A Study of Flow and Phase Change in Porous Media:
CFD Simulations and Experiment Design

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Requirements for the degree of Master of Science in
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by

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Entitled

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Abstract

NASA space systems have continuously experienced evolution in different aspects of the technologies used for the overall mission goal. To keep up with the increased capability and performance of the systems used, there is a need for developing better thermal control systems to manage the thermal needs. Current thermal control systems, mechanically pumped single-phase cooling loops, work on the principle of sensible heating and cooling. Increased precision and amount of thermal management requirement is difficult to handle by using such single-phase cooling systems.

Loop Heat Pipes (LHP), working on the principle of phase-change of coolant to extract heat from a hot surface are the way forward. LHPs are smaller in size than single-phase systems while easily being able to maintain the required isothermality of the heat generating devices. In this work, a computational study on different aspects of flow and phase change in a portion of the LHP evaporator wick is conducted. Different computational simulations were carried out for a portion of the wick to better understand the heat transfer and flow patterns in the porous wick. A study on parasitic heat flux was conducted by using the results of computational simulations. An experimental setup is also designed to visualize the flow and phase-change in a porous media. A porous stainless steel sample was produced using loose powder sintering method. ANSYS Fluent is used as the computational platform.
Acknowledgement

Firstly, I would like to thank my parents for their constant support and motivation for every endeavor during my life. This achievement would not have been possible without them.

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Chapter 1

Introduction

1.1 Overview

Thermal control of different power electronics onboard NASA space exploration systems is a key enabler for the space mission success. One example of such a system is the Surface Water Ocean Topography (SWOT) Project scheduled for a launch in 2020. The main objective of this project is to perform a global survey of Earth’s surface water. The survey will cover at least 90% of the globe and will collect detailed measurements of how water bodies on Earth change over time. The data gathered will help to improve ocean circulation models, weather and climate predictions and aid in freshwater management around the world. Nadir Altimeter and KA-band radar interferometer are the electronic components which will help achieve the aforementioned objectives. As with any other electronic components, these components also need to be maintained at a particular temperature range for proper operation. In fact, the allowable KaRIn temperature rate of change is 20 times smaller than typical spacecraft instruments.

NASA space systems usually employ single-phase liquid cooling loops, which remove heat from heat-generating components and transport it to cooler components by utilizing the sensible heating and cooling properties. However, due to the continuous increase of and the more precise demands on cooling, single-phase cooling is not sufficient to meet these demands and phase-change cooling systems are considered good alternatives. Loop Heat Pipe (LHPs) are two-phase cooling systems, which require lesser size and weight than single-phase systems while they can achieve
higher temperature stability. LHPs unlike single-phase systems work on the principle of phase-change of coolant during heat transfer processes. Heat is removed by the LHP’s evaporator section which contains a porous wick that delivers the coolant to the heated plate of the evaporator. The coolant is then vaporized to extract heat from heated plate and the resulting vapor is transported to a condenser, which rejects heat to cooler components or space. The vapor condense in the condenser and the resulting liquid is sent back to the evaporator in a closed loop. Due to the use of latent heat, LHP provide efficient spatial and temporal stability as well.

To enhance the performance of the LHP, the heat transfer processes and flow patterns in the porous wick of the evaporator needs to be well understood. The complex coupling of thermal and hydrodynamic mechanisms in the porous wick is the subject of this thesis. In this work, we attempt to better understand those phenomena through computational and experimental study of the flow and heat transfer in porous wick. Computational Fluid Dynamics (CFD) simulations using ANSYS/Fluent are carried out using VOF multiphase model. A quasi two-dimensional (2D) experimental setup is designed to study phase-change in porous wicks and to validate the computational results.

1.2 Literature Review

Loop heat pipes (LHPs) are two-phase heat transfer devices with capillary pumping of a working fluid. They possess all the main advantages of conventional heat pipes but due to the use of capillary structure, they are capable of transferring heat efficiently for large distances at any orientation. The main principles of LHPs are [1]:

- Use of porous wicks to transport fluid
• minimum distance travelled by fluid in wick
• Minimum pressure losses in the cooling loop

Figure 1.1: Principle scheme of an LHP [1]

The principle scheme of an LHP with flow directions is shown in Figure 1.1. Coolant fluid is supplied to the evaporator section by the liquid line. The heat input applied to the evaporator vaporizes the coolant and change phase to vapor. The vapor leaves the evaporator through the vapor removal channels and reaches the vapor line. The vapor line is connected to a condenser where the vapor cools down and change back to liquid phase. The cycle is repeated after that and in this way, heat is transported from heat generating components via evaporator and rejected via condenser to heat needing components.

