Flow and heat transfer of fluid in a pulsating heat pipe

Megha Sharma and Basant Singh Sikarwar
Department of Mechanical Engineering, Amity University Uttar Pradesh, Sector 125, Noida 201313, India
msharma@amity.edu

Abstract: Cooling of electronic components are preferred heat pipes imbedded in heat sink because of high heat transfer through them. The oscillating motion of the working fluid due to continual phase change facilitates both sensible and latent heat transfer in pulsating heat pipes. This enables transfer of high heat flux efficiently over long distances. In this work, a mathematical model is developed to optimize the parameters which involves in heat pipe design. The parameters are filling ratio, size of tubes and degree of temperature. In House Finite Volume based CFD solver is used for simulation. The obtained simulation results are validated against experimental results present in literature. These results are useful for understanding thermos-hydrodynamics of pulsating heat pipe for achieving maximum thermal performance

1. Introduction
With the advent of technology and use of portable systems, microelectronics has become a significant part of our day-to-day lives. Advanced electronic components are subjected to high heat loads due to increased electrical demand and smaller sizes. A major concern is the inefficient heat transfer in these devices which causes over-heating of the components. As a result, there are various cooling systems used in electronic components such as air/liquid cooling, synthetic jet cooling, spray cooling, solid cooling, passive cooling using PCM, cold plates, heat pipes etc. Among these cooling techniques, heat pipes have been widely employed in various industries, since they can carry high heat loads[1]. Out of the various heat pipes used for heat transfer, Oscillating heat pipes (OHPs) are very popular due to their simple design and low fabrication cost as compared to conventional designs. Moreover, they do not require an additional wick structure to transport condensed liquid back to the evaporator.

An Oscillating Heat Pipe (OHP) utilizes phase change for effective heat transfer over long distances. It was invented by Akachi in 1996 [2] and since then it has found numerous applications such as solar energy collection, thermal energy storage, cryogenics, drying, electronics cooling, etc. A typical OHP consists of a working fluid inside a meandering channel, which passes through a heat reception and a heat rejection area[3]. The high heat input from the reception area (evaporator section) triggers nucleate boiling of the working fluid and forms vapor bubbles inside the heat pipe. The energy from the evaporator provides the driving force for fluid motion, which causes the vapor bubbles to move towards the heat rejection area (condenser section). The vapor condenses as it encounters the cold surface of the condenser and the temperature is reduced below saturation temperature. The continuous evaporation and condensation of the working fluid creates a circulatory and oscillatory motion of vapor and liquid plugs which facilitates both sensible as well as latent heat transfer through the OHP[4]. Hence, OHPs can transport a high amount of heat flux over long distances.
The hydrodynamics and thermal performance of heat pipes have been studied extensively over the years. Tong et al. [5] conducted an experimental study on a vertically oriented Pyrex glass OHP with Methanol as the working fluid. They observed nucleation boiling at the evaporator, coalescence of bubbles and movement of liquid-vapour in a “slug-train” arrangement inside the heat pipe creating an oscillating flow inside the heat pipe. Some other researchers such as Pouryoussefi[6] have developed a numerical model to simulate chaotic flow in a closed loop pulsating heat pipe under several operating conditions. Lin et al.[7] compared Volume of Fraction (VOF) and Mixture models to study the heat transfer mechanism of miniature oscillating heat pipes via simulations. They reported that the mixture model was more suitable for predicting the thermal performance. However, the VOF model was more suitable for flow visualization as it does not allow interpenetration of phases and liquid-vapor slug flow can be clearly seen. Although several analytical and experimental studies have been carried out to analyse the performance of an OHP[8–10], computational analysis demands more attention. In this background, numerical simulations are performed in this paper using Volume of Fluid (VOF) multiphase model to analyse the effect of heating wall temperature (393-413K) for water as working fluid at 50% filling ratio inside the oscillating heat pipe.

2. METHODOLOGY
2.1 Computational model: The transport phenomenon inside an OHP has been simulated to analyse the effect input heat flux and filling ratio of water on the flow behaviour and thermal performance of the OHP. The 2D computational model of a three-turn closed loop OHP is created. The OHP has an evaporator and a condenser each of length 20mm separated by a 110mm adiabatic section with an inner diameter of 1.3 mm as shown in Fig 2. It is partially filled with the working fluid. The computational domain is discretised using a structured grid with approximately 30,000 elements as shown in Figure 2.

