formation time and poor performance. It is reasonable to conclude then, that a larger droplet would require larger pins to sufficiently slow down its fall, and this can then explain the shift of the steep increase area.

Between 6-7 mm the values of mass absorbed suggest that the area of stabilized values, seen in Fig. 5.20, has been reached. However, the experimental results for the 8-10 mm could not be obtained because of the same additional component of film flow that was observed in the previous round of experiments.

5.5.2.2 *Results of Jeong and Garimella model for the 1.5 mm orifice case*

In this case, the prediction of the model in [77] is closer to the experimental values than the proposed model. However, large differences are observed for pin widths larger than 5 mm. This difference can be attributed to the assumption of a half sphere geometry used, which differ from the real droplet profile observed. The larger solution-vapor interface surface area of the real droplet with respect to a half sphere returns a higher absorption.

5.5.2.3 *Results of proposed model for the 1.5 mm orifice case*

The results of the proposed model instead (blue line in Fig. 5.20), largely underestimate the experimental data. A cause of this can be found in the sub-optimal experimental conditions during the testing of the 1.5 mm orifice. It was in fact observed that during the tests, the droplet generation rate could not be slowed down to the same level achieved when the 1 mm orifice was tested.

The faster generation rate has two root causes. First, the larger orifice requires a lower static solution head in the reservoir of Fig. 5.14 to be maintained to ensure a stable droplet flow. This can be seen in Fig. 4.8, which reports the critical static head separating dripping and jetting flow regime. Secondly, the metering valve only allows a relatively coarse regulation, which causes a high admission of fluid into the orifice reservoir.

The result of the combination of these two causes is that even with a small valve opening, too much solution would be allowed into the reservoir. This would then make the orifice reservoir approach the critical static head point and would then trigger a faster rate of droplet generation.

Because of the faster generation rate, the solution mixing-inducing phenomena listed by Skelland and Minhas [105] could not be effectively mitigated like for the smaller orifice. For example, the previously mentioned impossibility in reducing the fast rate of droplet generation would increase the mixing effect due to the consequent higher droplet growth speed.

Furthermore, the faster generation rate would as well increase the risk of a forming droplet to be perturbed by the following impinging droplet. All these factors can contribute to increase the droplet solution internal mixing, which in turn explains the high measured values of mass absorbed.

As mentioned in the model behavior description in Section 5.2.1, the influence of these factors was not considered in the model development, causing then the discrepancy observed in Fig.5.20. In fact, unless these sources of disturbance could be mitigated, complex numerical models would be required to describe the magnitude of their impact on the absorption process. Consequently, the use of the presented model as a tool for absorber preliminary design optimization would be hindered by the potentially large increase in the computational effort required.

For these reasons it was chosen to retain the relatively less complex model and benefit from its lower computational requirement and identify an alternative approach to improve the accuracy of the model.

In their work, Skelland and Minhas [105] state that with respect to an unperturbed case, values of mass absorbed two to three times larger can be obtained. Following this rule of thumb, Fig. 5.20 reports two additional curves corresponding to a double and triple value of mass absorbed during droplet formation, respectively.

It is then observed from the figure that most of the experimental data collected falls within the band comprised by these two curves, suggesting once more that the discrepancy is caused by the enhanced solution internal mixing.

If the internal mixing is not considered, it is however shown that the 1 *mm* orifice is more effective than the 1.5 *mm*. This can be measured if the ratio of the modeled mass of water vapor absorbed and the droplet volume are compared. The values in Table 5-4 in fact show that although the gross amount of water vapor absorbed is higher for the 1.5 *mm* orifice, the solution with the 1 *mm* orifice is used more effectively and a lower circulation ratio is obtained.

Table 5-5 Ratio of mass absorbed and droplet volume

Orifice\Pin	Mass absorbed to droplet volume ratio [g/m ³]					
	2	3	4	5	6	7
1.5 mm	9.76	10.66	11.24	11.79	12.76	13.41
1 mm	11.10	11.86	12.93	13.69	13.50	14.03

If the internal mixing of droplet generation were to be promoted for the 1 mm orifice by increasing the droplet generation rate, the same multiplication factor on the absorption rate could be obtained.

Like it was observed for the 1 mm orifice case, here an optimal range of pin widths can be found, although shifted towards larger values of the pin widths. In the 1 mm orifice case, this range is found between 5 and 7 mm. In the present case of 1.5 mm orifice, the range lower bound can be located at 6 mm, but it is unclear whether the range upper bound is located at 7 mm or at a wider pin.

5.5.2.4 Experimental results for the optimal range location

To clearly identify the optimal width range for the 1.5 mm orifice case, the experimental setup was modified to accommodate the larger pins and avoid the film flow phase observed in the past tests (see Fig. 5.19). The glass fitting was substituted with a larger, KF25 type, and the experiments were performed in the range of interest between 5 mm and 10 mm. The smaller pins were not considered, as the lower bound of the optimal range was already established at 6 mm in the previous round of tests shown in Fig. 5.20. As a result, the 5 mm pin is a sufficient starting point to identify the increase and stabilization of the water absorbed values.

The results of these experiments are given in Fig. 5.21.

Fig. 5.21 Results for the second run of experiments on the 1.5 mm orifice. Identification of upper bound of optimal width range

The experimental values collected present values of the mass of water absorbed that are lower than the ones of Fig. 5.20. This difference is caused by the lower LiBr mass fraction used for these experiments, which reduced the concentration gradient between the equilibrium value and the solution real value. Details of the experimental conditions and results can be found in Appendix C.

From the results observation it is now possible to identify the upper bound of the optimal range. In fact, between the 8 mm and 9 mm pins a dramatic drop in the value of mass absorbed is seen, with a partial recovery observed at the 10 mm pin, which might be explainable with a more favorable tradeoff between the worsening effect of the partial pin wetting during droplet formation, and the additional mixing obtained in the upper half of the pin via droplet impingement on the retained fluid. As explained for the 1 mm orifice case, the drop is due to both the partial wetting of the pin, that leads to a non-optimal droplet profile during formation, and to the excess solution being retained on the upper half of the pin, which prevents an effective utilization of the solution for the absorption.

With this second round of tests it was then finally possible to identify the optimal range of width between 6 *mm* and 8 *mm*, which is shifted by 1 *mm* with respect to the optimal range found for the 1 *mm* orifice.

5.5.2.5 *Discussion on findings*

The validation process on the 1.5 *mm* orifice has shown that the proposed model underestimates the experimental data when internal mixing of the solution is present. The Jeong and Garimella [77] model is supposed to include this mixing effect in the estimation, via an experimental correction factor for the mass transfer coefficient. Despite this, their prediction is still far from the experimental data, especially for wider pins, because of the simplification of the droplet geometry used.

However, if the multiplication factor suggested by Skelland and Minhas [105] is applied to the proposed model the accuracy of the prediction is enhanced, and for wider pins (6-7 *mm*) the prediction is close to the experimental observation, with an error of 2-3%.