The processes happening in the loop are thermally and hydro-dynamically coupled. Changing a parameter in one of the components will affect the whole performance of the loop. Siedel et al [2] developed a steady-state analytical model of an LHP with a flat evaporator using energy balance equations and thermodynamic relationships in different components of the LHP.
Their model is based on the nodal network presented by Launay et al. [3] shown in Figure 1.2, in which the thermal links are represented by equivalent thermal resistances. Seidel et al. [2] combined all the equations for the nodal network and developed a quadratic equation for vapor temperature ($T_v$). All the other parameters would then be calculated based on the vapor temperature, $T_v$. 

Figure 1.2: LHP Schematic nodal network [3]
Figure 1.3 shows the relation between temperatures at different locations of the LHP with an applied heat flux ($Q_{in}$). As evident, the evaporator temperature ($T_{e}$) displays an almost linear relation with the applied heat flux for $Q_{in} > 60W$ with a slight non linearity seen for $Q_{in} < 60W$. This is important because LHP design is governed by this temperature.

He et al. [4] performed simulation of transpiration cooling with phase-change inside a porous wick by developing a one-dimensional (1-D) steady-state model that is useful is finding the saturation and temperature within the porous media domain. The assumptions made in their model were:

- Isotropic porous plates with fully saturated pores
- Solids maintain the same temperature as the local coolant at any point
- Coolant in the liquid state is incompressible and it is ideal gas in vapor state
Figure 1.4: Saturation profile of the porous wick [4]

Figure 1.5: Temperature profile of the porous wick [4]

Their results shown in Figure 1.4 and Figure 1.5 reveal that the saturation temperature is not truly constant in the two phase region and the increase in coolant flow rate pushes the profile to the heated area. That means the higher the coolant flow rate the smaller the area for the two-phase and vapor region as more area will be occupied by liquid coolant. This is an important result and shows that non isothermality needs to accounted for while modelling the porous wicks.
Chapter 2

CFD Simulations

This chapter presents the detail steps taken in order to better understand the mechanisms occurring inside a porous wick. The principles involved in all the computational studies will be discussed in this chapter.

2.1 Computation Fluid Dynamics (CFD) Simulations

CFD simulations using ANSYS/Fluent as the computational platform are carried out to simulate the flow and heat transfer in porous wicks. ANSYS/Fluent software contains broad physical modeling capabilities to model flow and heat transfer in multiphase systems. The capabilities of ANSYS/Fluent in modelling multiphase flows can be broken down into different predefined models built in the code:

1. Mixture Model
2. Eulerian Model
3. VOF Model

2.1.1 Mixture Model

The mixture model is a simplified multiphase model that can be used to model multiphase flows where the different phases move at different velocities, but assume local equilibrium over short spatial length scales. It can also be used to model homogeneous multiphase flows with very strong coupling while the phase move at the same velocity. The mixture model can model $n$ phases by solving the momentum,
continuity and energy equations for the mixture, the volume fraction equations for the secondary phases, and algebraic expressions for the relative velocities [5].

2.1.2 Eulerian Model

The Eulerian model in ANSYS/Fluent allows for the modelling of multiple separate, yet interacting phases. The phases can be liquids, gases, or solids in nearly any combination. Eulerian model doesn’t distinguish between fluid-fluid and fluid-solid multiphase flows. Momentum and continuity equations are solved for each phase while the pressure is shared by all phases. With the Eulerian model, the number of secondary phases is limited only by memory requirements and convergence behavior [6].

2.1.3 VOF Model

The VOF model can model two or more immiscible fluid by solving a single set of momentum equations and tracking the volume fraction of each of the fluid throughout the domain (on all the cells) [7]. Momentum equation and thus the resulting velocity is shared by the phases. The tracking of the interface between the phases is accomplished by the solution of the volume fraction of one (or more) of the phases [8]. Due to the focus being on tracking the interface between the phases, VOF model was deemed most suitable for our need.

