PARAMETRIC INVESTIGATIONS ON THE FLOW CHARACTERISTICS OF A CLOSED LOOP PULSATING HEAT PIPE - A NUMERICAL STUDY

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ABSTRACT

Pulsating Heat Pipe (PHP) is two phase passive heat transfer device for low temperature applications. Even though it is a simple, flexible and cheap structure, its complex physics has not been fully understood and requires a robust, validated simulation tool. In the present work the basic theoretical model by H.B. Ma et al has been updated with the inclusion of capillary effect in order to characterise the pulsating flow for various refrigerants which could be used in PHPs. The mathematical model is solved using explicit embedded Range-kutta method and the slug displacement and velocity are investigated under various influencing parameters.

Keywords: Mathematical Modelling, Pulsating Heat Pipe (PHP), Slug Flow.

1. INTRODUCTION

Heat flux levels continue to increase because of rapid increase in chip and power density along with continuous miniaturisation of modern electronic devices. Thermal management of such Micro electronics systems is becoming a challenge of the day and caught attention of researchers to develop efficient cooling systems. The Pulsating (or) Oscillating Heat Pipe is a passive two phase heat transfer device developed by Akachi [1, 2]. It is simple in structure with a coil of capillary dimensions filled with certain working fluid in it under evacuating conditions and extended from the heat source to sink. Unlike a conventional heat pipe, PHP does not contain the wick structure to return the condensate back to the evaporator section. Instead, PHP works on the principal of fluid pressure oscillations that are created by means of differential pressure across the vapour plugs from evaporator to condenser and back. The vapour formed at the evaporator pushed towards the condenser in the form of discrete vapour bubbles amidst pockets of fluids. The vapour gets condensed at the condenser releasing the latent heat of vaporization and returns to the evaporator to complete the cycle. The thermal performance of an actual PHP depends upon the temperature gradient prevails between the evaporator and condenser section. Various influencing parameters[3] that are directly or indirectly affect the pulsating flow in a Closed Loop Pulsating Heat Pipe are listed below:

1.1 Design/Geometrical parameters

- Tube diameter and material
- Orientation of PHP
- Number of turns
- Length of evaporator and condenser section
- Bend radius
1.2 Operating parameters

- Fill ratio
- Heat input
- Working fluid
- Dry out condition

Out of all, the diameter place a vital role in deciding whether or not the capillary flow exists inside of PHP. If the diameter of a PHP is less than its critical diameter, capillary flow exists otherwise, flow gets stratified. The critical diameter is defined as

\[ D_{crit} = \sqrt{\frac{\sigma}{\vartheta (\rho_l - \rho_v)}} \]  (1)

A Pulsating Heat Pipe system looks simple, However understanding of its physics related to various processes such as Thermo-Hydro dynamics, Two phase flow capillary actions, Phase change etc. is relatively complex. Therefore many challenging issues remain modelled unsatisfactorily. The understanding of physical phenomena occurring in a PHP is explained better from the experimental results. Quite a few attempts have been presented in the literature to model a PHP system with various degrees of approximation and success. Shafii et al. [5] presented a multi plug model for heat transfer estimation both for CEPHP and the CLPHP. In their model the total number vapour plugs were decreased to total number of heating sections during simulation. Zhang and Faghri [6] proposed a multi plug model for Vertical CEPHP and studied the effect of number of turns on the fluid oscillation frequency. Jang-Soo kim et al [7] developed a theoretical model based on a separated flow model with two liquid slugs and three vapour plugs and investigated the effect of diameter surface tension and fill ratio on the performances of the PHP. Khandekar and Groll [8] proposed lumped parametric model for a single closed loop PHP. Xin-She Yang et al [9] devised a mathematical model to make the predictions regarding the start-up characteristics.

In almost all the numerical studies the Thermo-Hydro Dynamic system in PHP is considered analogous to forced damped vibration system. In some cases an attempt of applying the mass, momentum and energy equations to a control volume of PHP has been done to obtain the mathematical model. But the experimental results have not been able to match with mathematical models developed. A comprehensive mathematical model which could be used in the design of PHP has eluded researches till now.

In all the above works pressure difference in evaporator and condenser is considered as driving force while the heat transfer in PHP is studied. Whereas a mathematical model based on the temperature difference between evaporator and condenser was proposed by Ma et al [10] in order to deal with the pulsating behaviour of the fluid in a PHP. The authors relate the pressure difference between evaporator and condenser with the temperature difference using Clasius-Clapeyron’s equation. The model was solved using Laplace Transforms and the slug displacements were obtained at different operating parameters. The flow characteristics were derived for the PHP in the saturation region.

