Proc. 5th Minsk International Seminar (Heat Pipes, Heat Pumps and Refrigerators), Minsk, Belarus, 2003.

ON THE DEFINITION OF PULSATING HEAT PIPES: AN OVERVIEW

Sameer Khandekar and Manfred Groll
Institut für Kernenergetik und Energiesysteme, Universität Stuttgart
70569 Stuttgart, Germany.
Tel: (+49)-711-685-2142, Fax: (+49)-711-685-2010, E-mail: khandekar@ike.uni-stuttgart.de

Abstract

Pulsating heat pipes (PHPs) have emerged as interesting alternatives to conventional heat transfer technologies. These simple looking devices have intriguing thermo-hydrodynamic operational characteristics. In fact, it is rare to find a combination of such events and mechanisms like bubble nucleation and collapse, bubble agglomeration and pumping action, flow regime changes, pressure/ temperature perturbations, dynamic instabilities, metastable non-equilibrium conditions, flooding or bridging etc., all together contributing towards the thermal performance of a device. Recent literature suggests that important milestones have been achieved in characterization of these devices. Yet, the very definition of PHPs is quite vague. The paper addresses this fundamental issue and attempts to define the device in terms of controllable thermo-mechanical boundary conditions. Such an exercise is deemed necessary to benchmark the operational performance limits and to help in system analysis.

KEYWORDS

pulsating heat pipes, thermo-hydrodynamics, electronics cooling.

NOMENCLATURE

D	tube diameter, m
g	gravitational acceleration, m/s^2
u_∞	terminal velocity, m/s

Greek symbols

μ	dynamic viscosity, Pa·s
ρ	density, kg/m^3
σ	surface tension, N/m

Subscripts

crit	critical
liq	liquid
vap	vapor

INTRODUCTION

Instabilities in two-phase systems has been an area of long time research [1]. The phenomena of thermally induced two-phase flow instabilities have been of interest in many industrial systems, nuclear reactor flow dynamics, steam generators, thermosyphon reboilers and other chemical process units. A generation of researchers have directed their efforts in understanding these instabilities with the primary motive of reducing their undesirable effects on system performance. Pulsating heat pipes present a strong contrasting case in which desirable effects of thermally induced two-phase flow instabilities are harnessed for heat transfer augmentation. Presently such systems have found niche applications in micro-/power electronics cooling and the future prospects seem quite promising.

For various reasons a closed loop pulsating heat pipe (CLPHP) is thermally a better option than an open loop device. These two types of possible designs are shown in Figure 1. Therefore, unless otherwise stated, the discussion in this paper will be focussed on CLPHPs, in which case a simple tube of capillary dimension is bent in a serpentine manner and the ends are joined. It is evacuated and then

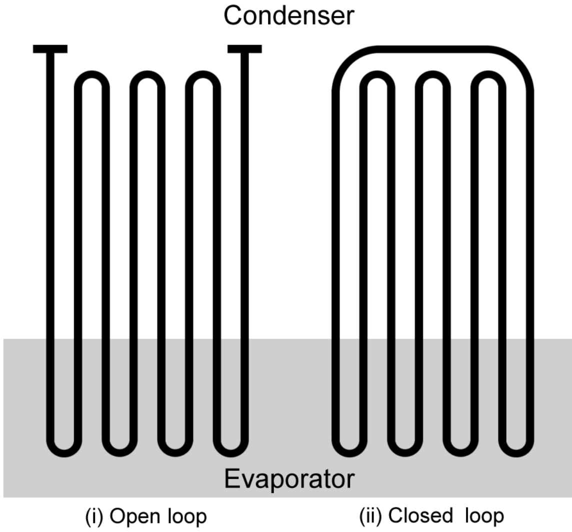

Figure 1: Two configurations of pulsating heat pipe (i) Open loop (ii) Closed loop

filled partially with a suitable working fluid which distributes itself naturally in the form of a liquid-vapor slug-plug system. Heating the U-sections at one end and simultaneously cooling the other end, produces flow instabilities resulting in pulsating fluid flow in the tube. This causes heat transfer, as a combination of sensible and latent heat portions. The flow instabilities are a superposition of various underlying effects. Static instabilities occur because the bubble pumping characteristics get affected by the pressure drop characteristics of the tube sections (direct analogy with Ledinegg instability is not possible since ‘pump’ characteristics are not explicitly known). This may also lead to relaxation type instabilities if the conditions are close to transition between slug and annular flows. In addition, dynamic instabilities, especially the density wave oscillations, are inherent as a direct consequence of the link between vaporization/ condensation processes and the two-phase flow behavior. The range of instabilities is also manifested through the metastable non-equilibrium conditions which are always expected in real systems. The heat input is the ‘cause’ and the bubble pumping action is the primary ‘effect’. The pumping velocity is governed by the input heat flux and is not known a priori. Therefore, system analysis is difficult and cannot be extrapolated from heat transfer experiments and modeling in open micro/mini single and parallel channels.

The emerging literature so far has addressed various issues related to subtle operational thermo-hydrodynamic characteristics of PHPs [2-5]. It is interesting though, that a primary definition of PHPs has not been presented so far. What is a CLPHP? Is it really a ‘heat pipe’? What can we expect, in terms of thermal performance, from such devices? Answering these and similar questions is of fundamental importance for future developmental activities. An attempt has been made in this paper to highlight the critical issues involved towards finding the answers.

COOLING PHILOSOPHY OF PULSATING HEAT PIPES

To achieve the objective of this paper, it is worthwhile to begin with an overall perspective on the principal ideas behind various heat transfer technologies. This will ascertain the relative position of PHPs in the hierarchical structure of modern day heat transfer solutions. This is necessary to benchmark the operational and performance limits and to help in system analysis.

