Performance Characterization of Water Heat Pipes and their Application in CubeSats

Solving the Thermal Challenge of Next Generation CubeSats

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PERFORMANCE CHARACTERIZATION OF WATER HEAT PIPES AND THEIR APPLICATION IN CUBE SATS

Solving the Thermal Challenge of Next Generation CubeSats

by

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in partial fulfillment of the requirements for the degree of

Master of Science
in Space Engineering

at the Delft University of Technology,
to be defended publicly on Wednesday 18 May, 2016 at 10:00h.

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Project duration: September, 2015 – May, 2016
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An electronic version of this thesis is available at http://repository.tudelft.nl/.
To my grandfather, who would be immensely proud and whose love and care for others remain my true source of inspiration, even up till this very day...
I would like to express my sincere gratitude to my supervisors who have helped me turning this project into a success: J. Guo for his guidance and support throughout the entire project and pointing me in the right direction when I tended to stray off the path set out; H. van Gerner for his inspiring enthusiasm and knowledge about the matter, which motivated me and made me strive for better results; Z. de Groot for providing me the opportunity to work on this, sharing his knowledge, and expressing his confidence in me, which kept me going and boosted my own confidence.

I would also like to thank all my colleagues at both the NLR and ISIS, with whom I have had a great time during the project. In specific, I would like to mention A. Pauw, who has helped me setting up the tests at NLR and kept his patience while answering an endless stream of questions from my side.

I owe my deepest gratitude to my parents who have always been there for me and supported me not only financially, but also with their unconditional love, their encouragement, and by keeping faith in me. The values and norms they stand for will remain a true source of inspiration throughout my life. I will be forever in your debt.

Finally, I thank my girlfriend for always supporting me (often unknowingly) and being there when needed most and listening to my often boring and incomprehensible, work-related stories.

Thank you.

-Hugo Brouwer
Utrecht, 2016
The master track Space Engineering is finalized by the master thesis project: A project that puts the acquired academic skills and knowledge by the student to the test. The project offers the possibility to innovate and perform research in a scientific area related to the Space Engineering master track.

CubeSats and the field of thermal control have sparked my interest during my earlier internship at ISIS, Innovative Solutions In Space. The design-philosophy of CubeSats allows an engineer to get involved in almost all the different aspects of satellite design. The fun of being actively involved in satellite projects during this internship and obtaining hands-on experience led me to choose for an external thesis project.

After my successful internship at ISIS there was a mutual benefit for both ISIS and me to work on a new project as part of graduation. Continuing in the domain of thermal control required bringing in the expertise of the Netherlands Aerospace Centre, NLR, who have years of experience with thermal control systems and testing. This combination proved to be the perfect match and the project led to this thesis report in which all the work performed is accumulated.
ABSTRACT

The CubeSat platform has long since its introduction surpassed its original intend of being an educational and technology demonstration platform only. The on-going trend of miniaturization has enhanced the capabilities of CubeSats and thereby sparked the interest of the entire space sector. The recognition and acknowledgement of its potential has led to an everlasting hunger for more performance which translates directly into a demand for more power. While the CubeSat form factor grows only slowly the power density of CubeSats is rapidly increasing leading to thermal problems that can be destructive for any CubeSat mission. Up till now thermal control has hardly been part of the CubeSat design chain and therefore has not been able to keep up with this trend. As thermal control will become vital for the continuation of the aforementioned trend new solutions have to be brought to the table. The solution proposed to cope with the increase in power density is the heat pipe. The heat pipe is a two-phase, passive thermal control device that is able to transport a large amount of heat without introducing a large thermal gradient over the heat source and sink. Specifically heat pipes with water as fluid are extensively used in Earth applications, low cost, and unrivaled in terms of performance. Although in larger satellites water heat pipes have often been disregarded because of their dis-functionality below 0 °C, this property might be turned into an advantage if the heat pipe acts as a heat switch.

The thesis is centered around the possibility of employing water heat pipes in CubeSats, with the focus on performance characterization and integration into the CubeSat platform and poses the following research question:

Can commercial water heat pipes solve the thermal challenge of high performance CubeSat missions?

In order to find an answer to this question performance characterization, bending, and gravity-tilt tests were carried out. Transient start-up tests were performed and repetitive freeze/thaw cycles to observe the effect of the LEO environment on heat pipes. Finally, heat pipe integration in a CubeSat structure was done along with testing.

Following from the different performance characterization tests and the heat pipe integration experiments it becomes clear that commercial water heat pipes can indeed solve for the thermal challenge that is upcoming in high performance CubeSat missions. The water heat pipe is able to passively transport the heat loads expected (up to 10 W) in the next-generation CubeSats, thereby keeping the heat source within its temperature limits. Bending was found to have negligible influence on the heat pipe’s performance, which gives a large degree of freedom during integration and therefore poses no additional constraints on other internal (sub)systems. The performance of heat pipes is characterized by an increase in heat transfer coefficient for higher heat pipe temperatures, while at temperatures below the freezing point of water only pure conduction remains. The latter is found to be beneficial as it prevents the heat source from cooling down too quickly and even with an instant 10 W heat load the heat pipe is able to thaw way before the temperature limit of the heat source is reached.

The critical aspect in integrating a heat pipe in the CubeSat platform is attaining an efficient heat transfer between the heat pipe and the source and sink. For this, a proper design is necessary to reduce the thermal gradients between these interfaces. Furthermore, the heat pipe is perfect for heat transport but will create a hotspot at the chosen heat sink element of the CubeSat if no other thermal control mechanisms are involved which are able to remove this heat from the satellite.
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NOMENCLATURE

Acronyms
ADCS Attitude Determination and Control System
CDHS Command and Data Handling System
COTS Commercial Off-The-Shelf
EO Earth Observation
EPS Electronic Power System
FM Figure of Merit
ISIS Innovative Solutions In Space
LEO Low Earth Orbit
LTAN Local Time at Ascending Node
NLR Netherlands Aerospace Centre
PCB Printed Circuit Board
TC Thermo-Couple
TEC ThermoElectric Cooler
TTC Telemetry Telecommand and Control

Symbols
\(\alpha\) Tilt angle °
\(\alpha_s\) Solar absorptivity -
\(\beta\) Solar beta angle °
\(\Delta p\) Pressure gradient N m\(^{-2}\)
\(\Delta p_l\) Liquid pressure drop N m\(^{-2}\)
\(\Delta p_v\) Vapor pressure drop N m\(^{-2}\)
\(\Delta p_g\) Gravitational head N m\(^{-2}\)
\(\Delta T_{\text{crit}}\) Critical fluid temperature K
\(\delta\) Rectangular groove depth m
\(\delta_b\) Thermal/boundary layer thickness m
\(\dot{m}\) Mass flow kgs\(^{-1}\)
\(\dot{Q}\) Heat flow W
\(\epsilon\) Wick porosity -
\(\mu\) Viscosity Pa s\(^{-1}\)
\(\mu_l\) Liquid viscosity Pa s\(^{-1}\)
\(\mu_v\) Vapor viscosity Pa s\(^{-1}\)
\(\omega\) Rectangular groove width m
\(\rho\) Density kg m\(^{-3}\)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma_1$</td>
<td>Surface tension</td>
<td>N m$^{-1}$</td>
</tr>
<tr>
<td>$\sigma_b$</td>
<td>Stefan-Boltzmann constant</td>
<td>Js$^{-1}$ m$^{-2}$ K$^{-3}$</td>
</tr>
<tr>
<td>$\tau_1$</td>
<td>Shear stress liquid</td>
<td>Pa</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Liquid contact angle</td>
<td>rad</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Emissivity</td>
<td>-</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Surface roughness</td>
<td>m</td>
</tr>
<tr>
<td>$a$</td>
<td>Speed of sound</td>
<td>ms$^{-1}$</td>
</tr>
<tr>
<td>$A_{hp}$</td>
<td>Heat pipe copper cross-sectional area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$A_v$</td>
<td>Vapor cross-sectional area</td>
<td>m</td>
</tr>
<tr>
<td>$A_w$</td>
<td>Wick cross-sectional area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$C_{ij}$</td>
<td>Conductive coupling</td>
<td>WK$^{-1}$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Thermal capacity</td>
<td>JK$^{-1}$</td>
</tr>
<tr>
<td>$d$</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>$d_h$</td>
<td>Hydraulic diameter</td>
<td>m</td>
</tr>
<tr>
<td>$f_D$</td>
<td>Darcy friction factor</td>
<td>-</td>
</tr>
<tr>
<td>$h$</td>
<td>Height</td>
<td>m</td>
</tr>
<tr>
<td>$h$</td>
<td>Satellite orbital altitude</td>
<td>m</td>
</tr>
<tr>
<td>$h_c$</td>
<td>Convective heat transfer coefficient</td>
<td>Wm$^{-2}$ K$^{-1}$</td>
</tr>
<tr>
<td>$h_{l,v}$</td>
<td>Latent heat of evaporation</td>
<td>kJkg$^{-1}$</td>
</tr>
<tr>
<td>$h_l$</td>
<td>Heat transfer coefficient</td>
<td>Wm$^{-2}$ K$^{-1}$</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity</td>
<td>Wm$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$k_{eff}$</td>
<td>Effective thermal conductivity</td>
<td>Wm$^{-1}$ K</td>
</tr>
<tr>
<td>$K_L$</td>
<td>Loss coefficient</td>
<td>-</td>
</tr>
<tr>
<td>$l$</td>
<td>Length</td>
<td>m</td>
</tr>
<tr>
<td>$l_a$</td>
<td>Adiabatic section length</td>
<td>m</td>
</tr>
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<td>$l_c$</td>
<td>Condenser section length</td>
<td>m</td>
</tr>
<tr>
<td>$l_{eff}$</td>
<td>Effective heat pipe length</td>
<td>m</td>
</tr>
<tr>
<td>$l_e$</td>
<td>Evaporator section length</td>
<td>m</td>
</tr>
<tr>
<td>$N$</td>
<td>Number of grooves</td>
<td>-</td>
</tr>
<tr>
<td>$p_{c,a}$</td>
<td>Vapor pressure drop in adiabatic section</td>
<td>Nm$^{-2}$</td>
</tr>
<tr>
<td>$p_{c,c}$</td>
<td>Vapor pressure drop in condenser section</td>
<td>Nm$^{-2}$</td>
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<td>Vapor pressure drop in evaporator section</td>
<td>Nm$^{-2}$</td>
</tr>
<tr>
<td>$p_{c,m}$</td>
<td>Maximum capillary pressure</td>
<td>Nm$^{-2}$</td>
</tr>
<tr>
<td>$p_c$</td>
<td>Capillary pressure</td>
<td>Nm$^{-2}$</td>
</tr>
<tr>
<td>$r$</td>
<td>Radius of curvature</td>
<td>m</td>
</tr>
<tr>
<td>$r_b$</td>
<td>Radius bubble</td>
<td>m</td>
</tr>
<tr>
<td>$R_E$</td>
<td>Radius of the Earth</td>
<td>m</td>
</tr>
<tr>
<td>$r_{h,l}$</td>
<td>Liquid hydraulic radius</td>
<td>m</td>
</tr>
<tr>
<td>$r_{h,v}$</td>
<td>Hydraulic vapor radius</td>
<td>m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
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<tr>
<td>--------</td>
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</tr>
<tr>
<td>$r_i$</td>
<td>Inner radius</td>
<td>m</td>
</tr>
<tr>
<td>$r_o$</td>
<td>Outer radius</td>
<td>m</td>
</tr>
<tr>
<td>$r_{c,c}$</td>
<td>Effective capillary radius at condenser end</td>
<td>m</td>
</tr>
<tr>
<td>$r_{c,e}$</td>
<td>Effective capillary radius at evaporator end</td>
<td>m</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$t_δ$</td>
<td>Groove wall height</td>
<td>m</td>
</tr>
<tr>
<td>$t_ω$</td>
<td>Groove wall width</td>
<td>m</td>
</tr>
<tr>
<td>$T_{sat}$</td>
<td>Fluid saturation temperature</td>
<td>K</td>
</tr>
<tr>
<td>$T_v$</td>
<td>Vapor temperature</td>
<td>K</td>
</tr>
<tr>
<td>$T_w$</td>
<td>Heat pipe wall temperature</td>
<td>K</td>
</tr>
<tr>
<td>$t_w$</td>
<td>Heat pipe wall thickness</td>
<td>m</td>
</tr>
<tr>
<td>$u$</td>
<td>Local velocity</td>
<td>m s$^{-1}$</td>
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<td>$v$</td>
<td>Velocity</td>
<td>m s$^{-1}$</td>
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This chapter provides the introduction to the master thesis project and presents the research questions and objectives that will serve as guidelines throughout the project. The employed research methodology will be discussed along with the chapter division of the report.

1.1. General Introduction
The CubeSat platform has long since its introduction surpassed its original intend of being an educational and technology demonstration platform only. The on-going trend of miniaturization has enhanced the capabilities of CubeSats and thereby sparked the interest of the entire space sector. The recognition and acknowledgement of its potential has led to an everlasting hunger for more performance which translates directly into a demand for more power. While the CubeSat form factor grows only slowly the power density of CubeSats is rapidly increasing leading to thermal problems that can be destructive for any CubeSat mission.
Up till now thermal control has hardly been part of the CubeSat design chain and therefore has not been able to keep up with this trend. As thermal control will become vital for the continuation of the aforementioned trend new solutions have to be brought to the table. The solution proposed to cope with the increase in power density is the heat pipe.
The heat pipe is a two-phase, passive thermal control device that is able to transport a large amount of heat without introducing a large thermal gradient over the heat source and sink. Specifically heat pipes with water as fluid are extensively used in Earth applications, low cost, and unrivaled in terms of performance. Although in larger satellites water heat pipes have often been disregarded because of their dis-functionality below 0 °C, this property might be turned into an advantage if the heat pipe acts as a heat switch.
This thesis is centered around the possibility of employing water heat pipes in CubeSats, with the focus on performance characterization and integration into the CubeSat platform.

1.2. Research Question and Scope of the Project
The application of this thermal control system to CubeSats needs to be thoroughly investigated. To guide this process the following research question is posed:

*Can commercial water heat pipes solve the thermal challenge of high performance CubeSat missions?*

This research question is broken down into several sub-questions that will aid in the search for an answer. The first set of sub-questions explores the theoretical functionality of heat pipes and the design that is necessary to cope with the projected heat loads in the next-generation CubeSats.

1. What heat pipe is needed for CubeSat thermal control?
   1.1 What are the limits for the heat pipe performance (max heat flux, heat transport) as a function of temperature?
   1.2 How do geometry, material and fluid affect these limits?
   1.3 What heat pipe design is required to solve for the expected heat loads?
1.4 Is there a commercially available heat pipe that fits the desired heat pipe design?

The second set of sub-questions deals with the experimental performance of commercial water heat pipes under various test conditions.

2 What is the experimental performance of commercially available heat pipes?
   2.1 How does the performance scale with heat pipe temperature?
   2.2 What is the heat transfer coefficient at the heat pipe hot- and cold interfaces?

3 What is the transient start up behavior of water heat pipes around freezing point?
   3.1 At what temperature does the heat pipe starts functioning?
   3.2 What is the start-up time of a heat pipe when frozen?

4 What is the effect of freeze/thaw cycling on a heat pipe?
   4.1 Does repetitive freeze/thaw cycling lead to external physical damage?
   4.2 How does freezing of the liquid impact the wick structure?
   4.3 What is the effect of freeze/thaw cycling on heat pipe performance?

5 Can heat pipe freezing be beneficial for CubeSat thermal control?

Another aspect that is investigated is the integration of a heat pipe into the CubeSat platform, which is captured by the following set of sub-questions:

6 Are heat pipes suitable for use in CubeSats?
   6.1 Can a heat pipe deal with the projected heat loads?
   6.2 How can a heat pipe be integrated into the CubeSat platform?
   6.3 What is the impact of interfaces on the thermal performance of heat pipes?

The last sub-question concerns with the effect water heat pipes have when employed in CubeSats on a possible mission.

7 What are the critical aspects for a CubeSat mission when employing heat pipes?

The scope of the project is captured by the entire set of sub-questions. The first two sets of sub-questions make up for the larger part and will be the prime focus of this project. The last set of sub-questions will be important as well, but the focus will lie on the proof of concept of the heat pipe in a CubeSat rather than trying to come up with the best design as this is a study in itself.

1.3. RESEARCH METHODOLOGY

In order to reach satisfying conclusions that answer the research question posed this project employs the following research methodology:

- Literature Review
  The thesis project starts off by carrying out an extensive literature study to obtain sufficient knowledge on the subject. Research papers and books will be consulted and reviewed and personal contact with people from industry will be sought. Sources consulted will be documented along with proper references and descriptions.

- Theoretical Approach and Analyses
  Theoretical analyses will be performed to understand the heat pipe's operating principle. Basic theory will be extracted from books and derivations will be made by hand and with the help of software tools, such as Matlab. To model the thermal behavior of a CubeSat ESATAN ThermXL is employed, a plugin for Microsoft Excel which allows computation of transient and steady-state analyses. Finally, 3d-designs and technical drawings will be made with the modeling software CATIA.
Experiments and Data Collection
In order to validate the found theoretical results experiments will be carried out. The data resulting from these will be used for comparison and validation of the theoretical values. The software program LabView will be used to run the heat pipe tests and collect the data. Furthermore, heat pipe integration tests will make use of a thermal chamber.

Data analyses
The data, stored in TDM or CSV files, will be processed by Matlab and theoretical and experimental data will be numerically and visually compared, with the help of tables and graphical plots.

1.4. REPORT OUTLINE
This report is divided into multiple chapters: Chapter 2 discusses the problem given in detail by providing an overview of the trends observed and expected in the near future in the CubeSat market, while Chapter 3 discusses thermal control theory basics and provides numerical analyses to support the earlier made claims about thermal challenges. Then, Chapter 4 explores the physics and mathematics behind the operating principle of heat pipes in depth, after which Chapter 5 reports on considerations necessary to take into account when employing heat pipes, such as fluid selection, geometry, and wick types. The availability of suitable heat pipes for CubeSats is explored in Chapter 6. Finally, theoretical analyses are carried out in Chapter 7 and the integration of a heat pipe in a CubeSat is discussed in Chapter 8. The tests are carried out in Chapters 9 and 10, which describe and discuss the performance characterization of heat pipes and their functionality when integrated in the CubeSat structure, respectively. The report is then finalized by the chapters Conclusions, 11 and Recommendations, 12.
THERMAL CHALLENGES IN HIGH PERFORMANCE CUBESAT MISSIONS

In the past two decades the satellite market has rigorously changed: The interest in small satellites has taken a flight and has led to many new technological developments in this field. In turn, this has led to a continuous growth of the market and a persisting demand for more performance. From this, new challenges and problems have arisen that need solving to continue the development of more powerful small satellites.

The growth and evolution of the small satellite market in the past decade is discussed in Section 2.1. Hereafter, Sections 2.2 and 2.3 discuss the thermal challenges that lie ahead and the requirements that are being posed on a thermal control system.

2.1. TRENDS IN THE CUBESAT MARKET AND TECHNOLOGY

The evolution of the nanosatellite market in the last decade finds its origin in the introduction of the CubeSat standard [1]. This standard, being proposed as a platform for educational and low-cost space experimentation for the industry, was quickly adopted and its growing popularity is reflected in the increasingly yearly launch rates since its introduction [2]. This trend has gone hand in hand with the trend of miniaturization, which has enabled the development of miniaturized subsystems thereby enhancing the capabilities of CubeSats [3].

Currently the CubeSat platform has surpassed its original purpose and is slowly winning ground over larger satellites by providing similar functionalities. Their potential has been recognized by the industry and the demand for more high performance missions is increasing. This is supported by not only the current mission proposals arising at the Dutch CubeSat company Innovative Solutions In Space (ISIS), but also by the forecasts made for the future market [4–6]. These forecasts predict an increase in CubeSats to be launched over the next few years, an increase in CubeSat form factor size, and a shift in primary mission objective from education and technology demonstration towards Earth observation (EO).

The interest of the industry and the shift in mission objectives runs parallel with the advances in the ongoing miniaturization trend and the development of deployable solar panels which have paved the way for the employment of more powerful instruments, systems, and components. Extensive literature review has revealed that the power level of CubeSats is increasing rapidly and that this trend does not adhere to the mass versus power trend lines established in the past, which is shown in Figure 2.1 [7].

The figure shows the peak power values for different CubeSat form factors. From these and other examined missions it has been derived that peak power levels will rise to values of approximately 20 to 40 W. While a shift is visible towards larger CubeSats, such as 6U and 12U, the 3U will remain popular in the next few years, especially for constellations. With these power levels it means that power density will increase significantly.

2.2. THERMAL CHALLENGES OF NEXT GENERATION CUBESATS

The thermal challenges of CubeSats have three main causes each leading to a different thermal problem: A low thermal mass, limited surface area, and a high density. The first makes the satellite rapidly responsive to thermal fluctuations as the heat capacity is limited. This leads to thermal cyclic loading, which can be
The second cause, a limited surface area, leads to problems when heat loads increase beyond the radiating capability of the outer panels. In that case the satellite will not be able to remove all its excess heat and continue to heat up.

The third and last property of the CubeSat platform can result in local hot-spots when heat load levels increase on, for example, electronic chips. With high heat loads on a small area and insufficient conduction paths the local temperature will quickly rise to levels beyond the limit.

While all three causes lead to significant thermal problems it is expected that the increase in power density will be most critical: When looking at the shift in mission objectives towards EO it is expected that the payload electronics, transmitter and receiver systems, and batteries become the most critical components from a thermal point of view [7]. High performance EO missions will need dedicated instruments that are able to collect a large amount of scientific data. For down-linking all these data high data rates are necessary, which in turns requires a lot of (stored) power. The thermal load will hereby lie on the level of an electronic chip, such as the amplifier of a transmitter. For high power densities the temperature will quickly rise to levels beyond the operating limit of the electronics due to the low conducting printed circuit board (PCB) material. This is highlighted by Figure 2.2 which shows the thermal gradient and temperatures that will arise at steady-state conditions on a PCB with a heat load of 10 W applied at the center and a 15 °C boundary temperature at the four corners.

Up till now thermal control solutions on CubeSats have mostly been limited to the usage of simple heaters, thermal tapes and coatings [6]. As power levels have been relatively low these solutions were sufficient to keep the subsystems and components within operating temperature limits. For the increasing demand for high performance CubeSat missions, which simultaneously lead to an increase in power density, these methods fall short in maintaining suitable temperature ranges, therefore different solutions are required.

2.3. THERMAL REQUIREMENTS AND PROPOSED SOLUTION

Based on the expected power levels of CubeSats and the steady-state temperature of a PCB with these heat loads, it becomes apparent that a solution is necessary to reduce the temperature extremes attained. From literature review it has been found that the heat pipe is the prime candidate for solving this problem [7].

\[ \text{Thermal gradient is modeled with an along-plane conductivity of } 35 \text{ W m}^{-1} \text{ K}^{-1} \]
2.3. Thermal Requirements and Proposed Solution

A heat pipe is a so-called two-phase system in which a fluid, such as water or methanol, exists in both liquid and gas form. It consists of an evaporator section, at which heat is applied and a condenser section at which heat is removed from the pipe [9]. A heat pipe is able to transport a large amount of heat from one point to another without requiring a large temperature gradient between these two points, contrary to simple solid conduction paths.

The question that now arises: What requirements are being posed on the heat pipe as thermal control solution? For the problem at hand several requirements have been derived to which the thermal control system must comply [7]. These have been listed in Table 2.1.

The thermal control requirements have been found from reviewing the power budget of existing and announced CubeSat missions. The expected heat loads are mentioned in requirements 1.2.1.1 and 1.2.1.2. The requirements on structure and safety have all been derived from the constrains of the CubeSat standard. This standard limits, among others, the total available mass and volume, allowed on-board fluids, and used materials.

Figure 2.2: Steady-state temperature distribution for a standard CubeSat PCB with a 10 W centered heat load.
### 2. THERMAL CHALLENGES IN HIGH PERFORMANCE CUBESAT MISSIONS

#### Table 2.1: Requirements for thermal control.

<table>
<thead>
<tr>
<th>Category</th>
<th>Requirement</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal</td>
<td>The TCS shall keep the satellite and its components within survival temperature ranges.</td>
<td>Mission and subsystems defined.</td>
</tr>
<tr>
<td></td>
<td>The TCS shall keep the designated component within its survival operating limits.</td>
<td>Component: transmitter or processor.</td>
</tr>
<tr>
<td></td>
<td>The TCS shall keep the designated component within its predefined operating limits.</td>
<td>Component: transmitter or processor.</td>
</tr>
<tr>
<td></td>
<td>The TCS shall be able to transport excess heat from source to sink.</td>
<td>4.1.1</td>
</tr>
<tr>
<td></td>
<td>The TCS shall be able to remove 20 W peak heat dissipation.</td>
<td>4.1.2.2</td>
</tr>
<tr>
<td></td>
<td>The TCS shall be able to remove 10 W of other exceed heat.</td>
<td>4.1.2.1</td>
</tr>
<tr>
<td></td>
<td>The TCS shall not contain hazardous materials.</td>
<td>4</td>
</tr>
<tr>
<td>Structure</td>
<td>The TCS shall adhere to the CubeSat dimensional constrain 10x10x30 cm (lxwxh).</td>
<td>Ref. [10]</td>
</tr>
<tr>
<td></td>
<td>The TCS shall adhere to the mass restriction of a 3U CubeSat 3.9 Kg.</td>
<td>3.2</td>
</tr>
<tr>
<td>Safety</td>
<td>The TCS shall not pose risk to the satellite mission.</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>The TCS shall not pose risk to the present components of subsystems, both physically and functionally.</td>
<td>3.3.1.1</td>
</tr>
<tr>
<td></td>
<td>The TCS shall not interfere with the present structural elements.</td>
<td>3.3.2</td>
</tr>
</tbody>
</table>

---

2. THERMAL CHALLENGES IN HIGH PERFORMANCE CUBESAT MISSIONS

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Table 2.1: Requirements for thermal control.
Thermal Control in the Space Environment

The environment in low-Earth orbit (LEO) places a heavy burden on the satellite from a thermal point of view. Heat fluxes impinging on various surfaces and thermal cyclic loading necessitate delicate measures to ensure continuous operation of a satellite. Therefore, a thorough understanding of heat transfer theory and identification of critical thermal aspects in a CubeSat in this altitude range is vital. Section 3.1 will discuss the different mechanisms that are crucial for heat transfer, while Section 3.2 will discuss the thermal environment in LEO. Last, Section 3.3 will present the first analyses on the temperature behavior of a component under the applied heat load.

3.1. Heat Transfer Mechanisms

Heat transfer between two nodes can occur in three ways: convection, conduction, and radiation. Which mechanism comes into play depends on the heat path between the nodes under consideration.

3.1.1. Convective Heat Transfer

Convective heat transfer occurs between nodes that have a movable fluid as heat load path between them. When a fluid is heated the internal molecules will vibrate and collide, transferring heat from one place to the other. The heat that can be transferred is dependent on the convective heat transfer coefficient \( h_c \) and the heat transfer surface area. The total heat that can be transferred via convection is given by Equation (3.1).

\[
Q = h_c A \Delta T
\]  

(3.1)

In space this type of heat transfer is only of importance for spacecraft that carry fluids along with them or in propulsion systems. In LEO, especially at low orbital altitudes the atmosphere is still of some influence and convection may take place between it and the spacecraft. However, this effect is only small and hardly occurs at altitudes higher than 300 km and can therefore be neglected during analyses.

3.1.2. Conductive Heat Transfer

Conductive heat transfer occurs between nodes with a solid heat path between them. The addition of heat to a solid makes the internal molecules of the solid vibrate thereby passing on the heat. The total heat that can be transferred is dependent on the surface properties of the solid and heat path dimensions. The conductive transfer between two surfaces is given by Equation (3.2).

\[
Q_{ij} = C_{ij} (T_i - T_j)
\]  

(3.2)

The term \( C_{ij} \) is the conductive coupling between surface i and j. This term can be calculated by using Equation (3.3), which is a function of the thermal conductivity \( k \), the heat path cross-sectional area, and length [11].

\[
C_{ij} = \frac{kA}{T}
\]  

(3.3)
Mathematically, for different surfaces or material properties, heat transfer via conduction is analog to the series- and parallel-equations applicable to electric circuits [12]. For example, the total conductive coupling of multiple conductive paths in series is given by Equation (3.4).

$$\frac{1}{C_{ij}} = \left( \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + ... \right) \quad (3.4)$$

For parallel conductance simple addition of conductive couplings to find the total conductance applies. The contact between two surfaces is never ideal and heat loss will occur at this site. Therefore, an additional coupling has to be incorporated that accounts for this heat loss, which is the so-called contact conductance. This conductance is dependent on a variety of parameters, such as surface roughness, hardness, contact pressure, and fill medium, which may differ in each situation, making it hard to assign a value to it [13].

### 3.1.3. Radiative Heat Transfer

The third method of heat transfer occurs via radiation. Any heat source radiates energy in the form of radiation, its strength dependent on the heat source's temperature. The total amount of energy radiated by a heat source is a function of the Stefan-Boltzmann constant $\sigma_b$ and the temperature to the power fourth, as shown by Equation (3.5).

$$E = \sigma_b T^4 \quad (3.5)$$

The amount of energy radiated by a heat source and incident on a surface dictated by the temperature difference between the two objects and the line of sight. This line of sight, or view is characterized by the view factor, $F_{ij}$, which is a dimensionless factor dependent on the geometry of both objects [11]. This factor can be calculated with the help of Equation (3.6).

$$F_{ij} = \frac{\text{Radiation emitted by surface } i \text{ and incident on surface } j}{\text{Total radiation emitted by surface } i} \quad (3.6)$$

The difficulty in determining this factor lies in the fact that one has to trace back the path of each individual ray from an infinite set of rays. Therefore, simplified equations exist for view factors for specific problems. As an example, heat fluxes incident from the Earth on a surface in LEO can be represented as shown in Figure 3.1.

