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Pulsating Heat Pipes: Attractive Entrants in the Family of Closed Passive Two-Phase Systems

Sameer Khandekar and Manfred Groll*
Institut für Kernenergetik und Energiesysteme (IKE)
Universitat Stuttgart, Pfaffenwaldring 31
D-70569 Stuttgart, Germany

Abstract

The electronics semi-conductor technology is changing the face of the world in an unprecedented manner. Thermal management of such systems is the elixir to transform dreams and imaginations of the designers into reality. Motivation and need for research in the development of novel cooling strategies for modern electronics is of paramount importance. Pulsating heat pipes are one such promising example; they are new entrants in the family of closed passive two-phase heat transfer systems. This paper highlights the thermo-hydrodynamic characteristics of these devices. State of the art indicates that at least three thermo-mechanical boundary conditions have to be met for the device to function properly as a 'pulsating' heat pipe. These include the internal tube diameter, the applied heat flux and the amount of the working fluid in the system. Additionally, the number of turns of the device, the orientation with respect to gravity and the thermo-physical properties of the working fluid also play a vital role in determining the thermal behavior. Apart from these thermo-mechanical boundary conditions, research also indicates that two-phase flow patterns inside the device have a decisive effect on the effective thermal conductivity of the system. It is concluded that with the progress in the development of these devices achieved so far, the future prospects seem quite promising.

INTRODUCTION

Miniaturization is in vogue and this euphoria has especially gripped the electronics and allied industries. The insatiable urge for 'going nano' does come with associated inter-disciplinary technological problems. Although this 'increase power-decrease size' scenario has been prevalent for many decades, in recent years, thermal management has become the major feasibility bottleneck for micro/power electronics. Each conceived new design coming up in the market is with higher power dissipation levels. In addition, total dissipated power is not the only problem; heat

* Corresponding author. Tel.: +49 711 685 2481; fax: +49 711 685 2010;
E-mail address: groll@ike.uni-stuttgart.de

NOMENCLATURE

Cp specific heat at constant pressure (J/kg·K)

D characteristic duct dimension, diameter (m)

g acceleration due to gravity (m/s²)

h_{fg} latent heat (J/kg)

k thermal conductivity (W/m·K)

L_e evaporator section length (m)

Le_{ff} effective length of PHP (m)

N number of turns

ΔP pressure difference (N/m²), in this case between evaporator and condenser

$\dot{Q}$ heat throughput (W)

$\dot{q}$ heat flux (W/m²)

ΔT temperature difference (°C or K), in this case between evaporator and condenser

Greek symbols

ρ density (kg/m³)

μ dynamic viscosity (N×s/m²)

n^* characteristic velocity (m/s)

Non-dimensional numbers

Bo Bond number = $D\times(g(\rho_{liquid}-\rho_{vapor})/\sigma)^{0.5}$

$Eö$ Eötvös number = $(Bo)^2$

Ja Jakob number = $h_{fg}/(Cp\times\Delta T)$

Ka Karman number = $f\times Re^2 = (\rho\times\Delta P\times D^3)/\mu^2\times L_{eff}$

Pr Prandtl number = $(C_p\times\mu)/k$

Re Reynolds number = $\rho\times v^*\times D/\mu$

density (power per unit area) is complimentary to it. Market expectations towards higher functionality at reduced package sizes have led to denser electronics and increased clock frequencies which pose a simultaneous challenge to thermal management of increased power levels coupled with high heat fluxes.

As shown in Figure 1, the Moore's law states that the semiconductor transistor density, and hence the performance, doubles roughly every 18 months [1-3]. History has supported the conclusions of this law ever since its inception in 1970. In addition, an unprecedented exponential growth of power densities has occurred from 0.5 W/cm² in the eighties to more than 15 W/cm² by the turn of the last century as seen in Figure 2 [4]. All future predictions point towards die level heat fluxes of 40 to 60 W/cm² as the norm rather than exception. While the power consumption per typical gate has come down, the number of gates per device has dramatically increased with the 'system on chip' concept bringing the net dissipated power to over 100 W/module. Thus, thermal management has become a major challenge in the electronics industry. To address the future issues and projections, technological advancement is likely to be the only savior. This includes development of new materials, novel cooling strategies and a paradigm shift in the cooling technology concepts and modes of implementation, e.g. development of metal matrix composite materials, silicon nano-technology and MEMS, mini/micro heat pipes, loop heat pipes, microscale boiling/condensation, etc.

