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COMPARATIVE STUDY OF AXIAL HEAT CONDUCTION ON THE WALLS OF A MICRO GROOVED HEAT PIPE

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ABSTRACT

This paper describes a computational procedure to solve the constant temperature at which evaporation and condensation of the working fluid takes place inside a heat pipe. The numerical method used is a cell centered finite volume method. The temperature of the heat pipe is found to be more controlled by the heat transfer coefficient at the condenser. At the adiabatic section it was found that the effective liquid film heat transfer coefficient and the thickness of the wall has profound influence on the spreading of the heat. It was also found that by increasing the thickness of the wall, high heat spread can be achieved. Studies were also conducted to analyse the effect of thermal conductivity of the heat pipe material on its heat transfer characteristics.

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NOMENCLATURE

h Heat Transfer Coefficient at Condenser, $W/m^2 K$

h_{eff} Film heat transfer coefficient, $W/m^2 K$

k Thermal conductivity of cover plate, W/mK .

L_{hp} Length of the Heat Pipe, m

q Heat flux into the working fluid, W/m^2

T Temperature, K

T_i Interface Temperature, K

T_∞ Ambient temperature, K

t Thickness of cover plate, mm

1. INTRODUCTION

Heat pipes are an effective passive heat transfer technology in the cooling of high heat flux electronic devices and systems because of its high thermal conductivity, reliability and low weight. It essentially consists of different sections viz: evaporator, transmitter and condenser.

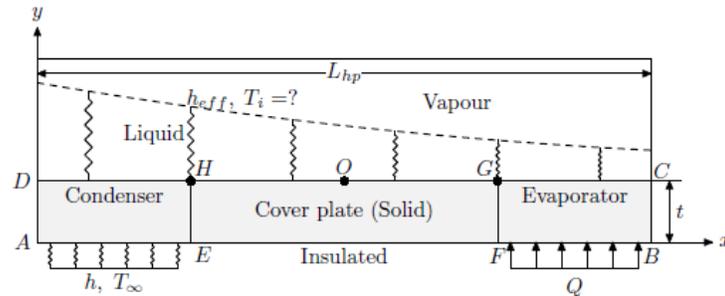


FIGURE 1. SCHEMATIC DIAGRAM SHOWING THE HEAT TRANSFER IN A HEAT PIPE

A schematic diagram showing the heat transfer process through the wall of a typical heat pipe is given in Fig. 1. Do et al. [1] have noted that several prior investigations have assumed a uniform evaporation and condensation in the respective sections and no phase change in the transmitter section (adiabatic). Vadakkan et al. [2, 3] have demonstrated that there occurs heat transfer to the working substance at the axial locations of adiabatic section as well, due to spread of the heat flux in the axial direction. The distribution of heat flux $q(x)$ into the fluid from the inner surface of the cover plate depends on the temperature of inner surface of the wall, which in turn, depends on the constant liquid-vapor interface temperature (T_i) of the working substance and film resistance coefficient. Aghvami and Faghri [4] have presented an analysis of heat conduction through the cover plate of a flat heat pipe by solving two-dimensional heat conduction equation. They specified a constant temperature for the liquid vapor interface and used a linear interpolation to get the inner wall temperature. This is equivalent to treating the wick and liquid within the wick as a thermal resistance. The constant liquid-vapor interface temperature is found out by solving separate equations in liquid and vapor regions. Hence the heat transfer to the liquid and axial heat conduction depend solely on the value of the temperature prescribed at the inner surface of the cover plate. Sobhan et al. [5] have employed a transient computational methodology using ambient temperature as the initial temperature and presented the results for steady state conditions as well. In this study a constant liquid vapor interface temperature computed using Clausius-Clapeyron equation is taken as the boundary condition. Vadakkan et al. [3] have also used a transient approach in which the wick-vapor interface temperature is computed from energy balance at the interface and liquid pressure at the interface is computed using Clausius-Clapeyron equation. The above researchers have neglected the axial conduction through cover plate, because of its small thickness.