2.1.3.1 Volume Fraction Equation

The equation solved in order to calculate the volume fraction in the VOF model is as follows:
\[
\frac{1}{\rho_q} \left[ \frac{\partial}{\partial t} (\alpha_q \rho_q) + \nabla \cdot (\alpha_q \rho_q v_q) \right] = S_{\alpha_q} + \sum_{p=1}^{n} (m_{pq} - m_{qp})
\]

(2.1)

where \(m_{pq}\) represents the mass transfer from phase \(q\) to phase \(p\). \(S_{\alpha_q}\) is the mass source for the \(qth\) phase. The Volume Fraction equation is solved for every secondary phase and the primary phase volume fraction is determined from the fact that the sum of volume fraction of all phases equals 1.

\[
\sum_{q=1}^{n} \alpha_q = 1
\]

(2.2)

### 2.1.3.2 Phase Change Model

In order to simulate the phase change from liquid to vapor and vice versa, a User Defined Function (UDF) based on Lee Model was formulated. Mass transfer from one phase to another was calculated using this model and it was inputted to the Volume Fraction Equation as a source term \(S_M\). Also the energy associated with that mass transfer was included as source term \(S_E\) in the energy equation for that particular phase.

#### 2.1.3.2.1 Lee Model

The Lee Model is used with mixture and VOF models to compute the mass transferred from one phase to another during phase change process. The liquid-vapor mass transfer (evaporation and condensation) is governed by the vapor transport equation [9]:

\[
\frac{\partial}{\partial t} (\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v v_v) = \dot{m}_{lv} - \dot{m}_{vl}
\]

(2.3)

where,

\(v = \text{vapor phase},\)

\(\alpha_v = \text{vapor volume fraction}\)
\[ \rho_v = \text{vapor density} \]

\[ V_v = \text{vapor phase velocity} \]

\[ \dot{m}_{lv}, \dot{m}_{vl} = \text{rates of mass transfer due to evaporation and condensation} \]

If \( T_l > T_{sat} \) then it indicates evaporation and the mass transfer is given by

\[ \dot{m}_{lv} = \text{coeff} \times \alpha_l \rho_l \frac{(T_l - T_{sat})}{T_{sat}} \tag{2.4} \]

If \( T_v > T_{sat} \) then it indicates evaporation and the mass transfer is given by

\[ \dot{m}_{vl} = \text{coeff} \times \alpha_v \rho_v \frac{(T_{sat} - T_v)}{T_{sat}} \tag{2.5} \]

where,

\[ \text{coeff} = \text{coefficient that must be fine tuned, 0.1 by default in ANSYS Fluent} \]

\[ T = \text{temperature} \]

### 2.2 Geometry and Boundary Conditions

The geometry used in the CFD simulations is a 1×1 cm\(^2\) porous domain with a fluid inlet on the top horizontal edge and fluid outlet from the bottom left horizontal edge. The right horizontal edge acts as the heat inlet for the wick. Symmetry is assumed for the vertical edges. The porous domain has a predefined porosity and permeability associated with it. The computational domain for these studies is shown in Figure 2.1.
2.3 Parasitic Heat Flux

Parasitic Heat Flux \( Q_p \) is defined as the heat flux lost by conduction at the inlet of the wick [10]. In most cases, a low amount of parasitic heat flux is desirable since that would indicate most of the heat applied being used up in phase change process. The parasitic heat flux depends on various wick properties (porosity, conductivity etc.) as well as different flow properties (liquid temperature at inlet, mass flow rate etc.). ANSYS/Fluent simulations were performed to determine the dependency of parasitic heat flux on these different parameters.
Chapter 3

Analytical Code Development

This chapter presents the details regarding the development of analytical code in a 1D porous wick domain. In order to gain more control and flexibility over characterization of the porous wick, a 1D numerical model of the porous wick was developed. Based on mass, momentum and energy conservations equations for different regions (liquid, vapor and two phase) present inside the domain, we were able to obtain the temperature and saturation profiles in the domain. The solution of these equations in single phase regions (liquid and vapor) is straightforward but the solution in the two phase region is complicated due to more number of variables and the non-linearity of the equations.

