Development and Analysis of Sulfur Based McGill Heat Pipe

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By

Hujun Zhao

Department of Mining, Metals and Materials Engineering McGill University, Montreal, Canada



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Abstract

The development of a mid-temperature range (250°C - 500°C) heat pipe for high heat flux applications has been the focus of numerous researchers during the last 40 years. However, until this work a viable working substance for the heat pipe has eluded researchers. While the most mentioned element has been sulfur, its unusual viscosity-temperature relationship has prevented the commercialization of a sulfur-based heat pipe.

The recent development (and patenting) of the McGill heat pipe revived the question of whether sulfur would be viable in such a unit. Extensive testing showed that it is possible to make a high heat flux heat pipe with sulfur as the working substance. Given the lack of scientific details about the McGill heat pipe, a focused research program was undertaken to quantify the operation of the McGill heat pipe prior to studying the sulfur based unit.

One study looked at the two-phase flow characteristics of the McGill heat pipe. Both qualitative (videos) and quantitative data like the pressure drop and returning velocity were measured as a function of gas flow rate. Moreover, a new non-dimensional parameter, the modified swirler number was proposed. Further, the Lockhart-Martinelli method was used to analyze the pressure drop.

In the McGill heat pipe, the centrifugal force that is produced by the vortexing flow pushes liquid up against the walls and increases the critical heat flux. A theoretical model consisting of 4 sub-models was developed to predict the critical heat flux for defined situations.

The development of the sulfur-based heat pipe followed the empirical and mathematical modeling work that was carried out. A McGill heat pipe with sulfur as the working substance was designed, built and tested. The design was arrived at by considering the modeling work that was originally carried out. A number of interesting features were discovered with the sulfur-based heat pipe. A model based on mass, energy, and flow balances between the condenser and the evaporator was also developed. The model can be used to calculate the void fraction, quality, wall temperature, local heat flux distribution, heat load, cooling flow rate, and working substance temperature. The experimental results fit well the calculated ones.

Résumé

Le développement d'un caloduc opérant dans la zone de température moyenne $(250^{\circ}C \sim 500^{\circ}C)$ pour des applications de transfert de chaleur élevé a été le centre d'intérêt de nombreux chercheurs pendant les 40 dernières années. Cependant, jusqu'à ce travail une substance de travail viable pour le caloduc a été éludée des chercheurs. Bien que l'élément le plus mentionné soit le soufre, sa relation viscosité-température inhabituelle a empêché la commercialisation d'un caloduc à base de soufre.

Les développements récents (et la demande de brevet) du caloduc de McGill ont remis en question la viabilité du soufre dans une telle unité. Un important programme d'essais a prouvé qu'il était possible d'obtenir un transfert de chaleur élevé avec un caloduc utilisant le soufre comme substance de travail. Étant donne le manque de détails scientifiques sur le caloduc de McGill, un programme de recherche ciblée a été élaboré pour quantifier le fonctionnement du caloduc avant d'étudier une unité à base de soufre.

Une étude a été faite sur les caractéristiques biphasées d'écoulement du caloduc de McGill. Des données qualitatives (des vidéos) et quantitatives comme la baisse de pression et la vitesse de renvoi ont été mesurées en fonction du débit de gaz. De plus, un nouveau paramètre non-dimensionnel, le nombre de tourbillonnement (swirler number) a été proposé. D'autre part, la méthode Lockhart-Martinelli a été utilisée pour analyser la baisse de pression.

Dans le caloduc de McGill, la force centrifuge qui est produite par l'écoulement en vortex pousse le liquide vers le haut contre les murs et augmente le flux critique de la chaleur. Un modèle théorique se composant de 4 sous-modèles a été développé pour prévoir le flux critique de la chaleur pour des situations définies.

Le travail de modélisation empirique et mathématique complété, un caloduc à base de soufre a été développé. Un caloduc selon le concept de McGill avec du soufre comme substance de travail a été conçu, construit et examiné. La conception a été atteinte en se basant sur le travail de modélisation qui a été fait dans un premier temps. Un nombre de caractéristiques intéressantes a été découvert avec le caloduc à base de soufre. Un modèle basé sur la masse, l'énergie, et des équilibres d'écoulement entre le condensateur et le vaporisateur a été également développé. Le modèle peut être employé pour calculer la fraction vide, la qualité, la température du mur, la distribution locale de flux de la chaleur, la charge thermique, la vitesse de refroidissement, et la température de la substance de travail. Les résultats expérimentaux confirment les prédictions calculées.

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Nomenclature

Symbol	Meaning	Units
A	Flow area	m^2
a_l	Constant in the regressed correlation (chapter 3)	
a_2	Constant in the regressed correlation (chapter 3)	
b	Wetted ratio in one wave	
Bo	Boiling number	
С	Constant in Chisholm equation	
С	Complex wave speed	
C_0	Distribution parameter	
Ci	Imagine part of the complex wave speed	
c _{pl}	liquid heat capacity	$J/(kg \cdot C)$
C _r	Real part of the complex wave speed	
D	Diameter	m
D_h	Hydraulic diameter	m
Ε	Coefficient in the Gungor and Winterton correlation	
e	Diameter of the coil	т
F	Force	Ν
f	Friction factor	
F_m	Multiplier in Chen's correlation	
F_{v}	view factor	
G	mass flux G	$kg/(m^2s)$
g	Gravity acceleration	m/s^2
Η	Width of the spiral whole tunnel	m
h	Vertical distance between the inlet and outlet of return(Chapter 3)	М
	Heat transfer coefficient (Chapter 6)	$W/m^2 \cdot C$
h _{FZ}	Heat transfer coefficient in Forster and Zuber correlation	$W/m^2 \cdot C$

h _{fg}	Latent heat	J/kg
J	Momentum flux	kg/(m·s)
K	Conductivity	$W/m \cdot \mathcal{C}$
k	Wave number in one period	
k_p	Minor pressure factor	
L	Length	m
М	molecular weight	
т	Mass flow rate	kg/s
Р	Pressure	Ра
p	Pitch of the coil	m
<i>P1</i>	Pressure at the inlet of the returning line	Ра
<i>P2</i>	Pressure at the outlet of the returning line	Ра
Pr	Prandtl number	
q''	Heat flux	kW/m^2
Re	Reynolds number	
Ri	inner radius of the spiral tunnel	m
Ro	Outer radius of the spiral tunnel	m
S	Coefficient in Chen's correlation	
S_{gw}	Coefficient in the Gungor and Winterton correlation	
Sw	Swirler parameter	
Т	Temperature	${}^{e\!c}$
T _{fu}	Furnace temperature	K
T_p	Pipe temperature	K
U	Initial liquid velocity in the evaporator	m/s
U_{f}	Vapor velocity in the separate model	m/s
U_{v}	Vapor velocity in the separate model	m/s
V	Velocity	m/s
V _g	Local superficial velocity of air	m/s
V _{gj}	Mean drift velocity	m/s
V _I	Local superficial velocity of water	m/s

' r

 $\sum_{i=1}^{n}$

Subscripts

ас	Acceleration
amb	Ambient
С	Condenser
Con	Convection
f	fluid
FC	Forced convection
fr	Friction
fu	Furnace
g	gas(air)
gr	Gravitation
i	Inner
l	liquid
NB	Nucleate boiling
0	Outer
r	Return
Rad	Radiation
S	Smooth tube
Sat	Saturation
tp	Two phase
total	Total
v	vapor

W	Pitch of the spiral whole tunnel	m
x	Coordinate (In chapter 3 it means quality)	
x_0	Position of the first wave occurs	m
X_{tt}	Lockhart-Martinelli parameter	
<i>x</i> *	First wetting location	m

Greek characters

α	Void fraction	
δ	Vapor thickness in separate model	m
Е	Emissivity of the heat pipe surface	
$\phi_{\scriptscriptstyle L}$	Two-phase flow friction multiplier	
η	Complex function of the interface	
$\eta_{\scriptscriptstyle 0}$	Maximum amplitude of the sinusoidal perturbation	
φ	Slope angle of the spiral tunnel	
λ	Wave length	m
μ	Viscosity	$Nm^{-2}s$
θ	Coordinate in chapter 4	
ρ	Density	kg/m ³
$ ho_f'$	Modified liquid density	kg/m^3
$ ho_{v}^{\prime}$	Modified vapor density	kg/m ³
σ	Surface tension	N/m
$\sigma_{\!sb}$	Stephan-Boltzman constant, $5.67 \times 10^{-8} Wm^{-2}K^{-4}$	
τ	Shear stress	N/m^2
τ_i	Interfacial shear	N/m^2
ΔP_r	Pressure drop in the returning line	Pa

Chapter 1 Introduction

1.1 Heat pipe basics

As the global energy crisis persists and the global warming debate deepens, heat transfer devices with high efficiency have taken on a pivotal role in contemporary engineering applications. Being a device of very high thermal conductance, a heat pipe with phase-change (i.e. two-phase system) offers significant advantages over traditional single-phase heat transfer equipment. In a heat pipe, considerably smaller mass flow rates are required to transport equivalent amounts of energy than in single-phase liquid or gas systems. Heat transfer coefficients in two-phase systems are much higher than in single-phase systems. Lower mass flow rates and enhanced heat transfer can be achieved in smaller system sizes while providing increased capability. The thermal capacity of a single-phase heat transfer device depends on the enthalpy change of the working fluid, and this may require both a large temperature gradient and flow rate. On the other hand, a heat pipe can be operated isothermally regardless of variations in the heat load.

Traditional heat pipes have three major components, which are a sealed container, a wick structure and a working fluid. As the heat is transferred from a heat source (evaporator) to a heat sink (condenser), a heat pipe can be divided into three sections, an evaporator section, a transport section and a condenser section. When heat is added to the evaporator section, working fluid is evaporated and the vapor moves to the condenser section. In the condenser the vapor is condensed and the condensate is returned to the evaporator by capillary force or gravity.

1.2 The concept of the McGill heat pipe

The two-phase system in a heat pipe typically increases thermal capacity associated with the phase change of a working fluid. However, much of the metallurgical industry operates beyond the operational range of the device because the heat fluxes encountered are often above the critical heat flux (CHF). As a result, the classical (i.e. conventional) heat pipe is prone to film boiling within the evaporator. When this happens, failure of the unit is often the result. Much research was devoted to finding an alternative heat pipe configuration that was capable of handling the heat fluxes typically encountered in the metallurgical industry. The result was the development of the 'McGill Heat Pipe', which is documented in US Patent 7,115,227 [1] that was issued on Oct.3, 2006. Several industrial engineering applications of the McGill heat pipe have demonstrated its effectiveness and viability.

A McGill heat pipe is a sealed vessel that consists of three main parts: the condenser section, the evaporator section and the working fluid within the vessel. The working fluid absorbs heat from the heat source and vaporizes. The vapor moves up to the condenser section where it condenses and releases its latent heat of vaporization. The heat that is released in the condenser is dissipated by conventional heat transfer techniques. The condensate gathered in the reservoir is returned to the evaporator through a return pipe. Comparing with the traditional heat pipe, the two distinguishing characteristics of the McGill heat pipe are the flow modifier in the evaporator and the return pipe connecting the reservoir and the evaporator.

The spring shaped or helical shaped flow modifier causes the fluid within the evaporator to flow in a helical fashion. Thus, the denser liquid is forced to contact the evaporator wall because of the centrifugal force that is generated. In the nucleate boiling regime, the heat flux and the bubble formation rate increase with the temperature difference between the evaporator wall and the working fluid. As the heat flux continues to increase beyond the critical heat flux, the bubble population becomes so high that

portions of the heating surface are covered by a layer of insulating vapor. This is known as partial film boiling. In this region, the heat flux decreases as the temperature difference between the evaporator wall and the working fluid increases. This mode of operation is undesirable in the operation of heat pipes. Hence the centrifugal force generated by the rotating movement overcomes the formation of a vapor film layer and consequently the limitation of film boiling is improved.

The role of the returning pipe is to provide a separate path through which the liquid can travel down to the bottom of the evaporator. In the traditional heat pipe the shear force existing at the liquid-vapor interface because of the countercurrent flow condition between the vapor and the liquid prevents the condensate from returning to the evaporator. Especially when the vapor velocity is sufficiently high, the condensate cannot be returned to some of the extremities of the evaporator. This condition is known as the entrainment limit. With the return pipe, the liquid can be returned to the bottom of the evaporator without any entrainment from the rising vapor. It should be mentioned that an excess of liquid typically flows through the return pipe to the evaporator. The excess liquid is carried by the vapor back to the condenser. This has the effect of transforming the flow in the evaporator of McGill heat pipes into co-current, rather than counter-current as that in traditional heat pipes.

1.3 Working fluids for the intermediate temperature range

Depending on the operating temperature, four temperature ranges for heat pipes are usually described and associated to working fluids. These are the cryogenic range (<200K), the low temperature range (200~550K), the intermediate temperature range (550~800K) and the high temperature range (>800K). In the low temperature range, water is the typical working fluid. But when water is used as the working fluid, cold water may be required to work as the coolant in the condenser for high heat fluxes, and this may not be permitted in many metallurgical engineering cases. Alkali metals are used in the high temperature range, but high quality materials are necessary for the manufacture of such a heat pipe. So the intermediate working temperature is suitable for heat pipes that can be

used in many metallurgical cases. However, it has long been the difficult range for heat pipes to work with. Mercury and some organic fluids have been advanced as working fluids. But in reality mercury vapor is extremely poisonous, and the organic fluids will decompose after operating for some time and contaminate the heat pipe with non-condensable gas. Consequently, mercury and organic fluids are not suitable candidate fluids.

Sulfur is an element with appropriate melting and boiling points (119°C and 444°C). But it was abandoned by the researchers because of its high viscosity.

In a McGill heat pipe, the expanded cross-section in the condenser reduces the vapor velocity and makes the condensation effective; the reservoir collects all the condensed liquid, and a separate pipe returns the liquid to the bottom of the evaporator. The centrifugal force produced by the rotating movement restrains film boiling. All these specific structures allow sulfur to work in a McGill heat pipe as will be demonstrated in this thesis. It is important to note at this point that there are no published reports of the use of sulfur in high heat flux applications with conventional heat pipes. The research to be described will show why this was so.

1.4 Objectives and approach

The objectives of the current research are summarized by the following statements:

• Air-water simulation of two-phase flow in the McGill heat pipe

Air and water are used to simulate the two phase flow in the McGill heat pipe. A new non-dimensional parameter, the swirler parameter, is proposed in the analysis of the friction factor in single-phase flow. The Lockhart-Martinelli method is adopted to analyze the two-phase pressure drop.

• Develop a theoretical model to predict CHF(critical heat flux) in a spiral full tunnel Critical heat flux is predicted by using instability analysis in a separate model, which proves that the CHF is improved substantially in a rotating flow. • Develop a sulfur based McGill heat pipe

A sulfur/iodine mixture as promoted in the literature is tested in a McGill heat pipe and proved to be not suitable. The old design of McGill heat pipe is altered to allow sulfur to be a viable working substance. Experiments are carried out to test this new system at high heat fluxes.

• Generate a theoretical model to predict the working properties of a sulfur based McGill heat pipe

Based on the results from the air-water modeling experiments, a theoretical model is developed to predict the working capacity, and the operating temperature of the sulfur based McGill heat pipe. Experimental results are compared to the results from the model.

Chapter 2 Literature survey

2.1 Heat pipe technology

2.1.1 Basic Concept of a heat pipe

The concept of the heat pipe was first presented by R.S. Gaugler [2] in1944, but it was not widely publicized until an independent development by G.M. Grover [3] at the Los Alamos Scientific Laboratories. In general terms, a heat pipe is a heat transfer device comprising three basic sections: the evaporator section, the adiabatic section and the condenser section, as shown in Figure 2.1. It utilizes the vaporization and condensation of a working substance contained within to move energy from the evaporator to the condenser. The evaporator section absorbs heat from a heat source and thus causes some of the liquid to vaporize. The vapor moves to the condenser section, where it condenses on the pipe walls and releases its latent heat of vaporization. The heat is extracted by cooling air or other coolants. The condensate is returned to the evaporator by capillary force. The heat pipe is, in effect, a 'superconductor' of heat energy. Tests have shown that a heat pipe can be as effective in transporting energy as 1,000 times the equivalent quantity of copper under similar heat transfer conditions.



Figure 2.1 Schematic diagram of operating principles of a heat pipe

Typically, many heat pipe applications involve the use of heat pipes in vertical or inclined positions. During heat pipe operation, the working substance is evaporated and flows to the condenser section, where it condenses and returns by gravity in liquid form to the evaporator.

Numerous papers and patents have been issued since heat pipes gained prominence in the early 1960s. In addition, a number of textbooks have also been written on the subject. The reader is referred to References [4] [5] [6] for a detailed account of heat pipe principles. Several points are worth noting about classical heat pipes: 1) the heat pipe chamber is completely evacuated after it is charged with the working substance, and 2) the classical heat pipe is a two phase, counter-current flow system in which vapor moves up while liquid moves down.

2.1.2 Limitations of Traditional Heat Pipes

Generally, there are five limitations for heat pipes. These are: Viscous limitation, sonic limitation, capillary limitation, entrainment limitation and boiling limitation.

Viscous limitation

The viscous limitation represents the highest heat transport that can be sustained in a heat pipe for a specific vapor viscosity at the evaporator of the heat pipe. Viscous limitation occurs when the operating temperature is relatively low and the applied heat load is small. It refers to a limitation in which the viscous forces are larger than the driving forces caused by the applied heat load. Under this condition, there is a low flow in the system and the heat transport capability is limited. This situation is usually observed in cryogenic applications or in start-up from a frozen state.

Sonic limitation

Cotter [6] proposed that compressible flow in a duct of constant cross section with mass addition and removal (i.e., vapor flow in a vapor channel) and constant mass flow in a duct of variable cross section (compressible flow in a converging-diverging nozzle) share a number of common properties. Therefore, for some LHP (Loop Heat Pipe) applications, especially those with liquid metal working fluids, the vapor velocity in the vapor channel may reach sonic values during the start-up or steady-state operation. Under this condition, the mass flow rate in the system reaches its maximum value and the flow is choked.

Capillary limitation

The capillary limitation is the highest heat transport rate that can be sustained by the capillary pressure in a heat pipe wick. A higher mass flow of liquid that can cover the entire evaporator cannot be generated by the capillary pressure once the limitation threshold is reached. Operating a heat pipe above its capillary limit will result in a dry out region without working fluid wetting the inner surface. The dry out region is often called a hot-spot. If the total system pressure drop surpasses the capillary limit, the wick dries out and operation of the heat pipe becomes unstable.

Entrainment limitation

The entrainment limitation is often referred to as the counter-current flooding limitation in a vapor-liquid flow system. In traditional heat pipe operation, the vapor and liquid flow in opposite directions, causing surface tension and shear forces to interact across the liquid-vapor interface. When the vapor velocity is sufficiently high, the shear force may tear the liquid from the wick and entrain it in the vapor flow. This phenomenon inhibits the return of liquid to the evaporator.

The entrainment phenomenon leads to the partial and total stoppage of liquid flow or dry out. It is generally characterized by the Weber number which compares the vapor inertial force to the liquid surface tension force.

Boiling limitation

In traditional heat pipes, the heat has to conduct all the way through the wick structure, which is saturated with liquid, and evaporate the liquid in the core area. When the applied heat load or the wall temperature becomes excessively high, boiling of the liquid in the wick structure may occur. The vapor bubbles generated inside the wick structure may block the liquid return paths and the wick can dry out.



Figure 2.2 Limitations to heat transport in the heat pipe[6]

All these limitations on maximum axial heat transport are illustrated in general form in Figure 2.2.

2.2 The development of the McGill Heat Pipe

The start of the development of the McGill heat pipe can be traced back to 7 years ago (1.e. 1999). G. Zheng [7] tried various types of flow modifiers in a traditional heat pipe. Four different flow modifiers were tested (Figure 2.3): 4-way groove twisted bar, twisted tape, coiled wire and helical rabbit.



Figure 2.3 Various types of swirlers

Experiments were carried out to prove that a water based heat pipe with swirling flow can be operated at heat fluxes up to an order of magnitude greater than those typically associated with a water-based, conventional heat pipe. The flow modifier caused the fluid within the evaporator to flow in a helical fashion, causing the denser liquid to be pressed against the outer tube wall due to the centrifugal forces generated. During operation, the liquid is pressed onto the evaporator wall as it rises. The liquid wets the wall and absorbs the heat input by vaporizing. This not only avoids the formation of hot spots but also suppresses film boiling at least at fluxes that would cause film boiling under normal circumstance.



Figure 2.4 Concept of the novel heat pipe

In Zheng's research, the concept of the novel loop heat pipe (Figure 2.4) was also presented. In that heat pipe a reservoir in the condenser was incorporated to handle the condensing liquid, and a return pipe was connected to the evaporator bottom. The pressure head between the reservoir and the return line discharge made the returning flow possible. Compared to the traditional heat pipe, the two-phase flow in the evaporator of this heat pipe is co-current instead of counter-current. Thus the shear forces between the reverse flow streams are eliminated and no entrainment limitation occurs.

Zhang [8] [9]used the following design (Figure 2.5) in the controlled cooling of permanent mold castings of aluminum alloys. The return pipe came out of the reservoir and extended to the end of the evaporator. A valve was incorporated in the return line to control the return flow, and also to allow the heat pipe to be turned on or off at will. The valve played a very important role in the process of permanent mold cooling because the mold does not need to be cooled, for example, when it is preheated.



Figure 2.5 Basic configuration of the novel heat pipe

Z. Yuan[10][11] expanded the condenser, and made it look like a standard shell and tube heat exchanger with two cooling circuits: an external jacket and a series of internal cooling pipes. The coolant (air or water) came into a chamber beneath the condenser and flowed through the internal cooling pipes to the chamber above the condenser, and then down through the jacket to the outlet (Figure 2.6). The expanded cross-section reduced the vapor velocity, and the condensing efficiency was consequently improved.



Figure 2.6 A McGill heat pipe with the expanded condenser

The advantages of the McGill Heat pipe can be summarized as follows:

- The centrifugal force produced by the rotating movement restrains film boiling.
- The returning pipe eliminates the shear force between vapor and liquid.
- The reservoir collects all the condensed liquid to maintain a continuous flow.
- The enlarged cross-section of the condenser reduces the vapor velocity.

2.3 Applications of McGill heat pipes

The McGill heat pipe has been used in several metallurgical applications where temperatures and heat fluxes reach the maximum safe range of conventional cooling technology.

C.Zhang [12] used the flexible heat pipe cooling system for low pressure die castings on an industrial scale casting machine. In this system a valve was inserted in the return line to control the return flow, and also to allow the heat pipe to be turned on or off. A flexible connection between the condenser and the evaporator made it possible to change the level of the reservoir, allowing the pressure head for the returning flow to be adjusted according to need (Figure 2.7).



Figure 2.7 Flexible heat pipe cooling system for the low pressure die casting

K. Elalem et al [13] [14] used a similar system in the cooling of permanent molds for the casting of magnesium alloys.

P. Navarra et al [15] applied McGill heat pipes in the industrial tests of the cooling of a slag taphole (Figure 2.8). A copper test block was designed with dimensions similar to those of an industrial slag tapping block (1.1m×1.0m×0.16m). In the experiments the

heat pipe was able to extract 135kW of heat at steady state when operated in the range of 120°C to 130°C. Thermal modeling of the system indicated that a continuous local heat flux of 2.4 MW/m² was dissipated by the water based heat pipe.



Figure 2.8 The schematic diagram of the taphole cooling system

P. Navarra et al. [16] also reported on the modeling work and the experiments with the heat pipe cooling of a slag launder (Figure 2.9). Three evaporators were positioned at the bottom of the slag launder, normal to the slag flow. The 3 heat pipes were connected to one condenser.



Figure 2.9 The schematic diagram of the slag launder cooling system

2.4 Spring flow modifier

Spring flow modifiers have been used in heat exchangers, pre-heaters or boilers as heat transfer enhancement devices. In addition, they also possess several other advantages: low cost; easy installation and removal; and preservation of the original plain tube mechanical strength. A spring flow modifier acts as a swirl flow generator, which produces a helical flow at the periphery of the evaporator. This rotating flow will cause centrifugal forces. In single phase flow and heat transfer, the density of most liquids decreases with temperature, so the centrifugal force moves the heated liquid from the wall toward the axis, which enhances the heat transfer. In a McGill heat pipe, this effect is more obvious, since the density of the vapor produced on the heated wall layer is much smaller that of the liquid in the central parts. The centrifugal force pushes the liquid to the walls, which not only produces a heat transfer augmentation but also overcomes the film boiling limitation.