However as the thickness of the cover plate increases the axial conduction and heat spread has to be taken into consideration in the design of heat pipes. Hung and Tio [6] have investigated the effect of axial conduction through the solid walls of a micro heat pipe using an analytical solution of a one dimensional steady state model. In this study, by considering a convective resistance corresponding to $Nu = 2.68$ for the convective films at the evaporator and condenser, the temperature distribution of the solid is found as a function of a constant liquid temperature. Then they prove by equating the heat added and removed at the evaporator and condenser sections that the liquid temperature is equal to the average solid wall temperature. They observed a significant axial heat conduction for different materials of the walls such as Copper, Nickel and Monel.

The foregoing literature confirms that the heat conduction through the cover plate is important in setting the temperature of the working fluid of a heat pipe. Therefore it is of utmost importance to analyse the axial conduction in the heat pipe. The present work is concerned with development of a numerical scheme for the determination of the distribution of heat flux along the axial direction of the heat pipe by considering axial heat conduction through the wall. Simulations are performed by varying the material for the heat pipe and its thickness, to study its impact on heat spread along the axial direction.

1.1 Modelling

In Fig.1, the rectangle ABCD represents the metallic cover plate of a heat pipe that separates the working substance and the heat exchanging surfaces. The edge FB represents the evaporator of the heat pipe through which a uniform heat flux enters the cover plate. The condenser is represented by the cooling edge AE, where a fluid at temperature T_{∞} with a convective heat transfer coefficient h removes heat. The edge EF is insulated and the edge DC is exchanging heat to the working substance. It is assumed that the liquid vapor interface remains steady so that the net heat transfer across CD is zero for a given constant heat flux through FB. In this manner the edge DC is exchanging heat to a constant temperature reservoir which acts simultaneously as a heat sink and heat source. An effective heat transfer coefficient denoted as h_{eff} for the exchange of heat from the top edge DC to the constant temperature liquid vapor interface is considered. It has been reported by Ma and Peterson [7] that for triangular grooved channels, the heat transfer coefficient for heat exchange with the boiling liquid over CD could vary typically in the range $3.7 - 6.7 \text{ W/cm}^2\text{K}$. For flat heat pipes an effective heat transfer coefficient should be computed based on the thermal conductivity of the materials of liquid and wick and the liquid film heat transfer coefficient. The heat transfer problem consists of solving the constant temperature of the liquid-vapor interface, T_i and determination of the distribution of heat flux, $q(x)$ to the liquid through CD.

From the earlier section it is noted that most of the previous investigations on flat heat pipes and micro heat pipes have assumed the resistances of the metallic plate and convective film resistances as zero. Determination of $q(x)$ for a known value of liquid temperature T_i is a direct heat transfer problem which can be easily solved. But in the present work T_i is an unknown. For the evaporation and condensation of the liquid, the temperature T_i should have a constant value over the entire length of the cover plate. Another known condition is that the net heat exchange to liquid through the edge CD is zero, which can be mathematically expressed $\int_C^D q(x) dx = 0$. Typical variations of heat flux across the edge CD of the cover plate are shown in Fig. 2.

The mathematical model of the problem consists of the Laplace equation for the domain subjected to boundary conditions. The governing equations and boundary conditions are shown in the following:

$$\nabla^2 T = 0 \quad \text{at all the points inside the cover plate} \quad (1)$$

$$k \frac{\partial T}{\partial y} = -h(T - T_{\infty}) \quad \text{and} \quad k \frac{\partial T}{\partial y} = -q \quad (2)$$

at points on AE and on FB respectively

$$\text{At point on CD, } Q_2 = - \int_0^C k \frac{\partial T}{\partial y} dx = \int_D^0 k \frac{\partial T}{\partial y} dx \quad (3)$$

