

École polytechnique de Louvain

Performance evaluation of a curved slab-shaped macro-encapsulated phase change material through experimental analysis and CFD simulations for residential latent heat storage applications

Author: **Samuel DEL FORNO**

Supervisor: **Hervé JEANMART**

Readers: **Matthieu DUPONCHEEL, Nicolas PARMENTIER**

Academic year 2023–2024

Master [120] in Electro-mechanical Engineering

Acknowledgements

I would like to express my sincerest gratitude and appreciation to the following individuals who contributed significantly to the completion of this work.

First and foremost, my heartfelt and special thanks goes to Nicolas Parmentier for his invaluable mentorship throughout this project. The many insightful discussions, his great availability, his sincere desire to help and clarify my question marks, and his trust have all enhanced to the depth and quality of this paper. He has been a real pillar of support for me.

Secondly, I would like to acknowledge Prof. Hervé Jeanmart, who provided guidance and constructive feedback on numerous occasions. His profound knowledge and expertise were instrumental in significantly improving the clarity and coherence of this master's thesis.

I am also grateful to those who ensured that this work could include an experimental part. In particular, I would like to extend my appreciation to Arnaud for his assistance in drawing up the plans and modelling the capsule, to Hervé for his dedication to the construction of the device with precision, and finally to Frank for his help with the measurement systems.

Moreover, I would like to extend my gratitude to the Destore team, in particular to Arnaud Latiers, for persuading me to become interested in thermal storage technologies, and to Achille Mathot for sharing his knowledge of the subject.

Finally, I would like to express my gratitude to my close friends and family, in particular my girlfriend and my parents, for their unwavering support and encouragement throughout this academic journey. Their assistance has been of the utmost value, and without it, this work would not have been completed in a timely manner.

Abstract

One of the strategies for achieving climate targets is the development of renewable energies. However, the intermittent nature of these sources implies a coupling with storage systems to ensure a balanced supply/demand. Among the existing technologies, latent heat thermal energy storage (LHTES) with macro-encapsulated phase change materials (PCM) is considered here. The objective of this work is to evaluate the thermal performance of a curved slab-shaped capsule in the context of residential heating applications.

Firstly, the flow conditions experienced by a capsule stacked in a thermal storage tank for the instantaneous heating of a shower were determined. The capsule was then positioned at the centre of a cylindrical experimental device, where the pre-established flow conditions were applied. The response was quantified by means of temperature sensors. Finally, the configuration was replicated in a simplified manner in computational fluid dynamics (CFD) simulations. By assuming a constant boundary temperature, the convection around the capsule was modelled in stationary mode, and the phase change was analysed in transient mode.

Experimental tests demonstrated that the phase change duration of this capsule is five times shorter than that of an equivalent volume conventional geometry. Simulations allowed to establish an empirical law for heat transfer around this specific shape. Finally, a model is proposed for determining the effective heat transfer coefficient of a storage tank based on the equivalent packed bed's porosity and the capsule's geometry.

Contents

1	Introduction	1
1.1	Context	1
1.2	Background	3
1.3	Objectives and outline	4
2	Packed bed modelling of a thermal storage	5
2.1	Reference situation	5
2.2	Specifics of the PCM capsule selected	5
2.3	Intrinsic velocity in a packed bed	7
3	Experimental analysis	9
3.1	Design overview	9
3.2	Measurement	13
3.2.1	Temperature	13
3.2.2	Flow rate	15
3.3	Experimental test rig & procedure	15
3.4	Results and discussion	16
3.4.1	Charging and discharging behaviour	16
3.4.2	Charge dynamics	21
3.4.3	Flow asymmetry analysis	23
3.4.4	Heat transfer & power characterisation	24
4	CFD simulations	27
4.1	Computational domain definition	27
4.1.1	Geometry selection	27
4.1.2	Mesh	28
4.2	Part A : Steady determination of the heat transfer coefficient	32

4.2.1	Calculation model	32
4.2.2	Grid sensitivity analysis	33
4.2.3	Results	34
4.3	Part B : Transient simulation of solidification in experimental context	38
4.3.1	Phase change modelling	38
4.3.2	Results	40
4.4	Effective heat transfer coefficient correlation	45
5	Confronting simulations and experiments	47
5.1	Comparison of transient simulation and experimental results	47
5.2	Upscaling proposal	48
6	Conclusion	51
A	Appendix	I
A.1	Additional pictures related to the experimental process	I
A.2	Test device heat capacity calculation	IV
A.2.1	Flanges temperature evolution during a charge test	V
A.3	Power losses extent determination	VI
A.4	Experimental temperature curves	VIII
A.4.1	Charges	VIII
A.4.2	Discharges	XI
A.4.3	Symmetrical sensors	XIII

Nomenclature

Symbols

T	Temperature	$^{\circ}\text{C}$
P	Power	W
E	Energy	J
t	Time	s
m	Mass	kg
V	Volume	m^3
A	Surface area	m^2
D	Diameter	m
L	Length	m
e	Thickness	m
$\dot{V}$	Volumetric flow rate of heat transfer fluid	L/min
$\dot{m}$	Mass flow rate of heat transfer fluid	kg/s
u_f	Intrinsic velocity in a packed bed	m/s
R_{th}	Thermal resistance	K/W
L_c	Characteristic length of the capsule	m
D_p	Effective particle diameter	m
Δh	Latent heat of fusion	J/kg
c	Specific heat capacity	J/(kg K)
k	Thermal conductivity	W/(m K)
h	Heat transfer coefficient	W/(m^2 K)
$\bar{T}_-$	Mean temperature over the entire test duration	$^{\circ}\text{C}$
Re	Reynolds number	
Nu	Nusselt number	
Pr	Prandtl number	
Ste	Stefan number	
$\bar{t}$	Dimensionless time	
$\bar{T}$	Dimensionless temperature	

Greek symbols

ρ	Density	kg/m^3
α	Thermal diffusivity	m^2/s
μ	Dynamic viscosity	$\text{kg}/(\text{m s})$
ϵ	Porosity	
ψ	Shape factor	
δ	Uncertainty	
Δ	Difference	

Subscripts

PC	Phase Change
inlet	At the inlet of test device
outlet	At the outlet of test device
stream	Mean of inlet and outlet
start	At the start of the test
stop	At the end of the test
bed	related to a packed bed
eff	effective
water	related to the water
capsule	related to the capsule
shell	related to the capsule shell
glob	over the entire surface
loss	associated to a thermal loss
device	related to the test device
room	related to ambient conditions observed in a room
device	related to the test device
charge	related to a charge test
discharge	related to a discharge test
inner	on basis of the internal dimensions
outer	on basis of the external dimensions
shower	related to a shower
sp	related to a spherical geometry

Abbreviation

PCM	Phase Change Material
LHTES	Latent Heat Thermal Energy Storage
CFD	Computational Fluid Dynamics

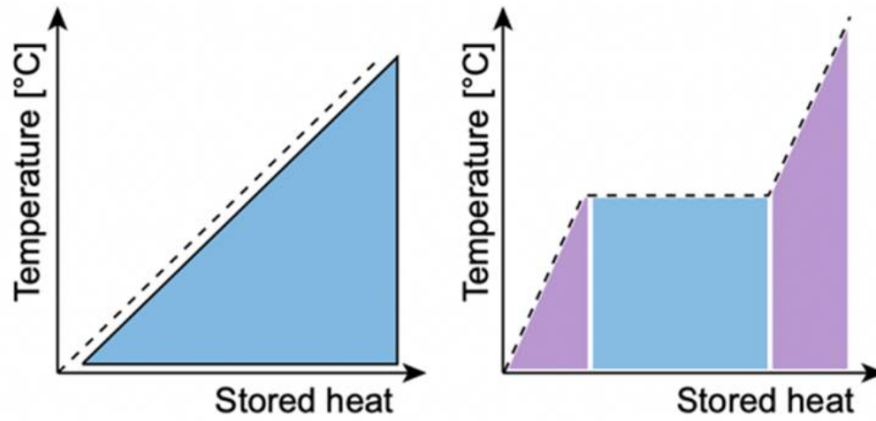
1.1 Context

In order to meet the global climate targets, it is necessary to consider a number of areas. One of these is the deployment of renewable energy production systems on a massive scale [1]. However, a significant challenge arises from this: unlike fossil fuels, energies such as solar and wind are not consistently available throughout the day, and when they are, their intensity can be unpredictable. Consequently, the development of energy systems based on these resources must be coupled with the emergence of short-term (hourly, daily, weekly) and medium-term (seasonal, annual) storage technologies. Without these, it is not possible to guarantee a supply-demand equilibrium with a large fraction of renewables in the energy mix [2].

The International Energy Agency (IEA) has recently reported [3] that there has been a continuous increase in the consumption of energy to meet heating needs, amounting to 6% over the last five years. It is therefore evident that thermal energy storage systems have a crucial role to play among all the existing storage technologies. Moreover, an analysis [1] of the residential sector reveals that sales of heat pumps have increased by 11% globally in 2022, indicating a growing preference for this technology among consumers. In this context, it has been demonstrated that the coupling of a heat pump installation with short-term thermal storage is highly relevant in order to optimise the consumption of local renewable resources, such as photovoltaic or solar thermal, or to stagger demand in relation to peak tariff times ([4], [5]). This further emphasises the necessity of considering thermal storage.

Three methods of storing energy in the form of heat can be identified ([6]–[8]). The first is thermochemical storage, whereby energy is transferred through a reversible endothermic/exothermic process. However, this technology is still in development, and its potential application in the residential sector has not yet been explored [9]. The second method consists of storing sensible heat. This is the simplest and cheapest method, as it only involves applying a temperature difference to a medium to cause it to accumulate or give off energy. The quantity of storage is therefore proportional to the volume of the medium. However, this can become problematic where technical rooms of buildings are limited in size, particularly in densely populated urban areas [10]. In such instances, latent heat storage can be highly effective. It consists of exploiting the energy associated with the phase change of a material from liquid to solid state [11]. This results in a significant increase in energy density compared to sensible heat storage ([12]–[16]), from 1.4 [17] to 5.6 [10] times better. Furthermore, as illustrated in Figure 1, latent heat storage

is associated with constant temperature energy exchange, which is advantageous for residential applications.



<https://www.ijraset.com/research-paper/pcm-thermal-storage-technology-and-its-military-applications>

Figure 1: Comparison of how energy is stored as a function of temperature in the case of sensible heat storage (left) and latent heat storage (right).

Phase change materials used in latent heat storage are commonly designated as "PCM" and will be abbreviated as such throughout this document. Within the temperature range of residential applications, spanning from 0 to 100 °C, three types of materials emerge as particularly suitable for use as PCM : salt hydrates, paraffins and fatty acids ([9], [10], [18]). However, regardless of their specific type, PCM all have a low thermal conductivity, with values ranging from 0.2 to 1 W/(m K)([11], [16], [17]). As a result, their thermal response is markedly slow. This represents a significant challenge, particularly in the context of peak load shifting ([10], [12], [13]). To address this issue, two potential solutions can be considered. The first approach involves the combination of PCM with an additive (nano-particles) characterised by a high conductivity. However, this often leads to problems of agglomeration and large losses in energy density [10]. The second approach consists of maximising the exchange surface area between the PCM and the heat transfer fluid. This can be achieved, for instance, through the use of an exchanger [11]. Exchangers represent a mature and cost-effective technology, yet this solution lacks flexibility in adapting to varying storage capacities [10]. As an alternative, it is possible to place the PCM in capsules and arrange them in a stacking configuration, thus creating a bed with a substantial exchange surface. If the capsule is less than 1 mm in width, this will be considered to be micro-encapsulation. Otherwise, it will be designated as macro-encapsulation ([16], [19]). Given that changing phase implies a change in density, the encapsulant is subjected to significant stress. Consequently, it is not uncommon to observe ruptures and leaks in the case of micro-encapsulation [15]. Conversely, macro-encapsulation is more dependable in this regard, and additionally allows for a lower ratio of shell to PCM [13].

The market is currently witnessing the emergence of PCM macro-capsules. These are sometimes presented in relatively simple shapes, such as spheres or cylinders, or the geometry can be more complex, with the aim of achieving the maximum exchange surface area within a given capsule volume. Among the manufacturers, Rubitherm/Axiotherm® is offering one of the largest selections of PCMs on the market [12]. Two of their products, in particular, stand out for their distinctive designs: HeatStixx® and HeatSel® (Figure 2). Both are available for purchase and can be installed in a water tank to convert it from sensible to latent heat storage in the simplest way possible.

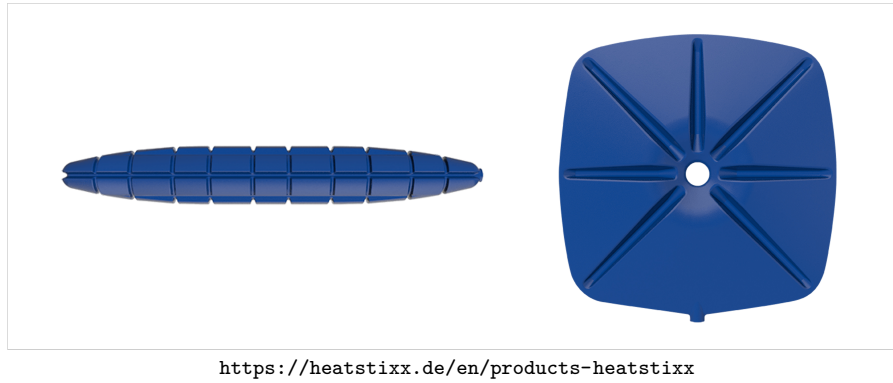


Figure 2: Photo of HeatStixx[®] (left) and HeatSel[®] (right). Two macro-capsules marketed by Axiotherm[®] to create customised latent heat thermal energy storage.

1.2 Background

A substantial corpus of references has analysed the thermal performance of PCM macro-capsules. However, the majority of these articles concern conventional geometries. A review of the literature demonstrates that since thermal storage has gained interest, an average of 17 out of 18 papers have focused on the sphere or cylinder, despite their suboptimal surface-to-volume ratios [20].

Still, four references analysing Axiotherm[®]'s capsule geometries can be found. Firstly, one paper used the aforementioned statement as a starting point, proposing a method to convert any capsule geometry to that of an equivalent sphere [11]. The authors applied this method to a bed of HeatStixx[®]s and HeatSel[®]s and compared it with experiments, showing that the model deviated from reality by 10%. This was primarily due to the fact that by reducing the capsules to equivalent spheres, non-punctual contacts between particles are not modelled. Another study focused on the HeatStixx[®] and used them to store heat for a typical Swedish house [13]. The capsule was disassembled to identify all the materials composing it. An experimental storage tank was set up with a 380 L tank and 285 capsules to determine the charging time and the amount of energy that can be exchanged. A 1D model based on the assumption of perfect stratification was validated by the experimental tests and the authors concluded that the phase change time for a HeatStixx[®] is 60% shorter than for a cylindrical capsule. Thirdly, a numerical comparison was proposed between the heat exchanger configuration, HeatStixx[®] and HeatSel[®], for storage in a combined cycle power plant [14]. It was shown that in this context, HeatSel[®]s have the best thermal response and is the cheapest solution, but the heat exchanger will always have the best energy density. Finally, experimental tests were carried out on a thermal storage system consisting of 160 HeatSel[®]s [19]. It was shown that by applying a controlled variation of the flow rate it is possible to maintain a high power exchange for a longer period of time.

In the aforementioned papers, it is evident that Axiotherm[®]'s capsules have not been subjected to a comprehensive thermal characterisation. Instead, they have been examined in the context of specific applications. Yet, in order to assess the potential of these capsules whatever the type and size of storage being considered, it is necessary to understand their intrinsic behaviour. Besides, several references explicitly mention ([13], [19]) that there is a lack of component-scale heat transfer performance validation. It is in this context that this study assumes its full significance, since it has been chosen to focus on the fundamental thermal behaviour of HeatSel[®] on the basis of experiments and CFD simulations.

1.3 Objectives and outline

It is important to emphasise that this type of capsule has been designed for use in groups. Therefore, a fundamental characterisation at the component-scale must be conducted under conditions analogous to those of a stacked configuration. In this context, this paper begins with an investigation of the flow conditions experienced by the capsule when placed in a thermal storage tank sized for a residential application.

Subsequently, an experimental setup is created to observe the behaviour of the capsule when subjected to the previously determined conditions and varying temperatures. The objective is to gain insight into the differences between the solidification and melting in terms of the shape of the phase change plateau and the total duration of the process. Furthermore, a comparison of charging times as a function of capsule geometry is to be derived.

The experimental situation is then ideally replicated in CFD simulations, which involve two scenarios. The first determines the convective heat transfer coefficient around the HeatSel[®] for different types of flow. No phase change is modelled, and thus the capsule is assumed to be at a constant, uniform temperature. The second scenario simulates the complete behaviour during a solidification to understand how the phase change front evolves within the geometry and the local influence this has on the power exchanged.

Following this, a comparison of two experiments with the solidification simulation is presented, with the objective of identifying similarities and any assumptions that result in the simulations deviating from reality.

The final section of the paper presents a methodology for determining the heat transfer coefficient of an entire storage tank, irrespective of its dimensions, based on the convection coefficient of a single capsule maintained at a constant temperature.

Packed bed modelling of a thermal storage

This work will not attempt to demonstrate how a thermal storage system can be integrated into a residential hydraulic circuit, as this has already been shown to be a feasible proposition [5]. However, it is necessary to define a specific scenario that will be considered as a reference for the thermal study of the HeatSel® within the context of residential heating applications.

2.1 Reference situation

A simplified case of a boiler room is considered here, whereby latent thermal storage would be used to instantly supply the power required to heat mains water for showers. While other applications could have been identified, the objective here is simply to establish a framework in which the key variables are of a similar order of magnitude to those observed in a storage facility used for buildings. The chosen case is thus sufficiently representative for this purpose.

It is often recommended that a PCM with a phase change temperature close to the one desired for the chosen application to be selected in order to exploit latent phenomena to the greatest extent possible. In the context of a thermal storage system for domestic hot water, it is therefore advisable to select the phase change temperature within the range of 55 to 70 °C [15]. However, this assumption is based on the premise that the heat transfer fluid is the one used for consumption. In this case, it is logical to recommend such a temperature range in order to avoid all the health risks associated with legionella¹. Nevertheless, in the present case study, it is necessary to separate the storage and consumption circuits by an exchanger due to the incompatibility of the capsule shell material with the hot water sanitary regulations. It is therefore unnecessary to consider such high temperatures. A conventional shower mixer set to a comfortable temperature indicates that water of approximately 40-45 °C is sufficient². With this in mind, a paraffin melting at 43 °C, the APT43, was selected from Axiotherm®'s extensive range of HeatSel®s.

2.2 Specifics of the PCM capsule selected

The ATP43 HeatSel® is a curved slab-shaped capsule with a lateral length of 180 mm and with a 12.7 mm diameter hole in the centre. It is thick around the central hole and thin around

¹<https://guardian-group.co.uk/blog/legionella-water-temperature/>

²<https://www.delabiebenelux.com>

the periphery. Grooves are present to extend the surface area available for heat transfer relative to the volume of paraffin. A computer-aided design (CAD) of the capsule is depicted in Figure 3.

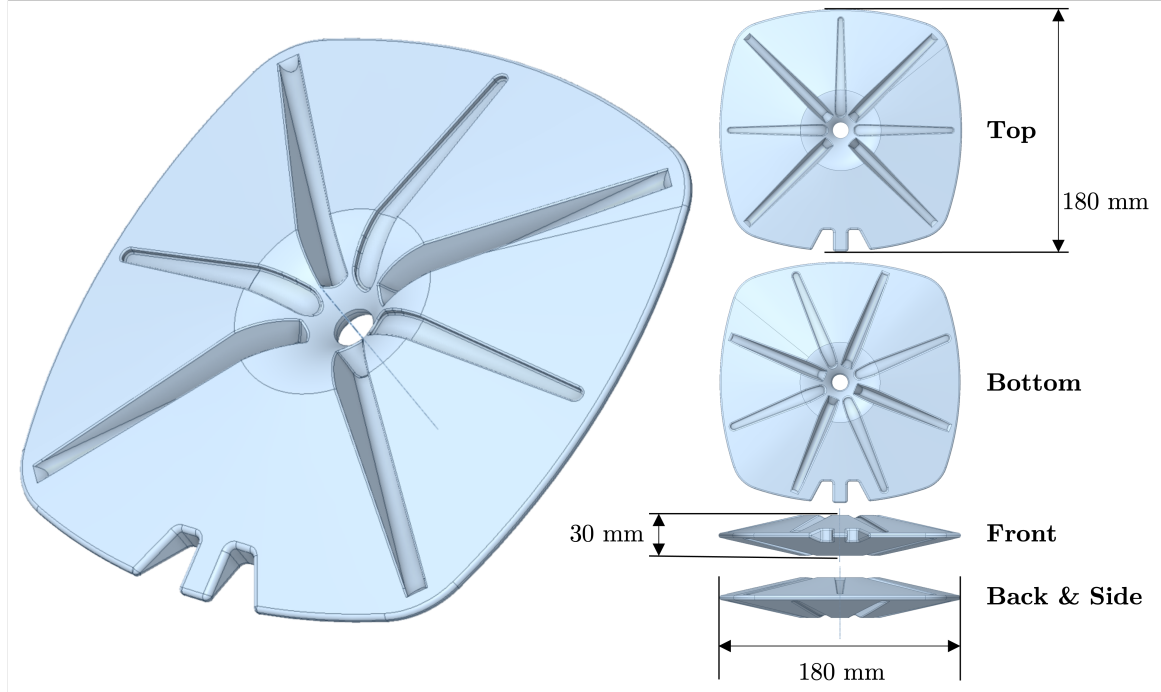


Figure 3: Several views of the 3D model of the capsule. Its dimensions are $180 \text{ mm} \times 180 \text{ mm} \times 30 \text{ mm}$. Note that it does not have any axis of symmetry. In one direction, the number of grooves on the lower face is not identical to that on the upper face. In the other direction, one groove out of two is deeper.

The properties of the paraffin contained within the capsule are enumerated in Table 1 and are directly derived from the manufacturer's datasheet. No public information is given on the type of encapsulant used, although some references mention a 1 mm-thick polyethylene shell ([11], [13], [14]). The parameters of this shell are given in Table 2.

Axiotherm® PCM ATP43 HeatSel®			
Melting temperature		41 to 43	°C
Congeeing temperature		39 to 43	°C
Heat storage capacity	Δh_{PCM}	265	kJ/kg
Specific heat capacity	c_{PCM}	2000	J/(kg K)
Density	ρ_{PCM}	760	kg/m ³
Thermal conductivity	k_{PCM}	0.2	W/(m K)
Thermal diffusivity	α_{PCM}	1.316×10^{-7}	m ² /s
Dynamic viscosity	μ_{PCM}	0.035	kg/(m s)
Volume	V_{capsule}	0.383×10^{-3}	m ³
Surface	A_{capsule}	0.698	m ²

<https://www.axiotherm.de/en/produkte/axiotherm-pcm/>

Table 1: Thermal properties and dimensions of the selected PCM capsule, designated as Axiotherm® ATP43 HeatSel®. The thermal diffusivity is derived from the others parameters using the following formula : $\alpha = \frac{k}{\rho c}$.

Capsule shell material : Polypropylene			
Specific heat capacity [13]	c_{shell}	1930	J/(kg K)
Density	ρ_{shell}	900	kg/m ³
Thermal conductivity [13]	k_{shell}	0.22	W/(m K)
Thickness	e_{shell}	1	mm

<https://plasticseurope.org/plastics-explained/a-large-family/polyolefins/>

Table 2: Thermal properties and thickness of the encapsulation.

In general, the data supplied by the manufacturer lacks the requisite accuracy and precision in terms of temperature conditions and uncertainty range. Nevertheless, it will be considered in the context of this work. To obtain more detailed information, it would have been necessary to re-measure properties using for example a Differential Scanning Calorimetry (DSC) method for specific heat capacity and latent heat of fusion, a pycnometer for liquid and solid densities and a Transient Plane Source (TPS) method for conductivities [13].

It is important to highlight that, in addition to paraffin, the capsule contains sand. The rationale behind its inclusion in the mixture remains unclear and is not explicitly elucidated in this study. In any case, the presence of sand reduces the PCM content for a given volume of capsule, thereby decreasing the energy density. In the context of this work, the capsule was not disassembled to determine the precise quantity of sand. However, it can be hypothesised that the fraction of sand included is comparable across all Axiotherm[®]'s products, and that the reference [13] determining precisely the properties of the ATP60 Heatstixx[®] can be used as a basis. In this instance, a non-negligible sand mass fraction of 36% could be considered, but it should be noted that this assumption is, a priori, unverified. A picture of a real capsule exhibiting the presence of sand is provided in Appendix A (Figure 33).

The density of paraffin and thus its volume vary depending on the state of matter. To accommodate this, the encapsulant must either be flexible with the capacity to expand its surface area, or a proportion of air must be incorporated into the capsule to offset the expansion [16], [21]. Given the rigidity of the HeatSel[®] shell, it was deemed appropriate to assume that air is also present within the capsule.

2.3 Intrinsic velocity in a packed bed

It has been shown that a thermal storage tank using macro-encapsulation technology can be numerically modelled as a packed bed heat exchanger, with PCM capsules acting as packing particles ([13], [14]). The key parameter in such a model is the porosity (ϵ), defined as the ratio of the volume of heat transfer fluid to the total volume [11]. If uniform spaces between all capsules are assumed, it is possible to determine the flow velocity experienced by a particle inside the bed as a function of the total mass flow rate.

Two studies have already conducted experimental tests with a tank filled with HeatSel[®]s. The first concluded that a porosity of 0.63 was obtained with ATP60 [19], while the other concluded a porosity of 0.57 with ATS57 [11]. The geometry of these two capsules is identical to that of the ATP43 considered here, but the size is slightly larger in their case. However, in the absence of other examples, this variation will not be considered here. Therefore, $\epsilon = 0.60$ appears to be a reasonable estimation to model a packed bed composed of ATP43 HeatSel[®]s.

The reference situation selected (Section 2.1) to illustrate the thermal behaviour of the capsule is depicted in a schematic manner in Figure 4 and encompasses the following parameters:

$$\begin{aligned} \dot{V}_{\text{shower}} &= 10 \text{ L/min} & T_{\text{mains water}}^3 &= 15^\circ\text{C} & T_{\text{shower}} &= 45^\circ\text{C} \\ & & T_{\text{cold,bed}} &= 35^\circ\text{C} & T_{\text{hot,bed}} &= 50^\circ\text{C} \end{aligned}$$

where the inlet and outlet temperatures of the packed bed have been determined in such a way that the phase change temperature is approximately in the middle of the range. This is to ensure that the phase change process occurs in both directions in a complete manner.

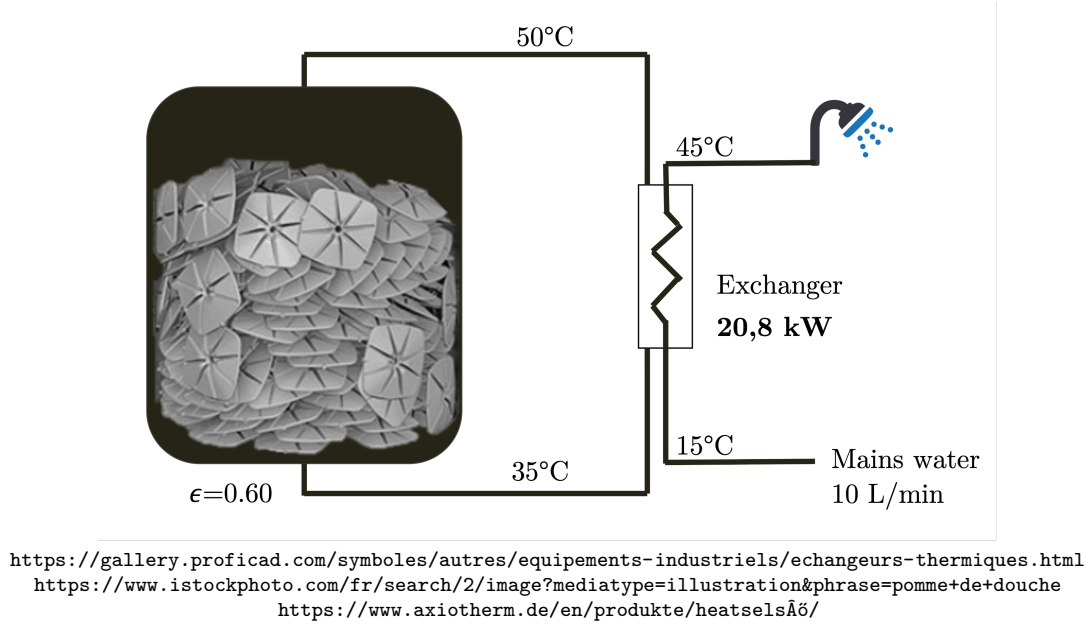


Figure 4: Illustration of a simplified scenario in which the latent heat thermal energy storage tank filled with macro-capsules must provide the instantaneous power required to take a shower.

