Numerical Analysis of Pulsating Heat Pipe Based on Separated Flow Model

Jong-Soo Kim*
School of Mechanical Engineering, Pukyong National University, San 100, Yongdang dong, Nam-gu, Pusan 608-737, Korea
Yong-Bin Im, Ngoc Hung Bui
Department of Refrigeration and Air Conditioning Engineering, Graduate School of Pukyong National University, San 100, Yongdang dong, Nam-gu, Pusan 608-737, Korea

The examination on the operating mechanism of a pulsating heat pipe (PHP) using visualization revealed that the working fluid in the PHP oscillated to the axial direction by the contraction and expansion of vapor plugs. This contraction and expansion is due to the formation and extinction of bubbles in the evaporating and condensing section, respectively. In this paper, a theoretical model of PHP was presented. The theoretical model was based on the separated flow model with two liquid slugs and three vapor plugs. The results show that the diameter, surface tension and charge ratio of working fluid have significant effects on the performance of the PHP. The following conclusions were obtained. The periodic oscillations of liquid slugs and vapor plugs were obtained under specified parameters. When the hydraulic diameter of the PHP was increased to $d=3$ mm, the frequency of oscillation decreased. By increasing the charging ratio from 40 to 60 by volume ratio, the pressure difference between the evaporating section and condensing section increased, the amplitude of oscillation reduced, and the oscillation frequency decreased. The working fluid with higher surface tension resulted in an increase in the amplitude and frequency of oscillation. Also the average temperature of vapor plugs decreased.

Key Words : Pulsating Heat Pipe (PHP), Separated Flow Model, Heat Transport

Nomenclature

A : Tube cross sectional area
f : Friction coefficiency
h : Heat transfer coefficient
h_{fg} : Laten heat of vaporization
k : Thermal conductivity
L : Length
m : Mass
$\dot{m}$: Mass flow rate
R : Gas constant

Greek symbols

μ : Dynamic viscosity
ρ : Density
σ : Surface tension
τ : Shear stress

Subscripts

cond : Condensation
evp : Evaporation
f : Liquid
fg : Vapor plug
g : Gas
l : Liquid slug
lv : Left end of vapor plug
rv : Right end of vapor plug

* Corresponding Author,
E-mail : jskim@pknu.ac.kr
TEL : +82-51-620-1502; **FAX** : +82-51-611-6368
School of Mechanical Engineering, Pukyong National University, San 100, Yongdang dong, Nam-gu, Pusan 608-737, Korea. (Manuscript **Received** January 25, 2005; **Revised** August 5, 2005)

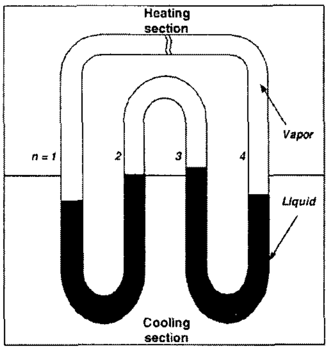
1. Introduction

The pulsating heat pipe (PHP) is a very promising heat transfer device (Akachi et al., 1994). In addition to its excellent heat transfer performance, it has a simple structure: in contrast with conventional heat pipe, there is no wick structure to return the condensed working fluid back to the evaporating section. The PHP can operate successfully for all heating modes. The heat input, which is the driving force, increases the pressure of the vapor plugs in the heating section. In turn, this pressure increase will push neighboring vapor plugs and liquid slugs toward the cooling section, which is at a lower pressure. However, due to the continuous heating, small bubbles formed by nucleate boiling grow and coalescence to become vapor plugs. Nagata et al. (1998) analyzed the vapor plug propagation phenomenon by the one-dimensional homogeneous flow model. This phenomenon indicated that the self-excited oscillation of liquid columns might occur as the turn number was reduced. Miyazaki et al. (1998) proposed a theoretical model, which was supported by experimental results, to predict the oscillatory flow characteristics of PHP. Maezawa et al. (2000) presented studies, which propose the existence of chaos in PHP under some operating conditions. Dobson et al. (1999) have applied the governing equations of mass, momentum and energy to a simplified PHP consisting of single liquid slug with vapor bubbles on both the sides.

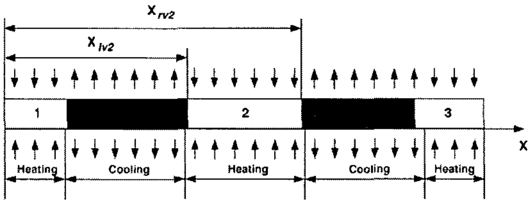
In this study, a theoretical model of PHP was presented that based on the separated flow model with two liquid slugs and three vapor plugs. It studied that the diameter, surface tension and charge ratio of working fluid have significant effects on the performance of the PHP.

