Study of Transient Behavior of the Evaporator of the Planar Micro Loop Heat Pipe

Priyanka Ponugoti, Frank M. Gerner
Department of Mechanical, Industrial and Nuclear Engineering
University of Cincinnati
Cincinnati, Ohio 45221-0030

H. T. Henderson
Department of Electrical and Computer Engineering and Computer Science
University of Cincinnati
Cincinnati, Ohio 45221-0030

ABSTRACT

The Micro Loop Heat Pipe (LHP) is a self-circulating cooling device with extremely high thermal conductivity where heat is removed by phase change and the working fluid circulates by means of the thermodynamic pressure difference developed between the evaporator and the condenser. There exist many models in the literature, which explain steady-state behavior of the micro LHP. Since the evaporator package is a large thermal mass, the time required to reach steady-state is large. In this work, a transient state model is proposed for the planar micro LHP. Two-dimensional transient state analysis is performed on the flat plate evaporator of the LHP with necessary boundary conditions using Ansys 9.0. Analysis is performed in two steps with change in the boundary conditions on the surface of the silicon wick. In the first step, the surface of the wick is assumed to be adiabatic until it reaches 70^0C where it is assumed to be evaporating. Evaporating boundary condition is then applied on the surface of the wick, which constitutes the second step. It is analyzed for different values of 'h' ranging from 0.45 W/m^{2}K to 4500 W/m^{2}K to study the effect of convection cooling.

Nomenclature:

h – Heat Transfer coefficient (W/m^{2}K)
Q – Power (W)
T_∞ – Ambient Temperature (K)
q" – Heat flux (W/cm^2)
t – Time (sec)
k – Thermal Conductivity (W/mK)
Bi – Biot number (hl/k)
Cp – Specific Heat (J/Kg-K)
ρ – Density (Kg/m^3)

Introduction:

The growing trend of miniaturization of electronic components and the rapid increase in power density of advanced micro-processors and electronic components have created a need for more novel and high performance cooling techniques. The next-generation microprocessors and electronic components are projected to dissipate over 1000 W/cm^2. A recent trend in cooling technology is to use phase-change devices such as heat pipes, micro heat pipes, capillary pumped loops (CPL) and loop heat pipes (LHP). The Micro Loop Heat Pipe is a passive device with extremely high thermal conductivity where heat is removed by phase change and the working fluid circulates by means of the thermodynamic pressure difference developed between the evaporator and the condenser. The most important advantage of the micro LHP is its ability to be integrated into the same package as electronic components. NASA is highly interested in the phase-change and light weight cooling equipment like micro loop heat pipe which is a self-circulating device requiring no external pump.

The basic components of the Micro Loop Heat Pipe are evaporator, condenser, vapor line and liquid line as shown in the Fig (1).

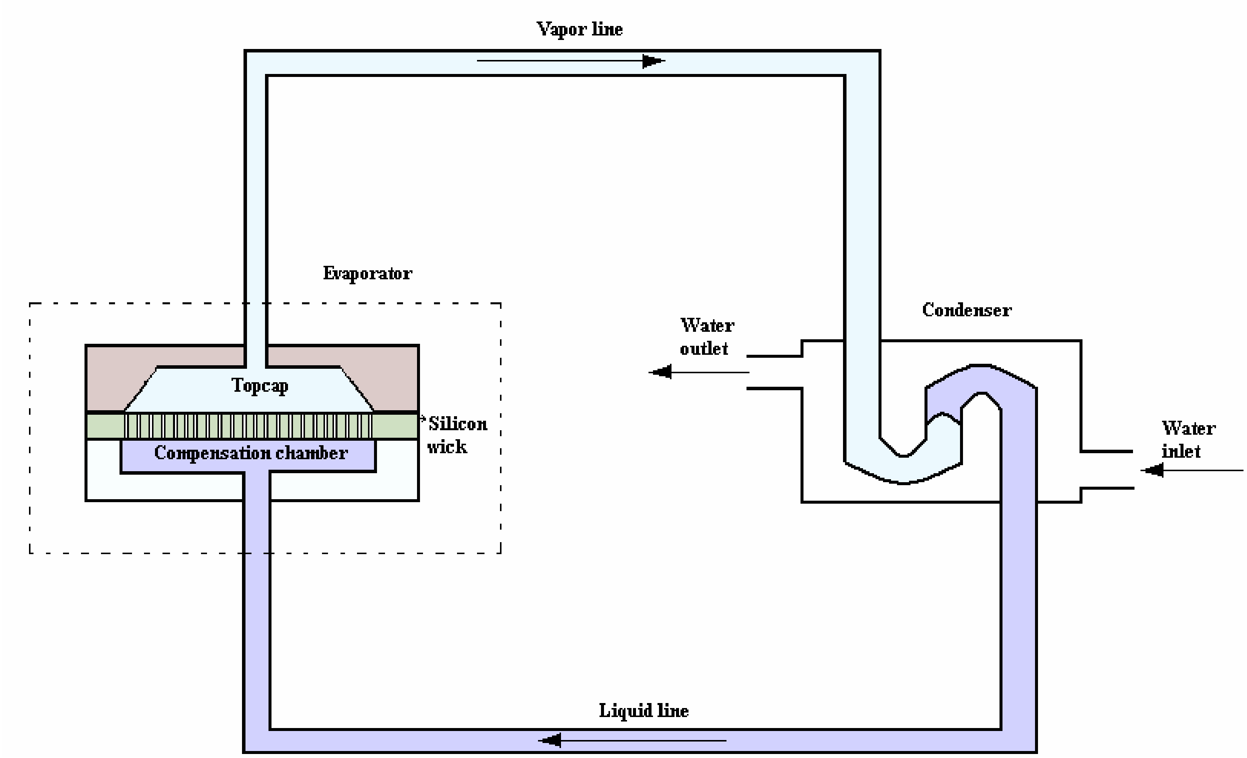


Fig (1): Working of Micro Loop Heat Pipe

As seen in the Fig (1) the evaporator consists of a top cap, coherent porous silicon wick (CPS) and the compensation chamber which acts as a reservoir for the working fluid. The CPS wick as showin in Fig(2a) is an array of micron-range silicon dioxide capillaries micromachined using KOH through ordinary (100) electronic quality silicon wafers. Compensation chanber shown in Fig(2b) consists of quartz wool fiber acting as secondary wick ensuring continuous supply of water to the primary CPS wick.