In the wake of the ongoing trends, the demands from the contemporary thermal solutions of electronic equipment are as follows: (a) Thermal resistance from chip to heat sink < 1 K/W, (b) High heat transport capability (5 -250 W), (c) Heat flux spreading (1-40 W/cm^2), (d) Mechanical & thermal compatibility, (e) Long term reliability, (f) Miniaturization and (g) Low cost.

The primary goal of any cooling system is to enhance the package or module performance and the reliability, which incidentally are strong functions of temperature. The temperature plays a role in the following:

- Device functionality: since many electrical parameters are temperature dependent.
- Device safety: as all exposed parts etc. must abide by the international standards of operating temperature.
- Device failure: since most mechanisms of physical failure are dependent on absolute temperature, temperature differences or spatial/temporal gradients, common examples being electro-migration, corrosion and solder fatigue, to name a few. This aspect is the most critical since quality and reliability issues are inherently linked to the junction temperature or temperature gradients.

Further, modern electronics cooling problems can be broadly divided primarily into three levels (refer Figure 3) [5]:

- First level cooling is concerned with heat dissipation from a chip to a directly connected chip carrier. The carrier is either the chip package (plastic, ceramic, etc.) in case of single chip packages, or a substrate in case of multi chip modules.
- Second level cooling involves the thermal path from a chip carrier to a casing/cold plate.
- Finally, third level cooling is the heat rejection from casing/cold plate to the ambient.

With these boundary conditions in mind, novel cooling strategies are continuously being developed. Considerable research in new material development and characterization has drastically improved the performance of the first level cooling. Historically, the second and third level cooling were primarily based on natural convection or forced convection air-cooling techniques. Continued stringent demands led to the development of phase change techniques such as pool boiling, jet impingement cooling and more recently mini/micro channel flow boiling concepts [6, 7]. In parallel, heat pipes in various configurations and designs, have played a decisive role in many second/third level cooling applications [8, 9]. Introduction of micro heat pipes has led to the development of many hybrid-cooling concepts combining different levels of cooling [10].

Meandering tube Pulsating Heat Pipes (PHPs) is another novel concept proposed by Akachi [11-13] and shown in Figure 4, which seems to meet all the present day cooling requirements. Typically suited for second/third level management, PHPs have already found some applications in micro and power electronics applications owing to the unique operational characteristics coupled with relatively cheaper costs. Although grouped as a subclass of the overall family of heat pipes, the complexity of thermo-hydrodynamic coupling is distinctly unique. A completely different research outlook is needed due to its singular thermofluidic behavior. The subtle nuances of this conceptually simple device are considerably intriguing for theoretical and experimental studies alike.

This paper aims to summarize the fundamental thermo-hydrodynamic operational characteristics of pulsating heat pipes. Attention is focussed particularly on Closed Loop Pulsating Heat Pipes (CLPHPs) without directional flow control check valve. Before proceeding further, a review of contemporary cooling strategies is presented so that the concept of CLPHPs can be better appreciated in the context of prevailing trends.

ELECTRONICS THERMAL MANAGEMENT STRATEGIES: A REVIEW

Electronics thermal management techniques have undergone a drastic evolutionary change over the last fifty years. As a result, several techniques have emerged depending on the need and applicability, as summarized in Table 1 [6, 7]:

Table 1. Summary of electronic cooling methods [6, 7].

Single-phaseTechnology	Two-phase Technology	Special Technology
Natural convection (air or liquid)	Pool boiling Falling film	Thermoelectric devices Heat pipes, Thermosyphons,
Forced convection (air or liquid)	Liquid jet impingement/ Spray/mist cooling Flow boiling	(various designs) Pulsating heat pipes

A further distinction may be made between indirect cooling, in which the heat-dissipating electronic components are physically separated from the coolant, and direct cooling, for which the components come in direct contact with the coolant.

Without going into the details of each technology, we will directly focus our attention on the special cooling technologies encompassing passive two-phase devices.