![Figure 3.1: Simplified situation for view factor calculation for a surface in LEO [14].](image)

As shown, the view factor is a function of the radius of the Earth and the satellite's altitude and attitude. The view factor is then given by equation 3.7.

$$F_{dA_1-dA_2} = \frac{1}{\left(1 + \frac{h}{R_E}\right)^2} \cos \theta \quad (3.7)$$

For a satellite orbiting Earth at an altitude of 600 km and nadir-fixed attitude ($\theta = 0$) the value is 0.84. The other view factors that are present are the view factor between the satellite and the Sun and deep space.
3.2. THE THERMAL ENVIRONMENT IN LOW EARTH ORBIT

In both these cases the view factor is taken equal to 1 as it is assumed that for the Sun the solar rays are arriving parallel at every satellite’s surface due to the large distance to the Sun. For the coupling with deep space it is assumed that the satellite always has an unobstructed view towards it.

The total amount of energy taken up by a spacecraft does not only depend on the strength of the incident heat fluxes and line of sight between two objects: Surface properties have a large influence on this value as well. Each surface has a specific emissivity and absorptivity, which represent the fraction of energy emitted or absorbed, compared to that of a black body. The emissivity is expressed by the parameter \( \varepsilon \), which, for convention, is taken as both the emissivity and absorptivity in the infrared spectrum. The \( \alpha \) is the absorptivity in the visible wavelength and therefore also called the solar absorptance.

With the view factor explained the radiative coupling between two surfaces is then given by Equation (3.8).

\[
Q_{ij} = A_i F_{v,ij} \varepsilon \sigma_b (T_i^4 - T_j^4)
\]  

(3.8)

Although in space the same heat transfer mechanisms apply as on Earth, the role of each mechanism differs: For a satellite’s internal heat transfer is dictated by conduction, while externally radiative heat transfer dominates.

In LEO two distinct heat sources are present that radiate heat towards a spacecraft: the Sun and the Earth. The heat flux incident on the satellite’s surface from the Sun is strongest, whose value fluctuates along the year with a maximum and minimum around January and July, respectively. This heat flux is, due to the Sun’s temperature, strongest in the visible wavelength range [12]. Solar flux finds its way to the satellite in two ways: directly or by reflection from the Earth. The latter is called albedo whose value is dependent on surface properties. For Earth the average albedo factor lies in the order of 0.3 [11].

The Earth has a substantially lower temperature than the Sun and emits its radiation in the infrared range [11]. This heat flux is present along the entire orbit of the satellite.

Common values for heat fluxes present in LEO for the different heat sources discussed are shown in Table 3.1.

<table>
<thead>
<tr>
<th>Heat Source</th>
<th>Heat Flux ([\text{Wm}^{-2}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar flux</td>
<td>1366</td>
</tr>
<tr>
<td>Albedo</td>
<td>410</td>
</tr>
<tr>
<td>Earth infrared</td>
<td>237</td>
</tr>
</tbody>
</table>

The complete thermal environment of a satellite in LEO is shown schematically in Figure 3.2.

Figure 3.2: The thermal environment of a satellite in LEO [13].
3.3. PRELIMINARY THERMAL ANALYSIS OF A HIGH PERFORMANCE MISSION

In order to get a feeling for the temperatures attained by CubeSats in LEO it is interesting to look at their temperature profiles from several orbits. A tool has been developed that enables calculation of orbital temperatures along any chosen orbit. The tool incorporates a thermal model of a 2U CubeSat that allows computation of a first order thermal analysis and is validated with in-orbit data. It can be used to illustrate the temperature levels that will be attained both with and without the aforementioned, expected heat dissipation loads.

3.3.1. ORBITAL TEMPERATURE PROFILES OF A LEO CUBESAT

The temperature profile of a satellite in LEO is dependent on its orbital parameters and its attitude. The orbital parameter that plays a crucial role here is the solar beta angle, which is the angle between the Sun vector and the orbital plane. The picture at the top of Figure 3.3 shows how this angle is defined. The bottom two pictures are drawn from the perspective of the Sun and show what an orbit looks like when the solar beta angle is 0° and 90°, respectively.

An orbit with a solar beta angle of 0° is a so-called noon/midnight orbit as the satellite crosses the subsolar point and experiences the longest eclipse time. The opposite is a LEO with a solar beta angle of ± 90°. This orbit is a dawn/dusk orbit in which the satellite flies along the day/night terminator. In this orbit the satellite continuously receives sunlight on one of the satellite’s sides while the opposite side is continuously facing deep space.

The heat fluxes incident on the satellite’s external structure are dependent on the orientation of the orbit, which is thus dependent on the solar beta angle. Figure 3.4 shows how the strength of the heat fluxes change as well as the eclipse time as a function of this angle. For increasing solar beta angles the eclipse time drops and the satellite spends more time in sunlit conditions, therefore the orbit average solar and albedo fluxes increase [9].

Instead of the solar beta angle an orbit can also be characterized by the local time at the ascending node (LTAN). A LEO with a solar beta angle of 0° would correspond to an LTAN of 12:00 h as the longitudinal ascending node lies around noon. A ± 90° orbit has an LTAN of 06:00 or 18:00 h depending on whether the solar beta angle is positive or negative, which implies a different flight direction of the satellite.

As shown the orbit of a satellite affects the heat inputs on a LEO satellite and therefore dictates its temperature profile. However, the profile itself is also dependent on the attitude of the satellite: For a Sun-pointing attitude one can imagine that the surface facing the Sun will attain a higher temperature than the surfaces facing deep space. Similar, the surfaces of a rapidly spinning satellite will see heat sources and deep space alternating in its view leading to a smoothed overall temperature profile. For these reasons it is important to know the attitude and orbital parameters as the temperature profile of the satellite’s outer panels are directly related to these.

1This is only true for a 90° orbit at one of the equinoxes. Due to the tilt in axis of rotation of the Earth the beta angle will differ over time and season and inclination will thus affect this.
3.3. PRELIMINARY THERMAL ANALYSIS OF A HIGH PERFORMANCE MISSION

Figure 3.4: Heat fluxes and eclipse duration as a function of solar beta angle for a cylinder in LEO [9].

For the computation of the temperature profile two different attitudes have been chosen: nadir-fixed and Y-Thomson mode. In the nadir-fixed attitude the satellite always points one and the same point or surface towards the Earth. The Y-Thomson mode is here defined as a constant spin (-2.2 pitch rate) around the satellite’s Y-body axis. The resulting external temperature profiles for the two different attitudes defined and for several LTAN orbits have been computed. The two extreme cases, LTAN 06:00 and 12:00 h are shown in Figures 3.5 and 3.6, respectively. The complete set of figures along with the temperature profiles for a 09:00 h LTAN orbit can be found in Appendix B.

The dawn/dusk orbit in nadir-fixed mode shows what temperatures will be attained by the different surfaces of the CubeSat. It is clear and logical that the panel orientated towards the Sun (-Y) attains the highest temperature of approximately 32°C, while the opposite panel is coldest as it is pointed continuously towards deep space. Panel +Z is the nadir-fixed panel and therefore experiences the highest Earth infrared flux and at one point along the orbit a small part of the solar heat flux. The Y-Thomson mode shows a picture not very different from the nadir-fixed mode except for the fact that, due the spin, each of the panels in flight direction are alternating between Earth and deep space view. Therefore, their temperatures are close together and show that the temperatures of each of the panels averages out.

The noon/midnight orbit shows a very different picture from the dawn/dusk orbit as now the temperature of the different panels fluctuates between two extreme values. This is due to the eclipse and sunlit period of the orbit. The panel directed towards the Sun sees a sudden increase in temperature. From the graph it can be derived that panel +X is pointed along the flight direction vector and will first see the Sun when the satellite exits eclipse. A bit further surface -Z faces the Sun before -X takes over and spends the last part in sunlit conditions before the satellite enters eclipse again. In eclipse the CubeSat cools down to approximately -30°C.

For a noon-midnight orbit the temperatures of the different panels are quite different and induce a large thermal gradient over the satellite’s external. For the Y-Thomson attitude mode this is quite different as the temperatures of all the panels remain close to each other over the entire orbit, except for the Y-panels which are facing deep space.

3.3.2. INTERNAL TEMPERATURE PROFILES OF A CUBESAT

The dependency on the orbital parameters and satellite attitude is evident for the external temperature profiles. Internally, the temperature profile is also dependent on the conduction paths between the internal components and the outer structure. The same calculations have been carried out to investigate the temperatures attained by the internal components for the same different orbits and attitudes. The results are shown
in Figures 3.7 and 3.8, respectively and again in Appendix B.

Figure 3.7 shows a similar profile as the outer structure (Figure 3.5). Due to the fact that the temperatures of the outer panels remain fairly constant over the orbit, no rapid fluctuations in temperature occur in the subsystem temperatures as well.

The internal temperature profile for a 12:00 h orbit, shown in Figure 3.8 shows severe fluctuations in subsystem temperature similar to what is experienced by the outer structure. More interesting to note, however, is the fact that satellite attitude hardly has any effect on the internal temperature profile: for both the nadir-fixed and Y-Thomson mode the internal temperatures are almost identical.

3.3.3. Temperature Profiles of the Next Generation CubeSat
The last scenario that needs to be analyzed is the one where a heat load is applied that is expected for near-future CubeSats. Requirement 1.2.1.1 states an internal heat load dissipation of 10 W orbit average. Again, as before, the same analyses have been performed but now with an orbit average internal heat load dissipation of 10 W located on the payload subsystem node.

The external temperature profiles for the different orbits and attitudes (shown in Appendix B) show an overall increase in temperature of 20 °C. This is the effect of the increased internal heat dissipation that conducts towards the outer structure from where it is radiated away. This overall distributed increase can be explained by the fact that the CubeSat's internal and external structure are symmetric in both directions.
3.3. PRELIMINARY THERMAL ANALYSIS OF A HIGH PERFORMANCE MISSION

The internal temperatures are again independent of satellite attitude. The profiles for the same orbits as before are visualized in Figure 3.9 and show that in both orbits the subsystems attain roughly the same
steady-state temperature. More interesting to see is that the thermal cycle becomes less apparent for the noon/midnight orbit due to the large amount of heat dissipated. Most evident of course is the high temperature that is reached by the payload subsystem.

It has to be noted however, that the model does not incorporate internal view factors. While this is safe to assume for low levels of heat dissipation, as thermal gradients will be relatively low, this is not true anymore when the heat load increases leading to significant thermal gradients. Internal view factors lead to radiative heat transfers between the PCBs themselves and the PCBs and the external panels. The effect of including these radiative couplings is that the temperature of the payload PCB will be slightly lower than shown in the graphs for a 10 W heat load, while the PCBs located directly beneath and above the payload PCB will have a higher temperature. The external panels, experience little influence of the additional radiative couplings.

The analyses show the problem at hand: temperature rises quickly and leads to a local thermal hotspot on the payload subsystem that cannot be solved by the existing conduction paths. Because of this new thermal control methods or systems are necessary and the heat pipe is proposed as solution.
Flow Theory and Heat Pipe Operation

Fluid interaction and behavior form the basic principles of heat pipe operation. It is therefore vital to understand the mechanics of fluid flow, which eventually determines the heat pipe’s performance. This chapter will explain the basic principles of fluid mechanics. In Section 4.1 the phenomenon of surface tension is addressed. Then in Section 4.2 fluid flow in heat pipes, required for operation, is mathematically derived and explained. Hereafter, in Section 4.3 the different limits are described that constrain the heat pipe’s performance.

4.1. Liquid Behavior Due to Surface Tension

The operating principle of a heat pipe is based on capillary forces arising from liquid surface energy. Molecules inside a liquid are all attracted by each other due to molecular forces. Molecules situated at the surface edges will experience a net force inward due to the inner molecules and therefore a free floating liquid will form a round droplet.

When a liquid comes into contact with a surface the molecules at the edge of the liquid interact with the molecules located at the solid surface. The strength of the molecular forces determines whether there is an attractive or repulsive force. When the solid molecules have a greater attractive force than the attractive force between the molecules of the liquid, the liquid is wetting the solid. When the solid molecules exert a repulsive force, the liquid is non-wetting the solid. Both situations are depicted in Figure 4.1a.

![Figure 4.1: The principle of wetting (a) and capillary force due to wetting and non-wetting (b).](image)

Figure 4.1b shows how a wetting or non-wetting liquid behaves in a pipe. For a neutral fluid (e.g. not wetting or non-wetting) the forces between molecules inside the liquid are equal and the surface remains horizontal. For a wetting material, the surface molecules of the liquid are attracted by the attraction of the molecules from the solid surface. Due to this attraction the liquid crawls up at the wall of the pipe. The surface forms a concave and the inter molecular attractive force is now orientated tangential to the liquid’s surface. Decomposing this tangential force into a horizontal and vertical component results in a vertical net force upwards, as the horizontal forces are canceled out by each other. The fluid draws upwards and a pressure difference is created between the concave and convex side of the fluid surface.
Exactly the opposite is true for a non-wetting situation in which the fluid is withdrawn due to repulsive forces between the liquid and solid.

4.2. Fluid Mechanics in Heat Pipes

The next step is to analyze the liquid and vapor flows inside a heat pipe. A schematic displaying the basic elements and working principle of a heat pipe is shown in Figure 4.2. The heat pipe consists of three sections: an evaporator, condenser, and adiabatic section. When the evaporator section is subjected to a heat input, the present liquid will evaporate. This, along with the wetting of the wick structure by the liquid, leads to a depression of the remaining liquid and a decrease in pressure. The evaporation of the liquid simultaneously leads to an increase in vapor pressure in the evaporator section.

![Figure 4.2: Schematic of the heat pipe elements and basic operating principle [9].](image)

In the condenser section, both vapor and liquid have remained in equilibrium and no pressure difference has occurred in this part. However, as the vapor pressure is higher in the evaporator area the vapor will flow towards the lower pressure area in the condenser section. At the same time, the liquid pressure in the condenser area is higher and will therefore flow towards the evaporator area, due to capillary action. This entire process complete the heat pipe’s operating cycle.

In order for the heat pipe to work the pressure difference or gradient arising between the condenser and evaporator section needs to be higher than the pressure losses present during operation, according to Equation (4.1). These pressure losses are the liquid pressure drop, $\Delta p_l$, vapor pressure loss, $\Delta p_v$, and the pressure loss due to gravity, $\Delta p_g$, which is present when the heat pipe operates under an angle. Each of these pressure terms will be discussed separately in the sections hereafter.

$$\Delta p_c \geq \Delta p_l + \Delta p_v + \Delta p_g \quad (4.1)$$

4.2.1. Capillary Pressure

The capillary pressure is the driving force behind the heat pipe operating principle. The capillary force draws the liquid from the condenser section towards the evaporator section where it evaporates and flows back to the condenser section, completing the cycle.

The capillary pressure can be derived from the forces acting on a curved liquid surface. In order to remain in equilibrium the pressure difference between the convex and concave surfaces must balance the surface tension force. This situation is depicted in Figure 4.3.

![Figure 4.3: Balance between the pressure difference and surface tension on a curved liquid surface [15].](image)

The mathematical description of this situation is given by Equation (4.2), where $\Delta p_c$ is the pressure difference between $p_1$ and $p_2$. Solving for $\Delta p_c$, shows that the capillary pressure is only a function of the liquid...
surface tension and the radius of curvature.

\[ 2\pi R \sigma_1 = \Delta p_c \pi R^2 \] \hspace{1cm} (4.2)
\[ \Delta p_c = \frac{2\sigma_1}{R} \]

Figure 4.4 shows a cross-section of one groove of the heat pipe. From this the relation shown in Equation (4.3) can be established between the radius of curvature, the effective capillary radius \( r_c \), and the contact angle \( \theta \).

\[ \cos(\theta) = \frac{r_c}{r} \] \hspace{1cm} (4.3)

Substituting Equation (4.3) into (4.2) results in (4.4). This is the general equation for the calculation of capillary pressure.

\[ \Delta p_c = \frac{2\sigma_1}{r_c} \cos(\theta) \] \hspace{1cm} (4.4)

During steady-state operation the radius of curvature is different for the evaporator and condenser section. This is shown in Figure 4.5, where it can be seen that the surface tension is different at the evaporator section and the condenser section due to evaporation of the liquid and condensation of the vapor, respectively. Therefore a different relation is established for the total capillary pressure. This is shown by Equation (4.5).

\[ \Delta p_c = 2\sigma_1 \left\{ \frac{\cos(\theta_e)}{r_{c,e}} - \frac{\cos(\theta_c)}{r_{c,c}} \right\} \] \hspace{1cm} (4.5)

Figure 4.5: Interface between the liquid and vapor inside a heat pipe during steady-state operation [16].
To obtain the maximum capillary pressure of a heat pipe operating at steady-state Equation (4.5) can be simplified by using the following two assumptions:

i At the condenser section the radius of curvature approaches infinity.

ii At the evaporator section the contact-angle approaches unity.

Applying these assumptions to Equation (4.5) leads to:

\[ \Delta p_{c,m} = \frac{2\sigma_1}{r_{ce}} \]  

(4.6)

This relation shows that in order to achieve a high capillary pressure the liquid surface tension must be high, while the capillary radius must be small.

4.2.2. Liquid Pressure Drop

One of the pressure drops that needs to be overcome by the capillary pressure is the liquid pressure drop. The liquid flowing through the wick experiences a friction force at the interface with the solid heat pipe wall, whose value also depends on the viscosity of the liquid. This friction leads to a pressure drop decreasing the capillary flow through the wick.

The liquid pressure drop can be calculated using the Hagen-Poiseuille Equation. This equation, which is derived in Appendix C, describes the pressure drop over a cylindrical pipe as a function of its cross-sectional area through which the fluid flows and the square of the pipe’s radius.

\[ \Delta p = \frac{8\mu l\bar{m}}{\rho \pi r^4} \]  

(4.7)

For the liquid pressure drop, however, the cross-sectional area and radius are described by the geometry of the wick structure. For simplicity, an axial grooved pipe is assumed. For this type of wick structure, the cross-sectional area, \( A_w \) is defined by the dimensions of the wick. The radius is determined by the cross-sectional area of the wick divided by the wetted perimeter, called the hydraulic radius \( r_{h,l} \). For cylindrical grooves this radius is simply equal to the radius of the groove itself, while for rectangular grooves, with groove width \( \omega \) and depth \( \delta \), this radius can be derived to:

\[ r_{h,l} = \frac{\omega \delta}{(\omega + 2\delta)} \]  

(4.8)

Furthermore, the total mass flow \( \bar{m} \) is given by the summation of the mass flows through the individual grooves and hence, the total pressure drop is a function of the number of grooves \( N \).

The length of the pipe for has been written in terms of an effective length. This is because the mass flow at both ends of the pipe will differ due to acceleration and deceleration of the flow. This effective length is given by:

\[ l_{eff} = l_a + \frac{l_e + l_c}{2} \]  

(4.9)

Finally, substituting these aforementioned relations into Equation (4.7) leads to the final relation for the liquid pressure drop for rectangular axial grooves where the subscript \( l \) denotes the properties of the liquid.

\[ \Delta p_l = \frac{8\mu_l l_{eff}\bar{m}}{\rho_l N A_w r_{h,l}^2} \]  

(4.10)

This equation shows that the pressure drop for axial grooves is relatively easy to calculate: all parameters can be accurately measured or determined. For other wick structures, such as mesh or sintered wicks this is somewhat more difficult and the Darcy-Weisbach Equation is used (for the derivation see Appendix C):

\[ \Delta p_l = \frac{f_D \rho \mu_l l_{eff}\bar{m}}{8 \epsilon \rho_l A r^2} \]  

(4.11)

For these types of wicks an additional factor has come into play, which is the wick porosity \( \epsilon \). This is a factor that represents the “easiness” with which the liquid can flow through the porous wick. The wick
porosity is difficult to determine a priori and therefore experimental verification is required [15]. Another parameter that is now introduced is the wick permeability $K$. This parameter is a function of the Darcy's friction factor, the Reynolds number, the hydraulic radius, wick cross-sectional area, and the wick porosity.

$$K = \frac{8c_{f,b}^2}{f_D Re}$$ (4.12)

Substituting this into Equation (4.11) leads to the liquid pressure drop for pipes with a porous wick structure:

$$\Delta p_l = \frac{\mu_l l_{ef} \dot{m}}{\rho_l K A_w}$$ (4.13)

### 4.2.3. Vapor Pressure Drop

Next to the liquid pressure drop the vapor induces several losses along the length of the heat pipe as well. The vapor pressure loss $\Delta p_v$ can be written as the sum of the pressure losses occurring at the evaporator section, $\Delta p_{v,e}$ the condenser section, $\Delta p_{v,c}$ and the adiabatic section $\Delta p_{v,a}$ (Equation (4.14)).

$$\Delta p_v = \Delta p_{v,e} + \Delta p_{v,a} + \Delta p_{v,c}$$ (4.14)

The vapor pressure drop in the evaporator, condenser, adiabatic section are described by the Hagen-Poiseuille relation as well (Equation (4.15)). The only parameter that changes for these three sections is the length.

$$\Delta p_v = \frac{8\mu_v l_{ef} \dot{m}}{\rho_v \pi r_{h,v}^4}$$ (4.15)

However, for the condenser and evaporator section an additional term comes into play which accounts for the pressure gradient required to accelerate and decelerate the vapor flow to and from axial velocity. This term, also called the inertial term, is given in Equation (4.16). For simple analysis it may be assumed that these terms cancel each other out due to full pressure recovery [15]. If this is not the case, then this term has to be added to Equation (4.14).

$$\Delta p'_v = \rho v^2$$ (4.16)

The total vapor pressure drop is then found by the substitution of all these relations into the earlier stated Equation (4.14), leading to the following equation for laminar flow, with $A_v$ being the vapor cross-sectional area:

$$\Delta p_v = \frac{8\mu_v l_{ef} \dot{m}}{A_v r_{h,v}^2 \rho_v}$$ (4.17)

### 4.2.4. Gravitational Head

Another pressure drop that can occur during heat pipe operation is the pressure loss due to the weight of the fluid under the influence of gravity. The body force acting on the liquid due to gravity is dependent on the orientation of the heat pipe and can have severe influence on its performance. When the evaporator section is located above the condenser section the returning liquid has to work against the gravity resulting in a pressure loss. However, the opposite is true when the condenser section's relative position is above the evaporator section. In this case the gravity can be used to an advantage and the gravity pressure drop will have a negative value. The general relation for the pressure drop due to gravity is shown in Equation (4.18).

$$\Delta p_g = \rho g h$$ (4.18)

This can be rewritten and related to the orientation and total length of the heat pipe. Further, the gravity force acting on the fluid can be decomposed into two forces: one axial force and a normal or radial force
which leads to the relations for the axial and radial pressure drop as shown in Equations (4.18) and (4.20), respectively.

\[ \Delta p_{g,a} = \rho_l g l \sin(\phi) \]  \hspace{1cm} (4.19)

\[ \Delta p_{g,r} = \rho_l g d \cos(\phi) \]  \hspace{1cm} (4.20)

For a 25 cm length water-heat pipe with a 8 mm diameter the gravitational pressure drop has been calculated as a function of \( \phi \) (positive) in Figure 4.6 to show how the axial and normal drop evolve. It can be seen that the normal pressure drop remains fairly constant for different tilt angles, while the axial pressure drop increases rapidly. It is logical to assume that for large, positive tilt angles the axial pressure drop will predominate and the capillary pressure level will be insufficient to sustain operation.

![Gravitational Pressure Drop](image)

**Figure 4.6:** Axial and normal gravity pressure drop as a function of a positive tilt angle.

The normal pressure drop is only present when the liquid is able to flow radially and is not obstructed in anyway [17]. The gravity pressure drop is a term that poses some difficulty during heat pipe testing for space applications, for which this pressure loss is absent, as this requires a zero degree tilt of the heat pipe. While this is no problem when testing a single heat pipe, for a system with multiple heat pipes incorporated under different angles, this means that no testing can be performed with the complete system as gravity influences its performance.

### 4.3. Heat Pipe Operating Limits

At this point equations have been given that describe the working principle of a heat pipe. However, next to this the operation and performance of heat pipes is limited by several phenomena. These phenomena or limits are described hereafter, including the appropriate equations and, if possible, derivations.

#### 4.3.1. Capillary Limit

From the earlier derived equation for the capillary pressure (Equation (4.4)) it was seen that for normal operation the capillary pressure term must be higher than the summation of pressure drops along the heat pipe's length. As soon as the pressure drops exceed the capillary pressure the heat pipe will cease operation. The limit case would be the situation in which both these terms are equal to each other, which is the so-called capillary limit, shown in Equation (4.21).

\[ \Delta p_c = \Delta p_l + \Delta p_v + \Delta p_g \]  \hspace{1cm} (4.21)
All these terms have been derived in the previous sections and can be substituted into the previous equation leading to:

\[
\frac{2\alpha_1}{r_c} \cos(\theta) = \frac{8\mu_l l_{\text{eff}} \dot{m}}{\rho_l A_w r_{h,l}^2} + \frac{8\mu_v l_{\text{eff}} \dot{m}}{\rho_v A_v r_{h,v}^2} + \rho_l g l_{\text{eff}} \sin(\phi) \quad (4.22)
\]

At this point the question is what the contribution of the liquid and vapor pressure drop is to the capillary limit. In order to find this out, both pressure drops have been calculated for axial-grooved water heat pipes as a function of temperature. The result is shown in Figure 4.7.

![Figure 4.7: Value and behavior of liquid and vapor pressure drop as a function of the temperature for a heat pipe of 15 cm length and 6 mm diameter.](image)

The difference in magnitude between the pressure losses is evident. As the liquid pressure drop is substantially larger, it is valid to neglect the vapor pressure drop in calculating the capillary limit. The next step is to rewrite the mass flow \( \dot{m} \) into terms of heat transport capacity. This is done with the help of the following relation:

\[
\dot{Q} = \dot{m} h_{lv}
\]

By substituting this relation into Equation (4.22), assuming a maximum contact angle \( \cos(\theta) = 1 \), and solving for the heat transport capacity \( \dot{Q} \) one arrives at the final relation for the capillary limit.

\[
\dot{Q} = h_{lv} \left( \frac{\rho_l A_w r_{h,l}^2}{8\mu_l l_{\text{eff}}} \right) \left( \frac{2\alpha_1}{r_c} - \rho_l g l_{\text{eff}} \sin(\phi) \right)
\]

### 4.3.2. Viscous Limit

During start-up of a heat pipe or at low heat pipe temperatures the vapor pressure gradient that is created due to heat input at the evaporator end can be low. A low pressure gradient means a low vapor velocity. If the working fluid also has a high vapor viscosity then operation can be limited due to insufficient heat transfer by vapor flow. The heat pipe then encounters the viscous limit. The maximum heat transfer at this limit can be derived by starting with the equation of state.

\[
\frac{P_v}{\rho_v} = \frac{P_0}{\rho_0}
\]

\( (4.25) \)
The next step is to use the Hagen-Poiseuille relation again (Equation (C.7)). With the substitution of Equation (4.23) and solving it for \( \rho_v \), the following relation is found:

\[
\rho_v = \frac{8\mu_v \dot{Q}}{\pi r_h^4 \lambda \Delta P_v}
\]  

(4.26)

Equation (4.26) can now be inserted into the equation of state, leading to:

\[
P_v \Delta P_v = \frac{8\mu_v \dot{Q}}{\pi r_h^4 \lambda \rho_0}
\]  

(4.27)

The term \( \Delta P_v = \frac{dP_v}{dx} \) and by integrating along the heat pipe’s length the following is found:

\[
\int_0^L P_v \frac{dP_v}{dx} = \frac{8\mu_v \dot{Q}}{\pi r_h^4 \lambda \rho_0} \int_0^L dx
\]  

(4.28)

\[
\left[ \frac{P_v^2}{P_0^2} \right]_{L}^{0} = \frac{8\mu_v \dot{Q}}{\pi r_h^4 \lambda \rho_0} L_{eff}
\]

\[
P_v^2 - P_0^2 = \frac{16\mu_v \dot{Q}}{\pi r_h^4 \lambda \rho_0} L_{eff}
\]

\[
\dot{Q} = \frac{\pi r_h^4 \lambda \rho_v \dot{Q}}{16\mu_v L_{eff}} \left( 1 - \frac{P_v^2}{P_0^2} \right)
\]

(4.29)

In the last relation of Equation (4.28) the terms \( P_0 \) and \( P_L \) refer to the vapor pressure at the heat pipe’s beginning and end, respectively, where \( P_0 \) is the vapor pressure at the evaporator section. The maximum heat transfer capacity or viscous limit is achieved when the vapor pressure at the condenser section goes to zero and the vapor pressure at the evaporator increases. This leads to the term \( \frac{P_L}{P_0} \) going to 0. The final equation for the viscous limit can found to be:

\[
\dot{Q} = \frac{A_v \pi r_h^2 \rho_v \dot{Q}}{16\mu_v L_{eff}}
\]

(4.30)

The viscous limit can be reached when the pressure ratio \( \frac{P_L}{P_0} \sim 0.3 \) [18]. Furthermore, during normal operation, this limit can be avoided when Equation (4.30) holds.

\[
\frac{\Delta P_v}{P_v} < 0.1
\]  

(4.30)

For more background on the viscous limit and a more extensive derivation the reader is referred to Reference [18].