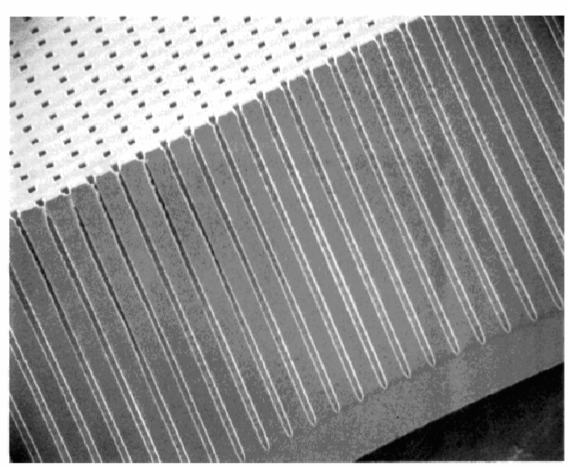


Fig (2a)- SEM Micrograph of a sample of CPS wick

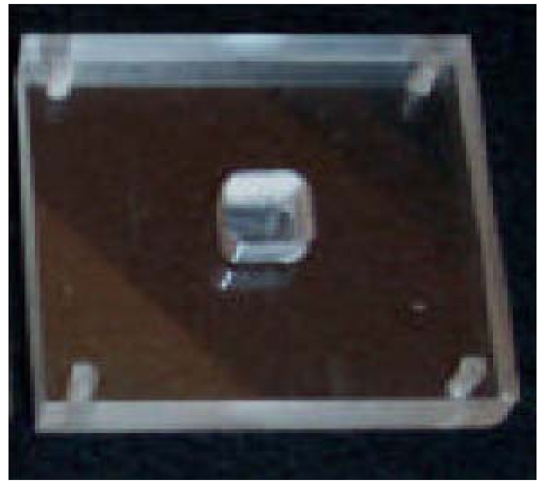


Fig (2b) – The compensation chamber with quartz wool located in secondary wick

The heat source from which the heat should be removed is placed on the top cap of the evaporator. Water in the wick gets heated by conduction of heat from the top cap to the silicon wick as shown in Fig(3). Significant fraction of this heat conducts to the compensation chamber due to the existing temperature difference across the wick. Major components of the evaporator package like top cap, CPS wick and compensation chamber gets heated up before the water in the wick due to their thermal capacitance. This transient phenomena is explained later in this paper.

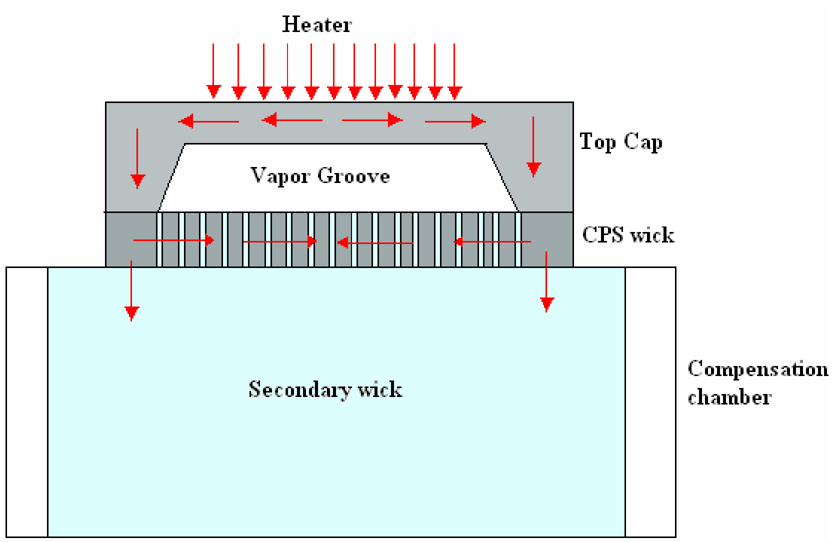


Fig (3) – Heat conduction in the evaporator package

In the steady operation, the energy distribution in the evaporator is as given in Eqn (1)

$$Q = Q_{evp}+Q_{amb}+Q_{cc} \qquad \text{---------------} \qquad (1)$$

Experimental studies conducted here at UC have shown a significant loss of heat to the ambience due to large temperature gradient existing between evaporator top cap and ambient conditions. Water in the wick absorbs the latent heat and evaporates in the saturated and superheated conditions. Generated vapor gets collected in the groves of the top cap. As more and more water gets evaporated, it results in pressure build-up in the top cap. The capillary forces prevent the burst through of the vapor through the wick. Generated vapor pressure forces the vapor out of the evaporator top cap. Due to the thermodynamic pressure difference between the evaporator and the condenser, the vapor is transported through the vapor line to the condenser. The condensed liquid due to pressure difference returns to the compensation chamber and by means of the capillary effect enters the wick. The pressure difference between the evaporator and the condenser maintained by capillary forces in the small diameter pores of the wick cause the working fluid to circulate in the loop. The fluid circulation is passive in the sense that there is no external pump required.

Heat conduction from the source to the silicon wick is shown in the Fig(3). Heat distribution on the surface of the wick is non-uniform due to the lateral conduction of heat within the wick. This leaves many of the pores at the center unheated resulting in high temperature gradients from the side to the center of the wick. The power dissipation ability of the micro loop heat pipe is drastically affected by this non-uniform heat distribution. This problem can be avoided by providing conduction pathways in the top cap as shown in the Fig(4). Arragattu[1] determined the optimal solutions of the various top cap designs feasible from a fabrication point of view. He derived equations to calculate temperature and pressure drop for the trapezoidal top cap geometry as a function of mass flow rate.

