Experimental Investigations of Techniques to Enhance the Cooling Rate of Hot Fluids By Utilizing Gallium as a Heat Sink Material

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EXPERIMENTAL INVESTIGATIONS OF TECHNIQUES TO ENHANCE THE COOLING RATE OF HOT FLUIDS BY UTILIZING GALLIUM AS A HEAT SINK MATERIAL

Abdallah M. S. Ghazal

This thesis is submitted in partial fulfilment of the requirements for the degree of Master of Science in Mechanical Engineering

Under the Supervision of Dr. Salahaddin B. Al Omari

November 2017
Declaration of Original Work

I, Abdallah M. S. Ghazal, the undersigned, a graduate student at the United Arab Emirates University (UAEU), and the author of this thesis entitled “Experimental Investigations of Techniques to Enhance the Cooling Rate of Hot Fluids by Utilizing Gallium as a Heat Sink Material”, hereby, solemnly declare that this thesis is my own original research work that has been done and prepared by me under the supervision of Dr. Salahaddin B. Al Omari, in the Collage of Engineering at UAEU. This work has not previously been presented or published, or formed the basis for the award of any academic degree, diploma or a similar title at this or any other university. Any materials borrowed from other sources (whether published or unpublished) and relied upon or included in my thesis have been properly cited and acknowledged in accordance with appropriate academic conventions. I further declare that there is no potential conflict of interest with respect to the research, data collection, authorship, presentation and/or publication of this thesis.

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Abstract

Heat transfer enhancement is crucial for many industrial applications. Liquid water has been known as the commonly used coolant for such applications. When water captures heat from the to-be-cooled heat generation source, it loses its cooling deriving force; hence, it needs to be cooled before it can be reused. This research proposes novel techniques to cool hot water in batch-wise operation, at cooling rates faster than those attainable by conventional means. In these techniques, the water is allowed to lose its heat in a direct-contact manner to a high thermal conductivity sink material, which is taken in this research to be gallium. When gallium is used as a heat sink material, while being in its solid state, it acts as a phase change material (PCM) due to its relatively low melting temperature. In other words, when the solid gallium captures heat from the hot source, it melts to a liquid phase that can experience some kind of superheating upon further dumping of heat into it. Consequently, this may cause appreciable diminishing of the temperature difference driving heat transfer from the source. To overcome such possible liquid gallium superheating issues, the present research proposes for the first time in the literature a new technique to integrate chunks of additional un-encapsulated PCM within the solid gallium; (up to 10% volume fraction in this work) that has lower melting temperature than that of the gallium.

Also, this research proposes new techniques to enhance the rate of heat transfer from the hot water to the gallium either by vibrating the heat sink (gallium and water) under a range of amplitudes (0.3, 0.5 and 0.7 mm) and frequencies (20, 35 and 50 Hz) or discretizing the hot water into small bubbles travelling through a gallium bath with height of 13 or 18 mm. For the water bubbling technique, three supporting
empirical models from the literature were tailored to the experimental conditions of the present study to predict the experimentally investigated bubbling process and water/gallium heat exchange scenarios.

The results have shown that when the PCM loading within gallium increases, higher water cooling rates can be achieved; with less important role played by the way the PCM is structured in the gallium. Also, the vibration experiments revealed that in the range of the studied parameters, the increase in the vibration parameters lead to higher heat removals; with more domineering effect of the amplitude over frequency. Finally, the bubbling technique had resulted in a promising performance when the liquid gallium bath temperature was maintained at low temperature levels. The predicted results obtained by the three empirical models used for the bubbling process were in very good agreement with experimental results.

In conclusion, this work provides better understanding of heat transfer enhancement applications and proposes in new concepts and unconventional techniques for effective cooling of hot fluids (e.g. hot water) that surpass the heat exchange concepts implemented in existing traditional heat exchangers.

**Keywords:** Heat Transfer Enhancement, Direct Contact Heat Exchange, Phase Change Materials, Water Bubbling, Vibration.
دراسات مختبرية لطرق تحسين معدل تبريد الموائع الساخنة عن طريق استعمال الجاليوم (Gallium) كمبدد حراري

الملخص

يُعتبر تعزيز الانتقال الحراري أمرًا بالغ الأهمية لعدة تطبيقات صناعية، وتعرف المياه بأنها المبرد الشائع الاستخدام لمثل هذه التطبيقات. عندما تنتقل الحرارة من مصدر التوليد الحراري إلى المياه، فإنها تفقد القوة الدافعة لعملية التبريد، لذلك فإنها تحتاج إلى إعادة التبريد قبل إمكانية استعمالها مرة أخرى. تقدم هذه الأطروحة تكنولوجيا لبريد المياه الساخنة بشكل دفعات بمعدل تبريد يفوق ما يمكن تحقيقه بالطرق التقليدية. كما وتسمح هذه التقنيات للمياه الساخنة بأن تفقد حرارتها عن طريق التعرض المباشر لمواد ذات موصفات حرارية عالية مثل الجاليوم (Gallium) عند استخدام الجاليوم كمبدد حراري، فإنه يعير من حالة الفزيائية ليصبح سائلاً نظراً لدرجة انصهاره، مما يعني أن تعرض الجاليوم وهو في حالتها الصلبة إلى حرارة، يؤدي إلى تشكيل طبقة من الجاليوم السائل والتي قد تسخن إلى درجات حرارة مرتفعة عندما تتزايد كميات الحرارة المتخلص منها من الموائع الساخنة. يؤدي الحدث الموصوف أعلاه إلى تنافص ملموس في الفارق الحراري والذي يثير على عملية الانتقال الحراري من المصدر.

يقترح في هذا البحث وأول مرة في المقالات الأكاديمية، تقنية جديدة للتغلب على الارتفاع الزائد في حرارة الجاليوم السائل والتي تعمل على إضافة قطع من المواد القابلة للتغيير الفيزيائي بدلاً من فصل الجاليوم الصلب (يُعتبر نسبًا إلى 10% من الحجم) والتي تغير من حالة الفيزيائية على درجة حرارة أقل من تلك في الجاليوم.
يُقترح في هذا البحث أيضاً تقنيات جديدة لزيادة معدل الانتقال الحراري من المياه الساخنة إلى الجاليوم إما عن طريق تعريض المبدد الحراري إلى نطاق الاهتزازات أو تسمية المناطق المختلفة من الموجات (0.3، 0.5، و0.7 مم) والترددات (20، 35، و50 هرتز) أو تجزئة المياه الساخنة إلى فقاعات صغيرة لكي تتصاعد في 13 أو 18 مم من الجاليوم السائل. تم تدعيم النتائج المختبرية لتغير تجزئة المياه الساخنة إلى فقاعات بنسبات لعملية الانتقال الحراري بين الماء والجاليوم بواسطة ثلاثة نماذج تجريبية من المقالات الأكاديمية والمطلقة مع الظروف المختبرية للدراسة الحالية.

أظهرت النتائج أن نسبة زيادة المواد القابلة للتغير الفيزيائي في الجاليوم تؤدي إلى زيادة معدلات التبريد، وقد تضمن النظر عن الدور الذي تلعبه طريقة توزيع تلك المواد. بالإضافة إلى ذلك، أظهرت نتائج الاهتزاز الحراري في نطاق المجال المدروس أن زيادة سعاتترددات الاهتزازات تؤدي إلى زيادة نسبة التبريد، بتبئير أكبر لزيادة السعات عند زيادة الترددات. أدى تجزئة المياه الساخنة إلى فقاعات لنتائج واعدة عندما تمت المحافظة على درجات حرارة متدنية للجاليوم. بالإضافة إلى ذلك، أظهرت نتائج النماذج التنبؤية تلك العمليّة تنافياً نسبياً مع النتائج المختبرية.

وفي الخلاصة، يقدم هذا البحث تصوراً أفضل لتطبيقات تعزيز الانتقال الحراري كما يقترح أفكاراً جديدة وتقنيات غير تقليدية لتبريد المواد الساخنة (المياه الساخنة على سبيل المثال) بفعالية تقوّع عمليات الانتقال الحراري المستخدمة في مبادلات الحرارة الراهنة.

مفاهيم البحث الرئيسية: تعزيز الانتقال الحراري، الانتقال الحراري بالاتصال المباشر، المواد القابلة للتغير الفيزيائي، فقاعات مائية، الاهتزازات.
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I am especially grateful to my parents, brothers, sisters in law and niece who helped me along the way and gave me the moral support whenever needed. In addition, special thanks are extended to Ghazal family for their assistance and friendship.
Dedication

To my beloved parents and family
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<tr>
<td>BWB</td>
<td>Back Wash Bubbling</td>
</tr>
<tr>
<td>ECT</td>
<td>External Cooling Tubes</td>
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<td>PCM</td>
<td>Phase Change Material</td>
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List of Symbols

A: bubble projected frontal area normal to the direction of bubble motion (m²)

Aₚ: Surface area of spherical bubble (m²)

a: Bubble’s acceleration (m/s²)

Cₐ: Drag coefficient

D: Bubble’s diameter (m)

Fₖ: Buoyancy force (N),

F₃: Drag force (N)

g: Gravitational acceleration (m/s²)

K: Gallium’s thermal conductivity (W/(m.K))

k: viscosity of fluid inside the bubble/viscosity of the fluid outside the bubble.

kₜ: thermal conductivity of bubble fluid (W/(m.K))

L: height of gallium bath (m)

m: Bubble’s mass (kg)

Nu: Nusselt number

Pe: Peclet number of bubble.

Pr: Prandtl number of gallium

Prₜ: Prandtl number of bubble

Q: Rate of heat removed from water bubble (W)

Qₜ: Rate of heat removal from hot water (W)

Re: Reynolds number

Tₜₙₚ: Outlet bubble temperature at the gallium’s bath upper surface (°C)

Tₚₙₚ: Inlet bubble temperature at the bubbling tube orifice (°C).
$T_{Ga}$: Temperature of Gallium bath (°C).

$T_s$: Bubble’s surface temperature (°C)

$T_{\infty}$: Free stream temperature of the gallium away from the bubble (°C)

$U$: Convective heat transfer coefficient (W/(m$^2$.K))

$V$: Bubble’s velocity (m/s)

$v_w$: Bubble’s volume (m$^3$)

$W$: Bubble’s weight (N)

$\mu_g$: Gallium’s dynamic viscosity (Pa.s)

$\nu_b$: Kinematic viscosity of the bubbled fluid (m$^2$/s)

$\rho_g$: Gallium’s density (kg/m$^3$)

$\rho_w$: Water’s density (kg/m$^3$)
Chapter 1: Introduction

1.1 Overview

Energy conversion, management and recovery are evident in many of the industrial and commercial applications at which heat transfer plays a pivotal role in their operation. Some of these applications could be found in power generation stations, chemical industry, automobile and advanced electronics industries that may involve: condensation in power and cogeneration plants, sensible heating and cooling of viscous media in thermal processing of chemical products, air and liquid cooling of engine and turbomachinery systems and cooling of electrical machines and electronic devices [1, 2].

As inferred, due to the demand of the industry, the need to research and investigate in order to enhance the heat transfer mechanisms is vital. This is because it will contribute to the enhancement of the overall thermal performance of heat exchangers and increases their efficiency; hence, influencing the economies of the direct-users industries [1-3].

1.2 Statement of the Problem

Conventional coolants such as water are widely used in thermal management applications such as electronics and others. When these coolants capture heat from the heat-generating source, they need to be cooled down again in a conventional heat sink so that they can be used repeatedly in closed loop systems. Having a thermal management system that can handle such a task efficiently and in a timely manner is considered as one of the major challenges in achieving the above task. This work experimentally investigates novel techniques to cool down heat sources efficiently.
These heat sources can be of different natures; hot fluids or solid objects. Yet, this research focuses particularly on cooling hot fluids; ‘hot water’ in specific.

There is quite good evidence in the literature [4-8] indicating that when two immiscible fluids are brought into direct contact, they exchange heat more effectively than when being separated by an intervening solid surface (e.g. tube wall). Moreover, this direct heat exchange is even further enhanced when a hot, to-be-cooled, fluid (e.g. hot water) loses its heat to a metallic heat sink material that has relatively high thermal conductivity and undergoes a phase change process at the heat transfer process (i.e. gallium).

The current research builds on the above observations presumed from the literature and attempts to advance the research further in the direction of direct contact heat exchange between immiscible liquids (liquid metal/liquid water) targeting coming up with ways to enhance the cooling of hot fluids (e.g. hot water).

When liquid metals are used as heat sink materials, they suffer from a major drawback; namely their limited specific heat capacity. This limitation may seriously offset any superiority attributed to the high thermal conductivity these media may have. More specifically, their very limited specific heat capacity makes it almost impossible for a certain amount of a liquid metal to capture any significant quantity of heat without experiencing an appreciable rise in temperature. This temperature rise leads to diminishing heat transfer driving potential; i.e. to a small temperature difference, that subsequently hinders the effective use of such materials as heat sinks for extended operating periods.

In order to address the above issue, namely the diminishing heat transfer, two main proposed solutions were implemented. The first is to directly boost the thermophysical properties of such materials; in particular their specific heat capacity,
while the second is to employ innovative operational design concepts that actively facilitate effective heat transfer and resulting in faster heat removal from the to-be-cooled hot fluids.

The experimental research conducted at the present work aims to provide valuable contributions to the above two fascinating areas. In regards to the first area, the usage of phase change materials (PCMs) with relatively higher specific and latent heat than the heat sink material (here solid gallium) is proposed in a way that has not been disclosed before in the literature. In regards to the second area, two main directions have been considered to augment the feasibility of utilizing gallium as a heat sink material, without directly imposing any modifications to its thermophysical properties. In particular, two active approaches are implemented in the heat sink’s design that can lead to enhancing the heat transfer between the to-be-cooled water and gallium. Hereby, vibrational effects are introduced to cause an external excitation to the built heat sink that involves hot water in direct contact with a gallium body. The other approach that is implemented in this context addresses disintegrating the hot water mass into small droplets and bubbling them through a column of a cooler liquid gallium such that the rate of heat exchange between the two media will be significantly enhanced due to the different effects that will be further elaborated in next sections of the thesis.

It is anticipated that the above techniques will lead to facilitate the heat transfer enhancement to the extent that it would allow for a high exploitation of the inherent high heat exchange potential of the materials and design concepts implemented. The use of small quantities of the PCMs in this work targets alleviating the limitations of the specific heat capacity of liquid metals used (here, gallium). Moreover, the new hardware and process design concepts adopted will allow for
faster heat removal (i.e. heat transfer enhancement) from the hot fluid; for situations where heat transfer time plays an important role.

Although this study addressed separate individual elements to shed light on approaches that can be followed to enhance the feasibility of utilizing gallium in heat sink applications, these different approaches (depending on the application) can all, or at least in part, be integrated together in one whole complete heat sink design.

1.3 Relevant Literature

This section includes relevant literature to the work done in this thesis; covering the main areas of relevance to the topic addressed. It is further sub sectioned to background, at which the basic knowledge about heat exchangers and phase change materials is covered, and literature review, at which recent advances in the researched topic is summarized.

1.3.1 Background

Direct contact heat exchangers and phase change materials are of much relevance to the thesis work. Hence, it is worth providing the basic general physics behind them including their: classifications, principles of operation as well as their usage advantages and disadvantages. However, detailed literature addressing the state of the art for the advanced engineering knowledge and applications directly pointing at the techniques used in this research work will be provided in the literature review subsection afterwards.
**Heat Exchangers**

This section provides an overview of heat exchangers focusing on their classification based on the transfer process (e.g. direct and indirect) as will be shown later in this section. The overview starts with a definition of heat exchangers. Then, the general classification basis of heat exchangers is provided. After that, examples of direct and indirect contact heat exchangers will be mentioned. Finally, advantages of direct contact heat exchangers over the indirect type will be highlighted.

“Heat exchangers are devices designed to facilitate the flow of thermal energy between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact” [9]. These exchangers range from simple devices made of plastic or paper (for low-temperature applications) to the sophisticated ceramic exchangers (for high-temperature applications) such as gas turbine regenerators [10].

There are many classifications of heat exchangers based on the following basis: Transfer Processes, Number of Fluids, Surface Compactness, Construction Features, Flow Arrangements, and Heat Transfer Mechanisms [11]. Figure 1.3.1 shows a flow chart that summarizes the above classifications [12].
Figure 1.3.1: Classifications of the Heat Transfer Processes [12]
The study of heat exchangers classifications is out of the scope of this study; however, the most related classification, which is based on the transfer process, will be touched as in below.

According to the flow chart given in Figure 1.3.1 heat exchangers, based on the transfer process, can be classified into direct and indirect contact types. As the name implies, for the direct contact heat exchangers, the two streams are exposed to each other directly without any thermal barrier between them unlike the indirect type at which impervious dividing wall separates the streams \[11, 12\].

Indirect contact heat exchangers could be further classified into direct transfer type, storage type and fluidized bed. First, at the direct transfer type, the hot stream is continuously supplied in a separate passage than the cold stream, and it is allowed to transfer heat through a solid wall. Second, unlike the direct transfer type, the heat transfer is intermitted at the storage type. The basic principle of operation at this type is based on the storing and releasing of heat from a solid matrix from which heat is transferred from the hot gas passing through the passage tubes. Third, at the fluidized bed, especially in coal combustion fluidized beds, a fluid (e.g. water) flows in a tube that is submerged in a coal bed or any combustible agents. Then, the heat is allowed to transfer from the hot gases resulting from the combustion to the to-be-heated water tubes \[12\].

Direct contact heat exchangers are sub-classified according to the phase characteristics into: gas-liquid, liquid-vapor and immiscible fluids. First, the gas-liquid heat exchanger can be seen in the famous wet cooling tower application, where warm/hot water is pumped to the top of the tower and splashed so that it gets
in contact of air that is circulated using fans; either in counter or cross direction as shown in Figure 1.3.2.

Figure 1.3.2: Direct Contact Heat Exchangers (a) Counter Flow Wet Cooling Tower, (b) Cross-Flow Wet Cooling Tower [13]

Second, the liquid-vapor heat exchanger is better understood by having a close look at water spray desuperheaters (see Figure 1.3.3) at power plants, where superheated steam is directly exposed to sprayed cold water; resulting in cooling the steam from its superheating to saturating state.

Figure 1.3.3: Single Point Radial Injection Spray Desuperheater [14]
Third, the immiscible fluids heat exchanger (see Figure 1.3.4), that is the most related type of direct contact heat exchangers to the work done in this thesis from the physics perspective, allows the streams to be brought into direct contact in either a single phase or condensation/evaporation [12]. The main driving force of separation in this case is gravity (e.g. variation in density) [10].

![Figure 1.3.4: Schematic of: (a) Agitated Column, (b) Perforated-Plate Column](image)

The advantages of direct contact heat exchangers are raised from the limitations of the indirect type. As mentioned earlier, in the indirect heat exchanger, the two streams are allowed to exchange heat through a wall separating them; that could be disadvantageous to use due to the thermal resistance on the heat transfer process imposed by the separation wall. Moreover, at elevated temperatures, such walls may corrode or foul, which decrease the heat transfer effectiveness. In addition, the ability for the two streams to be directly contacting each other provides a larger
surface area for heat transfer to take place; hence, the sizes of the direct contact heat exchangers are usually smaller than those of the indirect type [10, 15].

It should be noted again that all the work done in the thesis is focusing on the direct contact heat exchangers due to the advantages highlighted above. More specifically, in all experiments done, hot water is allowed to directly transfer heat to superior heat transfer media (e.g., gallium) so that higher heat removal rates could be achieved. Detailed description of the experiments will be mentioned in the experimental setup section. Also, the rate of enhancement in heat transfer will be addressed in the results and discussion section as well be seen later in this thesis.

**Phase Change Materials (PCMs)**

This section provides an overview of typical phase change materials focusing on the solid-liquid PCM type. It provides its basic fundamentals and applications for thermal management applications, the most related applications to the work done at this thesis. The overview starts with a definition of phase change materials and a differentiation between the latent and sensible heat. This was done along with an introduction of the PCM supercooling phenomenon. Then, the types of PCMs, PCM desired thermophysical properties, PCM selection criteria and PCMs thermal conductivity issue in thermal management application will be highlighted.

PCMs are materials that experience phase change (e.g., from solid to liquid) at a desired constant temperature depending on the operating temperature of the application [16].

In order to understand the heating and cooling cycles of the PCM, it is worth distinguishing between latent and sensible heat. Latent heat is the one at which the
material absorbs heat/energy without experiencing an increase in the material’s temperature (e.g. it absorbs heat at constant temperature). It could be either a fusion type, when the material undergoes a phase transition from solid to liquid, or a vaporization one, when the material undergoes a phase transition from liquid to gas (see Figure 1.3.5 (a)). A closer look into the atomic level explains the above phenomenon. The absorbed heat in the case of fusion, for example, is responsible for increasing the energy level of the solid state atoms to a level at which the bonds among the atoms are loosen to the liquid state. Unlike latent heat, sensible heat is the one responsible for increasing the temperature of the material due to the further increase of the atoms’ energy (see Figure 1.3.5 (b)).

Figure 1.3.5: (a) Standard Heating Curve, (b) Atomic Melting and Solidification Process [16]
There are many varieties of PCMs depending on the phase transition states; however, only solid-liquid PCMs will be addressed here; since it is the most relevant one to the thesis work. This kind of PCMs can be classified to organic, inorganic and eutectic types. The organic type can be further classified into paraffin and nonparaffin. The inorganic type; however, can be either salt hydrate or metals. Finally, eutectics are classified to: organic-eutectic, inorganic-eutectic and organic-inorganic mixtures as shown in Figure 1.3.6.