### 4.3.3. Sonic Limit

Sonic limitation in heat pipes occurs when a shock wave emerges from sonic flow conditions. This shock wave chokes the flow and limits the heat transfer from evaporator to the condenser end. The existence of sonic flow is dependent on the speed of sound and the velocity of the vapor. Both are temperature dependent and it can be shown that sonic flow occurs only at low heat pipe temperatures. While it is argued by [19] that the speed of sound decreases at lower temperatures and therefore sonic flow is earlier achieved, it is actually the vapor density that dictates this phenomenon. This can be shown by computing the speed of sound, \( a \), the vapor density, and the latent heat of vaporization at a temperature of 274 K and 343 K for water (Table 4.1).

The variation in the value of the latent heat of vaporization is subtle between the two temperatures. The same is true for the speed of sound, where the drop is in the order of 10% from the high to low temperature. The vapor density however, is approximately an order of magnitude of 2 smaller than the density at a temperature of 343 K. Then, with the relation for the vapor velocity (Equation (4.31)) it can be seen that the vapor velocity will increase substantially when the vapor density is low.

\[
v = \frac{\dot{Q}}{\rho_v A_h \dot{h}_v}
\]

(4.31)
Table 4.1: Values for different parameters influencing the sonic limit of water at two different temperatures [20].

<table>
<thead>
<tr>
<th>Temperature [K]</th>
<th>274</th>
<th>343</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>409.6</td>
<td>455.5</td>
</tr>
<tr>
<td>(h_{lv})</td>
<td>(2.499 \times 10^6)</td>
<td>(2.333 \times 10^6)</td>
</tr>
<tr>
<td>(\rho_v)</td>
<td>0.0051</td>
<td>0.1972</td>
</tr>
</tbody>
</table>

This increase in velocity is considerably larger than the decrease in speed of sound. Hence, the sonic limit is mostly affected by the drop in vapor density at low heat pipe temperatures.

The derivation of the sonic limit is initiated with the inertial pressure drop given by Equation (4.16). Substituting the heat transport capacity as a function of mass flow into the right-side term for velocity and inserting the equation of state (Equation (4.25)) into it as well, leads to:

\[
p_v - p = \frac{p_v}{\rho_v p} \frac{\dot{Q}^2}{\dot{\mathcal{A}}_v h_{lv}}
\]

With further algebraic rearranging and solving for \(\dot{Q}\) this can be rewritten as follows:

\[
\dot{Q} = \dot{\mathcal{A}}_v h_{lv} (\rho_v p_v)^{1/2} \left[ \frac{p}{p_v} \left( 1 - \frac{p}{p_v} \right) \right]^{1/2}
\]

The heat transport capacity at the sonic limit can now be found by setting the first derivative of \(dq/dp\) to zero. This leads to a final relation for the sonic limit:

\[
\dot{Q} = 0.474 h_{lv} \rho_v \alpha_v (\rho_v p_v)^{1/2}
\]

Another formula for the heat transport capacity at the sonic limit can be found by combining the momentum equations and the conservation of energy law [21]. After some algebraic rearranging this eventually results in:

\[
\dot{Q} = \dot{\mathcal{A}}_v \rho_0 h_{lv} \left( \frac{\gamma_v T_0}{2 (\gamma_v + 1)} \right)^{1/2}
\]

This relation is based on the assumption that the frictional forces can be neglected and the vapor can be regarded as an ideal gas. This expression is valid for both the evaporator and the condenser section.

4.3.4. ENTRAINMENT LIMIT

The entrainment limit is a limit that comes into play at high vapor velocities. When the vapor velocity is high enough it is capable of taking liquid along with it due to high shear forces at the vapor-liquid interface, against the liquid flow direction. This results in a dry out of the liquid in the evaporator section and leads to a cease of operation.

The entrainment limit can be found by starting with Equation (4.36), which gives the Weber number. This number represents a relation between the viscous shear force and the forces resulting from the liquid surface tension.

\[
We = \frac{2 r_{h,l} \rho_v v^2}{\sigma}
\]

In general for Weber values close to or above 1 entrainment is likely to occur [15, 17]. By setting the Weber number to unity one can solve for the limiting vapor velocity:

\[
v = \sqrt{\frac{\sigma}{2 r_{h,l} \rho_v}}
\]
By introducing again the relation between mass flow and heat flux (see Equation (4.23)) and solving for the vapor velocity equation (4.38) is obtained.

\[ v = \frac{\dot{Q}}{A_v \rho_v h_v} \]  

(4.38)

By substituting Equation (4.38) into Equation (4.37) and solving for the heat flux Equation (4.39) is obtained, which gives the maximum heat transport capacity for the entrainment limit.

\[ \dot{Q} = A_v h_v \sqrt{\frac{\sigma \rho_v}{2 \tau}} \]  

(4.39)

**4.3.5. Boiling Limit**

While the aforementioned heat pipe limits are all axial limits, the boiling limit is a radial heat flux limit that occurs when heat fluxes are high such that the temperature of the surface of the heat pipe wall becomes higher than the liquid saturated (boiling) temperature. When this occurs, nucleate boiling emerges, which increases the convective heat transfer, but can disturb the liquid flow in the evaporator, leading to a dry-out of the wick structure. For this to happen there is a so-called critical temperature difference, \( \Delta T_{\text{crit}} \) which is the temperature elevation with respect to the boiling temperature for which nucleate boiling starts.

The formation of bubbles during nucleate boiling is governed by the relation between the bubble radius and the pressure difference across the curved surface according to Equation (4.40), where \( r_b \) is the radius of the bubble or nucleation site. The situation is depicted in Figure 4.8.

\[ \Delta p = \frac{2\sigma}{r_b} \]  

(4.40)

![Figure 4.8: Schematic of a cavity serving as a nucleation site for nucleate boiling [15].](image)

The temperature difference required from the saturation temperature of the fluid for the bubble to grow is given by the Clausius-Clapeyron relation:

\[ \frac{\Delta p}{\Delta T_{\text{crit}}} = \frac{h_v}{T_{\text{sat}} \left( \frac{1}{\rho_v} - \frac{1}{\rho_l} \right)} \]  

(4.41)

At high temperatures the vapor density is much smaller than the liquid density and therefore the inverse liquid pressure term can be neglected. Now, by substitution of Equation (4.40) and solving for the critical temperature difference \( \Delta T_{\text{crit}} \) one arrives at Equation (4.42), which gives the critical temperature at which nucleate boiling starts. (A more extensive derivation can be found in Reference [22].)

\[ \Delta T_{\text{crit}} = \frac{2\sigma T_{\text{sat}}}{\rho_v h_v r_b} \]  

(4.42)

The critical temperature difference calculated with this relation gives values that are quite strict, compared to experimental results [23]. The reason for this is that this theory is based upon an initial existence of gas bubbles in the cavities. Theoretically this means that nucleate boiling starts at a lower critical temperature difference than is experimentally verified.
Another empirical relation for the calculation of the critical temperature difference has been given by [24]. This relation is shown in Equation (4.43). Its only difference is the addition of a multiplication factor and the usage of a boundary layer parameter instead of the bubble radius.

\[ \Delta T_{\text{crit}} = \frac{3.06 \sigma l T_{\text{sat}}}{\rho v h_{\text{lg}} \delta_b} \]  

(4.43)

Here, \( \delta_b \) represents the thickness of the thermal layer (boundary layer) that emerges at the surface of the heat pipe wall. This value can be taken in the order of 25µm for typical heat pipe evaporators [15].

The last step is to relate the critical temperature difference to the maximum heat transport capacity. This is done with the help of Equation (4.44), which relates the heat transport capacity to the conductivity of the liquid and solid wall (\( k_{\text{eff}} \)) and the wall thickness, \( t_w \).

\[ \dot{Q} = \frac{A k_{\text{eff}}}{t_w} \Delta T_{\text{crit}} \]  

(4.44)

Here, \( A \), denotes the cross-sectional area through which the heat load is applied and \( t_w \), the heat load path (wall thickness). The effective thermal conductivity is dependent on the wick structure and is a function of the thermal conductivity of the wick material and liquid.

By substituting Equation (4.42) into (4.44) one ends up with the final relation for the critical heat flux [25]:

\[ \dot{Q} = \frac{2 \pi l e k_{\text{eff}} T_v}{h_{\text{lg}} \rho_v \ln \left( \frac{r_i}{r_v} \right)} \left( \frac{2 \sigma}{r_b} - \Delta P_c \right) \]  

(4.45)

The boiling limit is one of the most difficult limits to determine. This is due to the big unknown in this equation, which is the nucleation site radius \( r_b \). The presence and size of nucleation sites is dependent on the manufacturing and cleaning processes and wick structure. This value is usually in the order of \( 2.54 \times 10^{-5} \) and \( 2.54 \times 10^{-7} \) [26]. However, this differs a factor 100, which is translated directly to the same difference in result of Equation (4.45). Therefore, the nucleation site radius has to be determined experimentally in order to determine the boiling limit accurately. For boiling water at atmospheric pressure the critical temperature difference \( \Delta T_{\text{sat}} \) is approximately 5 K. Substituting this into Equation 4.42 and solving for the bubble or cavity radius \( r_b \) results in \( 6.5 \times 10^{-6} \) m for the smallest active cavity radius [15].

4.3.6. Heat Pipe Operating Region
The aforementioned limits all place restrictions on the performance of the heat pipe. When the limits are calculated as a function of temperature a plot can be made which depicts the operating region of a heat pipe. This plot will look similar to Figure 4.9, where the different operating limits demarcate the operating region of the heat pipe.

![Figure 4.9: Operating range of a heat pipe demarcated by the different limits [15].](image_url)
While the plot will look slightly different for different heat pipe configurations it does give an indication of when to expect which limit. At lower heat pipe temperatures the heat transport is mostly limited by the viscous and/or sonic limit, as was explained already in Sections 4.3.2 and 4.3.3. High velocity vapor flow will develop when the axial heat flux becomes large, which eventually leads to the entrainment limit. At high heat pipe temperatures the boiling limit will restrict the heat transport capacity as the critical temperature difference will be attained quicker at higher wall temperatures. It is for this reason the boiling limit decreases substantial with increasing heat pipe temperature. Finally, the capillary limit is much dependent on the configuration of the heat pipe. In some cases other limits will be reached first, while in others the capillary limit will be the restricting limit.
From the mathematical descriptions of the different heat pipe operating limits it was found that fluid properties and heat pipe geometry dictate heat pipe performance. Considering the first, a fluid has to be selected tailored to the heat pipe performance need. However, another aspect plays a role during this selection, which is the compatibility of a fluid with the material of the heat pipe itself. In terms of heat pipe geometry, the geometrical values that influence the different limits originate from the chosen type of wick structure. Furthermore, the external geometry of heat pipes can differ as well and impact its performance. All these aspects are investigated in Section 5.1. Finally, Section 5.2 describes the influence of transient behavior of heat pipes on performance when close to the freezing temperature of the fluid.

5.1. HEAT PIPE DESIGN AND GEOMETRY

The limits, confining the operating range of a heat pipe, are dependent on the internal working fluid. Many fluids can be chosen that all result in a different heat pipe performance. Next to the fluid the heat pipe housing plays an important role as well. The housing should be able to withstand the internal pressures and be compatible with the working fluid.

5.1.1. FIGURE OF MERIT FOR FLUIDS

The working fluid is the driving mechanism behind the heat pipe principle, therefore attention has to be paid during selection. As heat pipe transport capacity is most often the design driver a fluid tailored to this need has to be selected. The performance of different fluids can be analyzed with the help of a figure of merit. This parameter can be found from the earlier relation derived for the capillary limit in Section 4.3. By algebraic rearranging of Equation (4.24) one arrives at:

$$\dot{Q} = \left( \frac{NA_w r_w^2}{l_{eff}} \right) \left( \frac{\sigma_l \rho_l h_{le}}{\mu_l} \right) \left( \frac{2}{r_c} - \frac{\rho_l g l_{eff} \sin(\phi)}{\sigma_l} \right)$$  \hspace{1cm} (5.1)

This relation now consists of a term that is entirely dependent on the chosen fluid and is used as a figure of merit for comparison between different fluids. This term, explicitly shown in Equation (5.2) shows that it is desirable to utilize a fluid that has a high latent heat of vaporization, surface tension, and a low viscosity.

$$M = \frac{\sigma_l \rho_l h_{le}}{\mu_l}$$  \hspace{1cm} (5.2)

The comparison of the FM of different fluids, as a function of temperature, is shown in Figure 5.1. It can be seen that the FM is different for each fluid with its own specific operating temperature range.

Water has the highest FM and is thus performance-wise superior to the other fluids in the temperature range shown in Figure 5.1. This high FM is due to the high latent heat of vaporization and surface tension of water. Water is a common good and is not hazardous, therefore special handling during usage or manufacturing is unneeded. This makes it a commonly used fluid in heat pipes for Earth applications.
A downside of water can be its useful operating temperature range (approximately between 5 and 340 °C [28]) prohibiting operation at temperatures below zero °C (For an elaboration on heat pipe transient behavior and freezing see Section 5.2). This explains the large absence of water heat pipes in space as temperatures frequently go below 0 °C. Other fluids however, such as ammonia and methanol, lack a high FM and are in most cases hazardous or require special care during manufacturing [15].

The choice for a fluid is dependent on more than just the FM: Among others, the operating temperature range, material compatibility (see next section), and safety can all play a role in determining which fluid is most suited.

5.1.2. **Heat Pipe Case Material and Fluid Compatibility**

The case material or heat pipe housing has multiple functions such as preventing leakage of the fluid, withstanding the pressures attained during operation, and providing the interface for generating the capillary pressure. While these aspects are all important, the most crucial requirement for material selection is its compatibility with the working fluid.

Extensive research has been carried out on the compatibility of fluids with different materials [29]. Life tests have been executed to assess the effect of different fluid-material combinations on the operating performance of heat pipes. Certain fluid and material combinations may lead to chemical reactions from which non-condensible gas can be generated, although this can also be generated from material outgassing [28]. Subsequently, these gas particles are adopted in the liquid-vapor cycle and will be transported to the condenser section, where they remain and block part of it, effectively shortening the condenser area. This causes a sharp decrease in the temperature at the blocked part, shown in Figure 5.2 and leads to a decrease in thermal conductance [15, 17].

The decrease in heat pipe performance can also emerge from corrosion or erosion between material and fluid and lead to a decrease in surface tension and wetting angle of the liquid, while the wick structure's porosity, permeability, or capillary pore size may be adversely affected as well [28]. Furthermore, similar to the existence of non-condensible gas particles, the presence of solid or liquid impurities can degrade performance too. These impurities are transported along with the liquid flow to the evaporator section where they are deposited. This can obstruct the liquid flow in the arteries eventually speeding up dry-out of the wick or lead to further corrosion of the heat pipe wall [17, 28].

The importance of a suitable fluid material combination is covered by the aforementioned reasons. Table 5.1 lists several fluids and commonly used materials for heat pipe casing and shows their compatibility.
5.1. Heat Pipe Design and Geometry

![Figure 5.2: Effect on temperature behavior with the presence of non-condensible gas at the condenser end [28].](image)

Table 5.1: Fluid and material compatibility of heat pipes [26].

<table>
<thead>
<tr>
<th>Material</th>
<th>Water</th>
<th>Acetone</th>
<th>Ammonia</th>
<th>Methanol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper</td>
<td>RU</td>
<td>RU</td>
<td>NU</td>
<td>RU</td>
</tr>
<tr>
<td>Aluminum</td>
<td>GNC</td>
<td>RL</td>
<td>RU</td>
<td>NR</td>
</tr>
<tr>
<td>Stainless Steel</td>
<td>GNT</td>
<td>PC</td>
<td>RU</td>
<td>GNT</td>
</tr>
</tbody>
</table>

RU: Recommended by past successful usage
NU: Not used
RL: Recommended by literature
NR: Not recommended
GNC: Gas generation
GNT: Gas generation at elevated temperatures

The table shows that the copper water combination has been successfully tested in the past. It also shows that aluminum and steel are not used in combination with water as fluid due to the possibility of gas generation. Methanol can also only be used together with copper, while ammonia is applicable to both aluminum and stainless steel.

5.1.3. Different Types of Wick Structures

Many different types of wick structures have been researched and analyzed to increase the overall performance of heat pipes. The three most commonly used wick structures, also identified as being widely commercially available (Section 6.2), are depicted in Table 5.2, along with several performance features.

Table 5.2: Commonly used wick designs and their performance characteristics [22].

<table>
<thead>
<tr>
<th>Wick design</th>
<th>Capillary pumping</th>
<th>Thermal conductivity</th>
<th>Permeability</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial grooves</td>
<td>Low</td>
<td>High</td>
<td>Average-High</td>
<td>Rectangular, circular, triangular, or trapezoidal grooves</td>
</tr>
<tr>
<td>Screen mesh</td>
<td>High</td>
<td>Low</td>
<td>Low-Average</td>
<td>Single or multiple layers of screen mesh</td>
</tr>
<tr>
<td>Sintered</td>
<td>High</td>
<td>Average</td>
<td>Low-Average</td>
<td>Packed spherical particles or powder</td>
</tr>
</tbody>
</table>

The wick structures shown in Table 5.2 are the simplest in form and called homogeneous wicks as they are constructed by one machining technique or made from a single material [17, 22]. More advanced wick structures exist as well and often employ a combination of materials and layout (composite wicks) to improve on performance. However, the homogeneous wick designs are widely employed and are expected to satisfy the performance requirements of CubeSats (See Chapter 7). In this light (and in general) it is logical to argue that if satisfactory the simplest designs should be used [25]. Creating an axial grooved wick structure is achieved by extrusion or deposition [9, 22]. The simplicity of this design allows for easy and quick upfront calculations on expected performance. The axial grooves employ high permeability and thermal conductivity due to the large groove width, thereby making up for the rel-
Relatively low capillary pumping pressure compared to mesh and sintered wicks (See also Section 7.2 for an elaborate discussion on the relation between groove width and the different heat pipe limits.) [22]. Mesh or screen wicks are produced by inserting a metal cloth into a heat pipe and are the most commonly used wicks and simplest to manufacture [15, 22]. The metal cloth or screen is convoluted and inserted in the heat pipe, which is then held in place under its own tension. The performance of the screen wick structure is determined by the number of screen wraps and the pore size between the individual threads of the screen [22]. While the dimensions of these pore sizes and of the screen itself is known, the resulting generated capillary pressure is difficult to determine in advance as these pore sizes and thus permeability are also dependent on the loose- or tightness of the screen [15, 22]. Sintered wick structures are constructed by depositing metal particles on the heat pipe wall with the help of a mandrel [22]. Heat is then applied so that these particles fuse partly together thereby leaving small pores in the structure. This process leads to a high thermal conductivity of the wick and to fairly uniform properties making the prediction of performance easier than for screen wicks, however, manufacturing is more complex compared to the axial grooved or mesh wicks [22].

Although manufacturing processes are different for the three types of wicks examined and one may be more difficult to create than the other, this is not reflected in the prices of these commercial-off-the-shelve (COTS) heat pipes, as these are all in the same range (see Table 6.1). Thus, the choice for a wick structure will be mostly based on their performance.

5.1.4. The Effect of Geometry on Heat Pipe Performance
In most systems heat source and sink do not line up to allow for the implementation of a straight heat pipe. Moreover, implementation often occurs in systems which are volume limited, such as CubeSats or laptops. Both these aspects can call for the necessity of a different heat pipe geometry or orientation. As seen in Figure 6.1, heat pipes are bent to comply with the dimensional constraints that lay upon them. For most applications heat pipes are custom made and bends are introduced during manufacturing. Bends in pipes or tubes alter the flow path of the fluid. This change in geometrical flow path impacts the way the fluid flows and flow separation may occur leading to a pressure drop over the bend [30]. The pressure loss that occurs due to this phenomenon is most evident for sharp or rough inner surface corners and flows with high Reynolds numbers [31].

Figure 5.3: Reynolds number of the vapor flow as a function of heat pipe temperature and corresponding maximum heat transport capacity for a 15 cm heat pipe with outer diameter of 4 mm.

In heat pipes bends will inevitably lead to an altered flow path as well but now this holds for both the liquid and vapor flow. However, the flow velocities are low and Reynolds numbers still lie within the laminar domain as shown in Figure 5.3 for which the vapor flow Reynolds number is plotted as a function of heat pipe
temperature. The liquid flow velocity in heat pipes is even lower than the vapor velocity and liquid pressure loss due to bending can therefore be neglected [32]. The vapor pressure loss due to a bend is, among others, dependent on the vapor velocity and density, the loss coefficient, $K_L$, and the bending angle [31, 33]. In turn, the loss coefficient is dependent on the geometry of the bend, such as the bend radius and bend angle [31]. The vapor pressure loss is most evident on the capillary limit when the bend is applied in the adiabatic section, due to the fact that the capillary limit is often the restricting limit on performance [34]. Still, the other limits are affected by a bend as these are dependent on the vapor pressure term [32, 33].

Theoretically, considering the low Reynolds numbers occurring in heat pipes and the smoothness of the bends applied, it is not expected that the pressure loss due to bending is significant. Still, many tests have been carried out investigating the influence of bends, including the effect of differences in angles, positions, and radii, with varying outcomes [30, 33, 35]. From these tests it can be concluded that bends indeed can have an influence on heat pipe operation. This is mostly marked by an increase in temperature drop over the heat pipe for increasing heat loads. Theoretically, heat transfer capacity is affected as well due to the additional pressure loss term, but this is not frequently reflected in practice. This can be due to the fact that many more parameters play a role in this, such as flow velocities, pressures, and surface roughness, which can be different for each tested heat pipe.

5.2. TRANSIENT BEHAVIOR AT LIQUID FREEZING TEMPERATURE

The thermal environment in space can lead to satellite temperatures well below and above 0 °C. With water as a working fluid this is problematic as its freezing temperature is 0 °C. Therefore, it is necessary to understand the heat pipe’s behavior close to this temperature. Most analyses on heat pipes are carried out for a so-called steady-state situation in which the temperature of the heat pipe has been uniform and remained constant. Near freezing point of a working fluid, these conditions will not hold anymore and transient behavior occurs. Only few experiments have been carried out in the past that studied transient start up behavior of heat pipes [36–38]. Results of these different experiments has led to the conclusion that dry-out of the wick occurs for a high or continuous heat input at the evaporator end, resulting in a start-up failure. Due to the frozen liquid in the condenser section the heat removal rate remains high and returning vapor immediately freezes again. Solutions to aid successful start-up were found in the form of applying heat to the evaporator section in pulses and adding heat to the condenser section [38]. Furthermore, the injection of non-condensable gases have shown some effect on the start-up of frozen heat pipes as well. These gases block part of the condenser section, allowing a gradual start-up of the heat pipe [38]. The accomplishment of successful start-up however, is also dependent on a number of other parameters such as the heat pipe geometry, the working fluid, and the condenser/evaporator length ratio [39]. In the experiments carried out most of the heat pipes were in the order of 1 m length, while not always similar fluid, material, and wick combinations were used. It can therefore be expected that the outcome in similar tests for the heat pipes envisioned for use on CubeSats is different. Because of this, similar tests are required to analyze the transient start-up behavior of these smaller heat pipes.

Previous experiments have only investigated the start-up behavior of heat pipes. Another interesting aspect is thermal cycling, especially freeze/thaw cycling, which has been studied by [40]. 5 COTS copper-water heat pipes with a sintered wick structure of length 305 mm and outer diameter of 4 mm have been freeze/thaw cycled and tested. The heat pipe condenser and adiabatic section were frozen to a temperature of -30 °C while the evaporator section was held between 3 and 11 °C. Afterwards, a heat load of 0.75 W was supplied to the evaporator end until the entire heat pipe was unfrozen. This was done a 100 times and after each cycle the overall conductance of the heat pipe was measured. After 60 cycles, two heat pipes failed as the condenser ends ruptured, as shown in Figure 5.4. The other heat pipes reached 100 cycles, but similar, physical damage was found to the condenser ends. It was concluded that this was due to expansion of the liquid in the frozen state which slowly deformed the copper housing of the heat pipe.

This test shows that sintered COTS heat pipes can only sustain a limited number of freeze/thaw cycles: The expansion and retraction of water during phase transition places a heave burden on the wick structure. The question is: will axial, open grooves experience the same failure? A couple of remarks can be made concerning the test set up used. The heat pipe has been cooled down to -30 °C at both the condenser and adiabatic section. In a CubeSat LEO environment the cooling down will occur from the condenser end, which means that all the liquid will be gathered at this section, which can be an even worse case. Next to this, the heat load applied is substantial lower than the expected heat load of 10 W.
5.3. CONCLUSION ON PRACTICAL CONSIDERATIONS
The FM of fluids shows that water is superior to the other fluids when considering performance, safety, and handling and care during manufacturing. Its useful operating region is the only aspect that can be seen as a downside for space applications, which is also endorsed by the different transient start up tests carried out. However, none of these experiments governed small, commercial heat pipes and conditions similar to those present here.
The geometry does not seem to significantly affect the heat pipe’s performance, which results in a lot of freedom when employing heat pipes from a design point of view. Each wick structure has its own advantages and disadvantages and the suitability of each of these for space application will need to be investigated through experiments.
HEAT PIPES FOR CUBESAT APPLICATION

The thermal analyses have illustrated the high temperatures that can be reached on subsystem and PCB level for high performance missions. Heat pipes have been brought to the table as solution for bringing down these high temperatures. However, it is important to also explore the market for heat pipes and to investigate its applications. This will aid in identifying suitable heat pipes for CubeSats. First, Section 6.1 will present examples of small applications in which heat pipes have been employed. Second, Section 6.2 will investigate the commercial availability of these small heat pipes. Both sections will help identify the feasibility of using these heat pipes in CubeSats.

6.1. SMALL HEAT PIPE APPLICATIONS

Considering the posed requirements on the TCS in a CubeSat one has to look at similar, small scale applications that employ heat pipes. A well-known device, which utilizes heat pipes and is volume limited similar to CubeSats, is the laptop. This device contains several processing units that can dissipate substantial amounts of heat during high workloads. Since the early laptops heat pipes have been used to cope with this heat dissipation by transporting the heat from these hot spots towards a sink, which is cooled by forced convection through a fan. A schematic of a heat pipe thermal solution and a real life example in a laptop is shown in Figure 6.1.

Figure 6.1: Basic schematic of a thermal solution in a laptop with a heat pipe [41] (a) and real-life example (b).

While heat pipes in laptops have become a common sight, the mobile phone industry is just at the forefront of realizing the necessity of proper thermal management [42]. Processing units in mobile phones are getting more powerful than ever, while the volume becomes or remains even more limited than in laptops. Also in this industry heat pipes have been used or are being proposed to prevent overheating of processors. Small heat pipes for usage in mobile phones or hand-held devices and an example are shown in Figure 6.2.
6.2. COMMERCIAL AVAILABILITY OF SMALL HEAT PIPES

There exist many companies who offer thermal control solutions in the form of heat pipes. While they all provide the possibility of custom heat pipe solutions, only few of them offer standard off-the-shelve heat pipes. Among these few is a Germany-based company called Quick-Ohmm who offers both standard and custom-made copper-water heat pipes with sintered, mesh, or grooved wick structures (see Section 5.1.3 for an elaboration on wick structures) [45]. The prices in Euro for standard copper-water heat pipes available for purchase are shown in Table 6.1.

Table 6.1: Prices (€) of standard copper-water heat pipes offered by the company Quick-Ohm [45].

<table>
<thead>
<tr>
<th>Length [mm]</th>
<th>Grooved Ø 4</th>
<th>Grooved Ø 6</th>
<th>Grooved Ø 8</th>
<th>Mesh Ø 4</th>
<th>Mesh Ø 6</th>
<th>Mesh Ø 8</th>
<th>Sintered Ø 4</th>
<th>Sintered Ø 6</th>
<th>Sintered Ø 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>-</td>
<td>12.30</td>
<td>12.40</td>
<td>11.21</td>
<td>12.40</td>
<td>12.88</td>
<td>11.67</td>
<td>12.88</td>
<td>14.86</td>
</tr>
<tr>
<td>150</td>
<td>-</td>
<td>12.30</td>
<td>12.40</td>
<td>11.21</td>
<td>12.40</td>
<td>12.88</td>
<td>11.67</td>
<td>12.88</td>
<td>15.56</td>
</tr>
<tr>
<td>200</td>
<td>-</td>
<td>12.40</td>
<td>12.88</td>
<td>11.67</td>
<td>12.88</td>
<td>13.33</td>
<td>12.30</td>
<td>12.88</td>
<td>15.56</td>
</tr>
<tr>
<td>250</td>
<td>-</td>
<td>12.40</td>
<td>12.88</td>
<td>11.67</td>
<td>12.88</td>
<td>13.33</td>
<td>12.30</td>
<td>12.88</td>
<td>15.56</td>
</tr>
</tbody>
</table>

The offered prices are for a purchase of a single heat pipe only. The price drastically drops when the bought quantity is increased: For 3 heat pipes the individual prices already drop 30%, while for a quantity of over 100 the prices drop approximately 60%. It is interesting to note that the price differences between the heat pipes with different wick structures are small. Only for a diameter of 8 mm the sintered heat pipe is slightly more expensive than the other two.

