



NUMERICAL ANALYSIS OF TWO PHASE FLOW IN FLAT HEAT PIPE

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ABSTRACT

Heat pipe is a thermodynamic device of very high thermal conductance. In a heat pipe the working fluid experiences phase change both in the evaporator and condenser sections. The flat heat pipe (FHP) is a transmutation of the conventional heat pipes that finds applications to deal with high density electronic cooling problem due to their high thermal conductivity and reliability where large heat flux is to be transported through limited space. The paper presents a numerical analysis of phase change phenomenon in a flat heat pipe. The heat pipe modeling involves the principal physical processes of wick-liquid dynamics and movement of resulting vapour away from the evaporator through the heat pipe vapour space to the condenser. The analysis involves the simultaneous solution of two dimensional continuity, momentum and energy equations in the liquid wick and the vapour core regions with respective boundary conditions. Vapour flow in the vapour passage is assumed to be laminar and the heat dissipation at the condenser side is by forced convection. The axial velocity distributions and velocity vector plots are obtained along the vapour core and liquid wick region at steady state conditions. Two phase mixture model is formulated to investigate the phase change phenomena in the porous wick structure. The flow equations for liquid-vapour interface are solved using appropriate numerical technique. Nusselt number and Sherwood Number are plotted against wall saturation for boiling and condensation.

NOMENCLATURE

d	Liquid wick thickness, m
h	Vapour Core Thickness, m
k	Thermal Conductivity, W/mK
K	Wick Permeability, m ²
L	Length of Heat Pipe, m
P	Pressure, Pa
s	Saturation
t	Time, s
T	Temperature, K
u	Axial Velocity, m/s

v	Transverse Velocity, m/s
x	Axial Distance, m
y	Transverse Distance, m
ε	Porosity
μ	Dynamic Viscosity, Ns/m ²
ρ	Density, kg/m ³
ϕ	Viscous Dissipation

Subscripts

a	Adiabatic
c	Condenser
e	Evaporator

1. INTRODUCTION

In recent years flat heat pipes (FHP) are widely used for electronics and space craft cooling applications due to their high heat transport capability, effective performance and adaptable structure. A flat heat pipe is an enclosed chamber whose inner surfaces are lined with a capillary wick structure with the remaining volume containing vapour. The working fluid is vaporized by the external heat source at the evaporator, the generated pressure difference drives vapor from the evaporator to the condenser, where it condenses and enters the wick structure. The capillary pressure in the wick pumps the condensed liquid back to the evaporator for re-evaporation. This internal phase-change circulation will continue as long as flow passage is not blocked and sufficient capillary pressure is maintained. To face the new challenges in electronic cooling, the heat transfer limit need to be increased by optimizing the geometric and operating parameters. Compared to small cylindrical heat pipes, flat heat pipes can be easily fixed onto PCB cooling device, because of the thickness shrinking. The thickness of the FHP can be made in the range of a few millimeters depending on the wick structure type, which can reduce the space of cooling devices significantly.

A review of the published literature on heat pipes reveals that numerical studies on heat pipes have been performed by various researchers. A detailed, three-dimensional model to analyze the thermal hydrodynamic behaviour of flat heat pipes without empirical correlations was carried out Bin et al. [1]. A computational model for the analysis of the transient operation of a flat plate heat pipe was presented by Sobhan et al. [2]. The analysis involves the solution of two-dimensional continuity, momentum and energy equations coupled with the state equation in the vapour core, the transport equations for a porous medium in the wick, and the two dimensional heat conduction equations in the heat pipe wall. A two-phase analysis is carried out by Mehta et al. [3] for the heat pipe to study the vapour dynamics of the working fluid. The compressible flow equations for vapour -phase interaction with water particle phase are solved using finite volume method. Wang and Beckermann [4] presented a boundary layer analysis in porous media considering boiling and condensation. A set of boundary layer equations that has been derived from two-phase mixture motion are transformed into differential equations by means of similarity transformation. These equations are solved for different wall saturation and velocities. A two-dimensional, unsteady formulation was presented by Unnikrishnan et al. [5] for modeling the liquid wick region with the porosity and permeability factors in the governing differential equations. The model predicted the transient distribution of the field variables in the core and wick regions. TaoufikBrahim et al. [6] presented three-dimensional simulation of cylindrical heat pipe which incorporates the phase change mechanisms at the liquid and vapour interface. The model predicted the behaviour under critical conditions where a high heat input is exposed to its evaporator region. DehaoXu et al. [7] presented a three-dimensional unsteady model of vapour-liquid two-phase flow and heat transfer in a flat-plate pulsating heat pipe (FP-PHP). The volume of fluid method is introduced to model the unsteady vapour-liquid two-phase flow in FP-PHP and the thermo-hydrodynamic characteristics under steady