J.P. Chiou [17] carried out experiments with a class of spiral spring coil that was used as a heat transfer augmentative device for a single phase cooling system. He found that the ratio of the heat transfer coefficient of the oil flow in a tube with spring insert to that in a bare tube reached 3 when the Reynolds number was lower than 1200, and a constant value of about 1.5 when the Reynolds number was greater than 6000.

By using water and water-propylene glycol mixtures at different temperatures, A. Garcia et.al [18] carried out experiments on six coils inserted in a smooth tube, covering a single phase flow in the laminar, transition and turbulent regimes: Re= 80 to 90000 and Pr= $2.8 \sim 150$. They correlated the experimental results for the friction factor for Reynolds number from 2000 to 30000 as:

$$f = 5.76(e/d)^{0.95} (p/d)^{-1.21} \operatorname{Re}^{-0.217}$$
(2.1)

where f is the friction factor, e is the wire diameter, p is the pitch, d is the tube diameter and Re is the Reynolds number. For the heat transfer enhancement, they obtained the correlation within a Reynolds number range from 1700 to 80000 and Prandtl number range from 2.5 to 170:

$$Nu = 0.303 (e/d)^{0.12} (p/d)^{-0.377} \text{ Re}^{0.72} \text{ Pr}^{0.37}$$
(2.2)

Here Nu is the Nusselt number, and Pr is the Prandtl number.

Yukitsugu Shoji et.al [19] performed experiments to study heat transfer enhancement in a round tube using a wire coil as a tabulator. They presented their experimental results with the following correlation in the range of Reynolds number from 1800 to 10000:

$$Nu = 0.103 \cdot K^{0.568} \cdot \text{Re}^{0.723} \cdot \text{Pr}^{1/3}$$
(2.3)

$$K = \left[N \cdot l + (N-1) \cdot p \right] / L \tag{2.4}$$

Where N is the number of the coils, l is the length of wire coil, L is the length of the tube and p is the pitch. Obviously, the diameter of the coil was not considered in their correlation.

2.5 Critical heat flux

Boiling is an effective method of extracting heat. To better understand boiling one can consider a generic representation of boiling such as shown in Figure 2.10.



Figure 2.10 Boiling curve

The onset of nucleate or free boiling begins when convection from the free surface is insufficient to dissipate the heat load. Liquids of low thermal conductivity with thin thermal boundary layers (i.e. high Prandtl numbers) are more readily susceptible to boiling even at low heat fluxes. Water is an example of such a liquid. Thus, as the heat flux is increased, some of the liquid water at the interface is converted to vapor which then forms bubbles. As the heat flux is increased, a critical or peak heat flux is reached, which is known as the critical heat flux (CHF). At this point the solid surface is about 60°C hotter than 100°C for water at 1 atm. An increase in the heat flux beyond this point quickly gives rise to film boiling, which is characterized by the formation of a vapor layer over the heat source. The result is that there is a dramatic drop in the heat load that can be dissipated, which in turn causes the temperature of the solid interface to increase rapidly until a minimum heat flux is reached (assuming the heat source is of sufficient temperature). As the solid surface heats further, the heat flux again begins to increase because of the increase in heat transfer by radiation across the vapor film. Visually, the
behaviors of liquid and vapor in the nucleate boiling and the film boiling regimes can be shown as in Figure 2.11.



Figure 2.11 Visualization of nucleate and film boiling: (a) Nucleate boiling-the isolated bubble region; (b) Nucleate boiling-the slugs and columns region; (c) Film boiling

It is important to mention that although the film boiling limitation of the McGill heat pipe has been raised as compared to an equivalent classical heat pipe, this limitation nevertheless still exists. In other words, the CHF is raised to some extent but it is not infinitely large. At sufficiently high heat fluxes, the liquid working fluid still cannot wet the evaporator wall and film boiling occurs. The film boiling limitation is less important for working fluids with low Prandtl numbers (Pr) such as sodium, than those with higher Prandtl numbers such as water.

2.5.1 The mechanisms of CHF

The mechanism of the critical heat flux phenomenon has been the subject of widespread interest and controversy. Katto [20], Dhir [21], and Bergles [22] presented reviews relating to the mechanism of pool boiling. Hewitt [23], Tong and Tang [24],

Collier and Thome [25], Carey[26] and Katto [27] gave detailed reviews of critical heat flux mechanisms in forced convective boiling. The postulated mechanisms can be approximately classified into the following types:

1. Hydrodynamic instability mechanism.

Here, instabilities occur in the vapor-liquid interfaces leading to the breakdown of the vapor release mechanisms and to vapor accumulation at the surface, leading to critical heat flux. This mechanism was originally suggested by Zuber [28],[29]. Zuber postulated that provided vapor can escape from the layer of bubbles near the surface, thus preventing it from becoming too thick, the liquid phase can penetrate the layer, wetting the surface and preventing the overheating which would lead to the critical heat flux phenomenon. He suggests that these columns occur because the vapor-rich layer adjacent to the surface is fundamentally unstable, i.e., a small disturbance in the interface between the layer and the surrounding liquid is amplified at a rate that depends on the wavelength of the disturbance λ . Zuber hypothesized that a rectangular square ray of jets was formed with a pitch λ .

Eventually, the velocity of vapor in the jets becomes so large that the jets themselves become unstable near the interface as a result of Helmholtz instability. The breakup of the jets destroys the efficient vapor-removal mechanism, increases vapor accumulation at the interface, and leads to liquid starvation at the surface and to the critical heat flux phenomenon. If jet breakup occurs at a vapor velocity U_H within the jets, the critical heat flux is given by:

$$q_{crit}'' = \rho_g H_{fg} \frac{A_j}{A} U_H$$
(2.5)

where A_j is the area occupied by the jets and A is the total surface area. Helmholts instability theory gives U_H as:

$$U_{H} = \left(\frac{2\pi\sigma}{\rho_{g}\lambda_{H}}\right)^{\frac{1}{2}}$$
(2.6)

where σ is the surface tension.

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2. Bubble crowding at a heated surface

In this mechanism, close packing of bubbles leads to a reduction in the heated surface area that is in contact with the liquid phase and, hence, to a fall in heat flux with increasing wall superheat. Dhir and Liaw [30] measured void fraction as a function of distance from the wall. The void fraction passed through a maximum value and this maximum depended on the heat flux.

For well-wetting fluids, the peak void fraction reaches a value close to unity at the critical heat flux condition, and this implies that no liquid may be transferred to the surface under these conditions. For systems with larger contact angles, the peak void fraction was lower than unity, its value at the critical heat flux condition decreasing continuously with increasing contact angles.

3. Hot-spot heating

This mechanism has been investigated by Unal et al. [31]. On the departure of the vapor mushroom in the fully developed boiling region, the surface is wetted by fresh liquid and the vapor stems are recreated. At the bottom of the stems, a liquid micro-layer is evaporated; the base of the stem is dry and its center temperature rises quickly as a result of the continued heating of the wall. When the next vapor mushroom departs, the dry zone at the base of the vapor stem is rewetted provided that the temperature it has reached during the mushroom hovering period is low enough to allow such rewetting. If the temperature is too high, then rewetting is inhibited and the hot spot becomes permanent and may grow, resulting in the CHF.

4. Film dry-out

In the annular flow regime, the critical heat flux condition is reached as a result of film dry-out. The film dries out because of the entrainment of droplets from its surface and as a result of evaporation, and despite the re-deposition of droplets counteracting the effect of the entrainment. As the heat load to the channel is increased, annular flow begins, and the flow rate of the annular liquid film decreases with further increases in heat input.

Eventually, the film flow rate reaches zero at a point corresponding to the critical heat flux (burn out).

It is probable that all of the above mechanisms play a role, their influence depending on the flow and thermodynamic conditions within the channel.

2.5.2 The correlations for CHF

In the published literature many researchers have presented their correlations to predict the CHF under specific conditions. M.A.Islam and M.Monde [32] presented the correlation for prediction of CHF in a vertical concentric annular passage. They found the following:

$$\ddot{q}_{cr} = \rho_{v} H_{fg} \sqrt[4]{\frac{\sigma(\rho_{f} - \rho_{v})g}{\rho^{2}_{v}}} \times 0.64 \times \frac{D_{i}}{4L} \left(\frac{\rho_{f}}{\rho_{v}}\right)^{0.13} \times \left\{1 + 0.803 \left(\frac{L}{\sqrt{\sigma/g(\rho_{i} - \rho_{v})}}\right)^{0.5} \frac{d_{i}}{D_{i} + d_{o}}\right\}$$
(2.7)

L.S. Tong and Y.S. Tang discussed flow boiling crisis [24] and introduced many correlations for specific working fluids in different conditions. The parameter effects on CHF in experiments and the operating parameter correlations for CHF predictions in reactor design were also discussed.

L. Cheng and G. Xia [33]did the experiments on CHF with a smooth tube and a four-head, spirally, internally ribbed tube. They concluded that CHF in the spirally, internally ribbed tube could be enhanced by a factor of 1.5 as compared with that in the smooth tube at 10, 17 and 19MPa, but the CHF could not be enhanced near the critical pressure in the spirally internally ribbed tube.

Y. Katto [27] presented an empirical correlation based on Tong's correlation to predict the CHF of water in the pressure range of 0.1-20MPa.

G.P. Celata, K.Mishima and G.Zummo [34] presented an analytical model for the prediction of CHF in water saturated flow boiling in round, vertical and uniformly heated pipes. The CHF is assumed to occur in annular flow when the liquid film vanishes at the exit section of the heated channel. In the calculation of the entrainment of liquid flow, two mechanisms, liquid-vapor interfacial waves and boiling in the liquid film were considered.

J. Boscary, J. Fabre, and J. Schlosser [35] provided correlations for predicting CHF. In their paper the CHF was reached when the length of the vapor blanket achieved the critical wavelength of the Kelvin-Helmholtz instability at the liquid-vapor interface.

J.C. Sturgis and I. Mudawar [36] predicted the CHF in long, rectangular channels. They put forward the concept of lift-off heat flux at the critical heat flux and test results performed with FC-72 liquid were obtained.

There are many other researchers who have presented their correlations or models to predict CHF in their specific conditions, however, a model to accurately predict the CHF in a situation such as the evaporator of the McGill heat pipe is not yet available from the literature.

2.6 The working materials for intermediate temperature range

The selection of the working fluid should be based on thermodynamic considerations which are concerned with the various limitations to heat flow occurring within the heat pipe. Table 2.1 from P. D. Dunn and D.A. Reay's book [4] lists the working fluids in different temperature ranges.

Medium	Melting point °C	Boiling point °C	Useful range °C
Helium	-272	-269	-271~ -269
Nitrogen	-210	-196	-203~ -160
Anmonia	-78	-33	-60~ 100
Freon 11	-111	24	-40~120
Pentane	-130	28	-20~120
Freon 13	-35	48	-10~100
Acetone	-95	57	0~120
Methanol	-98	64	10~130
Flutec PP2	-50	76	10~160
Ethanol	-112	78	0~130

Table 2.1 Heat pipe working fluids

Heptane	-90	98	0~150
Water	0	100	30~200
Toluene	-95	110	50~200
Flutec PP9	-70	160	0~225
Thermex	12	257	150~395
Mercury	-39	361	250~650
Caesium	29	670	450~900
Potassium	62	774	500~1000
Sodium	98	892	600~1200
Lithium	179	1340	1000~1800

In reality, water has been used as a working fluid in the lower temperature portion (30~200°C), and alkali metals in the higher range (>500°C). Water is the optimal working fluid when the temperature is below 200°C, but its saturated pressure above 200°C (>15.5 atm) requires thicker walls for safety reasons. Also, its critical point (374.15°C,221.2 atm) restricts its high temperature working range. The alkali metals such as cesium, potassium, and sodium are suitable working materials at higher temperatures (>500°C), however, when the temperature is decreased below 500°C, the vapor pressure and vapor density are decreased enough so that the vapor sonic velocity limits the heat transfer. The temperature range of 200~500°C has long been the difficult range for heat pipes. William G. Anderson [37] presented a few potential working fluids for this intermediate temperature range. In their paper, they summarized the requisite properties of potential heat pipe fluids for intermediate temperatures:

- High latent heat of evaporation
- Melting point below ~400K
- Critical point above ~800K
- Chemically stable in the above temperature range, i.e. no ionization and dissociation
- Acceptable liquid viscosity
- High surface tension
- Non-toxic, non-carcinogenic
- Non-volatile/low volatility

Organic fluids

Organic fluids such as aniline (C₆H₇N), naphthalene (C₁₀H₈), phenol (C₆H₆O),

toluene ($C_6H_5CH_3$), were introduced as working fluids by Saaski and Tower [38]. Table 2.2 shows some properties of the organic fluids. From the table we can see that the fluids have good boiling points, but the critical temperatures limit the upper working range. During operation, the organic fluids are likely to decompose and generate non-condensable gas when heated. This is typically not acceptable for long term operation.

		Melting point °C	Boiling temp. °C	Critical temp. °C	Critical pressure MPa
Aniline	C ₆ H ₇ N	-6°C	185°C	426°C	5.3
Naphthalene	C10H8	81°C	218°C	512°C	11.6
Phenol	C ₆ H ₆ O	41°C	182°C	421°C	6.13
Toluene	C ₆ H ₅ CH ₃	-95°C	114°C	319	4.1

Table 2.2 Some organic fluids of heat pipes

Halides

A halide is a compound of the type MX, where M may be another element or organic compound, and X may be fluorine, chlorine, bromine, or iodine. Unfortunately, the physical property data for many of the halides are incomplete. The known properties of one halide in a family may be used to estimate the properties for related halides, i.e. use the AlCl₃ data to estimate AlBr₃ and AlI₃ properties. In some cases, there are no data for any fluids in a particular family. In the literature the halides were not considered further, since there were no data (typically surface tension and/or liquid viscosity) for any member of the family.

Iodine

An iodine heat pipe was tested by Rosenfeld and Lindemuth [39] with a 304SS wall. With melting point at 114°C, boiling point at 185°C, critical temperature at 512°C and critical pressure of 11.6MPa, iodine was reported to work very well in the traditional heat pipe. However, when the working temperature is higher than 185°C, the pressure in the heat pipe is positive, and iodine vapor is extremely hazardous, which raises the safety problem in high pressure cases. The other major unknown with iodine is long term

corrosion, since it is a very reactive fluid.

Sulfur

Sulfur was originally proposed by Polasek and Stulc [40] as a working fluid in heat pipes. They constructed a number of stainless steel/sulfur heat pipes having an outside diameter of 3.2cm and a length of 2m, of which 0.5m was used as the evaporator section and 1m as the condenser. The viscosity of sulfur is very temperature sensitive. It melts at 119 °C, the viscosity at first decreases (in the usual fashion for liquids) from $0.0017Nsm^{-2}$ at about 120 °C to a minimum value at 157 °C, but above this temperature the viscosity rises extremely steeply to a maximum value of 93 $Pa \cdot s$ at 186-188 °C, at which point the melt is very viscous and dark red. When heated further, the viscosity decreases once again. In Anderson's 2005 report [41], they claimed "The viscosity of sulfur is too high in this temperature range to make it a candidate fluid". The viscosity makes pure sulfur unsatisfactory as a heat pipe working fluid.

Sulfur/iodine

When a small quantity of iodine is added to the sulfur, the sulfur viscosity decreases dramatically. Around the peak point $(200 \, \text{C})$, the viscosity of pure sulfur is 1000 times that of a sulfur 3% iodine mixture (this will be discussed in detail in Chapter 5). Anderson had reported [41] that a sulfur/iodine heat pipe can be successfully operated.

2.7 Modeling of heat pipes

In the literature there are many papers about the modeling of conventional heat pipes, in which a liquid film and a vapor flow exist in a counter-current annular flow. J.G. Reed [42] presented a comprehensive model to predict the steady-state and transient performance of the two-phase closed thermosyphon, which was essentially a gravity-assisted, wickless heat pipe.

M.Monde's model [43] showed that the CHF point on the heat pipe corresponded

to a maximum falling liquid rate.

M. S. El-Genk et al. [44] developed a one-dimensional steady-state model to predict the operation envelope and performance isotherms of closed two-phase thermosyphons.

M. Mahfoud, F.Mucciardi and J.Gruzleski [45] did the numerical simulation of heat pipe assisted solidification. A two-dimensional transient model was written from first principles and used to simulate the solidification and heat transfer aspect of an aluminum sample inside the core of a heat pipe.

S. I. Haider, Y. K. Joshi, W. Nakayama [46] developed a model for two-phase flow and heat transfer in the closed loop thermosyphon involving co-current natural circulation. When they were calculating the pressure drop and heat transfer in the evaporator, they adopted a homogeneous two-phase flow model, which does not accurately describe the two-phase flow in the evaporator.

In a McGill heat pipe, the heat flux, the swirling flow, the two-phase pressure drop, the return flow rate, the cooling flow rate and the operating temperature are all related. A change of one factor will affect the operating state of the system. For example, a change of the cooling flow rate will lead to a change in the saturation temperature in the condenser and the evaporator. The saturation pressure will also be changed correspondingly, which will influence the vapor velocity, the two-phase pressure drop and the return flow rate. So far a theoretical model to predict the working parameters of a McGill heat pipe is not available.

Chapter 3 Air-water simulation of two-phase flow in a McGill heat pipe

3.1 Introduction

Current industrial cooling technologies such as conventional forced-convection water cooling can be unsafe when incorrectly operated. To maintain comfortable safety margins, such systems are operated with large safety factors and are often grossly inefficient from an energy standpoint. It is clear that thermally intense applications can benefit from heat pipe evaporative cooling because of the increased safety and energy efficiency of such a device.

It is well established in the literature that the most significant limitations of classical heat pipes are the capillary, entrainment and film boiling limitations. These limitations together with an unwillingness to use hazardous working fluids have largely prevented the use of heat pipes in high-heat flux and high-heat load applications.

Similar to a classical heat pipe, the McGill heat pipe consists of an evaporator section that is subjected to a heat load, and a condenser section that is cooled externally. The interior of the device is sealed and contains only a working fluid (water, sulfur or sodium) existing as both liquid and vapor. Vapor is formed at the evaporator and travels towards the cooled condenser where it condenses. The liquid working fluid is then returned to the evaporator by gravity via a central return line. The return line shields the returning liquid from being entrained by the shear of the rising vapour, similar to a closed-loop thermosyphon.

It should be noted that unlike the classical heat pipe, the *McGill heat pipe* is co-current in nature. Liquid working fluid is discharged from a return line at the bottom end of the evaporator. As vapor begins to form in this region, excess liquid is entrained

upwards with the vapor co-currently.

The wick of the classical heat pipe has been replaced by a helical-shaped flow modifier in the evaporator section of a McGill heat pipe. As the two-phase working fluid rises through the evaporator, the flow modifier imposes a resistance causing the working fluid to rotate as it rises. The centrifugal forces that develop propel the liquid towards the evaporator wall, ensuring homogenous wetting and suppressing film boiling.

Results from laboratory and industrial-scale testing indicate that dry-out is the main limitation of McGill heat pipes. Therefore understanding how the evaporator configuration affects the two-phase flow resistance is of paramount importance in the continued development of this device. The purpose of this chapter is to provide a general understanding of the fluid flow within a McGill heat pipe, and to describe the effect of the helical flow modifier on the flow characteristics of the vapor-liquid two-phase flow.

3.2 Experimental apparatus and test section

A physical model of the McGill heat pipe was designed to study the fluid flow,. The model focused on the fluid mechanics. There was no heat transfer. The vapor phase that is formed in a heat pipe was simulated by injecting air into the model. The working liquid was simulated with water.

A semi-closed test loop of air-water, two-phase flow is schematically illustrated in Figure 3.1. The compressed air passes through a valve (1) to the air flowmeter (2) (variable area, Omega FL4513). The air was injected into the tubular test section (plexiglass, outer diameter ϕ 40mm, thickness 2.5mm) via the air injector (3) shown in Figure 3.2, where 8 holes (ϕ 2mm) were bored evenly along the edge.

Water is fed into the test section through a return line originating from the reservoir (10). Thus the air is mixed with the water at the bottom of the test section and the two-phase flow moves upward to the head tank (17). Both the test section and the head tank were fabricated from transparent plastic pipe to allow for visual observations.

The flow modifier (Figure 3.3) is placed in the tube. The mixture of water/air expands in the head tank and is separated by the filter(16) at the top. The air exits the system into the environment while the water is collected into the reservoir.

The water flows from the reservoir to the bottom of the test section through one of two lines: either the return line where flow is induced solely by the gravity head (8), or the pumping line where the flow is enhanced by a pump (7). A transparent section of pipe was installed in the return line. During the experiments some particles (ϕ ~2.5mm green beans) were added into the reservoir and mixed with the water. A video camera (JVC GR-DVL920U) (9) was used to film the passing particles (Figure 3.4) at 30 frames/second. Consequently, by accounting for the particle sinking velocity, the return velocity of the water can be measured from the particle vertical position difference between frames. A light (11) is used to facilitate the photography, and a valve is used to control the returning flow rate of the water in the return line (returning velocity).

The pressure differentials between the bottom, the middle, and the top of the test section were measured by differential pressure transducers (PX2300,Omega)(12). Five flow modifiers were used in the experiments; their specifications given in Table 3.1.

Spring	1	2	3	4	5
Pitch(mm)	35	30	30	20	10
Coil diameter(mm)	3.5	3	5	5	3
Outer diameter(mm)	35	35	35	35	35

Table 3.1 Specifications of flow modifiers



(1)valve	(2) air flow rate meter	(3) air injector
(4) flow modifier	(5) test section	(6) pump
(7) pumping line	(8) returning line	(9) video camera
(10) reservoir	(11)light	(12) diff. pressure transducer
(13) signal processor	(14)24V power	(15)computer
(16) filter	(17) expanding section	

Figure 3.1 Schematic diagram of the experimental apparatus



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Figure 3.2 Air injector



Figure 3.3 Spring flow modifiers evaluated



Figure 3.4 Measuring the returning velocity by the positions of the particle in adjacent frames

The void fraction is an important parameter in two-phase flows. By abruptly closing the on-off valves in the return line and the compressed air pipe simultaneously, the volume of water left in the test section falls to the bottom, and can be measured. By correcting for the return line volume, it is possible to calculate the void fraction. One uncertainty of this method is that the inertia of the water causes a loss at the top part of the test section, the water being ejected into the expanding section and reservoir. This tends to exaggerate the void fraction measured. However, this error should be relatively small from visual observations, and is therefore ignored.

In a real McGill heat pipe, heat is added into the evaporator along its length causing the void fraction and quality to increase. It is difficult to measure the frictional and acceleration pressure drop of two-phase boiling flow in these devices. In the experimental setup the void fraction and quality are constant as there is no mass transfer in the test section. Therefore it is much easier to measure the gravitational and frictional pressure drop as a function of the specific void fraction.

By discretizing the evaporator of a real McGill heat pipe into many one dimensional elements and assuming the void fraction and quality as constant in each element, the results of the present study can be used in the analysis of pressure drop in every element. Therefore the total gravitational and frictional pressure drop can be determined by integration.

Regarding the acceleration pressure drop in the evaporator, the air-water simulation does not provide any information as no accelerating movement exists for the air and water flow. Future work will involve using CFD to study acceleration pressure drop in more detail.

3.3 Experimental results and discussion

3.3.1 Flow patterns

Traditionally, five different flow patterns are defined based on flow behavior: bubble, slug, churn, semiannular and annular flow. The flow modifier has a significant effect on the flow pattern, especially when the air flow is relatively high.

Figure 3. 5 and Figure 3. 6 show the flow patterns in the test section when the air flow rate was 0.236 *Nl/s* and 4.719 *Nl/s*, respectively.

In two-phase flow, the *Re* number is defined as:

$$\operatorname{Re}_{tp} = \frac{GD_h}{\overline{\mu}}$$
(3.1)

where Re_{ψ} is the two-phase Reynolds number, D_h is the hydraulic diameter (m), and the mass flux G ($kg/(m^2s)$) is given by:

$$G = \frac{m_l + m_g}{A} \tag{3.2}$$

The two-phase mean viscosity $\overline{\mu}$ ($Nm^{-2}s$) can be calculated by:

$$\frac{1}{\overline{\mu}} = \frac{x}{\mu_g} + \frac{1-x}{\mu_l}$$
(3.3)

In Equation (3.2), m_1 and m_g are the mass flow rate (kg/s) of water and air, A is the cross-section (m^2); In Equation (3.3), x is the quality.