Figure 1.3.6: Solid-Liquid Phase Change Materials Classifications [17]

Organic and inorganic types are the most used PCMs in applications that could be of relevance to the thesis work. Hence, a comparison between them is summarized in Table 1.3.1 and a list of metallic PCMs is provided in Table 1.3.2.
Table 1.3.1: Comparison between Organic and Inorganic PCMs

<table>
<thead>
<tr>
<th>Materials and Chemical Structure</th>
<th>Organic</th>
<th>Inorganic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Paraffin</td>
<td>C_{n}H_{2n+2}</td>
<td>CH_{3}(CH_{2})_{2n}COOH</td>
</tr>
<tr>
<td>NonParaffin – Fatty acids</td>
<td>&lt; the paraffin</td>
<td></td>
</tr>
</tbody>
</table>

| Melting temperature              | 35 °C to 70 °C | < the paraffin | 10 to 900 °C | 28 to >700 °C |

| Advantages                       | • Have high latent heat of fusion allowing them to store a high amount of energy in a smaller mass  
• No supercooling  
• Chemically stable  
• Good compatibility with other materials | •Sharper transitions at the melt point than organics  
• Higher thermal conductivities than organics  
• Higher densities than organics - leading to greater energy densities | • Sharper melt transition than the organics  
• Excellent chemical and physical stability  
• High thermal conductivity range of 8–40 W/m K  
• High densities |

| Disadvantages                    | • The melt occurs more slowly over a range of temperatures  
• Extremely low thermal conductivities. The low thermal conductivity (for example ~0.2 W/m K. for most paraffin waxes) decreases their effectiveness in both energy storage and thermal management applications  
• Large volume change | • Significant instabilities  
• Causing corrosion  
• Supercooling | • Low latent heats of fusion |
Table 1.3.2: Thermal Properties of Common Metallic PCMs [16]

<table>
<thead>
<tr>
<th>Name</th>
<th>$T_m$ (°C)</th>
<th>Latent Heat (kJ/kg)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$C_p$ (kJ/kg)</th>
<th>$k$ (W/m.k)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cesium</td>
<td>28.65</td>
<td>16.4</td>
<td>1796</td>
<td>0.236</td>
<td>17.4</td>
</tr>
<tr>
<td>Gallium</td>
<td>29.8</td>
<td>80.1</td>
<td>5907</td>
<td>0.237</td>
<td>29.4</td>
</tr>
<tr>
<td>Indium</td>
<td>156.8</td>
<td>28.59</td>
<td>7030</td>
<td>0.23</td>
<td>36.4</td>
</tr>
<tr>
<td>Tin</td>
<td>232</td>
<td>60.5</td>
<td>730</td>
<td>0.221</td>
<td>15.08</td>
</tr>
<tr>
<td>Bismuth</td>
<td>271.4</td>
<td>53.3</td>
<td>979</td>
<td>0.122</td>
<td>8.1</td>
</tr>
<tr>
<td>Zinc</td>
<td>419</td>
<td>112</td>
<td>7140</td>
<td>0.39</td>
<td>116</td>
</tr>
<tr>
<td>Mg</td>
<td>648</td>
<td>365</td>
<td>1740</td>
<td>1.27</td>
<td>156</td>
</tr>
<tr>
<td>Al</td>
<td>661</td>
<td>388</td>
<td>2700</td>
<td>0.9</td>
<td>237</td>
</tr>
</tbody>
</table>

As shown from Table 1.3.2, comparing to all metallic PCMs presented, gallium shows the most advantageous thermal features for thermal management applications (e.g. cooling electronics). In regards to the melting point temperature, the only competitor for gallium is cesium; however, a comparison between both metals shows that gallium is more superior than cesium. It is about five times and near twice as much comparing to cesium in terms of the latent heat of fusion and thermal conductivity respectively. Hence, the present work has addressed gallium as a metallic PCM since it left no room for other competitors to be used in applications mentioned above.

The heating and melting curve of a solid-liquid PCM (see Figure 1.3.7) shows that when the PCM is heated, its temperature increases due to sensible heat. This occurs till the solid reaches its melting point at which the PCM starts changing its phase at constant temperature upon adding further energy; in the form of latent heat. When the phase change process is complete, an increase in the temperature of
the liquid phase starts to be noticed due to sensible heat again. Solidification; however, is the reverse process of the above, at which the liquid phase is cooled by rejecting sensible heat till it reaches its solidification temperature. At this stage, by further removal of heat, the PCM starts to change its phase from liquid to solid by restructuring the atoms at constant temperature and the heat removed in this case is latent. Finally, when complete solidification is reached, the solid phase is cooled to the desired temperature by losing sensible heat again (see Figure 1.3.7).

![Melting and Solidification Process of PCMs](image)

**Figure 1.3.7: Melting and Solidification Process of PCMs [18]**

It should be noted that when the two processes are combined, melting and solidification, a phenomenon called supercooling or subcooling may be present. Supercooling occurs when the liquid phase is cooled below its phase transition temperature without being transferred to the solid phase (see Figure 1.3.8). In this scenario, the heat will only be sensible till the phase transition starts [18, 19]. Generally, supercooling effect is not a desirable kinetic property of the PCM in the application of thermal management as will be shown later in this section.
The selection criteria of the PCM depends on the desired needs of the users as well as the properties of the heat storage materials (e.g. PCMs) as shown in Table 1.3.3. The arrows drawn in solid lines show the strongest dependency between the desired need and the PCM property; however, the dashed arrows represent a weaker connection than the solid line.

Table 1.3.3: PCM Selection Criteria [20]

<table>
<thead>
<tr>
<th>Desired Output</th>
<th>PCM Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input/Output Temperature</td>
<td>Phase Change Temperature</td>
</tr>
<tr>
<td>Heat Accumulation Density</td>
<td>Latent Heat</td>
</tr>
<tr>
<td>Load Responsiveness (Dynamic Response)</td>
<td>Volumetric Change in Phase Change Process</td>
</tr>
<tr>
<td>Exergy Efficiency</td>
<td>Dynamic Characteristics in Phase Change</td>
</tr>
<tr>
<td>Reliability</td>
<td>Heat Transfer Characteristics</td>
</tr>
<tr>
<td></td>
<td>Thermal Stability</td>
</tr>
</tbody>
</table>
The thermophysical properties of the PCM depend on the applications at which they are used. As mentioned earlier, the usage of PCM in the thesis work is intended for thermal management applications. For this reason, the selected PCM should meet the thermal, physical, kinetic and chemical performance of the targeted application as shown in Table 1.3.4. The table summarizes the main desirable properties of PCM in thermal management applications. Comparing many PCMs, gallium proves to be the most suitable candidate satisfying the majority of the PCM desired properties as shown in Table 1.3.4.

### Table 1.3.4: Desirable Properties of PCM [17, 20]

<table>
<thead>
<tr>
<th>Thermal Properties</th>
<th>Physical Properties</th>
<th>Kinetic Properties</th>
<th>Chemical Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>High Latent Heat of Fusion</td>
<td>Small Volume Change</td>
<td>Sufficient Crystallization Rate</td>
<td>Long Term Chemical Stability</td>
</tr>
<tr>
<td>High Thermal Conductivity</td>
<td>Low Vapor Pressure</td>
<td>No Supercooling</td>
<td>No Toxicity</td>
</tr>
<tr>
<td>Suitable Melting/Freezing Temperature</td>
<td>High Density</td>
<td></td>
<td>Non Flammable and Non Corrosive</td>
</tr>
</tbody>
</table>

Low thermal conductivity is one of the most challenging features of PCMs, specially, in applications related to thermal management where the response time of the PCM to capture the generated heat is a key factor (e.g. electronics). Thermal conductivity is a property of the material that is responsible for transporting the heat through its core, assuming the heat was subjected at its surface. The usage of low thermal conductivity PCM in such applications can deteriorate their thermal
performance and reduce the overall efficiency of the system as the PCM, in some cases, may act as a thermal insulator instead.

Below are two case scenarios (see Figure 1.3.9 and Figure 1.3.10) at which low thermal conductivity is disadvantageous. The first case is heating a PCM that was initially solid, whereas the second case is cooling liquid PCM by penetrating a pipe that carries a cold fluid.

Figure 1.3.9: Heat Pumping into Superheated Liquid Layer [16]

Figure 1.3.10: Solidification on a Cold Surface with an Increasingly Thick Solid Layer that Impedes the Heat Transfer [16]
In the first case, as shown in Figure 1.3.9, the low thermal conductivity of PCM prevents the heat flux from diffusing easily through the PCM material due to the thermal resistance that such materials imposes on the thermal energy transport. Hence, a layer of superheated liquid can be formed and act as a barrier for the heat transfer path, which may significantly slowdown the heat transfer rates and; consequently, may pose undesirable or even unsafe operation. In other words, the liquid layers formed due to melting, depending on the configuration and heat supply method, may induce internal convective currents that may enhance the heat transfer initially. However, ultimately the superheating effect of the liquid due to limited thermal conductivity may lead subsequently to adverse effect by deteriorating the driving temperature difference between the hot source and the liquid PCM layer.

In the second case, as shown in Figure 1.3.10, the liquid PCM starts to solidify at the circumference of the pipe. The solidification wave penetrates to the rest of the liquid PCM by conducting heat to the pipe at a very slow rate due to the high thermal resistance. Hence, in this case, the solid layer of PCM will act as a thermal insulator, which is not desirable [16].

In order to avoid such scenarios, the thermal conductivity of PCMs should be enhanced effectively. This could be done mainly using one of the following methods, namely; the use of macro scale metallic fins or nanoscale metallic particles forming a PCM nanofluid. The methods are further explained in [16].
1.3.2 Literature Review

This section includes a review on the state of the art available in the literature that is of most relevance to the studied techniques addressed in the problem statement section 1.2. In general, such techniques are used to design heat sinks that are able to enhance the heat transfer rates from hot fluids. A summary of the work done in using low melting point metallic phase change materials (e.g. gallium) in thermal management applications will be given. In addition, techniques as introducing vibrational effects for enhancing the convective heat transfer from very high melting point materials/objects (such as steel rods), in general, and relatively low melting point phase change materials, in specific, will be summarized. Finally, further techniques such as imposing vigorous relative motion between the to-be-cold fluid and the heat sink implemented by discretizing the hot fluid into discrete bubbles to facilitate the direct contact heat transfer will be mentioned.

1.3.2.1 Low Melting Point Liquid Metals as PCM (Metallic PCMs)

Conventional phase change materials (PCMs), the most frequent organic and inorganic types used, have proven to be promising in heat sinks’ materials nowadays. Yet, there are many factors that are attributed to their deficiencies in thermal management applications such as their limited thermal conductivity (recalling that the relatively high thermal conductivity of PCMs is a desired thermophysical property in such applications). Hence, the work summarized here has addressed low melting point metals (e.g. metallic PCMs) such as gallium. Several research papers have pointed on the above concept as will be shown in the summarized literature below.
H. Ge and J. Liu [21] have investigated the effect of adding low melting point metal, Gallium, to a USB flash memory on its operating temperature during data transfer. A commercially available USB was used to transfer 3.63 GB data from a desktop computer. A thermocouple was fixed at the outer wall of the USB to monitor its temperature during the transferring process and continued to record the temperature 10 minutes after the data transfer. 0.668 ml of gallium was used to fill the fabricated PCM USB. The tests done were under ambient condition, approximately 25 °C. The results have shown that without using the gallium as PCM, the USB temperature was raised quickly to 42 °C during data transmission, then, dropped and stayed at 37 °C after the 10 minutes of no data transfer; however, the counter parts, when the modified PCM USB was used, of the aforementioned temperatures were 29 °C and 28 °C respectively. Hence, it was concluded that the modified PCM USB can sustain the operating temperature in safe mode during the data transfer size studied in the experiments.

Later, H. Ge and J. Liu [22] have conducted conceptual experiments on using PCM to cool smartphones. In that study, the authors had fabricated hand-held stainless steel container filled with sodium sulfatedehydrate, paraffin, N-eicosane or gallium. A heater was placed on the container to simulate the CPU ship of the smart phone and was controlled using a DC power supply. Four T-type thermocouples, connected to data acquisition, were used to record the temperature history of the fabricated container. The heating was imposed for 30 minutes and the container was allowed to cool for another 30 minutes. At this work, a temperature of 45 °C was set as the human comfort temperature for such applications. The power applied was varied from 1.0 to 3.5 Watts under ambient conditions. The results have shown that the holding time, the time at which the temperature was within the
comfort level, when gallium have been filled in the container was the maximum comparing to the holding time of the other PCMs used. Also, it was shown that the gallium filled container was successful in reducing the operating temperature below the human comfort level. More specifically, it was shown that 3.4 ml of gallium was sufficient to maintain the temperature of the container below 45 °C for 16 minutes at a power of 2.8 Watts. It should be noted that the authors had reported that the gallium can suffer from the supercooling effect and to overcome such issue, it was suggested to add 1.0 weight percent of powder SiO$_2$ to gallium to initiate its crystallization.

L. Fan et al. [23] have tested the performance of Pb–Sn–In–Bi metallic alloy in a thermal storage heat sink and compared the results obtained with organic PCM (1-octadecanol). It should be noted that both PCMs had almost the same melting temperature, 60 °C, and volumetric latent heat of fusion. The heat sink filled with the PCM was placed on top of a designed heating unit that is cylindrical in shape and had four embedded heaters. Two T-type thermocouples were placed to monitor the temperature variations while conducting the experiments; one at the heater’s surface and the other one in the middle of the PCM. All experiments were performed at 21 °C surrounding temperature. It was concluded that the metallic PCMs are better candidates than organic PCMs in thermal energy storage applications due to their relatively high thermal conductivity. In addition, it was mentioned that the usage of the metallic PCM allowed for 50 °C temperature difference between the hot surface and the PCM’s temperature, doubling the time needed for charging, heating, and shortened the time for discharging, cooling, when comparing to the performance of the organic PCM.
X. Yang et al. [24] have done a numerical analysis to investigate the phase change characteristics of liquid metal PCMs (LMPCM) as gallium. The paper started by comparing the melting performance of gallium and paraffin. Then, the melting process of the LMPCM was done through a dimensionless analysis. Commercial Fluent 6.3 was used to simulate the problem using the enthalpy-porosity method and Boussinesq approximation in solving the continuity and momentum equations. The authors were successful in developing dimensionless correlations for the Nusselt number and melting fraction for constant temperature and constant heat flux boundary conditions. The developed correlations are applied for low Rayleigh number (<2000-4000), and it was noted that further investigations should be done to characterize the melting at high Rayleigh number where natural convection becomes of a significance in the analysis.

Also, X. Yang. [25] have done a theoretical and numerical analysis to optimize the performance of low melting point metals, (e.g. gallium) when they are subjected to thermal shocks, order of 100 W/cm². The heat sink studied had internal copper fins structure. Factors including fins’ structure, number, width and base thickness were investigated. The theoretical analysis conducted at this study was based on solving 1-D Stefan problem under constant temperature boundary conditions and the numerical analysis was validated against its results. It was concluded that the optimized heat sink design had experienced maximum temperature of 46 °C when a thermal shock was applied to it at room temperature, 25 °C. Also, it was pointed that the geometry of the internal fins greatly affects the thermal performance of the heat sink. Increasing the number of fins resulted in better performance till a limit value at which the lowest increment in temperature is
observed. Moreover, the fin area fraction was noted to be a key factor in the design of fins and its optimum value ranges from 0.45-0.6 based on the heating condition.

At a later study, X. Yang et al. [26] have experimentally and numerically studied the cooling performance of low melting point metallic PCM, E-BiInSn ~60 °C, filled heat sink with internal fins structure and compared it to organic PCM, octadecanol, that has nearly comparable melting temperature, ~55 °C. Three fins configuration were analyzed at this study including 1×1, 2×2 or 3×3 structures with 2 mm thickness fins. In the experimental part of the study, the heater was designed the same way found in [23]. The sink was placed on top of the heater and thermal grease was added in between to reduce the contact resistance. The heating conditions applied were: 80 W, 200 W and 320 W. All the experiments involved 100 ml PCM and performed at room temperature, 24 °C. As far as the numerical analysis is concerned, 3D conjugate numerical model was developed and tested using the method found in [24]. The results revealed that the performance of E-BiInSn is twice better compared to the one for octadecanol. Moreover, it was noted that experiments with E-BiInSn had shown good repeatability and supercooling effects were not observed.

L. Shao et al. [27] have investigated the effect of adding organic and metallic phase change materials in thermal-managing the high power heat flux devices at which microprocessors experience peak operation in less than one second, (e.g. operating in discontinuous form). The fabricated microprocessor with PCM was tested under real boundary conditions and integrated to Samsung Galaxy S3 smartphone for assessment. Rotometal, Bi, 21% In, 18% Pb and 12% Sn, and Paraffin wax BW-407 were used as metallic and nonmetallic PCMs respectively. It
should be noted that smart phones consume 1.2 – 1.3 Watts at maximum operations. The Testing Thermal Chip (TTC) was placed in the smart phone between the inner cover and CPU and operated in pulsating power form. The rests of the preliminary testing have shown that TTC with the Rotometal can lower the maximum temperature from 85 °C to 69 °C when comparing to the case without PCM.

Z.Li et al. [28] has addressed a relatively new concept of cooling electronics by combining liquid metal phase change material (LMPCM), 4.12 ml Gallium, to a flat heat pipe (FHP) in a module having the latter acting as a heat storage while the former acting as a thermal spreader. The authors had tested the module under both steady state and transient operation. Benchmark experiments of LMPCM and FHP where conducted for comparison purposes and to assess the performance of the suggested module. In order to achieve steady state condition, a DC power supply was turned on for 60 minutes to a 1.0 cm² heater that simulates the chip. However, the transient behavior was achieved by turning the DC power supply on for some time interval then switching it off for some time and so on. The whole process was monitored by T-type thermocouples placed at the surface of the testing apparatus. The results have shown that the proposed design is effective in cooling portable power electronics as laptops or pad PCs. Under the studied range of applied power, the FHP-LMPCM module was able to sustain the maximum surface temperature below 45 °C for 3468 seconds at 3.65 Watts and for 600 seconds at 7.25 Watts. Finally, the authors suggested conducting further studies at the microscopic level in order to understand the heat transfer process.

In this work, gallium has been used as the metallic phase change material in the designed heat sink (see section 2.1). As mentioned earlier in the background
section 1.3.1, gallium has favorable thermo-physical properties for thermal management applications; high thermal conductivity and low melting point. However, still gallium suffers from its relatively low specific heat capacity that might offset such advantages.

If solid gallium was brought into direct contact with the to-be-cooled hot fluid, an initial melted layer of liquid gallium will be formed after absorbing some of the energy content from the hot fluid in the form of latent heat. After this time, the melted layer will always be in direct contact with the hot source and might superheat due to the limited specific heat highlighted above.

Hence, at this work, it is proposed for the first time in the literature to integrate inorganic PCM (melting at 21 °C) within the solid gallium body in the form of structural arrangement or random distribution without any solid wall intervening them so that any disadvantages of indirect contact heat exchangers can be avoided. It is proved that this integration will result in lowering the temperature of the solid gallium body, which has not been melted yet, and the melting process in overall. This happens because some of the heat which has been dumped into the gallium will be partially absorbed by the inorganic PCM; resulting in partial relieving of the cooling load from gallium. More details about the designing process and the heat sink performance can be found in later chapters on the experimental setup and results and discussion.

1.3.2.2 Vibrational Effects to Enhance the Convective Heat Transfer

The application of vibration is considered as one of the active techniques (e.g. requires external power source) used to increase the convective heat transfer between the to-be-cold object and the heat sink. The idea above could be implemented to
regular or complex shape very-high melting point objects (which do not experience any phase change at the operating conditions of the conducted studies) or low melting point PCMs. In both cases, it results in heat transfer enhancement and, as a result, it increases the melting or solidification rates for systems that involve phase change processes in specific.

### 1.3.2.2.1 Heat Transfer Enhancement for Regular-Shape Objects without Phase Change

This section addresses the facilitation of the heat transfer from the hot source/objects by means of vibrating them. Early attempts to introduce the contribution of vibrational effects in the heat transfer process has started by testing heating wires [29, 30] as well as standard regular shapes as plates [31, 32] and cylinders [33, 34]. Then, the idea was developed to include complex geometries and applications [35]. References on the aforementioned point will be summarized below.

R. Lemlich [29] has studied the effect of vibration on the natural convection heat transfer experimentally. This study is considered to be one of the earliest in addressing the above phenomenon. In his experiments, the author had placed an electrically heated wire, exposed to air, such that it is stretched between two bridges. Only the center of the wire was allowed to vibrate and a thermocouple was placed at the same location, i.e. center of the wire, to monitor its temperature. Five variables had been studied in the experiments namely the wires’ diameters, temperature difference between the center of the wire and the surrounding environment, amplitude and frequency of the vibration, and the direction of oscillation, vertical or horizontal. It was concluded that the effect of vibration is evident in enhancing the
heat convective transfer. In addition, it was mentioned that both amplitude and frequency of the applied signals are directly related to the enhancement. Moreover, almost there were no effects of changing the direction of oscillations. Finally, it was mentioned that the vibration variables might be correlated in a dimensionless Reynolds number.

W. R. Penney and T. B. Jefferson [30], in a study similar to the one presented in R. Lemlich [29], have studied the effect of vibrating a platinum wire soldered across a copper yoke that was designed to allow for vibration in water and Ethylene Glycol. The applied vibrational signal was done with the aid of a scotch yoke that was driven by an electrical motor. The main objective of the study was to investigate the effect of low frequency, 0 to 4.5 Hz, high amplitude, 0 to 63.5 mm, and direction of oscillation, vertical or horizontal, on the heat transfer of the hot wire. Also, the study aimed to develop some correlation methods that can be applied to a range of fluid properties. It was concluded that the vibrational parameters can enhance the heat transfer from the hot wire. Also, it was mentioned that the effect of vertical oscillations on the heat transfer were more than the effect of the horizontal ones. In addition, a comparison between the findings of this study and the previous ones addressing water and Ethylene Glycol revealed, the authors had mentioned that they were successful in correlating the data obtained from their study and failed to do so for results from previous studies.

Unlike R. Lemlich [29] and W. R. Penney and T. B. Jefferson [30], K. K. Prasad and V. Ramanathan [31] has studied the effect of longitudinal vibrations of heated plates on the free convective heat transfer. In this study, a reaction type vibration machine was used to vibrate the tested plate. The basic principle of the
device is mainly dependent on the reaction of rotation unbalance masses to generate a force in the desired direction. Parameters including the signal’s amplitude and frequency as well as the temperature of the heated plate were studied. The ranges of the applied parameters were as follows: frequency (10 to 20 Hz), amplitude (0 to 635 mm) and temperature (38 to 94 °C). The authors have concluded that in the range of the applied parameters, a maximum enhancement of 33% in the heat transfer rate was recorded.

A. Dawood et al. [33] have investigated the effect of vibration on the heat transfer from a horizontal cylinder that has been heated using electrical resistive heaters. Two cylinders of different diameters were studied and a range of applied amplitude and frequency were employed. The testing setup was vibrated using electrodynamic exciter (B&K). Thermocouples were mounted on the surface of the cylinders and the surrounding environment so that the temperature difference between them can be measured. The range of amplitude, frequency and temperature difference for the first cylinder were: 0-17.15 mm, 0-63.7 Hz and 18-150 °C, while for the second cylinder were: 0-17.8 mm, 0-68 Hz and 22-90 °C. The results have shown that the vibration can increase the heat transfer coefficient to more than three times comparing to the cases of no vibration. Also, it was noted that if the amplitude to diameter ratio is more than 0.5, the heat transfer enhancement varies almost linearly with the amplitude to diameter ratio; irrespective of the applied frequency.