unidirectional circulation condition are numerically investigated and discussed. Wang et al. [8] in their experiment investigated the effect of evaporation and condensation length on the thermal performance of the flat heat pipes. The results showed that, the thermal resistance decreased and the heat transfer limit increased with the increase of evaporation section length, and similarly the increase in the condensation section length would make the heat pipe dry out at a lower heating power, which indicated the decrease in heat transfer limit.

2. THE PHYSICAL MODEL

A schematic diagram of the flat heat pipe analyzed, including dimensions of each region is shown in the Figure 1. Working fluid chosen for the present analysis is water. The wick and wall material is assumed as copper. The porosity of wick is taken as 0.6 for the analysis, with the voids saturated with water. A uniform heat flux of 50 W/m^2 is applied externally at the evaporator section, while the condenser section is cooled by imposing a heat transfer coefficient of $1000 \text{ W/m}^2\text{K}$, at the condenser end with the initial temperature of 300 K . The adiabatic section is assumed to be insulated such that there is no heat transfer between this region and the atmosphere.

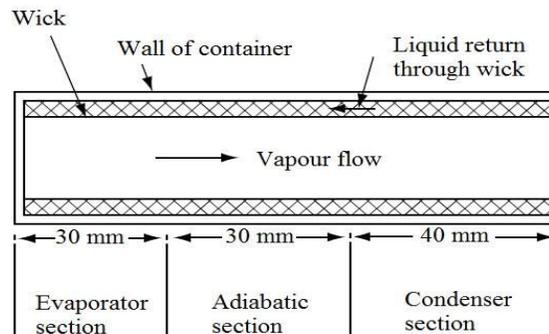


Figure 1. SCHEMATIC OF HEAT PIPE CONSIDERED

The assumptions in the model include: non-existence of body forces; two-dimensional flow fields; flow is laminar and incompressible; vapour is a saturated ideal gas, and thermodynamic equilibrium prevails at the vapour liquid interface.

Thermo-physical properties of the heat pipe material and working fluid are shown in Table 1. The thermo-physical properties are considered to be constant as the variations are negligible over the operating temperature range considered.

3. THE MATHEMATICAL MODELLING

The governing equations i.e. continuity, momentum and energy equations for vapour flow in fundamental variables for a laminar, two dimensional flow in a heat pipe is given as:

For vapour core:

$$\left(\frac{\partial u}{\partial x}\right) + \left(\frac{\partial v}{\partial y}\right) = 0 \quad (1)$$

$$\rho \left\{ \left(\frac{\partial u}{\partial t}\right) + u \left(\frac{\partial u}{\partial x}\right) + v \left(\frac{\partial u}{\partial y}\right) \right\} = - \left(\frac{\partial p}{\partial x}\right) + \mu \left\{ \left(\frac{\partial^2 u}{\partial x^2}\right) + \left(\frac{\partial^2 u}{\partial y^2}\right) \right\} \quad (2)$$

$$\rho \left\{ \left(\frac{\partial v}{\partial t} \right) + u \left(\frac{\partial v}{\partial x} \right) + v \left(\frac{\partial v}{\partial y} \right) \right\} = - \left(\frac{\partial p}{\partial y} \right) + \mu \left\{ \left(\frac{\partial^2 v}{\partial x^2} \right) + \left(\frac{\partial^2 v}{\partial y^2} \right) \right\} \quad (3)$$

