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STATE OF THE ART ON PULSATING HEAT PIPES

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ABSTRACT

In recent years, thermal management of microelectronics is becoming a major feasibility bottleneck and has shown the limitations and shortcomings of conventional solutions. Market expectations are posing a simultaneous challenge of increased power levels coupled with high heat fluxes. Active and passive systems incorporating mini-/micro channel flows are gaining ground to meet the challenges. Both single phase and two-phase flows are under consideration with the latter proving to be better alternatives. Inline with these developments are the Pulsating Heat Pipes (PHPs), which are very attractive entrants in the family of closed passive two-phase heat transfer systems. Research activity in this area has steadily increased after their introduction. These apparently simple looking cooling devices have offered considerable challenges in phenomenological and theoretical understanding. These devices have already shown very high promise for terrestrial applications. They also have a potential for thermal control applications for space. Yet, complete design rules and optimization procedures are still not available.

This paper highlights major progress and milestones achieved in the development of this promising technology of pulsating heat pipes in the last decade. A comprehensive review of design rules and modeling strategies available so far is presented. All the influence parameters affecting the thermal performance are explained in detail. Some recommendations for future research are also made.

KEY WORDS

Electronics thermal management, pulsating heat pipes, two-phase oscillating flow

NOMENCLATURE

C_p :specific heat at constant pressure (J/kg·K)
D : characteristic duct dimension, diameter (m)
f : friction factor
g : acceleration due to gravity (m/s^2)
h_{fg} : latent heat (J/kg)
k : thermal conductivity (W/m·K)
L : length (m)
N : number of turns
$P, \Delta P$: pressure, pressure difference (N/m^2)
$\dot{Q}$: heat throughput (W)
$\dot{q}$: heat flux (W/m^2)
$T, \Delta T$: temperature, temperature difference (°C or K)

Greek letters

β : inclination angle with the horizontal axis (rad)
ρ : density (kg/m^3)
σ : surface tension (N/m)
μ : dynamic viscosity (N·s/m^2)
υ^* : characteristic velocity (m/s)

Subscripts

a : adiabatic section
c : condenser
crit : critical
e : evaporator
eff : effective
i : inside, internal
liq : liquid
sat : saturation state
vap : vapor

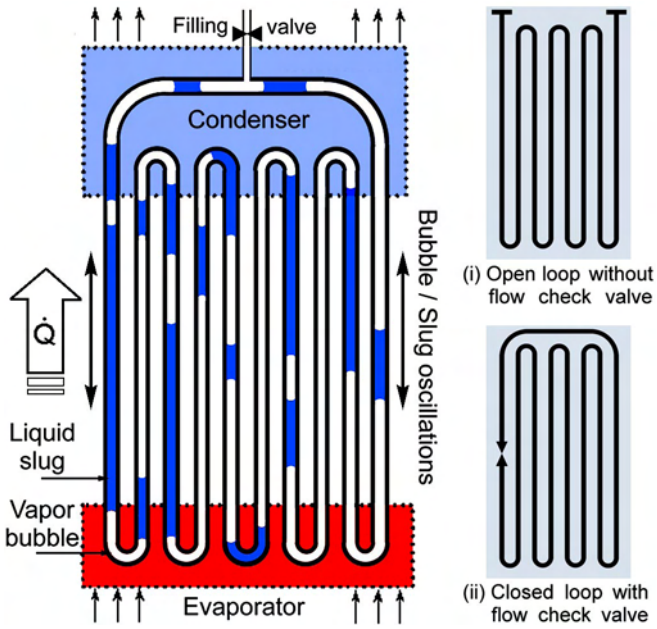

Figure 1: Schematic of a pulsating heat pipe

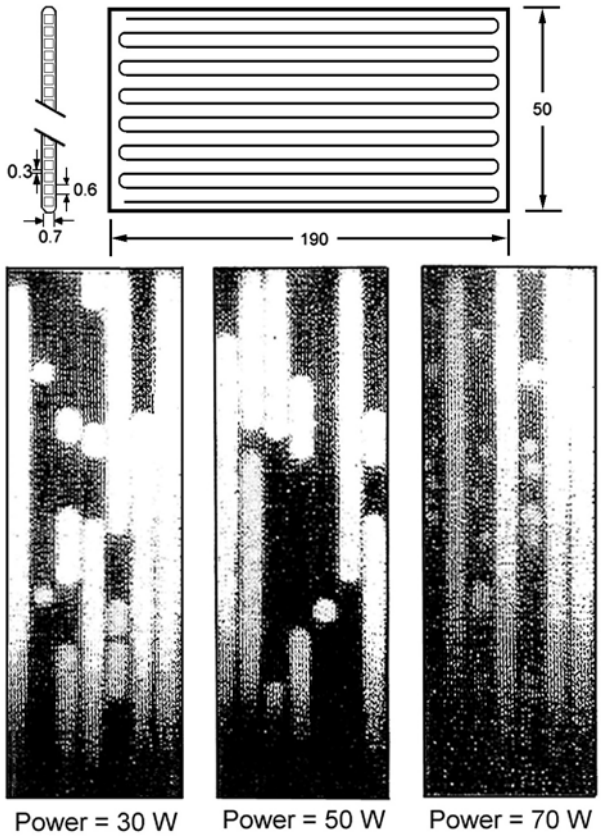

Figure 3: Setup details and results by Kawara et al. [12]

INTRODUCTION

Thermal management of electronics is the elixir to transform dreams and imaginations of the designers into reality. In this light, there is a need for research in the development of novel cooling strategies for emerging needs. Pulsating heat pipes (PHPs) are extremely suitable candidates for this very cause. These structures have the following basic features, as shown in Figure 1:

(a) Meandering tube of capillary dimensions with many turns. This tube may form either: (i) Open Loop PHP: tube ends are not connected to each other or, (ii) Closed Loop PHP: tube ends are connected to each other in an endless 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 can also be there in-between.

This serpentine tube is evacuated and then partially filled with a working fluid resulting in natural unsymmetrical liquid-vapour plug-bubble distribution (uneven void fraction) in the tube sections. In operational mode, there exists a temperature gradient between the heated and cooled end. Due to local non-uniform heat transfer rates, always expected in real systems, temperature differences also exist amongst the 'U' bends of the evaporator and condenser. The net effect of all these internal temperature gradients is to cause pressure disturbances leading to thermally driven two-phase flow instabilities which generate the primary driving force for the thermofluidic transport. In the evaporator, the bubble generation process tries to continuously push the fluid elements towards the condenser. Simultaneously, reverse action occurs in the condenser. In this way, a sustained 'non-equilibrium' state exists between the driving thermal potentials and the natural causality that tries to equalize the pressure in the system. In addition, inherent perturbations are always present in real systems due to the presence of an approximately triangular or saw-tooth alternating component of pressure drop superimposed on the average pressure gradient in a capillary slug flow due to the presence of vapour bubbles. Thus, a self-sustained thermally driven oscillating flow is obtained in a PHP. It is to be noted that there occurs no 'classical steady state' in PHP operation as far as the internal hydrodynamics is concerned. Instead, pressure waves and pulsations are generated in each of the individual tube sections, which interact with each other possibly generating secondary and ternary reflections with perturbations.