Figure 2 shows a range of heat transfer/ cooling strategies in comparison to CLPHPs. If a given heater block is to be cooled, thereby maintaining it at a fixed temperature, e.g. by convective air-cooling (the external heat transfer coefficient is known and fixed), there are various techniques which may be adopted. The most primitive is using extended surfaces/ fins, thereby employing conduction heat transfer, in which case the thermal conductivity of the fin material limits the performance. The effective thermal conductivity may be greatly enhanced by replacing the solid metallic fins by wicked heat pipes or thermosyphons, thus making use of passive closed two-phase systems based on pure latent heat transfer mechanism. While the heat pipes may be made to operate at any inclination angle, since the capillary wick is the ‘pump’, (albeit with varying performance), traditional gravity assisted thermosyphons only operate in ‘heater down’ position. As far as the external air-cooling is concerned,

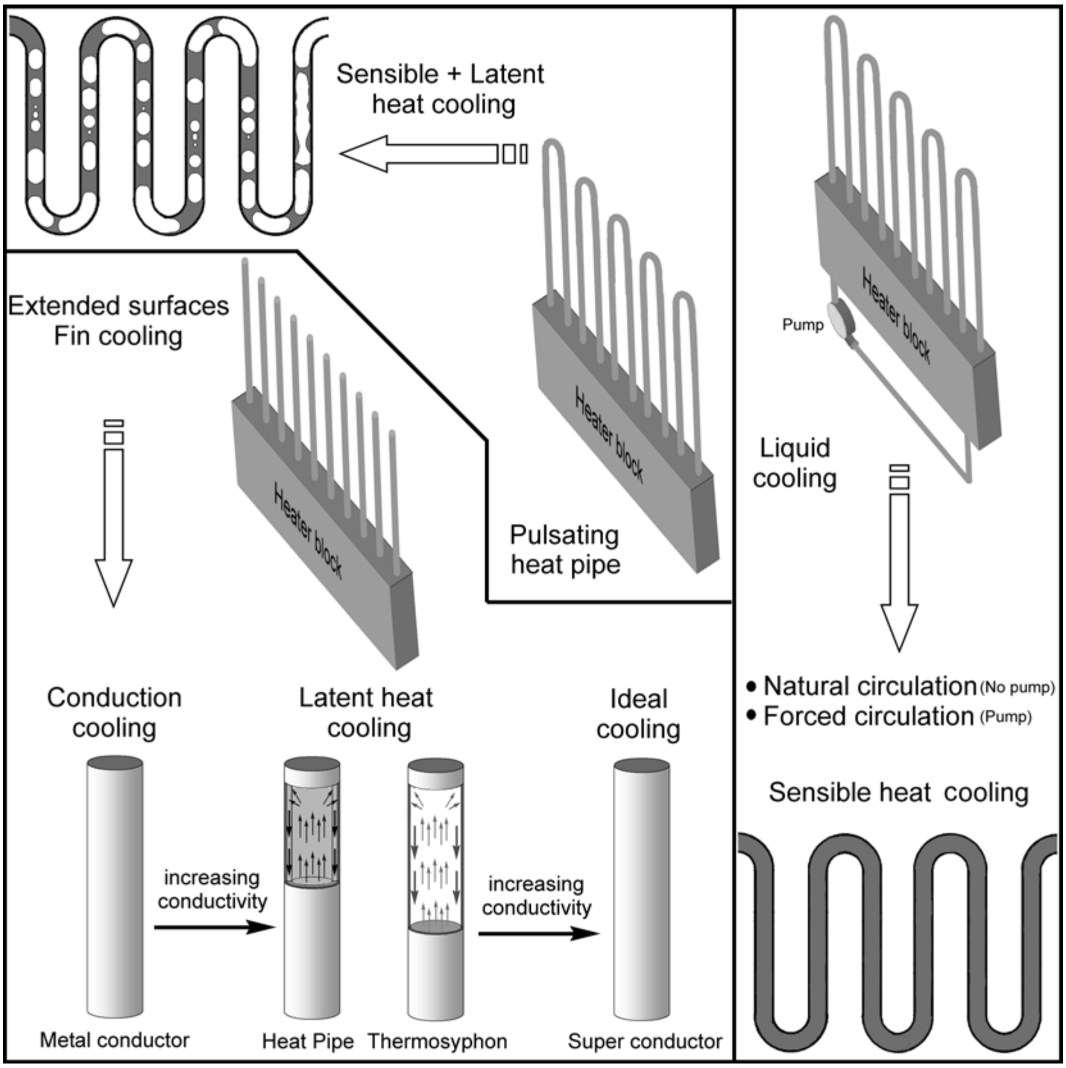

Figure 2: Comparison of various heat transfer technologies

the replacement of solid fins by heat pipe fins changes the longitudinal fin temperature profile which the ambient air (or any other coolant) experiences. In the ideal case of a super conductor, the entire fin will be at the heater base temperature, thereby maximizing the fin efficiency. Maximum heat transfer will be thus achieved as the effective thermal conductivity of the finning structure approaches infinity.

Another strategy to favorably change the longitudinal temperature gradient of the fins is to remove the solid material inside and circulate a suitable fluid. This system necessarily makes use of the sensible heat transfer of the fluid. Single-phase Nusselt analysis informs us that performance may be enhanced by increasing the flow Reynolds number and/ or choosing a fluid with a high Prandtl number. Thus, in stringent demand conditions, quite naturally, forced pump circulation overshadows natural buoyancy driven free convection. In addition, provided the reliability and cost handicap of the pumping system is acceptable, forced circulation cooling is not restricted by system orientation.

In between these two limits, i.e. latent heat cooling and sensible heat cooling, lies the present area of interest. The inspiration for research is to find ways to eliminate the pump as in case of liquid cooling, to drastically reduce the manufacturing complexity involved in heat pipes and try to achieve a structure with thermal performance independent of the operating orientation. The concept of CLPHPs is expected to addresses these very issues. The liquid-vapor system formed in the simple closed tube is capable of generating self-sustained thermally driven oscillations. How much better these structures can be made with respect to solid metallic fins and how close they can go to the conventional heat pipe will depend on sound understanding of the thermo-hydrodynamic behavior and the operational boundary conditions of these devices.

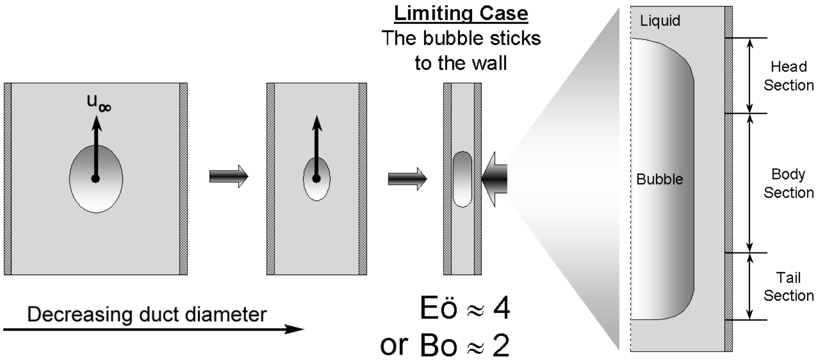

Figure 3: Effect of surface tension on the rise velocity of a cylindrical bubble in stagnant liquid contained in a channel

DIAMETER AS THE DEFINING PARAMETER

The expectations from CLPHP systems have been outlined in the previous section. To fulfill them, the system design must adhere to a series of specifications and boundary conditions in terms of geometry, operational modes and thermophysical properties of the working fluid. We begin the discussion by the most important geometrical dimension of the device, i.e. internal tube diameter, which essentially manifests the fundamental definition of CLPHPs, as explained below.