When the Re number is small (as shown in Figure 3.5), the effect of the flow modifier on the flow pattern is not obvious. There is no clear difference between the cases with or without a spring inside. When the Re number is larger (Figure 3.6), the spring inside the tube causes the two-phase mixture to flow in a helical manner and the centrifugal force generated by the rotating movement drives the water towards the outer wall. This is in contrast to a smooth tube, where the two-phase mixture tends to flow axially upwards i.e. there is no helical movement developed.

Although the test section discussed does not include boiling phenomenon, it can be assumed that the observed flow patterns roughly correspond to those present in a McGill heat pipe. It is therefore theorized that the observed swirling flow in the test section also occurs in the corresponding heat pipe. The swirling flow enhances liquid convection heat transfer at the wall and helps explain why the film boiling limitation is significantly greater in McGill heat pipes relative to their equivalent classical counterparts. Navarra and et al [47] have presented some results that substantiate the increase in the film boiling limitation.



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Figure	2	5 Photograph	of flow natter	at 0.236 1/s air flow rate
riguie	Э.,	J FIIOLOgraph	of now patient	1 at 0.250 l/s air now rate



Figure 3.6 Photograph of flow pattern at 4.719 l/s air flow rate

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3.3.2 Pressure drop and returning velocity

The water in the test setup travels through the loop defined by the return line, the test section and the reservoir, as seen in Figure 3.1. The driving force causing the flow of liquid from the reservoir through the return line is the pressure head, $\rho_1 gh$. To cause flow, this pressure must overcome the friction loss and minor loss in the returning line ΔP_r , and also the pressure differential between the bottom of the test section P_2 and the reservoir P_1 . This can be expressed as:

$$P_1 + \rho_I g h = P_2 + \frac{1}{2} \rho_I V_r^2 + \Delta P_r$$
(3.4)

where P_1 , P_2 are the pressures at the inlet, outlet of the returning line, ρ_1 is the water density(kg/m^3), h is the vertical distance (m) between the reservoir and the bottom of the return, V_r is the velocity of the return, and ΔP_r is the pressure loss in the return line.

The total pressure drop gradient $(dp/dz)_{total}$ can be divided into three parts corresponding to the acceleration term $(dp/dz)_{ac}$, the frictional term $(dp/dz)_{fr}$, and the gravitational term $(dp/dz)_{g}$, respectively. Thus:

$$\left(\frac{dp}{dz}\right)_{total} = \left(\frac{dp}{dz}\right)_{ac} + \left(\frac{dp}{dz}\right)_{fr} + \left(\frac{dp}{dz}\right)_{gr}$$
(3.5)

z is the vertical coordinate.

In terms of void fraction α , the gravitational term in Equation (3.5) can be written as:

$$\left(\frac{dp}{dz}\right)_{gr} = -\left[\rho_l(1-\alpha) + \rho_g\alpha\right]g\tag{3.6}$$

Collier [48] presents an expression for the acceleration pressure drop due to the change in quality x (the fraction of the gas mass flow rate) as follows:

$$\left(\frac{dp}{dz}\right)_{ac} = -\frac{1}{A}\frac{dJ}{dz}$$
(3.7)

where *J* is the total momentum flux defined by:

$$J = \frac{m_g^2}{\rho_g A_g} + \frac{m_l^2}{\rho_l A_l} = \frac{m^2}{A} \left[\frac{x^2}{\alpha \rho_g} + \frac{(1-x)^2}{(1-\alpha)\rho_l} \right]$$
(3.8)

 A_g , A_l is the flow area (m^2) of the air phase and the water phase, while A is the total flow area (m^2) , and m is the total mass flow flux $(kg/(m^2s))$. For this particular experiment, the quality x and the void fraction α can be assumed as constant along the test section; consequently, the acceleration pressure drop is canceled out.

The frictional pressure drop is difficult to evaluate and is discussed in the homogeneous model and Lockhart-Martinelli Method (sections 3.4 and 3.5). The overall pressure gradient in the test section is therefore given by:

$$\left(-\frac{dp}{dz}\right)_{total} = \left[\rho_l(1-\alpha) + \rho_g \alpha\right]g + \left(\frac{dp}{dz}\right)_{fr}$$
(3.9)

The relationship between the total pressure drop and the air flow rate is presented in Figure 3.7. When the air flow rate is zero, no circulation develops in the setup, and the void fraction, α , is zero. In this condition there is no frictional pressure drop, and the total pressure drop is equal to the gravitational pressure drop.

As the air flow rate increases, the void fraction α increases as well, leading to a decrease in the gravitational pressure drop. However this is partially offset by an increase in the frictional pressure drop, caused by the increased air/water velocity in the test section.

Therefore in summary, when the air flow rate is small $(0\sim3Nl/s)$, the gravitational pressure drop is the dominant component of the total pressure drop. Conversely the friction pressure drop is the dominant component when the air flow rate is large. This explains why the total pressure drop first decreases with air flow rate and then increases, as seen in Figure 3. 7.

Also, when the air flow rate is small $(0\sim 1Nl/s)$, the variance between the pressure drops of the different flow modifiers is small. Conversely, when the air flow rate is

increased (6Nl/s), the variance in the pressure drop between each flow modifier increases. Thus, the flow modifier has a more pronounced effect on the pressure drop at greater air flow rates.

Figure 3.8 presents the relationship between the returning velocity and the air flow rate. The returning velocity is determined by Equation(3.4). When the air flow rate is zero, $\Delta P = P_2 - P_1 = \rho_1 gh$, and no water flows down the return line. Initially as the air flow rate increases and the pressure drop decreases, the returning velocity increases correspondingly. However as the air flow rate continues to increase and the frictional pressure drop becomes dominant, the returning velocity decreases. If the air flow rate is sufficiently high, the pressure drop reaches $\rho_1 gh$, and the water cannot flow down to the bottom of the test section.



Figure 3.7 Relationship between total pressure drop and air flow rate

In reality when the heat load applied to a McGill heat pipe is large enough, the working fluid cannot sufficiently reflux the evaporator. This condition is called the dry-out limitation and can lead to failure of the heat pipe. A possible method to resolve

this problem is to install a pump in the return line.



Figure 3.9 Pressure drop across the pump line with the pump on or off



Figure 3.10 Returning velocity across the pump line with pump on or off

Figure 3.9 and Figure 3.10 show the pressure drops and returning velocities in the pump line when the pump was on or off and spring 3 (d=5mm, H=30mm) was used in the test section.

When a pump is included in the return line, Equation (3.4) can be rewritten as:

$$P_{1} + \rho_{l}gh + \rho_{l}gh_{pump} = P_{2} + \frac{1}{2}\rho_{l}V_{r}^{2} + \Delta P_{r}$$
(3.10)

Figure 3.9 and Figure 3.10 indicate that when the pump is on, both the pressure drop $\Delta P = P_2 - P_1$ and the returning velocity V_r are increased. The returning velocity also tends to change very little with changes in the air flow rate. Therefore a pump can theoretically ensure a sufficient flow in the return line and help prevent dry-out.

3.3.3 Single phase pressure drop

To understand the effect of the flow modifier on the friction factor of two-phase flow, it is helpful to first investigate the friction factor in single phase flow. The single-phase frictional pressure drop in a smooth tube can be expressed as follows:

$$-\frac{dP}{dz} = f_s \frac{2G^2}{\rho D_h}$$
(3.11)

The friction factor f_s in the smooth tube can be evaluated using the following conventional forms:

$$f_{\rm s} = 16 \cdot {\rm Re}^{-1}$$
 Re < 2300 (3.12)

$$f_s = 0.0791 \cdot \text{Re}^{-0.25}$$
 Re > 2300 (3.13)

The Reynolds number is defined as:

$$\operatorname{Re} = \frac{GD_{h}}{\mu}$$
(3.14)

Compared with the smooth tube, the frictional pressure drop with the presence of a flow modifier is increased because of the more pronounced rotational component of the flow. A non-dimensional parameter, *Sw* (swirler parameter) can be defined as:

$$Sw = \frac{\pi (D_o - e)e}{p \cdot D_h} \tag{3.15}$$

The physical meaning is an area ratio. The numerator is the horizontal cross section area occupied by the flow modifier; the dominator is the vertical section area in a single pitch.

The friction factor with the flow modifier can be expressed as:

$$f = f_s (1 + a_1 \cdot Sw^{a_2}) \tag{3.16}$$

where a and b are constants and are experimentally determined.

Figure 3.11 and 3.12 present the relationship between the friction factor and the Reynolds number in the turbulent regime (2300 < Re < 30000) and laminar regime (700 < Re < 2300), respectively. The results indicate that the swirler parameter and friction factor increase with a larger coil diameter *e* and a smaller coil pitch *p*. By using the method of least squares, the constants a_1 and a_2 can be calculated, and are determined as 12.1 and 1.13 respectively. In summary, the experimental results indicate that the friction factor of single-phase flow with the presence of the flow modifier is expressed as:

$$f = f_s \left(1 + 12.1 \times \left(\frac{\pi (D_o - e)e}{p \cdot D_h} \right)^{1.13} \right)$$
(3.17)

The predicted values of the friction factor are shown as solid lines in Figure 3. 13. Figure 3. 13 shows the comparison between the experimental and the predicted values of the friction factor with an error range of $\pm 20\%$.



Figure 3.11 Friction factor of water single-phase flow (turbulent flow)



Figure 3.12 Friction factor of water single-phase flow (laminar flow)



Figure 3.13 Comparison between the experimental and the predicted values of friction factor

3.3.4 Homogeneous flow model

The homogeneous flow model is used for two-phase flow, where it is assumed that the two phases are well mixed and therefore travel with the same velocities at any point in space. The two-phase mixture is treated as a single phase with mean fluid properties that are functions of the corresponding liquid and gas properties and of the quality of the fluid.

According to Equation (3.9), the friction pressure drop can be calculated as:

$$-\left(\frac{dp}{dz}\right)_{fr} = \left(\frac{dp}{dz}\right)_{total} + \left[\rho_l(1-\alpha) + \rho_g\alpha\right]g$$
(3.18)

In the homogeneous model, the frictional pressure drop can also be evaluated by:

$$-\left(\frac{dp}{dz}\right)_{fr} = f_{tp} \frac{2G^2}{\overline{\rho}D_h}$$
(3.19)

where f_{tp} is the effective two-phase friction factor and the mean two-phase density $\overline{\rho}$ is defined as a function of water density, air density, and quality as follows:

$$\frac{1}{\rho} = \frac{x}{\rho_g} + \frac{1-x}{\rho_l}$$
(3.20)

The quality x is defined as:

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} \tag{3.21}$$

The mass flux $G(kg/(m^2s))$ is given by Equation (3.2). In the homogeneous model the slip ratio between the two phases is assumed to be unity everywhere so that the void fraction is expressed as:

$$\alpha = \frac{x\rho_l}{\rho_g(1-x) + \rho_l x}$$
(3.22)

The Reynolds number for the two-phase flow is defined as Equation (3.1), where the two-phase mean viscosity can be calculated by Equation (3.3). Therefore the frictional pressure drop calculated using the homogeneous model in Equation (3.19) can be compared to the experimental results using Equation (3.18).



Figure 3.14 Comparison of experimental and theoretical frictional pressure drop when H=30mm d=5mm



Figure 3.15 Comparison of experimental and theoretical frictional pressure drop when H=35mm d=3mm

Figure 3.14 and 3.15 present the relationship between the frictional pressure drop and the two-phase Re number when H=30mm d=5mm and H=35mm d=3mm. At a constant air flow rate, the returning valve is adjusted to one of five settings and the two-phase Re number is calculated accordingly. From the results, it can be concluded that the homogeneous model is not suitable for predicting the frictional pressure drops observed in the experiments.

The homogeneous model assumes that the two phases (air and water) are perfectly mixed. However the rotational component of the flow tends to separate the water and air from the visual observations, the water tending to move towards the outside and the air tending to move towards the center. Because of the larger density of the water, the friction pressure drop of the actual system should therefore be larger compared with the homogeneous model. In addition, there is no doubt that some slip exists between the two phases in the real system, also contributing to a larger frictional pressure drop. These two facts help explain the consistent under-evaluation of the frictional pressure drop by the model compared with the experimental results, as seen in Figure 3.14 and 15.

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In summary, the homogeneous model is relatively simple to employ but was not satisfactory in predicting the frictional pressure drops observed experimentally.

3.3.5 Modified Lockhart-Martinelli Method

3.3.5.1 Correlation of void fraction

To determine the void fraction α , the Zuber-Findlay model [49] of drift flux is used.

$$\alpha = \frac{V_g}{C_0 (V_l + V_g) + V_{gj}} \quad or \quad \frac{V_g}{\alpha} = C_0 (V_l + V_g) + V_{gj}$$
(3.23)

 V_g and V_l are defined as the local "volumetric flux densities" (or local superficial velocities) of air and water respectively, and are given by:

$$V_g = \frac{m_g}{\rho_g A} \tag{3.24}$$

$$V_l = \frac{m_l}{\rho_l A} \tag{3.25}$$

 C_0 is the distribution parameter and V_{gi} is the mean drift velocity (*m/s*) which accounts for the relative velocity between the phases. P.B. Whalley [50] introduced the distribution parameter and the following drift velocity for vertical annular duct flow:

$$V_{gj} = 0.35(g \cdot D_h)^{1/2}$$
(3.26)

With $C_o = 1.2$. Therefore the friction pressure drop can be recalculated with Equation (3.18).

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<u> </u>	Pitch(mm)	Diameter(mm)	C	V_{g}	Error range
Spring 1	35	3.5	1.21	0.169	±12%
Spring 2	30	3	1.21	0.169	±12%
Spring 3	30	5	1.205	0.169	±14%
Spring 4	20	5	1.19	0.169	±13%
Spring 5.	10	3	1.19	0.169	±14%

Figure 3.16, 3.17 and 3.18 show the constant C_0 in the drift flux model for plots of the relationship between $(V_g + V_l)$ and V_l/α . Considering the drift velocity between the air and the water as constant, the distribution parameter C_0 is regressed according to the experimental results. This analysis indicates that the drift flux model with constant C_0 overestimates void fraction at low mixture velocity (bubble flow), and underestimates void fraction at highmixture velocity (annular flow).



Figure 3.16 C_0 in drift flux model d=3mm, H=30mm



Figure 3.17 C_0 in drift flux model d=5mm, H=20mm



Figure 3.18 C_0 in drift flux model d=5mm, H=30mm



Figure 3.19 Void fraction at different air flow rates

3.3.5.2 Lockhart-Martinelli method

It has been found convenient to use the two-phase flow friction multipliers developed by Lockhart and Martinelli [51]. They defined the Lockhart-Martinelli parameter X_n as follows:

$$X_{u}^{2} = \frac{\left(\frac{dp}{dz}\right)_{fl}}{\left(\frac{dp}{dz}\right)_{fg}}$$
(3.27)

where $(dp/dz)_{fl}$ and $(dp/dz)_{fg}$ are the pressure gradients due to friction if water or air flows alone, respectively.

The two-phase flow friction multiplier ϕ_L is defined as:

$$\phi_L^2 = \frac{\left(\frac{dp}{dz}\right)_{tp}}{\left(\frac{dp}{dz}\right)_{fl}}$$
(3.28)

where $(dp/dz)_{tp}$ is the friction pressure gradient for the two-phase flow.

Chisholm [52] recommended the following equation for predicting the frictional pressure drop during two-phase flow on the basis of the Lockhart-Martinelli parameter:

$$\phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{3.29}$$

where C takes the following values:

C=20 when flow of both the liquid and the gas is turbulent (turbulent-turbulent flow);

- C=12 when flow of the liquid is viscous and flow of the gas is turbulent (viscous-turbulent flow);
- C=10 when flow of the liquid is turbulent and flow of the gas is viscous (turbulent-viscous flow);

C=5 when flow of both the liquid and the gas is viscous (viscous-viscous flow).

Most of the experiments were performed in the viscous-turbulent flow regime (flow of the liquid is laminar and flow of the gas is turbulent), and C is taken as 12 for smooth tubes.

Depending on the results obtained for the single-phase flow and the void fractions measured in the experiments, we can calculate the Lockhart-Martinelli parameter X and

the two-phase multiplier ϕ_L . Figure 3.20 to 3.24 show the curves correlating X and ϕ_L by Equation (3.29) with varying values of C. It can be seen that the parameter C decreases when the swirler parameter (Sw) increases. So it is possible to obtain the correlation for C as a function of the swirler parameter.



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Figure 3.22 Spring 3 regressed constant C=6.2



Figure 3.23 Spring 4 regressed constant C=5.8



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Figure 3.24 Spring 5 regressed constant C=5.5



Figure 3.25 Relationship between Sw and C

Figure 3.25 shows the relationship between Sw and C. The following relationship is obtained by using the least-squares method.

$$C = 12(1 - 0.515Sw^{0.312})$$
(3.30)

Equation 3.30 describes the influence of the spring modifier on the constant C. Thus, by coupling Equation (3.30) and (3.29) one can predict the value of the two-phase friction multiplier ϕ_l .
Chapter 4 Theoretical model to predict CHF in the full tunnel McGill heat pipe evaporator

4.1 Introduction

Critical heat flux (CHF) appears as the peak in the boiling curve at the end of the nucleate boiling regime. It is one realistic limitation that a heat pipe should operate below the limit for it to be in a safe range. The existence of a flow modifier in the evaporator increases the magnitude of the CHF. The enhanced CHF depends on many factors, such as swirler parameters, the diameters of the evaporator and the returning pipe, and the working conditions. To develop an analytical model to predict the CHF in the evaporator with a spring flow modifier is very difficult because of the complicated two-phase flow state. In this chapter a specific flow modifier, i.e. a whole tunnel swirler, is considered to predict the CHF in a swirling flow.

4.2 Model description

The whole tunnel swirler between the returning pipe and the outer tube is shown as Figure 4.1. The theoretical model comprises four major components: a separated flow model, an instability analysis, a lift-off model and an energy balance model. To describe the two-phase heat transfer in the evaporator, they are all indispensable.



Figure 4.1 Full spiral tunnels in the evaporator

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Figure 4.2 The coordinate system and channel control volume

4.2.1 Separated flow

We define the coordinate system as one-dimensional, the x axis is the outer edge of the spring fin and the control volume is as shown in Figure 4.2.

Analysis of two-phase flow is often treated in an idealized method by assuming that the vapor and liquid phases are completely separated with a distinguished interface [34]. Due to boiling at the surface, mass is continually added to the vapor layer such that the mean vapor thickness increases along the flow direction. Due to the density disparity between the phases, the mean vapor velocity U_v tends to increase rapidly, surpassing the mean liquid velocity U_f . The velocity difference leads to the interfacial instability. In our case of flow in a spring channel heated on the outer wall, the vapor phase is adjacent to the heated wall and is supposed to have a mean thickness, δ , that increases in the streamwise direction. Even though a smooth surface between the vapor and liquid doesn't exist, the analysis can still provide information for the vapor and liquid phase.

The following assumptions are adopted:

1. Liquid and vapor are incompressible, the changes in physical properties are

negligible;

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- 2. The Pr number of the liquid is not very small;
- 3. Change of phase occurs only at the heated wall and no condensation happens at the interface;
- 4. Heat transfer from the spring fin to the liquid is neglected;
- 5. No condensation happens at the inner tube surface;
- 6. The liquid and vapor in the evaporator are at the same temperature, T_{sat} ;
- 7. Vapor generated at the wall is assumed to have no initial streamwise velocity;
- 8. Wall heat flux is uniform and constant along the evaporator wall.

From these assumptions we can conclude that the entire wall flux is used to convert saturated liquid to saturated vapor. So the continuity applied to a vapor control volume results in an expression for the average vapor velocity U_{y} .

$$U_{\nu} = \frac{q''x}{\rho_{\nu}\delta h_{fg}}$$
(4.1)

Applying mass conservation to the entire cross-section, we can obtain the average liquid velocity U_f .

$$\rho_f (H - \delta) U_f = \rho_f U H - \rho_v \delta U_v \tag{4.2}$$

Substituting (4.1) into (4.2), we get:

$$U_f = \frac{UH}{H - \delta} - \frac{q''x}{\rho_f (H - \delta)h_{fg}}$$
(4.3)

The momentum and force terms are shown for the vapor control volume in Figure 4.3 and for the full channel control volume in Figure 4.4. The interfacial shear(τ_i) is assumed to act solely in the streamwise direction and the wall shears($\tau_{w,v}$ and $\tau_{w,f}$) are accounted for on concave, convex and side walls. The pressure forces acting at the inner and outer walls are in the radial direction and do not contribute to the streamwise conservation equations. Also, vapor generated at the wall is assumed to have no initial streamwise velocity and consequently does not contribute to the streamwise momentum balance.



Figure 4.3 Momentum and force analysis for vapor phase control volume

Conversation of momentum applied to the vapor control volume (Figure 4.3) is:

$$\rho_{\nu}U_{\nu}^{2}A_{\nu} + \frac{\partial}{\partial\theta}\left(\rho_{\nu}U_{\nu}^{2}A_{\nu}\right)\frac{d\theta}{2} - \left[\rho_{\nu}U_{\nu}^{2}A_{\nu} - \frac{\partial}{\partial\theta}\left(\rho_{\nu}U_{\nu}^{2}A_{\nu}\right)\frac{d\theta}{2}\right] = \left(P_{\nu} - \frac{1}{2}\frac{\partial P_{\nu}}{\partial\theta}d\theta\right)\frac{\partial A_{\nu}}{\partial\theta}d\theta - \left(P_{\nu} + \frac{1}{2}\frac{\partial P_{\nu}}{\partial\theta}d\theta\right)\frac{\partial A_{\nu}}{\partial\theta}d\theta + F_{\nu} - \frac{\partial}{\partial\theta}F_{\nu}\frac{d\theta}{2} - \left[F_{\nu} + \frac{\partial}{\partial\theta}F_{\nu}\frac{d\theta}{2}\right] \quad (4.4)$$
$$-\tau_{i}(R_{o} - \delta)d\theta W - \left[\tau_{w,\nu}R_{o}d\theta W + 2\tau_{w,\nu}\left(R_{o} - \frac{\delta}{2}\right)d\theta\delta\right]$$

Since the vapor layer is thin and of very low density, it is reasonable to consider that the vapor pressure is constant with respect to r, So we have:

$$F_{\nu} \approx P_{\nu}W\delta \tag{4.5}$$

Also we have:

$$\frac{\partial}{\partial \theta} = R_o \frac{\partial}{\partial x}, \qquad A_v = W\delta, \qquad A_f = (H - \delta)W, \qquad (4.6)$$

and

$$\frac{\delta}{R_o} \ll 1, \quad \frac{H - \delta}{2R_i} \ll 1 \tag{4.7}$$

Hence, equation (4.4) can be simplified as:

$$-\frac{dP_{\nu}}{dx} = \frac{\rho_{\nu}}{\delta} \frac{d}{dx} \left(U_{\nu}^{2} \delta \right) + \tau_{w,\nu} \left(\frac{1}{\delta} + \frac{2}{W} \right) + \frac{\tau_{i}}{\delta}$$
(4.8)

Conversation of momentum applied to the entire channel control volume (Figure 4.4) is:

$$\frac{\partial}{\partial \theta} \left(\rho_f U_f^2 A_f \right) d\theta + \frac{\partial}{\partial \theta} \left(\rho_v U_v^2 A_v \right) d\theta = -\frac{\partial}{\partial \theta} \left(F_f \right) d\theta - \frac{\partial}{\partial \theta} \left(F_v \right) d\theta - 2\tau_{w,f} \left(R_o - \frac{H - \delta}{2} \right) (H - \delta) d\theta - \tau_{w,v} R_o W d\theta - 2\tau_{w,v} (R_o - \frac{\delta}{2}) \delta d\theta - \rho_f g \sin \varphi (H - \delta) \left(R_o - \frac{H - \delta}{2} \right) d\theta \cdot W - \rho_v g \sin \varphi \delta \left(R_o - \frac{\delta}{2} \right) d\theta \cdot W$$

$$(4.9)$$



Momentum termsForce termsfor channel control volumefor channel control volume

Figure 4.4 Momentum and force analysis for channel control volume

Here
$$\varphi$$
 is defined as $\varphi = \arcsin\left(\frac{h}{n \cdot 2\pi((R_o - R_i)/2)}\right)$

To calculate the force F_f , we can use the integration form:

$$F_f = \int_{R_i}^{R_o - \delta} P_f(r) W dr \tag{4.10}$$

From Euler's equation $\frac{u^2}{r} = \frac{1}{\rho} \frac{dp}{dr}$ and at the interface $r = R_o - \delta$, $P_f = P_i$, and so we

end up with:

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$$F_f = P_i(H-\delta)W - \rho_f U_f^2 W \left[(H-\delta) - R_i \ln \frac{R_o - \delta}{R_i} \right]$$
(4.11)

With (4.11), (4.6) and (4.7), equation (4.9) can be simplified as:

$$-\frac{dP_i}{dx} = \left(-\frac{dP_i}{dx}\right)_A + \left(-\frac{dP_i}{dx}\right)_F + \left(-\frac{dP_i}{dx}\right)_C + \left(-\frac{dP_i}{dx}\right)_G$$
(4.12)

where

$$\left(-\frac{dP_i}{dx}\right)_A = \frac{1}{H}\frac{d}{dx}\left[\rho_v U_v^2 \delta + \rho_f U_f^2 (H-\delta)\right]$$
(4.13)

$$\left(-\frac{dP_i}{dx}\right)_F = \tau_{w,v} \left(\frac{1}{H} + 2\frac{\delta}{WH}\right) + \tau_{w,f} \left(\frac{1}{H} + 2\frac{H-\delta}{WH}\right) \frac{R_i}{R_o}$$
(4.14)

$$\left(-\frac{dP_i}{dx}\right)_C = \frac{\rho_f U_f^2}{H} \frac{(H-\delta)}{R_o - \delta} \frac{d\delta}{dx} + 2\frac{\rho_f U_f}{H} \left[-(H-\delta) + R_i \ln \frac{R_o - \delta}{R_i}\right] \frac{dU_f}{dx}$$
(4.15)

$$\left(-\frac{dP_i}{dx}\right)_G = \frac{\delta}{H}\rho_v g\sin\varphi + \frac{H-\delta}{H}\rho_f g\sin\varphi$$
(4.16)

The subscripts A, F, C and G represent acceleration, frictional, centrifugal and gravity force respectively.