The tank flow rate required to ensure instantaneous heating of the mains water can be calculated as follows:

$$\begin{aligned} P_{\text{shower}} &= \rho c \dot{V}_{\text{shower}} (T_{\text{shower}} - T_{\text{mains water}}) = 20.8 \text{ kW} \\ \dot{V}_{\text{bed}} &= \frac{P_{\text{shower}}}{\rho c (T_{\text{hot,bed}} - T_{\text{cold,bed}})} = 20.09 \text{ L/min} \end{aligned}$$

On this basis, it is possible to calculate the intrinsic velocity (u_f), which is defined as the flow velocity perceived by a particle within a packed bed.

$$u_f = \frac{\dot{V}_{\text{bed}}}{\epsilon A_{\text{tank}}} = 1.88 \text{ mm/s} \quad (2.1)$$

where A_{tank} is directly derived from the diameter of a typical⁴ domestic hot water tank ($D_{\text{tank}} = 600 \text{ mm}$).

⁴<https://www.nextherm.fr/produit/ballon-ecs-1701-2701-3801/>

Experimental analysis

Having established the conditions under which the capsule must be placed, it is now possible to conduct laboratory tests to understand how the phase change occurs, how long it takes to complete and the key variables that influence this process. This chapter therefore begins by presenting the experimental device that has been constructed and all the measurements that are taken on it. Next, the test procedure is highlighted. Finally, a comparison is made between the temperature profiles measured during the charging and discharging processes, as well as an analysis of the phase change dynamic.

3.1 Design overview

The duct designed to experimentally characterise the HeatSel[®] is inspired by two references where a single PCM capsule is also tested ([21], [22]) and can be described as follows: water enters through a sleeve of the size of a typical $\frac{1}{2}$ ” hydraulic connector, then the cross-section gradually widens with a conic aperture of approximately 9.5° . This cone has been positioned to ensure a smooth transition between the inlet and the diameter of the 213.1 mm cylindrical section required to surround the 180 mm-wide capsule. Behind the capsule, an other conical part ensures that the outlet emerges with a connection size equal to that of the inlet. The device can be opened in order to place the capsule inside the duct, and this opening is closed by means of 2 flanges and 8 bolts. All parts are made of stainless steel, except the flanges, which are made of aluminium. All around is placed a 13 mm-thick layer of Thermaflex[®] insulation, as also used in [19]. These components can be seen in Figure 5, with additional views suggested in Figure 34 in Appendix A.1. The capsule has been chosen to be positioned horizontally, although alternative orientations could have been considered given that capsules are normally randomly positioned in a tank. Figure 6 shows that fixing the capsule using nuts and a threaded rod allows almost a perfect positioning of the PCM in the centre of the duct. The exact dimensions of all components are given in Table 3.

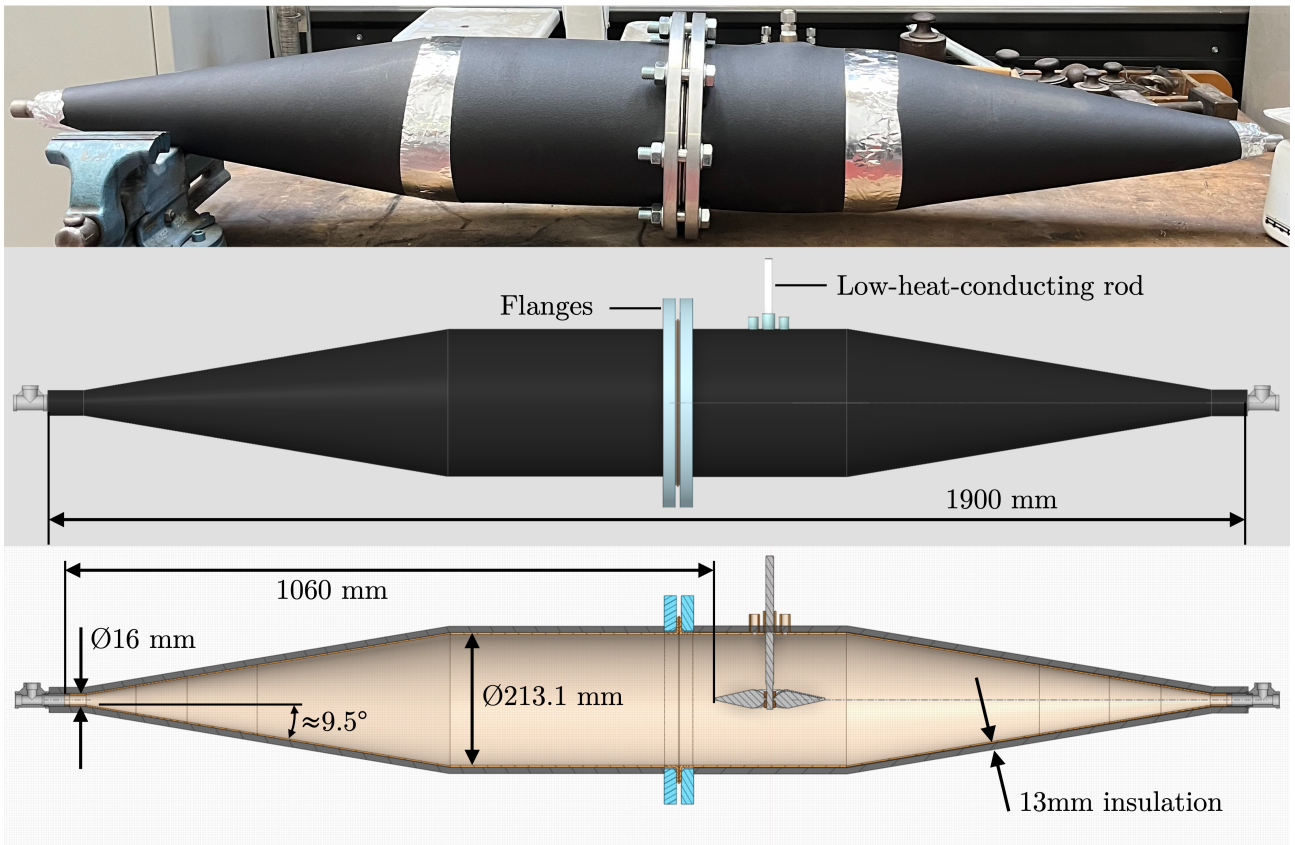


Figure 5: From top to bottom: the actual test duct, its 3D model and a cross-section showing the exact position of the capsule. The duct, measuring 1.90 m in length, begins and ends with a $\frac{1}{2}$ " sleeve. It contains two cones, positioned opposite one another, with an aperture of approximately 9.5° . These cones form a cylindrical section with a diameter of 213.1 mm, within which the capsule is placed. The rod that holds the capsule in the centre of the duct has a low thermal conductivity, and a layer of insulation has been added around the device. The two flanges facilitate access to the interior of the device, allowing the placement of temperature sensors and, eventually, the replacement of the capsule.

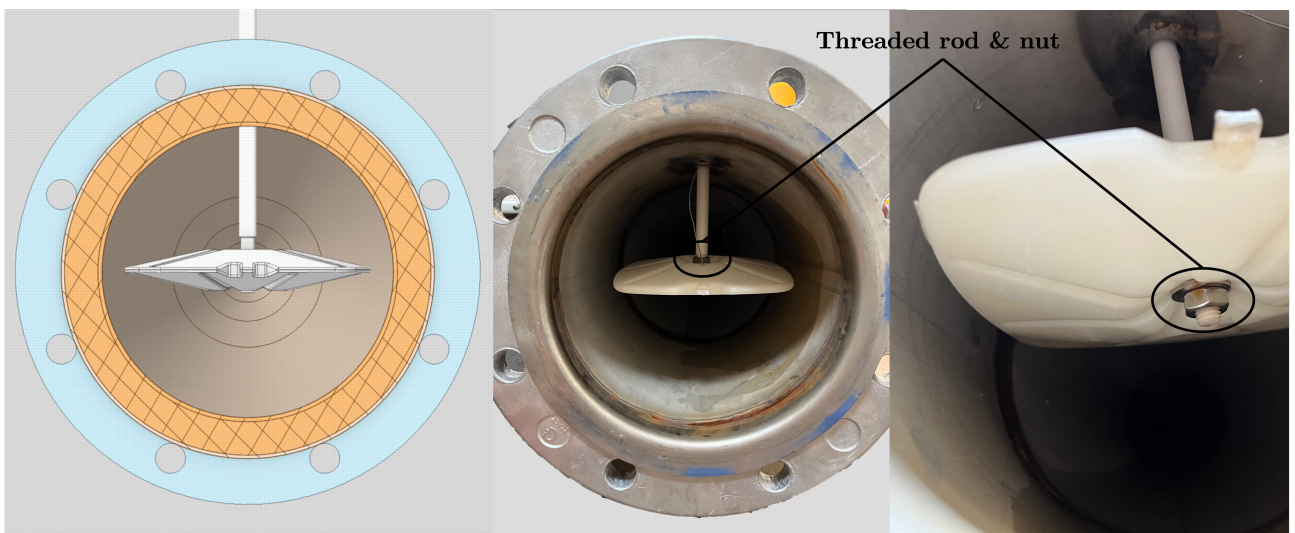


Figure 6: Cross-section view from the flange, with the model on the left and pictures of reality on the right. The capsule is ideally situated in the centre of the duct, thanks to the nuts fastened to the plastic threaded rod.

Component		[mm]
Tube	$L_{\text{inlet side}}$	350
	$L_{\text{outlet side}}$	250
	D_{inner}	213.1
Cone	L	593.4
	$D_{\text{inner, small}}$	17.24
	$D_{\text{inner, big}}$	215.04
Sleeve	L	35
	D_{inner}	16
Flanges	L_{tot}	48
	D_{inner}	220
	D_{outer}	270

Table 3: Dimensions of the 2 sleeves, 2 cones, 2 cylindrical parts and 2 flanges used to mount the experimental device. Note that the length L_{tot} refers to the total length of the junction, taking thus into account the joint, the 2 welded collars and the 2 flanges.

The data presented in Table 3 allows for the straightforward calculation of the total volume of water that will be contained within the device.

$$V_{\text{inner, device}} = 38.3 \text{ L} \quad (3.1)$$

Furthermore, by considering the mass of each component and its material, it is possible to determine the total specific heat capacity of the device. The exact calculation can be found in Appendix A.2 and the result is:

$$c_{\text{device}} = 15.48 \text{ kJ/K} \quad (3.2)$$

Finally, a model for characterising the total thermal loss of the device is proposed in Appendix A.3. The calculations demonstrate that the thermal resistance of all components can be approximated as:

$$R_{\text{th}} = 0.283 \text{ K/W} \quad (3.3)$$

Flow rate to be imposed in the device

Having established the dimensions of the experimental test device, it is now possible to determine the flow rate that must pass through it in order to subject the capsule to flow conditions analogous to those of a packed bed. By imposing the intrinsic velocity (u_f) on the cylindrical section in which the capsule is located, the flow rate can be deduced.

$$\dot{V} = u_f \times \pi \left(\frac{D_{\text{inner, tube}}}{2} \right)^2 = \boxed{4.02 \text{ L/min}} \quad (3.4)$$

It is of interest to note that if a plug flow is considered, combining $\dot{V}$ with $V_{\text{inner, device}}$ indicates that the water takes 9.5 min to pass through the device.

The Reynolds number of the flow can also be found by defining a characteristic length and specifying the properties of the water. The characteristic length has been defined as recommended ([13], [23]) for modelling flows around objects with complex geometries:

$$L_c = \frac{6V_{\text{capsule}}}{A_{\text{capsule}}} = 33 \text{ mm} \quad (3.5)$$

It is worth noting that the value for the characteristic length is, in fact, relatively close to that of the thickness of the capsule (30 mm).

The properties of the water are considered to be those at the phase change temperature of the ATP43 paraffin and are held constant for the rest of this work. Details are given in Table 4.

Specific heat capacity	c_{water}	4179.49	J/(kg K)
Density	ρ_{water}	991.93	kg/m ³
Thermal conductivity	k_{water}	0.629	W/(m K)
Dynamic viscosity	μ_{water}	644×10^{-6}	kg/(m s)

Table 4: Properties of water at 43 °C. These values are considered throughout this work and remain constant in the calculations.

This gives :

$$Re = \frac{\rho_{\text{water}} u_f L_c}{\mu_{\text{water}}} = 99.3 \quad (3.6)$$

indicating a laminar flow, as the limit is set at 2300 for the onset of turbulence in a cylinder¹.

It is also of interest to determine the relationship between the volumetric flow rate and the mass flow rate.

$$\dot{m} = \frac{\rho_{\text{water}} \dot{V}}{1000 \times 60} = 0.07 \text{ kg/s} \quad (3.7)$$

where the factors 60 and 1000 have been included in the equation because the volumetric flow rate is expressed in L/min.

Cone aperture analysis

A representative simulation of the device inlet (initial diameter of 16 mm and conical aperture of 9.5°) with 4.0 L/min as flow rate was carried out to analyse detachment. The result can be observed in Figure 7:

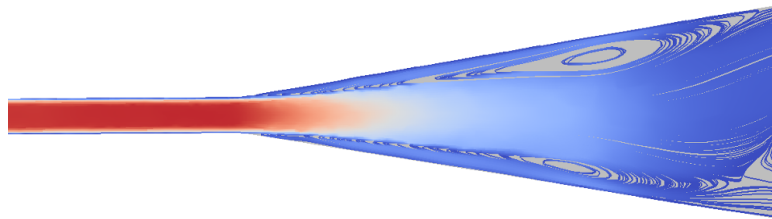


Figure 7: Simulation result of the inlet geometry with a water flow rate of 4.0 L/min. The streamlines are shown. The detachment is clearly observed, as recirculation zones are created, giving a local radial velocity to the flow, which was originally only axial.

¹<http://labman.phys.utk.edu/phys221core/modules/m8/turbulence.html>

In order to obtain a straight flow, a smaller conical aperture should have been considered, since detachment is clearly observed. However, since the total length of the device was already close to 2 m, it was decided not to extend it the context of this work. Still, it should be noted that the results depicted in Figure 7 are based the perfectly smooth walls assumption. Consequently, it can only provide information on the presence or absence of detachment, but cannot be used to quantify its significance, as in reality it is local pressure drops created by imperfections such as welds that determine the extent of detachment.

3.2 Measurement

In order to provide an accurate representation of the thermal behaviour of the capsule during the tests, it was deemed appropriate to measure the exact flow rate through the device and the temperatures at several key points.

3.2.1 Temperature

It was determined that measuring the temperature at the inlet and outlet of the device would be a logical approach. To this end, a T-shaped adapter was affixed to the sleeve on both sides, allowing water to flow simultaneously as a 1 mm-thick K-type thermocouple² was inserted. This ensures a representative inlet and outlet temperatures recording. Photographs illustrating the configuration are presented in Figure 8.

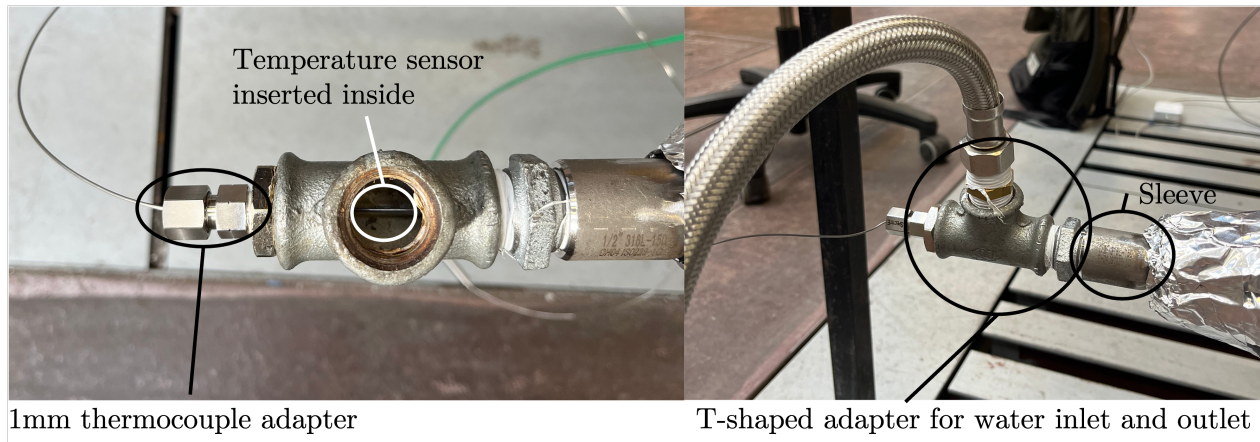
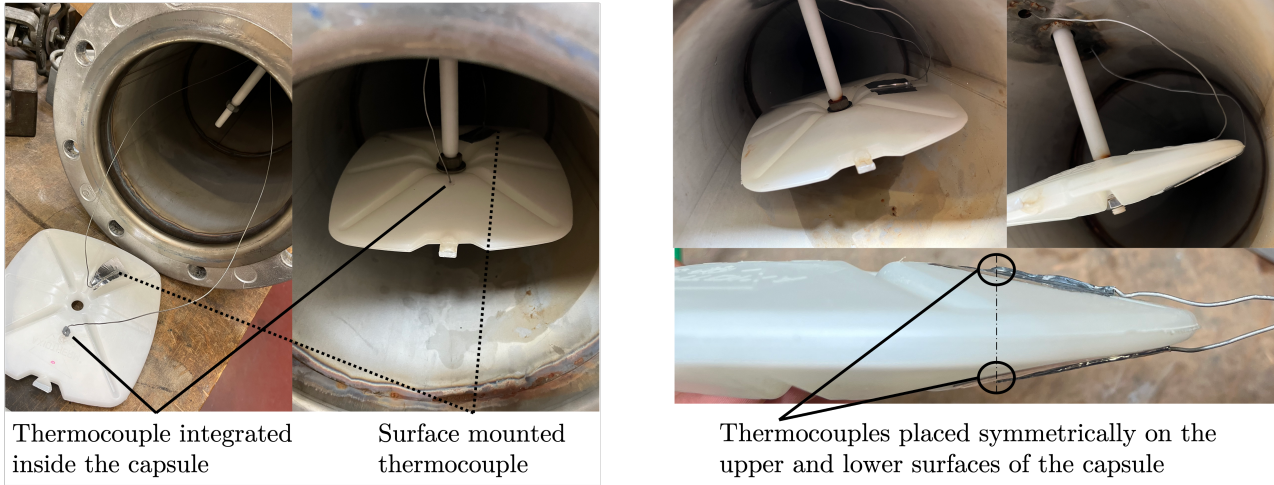


Figure 8: Pictures of the experimental device inlet. The T-shaped adapter mounted on the sleeve allows the thermocouple to be inserted inside while leaving a passage for the water. The K-type thermocouple is 1 mm wide, so an adapter is required to fix them.

Two configurations were considered for measuring the temperature of the capsule, as illustrated in Figure 9. The first configuration involves a thermocouple inserted inside the capsule, with a depth equal to half the thickness at the point under consideration (24 mm). The aim is to measure the temperature of the paraffin directly, thereby observing the phase change. A second sensor is also taped to the upper surface. The second configuration involves the placement of two thermocouples on the surface of the capsule, with one positioned on the upper and the

²<https://www.tcdirect.fr/product-2-190-1/Thermocouple-chemis%C3%A9-diam%C3%A9tre-10mm-15mm-ou-30mm-sortie-sur-c%C3%A2ble>

other on the lower part. The thermocouples are aligned on the same vertical axis in order to facilitate the observation of symmetry.



(a) Configuration 1: a K-type thermocouple is inserted 12 mm deep into the capsule (fixed with glue) and the other thermocouple is surface bonded at the back (good heat conducting adhesive).

(b) Configuration 2: the two K-type thermocouples are symmetrically positioned on the right side of the capsule. The vertical alignment between the one at the top and the one at the bottom is clearly visible.

Figure 9: Presentation of the 2 temperature measurement configurations in and around the capsule for the experiments.

All temperature sensors employed were identical and exhibited an uncertainty in measurement that could be quantified by the following relationship:

$$\delta_T = 0.001 \times T + 1.5 + 0.7$$

The quantity T represents the measurement displayed by the Thermosense® HH-521BT³ data logger. The term 1.5 is derived from the intrinsic uncertainty of the thermocouple, while the 0.7 is attributed to the acquisition error of the signal sent by the sensor to the data logger. For a temperature measurement of 20°C, $\alpha_T = 2.22^\circ\text{C}$. It therefore represents a high degree of uncertainty, which should be taken into consideration. The use of alternative temperature sensors, such as PT100⁴, is not a viable option for the measurement of surface and internal capsule temperatures. Nevertheless, they could have been considered as a potential enhancement for inlet and outlet measurements. A further method for reducing this uncertainty would have been to calibrate the measurement chain by comparing the data logger value with precisely known temperatures.

The data logger was connected to a computer, allowing for the real-time monitoring of fluctuations and the adjustment of the recording frequency. In this case, the interval between data samples was set to 3 s. A screenshot is provided in Figure 35 in Appendix A.1.

³https://www.thermosense.co.uk/public/pdfs/usermanuals/hh-521bt_instruction_manual.pdf

⁴<https://atp-instrumentation.co.uk/blog/pt100-accuracy-and-probe-classification/>

3.2.2 Flow rate

The Picomag[®] electromagnetic flowmeter⁵ was employed to quantify the volumetric flow rate through the device. Its uncertainty takes the following form:

$$\delta_{\dot{m}} = 0.008 \times \dot{m} + 0.001 \times 20$$

where the term 0.001×20 represents the uncertainty of 0.1% over the entire measurement interval of the instrument, which is 20 L/min. The term $0.008 \times \dot{m}$ represents the uncertainty on the measure itself.

3.3 Experimental test rig & procedure

Prior to the detailed description of the experimental procedure, it is essential to define the concepts of charge and discharge.

- Charge "C" : The capsule, initially at a **lower** temperature than the phase change temperature, is subjected to a water flow at a temperature **above** the phase change temperature, causing the paraffin to melt progressively.
- Discharge "D" : The capsule, initially at a **higher** temperature than the phase change temperature, is subjected to a water flow at a temperature **below** the phase change temperature, causing the paraffin to solidify progressively.

The test bench to which the experimental device is connected is shown in photographic form in Figure 36 (Appendix A.1) and in schematic hydraulic circuit form for the specific case of a charge in Figure 10. The case of a discharge can be observed in Figure 37 in Appendix A.1.

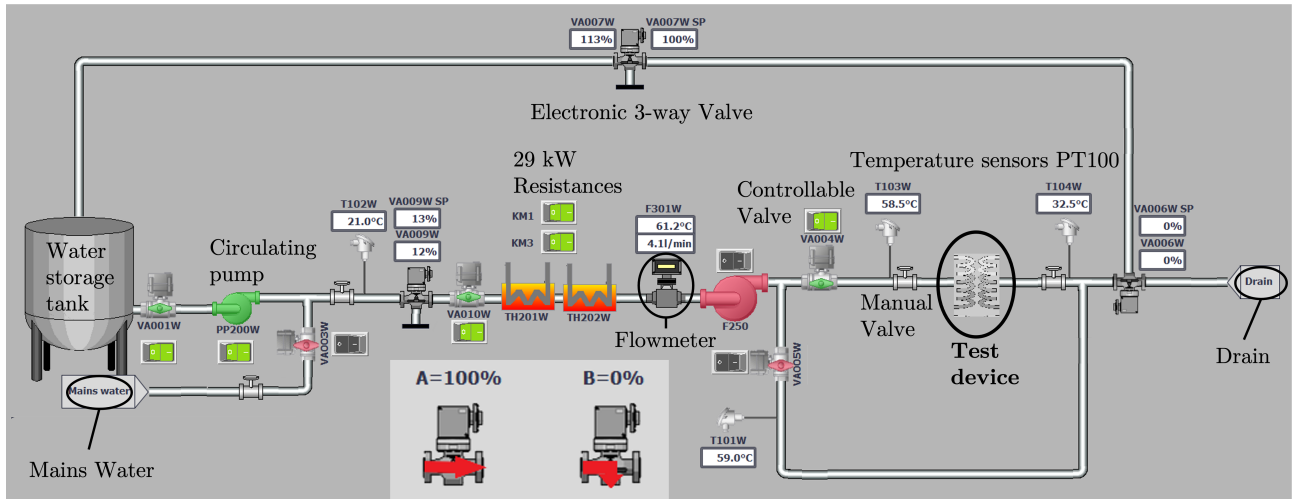


Figure 10: Hydraulic circuit scheme of the test bench used to carry out the experiments. The configuration shown is that of a charge: the bench is used in a closed circuit with a pump to compensate for pressure drops. In this case, the capsule, initially at ambient conditions, is subjected to a constant temperature of approximately 60 °C with a flow rate of 4.1 L/min.

⁵<https://www.be.endress.com/fr/instrumentation-terrain-sur-mesure/debit-mesure-produits/picomag-debitmetre-electromagnetique>

The test bench is presented as follows: the water can be brought into the device by either using mains water at ambient temperature or water at a higher temperature from a tank. Should the decision be made to use water from the tank, it is necessary to activate one of the pumps. With regard to the outlet, it is possible to direct the water either to the drain or to return to the tank. A hydraulic circuit is provided for bypassing the test device. Two resistances, one with a power rating of 20 kW and the other of 9 kW, can be activated separately or together to heat the water before it reaches the test device.

To control the flow rate, the 3-way valve located before the resistances was manually adjusted by changing the setpoint ("VA009W SP"), ensuring that the displayed value on the flowmeter corresponded to the desired value. In order to regulate the temperature of the water entering the test device and, consequently, the heating power of the resistances, an electronic control system was implemented. The system employed a proportional-integral-derivative (PID) controller, which used the temperature data from the flowmeter to adjust the resistance power.

For a charge test, the bench is set in closed circuit, with the inlet originating from the tank and the outlet returning to it. The water within the device and the capsule are initially at ambient temperature and isolated from the circuit. The flow is thus bypassing the device until the temperature setpoint is reached. At this juncture, the valves that isolate the cell are abruptly opened, allowing the introduction of hot water into the test device.

In the event of a discharge, the bench is employed in an open circuit configuration, with the inlet to the mains water and the outlet to the drain. The water and the capsule are initially at a high temperature and isolated from the circuit until the mains water has cooled the circuit and the resistances. At this point, the valves are switched, allowing the cold water to enter the system.

3.4 Results and discussion

This section presents the results and conclusions that can be drawn from all the tests carried out in the laboratory. Firstly, an analysis of the capsule's response to a charge and a discharge is presented. Secondly, a dimensionless relationship for phase change duration as a function of external flow temperature is derived from the measurements and compared with those for conventional capsule geometries. Finally, an analysis of the symmetry of the flow in the experimental device is presented, together with an attempt to characterise the thermal performance.

3.4.1 Charging and discharging behaviour

In order to examine the behaviour of the capsule following a charge or discharge test, measurement configuration 1, which involves the insertion of one of the sensors into the paraffin, is employed (Figure 9a). Post-processing of the temperature data was performed using the following parameters:

- Interpolation with B-Splines is applied to estimate the temperature every second, as the recorded values were only available at 3-second intervals.

- The start of the test is defined as the moment at which the difference between two successive temperature values at the inlet exceeds 2°C .
- Once the temperature difference between the interior of the capsule and the surface has fallen below 0.5°C , the test is deemed to have ended.
- It is well established that phase change occurs when there is a sudden change in the temperature gradient [8], so the following process is applied to the temperature measurements inside the capsule. First, a frequency analysis is conducted on the data to remove any noise, thus allowing only the underlying general trend to be discerned. Then, the first derivative is calculated on the basis of 6 successive data points, and the same process is again applied to obtain the second derivative. Finally, the maximum value of the second derivative is taken in the intervals $37 - 41^{\circ}\text{C}$ and $42 - 50^{\circ}\text{C}$. This approach allows the determination of the two inflection points, which represent the start and end points of the phase change.