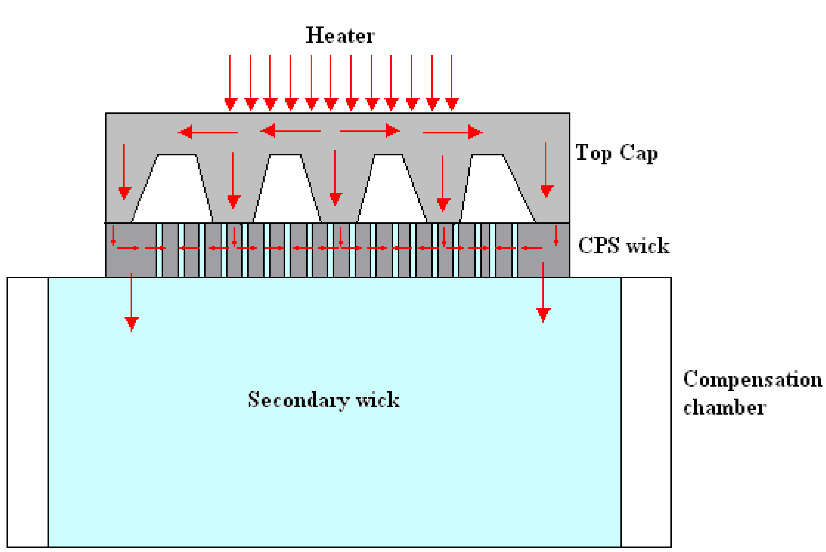


Fig (4)- Evaporator package consisting of top cap with rails

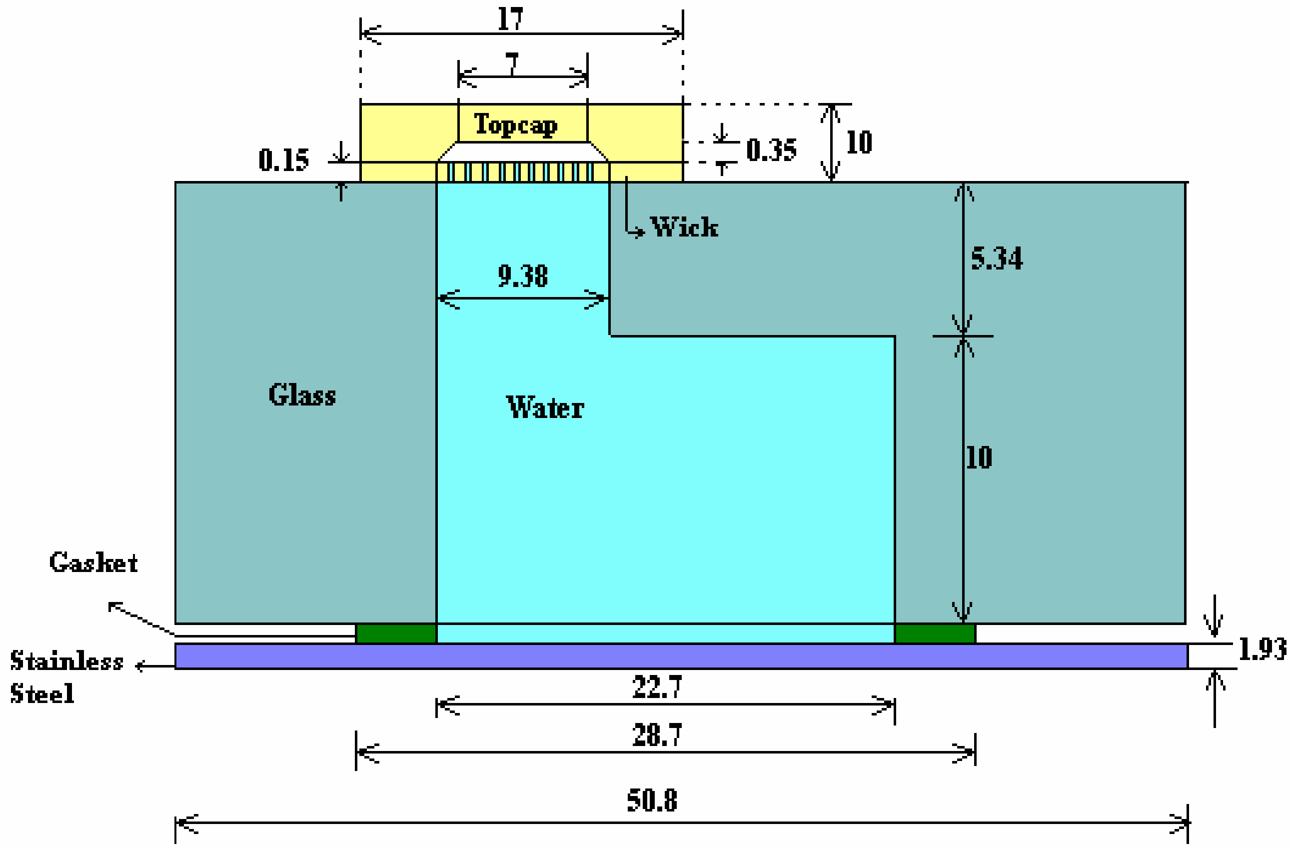
Non uniform wicks with relatively large diameter pores in commercially available loop heat pipes based on annular cylindrical sintered wick structure and metallic wick structure result in low burst through pressure. The micro loop heat pipe considered in this study is of planar shape in contrast to the cylindrical shape enabling it to cool most of the desired parts. Also more uniform pores of sub micron size can be realized in the wick using MEMS based fabrication techniques resulting in higher burst-through pressures.

The evaporator of the loop heat pipe is a large thermal mass. Therefore the time required to reach steady state is very large. The time to reach steady state depends on both the conduction within the evaporator package and convection to the surroundings. Very few studies published in the literature explain transient behavior of the loop heat pipe. Jake Kim[2] developed a system model capable of transient analysis which is able to accommodate wick level consideration on the loop performance and can determine ideal wick structure for the micro LHP using CPS wick. However, in developing the model no thermodynamic equations have been used. Hamdan et al[3] presented a model to predict the steady state behavior of Micro Loop Heat Pipes. He investigated the effects of different parameters on the performance of the LHP utilizing the conservation equations and thermodynamic cycle. Hőlke[4] showed an effective way of to increase the LHP's performance by reducing the pressure drop in the evaporator. The mathematical model of the steady state behavior of the loop heat pipe is presented by Kaya et al[5]. Thiago[6] presented a mathematical model which could predict the LHP operation temperature. Acetone was used as working fluid in their experiments. Nima et al[7] presented a network thermo fluid model of a loop heat pipe operating under steady state conditions. Using Ammonia as the working fluid, he proposed quasi one dimensional mathematical model of fluid flow and heat transfer in each of the elements of loop heat pipe. Suh[8] studied the phenomena of the nucleation occurring in the evaporator of the planar loop heat pipe . De-ionized water was used as the working fluid. Suh [8] showed that the vapor interface oscillates in the pore of the primary wick owing to the difference in the pressures existing between top and bottom of the wick.

