Thermal performance investigation of vapour chamber heat spreader

A thesis submitted in fulfilment of the requirements for the degree of Doctor of Philosophy

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Declaration

I certify that except where due acknowledgement has been made, the work is that of the author alone; the work has not been submitted previously, in whole or in part, to quality for any other academic award; the content of the thesis is the result of work which has been carried out since the official commencement date of the approved research program; any editorial work, paid or unpaid, carried out by a third party is acknowledged; and, ethics procedures and guidelines have been followed.

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Jason Velardo

September 2019
Acknowledgements

This has been quite a journey. I never planned to do a PhD when I started out at RMIT University as an 18 year old, yet here I find myself years later writing the Acknowledgements section of my thesis. I still don’t really know how I got here, but I’m glad I did. One thing I am certain of is that there were many people who helped me get here along the way.

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<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat</td>
<td>J/kgK</td>
</tr>
<tr>
<td>$\frac{dP}{dT}$</td>
<td>Slope of phase diagram</td>
<td>Pa/$^\circ$C or Pa/K</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient</td>
<td>W/m$^2$K</td>
</tr>
<tr>
<td>$h_{fg}$</td>
<td>Latent heat of evaporation</td>
<td>J/kg</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity</td>
<td>W/mK</td>
</tr>
<tr>
<td>$K$</td>
<td>Permeability</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$L_a$</td>
<td>Adiabatic length</td>
<td>m</td>
</tr>
<tr>
<td>$L$</td>
<td>Effective length</td>
<td>m</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>$\bar{M}$</td>
<td>Molar mass</td>
<td>kg/mol</td>
</tr>
<tr>
<td>$N$</td>
<td>Number of measurement points</td>
<td>-</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>Pressure difference, Pressure loss</td>
<td>Pa</td>
</tr>
<tr>
<td>$\Delta P_c$</td>
<td>Capillary pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>$q$</td>
<td>Heat flux</td>
<td>W/m$^2$</td>
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<td>$q'$</td>
<td>Volumetric heat generation rate</td>
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<tr>
<td>$r$</td>
<td>Radius, Capillary radius</td>
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<tr>
<td>$R$</td>
<td>Resistance</td>
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<td>Spreading resistance</td>
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### Greek symbols

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<td>Porosity</td>
<td>-</td>
</tr>
<tr>
<td>μ</td>
<td>Dynamic viscosity</td>
<td>Pa·s</td>
</tr>
<tr>
<td>ν</td>
<td>Specific volume</td>
<td>m³/kg</td>
</tr>
<tr>
<td>ν_lv</td>
<td>Specific volume difference between vapour and liquid phase</td>
<td>m³/kg</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>σ</td>
<td>Surface tension</td>
<td>N/m</td>
</tr>
<tr>
<td>σ̂</td>
<td>Accommodation coefficient</td>
<td>-</td>
</tr>
<tr>
<td>ω</td>
<td>Weighting coefficient</td>
<td>-</td>
</tr>
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### Subscripts

- **cond**: Condenser, Conduction
- **evap**: Evaporator
- **l**: Liquid
- **v**: Vapour
- **eff**: Effective
- **air**: Air
- **hs**: Heat source
- **mid**: Midpoint
- **surf**: Surface
- **int**: Interface
- **TC1**: Thermocouple 1 location in Figure 3.20
- **s**: Solid
- **∞**: Free stream

### Constants

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>π</td>
<td>pi</td>
<td>-</td>
</tr>
<tr>
<td>(\bar{R})</td>
<td>Universal gas constant</td>
<td>J/molK</td>
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Abstract

The seemingly endless development of integrated circuits has seen a dramatic increase in the usage of electronic devices over the past few decades. It would be difficult to imagine life without devices like computers, laptops, tablets and mobile phones to name a few. Yet all of these will unavoidably generate heat during operation. Removing this heat in an acceptable manner has been the subject of much research and development in order to cope with the ever increasing electronic demands. Two-phase heat transfer devices generally form part of the thermal management system for many electronic devices due to their excellent performance and passive operation. Of these two-phase heat transfer devices, the vapour chamber acts like a heat spreader which can effectively transfer heat from concentrated sources to larger sinks. The intention of this research was to perform an experimental and numerical investigation into vapour chamber heat spreaders. The experimental investigation consisted of the development of a vapour chamber with both sintered powder and mesh structures for the evaporator wick. This composite wick was designed such that the inherent strengths in each wick structure could be better utilised. An experimental setup was also developed to measure thermal performance. Through detailed measurements, spreading resistance was determined. It was found that the vapour chamber had superior performance to that of a solid copper heat spreader with reductions of up to 50% observed in spreading resistance. Further measurement of the vapour chamber temperature distribution highlighted some interesting heat transfer mechanisms that governed performance. Thereafter, a simple numerical model was introduced. It assumed that all heat transfer mechanisms were conduction based and thus the detailed aspects of two-phase heat transfer were replaced in favour of simple conduction mechanism. Through this model, which was validated against the solid copper heat spreader, the effective thermal conductivity of the vapour region was determined for a number of cases. It was found to be a function of heat source size, with reduced heat source sizes leading to reduced values for effective thermal conductivity of the vapour region. This value has been rarely explored in the literature however it provides useful information to designers of thermal systems which use vapour chambers. Some of the intricacies of this value were explored in further detail. Finally, an investigation was performed into the composite evaporator wick structure using the numerical model. This investigation helped determine practical guides for wick structures and provided suggestions for improving thermal performance of the vapour chamber. The overall research presented here will help in the continued development of the vapour chamber so that they can better achieve future demands.
Chapter 1 – Introduction

1.1 Research overview

Thermal management of electrical and mechanical systems has traditionally created problems for thermal designers. Removing heat from these systems is critical to maintaining safe and efficient operation. For the electronic industry, elevated operating temperatures are known to reduce performance and reliability of an integrated circuit (IC) [1-7]. This can also cause thermal expansion and thermal cycling issues that can lead to mechanical failures [7, 8]. Thus there is large demand for thermal management systems which can adequately cool these.

Heat transfer via natural convection of an IC is inadequate to keep it operating below its operational limit, commonly 80-100°C for an IC such as a CPU [9-12]. An external heat sink is used to increase the surface area through which heat is dissipated, thereby reducing the external convective resistance. This helps to transfer larger heat loads while maintaining safe operating temperatures. Conventionally, this can be achieved through a solid metal heat sink made from either copper or aluminium, which have been well studied [13-15]. Some typical designs of these have been shown in Figure 1.1. They generally consist of a solid base to which fins extend from. Heat conducts through the base of the heat sink to the base of the fins and then passes through the fins themselves. Heat is removed from the heat sink through convection over the fins and base. A fan is generally used to provide forced convection through the heat sink. Geometry and configuration of the heat sink base and fins vary widely among manufacturers.

Figure 1.1 – Some common heat sink designs using copper and aluminium technology (images courtesy of RMIT Thermodynamics Laboratory).
Improvement in heat sink performance is often achieved in heat sinks by employing phase change heat transfer devices such as heat pipes, loop heat pipes, pulsating heat pipes and vapour chambers (the vapour chamber will hereafter be referred to as a VC within this work). These devices operate on similar principles [16, 17] albeit with design differences. Heat is absorbed in the evaporator section, causing the working fluid inside the device to evaporate. The vapour then travels to the condenser section where it releases its heat, causing it to condense back into a liquid. The liquid then returns to the evaporator through capillary action via a wick or through gravity, thus allowing the cycle to repeat. As the latent heat of evaporation for working fluids is large and the phase change process is essentially isothermal, these devices are able to transfer large amounts of heat with minimal temperature drop. This makes them very attractive heat transfer devices. The concept of the heat pipe was first proposed in 1942 by Gaugler [18] and Grover first created a heat pipe in 1963 [19]. Through this experimentation, the excellent heat transfer capabilities of heat pipes became apparent. The use of two phase devices for thermal management of electrical systems is not novel. Heat pipes have long been used in desktop, laptop and mobile systems (among others) for thermal management. Some examples of this are shown in Figure 1.2.

As opposed to the well-established use of heat pipes for thermal management, the use of VCs is comparatively limited. The VC is a relatively new two phase heat transfer device which often takes the form of a thin plate. The general design of the VC is shown in Figure 1.3.
The objective of the VC is to spread heat from concentrated heat sources, thereby lowering the heat flux and creating a more manageable thermal scenario. This objective differs from the heat pipe which is designed to transfer heat along the length of the pipe due to its tubular design. The cross section shown in Figure 1.4, which includes the flow of heat and circulation of working fluid, illustrates how the VC is able to achieve this objective.

For the VC, heat (often from a high heat flux device) is input through the evaporator region. Then through two phase heat transfer mechanisms, which are discussed in great detail in this work, this heat is spread by the working fluid to the condenser region. It is through these mechanisms that VCs excel at thermal spreading scenarios. For these reasons, they are often
used as heat spreaders in thermal management systems for electronic devices. They are very well suited to these systems for a variety of reasons:

- High heat flux of the IC
- Strict space constraints on the system
- Reliable operation
- Lightweight
- Good performance

It is common in these scenarios that the thermal spreading resistance contributes significantly to the total thermal resistance of the solution [1, 13]. This thermal spreading resistance arises in the base of the heat sink which is subject to the high heat flux source. This resistance has been well studied for solid metal heat spreaders [20-24]. It is found to be mainly a function of geometry and conductivity. The geometry of the heat spreader often cannot be drastically altered by the designer and even with high conductivity metals such as copper and aluminium, the spreading resistance can still be significant. The VC can alleviate these issues by replacing the base of the heat sink in a conventional heat sink. This has been shown schematically in Figure 1.5 where a VC is located between the concentrated heat source and the heat sink.

*Figure 1.5 – The schematic setup used for most VC thermal management solutions where the VC replaces the traditional solid metal heat spreader. Note that the heat sink in this case still has its own base however the VC is responsible for thermal spreading.*
As it is highlighted in the following sections, the VC is able to improve performance by reducing the total thermal resistance of the thermal management solution. Some examples of thermal management systems incorporating VC as heat spreaders have been shown in Figure 1.6. Based on industry consultation, the cost associated with VCs makes them approx. 5-10 times more expensive than copper heat spreaders (depending on the size and requirements).

Figure 1.6 – VCs used as heat spreaders in thermal management solutions (left: image courtesy of Fujikura, right: image retrieved from [25]).

The VC is still the subject of research within the scientific community, with current commercial use mostly in electronic applications such as computing. This has extended to some degree into flight and space based applications [26-28], however the majority of their use is still land based. Some of the literature on the applications and benefits of VCs will now be briefly discussed. The term ‘total thermal resistance’ is commonly used in the following paragraphs; refer to section 2.1 for a definition of this term.

Nguyen et al. [29] showed that a VC system outperformed both copper and aluminium systems. The total thermal resistance for the VC system was 0.25K/W compared to 0.31K/W and 0.36K/W for the copper and aluminium systems, respectively. They also highlighted that these improvements could increase as the heat sink base grew larger or as the heat source area reduced i.e. the degree the spreading increased.

Wu et al. [30] investigated the use of a VC system for cooling of high density blade servers. These are thermally challenging environments, and the development of these servers has been limited by thermal management systems according to the authors. They identified that spreading resistance can contribute as much as 72% of the total thermal resistance in these systems. A model was introduced which suggested that VC solutions could improve
performance by 20-40% compared to conventional metal solutions. They also suggested that weight savings of 50-60% were possible with the VC system.

Wang [31] developed a VC thermal management system for a GPU that could reduce source temperatures by over 20°C compared to conventional systems, including a heat pipe system. At 240W, the total thermal resistance of the VC system was 0.265K/W compared to 0.335K/W for the heat pipe system.

Wang et al. [32] studied the performance of different thermal solutions for a board level high performance flip-chip ball grid array. The VC system had total thermal resistance of at least 14% better than the copper system, and further improvements were observed as the heat source size reduced.

Naphon et al. [33] performed further studies on the use of VC systems for CPU cooling. CPU temperature reduced by 6.89% at 90% operating load, and the total thermal resistance of the VC system was found to be 60-80% that of the solid metal solution. Further, the energy consumption of the VC system also reduced by over 10% at 90% load compared to the metal system. Naphon and Wiriyasart [34] compared the thermal performance of VC systems for hard drive cooling. The hard drive temperature was able to be maintained 10°C lower with the VC system at full load. Wang [35] highlighted the improvement of a VC system over a system using heat pipes embedded in the heat sink base. The VC system had total thermal resistance of 0.232K/W compared to 0.259K/W for the heat pipe based system.

Reyes et al. [27] constructed a VC heat sink for use in avionics applications and compared its use against an aluminium heat sink. At the same component hot side temperatures, the VC heat sink was able to transfer 95W compared to 89W for the aluminium heat sink. The VC heat sink was also found to comply well with the strict regulations of avionic systems and outperformed the aluminium heat sink in natural convection cases.

Wang [36] explored the use of VCs for applications involving LEDs with thermoelectric cells. At LED input power of greater than 6W, the VC system was advantageous. The maximum power output from the thermoelectric cell was also higher when using a VC, with 168mW output compared to 116mW for the non-VC solution. In the non-VC solution, a higher proportion of heat was rejected directly to the atmosphere, without first passing through the thermoelectric cell. It was also suggested that the VC solution could increase the lifespan of the LED by reduced operating temperatures. Tang et al. [37] investigated the use
of a grooved VC for LED thermal management. The VC was directly integrated into a heat sink. The VC system had total thermal resistance which was 16.5% less compared to an aluminium heat sink of the same dimensions. This also led to favourable LED performance in terms of luminous efficacy and radiant efficiency as the operating temperature of the LED was over 13°C lower at the highest input current. The VC system also had improved thermal response and temperature uniformity.

Thang-Long et al. [38] explored the suitability of VCs for use as the integrated heat spreader for a high power CPU. This is typically done with copper and can contribute significantly to total thermal resistance. Their numerical model suggested that VCs were only practical for this application when the spreader size was greater than 40mm x 40mm. At 300W, the VC integrated heat spreader had total thermal resistance of 0.178K/W compared to 0.201K/W for the copper integrated heat spreader. Horiuchi et al. [39] performed a similar study on the suitability of VCs for the integrated heat spreader. They found that the VC system could transfer 140W without the heat source temperature exceeding 100°C, compared to only 125W with the copper system. Robinson et al. [40] performed deeper analysis of the use of VC integrated heat spreaders over solid heat spreaders. They explored the geometric and thermal parameters which influenced the two technologies.

Naphon et al. [10] performed a parametric study on thermal solutions for electronic applications. The effect of CPU operation, coolant types, working fluids, charge ratios and other parameters on the performance of the thermal solution, including those with VCs, was explored.

The suitability of VCs for electronic based applications and some benefits associated with their usage is thus clear from this focused survey of the literature. For a more complete literature review, please refer to Chapter 2.

1.2 Research questions

Although the application of VCs for electronic devices has been well documented above, it is later explored in Chapter 2 that there are still many areas of VC design, modelling, performance and operation that require improvement. The development of VCs is an ongoing task within the research community. Although this work cannot explore every aspect within the field, it has attempted to identify needs within the literature and industry, and address some of these. Doing so has led to the development of the following research questions:
What aspects of vapour chamber technology are limiting their usage and future development?

This question has been extensively explored in the literature review of Chapter 2. Only a chosen aspect was addressed in further detail which formed the basis of this work. The wick has been identified as integral to VC performance; however in the literature not enough emphasis is placed on the thermal and fluid design of wick structures. Within the VC, the design of wick structures tends to follow conventional paths. Wick types inherently have different features which they excel at and exploiting these can help maximise both thermal and fluid performance of the VC. This research question has been further addressed in Chapter 3 and Chapter 4.

How can an effective thermal conductivity modelling scheme be improved to make it applicable to a wider range of scenarios?

Conduction based numerical modelling has shown great versatility due to its simplicity and ability to handle complex spreading scenarios. However, the contribution of the vapour region and associated processes is often overlooked. These aspects need to be explored in more detail to develop a better understanding of them in the context of effective conduction based schemes, as they are integral to VC operation. This research question has been addressed in Chapter 5.

How can the thermal performance of vapour chambers be improved through numerical methods?

The phase change processes of evaporation and condensation play an integral role in heat and mass transfer aspects of the VC. The wick structure governs and connects these processes and hence is vital to the overall operation and performance of the VC. While answering this question, the wick structure has been critically analysed such that VC performance can be further improved. This research question has also been addressed in Chapter 5.

1.3 Research methodology

This research focused on experimental and numerical methods to achieve its goals. At the initial stages, a survey of current literature was performed. This provided insights into the current issues with VCs and highlighted possible areas for research and development. Using these identified needs, a new design of VC was proposed with composite wick structures. A
dedicated experimental setup was developed and constructed in the RMIT Thermodynamics Laboratory in order to measure the performance of the VC. Detailed measurements of performance were carried out, and an equivalent sized copper heat spreader was tested as well for comparative purposes. Using these results, a numerical investigation was performed into VCs. More specifically, an investigation into the effective thermal conductivity of the vapour region was undertaken with a numerical model developed in Ansys. The model was validated with the experimental results of the copper heat spreader. This validated model was then used to explore the effect of evaporator wick properties on VC performance and to provide some guidance and suggestions for thermal designers of similar VCs.

1.4 Thesis outline

This thesis is structured into organised chapters. Chapter 1 – Introduction is a short overview of the motivations and methods for this research. Chapter 2 – State of the art includes the literature review into VCs used for thermal management of electrical systems. This chapter is broken into two sections which focus separately on the relevant experimental and numerical based literature. They focus on the design, operation, modelling and performance of VCs and the factors which influence these. Some concluding critical remarks are given for these sections. Chapter 3 – Methods focuses on the experimental and numerical methods and procedures which were used to achieve the aims of this research. The design of the experimental setup is discussed in detail, as is the design of VC developed in this work. The procedures used to capture data and measure the results are also explained. Further, the development of the numerical model for this work is also discussed in detail. In Chapter 4 – Experimental results and discussion, the experimental results of the heat spreaders are discussed. These discussions involve the spreading resistance and temperature distributions throughout each heat spreader. The performance of the VC is analysed in detail. Chapter 5 – Numerical results and discussion contains the results from numerical modelling. The model was first validated with the copper heat spreader and was then used for the VC heat spreader. The effective thermal conductivity of the vapour region was determined and its value was critically analysed. Further work in this chapter explored the delicate balance between the thermal, fluid and geometric properties of the evaporator wick which can influence the heat and mass transfer performance of the VC. Some guidelines and suggestions were made based off this. Chapter 6 – Conclusion and recommendations includes the concluding remarks for this work and also highlights some recommendations for future research and development in this field.
1.5 Summary of publications

Journal papers:


Conference papers:


Industry presentations:


Chapter 2 – State of the art

The VC is not a novel device as has been previously mentioned. It has been studied in the literature; the majority of these studies have been experimental work, with generally less focus on numerical work. A selection of these works has been included in the literature review that follows. A brief review of VC has also been provided in Ref. [41].

2.1 Terminology and definitions

In the literature on VCs and various thermal management systems, there are a few different methods through which thermal resistance can be measured. In general, thermal resistance can be measured through equation (2.1).

$$R = \frac{T_H - T_C}{\dot{Q}}$$  \hspace{1cm} (2.1)

Here $R$ is the thermal resistance, $T_H$ is the hot temperature, $T_C$ is the cold temperature and $\dot{Q}$ is the heat transferred. The temperatures $T_H$ and $T_C$ can vary based on the desired results. For example, the thermal resistance measured can capture the performance of the entire thermal management system, or it could seek to capture the performance of a specific component within the system such as the heat spreader or fins.

In this work, the term ‘total thermal resistance’ generally refers to the thermal resistance which develops between the heat source and heat sink of the system. This captures the performance of the entire thermal management system. The temperatures in this case generally refer to the temperature of the heat source ($T_H$), the component which the heat sink is removing heat from, and the temperature of the ambient air ($T_C$), where the heat is being rejected to. Hence the overall performance of the system is captured in this case.

In other scenarios, the term ‘spreading resistance’ is used. This term generally refers to just the heat spreader itself in which case $T_H$ is the temperature of the hot side of the heat spreader, where heat is being input into the spreader, and $T_C$ is the temperature of the cold side of the heat spreader, where heat is being removed from the spreader. This measurement is intended to quantify the spreading capability of the heat spreader, and it is commonly used in the results of this work (equation (3.6))

This can unfortunately cause some confusion as often the measurement of each of these temperature values varies between researchers. This can give slightly different meanings in
each case. Many researchers measure the hot side temperature of the heat spreader to be within the heat source. This is often separated from the heat spreader with an interface and the consideration of this interface if often neglected. It is also common to measure the cold side temperature of the heat spreader within the heat sink. This is often not integral with the heat spreader and as such another interface is present. Both of these factors can lead to errors in temperature measurement as thermal resistance develops between the desired surface and the measurement location. This has been shown in Figure 2.1. Note that this issue is common to solid metal heat spreaders (such as copper) and VC heat spreaders alike.

This is commonly done as it is simpler to measure temperature at these locations compared to the actual hot side and cold side surfaces of the heat spreader. The complexities of this are later discussed in section 3.1.3.2. When these simplifications are made, the spreading resistance measured may not be the true spreading resistance of the heat spreader since they consider some other system components in their measurement.

![Figure 2.1 – Some of the differences between spreading resistance measurements from literature. On the left are simpler measurements whereby the hot side temperature ($T_H$) is measured within the heat source and the cold side temperature ($T_C$) within the heat sink. On the right, more complicated measurements whereby the hot and cold side temperatures are measured on the heat input and removal surfaces of the heat spreader, respectively.](image)

In this work, spreading resistance has been clearly defined to only include the spreading performance of the heat spreader. Careful measurements were undertaken to ensure the correct values were measured as is discussed in section 3.1.3.2.

In the literature review sections of this work, the definitions as described above have been followed as much as possible, however for more clarity it is best to refer to the original work for clear definitions and measurements. This is particularly true for the spreading resistance which, as described above, can take on many definitions depending on the measurements performed by the author.
2.2 Experimental

This section of the literature review is separated based on the main wicking structure present in the VC. Much of the experimental work in the literature attempts to characterise the performance of the VC with measurements of spreading resistance, dry-out heat loads, heat transfer coefficients and heat source temperatures. These can provide insights into the heat transfer mechanisms within the VC and highlight its superiority as a heat spreader.

The spreading resistance ($R_{sp}$) of some VCs in the literature have been summarised in Table 2.1. In this, $t$ is the VC thickness, $A_{cond}$ is the condenser area, $A_{evap}$ is the evaporator area and the area ratio is $A_{cond}/A_{evap}$. The spreading resistance has been determined with equation (3.6), or the most similar measurement available. See section 2.1 for further discussion on this aspect.

Table 2.1 – Summary table of experimentally determined spreading resistance of VCs in the literature.

<table>
<thead>
<tr>
<th>$A_{cond}$ mm$^2$</th>
<th>$A_{evap}$ mm$^2$</th>
<th>Area Ratio</th>
<th>$t$ mm</th>
<th>Wick</th>
<th>Working Fluid</th>
<th>$Q$ W</th>
<th>$R_{sp}$ K/W</th>
<th>Ref.</th>
</tr>
</thead>
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<td>17220</td>
<td>1225</td>
<td>14</td>
<td>3</td>
<td>Evap (inner): sinter – copper</td>
<td>Water</td>
<td>40</td>
<td>0.07</td>
<td>This work</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Evap (outer): mesh – copper Cond: unwicked</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>17220</td>
<td>900</td>
<td>19</td>
<td>3</td>
<td>Same as above</td>
<td>Water</td>
<td>40</td>
<td>0.08</td>
<td>This work</td>
</tr>
<tr>
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<td>28</td>
<td>3</td>
<td>Same as above</td>
<td>Water</td>
<td>40</td>
<td>0.10</td>
<td>This work</td>
</tr>
<tr>
<td>17220</td>
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<td>43</td>
<td>3</td>
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<td>Water</td>
<td>40</td>
<td>0.13</td>
<td>This work</td>
</tr>
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<td>Acetone</td>
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<td>[42]</td>
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<td>-</td>
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<td>Water (30% FR)</td>
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<td>[44]</td>
</tr>
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<td>8</td>
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<td>Water (15% FR)</td>
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<td>[44]</td>
</tr>
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<td>Water (45% FR)</td>
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</tr>
<tr>
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<td>5</td>
<td>Sintered - copper</td>
<td>Water</td>
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<td>0.1</td>
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<tr>
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<td>FC</td>
<td>FC</td>
<td>R</td>
<td>Description</td>
<td>Fluid</td>
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<td>Note</td>
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<td>100</td>
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<td>Groove - silicon</td>
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<td>[53]</td>
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<td>-</td>
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<td>-</td>
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<td>-</td>
<td>Same as above</td>
<td>Methanol</td>
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<td>441</td>
<td>20</td>
<td>-</td>
<td>Same as above</td>
<td>Acetone</td>
<td>91</td>
<td>0.08</td>
<td>[56]</td>
</tr>
<tr>
<td>8900</td>
<td>441</td>
<td>20</td>
<td>-</td>
<td>Evap: mesh - copper</td>
<td>Water</td>
<td>350</td>
<td>0.04</td>
<td>[57]</td>
</tr>
<tr>
<td>5625</td>
<td>100</td>
<td>56</td>
<td>6</td>
<td>Evap: sinter groove - copper (leaf vein)</td>
<td>Ethanol</td>
<td>280</td>
<td>0.14</td>
<td>[58]</td>
</tr>
<tr>
<td>10000</td>
<td>1224</td>
<td>8</td>
<td>6</td>
<td>Evap: mesh - copper</td>
<td>Water</td>
<td>20</td>
<td>0.22</td>
<td>[59]</td>
</tr>
<tr>
<td>5806</td>
<td>94</td>
<td>62</td>
<td>-</td>
<td>Evap: sintered – copper (hydrophobic and hydrophilic)</td>
<td>Water</td>
<td>140</td>
<td>0.37</td>
<td>[60]</td>
</tr>
<tr>
<td>5806</td>
<td>94</td>
<td>62</td>
<td>-</td>
<td>Evap: mesh – copper (hydrophobic and hydrophilic)</td>
<td>Water</td>
<td>99</td>
<td>0.30</td>
<td>[61]</td>
</tr>
<tr>
<td>5806</td>
<td>94</td>
<td>62</td>
<td>-</td>
<td>Evap: unwicked (hydrophobic and hydrophilic)</td>
<td>Water</td>
<td>152</td>
<td>0.30</td>
<td>[61]</td>
</tr>
<tr>
<td>8100</td>
<td>100</td>
<td>81</td>
<td>2.5</td>
<td>Evap: sintered (copper) layer over mesh (copper)</td>
<td>Water</td>
<td>180</td>
<td>0.152</td>
<td>[62]</td>
</tr>
<tr>
<td>8100</td>
<td>100</td>
<td>81</td>
<td>2.5</td>
<td>Evap: sintered (copper) layer over mesh (copper)</td>
<td>Water</td>
<td>180</td>
<td>0.172</td>
<td>[62]</td>
</tr>
<tr>
<td>4900</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Evap: sintered – copper (super-hydrophilic)</td>
<td>Water</td>
<td>90*</td>
<td>0.31</td>
<td>[63]</td>
</tr>
<tr>
<td>4900</td>
<td>225</td>
<td>22</td>
<td>3</td>
<td>Evap: sintered - copper (hydrophilic)</td>
<td>Water</td>
<td>146</td>
<td>0.23</td>
<td>[64]</td>
</tr>
</tbody>
</table>

Note: FR denotes filling ratio. * denotes heat flux
2.2.1 Unwicked

Hsieh et al. [52] contributed to some of the earlier literature on VCs as heat spreaders. They tested the performance of a wickless copper-water VC under a range of heating loads with 3 different heating areas. The spreading resistance was reported to remain fairly constant at 0.20 K/W from heating loads 20-140 W. It must be noted that this VC was of overall thickness of approx. 100mm. Dry-out wasn’t observed within the tested range. The resistance was determined to be about 15% of an equivalent sized copper spreader, based on analytical results. They further studied the average heat transfer coefficients of the evaporator and condenser and found that both increased with heat input. There was also some reported temperature dependence, with higher operating temperatures leading to slightly larger heat transfer coefficients. Maximum heat transfer coefficients of approx. 10,000 W/m²K and 6,000 W/m²K were found for the condenser and evaporator, respectively.