$$(\rho C)_v \left\{ \left(\frac{\partial T}{\partial t} \right) + u \left(\frac{\partial T}{\partial x} \right) + v \left(\frac{\partial T}{\partial y} \right) \right\} = \left(\frac{\partial p}{\partial t} \right) + u \left(\frac{\partial p}{\partial x} \right) + v \left(\frac{\partial p}{\partial y} \right) + k_v \left\{ \left(\frac{\partial^2 T}{\partial x^2} \right) + \left(\frac{\partial^2 T}{\partial y^2} \right) \right\} + \phi \quad (4)$$

where,

$$\phi = 2\mu \left\{ \left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 \right\} + \mu \left\{ \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 \right\}$$

For the liquid wick:

The governing equations are

$$\left(\frac{\partial u}{\partial x} \right) + \left(\frac{\partial v}{\partial y} \right) = 0 \quad (5)$$

$$\frac{\rho}{\varepsilon} \left\{ \left(\frac{\partial u}{\partial t} \right) + u \left(\frac{\partial u}{\partial x} \right) + v \left(\frac{\partial u}{\partial y} \right) \right\} = - \left(\frac{\partial p}{\partial x} \right) + \frac{\mu}{\varepsilon} \left\{ \left(\frac{\partial^2 u}{\partial x^2} \right) + \left(\frac{\partial^2 u}{\partial y^2} \right) \right\} - \frac{\mu}{K} u - \frac{C_E}{K^{0.5}} \rho u |u| \quad (6)$$

$$\frac{\rho}{\varepsilon} \left\{ \left(\frac{\partial v}{\partial t} \right) + u \left(\frac{\partial v}{\partial x} \right) + v \left(\frac{\partial v}{\partial y} \right) \right\} = - \left(\frac{\partial p}{\partial y} \right) + \frac{\mu}{\varepsilon} \left\{ \left(\frac{\partial^2 v}{\partial x^2} \right) + \left(\frac{\partial^2 v}{\partial y^2} \right) \right\} - \frac{\mu}{K} v - \frac{C_E}{K^{0.5}} \rho v |v| \quad (7)$$

$$(\rho C)_m \left(\frac{\partial T}{\partial t} \right) + (\rho C)_f \left\{ u \left(\frac{\partial T}{\partial x} \right) + v \left(\frac{\partial T}{\partial y} \right) \right\} = k_m \left\{ \left(\frac{\partial^2 T}{\partial x^2} \right) + \left(\frac{\partial^2 T}{\partial y^2} \right) \right\} \quad (8)$$

Boundary Conditions

For vapor core:

At $x=0, y=0$ to $h, u=0, v=0,$

At $y=0, 0 \leq x \leq L_e, u=0, v=v_o, dT/dy=q/k$

At $y=0, L_e \leq x \leq (L_e+L_a) u=0, v=0, dT/dy=0$

At $y=0, L_a \leq x \leq L_c, u=0, v=-v_o, T=T_c$

At $x=0, y=0$ to $h, dT/dx=0$

At $x=L, y=0$ to $h, u=0, v=0$

At $y=h, 0 \leq x \leq L_e, u=0, v=-v_o, dT/dy=-q/k$

At $y=h, L_e \leq x \leq (L_e+L_a) u=0, v=0, dT/dy=0$

At $y=h, L_a \leq x \leq L_c, u=0, v=v_o, T=T_c$

At $x=L, y=0$ to $h, dT/dx=0$

For liquid wick:

At $y=-d, x=0$ to $L, u=0, v=0$

At $y=(h+d), x=0$ to $L, u=0, v=0$

Initial Condition

At time $t = 0, T = T_{amb}$, and in the liquid and vapour $p=p_{sat}$

Table 1. THERMOPHYSICAL PROPERTIES OF THE HEAT PIPE MATERIAL AND WORKING FLUID

	Thermal conductivity (W/m K)	Specific Heat (J/kg K)	Density (kg/m ³)	Viscosity (Ns/m ²)	Latent heat of vaporization (kJ/kg)
Copper (wall/wick)	401	385	8978	-	-
Water	0.608	4182	998	0.001003	-
Water vapour	0.0261	1418	0.5542	1.34X10 ⁻⁵	2309

4. CFD SIMULATION

Properties of liquid are considered to be constant throughout the simulation. Since the operating pressure inside the heat pipe is low corresponding to saturation temperature of the fluid, the density variation of the vapour can be calculated by ideal gas equation. Vapourflow in the vapour core region is assumed to be in laminar. Liquid flow through the wick is considered to be laminar and incompressible. Heat rejection at the condenser side is by imposing forced convection. The simulation of flow through the heat pipe is modeled by CFD software package ANSYS Fluent. The user defined functions (UDF) are linked to invoke the heat and mass transfer phenomena happening inside the flat heat pipe. Steady state operation of the heat pipe is achieved starting from transient simulation. Calculations are performed by combination of the SIMPLE algorithm for pressure–velocity coupling and a first-order upwind calculation scheme for the determination of momentum, volume fraction and energy. Convergence is based on the residual value of the calculated variables, namely mass, velocity components, volume fraction and temperature.

Assuming a two-phase zone of liquid and vapour phase co-exist and appear adjacent to the conducting wall of porous media, the phase change phenomena is obtained with the respective phase change equations. Taking into consideration that as the heat transfer at the wall gradually enhances, the liquid saturation at the wall approaches its limiting value, zero for boiling and one for condensation. Nusselt number which is a measure of heat transfer coefficient is found out for two phase region of both the evaporator and condenser section. Sherwood number which gives the measure of wall liquid flux and it also specifies the dimensionless phase change rate at the wall, is also computed for both the cases.

Numerical computation is considered to be converged when the scaled residuals of the variables mass, velocity components and volume fraction have lowered by six orders of magnitude and when the residuals of the temperature variable have lowered by seven orders of magnitude.

5. RESULTS AND DISCUSSION

Steady State Performance

The variation of velocity in the vapour core region along the center line is shown in the Figure 2. It can be observed that the vapour velocity monotonically increases in the evaporator section, which is due to the addition of the vapour formed. Similarly there is a fall in the velocity along the condenser section which is due to the mass depletion. The velocity in the adiabatic region is found to be almost constant and is substantially larger than that elsewhere.