This paper focuses on the thermal analysis which determines the transient time scales of Micro Loop Heat Pipe Geometry being fabricated at the Center for Micro Electronics and MEMS, University of Cincinnati. Experiments conducted by Suh[8] prove the fact that these time scales are very large. A 2-D conduction model of the evaporator is analyzed without taking the fluid flow in to consideration.

Description of the Evaporator geometry:

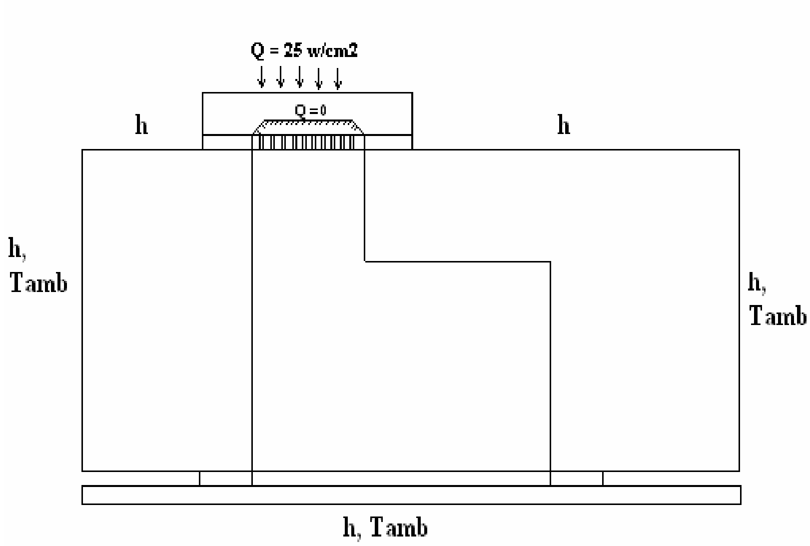
The evaporator of the micro LHP consists of a silicon top cap, CPS wick, pyrex glass compensation chamber, heater, silicone gasket (to prevent water leak) and a stainless steel plate acting as the back plate. To simplify the analysis, top cap used in this model is chosen without rails. Also, to correlate the experimental studies with the simulation, a secondary wick is not considered. The dimensions analyzed are shown in the Fig(5).



Fig(5) – Evaporator Geometry with dimensions in Microns

Boundary and Initial Conditions:

The evaporator was analyzed to determine the transient time scales using Finite Element Analysis Software, Ansys 9.0. The boundary and initial conditions applied to the 2-D model considered in this study are shown in Fig (6)



Fig(6) – Boundary conditions applied during the analysis

As seen in Fig(6), the outer surface of the evaporator was assumed to be exposed to the ambient at a film temperature ($T\infty$) of 298 K (25^0 C)) and a constant value of heat transfer coefficient 'h'. Heat flux of (q") 25 W/cm^2 was applied on the top cap. The surfaces enclosing the vapor in the top cap were assumed to be insulated to simulate the worst case failure analysis. In doing this, the obtained temperature drop values between the center of top cap and center of the wick are more conservative thereby enabling the working of LHP in the safe operational power input. Initially, the entire package was assumed to be at ambient temperature equal to 298 K (25^0 C) and the system was evacuated to remove non-condensible gases. So, the saturation temperature is expected to be much less than 100^0C. There is no direct method to find out the saturation temperature in the evaporator. In performing the simulation an optimum value of the evaporation temperature has to be assumed depending on the following criteria.

i. If evaporation temperature is very less in the order of 50^0C, then most of the energy goes to the evaporation
ii. If evaporation temperature is high in the order of 90^0C, then no evaporation takes place as all the energy input leaks to the compensation chamber

In view of the above criteria, a saturation temperature of 70^0C is assumed in this analysis to be more conservative. The boundary condition on the wick surface depends on the temperature at any point on the surface as described below.

i. If temperature on the surface of the wick is less than 343 K, the boundary condition is taken to be adiabatic.
ii. If the temperature on the surface of the wick is greater than 343 K, a constant temperature of 343 K is applied on the surface which implies an evaporating boundary condition.

Material Properties:

The properties of various materials that constitute the evaporator component of the Loop Heat Pipe are listed in Table (1)

Table (1): Material Properties used in the analysis

	Thermal Conductivity K(w/m-K)	Specific Heat C (J/Kg-K)	Density ρ (Kg/m3)
Silicon	140	700	2329
Glass	1.14	750	2230
Water	0.5	4180	1000
Wick	105.125	1570	1996
Silicone Gasket	138	1170	1090
Stainless Steel	18	500	8000

The porosity of the wick used in this study was taken as 25% based on the experiments performed here at University of Cincinnati. Therefore effective thermal conductivity of the CPS wick (25% water + 75% Silicon) was used in the calculations. For example,

Thermal Conductivity of the wick = 25% (Conductivity of Water) + 75% (Conductivity of Si)

$$= 0.25\,(0.5)+0.75(140)$$

$$k_{wick}=105.125\ \text{W/m-K}$$

Assumptions :

1) All the elements of the system are homogenous, isotropic, with invariant thermal properties and chemically stable.
2) There is no internal heat generation and no body forces acting on the system.
3) Viscous dissipation in neglected.
4) There exists no temperature gradient across the interface of two elements.
5) The elements of the package are perfectly bonded.
6) Saturation temperature of water is 70^0C.
7) Porosity of the wick is 25%.

Methodology:

To study the influence of convection on the transient time scale, the value of the heat transfer coefficient is varied from 0.5 to 4500 W/m^2K which ranges Biot number (Bi) from 0.01 to 100. Biot number is defined as the ratio of internal conduction resistance to the outer convective resistance. Methodology for a single value of 'h = 15 W/m^2K' is described below.