Finally, thanks to another round of tests performed on the orifice, it was possible to identify again an optimal range of pin widths that can ensure a high performance, as was already observed for the smaller 1 *mm* orifice.

5.5.3 *An explanation for the optimal width range identification*

The identification of an optimal range of pin widths for both the orifices used led to the final phase of the results analysis, in which a theoretical explanation is provided for the phenomenon observed.

To explain this behavior, an approach based on energy conservation was used. This approach is often used to describe the spreading of a droplet after impact on a flat surface, such as in Wildeman et al. [106]. In the case of the droplet impacting on the pin, the initial energy

of the droplet before impact is the sum of two terms, namely the kinetic energy EK of the droplet and the surface energy ES of the spherical droplet. These can be expressed by Eq. (5.28) and Eq. (5.29).

$$EK_i = \frac{\pi}{12} \rho d_d^3 u_i^2 \quad (5.28)$$

$$ES_i = \pi \sigma_s d_d^2 \quad (5.29)$$

Where u is the droplet speed before impact and d_d the droplet diameter.

After impact, the droplet can be assumed to spread over the surface and take the form of a thin disk. If the surface is large enough, the kinetic energy of the disc will eventually be entirely converted into the surface energy of the disc, that has then reached the maximum spread diameter. The corresponding energy components are then expressed by Eq. (5.30) and Eq. (5.31), assuming fully wetted conditions at the pin surface.

$$EK_{end} = 0 \quad (5.30)$$

$$ES_{end} = \frac{\pi}{2} \sigma d_{disc}^2 \quad (5.31)$$

By equating the total energy before and after the impact, the maximum spreading diameter of the solution disc D_{disc} can be calculated.

In the case of the pins under study, the available surface is not infinite, and the spreading width of the disc is constrained by a length value equal to the sum of the two edges of the upper half of the pin, as seen in Fig. 5.22.

Fig. 5.22 Scheme of the impact and spreading of a droplet onto the rhomboidal pin

As a result, if this length is substituted in Eq. (5.26), the maximum possible surface energy value of the solution disc is obtained (Eq. 5.32)

$$ES_{max} = \frac{\pi}{2} \sigma_s (W_{pin} / \cos(\pi/6))^2 \quad (5.32)$$

If the initial energy of the droplet $EK_0 + ES_0$ were to be higher than this value (ES_1), it would mean that, for energy conservation, EK_1 would be positive. Consequently, the solution disk would still be moving after reaching the pin edges, and this would result in a fast droplet formation phase as the initial velocity of the droplet would be higher. Conversely, if the initial energy of the droplet were equal or lower than the maximum surface energy of the disk, then the solution would slow down, and ideally stop, before reaching the pin edges. This would translate in a slower formation phase, caused by a lower initial droplet velocity.

Following this principle, the difference $EK_0 + ES_0 - ES_{max}$ was calculated for all the pins and orifices used. The speed V_0 was calculated as the terminal velocity of the droplet free fall, based on the distance between the orifice and the pin vertex (Fig. 5.22).

In Table 5-5 the results of the energy balance are presented.

Table 5-6 Difference between the initial energy and the maximum liquid disc energy. Values in Joule.

Orifice/Pin	2	3	4	5	6	7	8	9
1 mm	4E-06	3E-06	2E-06	0	0	0	0	0
1.5 mm	6E-06	5E-06	3E-06	1E-06	0	0	0	0

A negative value represents a condition in which the maximum surface energy of the disc is larger than the initial droplet energy before impact. This means that the fluid disc flow is ideally completely slowed down before it can reach the pin edge.

The calculated values show that indeed the point of arrest is placed between the 4 *mm* and 5 *mm* for the smaller orifice, and between 5 *mm* and 6 *mm* for the 1.5 *mm* orifice. These two points correspond to the transition points between the region with low absorption rates and the region high, stable absorption rates seen in Fig. 17 and Fig. 21 respectively.

The experimental results for both orifice sizes suggested that an upper limit of the optimal range exists as well, but this is affected by the wettability of the pin. Prediction of the wettability is a difficult task as it possesses an intrinsic random component. However, for the design of an efficient absorber, it is not necessary to identify the upper bound. For similar values of absorption rate, like they have been observed from the experiments, choosing the smaller pins size is the preferred choice as this would reduce the fabrication cost and weight of the absorber without incurring in performance loss.

This result then suggests that identifying the lower bound of this optimal range through an energy balance criterion represents a simple but effective tool to estimate the optimal size required for the rhomboidal pin. The proposed heat and mass transfer model can then be used to quantify the corresponding absorption rate and finally estimate the absorber performance.

5.6 Summary of results

In Chapter 4 and 5 the model equations describing the behavior of the proposed absorber design have been presented. The first part dealt with the identification of the most suitable geometry of the pins, that being the one which can retain the least amount of liquid after the droplet formation and detachment. This is to ensure that the absorber could operate with smaller droplets and maximize the liquid-vapor interface surface area.

An analytical model to describe the geometry of a forming droplet was used and validated with experiments. Experimental results showed that the model can accurately predict the droplet geometry closer to the tip; however, the disturbance created by the interaction between the droplet and the pins caused a lower accuracy near the liquid-pin contact point.

The final results showed that the 120° rhomboidal pin is the most suitable geometry for the absorber design, and it was then used as the basis for the second part of the modeling.

In the second part, the heat and mass transfer occurring in the absorber was modeled. An analytical model based on the energy and species conservation equations was used to describe the droplet fall, impact and formation phases. The model results were then validated through experiments to compare the real and expected mass of water vapor absorbed during operation.

The results showed that the model can predict the results with higher accuracy only if the droplet generation rate is slow enough to avoid the onset of internal mixing phenomena. As the droplet generation rate increases the effect of internal mixing is stronger. Using a multiplication factor, already observed in the literature, can be applied to the mass of water vapor absorbed during the formation phases to improve the model accuracy.

Experimental observation has also highlighted the existence of an optimal range of pin widths particular for every orifice size. A model to predict the minimum viable size to obtain a high absorption rate was then developed through an energy balance calculated on the falling droplet. The results showed good agreement with the experimental observation and identified the 5 mm wide pin and the 6 mm wide pin to be the optimal size for the 1 mm and 1.5 mm orifices, respectively.

The model results show a higher effectiveness in the absorption when smaller droplets are used. Internal mixing effects can greatly improve the absorption rate; this was observed for the 1.5 mm with a fast droplet generation rate, but it can be expected for any other orifice size.

For these reasons the 5 mm wide 120° rhomboidal pin, coupled with a 1 mm orifice is the optimal pin shape and size for the new absorber design.

