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CLOSED AND OPEN LOOP PULSATING HEAT PIPES

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ABSTRACT

Thermal management of electronics is a contemporary issue which is increasingly gaining importance inline with the advances in packaging technology. Material science, packaging concepts, fabrication technology and novel cooling strategies are some of the key areas requiring synchronal research for successful thermal management. Focusing on the latter area, this paper attempts to summarize the complex thermo-hydrodynamics of Pulsating Heat Pipes (PHPs), especially suited for thermal management of electronics. Considerable progress has been achieved in the last decade in the understanding of these devices but quite a few nuances of the device operation still remain unexplored or unclear. Nevertheless, with the progress achieved so far, the prospects for this exemplary and unprecedented technology seems quite promising.

KEY WORDS: pulsating heat pipes, open and closed loop, heat transfer, electronics cooling.

1 INTRODUCTION

Contemporary trends in thermal management of electronics are very demanding and the limits are being stressed in every aspect of design. Market expectations include: (a) Thermal resistance from chip to heat sink < 1 K/W, (b) High heat transport capability up to 250 W), (c) Heat flux spreading up to 60 W/cm^2, (d) Mechanical and thermal compatibility, (e) Long term reliability, (f) Miniaturization, and (g) Low cost. These demands pose a simultaneous challenge of managing increased power levels and fluxes [1, 2].

With such stringent boundary conditions in mind, neoteric cooling/ heat transfer strategies are continuously being demanded i.e. development of phase change techniques such as pool boiling, jet impingement cooling and more recently mini/micro channel flow boiling concepts [3]. In parallel, heat pipes in various configurations and designs, have played a decisive role in many applications. Inline with these developments is the introduction of pulsating heat pipes in the early-nineties [4-7], as a very promising heat transfer technology, especially suited for thermal management of electronics.

This paper summarizes the major milestones and progress achieved in understanding of pulsating heat pipes (PHPs).

2 CONSTRUCTIONAL DETAILS

PHPs are characterized by the following basic features (refer Figure 1):

(a) The structure is made of a meandering/ serpentine tube of capillary dimensions with many turns, filled partially with a suitable working fluid.

This tube may be either:

- Closed Loop: tube ends are connected to each other in an endless loop.
- Open Loop: tube ends are not connected to each other; essentially one long tube bent in multiple turns with both its ends sealed after filling the working fluid.

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

(c) At least one heat-receiving zone (evaporator /heater), heat-dissipating zone (condenser /cooler) and an optional in-between adiabatic zone are present.

The serpentine tube is evacuated and then partially filled with a working fluid. Filling results in a natural, uncontrolled, asymmetric liquid-vapor, plug-bubble distribution (uneven void fraction) in the tube sections, due to the dominance of surface tension forces [8].

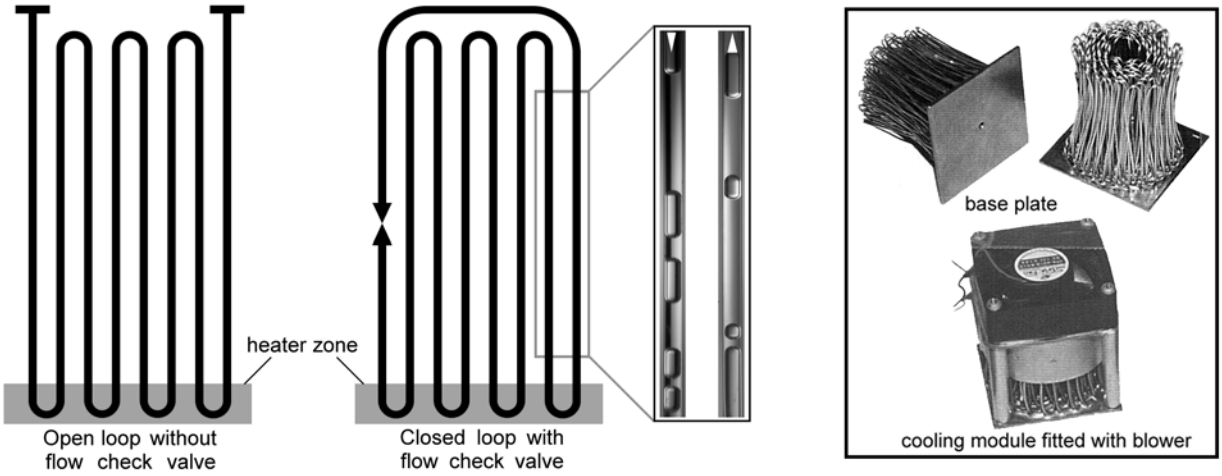

Figure 1: Schematic of open and closed loop pulsating heat pipe with a prototype

3 HEAT TRANSFER MECHANISM

One end of the PHP tube bundle receives heat, transferring it to the other by a pulsating action of the working fluid, generating, in general, a capillary slug flow.