As explained before, the problem was solved in two steps depending on the temperature of the surface of the wick.

1) Initially since the surface of the wick is at the ambient temperature of 298 K, an adiabatic boundary condition was applied on the surface of the wick until it reaches saturation temperature of 343 K. The model was then analyzed in Ansys 9.0. While the analysis was performed, all the elements of the evaporator geometry were meshed with a smart size of '3' on a scale of 1 to 10, 10 representing a coarser mesh. Transient temperature plot was generated for the central node on the surface of the wick to determine the value of time at which the node reaches saturation temperature as shown in the Fig(7).

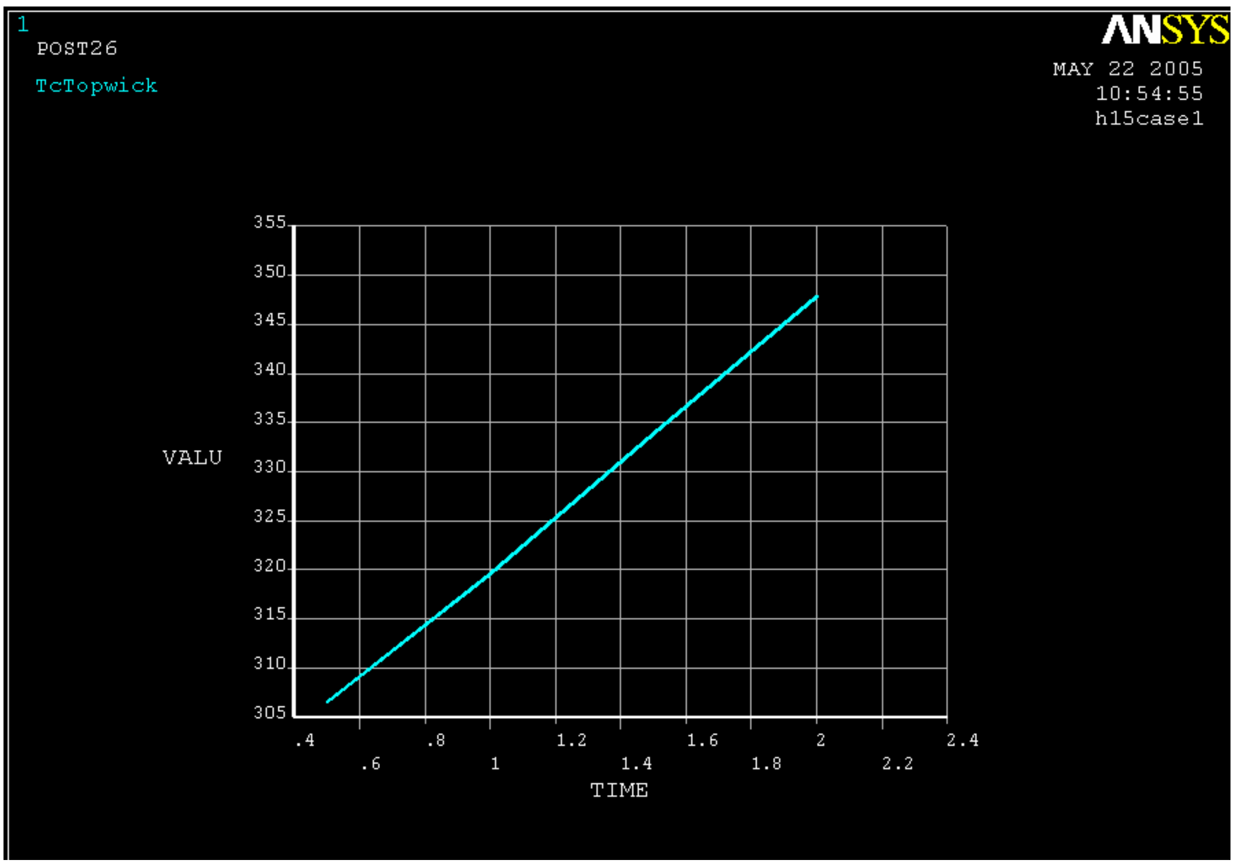
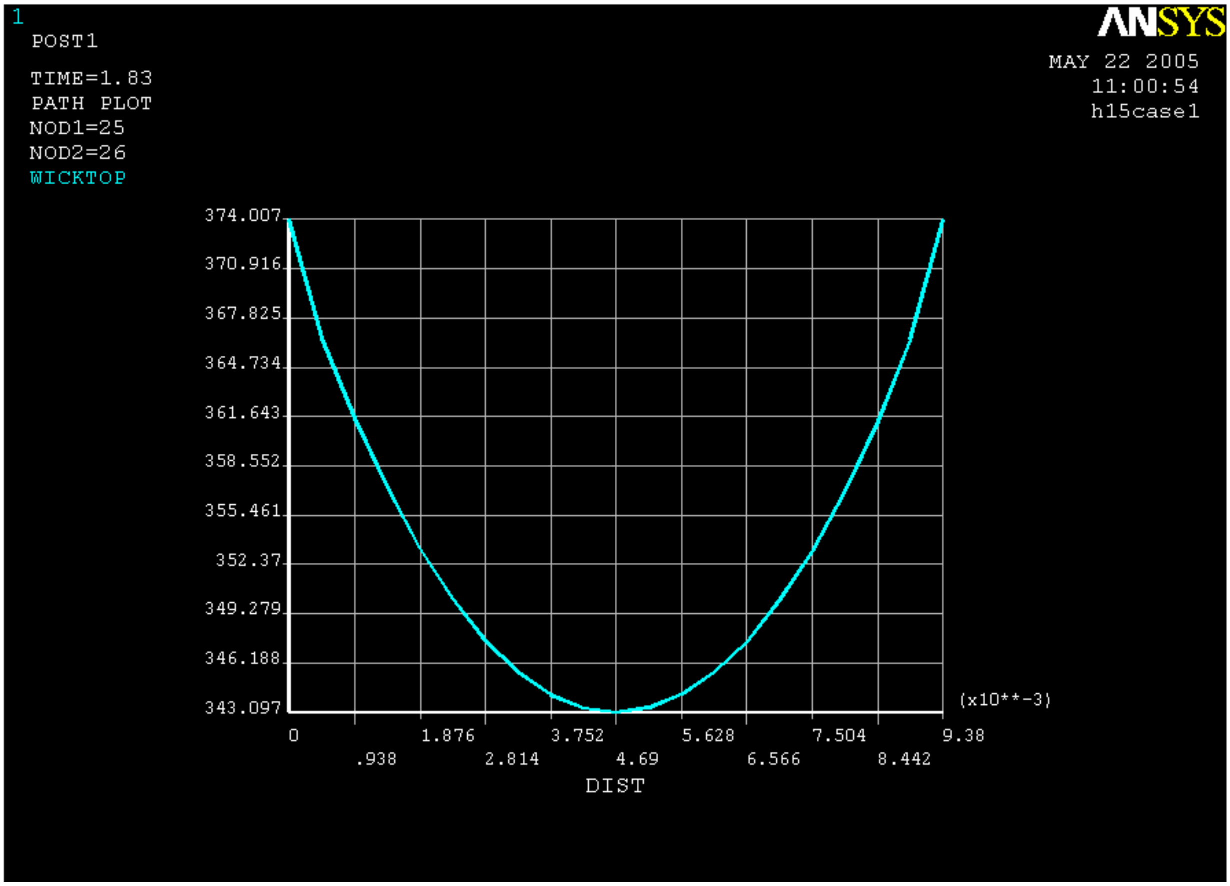