2. MATHEMATICAL MODEL

In the present work, parametric investigation on the flow characteristics of a single closed loop PHP has been carried out through a numerical study. The Internal flow patterns in a PHP are function of the applied heat flux. Since slug flow is the primary flow pattern in PHPs, most of the ongoing research works on modelling have focused on slug flow. The present work adopts the mathematical model proposed by Ma et al to study the slug flow characteristics of PHP. The governing differential equations have been derived from the physical model shown in figure 1.
The characteristic length of PHP consists of the length of evaporator section $L_e$, adiabatic section $L_a$ and condenser section $L_c$. Hence,

$$L = L_e + L_a + L_c$$  \hspace{1cm} (2)

As the fluid flows in a PHP, the fluid gets evaporated in the evaporator and condensed in the condenser. This results in the volume expansion and contraction of the bubbles. This causes an oscillating motion which affects the saturation temperature in the evaporator and condenser section. If the maximum and minimum temperature difference between the evaporator and condenser section are $\Delta T_{\text{max}}$ and $\Delta T_{\text{min}}$ respectively, then the temperature difference between the evaporator and condenser section will vary between $\Delta T_{\text{max}}$ and $\Delta T_{\text{min}}$ and is given by $\frac{\Delta T_{\text{max}} - \Delta T_{\text{min}}}{2}$. Considering the oscillating nature of PHP and system oscillation frequency as $\omega$, the thermal driving potential can be written as

$$\Delta T = \frac{\Delta T_{\text{max}} - \Delta T_{\text{min}}}{2} \left[ 1 + \cos(\omega t) \right]$$  \hspace{1cm} (3)

The pressure difference between evaporator and condenser can be related to the thermal driving potential using Clausius-Clapeyron’s equation as.

$$\Delta p = \Delta T \frac{h_f \rho_e}{T_e}$$  \hspace{1cm} (4)

Hence the driving force causing the pulsating motion in a PHP is expressed as

$$F_d = \left( \frac{Ah_f \rho_e}{T_e} \right) \left( \frac{\Delta T_{\text{max}} - \Delta T_{\text{min}}}{2} \right) \left[ 1 + \cos(\omega t) \right]$$  \hspace{1cm} (5)

This driving force overcomes the (i) viscous force which arises due to the interaction between liquid/vapour and the pipe walls (ii) force due to vapour pressure which arise due to volume contraction and expansion of bubbles and (iii) force due to inertia (iv) capillary force due to meandering tube diameter.

From Newton’s law, the governing equation for fluid flow in a PHP is

$$\left( \rho_i L_i + \rho_v L_v \right) \frac{d^2 x}{dt^2} + \left[ 32 \left( \frac{\mu_i L_i + \mu_v L_v}{D^2} \right) \right] A \frac{dx}{dt} + \frac{A \rho_v R_s T}{L_v} x + N \left[ 2\sigma (\cos \theta_v + \cos \theta_i) \right] A$$

$$= \left( \frac{Ah_f \rho_v}{T_e} \right) \left( \frac{\Delta T_{\text{max}} - \Delta T_{\text{min}}}{2} \right) \left[ 1 + \cos(\omega t) \right]$$  \hspace{1cm} (6)

The inertia force, the viscous force and the force due to vapour pressure in the above equation are given by

$$F_i = \left( \rho_i L_i + \rho_v L_v \right) \frac{d^2 x}{dt^2}$$  \hspace{1cm} (7)

$$F_v = \left[ 32 \left( \frac{\mu_i L_i + \mu_v L_v}{D^2} \right) \right] A \frac{dx}{dt}$$  \hspace{1cm} (8)

$$F_{vp} = \frac{A \rho_v R_s T}{L_v} x$$  \hspace{1cm} (9)
\[ F_{cap} = N \left[ 2\sigma \left( -\cos \theta + \cos \theta k \right) \right] A \]

The governing equation (6) for fluid flow in a PHP is similar to the governing equation of forced damped mechanical vibration with the following initial conditions:

\[ x = 0 \text{ and } \frac{dx}{dt} = 0 \text{ at } t = 0 \]

In the present study to solve both slug displacement and slug velocity Embedded Runge-Kutta formula as given in the MATLAB with the nomenclature ODE 45\(^{[11]}\) is used.