In metallic fin-cooling, the heat is 'pumped' by electron and phonon mutual interactions. In conventional heat pipes, while the vapor phase is pumped by the small pressure difference between the evaporator and condenser, the liquid phase is sucked back by capillary action. In a thermosyphon, the liquid returns by gravity assistance only. Single-phase natural circulation employs buoyancy driven 'pumping' while forced liquid cooling requires an external pump. As noted earlier, in designing CLPHPs the aim is to remove the external pump. This necessitates that some element of the device itself should act as a pump; the action being materialized by the generating bubbles in the evaporator and the collapsing bubbles in the condenser area. The heat input to the device itself provides part of the energy to run it. Thus, the thermo-hydrodynamic objective function for the design of a PHP is to ensure self-sustained thermally driven bubble pumping, ideally in any operating orientation of the device, which maximizes the heat transfer.

Before proceeding to discuss the actual design criteria of CLPHPs, it is worthwhile to take hints from the classical studies of cylindrical bubbles rising in isothermal static fluids (see Figure 3). A bubble rises through the denser liquid because of its buoyancy. The velocity u_∞ with which a single cylindrical bubble rises through stagnant liquid in a duct is governed by the interaction between buoyancy and the other forces acting on the bubble because of its shape and motion. If the viscosity of the vapor in the bubble is neglected, the only forces besides buoyancy, which are important, are those from liquid inertia, liquid viscosity and surface tension. The balance between buoyancy and these three forces may be expressed in terms of three non-dimensional groups [6]:

$$\text{Froude Number} = \text{Fr} = \frac{\rho_{liq} \cdot u_\infty^2}{Dg(\rho_{liq} - \rho_{vap})} = \frac{u_\infty^2}{Dg} \qquad \text{if } \rho_{liq} >> \rho_{vap} \tag{1}$$

$$\text{Poiseuille Number} = \text{Ps} = \frac{(u_\infty \cdot \mu_{liq})/D}{Dg(\rho_{liq} - \rho_{vap})} = \frac{(u_\infty \cdot \mu_{liq})/D}{Dg\rho_{liq}} \qquad \text{if } \rho_{liq} >> \rho_{vap} \tag{2}$$

$$\frac{1}{\text{Eötvös Number}} = \frac{1}{\text{Eö}} = \frac{(\sigma/D)}{Dg(\rho_{liq} - \rho_{vap})} = \frac{(\sigma/D)}{Dg\rho_{liq}} \qquad \text{if } \rho_{liq} >> \rho_{vap} \tag{3}$$

The Bond number is frequently used in place of the Eötvös number and is defined as,

$$\mathrm{Bo} = \sqrt{\mathrm{E\ddot{o}}} \tag{4}$$

In the above equations, D is typically the characteristic dimension of the duct cross section. For circular ducts, D represents the internal diameter. In situations where viscous forces and surface tension can be neglected, the rise velocity can be correlated only by Eq. 1 above. Similarly, when viscous force constitutes the only predominant factor, the bubble rise velocity is obtained by the Poiseuille number. The last case, when surface tension dominates, is the case of present interest. Interestingly enough, the Eötvös number has no velocity term in it. So, how can this number be used to find the rise velocity under the dominance of surface tension?

Since the general solution is governed by three non-dimensional parameters as defined above, it can be represented as a two-dimensional plot of any two chosen dimensional groups with the remaining third independent group as a parameter. The three parameters may also be combined to generate new dimensionless quantities for convenience. For example, a convenient Property group, not containing either D or u_∞ is frequently used and is defined as,

$$\text{Property number} = \mathrm{Y} = \frac{g \cdot \mu_{liq}^4}{\rho_{liq} \cdot \sigma^3} = \frac{\mathrm{Ps}^4 \cdot \mathrm{E\ddot{o}}^3}{\mathrm{Fr}^2} \tag{5}$$

When the above problem was first attempted analytically, it was thought that the Fr and Eö numbers should tend to zero together [7]. In simpler terms this means that for a given fluid-bubble system, as the tube diameter is reduced, thereby making Eö approach zero, the bubble rise velocity should follow the trend and become zero when Eö = 0. The experimental observations have negated this hypothesis and showed that there is a critical value of Eö below which no rise takes place at all (i.e. $u_\infty = 0$). Figure 4 shows experimental data for a wide range of fluids as reported by White and Beardmore [8]. Main conclusions of present interest are:

- As Eö increases beyond a particular value (≈ 70 for many common fluids e.g. water, ethanol etc.), the terminal bubble velocity approaches a constant value. The viscous forces and surface tension can be neglected and Eq. (1) takes the form $\sqrt{\mathrm{Fr}} \approx 0.345$.
- Below Eö ≈ 70, the terminal velocity continuously decreases.
- Around Eö ≈ 4, the terminal velocity becomes zero. This is the surface tension dominated zone given exclusively by Eq. (3), i.e.:

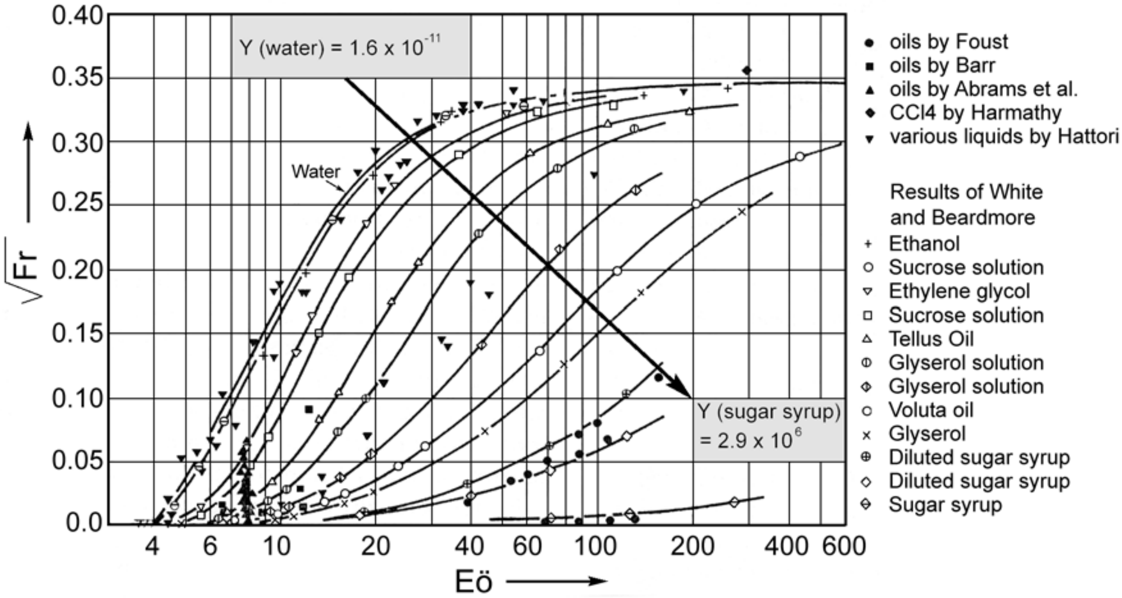

Figure 4: Parametric experimental results for rise velocity of cylindrical bubble in various stagnant liquids contained in a channel [8]

$$(\text{Eö})_{\text{crit}} \approx \frac{D_{\text{crit}}^2 g(\rho_{\text{liq}} - \rho_{\text{vap}})}{\sigma} \approx 4 \qquad \text{or} \qquad D_{\text{crit}} \approx 2 \cdot \sqrt{\frac{\sigma}{g(\rho_{\text{liq}} - \rho_{\text{vap}})}} \tag{6}$$