While in operation, there exists a temperature gradient between the heated and cooled end. Small temperature differences also exist amongst the individual 'U' bends of the evaporator and condenser due to local non-uniform heat transfer rates which are always present in real systems. Since each tube section between the evaporator and the condenser has a different volumetric distribution of the working fluid, the pressure drop associated with each sub-section is different. This causes pressure imbalances leading to thermally driven two-phase flow instabilities eventually responsible for the thermofluidic transport. Bubble generation processes in the heater tubes sections and condensation processes at the other end create a sustained 'non-equilibrium' state as the internal pressure tries to equalize within the closed system. Thus, a self-sustained thermally driven oscillating flow is obtained. There occurs no 'classical steady state' in PHP operation as far as the internal hydrodynamics is concerned. Instead, pressure waves and fluid pulsations are generated in each of the individual tube sections, which interact with each other generating secondary/ ternary reflections with perturbations [8, 9]. It will be appreciated that PHPs 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. The construction of PHPs is such that on a macro level, heat transfer can be compared to an extended surface 'fin' system. Simultaneously, the internal fluid flow may be compared to flow boiling in narrow channels.

PHPs may never be as good as an equivalent heat pipe or thermosyphon system which are based on pure latent heat transfer. If the behavior is well understood, the performance may be optimized towards classical heat pipes/thermosyphons, as a limiting case. At the least, the manufacturing complexities of heat pipes will be avoided. As compared to an equivalent metallic finned array, at the least there will be a weight advantage. Finally, there is always a reliability advantage because of the absence of an external mechanical pump [10].

The available experimental results and trends indicate that any attempt to analyze PHPs must address two strongly interdependent vital aspects simultaneously, viz. system 'thermo' and 'hydro-dynamics'. Figure 2 shows the genealogy of two-phase passive devices. Although the representation is not exhaustive, all the systems with relevance to the present interest are depicted. Although all the systems shown in Figure 2 have 'similar' working principles, there are decisive differences that significantly alter the course of mathematical analyses. The family can be subdivided into three major sub-groups, as shown.