This configuration is used in the next Section to evaluate the overall performance of the absorber. The case study used in Chapter 2 to compare the heat-to-cool conversion technologies is considered again, to show that the introduction of the new absorber design can further improve the characteristics of the absorption chiller, in particular regarding the cooling power flux (CPF) and the specific investment cost (SIC).

6 Application of results to a full-scale absorber

In this Chapter the results obtained from the modeling of the new absorber are used to evaluate the potential performance of a full-scale device. The optimal size of the orifice and the rhomboidal pin found, equal to 1 *mm* and 5 *mm* respectively, are used to evaluate the relationship between the absorber size and the corresponding solution flow rate allowed. Then, the heat and mass transfer model is used to provide an estimation of the cooling power obtainable with the current size and flow rate.

The case study presented in Section 2 is then considered again to evaluate the beneficial effect of the substitution of the original absorber with the newly designed one. In particular the improvement on the Cooling Power Flux (CPF) caused by the introduction of the new absorber design is investigated to show the downsize potential of the ABS chiller. Lastly, the performance maps developed in Section 2 are updated with the new performance indexes, to show the overall performance improvement.

6.1 Absorber design characteristics

With the chosen pin and orifice sizes, the overall design of the absorber and the optimal operating flow rate can be determined. The scheme of the absorber is presented again in Fig.

6.1.

Fig. 6.1 Front and side view for a pin finned adiabatic absorber

6.1.1 Solution distribution in the absorber

Similar steps as the ones followed in Section 4.4.2 are used: with Eq. (4.2-4.4) the critical mass flow rate for each of the 1 mm orifice is found, above which it will be impossible to generate a stable droplet flow. Eq. (4.5) then gives the minimum allowed distance between the orifices.

As the number of droplet generating orifices increases, correspondingly longer pins are needed to receive these droplets. The cumulative length of each row of pins is then varied and the corresponding number of orifices is calculated with Eq. (6.1).

$$N_{or} = L_{row,tot}/\lambda \quad (6.1)$$

In Eq. (6.1), N_{or} is the number of droplet generating sites on each row of pins, equal to the number of orifices. By multiplying N_{or} for the critical mass flow rate per orifice the maximum total mass flow rate is obtained. Fig. 6.2 maps the relationships between length and mass flow rate for the new absorber.

Fig. 6.2 Flow rate map for different total pin length

Given the 1 mm orifice diameter found in Section 4.4.1 of the design process, the map of Fig. 6.2 can be readily used to get an estimation for the absorber size. For instance, if a target mass flow rate of 0.05 kg/s is assumed, the map returns an overall pin length of approximately 20 m for an absorber based on 5 rows, while for an absorber with 20 rows the length increases to 40 m. This total pin length can be realized by using multiple shorter pins or few longer ones, depending on the geometrical constraints of the application. For the 20 rows absorber for example, 10 columns of pins, each 20 cm long, can achieve the desired total length ($20 \times 10 \times 0.2 = 40$).

This constitutes an important advantage for the absorber design. Since the absorption process is not depending on the cooling occurring at the tubes of a standard falling film absorber the droplet generation sites can be distributed with higher flexibility. This results in an absorber

that can offer a large variety of form factors and can thus be adapted each time to the required application. This is in line with the intended objective of the work, which is to develop a solution that can make this heat-to-cool conversion technology applicable in areas where standard systems could not be fitted due to geometrical constraints. The total pin length, along with the orifice size and the pin shape obtained from Chapter 5 constitute the main geometrical features needed for the absorber design.

6.1.2 Cooling power estimation

To better quantify the performance of the pin-finned absorber configuration, the cooling power obtainable during the droplet formation phase is calculated for the 1 mm orifice and the 5 mm wide rhomboidal pin, with varying total pin length and different number of rows to show how the flexible absorber layout described in Section 6.1.1 influences the potential cooling output.

The heat and mass transfer model described in Chapters 4 and 5 has been applied iteratively for each row of pins considered. The initial values for the solution temperature and concentration have been set equal to the ones used by Mortazavi et al. [62] and are reported in Table 6-1. In their study the authors reported the resulting absorption rate from several works available in the literature, and it is thus a good benchmark for comparison.

Table 6-1 Initial values for absorption testing

T_i [K]	X_i	P [kPa]
298	0.57	1.2

The model returns the solution concentration obtained after each row. Knowing then the initial concentration and the mass flow rate corresponding to a given total pin length and number of rows (Fig. 6.2) the total water vapor mass flow rate can be calculated with Eq. (6.2). Finally, the total cooling power associated with the water absorbed during droplet formation phase can be calculated with Eq. (6.3).

$$\dot{m}_w = \dot{m}_{sol,i} \left(1 - \frac{X_i}{X_{end}} \right) \quad (6.2)$$

$$\dot{Q}_{cold} = \dot{m}_w \Delta h_{lv} \quad (6.3)$$

In Fig. 6.3 the resulting cooling power obtained with the new absorber is shown.

Fig. 6.3 Effect of pin size, cumulative pin row length on cooling power, for different number of pin rows.

The figure shows that the cooling power behaves similarly to the mass flow rate shown in Fig. 6.2. The cooling power increases linearly with the total pin length, but different slopes are obtained for different number of rows. As a result, for the same total pin length, a lower number of rows produces a higher cooling power. A high value is obtained for 5 rows and the power progressively reduces for larger number of rows.

This can be explained by the adiabatic nature of the absorber. At the upper rows nearer to the absorber inlet, the concentration gradient driving the absorption is high. If the total pin length is distributed on fewer rows, each row would be longer and have more droplet generating

sites. Consequently, both a larger flow rate and higher benefit from the high concentration gradient are obtained, resulting in a large cooling effect.

Conversely, if the same total pin length were distributed on more rows, each row would become shorter and allow fewer droplet generating sites. The resulting flow rate would be smaller, and the higher number of rows would not be able to compensate for it, as the concentration gradient rapidly decreases after each row, making this configuration less effective.

To visualize the difference in the absorber effectiveness the absorption rates for the different configurations is given in Fig. 6.4. For all the curves considered, some fluctuations are observed. These are due to the discontinuities deriving from the fixed distance between the orifices. As the total pin length is varied between 0 and 40 m with using a smaller step than the orifice distance, values of absorption rates are shown to have a step whenever a new orifice is “added” in the absorber, i.e., whenever the length reaches an exact multiple of the orifice distance.

The figure shows that with lower number of rows a higher absorption rate can be obtained, making the absorber more effective. The values for every configuration stabilize above a total pin length of 5 m. The highest value is reached for the 5 rows case, with an absorption rate of $1.5\text{e-}3 \text{ kg/m}^2\text{s}$.

This is due to the adiabatic nature of the absorber, which causes the solution to approach equilibrium faster and with no discontinuities caused by intermediate cooling, as is the case of the standard falling film absorbers. From this it can be concluded that the flexibility in the solution distribution expressed in Section 6.1.1 is still reflected in the cooling power output generated, which can be achieved with different absorber layouts as seen in Fig. 6.3. At the same time, this flexibility would be partially limited by the loss in the absorption rates as more pin rows are used shown in Fig. 6.4.