Ming et al. [65] studied the effects of magnetic working fluid in a copper VC. The colloidal water working fluid had dispersed 10-100nm magnetic particles. An annular magnet placed on the evaporator (around the heat source) created a magnetic field which returned the condensed working fluid to the evaporator through a 1mm gap, replacing the need of a conventional wick. Temperature fluctuations were observed on the VC surfaces and it was suggested this was due to the evaporator producing discontinuous bubbles. They also determined the optimum charge ratio was 53.5% and performance was orientation dependent. More uniform condenser temperatures were recorded for this VC compared to a pure water wickless VC. The authors also commented on degradation of the working fluid through boiling processes.

Peng et al. [66] studied the effects of charge ratio and vacuum degree on a VC with internal finning. There was no typical wick structure, instead internal finning was present. The VC was made from aluminium and the working fluids were water and acetone. Start-up times were always reduced with larger input heat fluxes. Increasing the charge ratio from 15% to 50% resulted in heat source temperature reduction of nearly 4°C. Further reductions were not observed with higher charge ratios. Higher vacuum degrees also led to improved performance. The authors also reported that the acetone charged VC had slightly better performance than the water charged VC.

Liu et al. [67] performed an experimental study on evaporation and condensation heat transfer performance of a VC. The VC had no wick structure. The effect of filling ratio and
heat input on the phase change heat transfer coefficients was explored. It was found that both coefficients increased as the heat input increased. The influence of filling ratio was not as simple as the best evaporation and condensation performance was obtained with different filling ratios. For evaporation, the largest heat transfer coefficients were obtained with filling ratio of 33%. Values of approx. 23,000 W/m$^2$K were obtained at heat flux of 21 W/cm$^2$. For condensation, the largest heat transfer coefficients were obtained with filling ratio of 53%. Values of approx. 74,000 W/m$^2$K were obtained, also at 21 W/cm$^2$, the highest tested heat flux. The authors suggested that there were interactions between the evaporative and condensation heat transfer mechanisms within the VC, and these were strongly influenced by the filling ratio. The overall performance of the VC was best when the filling ratio was 33%.

### 2.2.2 Sintered

The sintered wick is the most common wicking structure used in VCs. When the term ‘sintered wick’ is used, it generally refers to a wick which was formed by the sintering of powder of a given material. The powder is usually compacted and formed into a porous medium at high temperatures (but below the melting point of the material). An example of a sintered wick has been shown in Figure 2.2.

![Figure 2.2](image)

*Figure 2.2 – A copper sintered powder wick (left) and view of this at X300 magnification (right). Adapted from [68].*

There are a large number of parameters which can influence the properties of the sintered wick. They include, but are not limited to; powder material, powder size, powder size distribution, powder shape, sintering pressure, sintering temperature, sintering environment and sintering time. This gives great flexibility in the design of such wicks. Although the properties of sintered wicks can vary greatly depending on these factors, it is generally found that they have:
- Moderate to large effective thermal conductivity
- Small to moderate porosity
- Small effective capillary radius
- Small permeability

Wang and Vafai [69] performed some of the earlier experimental and analytical investigations on a copper sintered powder wick VC. The wick had porosity, pore radius and permeability of 50%, $3.1 \times 10^{-5}$ m and $7 \times 10^{-12}$ m$^2$, respectively. An analytical conduction based model was introduced to predict the temperature distribution within the device. It agreed well with experimental measurements. The even temperature distribution on the condenser surface was highlighted, as was the large temperature drops through the evaporator wick of the VC (this is shown in Figure 2.3). Larger heat fluxes led to shorter start-up times of the device and the concept of a time constant was introduced to characterise this.

![Figure 2.3 – The temperature drop (θ) through a VC at different heat fluxes according to Wang and Vafai [69].](image)

Wei et al. [70] studied the performance of a sintered copper VC. Their study suggested that the main component of thermal resistance within the VC was the evaporator and advised this was because wall superheat was necessary to sustain boiling. The heat transfer coefficient was found to be linearly dependent on heat flux with a value of 40,000 W/m$^2$K observed at 43 W/cm$^2$. This was also dependant on orientation. They further investigated the use of VCs against copper and aluminium as heat spreaders. The VC was advantageous for scenarios with high spreading requirements and in cases where small thickness spreaders were required.

Hanlon and Ma [71] proposed a model for evaporative heat transfer in sintered porous media, similar to the evaporation process within a VC. Their combined thermal-fluid model
considered heat conduction, capillary limitations and nucleate boiling. It aimed to explore how wick parameters (particle size, porosity and thickness) influenced the evaporative heat transfer performance of the wick. An experimental system was developed to validate the numerical model and good agreement was observed between the two. The authors stated that performance would degrade at the onset of bubble nucleation or if liquid supply was insufficient. Onset of bubbles was strongly tied to the wick thickness and heat flux. Thinner wicks in particular were more affected by the onset of nucleate boiling within the wick. Once nucleation had occurred a vapour blanket would form at the bottom of the wick and prevent thin film evaporation from occurring on the top surface of the wick. Thin film evaporation was a very effective heat transfer mechanism, so preventing it would greatly reduce thermal performance. The boiling limit was found always to be reached before the capillary limit in this work. Improvements to the evaporation process could be achieved by decreasing the particle size. The dry-out heat flux was a function of wick thickness and porosity.

Xuan et al. [72] investigated the transient behaviour of a VC both experimentally and numerically. Experiments on an unwicked and sintered powder copper-water VC were carried out. The unwicked VC was found to have irregular temperature distributions at higher heat fluxes however moving to a sintered wick eliminated these irregularities. The authors suggested this was due to the sintered wick providing nucleation sites for bubble growth and capillary pressure to more evenly distribute the working fluid over the evaporator. A simplified numerical model was then introduced. The effect of liquid in the wick was neglected, as were vapour flow aspects within the vapour space. The working fluid vapour was further assumed to be an ideal gas. Transient response of the model did not agree well with experiments however the steady state solution was in good agreement.

Boukhanouf et al. [73] investigated a sintered copper-water VC under different heat fluxes with a non-centrally located heat source. The wick had porosity, pore radius and permeability of approx. 50%, 40μm and 1.5x10^{-11} m^2, respectively. They utilised IR thermal imaging to capture temperature plots of the evaporator surface in top heating mode. These showed that the VC has nearly isothermal evaporator surface temperature whereas very large temperature gradients were observed for an equivalent copper spreader as seen in Figure 2.4. The VC evaporator surface temperature was diminished by sudden drops in some corners of the device which the authors suggested was trapped non-condensable gases. The maximum heat flux of the VC was 40 W/cm^2 compared to 34 W/cm^2 for the copper spreader in order to keep
the heat source temperature below its limit. The measured resistance of the copper spreader was over 40 times larger than for the VC heat spreader.

Figure 2.4 – IR images of VC evaporator surface (left) and copper plate (right) at 28W/cm². Adapted from [73].

Chang et al. [74] studied the effects of non-uniform heating on a VC with sintered copper powder wick. The wick has porosity and average particle radius of approx. 53% and 32μm, respectively. They suggested that non-uniform heating conditions were more common in practice (i.e. silicon technology) than uniform heating conditions which are often seen in the literature. Their main finding was that thermal performance was relatively insensitive to heat flux distribution even with localised heat fluxes as high as 746 W/cm². The variations in evaporative resistance were less than 10% due to the non-uniform heat sources (Figure 2.5). The authors further reported that reduction in evaporative resistance with increased heat input was a consequence of the receding liquid level in the wick. This receding meniscus was affected by the wick pore size and wick particle arrangement. Different particle arrangements could lead to differing liquid recession characteristics according to the authors.

Figure 2.5- The effect of non-uniform heat sources on the evaporative resistance (θ_evap) expressed as a ratio to the uniform heat source evaporative resistance [74].
Ying-Tung et al. [75] studied the addition of diamond particles to a sintered copper powder wick. In general, the diamond-copper wick VC had lower total thermal resistance than a purely copper wick VC. The effect of diamond addition was further investigated with larger volumes of diamond leading to further reduced thermal resistance of the device. This was attributed to the very high thermal conductivity of diamond (2,400 W/mK) compared to copper, and the improvement in overall wick thermal conductivity by addition of a greater amount of diamond. The authors identified a limit beyond which further diamond addition had no effect on performance. They also found that the diamond-copper wick VC had larger dry out limit compared to the copper wick VC.

Chen et al. [76] evaluated VCs with radial grooved wicks against sintered powder wicks. Both were made from aluminium with acetone as the working fluid. The performance of both VCs was strongly influenced by charge ratio, and the optimum charge in both cases was different. The total thermal resistance of the sintered wick VC was constant at 0.69 K/W in the tested range of heat inputs, unlike the radial grooved VC whose performance was dependent on heat input. The total thermal resistance of this VC reduced with higher heat inputs, and a minimum of 0.72 K/W was measured at the highest heat input.

Sungtaek Ju et al. [77] studied a VC with advanced sintered wicks. The purpose of these advanced wicks was to separate the two functions of a wick: providing liquid to the evaporator and providing an evaporation surface for phase change. They achieved this by having an evaporation surface (thin sintered copper layer) and a separate liquid circulation structure (sintered vertical columns, sintered lateral arteries, sintered bi-porous structures). Refer to the original work for a more complete description of these wick structures. Experiments showed that up to 150 W/cm², all three types had similar performance. The authors suggested that at these low fluxes, evaporation was predominantly from the evaporation surface, not the liquid supply layer. At higher heat fluxes, the role of the liquid supply layer was highlighted and the sintered lateral artery VC performed worse than the other two VCs. The authors suggested that higher heat fluxes could have led to reductions in liquid layer thickness within the wick, as well as random bubble generation and liquid ejection from the wick. These factors also altered the performance of the VCs.

Tsai et al. [46] studied the thermal resistance in a copper-water VC with sintered copper wick and columns. They suggested that the spreading resistance within a VC was composed of two parts; lateral resistance and axial resistance, and that the lateral resistance was generally much
larger than the axial resistance. They additionally reported poor performance at low heat loads due to start-up difficulties. The effect of orientation was explored, with the worst performance of total thermal resistance of 0.893 K/W observed at 90°. Interestingly, the 180° orientation (top heating mode) had the best thermal performance with total thermal resistance of 0.850 K/W compared to 0.868 K/W at 0° orientation (bottom heating mode) however no reasons were given for this. It was further found that the axial resistance was independent of orientation, however spreading resistance was orientation dependent.

Tang et al. [47] constructed a sintered copper VC with sinter columns between the evaporator and condenser wicks. Maximum condenser temperature variations of 7°C were reported for the VC compared to 21°C for an equivalent copper spreader at the same heat load. They studied the effects of heat source size on thermal performance and found that with a heat source size of 4cm², the critical heat load was 180W. Partial dry-out occurred beyond this load however good performance was still reported up to 300W. Reducing the heat source size to 1cm² increased the critical heat load, however due to experimental limitations this load could not be found. This has been summarised in Figure 2.6. Non-uniform heat sources were also explored and copper spreaders were found to perform better for those scenarios.

![Figure 2.6](image.png)

Figure 2.6 – The effect of heat input and heat source size on the spreading resistance of the VC [47].

Boreyko and Chen [78] studied a sintered copper-water VC like device with super-hydrophobic condenser and super-hydrophilic evaporator. The condenser design caused
liquid drops to jump to the evaporator, eliminating the need for conventional wicking. Solely relying on this mechanism led to two new limits; the ability to jump across the vapour space against gravity (in top heating mode) and the limit imposed by high velocity vapour flow moving against the jumping drop. A simplified analytical model was introduced in the form of a series thermal resistance network that considered wick and phase change resistances. This model neglected spreading aspects since there was minimal thermal spreading in the experimental setup. There was good agreement between model and experimental results. Large wick thickness led to poor thermal performance due to low wick thermal conductivity. Phase change resistance was very small at higher vapour temperatures compared to wick resistance. At low temperatures however, the phase change resistance was significant due to the governing kinetic theory. The authors stated that approx. 2mm was the maximum gap which a liquid drop could jump in top heating mode. They also suggested that the VC performed well as a thermal diode due to the super-hydrophobic condenser, as 250 times more heat was transferred by heating the evaporator compared to heating the condenser at the same driving temperature difference. It was stated that the treated surfaces do deteriorate over time however it was not closely examined.

Sun et al. [64] fabricated a VC consisting of sintered copper evaporator wick and super-hydrophobic condenser wick. The condenser wick had contact angle of 150° with water as shown in Figure 2.7. They reported on the variations in thermal performance with respect to changes in initial charging pressure. The reduction in charging pressure led to better performance as this generally meant there were less non-condensable gases present and the lower pressure also led to earlier nucleate boiling according to the authors. At low charging pressure, the hydrophobic VC had lower thermal resistance compared to a screen mesh VC and copper spreader. The authors suggested that these improvements were due to the super-hydrophobic condenser which caused the vapour to condense into drops rather than a film. The associated heat transfer coefficient for drop-wise condensation was much larger than for film condensation and the drops can easily return to the evaporator via this mechanism.

Sun & Qiu expanded on this work in [63] by adding a nanostructure layer atop the evaporator wick. The addition of this layer reduced the resistance of the VC (compared to a VC without this treatment) for all heat inputs. It was suggested this was due to increased capillarity and nucleation sites caused by this layer. The authors trialled the addition of ethanol to the working fluid and found that this addition reduced thermal performance. It also reduced the critical heat flux value and led to early dry-out. Interestingly, the VC was found to transfer
heat better laterally using high ethanol ratios at low heat input. The authors also explored the use of a piezoelectric actuator on the evaporator side of the VC. It was found to slightly improve performance as it assisted in bubble departure and condensate return.

![Image](image_url)

Figure 2.7 – A plan view of the super hydrophobic condenser wick is shown in (a). The contact angle of 150° with water is shown in (b) and (c) shows a magnified view of the super-hydrophobic wick surface [64].

Shaeri et al. [60] studied a VC with both hydrophobic and hydrophilic features in the evaporator. A hydrophilic sintered wick was placed over a hydrophobic substrate in the evaporator. The wick was forced against the substrate via posts attached to the condenser section. This combination of structures was selected to decrease thermal resistance however experimental results suggested that the combination actually increased spreading resistance by approx. 50% compared to a VC with only hydrophilic evaporator. Furthermore, the critical heat flux was also reduced from 170 W/cm² to 54 W/cm². The authors suggested this may have been due to poor contact of the sintered wick over the substrate.

Kim and Kaviany [79] studied the receding liquid layer in a sintered wick VC by using minimum surface energy principles. Doing so, they were able to analyse the liquid recession with varying heat flux when considering sintered layers with different particle diameters and packing types. The properties of the wick were dependent on average liquid thickness and packing type, and thus could vary with heat flux during operation. The authors attempted to find the optimum wick, which maximised heat flux and minimised thermal resistance. A wick with particle diameter of 30-50μm and close packing arrangement gave the most optimal performance.
2.2.3 Grooved

This type of wick has groove like structures of varying cross section. The manufacturing techniques of these vary (conventional machining, etching, extrusion etc.) depending on the material and required geometry. An example of a groove wick has been shown in Figure 2.8.

![Grooved wick](image)

*Figure 2.8 – An example of a grooved wick for a VC. Adapted from [54].*

It is typically the groove geometry, pattern and material which govern the properties of these wicks. They generally have the following properties:

- Large effective thermal conductivity
- Large porosity
- Large effective capillary radius
- Large permeability

In 2002, Kang et al. [80] fabricated a VC from silicon wafers with radial grooves etched into the evaporator and condenser plates. The grooves were of trapezoidal cross section. An intermediate silicon wafer separated liquid and vapour regions while allowing circulation of working fluid. This separation wafer was expected to increase the entrainment limit of the VC. The wettability was improved by bonding a gold layer to the silicon. The performance of the water charged VC was compared to a plain silicon wafer, and at 27W, the evaporator temperature was 67°C for the VC compared to 92°C for the silicon. Further, the temperature difference between the evaporator and condenser of the VC was 33% lower than for the silicon. An optimal charge ratio of 70% was found based on these experimental investigations.
Murthy et al. [81] fabricated a VC for space constrained applications. The evaporator had a grooved structure to enhance phase change. It was made from copper and the working fluid was FC-72. There was no other wick present in the device, thus the design was such that the evaporator would always remain wetted with fluid, regardless of the orientation. This was achieved with some internal fluid storage systems. Rectangular grooves with width and depth of 0.31mm and 0.55mm, respectively, were cut into the evaporator to help initiate boiling at lower wall superheat. The VC outperformed an equivalent aluminium spreader for all heat loads. The junction temperature was able to be kept 47.1°C lower when using the VC at heat flux of 6.3W/cm². At low heat fluxes, the VC was found to be orientation independent, however higher heat fluxes would lead to boiling regimes and eventual partial dry-out. A high speed camera (Figure 2.9) was used to capture the phase change process from the wick and to see the effect of heat flux on this process. At higher fluxes, the horizontal VC slightly outperformed the vertical VC.

![High speed images of the evaporation surface at 90°C (vertical) orientation under progressively higher heat fluxes, (a) to (f). Boiling is observed at higher fluxes. At the highest flux value (5.8W/cm²) dry-out of the evaporation surface can be observed [81].](image)

Gillot et al. [82] studied the performance of a silicon VC with micro-capillary grooves. These grooves were formed via etching of a silicon wafer to width and depth of 90μm and 115μm, respectively. The VC was charged with differing amounts of water in order to investigate the optimal charge amount. Performance of the device was strongly dependent on charging
amount. At the optimal charge, up to 30W could be transferred without partial dry-out being reached. The best total thermal resistance of this device was approx. 0.80 K/W.

Go [42] studied the performance of an acetone charged VC with metal etched micro-wick structures. The etched wick was made of stainless steel and the container of aluminium. Go suggested that a charge ratio of 30% resulted in best operation. Reductions in resistance were observed up to 140W heat input. The author also studied the effect of orientation on performance and found that 90° orientation had the worst performance. It was suggested this was due to the wick drying out easily in this orientation. This VC could handle heat fluxes of 80 W/cm² without dry-out being observed.

Peng et al. [83] analytically investigated a VC which had a novel groove network based on the fractal architecture of leaf veins. The network consisted of ‘Y’ sections which fed into each other. Comparisons were made to a parallel groove network. Transport properties such as permeability and capillary pressure of the fractal network were much greater than the parallel network. The performance of the network was strongly influenced by geometric parameters of the fractals. The authors suggested that these networks were suitable for VC wicks not only due to their transport properties, but also as they enhanced condensation areas. Further work into the thermal-fluid performance of this leaf vein architecture VC was performed by Peng et al. [54] and Liu et al. [55]. They investigated the use of water and ethanol as working fluids and found that ethanol performed better using optimally determined charge amounts. They also found that VC performance was dependent on the manufacturing method used for the leaf vein network and highlighted some difficulties of the wick design.

Wei et al. [53] investigated the optimised design of a silicon based VC with micro scale grooves and overall thickness of 1.25mm. An optimisation model was introduced to balance the thermal and fluid performance of the VC by altering the groove geometry in the evaporator region. The optimised design was experimentally tested against non-optimal designs and found to have superior thermal performance, in particular at higher heat fluxes. Results from the model were in agreement to within 11% of the experiments. The spreading resistance of the optimised design was 0.53 K/W at heat flux of approx. 100 W/cm².

Zeng et al. [51] fabricated an aluminium-acetone VC with a micro-grooved wick structure. The grooves were roughly triangular in shape and were slightly different for the condenser and evaporator. The VC was found to have good start-up performance, even at low heat loads. The authors suggested that this was due to the design of the groove structures easily allowing
bubbles to flow from them. The spreading resistance of the VC was fairly constant at 0.06-0.065K/W through the testing range of 30-150W. The resistance also slightly decreased by inclining the VC by 30° but reduced in the vertical (90°) orientation; these changes were linked to the length of the flow path of the working fluid. The authors identified that the groove structure could be further developed to improve thermal performance and that optimisation between the fluid aspects would also need to be considered.

2.2.4 Mesh

The mesh wick tends to be quite common for heat pipes. When the term ‘mesh wick’ is used, it generally refers to a wick which is made from woven wires of a particular material (generally metals). This is sometimes also called ‘screen mesh’ or ‘wire mesh’ wick. An example of a mesh wick has been shown in Figure 2.10.

![Figure 2.10 - A layer of copper mesh with 100 openings per inch. Taken from [84].](image)

The properties of the mesh wick tend to be governed by the wire diameter and material, size of the opening and pitch. This gives a range of possible values for these properties however mesh wicks usually have the following:

- Small effective thermal conductivity
- Moderate porosity
- Large effective capillary radius
- Moderate to large permeability

In 2010, Wang et al. [85] observed through experiments combined with numerical analysis that a VC with copper mesh wick had better spreading performance than aluminium and copper for use in LED applications. The effective thermal conductivity of the VC was reported to be more than double that of copper. Transient tests were carried out and the VC responded to heat inputs faster than other spreaders. The influence of the external heat sink on the total thermal resistance of the VC system was also analysed. Further tests by Wang [86]
found that a VC system had better thermal performance above LED input power of 5W and that below this heat input, the copper spreader performed better.

Huang et al. [87] studied the effect of vapour region thickness on a device of similar design to a VC, however the device transferred heat in a manner similar to a heat pipe i.e. the objective was not to spread heat. It had multiple copper mesh wicks (#100 and #200) and was charged with water to a predetermined amount. The vapour region thickness was varied from 0.4mm to 1.2mm with all other thicknesses held constant. The total thermal resistance was found to vary strongly on the vapour region thickness. The smallest observed total thermal resistance at the minimum vapour region thickness (0.4mm) was 0.41 K/W compared to 0.19 K/W at the maximum vapour region thickness (1.2mm). The authors suggested these differences were due to the large vapour pressure losses associated with narrow flow paths of thin vapour regions. Increasing the vapour region thickness above 1.0mm generally had little effect on the total thermal resistance. The authors generalised these results based on the ratio of hydraulic diameter of the vapour region to that of the device interior (vapour and wick).

Bose et al. [84] investigated the thermal performance of a copper-water VC with wire screen mesh for electronic applications. The wire screen mesh had porosity and wire diameter of 63% and 0.07mm, respectively. The VC system was compared against a copper system and it was experimentally found that the VC kept the maximum temperature down by 26%, while increasing the heat transfer coefficient by 36%. Total thermal resistance of 0.82 K/W was measured for this VC system. The heat transfer limits were also explored by the authors in this work and the entrainment limit was found to be most critical although there was still significant margin of safety until this limit was reached.

Isaacs et al. [26] developed and tested a VC based heat strap for use in CubeSat applications. The VC had two layers of fine copper mesh (porosity and wire diameter of 67% and 50.8μm, respectively) which were separated via a coarse layer of copper mesh (porosity and wire diameter of 73% and 381μm, respectively) between them. Acetone was used as the working fluid. They observed that the heat strap had effective thermal conductivity of over 2,000 W/mK. A simplified theoretical model was developed to predict the maximum rate of heat transfer through the VC. It was then used for optimisation analysis through a genetic algorithm. The maximum adiabatic length, wire diameter and wire spacing of the mesh wick were explored through this optimisation.
2.2.5 Foam

Foams wicks are porous mediums that are similar to the sintered wick but with a different structure as can be seen in magnified views of the material. They can be manufactured from a variety of methods (for example bubbling, foaming and deposition methods) depending on the requirements of the foam. An example of a foam wick has been shown in Figure 2.11.

Figure 2.11 – The copper foam wick on the inside of a VC (left) and a magnified view of copper foam (right) which more clearly shows the foam structure. Adapted from [49] (left) and [44] (right).

The properties of the foam wick are determined by the material and manufacturing procedure of the foam. This can have a large influence on the final properties of the foam and can give significant variation in the values obtained. In the case of phase change devices it is the open cell foam structures which are of use. These types of foams wicks generally have:

- Large effective thermal conductivity
- Large porosity
- Moderate effective capillary radius
- Large permeability

In 2006, Lu et al. [45] investigated a graphite foam VC. The container was made from aluminium and the graphite foam was joined to the inner surfaces through soldering. The graphite foam had effective thermal conductivity of 140 W/mK in one direction and 70 W/mK in the other two directions. It had porosity, permeability and average pore diameter of 71%, 6x10^{-11} m² and 50μm. Ethanol was used as the working fluid since it naturally wets graphite. The influence of working fluid charge amount on the thermal performance was explored and an optimal charge was determined through experimentation. For this charge
amount, the spreading resistance of the VC was found to be approx. 0.4 K/W. The authors then studied the capillary and boiling limits of the VC and suggested that their VC was limited by boiling in the wick. The properties of ethanol in particular were linked to poor performance, and it was suggested that better performance would be observed with water as the working fluid. In this case, chemical treatment of the wick would be required to increase the wettability between water and graphite. The properties of the wick were highlighted as being particularly useful for VC operation.

Queheillalt et al. [88] investigated the use of a ‘multifunctional heat pipe sandwich panel structure’. This device combined the structural and functional applications of foams for aerospace application and will be referred to as a VC hereafter. It consisted of truncated, square honeycomb structures to provide structure support and working fluid paths. Refer to Figure 2.12 for clarity.

![Figure 2.12 – The ‘multifunctional heat pipe sandwich panel structure’ studied by Queheillalt et al. [88] provided structural and functional support by using open honeycomb structures lined with nickel foam.](image)

The cruciform posts were lined with compressed nickel foam, as were the evaporator and condenser plates. Refer to the original work for full discussions on the compressed foam properties. Aluminium plates were used for the container with water as the working fluid. The aluminium was internally coated with nickel to prevent non-condensable gas generation with water. Experiments were undertaken with a non-uniform heat source provided by a propane flame. Numerical results suggested that approximate heat inputs at the VC centre
reached 42kW compared to a minimum of 8.5kW at the edges [89]. The experiments were carried out at 90° orientation. The temperature response was measured over a time period of 40s and the authors reported more uniform VC condenser temperatures as compared to an equivalent aluminium spreader. The reported temperature difference between the centre and edges of the aluminium plate was over 100°C compared to approx. 20°C for the VC as can be seen in Figure 2.13. Further, the temperature of the VC condenser didn’t exceed 52°C in the reported time range, compared to 204°C for aluminium. This was linked to the large thermal mass of the VC due to phase change of the working fluid.

![Figure 2.13 – IR imaging after 40 seconds exposure to heat source (propane flame) for the aluminium heat spreader (left) and VC (right). Adapted from [88].](image)

Ji et al. [44] studied a VC with copper foam wick. This foam had porosity and effective pore diameter of 95% and 0.09 mm, respectively. The foam lined the inside surface of the VC and foam bars were sintered to provide direct contact between the upper and lower wicks. Copper columns were added to prevent deformation. The authors reported that water performed best as the working fluid, followed by acetone and ethanol. For the water VC, the maximum tested heat load was 170W (heat flux of 216 W/cm²) without dry-out occurring, however nucleate boiling was observed. The authors suggested that high flux could be sustained due to very high foam porosity and multiscale pore sizes in the foam. The ethanol VC had performance worse than an equivalent copper spreader below heat loads of 70W. See Figure 2.14 for detailed results from this work. The water VC also performed best at different orientations, and generally the effect of orientation was less pronounced at higher heat loads.

Li et al. [49] performed a comparative study of VCs with various sintered copper powder wicks against copper foam wicks. In general, the porosity of the sintered powder wick was between 55-59% and the porosity of the foam wick was nearly 70%. See the original work
for full details of wick parameters. The temperature uniformity on the condenser surface was superior for the foam VC compared to the powder VC. The authors suggested this was due to the high porosity and high permeability of the foam structure. Interestingly, at low heat loads the foam VC had lower spreading resistance than the powder VC however at higher heat loads the opposite was true. Across multiple heat source sizes, it was found that the powder VC generally outperformed the foam VC. The authors suggested this was due to the detailed design of the powder VC which incorporated sintered columns between the condenser and evaporator wicks. This feature was not present for the foam VC.

![Image]

Figure 2.14 – The spreading resistance against heat input for the copper foam VC of Ji et al. using water, acetone and ethanol. Also included is an empty VC and copper plate. Here the charge ratio (b) is equal to 0.30 and the orientation (θ) is horizontal [44].