2. Theoretical Model

Fig. 1(a) shows a theoretical model of the PHP. The bends are not considered and it is assumed that the PHP is a straight tube (Fig. 1 (b)). There are three vapor plugs and two liquid slugs in the PHP. The pressure and the temperature of all vapor plugs are initially equal. A control volume of a liquid slug bounded with two vapor plugs is shown in Fig. 2. Suppose the initial value of $\Delta p = p_{vi} - p_{v(i+1)}$ is greater than zero, and part of the vapor plug in the left side of the liquid slug is in contact with the cooling section, condensation occurring in the left vapor plug will result in a decrease in the pressure of the left vapor plug, p_{vi}. On the other hand, part of the right heating section is in contact with the liquid slug and boiling may occur at the contact area of the right heating section and liquid slug, which causes increasing the vapor pressure of the right vapor plug, $p_{v(i+1)}$. The liquid plug will be pushed back to the left side due to the pressure difference between the two vapor plugs, $\Delta p = p_{vi} - p_{v(i+1)} < 0$. When $\Delta p = p_{vi} - p_{v(i+1)}$ becomes zero, there is



(a)



(b)

Fig. 1 Theoretical model of the PHP

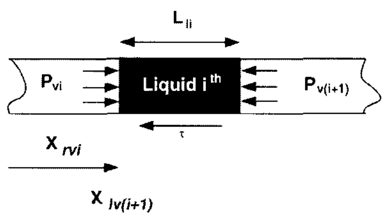


Fig. 2 Control volume of i^{th} liquid slug

no evaporation or condensation in the two vapor plugs, but the liquid slug keeps moving due to its inertia. When part of the liquid slug enters the left heating section, part of the right vapor plug will be in contact with the cooling section. At this point, the boiling in the left vapor plug and condensation in the right vapor plug will change the sign of Δp, and this will result in the motion of the liquid slug to the right side. The pulsation of the liquid slug can be maintained by alternative evaporation or condensation in the two vapor plugs.

The following assumptions are made in order to model the heat transfer and pulsation characteristics in the PHP :

(1) The liquid is incompressible and the vapor is assumed to behave as ideal gases

(2) Evaporative and condensation heat transfer coefficients are assumed to be constants

(3) Heat transport in the thin film is due only to the conduction in the radial direction

(4) The pressure losses at the bend are not considered

3. Govering Equations

3.1 Governing equations of vapor plugs

The continuity equation for the i^{th} vapor plug is

$$\frac{dm_{vi}}{dt}=\dot{m}_{in,vi}-\dot{m}_{out,vi} \tag{1}$$

Where $\dot{m}_{in,vi}$ is the mass flow rate transfer into the vapor plug due to evaporation, and $\dot{m}_{out,vi}$ is the mass flow rate transfer from the vapor plug due to condensation of the i^{th} vapor plug and can be calculated by the following equations :

$$\dot{m}_{in,vi}=\frac{Q_{evp,vi}}{h_{fg}} \tag{2a}$$

$$\dot{m}_{out,vi}=\frac{Q_{cond,vi}}{h_{fg}} \tag{2b}$$

Where $Q_{evp,vi}$ and $Q_{cond,vi}$ represent heat transfer resulting from the evaporation and condensation of the i^{th} vapor plug

$$Q_{evp,vi}=h_{evp}\pi dL_{hi}(T_h-T_{vi}) \tag{3a}$$

$$Q_{cond,vi}=h_{cond}\pi dL_{ci}(T_{vi}-T_c) \tag{3b}$$

Where h_{evp} and h_{cond} represent evaporative and condensation heat transfer coefficient, L_{hi} and L_{ci} are the length of i^{th} vapor plug in the heating and cooling section, respectively.

The energy equation for a vapor plug is

$$\frac{d(m_{vi}u_{vi})}{dt}=\dot{m}_{in,vi}h_{vi}-\dot{m}_{out,vi}h_{vi}-p_{vi}\frac{dV_{vi}}{dt} \tag{4}$$

Letting $u=c_vT$ and $h=c_pT$, may be rewritten as

$$\begin{aligned} m_{vi}c_v\frac{dT_{vi}}{dt}+c_vT_{vi}\frac{dm_{vi}}{dt} \\ =(\dot{m}_{in,vi}-\dot{m}_{out,vi})c_pT_{vi}-p_{vi}\frac{dV_{vi}}{dt} \end{aligned} \tag{5}$$

Substituting Eq. (1) into Eq. (5)

$$m_{vi}c_v\frac{dT_{vi}}{dt}=(\dot{m}_{in,vi}-\dot{m}_{out,vi})RT_{vi}-p_{vi}A\frac{dx_{vi}}{dt} \tag{6}$$

The pressure of the i^{th} vapor plug, p_{vi}, is calculated by using the ideal gas law

$$p_{vi}V_{vi}=m_{vi}RT_{vi} \tag{7}$$

3.2 Governing equations of liquid slugs

The continuity equation for i^{th} liquid slug bounded with two vapor plugs as shown in Fig. 2 can be found from the following equation :

$$\begin{aligned} \frac{dm_{li}}{dt}&=\dot{m}_{in,li}-\dot{m}_{out,li} \\ &=\frac{1}{2}\left(\frac{dm_{vi}}{dt}+\frac{dm_{v(i+1)}}{dt}\right) \end{aligned} \tag{8}$$