Fig. 6.4 Absorption rates for the proposed absorber design with no additional mixing considered

However, the adiabatic nature of the absorber allows an easier modular approach. In fact, since no cooling is provided directly at the absorber via a secondary cooling circuit (e.g. the cooling tubes of a falling film absorber), an array of smaller absorbers with an optimized layout could be used to achieve the desired performance. This constitutes an additional degree of design flexibility, since solution can not only be distributed onto an array of pins of different number and length, but also be split into more absorber modules.

6.1.3 Comparison with literature

If the absorption rate for the 5 row case is compared with the other literature results collected in [62], the absorption rate obtained for the 1.2 kPa operating pressure (point X1 in Fig. 6.5) falls in between the performance region of Sun et al. [107] and Islam [52], who investigated traditional tubular absorbers and the region of the plate absorbers studies by [62]

and by Karami and Farhanieh [61] and the works concerning the membrane absorber of Isfahani and Moghaddam [108] and Ali [109], presented in Fig. 6.5.⁶

Fig. 6.5 Absorption rates at different operating pressures. Image taken from [62] and modified with the current absorption rate value with (X2) and without additional mixing (X1)

For the proposed design, the linear mass flow rate is calculated to be 0.15 kg/min m. If compared with the range used in the works reported in [62], it can be observed that no data is collected in the lower range of linear mass flow rate by any of the studies. Indeed, falling film absorbers require a higher flow rate per unit of length as the heat convection around the tubes is of great importance for a good operation. Using a low flow rate would degrade the heat

⁶ Careful observation of the devices used by Mortazavi et al. [62] and Isfahani and Moghaddam [108] suggests that the surface area used for the absorption rate corresponds to the frontal area of the finned [62] and microchannel [108] plates and not to the total surface area available. This lower estimation might have been the cause of the relatively high absorption rates reported in Fig. 6.5.

transfer and consequently have a negative impact on the absorption rates. This has been shown by Hu and Jacobi [75] who identified a sharp decrease in the heat transfer coefficient as the flow rate decreases to the point of transition from jetting to dripping. The same conclusions have been experimentally confirmed again in a later study by Olbricht et al. [76]. The low heat transfer coefficients obtained for low linear mass flow rates would lead to a decrease in the absorption rate as the intermediate cooling at the absorber tubes would not be enough to reinstate a large driving concentration gradient.

This poses an important constraint on the absorber design, since one dimension will be constrained by the minimum linear flow rate required to achieve an effective heat transfer. This limits the flexibility of the absorber design and in turn the resulting size of the ABS chiller, which cannot easily be adapted to small-scale application that may require a non-standard form factor.

The proposed design however offers more flexibility in the design, as the pins can be organized in any combination of row and column number and pin length. For example, if a nominal power of 1 kW is considered, with distance between each pin row and column set at 10 mm, the resulting form factor of a 5 rows absorber and a 20 rows absorber would be very different. Table 6-2 presents the two possible and equivalent configurations.

Table 6-2 Possible configurations for an absorber based on 1 kW nominal power

[mm]	Width	Height	Depth
5 rows	800	50	50
20 rows	90	200	200

In addition to this, increasing the number of rows will reduce the effectiveness of the absorber (Fig.6.4). However, the reduction progressively decreases as the number of rows grows (Fig. 6.3-6.4) and thus no dramatic fall in the performance can be seen, leaving more room for the layout optimization. In addition, the adiabatic configuration used opens at the possibility of employing the ABS chiller in a modular approach, by using smaller systems that can be more easily adapted to respect the geometrical constraints of different applications. This

is thanks to the independence on the absorber performance to the solution mass flow rate and the resulting heat transfer coefficients, which would in turn greatly affect the performance of a standard absorber. As a result, the proposed adiabatic design is more suited to address the problem of the chiller downsizing and application in area where non-standard form factors are required.

Fig. 6.6 Absorption rates for the proposed absorber design with additional mixing considered

It needs to be reminded that the model presented in Section 5 was found to be underestimating the experimental results because of the additional internal mixing effect that improved the overall absorption. A multiplication factor of 2-3 was suggested by the literature [105] and its application on the proposed model provided a better prediction of the experimental results. This factor was introduced here, and the new absorption rates obtained can be seen in Fig. 6.6. The highest value is again reached for the 5 rows case, with an absorption rate of 2.5×10^{-3} kg/m²s.

The figure shows that the absorption rates are on average 50% higher than the previous case with no internal mixing, and thus the system performance is closer to the highest performing solutions seen in Fig. 6.5 (point X2).

Overall, the results show that the proposed design performs better than the most traditional solution of a tubular absorber, and it can be operated at low linear mass flow rate without incurring in performance losses due to poor heat transfer, thanks to its adiabatic nature. This allows higher flexibility in the design compared with the traditional devices, and it is thus a better solution for small-scale applications where both a small volume and particular form factors might be required.

6.2 New design comparison with baseline tubular absorber

To transfer the previous results in a real application, the scoring study presented in Chapter 2 is here considered again. The new absorber design has finally been introduced to evaluate the improvement in the cooling power flux (CPF) and its contribution to the increase in the total performance index.

The original model has been run using the equations provided in the Appendix I to establish the baseline values of cooling power obtained and mass flow rates required. Then, the baseline tubular absorber has been substituted with:

- A plate type heat exchanger for the solution sub-cooling
- The proposed adiabatic absorber

The cooling power obtained from the baseline has been used to size the adiabatic absorber accordingly, and the plate absorber has been sized to provide the required subcooling to the incoming $\text{H}_2\text{O-LiBr}$ solution.

6.2.1 Assumptions used

To show the comparison, the same assumptions made in Chapter 2 for the absorber have been used again here with some exceptions.

The input thermal power has been reduced to 1 kW to address the case of a small-scale unit. The tube diameter for the absorption chiller has been reduced to 5 mm, to make it comparable with the 5 mm wide pins used by the proposed absorber.

For the tubular absorber case a tube length of 10 mm has been used for the calculation of the linear mass flow rate required for the computation of the heat transfer coefficient [76]. During simulation, usage of longer pipes led to very poor heat transfer coefficients. The length has then been optimized not to create an unfair disadvantage for the baseline case. However further reduction below 10 mm was deemed not practical for manufacturing.

For the plate absorber, a 1 mm height of the plate has been used, and the heat exchanger has been sized to provide a 1 m/s solution speed. The heat transfer coefficient is calculated with the correlation by Marriot reported in [110].

6.2.2 Comparison results

The Cooling Power Flux obtained for the two options is shown in Fig. 6.7 for varying temperature of the heat source and condenser temperature, using the same ranges of the scoring study in Chapter 2.

