MODELING & SIMULATION OF OPEN LOOP PULSATING HEAT PIPE

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ABSTRACT

Pulsating Heat Pipe is a new concept in the field of heat exchangers that aims at extraction of large amounts of heat from comparatively small heat transfer areas in continuous modes. The device is passive in nature in the sense that it doesn’t involve use of any external power for pumping the working fluid. Moreover it can work in absence of gravity as well as in vacuum. PHP is a bend tube of capillary dimensions with either closed ends (open loop) or the two ends connected to form a closed loop. Vacuum is created inside the tube and working fluid is filled so that it flashes to form vapor plugs and liquid slugs. When this unit is placed between heat source and sink it serves as an efficient medium to transfer heat due to the pulsations present to pump the liquid slugs that serve as means of sensible heat carriers.

This present work is to study the phenomena taking place inside a pulsating heat pipe based on basic conservation laws of mass, momentum and energy for vapor plugs and liquid slugs; and thermodynamic aspects pertaining to it. A mathematical model was developed to simulate the physics behind the working of open loop pulsating heat pipe. The model on pulsating heat pipe is the very first attempt at RSD in India. The model tracks the motion of the plugs and slugs by determining their mass, temperature, pressure and velocity. Effect of gravity and surface tension force were also considered in modeling. The results are in good qualitatively agreement with those of the newly evolving models of the contemporary research in this field and the experimental observations made from actual working model of an open loop pulsating heat pipe. The effect of the various model parameters i.e. different working fluids, pipe diameter, filling ratio on its working were also looked upon to understand the characteristics of PHP.

INTRODUCTION

Pulsating Heat Pipe is a new concept in the field of heat exchangers that aims at extraction of large amounts of heat from comparatively small heat transfer areas in continuous modes. The device is passive in nature in the sense that it doesn’t involve use of any external power for pumping the working fluid. Moreover it can work in absence of gravity as well as in vacuum. PHP is a bend tube of capillary dimensions with either closed ends (open loop) or the two ends connected to form a closed loop. Vacuum is created inside the tube and working fluid is filled in partially, so that it flashes to form two phases, vapor plugs and liquid slugs. When this unit is placed between heat source and sink it serves as an efficient medium to transfer heat due to the pulsations present to pump the liquid slugs that serve as means of sensible heat careers. These structures have the following basic features:

(a) Meandering tube of capillary dimensions with many turns. This tube may be configured in any of the three forms: (i) Closed ends-Open Loop(CEOLPHP/CEPHP) : tube ends are closed and are not connected to each other or, (ii) Open ended-Closed Loop (OECLPHP/CLPHP): tube ends are connected to each other to form an endless closed loop. (iii) Closed loop PHP with a check valve (CLPHPW/CV) that incorporates one or more check valves in the loop.

(b) There is no internal wick as in conventional heat pipes.

(c) At least one heat-receiving (evaporator) and heat dissipating (condenser) zone is present; an adiabatic section may or may not be there in-between.

In this present era when the whole world is concerned about the ever increasing demand for energy, one of the ways to encounter the minas is to find ways for conservation of energy together with more efficient use and management of energy presently available with us. The present energy crisis has opened the floodgates of research in the field of passive and naturally driven systems. In this context many new ways to properly manage energy and to find ways to use the waste forms of it for benefits of mankind are looked upon, and pulsating heat pipe is the one which is rising fast on this horizon. As this is a passive means of heat transfer it can save considerable amounts of energy by the avoidance of extra pumping power for heat extraction. Moreover it can be used in the absence of gravity and hence it can serve as a revolutionary device for heat removal in space devices and similar operations. In the nuclear industry it can act as a containment cooling system during emergencies as LOCA, and also to cool the extensive electronics used along with the panels in the control room and associated machinery. More over as it is a
closed system; it has good reliability in terms of radiation shielding prevention of radioactive fluids exposed to the outside world.

The physical and mathematical models that have already been published in literature may be categorized according to the simplification scheme adopted. These may be summarized as follows:

- comparing a PHP action to an equivalent single/multiple spring-mass damper system [1];
- applying governing equations to a specified PHP control volume [2-6];
- modeling that takes into account the chaotic feature;
- modeling by artificial neural networks [7];
- modeling by semi-empirical correlations based on dimensionless numbers [8]; and
- modeling by the effect of capillary tubes and liquid film phenomena [9].