Fig (7) Transient Temperature plot for the central node on the top surface of the wick

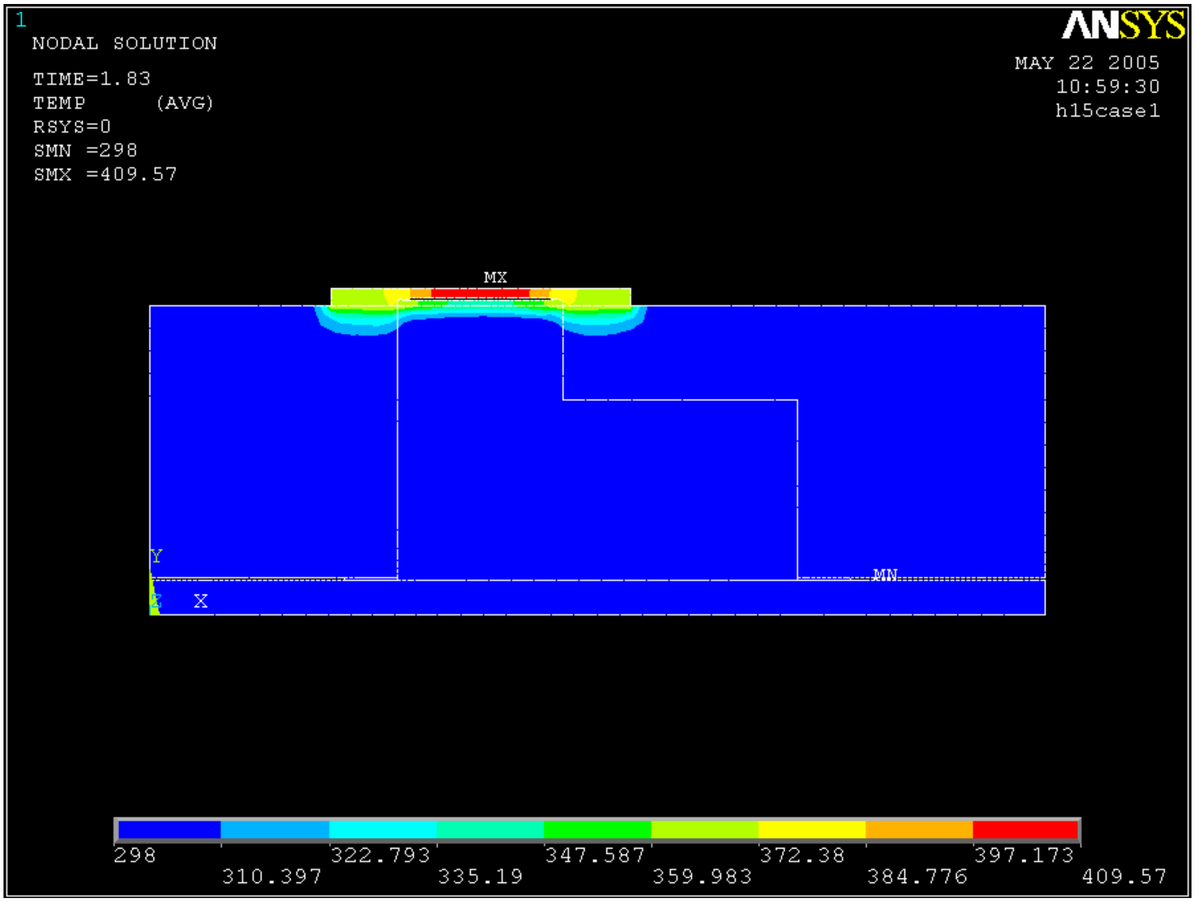
The graph in Fig (7) shows that the node reaches saturation temperature of 343 K at 1.83 sec. The temperature on the top surface of the wick at the end of 1.83 sec is plotted in Fig(8). It is seen from the figure that the center node is the last one to reach assumed saturation temperature. Fig(7) is plotted for the center node to simulate the worst case scenario.



Fig(8) - Temperature plot on the top surface of the wick at 1.83 sec

Also Fig(8) shows a huge temperature drop of 30^0C from the side to the center of the wick. As explained before, this is because of non-uniform heat distribution on the surface which can be avoided using the rails.

The contour plot showing the temperature distribution in the entire model at 1.83 sec is shown in the Fig (9). Since the surface of the wick reached 343 K at 1.83 sec, the boundary condition was changed at that instant of time.