The wall and interfacial shear terms can be evaluated as follows[53]:

$$\tau_{w,k} = \frac{1}{2} \rho_k U_k^2 \frac{f_k}{4} \tag{4.17}$$

$$\tau_{i} = \frac{1}{2} \rho_{\nu} \left(U_{\nu}^{2} - U_{f}^{2} \right) C_{f,i}$$
(4.18)

where k indicates the phase and i the interface. $C_{f,i}$ is a constant with a value of 0.5. The friction factor can be calculated by the following correlation [54]:

$$f_{k} = 0.079 \left(\frac{\rho_{k} U_{k} D_{h,k}}{\mu_{k}}\right)^{-0.25} + 0.0075 \left(\frac{D_{h,k}}{2R_{o}}\right)^{0.5}$$
(4.19)

Here D_{hk} is the phase hydraulic diameter.

$$D_{h,\nu} = \frac{2W\delta}{W+\delta} \tag{4.20}$$

$$D_{h,f} = \frac{2W(H-\delta)}{W+H-\delta}$$
(4.21)

The vapor and liquid layers are in mechanical equilibrium and, hence, the two gradients

are equal at all streamwise locations, i.e. $-\frac{dP_i}{dx} = -\frac{dP_v}{dx}$ we can get:

$$\left(-\frac{dP_i}{dx}\right)_A + \left(-\frac{dP_i}{dx}\right)_F + \left(-\frac{dP_i}{dx}\right)_C + \left(-\frac{dP_i}{dx}\right)_G = \frac{\rho_v}{\delta}\frac{d}{dx}\left(U_v^2\delta\right) + \tau_{w,v}\left(\frac{1}{\delta} + \frac{2}{W}\right) + \frac{\tau_i}{\delta} \quad (4.22)$$

After substitution of equation (4.13), (4.14), (4.15) and (4.16) into equation (4.22), we obtain the following first-order ordinary differential equation:

$$\frac{d\delta}{dx} = \frac{-\frac{2u_{v}q''}{h_{fg}}\frac{H-\delta}{H\delta} - \frac{2u_{f}R_{i}q''}{h_{fg}H(H-\delta)}\ln\frac{R_{o}-\delta}{R_{i}} + \tau_{w,v}\left(\frac{1}{H} + \frac{2\delta}{WH} - \frac{1}{\delta} - \frac{2}{W}\right) + \tau_{w,f}\left(\frac{1}{H} + \frac{2(H-\delta)}{WH}\right)\frac{R_{i}}{R_{o}} - \frac{\tau_{i}}{\delta} + \left(\frac{\delta}{H}\rho_{v} + \frac{H-\delta}{H}\rho_{f}\right)g \cdot \sin\varphi}{\rho_{v}u_{v}^{2}\frac{H-\delta}{H\delta} + \rho_{f}u_{f}^{2}\frac{R_{o}-H}{H(R_{o}-\delta)} - \frac{2u_{v}q''x}{h_{fg}}\frac{H-\delta}{H\delta^{2}} - \frac{2u_{f}R_{i}}{(H-\delta)^{2}}\ln\frac{R_{o}-\delta}{R_{i}}\left(\frac{\rho_{f}UHh_{fg}-q''x}{Hh_{fg}}\right)$$

(4.23)

So far we have equations (4.1) and (4.3) and the ODE (4.23), from which we can solve U_f , U_v and δ . These variables are functions of the physical properties of the system, the length, x and the heat flux.

4.2.2 Instability analysis for the critical point

The stability of a liquid-vapor interface can be determined from classical interfacial instability theory [53] once the local liquid and vapor velocities are known. To do this we can use the separated flow model.

Under the assumption of incompressible, irrotational flow, the velocity field within each phase may be determined by employing potential flow theory for liquid and vapor velocity potentials. In our case, the problem is linearized by expressing each velocity component as the sum of a mean value and a small perturbation from the mean. Since there is no mean motion in the radial direction, the radial velocity component is just the perturbation term. The mean potentials are easily constructed to recover the mean components, as determined from the separated flow model.

Solution of the perturbation potentials requires the imposition of velocities at the inner and outer tube walls as zero. The procedure for analyzing flow stability is to first solve for the perturbation potentials by applying the boundary conditions. Then the total velocity potentials will be known. These potentials then yield the velocity field which is used in the Bernoulli equation to determine the pressure field. This allows for the calculation of the pressure difference across the interface which is responsible for curvature and, consequently, interfacial instability. The pressure difference leads to an equation for interfacial wave speed which then provides the criterion for instability.



Figure 4.5 Wavy interface between liquid and vapor

The interface (as shown in Figure 4.5) has been assumed periodic which is the real part of some complex function:

$$\eta(\theta) = real \left\{ \eta_0 e^{ik\theta R} \right\}$$
(4.24)

When allowing for a complex wave speed $c = c_r + ic_i$, it is expressed as:

$$\eta(\theta,t) = \eta_0 e^{ik(\theta R - ct)} = \eta_0 e^{c_i k t} e^{ik(\theta R - c_i t)}$$

$$\tag{4.25}$$

where η_0 is the maximum amplitude of the sinusoidal perturbation, which is δ . k is the wave number and is expressed as: $k=2\pi/\lambda$.

Only the real part of $\eta(\theta, t)$ describes the actual interface; the imaginary part will provide information regarding wave stability.

The complex potential for the vapor and the liquid are[53] :

$$F_{\nu}(z) = -(U_{\nu} - c)z - \frac{\eta_0(U_{\nu} - c)}{\sinh(kH_{\nu})}\cos(kz + ikH_{\nu})$$
(4.26)

$$F_{f}(z) = -(U_{f} - c)z + \frac{\eta_{0}(U_{f} - c)}{\sinh(kH_{f})}\cos(kz + ikH_{f})$$
(4.27)

Then the complex velocity for the liquid and the vapor $U_{f,c} U_{v,c}$, neglecting η_0^2 , is given by:

$$U_{f,c}^{2} = \frac{dF_{f}}{dz} \cdot \frac{dF_{f}}{d\overline{z}} = (U_{f} - c)^{2} (1 + 2k\eta \coth(kH_{f})) \approx (U_{f} - c)^{2} 2k\eta \coth(kH_{f}) \quad (4.28)$$

$$U_{\nu,c}^{2} = \frac{dF_{\nu}}{dz} \cdot \frac{d\overline{F}_{\nu}}{d\overline{z}} = (U_{\nu} - c)^{2} (1 - 2k\eta \coth(kH_{\nu})) \approx -(U_{\nu} - c)^{2} 2k\eta \coth(kH_{\nu}) \quad (4.29)$$

By considering Bernoulli's equation at the interface with Eqs. 4.28 and 4.29, one gets:

$$\begin{cases} P_{f} + \frac{1}{2}\rho_{f}U_{f,c}^{2} + \rho_{f}g\eta = P_{f} + \rho_{f}(U_{f} - c)^{2}k\eta \coth(kH_{f}) + \rho_{f}g\eta = const \\ P_{v} + \frac{1}{2}\rho_{v}U_{v,c}^{2} + \rho_{v}g\eta = P_{v} - \rho_{v}(U_{v} - c)^{2}k\eta \coth(kH_{v}) + \rho_{v}g\eta = const \end{cases}$$
(4.30)

Solving the equation yields:

$$P_{f} - P_{v} = -\left[\rho_{f}'(U_{f} - c)^{2} + \rho_{v}'(U_{v} - c)^{2} + (\rho_{f} - \rho_{v})\frac{g_{n}}{k}\right]k\eta$$
(4.31)

Here ρ'_{v}, ρ'_{f} are the modified density terms:

$$\rho_f' = \rho_f \coth(kH_f) \tag{4.32}$$

and $\rho'_{\nu} = \rho_{\nu} \coth(kH_{\nu})$,

Now let's consider the pressure difference from another viewpoint. From point P

(4.33)

to Q (Figure 4.6), the pressure across a wavy interface may also be approximated by the product of surface tension and curvature:

$$P_{f} \delta s - P_{v} \delta s = \sigma \delta \theta \quad i.e \quad P_{f} - P_{v} = \sigma/R_{c}$$
(4.34)
Here 1/R_c is the curvature, which can be calculated as $\frac{\partial^{2} \eta}{\partial r_{c}^{2}} / \left[1 + \left(\frac{\partial^{2} \eta}{\partial r_{c}^{2}} \right)^{2} \right]^{3/2}$



Figure 4.6 The force balance between surface tension and pressure

With mild curvature $(\partial \eta / \partial x \ll 1)$, the pressure across the interface (as shown in Figure 4.6) is

$$P_{f} - P_{v} = -\sigma \frac{\partial^{2} \eta}{\partial (R\theta)^{2}} = -\sigma \frac{\partial^{2} \eta}{R^{2} \partial \theta^{2}} = -\sigma \eta_{0} e^{c_{i}kt} k^{2} e^{ik(\theta R - c_{r}t)}$$
(4.35)

Substituting equation (4.35) into equation (4.31) and equating the two expressions for pressure difference yields a quadratic expression for the wave speed c:

$$(\rho'_{f} + \rho'_{v})c^{2} - 2(\rho'_{f}U_{f,i} + \rho'_{v}U_{v,i})c + \left[\rho'_{f}(U_{f,i})^{2} + \rho'_{v}(U_{v,i})^{2} - (\rho_{f} - \rho_{v})\frac{g_{n}}{k} - \sigma k\right] = 0$$

$$(4.36)$$

Solving for the wave speed yields:

$$c = \frac{\rho_f' U_{f,i} + \rho_v' U_{v,i}}{\rho_f' + \rho_v'} \pm \sqrt{\frac{\sigma k}{\rho_f' + \rho_v'}} - \frac{\rho_f' \rho_v' (U_{v,i} - U_{f,i})^2}{(\rho_f' + \rho_v')^2} + \frac{\rho_f - \rho_v}{\rho_f' + \rho_v'} \frac{g_n}{k}$$
(4.37)

The expression for the interface, equation (4.25), indicates the conditions for stability. For $c_i < 0$, the amplitude diminishes leading to a stable interface, while if $c_i > 0$, the interface is unstable. The critical wavelength is defined for the condition of neutral

stability, i.e $c_i = 0$. An imaginary component of wave speed arises when the radicand in equation (4.30) is negative. Therefore, setting this term equal to zero results in the expression for critical wavelength, λ_{cr} .

$$k_{cr} = \frac{2\pi}{\lambda_{cr}} = \frac{\rho_f' \rho_v' (U_{v,i} - U_{f,i})^2}{2\sigma(\rho_f' + \rho_v')} + \sqrt{\left[\frac{\rho_f' \rho_v' (U_{v,i} - U_{f,i})^2}{2\sigma(\rho_f' + \rho_v')}\right]^2 - \frac{(\rho_f - \rho_v)g_n}{\sigma}}$$
(4.38)

If the body forces are negligible, then $g_n=0$ and the critical wavelength is given as:

$$\lambda_{cr} = \frac{2\pi\sigma(\rho'_{f} + \rho'_{v})}{\rho'_{f}\rho'_{v}(U_{v,i} - U_{f,i})^{2}}$$
(4.39)

4.2.3 Lift-off heat flux

CHF is postulated to occur when the momentum flux of vapor emanating from the surface overcomes the pressure force tending to maintain interfacial contact with the surface. The interface then detaches from the surface eliminating a wetting front which represents a path for heat transfer to the liquid. The heat fluxes in the remaining wetting fronts must increase to accommodate this loss with the eventual outcome being the lift-off of the interface at these locations as well. This lift-off of the interface and elimination of wetting fronts trigger the CHF condition.

An expression for this lift-off flux may be determined by equating the average pressure force acting to maintain interfacial contact with the vapor momentum tending to push the interface away, as shown in Figure 4.7. In equation (4.25), the real part (at t=0) describes the actual pressure difference:

$$P_{f} - P_{v} = -\sigma \delta \left(\frac{2\pi}{\lambda}\right)^{2} \cos \left(\frac{2\pi}{\lambda}x\right)$$
(4.40)

The force that opposes the momentum is approximated as the average pressure force existing over the length of the wetting front. Hence integrating equation (4.40) over the length $l = b\lambda$ centered at the wetting front yields an analytical expression for average pressure difference.

$$\overline{P_f - P_v} = \frac{4\pi\sigma\delta}{b\lambda^2}\sin(b\pi)$$
(4.41)



Figure 4.7 Average pressure difference and vapor momentum at wetting front

So,

$$\frac{4\pi\sigma\delta}{b\lambda^2}\sin(b\pi)A_l = \rho_v U_{v,n}^2 A_l \tag{4.42}$$

where A_1 is the area of the wetting front

i.e,

$$U_{\nu,n} = \sqrt{\frac{4\pi\sigma\delta}{\rho_{\nu}b\lambda^2}}\sin(b\pi)$$
(4.43)

The heat flux concentrated at the wetting front is to evaporate a mass of liquid into saturated vapor, which is expressed as:

$$q_{l}''A_{l} = \rho_{v}h_{fg}U_{v,n}A_{l} \tag{4.44}$$

Substituting equation (4.43) into equation (4.44), we have:

$$q_{I}'' = h_{fg} \sqrt{\frac{4\pi\sigma\delta\rho_{v}}{b\lambda^{2}}} \sin(b\pi)$$
(4.45)

Equation (4.45) provides a means for estimating the lift-off heat flux given the characteristic parameters λ , δ and b of the vapor layer.

4.2.4 Energy balance model

The energy balance is built with reference to the wavy vapor layer shown in Figure 4.8. The liquid contacts continuously with the heater over a distance x^* at the beginning. After that the wavelength increases along the flow direction, thus the wetting

fronts move further apart. Liquid wetting length also increases along the flow direction but remains a constant fraction of local wavelength. The energy balance model should account for this variation. We can divide the heating channel into n segments. This segmentation enables the energy balance to account for continuous stretching of the wavelength by using different average vapor lengths for each segment; essentially discretizing the heated length. Consequently, the length and the number of wetting fronts are accounted for in each segment.

The energy balance expresses the fact that the sum of energy dissipated at all wetting fronts, including the continuous wetting front, is equal to the energy dissipated by a flux acting over the entire heated surface. Thus,

$$q_{cr}'' = \frac{1}{L - x^*} \left(q_1'' l_1 n_1 + q_2'' l_2 n_2 + \dots + q_j'' l_j n_j + \dots + q_n'' l_n n_n \right)$$
(4.46)



Figure 4.8 Idealized vapor-liquid interface and heater segments

The number of wetting fronts is given by:

$$n_{j} = \begin{cases} \frac{L_{1} - x^{*}}{\lambda_{1}} & j = 1\\ \frac{L_{j}}{\lambda_{j}} & j = 2 \cdots j \cdots n \end{cases}$$

$$(4.47)$$

where L_j is the segment length.

Considering b = wetting front length /vapor wavelength = l_i / λ_i , we have:

$$\begin{cases} l_1 n_1 = b\lambda_1 \frac{L_1 - x^*}{\lambda_1} = b(L_1 - x^*) \\ l_j n_j = b\lambda_j \frac{L_j}{\lambda_j} = bL_j \qquad j = 2 \cdots n \end{cases}$$

$$(4.48)$$

Substituting the relations given in equation (4.48) into equation (4.46) and noting that the wetting front heat fluxes are given by $q_j'' = q_l''(x^*)$ yields a simplified expression for the estimation of the CHF:

$$q_{cr}'' = b \cdot q_1''(x^*) \tag{4.49}$$

Equation (4.49) states that the flux concentrated at the wetting fronts is 1/b times the critical heat flux since the latter is taken as an average value acting over the entire heated surface. For the curved channel subjected to concave heating, b can be selected as 0.27[55].

To identifying the first wetting location x^* , we can use the following relationship:

$$x^* = x_0 + \lambda_{cr}(x^*)$$
 (4.50)

Where x_0 is the location along the channel where the phase velocities are equal i.e, $U_f = U_v$, based on the separated flow model calculations. Beyond this location, the vapor velocity continues to increase faster than that of the liquid and instability between the phases is more likely to take place. The vapor then assumes the wavy formation one critical wave length beyond this point where the continuous wetting front ends. Since λ_{cr} is evaluated at x^* , which itself is an unknown, an iterative scheme must also be employed in determining x^* .

4.2.5 Solution Procedure

As described above, the model comprises four major components in the estimation of critical heat flux. The solution is not an explicit expression but rather a procedure requiring numerical integration along the spring heating channel and iterations at several levels.

After inputting parameters of the working liquid and dimensions of the heat pipe,

the solution process is initiated by guessing a value for the critical heat flux. This value is used in the separated flow model to determine the phase velocities U_f and U_v , and average vapor thickness δ for each x location along the heating channel. To obtain the solution of these values, we need to solve the ordinary differential equation (4.23). Whether the Euler single-step method or the Midpoint method is used to solve it numerically, iterations are required.

The output of the separated flow model is then used in the instability analysis to calculate the critical wavelength λ_{cr} and the extent of the first continuous wetting front x^* . In this step equation (4.32) is required to be solved with another series of iterations.

Next, the average interfacial pressure difference at x^* is calculated and then used in evaluating the lift-off heat flux at x^* . This heat flux, which is to be used in the energy balance, is a function of the pressure difference, vapor thickness and critical wavelength.

The energy balance yields a value for the critical heat flux, which is compared with the estimated value at the initial step. If these are sufficiently close, the critical heat flux for the given velocity has been predicted correctly. If not, a new estimated heat flux is used to re-initiate the process, which continues until the iteration converges on a certain value.

4.3 Results and discussions

With the CHF model, the CHF across the evaporator of a given heat pipe design can be predicted as a function of several parameters: namely the flow modifier pitch, the operating temperature of the heat pipe (T_{sat}) , and the evaporator to return line ratio (R_o/R_i) . The following results (Table 4.1) are based on the design with an evaporator length of 0.94 m, radius (R_o) of 2.5 cm and a return radius (R_i) of 1.25cm.

When we increase the pitch of the spiral fin, we can compute the corresponding CHF at different returning velocities. The following data is calculated for the saturation temperature of 100°C.

	Critical Heat Flux MW/m ²			
Returning velocity	Pitch	Pitch	Pitch	
U(m/s)	W=0.01 m	W=0.03 m	W=0.05 m	
0.5	4.404	2.914	2.543	
1.0	5.416	3.761	2.851	
1.5	6.080	4.370	3.103	
2.0	6.521	4.732	3.308	
2.5	6.859	5.031	3.480	
3.0	7.106	5.283	3.640	

Table 4.1 CHF for different pitches and returning velocities

From Table 4.1, one can see that whatever the pitch is, the CHF increases with the returning velocity. It can be explained that higher returning velocity leads to higher centrifugal force, which causes the CHF to increase.

In reality the presence of such a spring fin also has a "price" in terms of the fluid mechanics, in the form of an increased resistance to the flow of vapor and liquid returning to the condenser. Though the spring brings a higher CHF, the increased resistance causes a buildup of pressure at the bottom of the evaporator, which hinders the return of liquid through the return line to this region. Besides the heat flux and the evaporator dimensions, the resistance depends on the spring parameters. In order to achieve a higher CHF with the presence of a spring, the additional resistance should also be considered.



The results regarding pitch and saturated temperature are shown in Figure 4.9.

Figure 4.9 Theoretical CHF for several typical configurations and operating temperatures From the results presented in Figure 4.9, it can be seen that an increased operating temperature, a decreased pitch, and an increased return line radius, all have the effect of increasing the CHF. The effect of increasing the temperature can be explained by the associated higher saturation pressure, which increases the saturated vapor density and ultimately the CHF (we can see this point from equation (4.44)).

Decreasing the pitch of the flow modifier has the effect of increasing the CHF by causing velocities within the evaporator to flow in a more circumferential manner, rather than axially. Increasing the return line radius decreases the cross-sectional area of the vapor channel and thus increases the velocities within it by continuity. Both cases result in an increase in the centrifugal forces, promoting good contact between the liquid working fluid and the evaporator wall, which raises the CHF.

In terms of evaporator design, the results indicate that a flow modifier with a smaller pitch is beneficial in avoiding film boiling. Unfortunately, decreasing the flow modifier pitch may also have a penalty in terms of the fluid mechanics, in the form of an increased resistance to the flow of vapor. This increased resistance causes a promotion of pressure at the bottom of the evaporator, ΔP_{evap} , which may hinder the feeding of liquid through the return line to this region, causing dry-out. Therefore, the dilemma is that in general, a flow modifier with a small pitch ("aggressive") generates more centrifugal force and suppresses the film boiling limitation, but on the other hand it may cause a pressure buildup and promote evaporator dry-out. Similarly, a flow modifier with a larger pitch ("passive") may alleviate dry-out, but may be limited in suppressing film boiling.

Chapter 5 The development of a sulfur based heat pipe

5.1 Introduction

It has been proven that water is suitable to be used as the working substance in the relatively low temperature range (30~200°C). However, the vapor pressure of water is 2.6 MPa at 230°C, and rises rapidly with further increases in temperature. Also, the critical temperature of water (374°C) limits its working range. Moreover, in many metallurgical engineering cases, water is not permitted to be used because of safety issues.

The alkali metal sodium has been successfully used as working fluid at temperatures above 700°C. As the temperature is reduced, the vapor pressure and vapor density are decreased. The sonic velocity limits the heat transfer. Z Yuan carried out experiments [10] in an industrial furnace and showed that a high working temperature can adversely affect the heat pipe envelope and ultimately it can damage it. In the past, water and sodium have set the limits for low and high temperature operation. However, numerous researchers have searched for decades to develop a working fluid for the medium or intermediate temperature range. Thus, there is a tremendous need for a working substance that can operate in the temperature range of 200°C~500°C, which is defined as the intermediate temperature range. Mercury and some organic fluids are suggested in the literature. However, mercury vapor is extremely poisonous while organic fluids are prone to decomposition. Sulfur, with its physical and chemical properties, has been viewed as a candidate for decades. To date, little progress has been made to commercialize its use.

5.2 The basic structure of a McGill Heat Pipe

As discussed in Chapter 2, the McGill heat pipe (Figure 2.1) overcomes the film boiling and entrainment limitation by using a specific design. The condenser of a McGill heat pipe is normally very much like a shell and tube heat exchanger with two cooling circuits: an external jacket and a series of internal cooling pipes. The coolant (air or water) comes into a chamber beneath the condenser and flows through the internal cooling pipes to the chamber above the condenser, and then flows down through the interlayer to the outlet. Twisted swirlers can be used in the internal cooling pipes, and spiral swirlers can also be installed in the interlayer, both of which can improve the heat transfer in the condenser.

The returning pipe provides a separate line through which the condensate can travel down to the bottom of the evaporator. Thus, unlike the situation in the traditional heat pipe, the shear stress existing between the up-flow of the vapor and the down-flow of the condensate is eliminated. The bottom of the condenser acts as a reservoir for the condensate. The pressure head between the reservoir and the return line discharge makes the returning flow possible. Consequently the entrainment limit cannot happen in a McGill heat pipe.

