Design and Investigation of a pulsating heat pipe for electronic cooling

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Abstract

The purpose of this study is to design and investigate a pulsating heat pipe device (PHP). A PHP is a two-phase flow heat transfer device which can handle high heat fluxes with passive internal flow. It is made of a continuous loop serpentine, partially filled with a working fluid that can cool systems, such as electronic devices. It differs from a conventional heat pipe device in design and operation, potentially bringing some advantages when compared to the latter. Firstly, no wick structure is required to assist the condensed working fluid to flow to the evaporator, making it cheap to construct and flexible to integrate into different applications. Secondly, more working fluid is used in a PHP, potentially enhancing heat transfer in it.

The performance of a PHP can be measured by the thermal resistance, which is defined as the average temperature difference between the evaporation and condensation, divided by the supplied power to the device. The lower the thermal resistance, the more efficient the heat transfer in the PHP is. Despite the great efforts that many experimental and numerical works have put into fully understanding PHPs behaviour, since it was patented in the 90s, a lot remains unknown. This is mainly because of the chaotic two-phase, nonlinear internal flow of the device.

In this thesis, after the initial stage of designing and manufacturing a glass PHP, investigating how different parameters affect the thermal resistance was conducted. This included testing the PHP performance while varying different filling ratios for different working fluids (isopropanol and distilled water), different inclination angles of the device, in addition to different supplied power levels to the PHP. Results, validated by previous similar conducted studies showed that the optimal filling ratio for the designed PHP ranged between 40%-70% for both tested fluids. With increased power supplies, the thermal resistance of the device decreased. When comparing the performance of both working fluids, isopropanol seemed to perform better in all working conditions compared to distilled water. As for the inclination, a PHP in vertical position (assisted by gravity), had a more stable pulsating motion. While when in horizontal position, the device failed to operate in a fully pulsating mode, making the designed demonstrator gravity dependent.

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Nomenclature

Bi	Effective Biot Number
D	Diameter (m)
FR	Filling Ratio $(\frac{V_{liq}}{V_{pipe}})$
g	Acceleration due to gravity ($\frac{m}{s^2}$)
k	Thermal Conductivity $(\frac{W}{mk})$
$\mathbf{{}^{\circ}Q}$	Heat Flux $(\frac{W}{m^2})$
Q	Power (W)
h	Heat Transfer Coefficient $(\frac{W}{m^2K})$
L	Length (m)
$\mathbf{D_{int}}$	Hydraulic Diameter (m)
$\boldsymbol{\sigma}$	Surface Tension ($\frac{N}{m}$)
$\mathbf{B_o}$	Bond number
$\boldsymbol{\rho_L}$	Density of liquid ($\frac{kg}{m^3}$)
$\boldsymbol{\rho_v}$	Density of vapour ($\frac{kg}{m^3}$)
$\mathbf{R_{th}}$	Thermal Resistance $(\frac{^{\circ}C}{W})$
$\overline{\mathbf{T_e}}$	Average temperature of evaporator (°C)
$\overline{\mathbf{T_c}}$	Average temperature of condenser(°C)
$\mathbf{P_{sat}}$	Saturation pressure (Pa)
$\mathbf{p_v}$	Vapor pressure (Pa)
$(\frac{dP}{dt})_{sat}$	Ratio between saturation pressure difference and saturation temperature difference$(\frac{Pa}{k})$.

SLR	Single Lens Reflex
SLRC	Single Lens Reflex Camera
PHP	Pulsating Heat Pipe
OHP	Oscillating Heat Pipe
OLPHP	Open Loop Pulsating Heat Pipe
CLPHP	Closed Loop Pulsating Heat Pipe
IRC	Infrared Camera
Al	Aluminium
WF	Working Fluid
v	Kinematic viscosity ($\frac{m^2}{s}$)
α	Thermal diffusivity ($\frac{m^2}{s}$)
β	Thermal expansion coefficient of fluid (K^{-1})
$\mathbf{Ra_L}$	Rayleigh number
Nu	Nusselt number
VP	Vapour Plug
LS	Liquid Slug

Chapter 1

Introduction

Preventing overheating by spreading and rejecting waste heat of electronic devices is necessary for their optimal operation. With increased temperatures, electronic devices efficiency is significantly reduced. So far, micro heat exchangers as well as heat pipes and vapor chambers have been used to efficiently cool these systems. However, electronic manufacturing is becoming more compact, with increasing power levels and densities, thus, higher heat fluxes. Therefore, the spreading and rejecting of waste heat from these devices is also becoming more challenging and costly. For this purpose, pulsating heat pipe devices (PHPs), frequently named oscillating heat pipes (OHPs) have been gaining significant attention in the past years. Their simple structure, alongside having a high ability to remove heat from hotspots makes them an optimal candidate for cooling devices in applications with high heat fluxes, such as electronics [1].

A pulsating heat pipe is a two-phase flow device that consists of a capillary tube bent in several turns. The tube is placed between an evaporator and condenser section and is partially filled with a working fluid. When heat is supplied to the evaporator section, self- excited oscillations of liquid and vapour are created in the tube, transferring in this way heat from the evaporator to the condenser section [2]. The fluid motion is mainly driven by the thermal gradient between evaporator and condenser, which creates a pressure difference between both sides [1].

A pulsating heat pipe, and in particular a closed loop PHP, brings many advantages that conventional heat pipes don't. When compared to the latter, it differs in design and operation, since it contains more fluid, and no wick structure is required to assist the condensed working fluid flow to the evaporator. In addition to that, the thermal entrainment limit is reduced due to its ring circuit closed structure, which avoids the vapor-liquid convection [3].

Some of the unique features the PHP has when compared to a conventional heat pipe, is that vapour plugs don't interfere with the liquid slugs due to the capillarity so that both flow in the same direction. In addition, some of the heat supplied to the evaporator section of the PHP is converted to initiate and sustain the pulsating motion. The latter significantly improves the forced convection and the phase change, enhancing in this way the heat exchange in the device. Additionally, as mentioned previously, a PHP doesn't need a wicked structure such as a conventional heat pipe, which makes it less expensive and more flexible design wise. Finally, it can potentially be manufactured in a way to not depend on gravity [4]. Making it more efficient, and easier to integrate than a conventional heat pipe into different applications.

1.1 Scope/Aim

This thesis is intended to design and investigate a pulsating heat pipe as a demonstrator of the working principles of these devices. The different parameters that will be investigated are the filling ratio, power supply, inclination angle and the working fluid. This will give an insight to the main factors affecting the heat transfer in a PHP, in addition to the optimal operating conditions of the designed device.

The main research questions that are addressed are:

- What are the main geometrical factors to consider when designing a PHP?
- What are the optimal operating conditions for the designed PHP?
- What are the main flow patterns that can be observed in a PHP two phase flow?

1.2 Main delimiters of the work

Results of this work in the image processing section are meant to be qualitative. Extracted images are indicative of the flow development and can give an insight into the different flow patterns which could be further analysed in future work.

1.3 Outline and structure

The thesis is divided into 4 main sections, Chapters 2 and 3 give a general theory and background on heat pipes and particularly PHPs. Chapter 4 and 5 describe the main working principles of a PHP and design parameters. Chapter 6 describes the experimental setup of the conducted work. Results are presented in Chapter 7 and finally Chapter 8 gives the main findings of this study and possible future work.

Chapter 2

Literature review

Despite the simple concept and structure of pulsating heat pipe devices, the governing physical equations are quite complex. When looking at the research conducted in the field of PHPs, it has mainly been experimental, and the PHPs have been optimised based on their specific application [5]. A lot has been investigated regarding different variables effect on the performance of this device. The thermal performance of the PHP is dependent on different parameters which include the geometry and orientation of the PHP, the working conditions of the PHP [6], [7] and the thermo-physical properties of the working fluid [6], [8].

The thermal resistance (R_{th}), is a main indicator of the performance of a PHP, because it provides a direct measure of the average temperature difference between evaporator and condenser at a given heat load. The lower the thermal resistance is, the higher the performance of the PHP [9].

Firstly, when designing a PHP, careful attention has to be put into geometry parameters, as they have profound impact on the heat transfer performance of the device [10]. Specifically, according to [11], the inner surface of the PHP widely influences the capability of heat transport of the device. In this study, the inner and outer tube diameters of the PHP, the thickness of the tube, the number and bend radius of turns, in addition to the overall dimensions of the demonstrator were chosen based on the guidance provided by literature available in this field, comparing several studies that did an in-depth study on the effect each design parameter specifically had on the performance of the PHP. Taking into account the compatibility of the chosen parameters with the thermophysical properties of the selected fluids, isopropanol and water, which were used as working fluids during the experimental measurements conducted in this work.

During the experimental part, the R_{th} can vary with different parameters, such as the filling ratio (FR) of the working fluid. The optimal FR varies with different working fluids. However, for most PHP applications, a FR in the range of 40-60% is common [2], [12].

Another parameter that is often investigated is the inclination of the PHP. [13] observed that placing the PHP in a horizontal position resulted in a sharp increase of thermal resistance. When in this position, no bubble plugs were seen, which suggests that the gravity force can't be neglected, even though surface tension predominates in the experiments using the capillary tubes [13]. This is because in horizontal position, in order for the vapour bubbles to reach the condenser they have to work against the buoyancy force, causing an increase in thermal resistance [9]. To validate this, the designed pulsating heat pipe was tested with different inclinations (vertical and horizontal), to see how gravity impacted the thermal resistance of it.

The power level supplied to the PHP is often discussed in literature, as a parameter impacting the device operation. According to [4], when the power increases, the PHP heat transport capability increases. Different power levels were tested in this study, starting from 30 W up to 90 W of supplied electrical power. The effect the power variation had on the performance was investigated, by measuring the thermal resistance in each case, to obtain the optimal operating range of power for the designed device.

The effect of using different fluids, specifically isopropanol and distilled (DI) water, in addition to a mixture that contained a high percentage of ethanol was also investigated. When looking into previous conducted studies, finding a comparison between DI water and isopropanol wasn't easy. However, comparisons between DI water and other types of fluids were available. For example, [14] analysed the performance of an open loop pulsating heat pipe with both acetone and DI water at different filling ratios and power supplies. For 70% filling ratio, results showed that at lower heat inputs, acetone displayed almost uniformity of temperature. This is because acetone reaches start-up condition faster when compared to water. However, with increasing heat loads, at the same filling ratio, more bubbles were formed which slowed down the heat transfer. At different filling ratios, acetone usually performed better than water, by lowering the temperature difference between the evaporator and condenser section. Interestingly, water outperformed acetone at 50% FR [14].

After the initial stage of designing a PHP, the experimental investigation was conducted. Results of varying the previously discussed parameters are presented in Chapter 7. In addition to that, extracted images of the flow development are presented in section 7.8.

Chapter 3

Heat pipes

A heat pipe is a device where heat is transferred typically over long distances using evaporation and condensation. It contains a working fluid inside a container, which is divided into three sections: the evaporator, the adiabatic and the condenser section. Heat is added to the evaporator and rejected from the condenser, while no heat transfer occurs in the adiabatic section. The temperature difference between the condenser and the evaporator section, causes a difference in saturation pressure. This difference in pressure makes the vapour move from the evaporator to the condenser, the condensate is then pumped back to the evaporator section by forces such as: capillary, electrostatic, gravitational and others [4].

When it comes to the material used for constructing the container, it can range from metals like aluminium or copper and other materials such as plastics. The choice of fluid must also be compatible with the choice of container material. For example, water if used with an aluminium container can cause corrosion, since it contains many non-condensable gases, such as O_2. These different considerations must be well thought of when designing a heat pipe [4].

3.1 Heat pipes classifications

Several types of heat pipes exist, and they can be classified based on different parameters. If the driving force is the capillary force, heat pipes can be mainly classified into:

- Conventional heat pipes, which as mentioned previously, usually contain a wick structure made of a material such as aluminium or copper. A vacuum pump is often used to remove air from the heat pipe [15].
- Thermosyphon heat pipes: they are circulated by natural convection, thus dependant on gravity. The integration is increasing rapidly into systems such as solar panels for cooling [16].
- Pulsating (oscillating) heat pipes (PHPs), which will be of main focus in the following chapter [4].

When it comes to a conventional heat pipe thermodynamic cycle, heat transfer occurs in the following steps:

1) The fluid evaporates in the evaporator, absorbing the thermal energy.
2) The pressure change drives the vapour towards the condenser section with lower temperatures, along the cavity of the heat pipe.
3) The vapour is then condensed back to the fluid, releasing the thermal energy, the wick then absorbs the fluid.
4) The fluid flows back to the evaporator section through the wick structure [17].

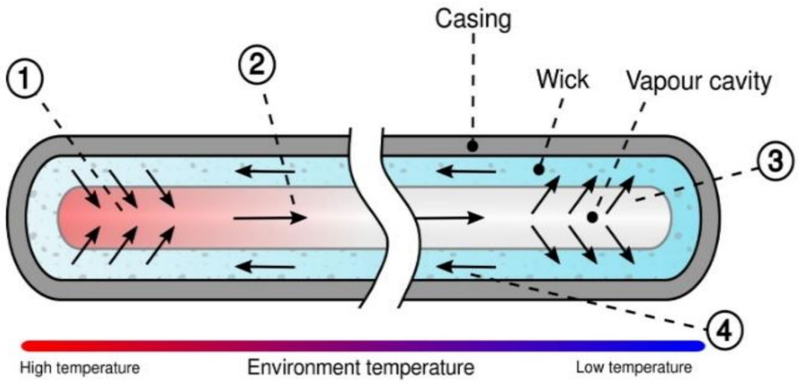


Figure 1 Conventional heat pipe structure. **cc**

As mentioned previously, with the growing need of cooling in different industries, conventional heat pipes designs have often shown some limits into integration with recent industry trends, which led to the evolution of a novel type of heat pipes, PHPs, being a promising cooling technology, which could meet current industry needs more efficiently [18].

Table 1 Conventional, pulsating and thermosyphon heat pipes [19].

	Heat pipes with sintered wicks	PHPs	Thermosyphons
Radial heat flux	Very high ≤ 250 W/cm²	Medium ≤ 30 W/cm²	High ≤ 100 W/cm²
Axial heat flux	High ≤ 600 W/cm²	High≤1200 W/cm²	-
Total power	Medium ≤ 200 W/unit	High ≤ 5000 W	High ≤ 10 kW/unit
Start-up time	Fast: 2-3 seconds	Medium: 2-3 minutes	Medium: 2-3 minutes
Effect of inclination angle	Medium: in bottom heating mode they perform better than in top heating mode	Proper design can reduce effect of inclination angle, bottom heating mode still preferred	Largely affected: only for complex system evaporator above (ex: valves)
Flexibility of 3D space	Low	Highly foldable	Medium/High (Limited by gravity)
Cost	Medium (wick structure required)	Low (no wick required)	Low/Medium

[19] provided a comparison between sintered, pulsating and thermosyphon heat pipes. As seen in Table 1, PHPs bring great advantage when it comes to flexibility of integration such as foldability, inclination angle, reduced costs of construction and handling high heat fluxes.

Chapter 4

Pulsating heat pipes

4.1 Introduction to pulsating heat pipes (PHP)

A pulsating heat pipe is a wickless system usually made of metallic material, in a serpentine form. It mainly consists of three sections: the condenser section, the evaporator section and the adiabatic section. It usually is made of several long continuous capillary tubes and has a filling/evacuating system, that first evacuates the heat pipe with a vacuum pump, then partially fills the tube with a working fluid. The later distributes itself in the form of vapour plugs and liquid slugs in the tube, these plugs and slugs evaporate and condense continuously creating a pulsating motion. The pulsating action of the liquid vapour system transfers heat from one end of the heat pipe to another [13]. It has been shown that the thermally excited oscillating motions increase with increased heat transfer in a PHP [4]. However, if the vapour velocity is bigger than a defined critical vapour velocity value, an annular flow is created due to the vapour penetrating the liquid plugs. When this happens, the oscillating motion of the PHP will stop due to the disappearance of the mass-spring system consisting of a train of liquid plugs and vapour bubbles. In that moment, the PHP heat transport capability has reached its maximum. The latter is called the operating limit of a PHP and is very different from a conventional heat pipe device [4]. Some of the main advantages a PHP brings when compared to a conventional heat pipe, is that being a wickless structure, it’s simple to construct and it can potentially operate in any orientation [15], making it more flexible to integrate into certain applications. In addition to that, a PHP can potentially handle higher heat fluxes than a normal heat pipe, making it optimal for integrating into applications that require large amounts of cooling as already mentioned [1].

PHPs are generally classified into two main categories:

- Open loop pulsating heat pipe device (OLPHP)
- Closed loop pulsating heat pipe device (CLPHP)

A CLPHP might enhance heat transfer when compared to the OLPHP, due to the possibility of fluid recirculation [6]. Therefore, in this thesis work, a CLPHP was designed and investigated.

4.2 PHP’s working mechanism (thermodynamics, fluid dynamics and heat transfer).

The working principle of a pulsating heat pipe is governed mainly by thermodynamics, fluid dynamics and heat transfer principles.