Charge

A total of 11 charge tests were conducted as part of this study. For each test, the flow rate was set to be as close as possible to the value of 4 L/min (Equation 3.4). However, the hysteresis associated with the valve position did not permit the exact flow rate to be consistently reached. Consequently, values between 3.8 and 4.2 L/min are to be considered. Also, the start temperature was dependent on the temperature of the mains water, ranging from 21 to 25°C . The completion times and characteristics of the phase change plateau are presented in Table 5.

#	$\bar{T}_{\text{inlet}} [^{\circ}\text{C}]$	$t_{\text{charge}} [\text{min}]$	$\Delta T_{\text{PC}} [^{\circ}\text{C}]$	$t_{\text{PC}} [\text{min}]$	$T_{\text{PC, start}} [^{\circ}\text{C}]$	$T_{\text{PC, stop}} [^{\circ}\text{C}]$
C1	45.2	> 200	x	x	39.4	x
C2	50.3	91.9	5.5	67.5	38.6	44.1
C3	50.5	88.1	7.0	59.5	38.6	45.6
C4	50.7	82.3	6.3	46.0	39.0	45.3
C5	54.6	61.9	7.9	30.1	38.8	46.7
C6	55.0	71.6	5.8	42.4	39.3	45.1
C7	59.6	49.6	x	x	x	x
C8	60.2	47.9	x	x	39.9	x
C9	64.5	43.0	10.9	15.7	38.6	49.5
C10	64.8	45.5	9.3	15.2	39.7	49.0
C11	75.6	27.4	12.8	7.9	37.8	50.6
Mean					39.0	47.0

Table 5: Time (t_{charge}) required for the complete melting of the PCM within the capsule as a function of the inlet temperature ($\bar{T}_{\text{inlet}}$). The characteristics of the phase change plateau are also presented, with the exception of C7 and C8, where it was not explicitly observable, and for C1, where the test duration was limited to 200.0 min. The quantity ΔT_{PC} represents the height of the plateau corresponding to the difference between $T_{\text{PC, stop}}$ and $T_{\text{PC, start}}$. t_{PC} is the length of the plateau, and therefore the duration of the phase change.

Table 5 illustrates that, on average, the phase change occurs at 39°C during a charge and ends at 47°C . For values of $\bar{T}_{\text{inlet}}$ that are slightly above the phase change temperature, the height of the plateau is approximately 5°C . However, it can be observed that this increases

as $\bar{T}_{\text{inlet}}$ increases. The temperature range in which melting occurs is therefore considerably broader than the one indicated by the manufacturer (Table 1). Moreover, although there may be discrepancies of up to 10 min for the same value of $\bar{T}_{\text{inlet}}$, it can be globally seen that the test completion time and, consequently, the phase change time decrease as $\bar{T}_{\text{inlet}}$ increases.

The temperature curves and the exact flow rate of each test are presented in Appendix A.4.1. However, for the sake of clarity, C10 is shown in greater detail here in Figure 11.

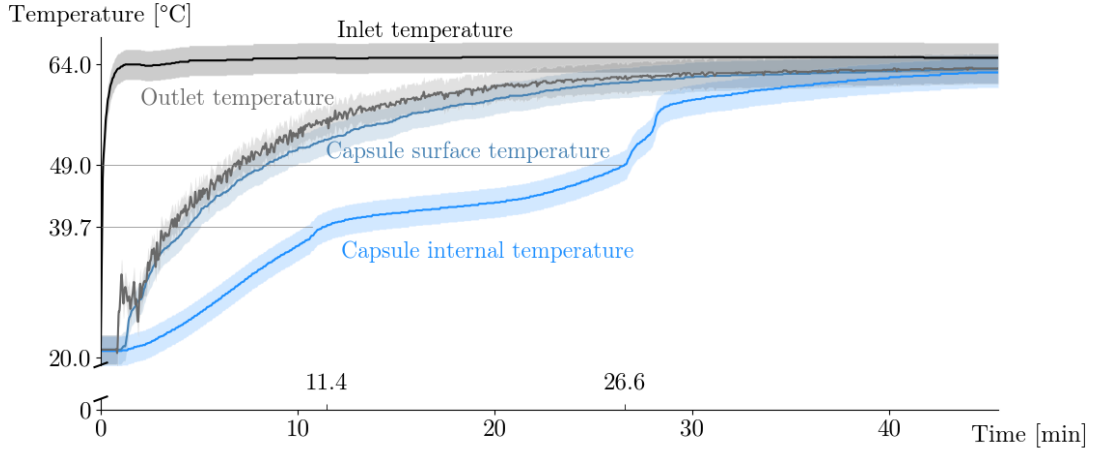


Figure 11: Example of the evolution of the 4 temperatures measured during a charge test (C10) with a constant flow rate of 3.9 L/min and an initial temperature of 21.1 °C. The temperature inside the capsule is measured with thermocouple inserted in the middle of the paraffin ; so where logically the phase change should take place in last. The 3 stages of phase change can be clearly seen: sensible heating and plateau followed again by sensible heating. Note that the highlighted area on either side of each curve represents the uncertainty of the measurement.

Figure 11 allows several conclusions to be drawn. Firstly, it is evident that the output temperature measurement exhibits a high degree of noise in comparison to the other measurements. The hypothesis that the thermocouple was faulty was disproven by replacing it. This phenomenon is more likely to be the result of unsteadiness in the flow, possibly caused by the detachment at the inlet of the device (Section 3.1). Secondly, the outlet temperature is observed to change approximately 1 min after the perturbation at the inlet and surface temperature. These observations indicate that the plug flow hypothesis is unlikely to be applicable to the experimental setup. If it were, a reaction would have been observed at the outlet after approximately 9.5 min. One potential explanation for this phenomenon is the significant impact of natural convection. It could be imagined that the hot water entering the device remains only in the upper part of the duct and remains separate from the cold water initially present. Consequently, the same flow rate applies to a smaller section, resulting in the hot water moving at a high velocity towards the outlet.

Discharge

The temperature of the mains water applied to the inlet throughout the 8 discharge tests varied between 20.5 and 22.5 °C. However, within the same test, the temperature remained constant. The flow rates varied from 4.0 to 4.4 L/min. The completion times and characteristics of the phase change plateau are presented in Table 6.

#	T_{start} [°C]	$t_{\text{discharge}}$ [min]	ΔT_{PC} [°C]	t_{PC} [min]	$T_{\text{PC, start}}$ [°C]	$T_{\text{PC, stop}}$ [°C]
D1	43.9	x	x	x	x	36.1
D2	48.8	65.0	6.7	47.6	44.4	37.7
D3	49.0	62.8	6.1	45.6	44.5	38.4
D4	54.2	64.0	6.3	38.1	42.6	36.3
D5	58.5	67.2	6.1	39.5	42.5	36.4
D6	61.6	68.2	7.3	46.9	44.4	37.1
D7	61.7	66.4	7.9	43.4	44.4	36.5
D8	73.3	65.7	8.0	41.2	44.1	36.1
Mean					43.8	36.8

Table 6: Time ($t_{\text{discharge}}$) required for the complete solidification within the capsule is as a function of the starting temperature (T_{start}). The characteristics of the phase change plateau are also presented, with the exception of D1, which followed C1 and thus the PCM was not completely liquid when the test started.

Table 6 indicates that the phase change begins at an average temperature of 43.8 °C and ends at 36.8 °C for a discharge. These values are therefore broader than those presented in the data sheet, but also distinct from those observed in the case of a charge. This phenomenon is known and has already been observed in the literature [13]. It can be explained by the fact that in both cases, the phase change front starts from the periphery and progresses towards the centre of the capsule. However, in the case of discharge, it means that the liquid is trapped inside the solid, whereas during melting, the difference in density causes the solid part to fall to the bottom of the capsule, allowing natural convection to have much more impact ([21], [24]–[28]). Therefore, charging and discharging cannot be considered as the same phenomenon.

Appendix A.4.2 shows the temperature curves and the exact flow rate value for each discharge test. Still, D6 is shown here (Figure 12).

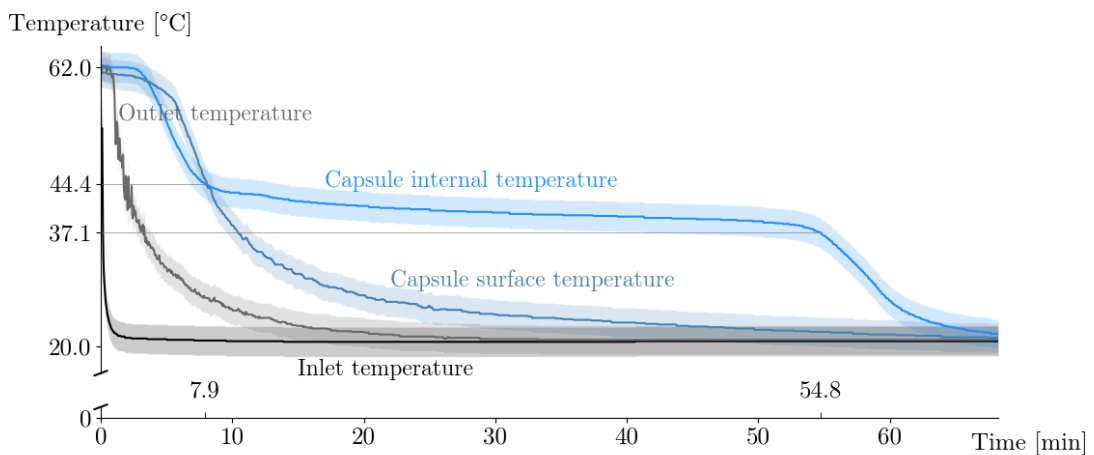


Figure 12: Example of the evolution of the 4 temperatures measured during a discharge test (D6) with a flow rate of 4.0 L/min. Note that the time scale presented here differs from the one of the aforementioned charge example.

As previously observed in the case of the charge, Figure 12 illustrates that the temperature measurement at the outlet is noisy and that a reaction at the outlet is observed almost

instantaneously. However, what is worthy of particular attention here is that for approximately 4 min during the sensible descent, the temperature inside the capsule is indicated as lower than the temperature at the surface. Two explanations can be identified for this phenomenon. Firstly, it is possible to consider as before that natural convection channels the cold water in the lower part. In this scenario, the lower surface would be considerably cooler than the upper one, thereby cooling the PCM while the upper surface sensor is measuring the temperature of hot water that remains in the upper region. Another potential justification could come from the fact that the stem of the thermocouple inserted into the capsule is not insulated. It is certain that the thermocouple only measures the temperature at its tip but its stem is made of refractory steel, which conducts heat very well ($k = 14.2 \text{ W/(m K)}$)⁶. This could result in the formation of a thermal bridge at the sensor location, thereby artificially accelerating solidification. Considering this, it would appear that the internal temperature measurement cannot be considered a reliable reference for determining the overall phase change behaviour of the capsule.

Influence of inlet temperature during charging

Figure 13 illustrates that for a given starting point, the higher the temperature set at the inlet (and therefore imposed around the capsule), the shorter the charging time and the less the shape of the curve is close to an ideal phase change plateau. This can be substantiated by references where comparable experiments have been conducted ([13], [29]).

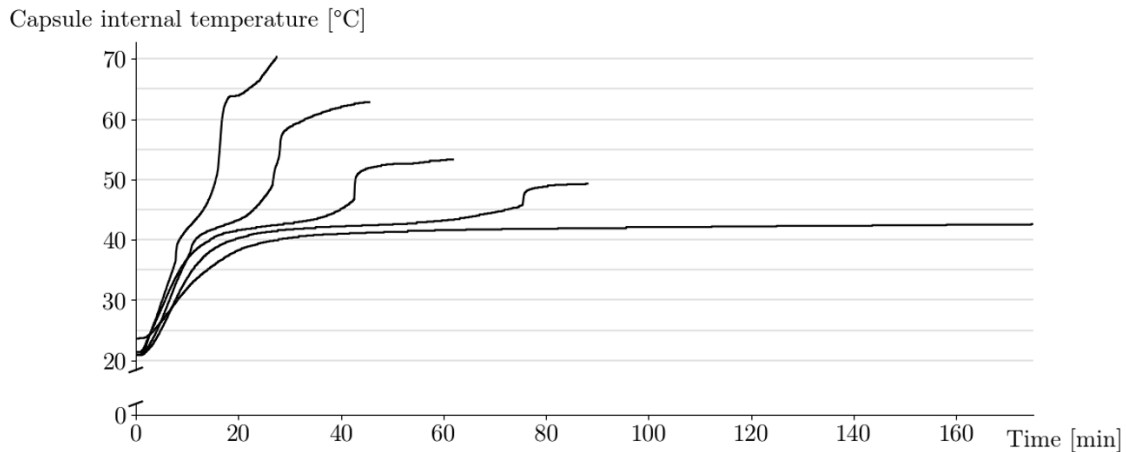


Figure 13: Evolution of the temperature inside the capsule for different inlet temperatures during charge tests. The C1, C3, C5, C10 and C11 are presented.

Influence of start temperature during discharging

Figure 14 illustrates that in the case of a discharge with a fixed inlet temperature, the initial temperature of the capsule has no impact on the solidification time. Indeed, all the plateaus exhibit a similar duration and shape.

⁶<https://fr.china-stainless-steels.com/stainless-steel-plate/310-1-4845-stainless-steel.html>

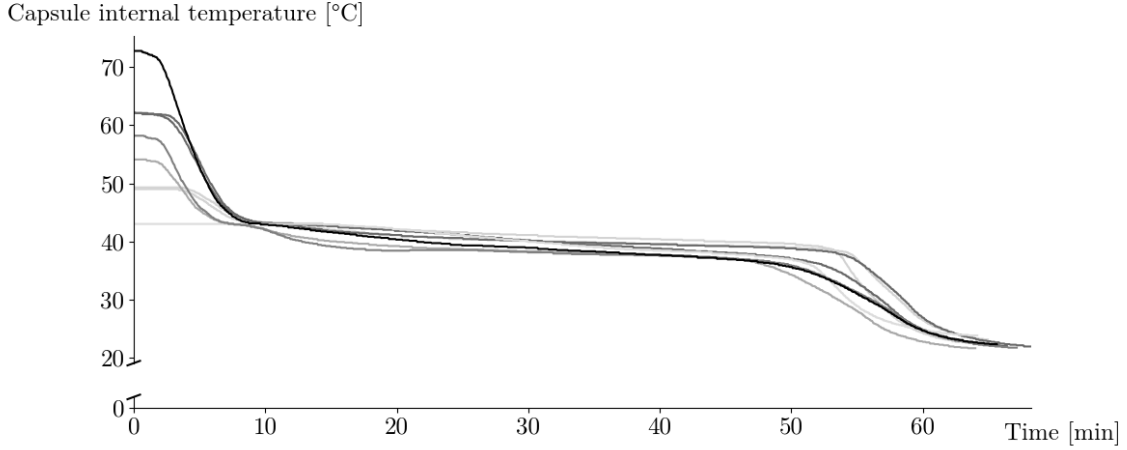


Figure 14: Evolution of the temperature inside the capsule for different starting temperatures during discharge tests. All D1-D8 discharges are included in the analysis.

The observations made on one side for charging and on the other for discharging can in fact both be applied to their counterparts [13]. Therefore, it can be established that the only variable influencing the phase change process is the temperature of the surrounding environment, while the starting temperature of the capsule has no effect.

3.4.2 Charge dynamics

In light of the preceding argument, there appears to be a possibility of establishing a correlation between the temperature applied around the capsule and the phase change duration. The thesis of B. Peremans [30] demonstrates how this can be achieved in the case of a charge.

Firstly, the paper indicates that the charging time can be dependent on all variables associated with a phase change problem :

$$t_{ch,PCM} = F(L_x, L_y, L_z, T_\infty - T_m, T_m - T_0, \rho_{PCM}, k_{PCM}, c_{PCM}, \Delta h_{PCM})$$

where the L_x, L_y, L_z represent the characteristic lengths in the 3 dimensions, T_∞ is the temperature imposed to the PCM, T_m is the phase change temperature, T_0 is the temperature at the start of the experiment, and the rest represent the intrinsic properties of the PCM, which are considered to be constant.

The thesis demonstrates, through dimensional analysis and the Buckingham-Pi theorem, that regardless of the shape of the PCM capsule, the previous equation can be simplified and expressed in the following manner:

$$\bar{t}_{ch,PCM} = f(Ste)$$

where $\bar{t}_{ch,PCM}$ is the dimensionless charging time that can be expressed using the following formula ([23], [31]):

$$\bar{t}_{ch} = \frac{\alpha_{PCM} t_{ch}}{L^2}$$

and where Ste is the dimensionless Stefan number characterising the heat transfer during the solid-liquid phase change and is defined as follows:

$$Ste = \frac{c_{PCM}(T_\infty - T_m)}{\Delta h_{PCM}}$$

As can be seen, this number basically expresses the ratio between the sensible heat and the latent heat of the PCM.

Given that the entire problem is now dimensionless, the temperature variable is written :

$$\bar{T} = \frac{T_m - T_0}{T_\infty - T_m}$$

Finally, the reference shows by numerical calculation that when $0 < \bar{T} < 5$, whatever the geometry of the capsule, its dimensionless charging time is proportional to the inverse of the Stefan number:

$$\bar{t}_{ch,PCM} = \frac{C}{Ste}$$

In the present particular case, defining $T_m = 43.0^\circ\text{C}$, $T_\infty = \bar{T}_{inlet}$ and $T_0 = T_{start}$ allows to derive the following interval for the dimensionless temperature of the 11 charge tests:

$$0.86 < \bar{T} < 3.64$$

An inversely proportional relationship between the dimensionless charge time and the Stefan number should therefore be observed, given that the dimensionless temperature value of all charge tests falls within the range of validity of the theory. Assuming $L = L_c = \frac{6V_{capsule}}{A_{capsule}}$ (from Equation 3.5), the Stefan numbers of the 11 charges can be calculated and the result is presented in Figure 15.

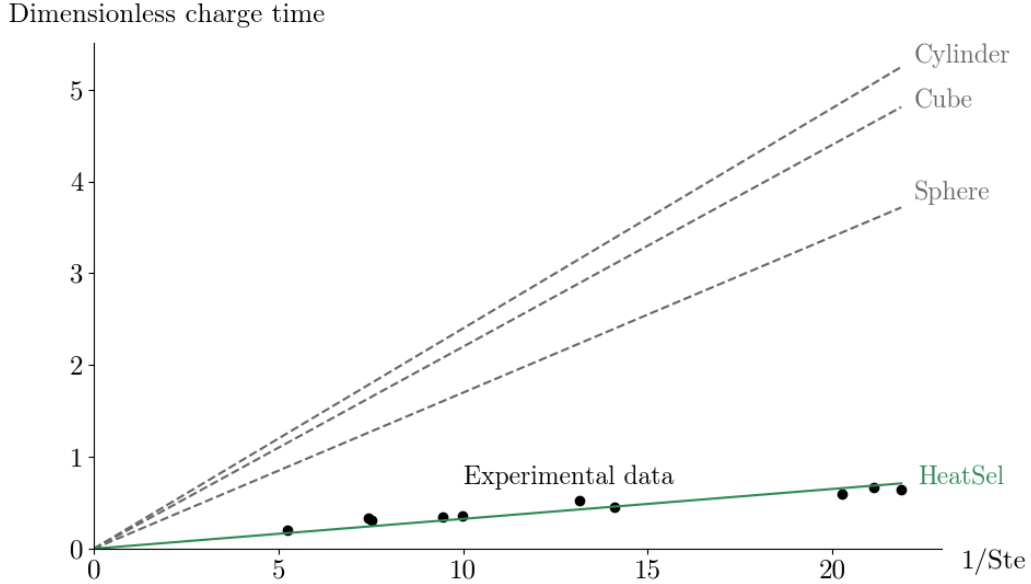


Figure 15: Experimental results provide evidence that the theory, which states that the dimensionless melting time of the PCM is inversely proportional to the Stefan number of the problem, regardless of the capsule geometry, is valid. The optimal surface-to-volume ratio of the Heatsel[®] is proving to be much more efficient than conventional geometries.

Figure 15 demonstrates that the experimental outcomes align with the theoretical framework presented in the thesis, which posits that regardless of the capsule geometry, the dimensionless charging time is inversely proportional to the Stefan number. A slope of 0.033 is obtained for

the Heatsel[®] while a slope of 0.17 was mentioned in the reference for the sphere. This indicates that the charge time of the HeatSel[®] is approximately $5\times$ shorter than that of an equivalent volume conventional geometry. The manufacturer of the HeatSel[®] has certainly designed the geometry to achieve the best surface-to-volume ratio, so it seems quite logical to observe better dynamics. Furthermore, studies on similar shapes (much longer than ticker and a hollowed-out centre) have also concluded that charging time is shorter for this type of geometry ([22], [32]).

It should be noted, however, that the hypothesis that the temperature sensor causes a thermal bridge and therefore an artificially faster phase change has not been ruled out. Consequently, the slope of the straight line may be underestimated, resulting in an overestimation of the performance.

3.4.3 Flow asymmetry analysis

In order to ascertain the relevance of the hypothesis that postulates that the temperature is not uniform around the capsule and that the upper side is always warmer than the lower side, the measurement configuration 2 presented in Figure 9b is employed. Results for a charge are indicated in Figure 16.

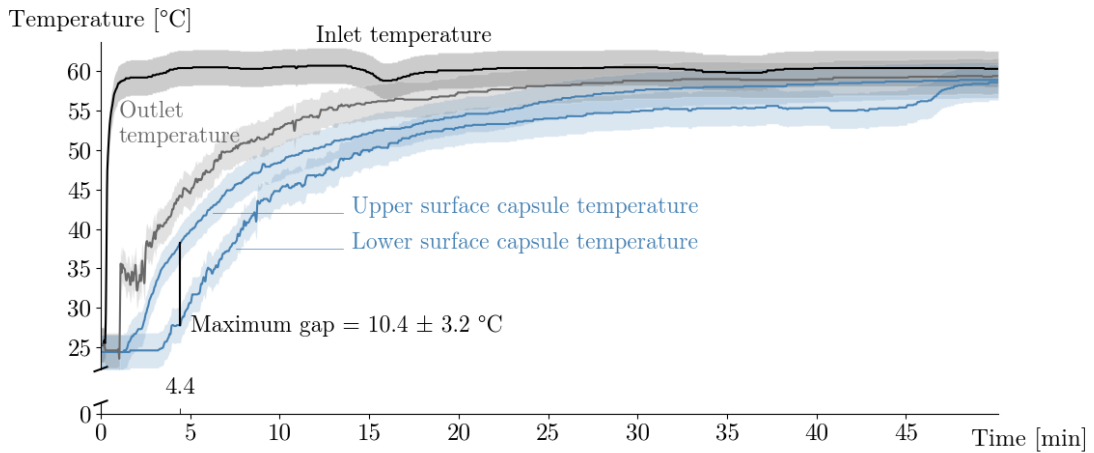


Figure 16: Comparison between the temperature measurements taken on the upper and lower surfaces of the capsule. The difference between the two readings remains significant for a period exceeding 45 min, which corresponds to approximately 2/3 of the total experiment duration. The maximum gap is highlighted.

A comparable phenomenon is observed in a discharge case, which can be seen in Figure 49 in Appendix A.4.3. Consequently, it can be concluded that the temperature measured on the upper surface during the experiments cannot be taken as a reference for the temperature around the entire capsule.

This observation can be attributed to a number of different causes. The first potential explanation is the presence of air on the upper surface of the capsule (as presented in Section 2.2), which would result in a change in conductivity of the upper portion only. The second potential cause is that the phase change process is not homogeneous or isotropic. Thirdly, the phenomenon may be attributed to natural convection causing stratification in the duct.

In order to exclude the possibility that it is mainly due to the phase change process, a charge test was conducted between 20 and 35 °C. In this instance, only sensible heating is considered. The results are presented in Figure 50 in Appendix A.4.3 and demonstrate that the gap between the two measurements is still present. It can therefore be concluded that the phase change phenomenon is not the dominant influence on the asymmetry.

To attempt to rule out the hypothesis that natural convection is the principal cause, a grid was placed in the test device. This can be observed in Figure 38 in Appendix A.1. The grid is a logical addition, as it diffuses radially the momentum through the abrupt opening associated with the exit of each mesh, therefore forcing flow to mix. Figure 17 indicates that a slight improvement is observed with this new component. In fact, the time required for the gap to close is reduced. But the maximum gap of 10.4 °C observed in Figure 16 persists. This indicates that the asymmetry is likely a consequence of natural convection within the device, although the grid may not be sufficiently effective in mitigating the phenomenon.

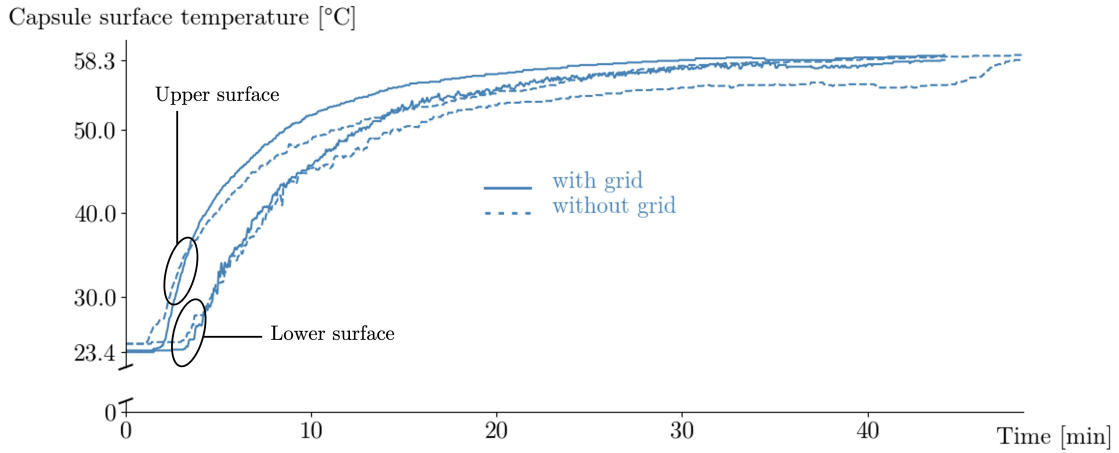


Figure 17: Comparison of the temperature on the lower and upper surface of the capsule when the grid is in place or not. The maximum difference is observed in both cases, but decreases more rapidly when the grid is in place.

To go further, one potential solution would be to orient the device vertically, with the hot water always injected from the top and the cold water from the bottom. This would result in a plug flow, as natural convection would in either case act in the desired direction of flow. This solution has already been considered in the literature [13]. An other approach would be to place small balls throughout the inner volume located before the capsule, thus distributing randomly the momentum in all directions before reaching the capsule.

3.4.4 Heat transfer & power characterisation

Previous observations have indicated that temperature measurements at the surface and within the capsule are not reliable. Furthermore, given that plug flow conditions are not observed, an analysis of the instantaneous power exchanged is not possible, since the distribution of the power injected at the inlet over time is unknown. In this context, the only viable approach to characterising the HeatSel[®] heat transfer would be to conduct an energy balance over the entire test period.

To illustrate the process, an example of a charge at 50°C is used. The losses can be characterised as follows:

$$P_{\text{loss}} = \frac{T_{\text{stream}} - T_{\text{room}}}{R_{\text{th}}}$$

with $T_{\text{stream}} = \frac{T_{\text{inlet}} + T_{\text{outlet}}}{2}$ and $T_{\text{room}} = 21^\circ\text{C}$ and R_{th} is determined in Section 3.1.

A test presented in Figure 39 in Appendix A.2, in which the temperature of the flanges was measured as a function of time, indicates that the device can be considered to rise in temperature by approximately 0.28°C/min. Based on this result, the energy lost in the device can be approximated as follows:

$$E_{\text{device}} = c_{\text{device}} \times \frac{0.28}{60} \times t_{\text{charge}}$$

where t_{charge} corresponds to the duration of the charge test considered.