Some features of the pulsating action of a PHP have been compared to the action of a spring mass-damper system. Zuo and North [1] represented it in the form of a second-order homogeneous differential equation with a time-dependent spring constant. The solution of the differential equation suggests that the spring stiffness coefficient increases with time for the entire range of filling ratios. Thus, the amplitude of oscillations has to decrease. Swanepol et al. [2] applied the governing equations of mass, momentum, and energy to a simplified PHP model consisting of a liquid slug in the middle, two vapor bubbles at both ends, and a liquid film separating the inside tube wall from the bubbles. After comparing the results from this model to the experimental results, it was found that the position of the liquid slug in the experiment reached the maximum value faster than that of the. The tendencies of the movement of liquid slug could be predicted with some degree of accuracy. Sakulchangsatjatai et al. [3] have constructed the mathematical model of a CEPHP with a bottom heat mode at different inclination angles. The solution for all of the basic governing equations of liquid film, liquid slugs, and vapor plugs, in which the effects of surface tension, viscous friction of the working fluid, and perfect gas were included, has been numerically obtained by solving a series of ordinary differential equations by means of the explicit method. It was noted that the maximum heat transfer rate of the CEPHP with bottom heat mode occurred at the highest evaporator temperature (150 °C for this study) and inclination angles of 70–80 degrees from horizontal axis. Shafii et al. [4, 5] introduced a numerical model using governing equations including the continuity, momentum, and energy equations for looped and unlooped pulsating heat pipes. They first investigated the effect of the number of vapor plugs and liquid slugs and found that eventually the number of vapor plugs reduces down to the number of evaporator sections. In their second model [5], the thin film evaporation and condensation models have been incorporated with the model to predict the behavior of vapor plugs and liquid slugs in the PHP. The results show that heat transfer in both looped and unlooped PHPs is due mainly to the exchange of sensible heat, and the contribution of evaporation and condensation heat transfer is less than 5% of the total heat transfer rate.

Arabnejad et al. [6] have investigated the performance of a U-shaped pulsating heat pipe (PHP) using numerical methods. The governing equations are derived analytically from the continuity, momentum, and energy equations and are solved implicitly. In this model, considering the liquid mesh, the rate of convection and boiling heat transfer in the U-shaped PHP, which has not been investigated as of yet, are examined. The results show that by increasing the evaporator temperature, due to the increase in pulse amplitude and frequency, the rate of heat transfer due to convection and boiling in the pipe will increase too. An artificial neural network (ANN) is a processing device, either an algorithm or actual hardware. ANN modeling has been successfully applied to predict the performance of a PHP, and an experimental setup for the validation of the model has also been built [7]. In the comparison between the results from this model and the experimental results, it was found that the thermal resistance increased when the filling ratio increased or the heat input decreased. Heat transfer in the evaporator and condenser sections of an open-ended pulsating heat pipe was modeled by analyzing thin film evaporation and condensation [9]. The simulation results showed that the heat transfer in a PHP is mainly due to the exchange of sensible heat. The amplitude of oscillation decreases with a decrease in diameter or a decrease in heat wall temperature. Although there already exists a large number of mathematical models, most of them are based on some unrealistic model that does not correspond to the full behavior of the working fluid inside a CEPHP, and all of the existing models cannot be applied to CEPHPs with bottom heat mode at several inclination angles. Further corrections and improvements with respect to the effect of surface tension as well as the increase in number of vapor bubbles generated in liquid slugs (nucleate boiling) and phenomena of evaporation inside the liquid film are required.

**WORKING PRINCIPLE**

A PHP is essentially a non-equilibrium heat transfer device. The performance success of the device primarily depends on the continuous maintenance or sustenance of these non-equilibrium conditions within the
system. The liquid and vapor slug transport is because of the pressure pulsations caused in the system. Since these pressure pulsations are fully thermally driven, because of the inherent construction of the device, there is no external mechanical power source required for the fluid transport. Generally there are heating, adiabatic and condensation sections. The heating section receives heat and the condensation section rejects heat to the heat sink or environment. The adiabatic section separates the heating and condensation sections. The surface tension dominates the flow and heat transfer inside the capillary tube. The pulsation mechanism is shown in Fig. 1.