LITERATURE REVIEW

While PHPs work in the two-phase regime, it is worthwhile to mention briefly about relevant studies in pulsating/oscillating single-phase flows. Experimental and numerical analyses of such flows require more stringent spatio-temporal resolutions; there exist few investigations of oscillatory flow heat/mass transfer. In general, oscillatory flows are grouped into two-categories: pulsating (modulated) and reciprocating (fully reversing). Pulsating flows are always unidirectional and can be decomposed into steady and unsteady components. For reciprocating flows, the flow direction cyclically changes. Hence these flows convect zero net mass flow [1-4].

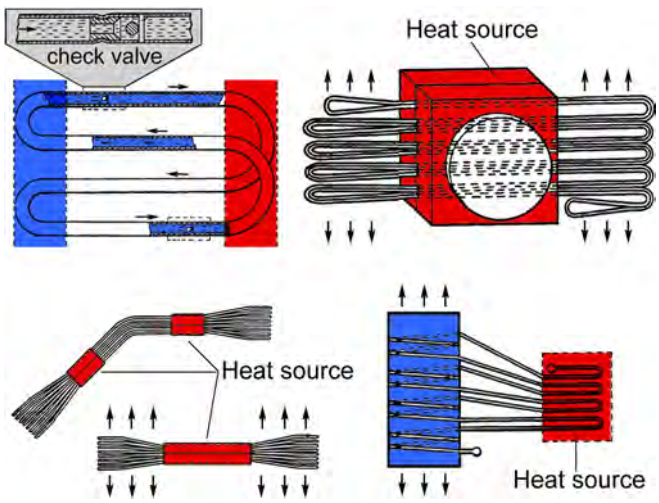

Figure 2: Examples of loop-type heat pipes [7]

Experimental Studies

'Drinking or Dunking Duck', a popular toy, may be the 'Neanderthal', symbolizing a vital link in the evolutionary chain of modern PHPs. This toy is a classic example of a closed passive two-phase system generating 'perpetual mechanical motion' through evaporative heat transfer [5].

The earliest known conceptual description of a 'Pulsating Heat Pipe' is contained in a patent by Smyrnov et al. [6]. Later, the engineering exploitation of this concept of self-induced thermally driven two-phase flow oscillations was done by Akachi [7]. Twenty-four different preferred embodiments of what was referred to as 'Loop Type Heat Pipe' were described, which were claimed to overcome some of the shortcomings of conventional heat pipes. Typical representative values of thermal resistance ranged from 0.082 to 0.233 K/W for water and 0.077 to 0.189 K/W for R11, in the power range of 920W to 310 W, R11 filled heat pipes operating better in all cases. It was also claimed that working fluids, which are unsuitable for conventional heat pipes, might be used in the 'loop-type heat pipes' with comparable or even better performance. All the proposed structures were characterized by the presence of at least one non-return flow check valve integrated in the tubes for imposing a preferred flow direction (refer Figure 2). Typically, tubes of ID 2.0 mm or more were employed, always ensuring that:

$$(\text{Eö})_{crit} = (\text{Bo})^2_{crit} \approx \frac{D^2_{crit} \cdot g \cdot (\rho_{liq} - \rho_{vap})}{\sigma} \approx 4 \qquad (1)$$

$$D_{crit} \approx 2 \cdot \sqrt{\frac{\sigma}{g \cdot (\rho_{liq} - \rho_{vap})}} \qquad (2)$$

This criterion is derived by equating the surface tension and gravity forces of a slug-bubble system in a capillary tube and ensures that a bubble suspended in a stagnant vertical column of liquid will not rise up but instead stick to the wall [8, 9]. In other words, individual liquid plugs and vapor bubbles are then formed in the tube. This criterion comes strictly from an adiabatic point of view and PHPs can still operate when $D>D_{crit}$ (refer later sections).

Long-term reliability issues of the flow check valves and their inability to perform reliably, if further miniaturization of the pipe cross section was done, led to the development of valve-less structures [10], as shown in Figure 1, representing the true PHPs as referred to in this text. Open loop and closed loop PHPs, having ID ≈ 1.0 mm, were proposed. Structures were fabricated with metallic tubes (ID/OD 0.7/1.0 mm) and filled with R142b. Experimental results for a power range of 5 to 90 W in top and bottom heating mode obtained an average thermal resistance ranging from 0.64 to 1.16 K/W.

After these introductory patents, various experimental investigations have been reported on PHPs. Their intriguing operational characteristics have prompted many to undertake visualization and flow characterization. Qualitative results with emphasis on phenomenological understanding are presently more abundant than quantitative data. Various geometrical configurations of PHPs have also been proposed.

Maezawa et al. [11] studied an OLPHP consisting of 20 turns of copper tube (ID 1.0 mm) of total length 24 meters. R-142b was used as the working fluid. Fill charge and inclination were varied and the temperature fluctuations at the adiabatic wall section were also recorded.

Kawara et al. [12] have undertaken a visualization study of an OLPHP employing proton radiography visualization. A 20.0 mm proton beam was passed through the test section and converted to visible light by a fluorescent screen. The PHP was formed of rectangular grooves of size 0.6 x 0.7 mm^2 in a 190 x 50 x 1.3 mm^3 base plate. The set-up details along with the radiographs are shown in Figure 3.

TS-Heatronics Co. Ltd., Japan [13, 14] have developed a range of PHPs including design variations termed as 'Heat Lane' and 'Kenzan' fins (refer Figure 4). Material combinations, e.g. SS-liquid N_2, Al-R142 and copper with water, methanol, R113 and R142b have been tested. Typical thermal resistance of ≈0.3 K/W at a cooling air velocity of 3 m/s was obtained for Kenzan fins (outside dimensions 60.0 x 60.0 x 65.0 mm^3) fabricated from copper tubes (ID/OD 0.7/1.0 mm) filled with R142b, having 152 turns and soldered to a copper heat input pad. Similar Kenzan fins, specifications of which are depicted in Figure 4, have been used for cooling MCMs and IGBTs.

Maezawa et al. [15] have tested another set of OLPHPs with R142b and water as the working fluid with a filling ratio of 50%. The heat pipes, both having 40 turns with a total length of 52.5 meters were made of copper tube of ID 2.0 mm and 1.0 mm respectively. Effect of diameter and the working fluid is shown in Figure 5. It can be seen that the performance for the bottom heat mode was better than horizontal operation mode. In addition, poor performance for top heat mode was observed.