This value of Eö_{crit} is certainly not unique and varies somewhat under different experimental conditions. It is reasonable to expect the contact angle of the liquid on the tube surface to have an effect on the conditions of zero velocity if wetting of the surface is incomplete. This factor does not appear in the dimensional groups, as outlined above. Factors such as cleanliness or tube surface roughness then may affect the experimental determination of critical Eö.

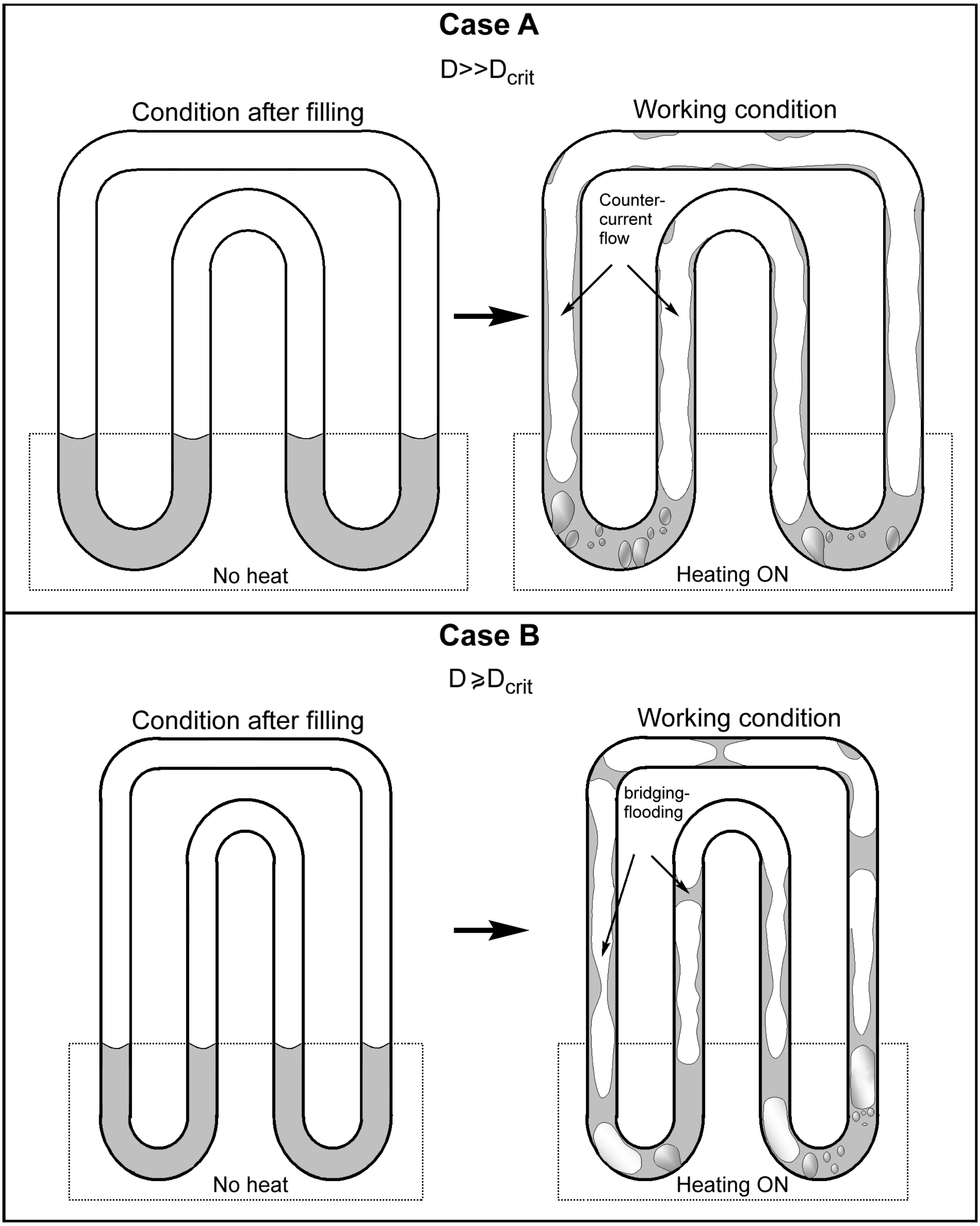

Figure 5: Effect of diameter on the fluid distribution inside circular tubes of closed loop pulsating heat pipes under adiabatic and operating conditions