4.2.1 Two-phase flow thermodynamics

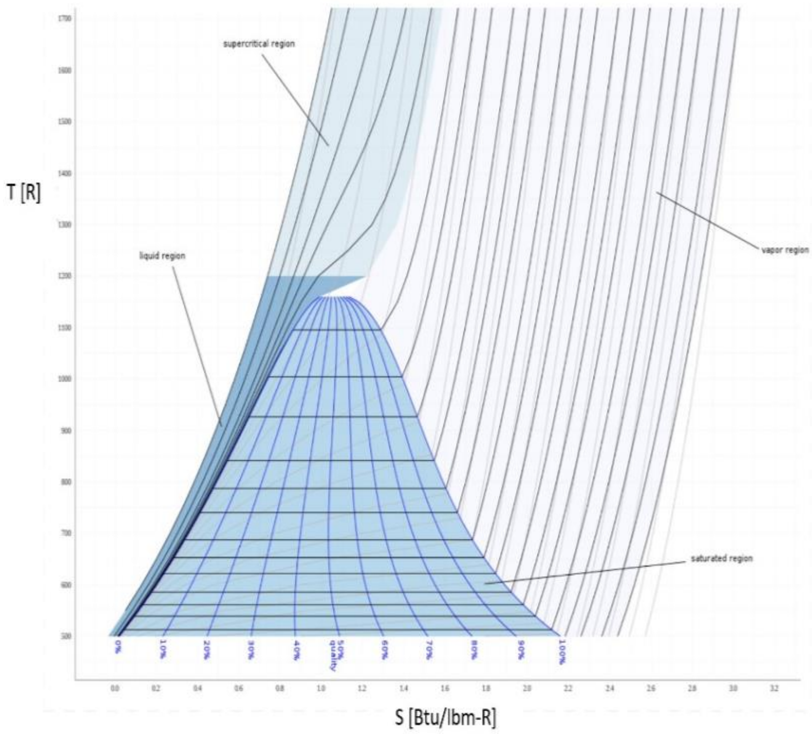


Figure 2 Temperature-entropy diagram for a 2-phase flow. **cc**

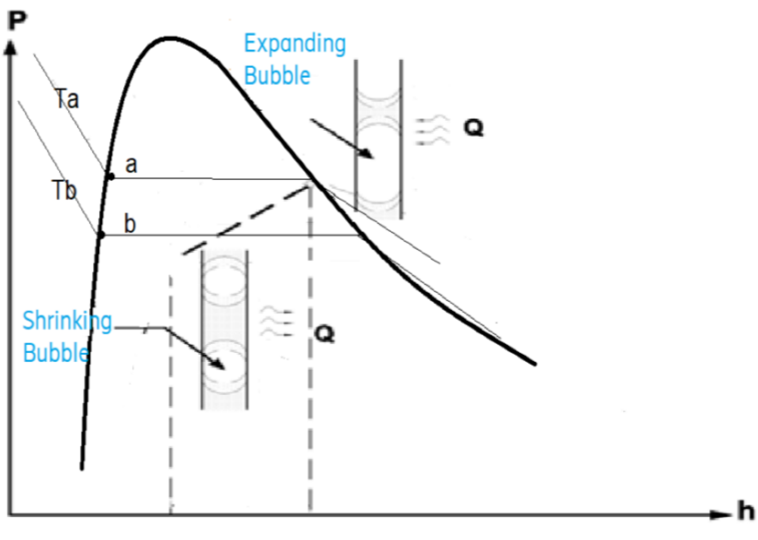


Figure 3 Enthalpy vs. pressure diagram of a control volume in non-equilibrium conditions of a WF.

A subcooled liquid can be defined as a liquid below the temperature of its boiling point. When heat is supplied to the subcooled liquid, it first absorbs sensible heat, it then starts to change phase from liquid to vapour and latent heat is absorbed. The fluid volume then expands due to the lower density of vapour. In systems such as PHPs, that are passive two-phase flow systems used for cooling, the working fluid (WF) pressure and temperature depend on both the operating conditions of the system in addition to its design parameters [20].

Despite the wide research conducted in the field of PHPs, a lot remains unknown when it comes to the exact thermodynamic process of these devices. However, the flow is mainly driven due to vapour bubbles growth and extinction, as seen in Figure 3, in addition to the heat rejection and addition that take place in the condenser and evaporator section of the device [21].

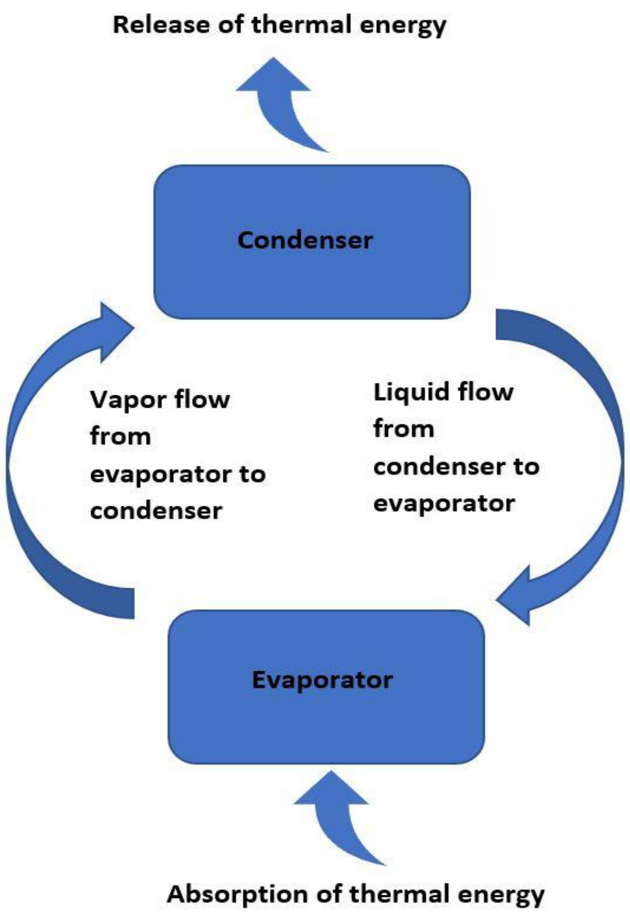


Figure 4 Thermodynamic cycle of a PHP.

The themodynamic cycle of a PHP is described in Figure 4. The fluid, which is distributed in the form of vapour plugs and liquid slugs, initially evaportes in the evaporator absorbing thermal energy, the vapour then flows to the condenser section through the adiabatic section, due to pressure change between both sides. The condenser condenses the vapour releasing thermal enegy. Finally, the liquid flows back to the evaporator section through the adiabatic section [21]. In this last stage, there is no wick structure that assists the liquid as in the conventional heat pipe. During evaporation, the main driving force is the pressure change due to liquid slug evaporating and isochoric compression in the bubble plug, which act as a pumping force. However, when the bubbles reach the condenser, since the PHP has a confined volume, bubbles condense and isochoric expansion of liquid slugs occurs, adittionaly, liquid already occupying the condenser setion will be pushed out of this region and back to the evaporator side, in this way, it can operate without a wick structure. The continuous evaporation and condensation will create this chaotic flow in the pulsating heat pipe, which moves continuously in between both ends through the adiabatic section.

4.2.2 Fluid dynamics

In a capillary tube such as a PHP, when partially filled with the working fluid, the fluid flow forms liquid slugs and vapor plugs, which distribute themselves in the tube. Surface tension forces overcome gravitational forces, making it possible for the liquid slugs to bridge the tube. The surface tension creates meniscus regions at the interface between the solid, liquid and vapour [21]. The trailing edge of the liquid slug when moving, leaves a thin liquid film on the wall of the pipe. The latter is vital, as the main driving force of the PHP pulsating motion is the condensation and evaporation on the thin liquid film [22].

Equation 1 describes the motion of a liquid slug in a pulsating heat pipe device:

$$\frac{dm_{li}v_{li}}{dt} = [p_{vi} - p_{v(i+1)}]A - \pi d L_{li}\tau \qquad 1$$

Where m_{li}, v_{li} and L_{li} are the i^{th} liquid slug mass, velocity, and length. $(p_{vi} - p_{v(i+1)})$ is the vapour pressure difference of two consecutive vapour plugs, which is of main importance for the oscillatory motion, while τ is the shear stress, which is dependent on the nature of the flow, whether it is turbulent or laminar [15].

4.2.3 Heat transfer principles

When the heat pipe is under the action of hot and cold sources, the working fluid performs self-excited motions [23]. When the fluid enters the evaporator section and power is supplied, sensible heat is transferred due to increasing temperatures. Evaporation occurs and the fluid moves to the condenser, where heat is rejected. The phase change takes place mainly in the thin liquid film between the VP and the wall, and in the meniscus region between the LS and VP [15]. The oscillating motion occurs, due to the differential pressure that is generated from latent heat transfer and at low heating powers, the flow will have oscillatory motion in each tube, with main patterns that alternate between bubble flow and plug flow. The fluid flow will change into unidirectional circulation flow with increasing heat input and the main flow patterns occurring will be plug, semi annular and annular flow [23].

The heat transfer performance of a PHP is measured by the thermal resistance, which can be calculated by the following equation:

$$R_{th} = \frac{(\overline{T_e} - \overline{T_c})}{Q} \qquad 2$$

Where R_{th} $[\frac{°C}{W}]$ is the thermal resistance of the PHP, $\overline{Te}$ [°C], is the average evaporator temperature, $\overline{Tc}$ [°C] is the average condenser temperature and Q [W] is the supplied power to the PHP [24]

In cooling applications, the objective is to find the best configuration that lowers the thermal resistance of the PHP. This means that more heat can be transferred due to the pulsation of the heat pipe, thus cooling is more effective, enabling in this way better performance of different devices that are subject to high power supplies, therefore temperatures.

4.3 Non dimensional numbers

Some of the dimensional numbers that give an indicator on the performance of the PHP are presented in the upcoming sections:

4.3.1 Bond number

In order to understand if due to the surface tension confined bubbles can occur, which is of main importance for the functioning of a pulsating heat pipe, the Bond number is calculated. The latter, is defined as the ratio between the gravitational forces and the surface tensions [9].

$$B_o = \frac{(\rho_L - \rho_v)\, g\, Dc^2}{\sigma} \qquad 3$$

Where Bo is the Bond number, σ is the surface tension $[\frac{N}{m}]$, Dc is the critical diameter [m], ρ_L is the density of liquid phase $[\frac{kg}{m^3}]$, ρ_v is the density of the vapour phase $[\frac{kg}{m^3}]$, and g = 9.8 $[\frac{m}{s^2}]$. To ensure capillary flow in the tube, the Bond number should be between 0.4-4 [9]. And the internal hydraulic diameter should be smaller than the critical diameter [9]. Therefore, when dimensioning the PHP tubes, the Bond number and fluids thermophysical properties must be considered, as the properties of the working fluid and geometry of the pulsating heat pipe have a profound impact on the PHP performance.

4.3.2 Weber number

The Weber number represents the interaction between the vapour and liquid phases, it can be defined as the ratio between the inertia and capillary forces. In two phase flows such as PHPs, quantifying it can give an important indicator on the nature of the flow. In literature, for PHP flows, a common proposed limit for the Weber number is 4 [15].

The Weber number can be defined according to the following equation:

$$W_e = \frac{\rho_L (u_l - u_v)^2 d_h}{\sigma} \qquad 4$$

Where u_l is liquid average velocity $[\frac{m}{s}]$, u_v is average vapour velocity $[\frac{m}{s}]$, d_h is diameter of the droplet [m], ρ_L is density of liquid $[\frac{kg}{m^3}]$ and σ is the surface tension $[\frac{N}{m}]$ [15].

4.3.3 Biot number

The Biot number indicates the applicability of certain methods in solving heat transfer problems, it can be defined according to Equation 5:

$$Bi = \frac{h_{eff} L}{k} \qquad 5$$

Where k is the thermal conductivity of the body [$\frac{W}{m.K}$], L is the characteristic length of the geometry [m], h_{eff} is the convective heat transfer coefficient [$\frac{W}{m^2.K}$] which can be defined as:

$$h_{eff} = \frac{\mathring{Q}}{(T_{avg} - T_{\infty})} \qquad 6$$

T_{avg} is the average surface temperature [K], T_{∞} is the ambient temperature [K] and $\mathring{Q}$ is the heat flux [$\frac{W}{m^2}$].

For the particle to be thermally thin, the Biot number can't exceed a certain limit. A thermally thin particle indicates that the temperature gradients inside the body can be neglected, since the heat convection at the surface of the body is much bigger than the conduction resistance inside the body [25]. According to [26] that calculated the Biot number for a similar experiment with a similar dimensioned PHP, in all cases the Biot number resulted to be <0.1. Thus, it's safe to assume that the temperature inside the tube is equal to the temperature on the outer surface of the tube. Similarly, the variation of the temperature of the wall over the angular coordinate is also negligible [26].

4.4 Fluid selection

The selection of an appropriate fluid coupled with the designed geometry is crucial for optimising the performance of a PHP. The appropriate choices will maintain a conventional reliability performance, inducing less stresses in the electronic packaging layers, which is important for optimal operation. Thus, the PHP has to generally be able to operate with dielectric fluids and with fluids at low pressures (~ 1 bar) [5]. According to [27], some other aspects which are impacted by the choice of fluid are:

- The share of sensible and latent heat in the total heat output.
- Flow patterns in the device.
- The pressure drops and average flow velocity.
- The shape, agglomeration, breakage and bubble nucleation [27].

Some of the main thermos-physical properties, which have to be considered when selecting the working fluid are the viscosity, the thermal conductivity, surface tension, boiling point and heat of vaporisation [6]. The fluid should have a high $(dp/dt)_{sat}$, so that a large P_{sat} is generated inside the bubble with a small variation of $\overline{T}_e$. This helps the device in the bubble pumping [28]. As to surface tension, it is also vital to consider. Surface tension is defined as "a force that operates on the surface and acts perpendicularly and inwardly from the boundaries of the surface, which decreases the area of interface", this results in the liquid forming a shape with a minimum area and In case of no gravity, the shape will be perfectly spherical [4]. For PHP fluids, low values of the surface tension and viscosity are desirable, since this helps avoid additional pressure drop and shear stress [28]. However, the selection of the fluid must be an optimum trade-off of these thermophysical properties, and different fluids work better in different operating conditions.

Since alcohol and water tend to differ in these properties, yet according to previous conducted experiments, both seem to be suitable as a working fluid for PHPs. Two fluids were selected to investigate their performance for this study case, distilled water and isopropanol.

Table 2 Thermophysical properties of DI water and isopropanol at 20 °C.

	Isopropanol	DI water	Unit
ρ_V	2.67	0.017	kg/m^3
ρ_L	785	998	kg/m^3
Surface tension σ	0.021	0.072	N/m
Melting point	-88	0	°C
Boiling point at 1 bar	82.5	100	°C

4.5 Geometry of the PHP

When designing a pulsating heat pipe, the main objective is that a sufficient amount of energy for running the device is provided through the heat input to it, in a way that the external pump is removed. To achieve that, the inner diameter of the device should be sufficiently small, in a way that capillarity plays a significant role so that vapour plugs and liquid slugs can be formed alternatively when filling the pipe with the liquid [29].

As mentioned previously, the Bond number is the main indicator whether capillary flow will occur in a tube. And in the conducted experiments, two fluids were tested, isopropanol and distilled water.

D_c was calculated with setting the Bond number to 4. According to Beardmore and White experiments as mentioned in [5], when the Bond number is equal to 4, the terminal bubble velocity is equal to 0, as the bubble velocity is inversely proportional to the diameter and Bond number. Therefore, for surface tension to dominate gravitational forces, the Bond number should be < 4. In view of this definition, the critical diameter can be calculated according to Equation 3 mentioned previously in section 4.3.1.

Additionally, for separation to occur, the hydraulic diameter should be smaller than the critical diameter for the specified fluid [5].

However, according to [15], referencing Qu et al. in “Thermal performance of micro pulsating heat pipe”, the internal diameter should have also a minimum limit, in fact the hydraulic diameter (D_{int}) should be in the range of:

$$0.7\sqrt{\frac{\sigma B_o}{(\rho_L - \rho_v)g}} < D_{int} < 2\sqrt{\frac{\sigma B_o}{(\rho_L - \rho_v)g}} \qquad 7$$

Based on the previous considerations and Table 2, in addition to considering that the PHP should function with both selected working fluids, the minimum and maximum critical diameters were calculated. Results are shown in Table 3, which suggests that the internal diameter of the PHP should be in the range of 1.89 < D_{int}< 3.33 mm. The final chosen internal diameter was of 2.4 mm.

Table 3 Internal and critical diameters of PHP based on Bond number.

	Isopropanol	DI water	Unit
Internal diameter	2.4	2.4	mm
Bond number	2.06	0.78	-
Maximum critical diameter (D_{cmax})	3.33	5.42	mm
Minimum critical diameter (D_{cmin})	1.17	1.89	mm
Gravity	9.8		$\frac{m}{s^2}$

4.5.1 Turns

When it comes to the choice of the number of turns, a critical number is required to make the PHP work independent of gravity. In literature there is no clear guidelines on the optimum number of turns. Generally speaking, a larger number of turns seems to increase the PHP performance and the evaporator heat supply area does too [6]. In addition to that, the larger the channel density the better the PHP seems to perform [1]. However, the fact that too many turns of the PHP can also limit the number of applications has to be considered, since integrating it in a compact system becomes more challenging. [7] suggests that a PHP with less than 5 turns can limit the oscillations amplitude and frequency. All the above should be evaluated when choosing the number of turns for a designed PHP. Based on previous considerations, for the designed PHP, a number of turns equal to 10 was chosen, as it's thought to be a suitable number to obtain efficient heat transfer.

4.5.2 Curvature of diameter

When a bend of a curve is smoother, the flow resistance is limited and pressure losses are reduced in that region [30]. This is because pressure loss is the sum of a $\frac{R}{D}$ term, where R is the radius of the bend and D is the diameter of the pipe [30], [31]. The objective when designing a PHP is having higher heat transfer, which occurs with lower pressure loss[30]. Thus, having a larger bending radius is more beneficial for PHP performance. Based on that, a bending radius of 7.5 mm was chosen for the PHP, which seemed to be adequate for the aimed size of the demonstrator, with a sufficient number of turns.