In actual working PHP, there are pressure and temperature gradients between the heating and condensation section. Temperature differences also exist amongst the “U” bends within the evaporator and the condenser due to local non uniform heating and cooling rates which are always expected in the real systems. The net effect of all these temperature gradients within the system is to cause non equilibrium pressure condition which, as stated earlier, is the primary driving force for thermo-fluidic transport. In addition to these, heat received from the heating section leads to uneven pressure distributions among the parallel channels due to the uneven vapor expansion in the parallel channels resulting from the unevenly distributed vapor plugs in the parallel channels. Such pressure differences between the heating and condensation section, and among the parallel channels, result in fluid transport among the parallel channels. Therefore self-sustained thermally driven oscillating flow is maintained in a PHP. Once thermally driven oscillating flow is initiated inside the whole PHP, heat is transferred from the heating section to the condensation section by the oscillating two-phase convection. Boiling takes place in the heating section, while condensation occurs in the condensation section. The working process of the PHP is a dynamic one, quite distinct from that of the conventional micro-heat pipes using wick structures.

THEORETICAL MODEL

The heat transfer in CEPHP can be predicted by solving mass, momentum and energy equations for each liquid slug and vapor plug. The appropriate maximum inner diameter of CEPHP can be calculated by

\[ D \leq 1.84\sqrt{\frac{\sigma}{(g(\rho_l - \rho_v))}} \]  

Assumptions:
1. Evaporative and condensation overall heat transfer coefficients are assumed to be constants.
2. The liquid slugs assumed to be incompressible and the vapor plugs are free to expand and contract also to move.
3. The vapor pressure in the each vapor plug is constant and pressure losses at the bends can be neglected so it can be assumed that the CEPHP are straight.
4. The amount of liquid slugs corresponds to the number of turns. The length of the liquid slug along the centerline of the tube is dependent on the filling ratio.

The geometry of the PHP is depicted in Fig. 2. The volume of the vapor plug and liquid slug may be written as

\[ V_{vi} = L_{vi} \pi d^2 / 4 \quad \text{and} \quad V_{li} = L_{li} \pi d^2 / 4 \] 

The mass of the vapor plug and liquid slug may be written as

\[ M_{vi} = \rho_{vi} L_{vi} \pi d^2 / 4 \quad \text{and} \quad M_{li} = \rho_{li} L_{li} \pi d^2 / 4 \]
The rate of change of mass of the vapor plug and liquid slug can be found from the following equations:

\[
\frac{dM_{vi}}{dt} = m_{in,vi} - m_{out,vi} \quad \text{and} \quad \frac{dM_{li}}{dt} = m_{in,li} - m_{out,li}
\]  

(4)

It is assumed that the rate of change of mass of liquid slug is equal to the average changes in mass of its adjacent vapor plugs.

\[
\frac{dM_{li}}{dt} = \left( \frac{dM_{vi}}{dt} + \frac{dM_{vi(i+1)}}{dt} \right) / 2
\]

(5)

where

\[
m_{in,vi} = U_e \pi d L_e(T_h - T_{li}) / h_{fg} \quad \text{and} \quad m_{out,vi} = U_e \pi d L_e(v_i - T_c) / h_{fg}
\]

(6)

Assuming appropriate values for the evaporative overall heat transfer coefficient, \(U_e\), and the condensation overall heat transfer coefficient, \(U_c\), the momentum equation for the \(i^{th}\) liquid slug is:

\[
\left( \rho v_i \right) C_i \frac{dM_{li}}{dt} = \left( p_{vi} - p_{vi(i+1)} \right) A - \pi d L_e \tau_{li} - mg_{li} g
\]

(7)

Where the friction coefficient \(C_{li}\) depends on Reynolds number. The energy equation for a vapor plug is:

\[
M_v C_v \frac{dT_{vi}}{dt} = \left( m_{in,vi} - m_{out,vi} \right) R T_{vi} - P_v A dX_v dt
\]

(9)

\[
P_v V_v = M_v R(T_{vi})
\]

(10)

Heat transfer in the CEPHP is defined as the total heat transferred from the evaporator sections to the condenser sections. It can be divided into two parts; first the heat transfer is due to the evaporation and condensation of the working fluid at vapor phase. The heat transferred into and out of the vapor plug is:

\[
Q_{in,vi} = m_{in,vi} h_{fg} \quad \text{and} \quad Q_{out,vi} = m_{out,vi} h_{fg}
\]