Liu et al. [50] developed a VC which used copper foam lined support columns to provide flow paths and mechanical strength. Additional micro-channels were machined into these foam structures. Copper foam with porosity of 95% was also used on the evaporator surface. The porosity of the foam for column lining was systematically varied. The largest porosity of 95% was found to have minimum spreading resistance of approx. 0.05K/W and was orientation independent. Foam porosity of 50% was found to increase spreading resistance to approx. 0.11K/W and was also orientation independent. When there was no foam lining the support columns, the spreading resistance increased to over 0.11K/W and performance became orientation dependent. The foam wicks were found to not strongly affect temperature uniformity; this was only affected by the filling ratio and device orientation.
2.2.6 Composite and other

In the literature review sections that preceded this, the wicking structure has generally been consistent throughout the VC; the same wick was present for the evaporator and condenser wicks. This is rather typical of VCs within the literature. More recently, designs have been introduced whereby different wick structures were present for the evaporator and condenser wicks. Thus there are combinations of wick types within a VC. There is not a generic name for these types of VCs and thus they will be referred to as ‘composite’ in this work. They have also been referred to by some as ‘hybrid’ within the literature. This section includes VCs of this composite or hybrid design, as well as some other unique wick structure VCs.

Lee et al. [90] experimentally analysed the performance of three VC solutions. The first utilised a grooved wick, the second utilised boiling enhanced wicks and the third had boiling enhanced multi-wick structures. The motivation behind the advanced wick was to balance the capillary needs and heat flow requirements for VCs. All were made from copper with water as the working fluid. The grooved VC had the largest spreading resistance of 0.19K/W, followed by the boiling enhanced VC (0.05K/W) and the best performance belonged to the boiling enhanced multi-wick VC (0.03K/W). These advanced wicking structure VCs also had dry-out limit beyond 330W, compared to 250W for the grooved VC. The performance of the advanced wick structure VCs was also relatively orientation independent.

Wong et al. [57] studied the performance of a VC with evaporator wick of 100 mesh screen placed over one or two layers of 200 mesh screen, sintered to the surface. The condenser consisted of parallel triangular grooves, the peaks of which contacted the mesh directly. Different groove spacing were trialled, with minimum spreading resistance observed when the spacing was 1.6mm. Larger groove spacing resulted in slightly worse performance at low heat loads however better performance at higher heat loads as it delayed the onset of dry-out. Larger charge ratios led to worse performance at lower loads, but also delayed dry-out and thus had better performance at higher heat loads. The authors suggested that poor performance at low heat loads was due to a thick liquid layer in the wick. They were also able to measure the heat transfer coefficients by approximating the vapour temperature. The evaporator and condenser heat transfer coefficients were found to vary from 38,000-65,300 W/m²K and 10,400-30,000 W/m²K, respectively. Still, the evaporative resistance was many times larger than condensation resistance due to the small heat input area. This has been shown in Figure 2.15. Both resistances were found to decrease at higher heat loads, with the reduction in evaporative resistance being more pronounced than condensation resistance.
Figure 2.15 – The evaporative ($R_e$), condensation ($R_c$) and spreading resistance ($R_{vc}$) for the VC tested by Wong et al. at different groove spaces ($W$) and charge amounts ($m$). Note the magnitude of condensation resistance is much smaller than evaporative resistance. Also note how evaporative resistance reduces with heat input [57].

Wong et al. [56] extended this study by investigating the use of water, methanol and acetone working fluids. The VC under consideration here had a double screen mesh layer on the evaporator and triangular grooves machined on the condenser. The water charged VC performed best, followed by methanol and acetone as shown in Figure 2.16. The working range of the water charged VC was also larger than methanol and acetone, which only had a narrow working range. The authors also studied the effect of heat source size on thermal performance, with small heat source sizes leading to larger spreading resistances.

Figure 2.16 – Variations in spreading resistance ($R_{vc}$) for the VC of Wong et al with heat source area ($A_e$). Water was the best working fluid here [56].
Weibel et al. [91] developed a VC which utilised carbon nanotube wicks for enhanced evaporative performance. The carbon nanotubes were grown through microwave plasma chemical vapour deposition methods and the authors discussed in detail the parameters which controlled this. The carbon nanotubes were naturally hydrophobic and as such a nanometre scale copper layer was deposited on them to improve the wettability properties. Experiments were run to characterise the evaporative performance of this wick, in a setup that simulated the conditions inside of a VC. The most successful wick design had the largest density of carbon nanotubes on the surface. The authors suggested this was because of the increased capillary pressure generated by the small effective pore size of the high density carbon nanotube surface. Heat fluxes of over 500 W/cm$^2$ were sustained without dry-out being observed. Similar experiments were run by Cai and Chen [92] to verify the evaporative performance of a carbon nanotube covered surface for phase change devices.

Deng et al. [58] developed a VC with a specially designed evaporator wick. This wick was made from copper particles that were sintered together to form uniform radial grooves which extended from the centre of the wick, thus giving an overall grooved pattern to the sintered wick. In total there were 16 grooves. The condenser had a uniform sintered wick on it, which directly contacted the peaks of the evaporator wick. This was done to aid the mechanical strength of the VC and to provide flow paths for condensate return. The authors extensively looked into the effect of copper particle shape and size on the thermal performance of the VC. Each variation had slightly different thermal performance, with the best performance obtained via a wick (porosity of 35%) with spherical powder of size 50-75μm. This VC had spreading resistance of less than 0.15 K/W for all heat inputs. Using a spherical powder of size <50μm (porosity of 34%) resulted in spreading resistance of greater than 0.22 K/W for all heat inputs so the performance was very sensitive to the powder selection.

Wiriyasart and Naphon [93] investigated VCs with both sintered and groove structures in the evaporator. In one VC, there were only groove structures in the heat source region, and the other VC also had sintered powder present between the grooves. All were made from copper with water as the working fluid. The VC with sintered structures present had lower total thermal resistance of approx. 0.20 K/W compared to 0.25 K/W for the VC without sintered structures present.

Yao et al. [94] developed a VC which had a tree-like groove network on the evaporator surface and screen mesh wicks that filled the vapour region. These mesh wicks helped the VC
perform in top heating mode, with only slight increases in the thermal resistances observed. Removing the mesh wick led to poor thermal performance and large temperature fluctuations on the evaporator and condenser surfaces. Increased thermal resistance by approx. 10% was also observed when the groove network was replaced with a smooth surface. The effect of filling rate, cavity height and mesh was also explored. A finer mesh (#200) surrounded by a coarser mesh (#80) was found to give the best thermal performance in top heating mode.

Shaeri et al. [61] furthered on their previous work [60] on wettability altered VCs by introducing two new VCs. One had hydrophilic screen mesh forced into contact with a hydrophobic evaporator surface and the other had no typical wick structures, instead the evaporator surface was chemically treated such the region above the heat source was hydrophobic and the surrounding region was hydrophilic. Performance of these was compared against a typical sintered wick VC which always outperformed the other two VCs; it had lower spreading resistance and higher critical heat flux. The authors suggested these VCs could be further improved by more research and development. However the manufacturing of wettability altered VCs was advantageous since they did not require high temperature sintering procedures to be performed like the sintered wick VC. Furthermore, the wettability of these surfaces was also maintained through several months of testing.

Li et al. [62] furthered this work by comparing the performance of two VCs under different cooling conditions. Both VCs had the same evaporator wick structure of sintered copper powder over a copper mesh (#200) layer. The powder layer was made using irregular shaped powder with average diameter of 60μm. It had lateral arteries to promote evaporation and provide flow channels. The final evaporator wick had porosity of approx. 57%. One VC had a condenser wick which consisted of compressed copper foam (porosity of approx. 60%) and the other had three layers of copper mesh (#200) with porosity of 60%. Higher cooling water temperatures led to improved temperature uniformity on both VCs. The foam VC had larger critical heat flux and more uniform temperatures on the condenser surface than the mesh VC. The effect of filling ratio on thermal performance was also extensively studied. The spreading resistance of the foam VC was generally slightly lower than that of the mesh VC. This improvement in most areas for the foam VC was linked to the small effective capillary radius and high permeability of foam compared to mesh.
2.3 Numerical

There have been fewer numerical studies of VCs performed in the literature compared to experimental studies. The numerical studies performed range from simple thermal models which attempt to predict the overall performance of the VC, to complex computational fluid dynamic (CFD) studies which perform detailed analysis of VC operation. The computational effort, resources and time vary according to the type of analysis undertaken which can influence a researcher or thermal designers choice of analysis. This review will be structured based on the type of numerical analysis performed.

2.3.1 Thermal

Wang and Vafai [95] developed a simplified analytical model of a phase change device similar to a VC. This model assumed that only conduction mechanisms were present in the wall and wick of the VC and neglected temperature drops within the vapour region. The purpose of the model was to predict the transient start-up and shutdown operation of the VC. These responses were strongly influenced by the thermal diffusivity and thickness of the wall and wick layers of the VC. It was also dependent on the external heat transfer coefficient. The start-up and shutdown periods were found to be roughly equal. They also found that the wick caused the largest thermal resistance of the VC. Experimental measurements [69] of the transient performance were in good agreement with the proposed model.

Sauciuc et al. [96] provided some preliminary analysis on the suitability of VCs as heat spreaders in thermal management systems. Two models were introduced to do this; one assumed that for a wickless VC, the resistance was made up solely of the evaporative resistance due to pool boiling in the evaporator. Reductions in heat source areas always led to increased evaporative resistances and evaporative heat transfer coefficients as high as 30,000W/m²K could be expected. This model agreed with test data to within 15%. The second model was suggested for VCs with wicks present and relied upon the thermal resistances associated with conduction through the layers of the VC. This conduction based model was itself based on the work of Prasher [97] where temperature drops due to the wick and vapour region were calculated based on some simplifying assumptions. Experimental data was again used to verify the model. They suggested that the use of a VC over copper was advantageous for heat spreading scenarios. The improvement of the VC in heat spreading scenarios was strongly dependent on the thickness of the heat spreaders used, with VCs being particularly useful for when thin heat spreaders were required (Figure 2.17).
Figure 2.17 – The improvement of the VC spreader over copper was explored by Sauciuc et al. The improvement was more pronounced at larger heat sink footprints and smaller base thicknesses [96].

Prasher [97] provided further insights into the use of VC thermal solutions. The author proposed a layered model of a VC which was based on conduction mechanisms. Each layer in the model had different thermal properties and special treatment of the vapour layer was required in order to consider vapour flow dynamics. The author suggested some simplifying assumptions could be made to this layer which would reduce the vapour flow to a medium with effective thermal conductivity of over 260,000 W/mK. This value was strongly dependent on the assumptions and analysis used. Experimental results also suggested reasonable agreement and that the layer model approach was appropriate for VC analysis. The author also provided some insights into VC optimisation. The wick thickness was found to strongly influence the thermal and fluid performance of the VC. Increased wick thickness would lead to reduced thermal performance but increased fluid performance and vice versa, thus there existed an optimum wick thickness for a given configuration as can be seen qualitatively in Figure 2.18.

Avenas et al. [98, 99] provided a simple thermal resistance network model of a VC which assumed that the VC was a structure composed of differing layers with differing thermal properties. This was done for a copper VC with sintered powder wick. The conduction through the side walls was neglected. A more advanced numerical model was also suggested which considered side wall conduction for a silicon VC with a grooved wick structure. This model was solved with commercial software. This study mostly compared the performance of
phase change devices to the performance of solid heat spreaders and found that phase change devices were well suited to high performance applications that required compact solutions.

Chen et al. [100] proposed a numerical model which assumed that conduction was the only heat transfer mechanism within the VC. They experimentally validated this model with a copper-water VC. They then suggested that the analytical solution to thermal spreading of their particular scenario could be used to determine the effective thermal conductivity of VCs along with their experimental results. Two types of bulk conduction based models were considered; one with orthotropic and one with isotropic thermal conductivity. The values for effective thermal conductivity were found to be a function of the ratio of the heat source area to the VC area. Larger heat source areas led to larger values for effective thermal conductivity. With the isotropic approach, a single value of effective thermal conductivity was determined for the entire VC region and this ranged from 436-558 W/mK depending on the heat source size. With the orthotropic approach, a value for effective thermal conductivity was determined for each direction within the VC. The axial thermal conductivity was 49 W/mK (independent of heat source size) and the lateral thermal conductivity was between 1148 W/mK and 2319 W/mK, depending on the heat source size.

Wang et al. [101] proposed an analytical relation for the effective thermal conductivity of VCs based on experimental data of a copper screen mesh VC with water as the working fluid. Buckingham Pi theorem was used to relate variables into non-dimensional groups and

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**Figure 2.18** – The simple optimisation analysis performed by Prasher showed that there existed an optimum wick thickness which would balance the thermal and fluid mechanisms within the VC [97].
experimental data was used to determine the constants of the relation. The resulting relation was a function of VC geometry and heat flux. A maximum effective thermal conductivity of 870 W/mK was reported in their work for the VC.

Wang et al. [59] showed that a bulk conduction based numerical model with orthotropic effective thermal conductivity values was appropriate for modelling VC performance. They first experimentally measured the thermal performance of a copper-water VC. This VC had grooves on the condenser surface and a screen mesh wick on the evaporator. The experimental results were then used to determine the effective thermal conductivity values in each direction based on some simplifying assumptions regarding the flow of heat within the VC. In the lateral direction a value of 6088 W/mK was obtained and in the axial direction a value of 5 W/mK was obtained. These values were then used in a conduction based numerical model and it was shown that the maximum deviation between the results was 3.1%.

Chen et al. [102] investigated the applicability of two thermal resistance models to a phase change device. The device under consideration was similar in design and construction to a VC however varied in its objective; it was supposed to transfer heat along its length, rather than spread heat like a VC. The first model was purely one dimensional and thus didn’t consider any spreading resistance contributions. The second model was quasi two dimensional and thus considered spreading resistance from the heat source. It was found that the spreading model was a better prediction of the device performance, especially in cases where there was large spreading present i.e. for smaller heat sources. This model was also appropriate when the material had low thermal conductivity i.e. silicon. The spreading model generally had error of less than 10% and the non-spreading model had error between 20-40% when compared to experimental results. Hence spreading was an important consideration when developing numerical models for these types of phase change devices.

Chang et al. [103] developed a semi-empirical relation to predict the thermal resistance of VCs with total thicknesses of less than 0.5mm. A conduction based model was introduced to assist with this. It was based on the assumptions of isotropic effective thermal conductivity, symmetric boundary conditions, and that the temperature did not vary through the thickness of the VC. The effective thermal conductivity could be calculated from this model based on steady state temperature derivatives observed from the experimental setup. From these experiments, two values for effective thermal conductivity were obtained; for the evaporator and condenser region. The evaporator region had values in the range of 2.34-34.1 times that
of copper and the condenser region had values in the range of 0.35-1.98 times that of copper. The authors suggested that low values for the condenser region were linked to flooding within the condenser. A logarithmic relation was proposed for the thermal resistance of the VC. This relation was dependent on heat input with coefficients proposed to account for the filling ratio, orientation and vapour region thickness based on experimental data. The thermal resistance was found to decrease with increased heat input and vapour region thickness. The authors stated that the determined values were only valid for their particular design of VC.

2.3.2 Thermal-fluid

Koito et al. [104] developed a CFD model of the VC to analyse the velocity, pressure and temperatures distributions inside a VC. Some simplifications were present in this model; vapour flow was considered laminar, the wick was modelled as an isotropic medium and the VC had not reached dry-out. Experimental measurement of a copper-water VC with sintered powder wicks was used to validate the model. Velocity distributions revealed that vapour velocity was an order of magnitude larger than liquid velocity, mainly due to differences in density. Pressure distributions showed that the majority of pressure losses in a VC were contributed to the liquid wick section. Temperature contours (Figure 2.19) suggested that the heat source region had the largest temperature drops due to the very low effective thermal conductivity of the wick and small liquid velocities. The vapour and condenser region temperatures were very uniform.

![Numerical temperature contours in the VC with central wick column studied by Koito et al. Note the large thermal gradients present in the centre of the VC [104].](image)

Koito et al. [105] also investigated the effect of heat source size on the performance of a VC using the previously developed model. The condensation phase change resistance was found to be hardly affected by heat source size or heat input, however the evaporation phase change resistance was strongly dependent on the heat source size. In general the condensation resistance was much smaller than the evaporation resistance with values of 0.01 K/W obtained compared to 0.11 K/W, respectively. The evaporation resistance was found to
decrease with increased heat source sizes, however it was unaffected by changes to the heat flux. This has been shown in Figure 2.20. Further, the effect of a wick column was investigated numerically in this work. The authors reported that due to its low effective thermal conductivity, the use of a central wick column was not recommended. The spreading resistance obtained was 13% higher with a central wick column compared to without one. Even though there were thermal performance disadvantages, there were mechanical advantages to having a central wick column as this provided structural strength to the VC.

Figure 2.20 – Thermal resistance contributions in a VC. The spreading resistance ($R_{vc}$), evaporation resistance ($R_e$) and condensation resistance ($R_c$) as a function of heat flux (for a given heat source size) and of heat source size (for a given heat flux) [105].

Lu et al. [106] also explored the use of a central wick column through CFD modelling. They found that a central wick column was detrimental to thermal performance of the VC due to its low effective thermal conductivity however it could aid in return of condensate to the evaporator and thus extend the working range of the VC. They suggested that the use of a central wick column with a solid copper centre would help alleviate the thermal issues created by a column made entirely from sintered powder. This type of central column also aided in the return of condensate to the evaporator so was helpful in improving the overall performance of the VC. The authors also performed some investigation into the velocity and pressure distributions within the VC.

Vadakkan et al. [107] studied a three-dimensional transient CFD model of a phase change device similar to a VC with multiple heat sources. The phase change interface was modelled using kinetic theory. The model was useful for finding the location of the dry-out region.
within the wick and the heat flux which caused dry-out. Capillary requirements of the wicking structure were then explored based on these findings. The authors also determined that having multiple heat sources which were close together generally led to worse performance of the device.

Carbajal et al. [89, 108] studied a heat spreader that was exposed to an intense flame-like heat source. They introduced a finite volume model to determine the transient temperature, velocity and pressure distributions in the heat spreader. A specialised procedure based on kinetic theory was used to model the phase change processes within the spreader. Special treatment was also introduced to consider the fluid flow within the wick and vapour regions. As the heat spreader was subject to very high flame temperatures, some radiation effects were also considered. There was reasonable agreement between the numerical model introduced and the experimental results of the spreader. The authors stated that numerical convergence and instability were difficult problems to overcome. Nevertheless, it was determined that the heat spreader had much more uniform temperature distributions on the heat sink side and smaller hotspots on the heat source side as compared to an aluminium heat spreader. The two-phase heat spreader also had superior thermal response to the aluminium plate which was linked to the large latent heat capacity of the working fluid and its phase change abilities.

Ming et al. [48] numerically and experimentally investigated a VC with machined grooves on the evaporator surface. The tops of the grooves directly contacted the condenser surface which the authors suggested increased the axial heat transfer capabilities of the VC. Experimental results showed that the VC should be used for cases with large heat load and large heat sink area for small thickness spreaders. The numerical model examined the liquid distribution within a grooved channel of the wick (Figure 2.21). This distribution was found to be a function of the initial and boundary conditions and poor performance could result if these initial conditions were not optimal. Similarly, large heat fluxes would also lead to poor performance due to dry-out of the groove channel near the evaporator. They also explored the pressure losses in the groove channel and found that the vapour had larger pressure drops due to the vapour velocity being much larger than that of the liquid.

Harmand et al. [109] introduced a transient numerical model of a VC with multiple heat sources. This model combined a 3D thermal model with a 2D hydrodynamic model for calculating the fluid flow in the wick and vapour regions. They were coupled through the mass flux of evaporation and condensation processes. The overall model was able to capture
both thermal and fluid aspects within the VC. Thermal cycles were imposed on the VC and a copper heat spreader and the responses were monitored. Long thermal cycling was better handled by the VC and the copper heat spreader better handled short thermal cycles.

![Figure 2.21](image)

Figure 2.21 – The liquid film thickness (δ) according to the numerical model of Ming et al. [48]. At higher heat fluxes, the thickness of the liquid film decreases at the centre of the VC (x=0) thus dry-out occurs. At low fluxes, the liquid layer is thick and heat must conduct through this thick layer.

Ranjan et al. [110] developed a numerical model which considered the microstructure effects of the wick on the phase change process. This microstructure model was coupled with the model of Vadakkan et al. [107] to account for macroscale effects. The model developed considered the effects of wick geometry on the evaporation process by including thin film evaporation, meniscus curvature and Marangoni convection in the model. The phase change processes were modelled with kinetic theory. Their results suggested that the thermal resistance associated with the phase change process may become significant as the device becomes smaller and also as the wick becomes thinner (around 0.1mm) and of higher effective thermal conductivity (greater than 100 W/mK).

Ranjan et al. [111] further developed this model to consider VCs with wick thickness from 0.05-0.25mm and vapour space thickness from 0.2-0.4mm. These thicknesses were deemed problematic as large thermal resistances developed within the vapour region. The device performance was strongly dependent on the heat sink temperature as this altered the thermal properties of the working fluid. Higher heat sink temperatures led to favourable properties that reduced vapour flow resistance. Some optimisation analysis was also performed into the balance of heat flux and wick thickness. It was found that small wick thickness was optimal for low heat fluxes. As the heat flux increased, the optimal wick thickness increased as well.
Patankar et al. [112] modified the model of Vadakkan et al. [107] for the purpose of improving the condenser temperature uniformity of ultra-thin VCs. This was identified as a problem particularly for mobile devices which are susceptible to hot-spots in the thermal management solution. An algorithm was introduced here in order to reduce the computational costs for this demanding transient problem and an order of magnitude reduction in computational time was obtained with its use. The temperature uniformity on the condenser was found to be effected by the vapour region thickness and the effective thermal conductivity of the condenser wick. The authors suggested that a bi-porous wick on the condenser with high permeability could reduce the condenser surface temperature variation by as much as 37%

Patankar et al. [113] developed a simplified transient numerical model for VCs through scaling analysis. This reduced the governing equations to a set of first order differential equations. This model was validated against a finite volume model with good agreement and three to four orders of magnitude less computational costs. It was then used to simulate the response of a VC to multiple transient heat sources.

Patankar et al. [114] extended this model to explore the transient thermal response of thin VCs. The simplified geometry in the model had total thickness of only 0.1mm, however only the vapour and wick regions were considered. Three mechanisms which governed the transient performance of such thin devices were identified to be the total thermal capacity of the device, the effective in-plane diffusivity and the effective in-plane conductance of the vapour core. These mechanisms were further explored and comparison against a copper heat spreader was presented.

Wang et al. [115] developed a simplified numerical model to explore the effect of the sintered wick porosity and powder size on the thermal-fluid performance of a copper-ethanol VC. The model assumed the phase change processes occurred at the interface between the wicks and vapour region only. It was further assumed that the vapour region temperature was uniform and laminar flow was present throughout the model. They extensively explored the working fluid velocity and pressure distributions within the wick and vapour regions. The best performance was obtained when the pressure losses within the VC were close to, but did not exceed the capillary pressure of the wick. The wick porosity had a larger effect than the powder size on VC performance. It was suggested that the porosity of the evaporator wick should be larger than the porosity of the condenser wick however the optimal values of
porosity were dependent on the powder size. For a powder size of 62.5\(\mu m\), the ideal porosity was from 35-40\%, however decreasing the powder size to 26.7\(\mu m\) increased the ideal porosity to 40-45\%.

2.4 Summary and remarks

Some concluding remarks, statements and suggestions concerning the gaps and deficiencies in knowledge can be made from the preceding review of VC literature in section 2.2 and 2.3.

2.4.1 Wick structures

From the preceding literature review it could be said that the wick structure is the most important factor in the heat and mass transfer performance of the VC for multiple reasons:

- The wick has been identified to be a major source of thermal resistance, particularly in the evaporator region, due to its relatively low effective thermal conductivity [69, 97, 104]. As a result of this, the axial heat transfer capabilities have been shown to be poor compared to the lateral heat transfer capabilities of the VC.
- The wicking structure is also the location where the phase change process occurs [16]. As described in detail below, there are many difficulties associated with this.
- The wick strongly influences the fluid performance of the VC as well due to its presence in the liquid flow path and responsibility for generating capillary pressure to circulate working fluid. These aspects are very important for the design of VCs, however thin and ultra-thin designs may also require additional considerations [111, 112, 114].

Based on the importance of these aspects to VC operation, further opportunities exist for innovative solutions, technology and materials to create improved wick structures to better VC performance:

- Advanced manufacturing techniques such as 3D printing are able to produce more functional and optimised components across many engineering fields. The use of 3D printing for phase change devices is relatively unexplored (see the works of Refs. [116-118] for some examples) however these techniques could be exploited for the design and manufacture of advanced wick structures.
- Although foam wick structures have been used in some phase change devices, due to their advantageous properties (high effective thermal conductivity, large porosity and
large permeability [44, 119-124]) they are ideal candidates for VC wick structures. They should be further explored since foam technology is constantly being improved.

- Carbon based technology [91, 92] has also been shown to have beneficial properties (high effective thermal conductivity, large porosity and small effective pore size) and are relatively unexplored in phase change devices. This technology is also constantly improving as it is an area of intense research.

- The grooved wick can have very fine groove geometry when made with silicon due silicon fabrication techniques such as etching [98]. It is generally very difficult to replicate such fine geometry with metals. With current advances in manufacturing, improved techniques could manufacture metal wicks with intricate groove geometry [51]. This could be advantageous due to the very high effective thermal conductivity of grooved structures, in particular ones made from metal, and the capillary pressure associated with very fine grooves.

- Bio-inspired designs are seeing resurgence in many fields of engineering. The leaf vein wick structure from Refs. [54, 55, 83] are one such bio-inspired design with clear advantages in fluid performance. It is expected that there are other areas where natural phenomena could be exploited within the VC wick structure.

- Surface treatment and alteration has also progressed significantly in the past few years thanks to advances in associated technologies. The surface structure plays an integral role in thermal and fluid aspects within the VC, and can greatly influence the phase change processes occurring from wick structures. Some literature has explored the use of engineered wettability surfaces [60, 61, 63, 64, 78]. These can have some interesting performance features and should be further investigated.

- Further development of existing wick types should be performed. The use of hybrid or composite wick designs is a step in this direction whereby the specific needs for different regions within the VC are addressed by a unique wick. These have been studied in the literature (as shown in section 2.2.5) however these designs can be better developed in order to better utilise existing wick structures and technology.

Regarding phase change processes from wick structures, evaporation and condensation are still relatively unexplored or unclear. These are inherently complex processes with both heat and mass transfer mechanisms to consider. This is further complicated by the wicking structure and the many intricacies which this can introduce. At their simplest, these processes
can be modelled as convective processes with associated heat transfer coefficients. Still, there are a number of issues in this regard:

- Considering the evaporative heat transfer coefficient (this is generally regarded as the more important phase change process), this value has been found to vary greatly between researchers (see for example Refs. [52, 57, 67, 70, 71, 78, 96, 105]). From the literature studied, the value generally fell between $10^3 - 10^5 \text{ W/m}^2\text{K}$, which are quite large variations.

- Some of the literature suggested that the heat transfer coefficient was heat input/flux dependent [52, 57, 67, 70, 71, 78], whereas other literature has suggested that it was heat input/flux independent [105]. Further studies have also suggested that this value was temperature dependent [52, 78, 96] and that there existed dependencies between the evaporation and condensation phase change processes [67]. Thus, there exists a need to further investigate these aspects in greater detail. Systematic studies on these phase change processes including the effects of wicking structures and properties (similar to the work of Ref. [71]) should be performed to better understand these processes as they are very important to characterise the thermal performance of the VC.

There is additionally some uncertainty in the literature regarding the importance of different mechanisms of heat and mass transfer for the wick structures. The most common mechanisms mentioned are; heat conduction through the liquid saturated wick layer and the evaporative phase change process from the liquid-vapour interface. Other aspects which are seldom mentioned are the effect of convection through the liquid saturated wick layer (this is thought to have negligible influence, see section 3.2.2.2), the effect of fluid recession in the wick (this has been discussed by Refs. [48, 74, 79]) and the effect of boiling processes if the required wall superheat is met (see for example Refs. [44, 64, 70, 71, 81]). Altogether, these can make for some very complex interactions within the wick region. Regarding the dominant mechanisms (conduction and evaporation phase change), there is some confusion between the relative importance of the two. In numerical modelling, in particular for simpler numerical models, there is often only consideration of one of these mechanisms, but not both together. Some literature has suggested that these mechanisms can actually influence one another [110] which introduces more complexities. Thus there exists a general need to further address these heat and mass transfer aspects within the wick regions.
2.4.2 Numerical modelling and optimisation

As has been shown in section 2.3, numerical modelling often takes on different levels of complexity based on the needs and required outcomes of the model. Some of these studies aim to develop a deeper understanding of the complex mechanisms within VCs and how they can be improved, whereas other studies aim to simplify these mechanisms such that they can be generalised for wider use. These models can then have different uses and outcomes. This will be accompanied by differing complexity and resources required. Hence some of these are better suited for fundamental research applications and others have found application in the VC industry.