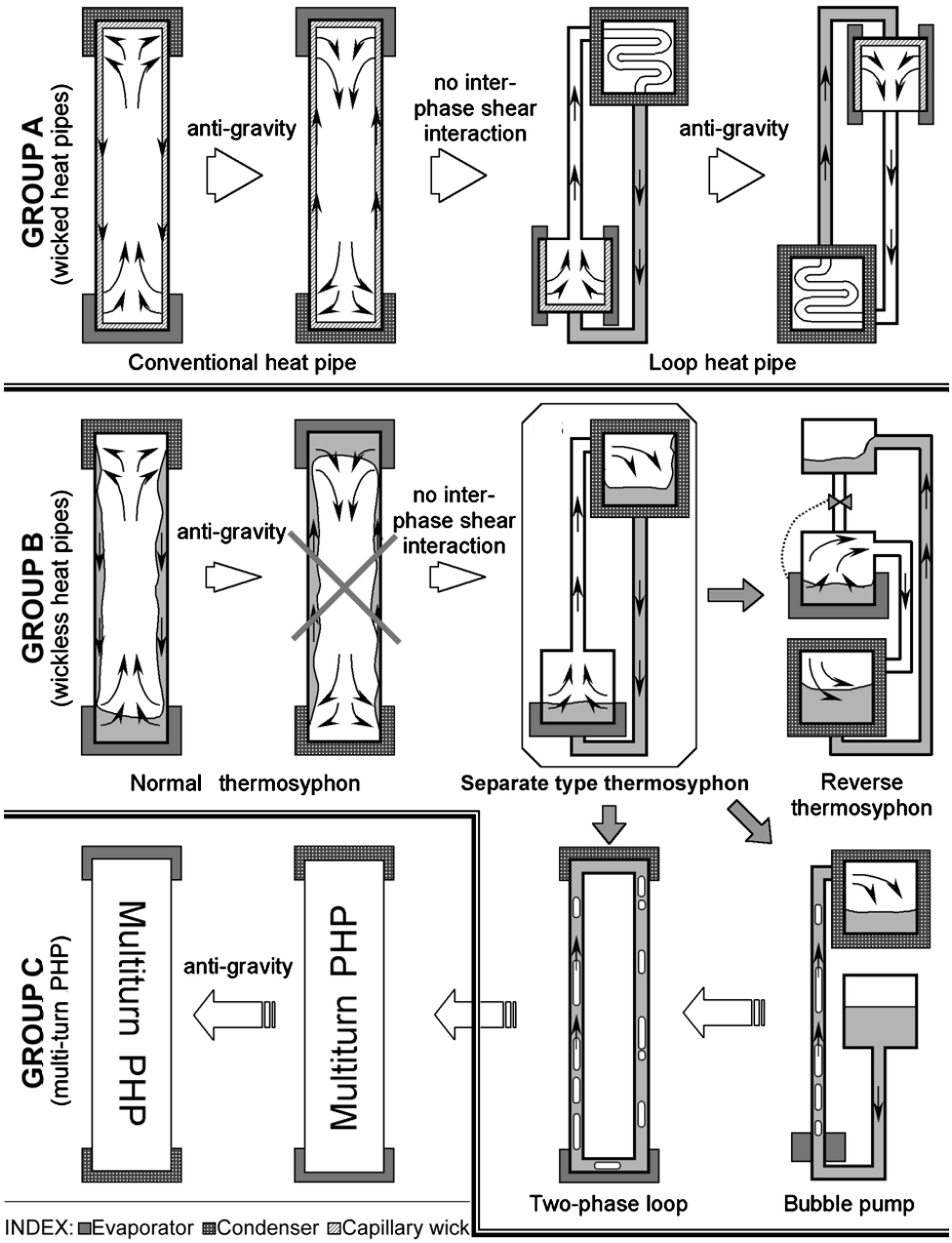

Figure 2: Genealogy of passive two-phase systems

A multiturn PHP is a thermo-hydrodynamic genesis of a reverse thermosyphon (without control valve), a bubble pump and a two-phase loop [11]. Although the relevant forces acting in this family of devices are common, their relative importance substantially differs because of: (i) the presence or absence of wick structure, (ii) the sizes of the tubing used for construction, (iii) liquid/ vapor interaction depending on the construction and, (iv) location of evaporator with respect to the condenser section. The applicable equations in each case have been outlined by Khandekar et al. [12].

4 INFLUENCE PARAMETERS

Looking into the available literature, it can be seen that six major thermo-mechanical parameters have emerged as the primary design parameters affecting the PHP system dynamics [13]. These include:

- Internal diameter of the PHP tube,
- Input heat flux to the device,
- Volumetric filling ratio of the working fluid,
- Total number of turns,
- Device orientation with respect to gravity, and
- Working fluid thermo-physical properties.

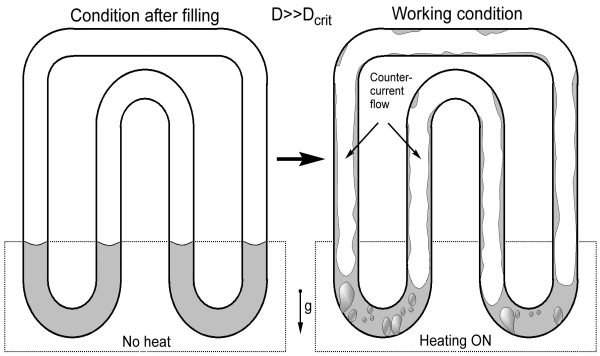

Figure 3: Thermosyphon mode of operation when $D >> D_{crit}$ [10]

Other conditions which influence the operation are:

- Use of flow direction control check valves,
- Tube cross sectional shape,
- Tube material and fluid combination, and
- Rigidity of the tube material, etc.

Apart from these variables, the performance is also strongly linked with the flow patterns existing inside the device (which in turn depends on the complex combination of other design parameters). Various flow patterns other than capillary slug flow, e.g. bubbly flow, developing or semi-annular flow and fully developed annular flow (in case of CLPHPs) have also been reported which have a significant effect on the thermal performance of the device [14-19].

The state of the art strongly suggests that a comprehensive theory of the complex thermo-hydrodynamic phenomena governing the operation of PHPs is not yet available [20]. Authoritative quantitative data base explicitly connecting the thermal performance with individual influence parameters is limited but growing continuously. With the available information, very preliminary conclusions regarding the complete design procedure of PHPs may be made.

4.1 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 critical Bond number (or Eötvös) criterion gives the tentative design rule for the diameter [7, 10]:

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

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

This criterion ensures that individual liquid slugs and vapor bubbles are formed in the device and they do not agglomerate leading to phase separation, if the device is kept isothermally in a non-operating period. This is most crucial, especially if top heating mode is employed. In bottom heat mode, though at Eö > 4 the tendency of slug flow diminishes as surface tension tends to reduce, a certain amount of liquid transport is still possible by the bubble pumping action thereby providing substantial heat transfer.

For a given specified heat power, decreasing the diameter will increase the dissipative losses and lead to poor performance. Increasing the diameter much above the critical diameter will change the phenomenological operation of the device. It will no more act as a pulsating heat pipe but will transform into an interconnected array of two-phase thermosyphons as shown in Figure 3. In this case then, only bottom heat mode of operation is possible.