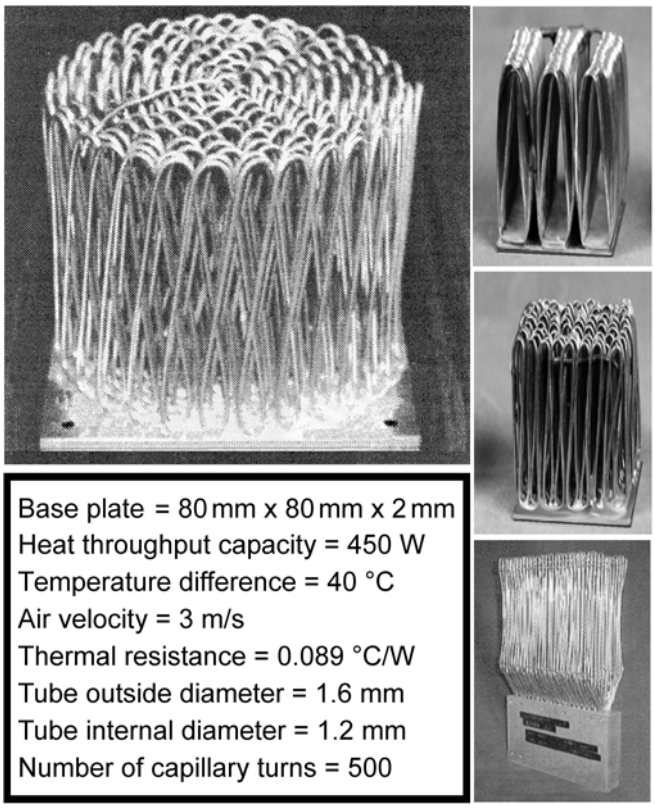

Figure 4: Examples of Kenzan fins [13]

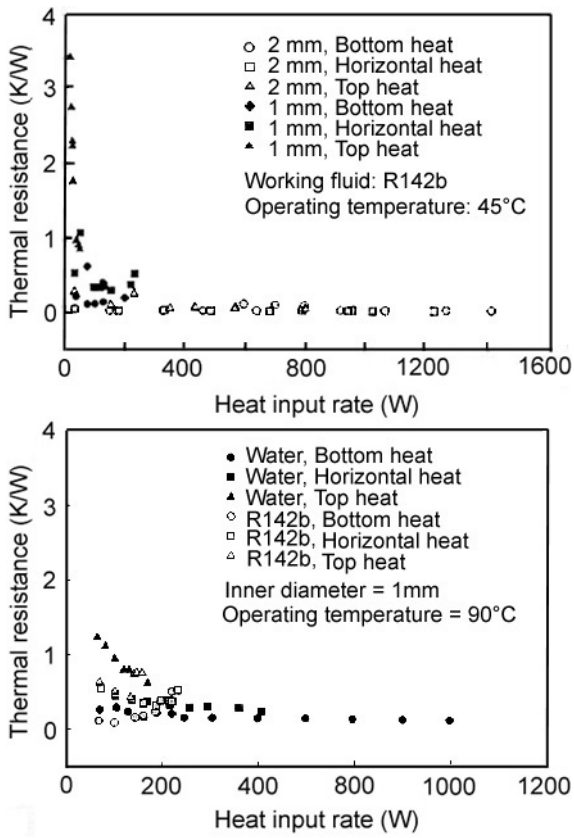

Figure 5: Results by Maezawa et al. [15]

Hosoda et al. [16] fabricated a CLPHP consisting of 10 turns with a glass tube having ID/OD 2.4/4.0 mm. Water was used as the working fluid with a small amount of black ink added for proper visualization. The evaporator and condenser were enclosed in an acrylic box and supplied with hot water (varied from 55-70°C) and cold water (30°C fixed). Results showed that thermal conductivity of the device was much larger than equivalent copper structure. The accuracy of measurement of input power in this experiment was not very high and so the data should be looked at as providing only quantitative trends.

Tong et al. [17] have undertaken a visualization study using a charge coupled device (CCD) on a CLPHP having tubes of ID 1.8 mm, 10 turns and made of pyrex glass. The distance between the evaporator and the condenser was 400 mm. The filling ratio was always 60%. The findings showed that uneven plug and bubble distribution and non-concurrent boiling at the evaporator contributed to the driving/restoring forces for fluid circulation and oscillations. In general, capillary slug flow was observed throughout the operation but bubbles smaller than the tube diameter were also observed (refer Figure 6). The study reveals that a minimum heat flux is needed to initiate sustained oscillations.

Qu and Ma [18] have also reported a detailed experimental study on a CLPHP having 8 turns and made of copper and glass tube (ID/OD 1.8/2.0 mm) alternatively placed to form a semi-visualization set-up. They report the effect of increasing input heat power and angle of inclination on the flow pattern and thermal performance of the device. Capillary oscillating slug flow as well as net-circulation is reported depending on the applied power. Preliminary qualitative results also show that the presence of non-condensable gases also affects the thermal performance.

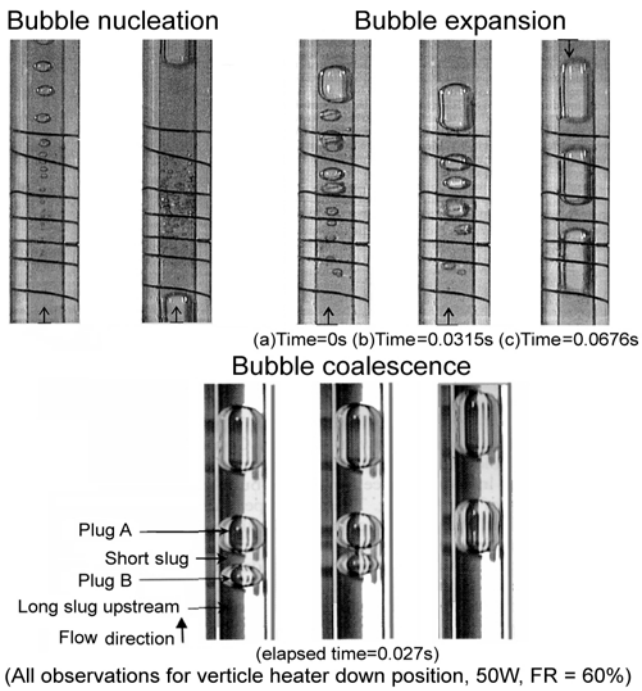

Figure 6: Bubble patterns observed by Tong et al. [17]

In another contemporary study, Charoensawan et al. [19] have reported results of copper CLPHPs with three working fluids (water, ethanol and R123) and two internal diameters (2.0 and 1.0 mm). In all the experiments, the filling ratio was maintained at 50%. In these experiments, instead of controlling the input heat power, the evaporator and condenser were always maintained at fixed temperatures at 80°C and 20°C, respectively. The net quasi-steady state heat transfer was then recorded. In parallel, visualization in glass tube set-ups was also undertaken. The effect of internal diameter, inclination angle (from vertical heater-down to horizontal operation) and number of turns was determined.