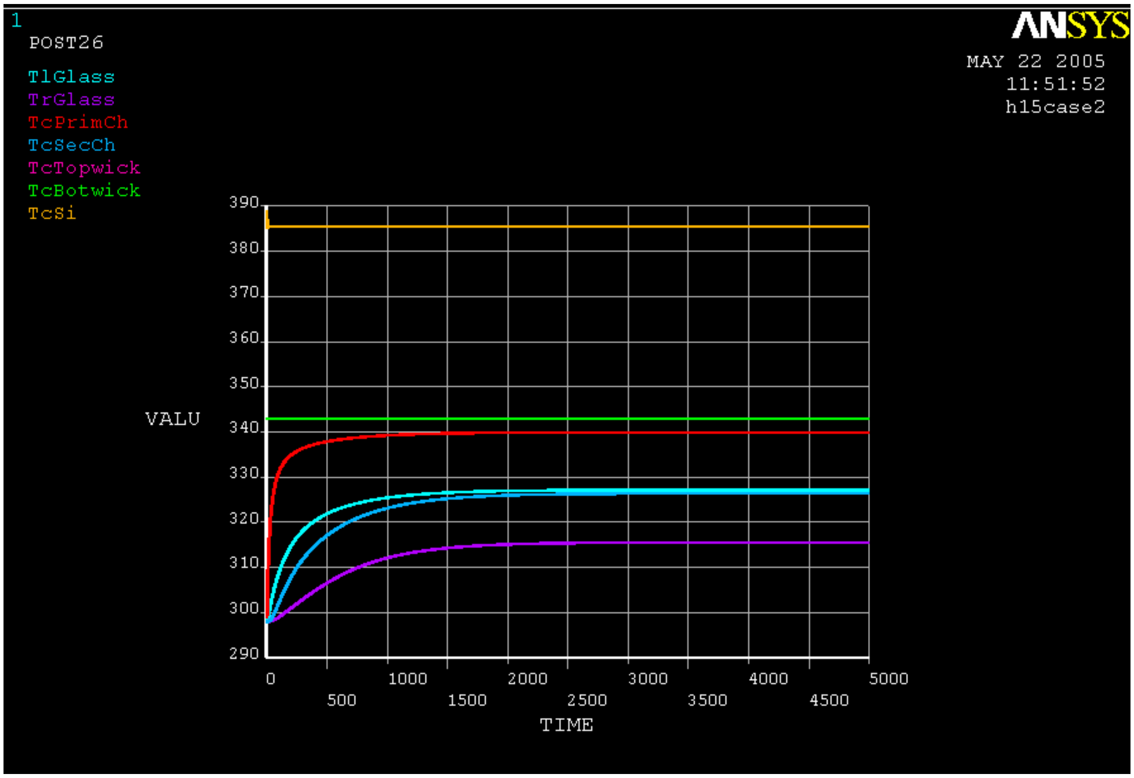


Fig(9) Contour plot at 1.83 sec when the central node on the top surface of the wick reaches saturation temperature

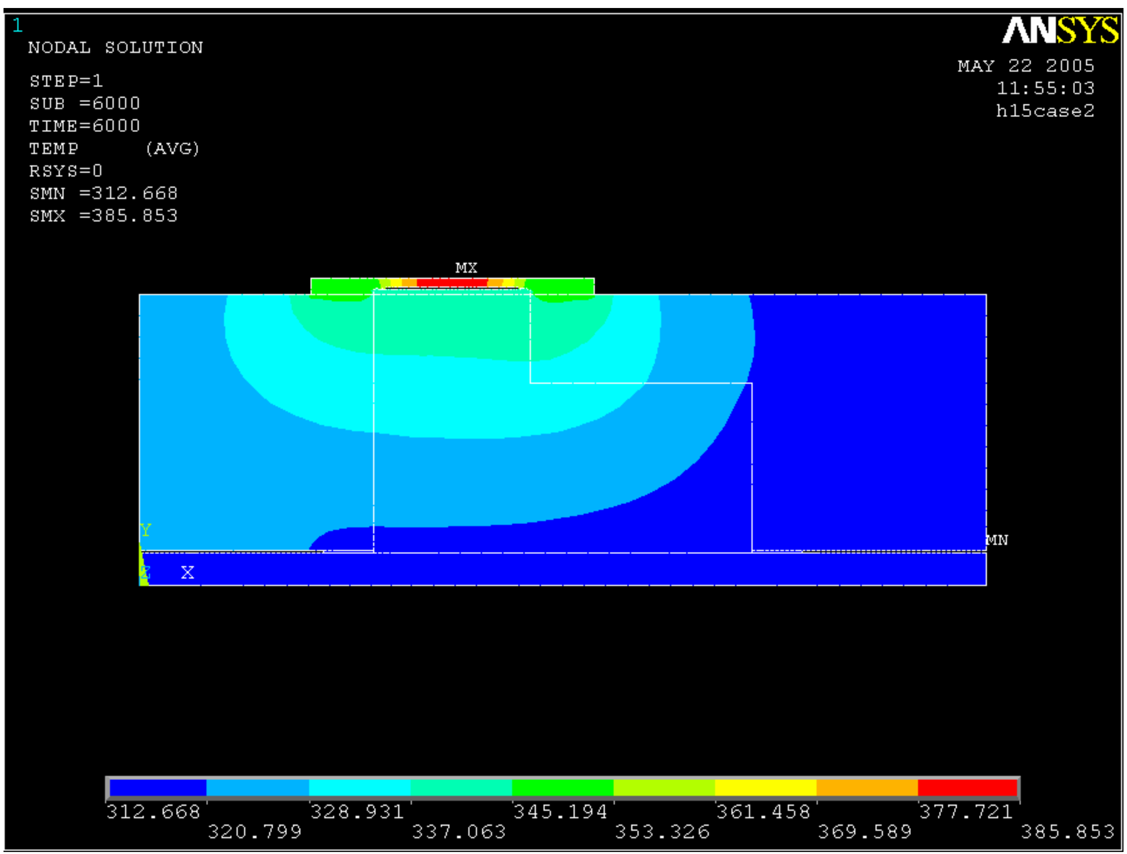
It can be seen from the above contour plot that there exists huge temperature gradient from the top cap so the bottom wick. Also we can see that the temperature of compensation chamber in 1.83 seconds has not varied which proves that it has very large capacitance.