4.2 Applied heat flux

The applied heat flux affects the following [20, 21]:

(a) Internal bubble dynamics, sizes and agglomeration/breaking patterns,

(b) Level of perturbations and flow instabilities, and

(c) Flow pattern transition from capillary slug flow to semi-annular and annular.

PHPs are inherently suitable for high heat flux operation. Since the input heat provides the pumping power, below a certain level, no oscillations commence. In case of CLPHPs, a unidirectional circulating flow has been observed at high heat fluxes. In addition, the flow also gets transformed from oscillating slug flow to annular flow. Once a flow direction is established, alternating tubes sections become hot and cold (hot fluid flows from evaporator in one tube and cold fluid from the condenser flows in the adjacent tube).

Further increase of heat flux will lead to some dry-out mechanism(s) induced by thermo-hydro-dynamic limitations. These have not been clearly identified and studied so far.

4.3 Working fluid filling ratio

Experimental results so far indicate that there is an optimum filling ratio for proper PHP operation (in the pulsating mode of operation). This optimum, however, is not sharply defined but generally is a plateau around 40% fill charge. For tube sections which have sharp angled corners, it is possible that the optimum filling ratio with respect to the overall thermal resistance is of the order of 5% to 15% (this FR is comparable to that of conventional heat pipes). This happens because the sharp angled corners act as a capillary pumps and the PHP starts behaving as wickless 'micro' heat pipe. In such a condition, the structure can no more be called as a pulsating heat pipe and although the thermal resistance is the lowest for such an operation, the maximum heat throughput is extremely limited.

A too high filling ratio above the optimum leads to a decrease in the overall degree of freedom as there are not enough bubbles for liquid pumping. All other parameters remaining fixed, an optimum effect of bubble pumping coupled with heat transfer from the resulting flow conditions is obtained for a certain range of the filling ratio.

At 100% filling ratio, the device acts as a single phase buoyancy driven thermosyphon [18]. In this mode too, substantial heat transfer can take place, but the action is limited to bottom heat mode only.

4.4 Total number of turns

The number of turns increases the level of perturbations inside the device. If the number of turns is less than a critical value, then there is a possibility of a stop-over phenomenon to occur. In such a condition, all the evaporator U-sections have a vapor bubble and the rest of the PHP has liquid. This condition essentially leads to a dry out and small perturbations cannot amplify to make the system operate self-sustained.

If the total heat throughput is defined, increasing the number of turns leads to a decrease in heat flux handled per turn. Thus, an optimum number of turns exits for a given heat throughput.

4.5 Operating orientation

Apart from simplicity of design, one of the strongest cases in favor of pulsating heat pipes is that their thermal performance is independent of the operating orientation. Nevertheless there are some contradicting trends in the literature. In some studies either there was a large variation of performance with device orientation, or horizontal as well as anti-gravity (heater-up) operation was not achieved at all [15, 17, 22]. Several results from other sources for a multi-turn CLPHP suggest that horizontal operation is possible albeit not as good as the vertical operation [14, 23, 24]. Some studies indicate near complete performance independence with orientation [6, 7, 25]. These apparently contradictory and uncomplimentary results seem to suggest that requirements for an orientation independent operation are:

(a) sufficiently large number of PHP turns, which is responsible for a higher degree of internal perturbations and inhomogeneity of the system,
(b) a high input heat flux leading to higher 'pumping power' and enhanced instabilities,
(c) these two aspects are not mutually exclusive and must simultaneously be satisfied.

The results of Akachi [6, 7], Charoensawan et al. [26] and Khandekar et al. [27] tend to support the first hypothesis. They conclude that a certain critical number of turns is required to make horizontal operation possible and also to bridge the performance gap between vertical and horizontal operation. This is attributed to increase in the overall level of internal perturbations. The second hypothesis is tentatively supported by the fact that even for vertical operation, there is a critical minimum input heat flux requirement to initiate self-excited oscillations [9-11, 16, 17, 20, 28]. In the absence of gravity, this minimum heat flux is likely to be higher.

4.6 Sensible vs. latent heat

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 (as in case of OLPHPs and in case of CLPHPs at low heat fluxes), then it has been demonstrated that latent heat will not play a dominant role in the overall heat transfer [9, 28]. If there is a transition to annular flow under the imposed thermo-mechanical boundary conditions (in case of CLPHPs), then the dominance of latent heat increases leading to better performance. 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 and the bulk flow takes a fixed direction. Strictly speaking, the term pulsating 'heat pipes' then becomes a misnomer.

5 SUMMARY OF RESULTS FOR CLOSED LOOP PHPs

The experimental results presented in this section summarize the general phenomenological trends for CLPHPs.