Also :

$$E_{\text{in-out}} = \sum_{i=0}^{t_{\text{charge}}} c_{\text{water}} \dot{V} (T_{\text{inlet},i} - T_{\text{outlet},i})$$

$$E_{\text{loss}} = \sum_{i=0}^{t_{\text{charge}}} P_{\text{loss},i}$$

Normally, the energy remaining in the balance should represent the impact of the capsule and should therefore be close to the sum of the latent heat of fusion and the sensible heating of the PCM associated with the test under consideration. The results are as follows:

$$\begin{aligned} \text{Energy balance : } E_{\text{capsule}} &= E_{\text{inlet}} - E_{\text{outlet}} - E_{\text{loss}} - E_{\text{device}} && \approx 4000 \text{ kJ} \\ \text{Expected : } &= \rho_{\text{PCM}} V_{\text{PCM}} (\Delta h_{\text{PCM}} + c_{\text{PCM}} \Delta T) && \approx 100 \text{ kJ} \end{aligned}$$

It is evident that an energy balance based solely on reliable measurements (T_{inlet} , T_{outlet} and $\dot{V}$) is unable to isolate the behaviour of the capsule. It is probable that a considerable proportion of the remaining energy is actually dissipated through the sensible heating of water ; considering that the hot water enters and, instead of pushing the cold water out, first forms a stratification and then gradually heats the cold water. Similar considerations apply in the opposite direction to a discharge.

This concludes that improvements are required to the experimental set-up in order to allow for an adequate characterisation of the heat transfer associated with the capsule. These include the isolation of the internal sensor, the use of plug flow conditions without natural convection and also an increase of the number of temperature sensors to gain further insight into the overall device behavior.

In addition to the experimental work, numerical simulations were conducted to represent the HeatSel[®] heat transfer problem in a simplified manner. The objective was to ascertain whether it was possible to obtain a close approximation to the observed behaviour in reality or to assess the influence that simplifications might have.

As the capsule does not have any axis of symmetry, a simulation in three dimensions was necessary. The Ansys[®] Fluent[®] computational fluid dynamics (CFD) software was selected for this purpose. It was deemed an appropriate choice for the objective, given that it includes viscous-laminar flow and solidification/melting models.

4.1 Computational domain definition

Fluent[®]'s approach to solving fluid dynamics problems is as follows. The initial step is to provide an input model of the body to be analysed, comprising fluid, solid or vacuum regions. Based on the conditions specified, Fluent[®] will divide the geometry into a large number of finite volumes. These volumes represent the computational domain within which fluid mechanics equations will be solved. Once all these volumes have been processed, a comprehensive solution for the entire domain can be obtained. This section consequently presents the geometry that has been selected to model the problem and the mesh that has been created.

4.1.1 Geometry selection

It was essential to select an appropriate geometry that excludes all extraneous influences, with the sole exception being the capsule. In light of this, a straightforward solution was to extract the volume of fluid within the device used during the experimental tests to create the domain. However, as a perfectly uniform flow can be imposed in the simulation, it is no longer necessary to consider a flow development zone, as was done in the experiment. This part has therefore been truncated. The resulting configuration is illustrated in Figure 18.

One might question the rationale behind maintaining the conical reduction in the geometry : during the simulation, conditions will be defined at the inlet, and the aim will be to observe the difference at the outlet by averaging the variables over the surface ; and it is generally accepted that averaging over a smaller area achieves greater accuracy.

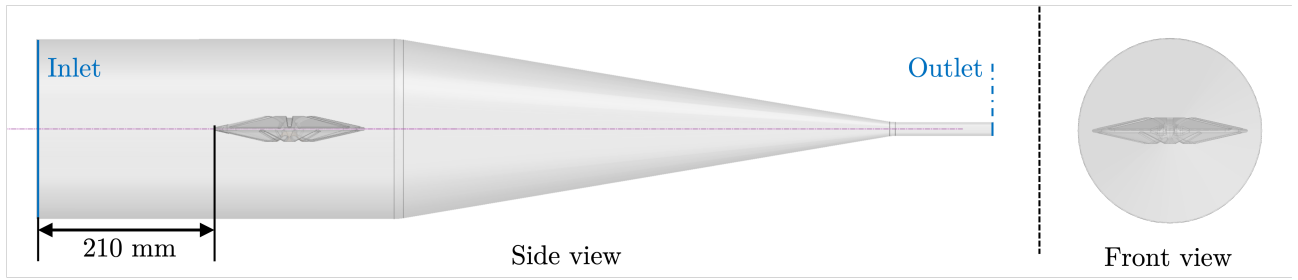


Figure 18: Geometry of the computational domain selected to represent the heat transfer between a fluid and the PCM capsule. The design is based on that of the experimental device, but truncated at the inlet.

As can be observed, the domain is constituted of two fluid regions, separated by a boundary in the form of the capsule. The enclosed region will be designated as containing PCM, while the water will be situated in the other region delimited by the inlet, outlet and the boundary with the environment.

4.1.2 Mesh

To mesh the entire domain, it is first necessary to divide the surfaces that define the boundaries. Following this, an extrusion process is applied to the face with a specified shape inwards, yielding to a volume mesh. However, it is important to define the dimensions of the elements in such a way that they can accurately capture all relevant flow features. In the context of this example, it is evident that there is no need to apply the same level of detail at the interface between the two fluids as further away where the solution will change slowly. With the aforementioned consideration in mind, a maximum face size of 10 cm and a minimum of 1 mm were set for the boundaries. The refinement between these two dimensions was determined by two conditions. First, the closer the face was to an edge, the smaller it had to be. Secondly, a straight face could not exceed a geometric curvature of 14° .

To generate the volume mesh, four boundary layers of cells were set in both PCM and water domains to ensure the precise capture of thermal effects around the capsule. The dimensions of the generated meshes can be found in Tables 7 and 8, with visual representations of the result presented in Figures 19, 20 and 21.

Surface Mesh	
# of faces	57 278
# of nodes	239 373
Target min size [mm]	1

Table 7: Boundary mesh size summary

Volume mesh	
Type of cells	Polyhedra
# of cells	728 308
# of faces	3 519 755
# of nodes	2 199 906
# of boundary layers	4

Table 8: Volume mesh characteristics

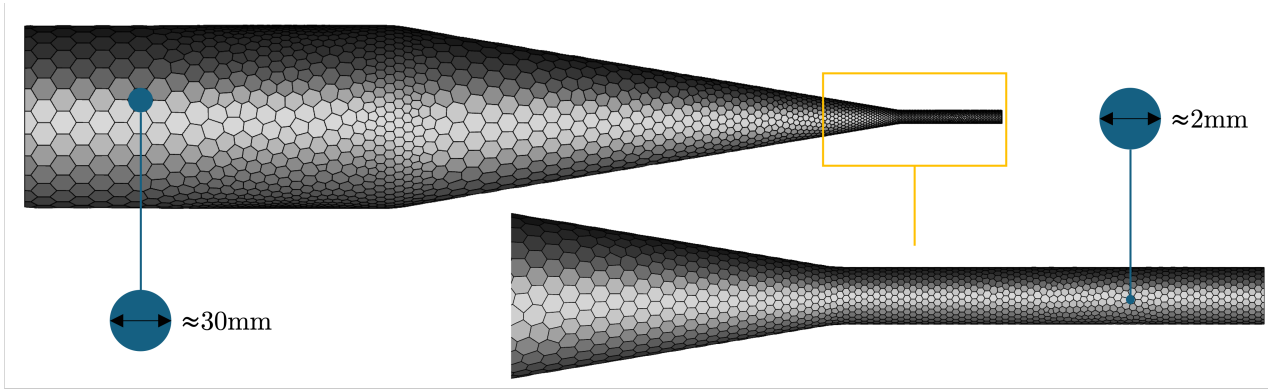


Figure 19: Mesh generated on the boundary between the water region and the outside. The faces on the 213.1 mm diameter cylinder are approximately 15 times larger than those on the outlet cylinder.

As can be observed in Figure 19, the constraint placed upon the curvature proved to be a more significant limitation than the maximum target size, as the width of the largest cells is around 3 cm and not 10 cm.

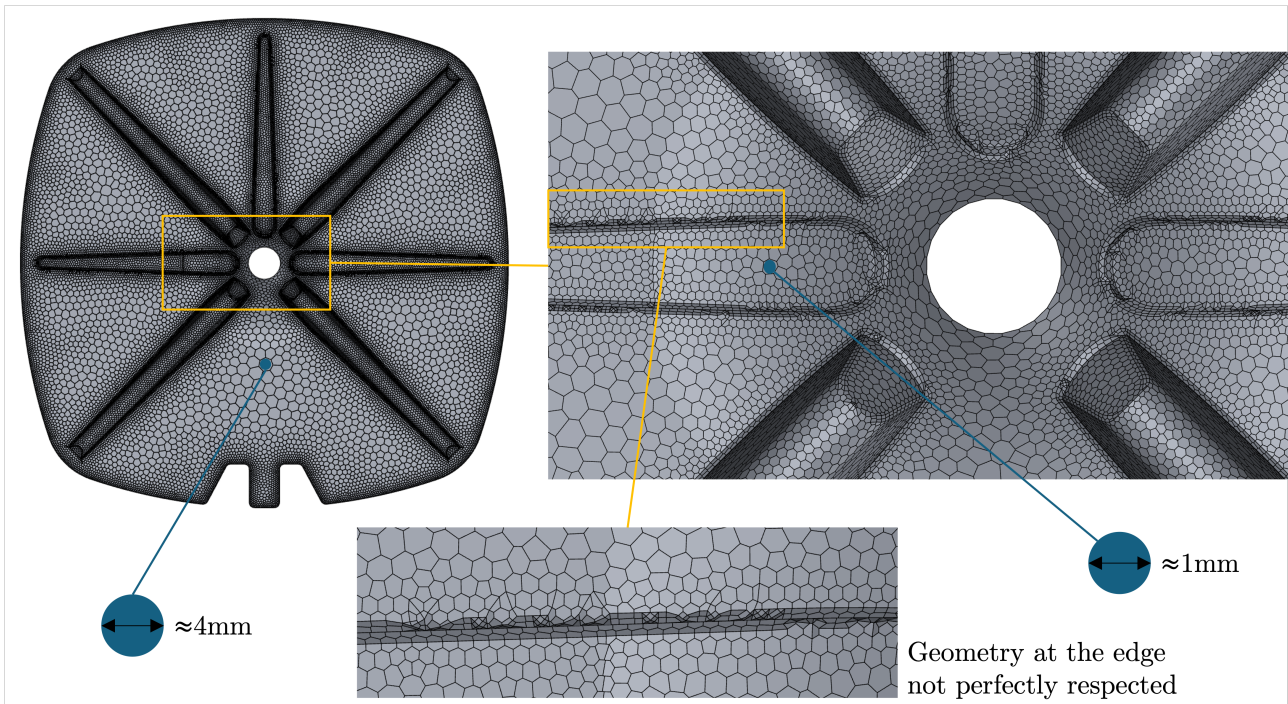


Figure 20: Visualisation of the surface mesh of the capsule boundary. The faces are smaller in the grooves than on the flat surfaces. Upon close examination, it is shown that the subdivision of the edges of the shallow grooves does not perfectly respect the initial geometry.

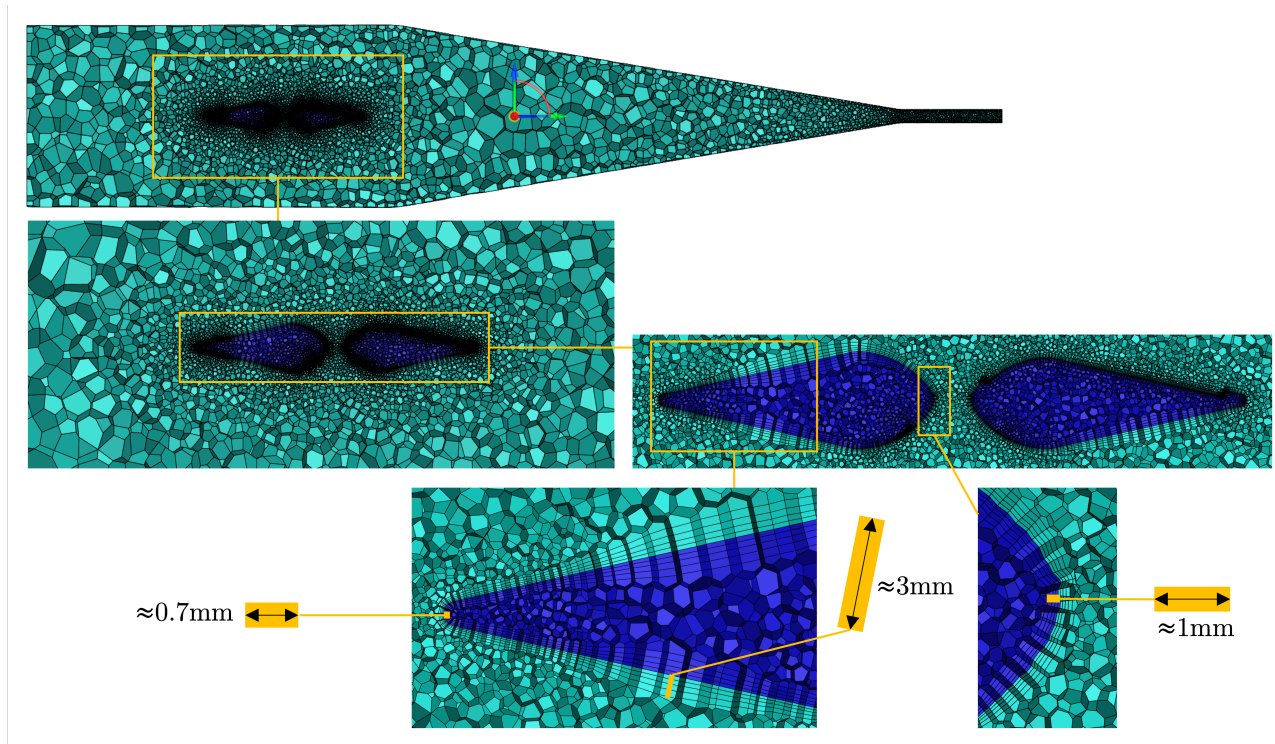


Figure 21: Representation of the volume mesh at all scales. Precise refinement around the capsule is clearly observed. The boundary layer of cells on either side of the boundary have a minimum size of 0.7 mm for 4 stacks.

Quality assessment

The irregularity displayed in Figure 20 illustrates that the division of the domain is not always optimal. Several indicators can be used to evaluate the quality of a mesh, with particular focus on the following:

- The **Skewness** value determines the degree of deviation of a face or cell from the ideal equilateral or equiangular configuration. Faces and cells with high skewness values are characterised by nodes that are almost coplanar, which is not acceptable because the equations solved assume that the cells are relatively equilateral or equiangular. The metrics^{1 2} are listed in Table 9.
- The **Orthogonal Quality** quantifies the angular relationship between cell faces and edges. A value closer to 90° indicates a more optimal alignment, which in turn enhances the precision of the solution. The values are subsequently normalised on a scale of 0 to 1. The metrics are reported in Table 10.
- The **Aspect Ratio** of a face or cell is defined as the ratio of the longest edge length to the shortest edge length. It is recommended³ that the average aspect ratio of all faces or cells be approximately 10, with a maximum of 20.

¹<https://romeo.univ-reims.fr/documents/fluent/tgrid/ug/chp15.pdf>

²https://www.researchgate.net/figure/Mesh-quality-recommendations-29_fig1_344721240

³<https://www.simscale.com/docs/simulation-setup/meshing/mesh-quality/>

Table 9: Skewness Spectrum

Range	
0-0.25	Excellent
0.25-0.50	Very good
0.50-0.80	Good
0.80-0.94	Acceptable
0.95-0.97	Bad
0.98-1.00	Unacceptable

Table 10: Orthogonal Quality Spectrum

Range	
0-0.001	Unacceptable
0.001-0.14	Bad
0.15-0.20	Acceptable
0.20-0.69	Good
0.70-0.95	Very good
0.95-1.00	Excellent

Poor-quality meshes will most likely lead to numerical errors and cause simulations to fail or give unreliable results. In light of this, it is important to analyse these parameters in the context of the present mesh (Tables 11 and 12).

	Best	Worst	Average	Min	Max
Area [m ²]			11	0.02	1090
Equivalent square side [mm]			3.3	0.15	33.0
Aspect ratio	1	∞	2.5	1.01	29.4
Skewness	0	1	0.028	2.30×10^{-8}	0.80785

Table 11: Surface Mesh Quality Metrics

	Best	Worst	Average	Min	Max
Volume [mm ³]			31	7.60×10^{-4}	2.90×10^4
Equivalent cube side [mm]			3.1	0.09	30.7
Aspect ratio	1	∞	4.8	1.4	56.7
Skewness	0	1	0.063	4.4×10^{-5}	0.829
Orthogonal quality	1	0	0.94	0.176	1

Table 12: Volume Mesh Quality Metrics

Both tables indicate that the mesh can be considered of good quality. Indeed, the minimum and average values all fall within the "Very good" and "Excellent" criteria. However, it should be noted that the maximum aspect ratio values are still somewhat elevated for the volume mesh. Upon closer examination, it becomes evident that only 705 cells exhibit an aspect ratio exceeding 5 out of a total of 728 308. This indicates that these are isolated instances that will not significantly impact the overall outcome. With regard to the surface mesh, Figure 22 illustrates that the quality is comparatively poorer in the edges of the shallow grooves, as previously indicated in Figure 20. However, this should not be considered a significant issue, as the irregularities are only present on the edges and, in addition, there are only 62 faces out of 57 278 with a skewness of more than 0.5.

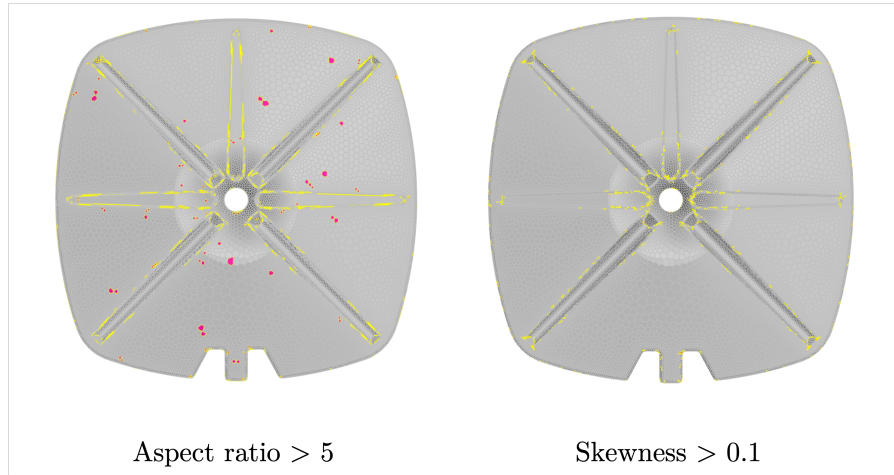


Figure 22: In red and yellow, the faces of the capsule boundary that are of poorer quality according to the Aspect Ratio and Skewness criteria.

4.2 Part A : Steady determination of the heat transfer coefficient

A stationary numerical simulation was carried out to determine the thermal performance of the HeatSel[®]. The objective was to identify an empirical law governing heat transfer by convection around the capsule, as a function of the type of flow around it and its geometry.

4.2.1 Calculation model

The finite volume method is employed here to resolve the common continuity, momentum and energy equations, also known as the Navier-Stokes equations. In general, these equations are challenging to solve using conventional mathematical methods. Here, they are first discretised, then linearised and finally solved using a CFD solver by iterative methods[33].

In this initial set of simulations, the precise behaviour of the PCM during phase change will not be analysed. Instead, a simplification considering that the capsule has infinite latent energy and remains at its phase change temperature regardless of the surrounding. In this context, a constant and uniform temperature throughout the capsule's boundary can be considered. Given this information, it is no longer necessary to solve the problem for the elements located inside the capsule. Consequently, the meshing of this region has been deactivated. In addition to this assumption, gravity has been neglected, the outer boundary has been considered to be adiabatic, and the water properties have been assumed to be constant and equal to those specified in Table 4.

The operational framework of the simulation is relatively straightforward : a constant flow of water at a specific temperature is directed from the inlet to the outlet, where it encounters the capsule, which is at a different temperature. This results in an exchange of heat. The outlet temperature and power will be monitored in order to characterise the phenomenon.

Flow rates ranging from 0.03 to 1 kg/s will be considered, resulting in Reynolds numbers ranging from 43 to 1400. Therefore, laminar flow will be assumed for all cases.

In light of the above considerations, a stationary, three-dimensional, laminar model is employed. The coupled pressure-velocity algorithm has been selected with a second-order upwind scheme, and gradients have been calculated using the least squares cell-based method.

Convergence criterion

In numerical simulation software such as Fluent[®], equations are typically solved iteratively. This process involves assigning discrete values to each of the variables on either side of the equation and then changing these values in a certain direction at each iteration. The objective of this approach is to reduce the difference between the left and right sides of the equation, which is known as the residual.

When a problem is simulated, a residual value for each of the equations solved is displayed, and as the iterations increase, this value is expected to decrease. If not, this may indicate that the problem is wrongly posed or that the mesh is not sufficiently dense to capture the key effects.

Prior to running the simulation, the author must therefore set a residual value for each equation at which they consider the solution to be sufficiently converged. By default, Fluent[®] suggests setting all residuals to 1×10^{-3} , with the exception of the one for the energy equation, which is set to 1×10^{-6} . In order to ensure the accuracy of the solutions, the criteria were increased to 1×10^{-4} and 1×10^{-8} for this study, as was also done in [33]. However, the default values could also have been chosen [34].

4.2.2 Grid sensitivity analysis

The mesh parameters presented in Section 4.1.2 were not arbitrarily selected; rather, they were derived from a sensitivity analysis. Indeed, it is generally accepted that a well-posed subdivision of a computational domain should not influence the solution. In this context, five different refinements were tested, each time with different values for the minimum size of a surface element, the number of boundary layer cells near the boundary of the capsule and the curvature normal angle. As previously discussed in Section 4.1.2, this final parameter specifies the maximum angle that an edge of an element is permitted to span in accordance with the specific curvature of the geometry.

Table 13 illustrates that the values of T_{outlet} and P_{glob} are very similar for the three most refined configurations, while they are more disparate for the first two. This indicates that these two options do not fully capture the details of the problem. Furthermore, it is clearly observable that an increased number of cells does not necessarily result in superior mesh quality. In point of fact, although the third configuration has half as many cells as the last one, it exhibits the best characteristics in terms of aspect ratio and orthogonal quality. In all five cases, the flow and heat transfer imbalances are of negligible order of magnitude and therefore do not require further consideration.

Given that the smaller the number of cells in the mesh, the shorter the calculation time, it is evident that the third configuration is the optimal choice. This configuration provides an equivalent solution and superior mesh quality compared to those with more cells.

Target min surface size [mm]	5	3	1	0.8	0.65
# of boundary layers	0	2	4	6	6
Curvature normal angle [°]	18	16	14	12	12
# of cells	25 485	74 307	407 436	798 414	912 444
max Aspect Ratio	57.0	65.5	56.7	69.3	112.1
min Orthogonal quality	0.08	0.07	0.20	0.17	0.08
T_{outlet} [°C]	21.6686	21.5811	21.5541	21.5534	21.5533
P_{glob} [W/m ²]	2863.4	2468.6	2338.8	2336.0	2335.6
# of iterations to converge	502	443	423	422	420
Flow rate imbalance $\times 10^{-11}$ [kg/s]	3.47	4.42	5.17	6.09	4.10
Heat transfer imbalance [W]	0.183	0.165	0.149	0.150	0.157

Table 13: Quantitative analysis of the change in key problem variables as a function of mesh size. Refinement was carried out using three parameters: the minimum surface size, the number of boundary layer cells and the curvature normal angle. The test conditions were defined as follows : $T_{\text{inlet}} = 21^\circ\text{C}$, $T_{\text{capsule}} = 42^\circ\text{C}$ and $\dot{m} = 0.07\text{ kg/s}$. P_{glob} represents the thermal power density considered over the entire boundary of the capsule. The column coloured blue indicates the configuration that was ultimately selected.

4.2.3 Results

The results of the 16 simulations are reported in Table 14.

	$\dot{m}$ [kg/s]	Re	T_{inlet} [°C]	T_{capsule} [°C]	T_{outlet} [°C]	P [W]	h [W/m ² /K]	Nu
D	0.03	43	80	50	78.7273	160.7	76.8	4.02
C	0.05	72	30	60	30.9530	200.5	95.8	5.02
C	0.07	100	40	50	40.2642	77.7	111.4	5.83
C	0.07	100	21	42	21.5541	163.1	111.4	5.83
D	0.07	100	60	42	59.5249	139.8	111.4	5.83
C	0.07	100	35	75	36.0550	310.7	111.4	5.83
D	0.1	143	50	25	49.4560	228.7	131.2	6.87
C	0.15	215	25	60	25.6151	387.9	158.9	8.32
C	0.2	287	30	39	30.1369	114.8	182.8	9.57
D	0.3	430	55	10	54.4421	704.3	224.4	11.75
D	0.4	574	60	42	59.8061	328.2	261.4	13.69
C	0.5	717	25	45	25.1956	412.7	295.8	15.49
D	0.6	860	70	30	69.6370	916.8	328.6	17.21
D	0.7	1004	45	30	44.8721	376.9	360.2	18.86
C	0.8	1147	15	40	15.2023	681.5	390.8	20.46
C	1	1434	21	42	21.1564	659.0	449.9	23.56

Table 14: Summary of all the cases simulated to determine the thermal performance of the capsule as a function of the imposed flow rate and its geometry, assuming a constant and uniform temperature at its boundary. The letters "C" and "D" represent the concepts of charge and discharge, respectively.

With T_{outlet} and P_{glob} [W/m²] as simulation output and T_{inlet} , T_{capsule} , and $\dot{m}$ fixed, the Reynolds number of the flow (Re), the convective heat transfer coefficient (h), and the Nusselt number (Nu) can be calculated using the following formulae:

$$P = P_{\text{glob}} \times A_{\text{capsule}} \quad Re = \frac{\rho_{\text{water}} \times \frac{\dot{m}}{\rho_{\text{water}} D} \times L_c}{\mu_{\text{water}}} \quad h = \frac{P_{\text{glob}}}{|T_{\text{outlet}} - T_{\text{inlet}}|} \quad Nu = \frac{h L_c}{k_{\text{water}}}$$

Table 14 indicates that the value of h , and consequently also Nu , varies only when the flow rate changes. As commonly understood [35], convection coefficients around objects are only influenced by the flow and the geometry of the object itself. In this case, the capsule boundary remains obviously the same throughout all tests, and given that water properties are assumed constant, the mass flow rate completely characterises the flow dynamic. Therefore, it is objective to observe a consistent value of h for a given flow rate under different temperature conditions.

It should first be noted that the lines in blue indicate the simulations where the imposed flow rate is identical to that imposed during the experimental procedure. For subsequent analysis, it is thus important to remember that the convective heat transfer coefficient under uniform, laminar flow around the HeatSel[®], with no losses to the surroundings and a mass flow rate equivalent to that of a packed bed is :

$$h = 111.4 \text{ W}/(\text{m}^2 \text{ K}) \quad (4.1)$$

Size of thermal boundary layer

The way thermal transfer occurs along the length of the capsule is depicted in Figure 23.