![Physical Model for the porous wick](image)

Figure 3.1: Physical Model for the porous wick [4]
3.1 Governing Equations

The equations characterizing the different regions of the porous wick as shown in Figure 3.1 are as follows:

In the liquid and vapor regions, the mass, momentum and energy conservations equations, are as follows respectively [11]:

\[ \nabla \left( \rho_k \varepsilon u_k \right) = 0 \] (3.1)

\[ -\nabla p - \rho_k g - \frac{\mu_k}{K} \varepsilon u_k = 0 \] (3.2)

\[ \nabla \left( \rho_k h_k \varepsilon u_k \right) = \nabla \cdot \left( k_{eff} \nabla T \right) \] (3.3)

where, \( k = l \) for liquid and \( v \) for vapor,

\( h = \) enthalpy,

\( u = \) velocity,

\( \mu = \) viscosity,

\( k_{eff} = \) effective thermal conductivity

The mass and energy conservation equations for the two phase region are as follows [11]:

\[ \frac{d}{dy} \left( \rho_l u_l + \rho_v u_v \right) = 0 \] (3.4)

and

\[ \frac{d}{dy} \left[ k_{lv} \frac{dT_{lv}}{dy} - \dot{m}_v h_{lv} \right] = 0 \] (3.5)

Here, \( k_{lv} = k_s \ast (1 - \varepsilon) + \varepsilon \ast k_l \ast s + k_v \ast \varepsilon \ast (1 - s) \) is the effective thermal conductivity of two-phase region with \( k_s \) being the thermal conductivity of solid. Also, \( h_{lv} \) is the latent heat of vaporization and \( s \) is the saturation.

The velocity for each phase in the two phase region can be related to the pressure gradient through the Darcy’s equation as:

\[ u_\alpha = -\frac{kK_{ra}}{\mu_\alpha} \left( \frac{dP_\alpha}{dy} + \rho_\alpha g \right) \] (3.6)
where, $K$ represents the permeability of the porous medium and is calculated by the Carman-Kozeny model as

$$K = \frac{\varepsilon^3 d_p^2}{180(1 - \varepsilon)^2} \tag{3.7}$$

Also, $K_{rl} = s^3$ denotes the relative permeability of the liquid and $K_{rv} = (1 - s)^3$ denotes the relative permeability of the vapor.

Substituting equation 3.6 into the equations 3.4 and 3.5, we get

$$\frac{d}{dy}\left( \rho_l \left( -\frac{KK_{rl}}{\mu_l} \left( \frac{dp_l}{dy} + \rho_l g \right) \right) + \rho_v \left( -\frac{KK_{rv}}{\mu_v} \left( \frac{dp_v}{dy} + \rho_v g \right) \right) \right) = 0 \tag{3.8}$$

and

$$\frac{d}{dy}\left[ K_{lv} \frac{dT_{lv}}{dy} + \left( \frac{K \rho_v h_{lv} K_{rv}}{\mu_v} \left( \frac{dp_v}{dy} + \rho_v g \right) \right) \right] = 0 \tag{3.9}$$

Now we have a system of coupled differential equations for variables $p_l$ and $T_{lv}$.

Integrating these equations once will result in following equations

$$\rho_l \left( -\frac{KK_{rl}}{\mu_l} \left( \frac{dp_l}{dy} + \rho_l g \right) \right) + \rho_v \left( -\frac{KK_{rv}}{\mu_v} \left( \frac{dp_v}{dy} + \rho_v g \right) \right) = m_0 \tag{3.10}$$

and

$$K_{lv} \frac{dT_{lv}}{dy} + \left( \frac{K \rho_v h_{lv} K_{rv}}{\mu_v} \left( \frac{dp_v}{dy} + \rho_v g \right) \right) = k_{ve} \frac{dT_v}{dy} \tag{3.11}$$

Also, we need the vapor-liquid pressure difference defined as

$$p_c = p_v - p_l \tag{3.12}$$

where, capillary pressure $p_c$ is a function of saturation $s$.