The evaporator section is a concentric circular pipe, in which the flow modifier is located. As mentioned earlier, the flow modifier causes the fluid within the evaporator to flow in a helical method, causing the denser liquid to be pressed against the outer tube due to the centrifugal forces that are generated. During operation, the liquid flows on the evaporator wall as it rises, wetting it in a uniform manner, and absorbing the heat input by vaporizing some of the liquid. This not only avoids the formation of hot spots but also suppresses film boiling by forcing the vapor film off the wall.

5.3 The properties of sulfur

Sulfur is one of the abundant elements that occur widely in nature. It is a yellow solid at normal temperatures (see Figure 5.1).



Figure 5.1 The appearance of sulfur flower

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The crystalline rhombic (α) form is stable at normal temperatures. There is a transition at 95.5°C from the rhombic to the crystalline monoclinic (β) form, which is stable up the melting point (119.3°C). Another crystalline modification—the metastable nacreous (γ) modification—is produced when hot, saturated solutions of sulfur in benzene or alcohol are cooled and when alkaline ammonium sulphide solutions are decomposed in contact with air. The structural element of these sulfur modifications is a non-planar S8 ring. Other crystalline modifications are also known, but they are unstable at all temperatures and have not yet been studied thoroughly. The physical properties of sulfur are shown in Table 5.1.

Melting point (monoclinic)	°C	119.2
Boiling point	°C	444.6
Density of solid at 20°C	kg/m ³	2070
Density of liquid		
125°C		1798.8
130°C	kg/m ³	1794.7
140°C		1786.5
150°C		1778.4
Vapor density (444.6°C,1atm)	kg/m ³	3.64
Saturated pressure		
120-325°C	bar	$\log P = 0.981(14.7 - 0.0062238 \cdot T - 5405/T)$
325-550°C		$\log P = 0.981(7.43287 - 3268.2/T)$
Surface tension		
120°C	N/m	0.06083
150°C		0.05767
Specific heat Cp		
Monoclinic,-4.5-118°C	J/kgK	$Cp=(3.58+6.24\times10^{-3}T)\times4.1868\times(1/0.016)$
Liquid,119-444.6°C		$Cp=(3.56+6.96\times10^{-3}T)\times4.1868\times(1/0.016)$
Heat of fusion	1-1/1-~	
Monoclinic (118.9°C)	KJ/Kg	38.52
Latent heat of vaporization		
200°C		308.6
300°C	1-1/1-0	289.3
400°C	KJ/Kg	286.4
420°C		287.6
440°C		290.1

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460°C	1	293.1
Dynamic viscosity of liquid		
120°C		0.017
140°C		0.008
158°C		0.0064
160°C	Pas	5.952
180°C		86.304
187.8°C		93.0
200°C		78.864
300°C		3.72



Figure 5.2 The properties of sulfur [4]

When heated, the monoclinic (β) sulfur is transformed at 119.3°C into a liquid, in which the molecules are still S8 rings. If the melt temperature is raised, the viscosity at first decreases (in the usual fashion for liquids) from 0.0017Pa.s at about 120°C to a minimum value at 157°C (See Figure5.2). But above this temperature the viscosity increases dramatically up to a maximum viscosity value at 186-188°C, at which point the melt is very viscous and dark red. When heated further, the viscosity decreases quickly again. This striking phenomenon is interpreted as follows. As the temperature is increased, above the melting point the S8 rings are progressively broken up into biradicals. These reach sufficient concentration at around 160°C to polymerize spontaneously into - sulfur chains which hinders flow. When the temperature rises further, the long chains are increasingly cracked, so that the viscosity will decrease again.

Figure 5.2 also shows other properties, such as the liquid and vapor densities, latent heat, surface tension, and the saturated pressure. Sulfur is insoluble in water; however, it is easily soluble in a number of organic compounds, especially carbon disulphide, hydrated naphthalene and chlorinated aromatics. Also disulphur dichloride will dissolve a large amount of sulfur. Consequently, one can clean the sulfur in a heat pipe if required, relatively easily.



Figure 5.3 Saturated pressure vs temperature of sulfur

Figure 5.3 shows the curve of saturated pressure vs temperature. The saturated pressure is 0.2atm at 400°C and 8atm at 500°C. Thus, sulfur is suitable for use in this range from the viewpoint of the saturated pressure and temperature.

5.4 Trials with sulfur/iodine heat pipe

5.4.1 The properties of iodine

Iodine is a bluish-black, lustrous solid, which is shown in Figure 5.4. When heated, iodine vapor is a blue-violet gas, which is extremely hazardous and has an irritating odor. Table 5.2 shows its properties.



Figure 5.4 The appearance of iodine

Table 5.2	The p	roperties	of ic	odine	[4]	1]	ł
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Melting point,	Boiling point,	Critical temp., °C	Critical pressure,atm	Latent heat,kJ/kg	Liquid density,kg/m ³	Viscosity, Pa.s
°C	°C					
114	185	512	116	171.4	3740(180°C)	0.127(200°C)

5.4.2 The specific character of sulfur/iodine mixture

From Figure 5.5 we can see that the addition of a small quantity of iodine to sulfur significantly decreases the viscosity of liquid sulfur. The dynamic viscosity of iodine/sulfur mixture is shown in the figure in proportions of 3, 5 and 10 percent by weight. Around the peak point (200°C), the viscosity of pure sulfur is 1000 times that of sulfur which contains 3% iodine. It would appear that the 3~10% iodine solves the problem of sulfur's high viscosity. So in the literatures, the mixture of sulfur and iodine is introduced as one working material.



Figure 5.5 The viscosity of sulfur or sulfur/iodine mixture[4]

5.4.3 The risk of explosion during the start-up

When a liquid mixture of sulfur/iodine is solidified, there is iodine located throughout the pipe and including at the bottom of the pipe. Both sulfur and iodine are not conductive. The thermal conductivity of sulfur is $0.21W/(m \cdot K)$ and iodine is 0.45

 $W/(m \cdot K)$. When the pipe is reheated, little heat is transferred to the condenser during the initial stage because of the low conductivities. So the contents at the bottom of the evaporator are readily converted to liquid while the top of the evaporator and the condenser still contain solid material. As mentioned before, the boiling point of iodine is 185°C. Even the temperature of the furnace is not high, the local pressure at the bottom of the evaporator can reach a high level even if it is in dilute form. For example, the corresponding saturated pressure for iodine at 512°C is 116 atms, which is the critical point. While the iodine is diluted by the sulfur it nonetheless can still exert a high vapor pressure. Consequently, the system has a risk of exploding especially around the bottom.

During the trial experiments, we did have two explosions that blew off the bottom tip. In Figure 5.6 we can see that the pipe is expanded and the bottom is blown off. In Figure 5.7 we can even see the imprint that was caused by the spring squeezing the return pipe, since the pressure outside of the return is much higher than the pressure inside.





Figure 5.6 The blown-off tip in the explosion

Figure 5.7 The squeezed return in the explosion

5.4.4 The working principle of the sulfur/iodine heat pipe

As we know, the boiling points of sulfur and iodine are 444°C and 185°C

respectively. When the sulfur/iodine heat pipe works at a temperature higher than 185°C (the boiling point of iodine), the process is very similar to distillation (Figure 5.8). The whole pipe can be imagined as two heat pipes.

One is "the sulfur heat pipe" (the lower section). The liquid sulfur is evaporated in the evaporator and the sulfur vapor moves up to the condenser. Once the sulfur vapor meets the cooler iodine vapor, condensation happens and a fog of sulfur develops. As the fog of mist droplets collect together, the sulfur droplets fall down to the bottom of the condenser, which works as the reservoir.

The upper section is "the iodine heat pipe". As the coolant flows through the condenser jacket, the iodine vapor is cooled down and condensed on the wall of the condenser. After the liquid iodine streams down to the reservoir, it is evaporated by the hot liquid sulfur and moves up to the top of the condenser.

The outcome of this phenomenon is that heat pipe temperature is not uniformly distributed. The iodine part is cooler than the sulfur part, which means that the condenser temperature is much lower than the evaporator. This causes the efficiency of the condenser to be reduced by a fair amount, with the result that the heat transfer capacity of the heat pipe is reduced.



Figure 5.8 The schematic working diagram of a sulfur/iodine heat pipe

Figure 5.9 shows a typical set of operating curves of the sulfur/iodine heat pipe when it was put into the gas furnace. Because the condensed liquid iodine flows down to the reservoir, the temperature at the bottom of the condenser is around 40°C lower than the temperature at the top of the evaporator. The top of the condenser is filled with iodine vapor, so the temperature is much lower (around 170°C) than that of the bottom. As the returning liquid flows through the part which is outside of the evaporator, it is cooled down. From the curve, we can see the temperature of the return is 80°C lower than that of the condenser bottom. In the sulfur viscosity figure, the viscosity decreases quickly with temperature when the temperature is higher than 200°C. The cooler return means more viscous liquid in the returning pipe, which tends to impede the flow.



Figure 5.9 The working curves of a sulfur/iodine heat pipe

Consequently, although adding some quantity of iodine into sulfur can control the high viscosity problem, it also brings a few setbacks:

- The risk of explosion during the start-up;
- The deficiency of the condenser;
- The possibility of development of non-soluble phases;
- The low heat transfer capacity of the device.

In conclusion, the sulfur/iodine binary mixture is not suitable as a working substance for a conventional or a McGill heat pipe that is to operate in high heat flux environments. Moreover, the conventional heat pipe configuration presents the added problem of experiencing film boiling at elevated heat fluxes because of the low thermal conductivities.

5.5 Development of Sulfur Heat Pipe

5.5.1 The improved design

In the old design, the return pipe comes out of the reservoir and then is inserted into the bottom of the evaporator. This design provides the convenience of measuring the returning flow rate and temperature. However, the liquid flowing from the reservoir will be cooled down by the environment, which means that the sulfur liquid viscosity in that section will be higher. And the returning flow resistance will be increased.

Let's consider a single test. In Figure 5.10a, the dry-out happens because not enough sulfur is returned to the evaporator. We put the pipe back into the furnace, keep heating the return pipe with a torch and take the pipe out, then we can see the dry-out is less serious (Figure 5.10b). Repeat this, we can see Figure 5.10c.

The cooled sulfur with a higher viscosity acts like a valve in the return line. To heat that part is equivalent to opening the valve, with the result that more sulfur can be returned to the bottom of the evaporator. This will restrain the dry-out.



Figure 5.10 The dry-outs in the sulfur/iodine heat pipe with old design

Maintaining a sufficient return flow is the key to making the sulfur heat pipe function adequately. An approximate evaluation of the return can be expressed as follows based on the Bernoulli Equation (which will be discussed in detail in Chapter 6):

$$P_1 + \rho_l gh = P_2 + \frac{1}{2}\rho_l V_r^2 + f \frac{h}{d_r} \frac{\rho_l V_r^2}{2}$$
(5.1)

where P_1, P_2 : Static pressure at the inlet and the outlet of the return pipe, Pa;

 ρ_l : Sulfur liquid density, kg/m^3 ;

V_r: Returning velocity in the return pipe, m/s;

h: Vertical distance between the inlet and the outlet of the return pipe,*m*;

 d_r : Return pipe inner diameter, m.

Suppose the flow through the return is laminar, then the friction factor is:

$$f = \frac{64}{\text{Re}} = \frac{64\mu}{\rho_l V_r d_r}$$
(5.2)

Re: Reynolds number;

 μ : Viscosity, *Pa*·s.

According to the experimental results from Chapter 3 (Figure 3.8), the pressure difference between the inlet and the outlet of the return pipe (P_2-P_1) can be estimated approximately as $\rho_1 gh/2$.

Then equation (5.1) can be rewritten as:

$$V_r^2 + 64 \frac{\mu \cdot h}{\rho_l d_r^2} V_r - gh = 0$$
(5.3)

Consequently the return velocity V_r can be written as:

$$V_{r} = \frac{1}{2} \left(-64 \frac{\mu \cdot h}{\rho_{l} d_{r}^{2}} + \sqrt{\left(64 \frac{\mu \cdot h}{\rho_{l} d_{r}^{2}} \right)^{2} + 4gh} \right)$$
(5.4)

And the mass returning flow rate \dot{m}_r can be expressed as:

$$\dot{m}_r = \rho_l V_r \left(\frac{\pi d_r^2}{4}\right) \tag{5.5}$$

Suppose the viscosity at 440°C is 0.08Pa·s, the elevation of the return pipe is 0.4m, and the density is 1780 kg/m^3 , the returning mass flow rates \dot{m}_r at different return pipe sizes are shown in Table 5.3 and Figure 5.11.

Return diameter	Return velocity	Return mass rate
(m)	(m/s)	(kg/s)
0.017	0.818	0.330
0.015	0.678	0.213
0.013	0.534	0.126
0.010	0.331	0.046

Table 5.3 Approximate evaluations of the return



Figure 5.11 The relationship between the return diameter and the return mass flow

Now we can see that the power relationship between the return diameter and the return mass flow rate exceeds an exponent of 2. From the experiments of water-based heat pipes, the heat load in the gas furnace was for a typical configuration around 10KW. So in designing a sulfur based heat pipe a loading of 10kW was also assumed, thus the sulfur amount needed for evaporation is:

$$\dot{m}_{v} = \frac{Q}{h_{fg}} = \frac{10kJ/s}{290.1kJ/kg} = 0.034kg/s$$
(5.6)

Based on the results from the literature [], the quality x is 30% in the two-phase flow when it approaches film boiling. Thus the required return of liquid sulfur is 0.034/0.3=0.113kg/s. If we consider the sulfur working range to be around 350~500°C, and that two-phase boiling in the evaporator should not reach film boiling, then for a quality which is lower than 30%, we selected the return diameter of 0.017m.



Figure 5.12 The improved design

Compared to the previous design of the McGill heat pipe, the new design (Figure 5.12 and Figure 5.13) has the following additional features:

• A built-in return pipe;

The built-in return pipe prevents the heat loss of the returning liquid. So the viscosity will not increase due to the temperature going down.

• A bigger diameter return pipe;

The bigger diameter decreases the flow resistance, which is shown in equation (5.2).

• The inlet of the return pipe is above the bottom surface of the reservoir;

The reservoir collects all the condensed liquid in a buffering area. Even if some solid residuals are present in the system, they will accumulate at the bottom of the reservoir and not be carried by the returning flow since the inlet is raised above the bottom surface of the reservoir.

• A heating passage along the axis;

The heating passage is a half inch pipe inserted from the top of the condenser, and its bottom is welded shut. Before starting up of the heat pipe, a heating element (STRI-3248/120, by Omega Company) is put into the heating passage, to heat the system to above the sulfur melting point.

• Inner thread in the evaporator section;

The traditional heat pipe wick structure uses capillary pressure to pump the working fluid and to evenly distribute the fluid. However, the pores of the wick can also trap some of the vapor and prevent the liquid from wetting the wall. The inner threads in the evaporator bring no contact thermal resistance and yet they can spread the liquid along the evaporator. This restrains the development of hot spots in high heat fluxes.



Figure 5.13 The photos of the new design

5.5.2 The start-up of a sulfur McGill heat pipe

After the whole pipe is heated up to some temperature which is above the sulfur melting point, the pipe can be put into the furnace. A typical set of working curves are shown in Figure 5.14 when the pressure inside the pipe is one atmosphere initially. The evaporator temperatures begin to increase. Once the evaporator temperatures reach the boiling point, the liquid sulfur starts to evaporate intensively, and the heat is transferred to the condenser rapidly. Now the working fluid commences to circulate. So the reservoir temperature climbs quickly, which is shown as position (1), also we can see that the evaporator top temperature continues to increase until it is higher than the evaporator bottom temperature. This phenomenon occurs because the sub-cooled liquid in the reservoir flows down to the evaporator bottom, and it manifests itself as a lower

temperature at the bottom end of the evaporator.

After the reservoir temperature reaches the boiling point, one can release the non-condensable gas in the system, which is tapped at the top of the condenser. Point ② shows the resulting changes in the temperatures with the opening-closing of a pressure relief valve that was located on the top cover of the condenser, with the rejection of most of the non-condensable gas, the sulfur vapor was able to occupy most of the entire volume. The temperature of the condenser top increases dramatically, on the other hand the evaporator and the reservoir temperatures decrease since the pressure is released, the saturated temperature drops off. At point ③ the valve is reopened, the condenser top temperature increases again since some non-condensable gas is released further. At point ④, since there is no non-condensable gas inside the system, the opening-closing of the valve does not change the temperatures of the heat pipe. After that, the temperatures are uniformly distributed.



Figure 5.14 The start-up curves of a sulfur heat pipe when the initial pressure is one atmosphere

Figure 5.15, 5.16 and 5.17 show the start-up curves when the sulfur heat pipe initial pressure is 0.02atm, 0.05atm and 0.1atm. It is interesting to see that there is a bump in the curve of the temperature at the evaporator top as shown in figures 5.15 and 5.16. During the experiments a hot spot at the top of the evaporator can be seen when the bump

appears. This means that the dry-out happens in the top region. While in Figure 5.17, with a 0.1 initial pressure in the heat pipe, the bump disappears and the hot spot cannot be seen during the start-up. This phenomenon is different from the viscous limit happening in a traditional heat pipe, which is more commonly known as the vapor pressure limit. It is stated that when the vapor pressure is very low, the minimum value of pressure occurs at the closed end of the condenser section. Thus, vapor pressure drop in the return can be constrained by this very low-effectively zero-pressure, and the low vapor pressure existing at the closed end of the evaporator section. Because the vapor pressure difference naturally increases as the heat transported by the heat pipe rises, the constraint on this pressure difference thus necessitates that the heat flux is limited.

Once some liquid is charged into the system, the reservoir, the return and the evaporator are acting as a "U" tube. When the system is sealed with a vacuum or very low pressure, the boiling point for the liquid inside will correspondingly be low. The vapor generated by the boiling entrains the liquid in evaporator to the reservoir. Thus, the liquid is carried to the other side of the U tube. So the evaporator is empty. On the other hand, because the pressure in the free space is effectively low, the vapor produced in the evaporator will move to the condenser with very high speed. The result is that the pressure drop caused by the high speed flow will prevent the liquid in the reservoir from returning. The returned liquid cannot completely wet the surface of the evaporator. Consequently, a dry-out happens at the top part of evaporator. The system pressure and temperature continue to increase as the heat is absorbed in. Once the pressure inside the pipe increases to some point, that the speed of the upward flow is sufficiently low, the returning flow is no longer blocked. The dry-out will disappear after that.

In Figure 5.17, the system was started up by charging 0.1 atmosphere nitrogen initially. The baseline for boiling is improved equivalently. During the operation, heat absorbed by the evaporator is used to increase the temperature of the liquid in evaporator and no boiling occurs. So no steep change of the evaporator temperature can be seen in the start-up curves and no dry-out happens during the experiment. The price of this method is to sacrifice part of the condenser, as is shown in Figure 5.17 where we see that the temperature at the top of condenser is much lower that in the reservoir and the
evaporator.



Figure 5.15 The start-up curves of a sulfur heat pipe when the initial pressure is 0.02 atmosphere



Figure 5.16 The start-up curves of a sulfur heat pipe when the initial pressure is 0.05 atmosphere



Figure 5.17 The start-up curves of a sulfur heat pipe when the initial pressure is 0.1 atmosphere

5.5.3 The steady state of a sulfur McGill heat pipe

Figure 5.18 and Figure 5.19 show the pipe temperatures in steady state in different furnace temperatures. One can see that whatever the furnace temperature is, the sulfur heat pipe can work stably. What needs to be adjusted is the cooling flow rate. Comparing with sulfur/iodine heat pipe working curves (Figure 5.9), the temperature difference between the evaporator and condenser is much smaller, and the heat transfer capacity is much higher.

It is necessary to mention that the temperature at the top of the evaporator is higher than that at the bottom. That is because the saturated sulfur liquid in the reservoir is sub-cooled as it returns back to the evaporator.



Figure 5.18 The steady state working curves at furnace temperature 980°C



Figure 5.19 The steady state working curves at furnace temperature 1300°C

5.5.4 The cleaning of sulfur McGill heat pipe

In the engineering cases, the heat pipes may need to be cleaned or the reason of repairing or disposal. It could be a difficult job to clean an alkali metal heat pipe. Sulfur is easily soluble in a number of organic compounds, especially carbon disulphide, hydrated naphthalene and chlorinated aromatics. In the reality another feasible way to clean sulfur pipe is by using water. The specific method can be operated as follows:

- Charge water into the pipe and close it;
- Heat the pipe up to 130~140°C (pressure 2.7~3.6atm); Sulfur is molten and mixed with water.
- Cool the pipe down to 100°C or lower quickly; Sulfur is solidified as particles or clusters in the water.
- Pour the water out with the sulfur.

5.6 Concluding remarks

While only a few results have been shown in this chapter, numerous experiments and tests were carried out to study the operation of the sulfur based McGill heat pipe. The objective of this work was to develop a sulfur based heat pipe for high heat flux applications. This objective was met as on can see from the steady state operating curve (e.g. Figure 5.19 and 5.20). During steady state operation, there was nothing unexpected that was noted. However, it was shown that the starting of a sulfur heat pipe requires that one takes precautions to avoid damaging the pipe. In addition, it was also shown that a sulfur heat pipe which is at a very high vacuum at room temperature will, when heated from a cold start, produce a "hot region" neat the top of the evaporator. This phenomenon can be avoided by charging the heat pipe with a small (e.g. 0.1 atm) quantity of inert gas when it is being constructed. One should note that the testing of a sulfur heat pipe as described above is not report in the literature. The present work constitutes a novel contribution in this regard.

Chapter 6 Experimental study and theoretical model for a sulfur based McGill heat pipe

6.1 Introduction

A McGill heat pipe, with its specific structures, not only allows large heat removal capability but also provides considerable flexibility in the positioning of the condenser and the evaporator. It involves a co-current natural circulation which contains a separate returning liquid flow, a two-phase boiling flow and forced-convection condensation. Most of the available modeling literature deals with two-phase thermosyphons with counter-current circulation with a closed, vertical, wickless heat pipe configuration. To design a McGill heat pipe for a specific industrial engineering case or to predict working parameters of a current McGill heat pipe, a theoretical model is a required necessity. In this chapter, an analytical model based on the mass, energy and momentum balance is presented, and sulfur is considered as the working material. The experimental results are compared with the theoretical ones.

In a McGill heat pipe evaporator, the absorbed heat is used to increase the liquid temperature if it is sub-cooled, and to provide the latent heat of evaporation. The twophase boiling flow moves through a concentric passage with a helical movement. As the mixture of vapor and liquid moves into the condenser from the evaporator,, the liquid is collected by the reservoir and the vapor is condensed by forced convection. In transient mode, if the condensation cannot be completed by the cooling flow, the system pressure will increase. This leads to the increasing of the saturation temperature, which improves the temperature gradient for the condensation until the latent heat released by the condensing liquid is balanced by the heat transferred from the condenser chamber to the coolant. Then the system reaches a steady state and a new saturation temperature is obtained. Located at the bottom of the condenser, the reservoir which contains saturated liquid is also cooled down by the cooling flow. So the liquid returned down to the evaporator is sub-cooled.

The height difference between the reservoir and the evaporator provides the pressure head for the return flow. This is very similar to the case simulated in Chapter 3, which is basically described as equation (3.4). However, the pressure drop of two-phase flow is an integration of each differential element since the quality x and the void fraction α are not constant along the evaporator.

To run this model, one needs to input the dimensions of the heat pipe, the heat source temperature, and the cooling flow rate and temperature. Then the model can calculate the heat load, outlet temperature of the coolant, working temperature, returning velocity, wall temperature, void fraction, quality and pressure drop in the evaporator. The process is very similar to a real experiment. To design a McGill heat pipe for a real industrial application, one can run the model with a different set of the heat pipe dimensions and select the optimal one. In this chapter, a gas furnace is used as the heat source, so radiation and convection are considered as the heat transfer methods. If the model is used in a liquid metal application, conduction will be considered.

6.2 Experimental setup

Figure 6.1 shows the schematic system of the experimental setup. The McGill heat pipe⁽⁵⁾ is inserted into the gas furnace⁽³⁾(MIFCO 6301-REV, 210KW). The furnace temperature is measured by R type thermocouple⁽⁴⁾ (RAT-24-12, Ω company). The cooling air flows through the condenser jacket. The flow is controlled by the valves⁽¹⁾ and the flow rate is measured by flow meters⁽²⁾ (variable area, Ω FL4513). The temperatures of the cooling inlet⁽⁷⁾ and outlet⁽⁸⁾ are measured by K thermocouples (KMTXL-032G-6, Ω company). The temperatures along the axis ⁽⁶⁾ of the heat pipe are measured by K thermocouples (KMQSS-020G-36). All the voltage signals are acquired by the processor⁽⁹⁾ (InstruNet 100) and transferred to the computer⁽¹⁾.