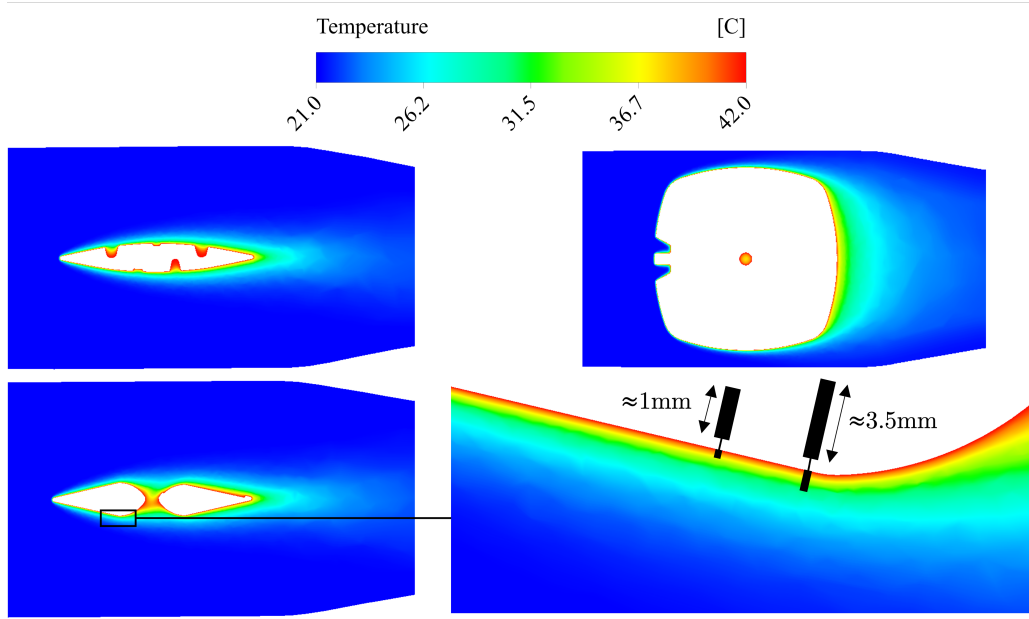


Figure 23: Evolution of the temperature profile around the capsule for $T_{\text{inlet}} = 21^\circ\text{C}$, $T_{\text{capsule}} = 42^\circ\text{C}$ and a $\dot{m} = 0.07 \text{ kg/s}$. The left-hand side of the figure displays two cross-sections of the capsule, with a side view. The right-hand side of the figure displays a top view and a zoom on the thermal boundary layer. The thickness of the thermal boundary layer is approximately 1 mm for a temperature difference of 5°C and 3.5 mm for a temperature difference of 10°C .

Only a single example is presented here in Figure 23, but analysing the temperature profile for the 16 simulated cases allows to posit that the impact of the capsule in the rear environment

is more pronounced at higher flow rates. Nonetheless, the influence remains relatively localised, and therefore it can be reasonably concluded that the assumption of an adiabatic boundary with the environment is not a limiting factor in the representation of the problem.

Furthermore, Figure 23 demonstrates that the boundary layer with $\Delta T = 5^\circ\text{C}$ is approximately 1 mm-thick, which is precisely the size of the temperature sensor used in the experiment. This, therefore, reinforces the conclusion that the experimental surface temperature measurements cannot be regarded as reliable.

Empirical law of heat transfer around the capsule

The heat transfer coefficient for flows around conventional bodies can be calculated with exactitude using the boundary layer equations. However, for flows where detachment occurs, for example around cylinders or spheres, the heat transfer coefficients are exceedingly challenging, if not impossible, to calculate and can therefore only be determined by experiments or simulations [36]. Frequently, calculated or measured results can be described by empirical correlations of the type $Nu = f(Re, Pr)$.

Figure 23 demonstrates that boundary layer detachment occurs with HeatSel[®], thereby refuting the hypothesis that heat transfer could have been characterised analytically. This is a logical conclusion, given the complex geometry of the capsule and the numerous grooves. Consequently, the values presented in Table 14 can be extracted to develop an empirical law for convective heat transfer in laminar flow around the capsule. It should be remembered that the properties of the water were considered to be constant throughout the tests, and thus the Prandtl, only depending on those properties, will not be involved in the relationship of this case. The least squares method was employed to identify a correlation between the Nusselt numbers and the Reynolds numbers, which is depicted in Figure 24.

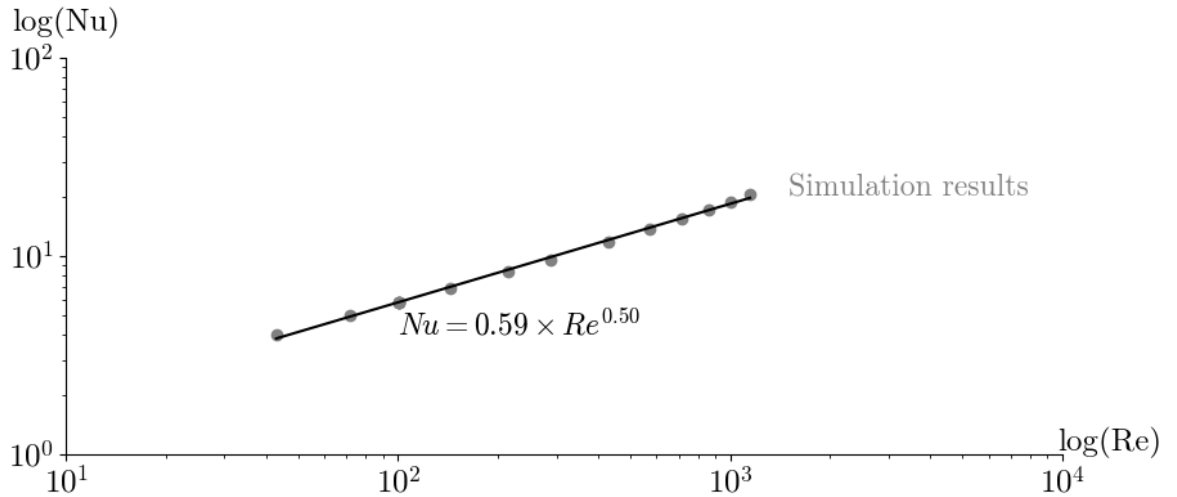


Figure 24: Empirical law for heat transfer by forced convection with a laminar flow of water around the capsule. The relationship is expressed in the form of $Nu = C \times Re^\alpha$, although a linear representation is achieved by the use of logarithmic axes.

In order to ascertain this relationship, it is necessary to compare it with other relationships that have already been validated in the literature. Given that the capsule is relatively thin in

comparison to its length (30mm vs 180mm, see Section section 2.2), a suitable comparison might be to examine convection transfer around a plate with laminar flow. This can be expressed as follows [36]:

$$Nu = 0.664 Re^{1/2} Pr^{1/3} \quad (4.2)$$

By determining the Prandtl number of the present problem, it is possible to isolate it from the constant and return to a form similar to Equation 4.2.

$$Pr = \frac{\mu_{\text{water}} c_{\text{water}}}{k_{\text{water}}} = 4.28$$

Giving :

$$Nu = 0.354 Re^{1/2} Pr^{1/3} \quad (4.3)$$

Equation 4.3 shows that the form of the empirical law generated for the HeatSel® is very likely to be relevant, since the dependence on the number of Reynolds is expressed with exactly the same exponent as for the plate. Furthermore, it seems reasonable to assume that for the same number of Nusselt and Reynolds, the heat transfer with the capsule would be approximately two times less efficient than with a perfect plate.

4.3 Part B : Transient simulation of solidification in experimental context

In order to model the phase change occurring within the capsule in response to an external flow, it was decided to carry out a transient simulation, considering now the two fluid regions and therefore the entire domain presented in Section 4.1.2. The aim is to evaluate the thermal performance of the capsule when the phase change is occurring. A comparison of the results with those of previous simulations is also presented.

4.3.1 Phase change modelling

With regard to the modelling of the water flow around the capsule, there is no difference from what was specified for the previous simulations in Section 4.2.1. Nevertheless, the modelling of the phase change must also be considered here. This can be achieved in the Fluent[®] software by using the model called Solidification & Melting. What is distinctive about this model is that the liquid-solid front is not tracked explicitly, but rather the enthalpy-porosity formulation is employed [33]. In this method, a quantity designated as the liquid fraction, which indicates the proportion of the cell volume that is in liquid form, is associated with each element within the domain. The energy equation of the PCM is therefore as follows [34]:

$$\text{Energy: } \frac{\partial(\rho H)}{\partial t} + \nabla \cdot (\rho V H) = \nabla \cdot (k \nabla T)$$

where k is the thermal conductivity and H is the specific enthalpy, which is defined as the sum of the sensible enthalpy (h) and the latent heat (ΔH):

$$H = h + \Delta H$$

with:

$$h = h_{\text{ref}} + \int_{T_{\text{ref}}}^T c_p dT$$

The content of the latent heat can vary between 0 (completely solid) and Δh_{PCM} (completely liquid), as specified in Table 1, following this equation:

$$\Delta H = \beta \Delta h_{\text{PCM}}$$

with the liquid fraction (β) defined as:

$$\beta = \begin{cases} 0 & \text{if } T < T_{\text{solidus}} \\ 1 & \text{if } T > T_{\text{liquidus}} \\ \frac{T - T_{\text{solidus}}}{T_{\text{liquidus}} - T_{\text{solidus}}} & \text{if } T_{\text{solidus}} < T < T_{\text{liquidus}} \end{cases}$$

By definition, solidus and liquidus temperatures are the maximum temperature at which the material is at solid state and minimum temperature at which the material is at liquid state, respectively.

This model is an effective way of easily addressing complex phase change problems. However, it is evident from its formulation that certain assumptions must be made before it can be applied.

- The PCM must be considered as a homogeneous binary mixture of solid and liquid, with constant properties regardless of the phase. In the present case study, this assumption has been accepted as it stands, as only constant values are listed in the datasheet (see Table 1) and the capsule has not been disassembled re-measure the properties, as mentioned in Section 2.2.
- Gravity and therefore buoyancy forces must be neglected. This is a rather strong assumption, given that natural convection can have a significant impact on phase change problems. However, it has been shown [37] that during the solidification process, the front moves from the periphery towards the centre of the object and the solid therefore encloses the liquid, preventing it from moving freely. In this context, natural convection can be reasonably neglected ([12], [38]). The demonstration was conducted on a sphere, but it is reasonable to assume that the same principles can be applied to the present geometry. Indeed, as the capsule is positioned horizontally with its small thickness serving as a representative length in the direction of gravity, it can be posited that little natural convection is occurring during the phase change. This consideration would certainly no longer be true if the capsule had been positioned vertically with the large dimension in the direction of gravity. In summary, neglecting natural convection is a suitable approach, provided that only horizontal solidification processes are considered.
- The phenomena of phase segregation and subcooling, which are often observed in reality with phase change materials ([8]–[10]), are not taken into account.

In the geometry presented in Section 4.1.1, the boundary between the water region and the PCM region is drawn as infinitesimal. To more accurately represent the real capsule, the software's "3D thin-wall conduction" option was used. This creates multiple layers of virtual mesh cells at the boundary, allowing a conduction model to be solved across and parallel to the wall ([21], [34]). In this case, the data in Table 2 were coded as parameters.

Transient problem characteristics

The simulation of PCM solidification due to laminar water flow around the capsule is as follows: the entire domain is initialised at a temperature of 62 °C, and at $t = 0$, water at 21 °C is injected into the inlet at a flow rate of 0.07 kg/s. The flow is first replacing the hot water that was initially present, and then the capsule, which is still at a high temperature, will be subjected to a permanent cold flow. This process will gradually cause the paraffin to solidify. After 7500 s, or 125 min, the simulation will end. The duration was selected based on the temperature evolution observed in a real experiment (see Figure 12), where the solidification time was approximately 70 min. This providing a sufficient margin for the simulation.

In transient simulations, it is necessary to select a time step at which the solution will be calculated. The smaller the time step in relation to the duration, the longer the calculation time will be, but a time step that is too large will result in lower accuracy, as some fast phenomena may not be modelled. As was done in Section 4.2.2 for meshes, it is generally necessary to perform a sensitivity analysis to demonstrate that the time step chosen does not

directly influence the simulation results. However, given that the computation time required on a conventional computer, such as the one used in this study, was approximately 18 hours, it was deemed impractical to conduct such an analysis. Instead, a reference study involving identical models was consulted [21], in which the same process was employed to solidify a capsule of a different shape with a solidification time of approximately 100 minutes. Their sensitivity analysis indicated that a time step of 5 seconds is necessary to capture the full thermal range of the problem, and this is therefore the value considered for this study.

The SIMPLEC (Semi-IMPlicit with Pressure Linked Equation Consistent) algorithm was selected with a second-order upwind scheme, and gradients were calculated using the cell-based least-squares method. For each of the 1500 time iterations, it was necessary to achieve convergence of the residuals. In this instance, the default values were chosen (1×10^{-6} for the energy equation and 1×10^{-3} for the rest). The maximum number of iterations for each time step has been set at 20. However, it should be noted that this value only served to limit the iterations in the initial stages. Once the simulation had reached a sufficient level of advancement, convergence was typically achieved within 10 or fewer iterations.

4.3.2 Results

To gain an insight into the thermal dynamics of the capsule during the solidification process, a detailed examination of the temperature profile, the evolution of the liquid fraction, and the spatial power distribution is conducted.

Temperature profile over the entire computational domain

Figure 25 depicts the temperature profile obtained, indicating that it takes approximately two minutes for the flow to reach the capsule position. At this juncture, the contact point divides the flow into two symmetrical parts. The findings of Figure 23, which concerned the detachment of the boundary layer and the region of influence in the rear environment, are also evident in this simulation. However, a noteworthy observation can be made here : frames 3 and 4 demonstrate that the process of solidification is not uniform. This is attributed from the observation that the zone of temperature around 40°C (yellow-green colour) is smaller on the left side than on the right side of the cross-section of the capsule. The cause of this phenomenon can be attributed to the fact that the flow originates from the left, resulting in the left side of the capsule being subjected to a greater temperature difference and, consequently, a greater heat flux. By way of metaphor, fluid arriving at the rear of the capsule will always be preheated by its passage next to the front part, resulting in an smaller temperature difference at the rear. This demonstrates the relevance of the simulation, as such behaviour could never have been observed in the case of uniform boundary conditions.

Evolution of the liquid fraction

Frame 2 of Figure 28 corroborates the non-uniformity of the solidification phenomenon, as it is readily apparent that the liquid fraction is lower on the left side of the capsule. Furthermore, the utility of the numerous grooves on the capsule boundary is apparent. Indeed, the cross-sections with the fewest grooves exhibit the longest phase change duration. This can be explained by the fact that the grooves, despite locally triggering a detachment of the boundary layer, allow the flow to enter the depth of the PCM, thereby facilitating heat exchange.

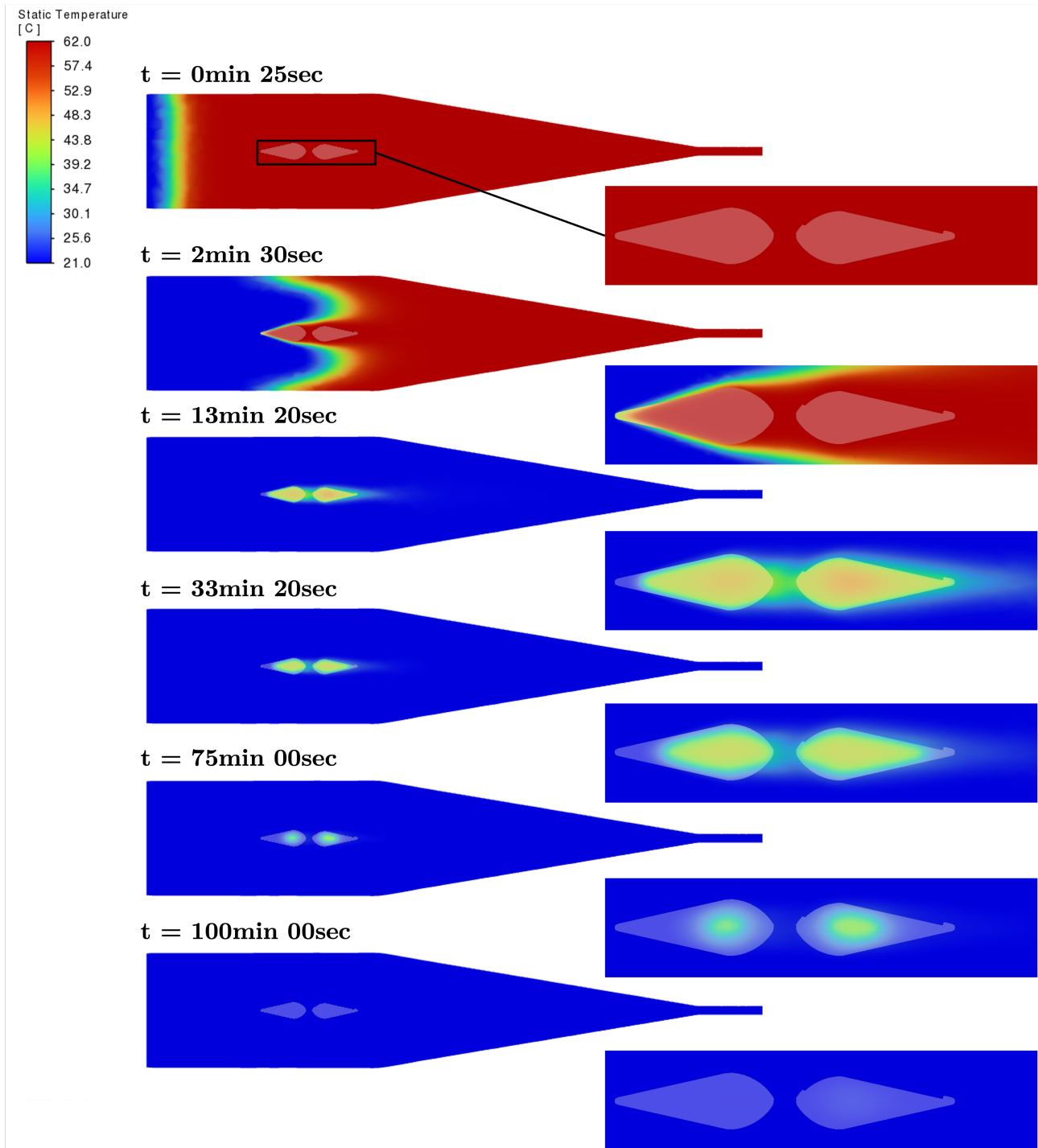


Figure 25: Temperature evolution on a cross-section (middle of the capsule) of the entire domain for six moments. The completion time is approximately 100 min. It can be observed that a phase change is occurring, as the external flow is at 21 °C, while a (decreasingly large) fraction of the PCM inside the capsule remains at around 40 °C. Frames 3 and 4 suggests a non-uniform solidification due to greater temperature gradients on the left side.

The continuous evolution of the liquid fraction as a function of time is illustrated in Figure 26, with the frames of Figure 28 highlighted. It should be noted that this graph demonstrates that the phase change process, in this case solidification, is not linear as a function of time. In contrast, it takes approximately 20 min to solidify the first half of the volume, followed by 60 min (3× longer) for the remaining half.

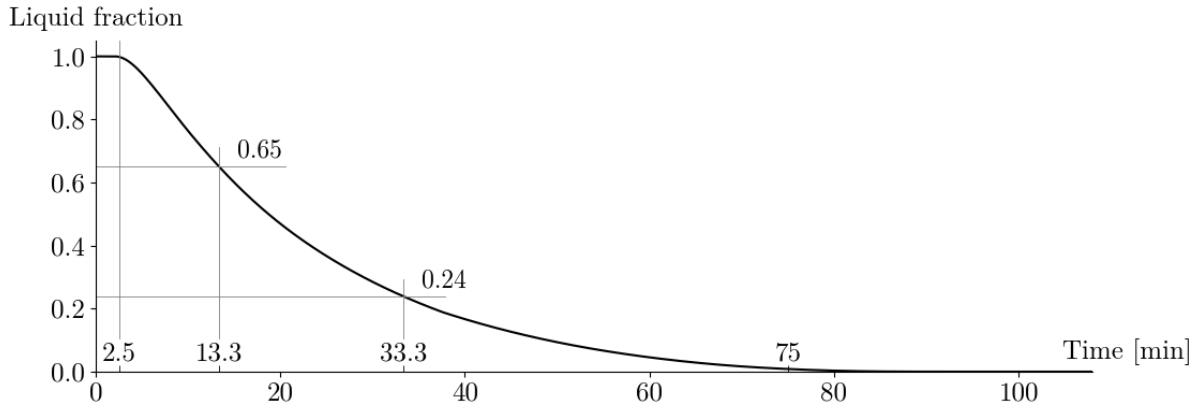


Figure 26: Continuous evolution of the liquid fraction over the entire volume of PCM. The moments represented in Figure 28 are highlighted on the curve. It can be observed that the second half takes approximately three times longer to solidify than the first.

Heat flux at the capsule boundary

The evolution of the spatial distribution of the power density at the wall is illustrated in Figure 29. It can be observed that the moment of maximum local power occurs when the flow at 21 °C reaches the first point of contact with the capsule, with a value of approximately 240 W (3452 W/m²). It is crucial to recognise that after this moment, such a temperature difference at the boundary will never recur, given the fact that the fraction already subjected to the flow will have cooled and the unexposed portion will be subjected to a preheated flow. After about five minutes, it can be observed that a relatively uniform exchange of heat is taking place throughout the wall. Then, the surface area undergoing exchange will gradually decrease as a consequence of the progressive solidification and cooling to 21 °C of the entire depth. This is corroborated by the constant decline observed in the curve following the peak in Figure 27.

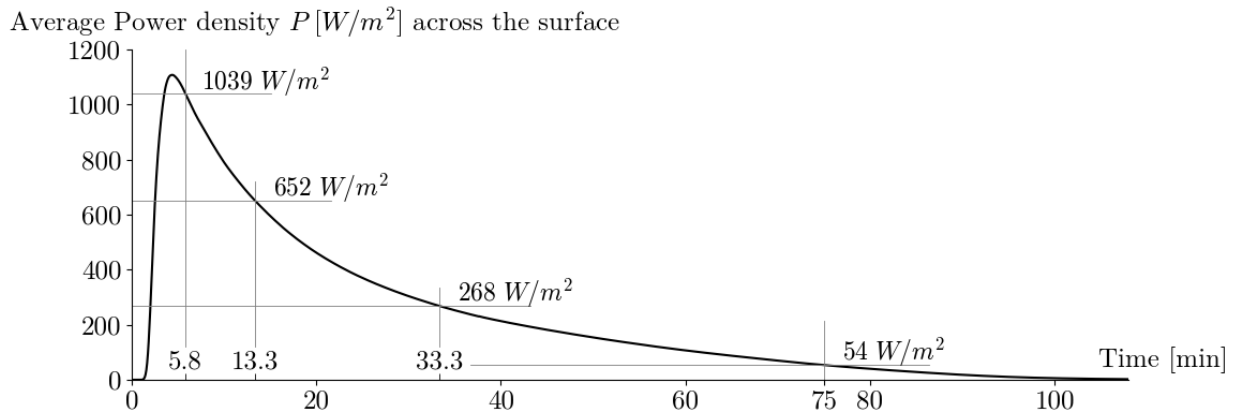


Figure 27: Continuous evolution of the power density over the entire boundary surface of the capsule. The last four moments represented in Figure 29 are highlighted on the curve. Following the peak, the power density decreases as the temperature difference between the paraffin volume and the external flow gradually decreases.

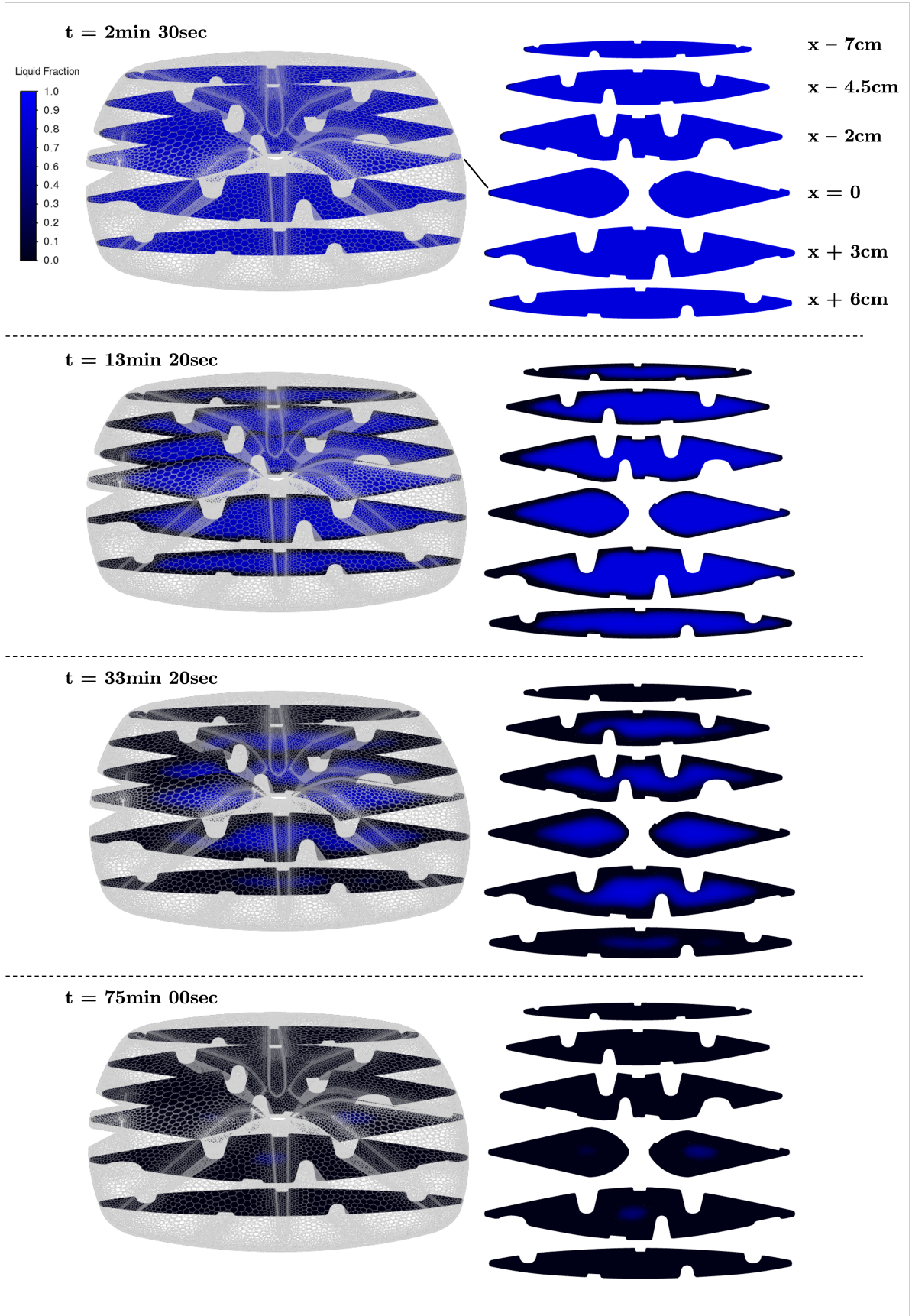
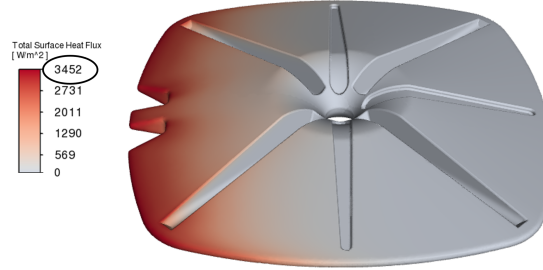
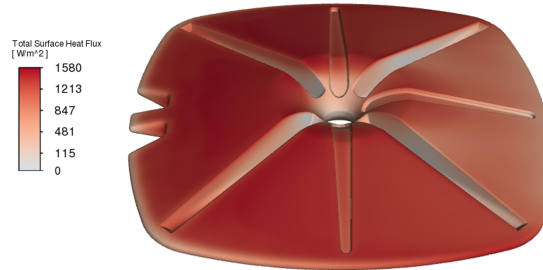


Figure 28: Evolution of the liquid fraction within the capsule at four distinct moments of the solidification process. The capsule is characterised using six planes, each representing a different depth. By observing the phase change on each of these planes, the overall behaviour of the capsule can be deduced. The second time step clearly demonstrates that solidification is not uniform, with the left side solidifying faster than the right.