The momentum equation for i^{th} liquid slug is

$$\begin{aligned} \frac{dm_{li}v_{li}}{dt}=(p_{vi}-p_{v(i+1)})A-\pi dL_{li}\tau \\ -\pi d\sigma-(-1)^n m_{li}g \end{aligned} \tag{9}$$

Where n indicate the tube number. Since it is assumed that the PHP is a straight tube, gravity has different signs at different locations. In the model, there are four parallel tubes, which are labeled from one to four (Fig 1(a)). At the first or third tube, the gravity term in Eq. (9) is positive but at the second or fourth tube, it is negative.

The shear stress acting between i^{th} liquid slug and the tube wall can be determined from (Abdul-Razzak et al., 1995)

$$\tau=\frac{1}{2}f_{li}\rho_l v_{li}^2 \tag{10}$$

Where the friction coefficient can be determined by from the Hagen-Poiseuille flow

$$f_{li}=\frac{16}{\mathrm{Re}_{li}} \tag{11}$$

3.3 Evaporation

The experimental results by flow visualization of the PHP in the works of Kim et al. (1999) reveal that the generation of bubbles in the evaporating section was a nucleate boiling process. The heat transfer rate in nucleate boiling is very high, but an adequate physical description of the thermo-dynamical process is still not available. Dario Delmastro and Alejandro Clausse (1994) proposed a general expression to correlate the nucleate boiling regime as the following:

$$q=C(p,\ \mathit{fluid},\ \mathit{surface})\,(T_w-T_{sat})^n \tag{12}$$

The recommended value for the exponent n is 3.

3.4 Condensation

The theoretical model of film condensation in the cooling section is shown in Fig. 3. Because every vapor plug is bounded by two liquid slugs, the vapor velocity in the model is low in most parts of the cooling section. Also the heat transfer in the thin liquid film is assumed due only to the conduction in the radial direction. Hence, the mean value of condensation heat transfer coefficient, h_{cond}, may be determined by the following expression (John, 1994)

$$h_{cond}=0.943\left[\frac{\rho_f(\rho_f-\rho_g)\,g\sin\theta h_{fg}k_f^3}{\mu_f L_c(T_{vi}-T_w)}\right]^{1/4} \tag{13}$$

Where, θ is the inclination angle of the PHP.

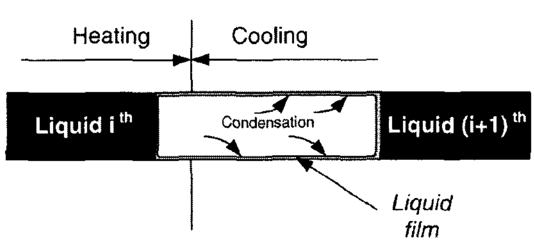


Fig. 3 Control volume of i^{th} vapor plug

3.5 Heat transfer

Heat transfer in the PHP is defined as the total heat transferred from the heating sections to the cooling sections due to evaporation and condensation of working fluid.

Evaporation and condensation heat transfer for each vapor plug can be calculated by

$$Q_{in,vi}=\dot m_{in,vi}h_{fg} \tag{14a}$$

$$Q_{out,vi}=\dot m_{out,vi}h_{fg} \tag{14b}$$

The total heat transferred into and out of the PHP can be calculated by

$$Q_{total,in}=\sum_{i=1}^{N}Q_{in,vi} \tag{15a}$$

$$Q_{total,out}=\sum_{i=1}^{N}Q_{out,vi} \tag{15b}$$

4. Numerical Procedure

The governing equations of vapor plugs and liquid slugs can be solved numerically by using explicit finite difference method (Necati Qzisik, 1994). Let m_{vi}^{new}, T_{vi}^{new}, Δx_{vi}^{new}, m_{li}^{new} and $\mathbf{v}_{li}^{new}$ denote the values of m_{vi}, T_{vi}, Δx_{vi}, m_{li} and $\mathbf{v}_{li}$ at the end of a time step. Then the finite difference equations corresponding to the governing equations are

$$m_{vi}^{new}=m_{vi}+(\dot m_{in,vi}-\dot m_{out,vi})\Delta t \tag{16}$$

$$T_{vi}^{new}=T_{vi}+\frac{(\dot m_{in,vi}-\dot m_{out,vi})RT_{vi}\Delta t-p_{vi}A(\Delta x_{vi}^{new}-\Delta x_{vi})}{m_{vi}C_v} \tag{17}$$

$$p_{vi}^{new}=\frac{m_{vi}^{new}RT_{vi}^{new}}{V_{vi}} \tag{18}$$

$$m_{li}^{new}=m_{li}+\frac{1}{2}[(\dot m_{in,vi}-\dot m_{out,vi})+(\dot m_{in,v(i+1)}-\dot m_{out,v(i+1)})]\Delta t \tag{19}$$