Conventional Heat Pipes

Heat pipes, due to their unique physical features, operational characteristics and applications, deserve a special mention in the field of microelectronics thermal

management. Historical development of heat pipes dates back to 1942 when the first patent for a heat pipe employing a capillary wick for pumping liquid against gravity was applied by Gaugler [14]. Grover [15], along with his co-workers reinvented the 'heat pipe' in 1963. A heat pipe is a passive two-phase heat transfer device capable of transferring large quantities of heat with a minimal temperature drop. In general, typical heat pipes utilize the continuous evaporation/condensation of a suitable working fluid for two-phase heat transport utilizing latent heat of vaporization in a closed system. By virtue of capillary pumping action, the heat pipe can be operated in a micro gravity field (as in satellites) or against gravity (on the ground with the evaporator above the condenser).

The conventional heat pipe dimensions range from a few centimeters in length and millimeters in diameter up to lengths of more than 10 meters and diameters of the order of many centimeters. Since a wick structure is used to transport the fluid from the condenser to the evaporator and the most significant performance limitation is the capillary limit, for high performance applications, complex wick designs have been developed (see Figure 5). Other limitations include the viscous, the sonic, the entrainment and the boiling limit. Some of these limitations can be avoided or the operating range of the device can be widened by using design variations such as Capillary Pumped Loops (CPLs) or Loop Heat Pipes (LHPs). The areas of application of conventional heat pipes include heat exchangers and boilers, electrical and electronics equipment cooling, medical devices, temperature control in satellites, de-icing and permafrost stabilization, solar heating systems, isothermal furnace liners, black body radiators, etc.

Gravity-assisted Thermosyphons

If the heat pipe is operated in the gravity field with the evaporator below the condenser, the external body force field can be utilized to drive the condensate back to the evaporator and therefore no capillary structure may be necessary. Such devices are called closed two-phase thermosyphons as shown in Figure 6 (a). The most important operational limitation comes from the counter-current flow of liquid and vapor phases.

Loop Heat Pipes

Like a conventional heat pipe, a LHP utilizes capillary action to circulate the working fluid in a closed loop by the action of the input heat. Hence, a LHP also contains no mechanical moving parts nor any electrical power for operation. But in contrast to a heat pipe, the wick structure of a LHP is confined only to the evaporator section as shown in Figure 6 (b) . The LHP design thus allows vapor and liquid to flow in separate smooth wall pipes, greatly reducing the dissipative losses. For these and other reasons, LHPs are thermally a better option than conventional heat pipes and have demonstrated reliability and robustness in both ground and micro gravity

applications. Currently, miniaturization of LHPs is at the forefront of an extensive research and development to provide cooling solutions to the high heat load/ flux problems of advanced electronics packaging [16,18].

Micro Heat Pipes

Stringent cooling requirements posed by the electronics industry concerning miniaturization and higher heat fluxes led to the conceptual introduction of a micro heat pipe (MHP) by Cotter [19]. MHPs are very small, wickless heat pipes having internal geometries containing sharp angled corner regions responsible for generating the required capillary pressure and which serve as liquid arteries (Figure 7). There are two supplementary definitions of a MHP. According to the first definition, the hydraulic radius of a MHP is of the order of (greater than or equal to) the capillary radius of the vapor-liquid interface [19]. The second definition reflects not only the geometry, but also the physical behavior and defines a heat pipe as 'micro' if, $Bo \leq 2$ [20]. Typical cross sectional dimensions of MHPs range from some 10 to some 100 μm with heat transport capabilities of some 1/100 to some 1/10 W [21]. It is emphasized that a sizable volume of the so-called 'micro' heat pipes reported in the literature does not represent the true 'micro' heat pipes as defined above. There exist various hybrid and transitional heat pipe designs in the 'meso' or 'mini' range which are frequently misnamed. Examples are the copper-water conventional heat pipes with typical outside diameters of 2.5 to 4 mm, which are routinely employed, for example, in cooling the CPUs in laptop computers.

Pulsating Heat Pipes

Pulsating heat pipe structures are characterized by the following features (see Figure 4):

(a) The structure is made of a meandering tube of capillary dimensions with many turns, filled partially with a suitable working fluid. This tube may be either: (i) Open Loop: tube ends are not connected to each other, (ii) Closed Loop: tube ends are connected to each other forming an endless loop.

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

(c) At least one heat-receiving (evaporator/heater) zone is present.

(d) At least one heat-dissipating (condenser/cooler) zone is present.

(e) There can be an optional adiabatic section between evaporator and condenser zone.