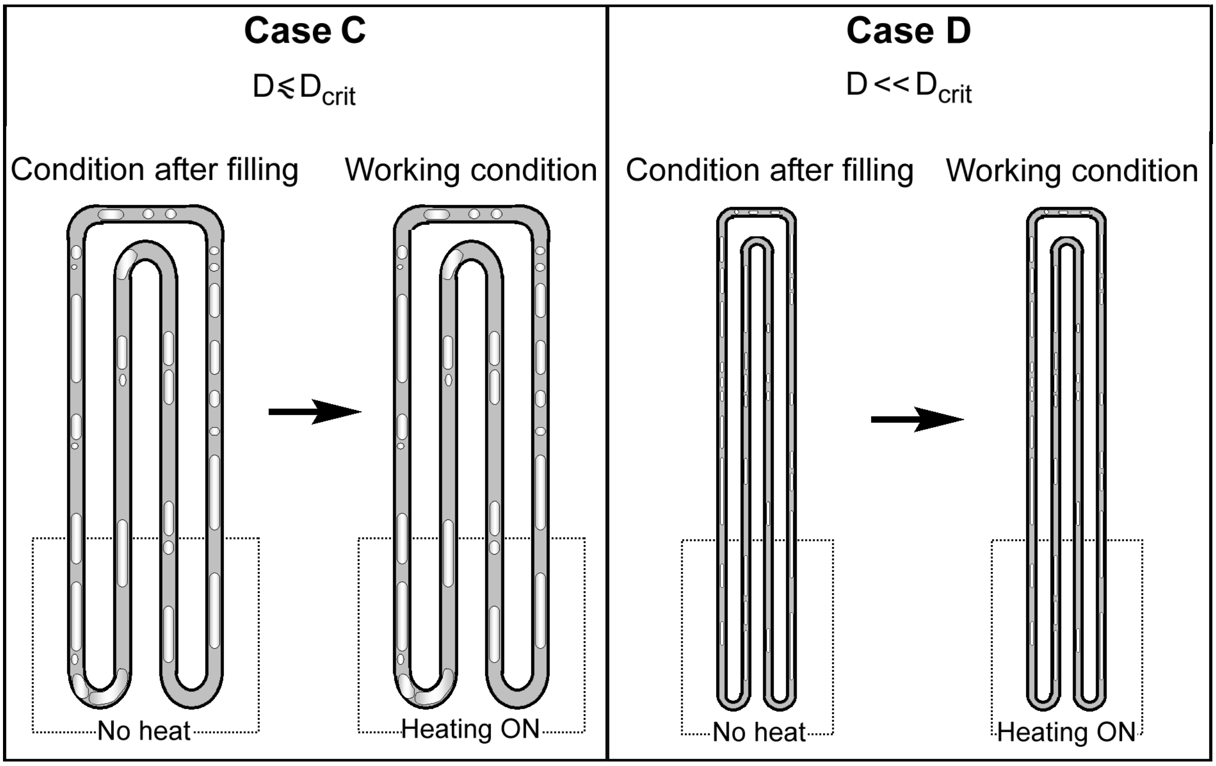

Figure 5: ...continued

The foregoing discussion has important implications in defining a PHP. The $Eö_{crit}$ criterion essentially means that for such systems, distinct liquid plugs and vapor slugs can be formed without separation, stratification or agglomeration under adiabatic conditions. This is schematically shown in Figure 5 which depicts four scenarios for $D >> D_{crit}$, $D \lesssim D_{crit}$, $D \gtrsim D_{crit}$ and $D << D_{crit}$. Following this discussion on critical diameter in adiabatic conditions, focus is now turned to real conditions in which heat is applied to such systems as described in Figure 5, with the aim to make CLPHPs. Which of the cases A, B, C and D will truly function as a CLPHP? One of the requisite design criteria was that bubbles should act as pumping elements. The success of bubble pumping action depends on the formation of distinct liquid-vapor plugs and slugs and so it is obvious that Case C, D will definitely function as CLPHPs. What about the cases A and B?

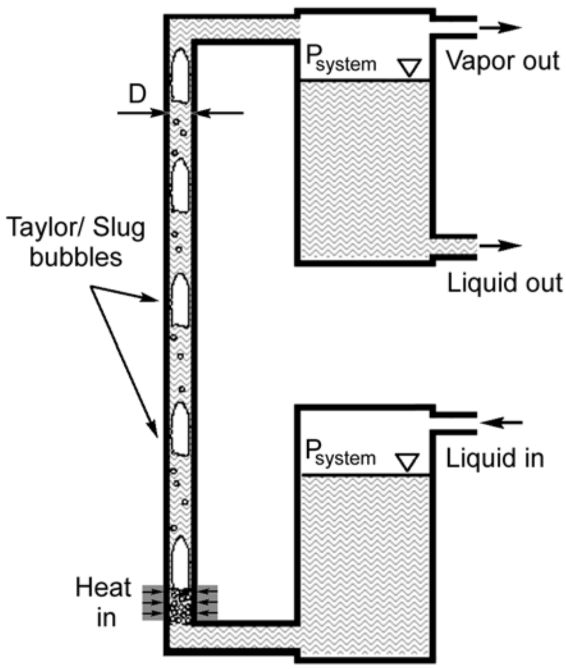

Figure 6: Schematic of a typical bubble pump

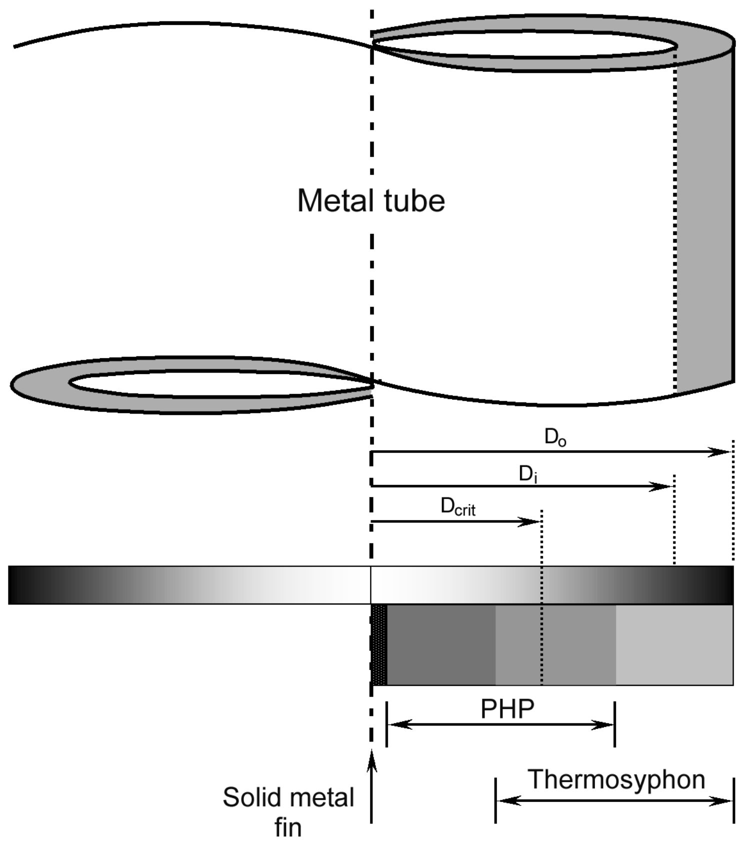

Figure 7: Overlapping zones of operating diameters for CLPHP and thermosyphon

Another case to consider, which answers the above question, is a standard bubble pump shown in Figure 6. Here the primary objective is to lift up/ pump the liquid by application of heat, by its own bubbles. The pumping success is achieved by restricting the flow pattern to the slug flow regime (which lies between bubbly flow and annular flow). The transition from bubbly flow to slug flow is characterized by churn-turbulent region. In this region, if agglomeration of smaller bubbles leads to the formation of stable Taylor-type bubbles, the flow transforms into the slug regime [9]. The larger bubbles, which nearly fill the tube diameter, are then able to transport liquid up the tube. Experiments on vertically oriented bubble pumps suggest that pumping is possible until a critical diameter, as given below, is reached [10],

$$D_{crit} \approx 19 \cdot \sqrt{\frac{\sigma}{g(\rho_{liq} - \rho_{vap})}} \tag{7}$$

It is interesting to compare this criterion, which is for diabatic flow boiling conditions to that given by Eq. (6) for static, adiabatic conditions. Both are correct under the respective boundary conditions they represent. There is indeed a large variation of tube diameter possible for generating slug flow conditions that can transport trapped liquid masses under the action of external heat flux.