$$m_{li}^{new}=m_{l1}+\left[(\dot m_{in,v1}-\dot m_{out,v1})+\frac{1}{2}(\dot m_{in,v2}-\dot m_{out,v2})\right]\Delta t \tag{19a}$$

$$m_{l(N-1)}^{new}=m_{l(N-1)}+\left[\frac{1}{2}(\dot m_{in,v(N-1)}-\dot m_{out,v(N-1)})+(\dot m_{in,vN}-\dot m_{out,vN})\right]\Delta t \tag{19b}$$

$$m_{li}^{new}\mathrm{v}_{li}^{new}=m_{li}\mathrm{v}_{li}+\lfloor(P_{vi}-P_{v(i+1)})A-\pi dL_{li}\tau-\pi d\sigma-(-1)^{n}m_{li}\cdot g\rfloor\Delta t \quad (20)$$

The displacement and the end positions of the vapor plugs at each time step can be found by using the liquid slug velocity obtained from Eq. (24) as follows

$$\begin{cases} x_{rv,i}^{new}=x_{rv,i}+\mathrm{v}_{li}\Delta t \\ x_{lv,i}^{new}=x_{lv,i}+\mathrm{v}_{l(i-1)}\Delta t \end{cases} \quad (21)$$

$$\begin{cases} x_{rv,N}=L \\ x_{lv,1}=0 \end{cases} \quad (22)$$

Where N is the number of vapor plugs, $x_{lv,i}$ and $x_{rv,i}$ are the left and right coordinates of i^{th} vapor plug measured from the origin as shown in Fig. 1(b)

$$\begin{cases} \Delta x_{rv,i}^{new}=x_{rv,i}^{new}-x_{rv,i} \\ \Delta x_{lv,i}^{new}=x_{lv,i}^{new}-x_{lv,i} \end{cases} \quad (23)$$

$$\begin{cases} \Delta x_{rv,N}^{new}=0 \\ \Delta x_{lv,1}^{new}=0 \end{cases} \quad (24)$$

$$\Delta x_{vi}^{new}=\Delta x_{rv,i}^{new}-\Delta x_{lv,i}^{new} \quad (25)$$

Where, Δx_{vi}^{new} is the change in length of the i^{th} vapor plug, $\Delta x_{rv,i}^{new}$ and $\Delta x_{lv,i}^{new}$ are the changes in location of the ends of the i^{th} vapor plug. Time step independent of the numerical solution for the model was varied by systematically varying the time step. It was found that varying the time step from 10^{-5} s to 10^{-6} s results in less than 0.2 percent variation in the locations of the liquid slugs. Therefore, the time step used in the numerical solution is 10^{-5} s.

5. Results and Discussion

The parameters used are listed in Table 1. Figure 4 shows the variations of pressure and positions of the two ends of the first vapor plug with respect to time. Initially the right end of the first vapor plug is located in the cooling section and condensation takes place. When the pressure is low enough, the adjacent liquid slug starts moving back toward the heating section. As it moves back, the pressure of the first vapor plug increases due to the compression. When the right end moves into the heating section, the condensation ceases.

Due to the evaporative of working fluid in the heating section, the pressure of the first vapor

Table 1 Initial values of the PHP

Initial temperature of vapor plugs	T_{vi}=35℃
Total length	L=0.88 m
Total number of vapor plugs	3
Total number of liquid slugs	2
Length of each vapor plug	$L_{v1}=L_{v3}$=0.12 m L_{v2}=0.24 m
Length of each liquid slug	$L_{l1}=L_{l2}$=0.2 m
Wall temperature in cooling section	T_c=20℃
Wall temperature in heating section	T_h=40℃
Diameter	d=0.0015 m
Total length of heating section	L_h=0.44 m
Total length of cooling section	L_c=0.44 m

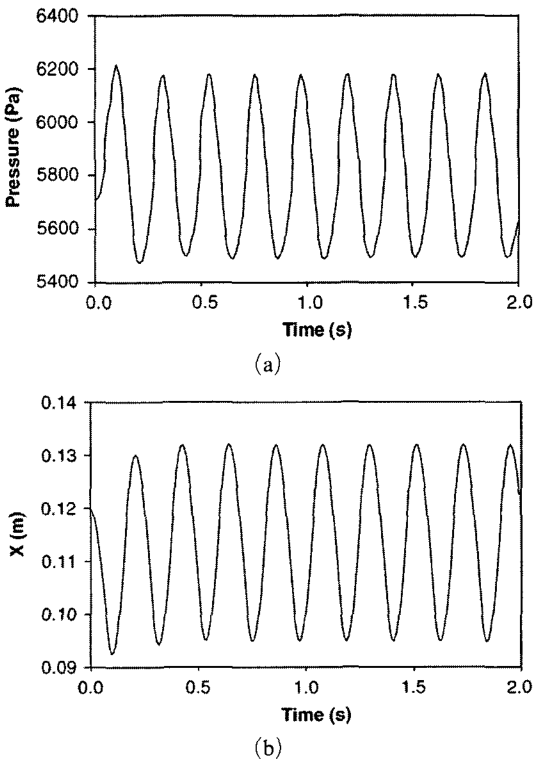


Fig. 4 Variation of pressure and the end positions of the first vapor plug with time