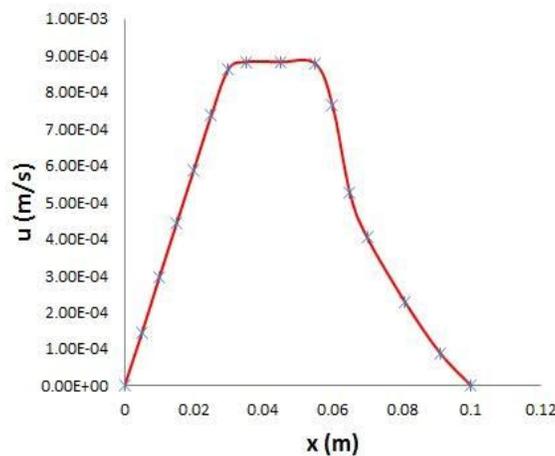


Figure 2. VAPOUR VELOCITY ALONG THE CENTERLINE OF THE VAPOUR CORE

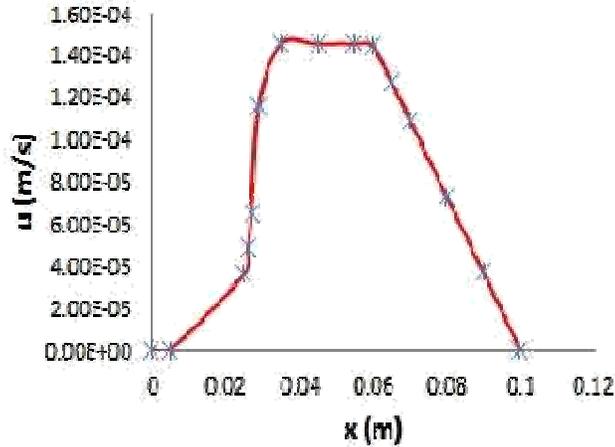


Figure 3. LIQUID VELOCITY ALONG THE WICK CENTERLINE

The variation of liquid velocity in the wick region along the length of the heat pipe, for a wick porosity of 0.6 is shown in Figure 3. The liquid velocity increases from the condenser to adiabatic region, where it is almost constant and there after decreases along the evaporator length. The magnitude of the liquid velocity is more than a hundred times smaller than the vapour velocity in the core due to the liquid-vapour density difference.

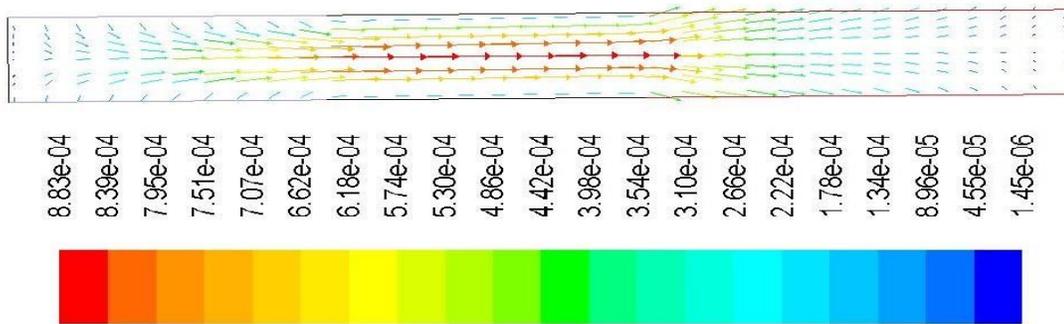


Figure 4. VELOCITY VECTOR AT STEADY STATE IN THE VAPOUR CORE



Figure 5. VELOCITY VECTOR AT STEADY STATE IN THE WICK

Steady state velocity vectors in the vapour core and the wick regions of the heat pipe are shown in the Figures 4 and 5. The vector plot could clearly capture the flow pattern of the vapour in the core from the evaporator to the condenser end, and the return of the liquid from the condenser to the evaporator.

Nusselt number represents the effectiveness of heat convection at the surface which is plotted against wall saturation for boiling and condensing flows and shown in Figures 6 and 7. For boiling its value goes on decreasing with the increase in wall saturation and it approaches a value of 1. For condensing flow, Nusselt number has a finite value at zero wall saturation. It increases with wall liquid saturation and reaches the maximum value at wall saturation equals unity.

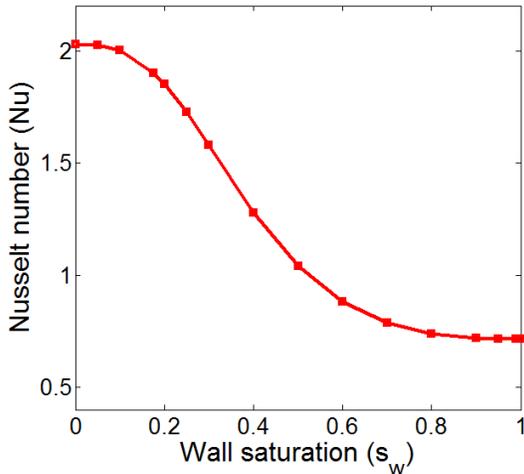


Figure 6. VARIATION OF NUSSELT NUMBER AGAINST WALL SATURATION (BOILING)