With this background we may conclude that rather than a certain fixed diameter which classifies the boundary between classical thermosyphons and CLPHPs, there is a finite transition zone, as explained in Figure 7. If the outside diameter is fixed and the internal diameter is reduced, the thermo-hydrodynamic behavior changes from classical thermosyphon to a CLPHP in a gradual manner. In the CLPHP zone, the input heat produces bubble pumping action. The retarding force to this pumping is the pressure drop inside the channel, which monotonously increases with decreasing diameter. Thus, optimum liquid pumping will be achieved at a certain diameter below which the pressure drop overshadows the pump yield; the CLPHP will tend towards a solid metal tube with further decrease in diameter. Thus, two conclusions can be made regarding the diameter:

- 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.
- In general, maximum heat throughput till some sort of evaporator dryout, will monotonously increase with increasing diameter. After a certain diameter range, the pulsating device will gradually loose its fundamental character. Instead, it will behave as an interconnected array of two-phase thermosyphons; there might still be some instabilities due to the interconnected tubes. Heat transfer will then be mainly governed by nucleate pool boiling characteristics. 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. It is important to note that the above mentioned optimum diameter is based on the premise that all other parameters (e.g. filling ratio) are optimally specified.

The following points may also be concluded for a true CLPHP in which bubble pumping results in liquid transport (i.e. heat transfer is result of sensible and latent portions):

- CLPHPs may never be as good as an equivalent heat pipe or thermosyphon system which are based on pure latent heat transfer. If the thermo-hydrodynamic characteristics are well understood, the performance may be optimized towards classical heat pipes or thermosyphons, as a limiting case. At the least, the manufacturing complexities of conventional 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.

HEAT INPUT AS THE DEFINING PARAMETER

The effect of input heat flux on various forms of two-phase flow instabilities is well documented. For example, experimental as well as analytical studies on density wave oscillations in single channel two-phase flow have indicated that these are strongly dependent on the heat flux variation; other factors being single and two-phase frictional pressure drop characteristics of the channel, inlet flow rate, level of subcooling, system pressure and inlet/ exit restrictions, if any [1, 11]. In such systems, with respect to input heat flux, the results may be summarized by saying that for a specified non-zero level of inlet subcooling, increasing the inlet heat flux above a certain limit induces flow instabilities. In the case of CLPHPs, for a defined geometry of the device, the input heat flux is also directly responsible for the type of flow pattern which will exist in the channel, thus affecting the fundamental relaxation instabilities. Furthermore, the static Ledinegg type instabilities are also affected by input heat flux in case of CLPHPs since this directly affects the bubble pumping characteristics. Thus, we may hypothesize, that the operating heat flux will directly affect the level of perturbations inside a CLPHP thereby affecting the thermal performance of the device.

Experimental studies on CLPHPs, coupled with visualization, have indeed indicated towards this trend. Figure 8 shows a typical phenomenological trend for a partially filled device (about 50%-70%). The figure has been adapted from the data reported in [12] and is representative of a range of working fluids like water, ethanol and R-123. The qualitative zones, as shown, may vary with actual fill charge, geometry and working fluid. Visualization experiments, in parallel, have also supported these trends [4, 13-15]. Low input heat fluxes are not capable of generating enough perturbations and the resulting bubble pumping action is extremely restricted. The bubbles only oscillate with a high frequency and low amplitude. There are periods of ‘no action’ intermission stage followed by some small bulk activity phase. Overall, this scenario results in a poor performance (i.e. very high thermal resistance). As the heat input is increased, slug flow oscillations commence whose amplitudes increase with increasing heat flux and become comparable to the length of the device. This improves the heat transfer coefficient to a marked degree. As the heat flux is further increased, the oscillating flow tends to take a fixed direction. The thermal resistance further reduces. Still higher input heat fluxes result in a transition of slug flow to annular flow at the outlet of the evaporator U-bends. This is true even for cases C and D in Figure 5. The bulk flow takes a fixed direction which does not reverse with time. The alternating tube sections are then hot and cold, with cold bubbly/slug flow coming down from the condenser to the evaporator in one tube and annular/semi-annular flow in the adjacent tube forming

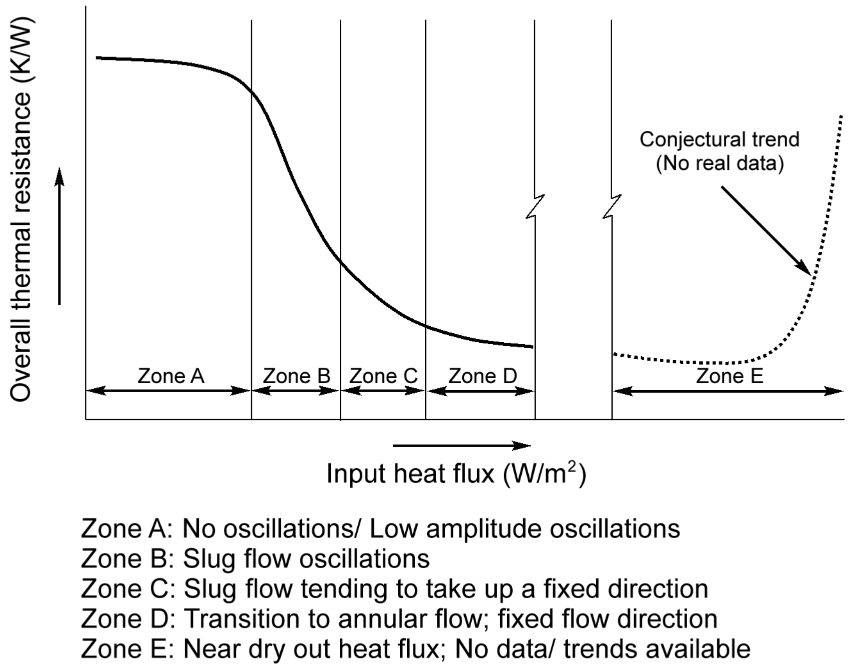

Figure 8: Phenomenological trends for the effect of input heat flux (based on [12])