$$\text{Such that } \int_D^C k \frac{\partial T}{\partial y} dx = 0$$

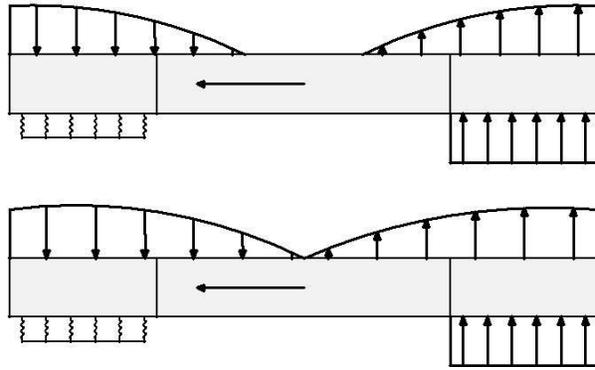


FIGURE 2. TYPICAL DISTRIBUTION OF HEAT FLUX OVER THE TOP SURFACE CD OF THE COVER PLATE, FOR CASES WITH AXIAL HEAT CONDUCTION. HERE NET HEAT FLOW ACROSS CD IS ZERO AND $Q = Q_1 + Q_2$.

$$\Rightarrow \int_D^C k \frac{\partial T}{\partial y} dx = - \int_D^C h_{eff}(T - T_i) dx = 0 \quad (4)$$

$$\Rightarrow \int_D^C \left(k \frac{\partial T}{\partial y} + h_{eff}(T - T_i) \right) dx = 0 \quad (5)$$

Where, $h_{eff} = kw\phi tw + hl(1 - \phi)$ is an equivalent heat exchange coefficient with ϕ , kw and tw are the porosity, thermal conductivity and thickness of the wick respectively. In the case of micro heat pipes the value of porosity ϕ is zero.

2. NUMERICAL METHODOLOGY

In the present work a cell centered finite volume method is applied for the solution of equation (1). Referring to the standard grid system for a general variable as followed by Patankar [8], the discretization equation for the control volume with geometric center marked as P, with East, North, West and South neighbors denoted by E, N, W and S,

$$A_P T_P = A_E T_E + A_W T_W + A_N T_N + A_S T_S \quad (6)$$

Although the solution of the above equation with the boundary conditions in equation (2) is straight forward. But application of the boundary condition on the top line CD as per equation (5), where the liquid-vapor interface with un-known temperature at the North neighbor requires special attention. The detailed procedure of discretizing the equation for liquid-vapor interface can be seen in Ramachandran et al. [9]

3. RESULTS AND DISCUSSIONS

3.1 Range of parameters

Computations are carried out by varying the heat load (Q) and heat coefficient (h) at the condenser in the ranges $[25 - 250]$ kW /m² and $[100 - 7000]$ W /m²K, keeping the effective film heat transfer coefficient (h_{eff}) at the inside surface of the cover plate where evaporation and condensation of the working substance takes place as constant, $[15000]$ W /m²K. The geometrical parameter varied in the present work is thickness of the cover plate and its range is $[0.5 - 15]$ mm The material of the cover plate

is varied (Cu = 401W /mK, Al = 204W /mK and Monel metal = 19.4W /mK) and the working fluid is Water.

3.2 Definition of different terms

Certain new terms used in the discussion of results of the problem are Heat Spread Factor (HSF), Heat Bypass Factor (HBF) and Effective Thermal Conductivity (ETC). Heat Spread Factor(HSF) is defined as the ratio of heat transferred to liquid from the adiabatic regions spread from evaporator or condenser to total heat. HSF for the evaporator and condenser are computed separately. Bypass factor (HBF) is related to the heat bypassing the liquid and flowing through the solid region of the cover plate. It is defined as the ratio of the difference between the total heat and the heat transferred to the liquid to the total heat. Effective Thermal Conductivity(ETC) is defined as the ratio of the heat flux to the linear temperature gradient between evaporator and the condenser.