Figure 7 summarize the thermal performance for $D_i = 2.0$ with respect to the inclination angle. For a given case, the performance was scaled by the maximum performance achieved for that case during operation in the full range of inclination angles. It can be clearly seen that the performance dependence with orientation is affected by the number of turns. For $D_i = 2.0$ mm devices, the effect could be clearly separated into two cases by using a certain critical value of number of turns (N_{crit}). In this case, N_{crit} was 16 turns (with the exception of L_e = 15 cm, 16 turns and ethanol as working fluid). In case of $D_i = 1.0$ mm devices, too, similar trends were recorded, but N_{crit} was higher than for 2.0 mm tubes. For $N > N_{crit}$, the CLPHP could satisfactorily operate in both the horizontal and vertical orientation. When $N < N_{crit}$, the highest thermal performance normally occurred at vertical bottom heating mode decreasing continuously as the device was turned towards horizontal. However, when $N > N_{crit}$, although the performance improved with increasing the inclination angle from horizontal orientation, it remained nearly constant from about 60° up to vertical position. In the visualization study the existence of various flow patterns vis-à-vis inclination angle and geometry of the device were observed. Figure 8 (top part) shows the phenomenological trend for heat throughput of the experimental data reported by Charoensawan et al. [19] This trend is the result of various flow patterns occurring in different subsections, as depicted in the bottom part of the figure. The images are taken in the evaporator of the visualization set-ups.

The authors of this review, along with co-workers, have also undertaken systematic study of CLPHPs identifying the different design parameters and isolating the various modes of proper operation and operational characteristics [20-24].

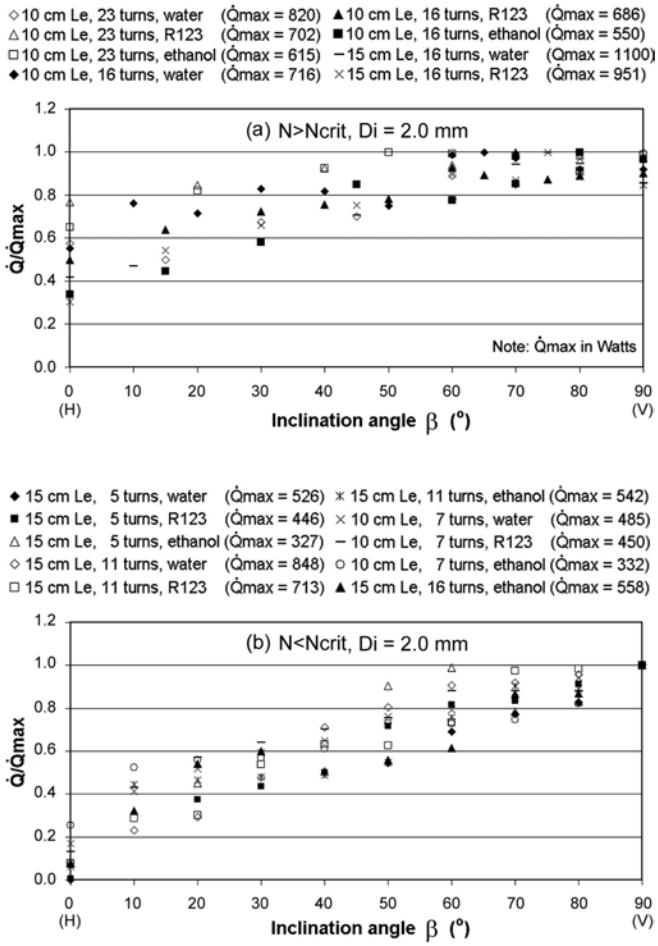

Figure 7: Results by Charoensawan et al. [19]

Mathematical modeling

Mathematical modeling and theoretical analysis of PHPs has been attempted in the recent past with many simplified approaches. The models that appeared so far in the literature may be categorized as follows:

(a) Comparing PHP action to equivalent single spring-mass-damper system [25],
(b) Kinematic analysis by comparison with a multiple spring-mass-damper system [26],
(c) Applying fundamental equations of conservation of mass, momentum and energy to a specified PHP control volume [27-29],

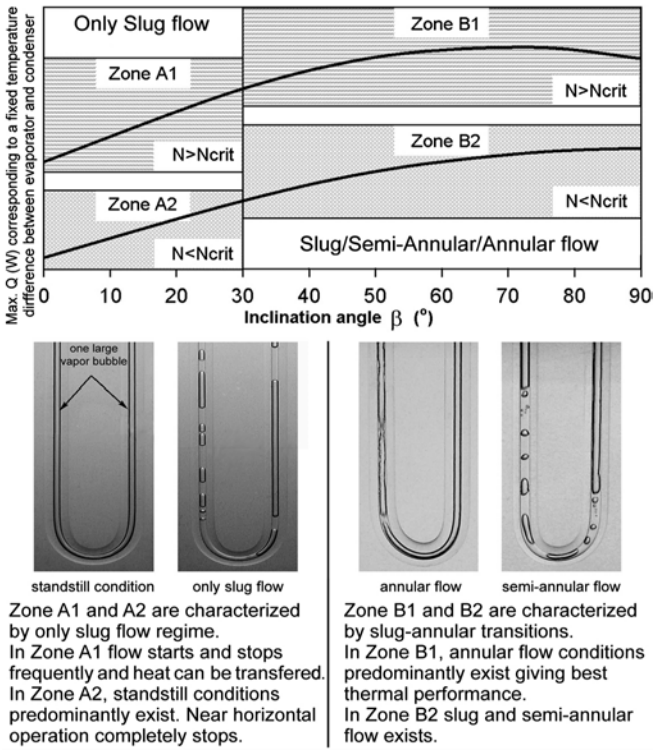

Figure 8: Operational regimes of CLPHPs [19]

(d) Mathematical analysis highlighting the existence of chaos under some operating conditions [30, 31],
(f) Modeling by semi-empirical correlations based on non-dimensional numbers [32] and,
(e) Modeling by artificial neural networks [33].

Zuo and North [25] have tried to model the pulsating action of a PHP by comparing the action to a single spring-mass-damper system represented by a second order homogeneous differential equation with time dependent spring constant. This oversimplified model has very limited applicability especially when compared to the experimental evidences of the flow patterns and visualization studies [34-36]. After the initial introduction of this model, no exhaustive results have yet been presented in the literature.

The modeling approach presented by Wong et al. [26] is without any heat transfer considerations and only predicts the kinematics of the liquid plug-vapor bubble system. In this case, an OLPHP is modeled as shown in Figure 9. The effect of imposed pressure pulses of the system is studied and results of parametric analysis with respect to plug lengths and filling ratios are presented. While the approach can give some insights into the device operation, the oversimplifications cannot be ignored. The model has no practical engineering applicability, i.e. it cannot be used for designing PHPs.

Swanepoel et al. [27] have applied the fundamental governing equations to a simplified OLPHP consisting of a single liquid plug with vapor bubbles on both its sides and a liquid thin film surrounding the bubbles. An experimental setup was also built for validation (refer Figure 10). It was found that the model does not give exact results of the movement of the plug but only predicted the general tendencies of the plug movement. No indication has been given whether the model can be extended to a multiple plug-bubble system.

Shafii et al. [28] and Zhang and Faghri [29] have also attempted to model open and closed loop PHPs with detailed numerical models. These models have a complex structure based on a multitude of assumptions and empirical and semi-empirical correlations governing the evaporation, condensation and liquid/vapor-plug/bubble dynamics. While some useful conclusions may be drawn supporting experimental evidence, there is a significant difference in the physical structure of the models as compared to real-time observations of experimental studies. The models provide no insight into the complex two-phase dynamics, bubble agglomeration, multiple flow patterns and chaotic oscillations that are present during actual operation.