Leverett showed that the capillary pressure can be related to the saturation in following way

$$J(s) = \frac{p_c}{\delta} \tag{3.13}$$

where $J(s)$ is the leverett’s function defined as

$$J(s) = 1.417(1 - s) - 2.12(1 - s)^2 + 1.263(1 - s)^3, \tag{3.14}$$

and $$\delta = \sigma \varepsilon^{1/2} / K^{1/2} \text{ with } \sigma \text{ being the surface tension.}$$
The vapor pressure \( p_v \) can be approximated using experimental data from Baggio et al. (1997) as:

\[
p_v = a \exp(bT_{lv}) \tag{3.15}
\]

where \( a = 0.1973 \) Pa and \( b = 0.03525 \) K\(^{-1}\) for fitting data near to 370K.

### 3.2 Non Dimensional Equations

The main focus is to obtain a profile for temperature and saturation inside the two phase region and assert whether it is perfectly isothermal region or not. For that the governing equations in the two phase region are modified in terms of some non-dimensional parameters as follows:

\[
z = \frac{y - L_l}{L_{lv}}, \quad \theta_{lv} = \frac{T_{lv}}{T_{sat}} \tag{3.16}
\]

Substituting these dimensionless parameters into equations 3.10 and 3.11 will result in

\[
\frac{s^3 Da f'(\frac{ds}{dz})}{Bo L_{lv}} = \frac{Q \exp(-Pe_{v}) Ca}{Bo} - \frac{f a_{lv} Da k^*}{Bo Le L_{lv}} \frac{d\theta_{lv}}{dz} + s^3 \left( \frac{Da}{Bo L_{lv}} \xi \exp(\varphi(\theta_{lv} - 1)) \frac{d\theta_{lv}}{dz} + 1 \right) + \frac{Ca}{Bo} \tag{3.17}
\]

and

\[
\left[ \frac{f a_{lv} Da k^*}{Ca Le L_{lv}} + \frac{Da}{Ca L_{lv}} \xi \exp(\varphi(\theta_{lv} - 1)) (1 - s)^3 \tilde{v} \right] \frac{d\theta_{lv}}{dz} = Q \exp(-Pe_{v}) - \frac{Bo \alpha (1 - s)^3 \tilde{v}}{Ca} \tag{3.18}
\]

These equations are subjected to following boundary conditions

\[
\theta_{lv} = 1 \quad \text{and} \quad s = 1 \quad \text{at} \quad z = 0 \tag{3.19}
\]

\[
s = 0 \quad \text{at} \quad z = 1 \tag{3.20}
\]

The dimensionless parameters in equations 3.17 and 3.18 are defined as follows:

\[
\xi = \frac{a \varphi \exp(\varphi)}{\delta}, \quad \varphi = b T_{sat}, \quad Da = \frac{(\epsilon K)^{\frac{1}{2}}}{H}, \quad Bo = \frac{K \rho g}{\sigma}
\]
\[ Ca = \frac{\dot{m}_0 v_l}{\sigma}, \quad J' = 2.823 - 4.240(s) - 3.708(1 - s)^2 \]

\[ \bar{v} = \frac{v_l}{v_v}, \quad Le = \frac{c_{pl}\sigma(\varepsilon K)^{\frac{1}{2}}}{k_l v_l}, \quad \bar{\alpha} = \frac{\rho_v}{\rho_l}, \quad Q = \frac{c_{pv}\Delta T_v}{h_{lv}}, \quad J_{lv} = \frac{c_{pl}T_{sat}}{h_{lv}} \]

\[ k^* = \frac{k_s}{k_l} (1 - \varepsilon) + \varepsilon s + \frac{k_v}{k_l} \varepsilon (1 - s). \quad (3.21) \]

### 3.3 Numerical Solution

In order to solve the equations in two phase region, an iterative method is employed. The method is listed below:

1. Discretize the domain into elements of desired size.
2. Prescribe values to different constant parameters involved.
3. Prescribe assumed values to the \( s \) and \( \theta_{lv} \) within the two phase region.
4. Assume length of Liquid Region to be equal to the element size.
5. Assume length of the Vapor Region to be equal to the element size.
6. Find the length of two phase region from the relation: \( L_l + L_v + L_{lv} = 1 \).
7. Iterate equations 3.17 and 3.18 in the whole two phase region.
8. Check whether the boundary condition 3.20 is satisfied at the last node of two phase region or not.
9. If no increase the value of vapor region by grid size. This happens every time 3.20 is not satisfied.
10. Repeat steps 5 to 8.
11. If 3.20 is not satisfied for all possible values of \( L_v \) then increase \( L_l \) by a size of grid and repeat from 5 to 10.
12. Repeat 11 until a solution is found.
13. Plot profiles of \( s \) and \( \theta_{lv} \) in two phase region.
Chapter 4

Experiment Design

This chapter presents the detail steps taken in order to design an experiment to study and visualize the phase change process inside porous wicks. The results will also be used to validate the computational model.

Fig. 4.1 shows a schematic of the experimental setup. The experiment consists of a thin porous wick sandwiched between two plexiglass plates. The porous wick is heated from the top using a cartridge heater inserted into a stainless steel casing. The coolant is supplied from the reservoir through the bottom portion. The vapor lines provide a passage for the coolant vapor to escape and contain in a reservoir after condensation.
Figure 4.2 shows the main portion of the experiment setup. It consists of a 100×50×1 mm³ porous wick sandwiched between two plexiglass plates. The middle 60 mm portion of the wick top edge is in contact with a stainless steel plate. A 250W cartridge heater is inserted in stainless steel plate's top portion to heat up the plate and subsequently the wick. An O-ring and screws are used in order to properly seal the experiment so coolant (either in liquid or vapor form) won't be able to exit the experiment from unintended areas. HFE 7100 is to be used as the coolant. The wick shown in the picture is a titanium wick of 40 micron rating.

An infrared camera and thermocouples are used to measure the temperature distribution of the heated plate and porous wick. A regular camera is used to monitor the interface of the liquid and vapor in the porous wick. The rate of liquid evaporation is measured by condensing the vapor living through the vapor lines and measuring the weight change of the resulting liquid. Various porous wicks with different permeabilities, porosities and materials are tested.
The characteristics of the wick and coolant used are listed in tables below:

**Table 4.1 Characteristics of Polyethylene Porous Wick**

<table>
<thead>
<tr>
<th>Material</th>
<th>UHMW Polyethylene</th>
<th>Dimensions</th>
<th>100mm<em>50mm</em>1mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Micron Rating</td>
<td>50</td>
<td>Temperature</td>
<td>Up to 210°F</td>
</tr>
<tr>
<td>Color</td>
<td>White</td>
<td>Compressibility</td>
<td>Compressible</td>
</tr>
</tbody>
</table>

**Table 4.2 Characteristics of Titanium Porous Wick**

<table>
<thead>
<tr>
<th>Material</th>
<th>Sintered Titanium</th>
<th>Dimensions</th>
<th>100mm<em>50mm</em>1mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Micron Rating</td>
<td>40</td>
<td>Temperature</td>
<td>Up to 3000°F</td>
</tr>
<tr>
<td>Color</td>
<td>Silver</td>
<td>Compressibility</td>
<td>Incompressible</td>
</tr>
</tbody>
</table>

**Table 4.3 Characteristics of Coolant**

<table>
<thead>
<tr>
<th>Material</th>
<th>3M 7100</th>
<th>Odor</th>
<th>Odorless</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boiling Point</td>
<td>142°F</td>
<td>Vapor Pressure</td>
<td>202mmHg</td>
</tr>
<tr>
<td>Color</td>
<td>Colorless</td>
<td>Surface Tension</td>
<td>13.6 dynes/cm</td>
</tr>
</tbody>
</table>
Chapter 5

Results and Discussion

This chapter presents results of the CFD simulations, analytical simulations and experiments. Based on these results, conclusions for different aspect of porous wick are given in the following chapter.