Fig. 1 Illustration of liquid-vapour slug flow in an oscillating heat pipe

Fig. 2 Structured grid on a 2D Oscillating Heat Pipe geometry with 30,000 elements. Location of temperature monitors is shown in evaporator, condenser and adiabatic sections.

A In House Finite Volume based CFD solver was used for modelling transport phenomenon inside the Closed Loop Oscillating Heat Pipe. The details and basic layout of CFD solver was reported
elsewhere[11]. The phase change of water into its vapor and vice-versa was modelled using the in-built multiphase Volume of Fluid (VOF) model, which uses the Euler-Euler approach to calculate phasic volume fraction and track the interface between two phases. An additional parameter ‘α’ whose value lies between 0 and 1, determines the volume fraction of one or more phases in each cell. The value of α in each cell for a phase, say liquid, corresponds to,

α = 0, absence of liquid phase in the cell, 0 < α < 1, interface between liquid and the other phase and α =1, cell is entirely filled with liquid

2.2 Governing equations: The solution of the continuity equation for the secondary phase is used to obtain the value of α which takes the following form,

$$\frac{\partial \alpha_i}{\partial t} + \nabla \cdot (\alpha_i \mathbf{u}) = \frac{s_m}{\rho_i}$$  \hspace{1cm} (1)

For the two-phase vapor-liquid oscillating heat pipe, the primary phase was set as water vapor and secondary phase as water. The density and viscosity of the mixture phase is calculated as using volume fraction and properties of individual phases as:

$$\rho = \alpha_i \rho_i + (1 - \alpha_i) \rho_g$$  \hspace{1cm} (2)

$$\mu = \alpha_i \mu_i + (1 - \alpha_i) \mu_g$$  \hspace{1cm} (3)

A single momentum equation is solved for the entire control volume and the resultant velocity is shared among the phases.

$$\frac{\partial}{\partial t} (\rho \mathbf{u}) + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\nabla p + \nabla \left[ \mu (\nabla \mathbf{u} + \nabla \mathbf{u}^T) - \frac{2}{3} \mu \mathbf{I} \right] + \rho \mathbf{g} + F_{CSF}$$  \hspace{1cm} (4)

Where, FCSF in the momentum equation is the Continuum Surface Force term, which is applied in conjunction with Wall Adhesion to model surface tension.

$$F_{CSF} = \sigma_{lg} \left( \frac{\rho_i \nu_i}{\rho_i + \rho_g} \right)$$  \hspace{1cm} (5)

Similarly, the energy equation is also shared between the two phases and is given as:

$$\frac{\partial}{\partial t} (\rho e) + \nabla \cdot (\rho e \mathbf{u}) = \nabla \cdot (k \nabla T) + \nabla \cdot (\mathbf{pu}) + S_e$$  \hspace{1cm} (6)

Where, k is the thermal conductivity of the mixture phase which is determined using the thermal conductivities and volume fraction of the phases in each cell.

$$k = \alpha_i k_i + (1 - \alpha_i) k_g$$  \hspace{1cm} (7)

The energy term ‘e’ and temperature T in equation (6) are treated as mass averaged variables by the VOF model as per equations (8-10).

$$e = \frac{\alpha_i e_i + \alpha_g e_g}{\alpha_i + \alpha_g}$$  \hspace{1cm} (8)

$$e_i = c_{p,i}(T - T_{sat})$$  \hspace{1cm} (9)

$$e_g = c_{p,g}(T - T_{sat})$$  \hspace{1cm} (10)

2.3 Solution Methodology: To model the boiling and condensation phenomena inside the heat pipe, a user-defined function reported in the literature[12] was used to incorporate energy and mass source terms in equations (1) and (6). Constant temperature boundary conditions were applied to the evaporator and condenser walls along with zero heat flux condition on the adiabatic walls. All the walls were treated as fully wetting with a contact angle of 50. The problem was initialized with a temperature of 368 K to speed up the boiling process. The evaporator was subjected to a range of temperatures (393-413K) keeping condenser temperature fixed in all cases at 298K. The various conditions can be altered as necessary to suit the requirements of the model [13].
A second order polynomial was used to model the surface tension \([20]\) as per equation (11):

\[
\sigma_{lg} = 0.09805856 - 1.845 \times 10^{-5}T - 2.3 \times 10^{-7}T^2
\]  

(11)