3.3 Effect of heat transfer coefficient at the condenser

Tables 1, 2 and 3 shows the effect of heat transfer coefficient at condenser h on different parameters at heat loads 25 kW /m^2 and 250 kW /m^2 , corresponding to an effective film heat transfer coefficient at the inside $h_{eff} = 15000 \text{ W /m}^2\text{K}$ and different thermal conductivity for the heat pipe material. The temperature at various points decreases with the increase of h . For the small thickness,

TABLE 1. EFFECT OF HEAT TRANSFER COEFFICIENT AT THE CONDENSER ON THE THERMAL PERFORMANCE OF THE HEAT PIPE MADE OF COPPER FOR HEAT FLUXES $q = 25\text{kW /m}^2$ AND $q = 250\text{kW /m}^2$, $t = 1\text{mm}$, $h_{eff} = 15000 \text{ W /m}^2\text{K}$

h	Temperature				HBF	HSF	
	Condenser	Evaporator	Interface	K_{eff}		Evaporator	Condenser
$\frac{W}{(m^2K)}$	K	K	K	$\frac{W}{(mK)}$	%	%	%
$q = 25\text{kW/m}^2$							
500	349.98	353.00	351.50	826.43	0.0821	12.99	13.04
1000	324.98	328.00	326.50	826.70	0.0823	12.99	13.09
2000	312.48	315.50	314.00	827.23	0.0826	12.99	13.19
3000	308.31	311.33	309.83	827.74	0.0829	12.99	13.29
4000	306.23	309.25	307.75	828.22	0.0832	12.99	13.38
5000	304.98	308.00	306.49	828.68	0.0834	12.99	13.46
6000	304.15	307.16	305.66	829.11	0.0837	12.99	13.55
7000	303.55	306.57	305.06	829.53	0.0839	12.99	13.63
$q = 250\text{kW/m}^2$							
500	799.84	830.09	815.06	826.43	0.0821	12.99	13.04
1000	549.84	580.09	565.05	826.70	0.0823	12.99	13.09
2000	424.85	455.07	440.03	827.23	0.0826	12.99	13.19
3000	383.18	413.39	398.35	827.74	0.0829	12.99	13.29
4000	362.35	392.54	377.50	828.22	0.0832	12.99	13.38
5000	349.85	380.02	364.98	828.68	0.0834	12.99	13.46
6000	341.64	371.67	356.64	829.11	0.0837	12.99	13.55
7000	335.57	365.71	350.67	829.53	0.0839	12.99	13.63

$t = 1 \text{ mm}$ of the cover plate considered in the present analysis, the HBF is found to be negligible and independent of the heat load. It can be inferred from Table 1 that the heat spread through the adiabatic regions (HSF) near the condenser and evaporator are nearly equal and is 13% of the total heat in the case

heat pipe made of copper. Same observation with reduced values are observed in Tables 2 and 3 for Aluminum and Monel metal. This is because of low thermal conductivity of wall material. It is worth to note that HSF is independent of h as well as q for the materials considered in the study. ETC is found to insensitive to variations of heat load and h it is found to be 20 times that of the thermal conductivity of the cover plate material made of Monel metal, due to its low thermal conductivity

TABLE 2. EFFECT OF HEAT TRANSFER COEFFICIENT AT THE CONDENSER ON THE THERMAL PERFORMANCE OF THE HEAT PIPE MADE OF ALUMINUM FOR HEAT FLUXES $q = 25\text{kW}/\text{m}^2$ AND $q = 250\text{kW}/\text{m}^2$, $t = 1\text{mm}$,