5.1 Analytical Simulations

Analytical calculations of flow and heat transfer through a 1-D porous media of 35% porosity are performed using a flow rate of 0.5 kg/s and heat input of $1.5 \times 10^6$ W/m$^2$. The profiles of the saturation and temperature in the two phase region of a porous wick is our main concern. The results from the simulation using MATLAB are shown below:

![Figure 5.1: Profile of Non Dimensional temperature in two phase region](image-url)
The temperature profile as shown in Figure 5.1 shows that the temperature in the two phase region is not perfectly isothermal. The temperature difference across the two phase length is only about 0.12% of the saturation temperature. There is steep temperature gradient at the beginning of the two phase region with the profile starting to equalize close to the middle of the region. This results shows that the assumption of adiabatic conditions in the two-phase region is valid within 0.12%.

The profile of liquid saturation is shown in Figure 5.2. The saturation profile depicts steep gradient at both ends of two phase region with the inflection point occurring around the middle of the two phase region.

**5.2 CFD Simulations**

As mentioned in Chapter 2, CFD simulations of a portion of the porous wick used in the evaporator are carried out. The results from these simulation are presented in term of volume fraction contours and parasitic heat flux.
5.2.1 Volume Fraction Contours

Figure 5.3 shows the volume fraction contour in the porous wick for different heat fluxes. The parameters used in these simulations are shown in Table 5.1. At small heat flux \( Q = 0.6 \text{ W/cm}^2 \), there is a little amount of vapor formed on the top of the heated section. As the heat flux applied increases, there is a rapid growth of the vapor bubble as the increased heat helps vaporize more liquid in two phase region. As the heat flux applied increase beyond 0.8 \text{ W/cm}^2, the rate of bubble growth decrease significantly and the heated portion is covered fully by vapor and the two phase region also shrinks its size.
Figure 5.4 Volume fraction distribution along the bottom edge of domain

Figure 5.5 Volume fraction distribution around the interface of outlet and heated wall
Figure 5.4 shows the distribution of volume fraction of vapor along the bottom edge of domain. There is a presence of steep gradient around the interface of outlet and heated wall as shown in Figure 5.5. The general trend is for the gradient to be sharper as the heat flux increases.

Table 5.1 Mass flow rate of vapor in outlet

<table>
<thead>
<tr>
<th>Flux (W/cm²)</th>
<th>Flow rate of Vapor in outlet (kg/s)</th>
<th>Flow rate as percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.8</td>
<td>9.3e⁻¹⁰</td>
<td>0.4%</td>
</tr>
<tr>
<td>0.9</td>
<td>3.7e⁻¹⁰</td>
<td>1.8%</td>
</tr>
<tr>
<td>1.0</td>
<td>3.2e⁻¹⁰</td>
<td>1.6%</td>
</tr>
<tr>
<td>1.3</td>
<td>6.5e⁻⁹</td>
<td>32.5%</td>
</tr>
</tbody>
</table>

The mass flow rate of vapor in outlet is shown in Table 5.1. With increase in the applied heat flux the amount of vapor leaving from the outlet is increasing as well.

5.2.2 Parasitic Heat Flux ($Q_p$)

The effect of different parameters influencing the parasitic heat flux to the liquid reservoir, are studied and the results are presented in this section. The base parameters used for this study are listed below:

Table 5.2 Base case parameters for parasitic heat flux study

<table>
<thead>
<tr>
<th>Material</th>
<th>Titanium</th>
<th>Liquid mass flow rate ($\dot{m}_0$)</th>
<th>$10^4$ Kg/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity ($k$)</td>
<td>10 W/m.K</td>
<td>Gravity (g)</td>
<td>0 m/s²</td>
</tr>
<tr>
<td>Density of wick material</td>
<td>2719 Kg/m³</td>
<td>Heat flux ($Q$)</td>
<td>0.7 W/cm²</td>
</tr>
<tr>
<td>Porosity ($\varepsilon$)</td>
<td>0.4</td>
<td>Liquid permeability ($K$)</td>
<td>$1\times10^{-11}$ m²</td>
</tr>
</tbody>
</table>
The temperature and volume fraction contours for the set of base parameters are shown in Figure 5.6 and Figure 5.7 respectively. Most of the region apart from the heated portion is filled by liquid while the heated portion of the wick is covered by two phase mixture and vapor as seen on Figure 5.7. Also, Figure 5.6 shows that there is a minimum amount of temperature gradient across the inlet of wick.
The relation between the non-dimensional parasitic heat flux, $Q_p/Q$ and the inlet mass flow rate $m_0$ is shown in Figure 5.8. It is evident that parasitic heat flux is inversely proportional to the coolant mass flow rate. Hence, more amount of liquid coming into the porous wick results in more amount of heat applied being used for phase change and less will be converted into parasitic heat. Thus, higher coolant mass flow rate is desirable for two phase flow in porous media.