### 3. RESULTS AND DISCUSSION

Ma et.al work was limited only to study the effect of parameters on the displacement. Thus the first step in the present study is to verify the patterns of present results with Ma et.al work. The refrigerants R12, R22, R123 and R134a are used as the working fluids throughout the study. The slug velocity has been obtained from the solution of the governing differential equations.

#### 3.1 Parametric Studies on Displacement

##### 3.1.1. Effect of working fluid

![Fig. 2 Effect of Working Fluid on Displacement](L = 304.8 mm, \( T = 30^\circ C \), \( D = 1.65 \text{ mm} \), \( \phi = 50\% \), \( \Delta T = 5K \))

The variation of displacement with respect to time for R22, R123 and R134a with diameter of 1.65 mm, fill ratio of 50\% and operating temperature of 30\(^\circ\)C is shown in Fig. 2. It is clear from the figure that the frequency of oscillation and amplitude are more in case of R22 compared to R123 and R134a. The system takes more time to reach the steady state in case of R22.

##### 3.1.2 Effect of fill ratio

![Fig. 3 Effect Of Fill Ratio On Displacement](R123, L = 304.8 mm, \( T = 30^\circ C \), \( D = 1.65 \text{ mm} \), \( \Delta T = 5K \))

3.1.2 Effect of fill ratio
The variation of fill ratio has significant effect on the performance of PHP. Fill ratio is basically defined as [12]

\[ \phi = \frac{\text{Volume of Liquid}}{\text{Total Volume}} = \frac{\pi D^2 L}{4} \frac{L}{L} \]

The heat pipe works as true pulsating device in the range of 20-80\% fill ratio [13]. Figure 3 shows the comparison of fluid displacements with respect to the time for fill ratios of 50\%, 60\% and 70\% with R123 as the working fluid at diameter of 1.65mm, operating temperature of 30°C and ΔT = 5K. It can be seen from Fig. 3 that the time to reach the steady state increases with decrease in fill ratio. The amplitude of displacement is inversely proportional to the fill ratio.

3.1.3 Effect of diameter

![Image](image.png)

Fig. 4 Effect of Diameter on Displacement
(R123, L = 304.8 mm, T = 30°C, D = 1.65 mm, \(\phi = 50\%\), ΔT = 5K)

The variation of displacement with respect to time for different diameters with R123 as the working fluid with fill ratio of 50\% at operating temperature of 30°C and ΔT = 5K is shown in Fig. 4. It is reported in the literature that the maximum diameter that can hold a vapour bubble in a PHP tube is 2.5 mm [5]. In the present study, three diameters of 1.14 mm, 1.65 mm and 2.16 mm are considered. It can be observed from Fig. 4 that the amplitude of displacement is directly proportional to the diameter. The frequency of oscillation remains same for all the diameters studied. With the increase in diameter, The system takes more time to reach the steady state.

3.1.4 Effect of operating temperature

![Image](image.png)

Fig. 5 Effect of Operating Temperature on Displacement
(R123, L = 304.8 mm, T = 30°C, D = 1.65 mm, \(\phi = 50\%\), ΔT = 5K)

The operating temperature has a significant effect on the fluid flow characteristics of a PHP. A change in the operating temperature results in considerable change in the energy levels of the fluid. Fig. 5 shows the variation of displacement as a function of time at different operating temperatures for R123 with diameter of 1.65 mm and fill ratio of
50%. With increase in the operating temperature an increase in the amplitude of displacements can be observed. It is also seen that with increase in the operating temperature the frequency of oscillation also increases.

3.2 Parametric Studies on Slug Velocity

3.2.1 Effect of fill ratio

![Figure 6: Effect of Fill Ratio on Slug Velocity](image)

Fig. 6 shows the variation of root mean square values of velocity of the slug with respect to time for different fill ratio with R123 as the working fluid at diameter of 1.65 mm and operating temperature of 30°C. Considering the pulsating nature of fluid flow in a PHP, the root mean square values of velocity are evaluated for each cycle and plotted with respect to time. The momentum of the fluid is less in the initial time steps resulting in lower slug velocity. The slug velocity increases with increase in time and reaches saturation at elapsed time as the fluid gains momentum as shown in Fig. 6.

The variation of fill ratio shows a significant effect on the performance of PHP. In the present study, the fill ratio is varied in the range of 50% to 70%. It is observed from Fig. 6 that due to the prevalence of more fluid displacement at lower fill ratios consequently higher slug velocities are observed at lower fill ratios.