(11)

second, heat transfer between the tube wall and the liquid slugs is obtained by solving the energy equation for a liquid slug

\[
(1 / \alpha_i) * \frac{dT_i}{dx} = \frac{d^2 T_i}{dx^2} - h_{li} \pi d(T_{li} - T_w) / (k_i A)
\]

(12)

The boundary conditions of the liquid slug temperature are:

\[X = X_{rel} ; T_i = T_{vi} \quad \text{and} \quad X = X_{rel(i+1)} ; T_i = T_{vi(i+1)}
\]

(13)

The heat transfer coefficient \(h_{li}\) can be determined by Nusselt number. For small Reynolds, laminar regime \(Re \leq 2200\), Nusselt number is considered as thermally developing Hagen–Poiseuille flow. In the transition and turbulent regions, the empirical equation used is different. The expressions used for the various regimes are as follows:

For laminar regime \(Re \leq 2200\) : \(Nu = 3.657 + (Re Pr D / L)^{0.8} / (1 + 0.117(Re Pr D / L)^{0.467})\)

For transition regime \(2200 < Re \leq 10000\) : \(Nu = 0.012(Re^{0.87} - 280) Pr^{0.4} (\mu_l / \mu_w)^{0.11} (1 + (D / L)^{0.33})\)

For turbulent regime \(Re > 10000\) : \(Nu = 0.0236 Re^{0.8} Pr^{0.43} (\mu_l / \mu_w)^{0.23}\)

The total heat transferred into and out of the liquid slug between the wall of the tube and the liquid slug is:

\[
Q_{out,li} = \int_{X_{rel}^{(i+1)}}^{X_{rel}} \pi d h_{li} (T_{li,X} - T_w) dX ; T_i > T_w \quad \text{and} \quad Q_{in,li} = \int_{X_{rel}}^{X_{rel(i+1)}} \pi d h_{li} (T_w - T_{li,X}) dX ; T_i \leq T_w
\]

(14)

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The total heat transferred into and out of the liquid slug between the wall of the tube and the liquid slug is:

\[
Q_{out,li} = \int_{X_{rel}^{(i+1)}}^{X_{rel}} \pi d h_{li} (T_{li,X} - T_w) dX ; T_i > T_w \quad \text{and} \quad Q_{in,li} = \int_{X_{rel}}^{X_{rel(i+1)}} \pi d h_{li} (T_w - T_{li,X}) dX ; T_i \leq T_w
\]

(14)

The temperature of the \(i^{th}\) liquid plug at location \(X\), \(T_{li,X}\), is obtained by solving the energy equation. The total heat transferred into and out of the CEPHP can be defined as:

\[
Q_{total,in} = \sum_{i=1}^{N} Q_{in,vi} + \sum_{i=1}^{N-1} Q_{in,li} \quad \text{and} \quad Q_{total,out} = \sum_{i=1}^{N} Q_{out,vi} + \sum_{i=1}^{N-1} Q_{out,li}
\]

(15)
An explicit finite difference scheme was used to solve the governing equation for the vapor and liquid phase. The time step used in all numerical simulations are $10^{-6}$ s because when varying the time step to $10^{-5}$ s the result is less than 0.3% in the location of the liquid slug. The new values at time step $t + \Delta t$ can be determined from the old values at time $t$ by

$$M_{vi}^{new} = M_{vi} + (m_{in,vi} - m_{out,vi})\Delta t$$  \hspace{1cm} (16)

$$T_{vi}^{new} = T_{vi} + ((m_{in,vi} - m_{out,vi})RT_{vi}\Delta t - P_{vi}A\Delta X_{vi})/(M_{vi}^{new}C_v)$$  \hspace{1cm} (17)

$$P_{vi}^{new} = M_{vi}^{new}R(T_{vi}^{new})/V_{vi}^{new}$$  \hspace{1cm} (18)

The new values replaced the old values and the procedure was repeated for the next time step. The locations of the vapor bubble and liquid slug can be known by the location of their ends. For CEOPHP the new position of each vapor plug is defined as

$$M_{li}^{new} = M_{li} + 0.5* ((m_{in,vi} - m_{out,vi}) + (m_{in,v(i+1)} - m_{out,v(i+1)})\Delta t$$  \hspace{1cm} (19)

$$v_{li}^{new} = (M_{li}v_{li} + ((P_{vi} - P_{v(i+1)})A - \pi dL_{li}^2\Delta_v - mg_{li}g)\Delta t) / M_{li}^{new}$$  \hspace{1cm} (20)