4.5.3 Evaporator and condenser section.

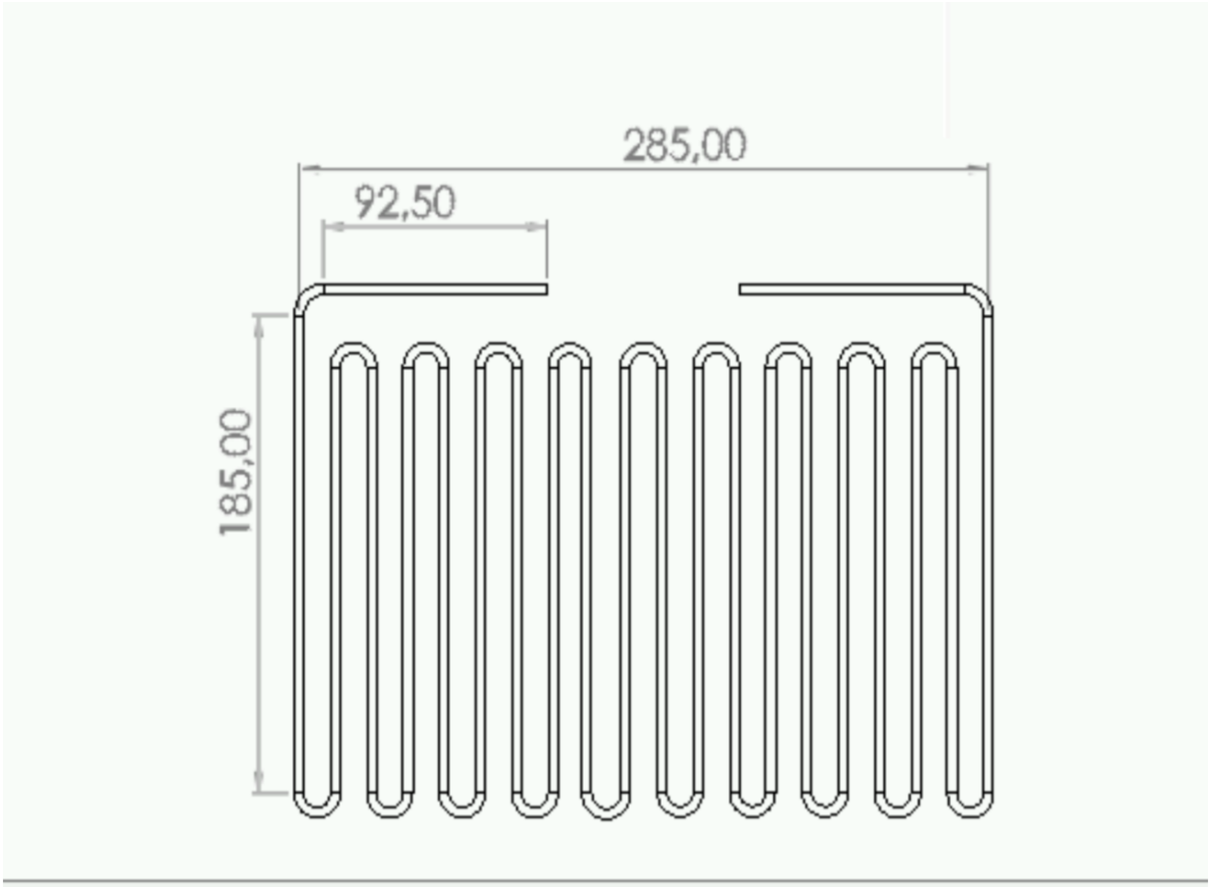
According to [21], when designing a PHP, it is vital to avoid dry out conditions, which is when the fluid completely vaporizes due to high heat input and low filling ratio, making conduction the only heat transfer method, thus decreasing the efficiency of heat transfer. The same study reported that with increased evaporator section length, the critical heat transfer flux decreased. Thus, for efficient heat transfer in a PHP, it is a thumb rule that the condenser section should have a larger area than the evaporator section. In the designed heat pipe, an aluminium plate was used to spread the heat between the evaporator and condenser. Thus, defining the exact area of evaporation and condensation was tricky. Since the fans used for cooling in the condenser section occupied a larger area that the resistors used for heating in the evaporator section, the condenser section had a larger size than the evaporator section.

As for where to place the evaporator section, [32] tested placing the heating (evaporating section) in three different positions: top, middle and the bottom of the pulsating heat pipe. Results showed that placing the evaporating section at the bottom yielded a smaller difference in temperature than the other two cases, indicating a better heat transport in the heat pipe. Therefore, in this study case, the evaporator section was placed at the bottom of the aluminium plate and the condenser section at the top, keeping the adiabatic section in between.

Table 4 shows the final design parameters, that were chosen for the PHP, after taking into account all the above mentioned.

Table 4 Design parameters of the PHP.

Design parameter	Numerical value
Internal diameter	2.4 mm
External diameter	4 mm
Number of turns	10 turns
Length	285 mm
Thickness of tube	0.8 mm
Height	185 mm
Bend radius of tubes	7.5 mm



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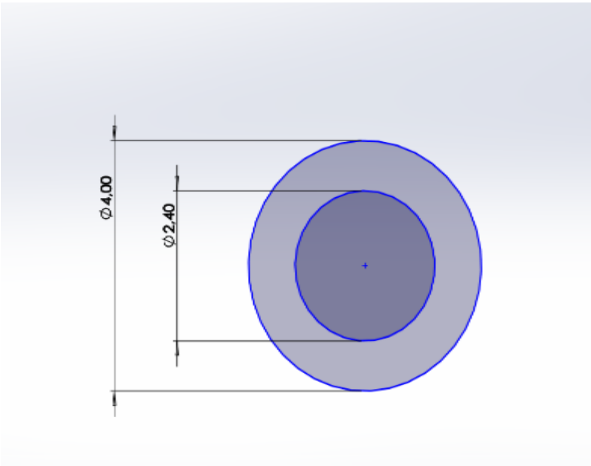
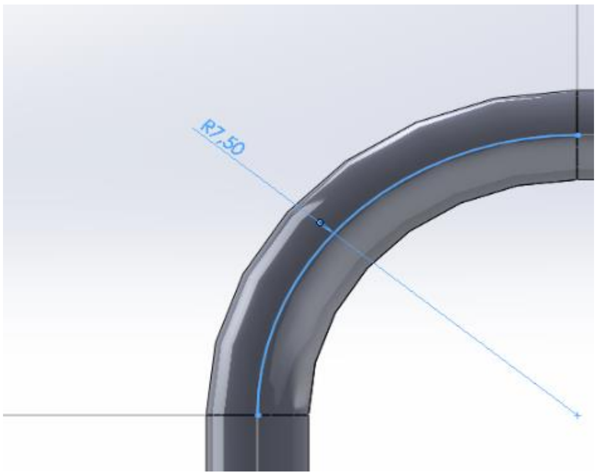




B **C**

Figure 5 Design parameters of the pulsating heat pipe [mm]. **A**: Geometry of the PHP, **B:** Tube internal and external diameter, **C:** Curvature of diameter.

4.6 Influence of operating parameters on PHP performance

4.6.1 Surface tension

Among all parameters affecting the performance of the PHP, the thermophysical properties of the fluid have a big impact. Particularly, the surface tension of the selected fluid seems to have a significant effect on the pulsating motion of the device. The capillary flow formation of liquid plugs and slugs are highly dependent on the surface tension of the fluid for a selected tube geometry. When the surface tension increases, the capillary resistance increases too, reducing the heat transfer in the device [29]. Surface tension also has a meaningful impact on the thin film thickness in both the evaporator and condenser section, in addition to affecting the evaporator temperature. Surface tension increase leads to vapour temperature decrease according to [22], while the effect the capillary force has on the performance can be neglected. Fluids with higher surface tensions are usually harder to obtain a pulsating motion under certain conditions, while fluids with very low surface tension can provide a pulsating motion faster, thus the start-up of a PHP can be facilitated when using such types of fluids at different operating conditions [33]. However, higher surface tensions increase the allowable critical diameter of the designed PHP, which indicates room for more fluid, thus potentially increase heat transfer for an application. Nonetheless, that also indicates a higher pressure drop in the pulsating heat pipe. Therefore, higher heat supplies might be needed to maintain the same pulsating motion if a bigger diameter is used [29], this might not be optimal for certain applications. All these considerations must be thought off when picking a fluid with specific surface tensions.

In the following study two fluids have been tested with different surface tensions. DI water, with a surface tension of 0.072 $\frac{\mathrm{N}}{\mathrm{m}}$, in addition to isopropanol, which has a surface tension of 0.021 $\frac{\mathrm{N}}{\mathrm{m}}$. A comparison is presented in section 7.7 of results and discussion.

4.6.2 Inclination angle in respect to horizontal position

Some studies suggested that a PHP, in some conditions, can work independently of orientation. Depending on the design parameters and working fluid properties. When the PHP is in horizontal position in respect to the surface $\theta = 0^{\circ}$, gravity is not accounted for. When the PHP is placed vertically $\theta = 90^{\circ}$, with the evaporator on the bottom, gravity is not absent thus will have an impact on the PHP performance [34].

A wide range of studies has been conducted on the effects gravity had on the performance of the PHP, specifically on tubular and planar geometries. Results of previous conducted studies showed that in these geometries, the position of both the condenser and evaporator sections with respect to gravity affects the performance of the device. Generally speaking, PHPs perform better when inclination is increased with respect to horizontal position, since gravity aids the device when inclined [35]. However, literature suggests, that with optimal design of the PHP, the effect gravity has on the performance can be reduced significantly. Making integrating the device into applications more flexible.

After placing the PHP in a vertical position, and measuring the performance for different filling ratios, power supplies and working fluids, the optimum operating condition has also been tested in a horizontal position of the PHP, to see the impact gravity has on heat transfer of the device. The comparison between both inclinations is presented in section 7.3 of results and discussion.

4.6.3 Filling ratio

The filling ratio has a big impact on the PHP heat transfer performance. For too low filling ratios, not enough liquid slugs are formed. While at too high filling ratios, not enough bubbles are formed to provide pumping action [36]. Testing different filling ratios for a selected fluid in a designed PHP is necessary for optimising the device when integrating into applications.

Previous studies have shown that PHP devices seem to operate best at a filling ratio ranging somewhere in between 40-60% for a selected fluid [12]. In section 7.2 of results and discussion, the optimal FR for the designed PHP is presented.

4.6.4 Heat input effect

A specified threshold value of input on heat flux exists at which the working of the PHP starts. At saturated conditions, the boiling of the fluid starts, thus small tube-size bubble formation occurs. The threshold value should be lower than the designed heat dissipated, so that the heat transfer device operates in time to prevent the failure of the electronic device due to overheating [17]. In this work, different heat inputs ranging from 30-90 W were tested for the different working fluids. The effect different power levels had on the pulsation of the heat pipe is presented in results and discussion section 7.4.

It’s important to mention that for the designed heat pipe, there’s a small percentage of lost power due to natural convection, which is caused by buoyancy forces, induced by density difference due to temperature difference in the fluid. This will result in a small amount of heat transfer, that will occur from the heat source (plate) in the adiabatic section to the surrounding air [37], causing a loss of power, which is neglected in the results. An estimation of that lost power can be calculated through the following formula:

$$Q_{loss} = hA(T_s - T_f) \qquad 8$$

Where:

Q_{loss} is the rate of heat transfer by natural convection from the Al plate to the air [W].

h is the natural convection heat transfer coefficient of solid to gas ~ (2-5) [$\frac{W}{m^2.K}$].

T_s is the average surface temperature of the adiabatic section (for ex. At 90 W with cooling fans, T_s=55°C).

T_f is the air temperature at a distance from the surface ~20℃.

A is the area of the adiabatic section on both sides of the plate (A=0.1032 x10^{-6} m^2).

To calculate the exact heat transfer coefficient, we assume having a vertical plate of 150 mm.

for air at ambient temperatures:

α=2.315 x 10^{-5} $\frac{m^2}{s}$	α is the thermal diffusivity.
v=1.643 x 10^{-5} $\frac{m^2}{s}$	v is the kinematic viscosity.
k=0.02672 $\frac{W}{m.K}$	k is the thermal conductivity.

The thermal expansion β , can be described according to Equation 9:

$$\beta = \frac{1}{T} \qquad 9$$

At ambient conditions, $\beta = \frac{1}{273+20} = 0.00341\ K^{-1}$.

The Rayleigh number, can be defined according to Equation 10:

$$Ra_L = \frac{g.B.\Delta T.L^3}{\alpha.v} \qquad 10$$

For this case, $Ra_L=\frac{9.8*0.00341*30*0.15^3}{1.643*2*315*10^{-10}}$ =0.9*$10^7 < 10^8$, the flow is laminar.

Using The Churchill-chu correlation [38], with Prandtl number (Pr)=0.71 for air:

$$\overline{Nu_L} = 0.68 + 0.67Ra_L^{1/4}[\ 1 + \left(\frac{0.492}{Pr}\right)^{9/16}]^{-4/9} \qquad 11$$

the Nusselt number ($\overline{\text{Nu}}$) can now be calculated:

$$\text{Nu} = 0.68 + 0.67 * (0.9 * 10^7)^{1/4}[\,1 + \left(\frac{0.492}{0.71}\right)^{9/16}]^{-4/9} = 28.85$$

The coefficient of natural heat convection can be defined according to Equation 12:

$$\text{h} = \frac{\text{Nu.k}}{\text{L}} \qquad 12$$

Substituting the values in Equation 12, h= 5.14 $\frac{\text{W}}{\text{m}^2\text{.k}}$.

Finally, using Equation 8, the heat loss due to natural convection can be calculated:

$$\text{Q}_{\text{loss}} = 5.14 * 30 * 0.1032 * 10^{-6} = 1.59 * 10^{-5}\ \text{W}.$$

When compared to 90 W of supplied power, the amount of heat loss due to natural convection is totally neglectable. Therefore, for the thermal resistance calculations and all results presented, the electrical power supplied from the power source is assumed to be the same power that the pulsating heat pipe receives.

4.6.5 Cooling effect

Trials were conducted for the different filling ratios and fluids with and without cooling. By attaching fans to the condenser section, to see the impact that would have on the pulsating motion and the thermal resistance of the PHP.

Cooling was tested by:

- Fixing the rotational speed of the fans for the different power supplies.
- Maintaining the condenser section at a constant temperature (~32° C), by increasing the rotational speed of the fans for the different power supplies.

Conclusions are presented in results and discussion section 7.5, on how cooling the condenser section affected the R_{th} of the PHP in both cooling configurations, comparing them to the case of no cooling in the condenser section.

4.6.6 Evacuation pressure effect

The effect the evacuation pressure had on the start-up of the pulsating heat pipe was investigated. Initially, the heat pipe was not evacuated and was partially filled with distilled water at 40% filling ratio, different power levels were then supplied to the device. The average temperature of condenser and evaporator where measured, and the thermal resistance of the PHP was calculated. No pulsating occurred during the different power supplies, which aligns with what [33] concluded, when using water as a working fluid in a PHP at atmospheric pressure (760 Torr), the thermal resistance of the PHP is too high to make it pulsate. However, when using different fluids in the PHP, such as HFE-7000 refrigerant, which has a very low surface tension, the pulsation can occur even at atmospheric pressure without evacuation. The same study compared the thermal resistance of water in a straight portion of the PHP, both at atmospheric conditions and when evacuated. Results showed that the thermal resistance of water at atmospheric pressure, was about 10 times bigger than the thermal resistance of water in evacuated conditions. Therefore, at atmospheric pressure conditions, the capacity of water in removing heat is extremely low, and the heat pipe can't act in a pulsating mode [33].

This was validated by results obtained in the current study, where different power supplies were tested in case of no evacuation. However, at the largest supplied power of 90 W, in the case of DI water, the heat pipe produced a slight moving motion but not a pulsating one. [33] also states that this is related to non-condensable gases which may deteriorate the heat transfer noticeably. Since water when compared to other certain fluids has a higher solubility of non-condensable gases, when using water in a PHP, the non-condensable gases accumulate next to surface, which might prevent the water vapour molecules from condensing, reducing the heat transfer coefficient in this way. [39], [40] further explain that a non-condensable gas boundary is created when these non-condensables accumulate on the interface of gas and liquid, hindering in this way the condensation process due to mass transfer resistance. While in some other fluids, vapour molecules can still condense on the liquid film, in this way maintaining the circulation of the working fluid between the evaporator and the condenser, thus providing a pulsating motion even at atmospheric pressure [33]. When using isopropanol, at higher power supplies with an average evaporator temperature of around 85°C, which is above the boiling temperature of isopropanol. Pulsation occurred even with no evacuation. Therefore, the correct selection of working fluid can facilitate the start-up of a pulsating heat pipe, at different evacuation conditions.

Chapter 5

PHP operation

5.1 Start-up of pulsating heat pipe

When filling the pulsating heat pipe with the working fluid, the surface tension makes the fluid distribute itself in the form of plugs-slugs in the tube. At ambient conditions, before supplying power, the vapor is at a saturated state inside the tube. When the fluid receives heat, different frictional forces for the directional motion are created due to the uneven distribution of the plug-slug in the tube. With continuous heat supply, temperature rises and the liquid slugs evaporate, forming tube size bubbles which expand. In the meantime, the condenser section simultaneously cools down the fluid causing heat dissipation. This makes the vapour volume and low-pressure zone decrease. Since the device is isochoric, the same generated vapor at the evaporator section makes the moving of the working fluid possible, due to random pressure and volume distribution in the tube [17].

During the start-up phase of the PHP, the process typically changes from pure heat conduction to oscillating movement of the working fluid, due to phase change. In the initial stage, the temperature of the evaporator increases, while no pulsation occurs. After some time, the temperature difference between both the evaporator and condenser section, thus pressure imbalance between them will cause the PHP to successfully start-up. With continuous pulsation, the evaporator temperature drops and reaches stable operating conditions. Stable circulation of the fluid eventually occurs, enhancing in this way the heat transfer of the two phase flow device [41].

5.2 Pulsating mode of operation

In the pulsating motion of the PHP, 3 distinct modes exist:

a) PHP with too high filling ratio (close to 100 %): When the volume of the tube is filled with working fluid. In this condition, the fluid is mostly in liquid form and a small number of bubbles is present, thus pulsating mode doesn’t occur.

b) PHP with too low filling ratio (close to 0 %): When the pipe is not filled with enough working fluid, the evaporator can risk dry-out. Unstable and undesirable conditions occur in this case.

c) Working range of PHP: for the device to operate at a fully pulsating mode, it should be filled between 10% to 90% of fill charge, depending on the working conditions and design of the device. For lower filling rates, more bubbles are formed, but less liquid mass for sensible heat transfer can occur. With higher filling rates, less bubbles are present, thus less perturbations and chaotic flow occur in the PHP, causing a drop of performance. The optimal filling ratio is found by testing different ones, for the different fluids in the different working conditions of the PHP [13] .

5.3 Performance determination

When supplying electrical power to the thick film resistors in the evaporator section, the PHP will receive an amount of electrical power which is equal to:

$$Q = V \times I \qquad 13$$

Where Q is the electrical power [W], V is the voltage [V] and I is the current [A] of the power source.

Neglecting a small amount of power loss which can classified into the following:

- Heat loss due to natural convection in the adiabatic section, as mentioned previously.
- The contact of the PHP brings an additional thermal resistance which reduces the power supply. However, as [15] states, a properly mounted heat exchanger has a very small contact thermal resistance, thus the power which might be dissipated due to contact has been neglected in the performance calculations.
- The aluminium plate that was used as a heat spreader is 2 mm thick. The thickness is considered negligible as the heat flux on both sides of the plate is assumed to be equal.

Using the Fourier law of heat conduction, the effective thermal conductivity can be used to estimate the performance of a pulsating heat pipe [4], [17]. The filling effect K_{eff} $[\frac{W}{m.K}]$ on the performance of the closed loop pulsating heat pipe can be described by Equation 14:

$$K_{eff} = \frac{Q \text{ x } L_{eff}}{A * (\overline{T}_e - \overline{T}_c)} \qquad 14$$

$\overline{T}_e$ is the average surface temperature of the evaporator section and $\overline{T}_c$ is the average surface temperature of the condenser section.