Generally, simpler numerical analysis can only model thermal aspects of the VC. These models generally aim to characterise the overall thermal performance of the VC by exploring the aspects which contribute to this. This is normally done with some form of the energy equation. Consideration of thermal spreading typically introduces complexities that prevent the use of simple one-dimensional thermal resistance networks or analytical solutions since spreading is inherently multi-dimensional. Introducing multi-dimensional heat transfer into these problems often requires more advanced schemes (finite difference, finite volume etc.) which need to be solved numerically. This is typical for most thermal models of VCs. Some thermal models have been explored in section 2.3.1.

Considering that heat is transferred through multiple mechanisms within the VC (which also involve mass transfer) most thermal models simplify these into conduction mechanisms. Thus all processes within the VC are replaced with effective conduction based heat transfer such that only the energy equation is required for a numerical solution to be found. Still, there are a few different approaches for this in the literature:

- One such approach was to consider the entire VC to be a bulk continuous medium with an effective thermal conductivity applied to this whole region [84, 100, 101, 103]. In some cases, the effective thermal conductivity was further broken down into separate, directional dependent values (i.e. lateral and axial directions) [59, 100].

- Another approach was to consider that the VC could be modelled as a layered structure where each region of the VC was a distinct layer [69, 95, 97-99]. This way the VC is composed of separate layers which are thermally connected to each other. In these layered schemes, knowledge of the effective thermal conductivity of each region within the VC is required. Depending on the region of interest, these values may not
be trivial to find and further work may be required in determining these values. This is particularly true for the wick and vapour regions of the VC, since these regions have inherent fluid and mass transfer aspects associated with them as well:

- For further discussions on the wick regions, please refer to section 3.2.2.2.
- For further discussions on the vapour region, please refer to section 5.2.

The literature involving these models typically uses experimental results in order to validate the conduction based model introduced. This is then used to determine the effective thermal conductivity (in either the bulk or layered approach) which can be generalised such that it can be used for a wider range of scenarios.

When the thermal and fluid aspects need to be considered in detail, generally a computational fluid dynamics (CFD) solution has been used in the literature. In order to model fluid aspects, the momentum and mass equations must also be considered. This transition introduces considerably more complicated schemes which demand significant computational resources and time. Hence these are often done with CFD software of which there are a wide variety of numerical approaches, methods and schemes available. These solutions can provide detailed information about variables (temperature, pressure, density, velocity etc.) at all points in the computational domain which the thermal scheme cannot. Some of these thermal-fluid models have been explored in section 2.3.2.

Still, there are many complexities associated with CFD solutions which require further assumptions. Additional complexities can be introduced by the modelling of wick geometry, phase change processes, vapour flow dynamics and any turbulence effects. These aspects are often too complex to completely model and thus they are simplified for numerical modelling purposes. As the design of VCs and their components progress, and they become subject to more challenging conditions, proper consideration of these aspects could become vital to meeting demands.

The VC is governed by thermal and fluid dynamics which result in the heat and mass transfer required for operation. There are intricate links between these aspects; the work of Ref. [97] provided a brief insight into how the wick structure was affected by some thermal and fluid factors that are competing within the wick. Other scenarios like this exist within the VC (for example Refs. [110, 111, 114, 115]). These situations are ideal candidates for optimisation analysis. A key here is the computational effort and time required to run these studies; the former work only required a simple numerical model compared to the latter works which
required complex numerical models. This could become a determining factor for the usefulness of optimisation studies. The development of a simple numerical model which can be used for optimisation and sensitivity analysis will be an invaluable tool for thermal designers of VCs.

In the development of such models, the distinction between VCs of regular against “thin” or “ultra-thin” design cannot be overstated. These differences can lead to varying modelling requirements and demands since the physical mechanisms at play can change. Some aspects of these governing mechanisms have been explored in the literature (for example in Refs. [110-112, 114]. Ultimately, a VC designed for a PC application might be drastically different to one designed for a mobile application. In general, the thickness would be a good way to categorise these devices since this can have such a large effect on these aspects. As a general rule of thumb, it could be said that designs of less than 1.5mm thickness could be categorised as “thin” and designs of less than 0.5mm thickness could be categorised as “ultra-thin”.

2.4.3 Research focus
The previous sections are by no means exhaustive of the potential opportunities regarding the VC. There are still many areas for further improvement, research and development in both academic and industrial senses. This is only intended to be a brief guide of some current issues from the literature regarding VCs for future researchers.

The research undertaken in this work has only considered some of these gaps. These have been formulated through the research questions posed in section 1.2. Answering these research questions will address the gaps discussed in the previous sections. More specifically, the development of a composite wick VC and a simple conduction based thermal model helped to answer these research questions and clarify the gaps.
Chapter 3 – Methods

In order to fulfil the aims of this research, a combination of experimental and numerical methods were required. This chapter discusses the development and details of these methods. A copper-water VC was designed and manufactured for this research. This VC formed the basis for both experimental and numerical methods of this work. Detailed measurements of the VC were taken within the experimental setup which allowed for its thermal performance to be characterised. This further permitted the use of the developed numerical model to carry out a deeper investigation and analysis into VC operation and performance.

3.1 Experimental method

These sections detail the development of the copper-water VC used in this research. The VC was designed with the intention of being a heat spreader that efficiently transfers heat from small heat sources (or high heat fluxes) to large heat sinks (or small heat fluxes) with minimal spreading resistance. Furthermore, a special purpose experimental setup was also developed in order to accurately measure the spreading resistance of the VC. The philosophy, principles and details behind these designs are discussed in depth in the following sections, as the procedures followed in order to manufacture and make use of these experimental components.

3.1.1 Development principles and philosophy of vapour chamber

As has been discussed previously, a composite wick VC was introduced in this work. The aim of this composite design was to use the inherent advantages in properties of differing wick types in regions where they could be exploited. This led to the wick design that employed multiple wick structures on the evaporator side (i.e. a composite wick) and no wick structure on the condenser side.

Sintered powder wicks are generally of high effective thermal conductivity, high effective capillary radius and low permeability (see Chapter 2 and Ref. [16]). The best location for this wick structure was deemed directly above the heat source. This could exploit the high effective thermal conductivity of the structure and minimise spreading resistance. It has been explored in the literature review that the evaporator wick often has large temperature gradients develop in it due to the relatively low effective thermal conductivity of the wick compared to the container region. This will lead to large thermal resistance associated with the wick. This can be appreciated by considering the thermal resistance associated with one-dimensional heat conduction which derives from Fourier’s law as shown in equation (3.1).
\[ R = \frac{x}{kA} \]  

Here \( x \) is the conduction distance, \( A \) is the area and \( k \) is the (effective) thermal conductivity. Low effective thermal conductivities will lead to large thermal resistances. Note that equation (3.1) is technically only valid for one-dimensional conduction and that conduction within the VC multi-dimensional. Even in the very thin container and wick regions of the VC, there will be some degree of multi-dimensional heat transfer i.e. spreading, and (3.1) will not be strictly valid. However, this is intended as qualitative analysis only and the outcomes will be similar in the multi-dimensional case where spreading is considered. The equations for spreading are far more complicated in nature than (3.1). Thus this was used for brevity. Further, due to the thinness of the wick, the conduction distance is quite small so the one-dimensional assumption is a reasonable approximation. The validity of this is later explored in section 5.3.

This is further amplified by the high heat flux experienced in this location since it is exposed to the heat source. The only spreading potential to this point is in the container which is quite thin. It is the two-phase mechanisms which are responsible for the spreading ability of the VC and at this location these have not yet been encountered. Thus heat must first conduct through the copper container, and then conduct through the wick layer before it reaches the liquid-vapour interface and phase change heat transfer can occur. A simple representation of this has been included in Figure 3.1. For some further discussions on heat transfer mechanisms within the VC, refer to section 3.2.2.2 and the results in Chapter 4 and Chapter 5.

![Simple heat transfer schematic of the evaporator region. Heat conduction dominates in the container and wick regions. The thermal resistance associated with this is influenced by the (effective) thermal conductivity of the region.](image-url)

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Using the one dimensional assumption of equation (3.1), the temperature drop could be simplified with the process below which results in equation (3.2).

\[
\Delta T = \dot{Q}R
\]

\[
\Delta T = qA \cdot \frac{x}{kA}
\]

\[
\Delta T = \frac{qx}{k}
\]

(3.2)

Here \(\Delta T\) will be amplified by the large heat flux \((q)\) and small effective thermal conductivity. Thus, it can be seen that it is advantageous to use wicks with high effective thermal conductivity in this region as the temperature drops can be minimised as a result.

The wire screen mesh generally has low effective thermal conductivity and large permeability (see Chapter 2 and Ref. [16]). This would lead to large thermal resistance in (3.2). Thus the best location for this structure was away from the heat source region where it was not expected to experience high heat flux. Some spreading effects may carry heat to this region however the majority of heat was expected to transfer through the sintered powder wick. Hence even with the apparent large thermal resistance of this wick, it should not contribute significantly to the VC thermal resistance. This has been further explored in section 5.3.2.

The large permeability of the mesh wick was important from a fluid perspective since the permeability of the wicking structure was linked to the pressure losses within the VC. Liquid pressure losses arise when fluid flows through the wicking structure as it must do to return to the evaporator region. These liquid pressure losses, in a simplified geometry such as a heat pipe [16], can be expressed through equation (3.3).

\[
\Delta P_l = \frac{\mu_l L \dot{m}_l}{\rho_l KA}
\]

(3.3)

Here \(\Delta P_l\) is the liquid pressure loss, \(\mu_l\) is the liquid viscosity, \(L\) is the effective length (relevant for heat pipe geometry), \(\dot{m}_l\) the mass flow rate of liquid, \(\rho_l\) is the liquid density, \(K\) is the wick permeability and \(A\) is the flow area. By maximising the permeability, the liquid pressure losses can be reduced. The permeability of sintered powder wicks is generally smaller than the for wire screen meshes; thus by placing the wire screen mesh around the sintered powder mesh, the liquid pressure losses can be minimised in this region. In this VC, much of the fluid flow path is through the wire screen mesh with high permeability such that liquid pressure losses can be minimised. This can be seen by the respective wicking areas in Figure 3.3a.
The importance of liquid pressure losses is seen from the capillary limit for VCs. The capillary limit is the condition which allows the wicking structure to sufficiently circulate working fluid in the VC without drying-out. Exceeding this limit can lead to poor thermal performance. This can be expressed in equation (3.4).

$$\Delta P_c \geq \Delta P_v + \Delta P_l$$  \hspace{1cm} (3.4)

Here $\Delta P_c$ is the capillary pressure and $\Delta P_v$ is the vapour pressure loss. Note that gravitational pressure losses have been neglected here. This equation states that the capillary pressure generated by the wick must be greater than the sum of the vapour and liquid pressure losses in order for the capillary limit to not be exceeded. These terms have been represented schematically in Figure 3.2 for a section of the VC.

![Figure 3.2](image.png)

*Figure 3.2 – The various pressure loss components within the VC are important for ensuring that the capillary limit is not exceeded. It is assumed here that the evaporative interface is located above the heat source in the sintered powder wick region and that the condensed fluid flows through the wire screen mesh and sintered wick before reaching the evaporative interface.*

The capillary pressure generated by the wick is also influenced by the choice of wicking structure. Capillary pressure can generally be expressed in the form of equation (3.5).

$$\Delta P_c = \frac{2\sigma_l}{r_{eff}}$$  \hspace{1cm} (3.5)

Here $\sigma_l$ is the liquid surface tension and $r_{eff}$ is the effective capillary radius. The effective capillary radius is a function of the wick type. It can depend on a variety of factors and is
often found through experimental measurements. For sintered particle structures it can be quite small which will maximise the capillary pressure generated by the wick. It tends to be larger for wire screen mesh structures. Thus placing the sintered powder wick at the heat source has the additional benefit of increasing the capillary pressure generated here. This will help to maintain fluid circulation in this region so that it won’t easily dry-out.

Thus, the wicking arrangement on the evaporator side was chosen with consideration to maximising the thermal and fluid performance of the VC. A composite wick structure was selected for the following reasons:

- A high effective thermal conductivity sintered powder wick above the heat input region would help to minimise temperature drops through this region.
- The small effective capillary radius of this wick would also ensure that capillary pressure generated at the evaporator region was large.
- Away from this region, where thermal aspects were expected to be less important, the large permeability of the wire screen mesh would reduce liquid pressure losses. Thus this was placed around the sintered powder wick.

For this particular arrangement of VC, there was no wick placed on the condenser side. This was done for a few reasons:

- All wick structures have low effective thermal conductivity compared to base metals and thus temperature drops through the wick will lead to reduced performance. Conduction losses through the wick were eliminated by having a wickless condenser.
- The copper columns (later discussed) helped to return condensed liquid to the evaporator side. This task would conventionally be performed by a wick.
- The VC was tested in a gravity assisted horizontal position in this work.

### 3.1.2 Design, manufacture and assembly of vapour chamber

With all the considerations from the previous section, the following VC was constructed for this work. The overall dimensions of the VC were 123mm x 140mm with total thickness of 3mm. The container was made of 0.8mm thick copper (C10200). As described previously, a sintered copper powder wick was used on the evaporator side directly above the heat input region. It was of size 40mm x 40mm x 0.5mm. It was made from copper particles with size of 150-300μm and had porosity of approx. 40%. Two layers of copper wire screen mesh (#60) were laid around the central sintered powder wick on the evaporator side. The final thickness
of the wire screen mesh was approx. 0.5mm. No wicks were laid on the condenser of this VC. Refer to Figure 3.3 for a representation of this wick and overall VC arrangement.

![Diagram of VC arrangement](image)

Figure 3.3 – Representation of the evaporator wick (a) and a cross section of the arrangement of the VC developed in this research (b). Sintered columns not shown.

In order to manufacture this VC, the top and bottom copper plates were first machined out of copper block. The bottom copper plate was machined so that the side walls of the VC were integral with the bottom plate, whereas the top copper plate was flat. Copper columns were added to the VC in order to provide mechanical strength and prevent deformation under vacuuming, charging and operational conditions. Furthermore, they helped to maintain flatness of the device during these conditions and provided a return path for the condensed fluid. These columns with diameter of 3mm were machined directly into the bottom copper plate such that they too were integral with the bottom plate. The column height was dictated by the thickness of the wick and vapour regions to ensure there was direct contact with the top copper plate once assembly was completed.

The respective wicking structures were then sintered to the bottom plate which formed the evaporator side. This sintering procedure created integral connections between the wicks and bottom plate. A connection was also formed between the inner powder wick and outer mesh wick during this procedure. The bottom plate (with wicks sintered onto it) and top plate (wickless) were then joined together along their edges (the side wall was directly machined into the bottom plate) by a brazing procedure in order to create a vacuum tight vessel. A small filling port made from copper capillary tube was also provided in order to allow for vacuuming and filling of the working fluid. See Figure 3.4 for an image of the fully assembled VC.
Before charging the VC, it was tested to ensure that it was leak proof. This was done with a helium leak detection method whereby helium was injected into the sealed VC unit through the charging port. A probe was used to detect if any helium was leaking through the seals of the VC. These seals which were tested included the main brazed joint which connected the bottom and top copper plates, and the seal which joined the capillary tube to the VC. No helium was detected during this which validated that the seals were leak proof.

In order to charge the VC, it was first connected to both vacuum pump and piston-cylinder via an arrangement of valves. Inside the piston-cylinder setup was distilled, deionised and degassed pure water. The vacuum pump was turned on and allowed to run until the vacuum gauge read -100kPa. After this, a fixed amount of water was charged into the VC via the piston-cylinder. The filling port was then crimped and sealed to be leak tight. The amount of water filled into the VC was then confirmed by measurements of the VC weight before and after this procedure. A filling ratio of approx. 30% was obtained.

The machining, assembly and charging of the VC was performed by Fujikura Ltd who have many years of experience and expertise in the manufacture of phase change devices including VCs.
3.1.3 Experimental setup

An experimental setup was developed in order to determine the spreading resistance of heat spreaders as required by this research. Spreading resistance is commonly used to measure the ability of a material or device to spread heat from concentrated heat sources. Not only was this value used to discuss the thermal performance of the heat spreaders in section 4.1, however it also formed an integral part of the numerical work in sections 5.1 and 5.2. Spreading resistance ($R_{sp}$) is defined with equation (3.6). Note that it follows the general definition of thermal resistance as described in equation (2.1).

$$R_{sp} = \frac{T_{evap} - T_{cond}}{\dot{Q}}$$  \hspace{1cm} (3.6)

Equation (3.6) states that the spreading resistance is equal to the average temperature of the evaporator surface ($T_{evap}$) minus the average temperature of the condenser surface ($T_{cond}$) divided by the heat transfer rate ($\dot{Q}$) through the heat spreader. The measurement of these values is later discussed in this section.

Note that the common nomenclature for VC has been followed in the definition of spreading resistance. That is, evaporator and condenser terms have been used in equation (3.6). This definition is still valid for the copper heat spreader tested in this work (section 3.1.3.3), with a slight adjustment required to the terminology. For copper, the use of evaporator and condenser terms is invalid since there are no phase change mechanisms occurring in solid metal heat spreaders. Rather, for the definition of spreading resistance for the copper heat spreader, these two temperatures are just the hot and cold side surface temperatures of the heat spreader, where heat was input and removed, respectively. These represent the exact same locations as for the VC heat spreader. The use of evaporator and condenser terms is common within VC literature and thus has been followed in this work, generally even in cases relating to the copper heat spreader.

Care was taken in the measurement of temperature terms $T_{evap}$ and $T_{cond}$. As was discussed in section 2.1, many authors prefer simpler measurements which are less accurate representations of the spreading ability of heat spreaders. However, in this work great care was taken to ensure that the correct surface temperature was measured for the heat spreaders. These methods have been discussed in detail in the following sections.
3.1.3.1 Design

The experimental setup developed in this work to measure the spreading resistance of heat spreaders has been shown schematically in Figure 3.5. It consisted of a wind tunnel with a test section. Each of these components will now be discussed in detail.

![Figure 3.5 – Schematic of the overall experimental setup with the wind tunnel and test section. Not to scale.](image)

The wind tunnel was used to provide forced air convection over the heat spreader and to measure the heat transfer rate through the spreader. In order to provide air flow, a DC fan was placed onto the outlet of the wind tunnel. This fan would pull air through the wind tunnel and was controlled with a variable DC power supply. The measurement of air flow is described in section 3.1.3.2.

In the design of this wind tunnel, it was deemed that an average air velocity of 3m/s was appropriate for tests based on uncertainty analysis and thermal considerations of the heat spreaders. The maximum average velocity was expected to not exceed 5m/s in this research. Based on the average air velocity in the duct being 3m/s, the Reynolds number ($Re$) within the circular duct can be found with equation (3.7).

$$Re = \frac{\rho U D}{\mu}$$  \hspace{1cm} (3.7)

Here $\rho$ is the air density, $D$ is the duct diameter and $\mu$ is the air dynamic viscosity. For this analysis, the properties of air were evaluated at 20°C since this was expected to be the average air temperature in the wind tunnel prior to the test section. Based on this and the velocity considerations, the Reynolds number was found to exceed 10,000 thus it was expected that turbulent flow would be present. From this, the entry length would be approx. 10 diameters downstream of the entrance [125]. A tube bundle flow straightener was placed near the entrance to the wind tunnel and the anemometer traverse was placed 15 diameters downstream of this flow straightener. It was expected that the flow was fully developed by
this point. The inlet and outlet thermocouple array are discussed in section 3.1.3.2. The actual average air velocity in the tests was measured to be 3m/s (refer to Table 4.1). This was achieved by varying the voltage supplied to the fan as required. Hence the above assumptions of entrance lengths and turbulent flow were realistic.

The heat spreader and heat source were located within the test section which was connected to the wind tunnel as shown in Figure 3.5. The test section was also where the required surface temperatures of the heat spreaders were measured. A plan view of the test section base (with the cover removed) has been shown in Figure 3.6.

![Figure 3.6 – Plan view of the test section base (without cover shown) and location of the non-central heat source.](image)

The location of the heat source on the underside of the heat spreader was non-central in this work. Many of the heat sources within the literature are centrally located for VC heat spreaders. Here, a non-central heat source was used to provide some more information to the literature about the thermal performance of heat spreaders with a heat source that was non-centrally located. The location of the heat source was driven by the location of the sintered powder wick. The centre of the heat source was 20mm from the geometric centre of the heat spreader in the x-direction and in the geometric centre in the y-direction. This also coincided with the centre of the sintered powder wick of the VC. This location was constant for all heat source sizes. The heat source sizes were variable in this work as shown in Table 4.1. They were made from copper (C10200) with multiple DC cartridge heaters embedded within them. These heaters were powered with a Manson SPS9400G variable DC power supply such that heat input could be varied. The voltage and current supplied could be monitored during tests.
Air flow from the wind tunnel passed through the flow channel of the test section. This flow channel had cross sectional geometry of 0.123m x 0.02m (width x height). It was created by placing the lid component onto the base of the test section as shown in Figure 3.7.

![Figure 3.7](image)

*Figure 3.7 – The front view of the test section base and cover. Placing the cover on the base created the flow channel which was connected to the wind tunnel. The contact points are later discussed.*

When the lid component was placed onto the base, it provided the side boundaries and top boundary of the flow region. The bottom boundary was provided by the test section base and heat spreader itself. The cover component had a transparent acrylic sheet placed over it to allow for visual inspection of the heat spreaders. During tests, insulation was placed over this sheet to minimise heat losses.

The cross sectional area in the test section was slightly larger than in the wind tunnel, thus the average velocity was slightly less in the test section (reduced by approx. 15%). The Reynolds number could be calculated using a similar procedure shown with equation (3.7). It was determined that the Reynolds number in the test section was nearly 6,000 and thus turbulent flow was expected here as well.

The test section base and cover components were made from calcium silicate which had been milled as required. This material had very low thermal conductivity of less than 0.08 W/mK according to manufacturer specifications, thus would minimise heat losses to the environment. Only the top of the cover component was made from acrylic sheet and this too was covered with insulation during testing.

The heat source itself was placed within the test section base on a set of four evenly distributed springs. When the test section had its cover component removed, the springs would cause the heat source and heat spreader to protrude out of the setup as shown in Figure 3.8.
Figure 3.8 – The spring setup would ensure that the heat source and heat spreader protruded out of the base when the cover component was removed.

When the cover component was placed onto the test section base, the heat spreader and heat source would recede slightly into the base. These would then be locked into place and the cold side surface of the heat spreader would sit flush with the upper surface of the base. The springs would compress slightly to compensate for this as shown in Figure 3.9. These springs would thus provide some pressure on the mating components and improve the thermal contact conditions.

Figure 3.9 – Once the cover component (not shown for clarity) was placed on the test section, small contact points (also not shown for clarity) would cause the heat source and heat spreader to recede into the base.
The cover component was designed such that it did not interfere too much with the cold side surface of the heat spreader. Thus there were only a few contact points along the edge (parallel to the flow direction) of the heat spreader. This has been shown in Figure 3.10. With these contact points, the cover component would cause the heat spreader to recede into the test section and the springs to compress as shown in Figure 3.9.

![Figure 3.10 – The cover component showing the approximate location of the contact points between the cover and heat spreader which lay beneath it.](image)

The cover component of the test section was held in place with separate spring loaded clamps on either side. These clamps would hold the entire test section together. They were compressed to the same level for all tests. Similarly, the springs which the heat source rest on were the same for all tests. It was thus expected that the pressure applied to the heat spreader and heat source were constant throughout testing.

Re-assembly of the experimental setup was only performed when changing the heat source sizes or changing the heat spreader. During any re-assembly, the heat spreader and heat source would be thoroughly cleaned with isopropyl alcohol to ensure surfaces were free from oil and grease. New thermal paste would be applied after the surfaces were cleaned. A small amount of thermal paste (8.5 W/mK) was spread evenly on the mating copper surfaces. After these steps, the test section would be re-assembled as highlighted above.
3.1.3.2 Measurement

Thermocouples were used commonly at multiple locations throughout the experimental setup for temperature measurement. Type T thermocouples with wire diameter of 0.2mm were used. All thermocouples were calibrated prior to use in an ice bath. If any thermocouple measured outside the range of -0.1°C to +0.1°C in the ice bath then it was discarded and a new thermocouple was used. They were monitored with a Yokogawa MV200 data acquisition system. Data was monitored during the entire test period, but only steady state data was recorded and used in this work. A hot wire anemometer (Testo 405i) was used to measure the air velocity in the wind tunnel for each measurement location as is described below.

The measurement of the heat transfer rate first required the average velocity \( (U) \) of the air to be determined. These measurements were taken in accordance to Refs. [126, 127] where the log-linear method was used to determine the average air velocity through a duct. Seven measurement points (1 in the centre) at set locations in the radial direction were used. Multiple traverses through the cross section were taken, each set at 60° apart from each other as seen in Figure 3.11.

![Figure 3.11](image)

*Figure 3.11 – The measurement locations in accordance with Refs. [126, 127] based on the log linear method. Each measurement point is denoted with a red dot.*

The choice of measurement locations based on the log-linear method meant that weighting coefficients were equal to 1 and the average velocity of air in the duct was equal to the
arithmetic average of the measured values. The average air velocity was thus found from equation (3.8).

\[ U = \frac{1}{N} \sum_{i=1}^{N} U_i \]  

(3.8)

Here \( U_i \) is the air velocity measurement at the \( i^{th} \) point and \( N \) is the number of measurement points. The average air velocity was then used to find the mass flow rate of air (\( \dot{m}_{air} \)) in the duct via equation (3.9).

\[ \dot{m}_{air} = \rho A U \]  

(3.9)

Here \( \rho \) is the air density and \( A \) is the cross-sectional area of the duct. The air density was evaluated at the average air temperature measured by the thermocouple array prior to the test section as this was the closest location to the anemometer traverse. From this the rate of energy addition to the air stream (\( \dot{Q} \)) could be found via equation (3.10).

\[ \dot{Q} = \dot{m}_{air} c_p \Delta T_{air} \]  

(3.10)

Here \( c_p \) is the specific heat of air and \( \Delta T_{air} \) is the temperature difference of the air stream measured across the test section. This temperature difference was measured with thermocouple arrays prior to and after the test section as seen in Figure 3.5. The average of multiple readings was taken for both array locations. As the test section was well insulated, the rate of heat transfer through the air stream was assumed to be equal to the rate of heat transfer through the heat spreader.

The rate of heat transfer through the air stream could be compared against the electrical power input to the heat spreaders to assess the effectiveness of the insulation. It was generally found that 90% of the electrical power input was transferred into the air stream, with the remaining 10% being lost to the environment. This heat leakage most likely occurred through the bottom surface of the test section base however was deemed acceptable given it was only a small amount of the total heat input. Nevertheless, for all calculations requiring heat input (for example equation (3.6)), the value used was equal to the rate of energy addition to the air stream (equation (3.10)) and not the electrical power input.

Measurement of the evaporator surface temperature (or the hot side surface temperature for the case of the copper heat spreader) will now be discussed. This measurement was complicated as the surface was also a heat input surface. This introduced an issue as it was in
direct contact with the heat source, and hence direct contact with thermocouples as well was
difficult. This was further complicated by the thinness of the copper container (0.8mm).

These factors complicated the direct measurement of this surface temperature. Instead, a
system was developed to deduce the hot side surface temperature of the heat spreader. A
cross section through this system has been shown in Figure 3.12. It consisted of a copper
heater block, copper connector block and the heat spreader. It also had multiple
thermocouples present for temperature measurement. The sources of heat were multiple DC
cartridge heaters embedded into the bottom of the copper heater block. The use of this system
for measuring the evaporator (hot side) surface temperature is now discussed.