The device is first evacuated and then filled partially with a working fluid, which distributes itself naturally in the form of liquid-vapor plugs and bubbles inside the capillary tube. There is no external control over the initial plug/bubble distribution inside the tube. One end of this tube bundle receives heat transferring it to the other

end by a pulsating action of the liquid-vapor system. A PHP is essentially a 'non-equilibrium' heat transfer device driven by a complex combination of various types of two-phase flow instabilities [22, 23]. The performance success primarily depends on the continuous maintenance or sustenance of such conditions within the system. The liquid plugs and vapour bubbles are transported because of the pressure pulsations caused inside the system. The construction of the device inherently ensures that no external mechanical power source is needed for fluid transport. The driving pressure pulsations are fully thermally driven. Although many of the arguments and discussions that follow in this paper apply to both open and closed loop systems, attention is primarily focussed only on CLPHPs, the principal candidate of present scrutiny.

OPERATIONAL CHARACTERISTICS OF PHPS

An actual working CLPHP is not isothermal; there exists a temperature gradient between the evaporator and the condenser section. Temperature differences also exist amongst the 'U' bends within the evaporator and the condenser due to local non-uniform heat transfer rates that are always expected in real systems. The net effect of all these temperature gradients within the system is to cause non-equilibrium pressure conditions that, as stated earlier, are the primary driving force for the thermofluidic transport. This leads to thermally driven two-phase flow instabilities. The heating process in the evaporator continuously tries to 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. Further, inherent perturbations are always present in real systems due to:

- Pressure fluctuations within the evaporator and condenser tube sections due to local non-uniform heating and cooling,
- Unsymmetrical liquid-vapour distributions causing uneven void fraction in the tubes, and
- 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 CLPHP. It is to be noted that there occurs no 'classical steady state' in CLPHP operation as far as the internal hydrodynamics is concerned. Instead, pressure waves and pulsations are generated in each of the individual tubes, which interact with each other possibly generating secondary and ternary reflections with perturbations. The self-exited thermally driven oscillations seem to be dependent on a plethora of variables.

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 of these devices 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 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 [24-26]. If there is a transition to annular flow under the imposed thermo-mechanical boundary conditions, then the dominance of latent heat increases. The most interesting (disturbing!) 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 takes a fixed direction. Strictly speaking, the term 'pulsating heat pipes' then becomes a misnomer. The construction of the device is such that on a macro level, heat transfer can be compared to an extended surface 'fin' system. Although such an analogy provides important insight of the mechanism of heat transfer, extrapolations cannot be done authoritatively unless more data are available.

The internal tube diameter is one of the parameters which essentially defines a CLPHP. The physical behavior adheres to the 'pulsating' mode only under a certain range of diameters. The Eötvös number (square root of the Bond Number) criterion, i.e. Eö H" 4, for surface tension dominated adiabatic slug flow [27, 28] only provides a tentative design rule for the internal diameter of a CLPHP. In some studies, although the Eötvös number was much below the prescribed maximum limit of $E \approx 4$, gravity forces were definitely seen to affect the performance [29, 30]. Similarly, for conditions where Eö was somewhat greater than the value of 4, the CLPHP 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 function as an interconnected array of normal gravity assisted thermosyphons. Below a certain Eö, the dissipative resistance of the flow will lead to a decrease in the thermal performance, i.e. increase in thermal resistance.

The applied heat flux to the system is the most vital parameter for proper operation. Not only because of the fact that it provides the driving energy, it also catalyzes two important phenomena, i.e. (i) flow pattern transition in the device and, (ii) affecting the two-phase flow instabilities and thereby the level of internal perturbations. A certain minimum heat flux is required to overcome the dissipative fluid flow losses. Thereafter, an increase in the input heat flux leads to a series of changes in the internal flow patterns (from slug to semi-annular and fully developed annular flow) which directly affects the heat transfer characteristics (refer Figure 8). Present studies [29-31] indicate that design of these devices should aim at thermo-mechanical boundary conditions which result in convective flow boiling conditions in the evaporator leading to higher local heat transfer coefficients. Also, in conjunction with a minimum number of turns, a minimum heat flux is additionally required to make the thermal performance nearly independent of the orientation.

The volumetric filling ratio of liquid also affects the thermal performance [32]. For a given heat throughput requirement, an operationally better performing and self-sustained thermally driven pulsating action of the device 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 9). 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, instead 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.