The effective length of the pulsating heat pipe (L_{eff}), can be defined as the distance between the condenser and evaporator section:

$$L_{eff} = 0.5 * (L_e + L_c) + L_a \qquad 15$$

Where:

L_e is the length of the evaporator section [m].

L_c is the length of the condenser section [m].

L_a is the length of the adiabatic section [m].

A is the cross-section area of the pulsating heat pipe [m^2], which can be defined according to Equation 16:

$$A = \frac{n\,\pi d^2}{4} \qquad 16$$

With n being the number of tubes in the PHP and d [m] is the hydraulic diameter of the channels [17].

Using the electrical analogy, the overall thermal performance of the PHP is therefore also determined by the thermal resistance, described in Equation 2 [17]. With increased thermal resistance, the performance of the PHP deteriorates. With more pulsating motion, the thermal resistance is generally decreased, since a bigger amount of heat is transferred due to convection and phase change.

Chapter 6

Methodology and experimental setup

Several trials with an entirely glass manufactured pulsating heat pipe device led to modify the choice of the material, due to many obstacles faced when installing and trying to start-up the PHP. As seen in Figure 6, the filling section was manufactured entirely in glass for the initial design of the PHP. Leakage occurred when filling it with the working fluid, thus completely evacuating it was not possible with the pump. This also made it impossible to completely distribute the fluid in all the pipe sections. When trying to evacuate the fluid, to change the filling ratio in the PHP, the first adopted solution was to place the PHP in an oven at 105 °C to evaporate the working fluid. When filling the PHP after completely evacuating it using this methodology, leakage occurred in the evaporator section U turns. This is probably due to the thermal glue (seen as blue colour in Figure 6) that was used when attaching the pulsating heat pipe to the aluminium (Al) plate, which acted as a heat spreader for the PHP. Since the thermal coefficient of expansion of aluminium is different from that of borosilicate glass, the aluminium plate expanded when heating it in the oven, causing the glass to expand with the plate, which lead to cracking of the glass in certain areas next to the heat source in the U turns of the evaporator section, as they were directly glued to the Al plate.

The thermal expansion that happened in the Al plate can be expressed according to the following equation:

$$\Delta L = L_O \alpha \Delta T \qquad 17$$

Where ΔL is the change in length of the aluminium plate [m], L_O is the initial length of the Aluminium plate [m], ΔT is the difference in temperature between the reference state and the new state [°C], and finally α is the thermal expansion coefficient of Aluminium ~ $22x10^{-6}$ C^{-1}.

For the Aluminium plate that was used, when placing it in the oven at 105 °C, the elongation of the plate was of about 0.4 mm. Since the PHP glass had a thickness of only 0.8 mm, the expansion was sufficient to cause it to crack in certain areas. A different solution was then adopted to attach the pulsating heat pipe to the aluminium plate. To solve this problem, two Teflon blocks were manufactured, and holes were drilled to the plate in the evaporator and condenser sections. M3 screws were used to attach the Teflon blocks to the plate. This allowed the thermal expansion of the aluminium plate when heated, without cracking the glass, since the pipe is not directly attached to the plate in this configuration, rather placed in between. However, this didn't entirely solve the issue, since the whole section was over dimensioned, causing too much mechanical stress in certain areas, especially when filling or evacuating. Cracking of the glass in the T-joint section eventually occurred.

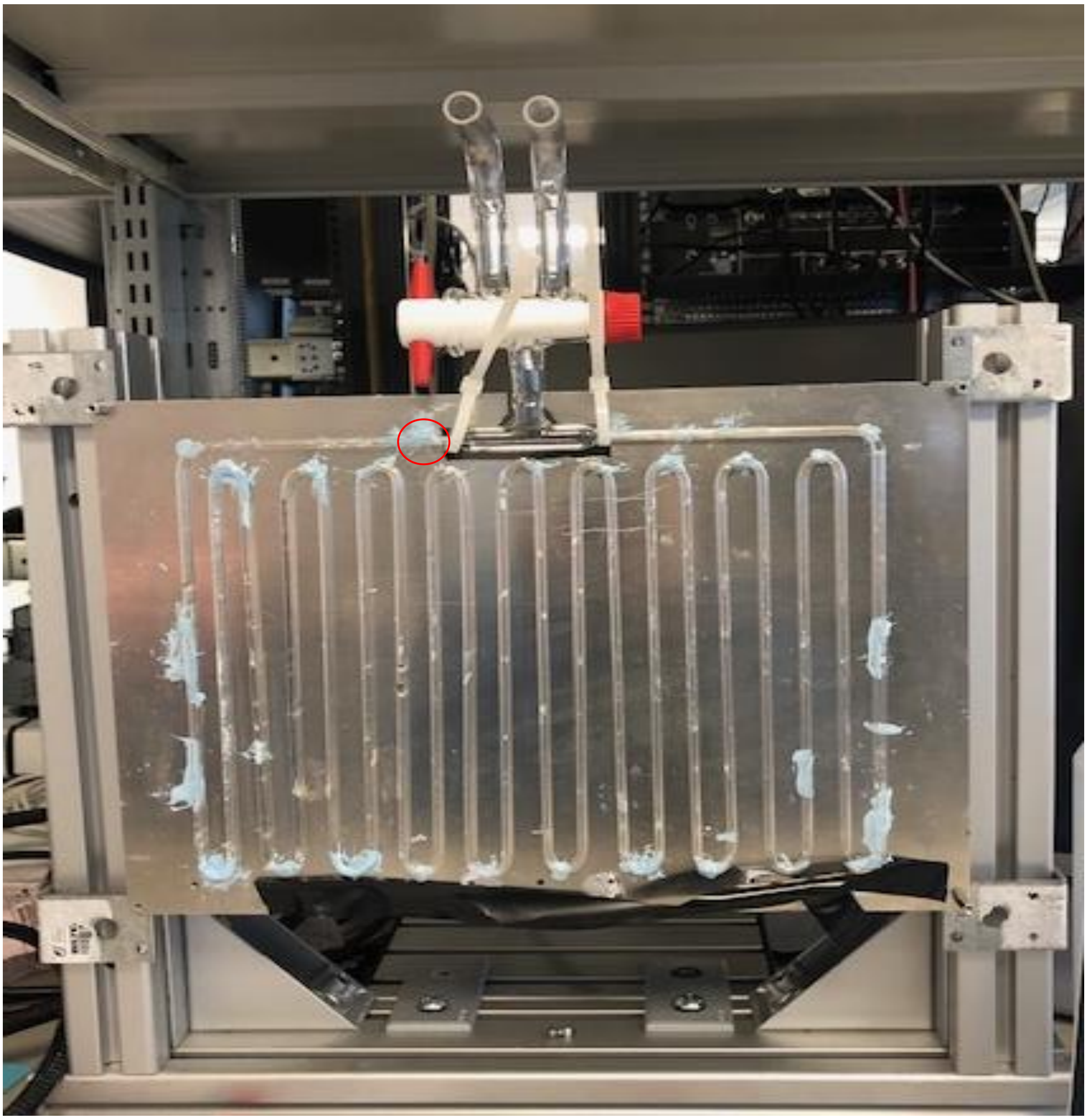


Figure 6 Cracks that occurred at the glass T-joints in the pulsating heat pipe.

All these complications led to rethinking the choice of a completely glass manufactured PHP. Thus, an alternative material choice for the filling port was adopted. In order to have a more durable pulsating heat pipe, yet still be able to view the formed bubbles, a copper-glass pulsating heat pipe was opted for. The broken glass manifold was removed using a diamond cutter and new filing part made of a CPS Kulventil BV Rak/ bojd ¼-FFL X 1/4-MFL metal valve was attached to 4 mm coper tubes, forming a new T-joint. The copper tubes were then attached to the glass pulsating heat pipe with a thermally shrinking silicon tube, instead of directly gluing them together. This was done to avoid any complications which could be caused by different thermal expansion coefficients of glass and copper, when subjected to heat, as had happened in the initial stage of the experiment.

To enhance heat transfer in the PHP and get a better thermal contact with the Al plate, copper tape and thermal putty were added to the heat pipe. In addition to that, two mechanical supports were attached on the frontside and backside of the plate to act as a support to the valve. This guaranteed that no force would be directly applied to the glass heat pipe when filling or evacuating it.

The final configuration of the PHP setup, that was opted for during the experimental part measurements, is shown in Figure 7.

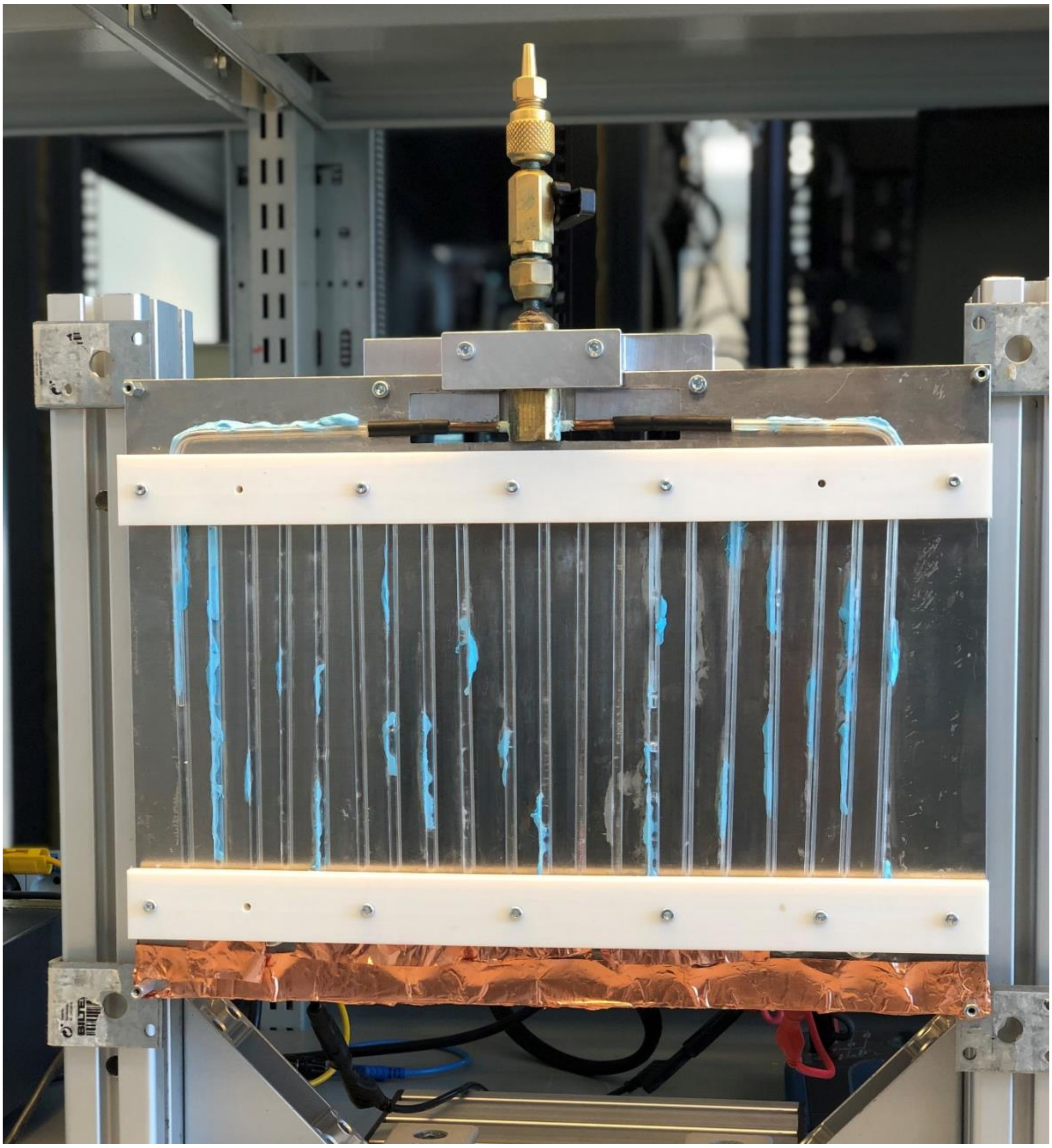


Figure 7 New glass-metal pulsating heat pipe design.

6.1 Experimental setup

A closed loop glass pulsating heat pipe, with an internal diameter of 2.4 mm and external diameter of 4 mm was manufactured for the purpose of this study. As mentioned previously, for the initial trials, a glass T joint shaped filling / evacuating port connected both ends of the PHP. This was later modified to a valve attached to the glass pulsating heat pipe, with copper and silicon tubes. A vacuum pump was used to degas the fluid and remove the non-condensable gases, completely evacuating the pulsating heat pipe. The evacuated tubes were then filled with the working fluid using a syringe, to a determined filling ratio.

The chosen material for PHP was borosilicate glass, in order to be able to view the plugs and slugs in the capillary movement. A 2 mm thick aluminium plate, with an area of (344 x 253) mm^2 was manufactured to act as a heat spreader. In a previous trial, thermal glue was used to glue the PHP to the plate. However, that resulted in glass cracking in the evaporator section, when heating the pipe, since the thermal coefficient of expansion of Aluminium and glass are different as mentioned previously. Thus, in the new configuration, the pipe was attached to the plate with 2 Teflon blocks, each 8 mm thick, using M3 screws, to allow the plate some expansion when heat is supplied. Equidistant holes were drilled to the plate and 8 TO-220 thick film resistors were placed on the bottom of the back side of the plate, along the evaporator section, connecting them in a combination of series parallel connections as seen in Figure 8. Supplying in this way power to the Aluminium plate at different rates. An ammeter was added in order to measure the voltage and current and a DC power source was connected to the circuit. The supplied electrical power was varied gradually from 30 W to 90 W. Two axial fans were attached to the condenser section of the pulsating heat pipe to enhance cooling. Since the motion of the working fluid when supplying power and cooling is the main reason behind the heat transfer performance of the PHP, proper choice and optimization of these setup parameters must be done to guarantee the best heat transfer in it.

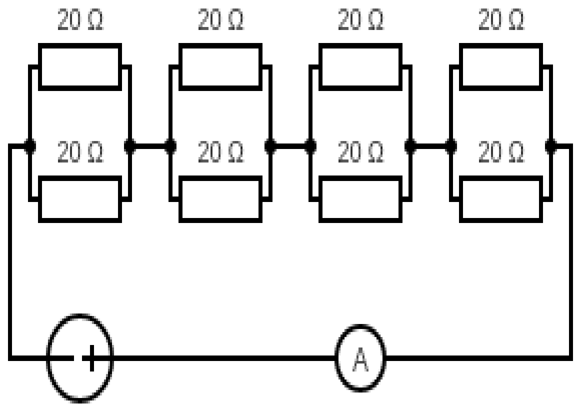


Figure 8 Electrical circuit of the PHP.

An infrared camera (IRC) was then used to measure the average temperature of evaporator and condenser sections for the different power supplies. The IRC readings were calibrated using T-type thermocouples. The thermal resistance of the PHP was then calculated, which indicates the thermal performance of the designed device.

Table 5 Calibrating average temperatures of IRC with T type- thermocouples, for 90 W power supply and 70% FR using DI water.

Section	T-type thermocouple	Readings	Average of readings	IR camera average temperature
Evaporator	Chanel 1	84.26 ºC	79.25 ºC	78.50 ºC
	Chanel 2	80.86 ºC		
	Chanel 3	77.43 ºC		
	Chanel 4	73.53 ºC		
Condenser	Chanel 5	33.40 ºC	33.40 ºC	35.00 ºC

Table 5 shows the averaged-out temperatures of the evaporator section using 4 T-type thermocouples placed equidistantly on the evaporator section, which was equal to 79.25 ºC. In addition to one thermocouple that was used to calibrate the condenser section reading, which was equal to 33.40 ºC. When comparing them to the average temperature the IR camera gave for both sections, which resulted in readings of 78.5 ºC for the evaporator section and 35 ºC for the condenser section, the deviation is small. Thus, the readings of the IRC with an emissivity set to 0.95, at the reference temperature of 20 ºC, can be considered accurate.

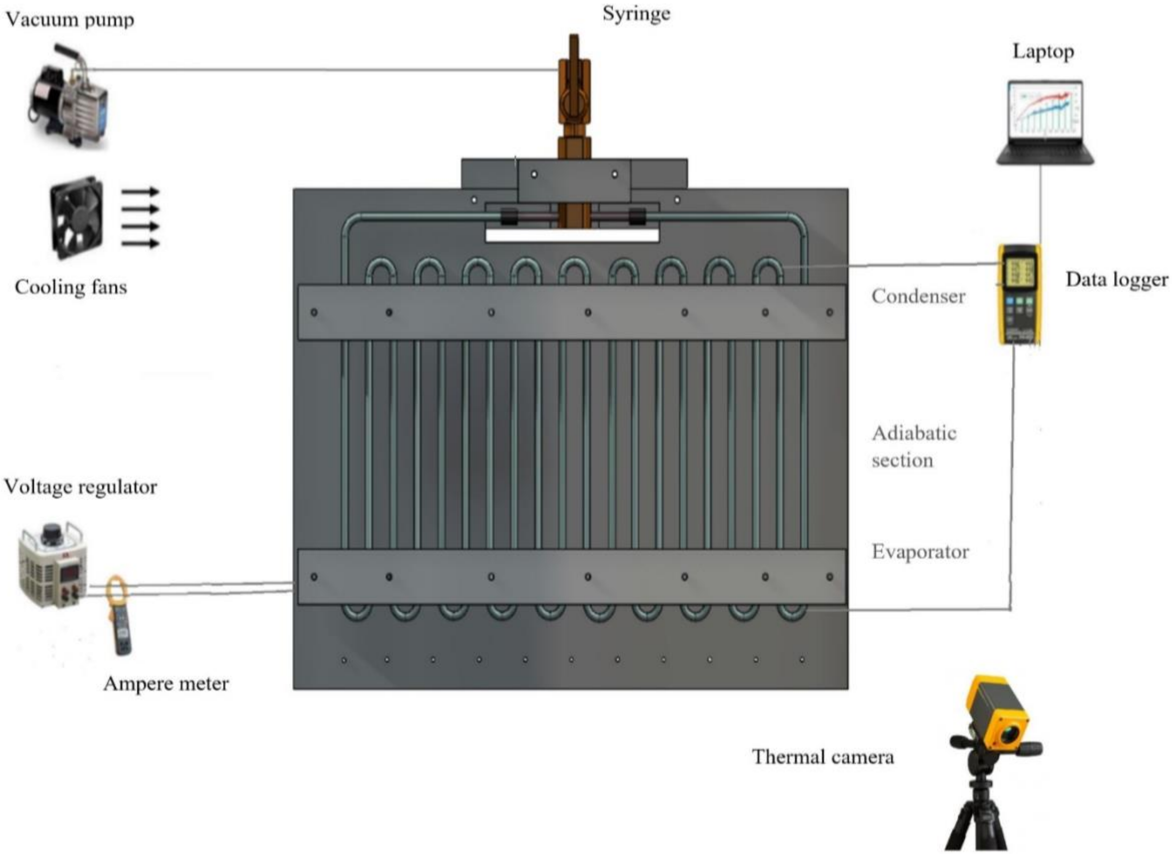


Figure 9 Schematic of setup at Ericsson laboratory.