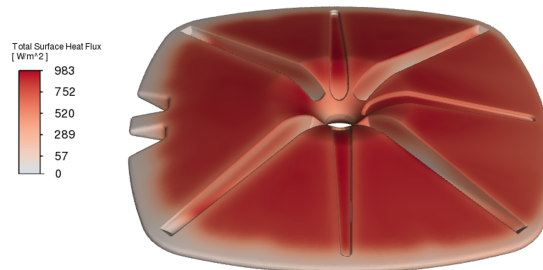
t = 2min 30sec – Local Power at maximum



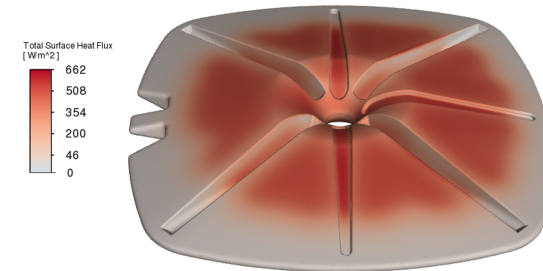
t = 5min 50sec – Power everywhere over the surface



t = 13min 20sec – Power where PCM not fully solid



t = 33min 20sec



t = 75min 00sec

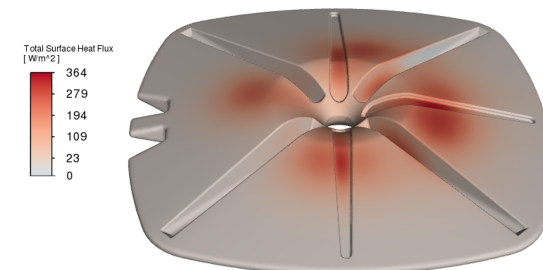


Figure 29: Visualisation of the spatial distribution of power density at five different times. The local power peak is observed at the moment of initial contact between the cold water and the capsule. Subsequently, the distribution becomes uniform across the entire boundary, and then gradually disappears at the points where the PCM is solidified across the entire depth. Note that the scales change from frame to frame.

4.4 Effective heat transfer coefficient correlation

To recall the situation, the initial set of simulations assumed a uniform and constant temperature at the wall in order to model heat transfer. Then, another simulation was conducted in which this assumption was lifted and heat transfer has been determined under solidification conditions. Consequently, it is now necessary to compare the results to determine the order in which the results can be related to one another.

Figure 30 presents the evolution of the heat transfer coefficient over time in the context of the solidification simulation.

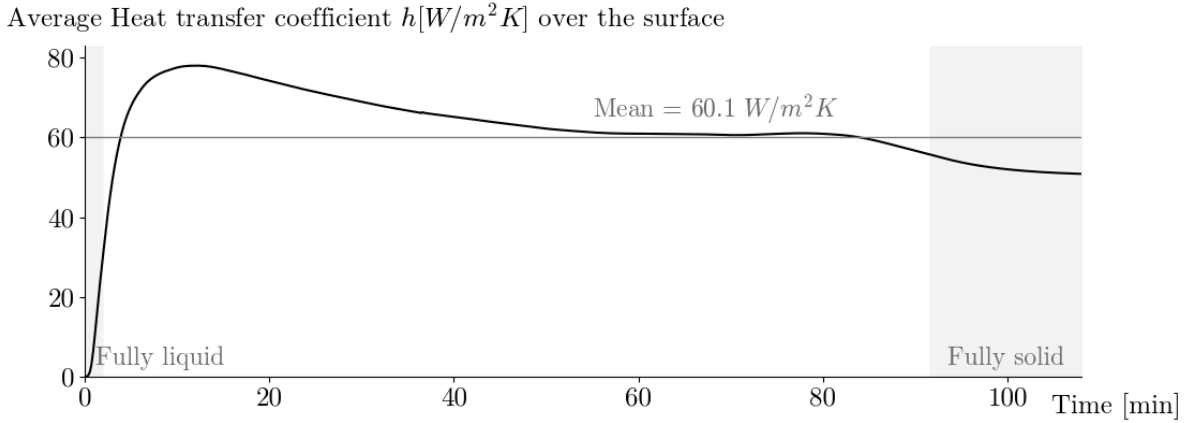


Figure 30: Heat transfer coefficient averaged over the entire capsule boundary for the solidification simulation. The time-averaged value over the entire duration is $60.1 \text{ W}/(\text{m}^2 \text{ K})$.

Given that the flow rate in the solidification simulation is 0.07 kg/s , it could have been anticipated that values of approximately $110 \text{ W}/(\text{m}^2 \text{ K})$ would be observed in Figure 30. Indeed, the initial simulations determined that convection around the capsule for such a flow rate should be $111.4 \text{ W}/(\text{m}^2 \text{ K})$ (see Equation 4.1). Upon further reflection, it becomes evident that the heat transfer coefficient of the transient simulation encompasses both external convection and conduction within the capsule, which is undergoing solidification. In contrast, the coefficient obtained in the initial simulations only considered external convection. To reconcile the two results, it is essential to estimate and model the thermal resistance due to conduction. By combining this value with the coefficient obtained from the initial simulations, it should be possible to obtain a result that is in close proximity to those depicted in Figure 30.

The expression of the heat transfer coefficient, which takes into account both convection and conduction, for a PCM capsule of any geometry can be expressed as follows:

$$\frac{1}{h_{\text{eff}}} = \frac{1}{h} + \frac{I D_p}{k_{\text{capsule}}} \quad (4.4)$$

where h is the convection heat transfer coefficient around the capsule and:

$$I = \frac{1}{A_v D_p + 4} \quad A_v = \frac{A_{\text{capsule}}}{V_{\text{capsule}}}$$

This formula is frequently cited in the literature and is commonly employed to identify the effective heat transfer coefficient of a capsule to use it in the lumped capacitance method, which allows for the numerical modelling of packed beds ([13], [14], [39]–[41]).

The capsule effective diameter is noted :

$$D_p = \frac{6V_{\text{capsule}}}{A_{\text{sp}}} = \frac{6V_{\text{capsule}}}{\psi A_{\text{capsule}}}$$

where A_{sp} is the surface area of the equivalent-volume sphere. Consequently, the shape factor, designated by the symbol ψ , can be written as follows:

$$\psi = \frac{\text{surface of sphere of equal volume to the particle}}{\text{surface area of the particle}} = \frac{A_{\text{sp}}}{A_{\text{capsule}}} = \pi^{\frac{1}{3}} \frac{(6V_{\text{capsule}})^{\frac{2}{3}}}{A_{\text{capsule}}}$$

In the present study, the conductivity of the capsule can be quantified in relation to the volume fractions of PCM and polyethylene :

$$\frac{1}{k_{\text{capsule}}} = \frac{V_{\text{shell}}}{V_{\text{capsule}}} \frac{1}{k_{\text{shell}}} + \frac{V_{\text{PCM}}}{V_{\text{capsule}}} \frac{1}{k_{\text{PCM}}}$$

Giving :

$h_{\text{eff}} = 53.9 \text{ W}/(\text{m}^2 \text{ K}) \tag{4.5}$
--

This value is clearly within the range shown in Figure 30. Consequently, this methodology may be regarded as a reliable approximation for the determination of the heat transfer characteristics of a capsule undergoing phase change, based on its external convection coefficient.

Confronting simulations and experiments

5.1 Comparison of transient simulation and experimental results

The solidification simulation was conducted using the identical boundary conditions as the D6 and D7 discharges, thus enabling a comparison between the experimental and numerical curves. The outcome is presented in Figure 31.

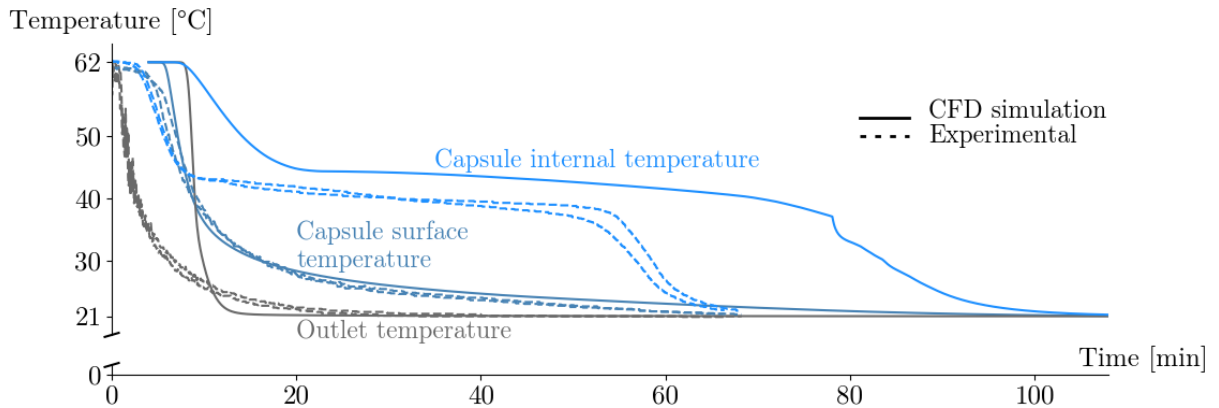


Figure 31: Comparison of the temperature curves of the experimental D6-D7 discharges and the transient solidification simulation carried out under the same boundary conditions ($\bar{T}_{\text{inlet}} = 21^\circ\text{C}$ and $T_{\text{start}} = 62^\circ\text{C}$). As the flow before the capsule was not simulated, the simulation curves were shifted of 9.5 min by considering a plug flow. The time of the disturbance at the inlet is therefore precisely replicated in both situations.

The figure above demonstrates again that the experimental tests cannot be considered to be in plug flow, as the temperature curves of the outlet would have reacted more or less at the same time as the simulation. Nevertheless, it can be observed that the surface temperature curves merge and follow a similar pattern. But, it is crucial to remember that the experimental value is highly dependent of the location of the measurement.

Furthermore, it can be observed that the phase change exhibits a comparable slope in both cases, although the duration is longer in the simulation. This phenomenon can be attributed to a number of factors. Firstly, it has been explained that there is a non-negligible sand mass

fraction in the capsule. In the simulation, this was not accounted for, and paraffin was defined throughout the volume. Consequently, there is a greater volume that requires phase change and thus more energy to extract in the simulation. Secondly, the natural convection of the liquid inside the capsule was not considered in the CFD. Although it has been demonstrated that this does not significantly influence the process, it is nevertheless a factor that contributes to a shorter experimental phase change. Furthermore, it is important to consider that the sensor may act as a thermal bridge, artificially accelerating the solidification process at the point being measured. Additionally, the temperature sensor was intended to be positioned 12 mm inside, but drilling 1 mm wide into the material proved challenging, potentially leading to discrepancies between the simulated and actual points of measurement. Finally, the values presented in the datasheet were found to be of limited accuracy, which raises the possibility that the characteristics of the paraffin encoded in Fluent® may not fully reflect the actual situation.

5.2 Upscaling proposal

By applying the methodology validated by the solidification simulation (Section 3.4.4), an order-of-magnitude value for the heat transfer coefficient of the phase changing capsule can be found for each of the external convection coefficients identified by simulations. The results are presented in Figure 32 and are compared with the values that have been found experimentally with a 380 L storage tank filled with HeatSel®s [14].

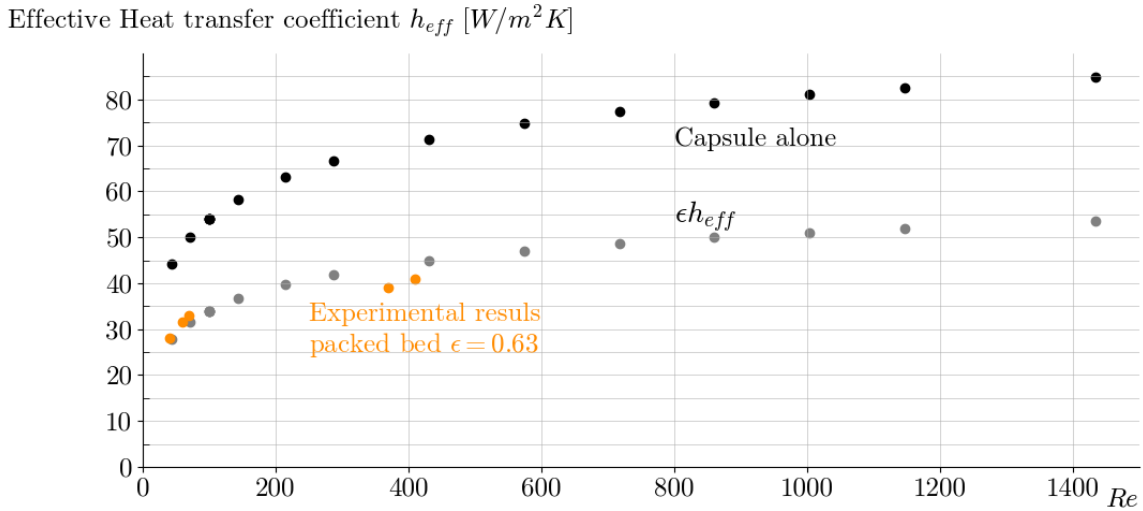


Figure 32: Evolution of the effective heat transfer coefficient of HeatSel® in relation to the Reynolds number. The points representing the capsule alone were obtained by combining the external convection coefficients obtained in the first part of the simulations with the effective conduction method. The points representing the packed bed come from a reference which had experimental values for a HeatSel®'s packed bed with a porosity of 0.63. It can be observed that if the values of a single capsule are multiplied by the porosity factor, the resulting predictions are in close agreement with the experimental observations reported in the literature.

Figure 32 indicates that it is possible to obtain a value for the heat transfer coefficient of a tank filled with capsules exclusively on the basis of the external convection coefficient of a single capsule and the porosity of the bed modelling the tank.

$$h_{\text{bed}} = \epsilon \times h_{\text{eff, capsule}} \quad \text{with} \quad \frac{1}{h_{\text{eff, capsule}}} = \frac{1}{h} + \frac{I D_p}{k_{\text{capsule}}} \quad (5.1)$$

where h represents the convection coefficient of the capsule when considered at a constant temperature, and I , D_p depends only on the dimensions of the capsule.

To confirm this postulate, experimental results must first confirm the effective heat transfer values of a single capsule. Then the effective heat transfer coefficient of several packed beds with different porosities must be determined in order to compare these values with the present upscaling method.

Conclusion

In the context of necessity to consider storage technologies to increase the renewable fraction in the energy mix, it was decided to evaluate the performance of a macro-capsule of phase change material with a complex geometry intended to be arranged in a tank to form a residential latent thermal storage system. From there, several objectives were defined. Firstly, experimentally determining the thermal behaviour of the HeatSel[®] when placed in conditions similar to those experienced in a tank. Secondly, finding the heat transfer coefficient of the capsule using CFD simulations. Thirdly, comparing experimental measurements with a simulation to identify commonalities and limitations. Finally, proposing a method for scaling up the capsule properties to a complete storage system.

Initially, by modelling a tank filled with capsules as a packed bed with constant porosity throughout the volume, it was demonstrated that the typical flow velocity around a capsule is 1.88 mm/s.

The capsule was then placed in the centre of a cylindrical device surrounded by two opposite conical reductions for experimental testing. The HeatSel[®] was subjected to a constant flow rate corresponding to the value given above and to different temperatures. This demonstrated that the phase change duration is not dependent on the initial temperature of the capsule but only on the temperature that is set to the flow passing around it. The greater the difference between the set temperature and the phase change temperature, the shorter the phase change and the less the shape of the curve is close to an ideal phase change plateau. It was also concluded that the phase change duration of the HeatSel[®] was approximately 5× shorter than that of an equivalent volume conventional geometry, although sensor inaccuracy suggests that this value is probably slightly overestimated. Furthermore, the experimental setup did not permit the determination of an equivalent heat transfer coefficient, as the capsule behaviour could not be isolated from power flows. In this context, proposals for a change of configuration were put forth.

Subsequently, stationary simulations using Fluent[®] software were conducted to characterise the convection around the capsule. An empirical *Re-Nu* law was derived, and the coefficients obtained are found to be in close agreement with those of the flat plate. Concurrently, a transient simulation of a complete solidification process, which did not consider natural convection, demonstrated that the front was not evolving uniformly within the capsule. Furthermore, it has been highlighted that the grooves present in the geometry of the HeatSel[®] limit the zones where phase change duration is prolonged.

The results of the solidification simulation could not be validated by the two experiments that had identical boundary conditions. This is mainly due to the inability to achieve a plug flow in the experimental setup, despite the conditions being simulated. In addition, a disassembly of the capsule would be necessary to determine the exact properties of the materials to be considered in the numerical model.

Finally, it is demonstrated that by combining the convection coefficient around the particle with an effective conduction model, it is possible to characterise the overall heat transfer of the phase changing HeatSel[®]. Moreover, it has been empirically demonstrated that the effective heat transfer coefficient of a packed bed can be calculated exclusively on the basis of its porosity and the convection around a single capsule.

In conclusion, the experimental tests determined the charging and discharging dynamics of the HeatSel[®], while the simulations identified the heat transfer coefficient and allowed to propose a packed bed upscaling method. However, the experiment failed to confirm the theoretical results. In order to validate these statements, it is therefore necessary to adapt the experimental set-up and to characterise the effective heat transfer coefficient of several packed beds with different porosities.

Bibliography

- [1] International Energy Agency (IEA), *Net zero roadmap: A global pathway to keep the 1.5 °c goal in reach*, License: CC BY 4.0, Paris, 2023. [Online]. Available: <https://www.iea.org/reports/net-zero-roadmap-a-global-pathway-to-keep-the-15-0c-goal-in-reach>.
- [2] European Commission, “Energy storage - the role of electricity,” European Commission, Brussels, Commission Staff Working Document SWD(2017) 61 final, Feb. 2017, Accessed: 2024-06-01. [Online]. Available: https://ec.europa.eu/energy/sites/ener/files/documents/swd2017_61_f1_other_staff_working_paper_en_v3_p1_878984.pdf.
- [3] International Energy Agency (IEA), *Renewables 2023*, <https://www.iea.org/reports/renewables-2023>, License: CC BY 4.0, Paris, 2024.
- [4] A. Raj, M. Srinivas, and S. Jayaraj, “Cfd modeling of macro-encapsulated latent heat storage system used for solar heating applications,” *International Journal of Thermal Sciences*, vol. 139, pp. 88–104, 2019, ISSN: 1290-0729. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1290072918306318>.
- [5] T. Xu, “Integrating latent heat storage into residential heating systems: A study from material and component characterization to system analysis,” Ph.D. dissertation, KTH, School of Industrial Engineering, Management (ITM), Energy Technology, Applied Thermodynamics, and Refrigeration, 2021, pp. 51–79.
- [6] L. Cabeza, A. Castell, C. Barreneche, A. de Gracia, and A. Fernández, “Materials used as pcm in thermal energy storage in buildings: A review,” *Renewable and Sustainable Energy Reviews*, vol. 15, no. 3, pp. 1675–1695, 2011, ISSN: 1364-0321. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1364032110003874>.
- [7] International Renewable Energy Agency, “Thermal energy storage,” International Renewable Energy Agency, Tech Brief E17, 2013. [Online]. Available: <https://www.irena.org/-/media/Files/IRENA/Agency/Publication/2013/IRENA-ETSAP-Tech-Brief-E17-Thermal-Energy-Storage.pdf>.
- [8] H. Mehling and L. F. Cabeza, “Heat and cold storage with pcm,” *Heat and mass transfer*, pp. 11–55, 2008.
- [9] A. de Gracia and L. F. Cabeza, “Phase change materials and thermal energy storage for buildings,” *Energy and Buildings*, vol. 103, pp. 414–419, 2015, ISSN: 0378-7788. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0378778815004338>.

- [10] B. Zhao and R. Wang, "Perspectives for short-term thermal energy storage using salt hydrates for building heating," *Energy*, vol. 189, p. 116 139, 2019, ISSN: 0360-5442. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0360544219318341>.
- [11] M. Grabo, E. Acar, and E. Y. Kenig, "Modeling and improvement of a packed bed latent heat storage filled with non-spherical encapsulated pcm-elements," *Renewable Energy*, vol. 173, pp. 1087–1097, 2021, ISSN: 0960-1481. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0960148121005346>.
- [12] F. Agyenim, N. Hewitt, P. Eames, and M. Smyth, "A review of materials, heat transfer and phase change problem formulation for latent heat thermal energy storage systems (lhtess)," *Renewable and Sustainable Energy Reviews*, vol. 14, no. 2, pp. 615–628, 2010, ISSN: 1364-0321. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1364032109002469>.
- [13] T. Xu, E. N. Humire, S. Trevisan, M. Ignatowicz, S. Sawalha, and J. N. Chiu, "Experimental and numerical investigation of a latent heat thermal energy storage unit with ellipsoidal macro-encapsulation," *Energy*, vol. 238, p. 121 828, 2022, ISSN: 0360-5442. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0360544221020764>.
- [14] T. Xu, V. G. Gkoutzamanis, H. Dong, *et al.*, "Performance evaluation of three latent heat storage designs for cogeneration applications," *Solar Energy*, vol. 225, pp. 444–462, 2021, ISSN: 0038-092X. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0038092X21006137>.
- [15] L. Navarro, C. Barreneche, A. Castell, D. A. Redpath, P. W. Griffiths, and L. F. Cabeza, "High density polyethylene spheres with pcm for domestic hot water applications: Water tank and laboratory scale study," *Journal of Energy Storage*, vol. 13, pp. 262–267, 2017, ISSN: 2352-152X. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S2352152X1730083X>.
- [16] D.-H. Yu and Z.-Z. He, "Shape-remodeled macrocapsule of phase change materials for thermal energy storage and thermal management," *Applied Energy*, vol. 247, pp. 503–516, 2019, ISSN: 0306-2619. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0306261919307342>.
- [17] J. Park, D. H. Shin, S. J. Lee, Y. Shin, and S. W. Karng, "Effective latent heat thermal energy storage system using thin flexible pouches," *Sustainable Cities and Society*, vol. 45, pp. 143–150, 2019, ISSN: 2210-6707. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S2210670718314926>.
- [18] J. R. Hirschey, N. Kumar, T. Turnaoglu, K. R. Gluesenkamp, and S. Graham, "Review of low-cost organic and inorganic phase change materials with phase change temperature between 0°C and 65°C," in *International High Performance Buildings Conference*, Purdue University, Herrick Laboratories, West Lafayette, Indiana, USA, 2021.
- [19] M.-H. Hu, T. Xu, and J. N. Chiu, "Experimental analysis of submerged coil and encapsulated slab latent heat storage," *Applied Thermal Engineering*, vol. 209, p. 118 259, 2022, ISSN: 1359-4311. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1359431122002198>.
- [20] A. F. Regin, S. Solanki, and J. Saini, "Heat transfer characteristics of thermal energy storage system using pcm capsules: A review," *Renewable and Sustainable Energy Reviews*, vol. 12, no. 9, pp. 2438–2458, 2008, ISSN: 1364-0321. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1364032107001001>.

-
- [21] N. Amin, F. Bruno, and M. Belusko, "Effective thermal conductivity for melting in pcm encapsulated in a sphere," *Applied Energy*, vol. 122, pp. 280–287, 2014, ISSN: 0306-2619. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0306261914001020>.
 - [22] X. Wang, U. Djakovic, H. Bao, and J. F. Torres, "Experimental evaluation of heat transfer performance under natural and forced convection around a phase change material encapsulated in various shapes," *Sustainable Energy Technologies and Assessments*, vol. 44, p. 101 025, 2021, ISSN: 2213-1388. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S2213138821000357>.
 - [23] A. Surya, R. Prakash, and N. Nallusamy, "Heat transfer enhancement and performance study on latent heat thermal energy storage system using different configurations of spherical pcm balls," *Journal of Energy Storage*, vol. 72, p. 108 643, 2023, ISSN: 2352-152X. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S2352152X23020406>.
 - [24] Z. Liu, Z. (Yu, T. Yang, *et al.*, "A review on macro-encapsulated phase change material for building envelope applications," *Building and Environment*, vol. 144, pp. 281–294, 2018, ISSN: 0360-1323. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0360132318305031>.
 - [25] R. Kasibhatla and D. Brüggemann, "Coupled conjugate heat transfer model for melting of pcm in cylindrical capsules," *Applied Thermal Engineering*, vol. 184, p. 116 301, 2021, ISSN: 1359-4311. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1359431120337807>.
 - [26] D. H. Shin, J. Park, S. H. Choi, H. S. Ko, S. W. Karng, and Y. Shin, "A new type of heat storage system using the motion of phase change materials in an elliptical-shaped capsule," *Energy Conversion and Management*, vol. 182, pp. 508–519, 2019, ISSN: 0196-8904. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S019689041930007X>.
 - [27] W. Li, S.-G. Li, S. Guan, Y. Wang, X. Zhang, and X. Liu, "Numerical study on melt fraction during melting of phase change material inside a sphere," *International Journal of Hydrogen Energy*, vol. 42, no. 29, pp. 18 232–18 239, 2017, Special Issue on The 5th International Conference on Energy Engineering and Environmental Engineering (ICEEEE2017), 15-16 April 2017, Xiamen, China, ISSN: 0360-3199. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0360319917315446>.
 - [28] E. Assis, L. Katsman, G. Ziskind, and R. Letan, "Numerical and experimental study of melting in a spherical shell," *International Journal of Heat and Mass Transfer*, vol. 50, no. 9, pp. 1790–1804, 2007, ISSN: 0017-9310. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0017931006005503>.
 - [29] J. Bédécarrats, J. Castaing-Lasvignottes, F. Strub, and J. Dumas, "Study of a phase change energy storage using spherical capsules. part ii: Numerical modelling," *Energy Conversion and Management*, vol. 50, no. 10, pp. 2537–2546, 2009, ISSN: 0196-8904. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0196890409002209>.
 - [30] B. Peremans, *Optimal parameter and fin design for pcm-based thermal energy storage*, eng, 2020.
 - [31] K. Ismail and J. Henríquez, "Solidification of pcm inside a spherical capsule," *Energy Conversion and Management*, vol. 41, no. 2, pp. 173–187, 2000, ISSN: 0196-8904. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0196890499001016>.

- [32] X. Cheng and X. Zhai, “Thermal performance analysis of a novel pcm capsule in red blood cell shape,” *Applied Thermal Engineering*, vol. 120, pp. 130–137, 2017, ISSN: 1359-4311. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S1359431117303885>.
- [33] M. Iten, S. Liu, and A. Shukla, “Experimental validation of an air-pcm storage unit comparing the effective heat capacity and enthalpy methods through cfd simulations,” *Energy*, vol. 155, pp. 495–503, 2018, ISSN: 0360-5442. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S036054421830745X>.
- [34] H. Sattari, A. Mohebbi, M. Afsahi, and A. Azimi Yancheshme, “Cfd simulation of melting process of phase change materials (pcms) in a spherical capsule,” *International Journal of Refrigeration*, vol. 73, pp. 209–218, 2017, ISSN: 0140-7007. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0140700716302870>.
- [35] T. L. Bergman, *Fundamentals of heat and mass transfer*. John Wiley & Sons, 2011.
- [36] H. Baehr and K. Stephan, “Heat and mass transfer, springer,” *Berlin, Heidelberg, New York*, 2006.
- [37] C. Chan and F. Tan, “Solidification inside a sphere—an experimental study,” *International Communications in Heat and Mass Transfer*, vol. 33, no. 3, pp. 335–341, 2006, ISSN: 0735-1933. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0735193305002046>.
- [38] E. Halawa and W. Saman, “Thermal performance analysis of a phase change thermal storage unit for space heating,” *Renewable Energy*, vol. 36, no. 1, pp. 259–264, 2011, ISSN: 0960-1481. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0960148110002855>.
- [39] X.-L. Ouyang, R.-N. Xu, and P.-X. Jiang, “Three-equation local thermal non-equilibrium model for transient heat transfer in porous media: The internal thermal conduction effect in the solid phase,” *International Journal of Heat and Mass Transfer*, vol. 115, pp. 1113–1124, 2017, ISSN: 0017-9310. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0017931017320793>.
- [40] B. Xu, P.-W. Li, and C. L. Chan, “Extending the validity of lumped capacitance method for large biot number in thermal storage application,” *Solar Energy*, vol. 86, no. 6, pp. 1709–1724, 2012, ISSN: 0038-092X. [Online]. Available: <https://www.sciencedirect.com/science/article/pii/S0038092X12001223>.
- [41] L. Li and W. Ma, “Experimental study on the effective particle diameter of a packed bed with non-spherical particles,” *Transp Porous Med*, vol. 89, pp. 35–48, 2011.