It is obvious that the thermophysical properties of the working fluid, in conjunction with the device geometry, have profound implications on thermal performance, affecting the following:

- The relative share of latent and sensible heat in the overall heat throughput,
- The possibility of having different flow patterns, e.g. slug-annular flow inter-transition,
- 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 [13, 24, 29-31]. 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 trade-off of various thermophysical properties has to be achieved depending on the imposed thermo-mechanical boundary conditions. This certainly requires further research.

A certain critical number of turns is required, in addition to the minimum heat flux requirement to make the performance of the CLPHP free of the operating orientation. This is attributed to the increase in level of internal perturbations. The phenomenological trend of the available data is depicted in Figure 10 [29, 30].

MODELING OF PHPS

There have been various approaches to describe and predict PHP performance but unfortunately all of these have not been successful. They can be summarized as [33]:

- The PHP system is compared to an equivalent single spring-mass-damper system in which the system specifications are affected by the heat transfer.
- In a similar approach as above, instead of single spring-mass-damper system, the PHP is compared to a multiple spring-mass-damper system. This model describes only the kinematics of the liquid-vapour plugs without considering any heat transfer.
- In another approach the PHP is modeled with the first principles by applying fundamental equations of mass, momentum and energy conservation to a specified control volume.
- PHP analysis is done showing the existence of chaos under some operating conditions.

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 attempt has been made to develop semi-empirical correlations, of the form given in Eq. (1) below, for the radial evaporator heat flux, to fit the data [27, 28].

$$\dot{q} = \left(\frac{\dot{Q}}{\pi \cdot D \cdot N \cdot (2L_e)} \right) = 0.54(\exp(\beta))^{0.48} Ka^{0.47} \Pr_{liq}^{0.27} Ja^{1.43} N^{0.27} \quad (1)$$

This correlation is valid for the entire set of data for CLPHP as represented phenomenologically by Figure 10. 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 to the wider research in mini/micro scale boiling heat transfer in open/closed systems.

In addition, Artificial Neural Network based modeling has also been successfully applied to predict the CLPHP performance with acceptable accuracy [34]. 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 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.

CONCLUSIONS

Two-phase passive systems have been serving the thermal management industry since quite some time. The future trends are stressing the existing designs and concepts of cooling. CLPHP is a highly attractive two-phase passive heat transfer technology, which due to its simple design, cost effectiveness and excellent thermal performance is expected to find wide applications in the years to come. Further research is definitely needed to control the operation and fully understand the thermo-hydrodynamics of the system under various operating conditions.

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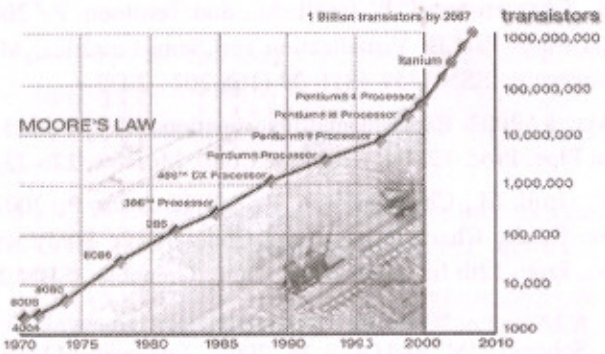

Figure 1 : The Moore's law for transistor count in electronics industry [1-3].

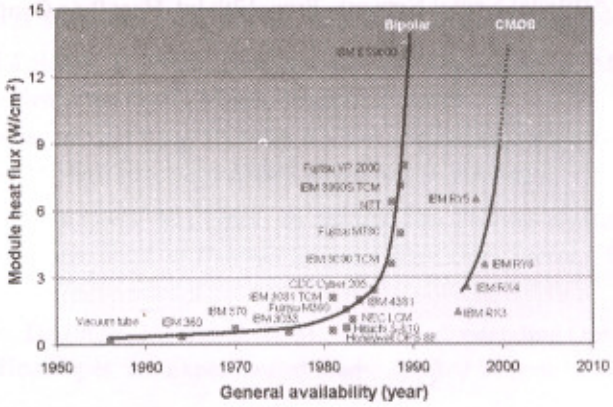

Figure 2 : Timeline of module heat fluxes in microprocessors [4].