There have been studies that propose the existence of chaos under some operating conditions in PHPs [30-31]. The time-temperature series at a specified location on the wall of PHP tube (adiabatic section) was analyzed with subsequent two-dimensional mapping of the strange attractor. Simultaneously, a theoretical study on a single PHP loop was also undertaken. It was concluded that the flow is governed by chaotic dynamics over a wide range of input heat power. While these studies have certainly added another dimension to the already complex PHP behavior, the results do not help in practical designing of PHPs.

These available models do not truly represent the complete thermo-hydrodynamics of the PHPs. In addition, models applicable for open systems are also not directly applicable for closed PHPs. The fact that PHPs are closed systems in which the velocity scale is dependent on the imposed thermal boundary conditions (and is not known a priori) makes it all the more difficult for analysis. Therefore, preliminary attempts, as described below, have also been made to develop a semi-empirical correlation based on non-dimensional groups [32].

It is obvious that some representative form of flow Reynolds number affects the heat transfer. As noted earlier, it is essential to find the characteristic velocity scale as per

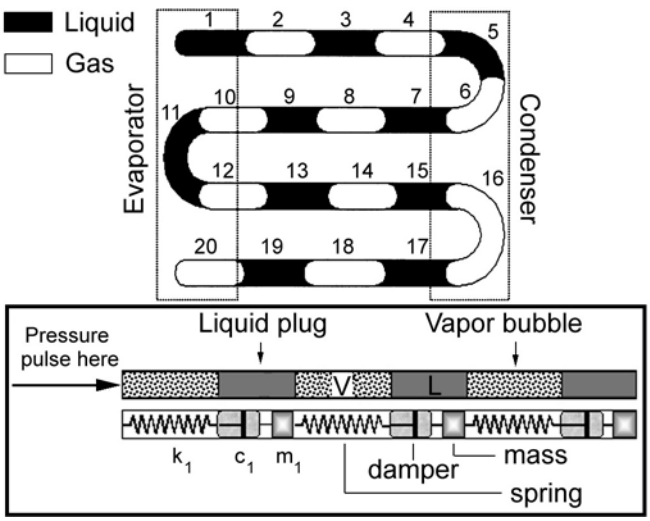

Figure 9: Model details by Wong et al. [26]

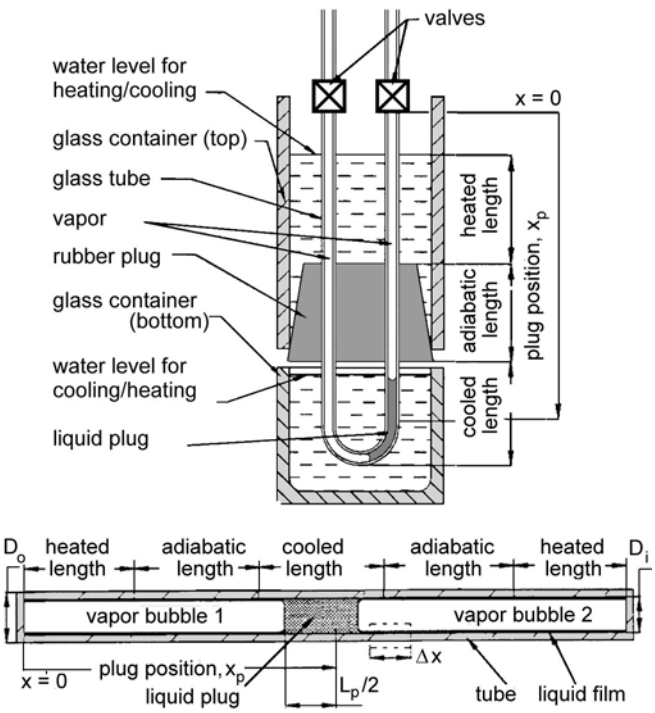

Figure 10: Set-up and model by Swanepoel et al. [27]

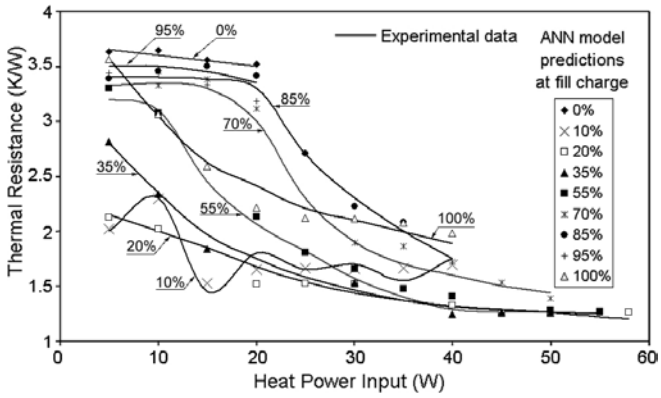

Figure 12: ANN predictions and experimental data [33]

the specified boundary conditions. The characteristic Reynolds number may be calculated as follows:

$$Re_{liq} = \left(\frac{\rho_{liq} \cdot \upsilon^* \cdot D_i}{\mu_{liq}} \right) \tag{3}$$

and since the frictional pressure drop in pipe flows is,

$$(\Delta P)_{liq} = \frac{f \cdot \rho_{liq} \cdot (\upsilon^*)^2}{(D_i / L_{eff})} \tag{4}$$

substituting Eq. (2) into Eq. (1) defines a dimensional group, referred to as Karman number in the context of pipe flows:

$$Ka_{liq} = f \cdot Re_{liq}^2 = \frac{\rho_{liq} \cdot (\Delta P)_{liq} \cdot D_i^3}{\mu_{liq}^2 \cdot L_{eff}} \tag{5}$$

where $L_{eff} = 0.5(L_e + L_c) + L_a$

The fact that the Karman number is calculated for the liquid phase (and not for an equivalent homogeneous two-phase mixture) is based on the assumption that out of the total fluid losses, $(\Delta P)_{liq} >> (\Delta P)_{vap}$. Therefore, in Eq. (5) above, $(\Delta P)_{liq} \approx (\Delta P)_{sat}^{e-c}$. Thus, the Karman number gives an appropriate velocity scale for CLPHP modeling.

The next numbers of interest, which do not need explanation, are the liquid Prandtl number and the Jakob number, defined respectively as,

$$Pr_{liq} = \left(\frac{C_{p,liq} \cdot \mu_{liq}}{k_{liq}} \right) \tag{6}$$

$$Ja = \left(\frac{h_{fg}}{C_{p,liq} \cdot (\Delta T)_{sat}^{e-c}} \right) \tag{7}$$

the thermophysical properties being calculated at $(T_e + T_c)/2$ and ΔT_{sat}^{e-c} is the steady state evaporator and condenser temperature difference. The liquid Prandtl number scales the single-phase convective effects on heat transfer while the Jakob number highlights the relative importance of sensible and latent heat portions.