Similar to A. Dawood et al. [33], C. Cheng et al. [34] have examined the effect of transverse vibration on the convective heat transfer for a heated circular cylinder. In order to get some insight to the above, a modified transient and flow visualization techniques were employed to predict the heat transfer coefficient and
visualize the flow pattern for the case above. Suction type wind tunnel was used to test the heated cylinder that was made of copper, to ensure homogenous temperature distribution, and heated by electrical heating resistance. The temperature was monitored using K-type thermocouples placed at the downstream so that effects of the disturbance due to the incoming flow can be minimized. In order to facilitate the flow visualization, smoke was generated by vaporizing kerosene oil. It was reported that enhancement in the heat transfer had reached 34% comparing to the baseline case with no vibration. Also, it was mentioned that the Nusselt number is dependent on the oscillations’ Reynolds number, amplitude and frequency; generally increases with their increase.

S. Dey and D. Chakrborty [35] have numerically investigated the effect of vibrating fins on the convective heat transfer enhancement. In brief, the problem formulation was done for a fin placed on top of a copper substrate heated from below at a constant flux of 20 W. The flow was assumed to be incompressible and laminar. In addition, the effect of body forces was neglected. The study had addressed both steady state and transient analysis. The simulation was carried out till a grid independent solution was achieved. In order to form baseline experiments for static fin testing, various Reynolds number flows were subjected to flow against the fin. The studied parameters of the work were: Reynolds number of the flow as well as amplitude and frequency of the actuated fin. The results have shown that thermal boundary layer is developed for the cases where no oscillations were employed, which acts as a thermal barrier for heat dissipation. However, when the fins are oscillated, such boundary layers are destroyed and an improvement in the heat transfer is obtained. The authors had shown that for both amplitude and frequency to a certain Reynolds number, there is a threshold value above which the enhancement
in the Nusselt number is observed. Finally, in a preliminary testing of multiple fins, it was concluded that the effect of the heat transfer enhancement from the fins that are far from the edges of the heated surface is more compared to the fins placed at the edges.

L. Zhang et al [36] have examined the effect of vibrating nanofluid on the convective heat transfer. At their study, the authors had designed their experiment such that the fluid is passed through a circular tube at which 400 Watt resistive wires are twined. In order to avoid any electrical issues related to leakage, for example, and to ensure uniform heating, the authors had placed a layer of mica sheet between the wires and the tube. The testing pipe was insulated from outside so that cooling due to natural convection is minimal. It should be noted that the heat transfer assessment was done based on temperature recording through T-type thermocouples connected to data acquisition. The system studied was in closed loop. As a benchmark, tests involving nanofluid effect without vibration and tests involving only pure water but with vibration, were done to assess the combined effect of vibrating the nanofluid. It was found that the incorporation of nanoparticles had enhanced the convective heat transfer by the Brownian motion, thermophoresis, and thermal diffusion. The enhancement increases by increasing the concentration of the nanoparticles. As far as the results of the vibrating water tests are concerned, it was noted that the effect of pure water vibration is dependent on the Reynolds number; when the Reynolds number increases, the effect of vibration decreases. Finally, the effect of combined techniques revealed further enhancements compared to the baseline experiments. This behavior was attributed to the slippage of nanoparticles, which are denser than water, and the enhancement of the radial inertial forces of the nanoparticles in addition to the ability of the nanoparticles to destroy the thermal boundary layer
developed when passing the fluid in the circular tube, which enhances the turbulent intensity.

A. Pilli et al. [32] like K. K. Prasad and V. Ramanathan [31] have studied the effect of vibrating a vertically mounted vibrating heated plate on enhancing the convective heat transfer; however, this study was done numerically using Large Eddy Simulations. It should be noted that the mass, momentum and energy equations were solved using ANSYS-FLUENT. The controlling factors of this study were the frequency and amplitude of the applied oscillation. The studied frequencies of the vibration signals were 5, 50 and 100 Hz; however, the studied amplitudes were ranging from 0.5 to 2.0 mm. All the experiments were conducted at constant flux of 400 W/m$^2$ heated plate. It was shown that the thermal boundary layer is dependent of the applied signal. Low frequency signals, 5Hz, almost had no effect on the thermal boundary when compared to the static testing. Also, it was reported that 107% increase in the heat transfer rate is achieved at 100 Hz and 2.0 mm amplitude. Finally, for the range of the studied parameters, it was concluded that the Nusselt number is dependent on the vibrational Reynolds number.

J. L. Lage and A. Bejan [37] have conducted both theoretical and numerical studies on two-dimensional square enclosure to examine the resonance frequency of the system. The analyzed problem of the investigated enclosure had one side maintained at cold temperature, while the other side was subjected to a pulsating heat flux. The authors concluded that the bouncy driven flow tends to resonate at a frequency similar to that supplied by the pulsating heating. This phenomenon occurred at relatively high Rayleigh number, when the convection heat transfer is dominating the heat transfer process.
1.3.2.2 Heat Transfer Enhancement for Phase Change Materials (PCMs)

Recall from previous sections that the heat transfer enhancement in the case of PCMs results in facilitating the propagation of the melting layer during phase change processes; hence, increasing the exploitation of the latent heat content therein. If such materials (when they are subjected to vibrational signals) were used in a heat sink, higher heat removals from the hot source will be achieved. In all studies addressed below, the hot source (e.g. heated plates) was allowed to transfer heat to a heat sink enclosure containing PCM. Unlike the case of objects with no phase change, section 1.3.2.2.1, most of the applied vibrational signals for PCMs were in the range of ultrasound as will be seen below.

L. Quan et al. [38] have experimentally studied the effect of vibration on the phase change process of n-octadecane in rectangular enclosures. The study aimed to investigate the effect of the aspect ratio, height to width, vibration parameters namely amplitude and frequency (up to 37.5 mm and 60 Hz respectively), and vibration direction, horizontal or vertical, on the melting enhancement of the phase change material. In order to achieve the above objectives, the phase change material was filled in an aluminum rectangular enclosure which is 1.0 cm thick, so that flow passages can be drilled to pass water to ensure isothermal conditions. The instantaneous melted layer was approximated using the images of a video-tape camera that were grabbed and digitalized to calculate the melted area. The result was compared with the melted portion of the material at the end of the experiment and 6% error was obtained. The authors have concluded that the aspect ratio is an important parameter to be considered in such situations; having aspect ratio of 1.0 reveals lowest melting enhancement. Also, it was concluded that the vibrational
effect is evident in increasing the melted PCM in all cases (e.g. possibly up to 95% comparing to the stationary testing). In addition, it was mentioned that the melting enhancement is proportionally depended on the acceleration of the applied vibrational signal. Finally, the horizontal oscillations were more effective that the vertical ones. In a later study [39], L. Quan has done similar analysis to ice instead of n-octadecane and have reached similar conclusions above; however, the melting enhancement of ice had reached 170% when comparing to the 95% obtained when n-octadecane had been used.

Similar to L. Quan et al. [38] however, in the range of ultrasound vibrational signals, K. Choi and J. Hong [40] have experimentally studied the effect of ultrasound vibration on the melting process of n-octadecane in rectangular vessel. The PCM was placed on top of a heated plate supplying constant heat flux in an insulated rectangular vessel open to atmosphere from one side. The heat transfer rate, solid-liquid interface and three dimensional temperature distribution were investigated when subjecting the PCM to various vibrational frequencies (15, 25, 35, 50 and 55 kHz). In order to serve the understanding of the above parameters, several thermocouples were placed in the PCM and the walls in three dimensional arrangements. In addition, aluminum probes, finely scaled and marked, were used to observe the solid-liquid interface. It should be noted that the experiments for interface observation and temperature recordings were done separately/independently. The results revealed that the application of 50 kHz signal can increase the melted amount of PCM to 1.8 times the amount obtained when no vibration signals were implemented. This is attributed to the agitation and acoustic streaming of the ultrasound signal. Moreover, it was concluded that the melting amount is proportional to frequency.
For the same purpose, M. Oka [41] has studied the effect of vertical oscillating vibration signals on the melting enhancement of phase change materials. The problem investigated at this study involved peripheral heating of a capsule filled with phase change material and exposed to vibration signals. The model was developed based on mass balance, linear momentum, and energy balance for Navier-Stokes-Fourier equations with the aid of the no-slip and thermal boundary condition. For simplicity, the model was approximated by two dimensional flat heating wall in cylindrical coordinates. The equations were nondimensionalized and approximated to simplified versions and a similarity solution was employed to solve them. The author’s general conclusion was that the heat transfer enhancement is evident when the heated walls are vibrated.

Y. Oh et al. [42] have modified the experimental setup mentioned in [40] to understand the effect of the ultrasound vibration on the melting process of the PCM. The authors have placed the setup in a water bath so that problems of electric overloading, that could occur otherwise, are avoided. It is worth mentioning that the vibratos were placed below the water tank. Also, a visualization window was adopted for the phase transition observance. It should be noted that the testing cavity was well insulated to ensure that the observation noted are only due to the effect of the heater and ultrasound vibration (avoiding any natural convection effects from the sides). Particle flow image velocimetry (PIV), to display the velocity field vector of the melted region, infrared thermal camera, to display the thermal oscillating flow, and thermocouples connected to data acquisition, to measure the temperature history of the PCM, were used. It was summed up that the usage of ultrasound vibration may accelerate the melting process upto 2.5 times comparing to cases without vibration (natural melting). This was attributed to the acoustic streaming and cavitation
resulting in thermal oscillating flow that had enhanced the heat transfer coefficients by introducing some flow instabilities that is able to destroy the diffusion-dominant melting mode from the heated plate. The authors have touched on the comparison of the consumed electricity in cases of with and without using the ultrasound, and they have concluded that the total consumed electricity is almost the same making the use of vibration favorable in applications where time matters.

Later in H. D. Yang and Y. K. Oh [43], the authors have done a numerical analysis so that the study of the acoustic pressure effect can be analyzed and referred to the acoustic streaming and cavitation mentioned in Y. Oh et.al. [42] It should be noted that in this study, the authors have used only two ultrasound vibrators and similar procedure in [42] was followed for the analysis. The model was developed using coupled FE-BEM (Finite Element-Boundary Element Method), a method to solve vibrational problems in infinite space. The study was conducted on SVS (Structural Vibration Simulator) programmed with Fortran language. The results were interpreted the same way as in the previous study [42]; however, in this study a parameter called “heat transfer enhancement ratio”, which is the ratio of the enhanced convective heat transfer to the natural one, was introduced and related to the temperature distribution of the PCM. The authors were successful in verifying the numerical results against the experimental ones, hence, the effect of the acoustic streaming to the thermal oscillating phenomenon as mentioned earlier. It was concluded that the high pressure imposed by the ultrasonic wave is responsible for developing intensive flows that resulting in flow instabilities and relative motions. Later. In the same year, the same authors of [43], have done similar analysis in [44]; however, at this study the authors have used four vibratos instead of two and have mentioned that numerical analysis was conducted only due to the difficulty of using
hydrophone to measure the acoustic pressure relating it to the elevated temperature the PCM can reach.

A year later, the authors of [43] and [44] have published [45] to clarify the flow visualization of the liquid part of the PCM using a thermal camera. It was mentioned that the thermal camera was placed vertically at irradiant cross section of the visualization window. In addition, laser illumination has been used and mounted vertically using Argon-Ion Laser and the obtained images were sent and analyzed on image board and personal computer. The authors have mentioned the basic classification of the acoustic streaming, Rayleigh and Eckart, and referred the phenomenon observed at the study to quasi-Eckart type. Similar conclusion drawn at the previous studies [43] and [44] were also mentioned.

As highlighted above, and up to the author’s knowledge, the work done in this area covers convective heat transfer enhancement either for hot very high-melting point solids, to cool the heated object, or relatively low melting point phase change materials, to study the phase change propagation. Hence, it is proposed for the first time, at this thesis work, to integrate the concepts of the two areas above to cool hot fluids and place them into direct contact with gallium while vibrating the whole system for a range of amplitudes and frequencies. This will facilitate the heat transfer process from the hot fluid to the gallium and result in increasing the melting wave at which much of the heat content of the hot fluid will be absorbed by the latent heat content of gallium. Detailed description of the system and the results of the study can be found in the experimental setup as well as results and discussion sections respectively.
1.3.2.3 Discretizing the Hot Fluid into Bubbles for Heat Transfer Enhancement

Bubbling is another active technique used to enhance the convective heat transfer by introducing vigorous relative motion between the hot fluid and the heat sink media. The idea implemented here allows the hot fluid to be discretized into small water bubbles so that the heat transfer rates can be maximized due to the influence of the increase in the convective heat transfer area and the dynamic motion between the hot fluid and heat sink media. In most of the literature found, air bubbles were injected into water at which the heat transfer process is of a significance. This concept is recently applied to shell and tube heat exchangers and their results are appealing. It should be noted that papers addressing the dynamics of bubbles’ motion were also evident in the literature; however, they were not included in this work since it is out of its scope. Following is a summary of the literature found.

H. Ghazi [46] has experimentally studied the effect of direct contact heat transfer of air bubbled in stagnant hot water tank. Room temperature air was injected via orifice placed at the bottom of a hot water filled tank. The hot water was heated by placing two electric heaters in the container. It should be noted that the testing setup was insulated so that changes in temperature can be only attributed to the effect of bubbling. The studied parameters were: the hot water tank temperature (e.g. 70 and 80 °C) as well as air injecting flow rate (e.g. 10 different rates were tested) and size. It was noted that an increase of 100 to 200% in the air bubbles’ temperature were observed. Also, it was mentioned that increasing the air flow rate and the hot water tank temperature would results in increasing the bubbles’ temperature. Moreover, when the size of the bubbles is reduced, higher heat transfer rates are achieved. Finally, in the studied parameters range, the authors were successful in
correlating Nusselt number of the above phenomenon to dimensionless quantities as Reynolds and Prandtl numbers.

Later, H. Inaba et al. [47] have conducted a relatively similar analysis as H. Ghazi [46]; however, this study was more elaborative and investigated more studied parameters that includes: the air inlet temperature and flow as well as the hot water tank temperature and level. The air temperature and humidity were measured before and after passing through the hot water so that conclusions related to the heat transfer mechanism can be drawn. It was shown that the bubbles diameter increase when the air velocity increases. In addition, it was noted that the bubbles’ temperature was independent of the hot water levels that were studied. That was attributed to the very short time of heat transfer near the bubbles’ injection place after which the air bubbles reach the equilibrium state with the hot water. Finally, a correlation for the non-dimensional correlations in terms of Reynolds, Prandtl and Schmidt numbers were developed for the above phenomenon in the studied parameters ranges.

A. Nandan and G. Singh [48] have applied the concept of bubbling in Shell and Tube heat exchanger in order to study its effect on the overall heat transfer enhancement. The experimental setup consisted from three parts namely test section, hot water loop and cold water loop. In all experiments, hot water was allowed to flow from 1.0 to 3.5 liter per minute (LPM) in steps of 0.5 LPM; however, constant flow rate was applied to the cold water. Air, at 15 °C, was injected at 0.05833 kg/s using aquarium pump. It should be noted that in the experiments involving air bubbles, it is either that the air was added at the entrance section or throughout the tubes. Four sets of experiments were done to assess the heat transfer performance namely baseline experiments without bubbling, air bubbles at the entrance of the cold water, air
bubbles at the entrance of the hot water and air bubbles throughout the cold water tube. It was concluded that the maximum enhancement of the heat transfer coefficient and Nusselt number occurred when the air bubbles were injected throughout the cold water tube followed by the air injection at the entrance of the cold tube and at the entrance of the hot water respectively. Also, it was noted that both the heat transfer coefficient and Nusselt number are dependent on the Reynolds number that was varied from 6600 to 23000.

Following the work done in [48], G. Thakuri and G. Singh [49] have done more elaborate study on the effect of air bubbling on the heat transfer enhancement when Al₂O₃ cold water based nanofluid was allowed to circulate in tubes section of the heat exchanger. Two volumetric concentration of the nanofluid were prepared for the study; 0.1% and 0.2%. The hot water temperature was varied from 30 °C to 60 °C at a step of 10 °C. The operating flow rates as well as the air injection technique were the same as in the previous study [48]. Three sets of experiments were conducted at this study; (1) without bubbling, (2) bubbling at the entrance of the cold tube and (3) bubbling throughout the bubbling tube. The effect of the above experiments was studied based on the heat transfer enhancement and the overall effectiveness of the heat exchanger. The results have shown that the heat transfer enhancement is linearly dependent on the nanofluid concentration; hence, 0.2% volume fraction had showed better results than 0.1% volume fraction. Also, it was noted that increasing the hot water temperature was found to enhance the heat transfer rate. Finally, as in the previous study [48], the maximum enhancement of the heat transfer coefficient and the overall effectiveness was observed when the air bubbles were injected throughout the cold tube.
H. Tokunaga et al. [50] has studied the heat transfer rate between bubbles and still molten wood’ metal. Nitrogen bubbles at 18 °C were injected from a single hole nozzle placed at the bottom of a cylinder filled with molten wood’s metal at 105 °C., kept constant using electric heaters. It should be noted that the melting temperature of the wood’ metal is 47 °C. In order to assess the heat transfer performance, the temperature, velocity and diameter of the bubbles were measured. Thermocouples were placed at different locations of the experimental setup to measure the temperature history of the raising bubbles. However, two-needle electro-resistivity probe was used to measure the velocity and diameter. The authors had pointed that the radiation heat transfer in such situation is negligible compared to the convection mode. Interestingly, it was shown that the Nusselt number’s correlation between air and water could be applied to the case of nitrogen and molten wood’s metal.

R. Winterton [51] has theoretically investigated the heat transfer effect of placing somehow adhering gas bubbles along the inner wall of a circular constant flux tube at which liquid metal is traveling. In order to solve the above problem, it was assumed that the axial conduction is neglected and the fluid properties are constant. The study had shown that under wetting condition, contact angle less than 60%, the Nusselt number could be reduced to 50% under low Peclet number. However, at high Peclet number, when the flow is high, few bubbles from the ones adhering at the surface can withstand the drag force exerted by the flow, in this case, the heat transfer might not be affected. Also, the authors had noted that further reduction in Nusselt number might be obtained when non-wetting conditions are applied.
As noted above, most of the papers found in the literature has addressed discretizing air flow in conventional cooling mediums as water. However, very limited work has been published in involving the technique done in this thesis; namely discretizing the to-be-cooled hot fluid (e.g. hot water) flow into small bubbles in a still cooler liquid gallium bath (e.g. heat sink).

Also, the work presented here takes care of the limited specific heat of gallium which has been mentioned previously. It is expected that after some time of exchanging heat between the hot water and the cooler liquid gallium, the gallium will be heated up and the cooling deriving force between the two streams will be diminishing and vanishing after reaching the equilibrium state. Hence, two cooling techniques for the heated gallium were proposed; one is to integrate cooling tubes to the container that hosts the gallium, which will be suitable for continues operations, and the other is to pump cold water through the same bubbling tubes from which the hot water was bubbled, which is suitable for batch operations. More information and further explanations on the designed heat sinks and their performance will be given in the experimental setup as well as the results and discussion.
Chapter 2: Experimental Setup and Mathematical Formulations

In this chapter, the experimental setup for the three techniques implemented in the thesis work will be mentioned. In addition, some mathematical formulations, especially for the bubbling technique subsection, will be illustrated. In all of the techniques adopted at this study, temperature measurements were recorded using K-type Omega’s thermocouples (accuracy ±1.0 °C) and the data were sent to a data acquisition, which is connected to a personal computer via LabVIEW software.

2.1 Gallium Heat Sink Integrating Un-encapsulated Discrete Inorganic PCM Chunks

In this section, the experimental setup and approach adopted for designing gallium heat sink integrated with inorganic PCM chunks are described. The testing heat sink apparatus is a cylindrical container made of plastic with its bottom base made of brass. Plastic was selected to minimize heat loss from the side walls of the sink and to restrict it to be mainly through the bottom metallic brass base of the container. The heat sink container is placed on top of a large water reservoir that is kept close to room temperature such that the brass base of the sink touches the surface of that water reservoir allowing thereby the heat to easily escape to the outer surroundings through the base. Several K-type thermocouples (accuracy ±1.0 °C) were used to measure the temperatures that are of relevance to this study. The collected data were sent to a computer through a data acquisition card using LabVIEW software. The frequency of data collection is 1000 Hz. Different experimental runs were conducted. In some of them hot water alone was poured in the test cylindrical container and allowed to cool down with time. In other runs, the amount of solid gallium used (without using PCM) was placed in the cylindrical
container and the hot water was poured on top of the gallium and allowed to cool down while being in direct contact with the gallium. It should be noted that the runs that do not include gallium; (i.e. those with only hot water cooling down in the cylinder), are considered as reference baseline cases to compare the other cases with. Moreover, the runs that include only gallium, without PCM, would serve as means to assess the role played by the PCM.

In the rest of the experimental cases, which are the main focus of this section, cavities filled with PCM are made within the lower half of the gallium’s body in the sink. In these cases, the hot water is poured on top of the gallium sink and allowed to exchange heat directly with the upper surface of the gallium, which is far enough from the PCM cavities. The cooling of the hot water in these cases is monitored over time and eventually compared with the cooling history of the water in the other cases that did not include PCM cavities.

Some of the heat removed from the to-be-cooled hot water source; for the cases with PCM, will be stored in the PCM cavities within the gallium. This way, the role played by the PCM cavities and the effect of the cavities distribution and structure within the solid gallium on the sink’s performance can better be appreciated. Indeed, simple heat transfer calculations made based on the operating temperatures considered in this work can easily reveal the insignificance of the amount of heat that might be escaping from the side walls to the outer environment. The extent to which heat is absorbed by the PCM in the cavities will decide the temperature levels attained within the gallium body and hence the overall rate of heat removal from the hot water source. This later aspect is the main focus of this study.
and the results obtained will help making judgment on the role played by the PCM on the sink’s performance.

The PCM cavities (as will be seen below) can be in the form of structured cylindrical holes drilled within the solid gallium, but also can be of any other shape. The PCM filling the cavities will be in direct physical contact with the cavities’ walls which basically are part of the gallium’s body. It should be noted and reemphasized that in the cases that include PCM cavities there is no need for any encapsulation shells to hold the PCM in the cavities as the region of gallium enclosing the cavities will be assured not to reach the melting state during the process of dumping heat into the sink. This point, which is one of the main strengths of the sink concept we are proposing hereby, will be further elaborated on in the subsequent section.

Two approaches to prepare the PCM cavities within the solid gallium sink were adopted. The first uses structured well defined pattern cavities in the form of cylindrical holes drilled within the lower half of the solid gallium body (see Figure 2.1.1(a)). In the second approach (see Figure 2.1.1(b)) the sink container was filled first with half of the total intended gallium amount. This gallium was ensured to be in the solid state at a lower temperature than the gallium’s melting point of about 29°C. Chunks of PCM in its solid state with different sizes were then randomly scattered in a layer on the top surface of the gallium. Subsequently, small pieces of solid gallium were placed in between and on top of the PCM chunks to hold them in place and at the same time to assure the presence of enough free gallium passageways interconnecting the different parts of the gallium’s body throughout the whole container; below and above that PCM layer. These gallium passageways are intended to ensure interconnectivity and smooth heat transfer throughout all gallium
parts in the sink both above and below the PCM cavities, hence effective cooling of
the hot water that will be poured on top of the gallium-PCM sink matrix. Liquid
gallium at slightly above its freezing point is then poured on top of the gallium pieces
covering the PCM chunks until the total amount of gallium needed in our
experiments is reached. In some of the randomly distributed PCM cases, double
layers of PCM were made to increase the volume fraction of PCM in the gallium and
hence give the gallium sink ample of margin to accommodate in the cavities more
heat from the hot source without having the gallium body experiencing much
temperature rise.