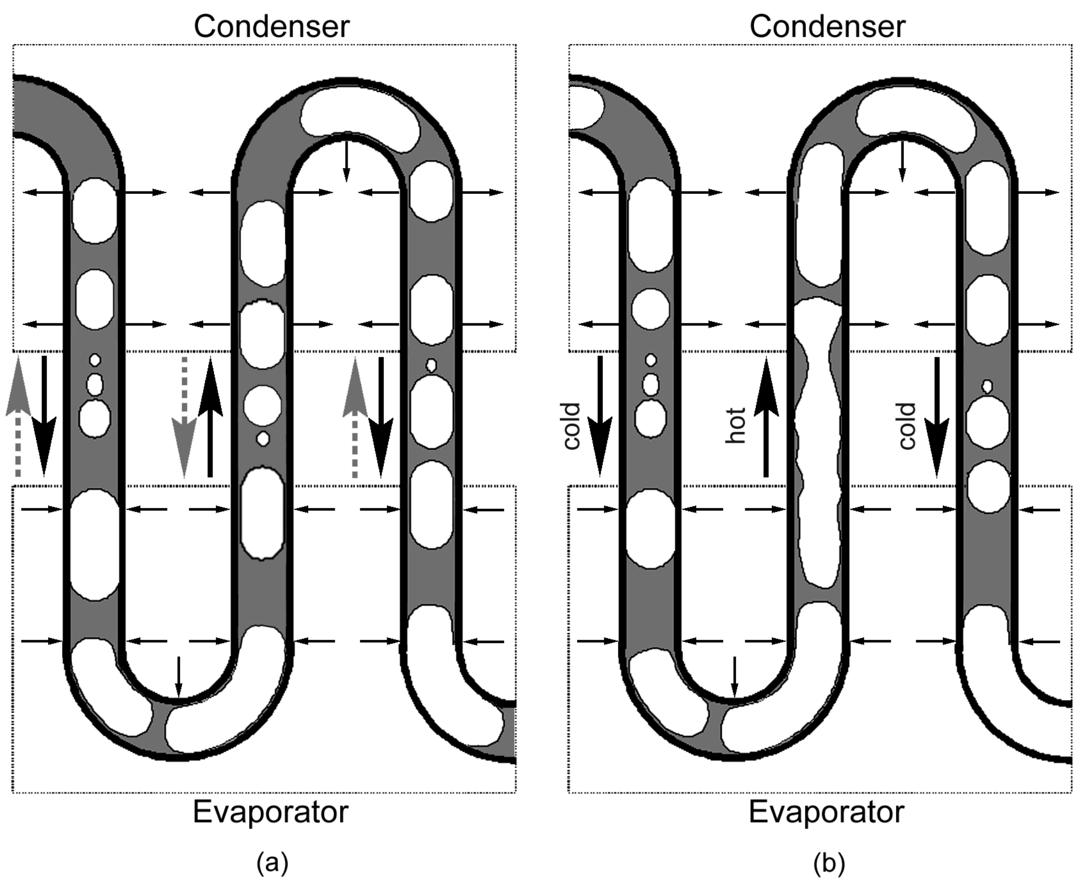

Figure 9: Flow patterns in a CLPHP (a) oscillating slug flow in all tubes (b) alternate tubes with slug and annular flow

the outlet of the evaporator U-tube (see Figure 9). This shows that the pulsating unstable slug flow behavior is again stabilized after a certain higher input heat flux. (In fact, this observation is in line with the classical experiments on two-phase flow instabilities in open systems [11]). Interestingly, in such a case, best performance of CLPHP (lowest thermal resistance) is observed. This is logical since the evaporator U-sections experience convective boiling through the thin liquid film rather than nucleate type boiling in slug flow regime. Thus, ironically enough, the best performing closed loop pulsating heat pipe no more remains a true 'pulsating' device. Further increase in heat flux will lead to some sort of evaporator dry-out phenomenon but quantitative experimental data and phenomenological trends are not available in these near dryout zones. It is indeed worthwhile to concentrate efforts in this direction. Thus we observe that the input heat flux governs the degree of pulsations in the device and essentially acts as a demarcation parameter.

FILLING RATIO AS THE DEFINING PARAMETER

The filling ratio (FR) of a CLPHP is defined as the ratio of working fluid volume actually present in the device to that of the total volume of the device (say at room temperature). Thus, a given CLPHP has two operational extremities with respect to the filling ratio, an empty device without any working fluid i.e. FR = 0 and a fully filled device i.e. FR = 1. It is obvious that at FR = 0, the empty CLPHP tubes constitute inefficient conduction fins and obviously have a very high thermal resistance. A fully filled PHP (FR = 1) is identical in operation to a single-phase thermosyphon. There exist no bubbles in the tube and so no 'pulsating' effect is present. Substantial sensible heat transfer can still take place due to liquid circulation in the tubes by thermally induced buoyancy [13].

In between these two extremities lies the present area of interest. In this region also there exist three distinct sub-regions [12]:

(a) Nearly 100% fill ratio: In this mode there are only very few bubbles present rest being all liquid phase. These bubbles are not sufficient to generate the required perturbations and the overall degree of freedom is very small. The buoyancy induced liquid circulation, which was present in a 100% filled PHP, gets hindered due to additional flow resistance due to a few bubbles. Thus, the device performance is seriously hampered and the thermal resistance is much higher than for FR = 1.

(b) Nearly 0% fill ratio: In this mode there is very little liquid to form enough distinct slugs and there is a tendency towards dry-out of the evaporator. The operational characteristics are unstable. The device may, under some operating conditions, work as a two-phase thermosyphon array.