An attempt was made to correlate the entire data sets (a total of 248 data) of the experimental matrix reported by Khandekar et al. [32] resulting in a correlation given by Eq. (8) below:

$$\dot{q} = \left(\frac{\dot{Q}}{\pi \cdot D \cdot N \cdot (2L_e)} \right)$$

$$= 0.54\,(\exp(\beta))^{0.48}\, Ka^{0.47}\, Pr_{liq}^{0.27}\, Ja^{1.43}\, N^{-0.27} \tag{8}$$

Equation (8) essentially determines the maximum heat transfer achievable for a given CLPHP (with FR = 50%), which is imposed to a specified temperature difference ΔT_{sat}^{e-c}. Alternatively, if heat flux and T_c are known, then an iterative solution by guessing T_e can be employed. This correlation is compared in Figure 11 with the entire set of CLPHP data [19, 32]. If the Bond number exceeds a critical value ($Eö_{crit} \approx 4$), stable liquid plugs will not form and the device will not function as a pulsating heat pipe. For Bo > 2 (for example in standard closed two-phase thermosyphons), the heat transfer limitations come from either nucleate pool boiling mechanism or counter-current flow limitations. Based on the facts it may be said, although without existence of evidence, in favor or otherwise, that Eq. (8) is only valid for situations where $Eö \leq Eö_{crit}$. In the wake of the critical issues regarding modeling of CLPHPs, this semi-empirical approach seems to be quite satisfactory. For a more fundamental modeling approach, an additional reliable data base is needed in congruence with the wider research in mini/micro scale boiling heat transfer in open/closed systems.

In addition, Artificial Neural Network based modeling has also been applied to predict the CLPHP performance with acceptable accuracy, as seen in Figure 12 [33]. Since modeling CLPHP behavior is rather difficult by traditional analysis, ANN based methods also appear to be good tools albeit with certain inherent limitations. To include all the

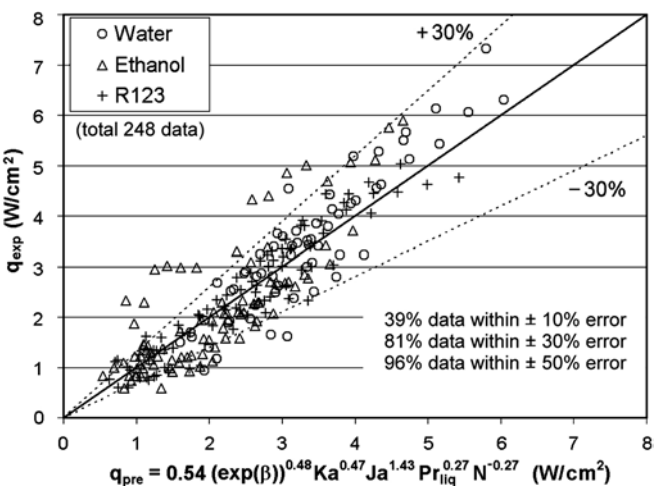

Figure 11: Comparison of Eq. (8) with data [19, 32]

performance parameters affecting the CLPHP performance in an ANN model, reliable and ample data are required. Here, there seems to be an inherent contradiction in the fact that while ANN can effectively model highly complex and non-linear systems, it is increasingly difficult to obtain reliable and abundant data for such complex systems.

PHP PERFORMANCE INFLUENCE PARAMETERS

It will be appreciated that CLPHPs are complex heat transfer systems with a very strong thermo-hydrodynamic coupling governing the thermal performance. The cooling philosophy draws inspiration from conventional heat pipes on one hand and single-phase forced flow liquid cooling on the other. Thus, the net heat transfer is a combination of the sensible heat of the liquid plugs and the latent heat of the vapor bubbles. If the internal flow pattern remains predominantly in the slug flow regime, then it has been demonstrated that latent heat will not play a dominant role in the overall heat transfer [28, 29, 36, 37]. If there is a transition to annular flow (for CLPHPs) under the imposed thermo-mechanical boundary conditions, then the dominance of latent heat increases. The most interesting aspect is the fact that the best performing CLPHP no longer behaves as a pulsating device. Alternating tubes then have slug flow and annular flow while the bulk flow circulates, taking a fixed direction (refer Figures 13 and 14). Strictly speaking, the term 'pulsating' heat pipes then becomes a misnomer [37-39].

The construction of PHPs is such that on a macro level, heat transfer can be compared to an extended surface 'fin' system. Although such an analogy provides an insight into engineering design of PHPs, extrapolations cannot be done authoritatively unless more data is available. The following points may also be concluded for PHPs:

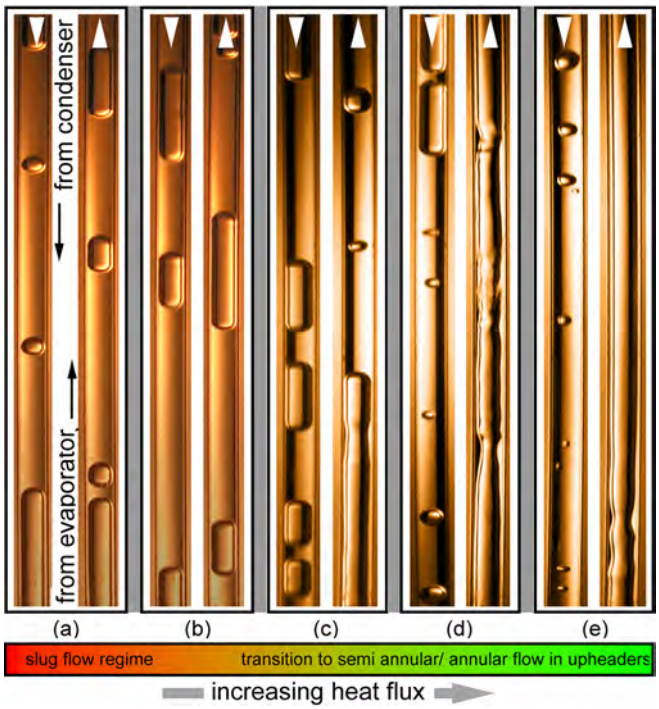

Figure 13: Observed flow patterns in CLPHPs [24]

- PHPs may never be as good as an equivalent heat pipe or thermosyphon system which are based on pure latent heat transfer. If the thermo-hydrodynamics is well understood, the performance may be optimized towards classical heat pipes, as a limiting case. At the least, the manufacturing complexities of heat pipes will be avoided.
- If the thermal performance is below that of an equivalent metallic fin array system (say of copper), at the least there will be a weight advantage.
- If the performance is below that of an equivalent single phase forced convection liquid cooling option, at the least there will be a reliability advantage because of the absence of an external mechanical pump.