Figure 4 shows the details of a CLPHP made of copper tube. The set-up consisted of a copper block of size 130 x 25 x 25 mm^3 forming the evaporator. This cooper block was fitted with four circular cartridge AC heaters (ø 10.0 x 25 mm). Five thermocouples, suitably placed as shown, measured the average evaporator copper block temperature. The U-turns of the CLPHP were soldered into the evaporator block. The safe maximum heat flux (limited by set-up safety) based on the U-turn tube section area soldered in the evaporator block was about 12 W/cm^2.

The CLPHP was formed from copper tube (ID 2.0 mm, OD 3.0 mm), with 20 turns on each side (a total of 40 tube sections), having a staggered pitch of 10 mm between the respective tube sections. The total length of the tube used was 5.4 m with a filling volume of 17.0 cc. Leaving apart the welded length of the U-turns embedded into the evaporator block, the entire CLPHP was cooled by forced air cooling. The average air velocity was 3.5 m/s with ambient air temperature of 27°C ±1.5°C.

The results are shown in Figures 5-8. While the results are self explanatory, the following are the main conclusions:

(a) A combination of large number of turns and high input heat flux ensures continuous CLPHP operation in any orientation without appreciable change in thermal performance. Both the requirements should be simultaneously satisfied to achieve this goal. In general, start-up by a step power level is only possible beyond a minimum heat flux. This minimum start-up power was much smaller in vertical the bottom heat mode than in the vertical anti-gravity mode. Beyond the critical heat flux, no 'stop-over' was ever detected and continuous operation was always achieved.

(b) In general, start-up by a step power level is only possible beyond a minimum heat flux. This minimum start-up power was much smaller in vertical the bottom heat mode than in the vertical anti-gravity mode. Beyond the critical heat flux, no 'stop-over' was ever detected and continuous operation was always achieved.

(c) The thermal resistance continuously decreases with increasing heat input until heat transfer gets limited by the external air-side heat transfer coefficient.

(d) Although an optimum filling ratio exists, the sensitivity of the filling ratio parameter is not very high within the limits of 30% to 70%. This sensitivity further reduces with increasing heat input. At high enough heat input (with FR between 30% to 70%), the performance is nearly independent of the global orientation.

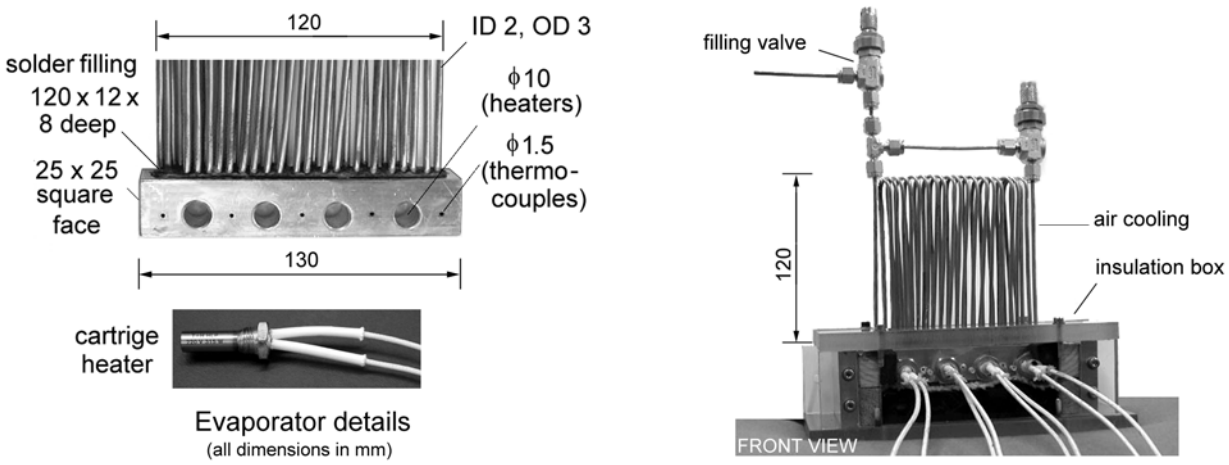

Figure 4: Schematic details of the experimental set-up

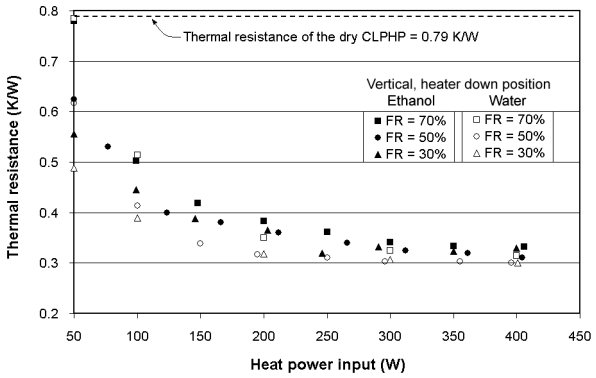

Figure 5: Effect of input heat power on thermal performance of the CLPHP. The performance becomes nearly independent of the flux (after 200 W, corresponding to about 6 W/m^2) and the filling ratio (between 30% and 70%) and becomes limited by the external air side heat transfer coefficient.