A.1 Additional pictures related to the experimental process

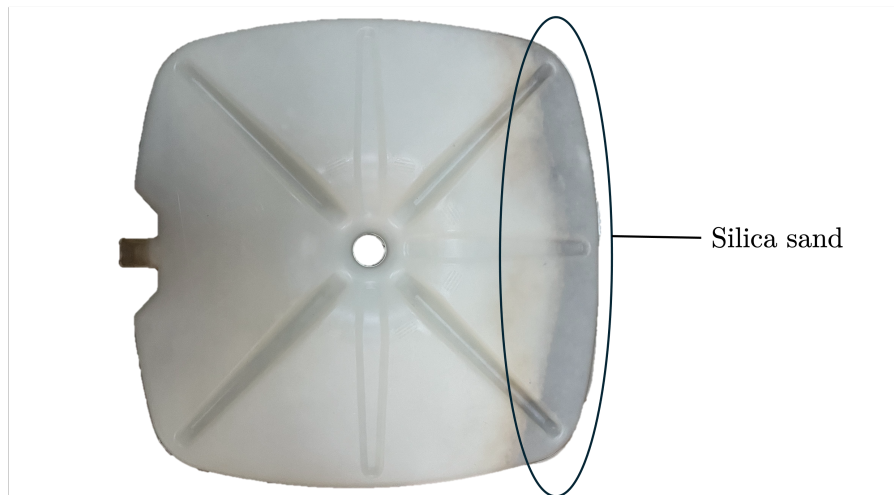


Figure 33: Photo of an ATP43 HeatSel®. The sand inside is clearly visible.

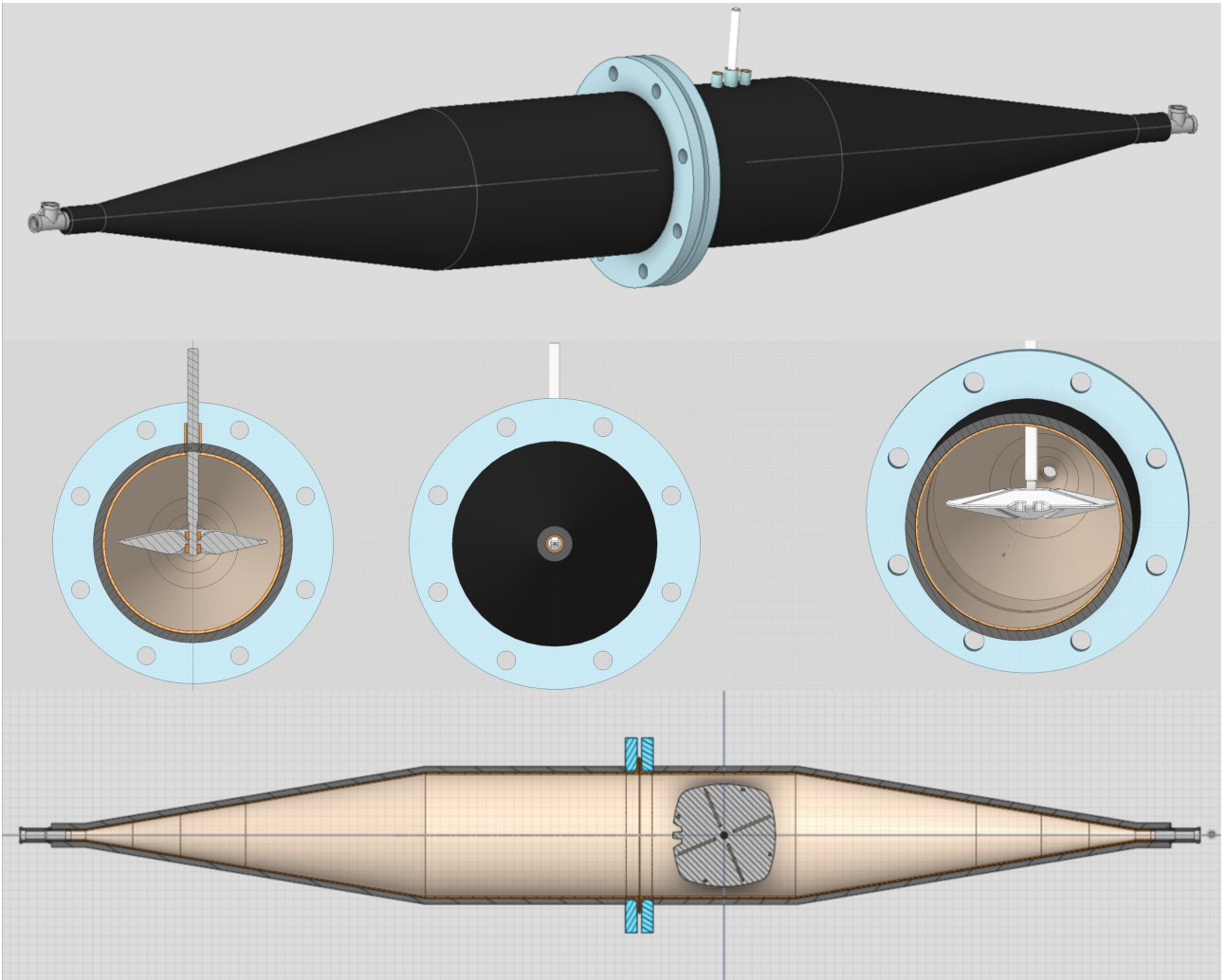


Figure 34: Additional views of the 3D model on which the experimental set-up plans were based. From top to bottom and from left to right: perspective view of the entire device, cross-section from the centre of the capsule towards the inlet, cross-section at the end of the inlet sleeve towards the inside of the device, perspective view of the capsule dimensions in relation to the cylindrical section with the outlet at the back, and cross-section from the top.

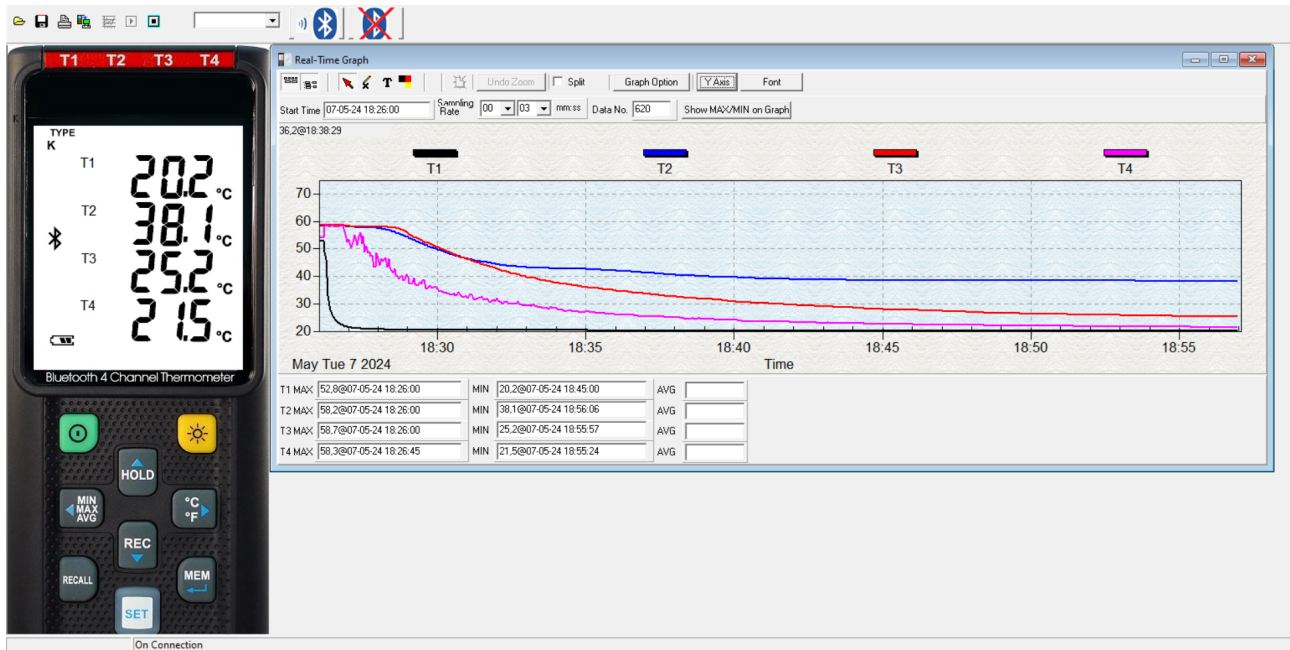


Figure 35: On the left is a photo of the data logger used to record temperatures during the experiments. Next to it is a view of the real-time data acquisition system, which allows the results to be viewed, exported to a text file and the recording frequency to be set.

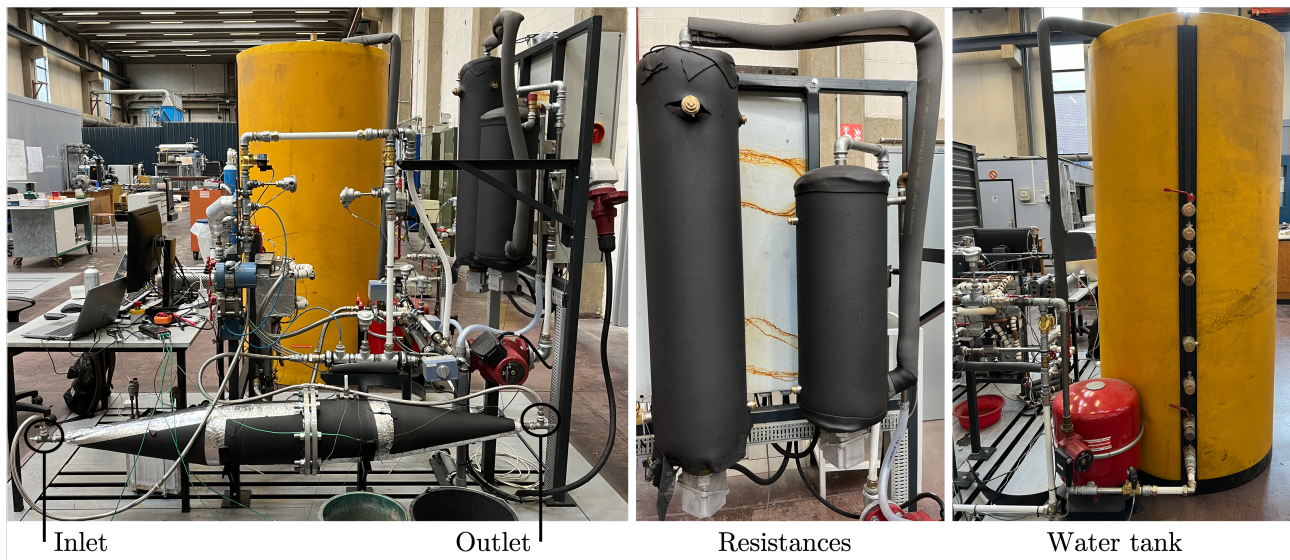


Figure 36: Photo of the test bench used. On the left are the computers used to control the electronic valves and collect the data. On the right are more detailed photos of the resistances and the tank, which are also in the background of the main photo. The inlet and outlet connections are shown.

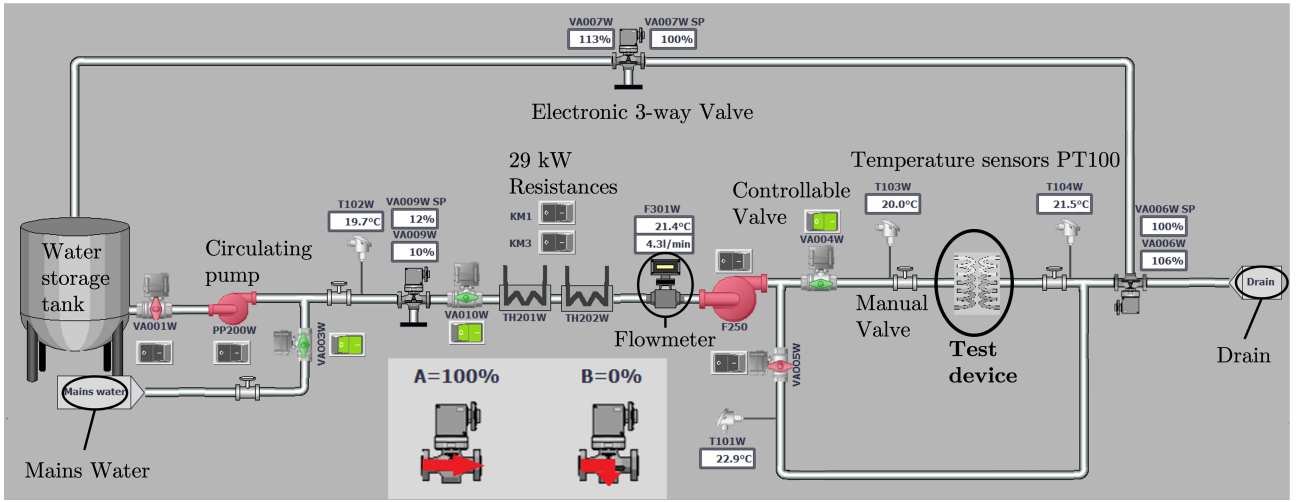


Figure 37: Hydraulic circuit scheme in a discharge configuration. The bench is used in an open circuit. The circulation is ensured by the pressure of the mains water, so no pump is needed in this case. The capsule, initially at high temperature, is subjected to a constant ambient temperature with a flow rate of 4.3 L/min.

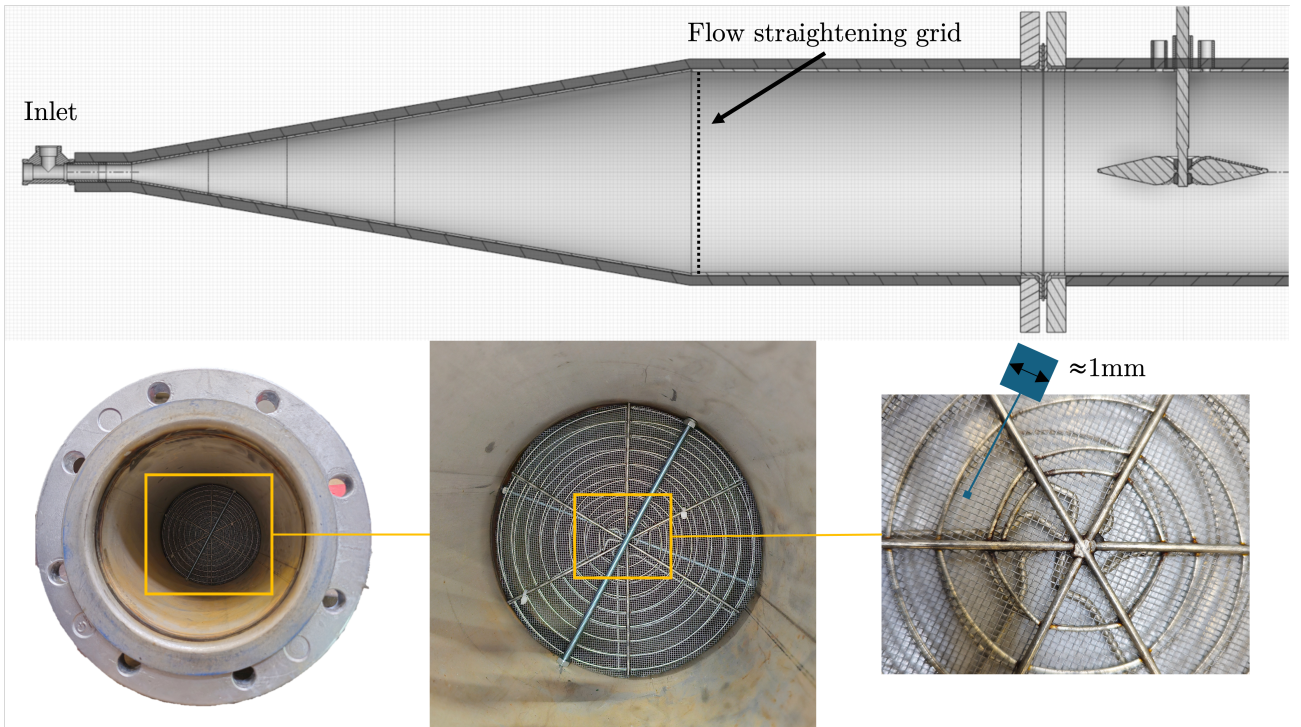


Figure 38: Visualisation of the placement of the grid in the experimental set-up. The grid consists of a mosquito net with 1 mm square mesh, resistant to a maximum temperature of 120 °C and fixed by means of 2 threaded rods.

A.2 Test device heat capacity calculation

Table 15 illustrates the components used for the construction of the test device. A total of three distinct materials were employed, and their respective heat capacities are presented in Table 16. The final cone is the result of the welding of 4 cones.

Component	Mass [g]
Flange	2567.26
Collar	797.69
Seal	93.33
Sleeve	39.67
Tube	9557.52
Cone DN200 to 100	2572.33
Cone DN100 to 65	530.51
Cone DN65 to 40	249.34
Cone DN40 to 15	131.89
Clamp Ring	16.47
Bolt	347.99

Table 15: Mass of all the components used to build the experimental device.

$$\begin{aligned}
 m_{\text{aluminium}} &= 2 \times m_{\text{flange}} \\
 m_{\text{stainless steel}} &= (2 \times m_{\text{collar}}) + (2 \times m_{\text{tube}}) + (3 \times m_{\text{sleeve}}) + (8 \times m_{\text{bolt}}) + (16 \times m_{\text{clamp ring}}) \\
 &\quad + 2 \times (m_{\text{cone DN200to100}} + m_{\text{cone DN100to65}} + m_{\text{cone DN65to40}} + m_{\text{cone DN40to15}}) \\
 m_{\text{graphite}} &= m_{\text{seal}}
 \end{aligned}$$

Material	c [J/kgK]
Stainless steel ¹	502
Aluminium ²	921
Graphite ³	707

Table 16: Heat capacities of stainless steel, aluminium and graphite.

$$c_{\text{device}} = (c_{\text{stainless steel}} \times m_{\text{stainless steel}}) + (c_{\text{aluminium}} \times m_{\text{aluminium}}) + (c_{\text{graphite}} \times m_{\text{graphite}})$$

Giving as a final result :

$$\begin{aligned}
 m_{\text{device}} &= 26.52 \text{ kg} \\
 c_{\text{device}} &= 15.48 \text{ kJ/K} \\
 c_{\text{flanges, collars, bolts, clamp rings}} &= 7.06 \text{ kJ/K} \\
 c_{\text{tube, cones}} &= 8.30 \text{ kJ/K}
 \end{aligned}$$

A.2.1 Flanges temperature evolution during a charge test

Figure 39 illustrates the change in flange temperature over time during a 50 °C charge. In consideration of the associated measurement uncertainty, the greatest potential increase is considered to quantify the energy that may be lost to the device during a test.

¹https://www.substech.com/dokuwiki/doku.php?id=stainless_steel_aisi_304

²https://www.engineersedge.com/materials/specific_heat_capacity_of_metals_13259.htm

³<https://gchem.cm.utexas.edu/data/section2.php?target=heat-capacities.php>

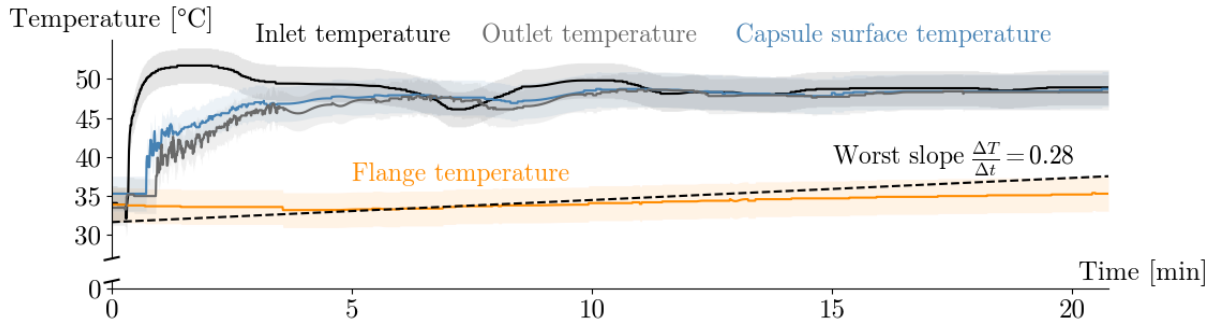


Figure 39: Change in flange temperature during 20 min of charge at 50 °C.

By combining this result with the heat capacity of the entire device, it is possible to estimate the energy loss as a function of charging time.

$$E_{\text{device}} = c_{\text{device}} \times \frac{\Delta T}{\Delta t} \times \frac{1}{60} \times t_{ch} = c_{\text{device}} \times \frac{0.28}{60} \times t_{ch}$$

where the factor 60 takes account of the minute-second conversion.

A.3 Power losses extent determination

A schematic representation of the power losses in the experimental device is presented in Figure 40, while Table 17 provides an overview of the thermal properties and dimensions of the components.

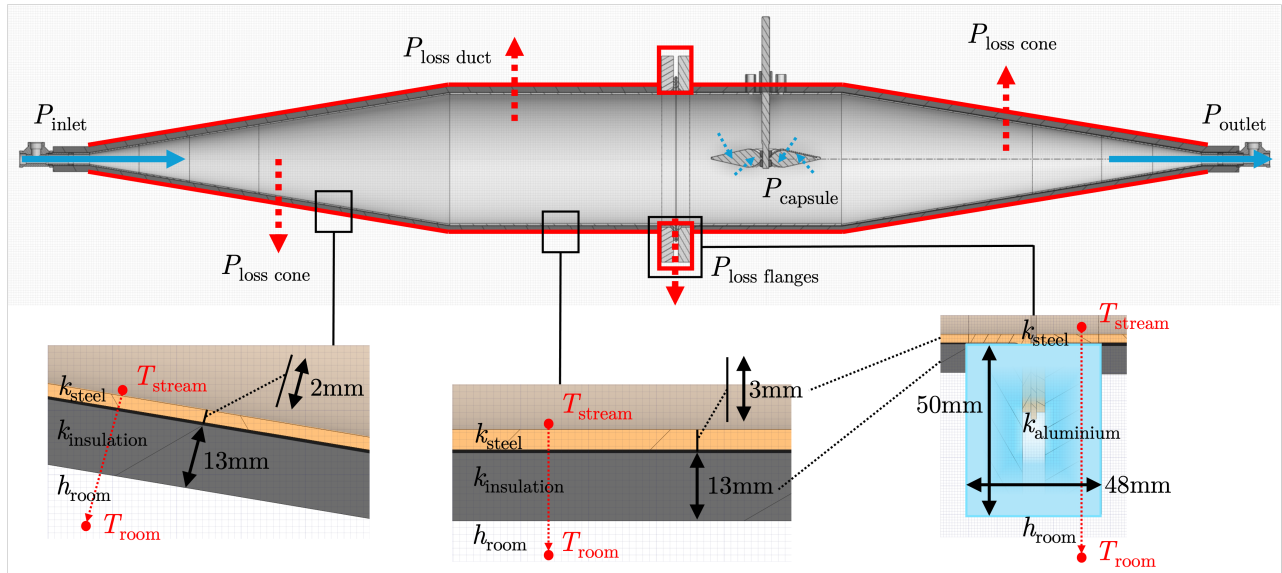


Figure 40: Schematic representation of how heat losses can be modelled in the experimental test device.

¹<https://local.armacell.com/fileadmin/cms/france/downloads/Catalogue/Produits/armaflex-xg.pdf>

²<https://www.mdpi.com/1996-1944/16/8/2972#:~:text=Aluminum%20has%20a%20thermal%20conductivity,1%2C2%2C3%5D>.

³<https://www.matweb.com/search/datasheet.aspx?MatGUID=abc4415b0f8b490387e3c922237098da&ckck=1>

⁴<https://www.sciencedirect.com/topics/engineering/convection-heat-transfer-coefficient>

Duct		
R	106.55	mm
L_{duct}	600	mm
$A_{\text{outer,duct}}$	0.462	m ²
$e_{\text{steel,duct}}$	3	mm
Cone		
$A_{\text{outer,cone}}$	0.276	m ²
$e_{\text{steel,cone}}$	2	mm
Flanges		
L_{flanges}	48	mm
$A_{\text{outer,flanges}}$	0.048	m ²

$e_{\text{insulation}}$	13	mm
$k_{\text{insulation}}^1$	0.035	W/(m K)
$e_{\text{aluminium}}$	50	mm
$k_{\text{aluminium}}^2$	237	W/(m K)
k_{steel}^3	16.2	W/(m K)
h_{room}^4	10	W/(m ² K)

Table 17: Thermal properties and dimensions of the components of the test device which need to be considered when calculating the total thermal resistance of the device.

The following formulas are employed to express thermal resistance by convection or conduction [35]:

$$R_{\text{conv}} = \frac{1}{Ah} \quad R_{\text{cond}} = \frac{e}{Ak} \quad R_{\text{cond,cylinder}} = \frac{\log(r_o/r_i)}{2\pi Lk} \quad (\text{A.1})$$

where R_{conv} is the thermal resistance by forced convection with a heat transfer coefficient h . R_{cond} represents the conduction through a constant surface of thickness e with a material that has a thermal conductivity k . $R_{\text{cond,cylinder}}$ is the expression of the thermal resistance of conduction through the thickness e of a cylinder of length L and internal radius r_i , such that $r_o = r_i + e$. This equation is obtained from Fourier's law expressed in axisymmetric coordinates with a non-constant exchange area varying according to the radial component r . It should be noted that all the logarithms in the formulas are Neperian logarithms.

For the duct part, which is purely cylindrical, with $L = L_{\text{duct}}$ it becomes :

$$\begin{aligned}
 R_{\text{th,duct}} &= R_{\text{cond,steel}} + R_{\text{cond,insulation}} + R_{\text{conv,room}} \\
 R_{\text{cond,steel}} &= \frac{\log\left(\frac{R+e_{\text{steel,duct}}}{R}\right)}{2\pi Lk_{\text{steel}}} \\
 R_{\text{cond,insulation}} &= \frac{\log\left(\frac{R+e_{\text{steel,duct}}+e_{\text{insulation}}}{R+e_{\text{steel,duct}}}\right)}{2\pi Lk_{\text{insulation}}} \\
 R_{\text{conv,room}} &= \frac{1}{A_{\text{outer,duct}}h_{\text{room}}}
 \end{aligned}$$

In regard to the flanges, it is reasonable to approximate the configuration as illustrated in Figure 40. This entails the omission of the void spaces and bolts. The mathematical formulas employed are thus analogous to the ones used for the duct, yet $L = L_{\text{flanges}}$ and $R_{\text{cond, insulation}}$ is replaced by $R_{\text{cond, aluminium}}$ with the same formula but with $e_{\text{aluminium}}$.

$$R_{\text{th,flanges}} = R_{\text{cond,steel}} + R_{\text{cond,aluminium}} + R_{\text{conv,room}}$$

The formula presented above ($R_{\text{cond}} = \frac{e}{Ak}$) is typically applicable to conduction scenarios where the cross-sectional area remains constant. However, in the case of the cone, this assumption does not hold true. Nevertheless, it is still employed here by considering the external cross-section around the insulator as a reference. This approach provides a relatively quick and somewhat

overestimated approximation of the losses, as the exchange area considered for conduction is greater than it actually is.

$$R_{th,cone} \approx \frac{1}{2 \cdot A_{outer,cone}} \left(\frac{e_{steel,cone}}{k_{steel}} + \frac{e_{insulation}}{k_{insulation}} + \frac{1}{h_{room}} \right)$$

It should be noted that the exchange area is doubled because the test device is comprised of two cones.

Giving finally :

$$R_{th} = \frac{1}{\frac{1}{R_{th,duct}} + \frac{1}{R_{th,flanges}} + \frac{1}{R_{th,cone}}} = 0.283 \text{ K/W}$$

A.4 Experimental temperature curves

A.4.1 Charges

This section presents all the temperature curves for all the charges carried out in the laboratory. The legend associated with each colour is specified on the first graph and then remains identical for all the tests. The graphs are presented in order of temperature at entry, starting with 45 °C and ending with 75 °C. Tests are grouped when they have a similar inlet temperature. Within these groups, the time scale is identical. However, it should be noted that the scale changes outside of the groups. Note that the highlighted area on either side of each curve represents the uncertainty of the measurement.