Fig. 6.7 Cooling power flux for the baseline tubular absorber and the current proposed design. Dependency on heat source temperature (left) and condenser temperature (right)

The curves show a net improvement on the CPF for the new proposed layout, with an average increase of 50% on the CPF value. The reason for the increase lies in the heat transfer occurring in the two solutions. For the baseline tubular absorber, the low mass flow rate required for a 1 kW power produces a low heat transfer coefficient that consequently leads to an increase in the necessary surface area. On the contrary, the proposed absorber is not affected by the small flow rate, and the plate heat exchanger can be designed to produce a high heat transfer with a small total area.

The effect of this improvement on the total performance index however cannot be identified directly on the ABS scoring map. The scoring maps obtained for the two solutions are in fact identical (Fig. 6.8), as the normalized indexes are the same. This is a result of the chiller advantage over the other two technologies investigate in Chapter 2, namely a coupled ORC and vapor compression system (ORC-VCC) and a coupled thermoelectric generator and vapor compression chiller (TEG-VCC). Since the ABS chiller performed best in most of the occasions, the normalized index would be one, regardless of any improvement done on it.

Fig. 6.8 Performance map for the absorption chiller.

To observe the improvement obtained on the absorption chiller then, it is necessary to see how the performance index of the other two technologies changed, so to have the new relative distance between these and the modified absorption chiller. The maps are shown in Fig. 6.9.

The maps based on the new absorber design show, on average, an improvement on the overall performance. This is shown by the reduced areas observed for each of the index level from 0.5 to 0.9 compared to the same areas relative to the baseline case, which confirm that, relative to the absorption chiller, these systems performance has decreased. Furthermore, the two cases studies (A and B) used in Chapter 2, whose weighting factors are provided in Table 6-3, can be observed.

Fig. 6.9 Performance maps for the ORC-VCC and TEG-VCC systems. A reduced colored area for any index value signifies a relative decrease in performance.

Table 6-3 Weights for two possible choice scenarios

Weights (%)	COP	CPF	SIC
Case A	80	10	10
Case B	30	20	50

The map shows that for the case B, which gives more importance to the CPF, the performance index has been moved from the 0.7 to the 0.6 region for both systems, indicating that their performance has decreased relatively to the modified absorption chiller. For case A with a lower CPF weight the index falls in the same 0.9 region for both the TEG-VCC and ORC-VCC systems, although the points have been shifted downwards, indicating that a small decrease in the index has occurred. This shows that the new absorber design can increase the performance of the ABS heat-to-cool conversion with respect to the other options considered, in particular if size reduction is the most critical criterion for the technology selection.

From this analysis it can also be then concluded that the choice of employing an adiabatic absorber coupled with a plate heat exchanger can offer more flexibility. The absorber can be independently optimized to offer the best performance for the particular application both in terms of power and also in terms of footprint, without being constrained by the necessity of tailoring the design to achieve a high heat transfer. This choice also proves to have a direct impact on the design of the entire chiller performance.

7 Conclusions and Future work

The rapid increase in cooling demand that will be faced in the next years, combined with a large availability of waste heat and a growing interest in its utilization have been the motivation of this study. The conversion of waste heat into cooling power has indeed the potential to address both trends and increase systems energy efficiency. However, the large size of these systems may affect the range of possible applications and thus limit the exploitable potential. The problem of downsizing a *heat-to-cool* conversion system was then chosen as the focus of this study, looking at solutions that can enable the development a small-scale device.

7.1 Heat-to-cool technologies comparison and scoring

The first part of the study has dealt with the identification of the most suitable *heat-to-cool* conversion system. Three systems have been compared, namely a H₂O-LiBr ABS chiller, a TEG-VCC system, and an ORC-VCC system.

Three parameters (COP, cooling power flux and specific investment cost) have been compared both individually and later combined into a single performance index, which resulted into a performance map for each of the systems considered.

The maps have been developed to offer a quick tool for comparing different systems by looking not only at the parameters values alone, but also including information on the relative importance given to each of them. Given that a system cannot be the best option for every situation, using different parameters weightage can give the design engineer more insight on the applicability of a system in the specific site considered.

The comparison of the three systems was done considering a nominal 10 kW of waste heat recovered with heat source temperature varying from 100°C to 200°C and condenser temperature varying from 35°C to 50°C. Results showed a higher performance of the absorption chiller for a temperature of the heat source below 140°C in terms of COP. This is a result of the direct heat-to-cool conversion this system can provide, while for the two alternative systems TEG-VCC and ORC-VCC, constituted by two not-integrated systems, an inefficient conversion process is obtained. As the vapor compression chillers proved the highest cooling COP, the main cause of the low performance is the power generation section of the systems, which requires a high heat source temperature to generate electricity with good efficiency. Indeed, the results showed that when the temperature increases above 140°C, these systems can be more performant than the absorption chiller.

Cooling power flux was calculated to be almost double for the absorption chiller with respect to the two alternative systems TEG-VCC and ORC-VCC, while the specific investment cost was between 25% and 50% lower than the worst case of a TEG-VCC system.

Performance maps were then developed considering a case study of 10 kW of waste heat recovered from a drying process in the textile industry. A heat source temperature of 170°C and ambient temperature of 25°C were assumed, and two different cases were considered: one which prioritizes efficiency (A) and one which prioritizes savings (B).

The resulting maps showed that for case A both the ORC-VCC and the TEG-VCC systems return an index in the 0.9-1 band, thanks to the high power generation efficiency; the absorption chiller returned an index in the lower 0.8-0.9 band.

For case B instead, the absorption chiller returned a performance index in the highest 0.9-1 band, while the ORC-VCC performance fell in the 0.8-0.9 band, and the TEG-VCC showed a lower performance, in the 0.7-0.8 band.

7.2 Absorber analysis

The absorption chiller was then chosen as the most suitable system for the development of a small-scale device. Review of past works on this technology has led to the identification of the absorber as the most critical component affecting the footprint of the chiller, as a consequence of the poor heat and mass transfer occurring inside and of the high sensitivity of the system to a change in operating pressure.

Different absorber designs have been reviewed, and the adiabatic absorber has been identified as a potential way to reduce the component size. This is thanks to the possibility to independently optimize the heat transfer part and the mass transfer part by efficiently pre-cooling of the H₂O-LiBr solution in a dedicated heat exchanger, and by designing an absorber section aimed at maximizing the solution utilization during absorption.

Furthermore, a flow of falling droplets has been found to be beneficial to the absorption process, in particular during the phase of droplet formation at the bottom of a solid surface. This is thanks to the maximization of the liquid-vapor interface surface area occurring during this phase. Additionally, interrupting the fall of droplets with solid surfaces can promote the

solution internal mixing at augment the total absorption process. Following these guidelines, an adiabatic, pin-finned absorber design to be operated with a droplet flow was proposed to evaluate its potential in reducing the absorber size.