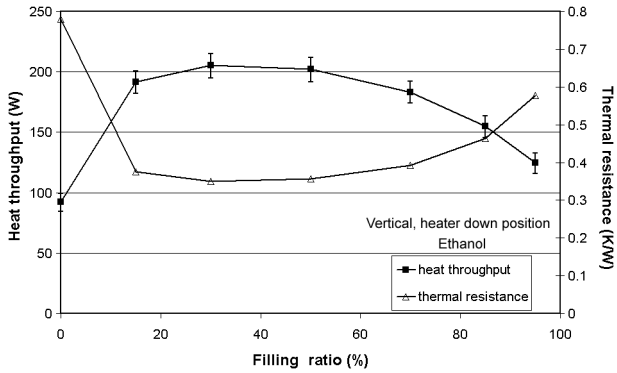

Figure 6: Effect of filling ratio on thermal performance of the CLPHP; heat throughput and thermal resistance correspond to an average T_e = 100°C.

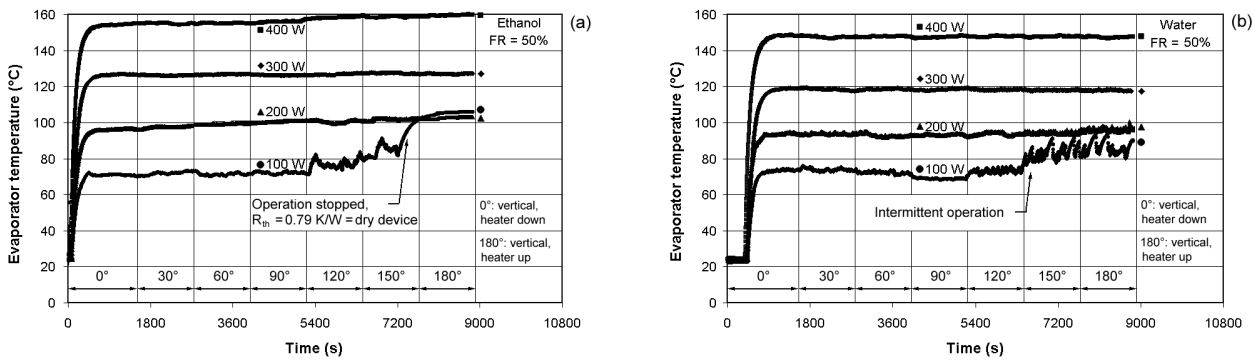

Figure 7: Effect of operating orientation on the average evaporator temperature of the CLPHP. (a) ethanol (b) water. Above 100 W, the performance is nearly independent of the orientation, at 100 W, large scale temperature fluctuations are present and anti-gravity operation is not ensured.

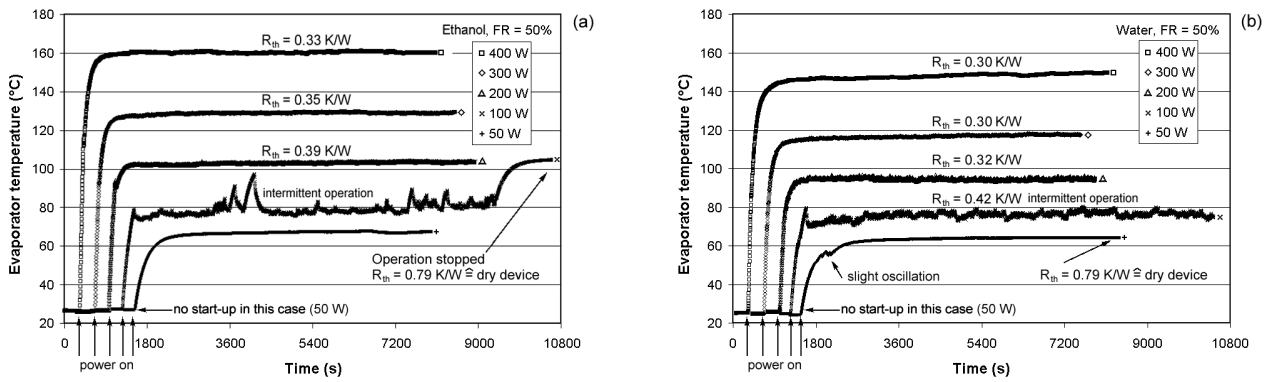

Figure 8: Anti-gravity (heater on top) start-up by step power input to the CLPHP, (a) ethanol (b) water. For very low input power (up to 50 W) no oscillations are initiated. For low input power (up to 100 W), start-up is not ensured (large temperature oscillations/ intermittent operation). Higher power levels ensure successful start-up and unhindered steady performance.