3.2.2 Effect of diameter

![Figure 7: Effect of Diameter on Slug Velocity](image)

Fig. 7: Effect of Diameter on Slug Velocity

(R123, L = 304.8 mm, T = 30°C, D = 1.65 mm, $\phi = 50\%$, $\Delta T = 5K$)

The variation of slug velocity with respect to time for different diameters with R123 as the working fluid with fill ratio of 50% at operating temperature of 30°C is shown in Fig. 7. Due to inertia, the velocity of the slug remains almost same at all diameters in the initial time steps. But with the increase in the momentum of the fluid, the velocity of the slug shows an increasing trend with an increase in diameter. Due to the reduction in viscous force the resistance to the fluid flow decreases with increase in diameter as seen from equation (6). Hence, higher slug velocities are observed at higher diameter. It is also seen that larger time has been taken place to attain the steady state by the system at higher diameter.
3.2.3 Effect of operating temperature

![Fig. 8: Effect of Operating Temperature on Slug Velocity](image)

Fig. 8: Effect of Operating Temperature on Slug Velocity
(R123, L = 304.8 mm, T = 30°C, D = 1.65 mm, ϕ = 50%, ΔT = 5K)

Fig. 8 shows the variation of velocity of the slug as a function of time at different operating temperatures for R123 with diameter of 1.65 mm and fill ratio of 50%. The thermal energy available for the momentum of the working fluid is directly proportional to operating temperature. This results in higher slug velocity at higher operating temperature. It is also seen that the momentum of the fluid is very less at 10°C and 20°C as the displacement of the fluid is less due to lower thermal energy. Hence it is advisable to operate PHP at higher operating temperatures.

3.2.4 Effect of working fluid

![Fig. 9: Effect of Working Fluid on Slug Velocity](image)

Fig. 9: Effect of Working Fluid on Slug Velocity
(R123, L = 304.8 mm, T = 30°C, D = 1.65 mm, ϕ = 50%, ΔT = 5K)

The variation of RMS slug velocity with respect to time for different working fluids with diameter of 1.65 mm, fill ratio of 50% and operating temperature of 30°C is shown in Fig. 9. It is observed that the magnitude of slug velocities is optimum for R32 compared to R22, R134a and R123. Higher values of slug velocities for R32 can be attributed to its lower value of latent heat.

3.2.5 Effect of temperature difference

![Fig. 10: Effect of Temperature Difference on Slug Velocity](image)

Fig. 10: Effect of Temperature Difference on Slug Velocity
(R123, L = 304.8 mm, T = 30°C, D = 1.65 mm, ϕ = 50%)
In the present study, the temperature difference between the evaporator and condenser is taken as the driving force for the fluid motion. A variation in this value of temperature difference results in a change in the momentum transport and heat transport values. Fig. 10 shows the effect of this temperature difference between evaporator and condenser on the slug velocity for R123 at fill ratio of 50%, diameter of 1.65mm and operating temperature of 30°C. It is clear from Fig. 10 that the slug velocities is directly proportional to the temperature difference. As the energy level increases at higher temperature difference, it results in better momentum and hence better heat transport. Thus it is desirable to operate the PHP with higher temperature difference between the evaporator and condenser.

3.2.6 Influence of forces on pulsating flow

Fig. 11 Influence of forces on pulsating Flow

4. CONCLUSIONS

The following conclusions have been made from the numerical study

- The pulsating flow characteristics in a PHP are analysed by the present study.
- Higher slug velocities are associated with lower fill ratio, higher diameter, higher operating temperature and higher temperature difference between evaporator and condenser.
- Higher values of slug velocities are observed in case of R123. This shows that R123 exhibits better fluid flow characteristics of PHP compared to other refrigerants.
- The effect of capillary forces found to be significant at lower diameter, lower operating temperature, higher fill ratio and lower temperature difference between evaporator and condenser.
- It has been observed that the inertia force is a predominant opposing force compared to other forces.

REFERENCES

3. S.Y Nagvase, P.R.Pachghare,” Parameters affecting the functioningof closed loopPulsating heat pipe-a review” Vol 2(1), 2013, pp. 35-39

11. **Ryuichi Ashino, Michihoro Nagase, Remi Vaillancourt** "Behind and Beyond the MAT LAB ODE suite", Expanded version of lecture given at Ritsumeikin University, Kusatsu, Shiga, Japan, 2000, pp. 525-8577.