plug keeps increasing until the pressure in the first vapor plug is high enough to push the adjacent liquid slug toward the cooling section. This phenomenon repeats itself and periodic oscillation occurs. Fig. 4(b) shows the positions of the two ends of the first vapor plug. Since the first vapor plug is located at the end of the OCHP, the left end is always located at $x=0$ m. The right end oscillates between $x=0.095$ m and $x=0.132$ m.

Figure 5 shows the variations of pressure and positions of the two ends of the second vapor plug. Since the two ends are free to move into the heating and cooling sections, the amplitude of the oscillation is low. Most of the time when one end is in the heating section the other end would be in the cooling section. This causes the variation in pressure of the second vapor plug smaller than that of the first vapor plug. Figure 5(b) shows the positions of the two ends of this vapor plug. The solid and dashed lines represent the location of the right and left ends, respectively. The two ends always move in the same direction, and when one moves into the heating section the other one moves into the cooling section.

Figure 6 represents pressure and position variations of the two ends of the third vapor plug with time. The oscillating trends of pressure and positions of the two ends are the same as those of the first vapor plug and the phase difference is 180°.

The rate of evaporative and condensation heat transfer of each individual vapor plug, when periodic oscillation is obtained, is shown in Figs. 7~9. It can be seen from Fig. 7 that when the right end of the first vapor plug moves into the heating section, the evaporative heat transfer increases. When the right end continues moving inside the heating section and compresses the first vapor plug, the evaporative heat transfer increases to its maximal value. As the vapor plug expands, its pressure drops and heat transfer decreases until the right end moves into the cooling section where

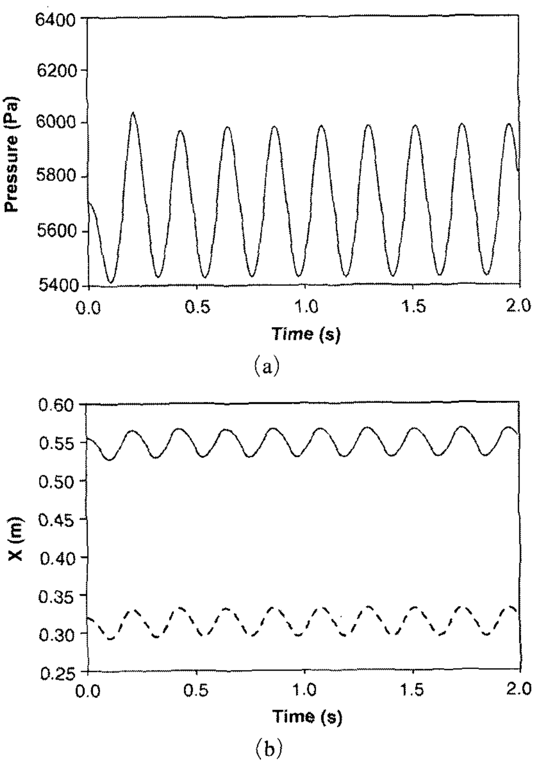


Fig. 5 Variation of pressure and the end positions of the second vapor plug with time

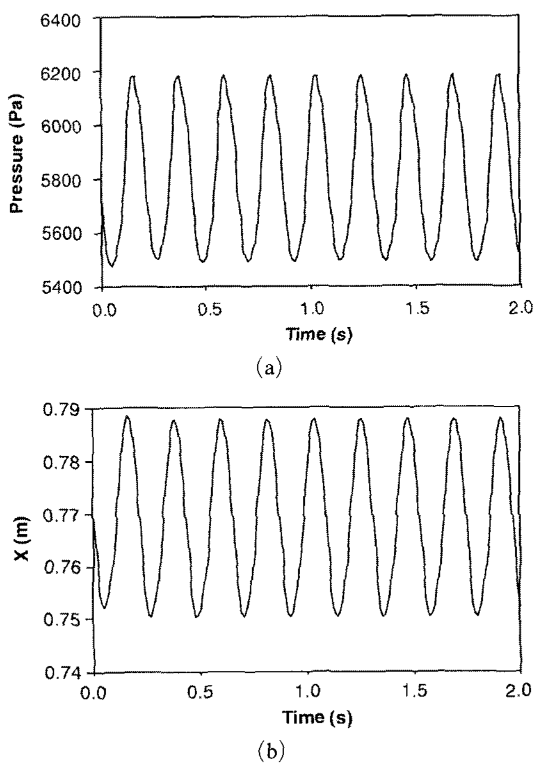


Fig. 6 Variation of pressure and the end positions of the third vapor plug with time

condensation occurs as well as increasing the condensation heat transfer.