6 SUMMARY OF RESULTS FOR OPEN LOOP PHPS

In contrast to CLPHPs, there is no possibility of an overall flow circulation to develop in OLPHPs. Thus, the possibility of the development of well structured circulating annular flow is also non-existent. The flow remains in the oscillating capillary slug flow regime with long vapor bubbles forming at higher heat fluxes. In addition, the possibility of counter-current flow is also enhanced in case of OLPHPs. Since local internal heat transfer usually enhances in the convective annular flow regime, CLPHPs in general show better performance than OLPHPs. Of course, if the overall heat transfer is limited by the external ambient heat transfer characteristics (as is usually the case in air cooling), then the performance of the two types may be nearly comparable.

Experimental results on OLPHPs have been reported in the original patent by Akachi [4-6] for a power range of 5 to 90 W in top and bottom heating mode with an average thermal resistance ranging from 0.64 to 1.16 K/W (R-142b).

Maezawa et al. [23] studied an OLPHP consisting of 20 turns of copper tube (ID 1.0 mm) of total length 24 m. 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. [29] 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 9.

TS-Heatronics Co. Ltd., Japan have developed a range of PHPs including design variations termed as 'Heat Lane' and 'Kenzan' fins [30]. Material combinations, e.g. SS-liquid N_2, Al-R142 and copper with water, methanol, R113 and R142b have been tested. Thermal resistance of ≈ 0.3 K/W at a cooling air velocity of 3 m/s was obtained for Kenzan fins (outside dim. 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, have been used for cooling MCMs and IGBTs.

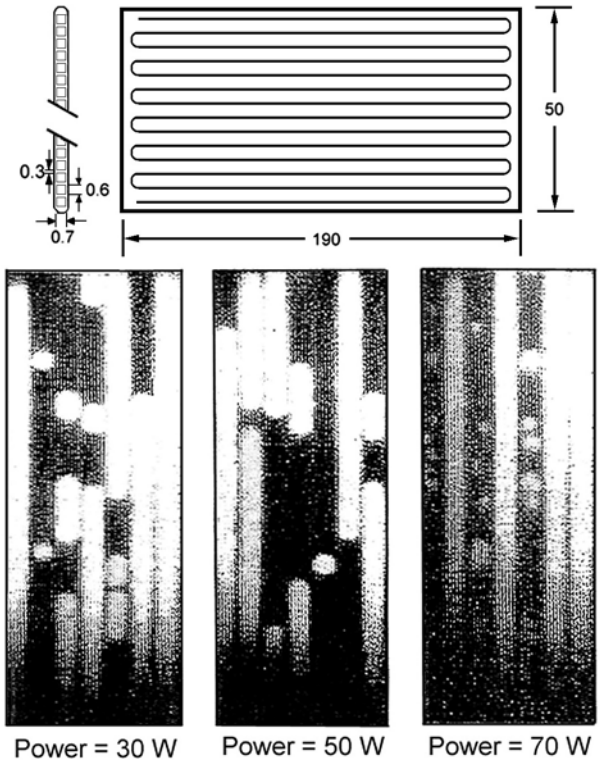

Figure 9: Setup details and results by Kawara et al. [29]

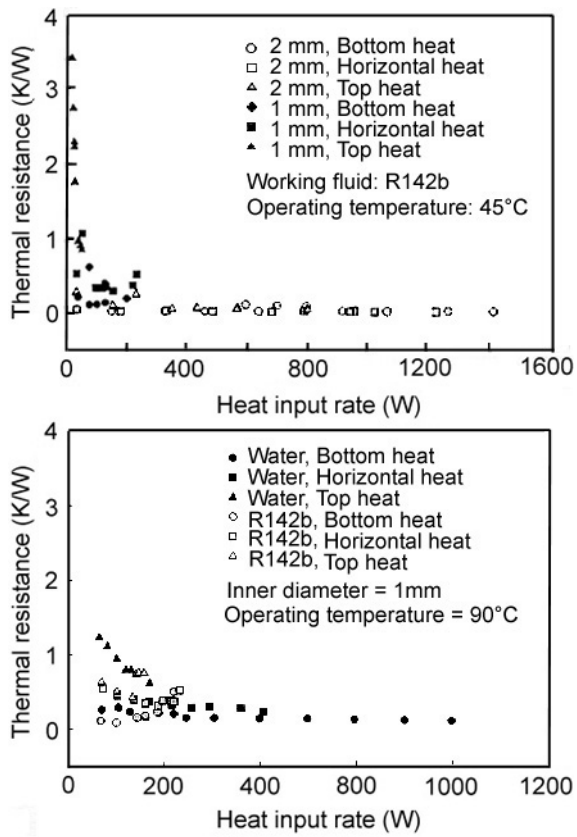

Figure 10: Results by Maezawa et al. [24]

Maezawa et al. [24] 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 m were made of copper tube of ID 2.0 mm and 1.0 mm, respectively. The effects of diameter and the working fluid are shown in Figure 10. It can be seen that the performance for the bottom heat mode was better than for horizontal mode. Poor performance for the top heat mode was observed.