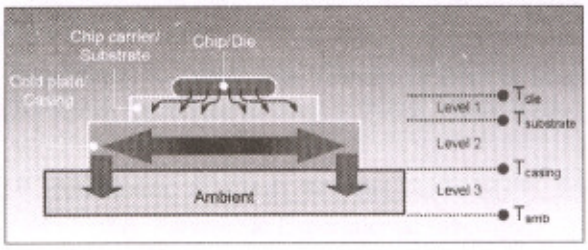

Figure 3 : Thermal management levels of a typical package (adapted from [5]).

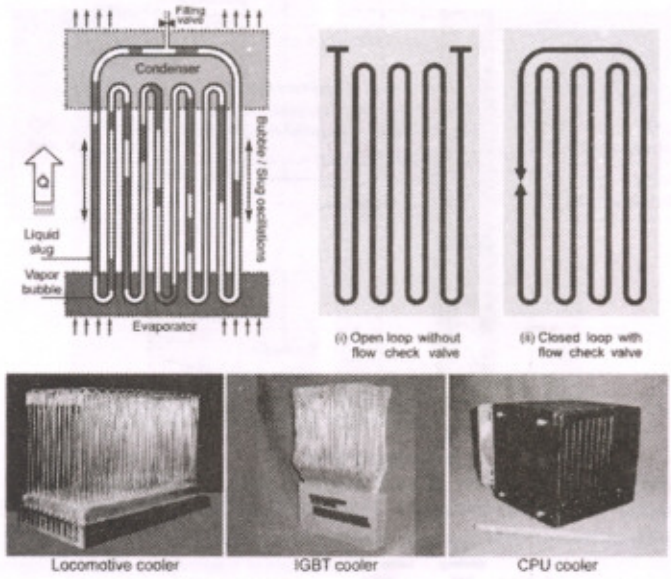

Figure 4. Schematic of a typical pulsating heat pipe with two design variations and some examples of its industrial applications [11-13, 23, 27].

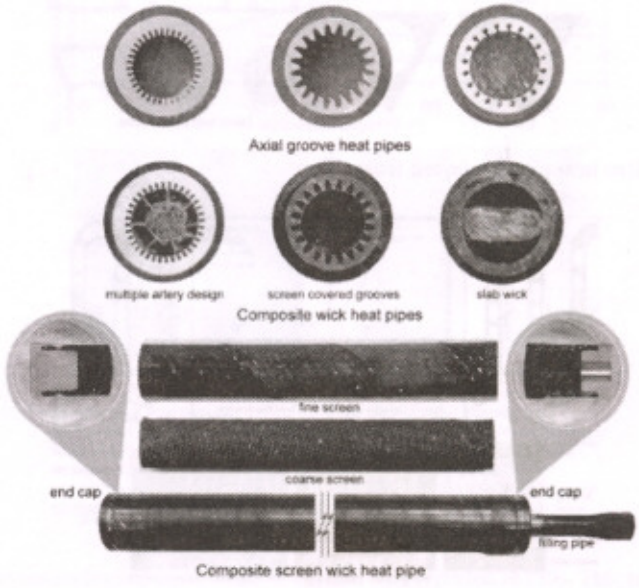

Figure 5. Details of conventional heat pipe with examples of various wick structures.

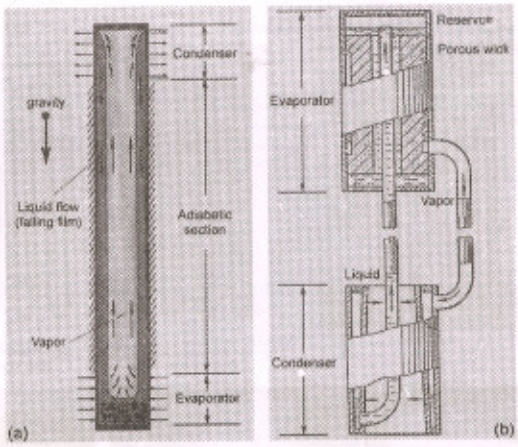

Figure 6 : Schematic of (a) gravity assisted thermosyphon (b) a loop heat pipe.