![Figure 3.12](image)

*Figure 3.12 – A cross section of the system developed to deduce the evaporator (hot side) surface temperature
of the heat spreaders. Insulation not shown.*

The heat source temperature \((T_{hs})\) was measured with four thermocouples placed in the
copper heater block. These thermocouples were soldered into 0.7mm holes that were drilled
into each face of the heater block, approximately 1.5mm from the top surface. The heater
block is shown in Figure 3.13. The average of the four thermocouple readings was taken to be
the heat source temperature.
Another component, the copper connector block, was placed on top of the copper heater block. It had the same dimensions as the heater block, however was of thickness 3mm. This also had four thermocouples soldered into 0.7mm holes on each face of the block. These holes were located midway through the thickness of the block as shown in Figure 3.14. The average of the four readings was taken to be the midpoint temperature ($T_{mid}$).

The heater block and connector block were thermally placed in series. The temperature drop between $T_{hs}$ and $T_{mid}$ would naturally depend on conduction losses and the interface conditions present between the two. This is shown with the series thermal resistance network in Figure 3.15.
The use of a series thermal resistance network was appropriate for this scenario as the heat transfer in the heater block was mostly one-dimensional, particularly in the region where the thermocouples were placed. Although the cartridge heaters were cylindrical in nature, they were embedded into the bottom of the heater block with a distance of over 35mm between the heaters and the thermocouples. Computational analysis of the heater block was performed to assess the one dimensional flow assumptions. Through this it was validated that there was sufficient distance to allow heat to diffuse through the heater block and create one-dimensional flow of heat in the region near the thermocouples as required. The sides and bottom surface of the heater block were well insulated to reduce heat leakage and improve the one-dimensional flow of heat within the block.

Figure 3.15 – The series thermal resistance network created by joining the copper heater block to the copper connector block. Note that in reality there was no gap between the surfaces, this was done in the figure to make clear the thermal interface present.

Here $R_{cond}$ and $R_{int}$ are the thermal resistances associated with conduction and the interface, respectively. The two conduction resistances were equal in size since the conduction lengths were equal. $T_{surf,1}$ and $T_{surf,2}$ are the temperatures on each side of the interface i.e. $T_{surf,1}$ is the surface temperature of the heater block (just prior to the interface) and $T_{surf,2}$ is the surface temperature of the connector block (just after the interface). The temperature difference between these two was dependent on the interface conditions present. This resistance is typically difficult to predict as it depends on a variety of factors which are
generally hard to measure [125]. This is in contrast to the conduction resistances, which are quite easy to predict for this one-dimensional case (as is later shown). However, by measuring the temperature difference between $T_{hs}$ and $T_{mid}$ there was no need to predict this interface resistance since it could be measured.

At this point, the heat spreader under consideration was placed in series on the copper connector block to form Figure 3.12. Consequently the thermal resistance network of Figure 3.15 can be updated to include this component. This results in the new thermal resistance network shown in Figure 3.16.

![Figure 3.16 – Thermal resistance network of overall setup shown in Figure 3.12. Up until $T_{mid}$ this is the same network presented in Figure 3.15. Note that in reality there was no gap between the surfaces, this was done in this figure to make clear the thermal interface present.](image)

Here $R_{int,1}$ and $R_{int,2}$ are the thermal resistances associated with the interface between the heater block and connector block, and the connector block and heat spreader, respectively. $T_{surf,3}$ and $T_{surf,4}$ are also introduced. These are, respectively, the surface temperature of the connector block (just prior to the second interface) and the surface temperature of the heat spreader (just after the interface). $T_{surf,4}$ is actually the surface temperature required in the calculation of spreading resistance i.e. it is equal to $T_{evap}$, the temperature of the heat input surface of the heat spreader.
This series thermal resistance network can now be simplified based on the one-dimensional heat flow assumption mentioned above. The three conduction resistances ($R_{\text{cond}}$) in Figure 3.16 are equal since the conduction lengths are equal. These resistances can be shown to be minor with some simple calculations using Fourier’s law. Consider the smallest heat source size in this work of 20mm x 20mm (this will lead to the largest resistance), and the conduction distance to be 1.5mm for the copper components. Using a one-dimensional heat flow assumption, the conduction thermal resistance will be:

$$R_{\text{cond}} = \frac{x}{kA} = \frac{0.0015 m}{390 \frac{W}{mK} \times (0.02 m \times 0.02 m)} = 0.01 \frac{K}{W}$$

If the heat load being transported was 40W (the maximum in this work), then the temperature drop through this resistive component will be:

$$\Delta T = \dot{Q} \times R = 40W \times 0.01 \frac{K}{W} = 0.4K = 0.4^\circ C$$

Thus, the maximum expected temperature drop due to conduction for the worst case scenario in this setup was only 0.4°C. Compared to the temperature drop through the interface, the temperature drop owing to conduction losses was minor. The temperature drop through the interface could be deduced from experimental measurements of $T_{\text{hs}}$ and $T_{\text{mid}}$. From these measurements, the observed temperature drop was much greater than what would be expected from only conduction losses. Thus, it could be safely assumed from these experimental measurements that the resistance of the interface was much greater than the conduction resistances.

Since the conduction resistances were minor, they could be neglected in the thermal resistance network. Removing these left only the interface resistances as has been shown in Figure 3.17.

![Figure 3.17 – Thermal resistance network neglecting the conduction losses.](image)

Furthermore, the thermal resistance of the interfaces can be assumed equal i.e. the interface resistance between the heater block and connector block ($R_{\text{int},1}$) was equal to the interface
resistance between the connector block and heat spreader \( (R_{int,2}) \). This was based on the fact that in this work all components were machined flat and cleaned thoroughly before use. They were all made from copper and so similar interface conditions could be expected. The assembly of the experimental rig was the same for all tests (see section 3.1.3.1) so factors like applied pressure were constant through testing. Hence it would be reasonable to assume that these two interface resistances were equal. This creates the thermal resistance network shown in Figure 3.18.

![Thermal Resistance Network](image)

*Figure 3.18 – Final thermal resistance network with equal interface resistances*

The measurement of \( T_{hs} \) and \( T_{mid} \) was undertaken by the average of four thermocouples for each value. Hence the resistance between these two points could be determined. Based on the previous discussions it would be reasonable to assume that the resistance between \( T_{mid} \) and \( T_{evap} \) would be equivalent, and thus the temperature drop between these points would also be equivalent. Making this assumption allowed for the evaporator surface (hot side) temperature to be calculated through equation (3.11) which can be derived from the series thermal resistance network of Figure 3.18.

\[
T_{evap} = T_{hs} - 2(T_{hs} - T_{mid})
\]  

(3.11)

Hence, in this work two interfaces were purposely introduced such that the measured resistance of the first interface would allow for a reasonable estimation of the second interface. Without such system, direct measurement of the hot side temperature would have been very difficult as discussed previously. With only a single interface, the interface thermal resistance could not have been found easily. This method allowed for the evaporator surface temperature to be experimentally deduced. It gave repeatable and consistent results through all tests.

Although there was no heat sink present on the condenser surface of the heat spreader, this surface temperature was similarly difficult to measure. The measurement of surface temperatures with thermocouples is not trivial and large errors may result if proper attention
is not given [128-130]. Some of the issues of thermocouple surface temperature measurement are listed below:

- The accuracy of results relies strongly on the thermal connection between the thermocouple tip and the surface to be measured. If there is poor thermal contact, then a large resistance could be present between the surface and tip and hence the measured temperature could deviate from the actual surface temperature.
- Heat leakage through the thermocouple could interfere with validity of the measurement. The flow of heat through and around a surface with a thermocouple attached to it can be altered by the presence of the thermocouple since they tend to act like extended surfaces (or fins) which conduct heat away from the surface and interfere with the boundary conditions.

These issues were addressed in accordance with Refs. [128-131]. In order to address the thermal contact issue, a number of different scenarios were tested as shown in Figure 3.19. They were; attachment with solder (Figure 3.19a), attachment with liquid metal (Figure 3.19b), attachment with thermal paste (Figure 3.19c) and attachment with polyimide tape (Figure 3.19d). The thermal conductivity of the solder, liquid metal and thermal paste used were approx. 50 W/mK, 70 W/mK and 10 W/mK, respectively. There was no filler material used with the polyimide tape method.

To assess which thermocouple had the best thermal attachment, the copper block was heated uniformly from the bottom side and all faces except the top surface were insulated. Some air flow was provided to help cool the block and mimic conditions present in the test section. The attachment method which measured the highest temperature was deemed to have the best thermal contact with the surface. All thermocouples were calibrated prior to these tests and all measured the same temperature before the test was run. After numerous trials, it was evident that both soldered attachment and liquid metal attachment were superior to the thermal paste and polyimide tape solutions. It was difficult to ensure good contact with the thermocouple tip and surface with these latter two methods, thus they were deemed unacceptable for measuring the surface temperature.
Figure 3.19 – The attachment methods tested to assess the adequacy of the thermal contact between the thermocouple tip and surface. The copper block was heated from the bottom (not shown) and insulated from all sides except the top surface where air flow was provided.

Both soldered and liquid metal attachment methods read the same temperature value for numerous tests so both were deemed to have good thermal contact with the surface. The liquid metal solution was chosen for the condenser surface temperature measurement as it was far easier to implement this method. Soldering the thermocouple tip to the surface was difficult to do and it was further complicated by the high temperatures required for soldering and possible issues this could create for the VC heat spreader.

The heat leakage problem was mitigated by using thermocouples with wire diameter of 0.2mm. It was expected that heat leakage through the thermocouples to the outside environment would have been less than 0.05% of the total heat input to the system which is a negligible amount of heat lost. In order to minimise disruptions to the surface of the heat spreader, the thermocouples were run along the surface before taking off from it. This is based on advice from Refs. [128, 130, 131]. This meant that the measurement location would be less effected by the presence of the thermocouple. Hence a variety of factors were considered in order to accurately measure the condenser (cold side) surface temperature of the heat spreaders.
Eight thermocouples were attached to the surface as shown in Figure 3.20. A weighted average was used to find the average surface temperature of the condenser ($T_{\text{cond}}$). This was done as the thermocouples were not evenly distributed on the condenser surface and thus the contribution from each thermocouple was not equivalent. The weighting coefficients ($\omega$) were area based and were calculated from equation (3.12).

$$\omega_i = \frac{A_i}{A_{TC1}}$$

(3.12)

Here $\omega_i$ is the weighting coefficient for the thermocouple, $A_i$ is the area associated with the thermocouple and $A_{TC1}$ is the area associated with TC 1. The weighting coefficients were scaled based on $A_{TC1}$ which meant that they had a minimum value of 1 as all other thermocouples had greater associated coverage areas than TC 1. This can be seen from the distribution of thermocouples on the surface in Figure 3.20. These weighting coefficients thus allowed a more accurate average over the surface to be found than would be attained with a simple arithmetic average. The weighted average was found with equation (3.13).

$$T_{\text{cond}} = \frac{\sum_{i=1}^{N} \omega_i T_{C_i}}{\sum_{i=1}^{N} \omega_i}$$

(3.13)

The differences between the methods for obtaining hot and cold side surface temperatures of the heat spreaders, and the reasons for these differences, should be clear from the preceding discussions. The methods for obtaining the heat transfer rate have also been discussed. These methods were the same for all heat spreaders tested in this work.
3.1.3.3 Solid copper heat spreader

Copper is commonly used as a heat spreader since it has very high thermal conductivity. Thus a copper heat spreader was manufactured in order to compare the performance of the VC against it. This copper heat spreader was also used to help validate the numerical model and methods introduced in this work. These aspects are later discussed in section 4.1 and section 5.1. The copper spreader was made from C10200 alloy which had thermal conductivity of 390 W/mK according to manufacturer specifications. It was machined from a copper sheet of 3.2 mm thickness down to its final thickness of 3.0mm to ensure flatness of the surfaces, particularly for the heat input surface which was in contact with the heat source. It had the same external dimensions as the VC.

3.1.4 Experimental procedure

The assembly of the experimental setup has been highlighted in section 3.1.3. After assembly, the experiments were ready to be run. The experimental procedure was as follows for both heat spreaders:

1. Turn on the fan and heater power supply units to achieve the required air flow and heat transfer rates as listed in Table 4.1.
2. Wait until steady state had been achieved. Steady state was deemed to have been achieved when the temperature of the heat source no longer changed more than 0.2°C over a 10 minute period.
3. Once steady state was achieved, record temperature data with the data logger (Yokogawa MV200). Steady state temperature data over at least a 10 minute time period was recorded.
4. During steady state, record air velocity measurements as per method described previously in section 3.1.3.
5. Repeat with different heat inputs or heat source sizes as required for the heat spreaders as listed in Table 4.1.

If the heat source size or heat spreader had to be changed, then full re-assembly of the experimental setup was required as highlighted in section 3.1.3.
3.1.5 Uncertainty analysis

Uncertainty analysis has been carried out according to the Taylor series method for error propagation as highlighted in Ref. [132]. According to this, the combined standard uncertainty in the determined result \( u_r \) can be expressed by equation (3.14).

\[
\begin{align*}
    u_r^2 &= \left( \frac{\partial r}{\partial X_1} \right)^2 u_{X_1}^2 + \left( \frac{\partial r}{\partial X_2} \right)^2 u_{X_2}^2 + \left( \frac{\partial r}{\partial X_3} \right)^2 u_{X_3}^2 + \cdots + \left( \frac{\partial r}{\partial X_J} \right)^2 u_{X_J}^2 \\
    u_r^2 &= \sum_{i=1}^{J} \left( \frac{\partial r}{\partial X_i} \right)^2 u_{X_i}^2
\end{align*}
\]

(3.14)

Here \( u_{X_i} \) is the combined standard uncertainty in the measured variable \( X_i \) and the partial derivatives present are referred to as the absolute sensitivity coefficients. The individual uncertainties are thus added in quadrature according to equation (3.14) to give the combined standard uncertainty. This Taylor series method for error propagation was used on the experimental data measured for temperature and air velocity to estimate the uncertainty in calculated values such as mass flow rate, heat input and spreading resistance. The uncertainty of the measurement devices used in these experiments was 0.3°C for the thermocouples and 0.3m/s for the hot wire anemometer. These calculations have not been shown here for brevity however Table 3.1 has been included to summarise the experimental uncertainty in the main calculated results from this work for each heat input.

| Table 3.1 – The relative uncertainty in the experimentally calculated values at each tested heat input. |
|---------------------------------------------------|---|---|---|---|
| Experimentally calculated value | 10W | 20W | 30W | 40W |
| Mass Flow Rate | ±1.8 | ±1.8 | ±1.8 | ±1.8 |
| Heat Input | ±16.9 | ±8.0 | ±5.4 | ±4.3 |
| Spreading Resistance | ±20.2 | ±12.5 | ±9.4 | ±7.1 |

It can be seen from Table 3.1 that there was particularly large experimental uncertainty for the 10W test. This was mainly due to the small temperature difference measured across the air stream of the test section. As the heat input increased, this temperature difference increased as well and the associated experimental uncertainty consequently reduced.

The preceding uncertainty analysis considered the systematic uncertainty from the measurement devices. Another component of uncertainty, the random uncertainty, has not been considered to this point. As is later discussed in Chapter 4, each test was run a minimum of 3 times to be confident in the data. The variations (standard deviation) between the tests
run at the same condition were many times smaller than the variations owing to systematic uncertainties. It can thus be said that the total uncertainty in these experiments was dominated by the systematic uncertainty component, and the random uncertainty component could be neglected. Hence only the systematic uncertainty was used to find the total uncertainty in this work (this uncertainty can be seen in Figure 4.1 and Figure 4.2).

3.2 Numerical

A numerical model of heat spreaders was introduced in order to address the aims of this research. The model introduced was conduction based which meant it relied solely on conduction mechanisms to transfer heat within the spreader, and neglected other mechanisms such as convection and radiation. The treatment of convection mechanisms has been discussed in section 1.2 and in further detail later in this work. Radiation mechanisms in particular are very minor for the majority of phase change devices which operate at low to moderate temperatures and thus have been neglected entirely here.

Such a model required numerical solution of the energy equation as shown in equation (3.15) which only considers conduction terms and neglects convection and radiation terms.

\[
\frac{\partial}{\partial x} \left( k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial T}{\partial z} \right) + q' = \rho c_p \frac{\partial T}{\partial t} \tag{3.15}
\]

Here \( T \) is the temperature, \( k \) is the thermal conductivity, \( q' \) is the volumetric heat generation rate, \( \rho \) is the density, \( c_p \) is the specific heat, \( t \) is time and \( x, y, z \) are the Cartesian coordinates in each direction. This general conduction equation can be simplified in this work based on further assumptions:

- Only steady state is considered, thus the time derivative is neglected
- No generation terms are present, thus \( q' \) is neglected

Doing so will yield the energy equation which needed to be solved within this work, as shown in equation (3.16). 

\[
\frac{\partial}{\partial x} \left( k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial T}{\partial z} \right) = 0 \tag{3.16}
\]

In order to solve for the unknown temperatures in the model, all regions required thermal conductivity to be known. This was trivial for the copper heat spreader since the thermal conductivity was known prior, but a more involved process was required for the VC model since there were many regions within this model. These will now be discussed in detail.
3.2.1 Copper model

The copper heat spreader only had a single region to consider without any internal interfaces. The thermal conductivity of the copper was known from manufacturer specifications to be 390 W/mK. This was applied throughout the entire region. The boundary conditions were equivalent to the VC numerical model and have been shown in Figure 3.21.

![Boundary conditions](image)

Figure 3.21 – Numerical model of the copper heat spreader in the plane of symmetry with boundary conditions.

The dependence of thermal conductivity on temperature for copper has been well explored [133-135]. In the range of temperatures observed here it was expected that thermal conductivity would vary by approx. 1% from minimum to maximum observed temperatures. This is an insignificant variation and thus a single value of 390 W/mK was used.

3.2.2 Vapour chamber model

The VC model was a conduction based replication of the physical VC which had been developed prior in this work (see section 3.1.2). This VC consisted of several regions with differing thermal properties. These were the bottom plate, inner evaporator wick, outer evaporator wick, vapour space and top plate. An equivalent 3D model was developed and is shown in Figure 3.22 (without relevant boundary conditions). It can be seen that each region present in the physical VC was given a region in the computational domain. As the VC was of rectangular shape, a Cartesian coordinate system was used with the x direction representing the length, y direction representing the width and z direction representing the thickness.
A symmetric model was developed in Figure 3.22 since there was a plane of symmetry through the centre of the VC (in the $xz$ plane). This symmetry can also be seen in section 3.1.2. There were no further planes of symmetry to exploit since the heat source was non-centrally located in the $x$ direction. A cross section through this plane of symmetry (similar to Figure 3.21) has been shown in Figure 3.23 with relevant boundary condition.

**Figure 3.22 – Equivalent 3D model of the physical VC showing the distinct regions (each with a different colour) present in the model.**

Another cross section in the $xy$ plane has been shown in Figure 3.24 with relevant boundary conditions. The height in the $z$-direction of this plane was taken such that the plane lay in the evaporator wick region. This has been included to clearly show the arrangement of the inner sintered powder wick.

**Figure 3.23 – Numerical model of the VC in the plane of symmetry with boundary conditions.**
As can be seen, the numerical model was a representation of the physical VC studied in this work. The numerical boundary conditions have been discussed in section 3.2.3.

Some additional simplifications have been made regarding geometric features of the VC. These include neglecting the posts present within the VC and neglecting the side walls of the VC. The heat transferred through these would be quite small in comparison to the heat transferred by the working fluid [105, 106] so it was deemed a reasonable assumption given that it greatly simplified the numerical modelling. In the condenser, the working fluid layer thickness was neglected. It was expected that the columns would help to draw the working fluid back to the evaporator side and away from the condenser region thus keeping this condensing layer thickness to a minimum.

The interior regions of the model as presented in Figure 3.22 were the bottom plate, inner evaporator wick (sintered powder wick), outer evaporator wick (screen mesh wick), vapour region and the top plate. It was in these interior regions that conduction was the only
mechanism present through which heat could be transferred and thus equation (3.15) was valid for these regions. The thermal conductivity of each region was required to solve the equation for temperature. As has been discussed in section 2.4, in the absence of thermal conductivity, the concept of effective of effective thermal conductivity can be used. Table 3.2 lists the values used for the effective thermal conductivity of each region in the model. These will now be discussed. The effective thermal conductivity of the vapour region was the subject of this research and it has been discussed in more detail in section 3.2.2.3 and in section 5.2.

Table 3.2 – Thermal conductivity or effective thermal conductivity values used in the VC numerical model for each region.

<table>
<thead>
<tr>
<th>Region</th>
<th>Thermal conductivity (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper (container)</td>
<td>390</td>
</tr>
<tr>
<td>Sintered Wick (inner evaporator)</td>
<td>32</td>
</tr>
<tr>
<td>Mesh Wick (outer evaporator)</td>
<td>1.5</td>
</tr>
<tr>
<td>Vapour</td>
<td>Determined in this work</td>
</tr>
</tbody>
</table>

3.2.2.1 Container

The bottom and top plates of the VC numerical model were the simplest regions since they were made from copper. The thermal conductivity of the copper was known from manufacturer specifications to be 390 W/mK. Hence an effective thermal conductivity was not required. This value was relatively temperature independent as discussed in section 3.2.1 since the copper container was made from the same alloy as used for the copper heat spreader.

3.2.2.2 Wicks

There were multiple wick regions present in the VC model. They were for the inner evaporator wick and outer evaporator wick. The wick regions are technically regions with both conduction and convection mechanisms present. There is conduction through the wick material itself (copper in this case), conduction through the working fluid which fills the pores of the wick (pure water in this case), convection associated with the bulk motion of the working fluid, and in some cases radiation between phases need be considered as well. This creates some complex interactions and mechanisms for heat transfer in wicks as was briefly discussed in section 2.4. Some of these aspects do warrant further discussion.
Conduction mechanisms are generally the dominant mechanism within the wick structure. Although there is some fluid motion within this layer due to the circulation of working fluid, convection effects within the wick are normally neglected as the liquid velocity within the wick is very small. From some of the sources in the literature review, the liquid velocity in the wick tends to be in the order of 0.5mm/s. Thus due to the very low liquid velocity observed within the wick region during VC operation, convective mechanisms are neglected. Furthermore it is common to neglect thermal radiation effects since the temperature of the wicking structure is quite low during normal VC operation. Thus conduction is considered the dominant heat transfer mechanism.

Even with these simplifications, conduction mechanisms throughout the wick structure are quite complex and pose many problems. At their simplest, they can be represented by two basic heat transfer scenarios. These are the series and parallel cases, and have been shown in Figure 3.25.

![Parallel and Series Network Structures](image)

*Figure 3.25 – The parallel and series network structures represent the maximum and minimum limits, respectively, for effective thermal conductivity of wick structures. Relative sizes have been chosen arbitrarily.*

Some simple analysis of the parallel and series cases leads to the development of equation (3.17) and equation (3.18), respectively, for the effective thermal conductivity of the medium. If it is assumed that one component is fluid (stagnant) and the other component is a solid, then the following equations result.

\[
k_{\text{eff}} = \varepsilon k_l + (1 - \varepsilon) k_s \quad \text{Parallel (upper limit)} \tag{3.17}
\]

\[
k_{\text{eff}} = \left[ \frac{\varepsilon}{k_l} + \frac{(1 - \varepsilon)}{k_s} \right]^{-1} \quad \text{Series (lower limit)} \tag{3.18}
\]
Here $k_{\text{eff}}$ is the effective thermal conductivity, $k_l$ is the thermal conductivity of the fluid (generally liquid for phase change devices) and $k_s$ is the thermal conductivity of the solid. The porosity is denoted by $\varepsilon$. For most wick structures in phase change devices, the porosity is based upon the volume occupied by the pores (which are fluid filled) relative to the total volume of the wick. This leads to the definition of porosity described with equation (3.19):  

$$\varepsilon = \frac{V_l}{V_l + V_s}$$  (3.19)  

Here $V_l$ is the pore (or fluid) volume and $V_s$ is the solid volume. The denominator ($V_l + V_s$) is also equal to the total volume of the wick. The porosity can be estimated through a variety of techniques, some requiring experimental measurement [136] and others relying on analytical treatment [137]. The parallel (3.17) and series (3.18) cases have been plotted in Figure 3.26. Here it has been assumed that the thermal conductivity of the solid (copper) was 400 W/mK and the liquid (water) was 0.6 W/mK.
The effective thermal conductivity shows large variations depending on the case (series or parallel) being considered. The superiority of the parallel case over the series case can be attributed to the heat flow path within the medium. For the series case, all heat must flow through the low conductivity component whereas for the parallel case the majority of heat can pass through the high conductivity pathway and bypass the low conductivity component. The parallel and series cases are also known to represent the upper and lower bounds for effective thermal conductivity of wick like structures, respectively. Although they do provide some limits on these values, these relations are not overly useful since wick structures are hardly designed like in Figure 3.25.

A more useful model was proposed by Maxwell [138] many years ago. Maxwell considered the case of a particle embedded within another continuous medium. As more particles were added, the contributions from each particle did not influence one another and thus were well dispersed. Systems of this type became known as the Maxwell model. Two cases could be considered with these Maxwell models; in one case, the ‘particles’ were considered to be fluids embedded within continuous solids, and in the other case the ‘particles’ were considered to be solids embedded within continuous fluids. These are shown in Figure 3.27.

Figure 3.27 – The Maxwell type models developed consider well dispersed particles of a particular phase embedded in a continuous medium of another phase.

The effective thermal conductivity for each of these cases is shown with equation (3.20) and equation (3.21). Note that many forms of these equations exist in the literature. These are the upper and lower Maxwell limits, respectively. They have also been included in Figure 3.26 where their behaviours can be clearly seen.
\[ k_{\text{eff}} = k_t \frac{2(k_s/k_l)^2 (1 - \varepsilon) + (1 + 2\varepsilon)(k_s/k_l)}{(2 + \varepsilon)(k_s/k_l) + 1 - \varepsilon} \]

Maxwell (upper)
Dispersed fluid in continuous solid

\[ k_{\text{eff}} = k_t \frac{2\varepsilon + (k_s/k_l)(3 - 2\varepsilon)}{3 - \varepsilon + (k_s/k_l)\varepsilon} \]

Maxwell (lower)
Dispersed solid in continuous fluid

In the upper Maxwell case, the dispersed phase has lower thermal conductivity than the continuous phase i.e. the disperse phase is fluid and the continuous phase is solid. Thus heat will tend to conduct through the continuous phase and avoid the disperse phase which creates more optimal heat transfer paths. This results in high effective thermal conductivity seen for the upper Maxwell case. In the lower Maxwell case, the dispersed phase has higher thermal conductivity than the continuous phase i.e. the dispersed phase is solid and the continuous phase is fluid. Although heat would prefer to conduct through the disperse phase due to its high thermal conductivity, it must conduct through the low conductivity continuous phase in between these disperse phases. This creates less than optimal heat transfer paths which result in low effective thermal conductivity of this structure. The Maxwell model is more practical than the series and parallel cases however there is still significant deviation between wick structures for phase change devices and the simplifications present in the Maxwell models.

Ideally, both phases are continuous in wicks to allow fluid motion within the pores and to provide mechanical strength to the structure. This invalidates the assumption that the disperse phase is sufficiently far apart that there are no interactions between this phase.

Other models to predict the effective thermal conductivity have also been introduced. Among the more common ones used for phase change devices are the models proposed by Bruggeman and Landauer (based on effective medium theory) [139], the model of Dul’nev (based on cubic unit cells) [140] and the model of Alexander (based on empirical data) [141]. These have been shown respectively with equations (3.22), (3.23) and (3.24). These three relations have also been included in Figure 3.26 to show their behaviour. They lie between the Maxwell limits.

\[ k_{\text{eff}} = \frac{1}{r} \left[ k_t(3\varepsilon - 1) + k_s[3(1 - \varepsilon) - 1] + \sqrt{[k_t(3\varepsilon - 1) + k_s[3(1 - \varepsilon) - 1]]^2 + 8k_s k_t} \right] \]

Based on effective medium theory
\[ k_{\text{eff}} = s^2 + \frac{k_l}{k_s} (1 - s)^2 + \frac{2(k_l/k_s)s(1-s)}{1-s(1-(k_l/k_s))} \] 

Based on cubic unit cells

with \[ s = \frac{y/(0.5-y)}{1+y/(0.5-y)} \] (3.23)

and \[ 4y^2 - 3y^2 + \frac{1-e}{4} = 0 \]

\[ k_{\text{eff}} = k_l \left(\frac{k_s}{k_l}\right)^{(1-e)^{0.33}} \] Based on empirical data (3.24)

The study of effective thermal conductivity of these types of materials in the literature is comprehensive and far too expansive to be covered in this work. Many more models have been proposed in the literature. These generally include the effects of, but not limited to:

- Contact resistances between the phases
- Distribution of the phases
- Geometry of the phases

The relations proposed can become very complex in nature and require in depth knowledge of the wick properties. Please refer to Refs. [142-151] for more in depth reviews and coverage of this topic.