2) In the second step, since the problem has to be continued after 1.83 sec from step (1), constant boundary condition is applied on the wick surface which means the evaporating boundary condition. The initial conditions for this step are taken from the result file of step(1) at 1.83 sec. All other conditions remain the same. The model is again analyzed in Ansys and the transient temperature plots at various locations in the model are plotted as shown in the Fig (10). And the contour plot after reaching the steady state is also shown in the Fig (11).

   The time taken to reach steady state is obtained from the graph based on 1/e criteria. The same procedure is repeated for different values of heat transfer coefficient using the same boundary conditions.



Fig(10) - Transient temperature plots for nodes at various locations in the evaporator



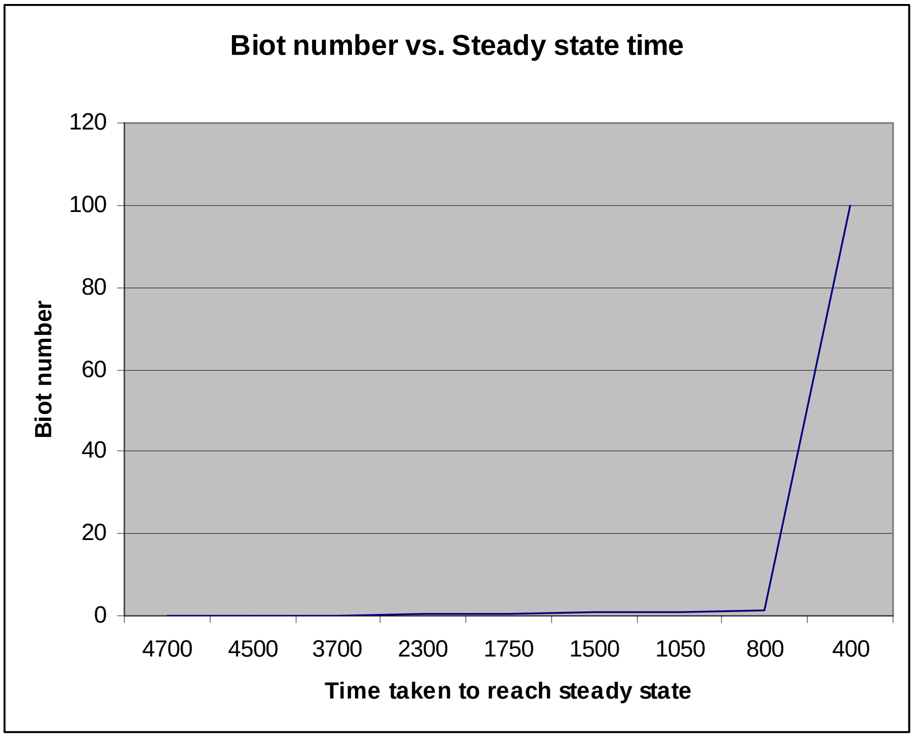
Fig(11)- Contour plot of the evaporator section at the steady state

Results & Conclusions:

The time taken to reach steady state for different values of heat transfer coefficient corresponding to various Biot numbers are shown in the Table (2) and a graph of Bi vs time taken to reach steady state is plotted as shown in the Fig (12). Higher Values of Biot number will tend to asymptote the curve.

Table (1): Data for Graph Biot Number vs. Time taken to reach steady state

Biot Number (Bi)	Convection Heat Transfer Coefficient (h)	Time taken to reach steady state
0.01	0.45	4700
0.022	1	4500
0.111	5	3700
0.333	15	2300
0.557	25	1750
0.779	35	1500
1.0026	45	1050
1.2254	55	800
100	4488	400



Fig(12) – Graph showing the variation of the time taken to reach steady state with Biot number

It was observed that the steady state temperatures for different values of convective heat transfer coefficient is the same. Also, it is seen that at smalls of heat transfer coefficient, time taken to reach steady state varies appreciably even with an insignificant change in heat transfer coefficient. As coefficient increases, the time taken to reach steady state varies insignificantly even with an appreciable change in the heat transfer coefficient. This is because as convection coefficient increases, the time taken for dissipating the heat by convection to surroundings is very less. The entire time is due to the conduction of the heat within the system.

Future work may involve studying the transient behavior of the planar micro loop heat pipe with rails in a full scale 3-D model.

Acknowledgements:

This research was sponsored by the NASA John H. Glenn Research Center at Lewis Field (GRC) under contract number C-80063-A and C-80075-A. A special thanks to Kenneth Mellott and Jake Kim for their continued support of and input to the project. Research team would also like to thank the following people Praveen Medis, Srinivas Parimi, Ahmed Shuja, Junwoo Suh , Praveen Arragattu and Monika Sharma at University of Cincinnati for their assistance with various activities of the project.

References:

[1]Arragattu P., Gerner F., Ponugoti P., Henderson T. 2004, "Competitive Designs of Evaporator Top Cap of the Micro Loop Heat Pipe", ASME International Mechanical Engineering Congress.

[2] Jake Kim, 2003, "Micro Loop Heat Pipe Design and Simulation", Final report - NASA Glenn Research Center, JINHA Sciences, Inc.

[3] Hamdan Mohammad Omar, 2003, " Loop Heat Pipe (LHP) Modeling And Development By Utilizing Coherent Porous Silicon (CPS) Wicks", PhD Thesis, University of Cincinnati, Mechanical Engineering.

[4] Holke A., Henderson T., Gerner F., and Kazamierczak M., 2003, "Analysis of Heat Transfer Capacity of a Micro-machined Loop Heat Pipe", ASME Journal Of Heat Transfer, submitted for publication.

[5] Kaya T., and Hoang T., 1999, "Mathematical Modeling of Loop Heat Pipes and Experimental Validation", J. of Thermo physics and Heat Transfer, Vol. 13, No. 3, pp. 314-320.

[6] Thiago D., and Roger R.R., 1998, "Analysis of the Thermal Performance of a Loop Heat Pipe", ASME International Mechanical Engineering Congress.

[7] Nima Atabaki and Rabi B Baliga, 2003 ," Steady State Operation of a Loop Heat Pipe : Network Thermofluid Model and Results", IMECE2003-43968, ASME.

[8] Suh J., Shuja A., Gerner F., Henderson T., 2004, "Nucleate Boiling Inside the Evaporator of the Planar Loop Heat Pipe", ASME International Mechanical Engineering Congress.