Internal tube diameter

The internal tube diameter is one of the parameters which essentially defines a PHP. The physical behavior adheres to the 'pulsating' mode only under a certain range of diameters. The Eötvös/Bond number criterion, i.e. Eö ≈ 4, for surface tension dominated adiabatic slug flow [8, 9, 13] only provides a tentative design rule for a PHP. In some studies, although the Eötvös number was much below the prescribed maximum limit of Eö ≈ 4, gravity forces were definitely seen to affect the performance [32, 37]. Similarly,

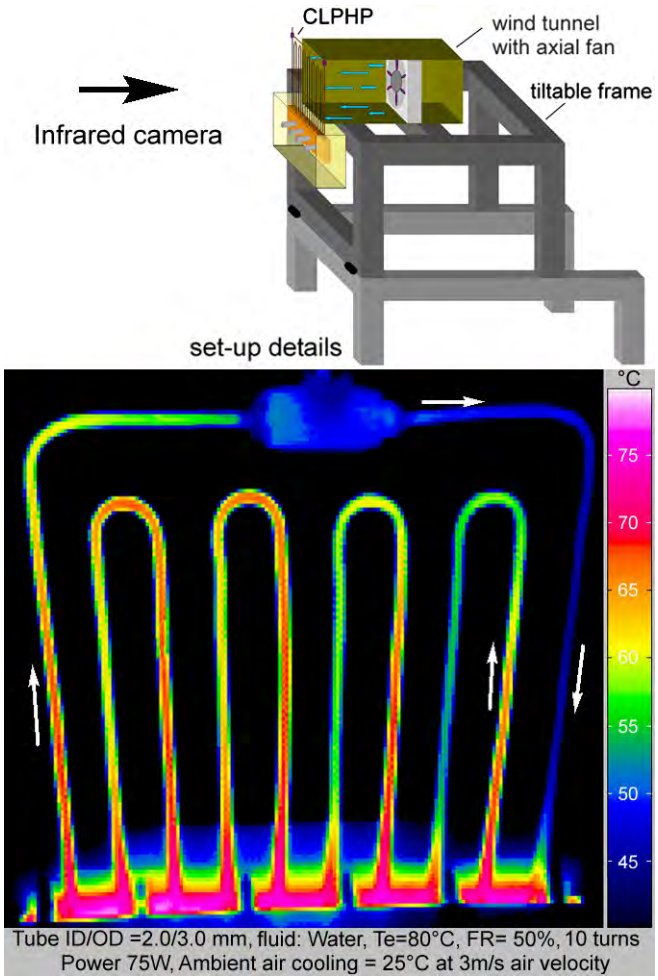

Figure 14: Unidirectional circulation in a CLPHP [39]

for conditions where Eö was greater than the value of 4, the PHP still worked quite effectively. This suggests that though at Eö > 4 the tendency of slug flow diminishes as the surface tension effect tends to reduce, a certain amount of liquid transport is still possible by the bubble pumping action in diabatic flows thereby providing substantial heat transfer. Beyond a certain maximum range of Eö, the device will gradually loose its fundamental character and function as an interconnected array of normal gravity assisted thermosyphons. There might still be some instabilities due to the interconnected tubes. Heat transfer will then be mainly governed by nucleate pool boiling mechanism. If the specified heat input can generate sufficient wall superheat creating favorable conditions for nucleate pool boiling then this interconnected array of thermosyphons may be thermally a better option, at least for a certain range of inclination angles. Below a certain Eö, the dissipative resistance of the flow will lead to an increase in thermal resistance. Thus, rather than a certain fixed diameter which classifies the boundary between classical thermosyphons and CLPHPs, there is a finite transition zone. Also, for a specified heat throughput and maximum allowable evaporator temperature and all other geometry remaining fixed, decreasing the diameter from an optimum value will decrease the performance. In addition, a smaller diameter tube amounts to less liquid inventory in the system and thus less sensible heat transport. It is important to note that the above mentioned optimum diameter is based on the premise that all other parameters (for example, the filling ratio) are optimally specified.

Two-phase instabilities

In addition to the internal tube diameter, two more parameters have emerged as demarcation parameters of these devices, i.e. the input heat flux and the volumetric filling ratio of the working fluid. PHPs loose their fundamental character if certain boundary conditions pertaining to input heat flux and filling ratio are not satisfied. This result is not surprising in the wake of the fact that these two parameters have also been explicitly responsible for affecting various types of two-phase flow instabilities (the effect of filling ratio in PHPs is analogous to the void fraction in open flow systems) [40-42]. A generation of researchers has directed their efforts in understanding these instabilities with the primary motive of reducing their undesirable effects on system performance. PHPs present a strong contrasting case; thermally induced two-phase flow instabilities are effectively harnessed for heat transfer augmentation.

In general, the following parameters/conditions affect the range of two-phase flow instabilities:

- Geometrical parameters: Channel length, size, inlet and exit restrictions, single or multiple channels,
- Operating conditions: pressure, mass velocity, inlet subcooling, heat power input, forced/ natural convection,
- Boundary conditions: axial heat flux distribution, pressure drop across channels.

A preliminary conjectural conclusion, looking at the thermo-hydrodynamic coupling in PHPs, is that ‘complex’ instabilities are inherent to system characteristics. The heat input is the ‘cause’ and the bubble pumping action (for capillary slug flow conditions prevailing in the system) is the primary ‘effect’, the secondary effect being (or ought to be) enhanced heat transfer. While in operation, heating the multiple U-sections at one end and simultaneously cooling the other end produces sustained flow instabilities. This results, as an end effect, in pulsating two-phase fluid flow inside the tube sections causing heat transfer, as a combination of sensible and latent heat portions. The flow instabilities are a superposition of various underlying effects. Static instabilities occur as the bubble pumping characteristics get affected by the pressure drop characteristics of the tube sections (direct analogy with Ledinegg instability is not possible since the ‘pump’ static characteristics are not explicitly known). This may also lead to relaxation instabilities if the conditions are close to transition between slug and annular flows. In addition, fundamental and compound dynamic instabilities, especially density wave oscillations and parallel channel instabilities, are inherent as a direct link between vaporization/ condensation processes and the two-phase flow behavior (these can occur for all types of flow regimes in the system). Certain instabilities are also manifested through the metastable conditions that are always expected in real systems.

Thus, all the parameters affecting the two-phase flow instabilities must affect the thermal performance of PHPs. It is important to remember that there are some parameters whose effect on PHPs may not be as explicitly stated as in the case of single-channel externally controlled flows. Since there is no separate pump in a PHP and the bubbles themselves are pumping liquid (and this action depends on the thermal boundary conditions), one of the major hurdles is the fact that the mass velocity of flow is governed by the input heat flux. In addition, there is no direct control over the fluid sub-cooling entering the respective evaporator U-turns. This is governed by various independent parameters including tube diameter, length, condenser cooling conditions, etc. Furthermore, inlet and exit restrictions are not present or not required in case of CLPHPs. These facts indicate that conventional (in)-stability analysis (e.g. method of small perturbations), as applicable to open single and parallel systems, seems unrealistic for PHPs.