As mentioned previously, the pipe walls are thermally thin since the Biot number < 0.1. Thus, uniform temperature distribution can be assumed, and the temperature measured at the inner surface of the tube can be considered equal to the temperature at the outer surface. The latter was measured using the IR thermal camera, calibrated by thermocouples readings.

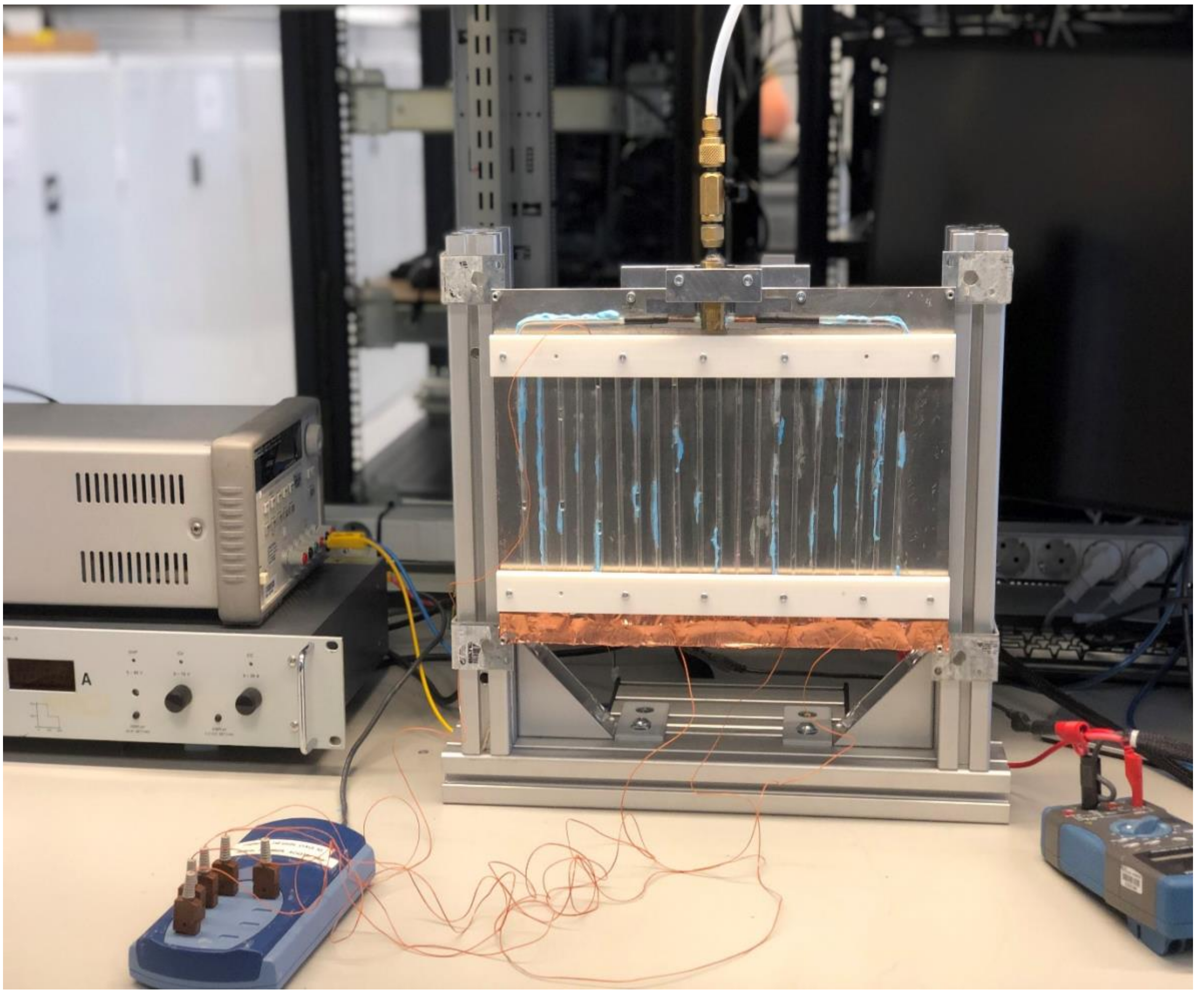


Figure 10 New pulsating heat pipe experimental setup at Ericsson laboratory.

6.2 Methods

6.2.1 Vacuum and filling method

One of the main problems faced during the filling and vacuuming part was leakage in the filling section. This led to the modification of the choice of an entirely glass manufactured filling section and was replaced with a metal valve, with an inlet fitting the pump pipe as seen in Figure 10. This gave the PHP structure more stability and filling or evacuating the PHP could be done without leakage. The evacuating pressure needed in order to remove 99% of the air in the PHP was of 1000 kPa. Once the heat pipe was evacuated, a syringe was used to fill the tubes with the working fluid using the same valve inlet. When evacuated, the fluid distributed itself in a capillary form along the entire heat pipe, alternating between liquid slugs and vapor plugs.

6.2.2 Infrared Thermal Camera (IRC)

The IRC is used to measure the average temperature of the evaporator and condenser sections since it gives additional insight into the spreading of heat. A defined area using a box was placed on the evaporator and condenser sections in the IRC, and the average temperatures of the readings were used to measure the thermal resistance of the heat pipe as seen in Figure 12. Thermocouples were used to calibrate the readings of the IRC in both the evaporator and condenser section as already mentioned. The emissivity of the IRC was set to 0.95.

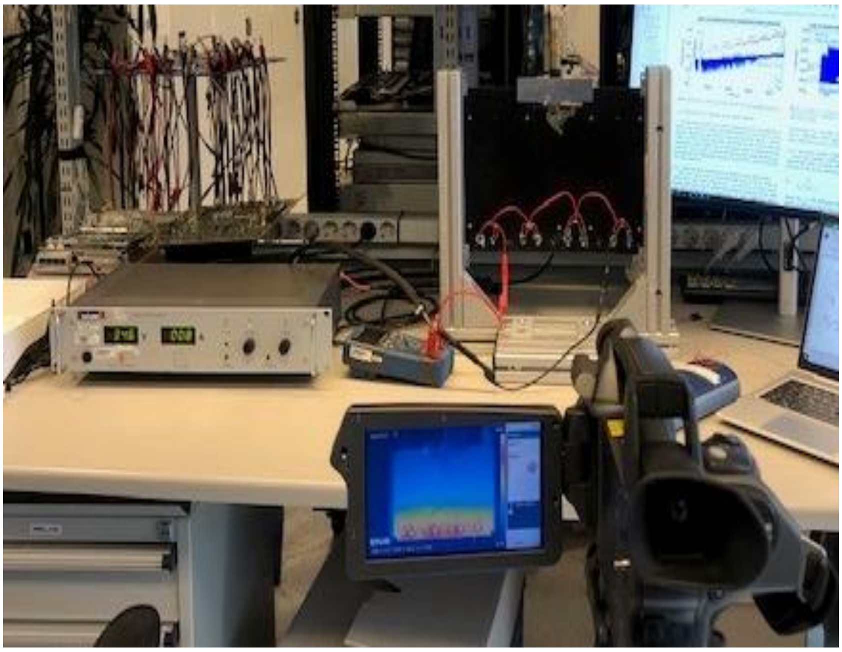


Figure 11 Thermal camera measurements at Ericsson laboratory.

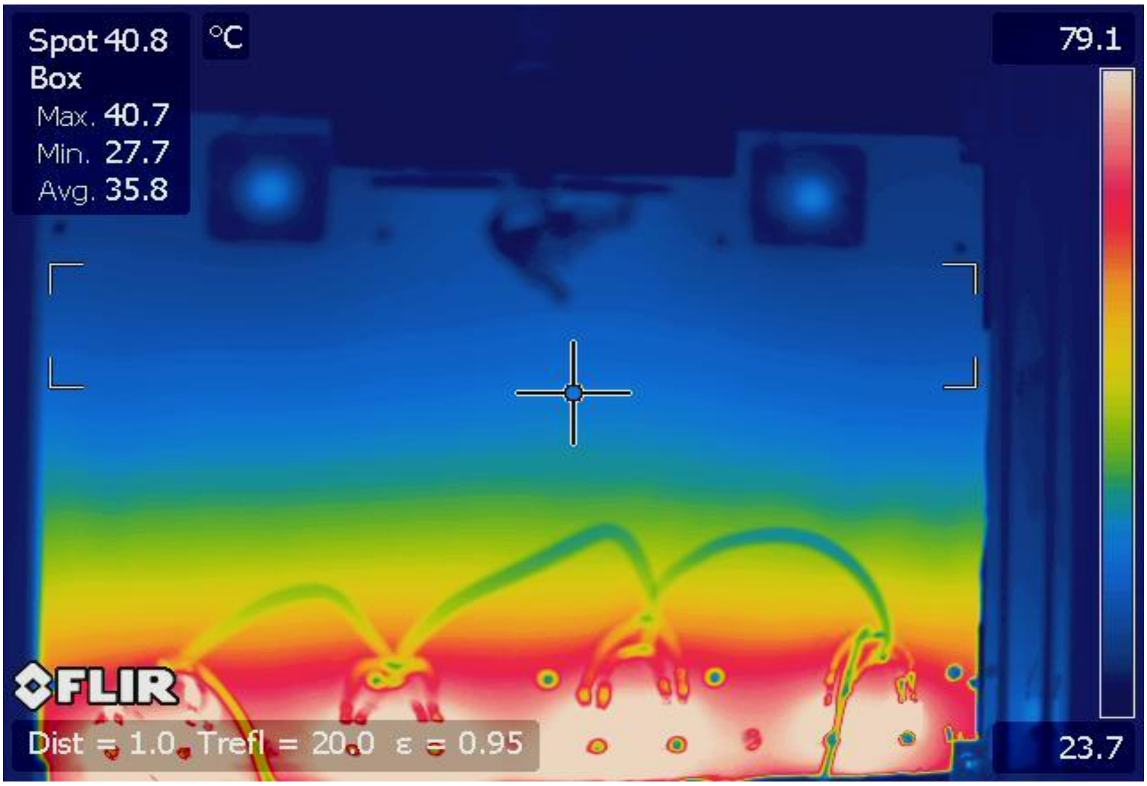


Figure 12 Thermal camera imaging for 90 W power supply at 70% FR.

6.2.3 Single Lens Reflex (SLR) camera

The SLR camera was used to visualize the bubble plugs and slugs of the fluid's pulsating motion. Capturing the flow provides an indispensable tool to study the complex movement of liquids and gases. The recognition of the bubble pattern, and several phenomenon that might occur like formation of thin liquid around the vapour, can be valuable input when mathematically modelling a pulsating heat pipe device [13].

Once the pulsating flow was captured, a MATLAB code, developed by Gaopan Kong and Shuo Yang, was used for image processing. Initially, the aim was to follow the procedure [42] conducted in image processing for a bubble in a microchannel. By extracting hundreds of images of the tube when filled with the liquid, which will act as a background for image processing. Then having raw images with the bubble inside the same tube, later removing the background from these images to separate the bubble. After that, the images are binarized, and the bubble inner area is filled to be able to distinguish it. However, for this study case, images obtained weren't clear enough. And the SLR camera failed to capture some instances of the very fast pulsating motion with a high resolution. Thus, results in section 7.8 are limited to presenting some extracted images form the pulsating motion, to be an indicator of the flow development, for qualitative analysis. Perhaps for future trials, a high-speed camera could be tested instead.

Chapter 7

Results and discussion

7.1 Flow pattern

The pattern of the two-phase flow in a pulsating heat pipe is variable and completely understanding it is complicated. As already mentioned, different factors affect the heat transfer, thus the flow nature in a PHP, such as the filling ratio of the working fluid, inclination of the PHP, power supply to the device and the evacuation pressure. However, there seems to be a general pattern of flow in a pulsating motion. The later can be classified into bubble, annular, plug and long plug flow. In between there is usually a liquid slug [43]. These patterns will be explained in detail in the upcoming sections.

7.1.1 Bubble flow

This type of flow mostly occurs due to nucleate boiling at the evaporator section, where the occurring bubbles have a diameter smaller than the tube width. They are usually created due to the non-condensable gases that escape the fluid when heated. As they are formed, they grow, leave the surface and rise, inducing a flow in the liquid [43]. They usually appear at low heating levels, where the main means of heat transfer is heat conduction. [44]. The bubbles pattern is difficult to maintain in a pulsating heat pipe and can easily change to another flow pattern when working conditions change.

- **Size**

Bubbles created in the PHP may have different sizes. Bubbles smaller than the tube diameter can often occur as the liquid pass the U turn in the evaporator section, since a small amount of liquid might remain entrapped. Bubbles equal to the diameter tube are also observed [13].

- **Contact angle**

The leading and lagging contact angles of the bubbles usually defer depending on the material of the tube and the working fluid. The overall frictional resistance of the flow can be affected greatly by this effect [13].

- **Merging patterns**

As mentioned previously, nucleate boiling produces small bubbles which rise in the adiabatic section of the PHP. The bubbles may collapse before reaching the condenser section or might reach it in a reduced size. Different bubbles with different sizes might merge, such as small bubbles merging with bigger ones. Similarly, bigger bubbles can shrink to a diameter smaller than the tube, then immediately float up due to buoyancy force. If two large bubbles encounter and merge, this might lead to subdivision of them into many smaller bubbles. Therefore, bubbles often have merging patterns, when moving through the PHP tubes [13].

7.1.2 Plug flow

When several bubbles which have a diameter equal to the channel width combine, plug flow occurs. The new formed bubble doesn't have a circular shape, but has a width equal to the channel size, and a length which is much bigger. When a plug flow passes through the condensation zone, it can be broken into several small bubbles. At low heating loads, plug flow can be present in a stable mode, the main means of heat transfer, in these conditions, is the heat conduction in the aluminium plate in addition to the vapour-liquid two phase flow in the channel [43].

7.1.3 Long plug flow

This pattern usually occurs at higher heating rates, when the bubbles generation becomes faster and faster, thus the distance between plugs is reduced as more plugs are formed, therefore the merging of plugs becomes more frequent [43].

7.1.4 Annular flow

This type of pattern occurs with the increase of the heating power further. Generally, after the fluid has completed a substantial circulating flow, the long plugs are converted to the annular flow. When nucleate boiling becomes more intense, the liquid film between the vapour plug and liquid will be torn, the annular flow is formed inside the channel at this stage. However, it's worth mentioning that at very low filling ratios, when increasing the power, annular flow is harder to form since the dry out phenomenon tends to occur [43].

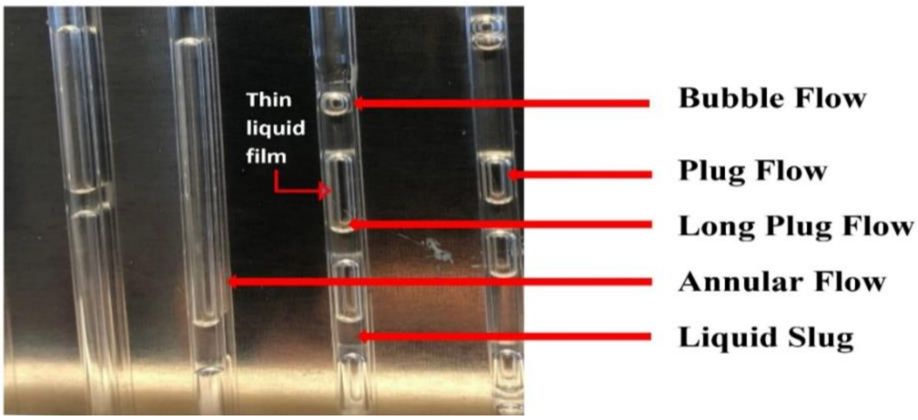


Figure 13 Visualisation of flow patterns of capillary flow inside the PHP.

7.2 Filling ratio

For very low FR such as 20%, the pipe doesn't act as a pulsating heat pipe. With increasing power, the thermal resistance initially decreases. However, if the filling ratio is too low, dry-out conditions will eventually occur, due to vapor space covering a large amount of the surface when increasing the power beyond a specific point, increasing the thermal resistance again [28]. This was validated experimentally when the PHP was filled with a small amount of DI water at 20% FR. Figure 14 below shows the thermal resistance in function of different powers supplied that was measured for the PHP.

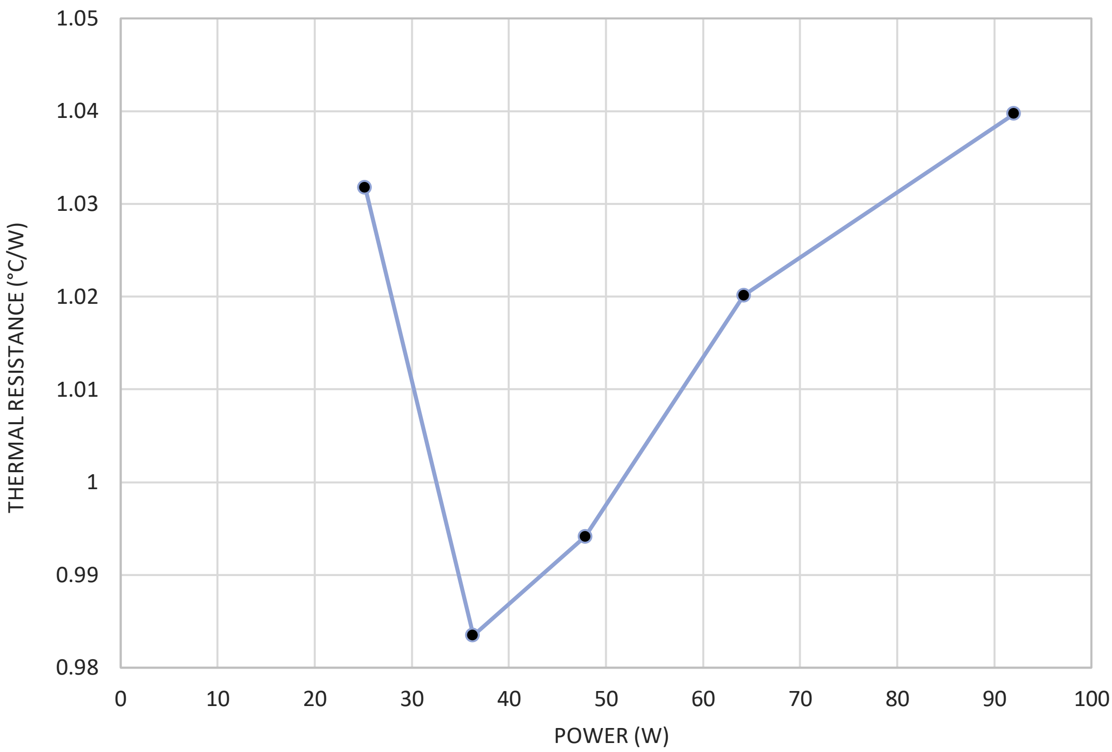


Figure 14 R_{th} of the PHP for 20% FR with distilled water as a working fluid.