Figure 6.1 Experimental set-up for a sulfur McGill heat pipe

The structure of the McGill heat pipe is shown in Figure 5.13. Some other important dimensions are shown in Table 6.1.

Evaporator	OD 48mm	Condenser	OD 158mm
	ID 40mm		ID 152mm
	Length 406mm		Length 381mm
Return	OD 21mm	Flow modifier	Coil diameter 3mm
	ID 17mm		Pitch 30mm
	Length 395mm		Length 250mm
Reservoir	Volume 1.46L		

Table 6.1 Dimensions of the McGill Heat Pipe

6.3 Model description

6.3.1 Heat transfer in the gas furnace

The heat transfer in the gas furnace consists of two parts: the radiation and the convection. The radiation heat flow \dot{q}_{Rad} is:

$$\dot{q}_{Rad} = \varepsilon F_{v} \sigma A_{p} (T_{fu}^{4} - T_{p}^{4}) = \varepsilon F \sigma A_{p} (T_{fu}^{2} + T_{p}^{2}) (T_{fu} + T_{p}) (T_{fu} - T_{p})$$
(6.1)

Where ε emissivity of the heat pipe surface;

 σ_{sb} Stephan-Boltzman constant, 5.67×10⁻⁸ $Wm^{-2}K^{-4}$;

 F_{ν} view factor of the pipe relative to the furnace wall, is taken to be 1;

- A_{fu} furnace area, m^2 ;
- A_p immersed area of the heat pipe, m^2 ;

 T_{fu} furnace temperature, K;

 T_p heat pipe temperature, K;

So equation (6.1) can be rewritten as:

$$\dot{q}_{Rad} = h_{Rad} A_p (T_{fu} - T_p)$$
(6.2)

where h_{Rad} is the effective radiative heat transfer coefficient, $W/m^2 \cdot C$.

$$h_{Rad} = \varepsilon F \sigma (T_{fu}^2 + T_p^2) (T_{fu} + T_p)$$
(6.3)

So the total heat transfer coefficient in the furnace can be expressed as:

$$h_{total} = h_{Rad} + h_{Con} \tag{6.4}$$

6.3.2Heat transfer in the evaporator

The condensate from the reservoir is sub-cooled. This means that the bulk temperature of the returning liquid is lower than the saturation temperature. So the boiling inside the evaporator can be divided into two sections: sub-cooled boiling and saturated boiling.

• Sub-cooled boiling

In the sub-cooled boiling section, vapor is formed at the heated wall in the form of single bubbles as a bubbly layer parallel to the wall. These bubbles are swept into the subcooled area of the sulfur liquid by the accumulation of bubbles or by the variable shear stress on their boundaries. To predict local heat transfer coefficients in the subcooled boiling regime, Gungor and Winterton[56] adapted their correlation by using separate temperature differences for driving the respective nucleate boiling and convective boiling processes. So the heat flux \ddot{q} is calculated as:

$$\ddot{q} = h_l \left(T_w - T_l \right) + S_{gw} \cdot h_{pool} \left(T_w - T_{sat} \right)$$
(6.5)

where h_l is the forced convection heat transfer coefficient, and it can be calculated as[18]:

$$h_{l} = 0.303 (e/D_{h})^{0.12} (p/D_{h})^{-0.377} \operatorname{Re}_{l}^{0.72} \operatorname{Pr}_{l}^{0.37} \frac{k_{l}}{D_{h}}$$
(6.6)

 T_w , T_l , and T_{sat} are the wall temperature, the liquid temperature and the saturation temperature, °C.

e: the diameter of the coil, m;

p: the pitch of the spring,m;

Re_l: Reynolds number of the liquid in the evaporator;

Pr_l: Prandtl number of the liquid in the evaporator;

 k_i : liquid conductivity, $W/(m \cdot C)$;

 S_{gw} can be calculated as:

$$S_{gw} = \frac{1}{1 + 1.15 \times 10^{-6} \cdot E^2 \cdot \text{Re}_I^{1.17}}$$
(6.7)

The expression for E is:

$$E = 1 + 24000 \cdot Bo^{1.16} \tag{6.8}$$

The boiling number Bo is defined as:

$$Bo = \frac{q}{H_{fg}G} \tag{6.9}$$

where \ddot{q} : heat flux, W/m^2 ;

$$G : \text{mass flux, } kg/(m^2 \cdot s);$$

$$H_{fg}: \text{ latent heat of vaporization, } J/kg;$$

$$h_{pool} = 55 \operatorname{Pr}^{0.12} (-\log_{10} \operatorname{Pr}_{l})^{-0.55} M^{-0.5} \ddot{q}^{0.67}$$

$$M: \text{ molecular weight.}$$

$$(6.10)$$

In the solution procedure, it is assumed that heat flux \ddot{q} is known and the wall temperature T_w is calculated. Once T_w is known, the heat flux between the wall and the environment can be calculated. A sequence of iterations is required. The heat transferred between the outer wall and the furnace equals the heat transferred inside the tube, so the outer wall temperature can be calculated.

• Saturated boiling

The most widely used correlation for forced convection boiling heat transfer is that of Chen [57], which is:

$$h = h_{FC} + h_{NB} \tag{6.11}$$

h: the total heat transfer coefficient along the inside wall, $W/m^2 \cdot \mathcal{C}$;

 h_{FC} : the forced convection heat transfer coefficient, $W/m^2 \cdot {}^{\circ}C$;

 h_{NB} : the nucleate boiling heat transfer coefficient, $W/m^2 \cdot C$.

 h_{FC} can be calculated as:

$$h_{FC} = F_m \cdot h_l \tag{6.12}$$

 F_m is a multiplier to consider the effect of the increase in liquid velocity which is caused by the existence of vapor flow, it can be expressed as:

$$F_m = 2.35 \left(\frac{1}{X_u} + 0.213\right)^{0.736}$$
(6.13)

 h_i is the heat transfer coefficient for the liquid phase flowing alone in the pipe and X_u is the Martinelli parameter. Because of the existence of a flow modifier, h_i is enhanced and can be calculated by (6.6).

The Martinelli parameter is given by:

$$X_{u} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_{g}}{\rho_{l}}\right)^{0.5} \left(\frac{\mu_{l}}{\mu_{g}}\right)^{0.1}$$
(6.14)

Where *x*: quality;

 ρ_{g} : vapor density, kg/m^{3} ;

 ρ_1 : liquid density, kg/m^3 ;

 μ_l : liquid viscosity, $kg/(m \cdot s)$;

 μ_g : vapor viscosity, $kg/(m \cdot s)$;

The nucleate boiling coefficient h_{NB} is given as

$$h_{NR} = S \cdot h_{FZ} \tag{6.15}$$

where h_{FZ} is the heat transfer coefficient for the ordinary type of nucleate boiling. It can be calculated by the Forster and Zuber correlation[]:

$$h_{FZ} = \frac{0.00122\Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75} c_{pl}^{0.45} \rho_l^{0.49} k_l^{0.79}}{\sigma^{0.5} H_{fg}^{0.24} \mu_l^{0.29} \rho_g^{0.24}}$$
(6.16)

 ΔT_{sat} : the difference between the wall temperature and the saturation temperature;

 ΔP_{sat} : the difference between the saturation pressure at the wall temperature and the saturation pressure of the liquid;

 c_{pl} liquid heat capacity, $J/(kg \cdot C)$;

 σ : surface tension, *N/m*;

S is a factor to consider the effect of the suppression of nucleate boiling due to the temperature gradient in the liquid film on the wall.

$$S = \frac{1}{\left(1 + 0.00000253 \cdot \operatorname{Re}_{l} \cdot F_{m}^{-1.25}\right)^{1.17}}$$
(6.17)

Undoubtedly, the forced convection is enhanced by the rotating movement. However, how the flow modifier affects the multipliers F_m and S is not clear thus far. The rotating movement, caused by the flow modifier, not only makes the liquid flow over the wall but it also helps the vapor to escape from the heated surface. This improves the nucleate boiling heat transfer. On the other hand, the lower wall temperature caused by a higher convective heat transfer rate reduces the effective number of bubble nucleation sites. From this point of view, the rotating movement can restrain the nucleate boiling. This is one uncertainty which suggested that we use the unchanged nucleate boiling heat transfer equation in the model.

The total heat transferred from the ambient to the liquid is equal to the heat transferred between the ambient and the outer wall, which can be expressed as:

$$d\dot{q}_{e} = \frac{\pi \cdot dz (T_{amb} - T_{l})}{\frac{1}{hD_{i}} + \frac{1}{2K_{w}} \ln \frac{D_{o}}{D_{i}} + \frac{1}{h_{tot}D_{o}}} = \frac{\pi \cdot dz (T_{amb} - T_{w})}{\frac{1}{h_{tot}D_{o}}}$$
(6.18)

dz: the element length,m;

 T_{amb} : the ambient temperature, °C; T_l : the liquid temperature in the evaporator, °C; T_w : the outside wall temperature, °C; *h*: the heat transfer coefficient along the inside wall, $W/m^2 \cdot \mathfrak{C}$

 K_{w} : the conductivity of the wall, $W/m \cdot \mathcal{C}$;

 h_{tot} : the total heat transfer coefficient, $W/m^2 \cdot C$;

 D_o : the outer diameter of evaporator, m;

 D_i the inner diameter of evaporator, m;

The wall temperature can be calculated by a sequence of iterations.

6.3.3 Evaporator energy balance

In the sub-cooled section, the heat transferred through the evaporator is equal to the increase of the enthalpy of the liquid.

$$d\dot{q}_e = \dot{m}_{re} c_{pl} \cdot dT \tag{6.19}$$

where $d\dot{q}_{e}$: the heat transferred in the element, W;

 \dot{m}_{re} : the returning mass flow rate, kg/s;

dT: the temperature change of the liquid in the element, \mathcal{C} ;

Once the liquid reaches the saturated temperature, the heat transferred through the evaporator should be equal to the latent heat, which can be expressed as:

$$d\dot{q}_e = \dot{m}_{re} H_{fg} \cdot dx \tag{6.20}$$

where x is the quality of the two-phase mixture.

At
$$T_{l} < T_{sat}$$
, x=0.

If the local actual quality x is known, the local void fraction α can be calculated from standard relationships for two-phase void fraction. The correlation in Carey's book [V. P. Carey, Liquid-Vapor Phase-change Phenomena, Taylor and Francis, Bristol, PA,1992]can be used:

$$\alpha = \left[1 + 0.28 \left(\frac{1-x}{x}\right)^{0.64} \left(\frac{\rho_g}{\rho_l}\right)^{0.64} \left(\frac{\mu_l}{\mu_g}\right)^{0.07}\right]^{-1}$$
(6.21)

6.3.4 Pressure drop from the evaporator to the condenser

The pressure drop consists of three parts: the gravitational pressure drop, the momentum acceleration pressure drop and the frictional pressure drop.

$$-\left(\frac{dp}{dz}\right)_{total} = -\left(\frac{dp}{dz}\right)_{g} - \left(\frac{dp}{dz}\right)_{a} - \left(\frac{dp}{dz}\right)_{fr}$$
(6.22)

The gravitational pressure drop gradient:

$$-\left(\frac{dp}{dz}\right)_{g} = \left[\rho_{I}(1-\alpha) + \rho_{g}\alpha\right]g$$
(6.23)

The momentum acceleration pressure drop gradient:

$$-\left(\frac{dp}{dz}\right)_{a} = \frac{1}{A}\frac{d}{dz}\left[\frac{G^{2}x^{2}A}{\alpha\rho_{g}} + \frac{G^{2}(1-x)^{2}A}{(1-\alpha)\rho_{l}}\right]$$
(6.24)

where G is the mass flux, $kg/m^2 s$.

The momentum acceleration pressure drop only exists at the part where the evaporator is heated. Assuming an adiabatic two-phase flow through the rising tube between the evaporator and the condenser, one then has a constant quality and a constant void fraction because of zero heat transfer assumption. This eliminates the acceleration term in that part of the heat pipe.

The best correlation for small mass fluxes (G<100kg/m²s) is the well-known Lockhart-Martinelli correlation. This correlation uses φ_l^2 , fractional multiplier for the liquid phase.

$$\left(\frac{dp}{dz}\right)_{fr} = \left(\frac{dp}{dz}\right)_{fr,l} \varphi_l^2 \tag{6.25}$$

where φ_l^2 is given by(3.28),(3.29) and (3.27).

The constant C in Lockhart-Martinelli method can be predicted by equation (3.30). The single-phase frictional pressure drop in the tube with a flow modifier can be expressed as follows:

$$-\left(\frac{dP}{dz}\right) = f \frac{2G^2}{\rho D_h} \tag{6.25}$$

where the friction factor f can be calculated by equation (3.17). Once the local quality and the local void fraction are known, the local Re number for the liquid phase and the vapor phase can be calculated. The single phase and the two phase frictional pressure drop can be obtained consequently.

6.3.5 Pressure balance on the returning pipe

The existence of the reservoir assures that the flow in the returning line is a single-phase flow. When water flows down to the bottom of the evaporator from the reservoir, the pressure head for the driving force is $\rho_l gh$. This pressure has to overcome the friction loss and minor loss in the returning line, and also the pressure difference between the bottom of the evaporator and the condenser. This can be expressed as:

$$-\Delta P_{e} - \Delta P_{re} + \rho_{I}gh = \frac{1}{2}\rho_{I}V_{re}^{2}$$
(6.26)

 ΔP_e : the pressure drop along the evaporator, Pa;

 ΔP_{re} : the pressure drop along the return, Pa;

h: the vertical distance between the inlet and outlet of return, *m*;

 V_{re} : the returning velocity, m/s.

 ΔP_{re} includes the friction loss and the minor loss in the returning line, which can be calculated by the traditional formulas.

$$\Delta P_{re} = f_{re} \frac{l}{d} \frac{1}{2} \rho_l V_{re}^2 + \sum k_p \frac{1}{2} \rho_l V_{re}^2$$
(6.27)

 f_{re} : the friction factor inside the return;

l: the length of return, *m*;

d: the inner diameter of the return, m

 $\sum k_{\rm p}$: the sum of the minor pressure loss factors.

6.3.6 Heat transfer in the condenser

The condenser acts as a heat exchanger. The cooling air flows into the bottom of the gap, and then up to the top. With the presence of a swirler, the cooling efficiency is improved. As we discussed in the introduction, complete condensation happens in the condenser. The heat transfer inside the condenser can be divided into two parts: the heat transferred by condensation and that transferred by the sub-cooling of the saturated liquid. When the flow rate and the inlet temperature of the coolant are given, the outlet temperature can be calculated by the energy balance:

$$G_{c}c_{p}(T_{o,c} - T_{in,c}) = h_{c,c} \cdot A_{c} \cdot (T_{w,c} - T_{sat}) + G_{re} \cdot c_{pl} \cdot (T_{sat} - T_{sub})$$
(6.28)

where G_c : the coolant flow rate, kg/s;

 c_p : the heat capacity of the coolant, $J/(kg \cdot C)$;

 $T_{in,c}$: the inlet temperature of coolant, \mathcal{C} ;

 $T_{o,c}$: the outlet temperature of coolant, \mathcal{C} ;

 $h_{c,c}$: the condensing heat transfer coefficient, $W/m^2 \cdot \mathcal{C}$;

 A_c : the condensing area, m^2 ;

 $T_{w,c}$: the condenser wall temperature, \mathcal{C} ;

G_{re}: the returning flow rate,*kg/s*;

 T_{sub} : the sub-cooled temperature, \mathcal{C} ;

The condensing heat transfer coefficient $h_{c,c}$ is given by classic Nusselt condensing theory:

$$h_{c,c} = 0.728 \left[\frac{\rho_l^2 g H_{fg} k_l^3}{\mu_l \Delta T_{w,c} d_c} \right]^{\frac{1}{4}}$$
(6.29)

hc,c: the condensing heat transfer coefficient, $W/m^2 \cdot C$;

 $\Delta T_{w,c}$: the temperature difference between the wall and the saturated vapor, \mathcal{C} ;

 d_c :the diameter of condenser, m;

The heat transfer coefficient in the coolant side $h_{l,c}$ can be calculated by (6.6) with the corresponding pitch, coil diameter and coolant properties.

To calculate $h_{c,c}$, we have to know the wall temperature in the condenser. The heat transferred between the vapor and the wall should be equal to that transferred between the vapor and the coolant. For example along the tubes distributed inside the condenser, we have:

$$\frac{T_{sat} - T_{w,c}}{\frac{1}{h_{c,c}\pi L_c d_o}} = \frac{T_{sat} - T_{I,c}}{\frac{1}{h_{c,c}\pi L_c d_o} + \frac{1}{2K_w\pi L_c}\ln\frac{d_o}{d_i} + \frac{1}{h_{I,c}\pi L_c d_i}}$$
(6.30)

 T_{Lc} : the coolant temperature, \mathcal{C}

 L_c : the length of the condenser, *m*;

 d_i : the inner diameter of the condenser, m;

 d_o : the outer diameter of the condenser, m;

To close the model, the heat transferred in the condenser should equal to that transferred in the evaporator.

6.3.7 Modeling the system saturation temperature

The saturation temperature is a key variable in the system. It depends on the heat transfer in the evaporator and in the condenser, and the fluid flow. The heat balance coupling between the evaporator and the condenser is what allows one to find the solution of the saturation temperature. T_{sat} is solved by iterations as the value that is low enough to absorb the heat input in the evaporator and high enough to reject the same in the condenser.

The present model has the feature of a highly coupled nature. It has a very iterative computational environment as shown in Figure 6.2. To perform the mass and energy balances on the evaporator, we have to calculate the sub-cooled temperature of the return as the inlet of the evaporator. The same balances in the condenser require the knowledge of heat load in the evaporator. At the same time, the evaporator and condenser balances require the mass flow rate, which is an unknown until the pressure balances are applied. The calculations of the pressure balances involve vapor quality and void fraction, which are unknown until the evaporator/condenser mass and energy balances have been applied.



Figure 6.2 Mass, momentum, and energy balance in the heat pipe

The non-linear system is solved by Matlab6.5 codes. The computational flow chart is shown in Figure 6.3.





6.4 Experimental and computational results

As shown in Figure 6.1, the sulfur based McGill heat pipe is inserted into the gas furnace. 1500 grams of sulfur was charged into the pipe, and the liquid volume was 830ml when sulfur was molten. The volumes of the reservoir and the evaporator were 1459ml and 337ml respectively. Initially, there was about 500ml sulfur in the reservoir when the system was heated up to a temperature above the melting point.

The temperature distribution along the axis of the heat pipe, the inlet and outlet temperatures and the flow rate of the cooling air, and the furnace temperature were measured. Because of some limitations of the experimental settings, some working parameters, for example, the returning velocities and the qualities could not be measured. However, the model can generate the predictions for these parameters.

6.4.1 Case 1

The sulfur based heat pipe works under the following working condition:

Furnace temperature 900° C;

Cooling flow rate: 0.0165m3/s(35 SCFM)

Length of evaporator inserted into the furnace: 0.3m (12 inches).

Cooling air inlet temperature:19°C.

Table 6.2 shows the cooling air outlet temperature and the heat load. Both the experimental results and those computed by the model are shown for this case.

Table 6.2 Results in Case 1

	Experimental result	Model result
Cooling air outlet temp. °C	216	227.8
Heat load kW	3.94	4.15

Figure 6.4,5,6,7,and 8 show the experimental data and the model results of the temperature distribution along the axis. One can see that the model prediction fits the experimental data very well. The slight decrease in temperature in the condenser ($\geq 0.4m$) may be due, in part, to the presence of a small quantity of non-condensable gas which is difficult to extract from the pipe. Moreover, it has been shown in the previous chapter

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that it is not desirable to exact it from the perspective of the starting of the heat pipe.

The presence of some inert gas will tend to show of as a decreasing temperature but one must keep in mind that this does not imply a decreasing pressure. Given that the model is based on fluid mechanics, it does not predict a significant drop in pressure, and jus temperature. However, the inert gas acts as a resistance and causes the drop in the temperature.



Figure 6.4 Temperature distribution along the central axis in case 1



Figure 6.5 The inside wall temperature distribution in case 1





6.4.2 Case2

Working conditions:

Furnace temperature:1100°C

Cooling air flow rate:0.0448 m³/s (95SCFM)

Cooling air inlet temperature:19°C

Length of evaporator inserted into the furnace: 0.3m (12 inches).

Table 6.3 Results for Case 2



Figure 6.9 Temperature distribution along the axis in case 2







Figure 6.11 The local heat flux distribution in case 2







Distance from the bottom, m Figure 6.13 The quality distribution in case 2

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6.4.3 Case3

Working conditions:

Furnace temperature:1350°C

Cooling air flow rate: 0.0472 m³/s (107 SCFM)

Cooling air inlet temperature:19°C

Length of evaporator inserted into the furnace: 0.15m (6 inches).

Table 6.4 Results in Case 3

	Experiment	computation
Cooling air outlet temp. °C	162	167.6
Heat load kW	8.76	9.04



Figure 6.14 Temperature distribution along the axis in case 3



Figure 6.15 Inside wall temperature distribution in case 3



Figure 6.18 The quality distribution in case 3

6.5 Discussions

The model presents an analytical method for modeling the performance of the McGill heat pipe. Although not all the working parameters can be measured during the experiments, the measured parameters agree well with the results generated by the model.

Heat load

Heat load depends on the furnace temperature and the heated area (the length inserted into the furnace). It can be found that the heat loads obtained from the experiments are lower than the ones from the model. One reason for this is that the section of the heat pipe that connected the evaporator and condenser was assumed to be adiabatic when in fact there was some finite heat loss.

Temperature distribution along the axis

One can see that the temperature at the bottom of evaporator is lower than the saturated temperature. In the condenser, the vapor is condensed and the condensate flows down to the reservoir, which is below the saturation temperature of the vapor. Thus, the condensate is sub-cooled before it returns to the evaporator. So the boiling in the evaporator can be divided into two parts: the sub-cooled part and the saturated part. In the sub-cooled part, the absorbed heat is used to increase the liquid enthalpy. Once the liquid temperature reaches the saturated point, the boiling turns into saturation boiling. The absorbed heat is equal to the latent heat of evaporation. From case 1,2 and 3, one can see that the sub-cooled part of the temperature increases with the heat load. An explanation for this is because at the higher heat loads larger cooling flow rates are required by the heat pipe. This, in turn, causes the condensate in the reservoir to be sub-cooled more greatly when the larger cooling flow are used.

Wall temperature and heat flux

The wall temperature and the local heat flux distribution can be divided into two parts too. In the sub-cooled boiling section, the evaporator has a lower wall temperature but a higher local heat flux. For the heat transfer between the furnace and the pipe, a lower wall temperature means a higher temperature gradient, so the local heat flux is consequently higher.

Void fraction and quality

In the sub-cooled boiling section, both the quality and the void fraction are zero. In the saturated boiling section, the quality increases linearly with the length of heated area. However, the void fraction increases dramatically because of the slip between the liquid and vapor.

6.6 Uncertainties

There are two uncertainties in this model. The first uncertainty is induced by the sub-cooled boiling correlation and the saturated boiling correlation. From the curves of the wall temperature in those three cases, one can see that there is a step change when the boiling is transmitting from sub-cooling boiling to saturated boiling. In reality, this step cannot exist, a transition section should be proposed. The second uncertainty comes from the cancellation of the function of inner wall machine threads on the heat transfer and flow resistance. Because of the existence of the flow modifier, the liquid and vapor move up helically. Compared with the effect of a flow modifier, the enhancement of heat transfer and flow resistance caused by the inner threads was assumed to not be significant. At low heat fluxes, the inner threads can be very helpful for the development and growth of bubbles. However, their influence is not considerable at high heat fluxes. The purpose for using inner threads was to make the liquid be distributed more uniformly so as to avoid the hot spots at relatively high heat fluxes.

Chapter 7 Conclusions

In the present work, McGill heat pipes are investigated by conducting both theoretical and experimental studies. Based on the results and discussions presented in the previous chapters, the following conclusions and observations can be formulated.

1. Two-phase flow characteristics in a McGill heat pipe

Water and air are used to simulate the two-phase flow in the evaporator of a McGill heat pipe. The conclusions are:

• The presence of a spring flow modifier has a significant influence on the flow patterns. The rotating movement (i.e. centrifugal force) propels the liquid move to the outer wall. The flow modifier promotes annular flow even at reduced air flow rates.

• The gravitational pressure drop is the dominant component of the total pressure drop in the evaporator section when the air flow rate is small. Conversely, the friction pressure drop is the dominant component when the air flow rate is large. The total pressure drop reaches a minimum when the air flow rate was 3.5NI/s for the physical system that was studied.