The density of liquid water was also modeled as polynomial \([20]\) as per equation (12):

\[
\rho_l = 859.0083 + 1.252209T - 0.0026429T^2
\]  

(12)

The pressure-velocity coupling was achieved using the SIMPLE scheme and the discretization schemes used for pressure, momentum, energy and volume fraction were Body Force Weighted, Second Order Upwind, First Order Upwind and Compressive respectively. Under-relaxation factors of 0.2, 0.5 and 1 for pressure, momentum and energy respectively were used. Convergence criterion was set to \(1e^{-6}\) for energy and \(1e^{-3}\) for momentum and continuity. This model was validated with experimental results shown in the literature \([14]\) for a thermosyphon and was extended to a multi-turn oscillating heat pipe upon agreement. Post validation, a parametric study was performed to study the effect of input heat flux on the hydrodynamics and thermal behavior of the heat pipe. Volume fraction of vapor and temperature distribution inside the OHP was studied and analyzed for evaporator wall temperatures (393-413K).

3. RESULTS AND DISCUSSION

Oscillating flow inside a closed loop three-turn OHP was simulated using volume of fluid multiphase model to study the behavior of OHPs at various operating conditions. The model was successfully able to imitate the oscillating motion of liquid and vapor plugs inside the OHP as shown in Figure 3.

![Fig. 3 Small water vapor bubbles, Taylor bubbles and liquid plugs formed in the OHP at \(T_{e,W}=393\) K.](image)

The evolution of volume fraction of vapor during start-up of a multi-turn oscillating heat pipe is illustrated in figure 4. Figure 4 (a) shows the initial state of the OHP filled with 50% water at temperature 368K. Formation of very small vapor bubbles can be seen in figure 4 (b). After 0.3 seconds, the ascent of vapor slugs towards condenser and transition to a vapor-liquid plug flow is observed.

![Fig. 4 Vapor fraction contours inside oscillating heat pipe during start up for \(T_{e,W}=393\)K and \(T_{c,W}=298\)K. Formation of distinct vapor bubbles and transition into plug flow can be seen.](image)

The flow is observed to follow a liquid-vapor slug regime. A circulatory and oscillatory motion of working fluid is observed as shown in Figure 5. Permanent evaporator dry-out is not observed for the range of evaporator wall temperatures tested (398-413K) in the present study. However, signs of
temporary evaporator dry-out can be seen in Fig 5(e) as it is subjected to a high evaporator wall temperature of 413K.

Fig. 5 Volume fraction contours of water vapor at 15 seconds for various evaporator wall temperatures.

Temperature at various locations (as per fig. 2) was measured and area-weighted average values were obtained at evaporator, condenser and adiabatic regions for a time of 15 seconds. These values of temperature exhibit an oscillating trend as shown in Fig 6. These temperature fluctuations are a characteristic of chaotic flow inside an oscillating heat pipe. Although not much variation is seen in the temperature of fluid in the evaporator section, the adiabatic zone shows an interesting trend of temperature fluctuations. Fig 6(a) and 6(b) show an almost periodic trend after 10 seconds in the adiabatic region. A periodic behavior is also observed in the condenser section for these two cases. Fig 6(a) also shows a similar trend. However, there are no periodic oscillations in the adiabatic section temperature. Due to a low temperature of 393K at the evaporator wall, there could be a delay in the oscillating trend of temperature in the adiabatic region.

Fig. 6 Temperature variations of fluid in oscillating heat pipe at evaporator, condenser and adiabatic sections for various evaporator wall temperatures.

Table 1 Temperature drop across evaporator and condenser sections and average temperatures in evaporator, condenser and adiabatic sections of OHP for various wall heating temperatures.

| Evaporator wall temperature (K) | $T_{f, E}$ (K) | $T_{f, A}$ (K) | $T_{f, C}$ (K) | $\Delta T = T_{f, E} - T_{f, C}$ (K) |
|-------------------------------|----------------|----------------|----------------|---------------------------------|
| 393                           | 391.37         | 366.09         | 314.45         | 76.92                           |
| 398                           | 396.06         | 392.31         | 327.97         | 68.09                           |
| 403                           | 396.98         | 337.72         | 309.16         | 87.81                           |
| 408                           | 403.50         | 353.30         | 314.53         | 88.97                           |
| 413                           | 409.63         | 407.29         | 339.61         | 70.01                           |

The average values of the temperatures of working fluid and temperature drop across in the evaporator, condenser and adiabatic sections are tabulated in Table 1. It is observed that for evaporator wall temperatures 403K and 408K where periodic fluctuations in temperature are seen in Figure 6(c) and 6(d), the temperature drop across the evaporator and condenser is more than 80K. However, for other cases, a lower value of temperature drop is seen from Table 1. These results can be used to
determine the optimum heat load range and can be useful for understanding thermo-hydrodynamics of pulsating heat pipe for achieving maximum thermal performance. Furthermore, a correlation can be obtained for predicting flow behavior in an oscillating heat pipe to prevent evaporator dry-out and maximize its performance.