h $\frac{W}{(m^2K)}$	Temperature				HBF	HSF	
	Condenser	Evaporator	Interface	K_{eff}		Evaporator	Condenser
	K	K	K	$\frac{W}{(mK)}$		%	%
$q = 25\text{kW}/\text{m}^2$							
500	349.98	353.24	351.62	765.61	0.0061	9.41	9.46
1000	324.98	328.24	326.62	765.84	0.0061	9.41	9.51
2000	312.48	315.74	314.12	766.30	0.0061	9.41	9.60
3000	308.31	311.57	309.95	766.72	0.0062	9.41	9.69
4000	306.23	309.49	307.86	767.13	0.0062	9.41	9.77
5000	304.98	308.24	306.61	767.51	0.0062	9.41	9.85
6000	304.15	307.40	305.78	767.87	0.0062	9.41	9.93
7000	303.55	306.81	305.18	768.22	0.0063	9.41	10.00
$q = 250\text{kW}/\text{m}^2$							
500	799.83	832.48	816.25	765.61	0.0061	9.41	9.46
1000	549.83	582.47	566.24	765.84	0.0061	9.41	9.51
2000	424.83	457.46	441.22	766.30	0.0061	9.41	9.60
3000	383.17	415.77	399.54	766.72	0.0062	9.41	9.69
4000	362.34	394.93	378.69	766.13	0.0062	9.41	9.77
5000	349.88	380.62	365.09	767.51	0.0062	9.41	9.85
6000	341.72	371.78	356.75	767.87	0.0062	9.41	9.93
7000	335.56	368.10	351.87	768.22	0.0063	9.41	10

TABLE 3. EFFECT OF HEAT TRANSFER COEFFICIENT AT THE CONDENSER ON THE THERMAL PERFORMANCE OF THE HEAT PIPE MADE OF MONEL METAL FOR HEAT FLUXES $q = 25\text{kW}/\text{m}^2$ AND $q = 250\text{kW}/\text{m}^2$, $t = 1\text{mm}$, $h_{eff} = 15000\text{W}/\text{m}^2\text{K}$

h $\frac{W}{(m^2K)}$	Temperature				HBF	HSF	
	Condenser	Evaporator	Interface	K_{eff}		Evaporator	Condenser
	K	K	K	$\frac{W}{(mK)}$		%	%
$q = 25\text{kW}/\text{m}^2$							
500	349.97	355.74	352.87	433.45	-2.9×10^{-6}	3.41	3.43
1000	324.97	330.74	327.87	433.56	-1.7×10^{-6}	3.41	3.46
2000	312.47	318.23	315.36	433.76	-2.8×10^{-6}	3.41	3.53
3000	308.31	314.07	311.20	433.93	-1.7×10^{-6}	3.41	3.58
4000	306.22	311.98	309.11	434.09	-1.4×10^{-6}	3.41	3.63
5000	304.97	310.73	307.86	434.24	-1.1×10^{-6}	3.41	3.68
6000	304.14	309.90	307.03	434.37	-1.3×10^{-6}	3.41	3.72
7000	303.55	309.30	306.43	434.49	-1.1×10^{-6}	3.41	3.76
$q = 250\text{kW}/\text{m}^2$							
500	799.74	857.42	828.71	433.45	-1.8×10^{-6}	3.41	3.43
1000	549.75	607.41	578.70	433.56	-1.6×10^{-6}	3.41	3.46
2000	424.76	482.39	453.69	433.76	-1.8×10^{-6}	3.41	3.53
3000	383.10	440.71	412.01	433.93	-4.1×10^{-6}	3.41	3.58
4000	362.28	419.87	391.16	434.09	-2.3×10^{-6}	3.41	3.63
5000	349.78	407.36	378.65	434.24	-1.4×10^{-6}	3.41	3.68
6000	341.46	399.01	370.31	434.37	-1.1×10^{-6}	3.41	3.74
7000	335.51	393.05	364.34	434.49	-8.5×10^{-6}	3.41	3.76

3.2 Effect of thickness of the cover plate

Tables 4, 5 and 6 shows the effect of the thickness of the cover plate on heat transfer for $h = 3000\text{W}/\text{m}^2\text{K}$ and $h_{eff} = 15000\text{W}/\text{m}^2\text{K}$, for different materials of the cover plate. For a heat load of $25\text{kW}/\text{m}^2$, it can be seen from the Tables that increase of the thickness of the cover plate causes the temperature of the heat pipe to reduce very slightly. Further, it can be seen from these Tables, by increasing the thickness of the heat pipe heat spread along the axial direction can be improved. This will result in reducing the intensity of heat flux entering the working substance and thereby chances of dry out can be reduced. As the spreading of heat is directly related to the thermal conductivity of the material dry out margin can be improved or reliability in operation can be maintained by using the wall material with high thermal conductivity (Copper). Similar observation is seen for higher heat flux but not shown here for brevity.