Figure 8 shows the rate of the evaporative and condensation heat transfer of the second vapor plug. As mentioned in Fig. 5 when one end of the second vapor plug is in the heating section, the other end would be in the cooling section. Therefore, when one end is moving in the heating section, the evaporative heat transfer increases and the condensation heat transfer also increases due to the other end moving in the cooling section.

Figure 9 reflects the behaviors of the third vapor plug. It can be seen that the third vapor plug behaves the same as those of the first vapor plug with the phase difference of 180°. The velocity variation with respect to time for each individual liquid slug is shown in Fig. 10

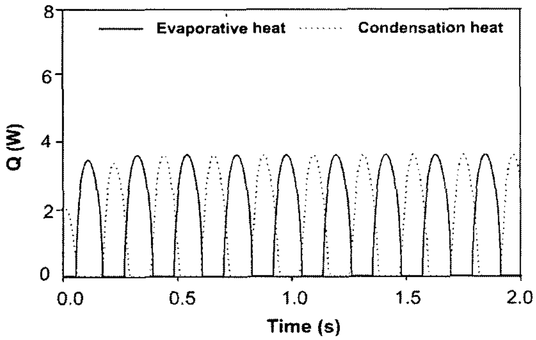


Fig. 7 Variation of the evaporative and condensation heat transfer rate of the first vapor plug with time

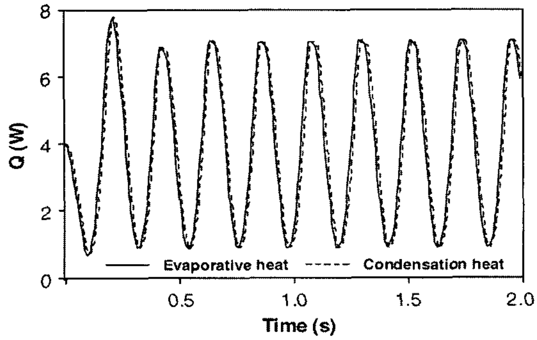


Fig. 8 Variation of the evaporative and condensation heat transfer rate of the second vapor plug with time

To investigate the effect of diameter on the performance of the PHP, the diameter of the tube is increased to 3 mm. The other parameters and initial values are not changed.

Figure 11 (a) shows the pressure variation with respect to time for the first vapor plug. It can be

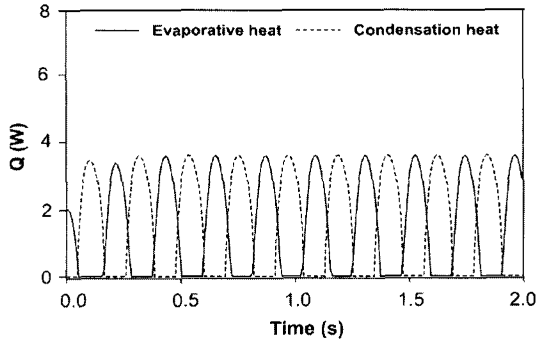
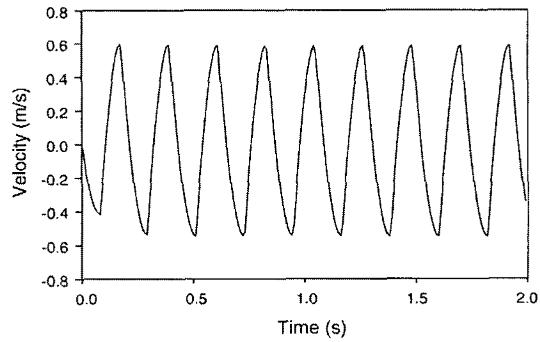
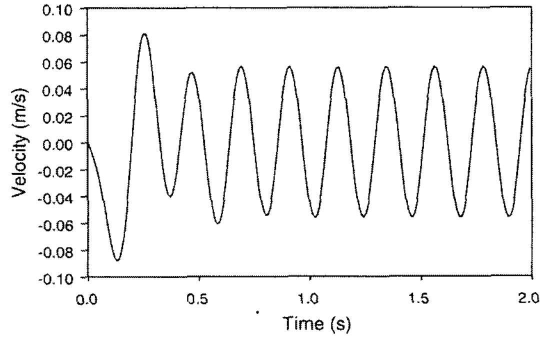


Fig. 9 Variation of the evaporative and condensation heat transfer rate of the third vapor plug with time



(a) First vapor plugs



(b) Second vapor plugs

Fig. 10 Variation of the velocity of each liquid slug with time observed that the frequency of oscillation is lower than that of the small diameter tube. From the Figs. 11(b) and 11(c) one can conclude that the average temperature of the large diameter tube is lower than that of the small diameter tube. Since the heating area and the temperature difference (between the heating wall section and vapor plugs) of the large diameter tube are greater than that of the small diameter tube, the evaporative heat transfer is higher for the first vapor plug as shown in Fig. 12(a). These results are similar for the second and third vapor plugs (Figs. 12(b) and 12(c)).