Figure 2.1.1: PCM-gallium matrix configurations: (a) structured arrangement; (b) random arrangement

Extra additional precautions while covering the PCM chunks with gallium
were taken to ensure as possible instant freezing of the gallium upon being poured on
top of the small solid gallium pieces covering the PCM chunks. For example, the
amount of gallium body below the PCM layer (which represents half of the total
gallium amount to be present in the sink, as mentioned above) was cooled to
sufficiently low temperatures (close to 5°C) before placing the PCM on top of it.
Recall that the melting point of the PCM used is ~23 °C and that of gallium is ~29 °C. Hence; due to the high thermal conductivity of gallium, it becomes much easier and controllable when pouring the rest of the gallium into the sink to get it solidified quickly to form a solid and firm ceiling cap above the cavities.

In the PCM cases, to start the tests while the PCM being in the solid phase, it has been decided to cool down the whole sink matrix to temperatures below the melting point of the PCM. Hence, the initial starting temperature for the whole gallium/PCM matrix was selected to be around 15 °C in all runs conducted; including the tests that include gallium but with no PCM cavities. Figure 2.1.2 shows a schematic drawing for the different configuration options of the gallium-PCM matrix considered. The amount of gallium used in all runs (both which have PCM within the gallium and those that have only gallium) is 120 ml. The different tests were conducted for two amounts of the hot water poured into the sink namely 40 ml, and 60 ml.

![Schematic drawing of different PCM-gallium matrix configurations](image)

Figure 2.1.2: Schematic drawing of different PCM-gallium matrix configurations studied: (a) Structured cylindrical holes; (b) Single layer random distribution; (c) Double layer random distribution
It should be recalled that in the cases with PCM cavities, the solid gallium layer placed as a cap that seals the cavities is chosen to be thick enough in order not to have any possibility of complete melting of gallium that would then cause cavities rupturing. As will be elaborated on more closely in the subsequent section with supporting basic calculations, the maximum depth to which melting may take place under extreme conditions of the conducted experiments is about 3.3 mm, which is clearly less than the thickness of the solid gallium layer placed on top of the cavities in our experimental runs. Figure 2.1.2 also shows a schematic of the mushy melted gallium region that separates the hot to-be-cooled water from the solid gallium beneath it. Further discussions on these aspects will be provided in the subsequent sections.

An overall schematic of the whole experimental setup with temperature instrumentations and the adopted testing process is shown in Figure 2.1.3. As noted, the thermocouples were placed in the following locations: (#1) cold water reservoir below the base of the gallium sink, (#2) hot water source sitting above the gallium sink representing the hot source in this study, (#3) solid gallium body, and (#4) gallium-water interface viewed from the gallium side. The above numbering of the thermocouple locations is also given in Figure 2.1.3. It should be noted that for each run, three thermocouples were placed in the hot water above gallium and the average value was taken. A closer look at the experimental cylinder in Figure 2.1.3 for the cases with structured PCM holes arrangement is shown in Figure 2.1.4 (a and b). The figure shows exact dimensions in millimeters for the container including the brass base and the gallium in the cylinder, the PCM holes and the locations of the thermocouples tips with the cover of the sink cylinder being the reference of measurement for both hot water amounts (i.e. 40 ml and 60 ml). The numbering of
the thermocouples’ locations shown in Figure 2.1.4(b) is in accordance with the description given above and in Figure 2.1.3.

Figure 2.1.3: Schematic of the Experimental Setup and the Used Instrumentations for PCM Experiments
Figure 2.1.4: CATIA 2D, side and top views, drawing of the experimental gallium/PCM-sink cylinder with structured cylindrical holes PCM cavities.

(a) 40 ml water, (b) 60 ml water

Table 2.1.1 summarizes the thermophysical properties of the used PCM; which is inorganic mixture of hydrated salts; PCM-HS22P, obtained from RGEES, LLC, whereas Table 2.1.2 lists the thermophysical properties of gallium.
Table 2.1.1: Thermophysical Properties of Used Inorganic PCM

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melting Temperature</td>
<td>23</td>
<td>°C</td>
</tr>
<tr>
<td>Freezing Temperature</td>
<td>22</td>
<td>°C</td>
</tr>
<tr>
<td>Liquid Density</td>
<td>1540</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Solid Density</td>
<td>1840</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Latent Heat of Fusion</td>
<td>185</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>Specific Heat: Liquid</td>
<td>1.05</td>
<td>kJ/kg.K</td>
</tr>
<tr>
<td>Liquid Thermal Conductivity</td>
<td>0.54</td>
<td>W/m.K</td>
</tr>
</tbody>
</table>

Table 2.1.2: Thermophysical Properties of Gallium

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
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<tbody>
<tr>
<td>Melting Temperature</td>
<td>29.76</td>
<td>°C</td>
</tr>
<tr>
<td>Boiling Temperature</td>
<td>2204</td>
<td>°C</td>
</tr>
<tr>
<td>Density</td>
<td>5.904</td>
<td>g/cm³</td>
</tr>
<tr>
<td>Latent Heat of Fusion</td>
<td>5.59</td>
<td>kJ/mol</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>371</td>
<td>J/kg.K</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>29</td>
<td>W/m.K</td>
</tr>
</tbody>
</table>
Some Basic Considerations Regarding the Proposed Gallium/PCM heat sink

Different kinds of materials can be used for the proposed heat sink. Important in this regard is that these materials will be receiving heat from the to-be-cooled hot source and thereby experience only a partial melting as a result of that heat. Using solid gallium in this research is just a sample case study intended to illustrate and validate the proposed concept, but using other high thermal conductivity metals such as copper, aluminum, etc. is also believed to be possible. However, in these later cases of high melting temperature metals the hot source and consequently the fabricated heat sink will be operating at temperatures above the melting points of these metals; much higher than the temperatures encountered in this research with a gallium-based sink. Moreover, the kind of the adopted PCM in these high temperature cases and their required thermophysical properties will need to be decided to fit the operation under such high temperatures.

The selection of the proper sink material (whether solid gallium or other metals) therefore depends on the range of operating temperatures of the to-be-cooled hot sources and the melting point of the used sink material. For example, the use of solid gallium with its melting point being close to room temperature would be suitable for applications where the to-be-cooled hot sources can experience temperatures say two or three orders of magnitude above that melting point (i.e. up to about 80°C) which are of relevance for applications like electronics. Considering much higher temperatures for the to-be-cooled sources than this range with much heat dumping needs to the sink would then dictate the need for bulky heat sinks with large mass of gallium in order to avoid having the melting wave of the gallium in the sink reaching the upper ceiling of the PCM cavities. Otherwise, there would be a
possibility of the rupturing of the cavities leading to spillage of the PCM out of them which would ultimately ruin the sink altogether. As such, a higher melting point metal would be required for the sink in such situations of high temperature high flux operation. In the conducted experiment, the above arguments and considerations have been taken into account while choosing gallium and its used amounts as well as the total heat that will be removed from the hot water. It is worth mentioning that, should the proposed sink concept be put at a wide scale of commercial mass production, more dedicated control systems supporting the operation of the proposed sink would then be included to ensure that the melting interface would not be allowed to reach the PCM cavities. Further discussions on this matter, however, are out of the scope of the present research.

There is quite significant evidence from previous researches reported in the literature [4-8] that indicate the superiority of heat transfer rates between immiscible liquids exchanging heat directly without interfacial solid barrier between them (e.g. tube wall or a solid metallic plate). In those papers, it was indicated that heat transfer would be clearly enhanced if we let a liquid metal like liquid gallium which has clearly a high thermal conductivity to act as a sink and receive heat from hot water that is placed in direct contact with it. However, irrespective of the superior thermal conductivity of gallium, its limited specific heat capacity can easily offset this favorable feature. In other words, it will leave the gallium sink ineffective after receiving little heat from the source. Thereby, the whole process would cease out quickly after the gallium reaches thermal equilibrium with the source.

To overcome the above limitation of low specific heat, we thought of using solid gallium instead of the liquid gallium. This would imply having the gallium
material in the sink to be initially (before dumping heat from the source into it) at a
temperature below gallium’s melting point of around 29 °C. Upon pouring the hot
water on top of the solid gallium block, the heat transfer would start taking place
rapidly between the hot water and the initially-solid gallium interface. When the
upper gallium layer reaches the gallium’s melting temperature it will be ready to start
phase change to liquid. Meanwhile, the heat transfer from the resulting mushy
gallium layer would continue to the bottom parts of the gallium that are still in the
solid state. As such, a competition between heat gain and heat loss from this melted
mushy gallium layer would decide the overall amount of gallium’s phase change
taking place. The high thermal conductivity of gallium would facilitate fast heat
transfer throughout the gallium body (both the liquid and solid parts) and once all
gallium body is heated to the melting temperature, the heat transfer from the hot
water to the gallium would then be solely utilized to melt gallium. Moreover, the
high thermal conductivity of gallium would try to smooth out any temperature
stratification within the gallium’s body. Yet, temperature gradients in the very thin
mushy layer interfacing the hot water and solid gallium (from the gallium side) are
still high to an extent that somehow will slow down heat removal from the hot water,
especially while having low specific heat. As will be seen in the results section, the
existence of PCM cavities within the gallium matrix; with their role acting as a sink
within the overall gallium sink, leads to lower temperature levels and to a less
temperature stratification in the very upper thin gallium’s interfacial layer.
Eventually, supported by gallium’s high thermal conductivity, this effect augments
further the homogenizing of the temperature distribution throughout the gallium
body. This effect is the main reason that differentiates the cases with PCM from the
ones without, where in the former cases this leads to faster rate of heat dumping into the sink.

It should be noted that the low specific heat of gallium would tend to cause appreciable temperature rise for the mushy melted gallium layer interfacing water with solid gallium. This effect will compete with the gallium phase change taking place which acts as a suppressor for temperature rise in that layer. Here comes into play the important role played by the PCM cavities proposed in this work that will act as a sink for the heat which will help avoiding any appreciable superheating and temperature rise in the gallium, most importantly in the upper part of the mushy region.

In addition to the above mentioned role played by the PCM cavities regarding controlling the temperature rise of the sink, by absorbing heat from the solid gallium next to them, they also restrict significant melting of solid gallium interfacing with the mushy region. Hence, it gives the whole gallium body a chance to last longer in its solid state while the sink is receiving heat from the hot source. Hence, this would assure longer operating time before marginal limits are reached upon dumping much heat into the sink that may cause the gallium’s melting wave to closely approach the solid gallium cap that seals the PCM cavities. Such a situation would then necessitate immediate cutting off of sink from the hot source to let it lose heat and get the melted gallium as well as the melted PCM in the cavities to re-solidify and restore their initial solid state once again for repeated operation of the sink to be possible.

Below are some basic representative calculations for the runs conducted in this work. It shows that melting of gallium under extreme conditions of the amounts of heat considered in the present runs would in no way reach the PCM cavities and
so the PCM cavities will always be intact and holding the PCM correctly, without needing to include extra supporting encapsulation shells.

Consider the case of 60 ml hot water undergoing in one of the conducted runs an overall temperature drop from approximately 70 °C to 30 °C (see results section) by giving its heat to the gallium in the sink below it. The latent heat of fusion of solid gallium is about 80 kJ/kg. The total amount of heat lost from the water that would be effecting the above water temperature drop by 40 °C, is about 10 kJ. By assuming that all of this heat would go from the water to the gallium and by considering the latent heat of fusion of gallium, the mass of gallium that would be melted as a result of that heat would be around 0.125 kg. By considering the sink dimensions used in the experiments (see Figure 2.1.4), this would mean a melted gallium thickness of about 3.33 mm.

Definitely not all of the 10 kJ removed from the above amount of hot water would exactly do only melting of gallium. Some of this heat will be used to raise the temperature of the gallium (sensible heating) for about 15 °C (the initial temperature of the sink at the beginning of the runs; see results section) to the melting point of around 29 °C. This kind of sensible heating would require about 4 kJ of heat. This means only 6 kJ of heat at max would remain and be available to do the subsequent melting for the solid gallium. This would be responsible for melting 0.075 kg of gallium which would correspond to a height of the melted layer in this case of about 2 mm. Once again, this is clearly smaller than the overall thickness of the solid gallium layer above the PCM cavities and clearly indicates that in the conducted experiments the chance for the gallium’s melting wave to cause any harm to the integrity of the PCM cavities is non-existing.
It should be noted further that the actual amount of melted gallium in our experiment would even be less than the above figures (i.e. less than a height of 2 mm) since some of the heat removed from the hot water would be captured by the PCM cavities. It should be recalled once more here that basic calculations can show that very minor amounts of heat will be escaping the sink through its other boundaries (other than the base) and these amounts would have a negligible effect on the phase change processes taking place inside the sink.

It should be noted further that the above scenarios of heat distribution among sensible and latent heat used for melting the gallium, as well as the heat captured by the PCM or that escaping to the surroundings will be taking place in parallel. Hence, it is not in the straightforward simplified sequence described above; in a way that is much more complex to be analyzed by the above simplified arguments. Yet, the above basic computations give hint and insight into the extreme scenarios regarding the melting of the gallium during the process. They also serve to illustrate the point we intended to highlight in regards to integrity of the PCM cavities and the fact that under certain proper precautions using encapsulation shells for the PCM in the cavities can easily be avoided leading thereby to better performance of the sink.

As for the option to use other metals as sink material other than gallium for example copper, it should be noted that copper has a specific heat capacity of 0.385 kJ/kg which is comparable to that of gallium but has much higher melting temperature than gallium. As we did with gallium, consider an equivalent mass of copper to replace the used mass of gallium in our experiments and consider the same initial starting temperature for copper of 15 °C. Attempting to let the hot water to dump 10 kJ of heat into the copper block; as was the case with gallium, this would
then mean that the copper block will experience a temperature rise by about 51 °C reaching thereby 66 °C. This ultimate temperature of copper is by far higher than the final temperature of the water upon losing 10 kJ namely 30 °C. This means, when using an equivalent copper block it would be impossible virtually to cool down the water to the limits that were possible when gallium was in use namely to 30 °C.

Of course the high thermal conductivity of copper would still facilitate passing part of the heat dumped into the copper block to the outer surroundings relatively quickly especially through the bottom base of the container. As a result, this would not allow for copper to reach the extreme limit of 66 °C mentioned above, so the to-be-cooled hot water contacting the copper block may still have the chance to cool down to low levels. Yet, the other enhancing features for heat transfer that are highlighted below for the case when gallium is used, will be missing in the case of a copper block which then would still suggest incompetency of a copper block to act as an alternative sink material for the ranges of temperature considered in this study, as compared with solid gallium.

Clearly, in regards to the above arguments, the advantageous role played by the low melting point of gallium and hence the latent heat absorbed by gallium in the mushy region during heat exchange with the hot water, is quite evident. In the case when gallium is used, the low melting point of gallium poses a threshold for the temperature of the mushy melted gallium layer close to about 30 °C, which guaranteed a continuous cooling process for the water until the water reaches desired temperatures as low as about 30 °C. Obviously, the high melting point of copper hindered this scenario from happening. Of course, to restrict temperature rise of the copper block and so still allow for further water cooling to as low as 30 °C, PCM
cavities can still be integrated within the copper block; the same way implemented in the case of the gallium. However, the high melting point of copper will still mean that there will be no existence of a mushy melted interface between the hot water and the copper block (unlike the mushy layer formed in the case of gallium). Hence, the advantages of enhanced heat transfer between two immiscible liquids in direct contact (as evidenced by previous research studies) would not be present when a metal such as copper is used to replace gallium under the operating temperatures considered in this study. All the above arguments are in favor of and support the use of gallium for our heat sink design when operating temperatures of the to-be-cooled hot source are, as described in the results section, in the range of 70 to 80 °C. These ranges can be of significant importance for applications like electronics, as highlighted earlier.

With the above being said, metals such as copper, that has a melting point above 1000 °C and with its almost 2.6 times higher latent heat of fusion than gallium, can still be used with advantage as described in the current study but for hot source operating temperatures that are above the melting point of copper.

For the cases that include PCM cavities, the interface temperatures reported in the results section are around 30 °C for most of the test time. This is still slightly above or at least at the melting temperature of gallium. It should be noted anyway that the tip of the thermocouples that is immersed in the gallium interface region is dipped about 2 to 3 mm below the water surface. When the measurements at the thermocouple location are ~29 °C or above, then this is a sure evidence that the gallium layer below it is in a melted state or at least that it is a solid gallium that is ready to melt when further heat is dumped into it.
For the cases that do not include PCM but only gallium, the gallium in the interface mushy region; due to some superheating attributed to the low specific heat capacity, attains higher temperatures above the melting point and so the temperature would rise to a point that will adversely affect heat transfer from the hot water to the gallium. Here the role played by PCM in moderating the temperature of the melted layer region and so maintaining higher driving temperature difference for heat transfer between the water and the gallium can be further appreciated.

2.2 The Application of Vibrational Signals to the Heat Sink

In this section, the experimental setup as well as the approach for the vibration experiments will be mentioned. In all experiments, a 100 ml of hot water was poured on top of fixed amount of solid gallium (e.g. 120 ml) either at a static condition or at specified amplitude and frequency vibration signals. The gallium and hot water were placed in a mild steel cylindrical cup of diameter 8.7 cm and height of 13.2 cm. The process was monitored by placing K-type omega thermocouples (accuracy = ±1.0 °C) at the locations of interest and the recorded data at a sampling rate of nearly one second were saved to a computer through data acquisition (DAQ).

In order to excite the system to vibrational signals, B&K LDS shaker was used. The device consists of different components namely field generator (to supply the shaker with the power needed for operation), power amplifier (to amplify the delivered input signals), controller/DAQ (to control the displaced motion and save the data), fan (to cool down the shaker) and two accelerometers, input and output, (to deliver the input signal and record the output one). The process starts by entering the input signal through a desktop computer via Shaker Control software. The signal is then passed to the controller, power amplifier and finally to the shaker. Then, the
input accelerometer gives a feedback signal to the controller to make the necessary adjustments and pass the signal to the power amplifier in a loop. The output accelerometer sends the output data to the DAQ in order to be saved in the computer. A simplified illustration of the above process including the names of the shaker’s components as well as a representation of the experimental testing can be shown in the schematic diagram found in Figure 2.2.1. It should be noted that the frequency of data collection is 1000 Hz.

<table>
<thead>
<tr>
<th>Component Letter</th>
<th>Name</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>Desktop Computer – Shaker’s Software: Input Vibrational Signal</td>
</tr>
<tr>
<td>b</td>
<td>Controller/DAQ</td>
</tr>
<tr>
<td>c</td>
<td>Power Amplifier</td>
</tr>
<tr>
<td>d</td>
<td>field generator</td>
</tr>
<tr>
<td>e</td>
<td>fan</td>
</tr>
<tr>
<td>f</td>
<td>shaker</td>
</tr>
<tr>
<td>g</td>
<td>Accelerometer</td>
</tr>
<tr>
<td>h</td>
<td>Desktop Computer – Temperature Data Collection</td>
</tr>
</tbody>
</table>

Figure 2.2.1: Schematic Diagram of the Vibration Testing Process
In order to ensure firm fixture of the testing setup and to accommodate for its occupied space, tapered Aluminum (Al) plate, small diameter = 15.5 cm (see Figure 2.2.2(b) and (c)); big diameter = 25.5 cm (see Figure 2.2.2(a)), was manufactured and nine holes were drilled into it so that bolts and accelerometers can be placed on the shaker’s base. Additional five holes were drilled in the Al plate to fix a 16.5 cm cylinder at which the 8.7 cm cylinder, the one that hosts the gallium, was placed (see Figure 2.2.2(a)). It is worth mentioning that the bigger cylinder was added to avoid any water leakage from the smaller cylinder or water spoilage while pouring the hot water on top of gallium. In order to achieve the above successfully, the base of the bigger cylinder was manufactured to be thick enough, 7.0 cm, to drill four holes so that the small cylinder can be firmly fixed. It should be noted that precautions such as the drilled holes’ depth in both the Al plate and the big cylinder were taken into account in order to avoid any water leakage to the shaker’s basement that might damage the device (e.g. they were not drilled through).

Figure 2.2.2: Design Features of the Vibrating System (a) and (b) Top and Back Views of the Manufactured Tapered Al Plate respectively, (c) Shaker’s Base Fixture

As depicted from Figure 2.2.1, the thermocouples were given numbers indicating the locations at which they were placed; (#1) Middle of the Gallium, (#2) Below the interface between the gallium and hot water (interface Up). (#3) Middle of the hot water. Three thermocouples were placed in water so that the average
temperature could be taken to minimize the experimental errors. Table 2.2.1 shows the locations of the thermocouples’ tips measured from the upper edge of the testing cup.

Table 2.2.1: Thermocouples’ Locations for Vibration Experiments

<table>
<thead>
<tr>
<th>Thermocouple #</th>
<th>Measured Heights (cm)</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12.2</td>
<td>Gallium</td>
</tr>
<tr>
<td>2</td>
<td>11.3</td>
<td>Hot Water-Gallium Interface</td>
</tr>
<tr>
<td>3</td>
<td>10.3</td>
<td>Hot Water</td>
</tr>
</tbody>
</table>

In this study, vertical sinusoidal vibrational signals were applied to the heat sink. Three levels of frequency (20, 35, and 50 Hz) and four values of the amplitudes (0.3, 0.5 and 0.7 mm) were adopted in the experimental runs. It should be noted that the amplitude values represent peak-to-peak displacement. Prior any experiment, the testing cup was placed on ice bath to reduce the gallium’s temperature below its melting point to 5 °C. Then, the gallium was allowed to heat up naturally, under ambient conditions of 23 °C, to 15 °C; the temperature at which the recording of the data was started. Each test was repeated three times and the maximum standard deviation for the hot water and gallium measurements were 0.2 °C and 0.18 °C respectively.

2.3 The Application of Bubbling Technique

2.3.1 Experimental Setup

Two designs for the experimental setup for the bubbling technique have been tested. In the first, the system consists of a main bubbling gallium cylindrical tank (inner diameter of 170 mm) and a set of four bubbling tubes connected to a header
that supplies the to-be-bubbled hot water to the four tubes. At each of the four bubbling tubes, 12 holes have been drilled with 3mm diameter each. Hence, the total bubbled flow would be emanating from 48 holes.