4. CONCLUSIONS

A constant liquid vapor interface temperature of the working substance of a heat pipe is solved by considering the axial spread of heat and the axial heat conduction through the cover plate. The temperature of the heat pipe is found to be more controlled by the heat transfer coefficient at the condenser and the thickness has very little impact on its temperature. By increasing the thickness of the heat pipe, heat spread along the axial direction can be improved. This will result in reducing the intensity of heat flux entering the working substance and thereby chances of dry out can be reduced. As the spreading of heat is directly related to the thermal conductivity of the material dry out margin can be improved or reliability in operation can be maintained by using the wall material with high thermal conductivity.

TABLE 4. EFFECT OF THICKNESS OF THE COPPER COVER PLATE ON THE THERMAL PERFORMANCE OF THE HEAT PIPE FOR $q = 25\text{kW}/\text{m}^2$, $h = 3000\text{W}/(\text{m}^2\text{K})$,

t (mm)	Temperature				HBF	HSF	
	Condenser K	Evaporator K	Interface K	K_{eff} $\frac{W}{(mK)}$		Evaporator %	Condenser %
0.5	308.32	311.41	309.87	807.72	0.0052	9.17	9.44
1	308.32	311.33	309.83	827.74	0.0829	12.99	13.29
1.5	308.32	311.29	309.81	840.67	0.2974	14.80	16.12
2	308.32	311.26	309.79	849.88	0.6534	18.09	18.38
5	308.32	311.19	309.76	871.64	4.79	26.11	26.32
10	308.32	311.20	309.76	868.43	14.61	31.96	32.10
15	308.32	311.24	309.78	856.42	24.85	34.95	35.06

TABLE 5. EFFECT OF THICKNESS OF THE ALUMINUM COVER PLATE ON THE THERMAL PERFORMANCE OF THE HEAT PIPE FOR $q = 25\text{kW}/\text{m}^2$, $h = 3000\text{W}/(\text{m}^2\text{K})$,

t (mm)	Temperature				HBF	HSF	
	Condenser K	Evaporator K	Interface K	K_{eff} $\frac{W}{(mK)}$		Evaporator %	Condenser %
0.5	308.31	311.56	309.94	770.92	0.0001	6.58	6.79
1	308.32	311.57	309.95	766.72	0.0062	9.41	9.69
1.5	308.32	311.61	309.97	758.48	0.0348	11.63	11.94
2	308.32	311.65	309.99	748.63	0.1007	13.52	13.84
5	308.32	311.96	310.14	685.24	1.4889	21.43	21.75
10	308.32	312.47	310.40	601.08	6.8897	28.73	28.99
15	308.32	312.91	310.62	544.05	14.47	32.87	33.08

TABLE 6. EFFECT OF THICKNESS OF THE MONEL METAL COVER PLATE ON THE THERMAL PERFORMANCE OF THE HEAT PIPE FOR $q = 25\text{kW}/\text{m}^2$, $h = 3000\text{ W}/(\text{m}^2\text{K})$, $h_{eff} = 15000\text{ W}/\text{m}^2\text{K}$

<i>t</i> (mm)	Temperature			K_{eff} $\frac{W}{(mK)}$	HBF %	HSF	
	Condenser	Evaporator	Interface			Evaporator	Condenser
	K	K	K			%	%
0.5	308.32	312.86	310.59	549.75	-1.4×10^{-6}	2.19	2.27
1	308.31	314.07	311.20	433.93	-1.7×10^{-6}	3.41	3.58
1.5	308.30	315.26	311.79	359.25	2.3×10^{-6}	4.5	4.76
2	308.30	316.44	312.38	307.17	9.28×10^{-6}	5.53	5.89
5	308.27	323.10	315.69	168.61	1.34×10^{-2}	11.35	12.29
10	308.24	332.78	320.47	101.89	0.716	20.13	21.59
15	308.23	340.85	324.45	76.62	3.65	26.85	28.34

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