7.3 Absorber modeling - Part I

Modeling of this absorber was subdivided into two parts. In the first one, a model of the droplet formation phase based on the Euler-Lagrange equation was developed to provide an accurate geometrical domain for the solution of the heat and mass transfer problem.

The model was validated with a series of experiments on circular, elliptical and rhomboidal pins. The validation showed that the model can accurately predict the droplet geometry close to the droplet bottom apex, but near the contact point between the droplet and the pin, larger deviations were observed. These are attributable to the distortion effect caused by the fluid present in the upper half of the pin and by an imperfect separation between the upper and lower pin halves.

In fact, results showed that for the rhomboidal pin with a 120° angle of aperture, so with an acute angle at the pin sides, the model prediction is more accurate, with an average error of 15.5%.

After validation, the model was used to identify the best pin geometry to be utilized in the absorber. The 120° rhomboidal pin coupled with an orifice of 1 mm diameter was found to be the configuration, as it can maximize the liquid-vapor interface surface area by reducing the risk of droplet coalescence through a reduction of the retained fluid volume, calculated to be 0.56 mm^3 .

Given the orifice and pin sizes, an estimation of the overall size of the absorber could be made through the production of a flow rate map. The results revealed that for an assumed solution mass flow rate of 0.05 kg/s, applicable to a small-scale absorption chiller, a total pin length of the first row of 4 m, which can be distributed in different ways to accommodate a

wide range of geometrical constraints. However, in order to ensure a stable droplet flow, very low static head has to be maintained in the upper reservoir; for the case considered, this value was calculated to be 1 *mm*.

7.3.1 Future work for the pin geometry optimization

Further optimization of the pin shape might be aimed at increasing the separation between upper and lower half of the pin. Thanks to this, higher accuracy of the droplet formation model could be achieved for these optimized shapes. However, a side effect of this would be the onset of droplet coalescence in the upper section of the pin, which would decrease the absorber performance.

The use of a non-symmetric pin shape seems then to be the best option for further optimization. Both the upper and lower halves of the pin should be optimized to avoid excessive liquid retention, however the different boundary conditions of the two problems related to the droplet-pin interaction would lead to different choices on the respective geometry features.

7.4 Absorber modeling – part II

In the second part of the modeling procedure, an analytical heat and mass transfer model was developed to describe the absorption process with the chosen rhomboidal pin geometry. The model is based on the solution of the energy and species conservation equations for the different phases of a droplet flow, namely free fall, impact on pin, and formation at the pin bottom surface. The model was applied to rhomboidal pins of different size, from 2 *mm* to 10 *mm* width, to find the optimal value for the component manufacturing.

An experimental setup was developed to reproduce the operation of the absorber and validate the absorption model on the different pins considered.

The experimental validation showed that for slower droplet generation rates the model can predict the results with good accuracy. This is the case of the 1 *mm* orifice; here a fast rise of the absorber water vapor was observed, with a subsequent stabilization at 210 μg which is predicted well by the model (9.5% error). Finally, a decay in the absorption is observed, caused by poor wetting of the pin surface. The previous model by Jeong and Garimella [77] was not able to accurately predict these results, because of the simplified prediction on the droplet shape during the formation phase (i.e. hemispherical shape).

However, as the generation rate increases, additional mixing contributions caused by phenomena such as the impact of droplets from above and surface turbulence produces larger absorption rates, in the order of 300-700 μg , with respect to the predicted values (200-300 μg). This was observed for the 1.5 *mm* orifice case, where the original model largely underestimated the experimental results. Application of a multiplication factor of 2-3, already identified in the literature, on the results concerning the droplet formation phase, improved the accuracy of the model with an error of only 2-3% for the 6 *mm* and 7 *mm* pins.

In general, operation of the absorber with a smaller orifice returned a better performance by utilizing the solution more effectively. This was measured as the ratio of the mass of water vapor absorbed and the droplet volume. Results showed that, on average, choosing the 1 *mm* orifice over the 1.5 *mm* one improves the effectiveness of the absorber by 11%.

In addition, an optimal range of pin width values was observed from the experiments. This range can be established with a simple energy conservation approach applied to the falling droplet. For smaller pins, the initial energy of the impacting droplet (surface and kinetic) is not completely converted into surface energy of the liquid disc spreading onto the pin. The residual kinetic energy of the droplet then causes a fast droplet formation phase, which produces low absorption rates. For larger pins instead, poor wetting properties were observed, leading to a decrease in performance.

In between these lower and upper bounds, a high and stable performance region exists. For the 1 mm orifice case, the smallest pin providing such high performance was found at 5 mm and thus chosen to complete the design of the absorber.

7.4.1 Future work for the absorber layout optimization

The results obtained from the application of this energy conservation approach suggest that another importance factor for the absorber optimization is the distance between pins. This value can in fact change the initial energy content of the impacting droplet, so that smaller distances might lead to the choice of smaller pins. This further optimization might have two beneficial results. First, the additional pins that would be introduced and the associated phases of droplet formation can lead to an increase in the absorption rates. Second, the use of smaller pins, although higher in number, can lead to a reduction in the total absorber area, further minimizing its footprint.

7.5 Application case

The proposed absorber design was compared with the current state of the art of the other absorber typologies investigated in the literature. The proposed design can offer absorption rates between $1.5\text{e-}3$ and $2.5\text{e-}3$ kg/m²s depending on the beneficial contribution of the mixing-inducing phenomena. With these values, the design can be positioned in between the most traditional tube-based falling film absorbers and the more performing membrane absorbers, but with an added advantage of a higher flexibility in distributing the absorber area in different number of pins and different individual length.

The new absorber design was finally used to evaluate the improvement it can offer to the original absorption chiller used for the initial systems comparison. The baseline absorber, based on a falling film onto a bank of tubes, was substituted with a plate heat exchanger and

by the proposed adiabatic absorber. Using an input heat source of 1 kW, an increase of 50% of the cooling power flux was calculated with the new configuration.

This affected the overall scoring index mapped in the first part of the thesis as well, and the modified chiller showed a relatively higher overall performance with respect to the two alternative systems. The modified absorption chiller returned a performance index between 0.87 and 0.97 for the Case A (priority to efficiency) and B (priority to cost and size). For the TEG-VCC and ORC-VCC, performance of Case B fell from the 0.7-0.8 band to the 0.6-0.7. No major difference was observed for case A, since cooling COP was not affected by the introduction of the new absorber.