For operating temperatures below 0 °C copper-methanol heat pipes are offered, although far less in quantity than the copper-water types. Only one copper-methanol heat pipe fitting the dimensions of the CubeSat is offered (Ø 8 mm and 250 mm length). This heat pipe has a price tag of € 58.40, which is substantially more costly than the copper-water heat pipes.

A second company is Enertron: A Taiwanese company with establishments in Taiwan and the USA who offers various thermal solutions. They can aid in design, testing, prototyping, and manufacturing of, among others, heat pipes. Standard copper-water heat pipes in different lengths and diameters, with different wick structures are offered [46]. Table 6.2 show the offered heat pipes on their webshop.

Although there is less choice compared to Quick-Ohm it is not unlikely that their stock changes from time to time. Furthermore, custom orders can be easily placed on their websites in case a different heat pipe is desired. The prices are in a similar range compared to those offered by Quick-Ohm.
Table 6.2: Prices ($S) of standard copper-water heat pipes offered by the company Enertron [46].

<table>
<thead>
<tr>
<th>Length [mm]</th>
<th>Grooved</th>
<th>Mesh</th>
<th>Sintered</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>⌀ 4</td>
<td>⌀ 6</td>
<td>⌀ 8</td>
</tr>
<tr>
<td>100</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>150</td>
<td>-</td>
<td>10.00</td>
<td>-</td>
</tr>
<tr>
<td>200</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>250</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
Theoretical Analyses of Heat Pipe Performance

The theory of heat pipes predicts the existence of a number of limits, which place a restriction on the heat pipe's transport capacity. While these can be examined at the hand of their equations a thorough understanding of the effect of certain parameters on these limits can better be obtained by visual observation. The operating regions for different heat pipe configurations are computed and examined in Section 7.1. Then, as the wick structure and vapor area both have a large influence on the maximum heat transport capacity of heat pipes analyses are conducted on the effect of varying these quantities, which is discussed and shown in Section 7.2. In Section 7.3 design parameters are derived from the theoretical analyses, while theoretical analyses are carried out in Section 7.4.

### 7.1. Heat Pipe Operating Performance

The performance of a heat pipe is demarcated by its operating limits. These in turn are dependent on a number of parameters identified in previous chapters. In order to perform analyses these parameters have to be known in advance. These parameters originate from the working fluid, the dimensions, and the wick structure of the heat pipe.

The requirements 2.1 and 2.2 stated in Section 2.3 lead to constraints on the dimensions of the heat pipe. The type of working fluid is chosen based on the commercial availability (Section 6.2) and the figure of merit and handling (Section 5.1.1). This leads to the conclusion that water is the preferred choice.

The wick structure is chosen based on the discussion about and analysis of different wick designs in Section 5.1.3. From this, and considering the fact that for some wick structures not all parameters are known in advance, analyses will be performed for an axial grooved wick structure. The groove width is chosen based on the theoretical analyses performed in Section 7.2.

The contact angle is set to the ideal value of 0°. For heat pipes with these parameters theoretical analyses will be carried out. The complete specifications are listed in Table 7.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Heat Pipe 1</th>
<th>Heat Pipe 2</th>
<th>Heat Pipe 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>Water</td>
<td>Water</td>
<td>Water</td>
</tr>
<tr>
<td>Wick design</td>
<td>Axial grooves</td>
<td>Axial grooves</td>
<td>Axial grooves</td>
</tr>
<tr>
<td>Length [mm]</td>
<td>150</td>
<td>150</td>
<td>150</td>
</tr>
<tr>
<td>Diameter [mm]</td>
<td>4</td>
<td>6</td>
<td>8</td>
</tr>
<tr>
<td>Groove width [mm]</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
</tr>
<tr>
<td>Evap/Cond length [mm]</td>
<td>30</td>
<td>30</td>
<td>30</td>
</tr>
</tbody>
</table>

Three heat pipes are chosen for analyzing the theoretical performance, which only differ by their outer diameter. Each performance limit has been calculated as a function of heat pipe temperature and plotted,
such that figures are obtained similar to 4.9. From these figures, the maximum heat transport capacity as a function of heat pipe temperature can be deduced. The results are shown in Figures 7.1, 7.2, and 7.3. Values for the different fluid parameters have been taken from [20].

Figure 7.1: Operating limits for a heat pipe with outer diameter of 4 mm.

Figure 7.1 shows the heat pipe limits of heat pipe 1, with outer diameter of 4 mm. It shows that the heat transport capacity is detained only by the capillary and boiling limit. The maximum heat transport capacity is 17.5 W at 317 K. At lower temperatures, the heat transport capacity quickly decreases to even below 10 W. At higher temperatures, the transport capacity is confined by the boiling limit. It has to be stressed, however, that the boiling limit is much dependent on the nucleation site radius. As was discussed in Section 4.3.5, this radius is the parameter that can induce large errors in the values for the boiling limit and can only be verified experimentally.

Figure 7.2: Operating limits for a heat pipe with outer diameter of 6 mm.
For heat pipe 2 (Figure 7.2) one can observe that the heat transport capacity has increased compared to the previous one and is detained by the capillary and the boiling limit as well. The larger heat pipe diameter leads to an increase in wick area with a fixed groove width. This larger wick area raises the capillary limit (see Equation 4.24) leading to a maximum value of 29 W at 316 K. At low heat pipe temperatures the viscous, entrainment, and sonic limit are all encountered at higher power levels compared to the 4 mm diameter heat pipe. This can be explained by the fact that the viscous, entrainment, and sonic limit are all dependent on the vapor area, which means that for an increase in vapor area the heat transport value at which these occur increases as well.

Figure 7.3, displaying the limits for heat pipe 3 with an outer diameter of 8 mm shows the same pattern: the capillary and boiling limit detain the maximum heat transport capacity and the maximum equals 40 W at 316 K.

Figure 7.3: Operating limits for a heat pipe with outer diameter of 8 mm.

Considering the CubeSat dimensions it is not unlikely that other heat pipe lengths are suitable as well or necessary due to constraints of the application or location of the heat source. Therefore, in Appendix D the same analyses have been carried out for heat pipes with lengths of 10 and 25 cm. From these analyses it is found that the length of the heat pipe has significantly less impact on the maximum heat transport capacity compared to a change in outer diameter. For longer heat pipe lengths the transport capacity slightly decreases, which can be attributed to the fact that the effective length has increased. This in turn increases the liquid and vapor pressure drops, thereby decreasing the capillary limit. The same holds for the viscous limit which is dependent on the effective length as well.

Finally, Appendix D also shows the performance of a methanol-copper heat pipe equal in dimensions to heat pipe 2. According to the FM in Figure 5.1 the performance of methanol is significantly lower than that of water. This is indeed reflected in the operating limits of the methanol heat pipe. Also, these limits mostly occur at low temperatures far below the freezing point of water. As methanol has a lower freeze point it is an interesting fluid for use in space, however, it is not capable of transporting the required heat levels of 10 W.

7.2. HEAT PIPE WICK GEOMETRICAL ANALYSES

With the computation of the performance of different heat pipes completed, the next step is to perform some analyses on different parameters that influence this performance. From the theory explained in Chapter 4 it was found that, next to fluid properties most operating limits are a function of geometrical parameters of which heat pipe length, vapor area, and groove width are most evident. Each limit has been analyzed to observe their behavior when varying these parameters for a heat pipe with properties as listed in Table 7.2. A more extensive set of results for heat pipes with different lengths can be found in Appendix D.
Table 7.2: Heat pipe specifications for limit analyses.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Heat Pipe</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>Water</td>
</tr>
<tr>
<td>Temperature [K]</td>
<td>313</td>
</tr>
<tr>
<td>Wick design</td>
<td>Axial grooves</td>
</tr>
<tr>
<td>Length [mm]</td>
<td>150</td>
</tr>
<tr>
<td>Evap/Cond length [mm]</td>
<td>30</td>
</tr>
</tbody>
</table>

7.2.1. Capillary Limit as a Function of Groove Width

The foremost aspect of the heat pipe's working principle is the necessity of generating capillary forces. Looking at the equation that describes the maximum capillary pressure (Equation (4.6)) one can see that this is a function of the liquid surface tension and the effective capillary radius (equal to the axial groove width). From this simple relation it can be deduced that the maximum capillary pressure is attained when the groove width approaches 0. Apart from the physical limitations that exist for groove-widths that approach this number, the liquid pressure drop is a function of the axial groove-width as well (Equation (4.13)). As this is related to the fourth power of the groove-width the liquid pressure loss for small widths will dominate. Thus, there exists an optimum range or value for the groove-width that enables a sufficient capillary pressure force while a minimal liquid pressure loss occurs.

Equation (4.24) enables the calculation of the maximum heat transport capacity as a function of capillary pressure and liquid pressure loss. For the heat pipe characteristics defined in Table 7.2, the maximum heat transport capacity is plotted in Figure 7.4 for several heat pipe diameters as a function of the axial groove-width. The number of grooves has been written as being a function of groove-width as well.

![Figure 7.4: Capillary limit for a 15 cm heat pipe in horizontal position.](image)

The figure shows a zoomed-in plot area with heat load levels closer to that which need to be transported in high-performance CubeSat missions. It can be seen that larger groove-widths enable a higher heat transport capacity, as expected as the liquid pressure drop decreases with the power fourth of the groove-width. The required 10 W of heat transport capacity requires a groove width larger than 0.15 mm in case a 4 mm heat pipe diameter is chosen. A groove width of 0.2 mm is therefore determined to be the minimum.

For the plot shown in Figure 7.4 it is assumed that the gravitational pressure drop is zero as the heat pipe's orientation is horizontal (similar to space condition). However, for testing it may be required to analyze the influence of gravity on the heat pipe's performance. Therefore, analyses have been carried out for different (positive) tilt angles. A positive angle is defined here as the elevation angle between an elevated evaporator...
section and the condenser end. The results can be found in Appendix D for different heat pipe lengths. The found trend is a decreasing capillary limit for higher elevation angles, as follows logically from the gravity pressure drop term in Equation 4.22.

7.2.2. ENTRAINMENT LIMIT AS A FUNCTION OF GROOVE WIDTH

The heat transport capacity at the entrainment limit is given by Equation (4.39) and is shown for different heat pipe diameters in Figure 7.5. Looking at both the equation and the figure it is seen that entrainment limit decreases for increasing groove-widths. This can be explained by the fact that the vapor flow can more easily collect droplets and transport these along with it from wider channels. Comparing the values for this limit with the capillary limit, one can easily see that for larger groove-widths the entrainment limit becomes dominant.

Figure 7.5: Entrainment limit for a 15 cm heat pipe in horizontal position.

7.2.3. BOILING LIMIT AS A FUNCTION OF GROOVE WIDTH

The boiling limit is a function of the groove-width due to the maximum capillary pressure term in Equation (4.45). (As for square axial grooves the thermal conductivity is only dependent on the thermal conductivity of the liquid and wick material.) As the maximum capillary pressure is lower for larger groove widths, the boiling limit will increase as well for larger groove widths (although the increase will only be small as the term \( \frac{2\sigma}{\rho L} \) will remain much larger than the term \( \frac{2\rho}{L} \) for maximum capillary pressure. The boiling limit for different heat pipe diameters is shown in Figure 7.6.

The figure shows that the diameter of the heat pipe (apart from the earlier mentioned value of the nucleation site radius) has the largest influence on the boiling limit, due to the \( \ln \left( \frac{2}{\rho L} \right) \) term. Another influential parameter is the length of the evaporator section. From the given equation for the boiling limit it can be seen that for smaller evaporator lengths the heat transport limit is reached sooner. This can be explained by the fact that an applied heat load on a small distance will lead sooner to boiling (higher heat flux density) than when the same heat load is applied over a larger distance.

At first sight, looking solely at the boiling limit it could be concluded that a heat pipe diameter of 4 mm is the minimum to ensure a sufficient heat transport of 20 W (requirement 1.2.1.2). However, as was explained in Section 4.3.5, the boiling limit is also the least easily determined limit. A solution when the boiling limit is limiting the heat transport capacity is to increase the evaporator length.
7.2.4. **Viscous Limit and Sonic Limit as a Function of Vapor Diameter**

Both the viscous and sonic limit are a function of the heat pipe radius. The heat transport capacity for the viscous limit is given by Equation (4.29). From this equation it can be seen that the heat transport capacity increases for larger vapor diameter radii. Furthermore, the effective length of the heat pipe affects this limit as well: for longer heat pipe lengths $\dot{Q}$ will decrease. The viscous limit is shown in Figure 7.7.

It can be seen from this figure that indeed the maximum heat transport capacity increases when heat pipe diameter increases. This capacity is sufficiently large and no limiting factor for the temperature level of 313 K used here. As was discussed in Section 4.3.2, the sonic limit mostly occurs at heat pipe start up and is much more dependent on vapor pressure (a function of temperature) than on vapor area.

The sonic limit can be calculated with both Equation (4.34) as (4.35). It is interesting to see how both
these compare with each other. Therefore, they are plotted both in the same graph as a function of vapor area diameter, shown in Figure 7.8. The result of both equations is only a small deviation, while the slopes are similar. For both it holds that the sonic limit is only reached at high heat transport capacities. However, as explained in Section 4.3.3, the sonic limit is much dependent on the vapor density, which is small at low temperatures. This means that at lower heat pipe temperatures this limit might become the dominating limit, as is the case for the heat pipe with outer diameter of 6 mm as shown in Figure 7.2.

7.3. Heat Pipe Design for CubeSats

Based on the analyses carried out one can define a heat pipe design tailored to the application’s needs. The analyses provide minimum values for the different parameters that are required to adhere to the needs of any application. The analyses, however, do not only give insight into the minimum requirements of a heat pipe they can also be used to optimize an existing heat pipe. In this case, however, the explicit desire to use COTS heat pipes limits optimization and a reversed procedure is followed: Minimum and optimal values are derived from the theoretical analyses after which a suitable, existing COTS heat pipe is sought.

The heat pipes analyzed in Section 7.1 have shown that they can all cope with the requirements on heat load dissipation (requirements 1.2.1.1 and 1.2.1.2), except for the 4 mm outer diameter heat pipe. The optimal groove width, based on the capillary limit (Figure 7.4) and the boiling limit (Figure 7.6) is found to be 0.2 mm. The length of the heat pipe was found to have insignificant impact on the heat pipe’s performance compared to the other dimensional parameters. Therefore the choice for heat pipe length is only restricted by the CubeSat dimensions but depends mostly on the distance between heat source and heat sink. The diameter of the heat pipe is determined by the maximum heat transport capacities found, which led to the conclusion that a minimum diameter of 6 mm is required. Larger diameter heat pipes, although able to transport more heat, are deemed not suitable as this will lead to an increase in mass, space occupation, and to be able to pass any on top laying PCB requires a relatively large cut-out. Table 7.3 sums up the optimal heat pipe specifications.

Table 6.1 from Section 6.2 shows that COTS heat pipes with these specification are available, except for the axial grooved heat pipes with an outer diameter of 4 mm.

7.4. Temperature Profile With Heat Pipe Integration

The analyses in previous sections have shown that theoretically the COTS heat pipes are sufficiently capable of transporting the expected heat dissipation levels. In practice, however, it can be expected that the heat transport capacity limits will be somewhat lower due to, for example, interface resistances (see Section 8.1.1).
Table 7.3: Optimal heat pipe specifications based on the theoretical analyses and requirements.

<table>
<thead>
<tr>
<th>Heat Pipe Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>Water</td>
</tr>
<tr>
<td>Groove width [mm]</td>
<td>0.20</td>
</tr>
<tr>
<td>Diameter [mm]</td>
<td>6</td>
</tr>
</tbody>
</table>

Before this is taken into account theoretical analyses are carried out to investigate the effect of the inclusion of a heat pipe on the temperature profile of a CubeSat. This has been accomplished by running thermal analyses with the same conditions as were done in Section 3.3. In the thermal model the heat pipe has been modeled as a direct high conductive path between the heat dissipating node and the designated structural nodes, the frame ribs, that act as a heat sinks (see Section 8.1.1).

![Transient Results Panels](image)

(a) 10 W no heat pipe

(b) 10 W with heat pipe

**Figure 7.9:** Panel temperature profile for a 06:00 h LTAN orbit in nadir-fixed attitude mode with and without a heat pipe. The results for the external structure (Figure 7.9) show no difference with and without the usage of a heat pipe, except for the fact that with a heat pipe the steady-state temperature will be reached quicker. This can be explained by the fact that heat from the subsystem is transported faster towards the outer structure, whereas without a heat pipe the heat transport goes slower and it takes more time to reach equilibrium conditions.
This holds for all attitude modes and orbit types. It is therefore more interesting to look at the temperature profiles of the subsystems for the different orbits and attitudes, shown in Figure 7.10. Comparison with Figure 3.9 shows a large drop in temperature of about 20 to 30 °C due to the inclusion of the heat pipe. The direct heat path between the external structure and the subsystems also leads to a more fluctuating temperature distribution for the 12:00 h LTAN orbit.

![Figure 7.10](image_url)

**Figure 7.10:** Internal temperature profiles with a heat pipe for the different orbits and attitudes.

The decrease in temperature is beneficial and leads to a safe temperature range for the components. However, there are several assumptions made, such as no interface resistances, no internal view factors (see Appendix A) and no detailed nodal breakdown of the subsystems. These simplifications can have substantial impact on the resulting temperature profiles, therefore, during integration testing (see Section 10.1.1) these simplifications have to be reconsidered including their effect on the outcome.
Theoretical analyses have shown the feasibility of using heat pipes to cope with the projected heat dissipation loads. The next step is to look at the integration of the heat pipe into the CubeSat structure. This brings along several aspects that need to be considered first, such as the impact that the interfaces between heat sink and source and the heat pipe have on the heat transfer. Another aspect is the choice of structural element that is going to be used as heat sink. Both these aspects are discussed in Section 8.1. Then, in Section 8.2 a preliminary design is made to visualize how integration could be realized in a CubeSat structure.

8.1. Integration and Interfaces on CubeSat Level

The theoretical analyses carried out assumed the ideal case in which the heat load is directly put in the heat pipe without any interface losses. In reality this is obviously not the case as the transfer of heat between different elements induces heat losses. Therefore, it is necessary to gain insight into the magnitude of these losses and how this affects the integration of the heat pipe into the CubeSat.

8.1.1. Thermal Aspects of Heat Source and Sink Interfaces

Apart from the challenge of designing a heat pipe tailored to the application’s needs it is even more important to look at the interfaces between the heat pipe and the source or sink. Interfaces are another resistance in the long chain of resistances present in a heat pipe, as shown in Figure 8.1.

![Figure 8.1: Thermal resistances present in a heat pipe set up and equivalent circuit [15].](image)

Each resistance leads to a small temperature drop. The summation of these temperature drops will determine the resulting temperature gradient between the heat source and sink. Table 8.1 lists the order of
magnitude for the different resistances, which shows that the resistances induced by the interfaces ($R_1$ and $R_9$) are the largest and approximately of the same order of magnitude as the resistance of the entire heat pipe itself [17]. This illustrates the influence of interfaces as a high resistance will lead to a large temperature gradient and thus a higher temperature of the heat source.

Table 8.1: Thermal resistances and their order of magnitude [15, 17].

<table>
<thead>
<tr>
<th>Resistance</th>
<th>Interface</th>
<th>°C/W</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_1$</td>
<td>Heat source - Evaporator</td>
<td>$10^3 - 10$</td>
</tr>
<tr>
<td>$R_2$</td>
<td>Heat pipe wall</td>
<td>$10^{-1}$</td>
</tr>
<tr>
<td>$R_3$</td>
<td>Heat pipe wall - Wick structure</td>
<td>$10$</td>
</tr>
<tr>
<td>$R_4$</td>
<td>Liquid - Vapor</td>
<td>$10^{-5}$</td>
</tr>
<tr>
<td>$R_5$</td>
<td>Vapor section</td>
<td>$10^{-8}$</td>
</tr>
<tr>
<td>$R_6$</td>
<td>Vapor - Liquid</td>
<td>$10^{-5}$</td>
</tr>
<tr>
<td>$R_7$</td>
<td>Wick structure - Heat pipe wall</td>
<td>$10$</td>
</tr>
<tr>
<td>$R_8$</td>
<td>Heat pipe wall</td>
<td>$10^{-1}$</td>
</tr>
<tr>
<td>$R_9$</td>
<td>Condenser - Heat sink</td>
<td>$10^3 - 10$</td>
</tr>
<tr>
<td>$R_s$</td>
<td>Heat pipe wall (axial)</td>
<td>$10^2$</td>
</tr>
</tbody>
</table>

While heat pipe resistances are largely fixed, the interfaces can be freely designed and are only limited by the application itself. The resistance of the interfaces is largely determined by their contacting area. Equation (3.3) shows that conduction (reciprocal of resistance) increases for a higher contacting area. This means that an interface works most efficiently when the areas between interface and heat source match.

Next to this, contact conductance (Section 3.1.2) plays a crucial role as well. As surfaces are never perfectly flat the actual contact area is less which leads to less efficient heat transfer. This contact conductance is, among others, dependent on the cleanness of the surfaces, the contact pressure, and the surface roughness [9]. Of these, the simplest method is proper clamping and persisting gaps can further be filled with, for example, thermal grease, glue, or a thermal cloth to improve the conductive path.

Figure 8.2: Different structural heat pipe interfaces (1: heat pipe casing, 2: mounting surface, 3: saddles, 4: thick flange) [9].

The difficulty of a heat pipe interface is that it needs to allow for clamping of a cylindrical structure, while a heat source in the form of a chip is flat. Several possible interfaces for heat pipe attachment to such a source are shown in Figure 8.2. Instead of resistance or conduction values the efficiency of heat transfer from the heat source to the heat pipe is often expressed by the heat transfer coefficient, $h_t$. This parameter is a measure based on the temperature gradient and heat flux, see Equation (8.1).

$$h_t = \frac{P}{AA\Delta T}$$
Here, $A$ is the contacting area between the interface and the heat pipe, while $P$ is the amount of heat in W. The power scales linearly with the temperature difference, meaning that for twice the power, the temperature difference between the evaporator section and the heat pipe will be twice as high as well. The constancy of the area and the linearity between the power and temperature difference lead to a constant heat transfer coefficient for different power levels.

Interestingly enough it is this parameter that can be used to check the heat pipe’s functionality: From theoretical analyses it was found that the capillary limit is the confining limit for the heat pipe’s performance. This means that above this maximum the capillary pressure is insufficient to overcome the liquid (and vapor) pressure drop. In other words the liquid is not able to fully return to the evaporator section and dry-out of the wick in the evaporator section will occur. When dry-out occurs the heat load subject on the evaporator section is not carried away by the heat pipe and will simply heat up the heat pipe wall and evaporator section. This results in a $\Delta T$ that does not scale linear anymore with the power input and, due to the following quick rise in $\Delta T$, the heat transfer coefficient drops (Equation (8.1)). While steady-state might still be reached at some point the benefit of having a heat pipe disappears when the $\Delta T$ becomes too large. Therefore, it can be concluded that at that point the functionality of the heat pipe is no more. This situation is visualized in Figure 8.3.

![Figure 8.3: Dry out occurrence during testing when the heat pipe limit is surpassed.](image)

### 8.1.2. Potential Heat Source and Sink Location in a CubeSat

Looking at the heat source, it was found in Section 2.2 that the amplifier is expected to become one of the most thermally critical components. The ISIS S-Band transmitter PCB, shown in Figure 8.4, highlights this even further as already a small heat sink is used, covering the amplifier, to distribute the generated heat over a larger surface area (depicted by the red box in the figure). Although the CubeSat poses many dimensional constraints on internal systems and components, the presence of this heat sink and other cut outs in the board indicate the possibility of adding additional components, such as a heat source interface, to a PCB.

A heat source interface for a heat pipe will look similar to the heat sink used on the amplifier. The only difference will be that the interface needs to allow for attachment of a heat pipe. The interface should furthermore cover at least the entire chip surface to achieve optimal heat transfer, but should not place a heavy burden on the mass budget. As the heat pipe can be bent in any direction and thus can be orientated such that it is routed across the location of the heat generating chip, it does not pose any restrictions on PCB design.

The heat sink interface needs to be a structural element which can both quickly distribute the heat to the rest of the structure and limits the possibility of creating a hot spot. The interface itself is again limited by the CubeSat structure and the dimensional constraints [10]. From these requirements and limitations the panels, frame structure and ribs, and stack ribs can be marked as potential heat sink elements, all of which are shown in Figure 8.5a.

So, which one is best suited to act as heat sink element? Looking at an integrated CubeSat it can be noted that there is limited space available for routing a heat pipe through. Physical limitations exist which prohibit a heat pipe connection to certain structural elements. This, for example, is the case when one wants to connect a heat pipe to the frame structure: The route is blocked by the steel stack rods, as shown in Figure 8.5b. The
Figure 8.4: The ISIS TXS S-Band Transmitter PCB with heatsink indicated in red [47].

Figure 8.5: Skeleton of a 1U CubeSat showing the different structural elements (a) [48] and internal stack layout showing the steel rod (b).

same holds for the outer panels, which would be ideal for quickly spreading out the heat but have lots of cut outs and wiring present necessary for, among others, the solar panels. This leaves the frame ribs and stack ribs as possible candidates. The inner stack of PCBs are all connected by an inter-stack connector. This connector covers one entire side path between the PCBs and thus prohibits routing a heat pipe through this side (the stack connector is shown in Figure 8.6 on the right side). This leaves the frame ribs as only plausible option for heat pipe attachment.

As the interest lies on employing water heat pipes one can imagine that temperatures around or below $0 \, ^\circ C$ are undesired as this might lead to cease of operation of the heat pipe (when no heat load is applied) (See also Section 5.2 for transient behavior of frozen water heat pipes.). Therefore, it is interesting to review the temperatures that these frame ribs attain at different heat loads. For a 12:00 h LTAN orbit in Y-Thomson mode their temperatures have been computed, see Table 8.2.

The lowest temperature attained by the frame ribs is substantially lower than the freezing point of water, which means that if a heat pipe is attached to it and no heat is being dissipated by the payload the heat pipe will freeze at the condenser end. This problem can be solved by making sure a high payload duty-cycle is used and temperatures below freezing point will not occur, however, one of the goals is to prevent the thermal solution for posing any restriction on the operation of subsystems (Req. 3.2). Therefore, it is necessary to gain knowledge on the freeze and thaw rate of water heat pipes.
Table 8.2: Temperature extremes attained by the frame ribs for different heat dissipation in a 12 h LTAN orbit in Y-Thomson mode.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard (&lt;1)</td>
<td>-25</td>
<td>11</td>
</tr>
<tr>
<td>5</td>
<td>-20</td>
<td>20</td>
</tr>
<tr>
<td>10</td>
<td>-5</td>
<td>40</td>
</tr>
<tr>
<td>10, with heat pipe</td>
<td>10</td>
<td>60</td>
</tr>
</tbody>
</table>

Another logical follow up from applying a heat pipe is that it leads to an elevated maximum temperature of the frame rib due to the additional heat transported by the heat pipe. This potentially leads to another thermal hot spot, albeit on a structural element only.

8.2. INTEGRATED HEAT PIPE DESIGN

With the interface locations and limitations known a design can be made to show how a heat pipe can be integrated into a CubeSat. Routing the heat pipe through the structure from heat source to sink is, as already discussed, constrained by several physical elements. Furthermore, to limit the heat fluxes in and out of the heat pipe it is desirable to have the contacting area with the interfaces maximized. Therefore, a length of 30 mm for both the condenser and evaporator sections is chosen. This choice subsequently confines the heat pipe’s orientation and path through the CubeSat structure.

The design is based on the fact that a satellite is often orientated such that one side sees deep space and the opposite experiences heat fluxes. For this reason attaching a heat pipe to both opposite sides of the satellite will make sure that heat is always able to find its way to a colder part. Even more for the reason that this ensures an equal heat distribution over the frame. While it seems that this requires the use of two heat pipes a single heat pipe can be used as well. The evaporator section of a heat pipe does not need to be at one of the heat pipe’s end. The thermal cycle inside a heat pipe is initiated as soon as heat is applied at a section of the heat pipe. When the evaporator section is located in the middle the liquid will evaporate at that point and the vapor will flow to both ends, effectively working as two separate, shorter heat pipes.

Interestingly enough, this freedom in placement of the heat pipe makes it also possible to adapt the design to the orbit itself. For example, in some orbits it might be that only one side is continuously exposed to deep space in which case one could come up with a design that integrates a single heat pipe from heat source to this panel only.

Figure 8.6 shows how the integration of a heat pipe could look. In this design the chip is assumed to be located at the middle of the PCB, but, as mentioned before, can be anywhere on the PCB due to the possibility of bending the heat pipe. The stiffners are designed such that they utilize the existing screw holes and do not interfere with the structure. Their job is to clamp the heat pipe thoroughly against the frame rib. More on the design and integration of each different element can be found in Chapter 10.

The integration of the heat pipe has only one impact on the rest of the structure, which is that it requires a small cut-out in the PCBs where the heat pipe is routed across. The small cut outs required are shown in Figure 8.7.
Figure 8.6: Heat pipe integration design in a single CubeSat stack.
Figure 8.7: Cut outs required to route the heat pipe across the on top lying PCBs.
While theoretical analyses give insight into and predictions on expected behavior, experiments are necessary to verify and validate this theory. The first experimental phase consists of executing different tests that will help in verifying and validating the theoretical analyses. In this phase the focus lies solely on the individual performance of the heat pipes under different circumstances. Section 9.1 will present an overview of the heat pipes that will be used during testing and the different tests that will be carried out. Thereafter, Section 9.2 will go through the setup used for carrying out the performance characterization tests, show the results, and discuss these. Subsequently, Sections 9.3 and 9.4 will present the tests covering the transient behavior of heat pipes, while this chapter is finalized by Sections 9.5 and 9.6 in which visual inspection of the different wick structures is done and a summation of the test results is given, respectively.