4. CONCLUSION
Numerical simulations have been performed on a three-turn oscillating heat pipe to visualize the oscillating flow of water and its vapor. The heat pipe was initially filled with 50% of water. Constant temperature boundary conditions were applied to the evaporator and condenser section along with a zero-heat flux boundary condition at the walls of the adiabatic section. The temperature of the evaporator wall was varied (393-413K) while keeping the condenser wall temperature constant at 298K. Simulations were run until quasi-steady state was observed and the solution reached convergence. The temperature of the fluid inside the oscillating heat pipe at various locations was monitored. Volume fraction contours of vapor phase show formation of vapor plugs, and no abnormality was observed. Even at high evaporator wall temperatures, pulsatile flow regime was observed with no signs of a permanent dry-out. The graphs obtained for the temperature of the fluid inside the heat pipe revealed a chaotic behavior inside the heat pipe. These results are useful in determining the range of heat load for the proper functioning of the heat pipe. However, the effect of other parameters such as filling ratio, working fluids, and geometry of the heat pipe must be incorporated for a comprehensive optimization study which forms the basis for further research in this domain.

References
[1] Zuo J, North MT, Wert KL. High Heat Flux Heat Pipe Mechanism. Inter Soc Conf Therm Phenom. 2000.
[2] Akachi H, Polasek F, Stulc P. Pulsating Heat Pipe. In: Proceedings of the 5th International Heat Pipe Symposium., Melbourne, Australia; 1996. p. 208–17.
[3] Ma H. Oscillating heat pipes. Oscillating Heat Pipes. 2015. 1-427 p.
[4] Khandekar S, Manfred Groll. On the definition of pulsating heat pipes: An overview. In: Proc 5th Minsk International Seminar (Heat Pipes, Heat Pumps and Refrigerators). 2003.
[5] Tong BY, Wong TN, Ooi KT. Closed-loop pulsating heat pipe. 2001;21:1845–62.
[6] Pouryoussefi SM, Zhang Y. Analysis of chaotic flow in a 2D multi-turn closed-loop pulsating heat pipe. Appl Therm Eng. 2017;1–8.
[7] Lin Z, Wang S, Shirakashi R, Zhang LW. Simulation of a miniature oscillating heat pipe in bottom heating mode using CFD with unsteady modeling. Int J Heat Mass Transf. 2013;57:642–56.
[8] Sun Q, Qu J, Li X, Yuan J. Experimental investigation of thermo-hydrodynamic behavior in a closed loop oscillating heat pipe. Exp Therm Fluid Sci [Internet]. 2016.
[9] Zhang XM, J. L. Xu, Z. Q. Zhou. Experimental study of a pulsating heat pipe using FC-72, ethanol, and water as working fluids. Exp Heat Transf A J Therm Energy Gener , Transp , Storage , Convers. 2004;17(December 2012):47–67.
[10] Rao Z, Wang Q, Zhao J, Huang C. Experimental investigation on the thermal performance of a closed oscillating heat pipe in thermal management. Heat Mass Transf. 2017;
[11] Sikarwar BS, Khandekar S, Muralidhar K. Simulation of flow and heat transfer in a liquid drop sliding underneath a hydrophobic surface. Int J Heat Mass Transf. 2013;57(2):786–811.
[12] W.H. Lee. A pressure iteration scheme for two-phase flow modeling. Vol. 1, Multiphase Transport Fundamentals, Reactor Safety, Applications. 1980.
[13] Lenhard R, Jandačka J. Two-phase modeling of interphase heat transport from the condensation to evaporation part of the heat-pipe. AIP Conf Proc. 2013;1558:2138–41.
[14] Fadhl B, Wrobel LC, Jouhara H. Numerical Modelling of the Temperature Distribution in a Two-Phase Closed Thermosyphon. :1–27.