As noted above, the thermal resistance initially decreases when increasing power from 25.2 W to 36.3 W. However, with further increase in power, the thermal resistance started increasing. With higher power supplies, at 92.2 W the thermal resistance increased further. The results align with [28] study which found that for low filling ratios, when the heat pipe doesn't act in a pulsating mode, the thermal resistance initially decreases slightly, but for power supplies between 30-40 W in the case of water, the thermal resistance of the pulsating heat pipe tends to increase. This is because of dry out conditions, where the surface is covered by vapour space as mentioned previously.

With increasing filling ratios, the device starts acting in a pulsating mode. The filling factor was further increased to 40%, 60% and 70% for both DI water and isopropanol. Figure 15 shows the thermal resistance for the different filling factors with DI water as working fluid.

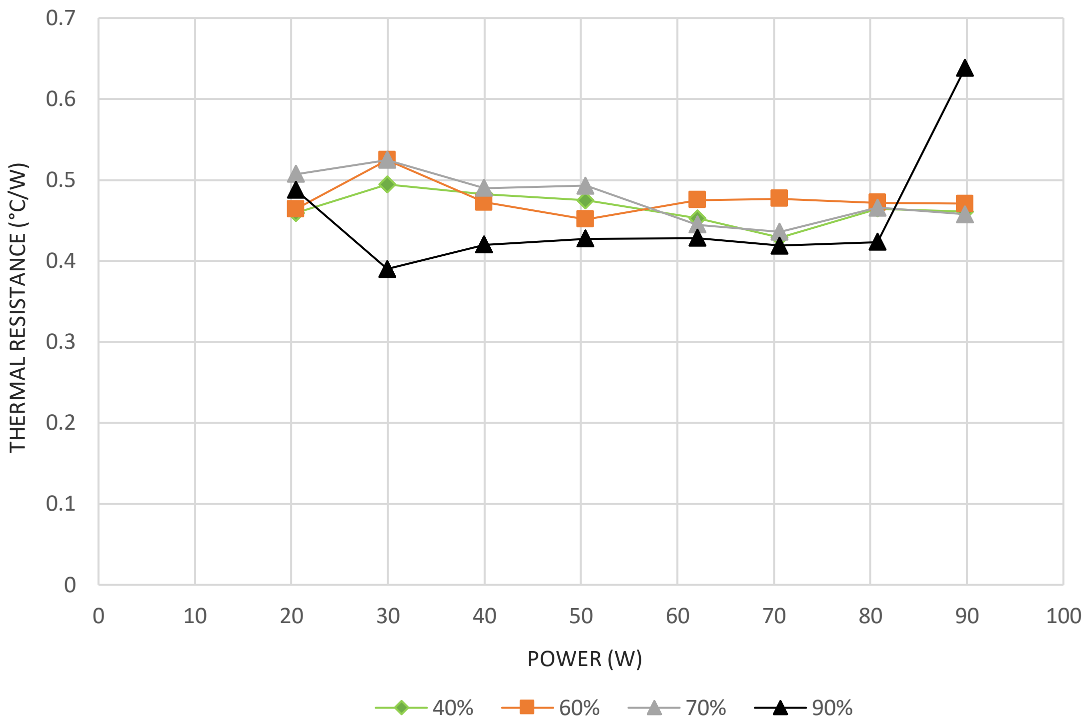


Figure 15 R_{th} of the PHP for different FR with distilled water as a working fluid.

As noted in Figure 15, the thermal resistance was similar in between 40-70% FR, which suggests the optimal filling ratio of the PHP is in between that range. When increasing the filling ratio to 90%, the thermal resistance seemed to increase with increasing power. According to [28], at too high filling ratios such as 90%, very few bubbles are present thus the performance of a PHP is subject to a remarkable drop. When filling the heat pipe further, almost no pulsation occurred. In these conditions the heat transfer occurs only by convection, since no bubbles are present.

When comparing the optimal operating range of the PHP using DI water as a working fluid, it seemed to perform slightly better with a lower filling ratio of 40%. According to [43], at lower filling factors, the two phase flow turns into a directional circulation flow, making the annular flow pattern occur more, enhancing in this way the phase change heat transfer, thus reducing the thermal resistance and improving the efficiency of the PHP. For higher filling ratios, such as 90%, more liquid plugs are present, making it more difficult for the vapour bubbles to break into these plugs. Thus, the two-phase flow heat transfer is reduced. This increases the thermal resistance and lowers the efficiency of the device.

The working fluid was then switched to isopropanol, and the PHP was tested for different filling ratios of 30%, 40%, 60% and 70%. The same procedure of measuring the R_{th} at different power supplies was repeated.

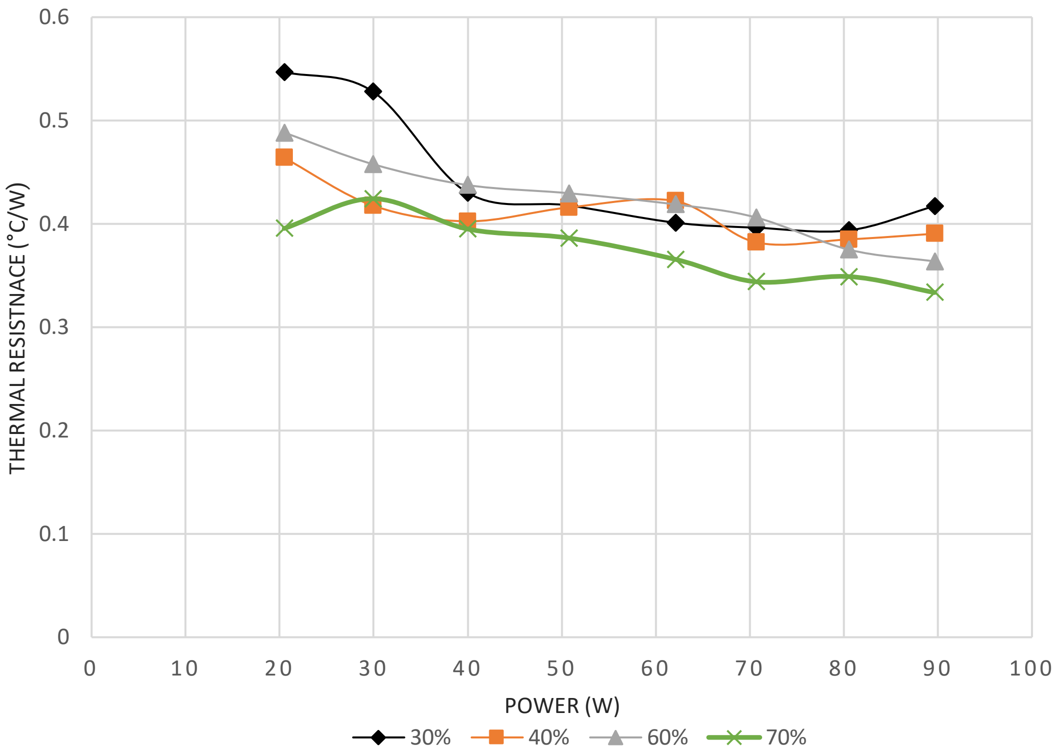


Figure 16 R_{th} of the PHP for different FR with isopropanol as a working fluid.

When using isopropanol as a working fluid, pulsation occurred at lower filling ratios, such as 30% and at lower power supplies of 30 W, since it has lower boiling temperatures and surface tensions when compared to DI water. As seen in Figure 16, with increasing filling ratios the thermal resistance decreased as the pulsation rate increased. For isopropanol, the lowest thermal resistance was achieved at 70% FR and 90 W of power supply, where the R_{th} was equal to 0.33℃/W. When comparing it to the R_{th} of 0.42℃/W which was achieved for 30% FR at the same power supply, this shows that the filling ratio has a notable impact on the performance of the PHP.

Finally, it’s important to note that measured results with isopropanol as a working fluid might give a better indication of the PHP performance with different filling ratios, mainly due to stable pulsation obtained with the fluid, whereas DI water pulsation wasn’t as uniform during the experiments.

7.3 Inclination angle

To test the effect the inclination angle had on the pulsating heat pipe performance, after finding the optimal operating conditions of the PHP placed vertically, with isopropanol as a working fluid, the PHP was then rotated 90˚, to see the effect gravity would have on the R_{th} of the device.

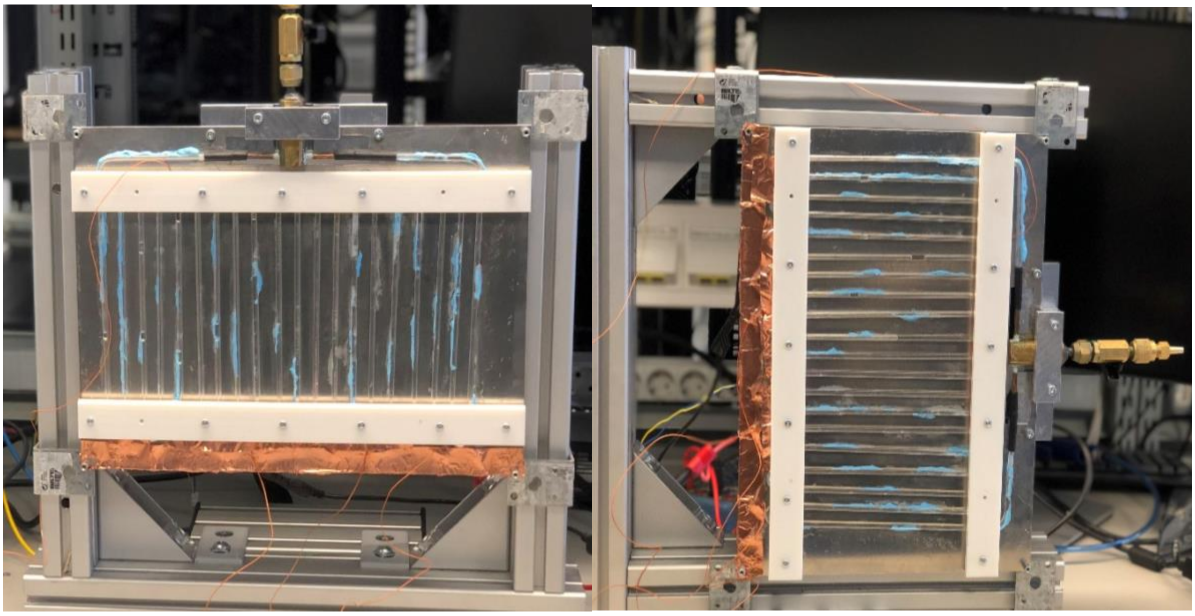


Figure 17 PHP in horizontal (right) and vertical (left) position.

When rotating the PHP with the channels in horizontal position in respect to the surface, most of the fluid settled in the tubes at the bottom, whereas the tubes at the upper side were left with little to no fluid. When supplying power, some pulsation occurred in the heat pipe, however it was very unstable and did not cover the entire PHP area. When testing lower filling ratios such as 40%, the start-up of the device wasn't possible. Thus, the heat pipe failed to act in a pulsating mode with this configuration under certain conditions.

Results align with several studies that tested different inclinations of the PHP. [19] concluded trough her experiments that gravity does play a crucial role in stabilising the pulsation of the PHP. When the fluid is not assisted by gravity, such as in horizontal configuration, oscillations of the fluid are less frequent, and the heat transfer is reduced, even in a heat pipe with a relatively high number of channels, such as the tested PHP. The main difference in both configurations is that when the gravity is perpendicular to the flow path, no direct effect is seen due to increased acceleration, as in horizontal position. While when placed in vertical position, the acceleration vector will be parallel to the fluid flow path, enhancing heat transfer and making the pulsating mode occur more frequently. For the designed PHP, stable pulsation was only obtained when using the vertical inclination. Thus, the current demonstrator, failed to be independent of gravity with the selected geometry and working fluids. [13] suggests that the effect of gravity can potentially be reduced using smaller tube diameters for the PHP channels.

7.4 Power supply

Generally, with increased power supplies the thermal resistance of the pulsating heat pipe tends to decrease, and pulsation tends to occur more often and more steadily, if there is a sufficient amount of working fluid, which avoids dry out conditions. This is because at higher power supplies, higher temperatures in the evaporator section are present, thus boiling and evaporation happens at a faster rate. When using distilled water, for power supplies below 40 W, no pulsation occurred but rather some motion in the working fluid. For the different filling ratios, the pulsation started occurring at around 40 W and above. While when using isopropanol, stable pulsation occurred at lower power supplies, such as 30 W. Nature of pulsation seemed to also be affected by the power supply level, as pulsation bursts were more frequent at lower power supplies. And at the highest tested power supply of 90 W, the pulsation was mostly steady, with little pressure bursts that occurred rarely.

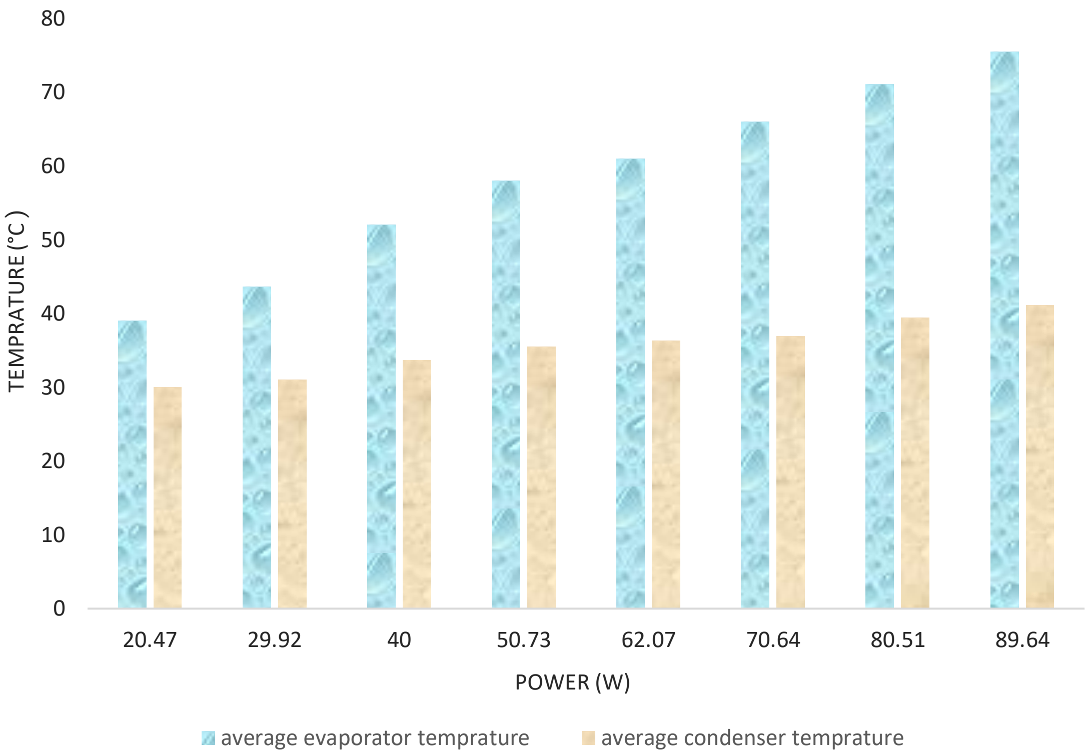


Figure 18 Power supply effect on average temperature of evaporator and condenser sections.

As seen in Figure 18, increasing the power supply increases the evaporator temperature steadily, while the condenser section remains at almost constant temperatures when cooling. This way, the evaporation rate and the pulsating motion is increased. This enhances the heat transfer of the PHP. And even though the temperature difference between both sections is increased, the thermal resistance of the device is still decreased as seen in Figure 19.

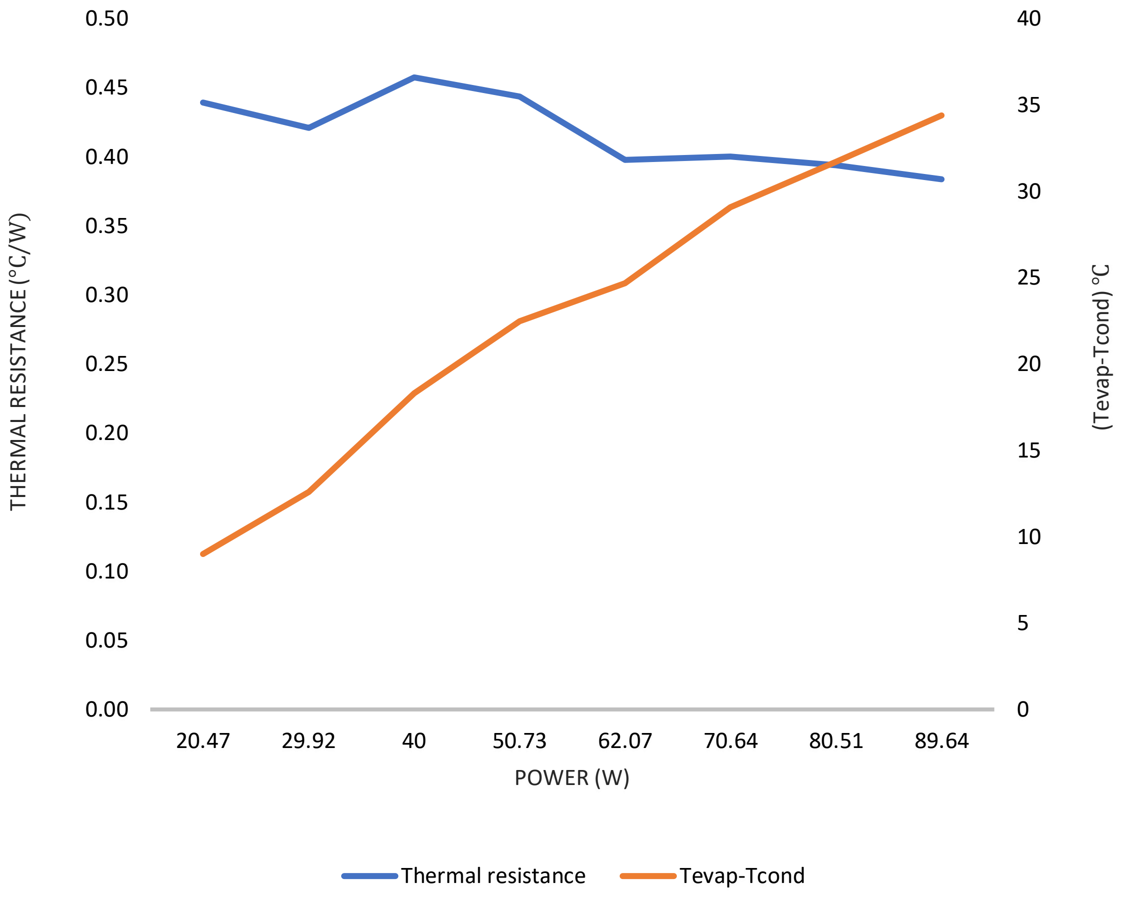


Figure 19 R_{th} and average temperature difference between the condenser and evaporator sections of the pulsating heat pipe at 60 % FR of isopropanol for different power supplies.