9.1. Overview of Heat Pipes and Experiments
The theoretical analyses carried out in Section 7.3 have shown that a minimum heat pipe diameter of 6 mm is required to cope with the heat loads. For this reason the main focus will lie on heat pipes with this diameter. The lengths of the heat pipes are based on the CubeSat constraints, which will lie in the order of 150 to 300 mm. For both the sintered and mesh heat pipes a 4 mm variant is included for performance comparison. Unfortunately, no 4 mm axial grooved heat pipe was available (Chapter 6) which resulted in the choice for a 6 mm axial grooved heat pipe with a length of 150 mm. This will allow for comparison between the two grooved heat pipes and validation of theory. A methanol heat pipe has been included as well for comparison of fluid performance. The dimensions of this pipe was again constrained by the availability.

<table>
<thead>
<tr>
<th>Heat pipe [#]</th>
<th>Wick type</th>
<th>Length [mm]</th>
<th>Diameter [mm]</th>
<th>Fluid</th>
<th>Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Groove</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>I</td>
</tr>
<tr>
<td>2</td>
<td>Mesh</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>I</td>
</tr>
<tr>
<td>3</td>
<td>Sintered</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>I</td>
</tr>
<tr>
<td>4</td>
<td>Groove</td>
<td>150</td>
<td>6</td>
<td>Water</td>
<td>I</td>
</tr>
<tr>
<td>5</td>
<td>Mesh</td>
<td>200</td>
<td>4</td>
<td>Water</td>
<td>I</td>
</tr>
<tr>
<td>6</td>
<td>Sintered</td>
<td>200</td>
<td>4</td>
<td>Water</td>
<td>I</td>
</tr>
<tr>
<td>7</td>
<td>Sintered</td>
<td>300</td>
<td>6</td>
<td>Methanol</td>
<td>I</td>
</tr>
<tr>
<td>8</td>
<td>Groove</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>I,II,III,IV</td>
</tr>
<tr>
<td>9</td>
<td>Mesh</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>I,II,III,IV</td>
</tr>
<tr>
<td>10</td>
<td>Sintered</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>I,II,III,IV</td>
</tr>
<tr>
<td>11</td>
<td>Groove</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>IV</td>
</tr>
<tr>
<td>12</td>
<td>Mesh</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>IV</td>
</tr>
<tr>
<td>13</td>
<td>Sintered</td>
<td>200</td>
<td>6</td>
<td>Water</td>
<td>IV</td>
</tr>
</tbody>
</table>
Based on the theoretical analyses it is expected that the 6 mm water heat pipes will outperform the others. Among these there will be differences expected in performance as well due to the different wick structures.

In total 13 heat pipes will be subject to various tests. For each heat pipe these tests have been listed in Table 9.1 as well. Figure 9.1 shows the heat pipes sorted on dimensions and type.

![Figure 9.1: Heat pipes used for the different tests.](image)

Figure 9.2 shows the logical flow of tests that each heat pipe will go through. Three dimensionally identical heat pipes, but with different wick structure will be subject to test I. Subsequently the heat transport limit for these will be measured under different conditions as well (tests Ia and Ib).

Then heat pipes with different dimensions and even a different fluid will be tested on their maximum heat transport capacity for comparison and verification of the theoretical analyses.

Heat pipes 8, 9, and 10, which are identical to the first three will be subject to test I as well. Hereafter, transient start up behavior will be investigated from the frozen state. Then, these same heat pipes will be exposed to several freeze/thaw cycles. Afterwards, again heat transport limit testing will be carried out to investigate the effect of these cycles on their performance. Last, they will be cut and internally inspected (test IV).

The final set of heat pipes will not be tested but only inspected internally by cutting the heat pipes in half. This in order to compare the wick structures of untested heat pipes and the ones that underwent freeze/thaw cycling.

![Figure 9.2: Flowchart showing the order of execution of the different experiments for each heat pipe.](image)
9.2. Test I Heat Pipe Performance Limit

Test I is intended to identify the maximum heat transport capacity of the heat pipe as a function of heat pipe temperature. For this test the following goals have been set to be reached upon completion:

1. Validate the theoretical performance of axial grooved water heat pipes.
2. Compare the performance of grooved, mesh, and sintered heat pipes.

The first goal is achieved by measuring the maximum heat transport capacity at different heat pipe temperatures. The second goal is accomplished by comparing the three heat pipes with different wick structures. This will mostly be a qualitative discussion.

9.2.1. Experimental Set Up and Considerations

The performance of the heat pipe is expressed as the maximum heat transport capacity as a function of heat pipe temperature. In order to measure this performance a test set up is needed in which both the heat input and heat pipe temperature can be controlled. Furthermore, data need to be logged and real-time visualization of several parameters is necessary. All these aspects are covered with the designed set up shown in Figure 9.3. The test set up consists of several elements and components, each will be addressed hereafter in detail.

**Figure 9.3:** Schematic of the test set up used for measuring the heat transport capacity limit of COTS heat pipe at different heat pipe temperatures.

Heat Source Section and Power Input

The heat source acts as the section from which heat is supplied to the heat pipe. At this section the most critical aspect one needs to consider is the transfer of heat from the used heaters into the heat pipe. This means minimizing losses by ensuring sufficient contact area between the heaters and the heat pipe and isolation of the heater section itself.

In this test set up the heat is supplied by two cartridge-heaters of 90 mm length. To ensure a uniform heat input to the heat pipe two heaters are used which are placed directly above and below the heat pipe in separate copper blocks. The length of these blocks match the length of the heaters. A cut out of the heat pipe diameter is made at one side of both of the blocks allowing the heat pipe to be clamped between them when the copper blocks are placed on top of each other. A small cavity of 1 mm between the blocks allows for sufficient clamping and ensuring a low contact resistance. The relatively long length of the blocks, compared to the heat pipe lengths, allows for a variation in heat pipe evaporator section length. The heat source section has been designed in Catia after which it has been manufactured at one of the workplaces of the NLR. Both the blocks and cartridge-heaters are shown in Figure 9.4.
HEAT SINK SECTION AND HEAT PIPE TEMPERATURE CONTROL
The heat sink comprises of two copper blocks between which the heat pipe is situated (Figure 9.5) and a temperature control device in the form of a thermo-electric cooler (TEC). The copper blocks, equal in size to the TEC plates, are clamped between the TEC, which ensures a proper heat transfer between them. Similar to the evaporator section length, the length of the heat pipe condenser section can be varied as well up to 52 mm, which is the length of the TEC plates. Again, this part was designed in Catia after which it was produced at the NLR. The TEC regulates the temperature of the heat pipe via a temperature feedback loop: The temperature at the middle of the heat pipe is measured and fed to the TEC controller which adjusts the power level of the TEC accordingly. The TEC can both cool down and heat up (via the controller), allowing a quick and precise control of the heat pipe temperature. The heat removed by the TEC is further carried away by a thermostat bath. While a thermostat bath would be sufficient on its own for controlling the heat pipe temperature, the addition of the TEC, which is able to heat up and cool down much more rapidly, allows for speeding up the measurements.

TEMPERATURE MEASUREMENTS
The final step to complete the test set up is to measure the temperature at different locations. For this, several thermocouples (TC) have been attached to the places of interest. In Figure 9.3 the TCs are represented by the black dots and show the locations where the temperatures will be measured. Two TCs are attached to the middle of the heat pipe of which one serves as feedback temperature for the TEC controller. This point is kept at the desired temperature. Two more TCs are placed at either side of these: one close to the condenser section and one close to the evaporator section. These TCs can help monitoring the thermal gradient over the heat pipe's length. This gives an indication of the functionality of the heat pipe:
If one of the TCs reads a substantial different temperature than the one from the middle it indicates that conduction dominates rather than the heat transfer through the heat pipe itself.

Finally, multiple TCs are placed in both the evaporator and condenser block. These will be used to determine the heat transfer coefficient from the interfaces to heat pipe.

**Complete Test Set Up**
The test set up, including the evaporator and condenser block, has first been designed in CATIA. The render shown in Figure 9.6 illustrates how the set up will look when these parts, the heat pipe, and the TEC are assembled.

![Figure 9.6: CAD drawing of the test set up used for heat pipe performance measurements.](image)

The white plates situated at the top and bottom of the condenser copper blocks are the plates of the TEC. The titanium top and bottom parts serve for both clamping and cooling of this section by water flowing through these parts from the thermostat bath.

The entire set up is embedded in Styrofoam to minimize heat leakage and closed off tightly with Velcro tape. The different TCs, hoses, and power wiring are routed through the Styrofoam. Figure 9.7 shows the test set up, with the heat pipe thermally isolated from the environment. Visible are also the water hoses from the thermostat bath and the different wires from the TCs.

**9.2.2. Heat Pipe Performance Test Description**
In order to carry out experiments successfully a test procedure and thorough preparation of the test set up is required. This includes making a clear test procedure and calibrating all the sensors used during testing. This way, one can make sure that the measurements are not influenced by external parameters.

The test is carried out by selecting a heat pipe and inserting it into both the evaporator and condenser section. The length of the heat pipe which is clamped in can be varied as explained before. The clamping strength should be sufficient to ensure proper heat transfer. At the predefined locations, TCs are placed to measure the corresponding temperatures. Then, the remaining voids are filled up with Styrofoam to minimize heat leakage and the set up is closed.

Before the test can be started several calibration checks need to be executed first. The accuracy of the TC sensors needs to be checked to prevent measurement read-out errors. As the ambient temperature is 20 °C the TCs should give this value as an output as well in steady-state condition. The value of each of the TCs is listed in Table 9.2. It was found that the maximum deviation is 0.12 °C from the expected and desired temperature of 20 °C. This is determined to be an acceptable deviation as an accuracy of 1 °C is deemed sufficient.
Next, the power input is checked: The given power input by the software is validated against the actual output given by the power supply. This way it is made sure that there is no off set in the expected power delivered to the heat pipe and the actual power provided. Validation was accomplished by comparing the output voltages and currents from both the software and the power supply. It was found that for all voltage and current levels these were within 1% of each other.

Another test that needs to be carried out is on heat leakage from the test set up to the ambient environment. The purpose of measuring the heat leakage is to be able to determine the exact amount of power that goes into the heat pipe during testing. Heat leakage is dependent on the $\Delta T$ between the set up and the environment, therefore it has to be measured at multiple, different temperatures. The heat leakage is measured by dis-coupling the condenser section. The heat pipe is only clamped in the evaporator section and no condenser section exists. Then the set up is closed and for three temperature points the heat input from the cartridge heaters is measured, needed to keep the temperature at the desired temperature level. This set up was devised specifically for measuring the heat leakage only at the evaporator section. The heat leakage further on at the set up is of no interest as at that point the heat has already been transferred through the heat pipe. The amount of power necessary to maintain this temperature is the heat leakage. The results are shown in Table 9.3.

Plotting these values into a graph results in a linear correlation, shown in Figure 9.8. As ambient air is
9.2. Test I Heat Pipe Performance Limit

Table 9.3: Heat leakage from the test setup for different temperatures.

<table>
<thead>
<tr>
<th>T [°C]</th>
<th>Heat leakage [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>-</td>
</tr>
<tr>
<td>40</td>
<td>0.7</td>
</tr>
<tr>
<td>60</td>
<td>1.5</td>
</tr>
<tr>
<td>80</td>
<td>2.3</td>
</tr>
</tbody>
</table>

At room temperature, any heat pipe temperature below 20 °C will result in a negative heat leakage, meaning that heat is drawn inwards from the outside. The figure shows that the heat leakage is minimal and only at higher temperatures, from 50 °C onwards, the leakage is more than 1 W. Still, this leakage has to be taken into account and will be subtracted from the measured value.

Figure 9.8: Heat leakage as a function of heat pipe temperature measured in the test setup.

Once the setup has been completed the different devices, such as the computer, heater power supply, TEC controller, and thermostat bath, are switched on. The thermostat bath’s temperature is permanently set to 5 °C. The test is then started by bringing the heat pipe to a temperature with the TEC. Once this temperature has been attained the heater power is slowly increased in steps of 2 to 4 W. After each power increase the temperatures along the heat pipe are allowed to reach a steady-state value.

The gravity test is carried out during the performance characterization test. When a 30 °C heat pipe temperature is set the entire setup will be elevated at the evaporator side. For three different power levels the vertical distance is measured at which the heat pipe temperature starts to change. This indicates the maximum deflection possible before the gravitational pressure drop becomes larger than the capillary pressure drop.

The final test for the first three heat pipes is carried out by bending each heat pipe and measuring its performance at two heat pipe temperatures. A bend of 90 ° is introduced at the middle of the heat pipe. The heat transport capacity is measured in the same way as before.
9.2.3. Test I Results
Test I consists of multiple tests. The results for each of these tests are presented and discussed separately. The results are derived from measuring the temperature at different locations of the test setup. For clarity these are shown schematically in Figure 9.9.

![Figure 9.9: Locations of the different TCs used for measuring the temperatures.](image)

Performance Testing
Figure 9.10 shows the temperature profile of the axial grooved heat pipe after performance testing. The plot shows the constancy of the heat pipe temperatures during the different set points, while the evaporator and condenser temperature slowly deviate from this for increased power inputs for each of the set heat pipe temperatures.

![Figure 9.10: Resulted temperature profile of the axial grooved heat pipe (HP8) after testing.](image)

Figure 9.11 shows a zoomed in part at a heat pipe temperature of 20 °C for one of the measurements. In Figure 9.11a the determination of the heat transfer coefficient is shown, while 9.11b shows the step-wise increase in power input.

The \( h_t \) for both sections is based on the temperature gradient between the temperature at the middle of the heat pipe, \( T_{\text{HP}} \) and \( T_{\text{evap}} \) and \( T_{\text{cond}} \). It was measured when steady-state was reached, after which the power was increased again. The power increase was done at first in relatively large steps (4 W) to decrease the total test duration, while closer to its predicted maximum the step size was reduced to 2 W. Unfortunately, it can be seen in Figure 9.11a that for the last two power increases steady-state has not been attained (marked in the figure with black circles). This will impact the measured heat transfer coefficient (explained later on). Another interesting aspect to be seen is the constant temperature of the heat pipe for this interval (and for every other temperature set point as well). Even when the temperature at the evaporator section starts to
9.2. Test I Heat Pipe Performance Limit

Figure 9.11: Zoomed in plots showing at which time the heat transfer coefficient was measured (a) (circles indicate no steady-state was achieved yet) and the power increased (b).

rise substantially, the heat pipe remains 20 °C (highlighted by the black square). This is an important feature and shows why this heat transfer coefficient is so important: A low heat transfer coefficient leads to a large temperature gradient over the heat pipe and heat source, something which is obviously undesired. The heat transfer can be improved via the methods described in Section 8.1.1.

For the grooved heat pipe for different heat pipe temperatures these measured heat transfer coefficients have been plotted for both the evaporator section (Figure 9.12a) and condenser section (Figure 9.12b) as a function of input power. The black dots show at which points the maximum heat transport capacities were determined.

Figure 9.12: Heat transfer coefficient \( h_t \) as a function of power for the axial grooved heat pipe.

The plots both show interesting features which give useful insight in the functionality and performance.
of the grooved heat pipe. At low heat inputs the $h_t$ stays fairly constant as predicted. Even though the errors induced on the measurement at low heat inputs are significantly larger due to, for example, heat leakage, which is shown by the error bars in Appendix E.2.

One of the interesting features is the behavior of $h_t$ with respect to heat pipe temperature: For higher temperatures the heat transfer coefficient increases meaning that the heat transfer will occur more efficient and lower temperature gradients will arise over the interfaces. At low heat pipe temperatures the $h_t$ is smaller meaning that heat is not easily transferred between the interfaces and the heat pipe. This, however, can be beneficial as this means that at low heat pipe temperatures a heat source will not cool down rapidly, thereby preventing it from reaching (too) low temperatures too quick. While this behavior follows logically from the given heat transfer coefficient equation, the real question is: why is the heat transfer coefficient heat pipe temperature dependent?

As the interface setup does not change this dependency must originate from the vapor and liquid behavior. At high heat pipe temperatures the vapor density increases and hence the vapor pressure increases as well. The increased vapor pressure depresses the liquid’s surface, effectively reducing its thickness and resulting thermal resistance. Simultaneously, the liquid’s conductivity drops at higher temperatures reducing the thermal resistance even more leading to smaller $\Delta T$ and thus a higher $h_t$.

A second feature to be seen from the plots is the quick drop in heat transfer coefficient at the evaporator section, which indicates the quick dry-out of this part of the wick (discussed in Section 8.1.1). This shows that the rate of vaporization suddenly overcomes the liquid return rate. The drop in heat transfer coefficient shown in the plots would be even more evident in case all the coefficients were measured at steady-state conditions. For high power levels this was not always the case, as highlighted by the large black circles in Figure 9.11a. Therefore, in most cases the last two or three $h_t$ points would lie lower.

The same measurements have been done for the sintered heat pipe (#10) and the resulting plots are shown in Figure 9.13.

![Figure 9.13: Heat transfer coefficient $h_t$ as a function of power for the sintered heat pipe.](image)

The same trend concerning the behavior of $h_t$ is seen for the sintered heat pipe as well. This holds for both the evaporator and condenser section. What is different though is the slope of the $h_t$ lines when the power increases. Where these saw an almost instant drop for the grooved heat pipe it only decreases slowly for the sintered variant. The reason for this might lie in the build up of the sintered wick: The sintered wick consists of an irregular pattern of pores, each different in size through which the liquid returns to the evaporator section. The heat input on the evaporator section will face more resistance due to the wick structure than for axial grooves. This leads to a slower rate of vaporization and thus dry-out will occur more gradual.
Although one might think that due to this slow rate of vaporization the heat pipe will function longer (and thus able to transport more power) this will not be the case as the permeability of the sintered wick also leads to a much lower liquid return-rate. The result of this phenomenon leads to a slowly decreasing heat transfer coefficient and while the heat pipe looks as if it is still operating, a large thermal gradient will arise over the interfaces. Because of the slowly decreasing $h_t$ it is also much more difficult to determine the maximum heat transport capacity of the heat pipe.

The next step is to compare the three different heat pipes with each other. 6 Heat pipes have been tested in total on their performance and their results are shown in Figure 9.14.

Figure 9.14: Experimental heat transport capacities of the three different heat pipes (1, 2, and 3) compared.

For the identical grooved and identical sintered heat pipes it holds that their performance is similar with a deviation of 5 W max, which is, considering the step-wise power increase of 2 W, which subsequently induces the same error, within close range of each other. The identical mesh heat pipes, however, performed very different from each other. Performance testing has been carried out several times and each time it proved difficult to reproduce previously found values. Furthermore, the measured heat transfer coefficient was found to be low compared to the other heat pipes and the mesh itself might be the reason for this: Although the mesh is held under its own tension against the heat pipe wall vapor can get trapped between them leading to a sudden rise in thermal resistance thereby losing performance capability, something experienced before during earlier projects at the NLR (J. van Es (NLR), personal communication, February 8, 2016). Because of the irregular performance pattern of the mesh heat pipes it was decided at this stage that the mesh heat pipe will dropped as candidate for further testing, except for the bending test.

Evident from the figure is the difference between the axial grooved heat pipes and the other ones, something which is not illogical as sintered heat pipes are known for their high capillary forces, but the permeability of the sintered wick is far worse than for the axial grooves. Furthermore, the thickness and irregular pattern of pores will increase the thermal resistance at both interfaces through which heat exchange will happen less efficient.

The Influence of Gravity on Maximum Heat Transport Capacity

The influence of gravity has been measured for the axial grooved and sintered heat pipe. At the evaporator section side the test set up has been elevated up to the point where temperature readings showed deviation from their steady-state values. Figure 9.15 shows how the entire set up was rotated.
At a heat pipe temperature of 30 °C and for 10, 20, and 30 W the tilt angle, $\alpha$ listed in Table 9.4 was measured for the axial grooved heat pipe.

Table 9.4: Measured tilt angle between the evaporator and condenser section for the axial grooved heat pipe.

<table>
<thead>
<tr>
<th>Power [W]</th>
<th>$\alpha$ [°]</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>17</td>
</tr>
<tr>
<td>20</td>
<td>9</td>
</tr>
<tr>
<td>30</td>
<td>2</td>
</tr>
</tbody>
</table>

The vertical distances are measured with respect to the entire test set up length to find the corresponding tilt angle, which subsequently can be used to find the absolute vertical distance between the evaporator and condenser section. This vertical distance, however, is dependent on the length of the liquid return path. While this can be set equal to the heat pipe’s length one can argue that the actual liquid return path is smaller due to the fact that the liquid does not necessarily need to return from the condenser end farthest away from the evaporator section as liquid is present along the entire length of the condenser section. This results in a different possible heat pipe length as shown in Figure 9.16.

While this is a valid argument, it is dependent on the amount of liquid present in the heat pipe, which is not known at this point. For the two possible lengths the gravity plots have been made and are shown in Figure 9.17.

The plot shows that the maximum heat transfer capacity decreases for an increasing vertical distance between the evaporator and condenser section. This can be explained by the fact that the returning liquid needs to work its way against gravity as discussed in Section 4.2.4. The linear relation between the heat transport capacity and vertical height is indeed covered by the gravitational pressure drop $\rho gh$, which reduces the maximum heat transport capacity for increasing vertical distances.

From the plots and with the help of Equation (9.1) (see Section 4.3.1) one can try to deduce the contact angle by extrapolating the experimental data.

$$\dot{Q} = h_{lv} \left[ \frac{\rho_l N A_w r_c^2}{8 \mu_l \epsilon_{eff}} \left( \frac{2 \sigma_l}{r_c \cos(\theta)} - \rho_l g h \right) \right]$$  \hspace{1cm} (9.1)

By extrapolation the maximum height can be found at a heat input of 0 W (also shown in Figure 9.17). This means that the left side in the aforementioned equation equals 0 and one of the terms on the right hand...
side must equal 0 as well. As the left term on the right hand side is entirely based on dimensional parameters or fluid properties it follows that the right term equals 0 and hence the gravitational pressure drop must equal the capillary pressure as shown in Equation (9.2).

$$\frac{2\sigma_l}{\cos(\theta)} = \rho_l g h \quad (9.2)$$

The corresponding heights at 0 W for both lengths, 156 and 186 mm, was found to be equal to 66.80 and 79.65 mm, respectively. Substituting these into Equation (9.2) and solving for the contact angles leads to a contact angle of 23.6 ° for a length of 156 mm while the larger length leads to a non-existent, imaginary angle. While the resulting contact angle of 23.6 ° seems plausible there is a large insecurity about what the exact liquid path length is. Furthermore, the amount of liquid also affects this as a large amount of liquid could result in a shorter path length. For these reasons it is impossible to make a valuable conclusion on the contact angle and correct liquid path length.

For the sintered heat pipe the same test was carried out and the values listen in Table 9.5 were measured. It can be seen immediately that the sintered heat pipe is far less affected by gravitation than the actual grooved heat pipe. This is something expected up front as the wick structure of the sintered heat pipe makes it less susceptible against gravity.

<table>
<thead>
<tr>
<th>Power [W]</th>
<th>$\alpha$ [°]</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>90</td>
</tr>
<tr>
<td>20</td>
<td>59</td>
</tr>
<tr>
<td>25</td>
<td>24</td>
</tr>
</tbody>
</table>

For the 10 W measurement the entire set up was placed vertically and still no affection was shown, meaning that the gravity pressure drop was still too low.
The Influence of a Bend on Heat Pipe Performance
Heat pipes 1, 2, and 3 have been bent 90° at the middle of the heat pipe and subsequently been subject to performance-limit testing to observe a possible deviation in performance. A bending radius of three times the heat pipe diameter has been applied. The bent heat pipes are shown in Figure 9.18.

Figure 9.18: A bend introduced in the middle of each of the three different heat pipes.

The maximum performance has been measured for two different heat pipe temperatures. The results are shown and compared with the straight heat pipes in Table 9.6. While a difference in maximum performance can be spotted it has to be noted that the error in performance measurements are in the order of 2 W. For the grooved and sintered heat pipes this means that a bend affects the performance up till 2 W. The mesh heat pipe does show a large drop in performance after bending. However, as was stated earlier the mesh heat pipes showed inconstant behavior during testing and therefore the drop in performance cannot be attributed to bending with full certainty. Nonetheless, looking at the three different wick types a bend is most likely to affect the mesh heat pipe as the mesh is not an integrated structure with the heat pipe wall; A bend can loosen the mesh wick from the heat pipe wall and thereby loose its capillary working.

Table 9.6: Measured maximum heat transport values in W before and after bending of the heat pipes.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>HP1 Grooved</td>
<td>18</td>
<td>16</td>
</tr>
<tr>
<td>30</td>
<td>HP1 Grooved</td>
<td>30</td>
<td>27</td>
</tr>
<tr>
<td>15</td>
<td>HP2 Mesh</td>
<td>15</td>
<td>8</td>
</tr>
<tr>
<td>30</td>
<td>HP2 Mesh</td>
<td>21</td>
<td>12</td>
</tr>
<tr>
<td>15</td>
<td>HP3 Sintered</td>
<td>16</td>
<td>13</td>
</tr>
<tr>
<td>30</td>
<td>HP3 Sintered</td>
<td>21</td>
<td>21</td>
</tr>
</tbody>
</table>

Heat Pipe Performance with Different Dimensions and Fluid
The heat transport capacities for the sintered and mesh heat pipes with a smaller outer diameter (4 mm) and the methanol heat pipe have been measured to be able to compare their performance against the others. Just as for the 6 mm mesh heat pipes the shorter axial grooved heat pipe (HP# 4) did not perform as expected and it was impossible to obtain any proper measurements. Therefore, this heat pipe has been left out. Figure 9.19 shows the performance-comparison of the 4 mm mesh and sintered heat pipes.

Clearly, the mesh heat pipes give again an inconclusive result as the heat pipe with the smallest outer diameter seems to perform just as well as the larger one. The sintered heat pipes do show an expected behavior as the decrease in outer diameter leads to a significant drop in maximum heat transfer capacity. It specifically shows that for the expected heat load of 10 W OAP the 4 mm heat pipe is not capable of removing this amount at lower temperatures.

Figure 9.20 shows the heat transport capacity of the methanol heat pipe. As expected it is able to function below 0 °C. The performance of methanol is clearly less than the water variant, something which is expected as one looks at the figure of merit of the two fluids.
9.2. Test I Heat Pipe Performance Limit

Figure 9.19: Comparison of the performance of mesh heat pipes (a) and the sintered heat pipe (b) for different outer diameters.

Figure 9.20: Maximum heat transport capacity of the methanol heat pipe.

9.2.4. Validation of Theory

The result of the axial grooved performance tests are compared with the theoretical prediction. The analyses carried out in Chapter 7, however, made some assumptions on certain parameters, such as the contact angle between the water and copper, the wick dimensions, and the exact length of the heat pipe. All of these influence the theoretical results and will therefore have to be determined accurately.

The heat pipe's length is slightly shorter than the given dimensions as both its ends are squeezed together to seal off the heat pipe during manufacturing and from the visual inspection carried out in Test IV (see Table 9.12 in Section 9.5), accurate values for the geometrical wick parameters have been determined. The contact angle has been found with the help of the results found in an earlier carried out project by NLR. In this project the contact angle was measured between a water droplet and a copper layer as a function of time. The contact angle was found to be in the order of 27° (See Appendix E), which is also supported by [49]. Table 9.7 lists the values found and used to compute the theoretical performance of the heat pipe.
Table 9.7: Accurate values determined for theoretical prediction of the heat transport capacity of the axial grooved heat pipe.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Pipe Length [mm]</td>
<td>186</td>
</tr>
<tr>
<td>Groove width [mm]</td>
<td>0.200</td>
</tr>
<tr>
<td>Groove depth [mm]</td>
<td>0.250</td>
</tr>
<tr>
<td>Groove-wall width [mm]</td>
<td>0.100</td>
</tr>
<tr>
<td>Heat pipe wall width [mm]</td>
<td>0.300</td>
</tr>
<tr>
<td>Number of grooves [-]</td>
<td>55</td>
</tr>
<tr>
<td>Contact angle [°]</td>
<td>27</td>
</tr>
</tbody>
</table>

With these accurate values the theoretical limits have been recalculated for the axial grooved heat pipe and plotted in Figure 9.21 along with the experimental results for the axial grooved heat pipes 1 and 8. It has to be noted, however, that the performance measurements of HP# 1 and HP# 8 have been a bit different: The performance of HP1 has been measured manually and by visual inspection only, while heat pipe # 8 underwent an automated, long test run in which the heat pipe was given sufficient time to reach steady-state conditions. For this reason the errorbars for HP#1 will be larger than for HP#8.

![Figure 9.21: Theoretical limits compared to the experimental found maximum heat transport capacities.](image)

Comparing theory with practice it can be seen that there is a difference present. The first important conclusion that can be drawn is that the boiling limit does not lie in the region of measured heat pipe temperatures. This was already expected as it was already stressed that the boiling limit is dependent on the nucleation radius which can vary significantly in magnitude (see Section 4.3.5). The existence of the boiling limit is thus shown to be only possible for heat pipe temperatures higher than 80 °C.