The separation of the heat and mass transfer parts using two dedicated devices can then be a viable option for the developments of a small-scale absorption chiller. Advancements in the microscale heat transfer can be fully exploited to provide effective subcooling of the working solution. This consequently grants the necessary flexibility in both the design of new and more complex geometries for the absorber, and in the choice of cheaper materials for its fabrication (e.g. polymers), thanks to the adiabatic nature of the device

Appendix A – Heat-to-cool technologies modeling equations

Models equations

ABS

Evaporator

$$\dot{Q}_{cool} = \dot{m}_{10}h_{10} - \dot{m}_9h_9 \quad \text{A.1}$$

$$\dot{m}_{10} = \dot{m}_9 \quad \text{A.2}$$

Absorber

$$\dot{m}_1 = \dot{m}_6 + \dot{m}_{10} \quad \text{A.3}$$

$$x_1\dot{m}_1 = x_6\dot{m}_6 \quad \text{A.4}$$

$$\dot{m}_2h_2 + \dot{m}_4h_4 = \dot{m}_3h_3 + \dot{m}_5h_5$$

$$\dot{Q}_{abs} = \dot{m}_{10}h_{10} + \dot{m}_6h_6 - \dot{m}_1h_1 \quad \text{A.5}$$

Generator

$$\dot{Q}_{gen} = \dot{m}_4h_4 + \dot{m}_7h_7 - \dot{m}_3h_3 \quad \text{A.6}$$

Condenser

$$\dot{Q}_{cds} = \dot{m}_7(h_7 - h_8) \quad \text{A.7}$$

ORC

Heater

$$\dot{Q}_{gen} = \dot{m}_{orc}(h_3 - h_2) \quad \text{A.8}$$

Turbine

$$\dot{W}_{tur} = \dot{m}_{orc}(h_3 - h_{4,is})\eta_{is} \quad \text{A.9}$$

Condenser

$$\dot{Q}_{cds} = \dot{m}_{orc}(h_4 - h_1) \quad \text{A.10}$$

Pump

$$\dot{W}_{pump} = \frac{\dot{m}_{orc}(p_2 - p_1)}{\eta_{is}} \rho_1 \quad A.11$$

TEG [36]

$$\dot{Q}_{gen} = \alpha_{sb} I T_H + K \Delta T - \frac{1}{2} R_{el} I^2 \quad A.12$$

$$\dot{Q}_{cds} = \alpha_{sb} I T_C + K \Delta T + \frac{1}{2} R_{el} I^2 \quad A.13$$

$$\alpha_{sb} = \frac{L_{leg}}{\sigma A_{leg}} \quad A.14$$

$$K = \frac{k A_{leg}}{L_{leg}} \quad A.15$$

Seebeck coefficient α_{sb} , electrical and thermal conductivity (σ_{el} , k_{el}) are calculated at the mean temperature:

$$T_{mean} = \frac{T_H + T_C}{2} \quad A.16$$

VCC [42]

Compressor

$$\eta_{is} = 0.85 - 0.046667(p_2/p_1) \quad A.17$$

$$\dot{W}_{comp} = \dot{m}_{ref}(h_{2,is} - h_1)/\eta_{is} \quad A.18$$

Condenser

$$\dot{Q}_{cond} = \dot{m}_{ref}(h_2 - h_1) \quad A.19$$

Expansion valve

$$h_3 = h_4 \quad A.20$$

Evaporator

$$\dot{Q}_{cool} = \dot{m}_{ref}(h_1 - h_4) \quad A.21$$

Correlations

Gnielinski [37]

$$Nu = \frac{(f/2)(Re-1000)Pr}{1+12.7\left(\frac{f}{2}\right)^{1/2}(Pr^{2/3}-1)} \quad A.22$$

$$f = (0.79 \ln(Re_{Dh}) - 1.64)^{-2} \quad A.23$$

Gungor-Winterton [37]

$$\frac{h_c}{h_{c,l}} = [1 + 3000 Bo_l^{0.86} + 1.12 \left(\frac{x}{1-x}\right)^{0.75} \left(\frac{\rho_l}{\rho_g}\right)^{0.41}] Fr_l^{(0.1-2Fr_l)} \quad A.24$$

$$Bo_l = \frac{q}{GH_{lv}} \quad A.25$$

Where h_l can be calculated using the Gnielinski correlation and G is the mass flux [$\text{kg/m}^2\text{s}$].

Chato – Jaster – Kosky [37]

$$h_c = \Omega \left\{ \frac{\rho_l(\rho_l - \rho_g)gh_{fg}k_l^3}{\mu_l(T_s - T_{wi})d} \right\}^{1/4} \quad A.26$$

$$\Omega = 0.728\alpha_g^{3/4} \quad A.27$$

$$\alpha_g = \frac{1}{1 + [(1-x)/x](\rho_g/\rho_l)^{2/3}} \quad A.28$$

Where x is the quality of the two-phase flow.

Bakhtiari et al. (ABS generator) [38]

$$h_c = 5554.3\Gamma^{0.236} \quad A.29$$

Fang (evaporating R134a) [39]

$$Nu = 0.00061(S + F)Re_lPr_l^{0.4}Fa^{0.11} / [\ln(\frac{1.023\mu_{l,bulk}}{\mu_{l,wi}})] \quad A.30$$

$$S = \begin{cases} 30000Bo^{1.13} & Bo < 0.0026 \\ 36 & Bo \geq 0.0026 \end{cases} \quad A.31$$

$$F = \left(\frac{x}{1-x} \right)^{0.95} \left(\frac{\rho_l}{\rho_g} \right)^{0.4} \quad A.32$$

$$Fa = \frac{(\rho_l - \rho_g)\sigma_s}{G^2 D_h} \quad A.33$$

Hoffmann (Absorber) [40]

$$h_c = 2000v_0^{-1.7} \quad A.34$$

$$v_0 = v/10^{-6}(m^2s^{-1}) \quad A.35$$

Double pipe HEX cost [45]

$$Co = 900f_m f_p (10.7639 * A_{hx})^{0.18} \quad A.36$$

Condenser cost [45]

$$Co = 24600(0.0108 * A_{hx})^{0.4} \quad A.37$$

Pump cost [46]

$$Co = (475.3 + 34.95\dot{W}_{pump} - 0.0301\dot{W}_{pump}^2)f_p \quad A.38$$

Where f_p is 1 for pressures under 1.03 MPa, 1.62 for pressures between 1.03 and 3.45 MPa, and 2.12 for pressures greater than 3.45 MPa.

Compressor cost [42]

$$Co = \left(\frac{573\dot{m}}{0.8996 - \eta_{is}}\right) \left(\frac{P_{high}}{P_{low}}\right) \ln\left(\frac{P_{high}}{P_{low}}\right) \quad A.39$$

Turbine cost [46]

$$Co = 0.31(0.001341\dot{W}_{tur})^{0.81} \quad A.40$$

Appendix B – Validation of droplet formation model for different pin shapes

Experimental errors

The only source of error encountered in this experimental work is related to the image quality and the post-processing activity. As explained in Section 4.2, image resolution is set at 768x768. During the profile marking, the closest pixel to the experimental droplet profile was chosen for the coordinate extraction. As a result, this translates in an accuracy of 0.13%.