In the first setup design, which does not include additional cooling water tubes, when appreciable amount of heat is dumped into the gallium, the gallium’s temperature would rise quickly and cause the temperature difference between the bubbles and the gallium bath to diminish. That results in ceasing out the heat exchange process in a short time after the start of the bubbling process. This design would then be most suitable mainly for batch operation in which a predetermined amount of heat is to be rejected from a hot source. This would imply that the amount of gallium in the bubbling tank is decided a priori to be sufficient for capturing the amount of heat dumped by the bubbled hot water that would cause a certain pre-decided temperature rise in the gallium; as decided by the design specifications of the system. In the second design, the above bubbling system is used but with additional header connected to a set of heat exchanger tubes being installed at the bottom of the bubbling tank so that cooling water can be circulated through these tubes. In this case, the purpose of the tubes would capture the heat dumbed into the gallium by the hot bubbled water. Thereby, the gallium temperature would be maintained at a desired fixed low level temperature during the bubbling of the hot water. This will enable the gallium to continuously act the role as a superior heat sink for the bubbled hot water for extended periods of time as needed and overcome the limitations imposed on gallium by its low specific heat capacity that otherwise would lead to appreciable temperature rise in the gallium upon dumbing heat into it. The number of cooling water tubes in this second design are selected to be five with inner diameter of 2 mm each. Such small diameter tubes have been selected to satisfy high surface
area to volume ratio. This will allow the cooling system to act in a timely manner to control the gallium’s temperature so that the gallium temperature is ensured to be within the desired limits decided by the design. Figure 2.3.1 shows the instrumentations and the locations to measure the different relevant temperatures demonstrated in this study.

Figure 2.3.1: Schematic Diagram for the Bubbling Experiments
Different K-type thermocouples with error of ± 1.0 °C were used. All the thermocouples were connected to data acquisition card (DAQ) and the measurements were recorded using the LabVIEW software. It should be noted that the frequency of data collection is 1000 Hz. The temperature measurement locations where the cooled water bubbles burst at the upper gallium surface were selected to be most closely on top of the bubbling holes from which these bursting bubbles basically originated. For each of the conducted runs the measurements were repeated different times in different gallium heating cycles and given the intermittent nature of the process the results showed good level of repeatability as will be seen in the results section. Variable speed pumps were used to circulate the bubbled water and for the cooling water in cases where external cooling tubes are used. The two pumps were connected to storage tanks one for the hot water and the other for the cooling water intakes and precautions were taken to prevent gallium from surging back to the bubbles.
distributing tubes at times of system startup and shut down by placing a ball valve as shown in Figure 2.3.1.

The to-be-bubbled hot water was poured into the hot water tank at temperatures selected to be in the range from about 40 °C to 60 °C. This is a range believed to be good enough to demonstrate the validity and effectiveness of the current bubbling heat exchanger design and at the same time it may be a good representation for a range of temperatures that might be encountered in applications relevant to thermal management (e.g. electronics cooling). More details on the operating temperatures of the different runs are given in Table 2.3.1.

For the runs which do not include external additional cooling tubes in the gallium, two hot bubbled water flow rates of 25 and 30 g/s were used. For these cases, to bring back the gallium to its initial starting temperature of around 30°C, every time the test is to be stopped to repeat the bubbling of the hot water again. Hence, “back-wash” is implemented by bubbling cooler water into the gallium. Some results that show the process of gallium cooling implemented by this back-wash process will be given in the results section. For the case when such cooling tubes were imbedded inside the gallium at the bottom of the bubbling tank, hot water flow rate of 11 g/s was used. For this later case, cooling water flow rate in the tubes of was used to remove heat dumped into the gallium by the hot bubbles at a rate that allowed for maintaining the gallium body at almost a fixed temperature of around 36°C during a continuous hot water bubbling process. Table 2.3.1 summarizes the operating conditions of the main three conducted runs and the way adopted to keep or cool gallium to its initial starting temperature.
To facilitate better appreciation and understanding of the bubbling results of the runs which do not include external cooling tubes imbedded in the gallium (i.e. the first experimental setup design), we give in what follows further description of the procedure adopted to conduct these runs. For these runs we considered two bubbling flow rates for the water namely about 25 and 30 g/s. At the onset of these runs the temperature of the gallium bath and that of the to-be bubbled water that resides in the tank which is connected to the bubbler tubes header, is made to be the same at about 30 °C. Afterwards the bubbled water temperature was gradually increased at a certain rise rate until the temperature reached about 70 °C. This temperature increase was

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Bubbled water flow rate (g/s)</th>
<th>Initial bubble temperature at bubbler orifice (°C)</th>
<th>Gallium bath initial temperature (°C)</th>
<th>Mechanism of cooling heated gallium</th>
<th>Nature of process</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>25.2</td>
<td>Variable: from about 30 to 60 °C</td>
<td>Varying: from about 30 °C to 70 °C</td>
<td>Back-wash; cold water bubbling (BWB)</td>
<td>Batch: Cycling cooling/heating of bubbled water</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>Variable: from about 30 to 60 °C</td>
<td>Varying: from about 30 °C to 70 °C</td>
<td>BWB</td>
<td>Batch: Cycling cooling/heating of bubbled water</td>
</tr>
<tr>
<td>3</td>
<td>11</td>
<td>Constant: at about 60 °C</td>
<td>Constant: at about 36 °C</td>
<td>External cooling tubes (ECT)</td>
<td>Continuous: Constant gallium and constant bubbling temperature</td>
</tr>
</tbody>
</table>
achieved by gradually pouring into the hot water storage tank, connected to the bubblor tubes header, water amounts at elevated temperatures that get mixed with the cooler water initially present in the tank. As we pour more and more hot water amounts into the storage tank, the water temperature at the bubbling orifices would increase up to the levels planned for these experiments. The temperatures at the bubblor tubes were displayed on the screen of the computer used for the data acquisition and this guided performing the experiments in the way reported in the results section. When a temperature around in the gallium bath of about 70 °C, the pouring of the hot water in the storage tank was stopped and gradually replaced by pouring cold water that gets mixed with the hot water present in the storage tank at this moment. This is the so-named in the previous section the “back-wash” process which replaces the external cooling water tubes to cool gallium in the second design of our bubbling system tested in this study. During this back-wash process, the temperature of the storage tank was gradually decreased at a certain desired rate by pouring more cold water into the tank until gallium bath and bubbler tube temperatures approach their initial values of around 30 °C; completing thereby the gallium heating then cooling process or alternatively said, the cycle of water cooling followed by water heating. With that, the gallium bath is ready once more to take another batch of hot water to cool it down by bubbling. This heating-cooling cycle was repeated at the same conditions of bubbling temperature rise and reduction rates to test repeatability of the obtained results of the bubbling heat exchanger built. These results will be presented and discussed further in the subsequent results section.
2.3.2 Mathematical Formulations

2.3.2.1 Instantaneous Heat Transfer Correlations Approach for the Bubbles

In this subsection, we present the approach followed to predict the bubbles’ instantaneous motion and heat exchange processes as they traverse the gallium bath from its bottom at the bubbling tubes orifices up to the upper surface of the gallium body. As highlighted earlier, there exist no such correlations to do so that are specifically built for the case of water bubbles moving through a gallium bath. For that reason, we refer at this stage to using theoretical considerations and some empirical correlations that are obtained specifically for two situations. The first is for solid spherical particles moving through a fluid media [52] and the second is for nitrogen gas bubbles injected into a molten metal bath [50]. The density ratio between the water bubbles and the liquid gallium in our experiments is somehow comparable to the density ratio between the solid spheres and the fluid media for which the correlation in [52] is constructed, which might give some justification to using this correlation for our case. However, due to the significant disparity of the density of the two media, it might look somewhat strange to consider the option of a correlation constructed specifically for gas bubbles injected into a molten metal bath that is reported in [50]. Yet, what encouraged us to still use the correlation given in [50] is the reported conclusion in [50] namely that irrespective of the material of the bubbles or the fluid through which the bubbles are injected, the results obtained by the correlation given therein are almost fluids-insensitive.

The first empirical correlation used for bubble’s Nusselt number obtained from reference [52] is valid for a wide range of particle’s motion conditions that includes the Reynolds number range encountered by the bubbles in the experiments.
conducted in this work and for a range of Prandtl number values for bath material that includes the liquid metal used here; namely liquid gallium. As mentioned above, at first glance the computed results based on solid sphere’s correlations may not look to be an appropriate fit for the case of the liquid water bubbles dispersed in gallium. Yet, given the lack in the literature of explicit correlations for water bubbles in liquid metals-like fluids, the performed computations are still expected to give some useful insight and guidance for understanding the experimental results obtained for water bubbles and would help interpreting them. Moreover, the obtained experimental results for the water bubbles in gallium and their possible deviations from the computed results, might help coming up with new updated empirical correlations for the bubbles case at hand that would enrich the literature in this important area.

In the performed computations we consider hot spherical water bubbles that start their motion at the bubbling orifice from zero speed and accelerate upwardly due to net force caused by buoyancy effects. Recall that the density ratio of the gallium to water is about 6. Hence, this will cause the bubbles, once they detach from the bubbling orifice rim, to begin their acceleration at a 5 times faster rate than gravitational acceleration, before they attain their terminal velocity. In selecting initial bubbles’ temperatures at the bubbling tube orifice, we chose in our computations values that are comparable to the initial temperatures encountered at the bubbling tubes root (orifices) in the experimental runs conducted.

Fixed gallium bath reservoir temperature values are considered for the performed computations; 30 °C, 40 °C, and 50 °C. These values are selected in such a way to reflect the impact of gallium bath temperatures on the cooling rate of water bubbles as well as on the minimum amount of gallium height that would be required
to cause a certain temperature drop for the bubbles during their trip through the gallium. Moreover, these temperature values are comparable to the gallium bath temperatures we used in the different experimental runs reported herein and would give the chance to assess used correlations by comparing experimental results with computed ones.

A range for the bubble diameter from 1 mm to 4 mm has been selected for the computations. In the conducted experiments the bubbling orifice diameter is made to be 3 mm. The bubble size at the moment it detaches from the bubbling orifice would be decided by interplay action of different forces including buoyancy, surface tension, gravity and pressure forces at the bubbling tube. By conducting simplified calculations for the bubble size at the moment of detachment from the bubbling orifice, one can conclude that the bubble emanating from the 3 mm-diameter bubbling hole would be approximately in the diameter range of 4 to 5 mm. For that reason, the computations performed based on 4 and 5 mm bubble diameter will appear in all of the figures that compare between experiments and computations in the results section.

Figure 2.3.3 presents a free body diagram for a spherical particle. The motion of this spherical solid particle in the surrounding liquid gallium bath is characterized by Newton’s second law:

\[ F_B - W - F_D = ma \] (1)

Equation 1 is given below in more explicit detailed form:

\[ \rho_g v_w g - \rho_w v_w g - \frac{1}{2} \rho_g V^2 AC_D = ma \] (2)
The drag coefficient \( C_D \) given as a function of bubble Reynolds number is calculated by using the following correlation which covers a wide range of interest ranging from creeping flow to high turbulent Reynolds number flows [29].

\[
C_D = \frac{24}{Re} + \frac{2.6x(Re)}{1^{1.52}} + \frac{0.411x(Re)}{1^{1.6}} - 7.94 + \frac{Re^{0.8}}{461x10^3}
\]

(3)

The Reynolds number is obtained as:

\[
Re = \rho g VD/\mu g
\]

(4)

The above equation of motion was further simplified and reduced to a final first order ordinary differential equation for the bubble velocity that was solved using MATLAB. In the solution, the bubble motion was analyzed at time intervals of 0.001 seconds.

For the heat transfer part of the computations, as mentioned earlier, Nusselt number was found in two ways. In the first we used the empirical correlation reported in [52] and in the second the one reported in [50]. The empirical equation from [52] is given as:

\[
Nu = 2 + 0.47Re^{1/2}Pr^{0.36}; \quad 3 \times 10^{-3} \leq Pr \leq 10; \quad 10^2 \leq Re \leq 5 \times 10^4
\]

(5)
In Equation 5, Re is the Reynolds number of the bubble and Pr is Prandtl number of the fluid through which the bubbles are injected (here gallium). The above ranges for bubble Reynolds and Prandtl number cover well the motion of the bubbles encountered in this study and the kind of the fluid medium (liquid gallium) through which they move.

The Nusselt number correlation that was obtained from [50] is given as:

\[
Nu = 1.1 \left[ \frac{Pe}{1+k} \right]^{0.7}
\]  \hspace{1cm} (6)

Where \( k \) is the viscosity ratio defined as the viscosity of the fluid inside the bubble (water in our case) divided by the viscosity of the fluid outside the bubble (liquid gallium in our case). In Equation 6, Pe is the bubble’s Peclet number which is given as:

\[
P_e = RePr_B
\]  \hspace{1cm} (7)

Where \( Re \) and \( Pr_B \) in Equation 7 are those of the bubble.

The convective heat transfer coefficient between the bubble and its surrounding medium (\( U \)) is obtained from the Nusselt number as:

\[
U = Nu \times \frac{k}{D}
\]  \hspace{1cm} (8)

Finally, the rate of heat removed from the water bubble (\( \dot{Q} \)) is found as:

\[
\dot{Q} = UA_s(T_s - T_\infty)
\]  \hspace{1cm} (9)

In the computed results reported, it was considered that the water bubbles are issuing from the bubbling tubes holes as discrete particles with spherical shape. The overall flow rate for the bubbled hot water and the frequency of the issuance of the water bubbles from any of the bubbling holes is such that the issued bubble will not
have any chance to collide with the previously issued bubble from the same bubbling hole. The following example calculations demonstrate this point further where we present below some calculations based on 30 g/s bubbling rate of the hot water, which represents the highest flow rate we considered in the conducted experimental runs:

The number of bubbling tubes in the present experiments is 4. At the top of these bubbling tubes, 12 holes with 3mm diameter each has been drilled. The spacing between every two successive holes on the tube is 10 mm. For hot water bubbling with total flow rate of 30 g/s a simple calculation would show that each of the bubbling tubes (assuming water is distributed uniformly among them) would receive a total flow of 7.5 g/s. Hence, each of the 12 holes on one tube would nominally have a share of 0.625 g/s. By doing a simple force balance at the exit of the bubbling hole for the moment where a water bubble is just ready to detach off the hole rim, one can conclude that a water bubble need to have a diameter of around 5 mm. For such a size of a bubble, the mass contained within the single bubble would be $6.45 \times 10^{-5}$ kg. Using the above water flow rate for each hole gives a frequency of bubbling from each hole of about 10 Bubbles/s. This means that one bubble will evolve and detach from the bubbling hole approximately every 100 ms. By considering the computed result for the time needed by 5 mm bubble to cover a vertical distance of about 13.2 mm in the gallium,(which is the gallium bath height above the bubbling tubes in our experiments), one would find that time to be about 45 ms. One could conclude from this that; with the bubble generation frequency of 10 Bubbles/second and this short time (45 ms) needed to cross the gallium height of 13.2 mm in our experiments, there would be no chance to have two successive bubbles from the same bubbling orifice hitting each other during their trip throughout the gallium bath. These simple
computations support our assumption of treating water bubbles in the performed computations as individual discrete spherical non-coalescing particles travelling through gallium. Moreover, the spacing between bubbling holes on one bubbling tube and the expected bubble diameter at the moment of detachment from the hole’s rim are chosen such that the possibility of touching and coalescence of adjacent bubbles issuing from neighboring holes on one tube, is minimized or even eliminated altogether. The issue with minimizing coalescence possibility in the practical designed bubbling system is to ensure successful formation of fine discrete bubbles with large enough surface to volume ratio so that within the given residence time for the bubble in the gallium bath (here 45ms), the bubble will have enough chance to drop temperature to the desired levels.

2.3.2.2 Global Average Heat Transfer Correlations Approach for the Whole Bubbling Process

In this section we present the mathematical formulation for the second approach we adopted in this study in which we account for overall average heat transfer in the whole bubbling process. Based on that we use a correlation reported in [46] to give average overall convective heat transfer coefficient for the bubble based on which we can predict the final temperature a bubble can attain after crossing the gallium bath. In what follows we give the equations based on the above approach:

Overall Nusselt number for the whole bubbling process is defined by [46]:

\[ Nu = UL/k_b \]  \tag{10}  

Where \( U \) is the average convective heat transfer coefficient for the bubble, \( L \) is the depth of the bath through which bubbles move (here gallium), and \( k_b \) is the thermal conductivity of the bubble fluid (water in our case).
The bubble Nusselt number is correlated as a function of Reynolds and Prandtl numbers as follows [46]:

$$Nu = 0.027(Re)^{1.36}(Pr)^{0.33}$$

(11)

Where Pr is the Prandtl number for the bubbles (water) but the definition of Reynolds number is given in a different from the bubble Reynolds number used in previous correlations above. The definition of Reynolds number appearing in Equation 11 is given as:

$$Re = VL/\nu_b$$

(12)

Where in Equation 12, V is defined to be the average velocity of the bubbled flow calculated based on the nominal flow rate value of the bubbled fluid as it flows through the bubbling orifice. L is the gallium depth, and $\nu_b$ is the kinematic viscosity of the bubbled fluid (water).

By using the convective heat transfer coefficient obtained from Equation 10, we can solve the following energy balance equation to obtain the bubble temperature as the bubble bursts from the upper surface of the gallium bath:

$$(UA)LMTD = \dot{m}c_p(T_{b,burst} - T_{b,root})$$

(13)

Where A in Equation 13 is a reference area arbitrarily taken in [46] to be the cross sectional area of the bubbling tube orifice which is used in the calculation of the bubbled water speed used in Reynolds number calculation given in Equation 12 above. The temperatures $T_{b,burst}$ and $T_{b,root}$ are the outlet bubble temperature at the gallium’s bath upper surface when the bubble leaves the gallium domain and the temperature at the bubbling tube orifice located at the bottom of the gallium bath.
when the bubble enters the gallium’s domain, respectively. The Logarithmic mean
temperature difference LMTD is obtained as:

\[
LMTD = \frac{T_{b\text{-burst}} - T_{b\text{-root}}}{\ln\left(\frac{T_{Ga} - T_{b\text{-root}}}{T_{Ga} - T_{b\text{-burst}}}\right)}
\]

(14)

Where \( T_{Ga} \) is the temperature of the gallium’s bath.
Chapter 3: Results and Discussion

3.1 Heat Sink Integrating Un-encapsulated Discrete PCM Cavities

In this section, the heat transfer performance results of the gallium heat sink with un-encapsulated PCM cavities will be presented and discussed. The experimental runs conducted are classified into two main groups; A and B. Each of these groups includes different cases mainly characterized based on whether PCM has been utilized and if so in which arrangement configuration, as well as on the amount of hot water used in each case. For both groups, in the cases that included gallium a fixed amount of 120 ml gallium has been used. Moreover, for each of the two groups, there is a baseline reference case (numbered as “0A and 0B; see Table 3.1.1”) to compare with, which does not include gallium nor gallium/PCM combinations; i.e. only water. To assure common basis of meaningful comparisons, the reference baseline cases are conducted under boundary conditions similar to those of the other cases in the groups. To assess the heat transfer performance of each case, different temperature measurements were recorded over tests’ time, as described in the experimental section; see Figure 2.1.3.

Table 3.1.1 summarizes the main features of each experimental group and the cases included therein. In the cases that integrate PCM within gallium, the PCM percentage is given on volume basis. A discussion of the results follows in the subsequent sections. It should be noted that runs for cases considered in this work were repeated three times. Table 3.1.2 shows the average standard deviation calculated. The results have shown that the tests are repeatable and the standard deviation for all cases was below the accuracy error of the thermocouples.
### Table 3.1.1: Summary of the Features for the PCM Experimental Runs

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Hot Water</th>
<th>Including Gallium?</th>
<th>Gallium Quantity</th>
<th>Including PCM?</th>
<th>PCM Arrangements</th>
<th>PCM Volume Percentage in gallium sink</th>
</tr>
</thead>
<tbody>
<tr>
<td>Group A</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0A</td>
<td>40 ml</td>
<td>No</td>
<td>N/A</td>
<td>No</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>1A</td>
<td>40 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>No</td>
<td>N/A</td>
<td>0%</td>
</tr>
<tr>
<td>2A</td>
<td>40 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>Yes</td>
<td>Structured cylindrical holes</td>
<td>5%</td>
</tr>
<tr>
<td>3A</td>
<td>40 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>Yes</td>
<td>Random; single layer</td>
<td>5%</td>
</tr>
<tr>
<td>4A</td>
<td>40 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>Yes</td>
<td>Random; double layers</td>
<td>10%</td>
</tr>
<tr>
<td>Group B</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0B</td>
<td>60 ml</td>
<td>No</td>
<td>N/A</td>
<td>No</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>1B</td>
<td>60 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>No</td>
<td>N/A</td>
<td>0%</td>
</tr>
<tr>
<td>2B</td>
<td>60 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>Yes</td>
<td>Structured cylindrical holes</td>
<td>5%</td>
</tr>
<tr>
<td>3B</td>
<td>60 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>Yes</td>
<td>Random; single layer</td>
<td>5%</td>
</tr>
<tr>
<td>4B</td>
<td>60 ml</td>
<td>Yes</td>
<td>120 ml</td>
<td>Yes</td>
<td>Random; double layers</td>
<td>10%</td>
</tr>
</tbody>
</table>
Table 3.1.2: Summary of Calculated Standard Deviation for PCM Experiments

<table>
<thead>
<tr>
<th>Case</th>
<th>Measuring location</th>
<th>Standard Deviation (± °C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water Only - Reference</td>
<td>Hot water</td>
<td>0.17</td>
</tr>
<tr>
<td>Gallium without PCM</td>
<td>Hot water</td>
<td>0.05</td>
</tr>
<tr>
<td>Gallium + PCM Structured Holes</td>
<td>Hot water</td>
<td>0.64</td>
</tr>
<tr>
<td>Gallium + PCM Random Single Layer</td>
<td>Hot water</td>
<td>0.69</td>
</tr>
<tr>
<td>Gallium + PCM Random Double Layer</td>
<td>Hot water</td>
<td>0.36</td>
</tr>
<tr>
<td>Gallium + PCM Random Double Layer</td>
<td>Gallium</td>
<td>0.17</td>
</tr>
</tbody>
</table>

It should be noted that the error bars shown on the temperature measurement curves given in the results below are estimated based on uncertainty analysis to give the best estimate of the true temperature value, provided by its average and uncertainty for that value. Hence, the temperature values are reported as the mean temperature ±1.36% uncertainty estimated at 90% confidence (probability) level.