The confining limit is thus indeed the capillary limit but why does this difference exist? The manufacturer notes maximum transport values of 45 to 55 W for the axial grooved heat pipe at a heat pipe temperature of 70 °C (343 K), although it is stressed that these values are much dependent on the set up of the user [50]. If one looks at HP# 8 it can be seen that this comes close to the measured value, but still lies above the theoretical prediction.

The answer for this difference should lie in the equation describing the maximum heat transfer due to the
capillary limit, which is here repeated for clarity.

\[ \dot{Q} = h_{lv} \left( \frac{\rho_l A_w R^2}{8 \mu_l \ell_{eff}} \right) \left( \frac{2 \sigma_l}{r_c \cos(\theta)} \right) \] (9.3)

Here, the vapor pressure drop and the pressure drop due to gravity are neglected. If one looks at all the parameters describing this limit then only the parameters describing the groove dimensions and the contact angle are the ones which can vary from the earlier found values. The contact angle was experimentally determined and is supported by other sources. However, dependent on the manufacturing process it is not unlikely that this angle is somewhat better than predicted. Concerning the groove dimensions, measurements of the wick photos show an almost unique groove-width and depth for each groove.

All these parameters pose uncertainties on the theoretical prediction, moreover their effect is hard to verify in practice. Still, the capillary limit has been recalculated for different values of these parameters. The result is shown in Figure 9.22, where the groove-width and depth and contact angle are all varied.

![Figure 9.22: Sensitivity analysis on the capillary limit by varying the groove dimensions.](image)

The plot shows that for a minimal increase in groove width and depth the capillary limit already changes significantly, which marks the sensitivity of both of these parameters. The difference between theory and practice can partly been attributed to these parameters. The change in contact angle shifts the capillary limit even further up, thereby closing the gap between theory and practice. Next to this the liquid-vapor ratio present in the heat pipe can also influence the heat transport capacity as this determines the vapor and liquid pressures inside the heat pipe.

The possible reasons presented are all likely to account for the difference between theory and practice. However, it remains difficult to point the exact cause of the difference present, due to the existing variation in the groove-widths and depths, the difficulty of measuring the contact angle, and the possible influence of the amount of liquid. Therefore, further investigation is required to identify the cause, which falls outside the scope of this project.
9.3. **Test II Transient Start Up Behavior of a Frozen Heat Pipe**

The second test that is carried out investigates the start up behavior of the heat pipe when partly frozen. As the satellite's exterior can reach temperatures below freezing point of water it is important to understand the behavior of the heat pipe under these conditions. The following goals have been established for this test:

1. Determine the start up time of frozen heat pipes without any heat input.
2. Determine the behavior of and thermal gradient over the heat pipe when 10 W heat load is applied in the frozen state.
3. Compare the start up time for the grooved and sintered heat pipes.

### 9.3.1. Test Set Up of the Transient Start Up Behavior Test

For this test, the same set up is used as was employed for Test I except for one difference: The temperature sensor used by the TEC has been detached from the heat pipe and placed into the copper condenser block (See Figure 9.23). This means that the TEC controls the temperature of the condenser block, just as would be the case for the condenser end in a CubeSat that gets below freezing point of water.

![Figure 9.23: Schematic of the test set up used for the transient start-up test.](image)

### 9.3.2. Transient Start Up Test Description

The test is prepared by setting the thermostat bath to a temperature of -10 °C, which will aid the TEC in cooling the section. The TEC cools the condenser block down to -10 °C. After steady-state temperatures are reached both the TEC and thermostat bath are switched off to prevent further cooling of the condenser block. For the first test no heat input is given and the time needed for the heat pipe to reach 0 °C is measured. For the second test a heat input of 10 W is given to see if this accelerates the start up of the heat pipe and what the temperature gradient will be before successful start up is achieved.

### 9.3.3. Test II Results and Discussion

Figure 9.24 shows the temperature profile for both the axial grooved and sintered heat pipe during the transient start up test. The first interesting feature to be noticed is the temperature gradient over the heat pipe. One would expect that the entire heat pipe eventually would cool down uniformly to -10 °C as no heat input is given and the TEC maintains this temperature at the condenser section. What can be seen, however, is that at steady-state conditions (Figure 9.24a) a temperature gradient is present over the heat pipe and the evaporator section remains 8.96 °C.
9.3. Test II Transient Start Up Behavior of a Frozen Heat Pipe

Figure 9.24: Start up behavior of the axial grooved (a) and sintered heat pipe with no heat applied (b).

So, why does the heat pipe show a different temperature profile than expected? One thing that has been touched upon earlier is the heat leakage from the test set up to the ambient environment (see Section 9.2.2). However, this obviously also works the other way around when the test object is at a lower temperature than the environment. From linear extrapolation of the values from Table 9.3 it is found that at -10°C 0.35 W of heat is drawn inwards at the heater section. This must mean that an equal amount of heat must be conducted away through the heat pipe. This situation is visually shown in Figure 9.25 with the equivalent thermal network.

Figure 9.25: Nodal network of the transient start up test explaining the steady-state temperature profile.

Conductive heat transfer will occur according to Equation (9.4), which is a combination of Equations (3.2) and (3.3).

\[ Q_{ij} = \frac{kA}{l} \Delta T \]  

Here, \( A \) is the cross-sectional area of the part through which the heat is transferred. For the axial grooved heat pipe this area consists of the heat pipe wall and 'teeth' between the grooves. This area is calculated as follows:

\[ A_{hp} = \pi r_o^2 - \pi r_i^2 + N t_\omega t_\delta \]  

Here, \( t_o \) and \( t_\delta \) are the width and height of the teeth between the grooves, which have been determined from the wick inspection (see Table 9.7 from Section 9.5). Filling in all the known parameters results in a heat pipe copper cross-sectional area of 7.22 mm².

With both equations the heat transfer through the heat pipe by pure conduction can be calculated, dependent on the heat path length and the temperature difference at the boundaries. Using the values for the different
parameters listed in Table 9.8 the conductive heat transfer between these nodes is found to be equal to 0.43 W. This is interesting as this means that only this amount of power is needed to keep the evaporator section at a temperature of 10 °C and prevent the heat load of freezing over.

**Table 9.8:** Parameters and values for the calculation of the conductive heat transfer through the heat pipe.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k$</td>
<td>390</td>
<td>Conductivity copper [WK$^{-1}$]</td>
</tr>
<tr>
<td>$l$</td>
<td>126</td>
<td>Conductive length of heat pipe [mm]</td>
</tr>
<tr>
<td>$A$</td>
<td>7.22</td>
<td>Heat pipe copper cross-sectional area [mm$^2$]</td>
</tr>
<tr>
<td>$T_{evap}$</td>
<td>8.96</td>
<td>Evaporator temperature [°C]</td>
</tr>
<tr>
<td>$T_{cond}$</td>
<td>-10.31</td>
<td>Condenser temperature [°C]</td>
</tr>
</tbody>
</table>

Comparing this to the extrapolated heat leakage value of 0.35 W it is obvious that there is a small difference. However, the heat leakage has been measured at a different point in time and many tests have been ran afterwards, therefore, it is not unlikely that the heat leakage has somewhat changed. Moreover, steady-state conditions are only reached after a long period which increases the time duration of each test significantly. It is plausible that the measured temperatures are not entirely steady-state yet. Both these aforementioned reasons can explain the small difference in the measured and calculated heat leakage value.

With the conductive heat transfer known from evaporator to condenser section the temperature at any point along the heat pipe can be computed and compared to the experimentally found values. **Table 9.9** shows the temperatures computed with ThermXL and the found experimental temperature values, with the aforementioned heat leakage of 0.43 W taken into account.

**Table 9.9:** Theoretical and experimental temperatures at different locations along the heat pipe.

<table>
<thead>
<tr>
<th>TC [K]</th>
<th>Theory</th>
<th>Experimental</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{evap}$</td>
<td>8.92</td>
<td>8.96</td>
</tr>
<tr>
<td>$T_{hp_e}$</td>
<td>6.18</td>
<td>7.03</td>
</tr>
<tr>
<td>$T_{hp_c}$</td>
<td>-0.69</td>
<td>1.43</td>
</tr>
<tr>
<td>$T_{hp_m}$</td>
<td>-6.80</td>
<td>-7.34</td>
</tr>
<tr>
<td>$T_{cond}$</td>
<td>-</td>
<td>-10.31</td>
</tr>
</tbody>
</table>

The theoretically found temperatures do explain the temperature gradient over the heat pipe, although a small difference is present between the theoretical and experimental values. This small difference, however, can be attributed to the fact that the temperature at any point along the heat pipe depends on the exact location of it. For example, $T_{hp_e}$ should be located exactly in the middle but the sensor’s location might be a few millimeters off, which alters the length of the heat load path and thus the resulting temperature. The temperature as a function of heat pipe length is shown in Figure 9.26.

The interest in this test lies in the total start up time of the heat pipe, which is defined as the time needed until the condenser section reaches a temperature of 0 °C. With the heat leakage at the evaporator end and the conductive heat transfer through the heat pipe known, the next step is to calculate the time it takes for the condenser section to reach 0 °C. For this a transient simulation needs to be performed which takes into account the thermal capacity, $c_p$ (385 JK$^{-1}$) and mass of the copper condenser section (0.298 kg).

From the analysis it is found that after 3779 s or 62.98 min the condenser section reaches 0 °C. If this is compared with the start up time found from the measurement (25.07 min), one can see that this is way off. Apparently the condenser section heats up faster than expected.

When the condenser section and thermostat bath are switched off there is no external heat source that influences the test result. Furthermore, the heat pipe is not functioning as the water is frozen. Thus, the only cause for the faster start up time is again heat leakage, but now at the condenser section. Looking at the test set up this is supported by the fact that the set up is less isolated from the outside world than the evaporator section. Unfortunately the heat leakage at this end has not been measured. However, by reverse engineering one can determine the heat leakage at this end and use this to validate the start up time with 10W heat leakage. The found heat leakage at the evaporator end for which the start up time equals 25 min (see Table 9.10) is 0.48 W.
9.3. TEST II TRANSIENT START UP BEHAVIOR OF A FROZEN HEAT PIPE

Figure 9.26: Steady-state temperature profile of the heat pipe at the transient start up test.

Considering the test set up and the earlier found heat leakage at the evaporator section this value is plausible.

Continuing with analyzing the test results: After 0 °C has been reached by the condenser section the heat pipe immediately starts functioning as can be seen from the sudden irregular temperature behavior of the heat pipe: As soon as the liquid thaws at the condenser section the thermal cycle is restarted and heat is drawn from the other sections. This is visualized by the drop in temperature at the different heat pipe locations. This process speeds up the temperature increase at the condenser section, which is clearly visible as well. Finally, the evaporator section’s temperature starts to drop (has largest mass) and the heat pipe attains a uniform temperature. This process is identical for both heat pipes, although the temperatures for the sintered heat pipe are different, which is due to the fact that steady-state has not been reached at the point the test was started.

From the first part of this test it can be concluded that when the heat pipe is frozen it will only thaw through pure conduction and the heat pipe immediately starts functioning as soon as the condenser section attains a temperature of 0 °C. The time to thaw is dependent on the temperature difference between the heat source and heat sink and affected by external heat inputs.

The next step is to look at what happens when the heat pipe is frozen and the heat source starts to dissipate 10 W heat load. The outcome of this determines whether any constraint is lain upon the duty-cycle of the payload: From earlier start up tests (see Section 5.2) it was found that successful start up was possible but gave rise to a severe thermal gradient between the evaporator and condenser section before the heat pipe was fully thawed. This is important for the case that is investigated here as the temperature of the electronic chip should remain within the chip’s limits.

Figures 9.27a and 9.27b show the start up behavior with a 10 W heat load of the axial grooved and sintered heat pipe, respectively. Comparing the grooved and sintered start up profile it can be seen that they follow the same temperature profile, except have different temperature values, which is again due to the fact that steady-state has not been reached yet (due to time constraints).

Comparing the start up time with the case when no heat is added one can see that start up is sped up significantly (Table 9.10). This is logical as the temperature elevation at the heat source section will increase the thermal gradient and subsequently increase the conductive heat transfer. The temperature elevation at the heat source section is especially important as the payload chip might become too hot when thawing of the heat pipe takes too long. The results, however, show that the heat source gets to a maximum of 50 °C, which is deemed within limits.

Again one has to keep in mind that heat leakage affect this outcome. This will also have its influence on the temperature that the heat source attains before complete thawing is achieved. The determined heat
leakage at the evaporator end is used here as well to check whether the start-up time matches for both cases. The found start-up time, with this heat leakage for the 10 W heat load case is 16.8 min. This value is close to the measured value and indeed confirms that heat leakage along the entire test set up significantly influences the outcome. Therefore, it is necessary to analyze the worst case in which the entire heat pipe is -10 °C and no external heat input is present except for the 10 W. With the correct conductivity values now known this simulation is carried out in ThermXL where all the nodes are set to -10 °C and node 1 (evaporator section) receives a heat input of 10 W. The resulting start up time is 31.3 min and the evaporator temperature rises to 51.74 °C, which is acceptable. The start-up time is twice the measured value, but interestingly enough the evaporator's temperature has not rose much higher than was measured experimentally with heat leakages.

The final check is performed by analyzing the temperature of the evaporator section before start-up is achieved with more realistic mass values and heat pipe length as will be used in CubeSats (see Chapter 10). The mass of the condenser and evaporator section are 7 and 20 g, respectively. For three cases the temperature of the evaporator section and the start up time have been simulated with the conduction values found from the experimental measurements above. Table 9.11 lists the results, which show that for all cases the temperature of the evaporator section stays below 90 °C. The low thermal capacitance enables a fast heat transfer, which is reflected in the start-up times. One thing that has to be kept in mind is that here it is assumed that the heat is directly input on the evaporator section without any heat transfer coefficient. In reality temperatures will be slightly higher due to additional parameter. However, the simulations ran here are for steady-state conditions where the payload is shut off for a longer period, something which is not likely to happen often during a mission, except during launch phase and orbit insertion.

So, what does all this mean for the application of a heat pipe in a CubeSat when temperatures below 0 °C are attained? The heat pipe will not function anymore as soon as the temperature drops to 0 °C and thawing is only achieved through conductive heat transfer between the heat source and heat sink, which is a relatively slow process compared to the heat pipe's functionality. For start up purposes this may be a disadvantage, but when the outer panels cool down and freeze the condenser section it also means that the payload does not cool down to quickly to temperatures below 0 °C as well.
Table 9.11: Theoretical start up times and temperatures for attained in LEO space conditions with a 10 W heat load.

<table>
<thead>
<tr>
<th>$T_{\text{cond}}$ [°C]</th>
<th>$T_{0\text{evap}}$ [°C]</th>
<th>$T_{\text{evap}}$ [°C]</th>
<th>$t_{\text{start-up}}$ [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10</td>
<td>-10</td>
<td>65.54</td>
<td>71</td>
</tr>
<tr>
<td>-30</td>
<td>-10</td>
<td>81.65</td>
<td>91</td>
</tr>
<tr>
<td>-30</td>
<td>-30</td>
<td>87.60</td>
<td>81</td>
</tr>
</tbody>
</table>

The temperature attained by the evaporator section stays within acceptable limits of the PCB and its components, showing that with an instant heat load of 10 W the heat pipe is able to successfully start up even in the most extreme condition.
9.4. Test III Freeze/Thaw Cycling of COTS Heat Pipes

An experiment is conducted that applies a predetermined number of freeze/thaw cycles to several heat pipes. This represents the LEO environment in which temperatures can fluctuate between temperatures below and above the freezing point of water. This test has the following goal:

1. Investigate the effect of freeze/thaw cycles on heat pipe performance.

9.4.1. Set Up of the Freeze/Thaw Cycling Test

The same set up is used as for test II, except for the fact that the measurements will be automated.

9.4.2. Freeze/Thaw Cycling Test Description

After the standard performance measurement of heat pipes 8 and 10, they will be subject to 10 consecutive freeze/thaw cycles after which the temperature of the heat pipe is elevated to 30 °C. Subsequently, a heat load just below their maximum transport capacity is supplied by the heater. This procedure is repeated 10 times and is automated as follows:

1. Setpoint = -15 °C [900 s]
2. Setpoint = 10 °C [900 s]
3. Cycle 10 reached? Yes, continue. No, go to [1]
4. Setpoint = 30 °C
5. Heater power = 20 W for axial grooved (15 W for sintered) [900 s]
6. Heater power = 0 W. 100 cycles reached? Yes, end test. No, go to [1]

The total number of cycles is based on the results found in an earlier cycling test carried out by [40] (see also Section 5.2), but is mostly constraint by the time it takes to complete the test. For these reasons a total number of 100 cycles was chosen.

9.4.3. Test III Results and Discussion

Figure 9.28 shows the temperature plot obtained from the test. The temperature profile shows neatly the execution of the 100 cycles and the intermediate performance tests.

Figure 9.28: Heat pipe temperatures due to the freeze/thaw cycles performed on the axial grooved heat pipe.

Figure 9.29 shows a zoomed plot from 10 cycles during the test. The temperatures show that the heat pipe only attains a temperature below 0 °C from halfway its length onward (Similar to the temperature gradient seen along the heat pipe during the start up test.). However, as soon as part of the condenser section gets below 0 °C all the liquid will freeze: Vapor that arrives at the condenser section will condensate and immediately...
freeze, preventing any liquid to return to the evaporator section. Therefore, the heat pipe can be regarded as frozen. From the consistency of the measured temperatures during the entire test it seems that the heat pipe is unaffected by the freeze/thaw cycles.

![Figure 9.29](image)

**Figure 9.29:** Detailed plot of 10 cycles and the temperatures of different heat pipe locations.

Figure 9.30 (sintered heat pipe) shows the same pattern as for the axial grooved heat pipe. Here, only 45 cycles are shown as data logging afterwards was done improperly. Eventually, 82 cycles were carried out (before the software crashed). The temperature results show a similar pattern to the axial grooved heat pipe, although the middle of the heat pipe does not reach the same low temperature. Still, freezing is accomplished as the $T_{hp}$ reaches a temperature of $-4^\circ C$.

![Figure 9.30](image)

**Figure 9.30:** Heat pipe temperatures due to the freeze/thaw cycles performed on the sintered heat pipe.

Just as for the axial grooved heat pipe the sintered heat pipe does not seem to be affected by the freeze/thaw cycling as temperatures remain the same even after so many cycles. The insensitivity to freeze/thaw cycling is also confirmed by visual inspection of the condenser section. The axial grooved heat pipe, shown in Figure 9.31, shows no external physical damage.
Although externally no damage is spotted it does not mean that internally no damage is present either. A wick inspection is done by cutting the heat pipes at the condenser section. The resulting pictures (see Section 9.5.2), however, do also not show any physical damage on the inside. The final check to see whether performance degradation occurred is to test the heat pipes once more. This is done by performing the gravity tilts again of which the results are shown in Figure 9.32.

Comparing the measured values with the earlier measured values one can see that there is no significant difference to be spotted for the grooved heat pipe. For the sintered variant, however, there is. Although earlier measurements and wick inspection did not indicate that the freeze/thaw cycling has an effect on the sintered heat pipe apparently the tilting does: For a 10 W heat input the earlier sintered heat pipe could be orientated vertical, while this is not the case here. Similary, for a heat input of 15 W the test set up has a maximum height of 228.0 mm, while this is roughly equal to the value found for a 25 W heat input before freeze/thaw cycling. There are a couple of remarks to be made concerning the comparison between the measured vertical heights: First of all, two different sintered heat pipes have been tested, which means that their performance might have differed somewhat from the start (something which is supported by the found heat transport capacities), because of a slightly different build up of the sintered wick structure. Second, the first test carried out was based on a change in evaporator temperature, while with the latter the heat transfer coefficient was taken into account as well. Both these remarks lead to a significant insecurity on the found results in the gravity test.
The overall conclusion regarding the effect freeze/thaw cycles have on heat pipe performance is that the axial grooved heat pipe is not affected up till at least 100 cycles. The sintered variant is more likely to be affected as the small sintered pores are likely to be damaged more easily by freezing of the water. However, as the anti-gravity test results and visual inspection are contradictory on this no conclusions can be made on the effect of freeze/thaw cycling and more investigation is necessary.
9.5. Test IV Visual Inspection of the Wick Structures

In order to compare theoretical data with experimental data it is necessary to have knowledge of the different parameters of the internal wick structure of the different heat pipes. With these known accurate calculations can be made. However, as these specifications are not listed or known by the manufacturer (N. Katenbrink (Quick-Ohm), personal communication, January 7, 2016) an alternative method is necessary.

9.5.1. Wick Inspection Test Set Up and Description

Wick inspection requires visual access to the inside of the heat pipe. However, simply sawing the heat pipe in half will damage the inner structure and therefore further processing is necessary. A small is piece is taken as a sample from the heat pipe and the end that is going to be inspected is sanded. Then, this sample is cast into a two-component resin under vacuum. After further finishing the sample is ready to be inspected under the light-microscope and photos can be taken. Figure 9.33 shows the cast-process and the light-microscope used to provide the photos for wick inspection. This process has been applied to each chosen heat pipe and carried out by one of the workshops of the NLR.

![Figure 9.33: Casting of heat pipe sample in vacuum (a) and the light-microscope used to observe the sample and create photographs (b) (Courtesy NLR).](image)

9.5.2. Test IV Test Results of the Untested Heat Pipes

The untested heat pipes (# 11, 12, and 13) have been inspected and the results are shown in Figures 9.34, 9.35, and 9.37. Figure 9.34 shows the grooved wick structure both completely as well as in detail. On the photos the grooves are clearly visible and also the relatively constant geometry of the grooves, indicating that these heat pipes are manufactured quite well. The scale in the photographs allows for direct measurement of the width of the grooves, the wall between the grooves, and the wall of the heat pipe. Furthermore, the number of grooves can be determined as well. The measured values from these photographs are listed in Table 9.12.

![Figure 9.34: Axial grooved wick structure of the untested heat pipe.](image)
Table 9.12: Measured dimensions of the wick structure of the axial grooved heat pipe.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \omega )</td>
<td>0.200</td>
</tr>
<tr>
<td>( \delta )</td>
<td>0.250</td>
</tr>
<tr>
<td>( t_{\omega} )</td>
<td>0.100</td>
</tr>
<tr>
<td>( t_{\delta} )</td>
<td>0.275</td>
</tr>
<tr>
<td>( t_{w} )</td>
<td>0.300</td>
</tr>
<tr>
<td>( N )</td>
<td>55</td>
</tr>
</tbody>
</table>

The second heat pipe that is cut is the mesh version of which the cross-section and a zoom of the mesh wick is shown in Figure 9.35. The figures reveal that the mesh consists of a single layer held against the heat pipe wall by its own tension. The cross-section shows clearly where the mesh layer starts and ends. The zoomed-in photograph visualizes in detail the single layer and the different fibers running in axial direction.

![Figure 9.35: Wick structure of the untested mesh heat pipe.](image1)

The mesh layer itself can be seen even better in Figure 9.36, where it is partly taken out of the heat pipe. The mesh is a finely woven metal cloth that is held under tension by its own structure. Unfortunately, it is not possible to retrieve the exact properties of the mesh from these pictures only, which makes it impossible to perform theoretical calculations on its performance.

![Figure 9.36: Mesh layer sticking out of the heat pipe.](image2)

The last heat pipe made ready for visual inspection is the sintered heat pipe. Figure 9.37 shows the sintered wick. Clearly visible is the thick internal wick layer which comprises of fused metal particles. Compared to the other two wick structures the sintered wick is relatively thick and occupies a large part of the heat pipe. This reduces the vapor cross-sectional area significantly.
Just as for the mesh heat pipe it is impossible to determine the wick properties of the sintered version from the photos only. The sintered structure consists of an irregular pattern of sintered copper and pores making it impossible to retrieve the porosity and permeability of the wick, with which its performance could be calculated (See Section 4.2.2).

Finally, the three different heat pipes are lain next to each other to show the difference between the wick structure. The cross-sections are shown in Figure 9.38. The figure shows the fine grooves, the looseness of the mesh wick, and the reduction in vapor area of the sintered wick compared to the others.

Figure 9.38: Cross-sections of the axial grooved, mesh, and sintered heat pipe.

9.5.3. TEST IV TEST RESULTS OF THE FREEZE/THAW CYCLED HEAT PIPES
Next to the inspection of the untested heat pipes, the heat pipes that underwent the freeze/thaw cycling test (see Section 9.4) are also inspected. Freeze/thaw cycling can place a heavy burden on the structural integrity of the heat pipe (see Section 5.2). It is therefore interesting to investigate the effect of continuous freezing and thawing on the internal wick structures.

Figure 9.39 shows the wick of the axial grooved heat pipe after the freeze/thaw cycling test. The wick does not show any sign of damage as all the ‘teeth’ and the heat pipe wall are identical to the ones before freeze/thaw cycling (Figure 9.34). Something which is also endorsed by the performance check carried out after the freeze/thaw cycling test. The photos do show that the heat pipe is slightly bent and not a perfect circle anymore. The grooves themselves are also not identical in dimensions if one closely looks at the photos. While this can be due to the bending of the heat pipe itself it can also be seen from the untested axial grooved heat pipe that its grooves are not perfectly identical as well.

The sintered wick, shown in Figure 9.40, displays a similar image to the axial grooved wick: There seems to be no real damage or loose parts that could indicate affection due to freeze/thaw cycling. What is interesting
9.5. Test IV Visual Inspection of the Wick Structures

Figure 9.39: Wick structure of the axial grooved heat pipe after freeze/thaw cycling.

tough is to see how this wick structure differs from the other sintered wick structure in terms of pore sizes and distribution of the sintered material. This highlights the difficulty of theoretically analyzing the performance of sintered heat pipes.

Figure 9.40: Wick structure of the sintered heat pipe after freeze/thaw cycling.

9.5.4. Discussion on Wick Inspection
Different heat pipes have undergone visual inspection by cutting them in half. The inspection of the axial grooved heat pipe allowed accurate determination of the different wick parameters, while this was impossible for the mesh and sintered variants.
The inspection of the two heat pipes that were freeze/thaw cycled showed no damage to the wick structure. What did became apparent was the difference in wick structure between the sintered heat pipes, which highlighted the difficulty of determining accurate wick parameters.
9.6. CONCLUSION ON THE PERFORMANCE CHARACTERIZATION TESTS
The different tests carried out have shown that the axial grooved and sintered heat pipe are capable of transporting the required heat loads. The mesh heat pipes showed irregular test behavior and the mesh wick was therefore deemed not suitable for further testing.

The measured heat transfer coefficients showed a heat pipe temperature dependency, beneficially for the CubeSat environment. Another important finding was the effect clamping strength has on this parameter: During testing a significant different in $h_t$ was seen when the heat pipe was not thoroughly clamped.

Bending was found to have negligible effect on heat pipe performance while the gravity tilt-tests illustrated why sintered heat pipes are the best choice in anti-gravity orientation.

The grooved and sintered heat pipes showed that they were able to successfully start up from frozen conditions and withstand repetitive freeze/thaw cycling, although the effect of these on the sintered heat pipe performance remains inconclusive.

The next step is to identify the heat pipe's performance in a CubeSat structure according to the design presented in Section 8.2. The performance identification with this integration, which includes bends and different interfaces, will have to proof the actual suitability of employing heat pipes in CubeSats.
Experiment II Heat Pipe Integration and Performance

In the previous chapter the performance of each individual heat pipe has been characterized. The next and final step is to integrate the heat pipe in a CubeSat structure and test its performance with a heat source and interfaces similar to what can be expected in the next-generation CubeSats. This will aid in determining the suitability of employing heat pipes in CubeSats. Section 10.1 will discuss the experimental set up that is going to be used to test the heat pipe's functionality in a CubeSat and will describe the procedure used to carry out the test. Section 10.2 will present the results and make conclusions on these. Then, in Section 10.4 a start up test from the frozen state is executed and a qualitative comparison with the earlier carried out transient start up tests is made. Finally, conclusions on the testing is given in Section 10.5.

10.1. Testing the Functionality of a Heat Pipe Integrated into the CubeSat Structure

The performance characterization of the heat pipes has revealed the maximum heat transport capacities that can be reached for different heat pipe temperatures. From literature (see Section 8.1.1) it was found (and confirmed during performance testing) that the interfaces between the heat pipe and heat source and sink play a crucial role in the heat transfer. Therefore, it is important to test the heat pipe in the environment similar to what it is going to operate in. The test that is going to be carried out has the following goal:

1 Proof of concept of the heat pipe functionality in CubeSats

This goal is achieved by carrying out a test which will measure the temperature of a heat generating chip inside a CubeSat, both with and without a heat pipe integrated.

10.1.1. Experimental Set Up and Considerations

A heat pipe will be integrated into a 2U CubeSat structure and tested in a thermally controlled environment with a heat source capable of generating 10 W heat load. The equipment and components necessary to realize this are described hereafter.

Heat Pipe

The theoretical analyses and experimental performance characterization of the different heat pipes have shown that a heat pipe is required with an outer diameter of at least 6 mm to meet Req. 1.2.1.1. This means that when the heat pipe needs to pass an above laying PCB a small cut-out will be necessary.

The type of wick that shows most potential is determined from the performance comparison computed in Figure 9.14. From this the mesh heat pipe is regarded unsuitable due to the fact that it showed irregular behavior during testing and performs the least of the three. The axial grooved heat pipe outperforms the sintered heat pipes significantly, but both heat pipes are capable of meeting the requirement on the heat load that needs to be removed. Furthermore, an aspect that is playing a crucial role during testing is the influence...
of gravity. From the gravity tests it was found that the sintered heat pipe performs best. Considering the orientation and bends required to integrate the heat pipe and to limit the influence of gravity on the heat pipe performance it is chosen to use the sintered heat pipe.