Even with such breadth in the literature, there is still disagreement on the appropriate models to use for wicking structures present in phase change devices. This can be mostly attributed to the fact that both phases are continuous, there exists contact resistances between phases and that the geometry is often very complex. Many authors rely on different relations which yield different results for the same wick structures.

Thus it was decided that experimental measurements for the effective thermal conductivity of the wick structure would be relied upon in this work. There is also a range of literature on this topic. The values obtained are strongly dependent on the type and properties of the wick structure. In the current VC the wicks were of two types; a sintered copper powder wick and copper screen mesh wick. A value of 32 W/mK was used for the sintered powder wick in this work based on the experimental measurement in Ref. [136]. Additional literature has been shown to support this value for comparable wicks [78, 111]. Measurements from Ref. [16, 137] suggested that the effective thermal conductivity of copper wire screen mesh with similar properties was approx. 1.5 W/mK. Please refer to section 5.3 for further discussions on these.
3.2.2.3 Vapour

The vapour region was the region of interest in this work as discussed in section 1.2 and thus was not known prior. The method for determining this and further discussions regarding the vapour region are presented in section 5.2. For some clarification, a deeper look into the significance of the value will now be provided.

The term ‘effective thermal conductivity’ implies that conduction is the mechanism through which heat is being transferred – this was the simplification present in the numerical model. The actual operation of a VC is not as straightforward as this as it involves not only conduction mechanisms, but also convective mechanisms. This is present in part through the phase change interface for evaporation, the flow of vapour throughout the vapour region and the phase change interface for condensation. This has been represented for the region around the heat source in Figure 3.28.

![Figure 3.28 – A representation of the actual heat and mass transfer processes around the heat source region of the VC.](image)

These processes are not considered separately, nor are their details described within the numerical model due to the simplifications presented by the model. The processes are instead lumped together into the value for effective thermal conductivity of the vapour region. This has been shown in Figure 3.29.
Figure 3.29 – The simplifications present in the numerical conduction based model regarding the convection processes around the heat source region. Notice that in this, no fluid flow aspects are considered, only heat conduction mechanisms through the wick and vapour region. Furthermore, there is no phase change interface.

3.2.2.4 Interfaces
There were several internal interfaces present in the model. These were located where different regions met and thus were present each time there was a material change. This can be seen in Figure 3.23. These interfaces were treated such that flux was maintained across the interface and no additional thermal resistances were imposed. At the interface between the bottom plate and bottom wicks (sintered powder and screen mesh) there was an integral connection formed during the sintering procedure as highlighted in section 3.1.2. A similar integral connection was formed between the inner (sintered powder) and outer (screen mesh) bottom wick. Sintered connections have very good thermal contact so the thermal resistance at these interfaces was neglected. Although there were phase change interfaces present for evaporation and condensation in the physical VC studied in this work, these were not considered in the numerical model. This was due to the simplifications present in the numerical model, which relied solely on conduction based mechanisms.

3.2.3 Boundary conditions
The boundary conditions (for example in Figure 3.21 and Figure 3.23) were required for a computational solution to be found. These boundary conditions were equivalent for both heat spreader models. There were four boundary conditions present; adiabatic, heat flux,
convective and symmetric boundary conditions. These were meant to represent the conditions present in the experimental setup, which have been discussed previously in section 3.1.3.

### 3.2.3.1 Adiabatic

The adiabatic boundary condition implies that no heat transfer can occur across the boundary. This is evident from the mathematical formulation as shown in equation (3.25). This equation states that the heat flux (i.e. the left hand side of the equation) is equal to zero (i.e. the right hand side of the equation). This implies that heat transferred across the boundary will also be equal to zero since there is no flux passing through the boundary.

\[-k \frac{\partial T}{\partial x} = 0\] (3.25)

Note that in (3.25) the temperature gradient in the $x$-direction was found. This implied that the boundary under consideration was perpendicular to the $x$ axis. As there were multiple adiabatic boundaries in the model, this direction changed depending on the boundary and its location. This can be seen in Figure 3.23 which also has adiabatic boundaries perpendicular to the $z$ axis; in this instance the temperature gradient in the $z$-direction is equal to zero.

In the experimental setup, these adiabatic boundaries were well insulated by calcium silicate which has very low thermal conductivity (less than 0.08 W/mK). Thus the assumption of adiabatic boundaries was made given the conditions of the experiment.

The mathematical formulation of the symmetric boundary condition was equivalent to the adiabatic boundary condition since no flux can pass through a symmetric boundary. Thus discussions for the symmetric boundary condition have been omitted from here.

### 3.2.3.2 Heat flux

The mathematical formulation of the heat flux boundary condition is shown in equation (3.26). This shows that the heat flux being conducted through the model at the boundary (i.e. the left hand side of the equation) is equal to the heat flux being transferred through the boundary (i.e. the right hand side of the equation).

\[-k \frac{\partial T}{\partial z} = q\] (3.26)

Here $q$ is the heat flux which was prescribed based on the heat flux in the experimental setup. The orientation of the heat flux boundary was always the same (although its size could vary
depending on the heat source size) and thus the temperature gradient in the \( z \)-direction was required based on Figure 3.23.

In the experimental setup, the heat flux was provided by the heat source. As has been discussed in section 3.1.3, a constant heat flux distribution was obtained at the top of the heat source based on the geometry and conditions provided. This matched the boundary condition in the numerical model which assumed a constant heat flux over the boundary.

### 3.2.3.3 Convection

The mathematical formulation of the convective boundary condition is shown in equation (3.27). This shows that the heat flux being conducted through the model at the boundary (i.e. the left hand side of the equation) is equal to the heat flux being transferred through the boundary due to convection (i.e. the right hand side of the equation).

\[
-k \frac{\partial T}{\partial z} = h(T - T_\infty)
\]  

(3.27)

Here \( h \) is the convective heat transfer coefficient and \( T_\infty \) is the free stream temperature. The convective boundary was the top boundary of the model and thus the temperature gradient in the \( z \)-direction was required based on Figure 3.23.

In the experimental setup, forced convection was provided by the wind tunnel. A fan controlled by a DC power supply would draw fresh ambient air into the system which would then pass through the test section and exchange heat with the heat spreader. This has been discussed in section 3.1.3. The value for the convective heat transfer coefficient was based on experimental measurements. An average value over the surface of 30 W/m\(^2\)K was determined based on experimental measurements from this work. See Appendix A for further details about this value and its calculation.

### 3.2.4 Solution

The numerical model was developed using Ansys software due to its wide use in heat transfer problems and robust capabilities.

A mesh independence study was performed in Ansys to find an appropriate mesh for the models. This was done by progressively decreasing the mesh size until the results for the average hot side surface and average cold side surface temperatures changed by less than 0.10%. With a further refined mesh, the changes to the calculated spreading resistance were less than 0.30%.
The mesh geometry changed slightly based on the heat source size, which varied in this work. A similar study as above was performed for each heat source size model to ensure a mesh independent solution.

For the VC heat spreader model with heat source size of 20mm x 20mm, the chosen mesh has been shown in Figure 3.30. In this particular model, there were a total of 61,500 hexahedron elements. Further studies were performed to ensure that the energy entering the model and leaving the model were consistent. It was found that the imbalance of energy was less than 1% for the model and thus good conservation was achieved. Similar procedures were followed for each of the heat source sizes to ensure good quality mesh and results were obtained in each case. Hence it can be said that the mesh type and geometry were sufficient for this thermal spreading scenario.

Figure 3.30 – The mesh geometry for the VC heat spreader model with heat source size of 20mm x 20mm consisted of hexahedra elements. Mesh geometry changed very slightly based on the size of the heat source.
For some cases, the chosen mesh and a more refined mesh were both run in order to assess the impact of meshing on the results for the effective thermal conductivity of the vapour region (later discussed in section 5.2). Using the more refined mesh, changes to the effective thermal conductivity of the vapour region were found to be less than 0.40%. This was much less than the experimental uncertainty (see section 3.1.5) and so the chosen mesh was deemed appropriate also based on this.
Chapter 4 – Experimental results and discussion

In this chapter the experimental results are discussed in detail for both heat spreaders. Using the experimental setup developed in section 3.1.3, systematic testing of a number of heat spreaders was carried out according to the aims of this research. The first heat spreader was the solid copper heat spreader as discussed in section 3.1.3.3 and the second heat spreader was the VC as discussed at length in section 3.1.2. The experimental procedure has been discussed in section 3.1.4. These experimental results also formed the basis for the numerical work of this research.

The tests were performed at a number of heat inputs and heat source sizes for each heat spreader. These have been summarised in Table 4.1 along with some additional test conditions. These conditions were the same for both heat spreaders. Each test was run a minimum of 3 times to be confident in the data and the averages of these tests have been used in the sections that follow.

The area ratio has also been defined in this work as the area of the condenser side surface (or heat sink) divided by the area of the evaporator side surface (or heat source). This is a useful way of denoting the amount of spreading for a given case and is used in the following sections.

Table 4.1 – Summary of test conditions for the copper and VC heat spreader.

<table>
<thead>
<tr>
<th>Heat input, $\dot{Q}$ (W)</th>
<th>10, 20, 30, 40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat source size, $A_{hs}$ (mm x mm)</td>
<td>20 x 20, 25 x 25, 30 x 30, 35 x 35</td>
</tr>
<tr>
<td>Area ratio, $A_{cond}/A_{evap}$</td>
<td>43, 28, 19, 14</td>
</tr>
<tr>
<td>Average air velocity, $U$ (m/s)</td>
<td>3.0</td>
</tr>
<tr>
<td>Ambient temperature ($^\circ$C)</td>
<td>20 ± 3.0</td>
</tr>
</tbody>
</table>
4.1 Spreading resistance

The definition and measurement of spreading resistance has been discussed in section 3.1.3. It has been calculated for both heat spreaders for all test cases and is shown in Figure 4.1 and Figure 4.2. The associated experimental uncertainty as discussed in section 3.1.5 has been shown in these figures by a numeric value just offset from the calculated data point.

4.1.1 Copper heat spreader

The spreading resistance of the copper heat spreader is shown in Figure 4.1. As can be seen, the measured spreading resistance was independent of heat input, however was strongly dependent on the heat source area. A reduction in the heat source area led to increased spreading resistance. This result could be expected based on theoretical analysis of thermal spreading in solid bodies by conduction, such as the analysis provided in Refs. [20-24]. In these works, the spreading resistance is found to be a strong function of the heat source size (usually expressed as the ratio of heat sink size to heat source size) as was observed here.

![Figure 4.1](image)

*Figure 4.1 – The experimentally measured spreading resistance of the copper heat spreader for all heat source sizes at all heat inputs. The values listed offset to the data points are the absolute uncertainty associated with that data point as discussed in section 3.1.5.*

The changes in spreading resistance with heat source size do not vary in an easily predictable manner. This result was also expected from the analysis of Ref. [20] and the resulting analytical relations. Unfortunately, these relations cannot be used to predict the spreading resistance of the copper heat spreader since in the experiments of this research the heat source
was offset from the centre of the heat spreader as discussed in section 3.1.3. However, many of the trends are still valid as discussed later, even though the exact results are not. The work of [21] developed analytical relations for heat spreading scenarios with non-central heat sources in Cartesian systems. The resulting analytical relations are too complicated to be used in most thermal design scenarios.

Other factors have been found to influence spreading resistance, such as the convective heat transfer coefficient on the heat sink side and the thermal conductivity of the solid. These factors have some degree of temperature dependence and thus could be affected by heat input since higher heat inputs led to elevated temperatures (refer to Table 4.3). The possible changes to spreading resistance with operating temperature are discussed below. In practice, the temperature dependence of these values (thermal conductivity and heat transfer coefficient) was far too weak for it to influence the spreading resistance.

As has been discussed in section 3.2.1, the thermal conductivity of copper is mostly temperature independent, particularly in the range of temperatures experienced in this work. Minor variations to the thermal conductivity of copper would have very minor effects on the spreading resistance. According to the analysis of Song et al. [20], a change of 1% in the thermal conductivity of the material would cause a change of approx. 0.9% in the spreading resistance for a scenario similar to this work. Hence any changes to the spreading resistance owing to temperature dependent thermal conductivity of copper would be negligible, or at the very least, impossible to measure with the experimental setup.

At higher heat inputs, the increased operating temperature of the heat spreader would create some changes to fluid properties of the air flowing over the heat sink surface. These fluid property changes would result in some change to the average heat transfer coefficient over the surface. The relations provided by Song et al. [20] show that spreading resistance is a very weak function of heat transfer coefficient. For appreciable changes to the spreading resistance, it is required that the heat transfer coefficient change by orders of magnitude. Considering the heat spreader to be a flat plate, some simple analysis was carried out according to Ref. [125] to explore the changes in convective heat transfer coefficient with fluid temperature. It was observed that the heat transfer coefficient changed by less than 10% between the extremes of this work. This was far below the orders of magnitude required for an appreciable change in the spreading resistance. Hence the effect of heat transfer coefficient on spreading resistance is very minor and would be barely affected by operating temperature.
4.1.2 Vapour chamber heat spreader

The spreading resistance of the VC heat spreader is shown in Figure 4.2. As can be seen, the measured spreading resistance was dependent on both heat input and heat source size. Similar to the copper heat spreader, a reduction in the heat source size led to increased spreading resistance. The dependency on heat input was a bit misleading at first glance. In fact, after the initial 10W tests, the performance of the VC was nearly independent of heat input. These results are later discussed in more detail.

Figure 4.2 – The experimentally measured spreading resistance of the VC heat spreader for all heat source sizes at all heat inputs. The values listed offset to the data points are the absolute uncertainty associated with that data point as discussed in section 3.1.5.

Neglecting the performance at 10W for the VC (the reason for this is later discussed in section 4.1.3), Table 4.2 has been included to summarise the experimental results of these heat spreaders. The values listed in this table were the average of the values at each heat input for each heat spreader. This was deemed acceptable as the values were close to constant at all heat inputs (except for the 10W test for the VC). It can be seen that in all cases, the VC outperforms the copper heat spreader. The spreading resistance of the copper heat spreader was at least 65% larger than the VC, and it was nearly twice as large for the largest heat source size of 35mm x 35mm. Generally, increasing area ratios led to decreasing spreading resistance ratios. This meant that there was less improvement of the VC heat spreader over the copper heat spreader when area ratio was large (heat source size was small), although it was still beneficial to use as the ratio was greater than 1.
Table 4.2 – Comparison of the experimentally determined spreading resistance of the heat spreaders at different heat source sizes.

<table>
<thead>
<tr>
<th>Heat source size</th>
<th>Area Ratio $A_h/A_{cond}/A_{evap}$</th>
<th>Spreading resistance (Cu) $R_{sp}$ (K/W)</th>
<th>Spreading resistance (VC) $R_{sp}$ (K/W)</th>
<th>Spreading resistance ratio $(Cu/VC)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>35 x 35</td>
<td>14</td>
<td>0.1342</td>
<td>0.0675</td>
<td>1.99</td>
</tr>
<tr>
<td>30 x 30</td>
<td>19</td>
<td>0.1561</td>
<td>0.0796</td>
<td>1.96</td>
</tr>
<tr>
<td>25 x 25</td>
<td>28</td>
<td>0.1823</td>
<td>0.0985</td>
<td>1.85</td>
</tr>
<tr>
<td>20 x 20</td>
<td>43</td>
<td>0.2163</td>
<td>0.1312</td>
<td>1.65</td>
</tr>
</tbody>
</table>

Note: For the VC, the 10W data point has been neglected.

The copper heat spreader relied on conduction mechanisms to transfer heat throughout the spreader, and although the thermal conductivity of copper is excellent (390 W/mK), the unfavourable spreading scenario led to poor performance. The VC doesn’t rely solely on conduction mechanisms, although these are still present within regions of the VC. The reason for improved VC performance over copper is due to the two-phase heat transfer mechanisms within the VC. As has been mentioned in the literature review and discussed in detail in section 5.2, two-phase heat transfer is an excellent way of transferring heat with minimal temperature drop. This led to the improved spreading performance of the VC.

From the results in Figure 4.2, the capillary limit had not yet been reached in the VC at 40W as the spreading resistance had not started to increase. It was expected that larger heat loads could have been sustained without dry-out being reached, however they were not trialled due to excessive temperature of the VC already present at 40W. Due to the convection conditions present on the heat sink, the temperatures of the heat spreader could reach over 110°C at heat inputs greater than 40W. This led to some issues with the VC, the main being that the flatness could not be maintained due to thermal expansion and the method in which the VC was held in the experimental setup. Had extended surfaces (such as a CPU heat sink) been used on the cold side of the VC, then larger heat loads could have been sustained. The main reason this wasn’t done was that it would interfere with the thermocouple attachment method on the cold side of the heat spreader (section 3.1.3.2). Hence testing was not performed above 40W for reasons regarding the thermal expansion of the VC, not due to any capillary limitation.

The spreading resistance of the VC developed in this work can also be compared to the results in the literature. It must be noted that direct comparisons of spreading resistance are difficult to make since measurements and definitions between authors tend to vary as discussed in section 2.1. The values included for spreading resistance in Table 2.1 somewhat
match the definition in equation (3.6), but for further clarity the original work should be referred to. Additionally, it must be considered that most experimental measurements are for centrally located heat sources, unlike the non-central heat source of this work. The device thickness must also be considered in any assessment. These aspects further complicate direct comparisons. Nevertheless, it can be seen that the performance of the VC in this work compares very well to other VCs in the literature. At higher area ratios it can be seen to have smaller spreading resistance than most VCs with comparable thickness in the literature which highlights the strength of this design.

4.1.3 Start-up difficulties

It can be seen that for all heat source sizes, the spreading resistance of the VC at 10W was much greater than for other heat inputs. Although there was particularly large experimental uncertainty for the 10W test (±20%), even with consideration of these uncertainties, the spreading resistance was still larger for the 10W test than for all other heat inputs for each heat source size. Hence there was in fact poor thermal performance experienced at 10W and this was not just due to experimental uncertainty. Note that even with this poor thermal performance of the VC at 10W, the spreading resistance was still less than that obtained with the copper heat spreader.

Poor performance at low heat inputs has often been encountered in phase change devices such as heat pipes [152], loop heat pipes [153, 154] and pulsating heat pipes [155]. They will now be explored in some detail for the VC under consideration. This also provides some interesting insights and practicalities regarding the operation and performance of VCs. This helps to better understand the underlying heat transfer mechanisms of the VC.

It was observed in all tests that 10W heat input led to the lowest operating temperatures of the heat spreader. As the heat input increased, so too did the operating temperature. The hot side surface temperature of the heat spreaders as a function of heat input for two heat source sizes has been included in Table 4.3. These hot side surface temperatures give an indication of the operating temperature of the VC. The actual temperature within the VC varies spatially throughout the device and so the temperatures listed in Table 4.3 are just an approximation for the operating temperature. The working fluid temperature would be slightly lower than the hot side surface temperature due to resistive components which heat must flow through associated with the VC container and wick regions.
### Table 4.3 – The hot side surface temperatures of the heat spreaders for the 20mm x 20mm and 35mm x 35mm heat source sizes. These temperatures also give approximations for the heat spreader operating temperatures.

<table>
<thead>
<tr>
<th>Heat Source Size</th>
<th>20mm x 20mm</th>
<th>35mm x 35mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat (W)</td>
<td>VC</td>
<td>Cu</td>
</tr>
<tr>
<td>10</td>
<td>49</td>
<td>50</td>
</tr>
<tr>
<td>20</td>
<td>68</td>
<td>69</td>
</tr>
<tr>
<td>30</td>
<td>87</td>
<td>90</td>
</tr>
<tr>
<td>40</td>
<td>107</td>
<td>111</td>
</tr>
</tbody>
</table>

Note: The averages of multiple readings have been used for each temperature. VC denotes the vapour chamber and Cu denotes the copper heat spreader.

As discussed previously in section 4.1.1, operating temperature had little effect on the performance of the copper heat spreader since its performance was mainly linked to the thermal conductivity of the copper. This had insignificant variation with temperature in the range observed in these experiments. Unlike copper, the performance of the VC heat spreader was dependent on operating temperature, mainly due to the effect of temperature on phase diagrams, working fluid properties and phase change mechanisms. These will now be discussed. In the following analysis, the vapour was assumed to remain in the saturated state and thus all properties were evaluated at saturated conditions. Relevant thermodynamic properties of pure water have been included in Table B.1 of Appendix B.

#### 4.1.3.1 Phase diagram

The phase diagram for water shows that lower operating temperatures are less favourable for VC operation. It can be seen from the slope of the phase diagram that higher temperatures are more favourable as the slope $dP/dT$ is very large. This means that a change in pressure ($dP$) is accompanied by a very small change in temperature ($dT$) since the gradient ($dP/dT$) must be large. At lower operating temperatures, the slope $dP/dT$ is smaller. Hence larger temperature changes ($dT$) can result from pressure changes ($dP$). For a given vapour pressure loss ($dP$), the resulting vapour temperature drop ($dT$) will be small when the operating temperature is high due to these gradients. Consequently, the vapour temperature drop ($dT$) will be large when the operating temperature is low. Approximate markers and gradients showing this effect have been included in Figure 4.3 which shows the phase diagram for pure water. Hence the low operating temperature at 10W would lead to poor vapour performance due to the location on the phase diagram at which the VC operates. This effect is minimised at higher temperatures which resulted from higher heat inputs.
In the previous discussion, there was no mention of the variation in vapour pressure losses with heat input; it was simply assumed that they were equal for all heat inputs. In fact, the vapour pressure losses are linked to the operating conditions of the VC as well. Analysis of vapour pressure losses can become quite challenging and often results in complex relations. For this qualitative analysis, the functional form for a heat pipe was assumed since it was only the trends that are of interest, not the exact value of vapour pressure loss. Even for a heat pipe, vapour pressure losses can become quite complex so a number of assumptions were made; only the adiabatic section was considered and vapour flow was incompressible, laminar and fully developed. Equation (4.1) results for the vapour pressure losses ($\Delta P_v$) for this tubular form. Derivation of this equation can be seen in Ref. [125].

$$\Delta P_v = \frac{8\mu_v\dot{m}L_a}{\pi \rho_v r_v^4}$$  \hspace{1cm} (4.1)

Here $\mu_v$ is the vapour dynamic viscosity, $\dot{m}$ is the working fluid mass flow rate, $\rho_v$ is the vapour density, $L_a$ is the adiabatic section length and $r_v$ is the radius of the vapour region. The relevant terms to consider here relate to the working fluid properties (vapour dynamic viscosity and vapour density) or the operating conditions (working fluid mass flow rate). The geometric terms (adiabatic section length and radius of the vapour region) are not required.
In order to explore the effect of operating conditions on the thermodynamic properties of the working fluid, one can consider the changes occurring between the 10W and 20W tests.

Regarding the mass flow rate of working fluid, this can be simply approximated for phase change devices with equation (4.2). This shows that the mass flow rate is equal to the heat input ($\dot{Q}$) divided by the latent heat of evaporation ($h_{fg}$) of the working fluid. Lower heat inputs lead to reductions in the mass flow rate which would clearly reduce the vapour pressure losses in equation (4.1). Different heat inputs also lead to different operating temperatures of the VC and thus temperature dependent working fluid properties must be considered. One might assume that the mass flow rate would double from 10W to 20W based on initial observation of equation (4.2), however the temperature dependence of latent heat of evaporation must first be considered.

$$\dot{m} = \frac{\dot{Q}}{h_{fg}}$$  \hspace{1cm} (4.2)

At higher temperatures the latent heat of evaporation for water tends to decrease. Thus at the higher heat input of 20W, due to the larger operating temperature, the mass flow rate will be further increased by the reduction in latent heat of evaporation. While the heat input doubled from 10W to 20W, the latent heat of evaporation decreased by about 2% due to temperature dependence. The change in heat input clearly dominated the mass flow rate, with only minor contribution from the latent heat of evaporation. The net effect on the mass flow rate between 10W and 20W would be an increase by a factor slightly greater than 2. The small increment above the factor of 2 here owed to the decrease in latent heat of evaporation. This increased mass flow rate would increase the vapour pressure losses (equation (4.1)). It can also be stated that for the mass flow rate, the increased operating temperature was detrimental since it tended to amplify the mass flow rate, although the effect was quite weak.

The next property to consider is the saturated vapour density of the working fluid. Vapour pressure losses are inversely proportional to vapour density as seen in equation (4.1). For water, this property has very strong temperature dependence. Between the temperatures observed for the 10W and 20W tests, the vapour density would greatly increase by a factor of approx. 2.5. This increase in vapour density will tend to greatly decrease the vapour flow pressure losses as shown in equation (4.1).

The last property to consider for the vapour pressure losses is the vapour dynamic viscosity which is shown to be directly proportional to vapour pressure losses in equation (4.1). The
variations in this property with temperature are quite weak. Between the temperatures observed for the 10W and 20W tests, the vapour dynamic viscosity would only increase by about 6%. Although this is minor, it would tend to increase the vapour pressure losses as seen in equation (4.1).

Considering all these factors within equation (4.1), the net effect owing to increased heat input (and consequent increased operating temperature) will tend to decrease the vapour pressure losses. Although this may seem counterintuitive, the increase in vapour density (inversely proportional to vapour pressure loss) is greater than the increase to the mass flow rate and vapour dynamic viscosity (which are both proportional to vapour pressure loss). From 10W to 20W, it is expected that pressure losses would decrease by approx. 15% based on this analysis. This is a very interesting insight on the operational characteristics of VCs and it helps to explain the start-up difficulties and poor performance of the VC heat spreader observed at 10W.

Combining the above discussion on vapour pressure losses with the previous discussion on phase diagrams, it can be seen how the increased operating temperature was very favourable. Not only would vapour pressure losses decrease at higher operating temperatures, but the location on the phase diagram was also desirable. This combination meant that the overall temperature drop within the vapour phase would be kept low, which would help to minimise the spreading resistance as well.

4.1.3.3 Phase change heat transfer coefficients

Another consideration for start-up difficulties are the phase change processes of evaporation and condensation. These are two complicated processes which can be influence by a large number of factors. These aspects were not the focus of this research and there are still many gaps in the knowledge, not only just for phase change devices which have complicated internal structures involved, but for a variety of applications and fields. Hence it is difficult to make absolute conclusions in regards to this. Please refer to section 2.4 for some remarks about the evaporation and condensation processes.

Continuing on with the discussions from previous sections, it can be seen that there is also temperature dependence on the phase change processes. Kinetic theory has been used to predict the heat transfer coefficient for interfacial transport such as evaporation and condensation processes in Ref. [156]. Based on this theory and some further assumptions, equation (4.3) results for the interfacial heat transfer coefficient ($h_{int}$).
\[ h_{\text{int}} \approx \left(\frac{2\hat{\sigma}}{2 - \hat{\sigma}}\right) \left[ \frac{h_{fg}}{T_v v_{lv}} \right]^2 \left[ \frac{\bar{M}}{2\pi\bar{R}T_v} \right]^{1/2} \left[ 1 - \frac{P_v v_{lv}}{2h_{fg}} \right] \] (4.3)

Here \( \hat{\sigma} \) is the accommodation coefficient, \( T_v \) is the vapour temperature, \( v_{lv} \) is the difference between the specific volume of the vapour and liquid phase, \( \bar{M} \) is the molar mass, \( \bar{R} \) is the universal gas constant and \( P_v \) is the vapour pressure. The accommodation coefficient takes into account the proportion of molecules which have sufficient energy to undergo phase change compared to the total number of molecules incident on the phase change interface. This can be slightly simplified for this section by assuming that \( v_v \gg v_l \) and thus \( v_{lv} = v_v - v_l \approx v_v \). These assumptions are reasonable for water. Then the relation \( v = 1/\rho \) can be used to yield equation (4.4) which contains more familiar terms that have been used previously.

\[ h_{\text{int}} \approx \left(\frac{2\hat{\sigma}}{2 - \hat{\sigma}}\right) \left[ \frac{\rho_v h_{fg}}{T_v} \right]^2 \left[ \frac{\bar{M}}{2\pi\bar{R}T_v} \right]^{1/2} \left[ 1 - \frac{P_v}{2\rho_v h_{fg}} \right] \] (4.4)

This equation also has temperature dependence, not only due to temperature being present in the equation, but also due to the temperature dependent working fluid properties (vapour density, vapour pressure and latent heat of evaporation). Although not immediately clear from equation (4.4), increased operating temperature will tend to also increase the heat transfer coefficient. This has been explored in Figure 4.4. Saturated fluid properties of water were assumed in this analysis (Table B.1 of Appendix B).