**Results of Group A; (120 ml Gallium and 40 ml Hot Water)**

Hot water temperature variation with time for the runs of group A is presented in Figure 3.1.1. In accordance with the earlier highlighted observations of previous researches [4-8], Figure 3.1.1 confirms that when hot water sits alone in the container, as in the reference baseline case (case 0A), the water loses its heat to the outer surroundings mainly via the container bottom brass base (see Figure 2.1.3). This happens at a clearly slower rate than that observed in the runs that include direct
contact between the hot water and gallium (case 1A) or gallium/PCM combinations (cases 2A-4A).

In case 1A hot water sits directly on top of a gallium block but in this case there is no PCM integrated within the gallium. Hence, this case will help as a benchmark to assess the extent to which PCM in the conducted experiments can influence the cooling of hot water for the cases where PCM is used (cases 2A-4A).

Figure 3.1.1: Hot water temperature variation with time for Group A cases

Figure 3.1.1 shows that from the very early stages; (about 10 seconds from the onset of the heat transfer process), the hot water temperature in the tests that include gallium (cases 1A and 2A-4A) starts deviating clearly from the temperatures recorded in case 0A, indicating thereby a clearly faster cooling rate for the water in these tests. Moreover, among the cases that include PCM, case 4A with the largest proportion of PCM in gallium (10%), shows the fastest water cooling rates. In this group of tests, hot water temperatures started from about 70 °C. The transient period
before a quasi-constant temperature level in the very late stages is observed, lasts for at least 260 seconds for all cases (see Figure 3.1.1). However, most of the thermal energy content reduction of the water takes place in the early stages of the process.

Table 3.1.3 presents the percentage temperature reduction for the hot water at some representative times during the process for cases 0A-4A. The two cases with PCM namely case 2A with 5% structured PCM and case 4A with 10% randomly distributed PCM, show (especially case 4A) clearly faster cooling rates for the hot water than the rest of the cases. For example, after 100 seconds from the start, case 2A results in a cooling rate that is about 44 percent faster than that in case 0A and about 7 percent faster than in case 1A. As time progresses further; (although it is still appreciable), the difference between these PCM cases and the two baseline cases (0A and 1A) decreases. For example, after 260 seconds from the starting point the overall cooling achieved in case 2A is about 4 percent faster than that of case 1A and about 35 percent higher than in case 0A. For all cases, especially the cases with gallium and PCM, most of the possible water cooling takes place in the early stages of the test time. Hence the appealing characteristics of the PCM/gallium combinations would then be judged mostly based on the extent to which they will facilitate fast heat removal from the hot water in these early stages.

Obviously, Table 3.1.3 shows even further superiority for the PCM integration in gallium when the volumetric PCM percentage in the sink is increased to 10%, as can be verified from comparison between case 4A and the rest of the cases. It is to be recalled that in case 4A the PCM is randomly structured as is the situation in case of 3A. However, in case 4A this is done in two layers. Stuffing only one single layer with more PCM content than what has been implemented in case 3A
aiming at increasing PCM percentage in the gallium may lead even to adverse effects, as will be discussed further below. Therefore, in case 4A two layers of the gallium were decided to accommodate the larger PCM content.

Table 3.1.3: The Percentage Temperature Reduction with Time for Group A

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>Case Number</th>
<th>% Temp. reduction</th>
<th>% Dev. from 1A</th>
<th>% Temp. reduction</th>
<th>% Dev. from 1A</th>
<th>% Temp. reduction</th>
<th>% Dev. from 1A</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0A</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>14.7%</td>
<td>17.5%</td>
<td>19.4%</td>
<td>10.9%</td>
<td>19.7%</td>
<td>12.6%</td>
<td>22.5%</td>
</tr>
<tr>
<td>50</td>
<td>25.1%</td>
<td>32.9%</td>
<td>36.3%</td>
<td>10.3%</td>
<td>34.8%</td>
<td>5.8%</td>
<td>38.1%</td>
</tr>
<tr>
<td>75</td>
<td>30.5%</td>
<td>40.8%</td>
<td>44.1%</td>
<td>8.1%</td>
<td>42.1%</td>
<td>3.2%</td>
<td>46.5%</td>
</tr>
<tr>
<td>100</td>
<td>34.2%</td>
<td>45.9%</td>
<td>49.1%</td>
<td>7%</td>
<td>46.9%</td>
<td>2.2%</td>
<td>51.9%</td>
</tr>
<tr>
<td>150</td>
<td>39.2%</td>
<td>51.9%</td>
<td>54.9%</td>
<td>5.8%</td>
<td>52.9%</td>
<td>1.9%</td>
<td>57.9%</td>
</tr>
<tr>
<td>200</td>
<td>42.4%</td>
<td>55.3%</td>
<td>57.9%</td>
<td>4.7%</td>
<td>56.2%</td>
<td>1.6%</td>
<td>60.6%</td>
</tr>
<tr>
<td>230</td>
<td>43.7%</td>
<td>56.7%</td>
<td>59%</td>
<td>4.1%</td>
<td>57.5%</td>
<td>1.4%</td>
<td>61.6%</td>
</tr>
<tr>
<td>260</td>
<td>45%</td>
<td>57.8%</td>
<td>59.7%</td>
<td>3.3%</td>
<td>58.5%</td>
<td>1.2%</td>
<td>62.2%</td>
</tr>
</tbody>
</table>

The slight superiority in regards to water cooling seen in case 2A over case 3A may hint to the possibility of even getting better cooling performance than that attained in case 4A if the larger 10 percent PCM content in case 4A be placed in a structured well-defined way. However, this would pose a challenge in finding reasonable space within the solid gallium body to accommodate this larger amount of PCM without sacrificing the internal heat transfer connectivity between the different parts of the gallium body in the whole sink matrix. Yet, further optimization of the
way to structure and arrange different discrete PCM pieces within the gallium matrix; based on the concept suggested in this study, would be a suitable issue to address in more extensive details in upcoming investigations.

Case 2A shows slightly better water cooling performance than case 3A (see Table 3.1.3). Yet, since this is only a slight difference, then this may indicate a less important role played by the exact way how PCM cavities are located within the solid gallium. Rather, it hints to a much more important role played by the actual total amount of PCM loading in the gallium instead (see cases 2A, 3A, and 4A in Table 3.1.3). The decision on the issue regarding PCM quantity and its arrangement structure would at the end be a function of various parameters including the input heat from the source as well as the temperature of the source itself in addition to probably the shape of the PCM cavities as well as the thermophysical properties of the used PCM. All these can be addressed in a future study, as mentioned above.

By considering the initial starting temperature of the hot water in the tests of this group which is about 70 °C and the ambient temperature, one can conclude that the maximum theoretical temperature drop that can possibly be attained in any of the runs of this group is the difference between these two temperature limits. Based on this maximum theoretical temperature drop we may define water cooling efficiency term for the water cooling process that may characterize the performance of each case at any instant of time during the test by dividing the water temperature drop at any moment by this theoretical maximum value. Figure 3.1.2 presents this cooling efficiency term for group A cases over 260 seconds of the test time. It can be concluded from the results presented in Figure 3.1.2 that the initial period is the most critical in determining the rate and amount of heat removed from the hot water. For
example, in case 4A, 75 percent of the total heat possible for removing is rejected from the hot water in the first 100 seconds of the test. About 15 percent of the heat content is however rejected in the remaining 160 seconds of the test time. It should be noted that in the very late stages of the process (after 260 seconds) the water’s temperature is becoming very close to thermal equilibrium with its environment and so the heat removal rate from the water would be very small. By all means, the validity of the PCM integration concept implemented in this study shows its most appealing potential in the initial stages of the process. The case without PCM (case 1A) clearly shows slower heat removal ability from the water as indicated by its lower cooling efficiency levels reported in Figure 3.1.2, especially when compared with cases 2A and 4A in particular.

As highlighted above, the late stages of the process, where the hot water temperature closely approaches the surrounding environment temperature, are characterized by a decline in the rate of change of the cooling efficiency. This may give a hint to use the proposed gallium/PCM sink concept in the temperature ranges that are associated with high efficiency rate of change namely the early stages of the process. Hence, the target in mind will in this case be to engineer the process to cool the hot source (water in this case) to desirable levels that are not closely matching the temperatures of the surrounding environment but nonetheless still serve the purpose of meeting acceptable cooling levels.
Obviously, from Table 3.1.3 and Figure 3.1.2 one can see that the performance of the case without gallium (only water cooling while setting alone in the test container; i.e. case 0A) is quite poor when compared with any of the other cases.

Case 3A with randomly distributed PCM in a single layer shows clear superiority over case 1A without PCM roughly only in the first 60 seconds. After that this superiority diminishes, although it is still observed to exist up to the end of the reported time of the test. Moreover, case 3A shows only in the very early stage of the process, within about 20 seconds, some marginal advantage over case 2A with structured PCM cavities. After this short time case 2A slightly surpasses case 3A up to almost the end of the reported test time.

Figure 3.1.2: Water Cooling Efficiency for Group A Cases
The reason behind the marginal superiority of case 3A over case 2A in the very short time at the beginning of the test might be explained as follows: in case 3A more PCM chunks are clustered near the upper surface close to the hot water body. Therefore, they act more effectively in the very early stages of the process in capturing the heat received from the hot water. Hence, they hinder any appreciable temperature rise in the gallium located in the region near the hot water. This effect facilitates maintaining a higher temperature difference between the gallium and the hot water to motivate slightly more cooling of the hot water within the first 20 seconds of the process. However, as time passes the above effects, which are perceived at first as an advantage, become more of adversity. The stuffing in case 3A of relatively large amount of randomly distributed and randomly sized PCM chunks within a limited single gallium layer space in the upper parts of the gallium matrix would be acting like a barrier preventing the heat captured from the hot water to escape to the lower gallium parts. After that, the heat will be escaped to the outer environment through the bottom base. Then, the above scenario will cause the gallium temperature in the very upper parts of the gallium region (the melted mushy region) to increase more clearly upon receiving further heat from the water. So, it will adversely affect the driving heat transfer temperature difference and cause the cooling process of the water to take place at a smaller rate than that observed in case 2A. Some slight superheating of the possibly melted gallium layer immediately below the hot water source may be present in this case that would lead to the above result.

The above events of superheating the melted gallium in the mushy region would not be happening in case 2A to the extent described above for case 3A. This is attributed to the fact that in case 2A the PCM amount is distributed over a larger
solid gallium body within the sink space in a more structured way that is associated with more free passageways throughout the gallium body. This would ensure more smooth interconnectivity hence more effective heat transfer inside the gallium sink matrix. Consequently, this later effect would then give more chance to benefit from the superior thermal conductivity of gallium in case 2A. The above argument is possibly the reason behind the ability of case 2A to exceed hot water cooling limits attained in case 3A, shortly after the onset of the process up to the end of the test time. With the above being said, once more, the difference between cases 2A and 3A is still of less significance as compared with the effect of the actual amount of PCM loading used. That is obviously a dominant factor as can be judged based on a comparison between the results of case 4A with 10% PCM loading and cases 2A and 3A with 5% PCM loading each.

In accordance with the above discussion, it seems that in case 4A the effect of the larger PCM loading and the distribution of the PCM in that case being in two distinct layers separated from each other, somehow offset the above adverse effects of stuffing PCM randomly in only one single layer, as has been done in case 3A. Yet, it should be noted that there would be expected to exist an optimum value for the PCM loading within the sink beyond which issues related to solid gallium interconnectivity highlighted above and the consequences this may entail might become important. Exploring these points in more details is a subject for future research work.

The time history of the temperatures measured in the middle of the solid gallium core for cases (1A-4A) is presented in Figure 3.1.3. As can be seen from the figure, the PCM in the gallium helped maintaining lower gallium temperatures than
the case without PCM (case 1A). This is very obvious especially when comparing the results in Figure 3.1.3 for cases 1A and 4A. The lower gallium temperatures in the cases with PCM are the reason for having larger temperature difference hence faster heat transfer from the water to the sink, as highlighted above in Figure 3.1.2 and Table 3.1.3.

Figure 3.1.3 shows that the gallium temperature results of case 2A are not far from those corresponding to case 3A, irrespective of the clearer (though still comparatively small) difference between the two cases when considering their overall bulk results related to hot water cooling history depicted in Figure 3.1.2 and Table 3.1.3. Yet, with the above being said, we still may expect some more non-negligible differences between the gallium temperature of the two cases in other locations within the gallium block (other than the location of temperature measurements reported in Figure 3.1.3). This is due to the different natures of PCM cavities arrangements within the sink in both cases and the effects this may have on the gallium interconnectivity and hence heat transfer and the possible trapping of heat within the sink, as highlighted above. Indeed, the existence of such differences in gallium temperature between cases 2A and 3A is one way or another to think of the possible reasons behind the (however small) somewhat overall global differences in water cooling rates in these two cases, as reported above in Table 3.1.3 and in Figure 3.1.1. In that sense, the structured way of distributing the PCM within gallium, the way done in case 2A, proves to be to some extent more effective in maintaining an overall lower bulk gallium temperature, hence enhancing the cooling of the hot water source the gallium matrix is in contact with.
Figure 3.1.3 shows further that after pouring the hot water on top of the gallium block, the gallium temperature starts rising very sharply, especially in the test with no PCM (case 1A). This very sharp increase is attributed to both the high thermal conductivity of gallium which passes the heat quickly to the location of the measuring thermocouple and at the same time to the limited specific heat capacity of gallium. For all cases after roughly 80 seconds from the start, the gallium temperature settles down to almost constant values spanning over the remaining tests’ time. The effect of larger PCM loading in case 4A in resulting in lower level gallium temperatures, is quite clear.

Figure 3.1.4 shows for group A cases the temperatures of the interfacial region between hot water and gallium, measured from the gallium side where the thermocouple was dipped about 2 to 3 millimeters below the interface between water and gallium.
Figure 3.1.4 shows that the lower interfacial region temperatures in the cases with PCM, especially case 4A, are consistent with the lower solid gallium temperatures of the whole solid gallium block reported in Figure 3.1.3. The difference in the interfacial region temperature between the cases with PCM and case 1A without PCM is of special significance in the initial period of the test (see Figure 3.1.4). It is in that initial period were most of the heat transfer from the water to the gallium block takes place, as has been observed from above figures and Table 3.1.3. The very sharp rise in the temperature of the interface in the case without PCM (case 1A) is very prominent and reflects the significant role played by PCM in dampening such rapid temperature rise impacting positively thereby the subsequent cooling of the hot water setting on top of the mushy gallium layer.

Figure 3.1.4 shows further that in most of the time of case 1A the interfacial gallium was in a melted state since the recorded temperatures were above the melting point of gallium (i.e. above 29 °C). In cases 2A and 3A the melting wave reaches the location of the temperature measurement (about 2 to 3 mm below the interface layer between water and gallium) very shortly after the start of the tests; as can be judged from the interfacial region temperatures reported in Figure 3.1.4 being around 29 °C. In this respect, it should be recalled that the nature of gallium being associated with low melting point, would serve two purposes when it comes to cooling hot sources. Firstly, gallium’s high thermal conductivity would facilitate fast cooling of the hot sources and secondly the latent heat captured by solid gallium during its melting would as possible restrict any significant temperature rise upon receiving heat from the source. This gallium’s phase change process augments and supports the role played by the used additional external PCM
material. In this regard, the role played by the PCM is appreciated by recognizing the clear superheating and temperature overshooting recorded in the interfacial mushy gallium region in a very short time after the onset of the test in case 1A were the temperatures rise sharply to about 40 °C. This scenario have not been seen to be taking place in the tests that included PCM (cases 2A-4A); see Figure 3.1.4.

![Group A: Interface Temperature (Ga 120ml; Water 40ml)](image)

**Figure 3.1.4: Interfacial Temperature Variation with Time for Group A Cases**

As evidenced by the lower interfacial temperatures being less than 29 °C, as seen in Figure 3.1.4, the melting wave did not reach the location of temperature measurement in the interfacial region in case 4A. Once more, this is attributed to the ability of the larger PCM amounts in case 4A to accommodate more heat within and so to suppress temperature rise in the gallium’s body.
Results of Group B; (120 ml Gallium and 60 ml Hot Water)

Hereby results of group B cases with larger amount of hot water (60 ml) will be discussed. The variation with time of the hot water temperature for cases 0B – 4B is shown in Figure 3.1.5. As noted in the previous groups’ results, the presence of gallium or gallium/PCM in the sink improves the cooling rates of the hot water that is brought into a direct contact with that sink. As depicted in Figure 3.1.5, the water temperature recording started from initial level of about 70°C, same as in the case of group A. The transient behavior in the results of group B, before a quasi-constant water temperature level is attained, would as a result of using larger amounts of hot water last longer than in the cases of group A.

Figure 3.1.5: Hot Water Temperature Variation with Time for Group B Cases

Table 3.1.4 presents the percentage reduction in temperature with time for cases 0B-4B. It also compares between the cases with PCM (2B-4B) and that with
only gallium (1B). Clearly, the presence of PCM in the gallium block has a favorable impact on the rate of cooling of the hot water, especially in the early stages of the process, which is the same conclusion drawn based on the results of group A above.

Taking the data in Table 3.1.4 at the sampling time of 100 seconds as an example, the temperature drop in case 2B is faster than that in cases 0B and 1B by about 38% and 20% respectively.

Table 3.1.4: The Percentage Temperature Reduction with Time for Group B

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>0B</th>
<th>1B</th>
<th>2B</th>
<th>3B</th>
<th>4B</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>7.4%</td>
<td>8.4%</td>
<td>12.1%</td>
<td>44.1%</td>
<td>12.5%</td>
</tr>
<tr>
<td>50</td>
<td>15.8%</td>
<td>17.9%</td>
<td>23.7%</td>
<td>32.4%</td>
<td>23.1%</td>
</tr>
<tr>
<td>75</td>
<td>21.0%</td>
<td>24.2%</td>
<td>30.2%</td>
<td>24.8</td>
<td>29.0%</td>
</tr>
<tr>
<td>100</td>
<td>25.3%</td>
<td>29.2%</td>
<td>35.0%</td>
<td>19.9%</td>
<td>33.5%</td>
</tr>
<tr>
<td>150</td>
<td>31.9%</td>
<td>36.6%</td>
<td>41.8%</td>
<td>14.2%</td>
<td>40.0%</td>
</tr>
<tr>
<td>200</td>
<td>36.9%</td>
<td>41.8%</td>
<td>46.6%</td>
<td>11.5%</td>
<td>44.6%</td>
</tr>
<tr>
<td>230</td>
<td>39.3%</td>
<td>44.2%</td>
<td>48.7%</td>
<td>10.2</td>
<td>46.7%</td>
</tr>
<tr>
<td>260</td>
<td>41.5%</td>
<td>46.2%</td>
<td>50.4%</td>
<td>9.1%</td>
<td>48.4%</td>
</tr>
</tbody>
</table>

By comparing the results in Table 3.1.4 (group B) with those of Table 3.1.3 (group A) one can conclude that including PCM in gallium is more effective when more heat is supplied, as is the case in group B where larger amount of hot water is used. The superiority of cooling for the PCM cases over those with only gallium is even more evident as the amount of PCM integrated in the gallium matrix increases.
(case 4B). Regarding the construction and arrangement of PCM cavities within the gallium block, the same conclusion made before based on group A results, structured arrangement shows some more superiority over the randomly distributed cavities. This, once more, may be attributed to the smoothness of the heat transfer interconnectivity between the different gallium parts and hence the PCM-free passageways for heat transfer in the gallium body, which apparently is of lower favorable attributes in the case of the random size and random distribution of PCM cavities (case 3B). Figure 3.1.6 presents the gallium’s temperature as a function of time for cases 1B-4B. Clearly, the inclusion of PCM has reduced the gallium’s temperature, as can be seen by comparing case 1B with cases 2B-4B. This observation has also been pointed out based on the results of group A above. However, one can see from Figure 3.1.6 that the difference between the gallium temperature in the PCM cases (2B-4B) and the case with only gallium (1B) with larger quantity of hot water being used, is more noticeable than in group A cases which once more reflect a more effective role played by the PCM in group B. Moreover, same as concluded from Figure 3.1.3 for group A cases, the case with more PCM loading in group B namely case 4B shows a clearer difference from the cases with less PCM in the same group (cases 2B and 3B). The steady levels of gallium temperature are attained at a slightly higher level than that reached in the cases of group A.
Figure 3.1.6: Solid Gallium Temperature Variation with Time - Group B

The interfacial region temperature measured from the gallium’s side (about 2 to 3 mm below the water/gallium interfacial layer) for group B is shown in Figure 3.1.7. Same conclusions drawn from Figure 3.1.4 for group A are still valid here. All the time recorded in case 1B the gallium in the interfacial region, where the temperature measurements are made, is in the melted state (above 29 °C) and so is the situation in cases 2B and 3B. In case 4B, however, the gallium in the location where interfacial temperature measurements are made, is always below the melting point temperature indicating that the melting wave coming from the upper interface did not reach deeper due to the heat capturing effects of the additional PCM quantities used in that case. This reflects positively on the cooling performance of the hot water in this case, as has been already outlined above.
Finally, Figure 3.1.8 shows the cooling efficiency results for group B cases where the advantages of the cases that integrated the PCM within gallium are quite clear, especially when larger amount of PCM is used. The efficiency values for group B are however less than those attained in the case of group A. The higher amount of hot water used in group B is responsible for these lower values.
3.2 The Application of Vibrational Signals to the Heat Sink

In this section, the water cooling results while applying vibrational effects will be presented and discussed. A reference baseline case under static conditions will be used to assess the extent to which vibration impacts the rate of water cooling. The different experimental runs conducted in this part and their operating conditions are summarized in Table 3.2.1.

As shown in the table, different test groups were considered. The baseline run is marked as test A1. The rest of the runs which include the effect of vibration are classified based on the vibration parameters namely frequency and amplitude.

Table 3.2.1: Summary of the Conducted Vibration Experiments

<table>
<thead>
<tr>
<th>Hot Water: 100 ml</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Group</td>
</tr>
<tr>
<td>Frequency (Hz)</td>
</tr>
<tr>
<td>Test Number</td>
</tr>
<tr>
<td>Displacement (mm)</td>
</tr>
</tbody>
</table>

It should be noted that the error bars shown on the temperature measurement curves given in the results below are estimated based on uncertainty analysis to give the best estimate of the true temperature value, provided by its average and uncertainty for that value. Hence, the temperature values are reported as the mean temperature ±1.36% uncertainty estimated at 90% confidence (probability) level.