The heat pipe length is determined by the location of the heat source and the heat sink. In this case the chip is located at the middle of the PCB. The distance to the heat sinks is dependent on the location of the PCB in the CubeSat stack. As the interest lies on testing at the worst case this location is chosen to be furthest away from the heat sinks. To connect the heat source with the heat sinks a heat pipe length of 300 mm is needed.

HEAT PIPE INTERFACES
The limited room available within the CubeSat limits the freedom of integrating a heat pipe into the platform. In Chapter 8 the choice for using the structural frame ribs as heat sink was explained. To ensure a high heat transfer coefficient, the contacting area between the frame rib and the heat pipe is maximized by employing a simple rectangular element in which the heat pipe is slit (Figure 10.1a). This piece is then clamped by using an aluminum strip that uses the present screw-holes which are normally occupied by the screws which in turn attach the outer panels to the frame structure.

![Condenser sections](image1)
![Evaporator section](image2)

**Figure 10.1:** Customly manufactured copper heat sink (a) and source interfaces (b) by the DEMO workshop at the TU Delft [51], used for integrating the heat pipe into the CubeSat structure.

The generated heat by the heat source needs to be transferred to the heat pipe as efficient as possible to prevent the occurrence of a large thermal gradient. For this reason a copper plate fitting the dimensions of the heat source and heat pipe is designed to act as an efficient interface (Figure 10.1b). While aluminum might seem a better candidate due to a three times lower density, copper has a thermal conductivity that is more than two times larger than aluminum. Therefore the benefit of using aluminum for mass savings is limited. Furthermore, designing an optimized interface is outside the scope of this project. The copper plate contains two diagonal ducts that allow for the placement of a 4 or 6 mm heat pipe (At the time this element was manufactured the performance characterization tests were not carried out yet, therefore it was not known which heat pipe diameter would be required.). The heat pipe is clamped with two simple aluminum strips, as shown in Figure 10.2.

![Complete interface between the heat source and the heat pipe.](image3)

**Figure 10.2:** Complete interface between the heat source and the heat pipe.
PAYLOAD REPRESENTATION AND HEAT DISSIPATION
The payload is represented by a single, custom designed PCB. A preliminary PCB layout was designed, specifying the required components and dimensions, after which a detailed schematic was made by E. Timmer (ISIS). PCB population of the PCB was done further in-house at ISIS by the integration team. The PCB houses a heat generating surface consisting of 12 resistors. The set of resistors represents a heat dissipating chip with an area of 20x20 mm and is able to dissipate over 10 W. Screw holes are located at the corners to clamp the copper heat sink to the PCB.

Four TCs (indicated in 10.3b) are integrated in the circuit and measure the temperature, while 9 external TCs can be connected to the board as well. By allowing connection of external TCs to the PCB itself, it can be used independently from the thermal chamber or any other test set up.

![PCB Schematic](image1.png)

![PCB](image2.png)

**Figure 10.3:** Schematic of the in-house designed PCB at ISIS (a) and its real-life counterpart (b).

Software was written in-house by P. Botma (ISIS) to power and control the circuit and the resistors. It furthermore enabled reading, logging, and outputting the measurement data for inspection and data analysis afterwards.

TEST FACILITY
The test is carried out by using one of the thermal chambers of ISIS. The thermal chamber that is used is able to reach temperatures above and below 0 °C, but not to create a vacuum. This means that convective heat transfer is present and will play a role if no countermeasures are taken. The chamber tracks the humidity of the air inside to prevent condensation which could be destructive for the test object.

In space convective heat transfer is absent and for the replication of a LEO environment a chamber which lacks the possibility of creating a vacuum seems a huge disadvantage. However, the test object is designed such that forced convection will not play a role: The internal structure of the CubeSat is closed off by the outer panels and these panels only need to attain a boundary temperature value, as the interest lies on the internal temperatures. Therefore, it does not matter how this boundary temperature is achieved and forced convective heat transfer does not need to be taken into account. Internally, the influence of radiation and natural convection is mitigated by the use of flexible Styrofoam which will completely fill up the remaining empty space inside the CubeSat.

INTEGRATION IN THE CUBE SAT ENGINEERING MODEL
The PCB is integrated in a standard CubeSat PCB stack, which is subsequently placed inside a 2U CubeSat model. The empty space inside the CubeSat is filled up with flexible Styrofoam which will limit the occurrence of natural convection and radiation inside the satellite. Finally, cables are routed through small cut-outs in the panels and the test object is placed inside the thermal chamber. The test object is elevated and isolated
from the chamber floor to limit conductive heat transfer and aid in achieving a uniform temperature over all the panels of 40 °C. The integration is shown in Figure 10.5.

For the second part of the test the heat pipe is integrated into the CubeSat structure according to the design presented in Chapter 8. The heat pipe has been bent with the help of a pipe-bending tool, which allows for the introduction of a gradual bend with a radius of four times the heat pipe’s diameter. Unfortunately, the tool flattened the heat pipe significantly in the bend itself. The actual integration proved to be difficult and further bending and twisting was inevitable.

The ends of the heat pipe were slit into the designed sinks and attached to the frame ribs with simple, aluminum stiffeners. The clamping of the heat pipe to the heat sink covering the resistors was achieved by the usage of two small aluminum strips and bolts and nuts. The heat pipe was not clamped as tight as desired because of the fear of breaking the PCB. The final step consisted, again, of applying flexible Styrofoam to fill up the remaining voids in the structure to prevent natural convection and radiation. The complete integration can be seen in Figure 10.6.

10.1.2. TEST DESCRIPTION

From the different orbit simulations carried out (See Appendix B) it was seen that with an internal heat load of 10 W the outer panel temperatures range from 30 to 50 °C, dependent on the LTAN and attitude mode.
10.1. Testing the functionality of a heat pipe integrated into the CubeSat structure

For testing a random tumble is assumed such that a temperature profile is achieved as for the 12:00 h LTAN orbit, except that now all panels will have the same temperature along the orbit. Along the orbit the highest temperatures are reached when the satellite is about to enter eclipse (just below 40 °C). This is marked as the worst case, even though at a fixed attitude and in a 6:00 h LTAN orbit the temperature of one of the panels will reach a higher temperature. It is easier, however, to design for this situation from a thermal point of view as the other panels have a substantially lower temperature, moreover, the satellite does not enter eclipse and is thus of less interest when exploiting water heat pipes.

For the test a random tumble is assumed such that all panels attain a uniform temperature and a temperature of 40 °C is chosen to represent this worst case scenario. The test will comprise of two parts: The first part of the test is carried out with a set up that consists of the 2U CubeSat and the PCB. The second part will have a set up in which the heat pipe is integrated.

For both parts of the test the procedure is exactly the same: The test is started by setting the thermal chamber to a temperature of 40 °C. The set up is left until the outer panels reach a steady-state temperature of 40 °C. Once this situation has been reached a heat load is applied and time is given to the test object to reach steady-state again. The heat load will be incremented in steps dependent on the thermal response of the PCB (mainly to prevent overheating and destroying the test object). As it is expected that the satellite’s outer panels will see a rise in temperature due to the internal heat load the chamber is actively controlled to maintain the panel boundary temperature at 40 °C.

10.1.3. Theoretical Prediction on Experiment Outcome

In previous chapters thermal analyses have been carried out which have shown the temperature profiles of a CubeSat with and without a heat pipe. Although these have been useful they cannot be used for validating the results of this test as the test conditions are different from the modeled LEO environment. Therefore, the used thermal model (see Appendix A.1) needs to be adjusted to reflect the test environment and a new thermal analysis needs to be carried out.

In the worst case described in previous section the satellite has attained an outer temperature of 40 °C. For the test a steady-state case is investigated at which the outer panels are given this 40 °C as a fixed boundary temperature. This means that external heat fluxes can be neglected which simplifies the thermal modeling. Further simplifications are achieved by reducing of the number of subsystems as the only subsystem present in the set up is the payload PCB. This PCB is discretized into 25 smaller nodes to be able to observe temperature gradients over the PCB itself and obtain more accurate results. The internal convection and radiation is mitigated by the Styrofoam which isolates the different internal nodes from each other. Therefore, only

---

1 Ideally, one would like to discretize the PCB in a much larger amount of nodes. However, ThermXL is not suited for assessing a large number of nodes.
Conductive heat transfer is taken into account in the model. The heat pipe is modeled as a direct conductive path between the heat source (chip) and the heat sink (frame rib) with a conduction value which is based on the found heat transfer coefficient from the heat pipe performance characterization tests and the resistance of 0.2 kW⁻¹ (which is the reciprocal of the conductance) given by the manufacturer [45]. The conduction value is calculated as follows:

\[
C_{t,\text{total}} = \left( \frac{1}{R_{t,\text{hp}}} + \frac{1}{h_{t,\text{source}}} + \frac{1}{h_{t,\text{sink}}} \right)^{-1}
\]  

(10.1)

The resulting conduction is found to be 4.99 W K⁻¹, which is close to the value of 4.01 W K⁻¹ resulting from a first estimate of the effective conductivity used by industry [52].

Figure 10.7 shows the complete thermal network for the test object (enlarged in Appendix A.3) with the aforementioned simplifications taken into account.

![Figure 10.7: Complete thermal network for the 2U test object.](image)

The structural breakdown of the structure into nodes is visualized in Figure 10.8 where the bottom half of the 2U CubeSat is shown with the payload PCB inside and their accompanying labels. Each structural element is broken down into one node only to keep the model as simple as possible (see Appendix A.1 for the consequences of this simplification).

With this nodal breakdown and the full thermal network the conduction values between the different nodes can be calculated. These values are dependent on the direction of the heat flow path and the material properties of each node. The material properties and resulting conduction values between nodes i and j can be found in Table 10.1.

With these conduction values steady-state simulations have been ran which led to the results listed in Table 10.2. The values listed are the highest temperatures found and are belonging to the structural elements closest to the location of the payload PCB.

As expected, without a heat pipe the heat generated on the PCB has to go via the steel rods towards the stack ribs before it spreads over the frame and outer panels. For the scenario where a heat pipe is present the heat is directly transported from the chip to the frame ribs, which therefore see a higher temperature compared to other case.

Table 10.3 lists the temperatures found for the TC nodes. A symmetric temperature profile was found on the PCB due to the symmetric nodal breakdown, but the presence of electrical components on the actual PCB are likely to lead to a raise in temperature locally. The numerical simulations predict a significant reduction in temperature on the PCB. As a consequence of the lower chip temperature the thermal gradient along the PCB...
10.1. Testing the functionality of a heat pipe integrated into the CubeSat structure

Figure 10.8: Nodal breakdown visualized for bottom half of the 2U test model (a) and the discretization of the PCB and location of the different TCs (b).

Table 10.1: Material properties for and conduction values between different nodes.

<table>
<thead>
<tr>
<th>Node i</th>
<th>Material</th>
<th>$k$ [W m$^{-1}$ K$^{-1}$]</th>
<th>Node j</th>
<th>Conductance [W K$^{-1}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>PCB</td>
<td>FR4</td>
<td>35.9</td>
<td>PCB</td>
<td>$6.25 \times 10^{-2}$</td>
</tr>
<tr>
<td>PCB</td>
<td>FR4</td>
<td>35.9</td>
<td>Rod</td>
<td>$3.11 \times 10^{-2}$</td>
</tr>
<tr>
<td>Rod</td>
<td>Steel</td>
<td>15.0</td>
<td>Stack rib</td>
<td>$5.04 \times 10^{-2}$</td>
</tr>
<tr>
<td>Stack rib</td>
<td>Aluminum (bare)</td>
<td>169.0</td>
<td>Frame</td>
<td>$5.04 \times 10^{-2}$</td>
</tr>
<tr>
<td>Stack rib</td>
<td>Aluminum (bare)</td>
<td>169.0</td>
<td>Panels</td>
<td>$4.91 \times 10^{-2}$</td>
</tr>
<tr>
<td>Frame rib</td>
<td>Aluminum (black anodized)</td>
<td>169.0</td>
<td>Panels</td>
<td>$4.91 \times 10^{-2}$</td>
</tr>
<tr>
<td>Frame</td>
<td>Aluminum (black anodized)</td>
<td>169.0</td>
<td>Panels</td>
<td>$2.85 \times 10^{-1}$</td>
</tr>
</tbody>
</table>

Table 10.2: Temperatures in [$^\circ$C] for the two different test parts predicted by the thermal model.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Chip</td>
<td>126.22</td>
<td>64.65</td>
</tr>
<tr>
<td>Rods</td>
<td>80.43</td>
<td>52.57</td>
</tr>
<tr>
<td>Frame rib</td>
<td>41.25</td>
<td>65.10</td>
</tr>
<tr>
<td>Stack rib</td>
<td>57.00</td>
<td>45.88</td>
</tr>
<tr>
<td>Frame structure</td>
<td>41.86</td>
<td>42.05</td>
</tr>
</tbody>
</table>

will be smaller as well. This reduction looks quite optimistic and it is expected that this will not be as high in practice as the numerical model neglects the presence of the heat source and sink interfaces and considering that the heat transfer coefficient is, among others, strongly dependent on the strength with which the heat pipe is clamped to the heat source. The steady-state simulation however, does not include thermal capacities and the actual heat transfer coefficient resulting from integration is hard to determine.

Table 10.3: Temperatures in [$^\circ$C] predicted for the four TCs located on the PCB.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
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</thead>
<tbody>
<tr>
<td>UC4</td>
<td>126.22</td>
<td>64.65</td>
</tr>
<tr>
<td>UC5</td>
<td>111.81</td>
<td>60.69</td>
</tr>
<tr>
<td>UC6</td>
<td>115.13</td>
<td>61.61</td>
</tr>
<tr>
<td>UC7</td>
<td>108.48</td>
<td>59.77</td>
</tr>
</tbody>
</table>
10.2. Test Results

Figure 10.9 shows the temperature measured at the center of the PCB (UC4) with and without a heat pipe. The power inputs given in time are indicated in the plot as well showing that for this first test the maximum power supplied was 3 W. The effect of the heat pipe is evident as a $\Delta T$ of 45.6 °C was achieved between both cases.

![Figure 10.9: Temperature measured at the center of the PCB (UC4) with and without a heat pipe for a heat load of 3 W.](image)

The positive effect of the integrated heat pipe is even more highlighted by Figure 10.10, which visualizes the temperatures measured along the PCB by the different TCs (see Figure 10.3a for the locations of these TCs). Not only can a huge temperature drop be seen but also the temperature gradient along the PCB is drastically reduced, which is beneficial as this will reduce the thermal stress on the PCB.

![Figure 10.10: Measured temperatures along the PCB without (a) and with a heat pipe (b).](image)

While the heat pipe has shown already its potential the real question is whether it can meet the require-
ment of removing a 10 W heat load. Figure 10.11 displays the measured temperature at the center of the PCB (UC4) for an instant heat load of 10 W. This is done in both assisted and anti-gravity orientation to check the functionality of the heat pipe.

Figure 10.11: Thermal response of the center of the PCB for a heat load of 10 W in both anti and gravity-assist orientation.

The results indicate that the heat pipe is not affected by its orientation as no substantial difference in temperature can be spotted between the two measurements. This was already expected as not only a sintered heat pipe was chosen for exactly this reason (see Section 9.2.3 on gravity tests), but also the liquid return-height against gravity is relatively short. What is more interesting though is the steady-state temperature at the center of the PCB under the 10 W heat load, which reaches 95.8 °C. Although this seems high one has to keep in mind that the sink temperature is 40 °C and represents the worst case as defined earlier.

Figure 10.12: Measured temperatures for a 3 W (a) and (an eventually) 10 W heat input (b).

Even though the temperature is within limits of the different components present on the PCB, it is quite
high and therefore necessary to see what the internal temperature gradients are and if the heat pipe behaves as observed during the performance characterization tests. For this reason, tests were run with TCs placed on the heat pipe at the evaporator section ($HP_e$), on the adiabatic section of the heat pipe ($HP$), and on the condenser section. The resulting temperature profiles are shown in Figure 10.12.

The temperatures measured indicate that the heat pipe operates as expected: the heat pipe temperature is the same along its length and only temperature gradients exist between the heat pipe and its interfaces. However, the temperature gradients are significantly higher than the ones measured during the performance characterization tests (in the order of 2-3 °C) while for a heat load of 10 W a gradient is found here of 24 °C. This again stresses the importance of the heat transfer coefficient and is directly a result of the relatively poor integration of the heat pipe. But, this also shows that there is significant room for improvements in both design and integration for which the resulting temperature at the center of the PCB will eventually be lower.

10.3. VALIDATION OF THEORY AND DISCUSSION

Table 10.4 compares the predicted and measured temperatures for the different TCs on the PCB. As the PCB already reached a temperature close to several components’ limits no temperature measurements could be done for the 10 W heat load without a heat pipe. Therefore, theoretical analyses have been redone for the 3 W heat load and listed as well.

Table 10.4: Comparison of theoretical and experimental temperatures in K of the PCB.

<table>
<thead>
<tr>
<th>TC</th>
<th>Theory 3 W</th>
<th>Exp. 3 W</th>
<th>Theory 3 W +HP</th>
<th>Exp. 3 W +HP</th>
<th>Theory 10 W +HP</th>
<th>Exp. 10 W +HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>UC4</td>
<td>65.87</td>
<td>103.10</td>
<td>47.74</td>
<td>57.46</td>
<td>64.68</td>
<td>95.75</td>
</tr>
<tr>
<td>UC5</td>
<td>61.54</td>
<td>83.36</td>
<td>46.49</td>
<td>51.73</td>
<td>60.69</td>
<td>77.75</td>
</tr>
<tr>
<td>UC6</td>
<td>62.54</td>
<td>72.33</td>
<td>46.78</td>
<td>49.27</td>
<td>61.61</td>
<td>69.18</td>
</tr>
<tr>
<td>UC7</td>
<td>60.54</td>
<td>60.33</td>
<td>46.20</td>
<td>46.36</td>
<td>59.77</td>
<td>60.33</td>
</tr>
</tbody>
</table>

The temperatures measured for the rod and the stack rib are listed in Table 10.5 and compared to the theoretical prediction of the 3 W heat load case.

Table 10.5: Comparison of theoretical and experimental temperatures [K] for the rod and stack rib connected to the payload PCB.

<table>
<thead>
<tr>
<th></th>
<th>Theory 3 W</th>
<th>Exp. 3 W</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rod</td>
<td>52.13</td>
<td>46.07</td>
</tr>
<tr>
<td>Stack rib</td>
<td>45.10</td>
<td>43.02</td>
</tr>
</tbody>
</table>

Obviously the measured and predicted temperatures do not agree with each other. The 10 W heat load case without a heat pipe could not even be measured due to the significant rise in temperature at a heat load of only 3 W. This already stresses the importance of thermal control and why a heat pipe is necessary at higher heat loads than currently experienced in CubeSats.

The question is now why there is such a large discrepancy between theory and practice. According to theory the chip temperature should have been way lower than was experienced experimentally. Apparently the PCB is not able to conduct the heat properly to the remainder of the structure. This is highlighted by the relatively low temperatures measured at the stack rib and rod, which are both close to the sink temperature of 40 °C (shown in Table 10.5). This means that the cause of the difference between theory and practice must lie with how the PCB is modeled. Looking at the nodal breakdown and properties of the PCB, the only parameter that can be responsible for this is the thermal conductivity $k$, as the length and width, material, and density are all accurately determined.

The conductivity of the PCB was based on the value used for the QB50p1 satellite ($k = 35.9$). A quick investigation on the conductivity of PCBs revealed that this is not a fixed value, but dependent on the build up of the board itself. Figure 10.13 shows this build up for the used PCB as seen from one of the edges of the PCB.

The PCB consists of two copper layers separated by a thick FR4 layer. The in-plane conductivity is calcu-
lated by Equation (10.2):

\[ k = \frac{2t_{Cu}k_{Cu} + t_{FR4}k_{FR4}}{t_{total}} \]  

Substituting the thicknesses given in Figure 10.13 and a FR4-conductivity of 0.9 Wm\(^{-1}\)K\(^{-1}\), leads to a total conductivity of 9.96 Wm\(^{-1}\)K. This conductivity is a factor 3 lower than the assumed value at first. The theoretical analyses are redone with this new found value and the resulting temperatures for the PCB TCs for a heat load of 3 W are shown in Table 10.6.

### Table 10.6: Comparison of theoretical and experimental temperatures in K of the PCB with the new conductivity value.

<table>
<thead>
<tr>
<th>TC</th>
<th>Theory 3 W</th>
<th>Exp. 3 W</th>
<th>Theory 3 W +HP</th>
<th>Exp. 3 W +HP</th>
<th>Theory 10 W +HP</th>
<th>Exp. 10 W +HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>UC4</td>
<td>101.39</td>
<td>103.1</td>
<td>50.10</td>
<td>57.46</td>
<td>70.78</td>
<td>95.75</td>
</tr>
<tr>
<td>UC5</td>
<td>85.80</td>
<td>83.36</td>
<td>47.61</td>
<td>51.73</td>
<td>63.23</td>
<td>77.75</td>
</tr>
<tr>
<td>UC6</td>
<td>89.40</td>
<td>72.33</td>
<td>48.19</td>
<td>49.27</td>
<td>64.97</td>
<td>69.18</td>
</tr>
<tr>
<td>UC7</td>
<td>82.20</td>
<td>60.33</td>
<td>47.04</td>
<td>46.36</td>
<td>61.49</td>
<td>60.33</td>
</tr>
</tbody>
</table>

With the new thermal conductivity of the PCB the values lie much closer to the experimental found values although the thermal gradient over the PCB remains different compared to the experimental results. The cause of this might lie with how conduction of the the heat pipe is modeled in ThermXL, which is difficult because of its dependency on heat pipe temperature.

A second cause can be found in the calculation of the PCB-conductivity: Routes, plated drills, and soldering of components can all influence the conductivity significantly. However, taking this into account requires a significant amount of modeling and is something which falls outside the scope of this project.

Finally, a more detailed nodal breakdown will lead to a more accurate temperature profile of the PCB, but requires an extensive amount of modeling in ThermXL.

These three causes can explain the remaining difference in temperature between the theoretical and experimental values. However, this requires more investigation and is left as recommendation for future work.

### 10.4. HEAT PIPE START UP FROM THE FROZEN STATE

With the proof of concept for the functionality of the heat pipe in the CubeSat almost completed, the final step is to perform a test in frozen conditions. This means that the entire 2U CubeSat is brought to a temperature below the freezing point of water and after steady-state conditions are reached the heat load is switched on and 10 W power is supplied. Two test runs were completed: one at -10 °C and one at -20 °C. In both cases the thermal chamber was kept at these temperatures, representing the worst case as the thawing of the heat pipe.
is entirely dependent on the dissipated heat of the chip. Figure 10.14 shows the resulting temperature plots for the center PCB temperature. The starting temperature of the middle of the PCB was around 0 °C for the -10 °C case while it reached a temperature of -8 °C at -20 °C. (For measuring the temperature power needed to be supplied to the payload which would directly elevate the temperature of the chip.)

![Temperature Plot](image)

**Figure 10.14**: UC4 temperatures measured during transient start up at -10 and -20 °C.

The results show that in both cases no temperature limits are breached and are significantly lower than for the hot case. This can be explained by the fact that the heat sink temperature is way lower as well (-20/-10 compared to +40 °C). Still, the results shed important light on the thaw-ability of the heat pipe as it indicates that the heat pipe is able to thaw before the temperatures of the chip go through the roof. Apparently, the limited thermal capacities present in both the interfaces as well as the CubeSat structure itself prove to be a benefit here and lead to a quick spread of the heat from one point to another.

![Temperature Profiles](image)

**Figure 10.15**: Temperature profile for the different interfaces at -10 °C start up (a) and -20 °C (b).
Figure 10.15 displays the resulting temperature profiles of the different interfaces for both cases. The difference between the found results here and the ones predicted in Section 9.3.3 are significant, but can be logically explained when considering that the dissipated heat in the earlier carried out theoretical test has to go entirely through the heat pipe via the evaporator section. Here, however, the heat dissipated is not only able to go into the heat pipe, but also into the PCB itself. Coupled with the additional thermal capacities present it is logical that the peak temperature is less than predicted.

Figure 10.16 shows the increase in temperature of the different panels due to the 10 W heat load being dissipated. Comparing this to the found temperature for the condenser section it can be seen that a hotspot is created there and the heat is not efficiently enough distributed to the external panels. This once more stresses the importance of the heat transfer coefficient.

The transient start up test has shown promising results in the worst cold case as the peak temperatures attained remain far below the limits of the chip. Still, more research and detailed modeling is necessary to verify and validate these results, which has unfortunately fallen outside the scope of this project.

10.5. CONCLUSION ON THE HEAT PIPE INTEGRATING TESTS

The tests showed the feasibility of the heat pipe for CubeSat application and highlighted the thermal problem that arises at component level for a continuous power output. Despite the relatively weak integration the heat pipe showed its potential by achieving a significant drop in $\Delta T$. Both in gravity assisted as well as anti-gravity orientation the sintered heat pipe achieved the same results.

The tests have also shed light on the importance of the conductivity of the PCB, which severely impacts the temperatures attained at PCB level, although more investigation is required to pinpoint the exact cause of this.

Just as experienced for the performance characterization tests, the heat pipe was able to successfully start up at temperatures below the freezing point of water, while keeping the temperature of the chip well within limits.

While the temperature of the chip was significantly reduced the removed heat actually led to a rise in temperature of the heat sink. To prevent the creation of a new thermal hot spot a more efficient heat sink interface is required with an improved conduction path towards the CubeSat structural frame.
The completion of the project is marked by answering the posed questions at the start. The questions and accompanying answers will together form a summary of the findings during this project. Each answer will be supported by references to the appropriate parts of the report. First, the sub-questions will be answered after which the chapter is concluded by answering the research question.

11.1. Answering the Sub-Questions
The sub-questions, which served as guidelines throughout the project are answered here with the help the earlier findings. Each question is answered directly or by answering its sub-questions.

1 What heat pipe is needed for CubeSat thermal control?

1.1 What are the limits for the heat pipe performance (max heat flux, heat transport) as a function of temperature?
The performance of heat pipes is confined by the different existing limits which have been identified from research and algebraically derived (Section 4.3). Each of these limits impacts the maximum heat transport capacity of the heat pipe and is function of heat pipe temperature. The maximum heat flux is covered by the boiling limit, which is dependent on the heat pipe material and surface finishing (Section 4.3.5).

1.2 How do geometry, material and fluid affect these limits?
The driving force behind the operating principle of heat pipes is the capillary pressure (Section 4.2.1), which is dependent on both the surface tension of the fluid and the capillary radius. As each fluid has a unique surface tension, which is also temperature dependent, a figure of merit is given to each fluid that describes its performance (Section 5.1.1). Next to the capillary pressure, each of the limit is dependent on the liquid and/or vapor viscosity (Section 4.3), which thus shows the influence and importance of the fluid.

Material or heat pipe casing has two influences: the wick conductivity, which plays a role in defining the boiling limit (Section 4.3.5) and the contact angle, which is a play between the fluid and the material (Appendix E) and directly affects the capillary limit.

Finally, a large influence is due to the geometry of the heat pipe both internal as external. The internal geometry concerns with the layout of the wick structure which directly influences the capillary pressure and all the limits (Sections 4.3 and 5.1.3). The number, size and shape of the grooves define the liquid hydraulic radius and the capillary radius, both which directly affect the capillary limit. The vapor space determines the vapor pressure drop, which is important for the viscous, sonic and the entrainment limit. External geometry is covered by heat pipe length, orientation, and bends. Length of the heat pipe has only a small influence on the different limits, while orientation significantly influences the axial grooved heat pipe performance in the presence of gravity (Section 9.2.3). In a micro-gravity environment this gravity pressure drop will be absent in any orientation and a decrease in performance due to this will thus not occur. Bends do not or hardly have any influence on performance (Sections 5.1.4 and 9.2.3).
1.3 What heat pipe design is required to solve for the expected heat loads?
Theoretical analysis concerning heat pipe limits have shown that a minimum outer diameter of 6 mm is required to cope with the project 10 W OAP (Section 7.1). Furthermore, a minimum groove-width of 0.2 mm was found to be the minimum in order to reach the 10 W heat transport capacity at all heat pipe temperatures (Section 7.2).
The properties of the mesh and sintered wicks are difficult to determine, therefore no conclusions can be made regarding the required properties of these for meeting the payload heat dissipation.

1.4 Is there a commercially available heat pipe that fits the desired heat pipe design?
Yes, it was found that low-cost, commercial heat pipes are available in the dimensions suitable for CubeSats (Section 6.2). Although not necessary, it is possible to custom manufacture heat pipes to suit the needs of any application.

2 What is the experimental performance of commercially available heat pipes?

2.1 How does the performance scale with heat pipe temperature?
The performance increases for increasing heat pipe temperatures (Sections 7.1 and 9.2.3). This increase in performance is due to the temperature dependence of the fluid properties.

2.2 What is the heat transfer coefficient at the heat pipe hot- and cold interfaces?
The heat transfer coefficient was found to be dependent on the heat pipe temperature and type of heat pipe (Section 9.2.3). The mesh heat pipe showed the worst heat transfer coefficient at both sides, while the axial grooved and sintered heat pipes exhibited similar values.

3 What is the transient start up behavior of water heat pipes around freezing point?

3.1 At what temperature does the heat pipe starts functioning?
During start-up from the frozen state the heat pipe starts functioning as soon as the condenser section reaches 0 °C (Section 9.3.3). Once this temperature is reached the liquid is thawed and the liquid-vapor cycle is reinstated.