The accommodation coefficients for water are typically difficult to determine and can vary largely between different authors. A range that represents typical values for evaporation of water [156, 157] has been included in Figure 4.4. This was sufficient for this qualitative analysis. The accommodation coefficients are found to have a large effect on the value for interface heat transfer coefficient. Assuming the accommodation coefficient to be 0.03, the heat transfer coefficient at 45°C is approx. 35,000 W/m²K compared to approx. 260,000 W/m²K at 105°C. Hence higher heat loads and consequently higher operating temperatures should lead to larger heat transfer coefficients of phase change processes and reduce the associated temperature drops across the phase change interfaces.

The operating pressure is thus dependent on the operating temperature (assuming saturated properties with the phase diagram). Most phase change devices with water as the working fluid operate in low to moderate temperatures and thus the operating pressures lie between vacuum and positive pressure. At low operating temperatures, a vacuum pressure may exist...
in the device and at higher operating temperatures (100°C and over) positive pressures may exist. This can be seen from the saturated fluid properties (water properties are shown in Table B.1 of Appendix B) and also from the phase diagram of Figure 4.3.

![Graph showing the effect of temperature and accommodation coefficients on the interface heat transfer coefficient predicted by kinetic theory.](image)

*Figure 4.4 – The effect of temperature and accommodation coefficients on the interface heat transfer coefficient predicted by kinetic theory.*

Note that the preceding analysis was intended to be qualitative only. The accuracy of the values predicted by equation (4.4) for wick structures in phase change devices is not guaranteed. Some values highlighted in the literature review for evaporative heat transfer coefficient show moderate agreement with the results of this kinetic model at low values (0.02) for accommodation coefficient. However this should not be taken to be general. Some of the complexities of this have been mentioned in section 2.4. Other difficulties have been explored in the literature [156-159]. Rather it is the trends, specifically the effect of operating temperature on phase change processes that are important here. It has been shown above that low operating temperatures can reduce the heat transfer coefficients associated with phase change processes, leading to poor device performance at low heat inputs. Similarly, in the previous sections it was shown that the low operating temperature would lead to larger temperature drops throughout the vapour phase due to the large vapour pressure losses and the unfavourable position on the phase diagram.
4.2 Temperature distributions

The previous section detailed the overall performance of the heat spreaders through the common metric known as spreading resistance and also delved into some of the complexities of phase change heat transfer devices. In this section, the temperature distributions within the VC will be discussed. These help to explain the performance of the VC in more detail with regards to the previous complexities and also give interesting insights into possible areas of improvement for VC development.

These temperature distributions have been defined to represent the heat transfer ability of the VC in each direction. The spreading performance was thus broken up into two components; namely the axial and lateral performance. The definitions presented in the following sections are simplifications of these aspects which were used as they provided useful insights. The axial performance was a measure of the heat transfer in the \( z \) direction and the lateral performance in the \( x/y \) directions (both directions can be seen in Figure 3.4). These can approximate the spreading performance of the VC. Only temperature differences were required for these simplified purposes.

First, some naming conventions need to be discussed. For both heat spreaders, the temperatures on the condenser surface (TC 1 – TC 8) follow the naming convention present in Figure 3.20. The average temperature of the evaporator surface \( (T_{evap}) \) has been determined with the method described in section 3.1.3.

4.2.1 Axial

Table 4.4 shows the axial heat transfer capabilities of the heat spreaders. This was measured by the temperature difference between the evaporator surface \( (T_{evap}) \) and the condenser surface directly above the evaporator (TC 1). This gives a good indication of the axial heat transfer capabilities of the heat spreaders, without considering the spreading effects.

It can be observed for the copper heat spreader that the temperature drop \( (T_{evap} – TC1) \) was much less than for the VC heat spreader in all cases. For lower heat loads at larger heat source sizes, the temperature drop for the copper heat spreader was so small that it was impractical to measure. In this case, it has been listed in Table 4.4 as less than 0.2°C as this was the maximum observed in these cases. At higher heat loads, the temperature drops increased and could be more easily measured. It was found that smaller heat source sizes led
to larger temperature differences. This was expected as the heat flux was higher as the heat source size decreased and the temperature drop would increase as a result. The two largest heat source sizes still had very small temperature differences at higher heat loads.

Table 4.4 – The axial heat transfer capabilities of the heat spreaders compared.

<table>
<thead>
<tr>
<th>Q (W)</th>
<th>20mm x 20mm</th>
<th>25mm x 25mm</th>
<th>30mm x 30mm</th>
<th>35mm x 35mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>VC</td>
<td>Cu</td>
<td>VC</td>
<td>Cu</td>
<td>VC</td>
</tr>
<tr>
<td>10</td>
<td>1.1</td>
<td>0.3</td>
<td>0.6</td>
<td>&lt;0.2</td>
</tr>
<tr>
<td>20</td>
<td>2.1</td>
<td>0.6</td>
<td>1.4</td>
<td>0.4</td>
</tr>
<tr>
<td>30</td>
<td>3.0</td>
<td>1.1</td>
<td>2.3</td>
<td>0.6</td>
</tr>
<tr>
<td>40</td>
<td>4.0</td>
<td>1.7</td>
<td>2.9</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Note: measurement locations are found in Figure 3.20, except for $T_{\text{evap}}$ (section 3.1.3). VC denotes the vapour chamber and Cu denotes the copper heat spreader.

The trends observed for the copper heat spreader were quite predictable which was to be expected since conduction was the heat transfer mechanism within this spreader. The trends for the VC heat spreader were more complicated to predict due to the complex heat transfer mechanisms occurring within the VC. It was generally seen that larger heat loads and smaller heat source sizes led to larger temperature differences as one would expect.

The axial temperature drops for the VC were generally much larger than for the copper spreader. In some cases they were as much as four times larger than the copper spreader which suggested that the copper spreader still had far superior axial thermal performance. This improved axial performance of the copper heat spreader was due to the very high thermal conductivity of the copper (390 W/mK). The conduction losses through this short path between the measurement points would be very small. The same cannot be said of the VC which was found to have quite poor axial performance. It was expected that the poor axial performance of the VC could be mainly attributed to the fluid saturated wick layer.

The wick in the VC of this work was a sintered copper powder wick. It had measured effective thermal conductivity of approx. 32 W/mK which was far less than that of the base copper material (390 W/mK) as is commonly found with wicking structures. It has been discussed in section 3.1.1 that large temperature gradients would develop in the wick for this reason, and also due to the large local heat flux. A similar result was observed in the numerical work of Koito et al. [104]. In that work, an evaporator wick with effective thermal conductivity of approx. 8 W/mK was used and large temperature gradients were observed in
the wick region above the heat source. Comparing these two, the effective thermal conductivity of the wick in this work was approx. 4 times larger than in Koito et al. and still large temperature gradients were observed. Hence this is still a weakness of the VC design in this work.

From these results and discussions it can be seen that the evaporator wick, particularly in the region above the heat source, is vital for the thermal performance of the VC. Poor VC axial performance can be linked to this region. Although the axial temperature drop within the VC is not the only aspect to consider for thermal performance (the lateral temperature drops must be considered as well, see section 4.2.2), it is still an important aspect which cannot be neglected. Hence, this has been further analysed and discussed in section 5.3.1.

4.2.2 Lateral

Table 4.5 shows the lateral heat transfer capabilities of the heat spreaders. This was measured with temperature differences using only thermocouples present on the condenser surface. It was best shown by the difference between the thermocouple directly above the heat source (TC 1) and the average of the thermocouples at a radius of 60mm from TC 1 i.e. TC 6, TC 7 and TC 8. This gave a measure of how well heat was being transferred laterally within the heat spreader and the uniformity of the condenser surface temperatures.

<table>
<thead>
<tr>
<th>Lateral temperature distributions</th>
<th>20mm x 20mm</th>
<th>25mm x 25mm</th>
<th>30mm x 30mm</th>
<th>35mm x 35mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>TC 1 – [average of TC 6, TC 7 and TC 8] (°C)</td>
<td>VC</td>
<td>Cu</td>
<td>VC</td>
<td>Cu</td>
</tr>
<tr>
<td>Q (W)</td>
<td>VC</td>
<td>Cu</td>
<td>VC</td>
<td>Cu</td>
</tr>
<tr>
<td>10</td>
<td>2.2</td>
<td>3.0</td>
<td>1.6</td>
<td>2.8</td>
</tr>
<tr>
<td>20</td>
<td>1.5</td>
<td>5.6</td>
<td>1.0</td>
<td>5.3</td>
</tr>
<tr>
<td>30</td>
<td>1.1</td>
<td>8.0</td>
<td>0.8</td>
<td>7.6</td>
</tr>
<tr>
<td>40</td>
<td>1.2</td>
<td>10.8</td>
<td>0.9</td>
<td>10.0</td>
</tr>
</tbody>
</table>

Note: measurement locations are found in Figure 3.20. VC denotes the vapour chamber and Cu denotes the copper heat spreader.

It can be seen from Table 4.5 that the lateral performance and temperature uniformity on the condenser surface of the VC was far superior to the copper heat spreader. In all cases, the lateral temperature measurements were more favourable for the VC. The temperature difference measured could be over 10 times larger for the copper heat spreader compared to the VC.
This has also been shown graphically for the 20mm x 20mm heat source case for the heat input of 30W in Figure 4.5. In this, the temperature distribution along the centreline of the heat spreaders has been shown (i.e. temperatures at TC 2, TC 1, TC 4 and TC 7 in Figure 3.20) for a particular test case. This figure also clearly shows the superior condenser temperature uniformity and lateral performance of the VC heat spreader.

![Graph showing temperature distribution along the centerline](image)

**Figure 4.5 – The surface temperature distributions along the centreline of the heat spreaders (30W and 20mm x 20mm heat source) indicated that the VC had superior lateral performance to the copper heat spreader. VC denotes the vapour chamber and Cu denotes the copper heat spreader.**

In the case of the copper heat spreader, the observations were mostly logical with larger heat inputs leading to larger temperature differences in a predictable manner. This was similar to the axial temperature differences from the previous section. Furthermore, a reduction in heat source size generally caused an increase to the measured lateral temperature difference.

For the case of the VC heat spreader, the temperature distribution on the condenser surface can be thought of as an indication for the vapour temperature distribution within the vapour region. As the vapour spreads throughout the vapour region, the temperature consequently drops depending on the flow conditions. This has been discussed in section 4.1.3.2. This can be best observed by the condenser temperature distribution since the vapour will condense on this surface. A representation of this has been included in Figure 4.6.
Hence the measurements in Table 4.5 are an indication of the vapour temperature distribution within the vapour region. It must be noted that these are not direct measurement of the vapour temperature; rather just an indication of it. Some of these measurements from the VC show interesting results. In particular, the results from the 10W test support the previous claim of start-up difficulties at low heat inputs due to low operating temperatures. The temperature differences measured at this low heat load were larger than what was measured at higher heat loads. They were comparable in size to that experienced by the copper heat spreader at the same heat load of 10W. Thus there were clear issues with the lateral performance of the VC at this heat load.

Since the measurements in Table 4.5 were an indication of the vapour temperature, this suggested that the vapour temperature drop was large for the 10W heat load. This result was predicted by the analysis presented in section 4.1.3. It was stated in this section that the low operating temperature of the VC (caused by small heat loads) was detrimental to VC performance, mainly due to the two following factors:

- This led to an unfavourable position on the phase diagram with a small value of \( \frac{dP}{dT} \) which meant that vapour temperature drops were very sensitive to vapour pressure losses.
- It also led to large vapour pressure losses due to unfavourable vapour properties, specifically the very small saturated vapour density.
These two factors led to the large temperature drops within the vapour region and thus the poor lateral performance of the VC as seen in Table 4.5 for the 10W heat load. At higher heat loads of 20W and 30W, the lateral temperature difference was seen to decrease in Table 4.5. This is consistent with previous discussions; higher operating temperatures led to favourable lateral performance due to the location on the phase diagram and vapour properties. This helped to reduce the vapour phase temperature drops. The 40W tests showed very similar results to the 30W tests. When considering the uncertainty of the thermocouples it was difficult to make absolute claims given that the measured temperature differences were so similar for the 30W and 40W tests. Another possible cause of this could have been that transition in the vapour flow regime had been reached. The higher working fluid mass flow rate associated with the 40W test would cause the vapour velocity to increase. These increased vapour velocities would eventually lead to transitional flow from laminar to turbulent within the vapour region. Pressure drops associated with turbulent flow are generally greater than laminar flow which gives another possible explanation to these results.

It is interesting to note that only the lateral results from Table 4.5 show poor performance at the low heat load of 10W. The same cannot be deduced from the 10W result of axial performance in Table 4.4. This suggested that the start-up difficulties of the VC were mostly linked to the lateral performance of the vapour i.e. vapour phase pressure and temperature drops, and not from other mechanisms.

4.3 Summary and remarks
From Table 4.4 and Table 4.5 it can be seen that the VC had worse axial performance however better lateral performance than the copper heat spreader. The overall spreading performance of the VC was superior to the copper spreader as seen in Figure 4.1 and Figure 4.2 (even for the 10W tests where the VC experienced start-up difficulties). This was due to the heat transfer problem proposed in this research involving thermal spreading. The area ratio was at least 14 in this work and thus spreading was a key factor. Even though the axial performance of the VC was found to be poor, the lateral performance of the VC was far more important for this heat transfer problem and thus the VC clearly outperformed the copper heat spreader. As discussed previously in this chapter, the copper spreader had to rely solely on diffusion mechanisms to spread heat from the source to the sink and these have been shown to be ineffective at doing so. The VC relied on two-phase heat transfer mechanisms which have been shown to be very effective in spreading scenarios such as this work.
Chapter 5 – Numerical results and discussion

In this chapter, the numerical model was validated against the experimental results of the copper heat spreader and then used to determine the effective thermal conductivity of the vapour region. Some further analysis and investigation has also been provided into this value and the factors which influence it. Lastly, the evaporator wick properties were examined in detail such that the effects of these on thermal performance of the VC could be better understood.

5.1 Model validation

The experimental results of the copper heat spreader were used to validate the conduction based numerical model developed in this work. Copper was used for validation since the thermal conductivity of the copper was known prior from manufacturer data to be 390 W/mK. Simulations of the model presented in Figure 3.21 were run with this value of thermal conductivity for each heat source size as tested in this work.

The validation procedure of the numerical model ensured that the spreading resistance obtained from both numerical and experimental methods were in agreement. If they were, then it could be safely said that the model was appropriately modelling the given thermal spreading scenario. The spreading resistance in the numerical model was obtained with equation (3.6); however the terms were obtained numerically instead of experimentally. Obtaining the surface temperatures of the spreader with Ansys software could be done with built in functionality which calculated the average temperature of the required surface. Once the spreading resistance was numerically found for the copper heat spreader, it could be compared against the experimental values for each of the heat source sizes. This has been done in Table 5.1.

Table 5.1 – Comparison of the numerical and experimental spreading resistance of the copper heat spreader for all tested heat source sizes.

<table>
<thead>
<tr>
<th>Heat source size</th>
<th>Spreading resistance (numerical)</th>
<th>Spreading resistance (experimental)</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_{hs} ) (mm x mm)</td>
<td>( R_{sp} ) (K/W)</td>
<td>( R_{sp} ) (K/W)</td>
<td></td>
</tr>
<tr>
<td>20 x 20</td>
<td>0.2125</td>
<td>0.2163</td>
<td>1.7%</td>
</tr>
<tr>
<td>25 x 25</td>
<td>0.1813</td>
<td>0.1823</td>
<td>0.1%</td>
</tr>
<tr>
<td>30 x 30</td>
<td>0.1565</td>
<td>0.1561</td>
<td>0.3%</td>
</tr>
<tr>
<td>35 x 35</td>
<td>0.1365</td>
<td>0.1342</td>
<td>1.7%</td>
</tr>
</tbody>
</table>
Note that since the copper heat spreader was found to have performance which was independent of heat input (Figure 4.1) only a single heat input was needed for the model. A single value for thermal conductivity of 390 W/mK was also used in this model, thus no changes to the numerical results would be expected at differing heat inputs regardless. The thermal conductivity of the copper was discussed in section 3.2.1. Furthermore, the heat transfer coefficient was assumed constant as well for reasons discussed in section 4.1.1.

From Table 5.1 it can be seen that the agreement between the numerically and experimentally determined spreading resistances for the copper heat spreader was excellent. At most, the difference between the two methods was 1.7%. This suggested that the numerical model developed was appropriate for modelling of thermal spreading scenarios as in this research. This also confirmed that the methods adopted (see section 3.1.3.2) for measuring the surface temperatures of the hot side and cold side of the heat spreader were appropriate. This meant that the experimentally determined resistance was a true measurement of the spreading resistance of the heat spreader and that no unwanted contributions from the experimental setup were included in this value. As discussed in section 3.1.3.1, the experimental setup in this work was unchanged for all heat spreaders and tests. Thus the conditions present for the VC were equivalent to that of the copper heat spreader. Hence it was expected that the spreading resistance of the VC could also be accurately determined through the same methods and techniques that were adopted for the copper heat spreader.

5.2 Effective thermal conductivity of the vapour region

From the discussions in this work, it can be seen that all thermal (Table 3.2) and geometric (Figure 3.23) parameters of the VC were known, except for the value of effective thermal conductivity of the vapour region. The experimental setup and numerical model were shown to have excellent agreement in the previous section, and thus the determination of experimental spreading resistance would allow for the effective thermal conductivity of vapour to be found numerically. The importance of spreading resistance and its accurate calculation is clear from this.

This was done through an iterative process for each heat source size tested in this work. The value used in the model for effective thermal conductivity of the vapour region was systematically varied until the numerically calculated spreading resistance was equal to the experimentally determined value.
The process used in determining the effective thermal conductivity of vapour has been summarised in Figure 5.1. The solid lines are the numerically determined spreading resistance plotted against the effective thermal conductivity of the vapour region. The dashed lines are the experimental results averaged over the 20W, 30W and 40W tests. The 10W test has been omitted from this since start-up difficulties were experienced as described in section 4.1.3. The remaining tests were nearly constant with heat input at a given heat source size and thus it was reasonable to use the average of these tests for the analysis. Thus the experimental point (round marker) and numerical line (solid line) were used to find the effective thermal conductivity of the vapour region by equating the spreading resistance of the two methods as can be seen in Figure 5.1.

Figure 5.1 – The iterative process used to determine the effective thermal conductivity of the vapour region. In this, ‘Exp’ denotes the experimental measurements and is shown with a round marker. ‘Num’ denotes the numerically calculated spreading resistance by varying the effective thermal conductivity of vapour.

It can be seen from Figure 5.1 that the value for effective thermal conductivity of the vapour region varied with the heat source size. All determined values were between 1,900 W/mK and 2,400 W/mK which clearly shows the excellent heat transfer capabilities of the vapour region. This value for thermal conductivity is at least 4 times larger than that of copper, and comparable to what can be obtained with some carbon compounds [75, 98, 160, 161].
At the smallest heat source size (largest area ratio) of 20mm x 20mm, a value for effective thermal conductivity of vapour of 1,900 W/mK was obtained. Increasing the heat source size to 25mm x 25mm, 30mm x 30mm and 35mm x 35mm (decreasing the area ratio) increased this value to 2,220 W/mK, 2,380 W/mK and 2,410 W/mK, respectively. These values can be practically used by thermal designers for a variety of thermal modelling scenarios involving VCs with similar features. The reduction in effective thermal conductivity values with reduced heat source sizes was also observed by Chen et al. [100], however in that work a bulk modelling conduction scheme was used, not the layered approach here. The causes of variations in the values of effective thermal conductivity of the vapour region with heat source size (or area ratio) will now be explored.

5.2.1 Variations with heat source size

The changes observed in the effective thermal conductivity of vapour with heat source size can be attributed to the processes which constitute this term, namely the phase change and vapour flow processes within the VC. These have been briefly discussed in section 3.2.2.3. A change in the effective thermal conductivity of the vapour region implied that there was a change in one of these processes. Identifying which process was the likely cause of this required that each process be analysed in some more detail. With this, the cause of the variations in the effective thermal conductivity of the vapour region with heat source size became clearer.

First looking at the phase change processes, it can be appreciated that these are quite complex as has been discussed in section 2.4. A reasonable simplification to this process is to represent it as a convective resistance (this has been explored in section 4.1.3.3) which can be expressed in the form of equation (5.1).

\[ R = \frac{1}{hA} \quad (5.1) \]

Here \( R \) is the thermal resistance of the phase change process. It can be defined for both evaporation and condensation phase change. Thus this process can be simplified by knowledge of two values, namely the heat transfer coefficient \( (h) \) of the phase change process and the area \( (A) \) associated with the phase change process. Any changes to this resistance must be due to some variations in either the area or the heat transfer coefficient associated with these aspects.
Changes to the heat source size were not expected to have had too large of an effect on the heat transfer coefficient. This value has been found to depend on the operating temperature and heat flux from the literature (see section 2.4). Although reductions in heat source sizes led to slightly larger operating temperatures (at the same heat input) as described in Table 4.3, the temperature increases were in the order of only a few degrees. The effect of temperature on the heat transfer coefficient has been explored in a simplified manner in section 4.1.3.3. When it is further considered that the performance of the VC was constant between heat loads of 20-40W for a given heat source size, this would also suggest that there was little dependence of heat transfer coefficient on heat flux as well. Thus, it was expected that the heat transfer coefficient would not vary too much as the heat source size varied.

On the other hand, changes to the heat source size were expected to have a large effect on the evaporative area for obvious reasons. As the heat source size decreased, the evaporative area would decrease too. This reduced evaporative area would lead to an increase in the evaporative resistance. This dependence of evaporative resistance on heat source area has also been confirmed by Koito et al. [105]. Changes to the heat source area would hardly affect the condensation resistance as the condenser is physically separated from the evaporator by the vapour region. The work of Koito et al. [105] also suggested that the condensation resistance was independent of heat source size.

The last process to explore for the change in effective thermal conductivity of the vapour region with heat source area was the vapour flow resistance. Not only did the heat flux and operating temperature increase as heat source size decreased, but for the vapour flow analysis it must also be considered that the mass flux increased. Mass flux was directly linked to the heat flux through the evaporator region and would increase as the heat source size decreased. As the mass flux increased, the vapour velocity would also increase and the vapour velocity distribution within the vapour region would change. From the previous discussions in section 4.1.3, it was expected that the resistance owing to vapour flow would be very small. As only the experimental results from the 20-40W tests have been included in this analysis, vapour flow resistance and associated temperature drops would benefit from:

- A favourable location on the phase diagram of water (Figure 4.3) with a relatively steep $dP/dT$ curve.
- Favourable working fluid properties (mainly high vapour density) that reduced vapour pressure losses.
These favourable conditions would result in small vapour pressure losses and temperature losses regardless of the increased mass flux. It was thus expected that the vapour flow resistance contribution would be minor in this case, and it would be negligibly affected by the reduction in heat source size.

From the preceding discussions on the phase change and vapour flow processes, it appeared as though the effective thermal conductivity of the vapour region would decrease with reduced heat source size mainly due to the increased evaporative resistance. The cause of the increased evaporative resistance was likely the decrease in evaporative area. Although the evaporative resistance was not directly included in the conduction based numerical model of this work, the influence of it was recognised within the value for effective thermal conductivity of the vapour region.

5.2.2 Comparison against literature
The values for the effective thermal conductivity of the vapour region have not been widely explored in the literature as has been briefly discussed in section 2.4. Table 5.2 shows some relevant values for conduction based schemes and highlights the large variations present in the literature for the value of effective thermal conductivity of vapour. Further discussion of this is warranted.

Of the items in Table 5.2, it is only the work of Prasher [97] that aimed to determine the value for effective thermal conductivity of vapour. In the first case, a value of over 260,000 W/mK was obtained at an operating temperature of 80°C. This value was re-calculated for a slightly different VC at an operating temperature of 40°C to be 23,000 W/mK. These values were calculated based on experimental measurements of a device with similar design and construction to a VC. The function of this device was however to transfer heat along its length rather than to act as a heat spreader like the VC. This factored into the experimental measurements which were then used to determine the values. Furthermore, the analytical relations used for this value were based on vapour flow conditions similar to what would be observed in a heat pipe, or were otherwise more one-dimensional in nature. Hence a combination of simplified experimental measurements and numerical analysis were used to determine these values.
### Table 5.2 – Comparison table for the values of effective thermal conductivity obtained in the literature.

<table>
<thead>
<tr>
<th>Value (W/mK)</th>
<th>Measurement</th>
<th>Note</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1,900 \text{ – } 2,400)</td>
<td>Effective thermal conductivity of vapour region</td>
<td>Value depends on heat source area</td>
<td>This work</td>
</tr>
<tr>
<td>(260,000)</td>
<td>Effective thermal conductivity of vapour region</td>
<td>Operating temperature = 80°C</td>
<td>[97]</td>
</tr>
<tr>
<td>(23,000)</td>
<td>Effective thermal conductivity of vapour region</td>
<td>Operating temperature = 40°C</td>
<td>[97]</td>
</tr>
<tr>
<td>(40,000 \text{ – } 50,000)</td>
<td>Effective thermal conductivity of vapour region</td>
<td>Based on heat pipe literature [162]</td>
<td>[98, 99]</td>
</tr>
<tr>
<td>Infinite</td>
<td>Effective thermal conductivity of vapour region</td>
<td>Vapour region temperature assumed uniform</td>
<td>[95]</td>
</tr>
<tr>
<td>(49)</td>
<td>Effective thermal conductivity of VC</td>
<td>Orthotropic value ((z\text{ direction in this work}))</td>
<td>[100]</td>
</tr>
<tr>
<td>(1,148 \text{ – } 2,316)</td>
<td>Effective thermal conductivity of VC</td>
<td>Orthotropic value ((x/y\text{ direction in this work}))</td>
<td>[100]</td>
</tr>
<tr>
<td>(436 \text{ – } 558)</td>
<td>Effective thermal conductivity of VC</td>
<td>Isotropic value</td>
<td>[100]</td>
</tr>
<tr>
<td>(5.3)</td>
<td>Effective thermal conductivity of VC</td>
<td>Orthotropic value ((z\text{ direction in this work}))</td>
<td>[59]</td>
</tr>
<tr>
<td>(6,100)</td>
<td>Effective thermal conductivity of VC</td>
<td>Orthotropic value ((x/y\text{ direction in this work}))</td>
<td>[59]</td>
</tr>
<tr>
<td>(50,000)</td>
<td>Effective thermal conductivity of VC</td>
<td>Isotropic value</td>
<td>[84]</td>
</tr>
<tr>
<td>(450 \text{ – } 870)</td>
<td>Effective thermal conductivity of VC</td>
<td>Isotropic value</td>
<td>[101]</td>
</tr>
<tr>
<td>(936 \text{ – } 13640)</td>
<td>Effective thermal conductivity of local evaporator region</td>
<td>Isotropic value for the local evaporator region</td>
<td>[103]</td>
</tr>
<tr>
<td>(140 \text{ – } 792)</td>
<td>Effective thermal conductivity of local condenser region</td>
<td>Isotropic value for the local condenser region</td>
<td>[103]</td>
</tr>
</tbody>
</table>
The values used by Avenas et al. [98, 99] were based on the results from heat pipe experimentation in the literature. In this referring work [162] the effective thermal conductivity of vapour for a heat pipe was determined, as was the effective thermal conductivity of the heat pipe itself. The high effective thermal conductivity of the heat pipe is not being questioned in this work; rather it is the applicability of these values to VCs which have different boundary conditions, operating conditions and design to a heat pipe. Wang and Vafai [95] assumed that the temperature in the vapour region was always uniform in their analysis which was equivalent to having a vapour region with infinite thermal conductivity. In their work however, the vapour region thickness was approx. 20mm which is far greater than the vapour region thickness in the current study and of most designs in the current literature. The effect of this vapour region thickness can be appreciated in section 4.1.3.2.