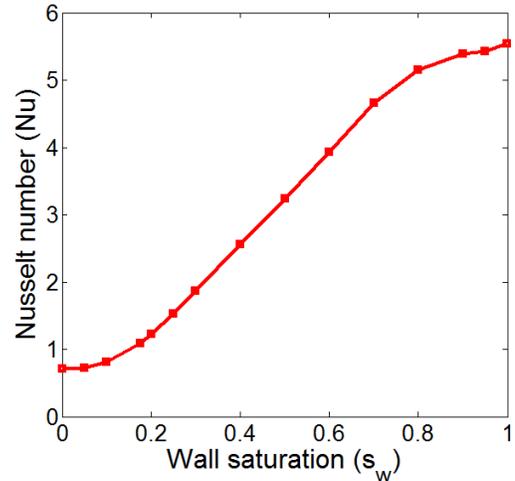


Figure 7. VARIATION OF NUSSELT NUMBER AGAINST WALL SATURATION (CONDENSING)

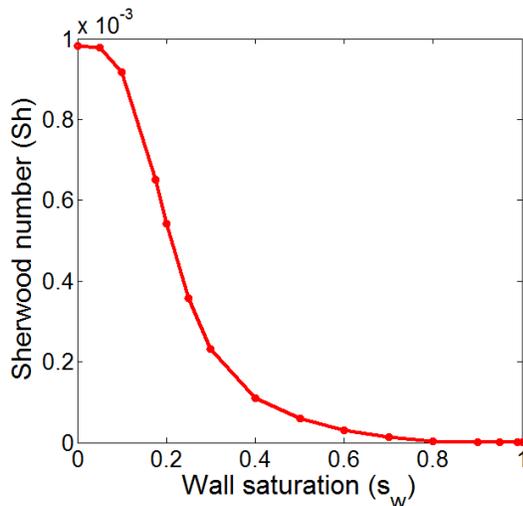


Figure 8. VARIATION OF SHERWOOD NUMBER AGAINST WALL SATURATION (BOILING)

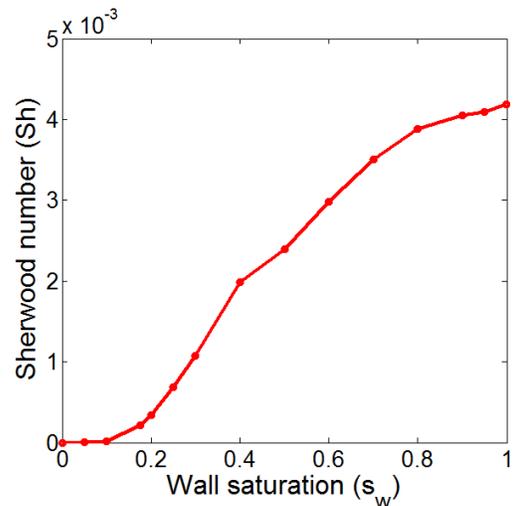


Figure 9. VARIATION OF SHERWOOD NUMBER AGAINST WALL SATURATION (CONDENSING)

Sherwood number which is related to the resistance to mass transfer in a fluid phase is plotted as a function of wall saturation for both boiling and condensing cases in the porous wick structure (Figures 8 and 9). For boiling, heat flux increases as wall saturation decreases and reaches maximum value at zero wall saturation which is the dry out heat flux for the heat pipe. In condensation flow Sherwood number is

zero when the wall saturation is zero and increases with wall saturation and finally reaches maximum value when the wall saturation is unity, which corresponds to maximum heat transfer rate.

6. SUMMARY AND CONCLUSION

In the present work, a two dimensional steady state model of flat heat pipe is numerically analyzed to solve the variation of velocity field in the liquid wick and the vapour core region, for a constant heat flux. The effect of fluid flow through the vapour core and wick which causes heat to flow is brought out in the analysis. The flow in the capillary porous media is solved using two-phase mixture model. The variation of Nusselt number and Sherwood number against the wall saturation for both boiling and condensing flows are obtained. It is also concluded that the flat heat pipe successfully spreads the heat uniformly at the condenser side of the structure.

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