7.5 Cooling effect

Following the procedure most studies conducted, as already mentioned, cooling fans were attached to the condenser section to increase the condensation rate. This will be referred to as cooling effect in the following sections. When using the cooling fans with distilled water as a working fluid, the thermal resistance decreased slightly. But the impact wasn't that noticeable. The fact that the experimental setup had an aluminium plate as a conducting base meant that cooling the condenser section would also lower the evaporator section temperatures and thus potentially decrease the evaporation rate. If the condenser section had a different setup, such as a water bath as a cooling method, lowering the temperature of condensation without impacting the evaporator temperature would have probably given different results and the effect of cooling the condenser section might have been more evident. With this setup, cooling seemed to decrease slightly the thermal resistance of isopropanol too, and pulsation occurred more steadily.

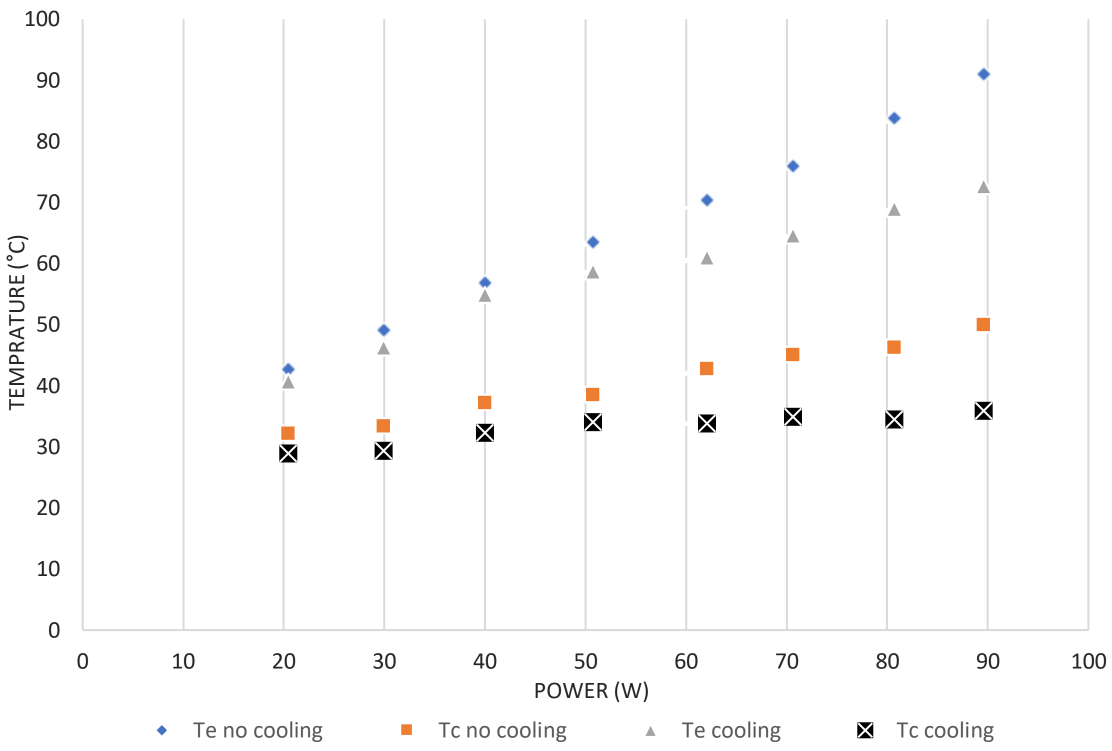


Figure 20 Cooling effect on the condenser and evaporator sections average temperatures for 70% FR of DI water.

As seen in Figure 20, when cooling, the condenser temperature was maintained at around (30--35)˚C, while without cooling, the condenser temperature reached 50 ˚C at higher power supplies. When attaching fans, the evaporator temperature was also subject to temperature decrease, however, less evident than the condenser section. The heat transfer was slightly enhanced with cooling fans and the R_{th} of the PHP decreased slightly.

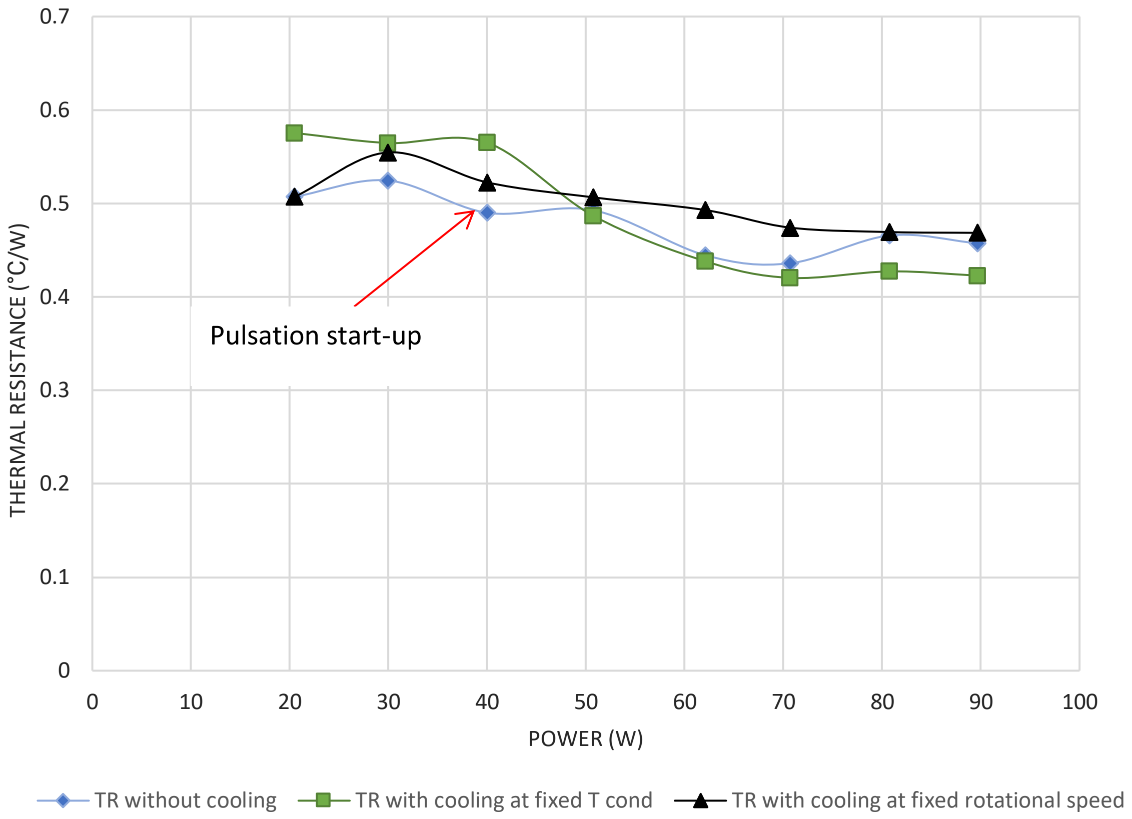


Figure 21 R_{th} of the PHP with 70% FR of DI water using different cooling methods.

As noted in Figure 21, cooling the condenser section at an almost fixed temperature decreased the thermal resistance of the PHP when it started pulsating, in the case of distilled water. The PHP performed better when fixing the condenser at a constant temperature than in the case of cooling with a fixed rotational speed of the fan, for the different power supplies. The decrease of thermal resistance was most notable when the heat pipe started pulsating at a bigger rate, which indicates that cooling the condenser section slightly enhanced the pulsating motion, thus the heat transfer.

Figure 22 shows the decrease in thermal resistance of the PHP when attaching fans and maintaining the condenser temperature almost constant, using isopropanol as a working fluid. As noted, the effect cooling had on isopropanol wasn't as notable as cooling effect on water pulsation. However, the decrease in thermal resistance was more linear than in the case of no cooling, which suggests heat transfer was still enhanced.

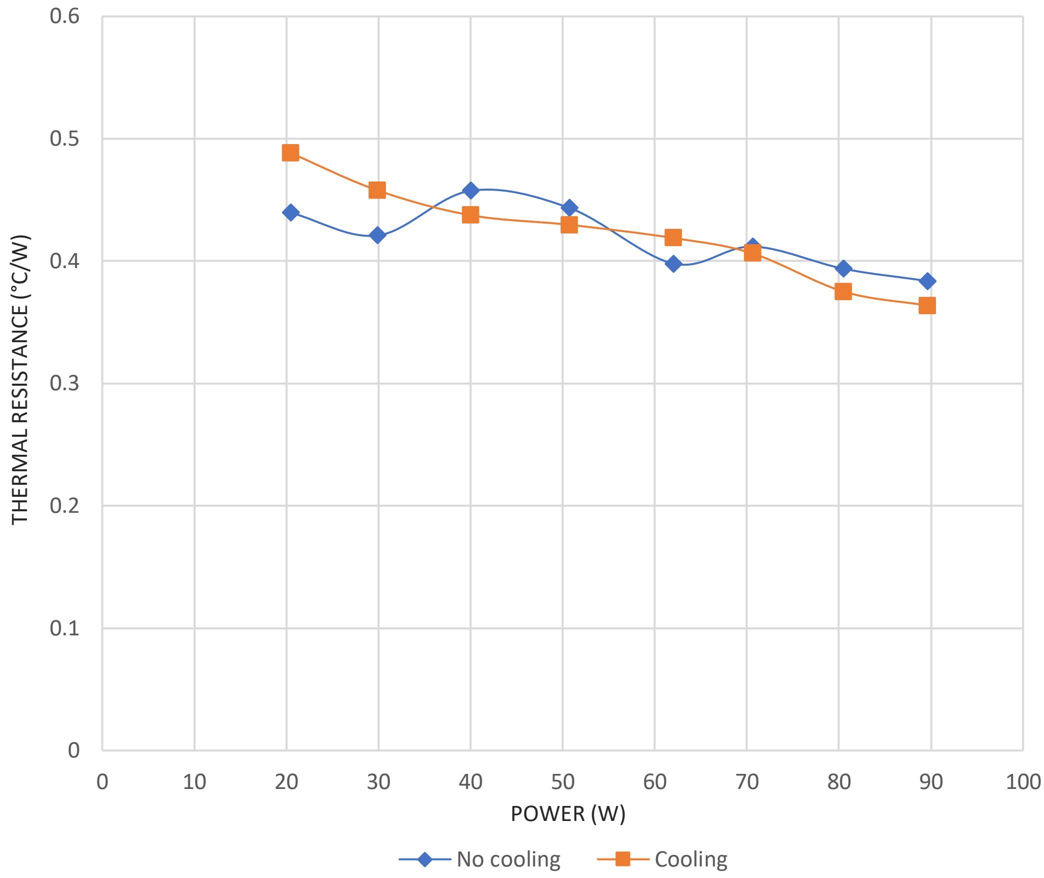


Figure 22 R_{th} variation with and without cooling for 60% FR of isopropanol.

Another aspect that is worth mentioning, even though attaching fans to the condenser section didn't decrease the thermal resistance at a big rate, when fixing the condenser section at a certain temperature, condensation of the fluid is guaranteed. For higher power supplies such as 90 W, without fans, the condenser temperatures reached 50 °C. Therefore, the pulsating motion that occurred could have possibly been due to pressure difference between the bottom and top of the PHP. As condensation of the fluid might not happen at such high temperatures for evacuated pressure levels. Thus, maintaining a constant condenser temperature is important to guarantee the heat pipe is evaporating and condensing simultaneously, as a PHP should have both sections, being of main importance for the device functioning.

7.6 Evacuation pressure

Initially it was unclear what effect the evacuation pressure had on the pulsation of the heat pipe. When filling the heat pipe with water, at high power supplies such as 90 W, the average temperature in the evaporator section reached 100 °C in some points, thus boiling was expected to happen, therefore a pulsating motion. However, when not evacuated, the start-up of the pulsation of the heat pipe didn’t occur even at higher heat loads. This led to completely evacuating the heat pipe and then supplying power, since that would lower the boiling temperature, hence facilitate the start-up of the PHP. When evacuated, for power supplies above 40 W a pulsating motion started occurring. And with higher heat loads more pulsating motion occurred. However, pressure bursts occurred randomly when using distilled water as a working fluid, and stable pulsation was not present. Axial fans were then attached to the condensing section to enhance the cooling, and more steady pulsation occurred when lowering the condenser section temperatures to a fixed range between (30-35) °C. Maintaining a stable pulsation for long periods was still challenging when using water, and some random pressure outbursts still happened at different heat loads. Another effect that occurred when using water as a working fluid, was a noticeable sound effect that accompanied the pressure bursts, of which the intensity of sound increased with higher power supplies. [45] observed the same sound, suggesting that this might be because when boiling water, the volume of the vapour bubbles pulsate, thus a compression and decompression of the surrounding liquid could occur, and it’s very likely to be the cause of this characteristic sound observed. The fluid was then switched to isopropanol which has a lower boiling point. However, evacuation was still needed to maintain a steady pulsation for most power supplies and the pulsating sound wasn’t observed for the second tested fluid.

When evacuated, isopropanol produced a stable pulsation at very low power supplies. 30% FR with 30 W of power supply was enough to get a steady pulsation. However, dry-out happened faster, thus a bigger filling ratio was necessary for testing the performance for prolonged times.

This validated the fact that evacuating the PHP has a big impact on the pulsation of it. Having the evaporator section reach boiling temperatures in some points might not be sufficient to start up the heat pipe at ambient pressure conditions, specifically in the case of DI water. As mentioned previously, the presence of non-condensable gases in the pipe, when not evacuated, might be a main reason for deteriorating the heat transfer in it.

7.7 Fluid selection

As already mentioned, DI water produced unsteady pulsation and higher power supplies were required to start up the PHP, in addition to the evacuation of the heat pipe that was needed for the pulsating motion to occur. Pressure bursts were also frequent alongside a characteristic sound that only accompanied water pulsation.

After completing DI water measurements, a different fluid was tested to see how the efficiency of the PHP and the nature of the flow would be affected. Isopropanol was used as a working fluid and the same steps were repeated. The physical properties, such as surface tensions and density are known, and the Bond number could be calculated and resulted for the dimensioned heat pipe < 4. Therefore, capillary flow should be present when using it as a working fluid for the PHP. With the second tested fluid, some pulsation occurred even at low filling ratios such as 15%, however dry out occurred very fast. At filling ratios of 30%, steady pulsation occurred and when comparing 40% filling ratio of isopropanol and DI water, results showed that the alcohol outperformed the water as a working fluid, with a decrease in thermal resistance of the PHP and almost steady pulsation at different power supplies. Sudden pressure bursts were less frequent than when using distilled water. However, dry out phenomenon occurred faster with the alcohol, thus maintaining a constant pulsation for prolonged periods required a larger amount of filling ratios when compared to DI water.

This validated the fact that the thermophysical properties of the working fluid have a big impact on the PHP performance. Water has a higher surface tension than isopropanol, allowing it to operate in a pulsating mode with bigger diameters of the PHP. However, this also indicates bigger pressure drops in the tube, thus sudden bursts with water can be more frequent. Bigger power supplies are also needed when using water to maintain a pulsating motion, when compared to another fluid with lower surface tensions [29], such as isopropanol. This also explains why the PHP was able to operate in a more stable mode when using the second fluid, even at lower power supplies. A comparison between the R_{th} of the PHP using distilled water and isopropanol is presented for three different filling ratios: 40%, 60% and 70%. These filling ratios were chosen being in the range of which the PHP performed best for both fluids.

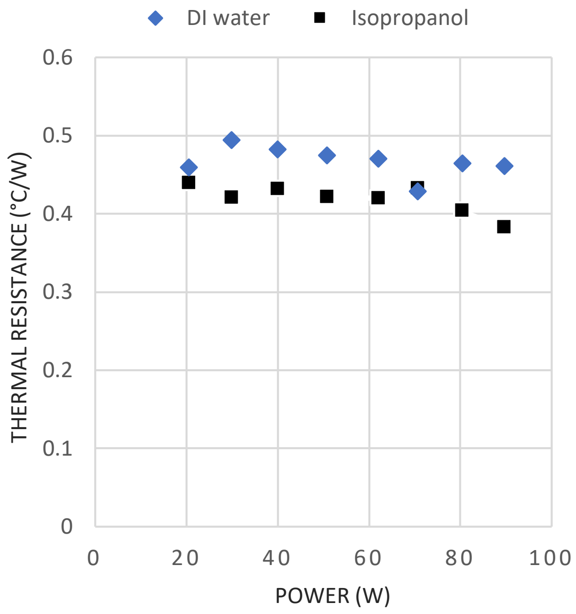


Figure 23 DI water and isopropanol PHP R_{th} at 40% FR.