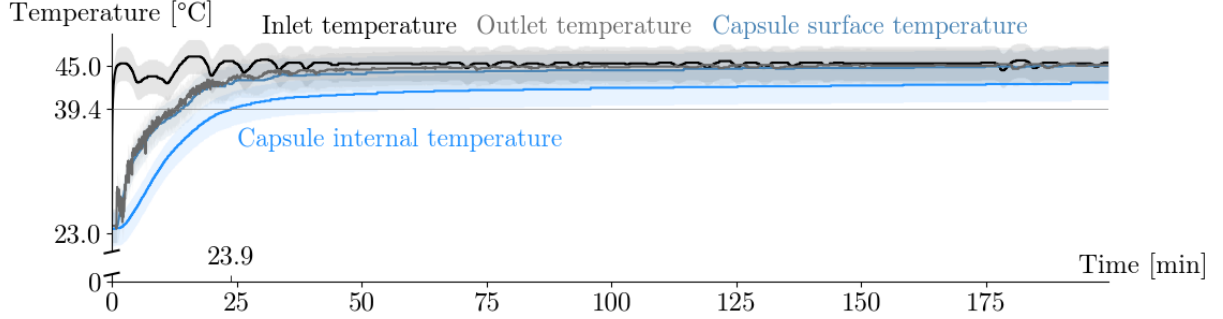


Figure 41: Charge C1 : $\dot{V} = 4.2 \text{ L/min}$, $\bar{T}_{inlet} = 45.2 \text{ °C}$, $t_{charge} > 200 \text{ min}$. An oscillation in the temperature measured at the input can be seen, highlighting non-optimal control of the test bench resistances. The test was stopped after 200 min, yet the shape of the curve suggests that the phase change was not complete.

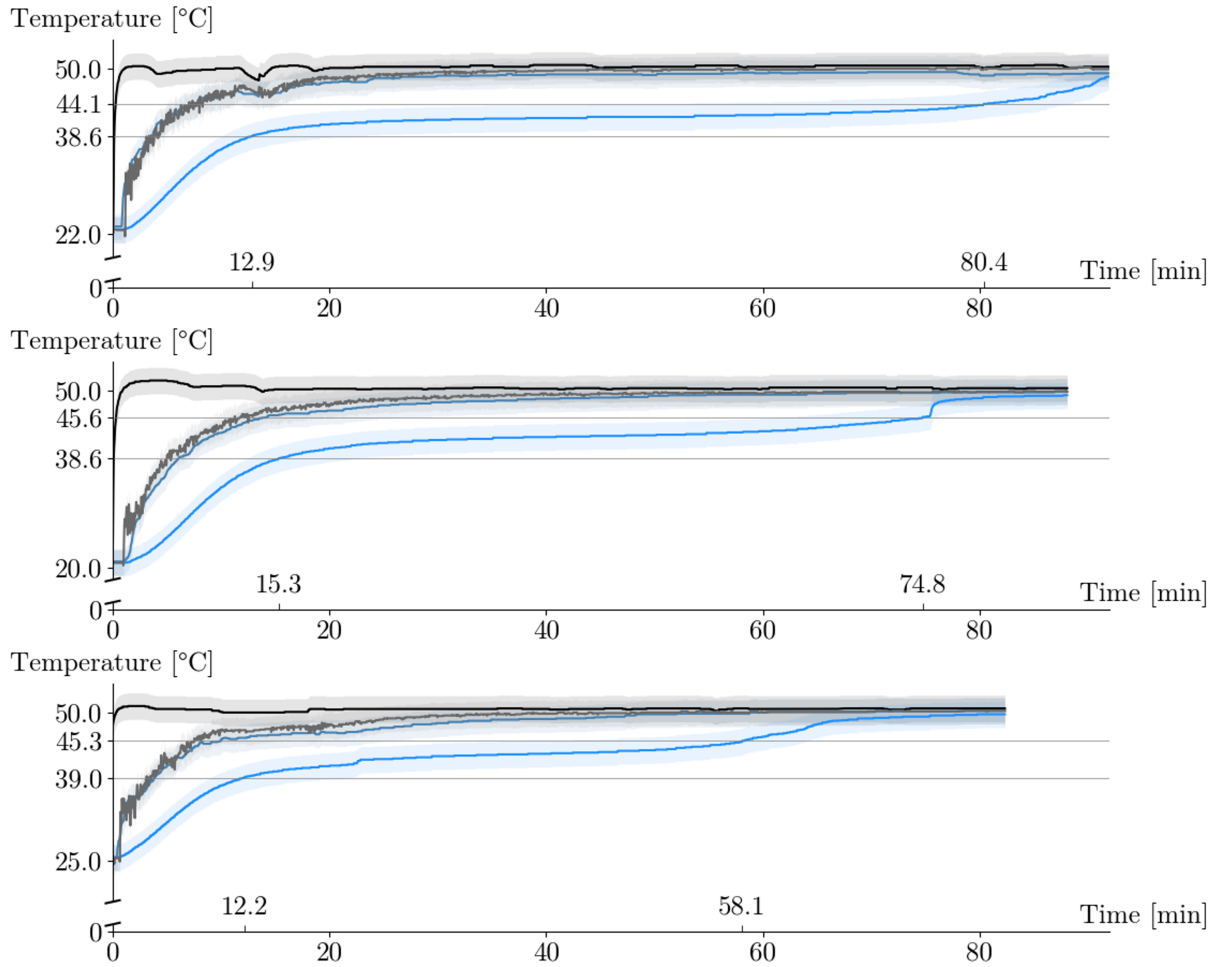


Figure 42: From top to bottom:

Charge C2 : $\dot{V} = 4.0 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 50.3^\circ\text{C}$, $t_{\text{charge}} = 91.9 \text{ min}$.

Charge C3 : $\dot{V} = 4.0 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 50.5^\circ\text{C}$, $t_{\text{charge}} = 88.1 \text{ min}$.

Charge C4 : $\dot{V} = 4.3 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 50.7^\circ\text{C}$, $t_{\text{charge}} = 82.3 \text{ min}$.

The fact that the C4 has a faster completion time for the same inlet temperature could be attributed to the slightly higher flow rate. It should be noted that the end of the phase change is not always as abrupt as on C3.

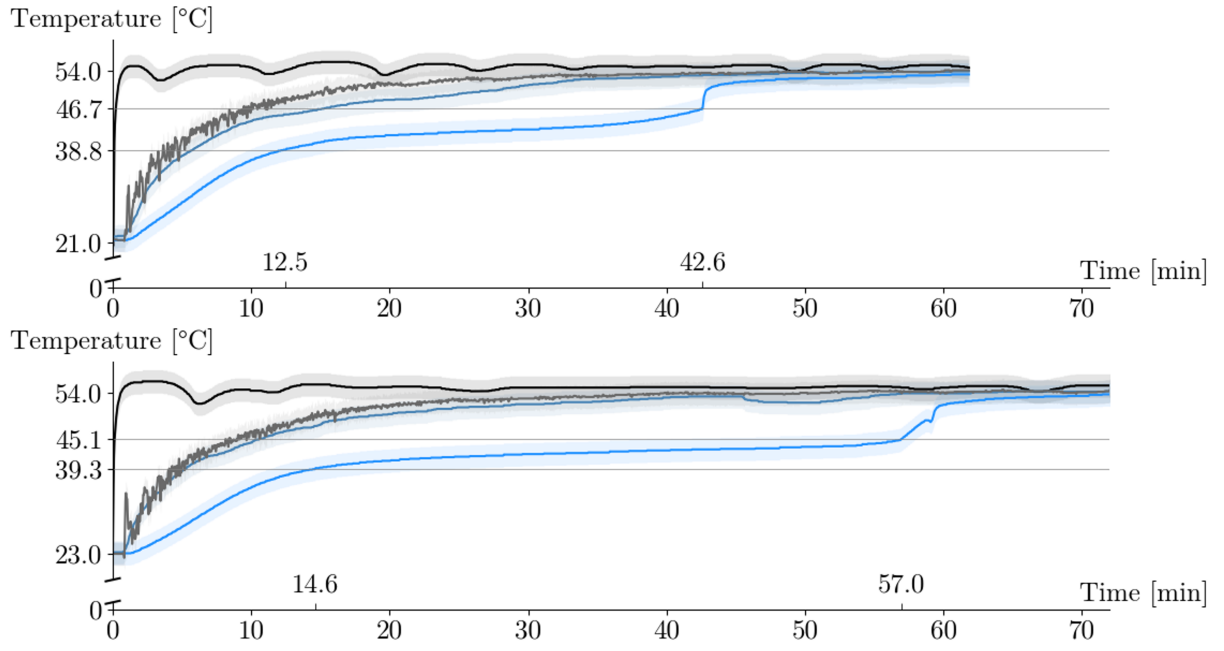


Figure 43: From top to bottom:

 Charge C5 : $\dot{V} = 4.1 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 54.6^\circ\text{C}$, $t_{\text{charge}} = 61.9 \text{ min}$.

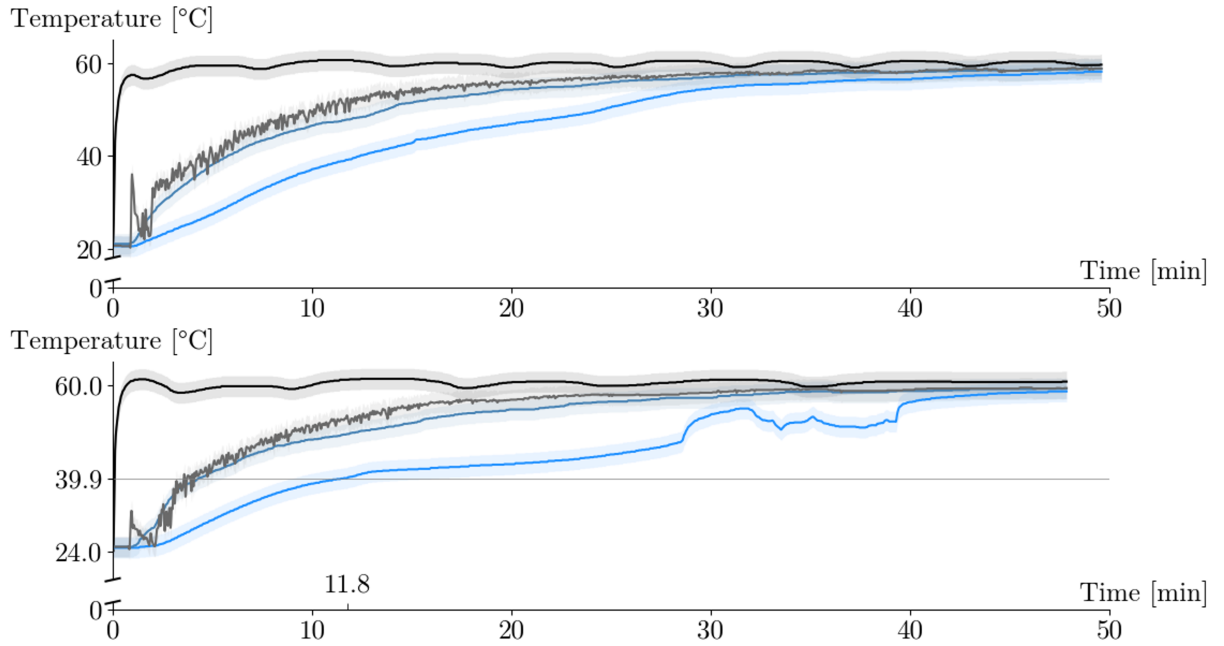
 Charge C6 : $\dot{V} = 3.9 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 55.0^\circ\text{C}$, $t_{\text{charge}} = 71.6 \text{ min}$.


Figure 44: From top to bottom:

 Charge C7 : $\dot{V} = 4.1 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 59.6^\circ\text{C}$, $t_{\text{charge}} = 49.6 \text{ min}$.

 Charge C8 : $\dot{V} = 4.2 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 60.2^\circ\text{C}$, $t_{\text{charge}} = 47.9 \text{ min}$.

As can be observed, for the two charges at 60°C , it is not possible to observe a plateau in either case.

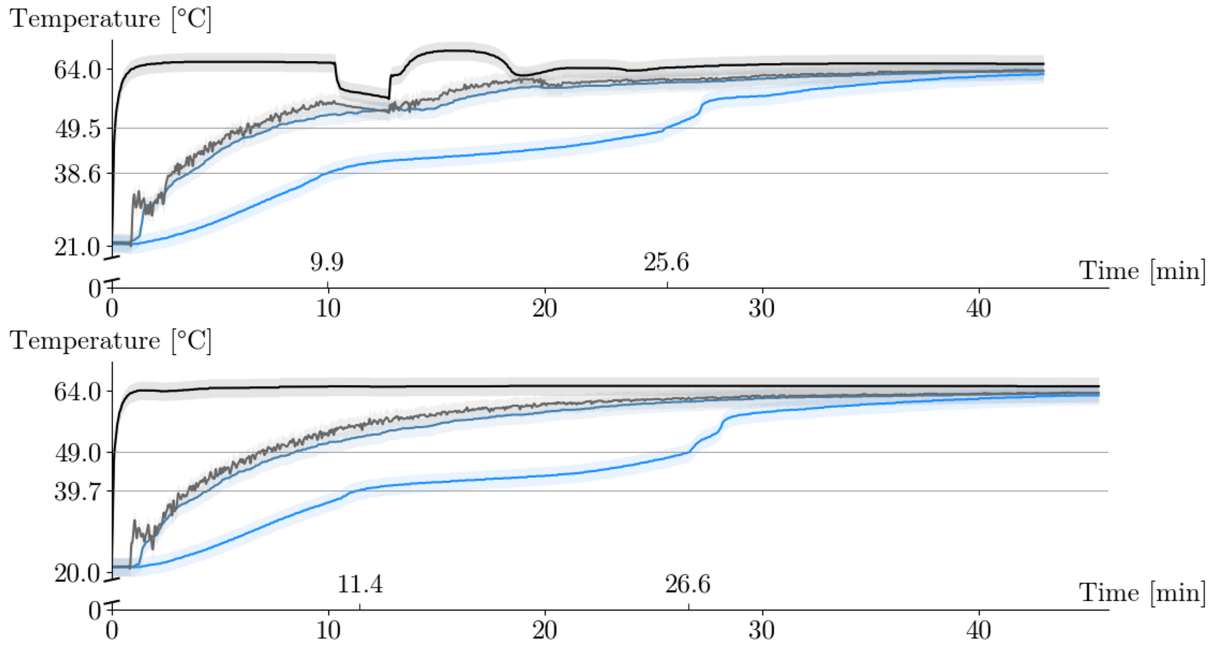


Figure 45: From top to bottom:

Charge C9 : $\dot{V} = 3.8 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 64.5^\circ\text{C}$, $t_{\text{charge}} = 43.0 \text{ min}$.

Charge C10 : $\dot{V} = 3.9 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 64.8^\circ\text{C}$, $t_{\text{charge}} = 45.5 \text{ min}$.

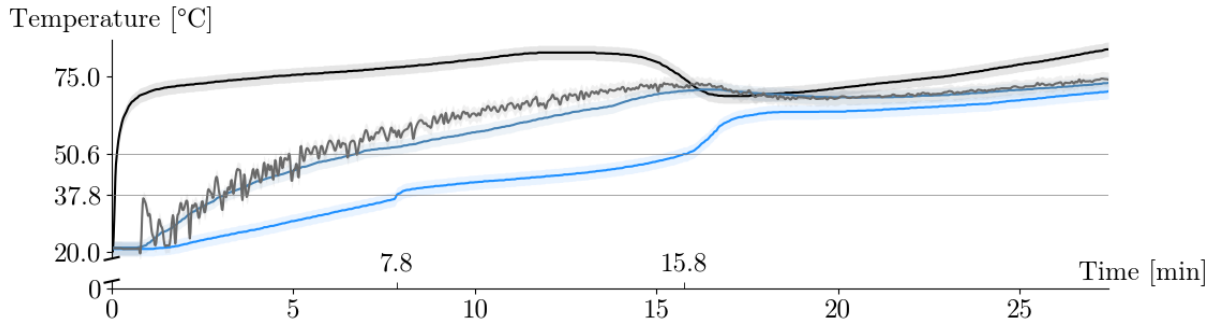


Figure 46: Charge C11 : $\dot{V} = 4.2 \text{ L/min}$, $\bar{T}_{\text{inlet}} = 75.6^\circ\text{C}$, $t_{\text{charge}} = 27.4 \text{ min}$. The controller is found to be unsuitable for the management of resistances when such high temperatures are required.

A.4.2 Discharges

As the discharge D1 followed the charge C1, the PCM was not completely liquid at the start of the experiment, so the graph is not shown here. It should be noted that the time scale is identical for the seven discharges presented, with the lowest starting temperature (49.0°C) placed at the beginning and the highest (72.0°C) at the end.

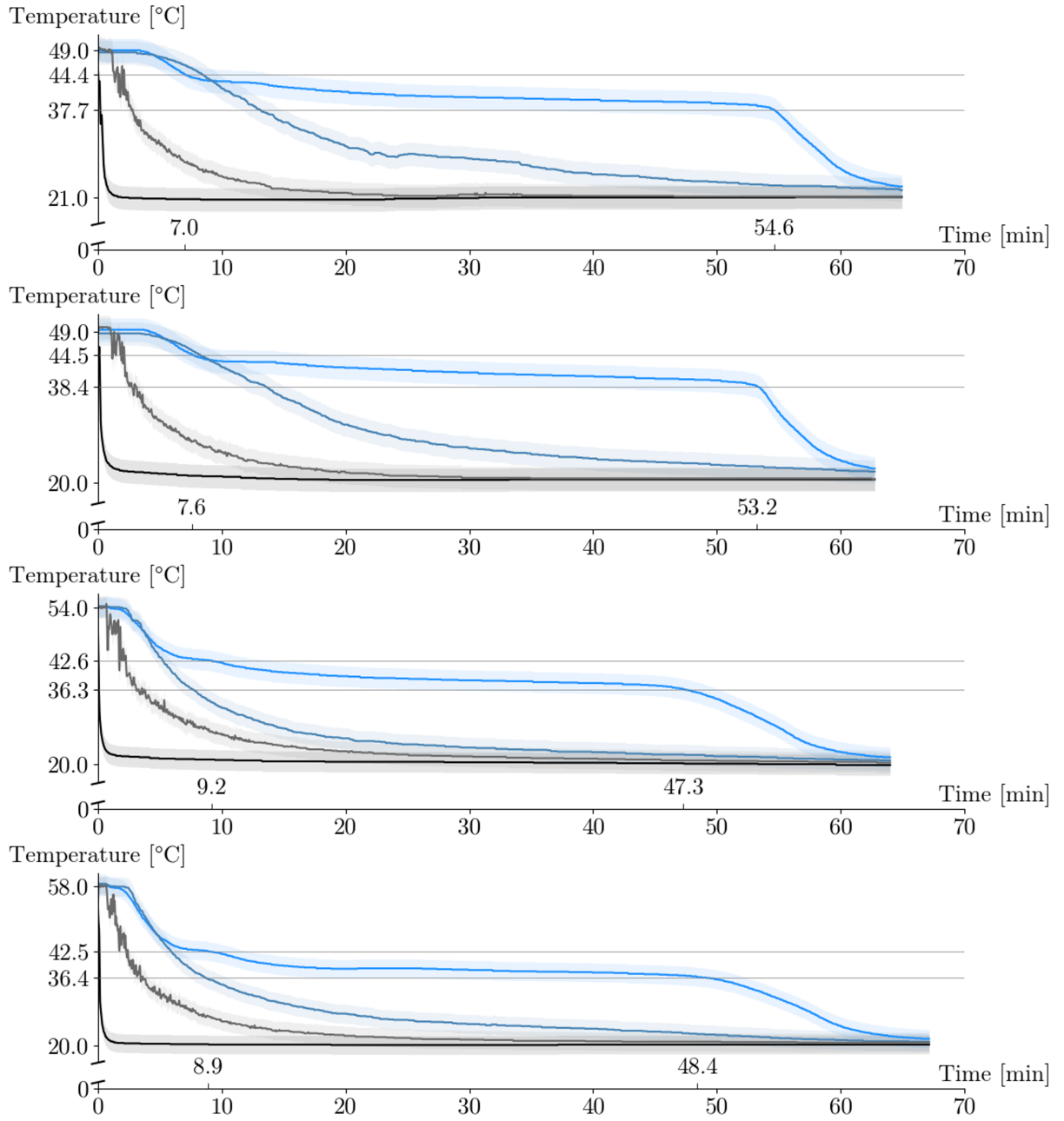


Figure 47: From top to bottom:

 Discharge D2 : $\dot{V} = 4.0$ L/min, $T_{\text{start}} = 48.8^\circ\text{C}$, $t_{\text{discharge}} = 65.0$ min.

 Discharge D3 : $\dot{V} = 4.0$ L/min, $T_{\text{start}} = 49.0^\circ\text{C}$, $t_{\text{discharge}} = 62.8$ min.

 Discharge D4 : $\dot{V} = 4.3$ L/min, $T_{\text{start}} = 54.2^\circ\text{C}$, $t_{\text{discharge}} = 64.0$ min.

 Discharge D5 : $\dot{V} = 4.4$ L/min, $T_{\text{start}} = 58.5^\circ\text{C}$, $t_{\text{discharge}} = 67.2$ min.

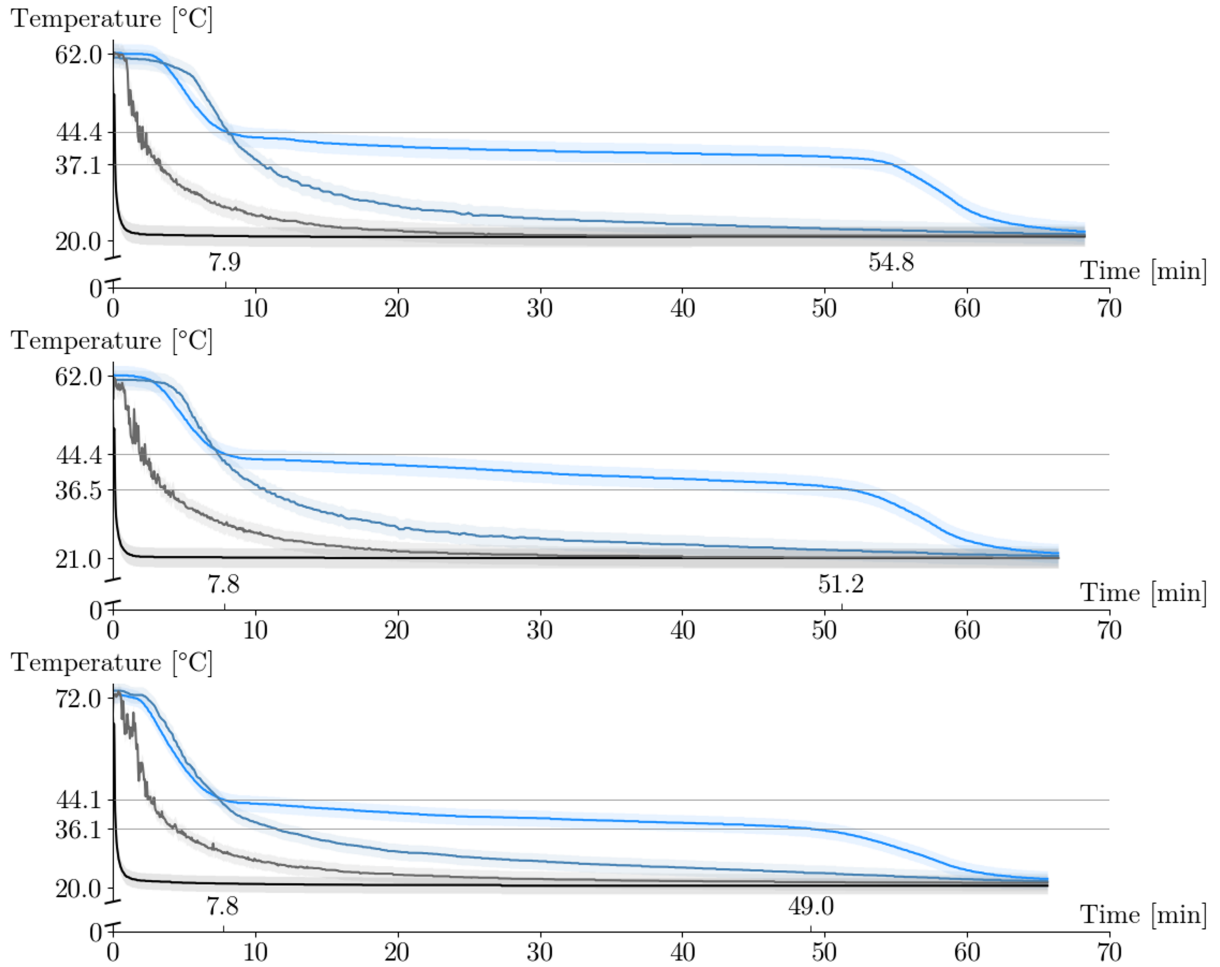


Figure 48: From top to bottom:

Discharge D6 : $\dot{V} = 4.0 \text{ L/min}$, $T_{\text{start}} = 61.6^\circ\text{C}$, $t_{\text{discharge}} = 68.2 \text{ min}$.

Discharge D7 : $\dot{V} = 3.8 \text{ L/min}$, $T_{\text{start}} = 61.7^\circ\text{C}$, $t_{\text{discharge}} = 66.4 \text{ min}$.

Discharge D8 : $\dot{V} = 4.3 \text{ L/min}$, $T_{\text{start}} = 73.3^\circ\text{C}$, $t_{\text{discharge}} = 65.7 \text{ min}$.

A.4.3 Symmetrical sensors

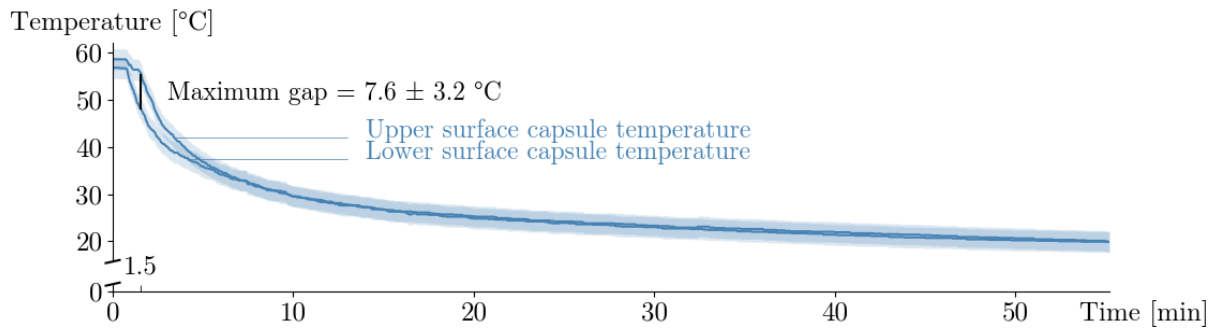


Figure 49: Example of discharge with the temperature sensors placed on a vertical axis in symmetry between the lower and upper surface of the capsule. The difference is cancelled out more quickly than in the case of a charge (Figure 16).

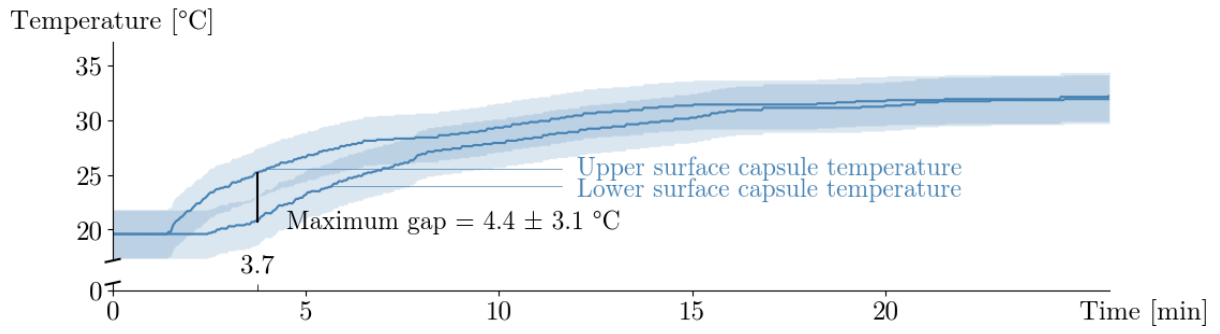


Figure 50: Illustration of a charge where the inlet temperature is limited to 35 °C in order to remain within a sensible heating regime. Despite the uncertainty observed, it can be postulated that a temperature difference exists between the upper and lower surfaces of the capsule, even in the absence of a phase change.

UNIVERSITÉ CATHOLIQUE DE LOUVAIN
École polytechnique de Louvain

Rue Archimède, 1 bte L6.11.01, 1348 Louvain-la-Neuve, Belgique | www.uclouvain.be/epl