Figure 3.2.1 (a) to (c) shows the gallium temperature, measured at the center of the gallium height, for the three vibrational groups considered at this study. For comparison purposes, this figure also includes the result of the baseline case 1A. For any of the groups considered, starting at 15 °C, the temperature at which the test had
started, the gallium’s temperature increases rapidly till it reaches the melting point, 30 °C, due to the sensible heat effect. After that, the gallium undergoes a phase change process and all the energy added to it from the hot water will be in the form of latent heat. It can be seen clearly that the rate of change for the gallium’s temperature in the sensible heat region increases as the amplitude (for the same group) or frequency (for different groups) increases. In other words, the tests conducted at higher amplitudes and frequencies reaches the melting point faster than the ones conducted at lower values of the studied parameters. This implies higher heat transfer rates when the vibration signals are applied and can be considered as indication for higher heat removal rates from the hot water as will be discussed later in this section.
Figure 3.2.1: Gallium Temperature Variations for Vibration Experiments: (a) Group B (20 Hz), (b) Group C (35 Hz), and (c) Group D (50 Hz)

In order to assess the heat transfer process between the gallium and the hot water above it, measurements for the gallium’s temperature at the water/gallium interface (see Figure 2.2.1) were taken.
Group B (20Hz): Water-Gallium Interface

- **1A** - Static (Reference)
- **1B** - 0.3 mm Amplitude
- **2B** - 0.5 mm Amplitude
- **3B** - 0.7 mm Amplitude

Group C (35Hz): Water-Gallium Interface

- **1A** - Static (Reference)
- **1C** - 0.3 mm Amplitude
- **2C** - 0.5 mm Amplitude
- **3C** - 0.7 mm Amplitude
The region considered by the temperature measurement of the interface, reported in Figure 3.2.2 above, represents the mushy melted gallium region between the hot water and the cooler solid gallium. This region plays a pivotal role in facilitating the heat transfer process that will only take place if the temperature of the interface is lower than that of the to-be-cooled hot water (e.g. temperature difference deriving force for the cooling process).

It should be noted that the starting temperature of the interface is the same as the one for the gallium bulk (near 15 °C) which implies that the whole gallium body system was in thermal equilibrium prior to the test starting time. This is attributed mainly to the relatively high thermal conductivity that makes the temperature to be distributed homogeneously at any point in the gallium. For all groups, the temperature of the interface raises very sharply (e.g. in few seconds) to almost 60 °C.
and then starts to cool down as the time passes. This sharp increase in the temperature is due to the limited specific heat capacity of gallium. The rate of change in the interface temperature is heavily dependent on the amount of heat that can be passed to the solid gallium underneath it. If the gallium at the interface fails to pass its heat quickly to the cooler solid gallium, the interface mushy region might be superheated as, in this case, the flow of heat prefers to accumulate as sensible heat in the liquid gallium at the interface considering it to be the less heat resistive option. Comparing the temperature profiles for static testing and the cases when the vibration was employed (see Figure 3.2.2), the fast heat removal rates when the heat sink is vibrated implies that the superheating phenomenon at the interface is minimized, which makes the method adapted here to be more appreciable.

Figure 3.2.3 shows the cooling temperature history of the to-be-cooled hot water for all groups of experiments considered in this study. The initial temperature of the hot water was unified at almost 70 °C so that the comparison among the tests can be done fairly. Moreover, this temperature represents a reasonable value as it may elucidate well the performance of the tested heat sink under the various conditions of interest to us in this study and at the same time may relate well to heat generating systems of interest, such as electronics. It can be seen clearly from Figure 3.2.3 that higher heat removal rates are achieved when the amplitude and frequency of the applied signal are increased. It should be recalled that the gallium temperature values in the three groups were almost identical (see Figure 3.2.1), over most of the time after the initial sensible heating period, while at the same time it is evident from Figure 3.2.3 that faster cooling rate of the water are achieved when the heat sink is vibrated. This may indicate that the heat conveyed from the water to the gallium in these cases (after the short sensible heating period) is solely utilized to
melt the gallium. It is interesting to notice the significant reduction in the time required to cool down the hot water to temperatures close to 40 °C and ultimately to temperatures close to being in thermal equilibrium with melted gallium (around 30 °C), when high levels vibration parameters are applied (Group D); Figure 3.2.3.
Figure 3.2.3: Hot Water Temperature Variations for Vibration Experiments: (a) Group B (20Hz), (b) Group C (35 Hz), and (c) Group D (50 Hz)

In order to appreciate the role played by the vibration in the cooling process of the hot water, it is important to highlight the time saved in the cases with vibration when cooling hot water from a certain initial temperature to a certain lower temperature, as compared with the static case, 1A. Table 3.2.2 shows the effect of the studied parameters on the achieved time saving percentages when the vibration signals were applied. For the sake of presenting clearly the above highlighted comparisons given in Table 3.2.2, the temperature profile of the cooling process for the hot water was divided into four main segments (a through d) starting from 70 °C and ending at 32 °C. It is noted from Table 3.2.2 that the time savings increases by increasing both the amplitude and the frequency of the applied signal. Also, it is interesting to note that, generally, for any of the frequency and amplitude levels presented, the time saving, when vibrational effects are applied, progressively
increases as the water gets cooler and ultimately it becomes most evident as the water temperature approaches (roughly) 40 °C (i.e. in segment d in Table 3.2.2). However, in absolute terms, the rate of water cooling in the last segment (i.e. segment d) is already much slower than the rates attained in the earlier segments (a and b) where in these early segments the temperature difference driving heat transfer between water and gallium is still sufficiently high. The above arguments highlight that vibrational effects enhance convective heat transfer appreciably, to the extent that compensate well for the progressively diminishing temperature difference driving heat transfer especially towards the late stages of the whole cooling process.

Table 3.2.2: Time Savings for the Cooling Process by Vibration as Compared with the Static Case 1A

<table>
<thead>
<tr>
<th>Test</th>
<th>Temperature Drop Segments</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>a (70-60°C)</td>
<td>b (60-50°C)</td>
<td>c (50-40°C)</td>
<td>d (40-32°C)</td>
</tr>
<tr>
<td>Group B</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(20 Hz)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1B (0.30 mm)</td>
<td>7.41</td>
<td>4.76</td>
<td>5.62</td>
<td>17.61</td>
</tr>
<tr>
<td>2B (0.50 mm)</td>
<td>22.22</td>
<td>33.33</td>
<td>48.31</td>
<td>67.44</td>
</tr>
<tr>
<td>3B (0.70 mm)</td>
<td>48.15</td>
<td>59.52</td>
<td>68.54</td>
<td>75.75</td>
</tr>
<tr>
<td>Group C</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(35 Hz)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1C (0.30 mm)</td>
<td>14.81</td>
<td>11.9</td>
<td>17.98</td>
<td>31.23</td>
</tr>
<tr>
<td>2C (0.50 mm)</td>
<td>40.74</td>
<td>38.1</td>
<td>49.44</td>
<td>68.11</td>
</tr>
<tr>
<td>3C (0.70 mm)</td>
<td>59.26</td>
<td>66.67</td>
<td>76.4</td>
<td>83.06</td>
</tr>
<tr>
<td>Group D</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(50 Hz)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1D (0.30 mm)</td>
<td>29.63</td>
<td>26.19</td>
<td>39.33</td>
<td>40.2</td>
</tr>
<tr>
<td>2D (0.50 mm)</td>
<td>59.26</td>
<td>66.67</td>
<td>75.28</td>
<td>82.06</td>
</tr>
<tr>
<td>3D (0.70 mm)</td>
<td>74.07</td>
<td>78.57</td>
<td>83.15</td>
<td>89.7</td>
</tr>
</tbody>
</table>

When the test is done under vibrating conditions, appreciable amount of the energy content of the hot water can be passed to the gallium in the mushy region which subsequently passes the heat, with the aid of vibration, easily to the cooler solid gallium below it. The role played by vibration in this case is to assist the melting wave to diffuse into the solid gallium body. It is anticipated that the heat
transfer dynamics due to the vibrational excitations at the molecular level, that take place in the interfacial region between water and mushy gallium as well as at the melted gallium/solid gallium interface, contribute to the enhancement of the cooling process heavily. In addition, the application of vibration is expected to disturb the interfacial surface of contact between the water and the melted gallium in the mushy region causing some kind of corrugated and wavy structures that ultimately lead to an increase in the area through which heat passes from the water to the mushy gallium. The above arguments explain the very rapid drop in water temperature observed in the case of the third test of each of the three vibration groups (B, C, D) where the vibration parameters are highest among all tests (see Figure 3.2.3 and Table 3.2.2).

Knowing the initial starting temperature of the hot water in the tests conducted (i.e 70 °C) and the ambient temperature in the immediate surroundings of the experimental setup during the experiments (i.e. 23 °C), one can conclude that the maximum theoretical temperature drop is the difference between these two temperature limits. Based on this maximum theoretical temperature drop water cooling efficiency can be defined, as done previously for the PCM results, for the water cooling process that may characterize the performance of each case at any instant of time during the test. This can be done by dividing the water temperature drop at any moment during the test by this theoretical maximum value. Figure 3.2.4 shows the cooling efficiency for the range of the three groups with vibration considered at this study and compares them with the reference baseline case with no vibration.
Group B (20 Hz): Cooling Efficiency

- **1A**: Static (Reference)
- **1B**: 0.3 mm Amplitude
- **2B**: 0.5 mm Amplitude
- **3B**: 0.7 mm Amplitude

Group C (35 Hz): Cooling Efficiency

- **1A**: Static (Reference)
- **1C**: 0.3 mm Amplitude
- **2C**: 0.5 mm Amplitude
- **3C**: 0.7 mm Amplitude
Figure 3.2.4: Cooling Efficiency for Vibrational Experiments: Case (a) Group B (20 Hz), (b) Group C (35 Hz), and (c) Group D (50 Hz)

As can be seen from Figure 3.2.4, the ultimate maximum cooling efficiency for all cases approaches around 84%, whether vibrational effects are applied or not. However, an important factor to note here (see Figure 3.2.4) is the time it takes to reach this maximum value. For any fixed frequency level, as the amplitude increases, the maximum efficiency is reached faster. For example, in group D, the maximum cooling efficiency can be reached in almost 65 to 70 seconds in case 3D, 0.7 mm amplitude; however, the same value can be reached by case 1D, 0.3 mm amplitude, in 120 seconds, which almost double the time. The same conclusions can be drawn when comparing the results across the groups, (e.g. effect of frequency). It should be note that the time it takes to reach the maximum cooling efficiency in any of the cases considered is equivalent to the time, which the interface temperature takes to reach close to that of the solid gallium. After that time further improvements in the cooling efficiency can be barely noticed because the system has reached thermal
equilibrium. If one reflected on the cooling efficiency the way defined above and considered the time it approaches about 70 percent, then the significant impact vibration has on the cooling rates can be further appreciated, especially when considering the runs with highest amplitude for any frequency level considered.

A general overall look at Figure 3.2.4 can show that the effect of amplitude is more dominant than the one for the frequency in the cooling process (i.e. cooling efficiency). In order to assist this argument on un-biased basis, one should examine the effect of increasing the frequency and the amplitude by the same relative percentages in reference to same common starting test. In order to illustrate the above concept, consider that the reference test is 3D, 35 Hz and 0.7 mm. On one hand, at fixed amplitude (i.e. 0.7 mm), if another test was done by increasing the frequency to 50 Hz, which corresponds to 42.86% relative change in frequency, the hot water cooling temperature difference between the two tests can be obtained at each instant of time. Similar analysis can be done if the amplitude was changed by the same relative percentage from 0.7 to 1.0 mm at fixed frequency (i.e. 35 Hz). For the sake of conducting the above analysis, additional preliminary test was done at 1.0 mm and 35 Hz and the results of the two cases above are presented in Figure 3.2.5.
It is clearly seen that the relative change in the amplitude resulted in higher temperature difference than when the frequency was changed by the same percentage. However, extensive study should still be done considering the dominancy effect of the applied signals at different ranges amplitudes and frequencies.

3.3 The Application of Bubbling Technique

The results section of the bubbling technique will be divided into two parts. The first part gives instantaneous results of the motion and heat transfer of single spherical water bubble released at the bottom of a gallium bath. These results were obtained based on the theoretical considerations and the two instantaneous empirical correlations of references [52] and [50] that were described earlier in the mathematical formulation section. In the second part, experimental results of the bubbling system for two groups of experimental runs were presented. In the first
group (Group A), we propose batch-wise operation of the bubbling system in which the experiments are conducted in a cyclic manner which alternates between hot water cooling and cold water heating; or alternatively described, gallium heating followed by gallium cooling strokes. In this case we implement water bubbling in both strokes; one time for the hot water and in the second for cold water. In the second group of experiments (Group B), we use external cooling tubes imbedded in the gallium bath at its bottom (at a horizontal plane almost close to the horizontal plane that contains the bubbling tubes). The purpose of these cooling tubes is to allow for continuous cooling of the gallium during the hot water bubbling such that a quasi-steady gallium temperature level can be attained all the time during the hot water bubbling process. More details about the conducted experiments are given in their respective sections below.

**Computed Results for Water Bubbles in Still Isothermal Liquid Gallium Bath**

At this subsection, a theoretical study of the water bubbles’ motion and heat transfer through cooler liquid gallium bath will be conducted. In order to do so, the bubbles’ diameter considerations were varied from 1 to 5 mm in a gallium bath of 30, 40 and 50 °C. However, Simple calculations based on force balance can show that in order for a bubble to detach from a 3 mm rim, the one used in the experimental study, its diameter has to be at least 4 mm. Despite that, other factors including the gallium height above the bubbling tubes can be attributed to the final (the biggest) diameter a bubble can reach till bursting. In the range of the studied parameters, 4 to 5 mm diameter bubbles are considered to model water bubbles travelling through 13.2 mm of cooler liquid gallium height.
As stated in the mathematical formulation section, one can get the velocity of the bubbles by solving Equation 2. The results can then be integrated to get the travelled distance, which is of importance to this study. Figure 3.3.1 shows the results of the integrated velocity, distance, for various bubbles’ diameters as a function of time. The curves of interest in Figure 3.3.1 are those corresponding to 4 and 5 mm diameter water bubbles. Another important factor that should be considered here, based on bubble’s diameter, is the time it takes for the water bubble to travel from the bubbler’s rim to the bursting point namely 13.2 mm. This time represents duration of heat transfer between the hot water bubble and the cooler gallium bath. In the case considered here, it is 53 and 48 ms for 4 and 5 diameters respectively. It should be noted that Figure 3.3.1 time spans till 60 ms, which covers the time required to travel 13.2 mm distance by both diameters considered. It is not expected that the assumed parameters such as the quiescent gallium conditions to be exactly encountered in the experimental testing. However, the analysis done for computing the distance can still give some insights to the real testing results and aid interpreting them as will be seen in later sections.

Figure 3.3.1: Vertical distance covered by water bubbles in the liquid gallium bath after the onset of bubble motion
Simple energy balance for a control volume of a bubble travelling in cooler gallium bath shows that the heat transferred from the hot water bubble to the gallium is equal to the heat released from the hot water bubble. For this energy balance to be solved for instantaneous bubbles’ temperature through travelling in the liquid gallium bath above the bubbler, one has to get the instantaneous Nusselt number from which the convective heat transfer coefficient could be obtained. For this purpose, two Nusselt number correlations found in [52] and [50] respectively were used and a comparison between their results was made in Figure 3.3.2(a-b). The results obtained using Nusselt number correlation of [52] and [50] are labeled as Model 1 and Model 2 respectively. In both models, the initial hot water bubble is assumed to be 60 °C traveling through gallium temperature ranging from 30 to 50 °C in a step of 10 °C, which is expected to be close to experimental conditions for both the water root (initial) and gallium. It is shown clearly that both models have resulted in a very similar temperature profile of the hot water bubble while travelling in the cooler gallium bath.
Generally, and irrespective of the model used, one can highlight few observations from the cooling rates of both diameters considered at this study. For each bubble size, it is shown that the lower the gallium temperature, the higher the heat transfer rates and, hence, the lower the bubble’s temperature, which is expected to occur. Comparing the two cases studied here, it is evident that the bigger bubble size requires more time in order to reach thermal equilibrium with the gallium bath. Recall that more travelling time means more gallium height above the bubbles (see Figure 3.3.1). In other words, 13.2 mm of gallium might not be enough, depending on the gallium’s temperature, for the bubble to transfer all its internal energy to the gallium and reach thermal equilibrium.
Experimental Bubbling Results

The bubbling experimental results obtained are divided into two main Groups: Group A and Group B. Group A includes cases 1 and 2 in Table 2.3.1 while group B includes case 3. In group A cases we alternate between cooling and heating strokes for the bubbled water as well as for the gallium bath in a batch-wise operation. On the other hand, in group B we consider quasi-steady operation of the process where by means of the external cooling tubes imbedded at the bottom of the gallium bath we could keep the gallium at almost a constant temperature level around 36°C while bubbling hot water into it at initial temperature of about 60°C.

In group A cases both the gallium and the bubbled water experience temperature variation during the process. When one of them experiences temperature drop the other experiences a temperature rise, and vice versa. Two bubbled water flow rates are considered in this group namely 25 and 30 g/s. In group B, a sample representative run was performed in which the gallium bath’s temperature was kept, by means of the external cooling tubes imbedded in the gallium bath, at almost a constant level of about 36°C during the continuous bubbling of hot water into the gallium.

In group A runs, both the bubbled water and its surrounding gallium bath start each bubbling stroke at the same initial temperature; (i.e. they start the bubbling stroke while being in thermal equilibrium with each other). This equity in temperature at the start of each bubbling stokes between water and gallium has been dictated by the nature of the procedure and the experimental setup design adopted in the conducted experiments in group A, which was described in the previous section in details.
In the water-cooling bubbling strokes, the initial temperature of the water and the gallium bath is taken to be about 30 °C. As the bubbling starts, the bubbled water temperature was gradually increased at a certain rate until it reaches about 60 °C. At each instant of this stroke, the hotter bubbles dispensed into the cooler gallium will exchange heat with the gallium and thereby become cooler as they burst from the upper gallium surface. It should be recalled that, depending on bubble diameter and the gallium bath height, the time scale at which a single bubble takes from the moment of detaching from the bubbler orifice until it emerges at the gallium’s upper surface, is in the order of 40 to 60 milliseconds. This time is by far shorter than the time at which the temperature results are collected by the data acquisition system used. Hence, the collected water bubbles temperatures at any measuring location in this work would be considered as a cumulative average of several bubbles emanating from the bubbling orifices and bursting at the upper surface of the gallium. This consequently gives a good representation of the overall bubbling process performance and hence would allow for using them as average input data for the models used to predict the bubbling process, as will be described in subsequent discussions.

Due to the low specific heat of gallium and in the absence of a dedicated mechanism to allow gallium to lose its heat during the heat exchange with hotter water bubbles moving through it, the temperature of the gallium will start building up gradually. This happens at a rate that depends on different parameters such as: the temperature of the initial hot water bubbles, the flow rate of the bubbled water as well as the size of the gallium bath. When both water bubbling temperature and that of the gallium bath more and more closely approach each other upon further dumping of hot water into the gallium, the temperature difference driving heat
exchange between the bubbles and the gallium will decrease and in the limit diminish. As this situation is approached, a call is made for “back-washing” process to bring the gallium temperature once again back to a reasonably low levels to facilitate more effective heat exchange between gallium and the hot water bubbles.

At the end of the water bubbles cooling stroke; before the back-wash process is started, the temperatures of both the water and the gallium are almost comparable and are in a high level range at around 70 °C, both. The back-wash process will now commence by pouring cold water in the water storage tank, where the pouring of hot water was stopped. The poured cold water will then mix with the hot water present in the tank at that moment and hence will lead to a gradual reduction in the temperature of the water received by the bubbling tubes from the storage tank. The resulting heat gain of the bubbled colder water into gallium will lead to gradual drop in gallium’s temperature until eventually both gallium and bubbled water approach a low level at which the gallium becomes ready once again to act as a cool sink for further hot water bubbling into it. Thereby the cycle of water cooling followed by gallium cooling is completed and a new cycle can be started. The variation of all relevant temperatures for us in this study namely the gallium bath, the bubbled water at the root of the bubbler orifices, and at the bubbles bursting sites at the upper surface of the gallium bath, was monitored and recorded over the time of the cycles conducted.

**Group A Bubbling Results**

Figure 3.3.3 presents the temperature history of the gallium, the bubbled water and the bursting temperatures for case 1 of group A (see Table 2.3.1). Two cycles with two water cooling strokes and two water heating strokes are shown in Figure 3.3.3. In this figure, The stroke of the hot water cooling for hot water
temperature range from 30 °C to about 60 °C (at a bubbling temperature gradual increasing rate of about 11 °C/min) and that for the bubbled cold water heating process (i.e. alternatively gallium cooling process) from about 65 °C to about 30 °C (at a bubbling temperature gradual decreasing rate of about 7 °C/min).

Figure 3.3.3: The Bubbling Results of Case 1 in Group “A” Over Two cycles; water flow rate of 25.2 g/s

Recall that the time scale of the bubbling at hand is much shorter than the time at which the temperatures in Figure 3.3.3 are sampled which means that at any instant in Figure 3.3.3 we can think of many bubbles completing their bubbling process during that time. Considering this, it is interesting to notice in Figure 3.3.3 that each bubble launched from the bubbling tubes orifices reach complete thermal equilibrium with its surrounding gallium bath in the case of the back-wash process; i.e. in the process cold bubble is dispensed into hotter gallium. However, some deviation from complete thermal equilibrium between bubbled hot water into its cooler surrounding gallium environment is noted. The difference between these two
opposing processes; water bubbles cooling Vs. water bubbles heating, in regards to
the ability of water bubbles to closely reach complete thermal equilibrium with
gallium in the bubbling system used in the experiments is something interesting that
the theoretical calculations reported in the previous section would not be able to
recognize. For the theoretical models used in previous section either bubble cooling
or heating would lead to the same temperature-variation scenarios but in opposite
directions, as long as the model computations were run under similar temperature
limits (difference) between the bubble and the gallium bath temperature.
Consequently, the difference observed in the two opposite directions in regards to
ability of water to attain equilibrium with the gallium has to do with some
phenomenon taking place in the bubbling gallium bath that have not been considered
in the theoretical calculations of the previous section.

In the case when hot water bubbles are initially injected into a cooler gallium
bath, as these hot bubbles move in the upward direction they cause the gallium
temperature in their immediate vicinity in the lower parts of the gallium bath to get
heated. The increasingly-becoming hot gallium in these lower parts of the gallium
body will try to convey their heat to the upper gallium regions. At the same time, the
convectional currents in the upward direction caused by the upwardly-rising hot
water bubbles will assist enhancing this heat transfer from the lower parts of the
gallium tank to the upper ones. As a result of these scenarios, as the water bubbles
progressively reach upper heights in gallium, they will encounter somehow gallium
that has already captured some heat from the bottom parts and got heated. As such,
the bubbles will encounter lower driving force for heat exchange (i.e. lower
temperature difference with gallium) as they travel upwards into the gallium which
ultimately will deprive them from the chance of reaching closer the temperature of
their gallium surrounding. The deviation between the bursting bubbles and the
gallium temperatures reported in Figure 3.3.3. Alternatively said, the bursting
bubbles reach thermal equilibrium with hotter gallium than the one present at the
bottom of the gallium tank.