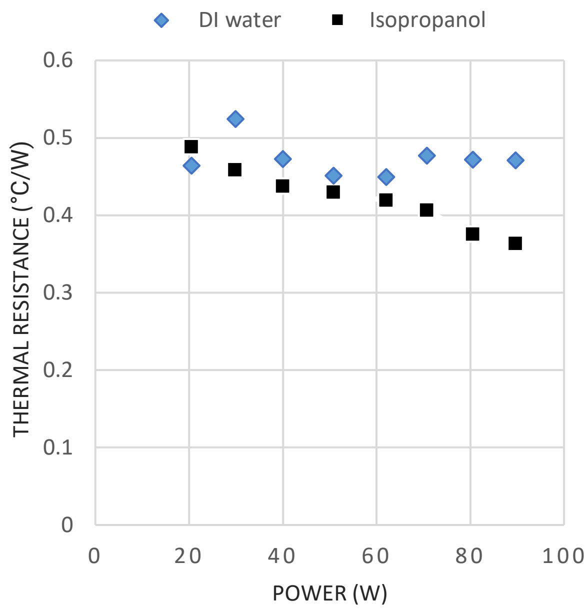


Figure 24 DI water and isopropanol PHP R_{th} at 60% FR.

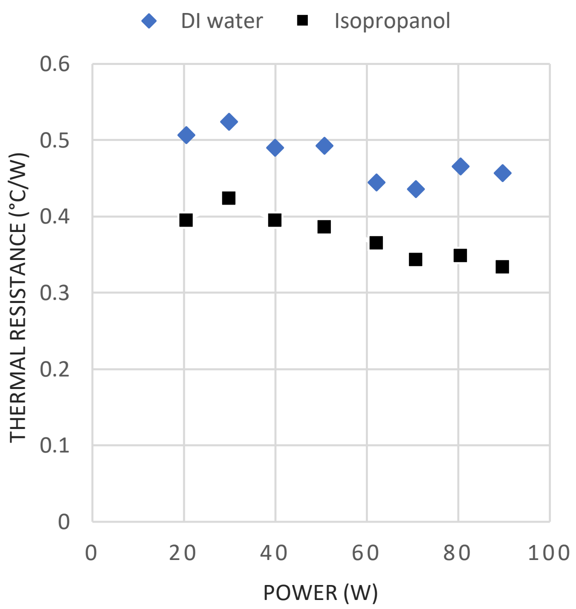


Figure 25 DI water and isopropanol PHP R_{th} at 70% FR.

Figures 23, 24 and 25 show that isopropanol outperformed DI water for different FRs at different power supplies. With increased power supplies, the R_{th} decreased almost linearly in the case of isopropanol due to steady pulsation, which was not observed in the case of DI water.

The best performance for isopropanol was at 70% FR with a R_{th} resistance of 0.33 °C/W, while water at the same power supply and FR had a R_{th} of 0.46 °C/W.

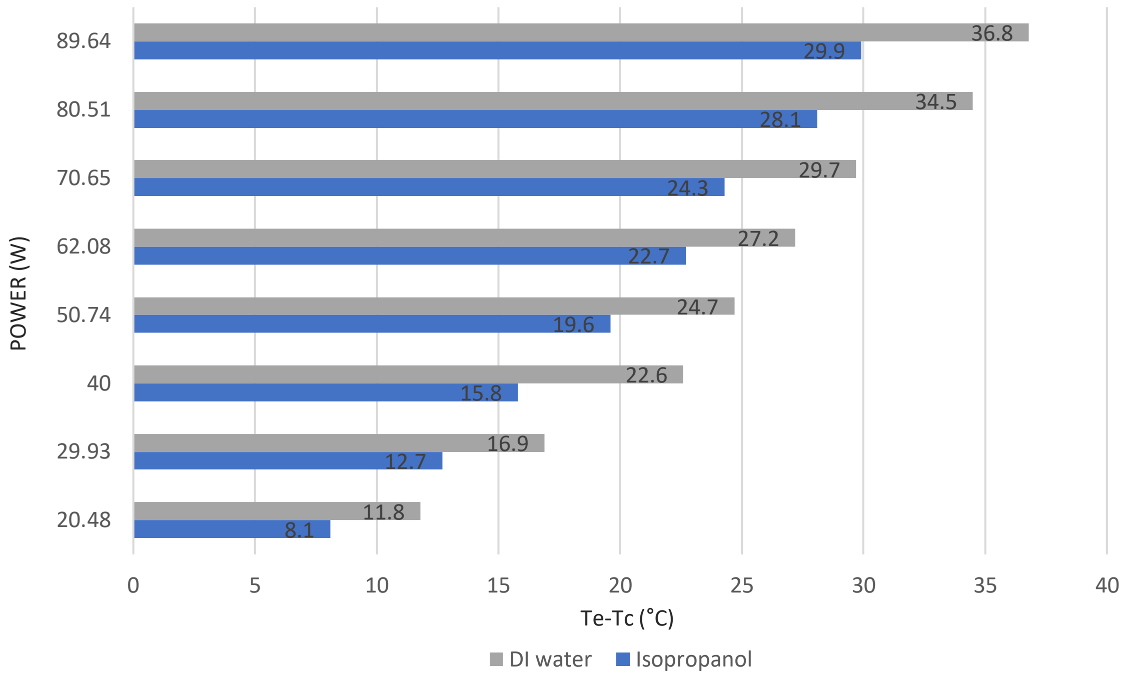


Figure 26 Condenser and evaporator average temperature difference for different power supplies at 70% FR of both working fluids.

As seen in Figure 26, the difference in temperature, and therefore pressure between the evaporating and condensing section when using DI water is bigger for all power supplies when compared to isopropanol. Since the thermal resistance is inversely proportional to the temperature difference, lower temperature difference improves the PHP performance, such as in the case of isopropanol.

Finally, during the initial design stage, ethanol was thought to be tested as a working fluid in the PHP, since the Bond number would be appropriate for the chosen dimensions. Instead of pure ethanol, a mixture which was available, that contained a high percentage of ethanol blended with several other alcohols was used. An interesting aspect that was observed is that this mixture of fluids did not distribute itself in a capillary form all over the heat pipe, despite containing a large amount of ethanol. Bubble plugs and liquid slugs were not formed in a uniform way all over the pipe. This might be because of inappropriate thermophysical properties of the mixture. Being a blend of different alcohols at unknown ratios, the Bond number was not known for this specific mixture, therefore a capillary flow was not guaranteed. This also emphasises the importance of picking the correct working fluid for a PHP. Alcohol blends might have lower boiling points, thus evaporation might occur faster. However, other thermophysical properties such as viscosity and surface tension might affect the performance of the PHP, making the distribution of the fluid more difficult, as more liquid can be present in certain areas and a steady pulsating motion can be harder to obtain. This aligns with what [4] states, that fluids which are pure and have a homogeneous chemical composition that is invariable, function better as a working fluid in heat pipes.

7.8 SLRC Results

In this section, some extracted images of the PHP adiabatic section under operation are presented.

Figure 27 Visualization of the chaotic flow and different flow patterns in the adiabatic section of the PHP when in operation.

Figure 27 displays the different flow patterns which can be seen in a PHP when in operation, which are bubble, plug, long plug and annular flow. As noted, all the flow patterns can be present at the same time.

Figure 28 shows the development of different flow patterns in a single channel of the PHP.

Figure 28 Flow pattern development in a single channel of the PHP.

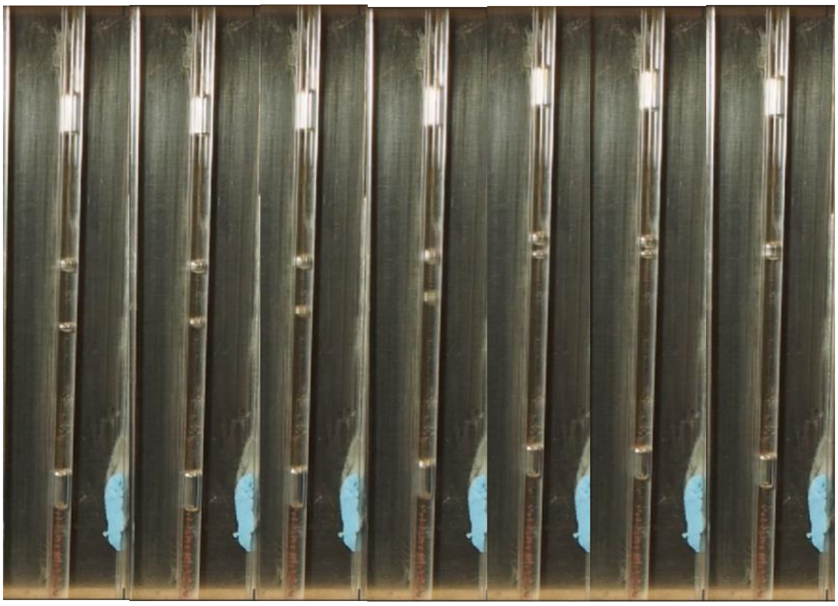
In ***1** and ***2** two bubbles merge into the plug flow.

In ***3** a new bubble is generated and travels through the liquid slug in ***4**, ***5** and ***6**. As noted, the bubble exhibits a reduction in size during this process.

In ***7** the bubble disappears, it might have merged with the plug flow, however the speed was too high to capture with the SLRC.

While ***8**, which was extracted at a later time frame, shows the flow heading towards the condenser section and an annular flow can be seen at the bottom of the tube, while no more bubbles are generated.

Figure 29 shows two bubbles which have different sizes, traveling through a liquid slug **(*1-*6**) and merging together into one bigger bubble in ***7**.



***1 *2 *3 *4 *5 *6 *7**

Figure 29 Merging of two different size bubble into a bigger one in a single channel of the PHP.

In Figure 30 the pulsating motion of a vapour plug in one channel of the PHP can be seen.

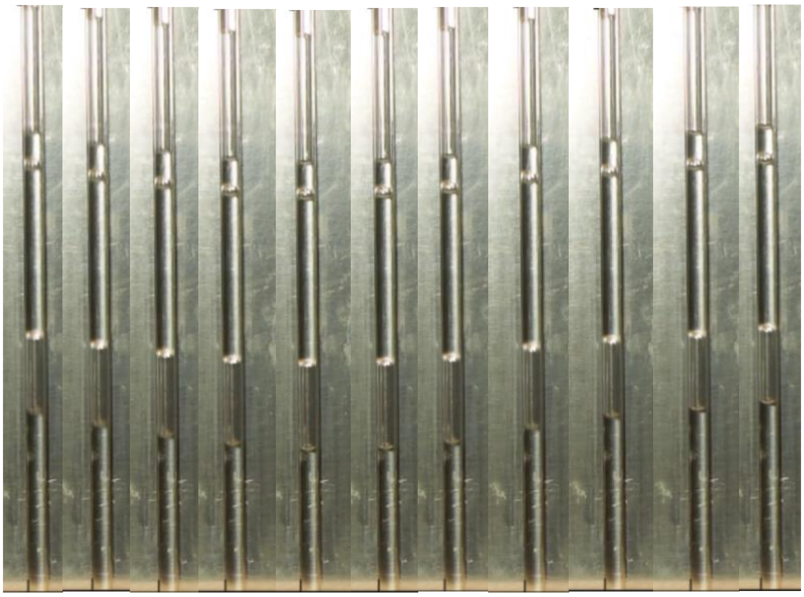
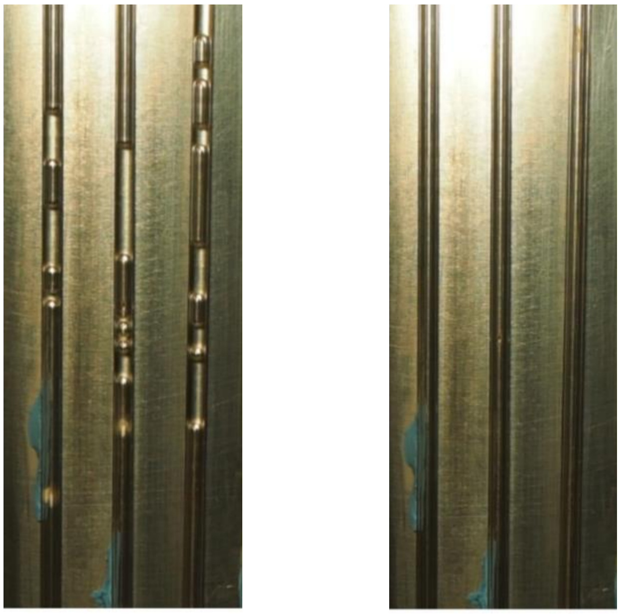


Figure 30 Oscillating movement of a vapour plug in one PHP channel at different time frames.

A **B**

Figure 31 Condensation process in two determined tubes of the PHP.

Figure 31 shows the vapour bubbles and plugs **(A)** condense into liquid slugs **(B)**, in two determined channels of the PHP.



C **D**

Figure 32 Evaporation process in three determined tubes of the PHP.

Figure 32 shows the liquid slugs in **(C)** evaporate, leaving the vapour cover the tubes entirely in **(D)** for three determined channels in a PHP.

Chapter 8

Conclusions and future work

The purpose of this work was to develop a demonstrator pulsating heat pipe and investigate its performance. After reviewing literature that conducted similar studies, the PHP was designed. Then the planning of experimental setup initiated. From the selection of the power and cooling supply methods based on desired range of operating temperatures, to the appropriate working fluids suitable to act in a pulsating mode.

Once the demonstrator was manufactured and the experimental setup was ready, a different set of parameters were varied to see the impact that had on the performance of the PHP. These parameters included the supplied power to the device, the filling ratio of the working fluid, the inclination angle of the PHP and the working fluid itself.

Results showed that with increased power supply, generally the PHP heat transfer performance improved. Pulsating motion occurred at a more frequent rate and the thermal resistance of the device decreased. An important aspect that was concluded is that an adequate amount of fluid has to be present at high power supplies, or else dry out conditions of the working fluid can occur, resulting in a decrease of the pulsating heat pipe performance.

When it comes to the filling ratio, for both tested fluids DI water and isopropanol, the device seemed to best operate at a filling factor ranging in between 40-70%. An interesting aspect that was observed is that DI water performed slightly better at the lower end of this range, while isopropanol seemed to perform better at the higher end of this range. This might be due to the fact that isopropanol reaches dry out conditions faster when compared to DI water, making a bigger amount of fluid more suitable in the case of steady pulsation for prolonged times.

As for the fluid selection, isopropanol performed better than DI water, for all tested filling ratios and power supplies. The best performance was obtained for 70% FR of isopropanol with TR =0.333 $\frac{°C}{W}$. This is mainly due to the surface tension of isopropanol being lower than that of DI water, which enhances heat transfer, as the increase of surface tension also means an increase in the capillary resistance of the fluid, sequentially reducing heat transfer. The pulsation of water was also very unstable when compared to isopropanol, and sudden pressure bursts occurred more often, accompanied by a characteristic sound that only occurred when water pulsated. Trials showed that other thermophysical properties of the fluid are also important to investigate, this was validated when the PHP was filled with a mixture of low surface tension alcohols yet failed to produce a capillary flow and start-up of the device wasn't possible. Which indicates the importance of choosing an appropriate fluid when using it in a PHP, as purity of the working fluid seems to also have a significant impact on the device performance.

When it comes to the inclination angle of the pulsating heat pipe device, both horizontal orientation and vertical orientation were tested. Results showed that for the designed PHP, gravity still played a crucial role in performance, as it failed to be a gravity independent device. When the device was assisted by gravity force, thus in vertical orientation, pulsation occurred in a stable way and start-up of the device was possible for different filling ratios and power supplies. However, when placing the device in a horizontal configuration, thus no gravity conditions, the start-up of the pulsating motion failed at low filling ratios and when the pulsating motion occurred for higher filling ratios, it was not uniform and not distributed equally along the entire heat pipe. Therefore, for this specific designed PHP, in order to obtain a fully operating device in a pulsating mode, vertical (gravity assisted) inclination is preferred over horizontal inclination.

Another important conclusion is that vacuuming the heat pipe before filling the PHP with a working fluid facilitates the start-up of the device. In the case of DI water, it was necessary for pulsation, as it failed to start-up at atmospheric conditions, even when the evaporator temperatures reached boiling temperatures in some points. This is probably due to non-condensable gases, which can deteriorate heat transfer and specifically the condensation rate in these conditions.

The effect cooling the condenser section had on the pulsating motion was also investigated, results showed that when maintaining the condenser section at a steady temperature range, the heat transfer was slightly enhanced, and the stable pulsation was easier to maintain.

Last step was using a single lens reflex camera, to capture the flow patterns in the adiabatic section when pulsating, decomposing the motion into a series of images using MATLAB, to be indicative of the flow development.

Concluding, pulsating heat pipes are promising devices for cooling applications. They offer a wide range of advantages when compared to conventional heat pipes, such as the ability to handle high heat fluxes, flexibility of design and low costs due to simple geometry. Some aspects of the operation and design of PHPs are widely agreed upon. However, a lot is still unclear when it comes to the optimal design and operating conditions of these passive two-phase flow devices. Nonetheless, the behaviour and the chaotic flow nature they exhibit, make them an intriguing topic to further research and investigate, both on an experimental and theoretical side.

8.1 Future work

Having designed and tested the PHP as a demonstrator for the two-phase flow passive heat transfer devices, there is room for improvement. Firstly, since the heat pipe was entirely manufactured in glass, a perfect thermal contact with the plate was very hard to maintain, with the PHP not being completely flat. For future experiments, the evaporator and condenser section U turns could be manufactured in copper, this would give a better thermal contact and a more even heat distribution, thus enhance heat transfer in the PHP, while still being able to maintain the adiabatic section in glass to visualise the flow.

Heating wires could also be directly attached to the U-turns of the evaporator, and a different cooling method could be tested out such as a water tank, in the condenser section, to see how that would affect the performance of the device.

In addition to that, attaching a pressure transducer would bring an additional value to the results, since that would facilitate knowing at what temperatures the fluid is evaporating. Doing that, the effect the pressure has on the PHP performance could be investigated numerically, which could bring further insight on the nature of the pulsating motion. As one of the main uncertainties of the current study was whether the evacuation pressure level in the PHP was constant. Having used thermally shrinking silicon tubes, which connected the copper wires to the glass pipe, heat affected them. Thus, when the PHP was tested without a cooling method, and temperatures in the condenser section reached 50 °C for high power supplies, that might have caused some expansion in that area of the tube, facilitating some leakage in the pipe, thus change in pressure. Perhaps using a different method for attaching the glass and copper, would have guaranteed maintaining a constant global pressure throughout different power supplies, making results more indicative of the performance of the PHP. Doing that, would solve another limit that the current design had, which is the inability to test working fluids such as acetone, since it could have impacted the thermally shrinking silicon tubes. Testing fluids with lower boiling temperatures, in future studies, could also enhance the heat transfer and the pulsating motion in the PHP.

Finally, for capturing the flow and being able to analyse the different patterns more accurately, perhaps a high-speed camera could be tested in future trials.

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