Appendix C – Experimental results for absorption process

1 mm orifice case

System pressure during pins testing

Experimental errors

The pressure sensor used (Inficon CDG020) has an accuracy of 1% of the reading, with a resolution 0.05% of the full scale (110 kPa), equal to 0.055 kPa.

2-mm

3-mm

4-mm

5-mm

6-mm

7-mm

8-mm

9-mm

10-mm

Sample mass measurements and processing

N	SAMPLE MASS [g]									
	Baseline	2 - mm	3 - mm	4 - mm	5 - mm	6 - mm	7 - mm	8 - mm	9 - mm	10 - mm
1	7.862	7.782	7.796	7.783	7.775	7.781	7.792	7.779	7.769	7.712
2	7.872	7.799	7.800	7.791	7.785	7.754	7.767	7.773	7.763	7.689
3	7.863	7.808	7.791	7.805	7.784	7.749	7.760	7.783	7.764	7.683
4	7.842	7.793	7.793	7.783	7.788	7.768	7.773	7.776	7.752	7.664
5	7.845	7.806	7.780	7.771	7.777	7.778	7.771	7.745	7.749	7.666
6	7.848	7.795	7.787	7.780	7.787	7.789	7.766	7.761	7.747	7.669
7	7.827	7.806	7.784	7.766	7.794	7.755	7.762	7.773	7.759	7.702
8	7.843	7.784	7.789	7.770	7.776	7.778	7.756	7.783	7.760	
9	7.801	7.772	7.795	7.791	7.788	7.785	7.781	7.776	7.758	
10	7.807	7.772	7.786	7.785	7.776	7.766	7.751	7.783	7.744	

	Base	2 - mm	3 - mm	4 - mm	5 - mm	6 - mm	7 - mm	8 - mm	9 - mm	10 - mm
AVERAGE SAMPLE MASS [g]	7.841	7.792	7.790	7.783	7.783	7.770	7.768	7.773	7.757	7.683
DENSITY	1.651	1.640	1.640	1.638	1.639	1.636	1.635	1.636	1.633	1.618
DROPLET VOLUME [m3]	1.4E-8	1.4E-8	1.4E-8	1.4E-8	1.4E-8	1.4E-8	1.4E-8	1.4E-8	1.4E-8	1.4E-8
DROPLET MASS [mg]	23.280	23.133	23.129	23.106	23.108	23.069	23.063	23.078	23.029	22.812
MASS ABSORBED [µg]		146.461	151.330	173.538	211.037	210.591	217.123	202.041	250.614	468.055
ST.DEV		25.385	22.656	24.476	26.176	25.596	24.672	24.608	23.249	28.107

Experimental errors

Measurements are affected by two sources of error, the syringe accuracy and the scale accuracy. A 1 ml Terumo syringe was used for the measurements, with an accuracy of 5% (or 0.05 ml) according to the norm ISO 7886-1. Since for each measurement the syringe was used five times to reach the volume of 5 ml, the error is estimated to be 0.25 ml. A Sartorius TE64 scale was used for weight measurements. The measuring range of the scale is 0 to 60 grams, with a resolution of 0.1 mg; reproducibility of the scale is equal to 0.1 mg, with an accuracy of 0.02 mg.

1.5 mm orifice case

System pressure during pins testing

2-mm

3-mm

4-mm

5-mm

6-mm

7-mm

8-mm

9-mm

10-mm

Sample mass measurements and processing

	SAMPLE MASS [g]							
N	Baseline	2 -mm	3 - mm	4 - mm	5 -mm	6 - mm	7 - mm	
1	7.8691	7.8199	7.7733	7.7753	7.7369	7.7163	7.7082	
2	7.8843	7.7677	7.7673	7.7622	7.7763	7.7028	7.6803	
3	7.8672	7.79	7.8076	7.7409	7.7532	7.6895	7.6676	
4	7.845	7.7792	7.7751	7.7521	7.6945	7.735	7.7081	
5	7.8312	7.8005	7.7458	7.7686	7.7412	7.6832	7.7475	
6	7.8411	7.7664	7.7717	7.7551	7.7422	7.7473	7.6585	
7	7.8424	7.7925	7.7627	7.7575	7.7403	7.6758	7.7001	
8	7.8641	7.7959	7.7731	7.7665	7.7622	7.6805	7.6666	
		Baseline	2 -mm	3 - mm	4 - mm	5 -mm	6 - mm	7 - mm
AVERAGE SAMPLE MASS [g]		7.8518	7.7890	7.7721	7.7598	7.7434	7.7038	7.6921
DENSITY		1.6530	1.6403	1.6362	1.6336	1.6302	1.6219	1.6194
DROPLET VOLUME [m3]		2.11E-08	2.11E-08	2.11E-08	2.11E-08	2.11E-08	2.11E-08	2.11E-08
DROPLET MASS [mg]		34.8271	34.5590	34.4736	34.4190	34.3462	34.1707	34.1189
MASS ABSORBED [μg]			268.0859	353.4928	408.0503	480.9045	656.3314	708.1721
ST.DEV			28.1208	28.2778	21.0887	36.54337	40.1966	44.1563

Sample mass measurements and processing – Additional measurements

N	Baseline	5 - mm	6 - mm	7 - mm	8 - mm	9 - mm	10 - mm
1	7.7902	7.7784	7.7544	7.7532	7.7346	7.7619	7.7741
2	7.7958	7.7636	7.7545	7.7517	7.745	7.7604	7.7647
3	7.7856	7.7743	7.7564	7.7542	7.7574	7.7708	7.7509
4	7.7821	7.7715	7.7552	7.7478	7.7468	7.7671	7.7441
5	7.7895	7.7607	7.7404	7.7692	7.7414	7.7563	7.7519
6	7.7973	7.7688	7.7526	7.7724	7.7411	7.776	7.7616
7	7.776	7.7591	7.7611	7.7599	7.7466	7.7813	7.7574
8	7.7843	7.7743	7.7583	7.7486	7.7362	7.7659	7.7593
9	7.7929	7.7715	7.754	7.7598	7.7455	7.7585	7.7652
10		7.7551	7.7411	7.7672	7.7315	7.7751	7.7657

	Baseline	5 - mm	6 - mm	7 - mm	8 - mm	9 - mm	10 - mm
AVERAGE SAMPLE MASS [g]	7.7881	7.7677	7.7528	7.7584	7.7426	7.7673	7.7595
DENSITY	1.5576	1.5535	1.5506	1.5517	1.5485	1.5535	1.5519
DROPLET VOLUME [m3]	2.11E-08	2.11E-08	2.11E-08	2.11E-08	2.11E-08	2.11E-08	2.11E-08
DROPLET MASS [mg]	32.8178	32.7316	32.6687	32.6923	32.6257	32.7299	32.6969
MASS ABSORBED [μg]		86.2095	149.1214	125.5242	192.0600	87.8950	120.9311
ST.DEV		12.0205	11.5434	12.6634	11.8914	12.3887	12.6503

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