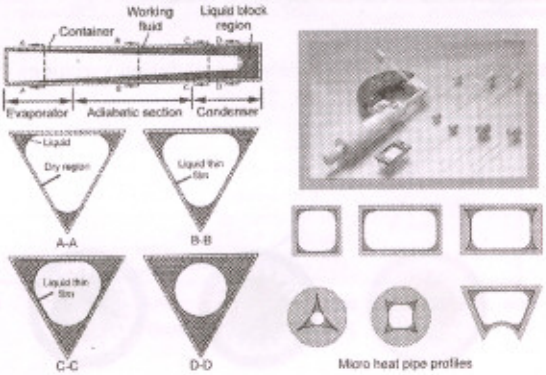

Figure 7 : Micro heat pipes (adapted from [7]).

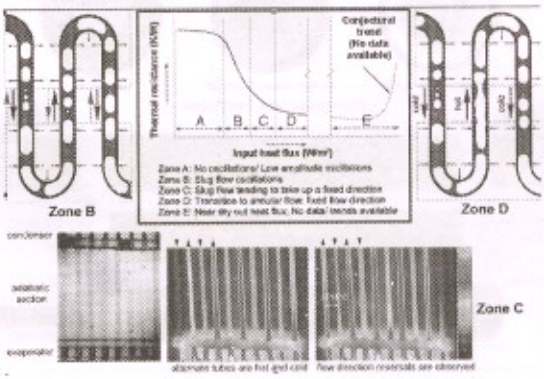

Figure 8 : Effect of input heat flux on the thermo-hydrodynamics of CLPHPs. Increasing input heat flux changes the flow patterns and affects the oscillating tendencies thereby decreasing the thermal resistance.

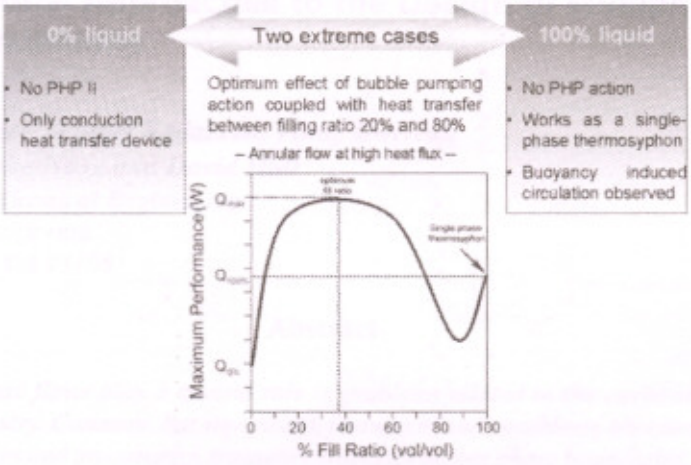

Figure 9 : Phenomenological trend for the effect of volumetric fluid filling ratio on the performance of a CLPHP. The maximum performance does not corresponds to dry-out power but to a specified safe evaporator temperature.

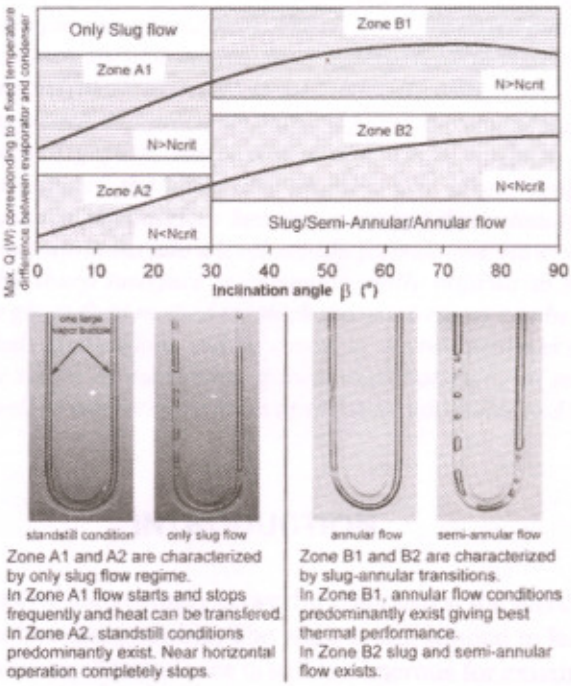

Figure 10 : Phenomenological trend of experimental data reported in [30]. This trend is the result of various flow patterns occurring in different subsections, as depicted. The images are taken in the evaporator section of the visualization set-ups.