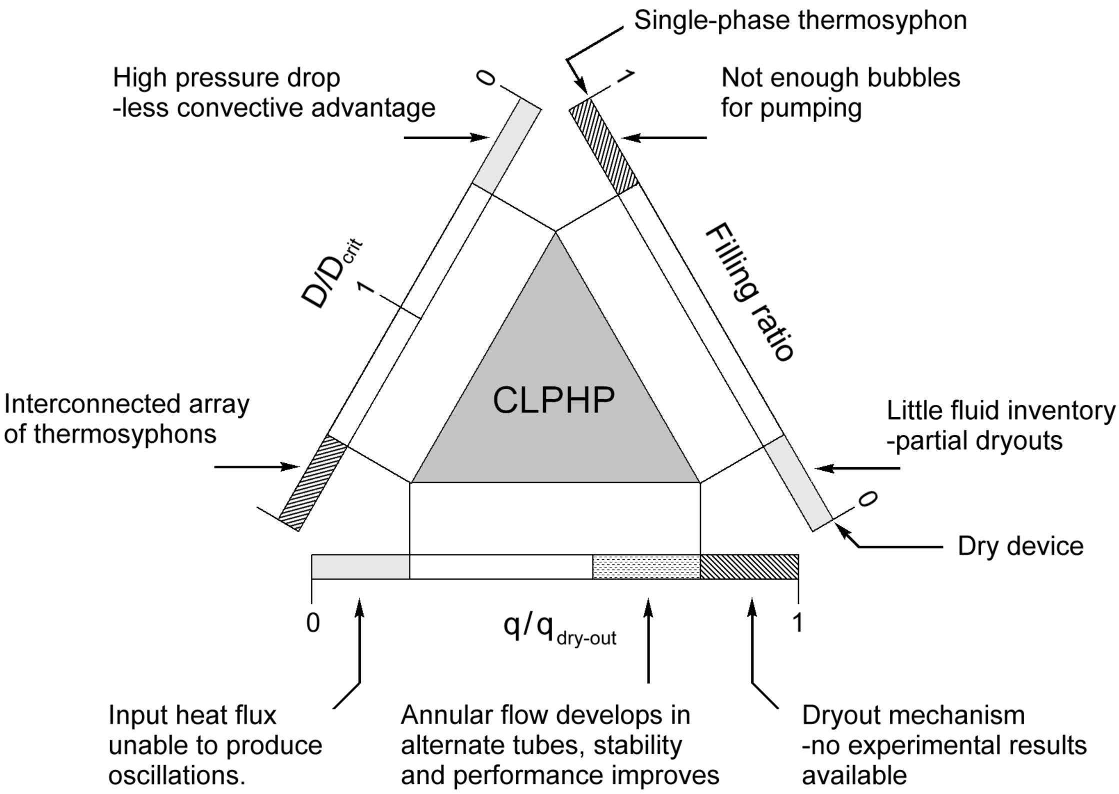

Figure 10: Boundary conditions for CLPHP operation

(c) PHP true working range: Between about 10% to 90% fill charge the PHP operates as a true pulsating device. The exact range will differ for different working fluids, operating parameters and constructional details. The more bubbles (lower fill charges), the higher is the degree of freedom but simultaneously there is less liquid mass for sensible heat transfer. Less bubbles (higher fill charges) cause less perturbations and the bubble pumping action is reduced thereby lowering the performance. Thus an optimum fill charge exists. It can therefore be concluded that the filling ratio is also an independent parameter which defines a closed loop pulsating heat pipe.

CONCLUSIONS

Figure 10 summarizes the main conclusions of this paper. From a thermal point of view, CLPHPs are devices which lie in between extended surfaces metallic fins and conventional heat pipes. Strictly speaking, the term pulsating 'heat pipes' is a misnomer. There are at least three thermo-mechanical boundary conditions i.e. internal tube diameter, input heat flux and the filling ratio, which are to be satisfied for the structure to behave as a true 'pulsating' device. The most interesting (disturbing!) aspect is the fact that the best performing closed loop pulsating heat pipe no longer behaves as a pulsating device. Alternating tubes then have slug flow and annular flow and the bulk flow takes a fixed direction. This aspect certainly requires further experimental evidence and supportive quantitative data is required which would also transform the loosely depicted phenomenological and qualitative definition in Figure 10 to more concrete foundations. Also, near dry-out behavior and the mechanism of dry-out itself requires further research.

ACKNOWLEDGEMENTS

The research work is partly supported by Deutsche Forschungsgemeinschaft (grant GR 412/33-1).

References

[1] Boure J., Bergles A. and Tong L., Review of Two-phase Flow Instability, Nuclear Engg. Design, Vol. 25, pp. 165-192, 1973 (also ASME paper No 71-HAT-42).

[2] Akachi H., Polášek F. and Štulc P., Pulsating Heat Pipes, Proc. 5th Int. Heat Pipe Symp. Melbourne, pp. 208-217, 1996.

[3] Groll M. and Khandekar S., Pulsating Heat Pipes: A Challenge and Still Unsolved Problem in Heat Pipe Science, Archives of Thermodynamics, Begell House, ISSN 1231-0956, Vol. 23/4, pp. 17-28, 2002. (also in Proc. 3rd Int. Conf. on Transport Phenomenon in Multiphase Systems (HEAT 2002), ISBN 83-88906-03-8, pp. 35-43, Baranów Sandomierski, Poland, 2002).

[4] Tong B., Wong T. and Ooi K., Closed–Loop Pulsating Heat Pipe, Applied Thermal Engg., ISSN 1359-4311, Vol. 21/18, pp. 1845-1862, 2001.

[5] Miyazaki Y. and Arikawa M., Oscillatory Flow in the Oscillating Heat Pipe, Proc. 11th Int. Heat Pipe Conf., Tokyo, pp. 367-372 , 1999.

[6] Wallis G., One Dimensional Two-phase Flow, Mc-Graw Hill, 1969.

[7] Bretherton F., The Motion of Long Bubbles in Tubes, J. of Fluid Mechanics. Vol. 10, pp. 167-188, 1961.

[8] White E. and Beardmore R., The Velocity of Rise of Single Cylindrical Air Bubbles Through Liquids Contained in Vertical Tubes, Chem. Engg. Science, Vol. 17, pp. 351-361, 1962.

[9] Chisholm D., Two-phase Flow in Pipelines and Heat Exchangers, George Goodwin NY, 1983.

[10] Delano A., Design Analysis of the Einstein Refrigeration Cycle, Ph. D. Thesis, Georgia Institute of Technology, 1998. (available at http://www.me.gatech.edu/energy/andy_phd/)

[11] Saha P., Ishii M. and Zuber N., An Experimental Investigation of the Thermally Induced Flow Oscillations in Two-phase Systems, ASME J. of Heat Transfer, Vol. 98, pp. 616-622. 1976.

[12] Khandekar S., Dollinger N. and Groll M., Understanding Operational Regimes of Pulsating Heat Pipes: An Experimental Study, Applied Thermal Engineering, Elsevier Science, ISSN 1359-4311, Vol. 23/6, pp. 707-719, 2003.

[13] Khandekar S., Groll M., Charoensawan P. and Terdtoon P., Pulsating Heat Pipes: Thermo-fluidic Characteristics and Comparative Study with Single Phase Thermosyphon, Proc. 12th Int. Heat Transfer Conference, ISBN-2-84299-307-1, Vol. 4, pp. 459-464, Grenoble, France, 2002.

[14] Qu W. and Ma T., Experimental Investigation on Flow and Heat Transfer of a Pulsating Heat Pipe, Proc. 12. Int. Heat Pipe Conf., pp. 226-231, Moscow, 2002.

[15] Khandekar S. and Groll M., An Insight into Thermo-Hydraulic Coupling in Pulsating Heat Pipes, Int. J. of Thermal Sciences, Elsevier Science, ISSN 1290-0729, (article in press), 2003.