• The returning velocity can be computed with the Bernoulli Equation as applied between the reservoir and the evaporator bottom. When the total pressure drop in the evaporator increases, the returning velocity decreases correspondingly.

• A non-dimensional parameter, Sw, is used in predicting the friction factor in single phase flow with a spring flow modifier. Eq. (3.17) is available with an error range of $\pm 20\%$.

• The homogenous model is not suitable for predicting the frictional pressure drop in two-phase flow. It under evaluates the frictional pressure drop.

• The void fraction can be calculated using the drift-flux model with the modified distribution parameter C_0 . Given the configuration of a swirler, the constant C in the Chisholm equation can be evaluated by Eq. (3.30). The two-phase

flow friction multiplier ϕ_L can be calculated, and the two-phase frictional pressure drop can consequently be predicted.

2. Theoretical model for prediction of Critical Heat Flux(CHF) in the full spiral tunnel

A specific flow modifier, i.e. a whole tunnel swirler, is considered to predict the CHF in a swirling flow.

• The critical heat flux(CHF) can be predicted by a model in which the interface of liquid and vapor is idealized as a sinusoidal curve with both amplitude and wavelength increasing in the flow direction. It is assumed that at the CHF point the vapor momentum emanating from the wetted surface is balance by the pressure difference which serves to maintain the interface in contact with the surface.

• An increased operating temperature, a decreased pitch, and an increased return line radius, all have the effect of increasing the CHF.

3. Development of a sulfur-based heat pipe

Sulfur, with its physical and chemical properties, can be considered as a working fluid for the medium temperature range applications.

• Though sulfur/ iodine mixture is recommended in the literature, it is not suitable to be a medium temperature working fluid because of the setbacks of the iodine vapor's poisonous and corrosive issues, and the low heat transfer capacity of the device.

• By introducing a heated passage through the heat pipe length, the sulfur can be melted before the start-up of the heat pipe. The risk of explosion during the start-up period is thus eliminated.

• The inner threads on the evaporator walls bring no extra contacting thermal resistance and they help to spread the liquid along the evaporator. This restrains the development of hot spots in high heat flux applications.

• The initial pressure in the heat pipe has a significant effect on the start-up curve. When the system is sealed with a vacuum or very low pressure, a dry-out event takes place at the top part of evaporator during the start-up. The hot spot may dissipate on its own or it may hunger for an extended period of time if it is produced

as steady state is attained. When the initial pressure is about 0.1atm, the dry-out will not occur.

• A sulfur based heat pipe can work stably in different furnace temperatures. The temperature difference between the evaporator and the condenser is very small ($\leq 12^{\circ}$ C). When the heat load is changed, the heat pipe can maintain the same working temperature if one adjusts the cooling air flow rate.

• A sulfur based heat pipe can be physically cleaned.

4. Experimental study and theoretical model for a sulfur based McGill heat pipe

• The operating properties of a sulfur based heat pipe can be described by an analytical model based on the mass, energy and momentum balance. The new correlation and modified Lockhart-Martinelli method for the calculation of friction pressure drop in two-phase flow is used to in this model to calculate the total pressure drop in the evaporator.

• Because of the specific structure of a McGill heat pipe, the boiling in the evaporator consists of the sub-cooled part and the saturated part. The temperature along the central axis increases linearly in the sub-cooled part and keeps constant in the saturated part. The results generated by the model agree well with the experimental results.

• The sub-cooled part of the temperature depends on the cooling flow rate, which causes the condensate in the reservoir to be sub-cooled.

• The distribution of the wall temperature, the heat flux, the void fraction and the quality also can be divided into two parts: the sub-cooled part and the saturated part.

• There are two uncertainties in this model. One is that the correlation for the transmit zone between the sub-cooled boiling and the saturated boiling is not available and not considered. The other one is the effect of the inner threads in the evaporator is neglected since their influence on heat transfer and flow resistance is not significant compared to the flow modifier.

Statement of originality

Prior to this study, the sulfur based mid-temperature range heat pipe for high heat fluxes had not been reported in the literature. Development of the sulfur based heat pipe is one of the significant highlights of this thesis. A McGill heat pipe with its specific structure can make use of sulfur as the mid-temperature range working fluid for high heat fluxes.

Given that there was little known about the McGill heat pipe, the initial research program focused on a study of the two-phase flow in a McGill heat pipe evaporator by simulating the fluid mechanics with an air-water mixture. This was followed by a theoretical study of the critical heat flux in a full spiral tunnel. With this knowledge, the research moved to the design, fabrication and testing of a sulfur-based McGill heat pipe. The major original contributions to knowledge made in this research are the following:

1. Two-phase flow simulation of a McGill heat pipe

• The two-phase flow patterns with a spinning motion were reproduced in a simulation model with transparent walls. Tests showed that the flow modifier promotes annular flow even at reduced air flow rates.

• A modified, non-dimensional parameter, *Sw*, the swirler number was proposed to describe the friction pressure drop in the annular passage in the presence of a spring flow modifier. A new correlation was obtained experimentally.

• For the two-phase flow in the presence of a spring flow modifier, the constant C in the Chisholm equation was determined experimentally. For the first time, the two-phase flow, pressure drop in a McGill heat pipe evaporator was evaluated. This procedure now provides an effective method for the estimation of the returning velocity, which was very helpful for the design of a sulfur based McGill heat pipe.

2. Theoretical analysis of the critical heat flux (CHF) in the full tunnel passage configuration

• A model for computing the CHF in a McGill heat pipe with a full tunnel evaporator was developed. The full tunnel configuration represents the extreme case scenario.

• It was shown that the CHF occurs when the vapor momentum emanating from the wetted surface is balanced by the pressure difference which serves to maintain the interface in contact with the surface.

3. Development of a sulfur based McGill heat pipe

• The sulfur/iodine mixture as described in the literature by a number of researchers was proven to be unsuitable as a working fluid in high heat flux applications.

• It is the first time that sulfur has been successfully applied in a McGill heat pipe. A sulfur based heat pipe has been shown to be viable and reliable. Researchers have been active in the search for suitable working fluids for the mid-temperature operating range for several decades. Sulfur had been abandoned by the research community. However, a viable replacement has yet to be found. A significant contribution of the present work has been to demonstrate the viability of sulfur for the mid-temperature range under high heat flux loadings.

• It was the first time that a threaded inner wall of the evaporator was used in the McGill heat pipe to distribute the liquid evenly over the walls, and to eliminate the occurrence of hot spots.

• The startup of the sulfur-based McGill heat pipe received specific focus as there are several safety issues that were uncovered and dealt with. Moreover, the startup of a highly evacuated sulfur heat pipe was found to develop hot spots near the top of the evaporator during startup. This occurrence was explained. A solution whereby a relatively small amount (e.g. 0.1 atm) of inert gas is initially charged in the pipe was shown to be viable in suppressing hot spot formation.

4. Theoretical model and experiments on a sulfur based McGill heat pipe

• A theoretical model was developed to predict the co-current flow in a McGill heat pipe. The model can be used to design and simulate the operation of a sulfur-based McGill heat pipe.

• It is the first time that high heat fluxes ($\sim 400 \text{kW/m}^2$) are handled by a sulfur based heat pipe. This has far reaching implications for those who design, build and use heat pipes.

References

- 1. F. Mucciardi et al., "McGill Heat Pipe", U.S. Paten 7, 115, 227, 2006
- 2. R. S. Gaugler, "Heat Transfer Devices", U.S. Patent 2,350348, 1944.
- G. M. Grover, "Evaporation-Condensation Heat Transfer Devices", U.S. Patent 3,229,759, 1964.
- 4. Dunn P. D., and Reay D.A., "Heat Pipe", 3rd Edition, Pergamon Press, 1982.
- 5. Peterson G. P., "An Introduction to Heat Pipes: Modeling, Testing, and Applications", John Wiley & Sons, Inc., 1994.
- Faghri A., "Heat Pipe Science and Technology", Taylor & Francis, Washington D.C., 1995.
- G. Zheng, "A Novel Flow-Modified Heat Pipe Development and Experimental Investigation", Ph.D thesis, Department of Mining, Metals and Materials Engineering, McGill University,2003
- C. Zhang, F. Mucciardi, and J.E. Gruzleski, "Controlled Cooling of Permanent Molds in the Casting of Aluminum", Light Metals Sym. Of MetSoc-CIM, Aug.2001, p431~441.
- 9. C. Zhang, F. Mucciardi, and J.E. Gruzleski, "Effects of Heat Pipe Cooling on Permanent Mold Castings of Aluminum Alloys", MetSoc-CIM, Aug.2002, p321~334.
- 10. Z. Yuan and F. Mucciardi, "Waterless Non-Consumable Thermopump Lances," International Symposium on Ecomaterials and Ecoprocesses, 2 (2003), 229-242.
- 11. Z. Yuan and F. Mucciardi, "Heat Fluxes from Aluminum Melts to Isothermal Surfaces", Light Metals (MetSoc), 2002, p. 335-349
- C. Zhang, "Controlled Cooling of Permanent Mold Castings of Aluminum Alloys", Ph.D thesis, Department of Mining, Metals and Materials Engineering, McGill University, 2003.
- K. Elalem, F. Mucciardi, and J. E. Gruzleski, "Development of Heat Pipe Technology for Permanent Mold Casting of Magnesium Alloys", COM 2002, P339~413.

- K. Elalem, F. Mucciardi and J.E. Gruzleski, "Industrial Applications of Heat Pipe Technology to the Permanent Mold Casting of Magnesium Alloys". *Light Metals*, 2003, 243-259.
- 15. P. Navarra and F. Mucciardi, "Heat Pipe Cooling of Copper Block Tapholes," International Symposium on Ecomaterials and Ecoprocesses, 2 (2003), 217-228.
- 16. P. Navarra, H. Zhao and F. Mucciardi, "Mathematical Modelling and Design of Heat Pipe-Cooled Matellurgical Furnace Equipment", TMS,2005
- 17. J.P.Chiou, "Experimental Investigation of the Augmentation of Forced Convection Heat transfer in a Circular Tube Using Spiral Spring Inserts", Transactions of the ASME, Vol.109, May 1987, P300-307
- 18. A. Garcia, P.G. Vicente and A. Viedma, "Experimental Study of Heat Transfer Enhancement with Wire Coil Inserts in Laminar-Transition-Turbulent Regimes at Different Prandtl Numbers", Int. J. Heat Mass Transfer, V48,2005 4640-4651
- 19. Yukisugu Shoji, Hyozo Sato and D.R. Oliver, "Heat Transfer Enhancement in Round tube using Wire Coil: Influence of Length and Segmentation", Heat Transfer-Asian Research, 32(2), 2003, P99-107
- 20. Y.Katto, "Critical Heat Flux in Pool Boiling", in Pool and External Flow Boiling, V.K. Dhir and A.E. Bergles eds., ASME, New York 1992 p151-164
- 21. V. K. Dhir, "Pool Boiling Heat Transfer: Recent Advances and Expectations for the Future", Enhanced and Multiphase Heat Transfer: A Festschrift for A.E. Bergles, R.M. Manglik and A.D. Kraus eds., 1996 New York, p99-126
- 22. A.E. Bergles, "What is the Real Mechanism of CHF in Pool Boiling", Pool and External Flow Boiling, V. K. Dhir and A.E. Bergles eds, ASME, 1992 New York, p151-164;
- 23. G.F. Hewitt, Burnout, "Handbook of Multiphase Systems", G. Hetsroni ed., McGraw-Hill Book Company, New York, 1982 Chapter 6.4
- 24. L.S. Tong and Y.S. Tang, "Boiling Heat Transfer and Two Phase Flow", 2nd ed., Taylor and Francis, Washington ,DC,1997;
- 25. J.G. Collier and J.R. Thome, "Convective Boiling and Condensation",3rd., McGraw-Hill, New York,1994
- 26. V.P. Carey, "Liquid Vapor Phase Change Phenomena", Hemisphere Publishing

Corporation, Washington ,DC,1992

- 27. Y.Katto, "Critical Heat Flux", Int. J. Multiphase Flow, V20, 1994, 53-90;
- N. Zuber, "Hydrodynamic Aspects of Boiling Heat Transfer", Report AECU-4439, Atomic Energy Commission (US), 1958
- N. Zuber, M. Tribus, and J.W. Westwater, "The Hydrodynamic Crisis in Pool Boiling of Saturated Liquids", International developments in heat transfer, ASME, New York, 1963, 230-236
- 30. V. K. Dhir and S.P. Liaw, "Framework for a Unified Model for Nucleate and Transition Pool Boiling", J. Heat Transfer V111,1989, 739-745
- 31. C. Unal, P. Sadasivan, and R.M. Nelson, "On the Hot-Spot Controlled Critical Heat Flux Mechanism in Pool Boiling of Saturated Fluids", Pool and External Flow Boiling, V.K. Dhir and A.E. Bergles eds., New York, ASME, 1992 193-201;
- 32. Md.Ashraful Islam, M.Monde, M.Z. Hasan and Y.Mitsutake, "Experimental Study of Critical Heat Flux in Concentric-Tube Open Thermosyphon", Int. J. of Heat and Mass Transfer V41(1998) 3691-3704;
- 33. L. Cheng and G. Xia, "Experimental Study of CHF in a Vertical Spirally Internally Ribbed Tube Under the Condition of High Pressures", Int. J. Thermal Science, V41, 2002 396-400;
- 34. G.P. Celata, K.Mishima and G.Zummo, "Critical Heat Flux Prediction for Saturated Flow Boiling of Water in Vertical Tubes", Int. J. of Heat and Mass Transfer V44(2001) 4323-4331;
- 35. J. Boscary, J. Fabre, and J. Schlosser, "Critical Heat Flux of Water Subcooled Flow in One-Side Heated Swirl Tubes", Int. J. of Heat and Mass transfer V42(1999) 287-301;
- 36. J.C. Sturgis and I. Mudawar, "Critical Heat Flux in a Long, Rectangular Channel Subject to One-Side Heating--II. Analysis of Critical Heat Flux Data", Int. J. of Heat and Mass transfer V42(1999) 1849-1862;
- 37. W. G. Anderson, "Evaluation of Heat Pipes in the Temperature Range of 450 to 700K", Space Technology and Applications International Forum-STAIF 2005, Edited by M. S. El-Genk, American Institute of Physics, 171-178;
- 38. E. Saaski and L. Tower, "Two Phase Working Fluids for the Temperature Range 100~350°C", American Institute of Aeronautics and Astronautics, 12th Themophysics

Conference, Albuquerque, NM., June 27~29, 1977;

- 39. J. H. Rosenfeld, and J. E. Lindemuth, "Sulfur Heat Pipes for 600K Space Heat Rejection System", Final Report for NASA LERC, Contract No. NAS3-26324, 1992
- 40. F. Polasek, P. Stulc, "Heat Pipe for the Temperature Range from 200 to 600°C", Proc., Second International Heat Pipe Conference, 1976, Bologna, Italy, p711~718;
- 41. W. G. Anderson, J. H. Rosenfeld, D. Angirasa and Y. Me, "Evaluation of Heat Pipe Working Fluids in the Temperature Range 450 to 700K", Space Technology and Applications International Forum-STAIF 2005, Edited by M. S. El-Genk, American Institute of Physics, 20-27;
- 42. J.G. Reed, C.L. Lien, "Modeling of the Two-Phase Closed Thermosyphon", Transactions of the ASME, Vol.109, Aug. 1987, p722~730;
- 43. M.Monde, "Analytical Study of Critical Heat Flux in Two-Phase Thermosyphon: Relationship Between Maximum Falling Liquid Rate and Critical Heat Flux" Transactions of the ASME, Vol.118, May 1996, p422~428;
- 44. M. S. El-Genk, H.H. Saber, "Determination of Operation Envelopes for Closed Two-Phase Thermosyphons", J. Heat Mass Transfer, V42 1999, 889-903;
- 45. M. Mahfoud, F.Mucciardi and J.Gruzleski, "Numerical Simulation of Heat Pipe Assisted Solidification", Int. J. Cast Metals Research, V18, 2005;
- 46. S.I. Haider, Y. K. Joshi, and W.Nakayama, "A Natural Circulation Model of the Closed Loop Two-Phase Thermosyphon for Electronics Cooling"; J. Heat Transfer, ASME, V124,2002 881-890;
- 47. P. Navarra, Hujun Zhao, and Frank Mucciardi, "Improvement of Flow Modifiers in McGill Heat Pipes", Multiphase Phenomena in Materials Processing Symposium, TMS, 2004 p33~44
- 48. J.G. Collier and J.R. Thome, "Convective Boiling and Condensation", 3rd ed., Clarendon Press, Oxford, UK, 1994;
- 49. Zuber N., and Findlay J. A., "Average Volumetric Concentration in Two-phase Flow System", ASME J. Heat Transfer, Ser. C, 87, 453-468, 1965
- 50. P.B. Whalley, Boiling, "Condensation and Gas-Liquid Flow", Oxford University Press, 1987
- 51. R. W. Lockhart and R.C. Martinelli, "Proposed Correlation of Data for Isothermal
Two-Phase Two-Component Flow in Pipes", Chem. Eng. Prog. 1949 V45 P39-48

- D. Chisholm, "A Theoretical Basis for the Lockhart-Martinelli Correlation for Two-Phase Flow", Int. J. Heat Mass Transfer, V10 1967 1767-1778
- 53. H.Lamb, "Hydrodynamics" (6th Edn), P.371, Dover Publications, New York (1945);
- 54. J.C. Sturgis and I. Mudawar, "Critical heat flux in a long, retangular Channel subjected to one-side heating-II, Analysis of Critical Heat Flux Data, Int. J. of Heat and Mass transfer V42(1999) 1849~1862;
- 55. J.C. Sturgis and I. Mudawar, "Critical heat flux in a long, curved channel subjected to concave heating, Int. J. of Heat and Mass transfer V42(1999) 3831-3848;

Appendix I: Matlab code for CHF calculation in full tunnels

% Using Euler single-step method in solving the ODE clc clear all format long

%Input the geometric dimensions Ro=0.0141; %the inner radius of evaporator Ri=0.007; %the outer radius of returning pipe ri=0.005; H=Ro-Ri; %the gap between concentric tubes W=0.09; %the distance between the fin h=0.27; % the height of evaporator phai=W/((Ro+Ri)*pi);%the angle after spreading the spring fin Ur=0.5; % the velocity of liquid from returning pipe U=(Ur*pi*ri^2)/(W*H);%the inlet velocity of the spring channel

%Input the estimate heat flux qq=2150000

% Input parameters of the working materials pv=0.5977; %the density of vapor pf=958.4; % the density of liquid muv=0.00001203;%the viscosity of vapor muf=0.002825; %the viscosity of liquid hfg=2257100; %the latent heat of saturated liquid sigma=0.05886; %the surface tension coefficient of saturated liquid

N=20000;%Define the number of elements for R-K method

K=0; hh=h*2*pi*Ro/(64*N*W);

%Define iteration times for I=1:6

for i=1:2000 if i==1 x(i)=0;y(i)=0: uv(i)=0; uf(i)=U; else x(i)=(i-1)*hh; y(i)=x(i)/100; uv(i)=qq*x(i)/(pv*y(i)*hfg); uf(i)=(U*H/(H-y(i)))-qq*x(i)/(pf*(H-y(i))*hfg); Dhv(i)=2*W*y(i)/(W+y(i));Dhf(i)=2*W*(H-y(i))/(W+H-y(i));fv(i)=0.079*(pv*uv(i)*Dhv(i)/muv)^(-0.25)+0.0075*(Dhv(i)/(2*Ro))^0.5; $ff(i)=0.079*(pf*uf(i)*Dhf(i)/muf)^(-0.25)+0.0075*(Dhf(i)/(2*Ro))^0.5;$ Twv(i)=(1/8)*pv*(uv(i))^2*fv(i); Twf(i)=(1/8)*pf*(uf(i))^2*ff(i); Ti(i)=(1/4)*pv*(uv(i)^2-uf(i)^2);

 $y1(i)=y(i-1)+hh^{*}((2^{*}uv(i)^{*}qq^{*}(H-y(i))/(hfg^{*}H^{*}y(i))+Twv(i)^{*}(H-y(i))^{*}(W+2^{*}y(i))/(H^{*}W^{*}y(i))+(2^{*}uf(i)^{*}qq^{*}Ri^{*}log((Ro-y(i))/Ri))/(hfg^{*}H^{*}(H-y(i)))-(y(i)^{*}pv^{*}9.8^{*}sin(phai))/H-(pf^{*}9.8^{*}(H-y(i))^{*}sin(phai))/H-Twf(i)^{*}(Ri/Ro)^{*}(1/H+2^{*}(H-y(i))/(W^{*}H))+Ti(i)/y(i))/(pv^{*}uv(i)^{2}(y(i)-H)/(H^{*}y(i))+pf^{*}uf(i)^{2}(H-Ro)/(H^{*}(Ro-y(i)))+2^{*}uv(i)^{*}qq^{*}x(i)^{*}(H-y(i))/(hfg^{*}H^{*}y(i)^{2})+(2^{*}uf(i)^{*}(U^{*}H^{*}pf^{*}hfg-qq^{*}x(i)))^{*}Ri^{*}log((Ro-y(i))/Ri)/(H^{*}hfg^{*}(H-y(i))^{*}));$

if y1(i)-y(i)>0

while (y1(i)-y(i))>y(i)/200000

K=K+1; y(i)=y(i)+(y1(i)-y(i))/20;uv(i)=qq*x(i)/(pv*y(i)*hfg); $uf(i)=(U^{H}/(H-y(i)))-qq^{x}(i)/(pf^{(H-y(i)))+hfg);$ Dhv(i)=2*W*y(i)/(W+y(i));Dhf(i)=2*W*(H-y(i))/(W+H-y(i));fv(i)=0.079*(pv*uv(i)*Dhv(i)/muv)^(-0.25)+0.0075*(Dhv(i)/(2*Ro))^0.5; ff(i)=0.079*(pf*uf(i)*Dhf(i)/muf)^(-0.25)+0.0075*(Dhf(i)/(2*Ro))^0.5; Twv(i)=(1/8)*pv*(uv(i))^2*fv(i); Twf(i)=(1/8)*pf*(uf(i))^2*ff(i); Ti(i)=(1/4)*pv*(uv(i)^2-uf(i)^2);

y1(i)=y(i-1)+hh*((2*uv(i)*qq*(H-y(i))/(hfg*H*y(i))+Twv(i)*(H-y(i))*(W+2*y(i))/(H*W*y(i))+(2*uf(i)*qq*Ri*lo g((Ro-y(i))/Ri))/(hfg*H*(H-y(i)))-(y(i)*pv*9.8*sin(phai))/H-(pf*9.8*(H-y(i))*sin(phai))/H-Twf(i)*(Ri/Ro)*(1/H+ 2*(H-y(i))/(W*H))+Ti(i)/y(i))/(pv*uv(i)^2*(y(i)-H)/(H*y(i))+pf*uf(i)^2*(H-Ro)/(H*(Ro-y(i)))+2*uv(i)*qq*x(i)*(H-y(i))/(hfg*H*y(i)^2)+(2*uf(i)*(U*H*pf*hfg-qq*x(i))*Ri*log((Ro-y(i))/Ri))/(H*hfg*(H-y(i))^2))); end

elseif y1(i)-y(i)<0 while (y(i)-y1(i))>y(i)/200000 K=K+1; y(i)=y(i)-(y(i)-y1(i))/20;uv(i)=qq*x(i)/(pv*y(i)*hfg); uf(i)=(U*H/(H-y(i)))-qq*x(i)/(pf*(H-y(i))*hfg); Dhv(i)=2*W*y(i)/(W+y(i)); Dhf(i)=2*W*(H-y(i))/(W+H-y(i)); fv(i)=0.079*(pv*uv(i)*Dhv(i)/muv)^(-0.25)+0.0075*(Dhv(i)/(2*Ro))^0.5; ff(i)=0.079*(pf*uf(i)*Dhf(i)/muf)^(-0.25)+0.0075*(Dhf(i)/(2*Ro))^0.5; Twv(i)=(1/8)*pv*(uv(i))^2*fv(i); Twf(i)=(1/8)*pf*(uf(i))^2*ff(i); $Ti(i)=(1/4)*pv*(uv(i)^2-uf(i)^2);$

y1(i)=y(i-1)+hh*((2*uv(i)*qq*(H-y(i))/(hfg*H*y(i))+Twv(i)*(H-y(i))*(W+2*y(i))/(H*W*y(i))+(2*uf(i)*qq*Ri*lo g((Ro-y(i))/Ri))/(hfg*H*(H-y(i)))-(y(i)*pv*9.8*sin(phai))/H-(pf*9.8*(H-y(i))*sin(phai))/H-Twf(i)*(Ri/Ro)*(1/H+ 2*(H-y(i))/(W*H))+Ti(i)/y(i))/(pv*uv(i)^2*(y(i)-H)/(H*y(i))+pf*uf(i)^2*(H-Ro)/(H*(Ro-y(i)))+2*uv(i)*qq*x(i)*(H-y(i))/(hfg*H*y(i)^2)+(2*uf(i)*(U*H*pf*hfg-qq*x(i))*Ri*log((Ro-y(i))/Ri))/(H*hfg*(H-y(i))^2)));

end%end of "while" end%end of "if" end%end of "if" end%end of "for i"

```
LL=length(x);
xx=x(:,2:LL);
yy=y1(:,2:LL);
uv=qq*xx./(pv*yy*hfg);
uf=(U*H./(H-yy))-qq*xx./(pf*(H-yy)*hfg);
figure(1);
plot(xx,yy);
figure(2);
plot(xx,uv,'--',xx,uf,':');
```