In contrast to the above scenarios related to hot bubble cooling in gallium
bath, very brilliantly the cold water reach complete thermal equilibrium with gallium
at the gallium temperature reported in the lower parts of the bubbling tank (see
Figure 3.3.3). This means that the cold water bubbles, although from momentum
point of view they may be under similar force conditions as the hot bubbles injected
into colder gallium, might be under clearly different thermal boundary conditions
during their trip from the bubbling orifices up to the upper gallium surface. Here
when the cold bubbles emerge from the bubbling tubes orifices at the bottom part of
the gallium bath, they will capture some heat from the hotter gallium surrounding
them. As a result, the increasingly-becoming colder gallium in the bottom parts will
try to capture heat from the still-hotter gallium in the upper parts of the bubbling
tank. However, the tendency of heat transfer from top to bottom parts of gallium
would be countered by the convectional currents and motions caused by the
upwardly travelling bubbles. This means that by slowing down the heat transfer from
the hotter gallium at the top of the tank in the downward direction by upwardly
travelling bubbles, the cold bubbles will have more chance to see hotter environment
as they travel higher heights. So, they will be granted a better chance to get heated
more and more up the limit of reaching complete thermal equilibrium with the
gallium bath as they emerge and burst at the top surface of the gallium. Such an
opportunity was not granted for the bubbles in the case of water cooling process,
discussed above. The above explanations highlight two important things: (1)
bubbling system will function more effectively when it aims at heating cold bubbled water into hotter gallium bath, and (2) the theoretical or empirical models that are based on assuming uniform temperature around the bubbles all the time during their trip in the gallium, would not be expected to exactly predict the instantaneous phenomena that takes place during the bubbling process. By providing these models with more accurate description of the special variations of the temperature throughout the bubbled-through bath (gallium in this case) the ability of these used models is expected to drastically improve. Yet, the used models with somewhat reasonable prescription of the bath temperatures, might still be performing well and predict in a good way the instantaneous heat exchange between the bubble and its surroundings. This is so especially when considering the level of empiricism in the coefficients included in these models which would mean that by some further tuning of these coefficients to fit the bubbling system at hand, more accurate predictions can be expected of these empirical models. More discussions on the above will be carried out based on the models’ results to be discussed in a subsequent section. Finally, it is interesting to see that in the beginning of the hot water bubbling into colder gallium, the water attains (although not complete equilibrium) a good level of temperature reduction at the moment they burst out of the gallium body. As the temperature difference between the gallium and the bubbled hot water diminishes more and more, the deviation form thermal equilibrium between the two media becomes more and more evident (see Figure 3.3.3).

In regards to the repeatability of the obtained cyclic results, taking into consideration the slight deviations in the way the two water-heating/water-cooling cycles are conducted, the level of repeatability attained is considered to be very satisfactory and reflect reasonable results that can be used to explain physical
phenomena related to the bubbling heat exchange processes studied. It should be noted that the slight deviation highlighted above is to a great extend attributed to the absence of a completely dedicated automatic system to conduct and control the different parameters as: flow rates, temperatures, water levels in the storage tanks, etc.)

Figure 3.3.4 shows the temperatures of the bubbling system for the case when the bubbled water flow rate was increased to 30 g/s. The results reported in Figure 3.3.4 were obtained in a continuous operation directly after we completed the collection of the data for case 1. Figure 3.3.4 show two consecutive cycles of the bubbling system. Both cycles start at the same temperature of about 30 °C, but in the first of these cycles we went to higher bubbling temperatures of about 80 °C while in the second cycle we conducted it under water bubbling temperatures comparable to those adopted in the second cycle of case 1 (see Figure 3.3.3 and Figure 3.3.4).

Figure 3.3.4: The Bubbling Results of Case 2 in Group “A” Over Two Cycles; Water Flow Rate of 30 g/s
Irrespective of the maximum temperature of the hot water adopted, both cycles presented in Figure 3.3.4 show the same trends and both are consistent with trends observed in Figure 3.3.3 for less flow rate. Simple calculations (done before) have shown that the kind of bubbling scenarios in case 2 still results in bubbling of finite number of discrete bubbles which will not be having chance to collide or merge with each other after being dispensed from the bubbling orifice. This may mean that, in principle, the physical phenomena governing the momentum and heat transfer between the released bubbles and their surrounding gallium medium are the same in both cases 1 and 2, leaving any quantitative differences between the results of the two cases to be attributed to the quantitative details that differentiate the operating conditions in both cases. As such, the overall general discussion on Figure 3.3.3 presented above, is still valid for the case of Figure 3.3.4. It should be noted that in case 2, relatively rapid increase rate of the bubbled water temperature was implemented in cycle 1 of about 20 °C/min (consequently rapid heating of the gallium would follow). However, for cycle 2 the rate of bubbling temperature rise was made to be comparable to the one implemented in the cycles of case 1 (about 12°C/min). Regarding the bubbling temperature decrease rate during the two back-wash strokes in cycle 2, they were in the range of 8 to 6 °C/min, which are comparable to the level implemented in the water-heating strokes of case 1 cycles. Yet, any slight quantitative differences between the cycles of cases 1 and 2 can still be attributed to the slightly larger mass flow rate of bubbled water in case 2 which will mean more rate of exchange with the gallium that would directly reflect on heating or cooling rates of both gallium and bubbled water.
Validation of Empirical Modeling Against Experimental Results

In this subsection we test the three empirical models described in the mathematical formulation section, namely the two instantaneous models from references [50, 52] and the global average model of reference [46], against the experimental results obtained in case 1. To facilitate clear comparison between the models predictions and the experimental results, we select representative sample segments from Figure 3.3.3 (case 1) spanning over time periods in which the gallium temperature did not show significant changes. This would be the case in the early stages of the water-bubbles cooling strokes and in the late phases of the water-bubble heating (the backwash) strokes. These segments are presented in Figure 3.3.5 (a-d).
Case 1: Bubbled Water Flow = 25.2 g/s; Ga above Bubbler tubes: 13.2 mm

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Case 1: Bubbled Water Flow = 25.2 g/s; Ga above Bubbler tubes: 13.2 mm

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Figure 3.3.5 (a-d): Comparison between experimental results of case 1 and the three empirical models

It should be recalled that the time scale for the bubbling process of one single bubble from the moment of detaching from the orifice until it bursts at the upper gallium surface is much smaller than the time at which the experimental data of Figure 3.3.3 are collected. This means that we can take data pairs of gallium temperature and bubbling root temperature (at the bubbler tube) from Figure 3.3.3 at any moment of time to be used in the models computations as self-standing individual case studies representing a single bubble moving throughout a stagnant fixed temperature gallium bath environment. As such, the experimental data points presented in Figure 3.3.3 would secure rich and wide enough database for testing the three empirical models at different conditions of gallium bath and initial bubbling temperatures.
Figure 3.3.5 (a) and (b) express reasonable level of repeatability of the experiments conducted in the range of temperatures considered in these figures. Moreover, they give results for the cooling process of hot water with gradual water temperature increase from 30 °C to about 45 °C, before gallium start gaining higher temperatures above its initial value of 30 °C. As such, the gallium temperature range presented in Figure 3.3.5 (a) and (b) represent almost a quasi-constant temperature of the gallium bath for the bubbled water in these two figures.

The agreement between the models predictions and the experimentally obtained bursting temperature of the bubbles in Figure 3.3.5 (a) and (b) is remarkable. All three models reproduce the experimental results so closely. Interestingly, all three models, regardless of the way they have been obtained nor the kind of fluids involved in their construction, they all merge and predict almost the same results. Moreover, the two instantaneous models predict more closely the experimental bursting temperatures when the bubble diameter is taken to be 4mm. As can be concluded from a simplified force balance at the moment of bubble detachment form the orifice, such an average diameter value of about 4mm would be a reasonably justified choice for the bubbling system and operating conditions adopted in this study. Variations of the bubble size as the bubble travel towards the upper surface of the gallium due to variations in the pressure with height have not been taken into account. The fact that bubbled water is incompressible material make us believe that using a fixed diameter for the bubbles throughout their whole trip in the gallium bath may not lead to any serious errors. This is especially true when considering the nature of the empiricism in the models used which may mask out such details and collectively lump sum them inside the empirical constants used. The same argument might also be generalized to the fact that we do not account for
distortions and exact deviations from the spherical shape of the bubble as it moves relative to its gallium environment which in the absence of detailed numerical simulations or visualization imaging, cannot be accounted for.

It is interesting to consider that the predictions based on the two instantaneous models [50, 52] are obtained based on the original form without the need to alter the empirical coefficients in these models, which consequently give these models the nature of being applicable to a wide range of conditions and materials (other than the ones based on which they were constructed). On the other hand, the global model proposed in [46] was used with a good level of success in the study and reproduced the experimental results for the bursting temperature. However, his was the case after having introduced a slight upgrade to its empirical coefficient multiplier (changed from 0.027 to 0.19), leaving the rest of the empirical coefficients in that model as they were given in the original form in [46]. This also reflects a wide applicability nature of that global model to cover different operating conditions and fluids, far beyond the original ones used to build it in the first place. The above discussion indicates that the used models are less sensitive to the exact materials used, leaving most influences to be coming from the bubbling hydrodynamics and temperature ranges considered in the experimental runs.

Figure 3.3.5 shows the experimental results against models predictions for the reverse process when the heat inside the gallium bath was flushed in the backwash process by bubbling colder water into gallium. Again, the time span of the selected two segments of the two cycles in case 1 were chosen such that little to moderate changes in the gallium temperature are observed. This furnishes reasonable infrastructure for the three models used over a good range of initial bubbling
temperatures. Figure 3.3.5 (c) shows the process of cooling the gallium, which has temperatures in the range of about 60 to 55 °C, by dispersing into it water bubbles that have initial temperatures gradually decreasing form levels of about 47 to 40 °C. Figure 3.3.5 (d) gives predicted results for the case of gallium bath and bubbled water temperatures at quasi-constant levels of about 45 °C and 35 °C, respectively. As concluded based on Figure 3.3.5 (a) and (b), similar conclusions can be drawn based on Figure 3.3.5 (c) and (d) namely that all three models superiorly predict the phenomena at hand with the instantaneous models being able to do so when the initial bubble diameter is taken to be 4 mm.

Testing models predictions based on comparison with the experimental results is shown in Figure 3.3.6 where segment from the cyclic results presented for case 2 in Figure 3.3.4, had been selected.
Figure 3.3.6 (a-c): Comparison between experimental results of case 2 and the three empirical models

Similar to the first case, 25.2 g/s, the models have predicted the bursting bubble temperature for the second case, 30 g/s, to a very good extent (±2 °C). The
selected segments for cooling the hot water bubbles, Figure 3.3.4(a), was intentionally presented to show higher operating temperatures than the first case. Yet, the models can still predict the bursting bubble temperature regardless of the flow rate selected, 25.2 g/s or 30 g/s, and as long as the bubbles do not hit each other and form a continuous flow from the bubbler to the bursting points as shown earlier in this text. Although the three models have given close predictions to the measured bursting bubbles temperatures, one can notice from Figure 3.3.4 (a-c) that the computed results of Model 3 and the 4 mm cases from Model 1 and 2 have given better predictions.

**Group B Bubbling Results with Validation of Models Predictions**

In this subsection, sample representative result is discussed for the bubbling system when the external cooling tubes are employed to keep gallium’s temperature almost fixed. This case applies while dumping heat into gallium by the hot bubbles; avoiding thereby the need for alternating in a batch-wise manner between water cooling and water heating bubbling processes and giving way for continuous non-stop quasi-steady hot-water-bubbling operation of the system. This approach, although suitable for continuous operation, faces the challenge of the need for extra equipment and tubing to be added to the system. Moreover, a serious challenge will be present if reasonably low gallium temperatures are to be maintained and at the same time if the cooling fluid that circulate in these cooling tubes is one that has temperatures that do not differ much from the gallium temperature level desired. In such a case, a very large tubes’ surface area and small diameter tubes would be required to cool down gallium during the bubbling of hot water. This will entail additional complexity in the system and consequently larger initial, maintenance, and
running costs of the system, especially if large amounts of hot water are to be bubbled into the gallium. Based on the above, the main reason behind including this section in this study is to demonstrate the capabilities of the three models employed to predict the experimentally obtained bursting water temperature over a good range of time. This happens under quasi-steady operation of the system with gallium and hot water being apart from each other in a way that was not possible to attain using the backwash operation of the system used in its current version.

Figure 3.3.7 shows the quasi-steady results for the case when the flow rate of bubbled hot water is 11 g/s. Reaching higher bubbling flow rates upto the levels considered in cases 1 and 2, given the high level of the intended quasi-steady initial bubbling temperature of about 60 °C, would require excessive cooling tubes matrix to be imbedded in the gallium bath. However, this is unplanned at this stage of this research. Yet, for the sake of demonstrating the capabilities of the computing tools used and for the sake of testing the performance of the system under temperature difference between gallium and hot water bubbles of about 25°C, the present setup is believed to be sufficient to fulfill this goal.
Figure 3.3.7: Comparison between experimental results of case 3 and three empirical models under quasi-steady bubbling operation using external cooling tubes for the gallium (group B)

Over almost all of the reported time segment shown in Figure 3.3.7, the three models predict fairly well the experimentally obtained bursting temperatures to within approximately ±2 °C. The bubble diameter of 4mm results in a remarkable match between the predictions of the first instantaneous model [52] and the experimental results over most of the time segment shown in Figure 3.3.6. However, contrary to the previous observation made based on Figure 3.3.5, the second instantaneous model [50] reproduces the experimental results in this case when the bubble diameter is taken to be 5mm in a better way than when a diameter of 4mm is adopted. This somewhat different conclusion (regarding the ability of the second model to relate to the experiments in a somewhat different way than observed before), would not, however, lead us here to make final conclusive arguments in this regard. But it will instead be interpreted as a normal consequence of the fact that we
are using empirical correlations that attempt to fit various operating situations and such variations would rather be something expected. With the above being said, we still can conclude that overall the three models perform very well in predicting the phenomena tested herein experimentally. Moreover, the fact that the quality of the predictions of the models in case 3 are a little bit less than what has been possible in case 1 reported in Figure 3.3.5 above, may be attributed to two factors. Firstly, the flow rate of the bubbled water in case 3 is clearly less than that in cases 1 and 2 which would imply less frequent bubbling and hence bursting events at the upper surface of the gallium. This may consequently have some impact on the measured bursting temperatures rendering them to be of a lower accuracy as good representation of actual bursting temperatures of bubbles under the studied conditions. The second factor might be attributed to the bigger difference between the gallium bath and the initial bubbles temperatures in case 2 which may lead to more pronounced error in the predictions. Finally, in the absence of more information, the prediction of the bubble diameter in the computations whether to be 4 mm or 5 mm is susceptible to uncertainty that can add to any deviations between the models predictions and the measured bursting temperatures.
Chapter 4: Conclusion

As highlighted above, cooling hot fluids is vital for thermal management applications. The work adopted in this thesis introduces relatively new techniques that are proven to be much more effective in cooling hot fluids (e.g. hot water) than the conventional ones (e.g. conventional heat exchangers). In this work, hot water was cooled by placing it in direct contact with superior heat transfer media (e.g. liquid gallium) to dump the heat from it easily. This technique provides promising results only if the liquid gallium’s temperature was at sufficiently lower level than the hot water (e.g. cooling deriving force). In order to do the above successfully, three main techniques were adapted. The first technique aimed to increase the apparent specific heat of gallium by integrating inorganic PCM into it. This had resulted in smoother temperature raise for extended time. However, the second and third techniques were intended to achieve the required heat removal in a short time before the gallium gets much heated. In the second technique, the heat sink, including a solid gallium block, was vibrated and resulted in higher exploitation of the latent heat content of the gallium. However, the third technique aimed to increase the convective heat transfer between the hot water and the liquid gallium by discretizing the to-cooled water into small bubbles travelling through a cooler liquid gallium bath. In what follows are some detailed conclusions and recommendations for the three techniques discussed above.
3.4 Research Conclusions and Implications

In this subsection, the main conclusions on the three techniques adapted in this thesis will be summarized under subsections corresponding to each of them.

3.4.1 Heat Sink Integrating Un-encapsulated Discrete PCM Cavities

A novel design of a heat sink incorporating discrete cavities filled with PCM material imbedded within a solid gallium block has been built and tested experimentally. The design makes it possible to benefit from the privileges gallium offers in regards to enhancing heat transfer from a hot source. It also gives the chance to benefit from gallium phase change during melting and the associated control over any significant tendency for temperature rise in the melted gallium. Incorporating PCM within solid gallium in the ways proposed in this section enables alleviating restrictions on the sink performance posed by the limited specific heat capacity of gallium.

The PCM cavities imbedded within the solid gallium were arranged in two different ways; structured as well as randomly sized and randomly distributed PCM chunks. The structured arrangement shows only slightly better sink performance than the random configuration. The obtained results with PCM integrated within solid gallium sink were compared with their counterparts using a gallium sink but without integrating PCM. The comparison reveals effective role played by PCM in gallium in relation to maintaining gallium’s temperature at lower levels during heat exchange with hot sources; hence, facilitating faster cooling rates of the source.
3.4.2 The Application of Vibrational Signals to the Heat Sink

The cooling performance of hot water under vibrating heat sink conditions was studied experimentally. The heat sink was designed such that it can be firmly fixed to the shaker’s base. Vertical sinusoidal waves with a range of frequencies (20, 35 and 50 Hz) and amplitudes (0.3, 0.5 and 0.7 mm) were applied to the setup. In addition, a reference baseline case under static conditions was conducted to assess the impact of vibration on the cooling performance. Prior to starting the test, the gallium was in a solid state at a 15 °C. In all tests, temperature measurements were taken at the gallium bulk, interface and hot water and a comparison among the different tests was done. Factors such as the cooling efficiency and time saving were calculated to quantify the results.

The results have shown that the vibration is a promising technique to cool hot fluids. The main contribution of the vibration was mainly attributed to the enhancement in the intermolecular activities between the liquid gallium-water and liquid gallium–solid gallium, interfaces. In general, the implementation of vibration in the range of the studied parameters resulted in higher cooling rates from the hot water if any of the amplitude or frequency was increased. This has resulted in much time savings in cooling the hot water in overall. Also, the results have shown that higher values of amplitudes and frequencies resulted in reaching the maximum cooling efficiency in much shorter time when comparing to the static testing. Simple comparison between the results obtained when increasing the frequency and amplitude one at a time with the same relative percentages has shown that the effect of amplitude is more dominant than frequency in the ranged of the studied parameters.
3.4.3 The Application of Bubbling Technique

Hot water was disintegrated into small particles that were dispersed into a cooler liquid gallium bath. Four bubbling tubes were used. On each tube drilled are 12 holes, each with a diameter of 3 mm. The amount of gallium used in the bubbling container is such that it makes a gallium height above the bubbling tubes of about 14 mm and in some experiments 18 mm. The rate of heat exchange between the bubbled water particles and their gallium surrounding was investigated by monitoring and recording the temperature of the gallium bath, hot root hot water and cooler bursting water. Two levels of hot water bubbling flow rates were considered in the experiments; in group A about 25 to 30 g/s and in group B about 11 g/s. The flow levels of group A were selected such that bubbling of the water will be taking place at a reasonable rate as discrete bubbles emanating from the bubbling orifices.

For the ranges of bubbling temperatures and flow considered (about 45 to 75°C) in a gallium bath with initial temperature of about 30 °C, clear enhancement in cooling of the water is attained while still requiring a gallium bath with very small depth of about 14 to 18 mm. For the tested cases, water cooling rates ranging roughly from about 1.0 to 2.0 kW were possible to achieve. Simple reasoning and heat transfer calculations can reveal a need for a much larger size system so that the same cooling rates can be achieved using conventional heat tubes-based heat exchangers. Three models were used to predict the cooled water temperature. Two of them were based on instantaneous prediction, however, the third one is based on a global overall effect of the whole process. All models showed outstanding performance as they, to a very good extent, could reproduce the experimental data.
3.5 Managerial Implications

In this subsection, the suggested future work for further developments in the addressed techniques will be provided.

3.5.1 Heat Sink Integrating Un-encapsulated Discrete PCM Cavities

- Including larger amounts of PCM in the gallium sink, up to about 10 percent by volume, gives better performance for the gallium/PCM matrix as a heat sink. However, stuffing the sink with further PCM content beyond this limit may cause issues regarding the smoothness of the interconnectivity as well as the free heat transfer passage ways among the different parts of the gallium body in the sink. Hence this can lead to adverse effects on the overall sink performance. Further optimization of the sink in relation to PCM loading is to be pursued in future research work.

- It is interesting to assess in a future work the impact of the proposed discrete PCM cavities approach by comparing it with one in which small scale encapsulated PCM particles (at the micro, nan, or mesoscale) are dispersed homogeneously within gallium. This later approach would be a challenging one to accomplish and might not be so needed, especially if the simpler-to-accomplish discrete approach proposed in this study showed a better performance in cooling hot sources. Coming up with conclusive results on the above point is worth considering in upcoming future researches.
3.5.2 The Application of Vibrational Signals to the Heat Sink

- It is recommended to study the effect of vibration on different hot fluids than water; having different viscosities (e.g. oils) and other thermophysical properties so that the impact of vibration on the cooling performance could be further understood for wider range of fluids under various vibration parameters.

- Also, it will be so interesting to examine the molecular-level dynamics and heat transfer phenomena taking place at the interface between the gallium and the cooled fluid and relate them directly to the heat transfer enhancement process. In addition, one can also try to locate the melting wave of gallium as a function of time so that the amount of heat passed to melt the gallium can be instantaneously calculated at any moment during the process.

- Finally, further studies with wider range of the applied signals shall be done so that the vibration parameters can be correlated with dimensionless quantities to quantify the cooling performance of the heat sink.

3.5.3 The Application of Bubbling Technique

- It is recommended to use a separate bubbling tank for the cold water used in the backwash process rather than using a single tank for both the backwash cold water and the to-be-cooled hot water. This will shorten the time needed to bring the liquid gallium back to its initial low temperature level for repeated use to cool another hot water batch.

- The empirical models used here to predict the heat transfer in the bubbling process where all constructed in the first place based on experiments that
used fluids and material different from the ones used in this thesis. Hence, the attained very close match between the present experimental results and the models’ predictions should be perceived with caution and (although very interesting and encouraging), still worth further considerations and validations based on more rigorous experimental testing under various fluids and operating conditions.
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List of Publications

1. S.Omari, A.Ghazal, E.Elnajjar, “A new approach using un-encapsulated discrete PCM chunks to augment the applicability of solid gallium as phase change material in thermal management applications,” Accepted in Energy Conversion and Management.

2. S.Omari, A.Ghazal, E.Elnajjar, “A novel concept to enhance the applicability of solid gallium as phase change material for heat sinks by integrating within it discretely distributed chunks of un-encapsulated PCM,” Accepted in International Communications in Heat and Mass Transfer.

Patent

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