The effect of charging ratio on the performance of the PHP is shown in Fig. 13. When the charging ratio is high in the PHP, the length of the liquid slugs are long, and a higher-pressure difference is needed to move more massive liquid slugs. Also the amplitude of oscillation increased and the oscillation frequency decreased.

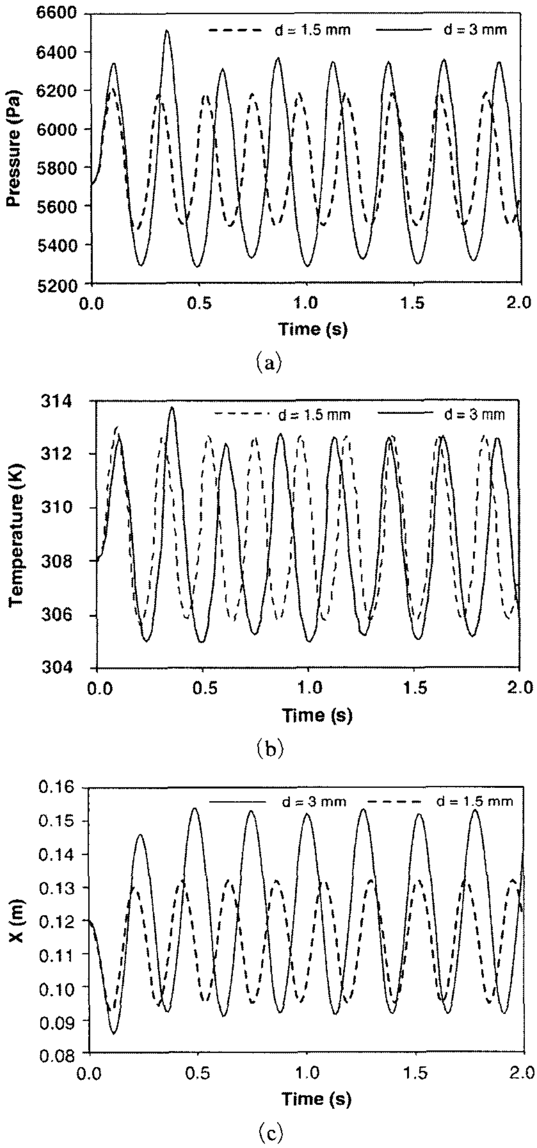


Fig. 11 Effect of diameter on the performance of the first vapor plug

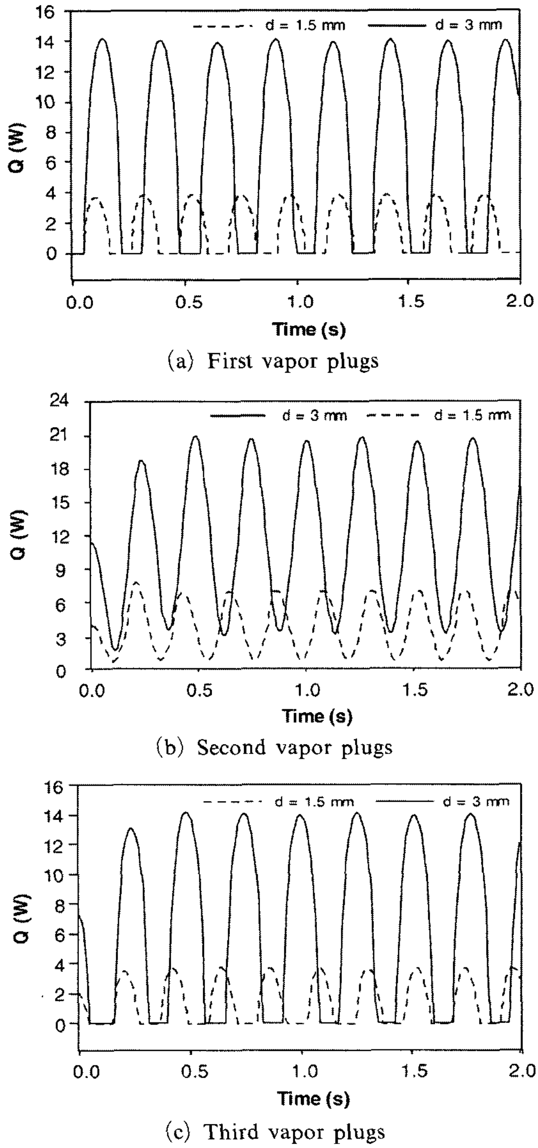
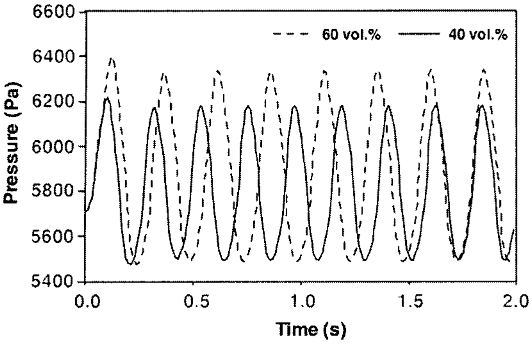
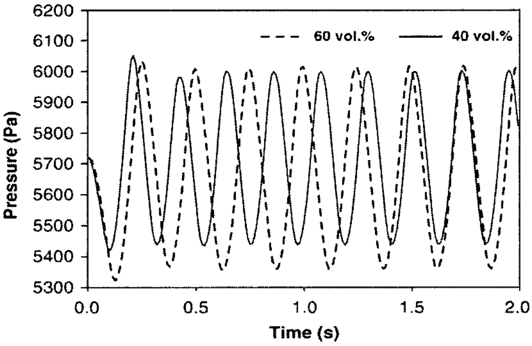