The results obtained for base case are shown in Fig. 3-4. These variation of various physical quantities with time show oscillatory behavior with the oscillation frequencies and amplitudes, that are in good agreement with those found by experiments done at IIT Kanpur and also with some of the most accurate models so far around the world in this field. After establishing the accuracy of the mathematical model, a parametric study was also carried out to find out the effects of various parameters on the behavior of the model and hence the PHP itself.
The effect of change in diameter

The inner diameter of PHP was increased to 2.5 mm keeping other parameters same. The results obtained for this case are shown in Fig. 5-6. It can be seen that by increasing the inner diameter, heat flux will increase. This is because when a tube inner diameter is large, the boiling phenomena will be like the pool boiling phenomena in which its heat transfer coefficient is higher than that of the boiling inside a confined channel with a small inner diameter.

The effect of change in filling ratio

After the analysis of results of the run for a filling ratio of 55% it has been observed that oscillations take some time to settle before attaining the stability in frequency and amplitude. Moreover the heat transfer rate also increases to some extent due to the rise in sensible heat taken and transferred by the liquid slugs which are larger in this case as compared to the case of 45% filling. The filling ratios of above 65% cannot sustain healthy oscillations and hence the heat transfer is poor, whereas the filling ratios below 40% are less effective because of the lesser contribution of sensible heat in the total heat transferred by the PHP.

The effect of change in working fluid

Although there are many properties of working fluid, the latent heat of vaporization is the main property that strongly affects the motion of liquid slug and vapor bubble in the tube and the heat transfer of the CEPHP. It can be seen that the working fluid can be arranged from the minimum heat flux to the maximum heat flux as water, ethanol and R123. The latent heat decreases, the phase change of vapor plug in the condenser and evaporator sections becomes increasingly active. Thus increased heat can be transferred by the working fluid. The vaporization latent heat has an effect on the mass transfer of vapor plug and the saturated working pressure has an effect on the velocity of liquid slug. The heat transfer rate from the model increases when the vaporization latent heat decreases and the saturated working pressure increases.
CONCLUSION

Thermal modeling of CEPHPs with multiple vapor plugs and liquid slugs was performed, and the behaviors of liquid and vapor plugs in the PHP were investigated. The total number of vapor plugs reduced to the total number of heating sections, no matter how many vapor plugs were initially in the system. Periodic oscillation was obtained under specified parameters. Heat transfer in CEPHPs is due mainly to the exchange of sensible heat. The role of evaporation and condensation on the performance of the CEPHPs is mainly on the oscillation of liquid slugs. By increasing the diameter of the CEPHPs the total average heat transfer increases and CEPHPs did not operate for higher charging ratios.

NOMENCLATURE:

- $A$: tube cross sectional area, m$^2$
- $C_l$: fiction coefficient
- $C_p$: specific heat (constant pressure), J/kg K
- $C_v$: specific heat (constant volume), J/kg K
- $d$: diameter, m
- $g$: acceleration due to gravity, m/s$^2$
- $h$: heat transfer coefficient, W/m$^2$ K
- $h_{fg}$: latent heat of vaporization, J/kg
- $k$: thermal conductivity, W/m K
- $L$: length, m
- $L_{ld}$: length of liquid slug in the heating section, m
- $L_{cv}$: length of vapor plug in the cooling section, m
- $m$: mass flow rate, kg/s
- $N$: number of turns
Transactions, SMiRT 21, 6-11 November, 2011, New Delhi, India

Nu Nusselt number
P pressure, N/m²
Pr Prandtl number
Q heat transfer rate, W
R specific gas constant, J/kg K
Re Reynolds number
time, s
T temperature, K
U overall heat transfer coefficient, W/m² K
V volume, m³
v velocity, m/s
X distance

Greek Letters
α thermal diffusivity of liquid, m²/s
µ dynamic viscosity, kg/m-s
σ surface tension, N/m
ρ density kg/m³
τ shear stress, N/m²

Subscripts
X distance
i inlet
li i th liquid plug
le left end
out outlet
re right end
vi i th vapor plug
l liquid
v vapour
w wall

REFERENCES