3.2 What is the start-up time of a heat pipe when frozen?
The start up time in test conditions without and with a heat load of 10 W for the axial grooved heat pipe were measured to be 25.07 and 15.24 min, respectively (Section 9.3.3). With numerical model validation (see Section 9.3.3) the start up time was modeled for a heat pipe in LEO environment and was found to be equal to 71 s. In both cases the heat pipe was brought to a temperature of -10 °C and the temperature at the evaporator section did not exceed the limits of a common electronic chip or component.
This was further shown with the carried out experimental tests (Section 10.4). With these tests it was shown that the temperature of the chip remained well below the limits of the different electrical components and the heat pipe was able to thaw successfully.

4 What is the effect of freeze/thaw cycling on a heat pipe?

4.1 Does repetitive freeze/thaw cycling lead to external physical damage?
After 80 cycles for the sintered heat pipe and 100 cycles for the axial grooved variant no external physical damage was observed (Section 9.4.3).

4.2 How does freezing of the liquid impact the wick structure?
Visual inspection of the grooved and sintered wick structure after freeze/thaw cycling showed no indication of wick damage (Section 9.5.2).

4.3 What is the effect of freeze/thaw cycling on heat pipe performance?
Between each 10 cycles the heat pipes were tested on performance (Section 9.4.3). From this and wick inspection (Section 9.5.3), no signs were spotted that indicated performance degradation. Tilt tests were carried out to double check whether its performance was decreased. From these only the sintered variant showed a difference in performance compared to before. For axial grooved heat pipes the conclusion was drawn that these freeze/thaw cycles have no impact on their performance, while for sintered heat pipes the effect remains inconclusive due to the contradictory test results.
5 Can heat pipe freezing be beneficial for CubeSat thermal control?
From the transient start-up test it was found that as soon as 0 °C was reached at the condenser end the heat pipe stopped functioning completely (Section 9.3.3). In all heat load cases the heat pipe was able to keep the heat source within temperature limits. In the case where the heat source is off and the condenser section freezes the heat pipe will stop function below 0 °C and cool-down of the payload will only occur via conductive heat transfer, which takes much more time compared to the heat transfer by normal heat pipe operation. This is definitely beneficial for CubeSat thermal control as it means that the payload will not quickly reach too low temperatures.

6 Are heat pipes suitable for use in CubeSats?
6.1 Can a heat pipe deal with the projected heat loads?
From test runs with a heat pipe integrated in a CubeSat it was found that the heat pipe is able to transport over 10 W continuous heat load (Section 10.2). This is achieved in a 'worst-case' scenario concerning the location of the PCB, heat pipe length, and integration.

6.2 How can a heat pipe be integrated into the CubeSat platform?
A simple design was proposed that covers the necessary elements for heat pipe integration in a CubeSat (Section 8.2). While the CubeSat platform has limited volume available the possibility of bending the heat pipe gives a sufficient amount of freedom of routing the heat pipe through the structure. As heat sinks the frame ribs were chosen as the frame is the prime interface between all the structural elements and thus ensures an optimal heat distribution from thereon. A single heat pipe was chosen with the heat source located at the middle.

6.3 What is the impact of interfaces on the thermal performance of heat pipes?
Theory showed that the impact of interfaces on overall performance is significant as it is the largest resistance in the entire chain (Section 8.1.1). This was further highlighted by the integration testing when a temperature difference of 26 °C was observed between the chip and heat pipe temperature (Section 10.2).

7 What are the critical aspects for a CubeSat mission when employing heat pipes?
From the different tests carried out it was found that the most critical aspect is the interface between the heat pipe and the heat generating source (Section 10.2). Heat transfer coefficients play a crucial role in keeping the payload temperature at a low value.

11.2. ANSWERING THE RESEARCH QUESTION
With all the sub-questions answered the final step is to look at the research questions posed at the start of this project, which is repeated here for clarity:

Can commercial water heat pipes solve the thermal challenge of high performance CubeSat missions?

By looking at the answers given to the sub-questions, following from the different performance characterization tests and the heat pipe integration experiments it becomes clear that commercial water heat pipes can indeed solve for the thermal challenge that is upcoming in high performance CubeSat missions. The water heat pipe is able to passively transport the heat loads expected (up to 10 W) in the next-generation CubeSats, thereby keeping the heat source within its temperature limits. Bending was found to have negligible influence on the heat pipe's performance, which gives a large degree of freedom during integration and therefore poses no additional constraints on other internal (sub)systems. The performance of heat pipes is characterized by an increase in heat transfer coefficient for higher heat pipe temperatures, while at temperatures below the freezing point of water only pure conduction remains. The latter is found to be beneficial as it prevents the heat source from cooling down too quickly and even with an instant 10 W heat load the heat pipe is able to thaw way before the temperature limit of the heat source is reached.

The critical aspect in integrating a heat pipe in the CubeSat platform is attaining an efficient heat transfer between the heat pipe and the source and sink. For this, a proper design is necessary to reduce the thermal gradients between these interfaces. Furthermore, the heat pipe is perfect for heat transport but will create a hotspot at the chosen heat sink element of the CubeSat if no other thermal control mechanisms are involved which are able to remove this heat from the satellite.
12 RECOMMENDATIONS

Recommendations are given to aid in further discovering the potential of heat pipes for CubeSat application. These recommendations will allow continuation of the path set out in this project or look into aspects that have fallen outside the scope of this project.

12.1. RECOMMENDATIONS ON HEAT PIPE TESTING

Regarding the different tests tailored to heat pipe performance testing, the following recommendations are given:

Heat pipe performance characterization
For several identical heat pipes their performance has been measured, with sometimes different results (Section 9.2.3). The limited amount of tested heat pipes does not allow for an overall verdict on the quality of commercial heat pipes. For this reason more performance tests need to be carried out with identical heat pipes to map the consistency in performance of commercial heat pipes.

Heat pipe performance validation
Reasons were presented that can account for the remaining difference between theory and practice (Section 9.2.4). However, due to the difficulty in determining these parameters and phenomena it fell outside of the scope of this project. Further research will be necessary to identify the exact cause of the difference and may even require one to build a custom heat pipe that can be internally inspected during operation.

Repetitive freeze/thaw cycling I
During the freeze/thaw cycling test both the axial grooved and sintered heat pipes experienced more than 80 cycles (Section 9.4). For a heat pipe operating in the LEO environment this is usually far from the amount of cycles experienced (although dependent on the orbit). The amount of cycles during this test was constraint by the test duration, but ideally one would like to test a lot more cycles to be able to conclude on the effect of these on the performance of the heat pipes. Therefore, it is advised that the commercial heat pipes are subject to more cycles than currently have been carried out.
Next to this, the cycle test could be altered to investigate what the effect is on heat pipe material strength by changing the cycle-time and/or the setpoint temperatures.

Repetitive freeze/thaw cycling II
The effect of freeze/thaw cycles on the sintered heat pipe was found difficult to observe and performance degradation due to this cycling was marked as inconclusive due to the contradictory results (Section 9.4.3). More research and accurate gravity tilt testing is required to be able to reach a valuable conclusion.

12.2. RECOMMENDATIONS ON HEAT PIPE INTEGRATION

Derived from the heat pipe integration test, the following recommendations are given:
**Structural integrity of heat pipes**
The structural integrity of heat pipes has not been touched upon during this project, but is specifically important for integration in the CubeSat. During launch phase, orbit insertion, and possibly other satellite modes vibrations are present in the satellite and affect the heat pipe structure or integration. Whether this can even be damaging is worth investigating. For this reason carrying out vibration tests simulating these environmental conditions are advised.

**Optimized integration design**
As the goal of the integration test was to show the proof of concept only a first design to integrate the heat pipe was presented (Section 8.2). While this proved to be sufficient for testing it was also shown that large temperature gradients were raised between the heat pipe and the heat source and sinks (Section 10.2). The importance of the heat transfer between these interfaces has been emphasized numerous times throughout the report and the used integration has shown that there is sufficient ground to win.

At the heat source section the heat transfer coefficient should be increased, which requires a better interface design and stronger clamping. This also holds for the heat sink area, where stronger clamping and an improved interface will lead to not only a smaller temperature gradient, but also has to lead to a better coupling between the heat pipe and the outer structure. The tests showed that most of the heat went into the clamping rib instead of directly into the frame structure of the CubeSat.

**Validation of thermal analysis**
The tests with a heat pipe integrated in the CubeSat structure had the goal to illustrate the proof of concept of it. For this reason no detailed analyses were carried out. Still, calculations were presented that partly showed why a discrepancy between theory and practice was present (Section 10.3). However, more investigations are necessary, specifically concerning detailed thermal modeling of PCBs. Furthermore, finite-element methods should be employed that allow for detailed thermal design of the CubeSat structure, and tests could be ran to experimentally verify all the conduction paths and values.

**Transient start up behavior of a CubeSat integrated heat pipe**
Transient start up tests were carried out showing promising results (Section 10.4), but lack detailed thermal analysis and model calibration. This is required to ensure verification and validation of the results found.

**From transport to radiation**
The heat pipe has shown its capability of removing sufficient heat from a payload chip to keep the temperature within limits (Section 10.2). However, although the heat pipe is able to remove the dissipated heat successfully from the chip, it only transports the heat from one location to another. The follow-up step would be to investigate how to remove this heat from the CubeSat itself. The CubeSat surface area is likely to be insufficient to radiate all this heat towards deep space. Therefore, opportunities lie in this area to explore the possibility of combining a heat pipe with, for example, a dedicated radiator. A dedicated radiator optimized for the expected heat dissipation in combination with a (or multiple) heat pipe(s) could lead to an efficient passive thermal control system.

Another option would be to look into the possibility of making a heat pipe part of the CubeSat structure. For example, a heat pipe integrated in the frame could rapidly distribute the heat around the entire structure and decrease the creation of local, external hotspots.

**Fly!**
The real proof of the usefulness of a heat pipe in a CubeSat is to fly it on an actual mission. The heat pipe has proven its use in many Earth applications and also during the tests carried out in this project. A demonstration flight could spark the interest of the CubeSat industry and be useful for further validation of theory. A demonstration flight can be any CubeSat mission which contains a payload or chip that has a significant heat dissipation. With an improved design for integration temperature gradients can be kept low and reduce the risk of breaching any temperature limits. Furthermore, to prevent mission failure, small heaters could be employed that keep the heat pipe at all times above 0 °C. The final step would be to combine the heat pipe with a system that is able to radiate the additional heat from the CubeSat structure.
BIBLIOGRAPHY


The thermal analyses are carried out with the help of a thermal model built in ThermXL, part of the ESATAN software suite. ThermXL is a Microsoft Office Excel plugin in which nodes and thermal couplings can be defined and steady-state and transient solutions can be run.

**A.1. Thermal Nodal Network of the QB50p1 CubeSat**
A.2. ASSUMPTIONS ON THE THERMAL MODEL

Several assumptions have been taken into account which have simplified the thermal model of the QB50p1 CubeSat. While simplification makes life easier it might also negatively affect the results. Therefore, one needs to carefully assess the effect of these when applied. The following assumptions have been taken into account along with their effect on the outcome.

- **No internal radiation**
  Internal radiation between components has been neglected. This assumption is valid for low thermal gradients between nodes as in that case conduction will be the dominant heat transfer mechanism. Modeling internal radiation is made difficult due to view factors that need to be taken into account. For higher thermal gradients radiation may become significant and does need to be taken into account or its effect needs to be qualitatively explained.

- **Single-node representation**
  Each structural element and subsystem is represented by a single node and thus regarded isothermal. While this prohibits examination of thermal gradients along a single structural element it makes the thermal model much more clear. It does have an effect on the resulting temperatures as the thermal gradient between nodes will be different.

- **Simplification of conduction paths**
  Conduction paths are viewed as series conduction only, which simplifies the thermal model. Assessing the path of heat transfer is one of the key challenges in thermal modeling and often most difficult, therefore simplification is often desired. Care has to be taken to ensure the proper conductance between nodes.

- **No contact conductance**
  Contact conductance is neglected as this is always a difficult parameter. Contact conductance is dependent on a number of factors and can only be measured experimentally. Its influence can be significant if elements are not properly connected to each other. This is one of the parameters that can be the bridge between theoretical and experimental differences.

While some of these assumptions may seem to have a significant impact on the resulting temperatures the conduction values used in the model account for most of them as these have been tweaked and validated with in-orbit data.
A.3. THERMAL NODAL NETWORK OF THE TEST OBJECT

A.4. CONDUCTION VALUES OF THE THERMAL TEST MODEL

To get a bit more grip on the thermal model used the conduction values calculated between the different nodes are listed here. The values are based on the thermal model of the QB50p1 CubeSat, which have been validated with in-orbit data.

<table>
<thead>
<tr>
<th>Node 1</th>
<th>Node 2</th>
<th>Conductance [WK⁻¹]</th>
</tr>
</thead>
<tbody>
<tr>
<td>PCB</td>
<td>PCB</td>
<td>$6.25 \times 10⁻²$</td>
</tr>
<tr>
<td>PCB</td>
<td>Rod</td>
<td>$3.11 \times 10⁻²$</td>
</tr>
<tr>
<td>Rod</td>
<td>Stack rib</td>
<td>$5.04 \times 10⁻²$</td>
</tr>
<tr>
<td>Stack rib</td>
<td>Frame</td>
<td>$5.04 \times 10⁻²$</td>
</tr>
<tr>
<td>(Stack) Rib</td>
<td>Panels</td>
<td>$4.91 \times 10⁻²$</td>
</tr>
<tr>
<td>Frame</td>
<td>Panels</td>
<td>$2.85 \times 10⁻¹$</td>
</tr>
</tbody>
</table>
ORBITAL TEMPERATURE PROFILES IN LEO

The temperature profile of a CubeSat in LEO is dependent on its attitude and orbital plane. The temperature profiles from a noon-midnight (LTAN 12:00 hour) to a dawn-dusk (LTAN 6:00 hour) orbit are plotted.

B.1. CUBE SAT SPECIFICATIONS

The temperature profiles calculated are based on the QB50p1 satellite, whose power and mass budgets and orbital parameters are shown in Tables B.1 and B.2, respectively.

Table B.1: Mass and power budget of the QB50p1 2U CubeSat.

<table>
<thead>
<tr>
<th>Subsystem</th>
<th>Mass [kg]</th>
<th>Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>EPS</td>
<td>0.225</td>
<td>0.22</td>
</tr>
<tr>
<td>ADCS</td>
<td>0.459</td>
<td>0.55</td>
</tr>
<tr>
<td>TTC</td>
<td>0.12</td>
<td>0.15</td>
</tr>
<tr>
<td>CDHS</td>
<td>0.155</td>
<td>0.35</td>
</tr>
<tr>
<td>Payload</td>
<td>0.399</td>
<td>0.824 - 10</td>
</tr>
</tbody>
</table>

Table B.2: Orbital parameters of the QB50p1 2U CubeSat.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Orbit type</td>
<td>SSO</td>
</tr>
<tr>
<td>LTAN [h:min]</td>
<td>12:00 - 09:00 - 06:00</td>
</tr>
<tr>
<td>Orbital altitude [km]</td>
<td>600</td>
</tr>
<tr>
<td>Mission date</td>
<td>18-03-2015</td>
</tr>
</tbody>
</table>
B.2. 12:00 HOUR LTAN ORBIT

External and internal temperature profile for the 2U LEO QB50p CubeSat in a 12:00 LTAN orbit with a nadir-fixed attitude mode.

**Figure B.1:** External panel temperature profiles for an 12:00 h LTAN orbit with nadir-fixed attitude.

**Figure B.2:** Subsystem temperature profiles for 12:00 h LTAN and nadir-fixed attitude.
External and internal temperature profile for the 2U LEO QB50p CubeSat in a 12:00 LTAN orbit with a Y-Thomson attitude mode.

Figure B.3: External panel temperature profiles for an 12:00 h LTAN orbit in Y-Thomson mode.

Figure B.4: Subsystem temperature profiles for 12:00 h LTAN in Y-Thomson mode.
B.3. 9:00 HOUR LTAN ORBIT
External and internal temperature profile for the 2U LEO QB50p CubeSat in a 09:00 LTAN orbit with a nadir-fixed attitude mode.

Figure B.5: External panel temperature profile in nadir-fixed attitude.

Figure B.6: Subsystem temperature profiles in nadir-fixed attitude.
External and internal temperature profile for the 2U LEO QB50p CubeSat in a 09:00 LTAN orbit with a Y-Thomson attitude mode.

Figure B.7: External panel temperature profile in Y-Thomson mode.

Figure B.8: Subsystem temperature profiles in Y-Thomson mode.
B.4. 6 Hour LTAN Orbit

External and internal temperature profile for the 2U LEO QB50p CubeSat in a 06:00 LTAN orbit with a nadir-fixed attitude mode.

**Figure B.9:** External panel temperature profile in nadir-fixed attitude.

**Figure B.10:** Subsystem temperature profiles in nadir-fixed attitude.
External and internal temperature profile for the 2U LEO QB50p CubeSat in a 06:00 LTAN orbit with a Y-Thomson attitude mode.

**Figure B.11:** External panel temperature profile in Y-Thomson mode.

**Figure B.12:** Subsystem temperature profiles in Y-Thomson mode.
FLOW IN HEAT PIPES

The heat pipes’ operation depends on the internal flow of the vapor and liquid. The behavior of these flows depend on their velocity: At low velocities the flow follows a laminar regime, while at higher velocities it becomes turbulent. Laminar and turbulent flow both have a different impact on the performance of heat pipes, therefore it is necessary to understand how to model the flow in these regimes.

For a cylindrical pipe laminar non-accelerating fully developed flow is a function of pressure and viscous forces only. This can be described by Equation (C.1), where $\tau$ denotes the shear stress and $r$ the radius of the cylinder.

$$\frac{dp}{dx} = -\frac{2\tau}{r} \tag{C.1}$$

The shear stress can be related to the velocity gradient by introducing Newton’s law of viscosity:

$$\tau = -\mu \frac{du}{dr} \tag{C.2}$$

By combining Equations (C.1) and (C.2) the following relation is obtained:

$$du = \frac{1}{2\mu} \frac{dp}{dx} rdr \tag{C.3}$$

Integration of this relation subsequently leads to a relation for the local velocity:

$$\int du = \frac{1}{2\mu} \int \frac{dp}{dx} rdr$$

$$u = \frac{1}{4\mu} \frac{dp}{dx} r^2 + C \tag{C.4}$$

The constant $C$ can be determined by acknowledging the fact that $u = 0$ at the heat pipe’s wall due to the viscosity of the fluid, with location $r = r$. Substituting this into Equation (C.4) results in an expression for $C$. The velocity $u$ can now be written as:

$$u = \frac{1}{4\mu} \frac{dp}{dx} (r^2 - r^2) \tag{C.5}$$

The next step is to evaluate the flow rate through the heat pipe’s cross-section. This is found by integrating Equation (C.5) for a small area element with constant velocity $dA = 2\pi rdr$:  

125
\[
\int u \, dA = \frac{1}{4\mu} \frac{dp}{dx} \int (r_0^2 - r^2) 2\pi r \, dr
\] (C.6)

With,
\[
\int_{r_0}^{r_0} (r_0^2 - r^2) 2\pi r \, dr = 2\pi \left[ \frac{r_0^4}{2} - \frac{r^4}{4} \right]_{r_0}^{r_0} = 2\pi \frac{r_0^4}{4}
\]
\[
u A = \frac{\pi r_0^4}{8\mu} \frac{dp}{dx}
\]

Finally, by integrating and solving for the pressure gradient term in Equation (C.6) over the heat pipe length \( l \) and noticing that \( u A = \frac{\dot{m}}{\rho} \), a final relation for the pressure drop is found:
\[
\Delta p = \frac{8\mu ll_{in}}{2\rho\pi r^4}
\] (C.7)

This relation is called the Hagen-Poiseuille Equation and describes the pressure gradient as a function of the fluid viscosity, the heat pipe radius and length, and mass flow for fully developed, laminar flow in a cylindrical pipe. (For a more extensive derivation from the Navier-Stokes Equations, the reader is referred to Reference [31].)

A second way of describing flow through a pipe mathematically is by using the Darcy-Weisbach Equation, which is valid for laminar as well as turbulent flow.
\[
\Delta p = f_D \frac{l}{d_h} \frac{\rho v^2}{2}
\] (C.8)

Here, \( d_h \) is the hydraulic diameter which is the cross-sectional area of the pipe divided by the wetted perimeter. For a cylindrical pipe this comes simply down to its diameter, as shown in Equation (C.9).
\[
d_h = \frac{4A}{P} \quad \text{(C.9)}
\]
\[
d_h = \frac{4\pi r^2}{2\pi r} = 2r
\]
\[
d_h = \frac{4r}{2} = d
\]

The term \( f_D \) is called the Darcy friction factor, which accounts for the friction losses in flow through both cylindrical and open channels. In Equation (C.8) the term \( \rho v \) can be rewritten in terms of the Reynolds number as follows:
\[
Re = \frac{\rho v d_h}{\mu}
\]
\[
\rho v = \frac{Re \mu}{d_h}
\] (C.10)

Furthermore, substituting twice the radius for the hydraulic diameter and writing the remaining velocity term in terms of mass flow, cross-sectional area, and density with \( \nu = \frac{\dot{m}}{\rho A} \), results in the relation for pressure loss in a cylindrical pipe.
\[
\Delta p = \frac{f_D Re \dot{m}}{8\rho Ar^2}
\] (C.11)

As said, this relation is valid for both laminar and turbulent flow. It can be easily seen that for the laminar flow regime the term \( f_D Re \) must equal 64 in order to obtain the earlier derived Hagen-Poiseuille Equation again, valid for laminar flow. The relation between the friction factor and the Reynolds number can be written as follows:
\[
f_D = \frac{64}{Re} \quad \text{for } Re < 2100
\] (C.12)
For turbulent flow multiple relations exist based on experimental findings. One of the most commonly used empirical relations is the Blasius Equation, as shown in Equation (C.13) [53]. The Blasius Equation is a simple relation for the Darcy friction factor which is only a function of the Reynolds number. As opposed to other found relations the Blasius Equation does not take into account wall roughness and is therefore only valid for smooth pipes and gives accurate results for \( Re < 10^5 \).

\[
f_D = \frac{0.316}{Re^{0.25}}, \quad 2100 < Re < 10^5
\]  

(C.13)

Another widely used empirical relation is the Colebrook Formula [54]. This relation, shown in Equation (C.14), does include the surface roughness, expressed by \( \varepsilon \). It is also valid at higher Reynolds numbers. However, the downside of this formula is that the friction factor can only be found by employing an iterative scheme, which makes it more complicated than the Blasius Equation.

\[
\frac{1}{\sqrt{f}} = -2.0 \log \left( \frac{\varepsilon D}{3.7} + \frac{2.51}{Re \sqrt{f}} \right)
\]  

(C.14)
D.1. Heat Pipe Length 10 cm
The following graphs are for an axial-grooved copper-water heat pipe with evaporator and condenser section lengths of 2 cm.

Operating Region

(a) OD 4 mm
(b) OD 6 mm
(c) OD 8 mm
CAPILLARY LIMIT
The following graphs are for a heat pipe operating temperature level of 40 °C.

(a) 0 deg tilt
(b) 5 deg tilt
(c) 10 deg tilt
(d) 15 deg tilt
(e) 20 deg tilt
(f) 25 deg tilt
D.1. HEAT PIPE LENGTH 10 cm

**Viscous Limit**

![Viscous Limit Diagram]

**Sonic Limit**

![Sonic Limit Diagram]
**Entrainment Limit**

![Entrainment Limit Graph]

**Boiling Limit**

![Boiling Limit Graph]
D.2. **Heat Pipe Length 15 cm**

The following graphs are for an axial-grooved copper-water heat pipe with evaporator and condenser section lengths of 3 cm.

**Operating Region**

![Graphs showing heat pipe operating limits for various ODs](image)

(a) OD 4 mm  
(b) OD 6 mm  
(c) OD 8 mm
CAPILLARY LIMIT
The following graphs are for a heat pipe operating temperature level of 40 °C.

(a) 0 deg tilt  
(b) 5 deg tilt  
(c) 10 deg tilt  
(d) 15 deg tilt  
(e) 20 deg tilt
**Viscous Limit**

![Viscous Limit Graph](image)

**Sonic Limit**

![Sonic Limit Graph](image)
**Entrainment Limit**

[Graph showing the entrainment limit with different groove widths and diameters]

**Boiling Limit**

[Graph showing the boiling limit with different groove widths and diameters]
D.3. Heat Pipe Length 25 cm

The following graphs are for an axial-grooved copper-water heat pipe with evaporator and condenser section lengths of 5 cm.

**Operating Region**

(a) OD 4 mm  
(b) OD 6 mm  
(c) OD 8 mm
CAPILLARY LIMIT
The following graphs are for a heat pipe operating temperature level of 40 °C.

(a) 0 deg tilt

(b) 5 deg tilt

(c) 10 deg tilt

(d) 15 deg tilt
VISCOS LIMIT

Sonic Limit
**Entrainment Limit**

![Entrainment Limit Graph](image)

**Boiling Limit**

![Boiling Limit Graph](image)
D.4. METHANOL HEAT PIPE LENGTH 20 cm
The following graph shows the heat pipe limits for a copper-methanol heat pipe.
CONTACT ANGLE MEASUREMENT

The contact angle between a liquid and solid influences the capillary forces generated. Therefore, knowing this angle is important. The contact angle is dependent on the liquid itself and the solid with which it is in contact.

The contact angle has been measured as part of an earlier project carried out at the NLR and, although not part of this thesis work, it gives useful insight into this phenomenon.

The test was carried out by applying a single droplet of water onto a copper surface after which a photo was taken. The copper surface went through a thorough cleaning process to remove any present contamination on the surface. This procedure was carried out many times and the time between applying a droplet and taking a photo was varied to study the effect of this.

The contact angle of a droplet of water on a copper surface was measured as a function of time. Figure E.1 shows the resulting contact angle after half an hour and after a week.

![Figure E.1](Property NLR)

Over a week the contact angle has increased to 93 °. This is due to the exposure to ambient atmosphere and the contamination of the copper surface over time. However, heat pipes are filled up in vacuum conditions and sealed off. Therefore, it can be assumed that the contact angle present after thorough cleaning is the angle that is most representative. For this reason a wetting angle of 27 ° can be taken as contact angle during calculations.
**EXPERIMENT I: PERFORMANCE TEST RESULTS**

**F.1. MAXIMUM HEAT TRANSPORT CAPACITIES**

The following maximum heat transport capacities in W were measured for the different heat pipes. (HP# 4 has been left out as no useful results could be obtained during testing.)

<table>
<thead>
<tr>
<th>Setpoint [°C]</th>
<th>HP1</th>
<th>HP2</th>
<th>HP3</th>
<th>HP5</th>
<th>HP6</th>
<th>HP7</th>
<th>HP8</th>
<th>HP9</th>
<th>HP10</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>1</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>-5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>2</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>0</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>3</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>8</td>
<td>3</td>
<td>13</td>
<td>-</td>
<td>5</td>
<td>-</td>
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<td>13</td>
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<td>16</td>
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<td>15</td>
</tr>
<tr>
<td>30</td>
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<td>21</td>
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<td>20</td>
<td>10</td>
<td>6</td>
<td>25</td>
<td>12</td>
<td>19</td>
</tr>
<tr>
<td>40</td>
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<tr>
<td>70</td>
<td>-</td>
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<td>-</td>
<td>-</td>
<td>9</td>
<td>-</td>
<td>-</td>
<td>-</td>
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<td>80</td>
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<td>-</td>
<td>22</td>
<td>-</td>
<td>71</td>
<td>25</td>
<td>37</td>
</tr>
</tbody>
</table>

**F.2. ERRORBARS FOR THE AXIAL GROOVED HEAT PIPE AND SINTERED HEAT PIPE PERFORMANCE MEASUREMENTS**

The plots below show the error bars for the measurements carried out for both the axial grooved and sintered heat pipes. The minimum and maximum errors are based on the estimated power error of 0.4 W (heat leakage) and an error of 0.2 ΔT. The resulting plots show that the error is larger at low heat inputs, which is logical as the error percentage gets smaller for higher power inputs.
$h_f$ behavior as a function of $P$ for the grooved heat pipe with errorbars

$h_f$ behavior as a function of $P$ for the sintered heat pipe
EXPERIMENT II: HEAT PIPE TEST RESULTS

The following sections show the resulting temperature plots for the different tests carried out.

G.1. TEST RUN 1: 3 W NO HEAT PIPE

![Temperature plots for PCB, structure, and panel temperatures for Test Run 1 with 3 W and 2 W heat inputs.]
**G.2. Test Run 2: 3 W No Heat Pipe**

![Graphs of PCB Temperatures](image1.png)

![Graphs of PCB Temperatures Through Plane](image2.png)

![Graphs of Structure Temperatures](image3.png)

![Graphs of Panel Temperatures](image4.png)
G.3. **Test Run 3: 3 W with Heat Pipe**

![Graph showing temperature changes over time for PCB, Structure, and Panel temperatures.](image)

The graphs illustrate the temperature changes over time for different components in the test run. The PCB Temperatures show a significant rise in temperature, with the 3W label indicating a specific condition. Similarly, the Structure Temperatures and Panel Temperatures show temperature variations with time, highlighting the performance under heat pipe conditions.
G.4. **Test Run 4: 10 W with Heat Pipe**
G.5. TEST RUN 5: 10 W WITH HEAT PIPE