%Find the point x0 where uv=uf mm=abs(uv-uf); for i=1:length(xx) mm0(i)=mm(i);if mm0(i)==min(mm) x0=xx(i)S=i; end

end

%Find the matrix of critical wavelength at some x for j=1:(1980) NN=0; z(j)=xx(j+2);

```
zz(j)=yy(j+2);
   uv1(j)=uv(j+2);
   uf1(j)=uf(j+2);
   rr(j)=z(j);
NNf(j)=(Ro-zz(j))/Ri;
   NNv(j)=(Ro-zz(j))/Ro;
   PP(j)=4*pi*(Ro-zz(j))/rr(j);
   ppf(j)=pf*((NNf(j)^PP(j)+1)/(NNf(j)^PP(j)-1));
   ppv(j)=pv*((NNv(j)^PP(j)+1)/(1-NNv(j)^PP(j)));
   rr1(j)=2*pi*sigma*(1/ppf(j)+1/ppv(j))/((uv1(j)-uf1(j))^2);
  if rr(j)-rr1(j)<0
        while rr1(j)-rr(j)>rr(j)/20000
             NN=NN+1;
             rr(j)=rr(j)+(rr1(j)-rr(j))/2;
             NNf(j)=(Ro-zz(j))/Ri;
             NNv(j)=(Ro-zz(j))/Ro;
             PP(j)=4*pi*(Ro-zz(j))/rr(j);
             ppf(j)=pf*((NNf(j)^PP(j)+1)/(NNf(j)^PP(j)-1));
             ppv(j)=pv*((NNv(j)^PP(j)+1)/(1-NNv(j)^PP(j)));
             rr1(j)=2*pi*sigma*(1/ppf(j)+1/ppv(j))/((uv1(j)-uf1(j))^2);
             %rr(j)=rr1(j);
       end
  elseif rr(j)-rr1(j)>0
       while rr(j)-rr1(j)>rr(j)/20000
            NN=NN+1;
            NNf(j)=(Ro-zz(j))/Ri;
            NNv(j)=(Ro-zz(j))/Ro;
            rr(j)=rr(j)-(rr(j)-rr1(j))/2;
            PP(j)=4*pi*(Ro-zz(j))/rr(j);
            ppf(j)=pf^{*}((NNf(j)^{PP}(j)+1)/(NNf(j)^{PP}(j)-1));
            ppv(j)=pv*((NNv(j)^PP(j)+1)/(1-NNv(j)^PP(j)));
           rr1(j)=2*pi*sigma*(1/ppf(j)+1/ppv(j))/((uv1(j)-uf1(j))^2);
           %rr(j)=rr1(j);
       end
    end
end
%Get the critical wavelength
       DD=length(z);
       ww=((z-rr1)-x0*ones(1,DD));
      figure(3)
      plot(z,ww)
for e=1: \text{length}(z)
     ww0(e)=abs(ww(e));
     if ww0(e)==min(abs(ww))
          rcr=rr1(e)
          xcr=z(e)
          yyy=zz(e)
          е
     end
end
 %Calculate the lift-off heat flux
 b=0.25;
 ppav=4*pi*sigma*yyy*sin(b*pi)/(b*rcr^2);
qql=hfg*sqrt(ppav*pv);
qqcr=b*qql
%Set iterative error evaluation
   if abs(qq-qqcr)<qq/1000
        return
   end
 qq=qqcr
end
```

Appendix II: Matlab code for the sulfur based

McGill heat pipe calculation

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Di=0.040; %the inner diameter of the evaporator,m Do=0.048; %the outer diameter of the evaporator,m do=0.021; %the outer diameter of the returning pipe,m di=0.017; %the inner diameter of the returning pipe,m d=0.003; %the diameter of the spring,m Dheater=0.0175; Dthermol=0.012; Dh=(Di^2-do^2-Dheater^2-Dthermol^2)/(Di+do+Dheater+Dthermol);%define the hydraulic diameter,m A=(pi/4)*(Di^2-do^2-Dheater^2-Dthermol^2); %the flow area H=0.030;%the pitch of the spring,m L=0.1500; %the length of the evaporator,m N=400; LL=0.400;%the vertical distance of the returning line,m

NN=1000; %define the elements along the evaporator

AFur=pi*0.4*0.5+pi*0.2^2; Apipe=pi*L*Do+pi*(Do/2)^2; Sw=pi*(Di-d)*d/(H*Dh);%the swirler parameter

%%%%%%%%%%Input the working conditions%%%%%%%%% Tamb=1350;%input ambient temperature; hoc=165;%input outer heat transfer coefficient Km=22; %the conductivity of steel,W/m c Ti=19; %input the coolant inlet temperature C Gc=107*28.317*0.001/60; %input the coolant flow rate,m3/s

Vre=0.57; %guess the returning velocity, m/s

%%%%Calculate the subcooled temperature of the condensing%%%%%%%%% Denlc=1900.8-0.816*Tsat; %liquid density in the reservoir HS=MS/(Denlc*3.14159*DDi^2/4)*0.35;%liquid level in the reservoir, 0.55 is the correcting factor Resub=1.203*Gc/(0.25*3.14159*(DSi+DDo)*0.0000181);%Renolds number in the subcooling section Prsub=0.70;% Pr number of cooling air Nusub=0.023*Resub^0.8*Prsub^0.4;

hsub=Nusub*2.59/(DSi-DDo); %heat transfer coefficient in the subcooling section

Asub=3.14159*DDo*HS;% subcooling area Cps=(3.56+6.96*0.001*Tsat)*4.1868*(1/0.016);%specific heat of sulfur,j/kg c Cpa=1005;%specific heat of air Denl=1900.8-0.816*Tsat;%density of sulfur liquid Are=3.14159*di^2/4; mmm=hsub*Asub/(Denl*Vre*Are*Cps*2); Tavesub=(Tsat+20*mmm)/(mmm+1); Tsub=Tavesub;

for j=1:40 %Iteration for returning velocity

> mug=1e-5*(0.0053*Tsat+0.4)*(1/9.81);% the sulfur vapor viscosity sigma=0.01*(-0.0067*Tsat+8);% The surface tension Pr=Cpl*mul/Kl; % Pr number

Gm=Vre*pi*(di^2/4)*Denl;% the returning mass flow,kg/s Gv=Vre*(di^2/(Di^2-do^2-Dheater^2-Dthermol^2))*Denl;% the initial mass flux, kg/m2s

for i=1:400 % Iteration for nodes on the evaporator

TL(1)=Tsub;

Tw(i)=TL(i)+35; for a=1:2000 Rel(i)=Gv*Dh/mul; %local liquid Re Nu(i)=0.303*(d/Dh)^0.12*(H/Dh)^(-0.377)*(Rel(i)^0.72)*Pr^0.37;% Nu number hl(i)=Nu(i)*KI/Dh; %subcooled part heat transfer coefficient qint(i)=270000;%input initial guess of heat flux

%Calculate the heat flux for t=1:2000 hpool(i)=55*Pr^0.12*(log10(Pr))^(-0.55)*32^(-0.5)*qint(i)^(0.67); Bo(i)=qint(i)/(Hfg*Gv);% Bo number E(i)=1+24000*Bo(i)^1.16; Rei(i)=Gv*Dh/mul; Si(i)=1/(1+1.15*10^(-6)*E(i)^2*Rei(i)^1.17); qqint(i)=hl(i)*(Tw(i)-TL(i))+Si(i)*hpool(i)*(Tw(i)-Tsat);

if abs(qint(i)-qqint(i))<qint(i)/500 break end%end of if qint(i)=qqint(i); end%end of t Twso(i)=qint(i)*log(Do/Di)*Di/(2*Km)+Tw(i); % the outer wall temperature ho(i)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twso(i)+273)^2)*(Tamb+273+Twso(i)+273)+hoc; qout(i)=ho(i)*(Tamb-Twso(i)); if abs(qout(i)-qqint(i))<qout(i)/500 break end%end of if

Tw(i)=Tw(i)+2*(qout(i)-qqint(i))/Hfg;

end%end of a

TL(i+1)=(qout(i)*pi*Do*L/N)/(Gm*Cpl)+TL(i); Tsatloc(i)=3268.2/(7.43287-(1/0.981)*log10((Psat+Denl*9.81*(LL-(i/NN)*LL))/100000)); if TL(i+1)>Tsatloc(i)

TL(i+1)=Tsatloc(i); Nsub(i)=i+1; break end% end of if

end%end of i

NNs=nonzeros(Nsub); NS=NNs(1);

qout; TWS=Twso(1:NS-1); TLS=TL(1:NS);

for k=1:(N+1-NS)

if k==1 TLb(k)=TLS(NS); Twb(k)=TLb(k)+35; PLb(k)=100000*10^(0.981*(7.43287-3268.2/TLb(k)));%the local saturated pressure Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa

 $hfz(k)=0.00122*((Twb(k)-TLb(k))^{0.24}*((Pwb(k)-PLb(k))^{0.75})*(Cpl^{0.45})*(Denl^{0.49})*(Kl^{0.79})/(sigma^{0.5}*Hfg^{0.24}*mul^{0.29}*Deng^{0.24});%Nucleate boiling heat transfer coefficient$

 $Nub(k)=0.303*(d/Dh)^{0.12}*(H/Dh)^{-(-0.377)}*(Relb(k)^{-0.72})*Pr^{-0.37};%$ Nu number hlb(k)=Nub(k)*Kl/Dh; %subcooled part heat transfer coefficient

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%local overall heat transfer coefficient

qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

> Xtt(k)=0; x(k)=0;

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k)));

while abs(Twb(k)-Twwb(k))>0.01 if Twb(k)>Twwb(k) Twb(k)=Twwb(k)+(Twb(k)-Twwb(k))/2; Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa

hfz(k)=0.00122*((Twb(k)-TLb(k))^0.24)*((Pwb(k)-PLb(k))^0.75)*(Cpl^0.45)*(Denl^0.49)*(Kl^0.79)/(sigma^0.5
*Hfg^0.24*mul^0.29*Deng^0.24);%Nucleate boiling heat transfer coefficient
hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%local overall heat transfer coefficient
qbint(k)=hb(k)*(Twb(k)-TLb(k));
Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k))); elseif Twb(k)<Twwb(k) Twb(k)=Twwb(k)-(Twwb(k)-Twb(k))/2; Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa

hfz(k)=0.00122*((Twb(k)-TLb(k))^0.24)*((Pwb(k)-PLb(k))^0.75)*(Cpl^0.45)*(Denl^0.49)*(Kl^0.79)/(sigma^0.5 *Hfg^0.24*mul^0.29*Deng^0.24);%Nucleate boiling heat transfer coefficient

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%local overall heat transfer coefficient

qbint(k)=hb(k)*(Twb(k)-TLb(k));

Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k)));

end %end if in the while end %end while

elseif k=2

TLb(k)=3268.2/(7.43287-(1/0.981)*log10((Psat+Denl*0.8*9.81*(LL-((NS+k)/NN)*LL))/100000));

Twb(k)=TLb(k)+30;%initial guess of wall temperature PLb(k)=100000*10^(0.981*(7.43287-3268.2/TLb(k)));%the local saturated pressure Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa

Xtt(k)=0; %F(i)=2.35*0.213^0.736;%factor for forced convection F(k)=1; Relb(k)=Gv*Dh/mul; %local liquid Re S(k)=1/(1+2.53e-06*(Relb(k)*F(k)^1.25)^1.17); %factor for nucleate boiling Nub(k)=0.303*(d/Dh)^0.12*(H/Dh)^(-0.377)*(Relb(k)^0.72)*Pr^0.37;% Nu number

hlb(k)=Nub(k)*Kl/Dh; % heat transfer coefficient

 $hfz(k)=0.00122*((Twb(k)-TLb(k))^{0.24}*((Pwb(k)-PLb(k))^{0.75}*(Cpl^{0.45})*(Denl^{0.49})*(Kl^{0.79})/(sigma^{0.5}*Hfg^{0.24}mul^{0.29}Deng^{0.24});$ %Nucleate boiling heat transfer coefficient

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%overall heat transfer coefficient qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

x(k)=x(k-1)+(L/N)*hb(k)*(Twb(k)-TLb(k))*pi*Di/(Gm*Hfg);

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k)));

while abs(Twb(k)-Twwb(k))>0.01
if Twb(k)>Twwb(k)
Twb(k)=Twwb(k)+(Twb(k)-Twwb(k))/2;

Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa

hfz(k)=0.00122*((Twb(k)-TLb(k))^0.24)*((Pwb(k)-PLb(k))^0.75)*(Cpl^0.45)*(Denl^0.49)*(Kl^0.79)/(sigma^0.5 *Hfg^0.24*mul^0.29*Deng^0.24);%Nucleate boiling heat transfer coefficient

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%overall heat transfer coefficient qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k)); x(k)=x(k-1)+(L/N)*hb(k)*(Twb(k)-TLb(k))*pi*Di/(Gm*Hfg);

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k))); elseif Twb(k)<Twwb(k) Twb(k)=Twwb(k)-(Twwb(k)-Twb(k))/2;

 $Pwb(k)=100000*10^{(0.981*(7.43287-3268.2/Twb(k)))}; where saturated pressure at the wall; Pahfz(k)=0.00122*((Twb(k)-TLb(k))^{0.24}*((Pwb(k)-PLb(k))^{0.75})*(Cpl^{0.45})*(Denl^{0.49})*(Kl^{0.79})/(sigma^{0.5})*(Hfg^{0.24}*mul^{0.29})*(Denl^{0.24}); where a statement of the saturated pressure at the sat$

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%overall heat transfer coefficient qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

x(k)=x(k-1)+(L/N)*hb(k)*(Twb(k)-TLb(k))*pi*Di/(Gm*Hfg);

Twwb(k) = TLb(k) + (1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k)) + log(Do/Di)*(1/(2*Km)) + 1/(Di*hb(k)));

end %end of if in the "while" end%end of while

else

TLb(k)=3268.2/(7.43287-(1/0.981)*log10((Psat+Denl*0.8*9.81*(LL-((NS+k)/NN)*LL))/100000));

Twb(k)=TLb(k)+10;%initial guess of wall temperature PLb(k)=100000*10^(0.981*(7.43287-3268.2/TLb(k)));%the local saturated pressure Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa

Xtt(k)=((1-x(k-1))/x(k-1))^0.9*(Deng/Denl)^0.5*(mul/mug)^0.1; if 1/(Xtt(k))<0.1 F(k)=1;

else

 $F(k)=2.35*(1/Xtt(k)+0.213)^0.736$;%factor for forced convection end

Relb(k)=Gv*(1-x(k-1))*Dh/mul; %local liquid Re S(k)=1/(1+2.53e-06*(Relb(k)*F(k)^1.25)^1.17); %factor for nucleate boiling Nub(k)=0.303*(d/Dh)^0.12*(H/Dh)^(-0.377)*(Relb(k)^0.72)*Pr^0.37;% Nu number

hlb(k)=Nub(k)*Kl/Dh;

 $\label{eq:hfz(k)=0.00122*((Twb(k)-TLb(k))^{0.24}*((Pwb(k)-PLb(k))^{0.75}*(Cpl^{0.45})*(Denl^{0.49})*(Kl^{0.79})/(sigma^{0.5}*Hfg^{0.24}*mul^{0.29}*Deng^{0.24});% Nucleate boiling heat transfer coefficient$

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%overall heat transfer coefficient qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

x(k)=x(k-1)+(L/N)*hb(k)*(Twb(k)-TLb(k))*pi*Di/(Gm*Hfg);

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k)));

while abs(Twb(k)-Twwb(k))>0.001
if Twb(k)>Twwb(k)
Twb(k)=Twwb(k)+(Twb(k)-Twwb(k))/2;
Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa
Xtt(k)=((1-x(k-1))/x(k-1))^0.9*(Deng/Denl)^0.5*(mul/mug)^0.1;
if 1/(Xtt(k))<0.1
F(k)=1;
else</pre>

 $F(k)=2.35*(1/Xtt(k)+0.213)^0.736$;%factor for forced convection end

hfz(k)=0.00122*((Twb(k)-TLb(k))^0.24)*((Pwb(k)-PLb(k))^0.75)*(Cpl^0.45)*(Denl^0.49)*(Kl^0.79)/(sigma^0.5

*Hfg^0.24*mul^0.29*Deng^0.24);%Nucleate boiling heat transfer coefficient

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%overall heat transfer coefficient qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

x(k)=x(k-1)+(L/N)*hb(k)*(Twb(k)-TLb(k))*pi*Di/(Gm*Hfg);

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k)));

elseif Twb(k)<Twwb(k) Twb(k)=Twwb(k)-(Twwb(k)-Twb(k))/2; Pwb(k)=100000*10^(0.981*(7.43287-3268.2/Twb(k)));%the saturated pressure at the wall;Pa Xtt(k)=((1-x(k-1))/x(k-1))^0.9*(Deng/Denl)^0.5*(mul/mug)^0.1; if 1/(Xtt(k))<0.1 F(k)=1; else

F(k)=2.35*(1/Xtt(k)+0.213)^0.736;%factor for forced convection

end

 $\label{eq:hfz(k)=0.00122*((Twb(k)-TLb(k))^{0.24)*((Pwb(k)-PLb(k))^{0.75)*(Cpl^{0.45})*(Denl^{0.49})*(Kl^{0.79})/(sigma^{0.5})*(Hfg^{0.24*mul^{0.29*}Deng^{0.24});%Nucleate boiling heat transfer coefficient}$

hb(k)=F(k)*hlb(k)+S(k)*hfz(k);%overall heat transfer coefficient qbint(k)=hb(k)*(Twb(k)-TLb(k)); Twbo(k)=qbint(k)*log(Do/Di)*Di/(2*Km)+Twb(k);

hob(k)=0.75*5.67*10^(-8)*((Tamb+273)^2+(Twbo(k)+273)^2)*(Tamb+273+Twbo(k)+273)+hoc; qbout(k)=hob(k)*(Tamb-Twbo(k));

x(k)=x(k-1)+(L/N)*hb(k)*(Twb(k)-TLb(k))*pi*Di/(Gm*Hfg);

Twwb(k)=TLb(k)+(1/(Di*hb(k)))*(Tamb-TLb(k))/(1/(Do*hob(k))+log(Do/Di)*(1/(2*Km))+1/(Di*hb(k))); end end %end of while end%end of if(from k==1)

end %end of for k

zzs=qout(1:(NS-1));%local heat flux in the subcooled part

zzb=hb.*(Twb-TLb);%local heat flux in the boiling part zzzb=qbout; QS=sum(zzs*pi*Do*L/N);%heat load in the subcooled part QB=sum(zzzb*pi*Do*L/N);% heat load in the boiling part

nn=N-NS-1;

x1=x(2:(nn+1));% the quality

aaa=1./(1+(1-x1)./(x1)*(Deng/Denl)^0.89*(mul/mug)^0.18); aa=(Denl/Deng)*x1./(1+(Denl/Deng-1)*x1); %calculate the void fraction with homogeneous model

DP=Denl*9.81*LL; %the pressure head for returning flow,pa dpgs=9.81*Denl*(NS/N)*L; %gravitational pressure drop in the subcooled part

fss=16/Resub*(1+12.1*Sw^1.13); dpfs=0.5*fss*((NS/N)*L)/Dh*Den1*(Vre*(di^2/(Di^2-do^2-Dheater^2-Dthermol^2)))^2;

ddpg=9.81*(Deng*aa+(1-aa)*Denl)*(L/(N));

dpg=sum(9.81*(Deng*aa+(1-aa)*Denl)*(L/(N)))+(aa(nn)*Deng+(1-aa(nn))*Denl)*9.81*(LL-L);%the gravitational pressure drop in the evaporator,pa

ddpfl=fsl*0.5.*(Gv*(1-x1)).^2/(Denl*Dh);%pressure gradient of liquid phase pa/m

Reg=x1*Gm*Dh/(mug*A);%the vapor phase Renolds number fsg=0.0791*Reg.^(-0.25)*(1+12.1*Sw^1.13); %friction factor when only vapor flows through

fsg(1)=1;

pril.

ddpfg=fsg*0.5.*(x1*Gv).^2/(Deng*Dh); %pressure gradient of vapor phase,pa/m ddpfg(1)=0.0000001; XX=ddpf1/ddpfg; %Lochart and Martinelli parameter;

C=12*(1-0.565*Sw^0.312); %the constant in Lockhart and Martinelli model

Phail=1+C./(sqrt(XX))+1./XX; %the multiplier in Lockhart and Martinelli model

ddpftp=Phail.*ddpfl; %two-phase frictional pressure gradient

dpftp=sum(ddpftp*(L/N))+ddpftp(nn)*(LL-L); %two-phase frictional pressure drop

dpmtp=sum(ddpmtp*(L/N))+ddpmtp(nn)*(LL-L); %momentum pressure drop

Rere=Vre*Denl*di/mul;

VVre=(1/(2*(1+1.25)))*(-61*mul*LL/(Denl*di^2)+sqrt((61*mul*LL/(Denl*di^2))^2+8*(DP-dpgs-dpfs-dpg-dpftp -dpmtp)/Denl));%returning velocity

if abs(VVre-Vre)>0.0001 Vre=(VVre+Vre)/2;

else break end %end of if

end %end of for j

To=Ti+QH/(1.20*Gc*Cpa); DTM=((Tsat-Ti)-(Tsat-To))/log((Tsat-Ti)/(Tsat-To)); Tc=(Ti+To)/2;

Ka=0.01*(0.0075*Tc+2.4683); Rec=Resub; Prc=Prsub; Nuc=0.303*(dc/DHc)^0.12*(Hpc/DHc)^(-0.377)*(Rec^0.72)*Prc^0.37;

Hc=Nuc*Ka/DHc;

Twwc=Tsat-8;%guess the outer wall temperature of inner tubes

for m=1:40

 $\label{eq:linear} Hcc=0.728^{*}((Denl^{(Denl-Deng)}*9.81^{*}Hfg^{Kl^{3}})/(mul^{(Tsat-Twwc)}*DDi))^{0.25};\%\ the\ condensing\ coefficient\ on\ the\ shell$

%Iteration for the wall temperature of the condenser TTwwc=(Hc*Tc*DDi+Hcc*DDo*Tsat-(Tsat*(Hcc*DDo)^2/(Km/log(DDo/DDi)+Hcc*DDo)))/(Hc*DDi+(Hcc*D Do*Km/log(DDo/DDi))/(Km/log(DDo/DDi)+Hcc*DDo)); if abs(TTwwc-Twwc)>0.01 Twwc=TTwwc;

else break end %end of if

 $\sum_{i=1}^{n}$

end%end of m

A2=pi*DDi*Lc;%the inner area of the shell,m2 A22=pi*DDo*Lc;%the outer area of the shell,m2

R2=1/(Hcc*A2)+1/(2*pi*Km*Lc)*log(DDo/DDi)+1/(Hc*A22);%thermal resistance through the shell

QC=DTM*(1/R2)+(Tsat-Tsub)*Cpl*Gm

if abs(QC-QH)>1
 Tsat=Tsat-20*(QC-QH)/(Hfg*Gm*x(nn+1))
else
 break
end %end of if
n
end
mm=(LL-L)/(L/N);
PP=100000*10^(0.981*(7.43287-3268.2/TLb(nn)));
for kk=1:mm

TLLb(kk)=3263.5/(7.43287-(1/0.981)*log10((PP+Denl*(1-aa(nn))*9.81*((LL-L)-((kk)/mm)*(LL-L)))/100000));

%TLLb(kk)=TLb(nn)-3238.2/(7.43287-(1/0.981)*log10((Den1*(1-0.98))*9.81*((LL-L)-(kk/mm)*(LL-L)))/10000 0)); end

TL=[TLS';TLb';TLLb'];