The values obtained in this work for the effective thermal conductivity of the vapour region vary from those obtained in the literature. Some possible reasons for this have been highlighted in the previous discussions. It is expected that the results of this work are more appropriate than other values in the literature since they were based on actual experimental measurements of a VC in thermal spreading scenarios.

Most other values listed in Table 5.2 are for conduction based schemes whereby the entire VC was considered to be a bulk continuous medium with effective thermal conductivity applied. In some cases, orthotropic approaches were taken and thus values in each direction were obtained. In other cases, isotropic approaches were taken. Direct comparisons should not be made between these values and the values obtained for effective thermal conductivity of the vapour region however some discussions can nonetheless be presented.

The work of Chen et al. [100] calculated the value for effective thermal conductivity of the VC through a combination of experimental and numerical methods. The analytical solution to thermal spreading was used to solve for the value in both isotropic and orthotropic approaches. Their results were indeed influenced by the size of the heat source. Smaller heat source sizes led to reduced values for the effective thermal conductivity of the VC, a similar outcome to this work.

Chang et al. [103] calculated the local value for effective thermal conductivity of the evaporator and condenser regions of the VC. This was done through experimentation on a thin VC with an isotropic numerical model introduced to determine these values. The heat source size was kept constant through testing. The values were found to vary strongly
depending on the region under consideration, as well as on the heat input, filling ratio, orientation and vapour region thickness.

Wang et al. [59] performed similar experiments but relied on simplified analytical relations to determine the value using an orthotropic approach. Wang et al. [101] also relied on experimental measurements, but used Buckingham Pi theorem to suggest a relation for the isotropic effective thermal conductivity of the VC. Bose et al. [84] did not calculate this value in their work; instead they assumed it for the purposes of modelling.

From the previous discussions on both layered and bulk (isotropic and orthotropic) conduction based schemes, some important points should be noted:

- Only a few works have actually been dedicated to determining the effective thermal conductivity values. From these, it was generally found that the value is dependent on geometric parameters of the VC for example the size of the heat source or heat sink. Decreasing the size of the heat source tends to lower this value.
- Operating temperature also seemed to have an important effect on the effective thermal conductivity of vapour. Such conclusions were also drawn in section 4.1.3.
- In the layered models, phase change resistances were neglected and the associated temperature drops were not considered.
- This research was based on experimental thermal spreading measurements of VCs whereas others have relied on simpler measurement and analysis for determining the value of effective thermal conductivity of the vapour region. Hence it is expected that the results from this work are more applicable for VCs.

**5.2.3 Further discussions**

The values for the effective thermal conductivity of the vapour region determined in the previous sections were all dependent on the experimental results for spreading resistance. These experimental results relied on the measurement of values of which there was some associated uncertainty. See section 3.1.5 for discussion on the uncertainty analysis. It would be logical that the numerically determined effective thermal conductivity of vapour would have some uncertainty as well, which could be related to these experimental results. This has been explored in Figure 5.2 and Figure 5.3 which shows the numerically determined spreading resistance against the effective thermal conductivity of vapour. The experimental results have been overlayed onto this, as has the associated experimental uncertainty.
The uncertainty used in Figure 5.2 and Figure 5.3 was the minimum uncertainty as found in section 3.1.5. It would be reasonable to assume the minimum experimental uncertainty given that the results for spreading resistance were independent of heat input (apart from the 10W test which has been neglected from this analysis as described previously). Thus the uncertainty associated with the 40W tests could be used with confidence as an accurate description of the uncertainty in spreading resistance.

Figure 5.2 – The influence of experimental uncertainty on the numerical value of effective thermal conductivity of vapour for the 20mm x 20mm heat source.

The way in which the experimental uncertainty influenced the value for effective thermal conductivity of the vapour region has been shown in Figure 5.2 for the 20mm x 20mm heat source (area ratio of 43). The determined value in this case was 1,900 W/mK however when taking into account the experimental uncertainty in the spreading resistance, the value could fall between 1,670 W/mK and 2,180 W/mK. These variations exist due to the gradients present in Figure 5.2 for the numerical line which themselves are a consequence of the physics associated with thermal spreading. This also resulted in a non-uniform distribution of values. This can also be seen in the results for the 35mm x 35mm heat source (area ratio of 14) in Figure 5.3. In this case, the determined value was 2,410 W/mK, however when considering the experimental uncertainty, the resulting value could lie between 2,160 W/mK and 2,710 W/mK for the effective thermal conductivity of the vapour region.
The influence of experimental uncertainty on the numerical value of effective thermal conductivity of vapour for the 35mm x 35mm heat source.

The gradients present in Figure 5.2 and Figure 5.3 can also be seen as quite favourable to the thermal designer as large changes to the effective thermal conductivity of the vapour region will only lead to minor changes in the spreading resistance value. If the value of effective thermal conductivity of vapour were to be overestimated by 10%, the results for spreading resistance would be affected by less than 10%. In the 20mm x 20mm case, overestimating the effective thermal conductivity of vapour by 10% would only lead to a 4% reduction in the spreading resistance of the VC. This gives thermal designers some leniency and flexibility in their designs.

The variations of effective thermal conductivity of the vapour region have been summarised in Figure 5.4 for each of the area ratios (or heat source sizes). Relative changes to the effective thermal conductivity of the vapour region were similar for all heat source sizes, although the absolute values of the changes were of differing magnitudes. It is expected that intermediate values could be interpolated with reasonable accuracy from the shaded region within Figure 5.4.
Figure 5.4 – The variations in effective thermal conductivity of vapour with area ratio and the associated uncertainty. The square markers represent the experimental data points for each heat source size.

Although it is expected that interpolation between the area ratio values would be valid, extrapolation outside of the tested range is cautioned. Particularly at low area ratios, when the effect of vapour flow is less important, the validity of the above data cannot be guaranteed. For this reason, it is expected that the minimum area ratio for which the above data could be extrapolated would be between 5 and 10. The maximum valid area ratio is expected to be less critical, however in order for the VC performance to exceed that of a copper heat spreader, the effective thermal conductivity of the vapour region should remain above approx. 1,000 W/mK. Extrapolating these results to the point where effective thermal conductivity reaches 1,000 W/mK leads to an area ratio of approx. 65-70. At these area ratios, the performance of the VC would be similar to that of a copper heat spreader, and beyond these it would have worse performance than copper. This figure could thus be very useful beyond this work for designers of thermal management systems using VCs. These results are expected to be valid for VC designs with thickness of greater than approx. 1.5mm i.e. not thin or ultra-thin designs.
5.3 Thermal study of evaporator wick

Now that the effective thermal conductivity of the vapour region has been determined in the previous sections, there exists many opportunities to perform some optimisation and sensitivity analysis regarding the thermal design of VCs. Some aspects concerning the evaporator wick have been explored in the sections that follow. The evaporator wick was chosen since it was identified in this work as a key component to VC operation and performance.

5.3.1 Effect of inner wick effective thermal conductivity

It has been seen in the literature, and further confirmed by the results of this research (see section 4.2.1) that the effective thermal conductivity of the evaporator wick has strong implications on the thermal performance of the VC. In this section, a study was performed to investigate the sensitivity of VC spreading resistance to the effective thermal conductivity of the inner evaporator wick region.

In this study, the VC under consideration had identical arrangement and geometry to the VC of the previous discussions (section 3.2.2). The most relevant geometry here was the size of the inner evaporator wick region, which was maintained at 40mm x 40mm x 0.5mm throughout this study. The effective thermal conductivity of this region was varied through ranges which could typically be expected from wicks used in phase change devices. Some of these wick structures have been discussed in the literature review.

These variations were performed for heat source sizes of 20mm x 20mm and 35mm x 35mm such that the effect of heat source size on this could also be explored. The corresponding effective thermal conductivity of the vapour region for these cases was found to be 1,900 W/mK and 2,410 W/mK, respectively, from section 5.2. The results have been shown in Figure 5.5.

There are many interesting observations to be made regarding Figure 5.5. It can be seen for both heat source sizes that as the effective thermal conductivity increased the spreading resistance appeared to asymptote towards a value. This asymptotic behaviour varied slightly for both heat source sizes. The larger heat source size appeared to approach this asymptote at lower values of effective thermal conductivity of the inner wick.
Another way of assessing this asymptotic behaviour was introduced by comparing the relative changes to spreading resistance with effective thermal conductivity. In this method, the spreading resistance when the effective thermal conductivity of the inner wick was 200 W/mK was found. This conductivity value was chosen as it appears to be above the upper limit of what can be attained with any wick structure, even advanced wicks using foamed metals or carbon based materials. Hence when the inner wick effective thermal conductivity was 200 W/mK, the spreading resistance of the VC could be considered a minimum. When the spreading resistance had increased by 10% from this value, the effective thermal conductivity of the inner wick region was re-determined. This value was deemed an ‘ideal’ effective thermal conductivity value; any conductivity below this would lead to sharp increases in spreading resistances and any conductivity above this would lead to minor decreases in spreading resistance. This has been shown in Figure 5.6. It can generally be said that mesh wicks with low effective thermal conductivity of less than approx. 15 W/mK should be avoided for all cases due to the sharp increases at these low values.

Figure 5.5 – The sensitivity of spreading resistance of the VC with the effective thermal conductivity of the inner wick for the 35mm x 35mm and 20mm x 20mm heat source sizes. Approximate ranges have been shown for each wicking structure.
Figure 5.6 – Beyond some ‘ideal’ value of conductivity of the inner wick, spreading resistance tends to asymptote. Prior to this, there are strong variations in spreading resistance with effective thermal conductivity. For the larger heat source size (35mm x 35mm) this ideal value appeared to be approx. 40 W/mK compared to approx. 65 W/mK for the smaller heat source size (20mm x 20mm). It was interesting to note that the ideal value tended to become larger as the heat source size reduced. This suggested that the inner wick effective thermal conductivity was more critical when the heat source size was small. As a result, the penalties for low conductivity wicks were more severe when the heat source size was small.

It can be seen from Figure 5.5 that the VC discussed in this research (which had inner wick effective thermal conductivity of 32 W/mK as shown in section 3.2.2.2) performed quite well. Had an inner wick with effective thermal conductivity of 15 W/mK been used, the spreading resistance for the smaller heat source size (20mm x 20mm) would have increased by over 16.5% compared to 14% for the larger heat source size (35mm x 35mm). Conversely, had a wick with effective thermal conductivity in the range of 100 W/mK been used, then a reduction in spreading resistance of slightly less than 15% would have been obtained for the smaller of the heat source sizes (20mm x 20mm). Doing the same with the larger of the heat source sizes (35mm x 35mm) would only lead to a reduction in spreading resistance by about 9%. The thermal performance of the VC was clearly more strongly influenced by the effective thermal conductivity of the inner wick when the heat source sizes were smaller. It
thus becomes more difficult to justify the use of higher effective thermal conductivity wicks for larger heat source sizes as there are less thermal benefits. At smaller heat source sizes (20mm x 20mm and less) these improvements in effective thermal conductivity may become worthwhile.

With the ever reducing sizes of ICs available on the market, the need for improving the effective thermal conductivity of wick structures becomes clear from this. Thus further research and development of wick structures must be carried out. As has been highlighted previously in this work, this could include the use of metal foam or carbon based wicks, or alternatively could involve the improvement in current sintered or grooved wick structures.

The preceding analysis only considers the thermal effects of the inner wick structure. For completeness, the fluid aspects which help to govern VC performance and operation should also be considered. However as this inner wick was only located at the heat source region, it was expected that thermal aspects were more important than fluid aspects. The most important fluid consideration here would have been the effective capillary radius of the wick which generates capillary pressure. The porosity and permeability would carry less importance for the inner wick since it was localised to the region above the heat source and thus only constituted a small amount of the overall fluid flow path. Some considerations for the outer wick region have been explored in the following section.

5.3.2 Effect of outer wick effective thermal conductivity
Continuing on from the previous discussions, the significance of the outer wick on thermal performance will now be explored. This has been done in a similar manner to section 5.3.1. The numerical model developed in this work was used to explore the sensitivity of spreading resistance to the effective thermal conductivity of the outer wick region. This was done for the 20mm x 20mm and 35mm x 35mm heat sources. The corresponding effective thermal conductivity of the vapour region for these was found to be 1,900 W/mK and 2,410 W/mK, respectively, from section 5.2.

The VC under consideration had the same geometry as in section 3.2.2. In this VC, the size of the inner evaporator wick was maintained at 40mm x 40mm x 0.5mm and thus the outer wick was placed around this. This general arrangement can be seen in Figure 3.3a and Figure 3.24. The effective thermal conductivity of this outer wick region was varied in a similar manner to section 5.3.1 in order to explore these effects. These results have been shown in Figure 5.7.
The results in Figure 5.7 are rather uneventful which in itself provided some interesting insights and confirmed some previous assumptions made in section 3.1.1 regarding the role of the outer wick region. It was assumed in that section that the outer wick did not need to be of high effective thermal conductivity due to its location being away from the heat source. These results support this claim and suggest that the role of the outer wick in the heat transfer process is minimal. The spreading resistance observed in Figure 5.7 was nearly constant which suggested that the outer wick region had negligible influence on this. There were very slight reductions observed, which tended to be larger for the larger heat source, however these reductions were still less than 2.5% between the extreme values of effective thermal conductivity of the outer wick. This was far less than the effect of the inner wick region as seen in section 5.3.1.

It can thus be confirmed that it was only the inner wick region (that is, the region around the heat source location) which impacted the thermal performance within the VC. This was due to the localised heat source and relative thinness of the container and wick layers which were not able to sufficiently spread heat within them. Thus heat transfer through the bottom container and evaporator wick region could be reasonably assumed as one-dimensional since the majority of heat tended to pass through the inner wick region above the heat source. This has been represented with Figure 5.8.
Based on these preceding results, it can be stated that the inner wick structure should focus mainly on thermal aspects and the outer wick structure should focus mainly on fluid aspects as it only contributes in a minor way to the thermal performance. A mesh wick is thus a good choice for this region since it has high permeability (it also has low effective thermal conductivity but this has negligible effect on thermal performance). This validates the design philosophy presented in section 3.1.1. The next step of this analysis would be to explore the sizing of the inner wick structure for given heat source sizes.

![Diagram of vapor and liquid flow through wicks](image)

*Figure 5.8 – Results from this work confirm that the thermal performance of the VC was more strongly influenced by the effective thermal conductivity of the inner wick compared to the outer wick. This suggested the majority of heat passed through the inner wick region as shown.*

5.3.3 Effect of central wick sizing for given heat source size

It has been previously shown that the inner wick plays a much larger role in the thermal performance of the VC than the outer wick. In the previous sections, the size of the inner wick structure has been fixed at 40mm x 40mm x 0.5mm with variations to the heat source size. In this section, the inner wick size has been varied for a fixed heat source size in order to explore if there exists any potential thermal improvements in doing this. In a sense, this section attempts to characterise the amount of thermal spreading which can be achieved within the evaporator wick region.

The VC under consideration had the same external dimensions and thickness as in section 3.2.2 however the internal dimensions of the inner and outer wick were changed as required. The general evaporator wick arrangement still followed that of Figure 3.3. The dimensions of the inner and outer wick regions were altered such that the two regions never overlapped. Hence increasing the size of the inner wick region meant that the size of the outer wick region had to consequently reduce. The centre of the heat source did not move in this analysis and it was still coincident with the centre of the inner wick structure.
For this analysis, the effective thermal conductivities of the inner and outer wick regions were held at 32 W/mK and 1.5 W/mK, respectively. The results of this can be seen in Figure 5.9 for the heat source sizes of 35mm x 35mm and 20mm x 20mm. The size of the inner wick was kept to greater than or equal to the heat source size. Inner wicks smaller than the heat source sizes were not trialled based on the results and discussions from the previous sections.

As can be seen from Figure 5.9, the effect of inner wick sizing was very small on the spreading resistance of the VC. When the inner wick size was equal to the heat source size, the spreading resistance was slightly higher than when the inner wick size was larger than the heat source size. Thus there was some improvement by having inner wicks that were slightly larger than the heat source size however this improvement was very small. At most, the spreading resistance decreased by approx. 2.5%. Extending the inner wick beyond 5mm in each direction of the heat source was found to have negligible effect on spreading resistance.

This information could provide guidance for thermal designers of similar VCs which use composite wick structures. The inner wick, of high effective thermal conductivity, should not be larger than the heat source by more than 5mm in each direction. Considering the coincident placement of the heat source and inner wick in this work, this meant that the inner wick should not extend more than 2.5mm from the heat source. Any extensions beyond this point may not be useful for reducing spreading resistance. A representation of this has been shown in Figure 5.10 for clarity.
Figure 5.10 – The guidelines for the sizing of the inner evaporator wick in side view (a) and top view (b). It is suggested that the inner evaporator wick should not exceed the dimensions of the heat source by more than 5mm (in each direction).

Conversely, extending beyond this point will actually reduce the fluid transfer capability of this particular VC. This inner wick was sintered powder which has very low permeability. The outer wick was mesh which has high permeability. Increasing the size of the inner wick meant that a larger portion of the fluid flow path was within the region of low permeability as the sintered powder now occupied more space. This would increase the liquid pressure losses. Hence the size of the inner wick should be kept to within the above mentioned limits for both thermal and fluid considerations.

5.4 Summary and remarks

The effective thermal conductivity of the vapour region was found to be a function of heat source size, with reductions in heat source sizes leading to reductions in the value for the effective thermal conductivity of the vapour region. The likely cause of these reductions was identified as the evaporative phase change resistance. Still, the values obtained for the effective thermal conductivity of the vapour region were many times that of copper as shown in Figure 5.4. This can be used by thermal designers of similar VCs for a variety of modelling scenarios. One such scenario was explored in this work; a study on the composite evaporator wick was performed. The most important parameter to optimise from this analysis was the effective thermal conductivity of the inner wick structure. This had strong influence on the thermal performance of the VC as shown in Figure 5.4. Most other parameters were relatively insignificant for thermal performance, and thus focus should be placed on these to maximise the fluid performance of the VC. The usefulness of such numerical models can be seen from this simple study.
Chapter 6 – Conclusion and recommendations

This research presented an experimental and numerical investigation into VC heat spreaders. For this, a design of composite wick VC which utilised both sintered powder and mesh wicks in the evaporator was developed and tested. The experimental performance of this VC was measured and then used to complete a numerical investigation with a validated conduction based model of the VC. The main conclusions and recommendations from this entire study are presented here.

6.1 Concluding remarks

The spreading resistance of the VC was experimentally determined to be lower than that of an equivalent sized copper heat spreader. When the heat source size was 35mm x 35mm, the spreading resistance of the VC was 0.067 K/W compared to 0.134 K/W for the copper heat spreader. Reducing the heat source size to 20mm x 20mm increased the spreading resistance for both heat spreaders to 0.131 K/W and 0.216 K/W for the VC and copper, respectively. In these cases, the spreading resistance of the composite evaporator wick VC was less than that of the solid copper, highlighting its superiority as a heat spreader.

The spreading resistance of the VC was found to be independent of heat input, except for the lowest heat input tested of 10W. The discussions above neglect this 10W result since it was identified that start-up difficulties of the VC were the cause of this poor performance. The start-up difficulties arose due to the low operating temperature of the VC at this low heat input. At this low operating temperature, several factors were identified to cause poor thermal performance associated with two-phase heat transfer devices. These were discussed in detail and have been summarised below:

- The low operating temperature of the working fluid led to a location on the phase diagram that was unfavourable. This adverse location meant that the slope of the saturation curve was rather flat. Consequently, vapour phase pressure losses would lead to large vapour phase temperature drops. At elevated operating temperatures, the slope of the saturation curve greatly increased and thus the vapour phase temperature drops became less influenced by vapour phase pressure losses.

- The low operating temperature also led to rather poor thermodynamic properties of the working fluid which tended to increase the vapour phase pressure losses. The temperature dependence of latent heat of evaporation, dynamic viscosity and vapour
density were found to influence these losses. The most relevant property was identified as the vapour density which had very strong temperature dependence. This tended to increase the vapour phase pressure losses as its effect outweighed the improvements in other relevant properties.

- Phase change mechanisms were also found to depend on operating temperature. The heat transfer coefficient associated with the phase change interface had similar thermodynamic considerations to the vapour phase pressure losses above. At low operating temperatures it was expected that the interface heat transfer coefficient would reduce due to adverse working fluid properties.

The above mentioned aspects were thought to be the main contributors to poor thermal performance at low heat inputs. The start-up difficulties associated with the vapour phase generally agreed with experimental observations of the condenser surface temperature, however it was more difficult to validate the role of the interface phase change processes.

Further measurements of the VC heat spreader confirmed that there were some aspects of thermal performance which required improvement, regardless of heat input or operating temperature of the VC. This was mainly linked to the axial temperature drops observed, which were many times larger for the VC heat spreader than that of solid copper. The poor capabilities of the VC in this respect were expected to be due to the relatively low value of effective thermal conductivity of the wick structure which heat must conduct through before reaching the evaporation phase change interface and two-phase mechanisms become involved.

Experimental results were used to validate the numerical model. The numerical conduction based model was able to determine the values for the effective thermal conductivity of the vapour region based on experimental measurements of the VC. The model was validated against the measurements of the solid copper heat spreader with very good agreement. The values for effective thermal conductivity of the vapour region were found to be a function of heat source size, with values of 1,900 W/mK, 2,220 W/mK, 2,380 W/mK and 2,410 W/mK resulting for the 20mm x 20mm, 25mm x 25mm, 30mm x 30mm and 35mm x 35mm heat source sizes, respectively.

The versatility of the determined range of effective thermal conductivity values was that it could be used for a variety of modelling applications with reasonable simplicity since it was purely a conduction term. Thus it can be applied much like thermal conductivity of metals or like the effective thermal conductivity commonly associated with wick regions of the VC.
The effective thermal conductivity of the vapour region has been seldom explored in the literature, thus this work provides guidance for designers of thermal management systems using VC heat spreaders about the range of values which could be expected. As the results from this work were based on experimental measurements of VCs, they were expected to be more appropriate than other values available in the literature which are often based on varying assumptions and conditions.

The value for the effective thermal conductivity of the vapour region was inherently a combination of multiple factors. These factors were simplified into conduction based mechanisms for the purposes of the numerical model introduced. Thus the effects of evaporative resistance, vapour flow resistance and condensation resistance were integral with this term. The effective thermal conductivity of the vapour region was found to be a function of the heat source size which implied that one of these aspects was influenced by the heat source size. The evaporative resistance was identified as the most likely cause of the changes to this value due to the intricate link between the heat source area and the evaporative area. The other aspects of vapour flow resistance and condensation resistance were not expected to have been the cause of these variations in effective thermal conductivity of the vapour region since they would not appreciably vary with changes to the heat source size.

The model developed was then used to perform a more thorough investigation into the composite evaporator wick. It was determined that the most important aspect which governed the performance of the composite wick was the effective thermal conductivity of the inner wick. There were large variations in spreading resistance observed with changes to the inner wick effective thermal conductivity. It was suggested that the ideal effective thermal conductivity was approx. 40 W/mK and 65 W/mK for the inner wick in the cases of the 35mm x 35mm and 20mm x 20mm heat source sizes, respectively. The magnitude of these values lay in the upper range of what could be expected from common wick structures including sintered and grooved wicks. Above these values, there was very little improvement in spreading resistance possible according to the model. The size of the inner wick should be at most 5mm larger than the heat source in each direction. Any larger would not lead to further reductions in spreading resistance, however could possibly limit the fluid performance of the VC. Instead, the outer wick structure should be devoted to maximising fluid performance as it was found to have negligible effect on thermal performance. The majority of the fluid flow path passes through the outer wick structure however only a marginal amount of heat actually transfers through this region during operation.
6.2 Recommendations for further work

There are a large number of recommendations for future work which exist based on this research. Some of the more important recommendations have been included below.

Investigation into phase change mechanisms
Understanding the phase change mechanisms which govern the evaporation and condensation processes should be a priority for two-phase heat transfer devices. Furthermore, the influence of complicated wick structures on these mechanisms needs to be better explored such that more optimal designs with minimal interface resistances can be introduced.

Prediction of wick structure effective thermal conductivity
Some correlations for predicting the effective thermal conductivity of wick structures (mainly sintered powder) have been discussed in this work. It was very difficult to find an appropriate relation to accurately predict this property and thus experimental data was relied on. It is recommended that analysis of different wick structures common to two-phase devices be performed and the effects of common wick parameters are included in this analysis. This would help to clear up the large amount of confusion surrounding these values.

Improvement of wick effective thermal conductivity
In the case of VC heat spreaders, one of the main limitations to the thermal performance was the effective thermal conductivity of the evaporator wick, particularly in the heat input region. Wicks must be developed with effective thermal conductivities of above 50 W/mK to improve this performance. This will become even more critical as future devices rely on even smaller heat generating components. Further research into metal foam and carbon based wick structures are suggested as they could provide the required improvement to this property.

Complete thermal-fluid model capable of optimisation analysis
The current numerical model employed a number of assumptions which greatly reduced the computational complexity of the model however also meant that specific details of VC performance were overlooked. A model needs to be developed which can accurately capture the heat and mass transfer aspects within the VC without relying on costly and demanding computational resources. The ability to perform optimisation analysis with such thermal-fluid models could drive future development of VCs within the industry.
References


Appendix A – Experimental heat transfer coefficient (external)

This appendix details the calculation of the experimentally determined external heat transfer coefficient on the cold side of the heat spreader i.e. the side which was exposed to the air stream as seen in section 3.1.3. This heat transfer coefficient was required for the boundary conditions in the numerical procedure as in section 3.2.3. The average heat transfer coefficient \( h \) over a surface can be defined as in equation (A.1).

\[
h = \frac{\dot{Q}}{A(T_{\text{surf}} - T_{\infty})}
\]  \hspace{1cm} (A.1)

Here \( \dot{Q} \) is the heat transfer rate, \( A \) is the surface area, \( T_{\text{surf}} \) is the average surface temperature and \( T_{\infty} \) is the average free stream temperature. All of these values were measurable through the techniques described in the body of this thesis (section 3.1.3.2). It was important to note here that the free stream temperature was evaluated as the average of the inlet and outlet thermocouple array temperatures.

Using a sample set of experimentally measured data for the VC heat spreader allowed for estimation of the heat transfer coefficient. In this sample set \( \dot{Q} = 31.50 \text{ W}, T_{\text{surf}} = 82.15^\circ\text{C} \) and \( T_{\infty} = 20.50^\circ\text{C} \). The surface area was \( A = 0.123 \times 0.140 = 0.0172 \text{ m}^2 \). Hence the average heat transfer coefficient over this surface can be found with equation (A.1):

\[
h = \frac{31.50}{0.0172(82.15 - 20.50)}
\]

\[h = 29.71 \text{ W/m}^2\text{K}\]

Using another sample data set, this time for the copper heat spreader, we have \( \dot{Q} = 21.88 \text{ W}, T_{\text{surf}} = 62.41^\circ\text{C}, T_{\infty} = 18.59^\circ\text{C} \) and \( A \) the same as before.

\[
h = \frac{21.88}{0.0172(62.41 - 18.59)}
\]

\[h = 29.03 \text{ W/m}^2\text{K}\]

Repeating this for many data sets yields an average heat transfer coefficient of around 30 W/m²K. Hence this value was chosen to represent the average external convective heat transfer coefficient on the cold side of the heat spreader. Besides, the influence of heat transfer coefficient on spreading resistance was very small unless there was an order of magnitude change (as seen in section 4.1.1). Thus even with small changes occurring during tests (possibly caused by changed operating conditions) the impact of this was negligible.
## Appendix B – Saturated water properties

Table B.1 – Some relevant properties of saturated pure water.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Saturation Pressure (kPa)</th>
<th>Liquid Density (kg/m³)</th>
<th>Vapour Density (kg/m³)</th>
<th>Latent Heat of Evaporation (kJ/kg)</th>
<th>Liquid Dynamic Viscosity (Pa.s)</th>
<th>Vapour Dynamic Viscosity (Pa.s)</th>
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Note: Properties obtained from REFPROP [163].