Applied heat flux and number of turns

While in an OLPHP, there is no possibility of overall fluid circulation, this can happen in a CLPHP. This makes capillary slug flow as the dominant flow pattern in OLPHP. The applied heat flux not only provides the driving energy for PHPs, it also catalyzes two vital phenomena, i.e. (i) flow pattern transition in case of CLPHPs and, (ii) affecting two-phase flow instabilities and thereby the level of internal perturbations. A certain minimum heat flux is required to overcome the dissipative frictional fluid flow losses, both for OLPHPs and CLPHPs. Thereafter, in case of a CLPHP, an increase in the input heat flux leads to a series of changes in the internal flow patterns (from slug to churn and fully developed annular flow) which directly affects the heat

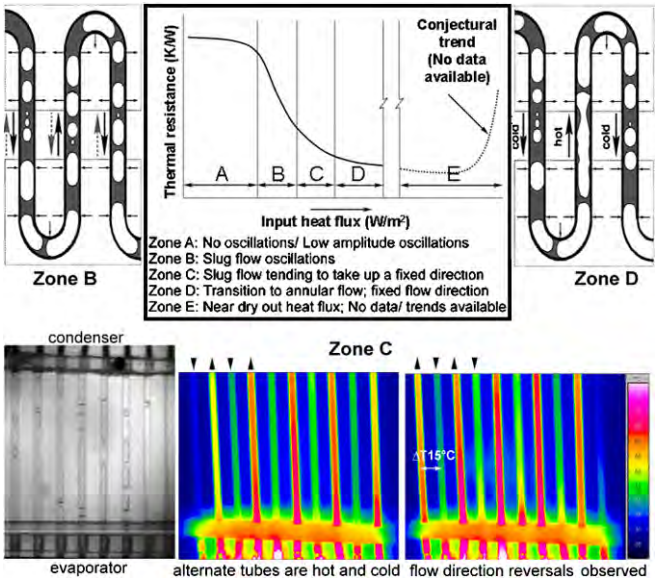

Figure 15: Effect of heat flux on CLPHPs

transfer characteristics (refer Figure 15). Studies [23, 24] indicate that design of CLPHPs should aim at thermo-mechanical boundary conditions which result in convective flow boiling conditions in the evaporator leading to higher local heat transfer coefficients.

A certain critical number of turns is required, in addition to the minimum heat flux requirement, to make the performance of both OLPHPs and CLPHPs nearly independent of the operating orientation. This is attributed to the increase in level of overall internal perturbations.

Volumetric filling ratio

For a given heat throughput requirement, an operationally better performing and self-sustained thermally driven pulsating action was only observed in the filling ratio range of about 20% to 80%, depending on the working fluid. Above this range, the overall degree of freedom and the pumping action of bubbles becomes insufficient for rendering good performance (refer Figure 16). Below a certain range of filling ratio, the evaporator dries out partially. Results also indicate that a 100% filled CLPHP (not working in the pulsating mode, but as a single-phase thermosyphon) may be thermally better performing than a partially filled device under certain operating conditions. As the input heat flux increases, this discrepancy reduces [24].

Working fluid thermophysical properties

Along with device geometry, thermophysical properties of the working fluid, profoundly affect the following:

- The relative share of latent and sensible heat,
- The possibility of having different flow patterns, e.g. slug flow, annular flow and inter-transition of flow-patterns,
- The average flow velocity and overall pressure drop,
- Bubble nucleation, collapse, shapes, agglomeration and breakage; pumping action; etc.

Literature reveals that many working fluids with distinctly varying properties have been tried. In general, the

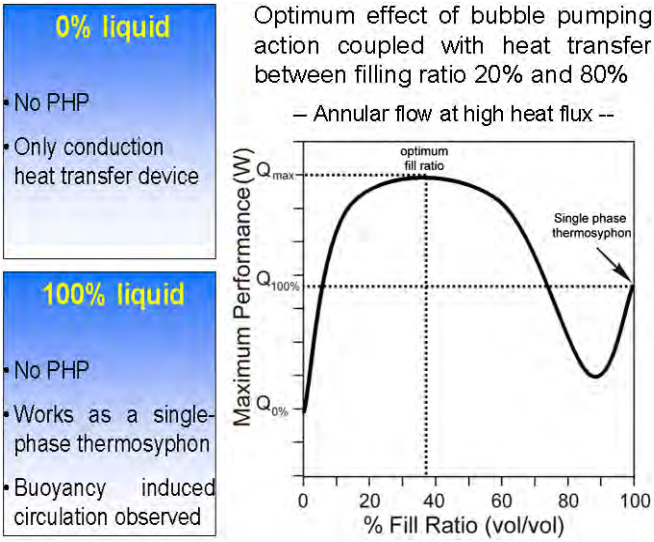

Figure 16: Effect of filling ratio on CLPHPs [37, 38]

following properties of the working fluid are favorable:

- High value of $(dP/dT)_{sat}$ for slug flow governed PHPs, ensuring that a small change in T_e generates a large change in P_{sat}, aiding the bubble pumping action of the device. The same is true in reverse manner in the condenser.
- High specific heat is desirable, given the fact that sensible heat is playing the major role in heat transfer in the slug flow mode of PHP operation.
- High latent heat for annular flow governed CLPHPs.
- Low dynamic viscosity for all types of PHPs.
- Working fluid-tube wall combinations with low dynamic contact angle hysteresis.

Since the domain of experimental activity is quite widespread, all the fluids have not been tested in the entire experimental parameter matrix and the amount of data is still growing. At this stage, it is certainly difficult to prescribe or proscribe a certain fluid unless all the boundary conditions are exactly known and individual effects have been explicitly quantified. Different fluids seem to be beneficial at different operating conditions. An optimum tradeoff of various properties has to be achieved depending on the imposed thermo-mechanical boundary conditions. This certainly requires further research.

CONCLUSIONS AND RECOMMENDATIONS

With the available trends on the performance of pulsating heat pipes, it may be safely concluded that the technology is very well suited for thermal management of high heat flux electronics. Authoritative quantitative data base explicitly connecting the thermal performance with individual influence parameters is although quite limited. With the available database, preliminary conclusions regarding PHP design procedure may be made for terrestrial applications.

Concerning mathematical modeling, extreme simplification has been adopted in all the modeling approaches developed thus far. The results have only limited validity and contribution in the device understanding , not to mention in performance prediction and optimization.

The following general recommendations are made for future research directions:

- The first and foremost aspect requiring immediate attention is the generation of more quantitative data for real time applications. This will throw more light on the suitability of this technology for contemporary and potential applications. While there is a gamut of terrestrial applications of the technology, simultaneous attention is also required for space and avionics applications. There are not enough studies so far which highlight the qualitative and quantitative trends of the pulsating flow in microgravity and varying gravity situations. Since the heat transfer characteristics are closely linked to the flow patterns, which in turn are affected by the body forces, any change in external body force field will have a profound effect on the thermo-hydrodynamics of the system. Such studies will certainly add another parameter in the thermo-mechanical boundary conditions governing the system performance, i.e. the gravity force.
- The PHPs tested so far, as is apparent in the literature, are all in the 'mini' diameter range. It will indeed be interesting to test a PHP in the sub-millimeter or even in the micrometer range. How small can a PHP be and still be thermally a useful device? In this connection, the importance of the fundamental understanding of two-phase heat transfer and fluid mechanics in mini/micro channels cannot be over-emphasized.
- The dryout mechanism of the device certainly remains unexplored and is one of the most vital information needed at this stage for further acceptance of the technology.
- There have been practically no reported studies on life testing of the device, effect of non-condensable gases, mixtures as working fluids, evaporator U-tube internal surface coating with porous material, PHPs as part of a satellite/space application radiator system, effect of vibrations, etc.
- Last, but certainly not the least, a comprehensive scheme for mathematical modeling of these intriguing devices from first principles still remains quite elusive and baffling.

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