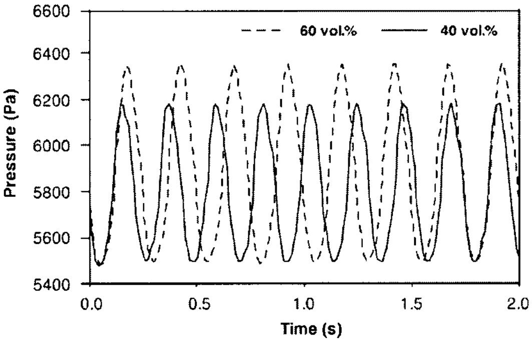
Fig. 12 Effect of diameter on the evaporative heat transfer rate



(a) First vapor plug

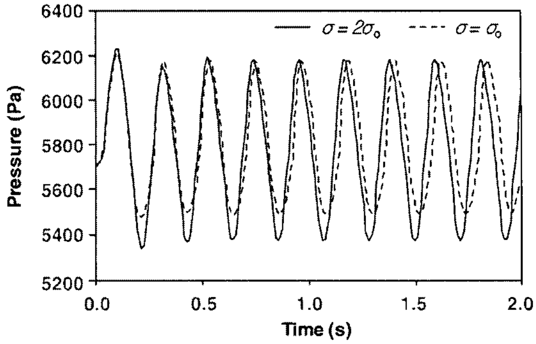


(b) Second vapor plug

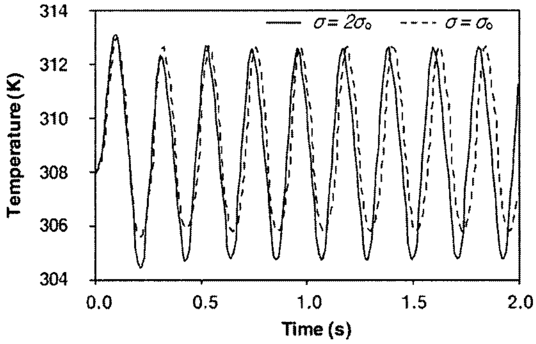


(c) Third vapor plug

Fig. 13 Variation of pressure of vapor plugs at different charging ratios



(a)



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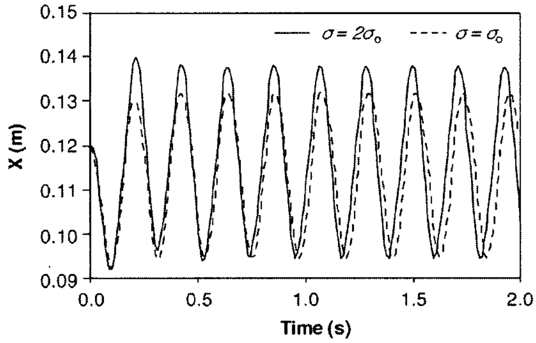

(c)

Fig. 14 Effect of surface tension on the performance of the first vapor plug

The effect of surface tension on the performance of the PHP is shown in Fig. 14. The amplitude and frequency of oscillation are increased for the higher surface tension case. Since the first vapor plug moves farther into the cooling section (Fig. 14(c)), the minimum temperature of the first vapor plug is lower (Fig. 14(b)). This may be explained that the working fluid with higher surface tension results in a thinner liquid film. These results are similar for the second and third vapor plug.

6. Conclusions

Analytical model of the PHP with two liquid slugs and three vapor plugs was presented. The behaviors of liquid slugs and vapor plugs in the PHP were investigated. The following conclusions were obtained.

(1) The periodic oscillations of liquid slugs

and vapor plugs were obtained under specified parameters.

(2) When the hydraulic diameter of the PHP was increased to $d=3$ mm, the frequency of oscillation decreased.

(3) By increasing the charging ratio from 40 vol.% to 60 vol.%, the pressure difference between the evaporating section and condensing section increased, the amplitude of oscillation reduced, and the oscillation frequency decreased.

(4) The working fluid with higher surface tension resulted in an increase in the amplitude and frequency of oscillation. Also the average temperature of vapor plugs decreased.

Acknowledgments

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