Figure 5.8: Relation of parasitic heat flux with coolant flow rate
The relation between $Q_p/Q$ and thermal conductivity of the wick, $k$ is shown in Figure 5.9 and the relation between the heater plate temperature, $T_{\text{hot}}$ and thermal conductivity of the wick, $k$ is shown in Figure 5.10, reveals some interesting results. The temperature of the heated surface decreases with increasing thermal conductivity while the portion of the heat flux converted into parasitic heat increases. This result reveal that the lower thermal conductivity wick will trap most of the heat near to the portion where the heat flux is applied which results in higher temperature of the hot surface while decreasing the amount of the parasitic heat.
The relation of $\frac{Q_p}{Q}$ with the porosity of the wick is inversely proportional as it can be seen in Figure 5.11. Higher porosity of the porous wick will result in lower thermal conductivity and thus will have lower parasitic heat flux as revealed by the relation of parasitic heat flux with thermal conductivity shown in Figure 5.9.
The dependence of parasitic heat flux on the applied heat flux is shown in the Figures 5.12 & 5.13. The total parasitic heat flux increases as the applied heat flux increases but the percentage of the applied heat flux converting to the parasitic heat flux decreases. This can be explained by the fact that with more heat flux applied, the
amount of vapor created is more and higher portion of the applied heat flux is used up in phase change rather than conducted to the liquid reservoir as parasitic heat.

5.2.3 Phase change rate evaluation

The results from ANSYS Fluent simulations at different heat flux were evaluated and vapor content on the porous domain was obtained. Using the vapor content in the porous wick for five different heat flux input, Figure 5.14 is drawn. This figure shows that the amount of water vapor created in the porous media increases with the applied heat flux, then levels up around 0.9 W/cm² and higher heat fluxes.

![Figure 5.14: Relation of vapor content to the applied heat flux](image-url)
5.3 Experimental Results

In this section, preliminary results from the experimental apparatus are presented. Experiments were performed using a plastic porous wick.

Figure 5.15: (a) Vapor Profile in a plastic wick, (b) Asymmetrical Vapor profile in a lower heat input

Fig. 5.15 shows the vapor profile in a plastic wick after vaporization. The vapor profile in Figure 5.15 (a) is bigger corresponding to higher heat flux of around 28W and looks almost symmetrical but at lower heat flux of 10W, asymmetrical profile can be seen as in Figure 5.15(b). This could most likely be due to some features/errors with the design or the contact between the stainless steel and wick, or contact between the wick and plexiglass. Volatility of the coolant used might be a factor in this as well. Obtaining thermal profiles have proved difficult to the incompatibility of the plexiglass with the IR thermal camera. Those profiles might help us explain why there is asymmetrical profile is seen in a lower heat flux and symmetrical profile in higher flux.
Chapter 6

Conclusions

In this thesis, CFD simulations of parasitic heat flux study, analytical code development and experiment design were performed.

Based on the CFD results, it can be concluded that a wick with lower thermal conductivity, higher coolant mass flow rate and higher heat flux applied are ideal in order to decrease the parasitic heat flux. Higher parasitic heat flux would also represent a higher area of wick being covered by the vapor and ideally we want the vapor zone to be smaller and limited close to the heated surface.

From the 1D analytical simulations using MATLAB, it became clear that the two phase region in a porous wick is not perfectly isothermal even though the temperature difference is very low.

The experiments performed on titanium and plastic wick showed an asymmetrical vapor bubble profile. Some troubles were encountered with getting accurate thermal reading through the plexiglass so there is a need to modify the current experiment or use an alternative material to replace the plexiglass. It would be ideal to use a material with a higher temperature range so as to incorporate water as the coolant so that volatility of other industrial coolant are removed from equation and it would be much easier to quantify the vaporization rate from the experiment.
Bibliography