More recently Rittidech et al. [31, 32] investigated the effect of inclination angles and working fluid properties for an OLPHP made of copper tubes (ID 2.03 mm, $L_e = L_a = L_c = 50$ mm, 100 mm and 150 mm, $L_t = 10$ m). R123, ethanol and water were used as working fluids with a filling ratio of 50%. The evaporator and condenser sections were maintained at fixed temperatures of 80°C (hot water) and 20°C (water + ethylene glycol 50:50 vol.), respectively. Simultaneously, a visualization study on a similar glass tube set-up was also undertaken. Figure 11 shows the effect of operating inclination angle on Q_{max}/Q_0 (Q_0: heat throughput for the horizontal operation). The maximum values of Q_{max}/Q_0 for R123, ethanol and water are 2.19, 2.15 and 1.98 respectively. In addition, it was found that a working fluid with lower latent heat of vaporization exhibited a higher Q_{max}/Q_0. It was found that, as the evaporator section length decreases from 150 to 50 mm the main flow changes from slug flow together with long slugs/ partial annular flow to slug flow with bubbly flow.

6.1 Dry-out of OLPHPs

At a certain high heat input to the OLPHP, the performance limit may be reached which results in a dry-out or burn-out in the evaporator. The associated heat flux is called the critical heat flux.

The thermo-hydrodynamic model of dry-out is not yet clear. At 'normal' operating condition the OLPHP operates by the simultaneous oscillation of bubble slugs and liquid plugs inside the capillary tube. If the heat flux at the evaporator section is increased to a very high level, violent boiling will occur with a very high evaporation rate. The oscillation is also violent. At a specific condition, the size of bubbles in the respective evaporator tubes increases leading to partial dry-outs in some tubes. The liquid gets prevented from entering the evaporator. The accumulated heat in the evaporator section results in a very high wall temperature leading to a burnout. The series of events is depicted in Figure 12 [33].

Figure 13 shows the critical heat flux at dry-out as reported by Katpradit et al. [34]. The OLPHPs were set to operate at both vertical and horizontal orientations with the working temperature maintained at 60 ± 5°C. For each OLPHP, $L_e = L_a = L_c$ and the FR = 50%. Two copper bars were welded onto each OLPHP to form the evaporator section while the condenser section used a cooling water jacket. The wall temperatures in each tube of the evaporator section were compared for each heating step. The procedure was repeated until one or more wall temperatures in the evaporator section started to increase rapidly indicating that the dry-out state had been reached. The results show that the critical heat flux decreased as the section length increased, and increased with increase in the latent heat of vaporization. It was suggested that that the dominating dimensionless parameters for heat transfer at horizontal heat mode were Ku, L_e/D_i, Ja and Bo while Ku, L_e/D_i, Ja, Bo and $1+(\rho_v/\rho_l)^{0.25}$ dominated the vertical heat mode. $1+(\rho_v/\rho_l)^{0.25}$ was chosen as an additional dimensionless parameter to represent flooding phenomena in the vertical mode.

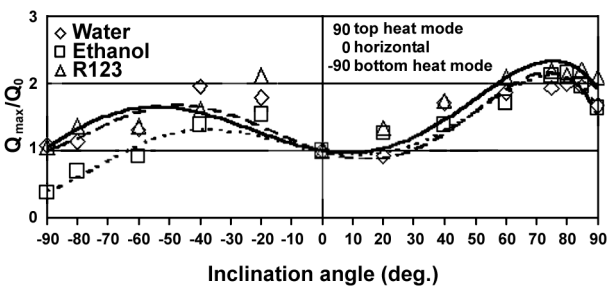

Figure 11: Effect of inclination angle on Q_{max}/Q_0 for the OLPHP tested by Rittidech et al. [31].

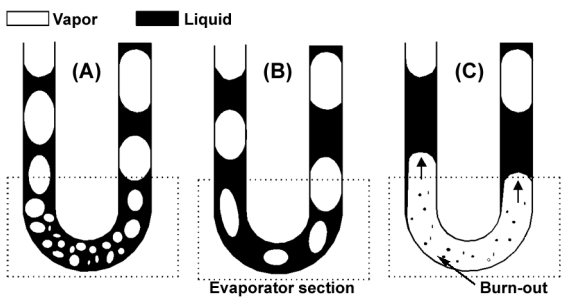

Figure 12: Operational states of OLPHP [33]

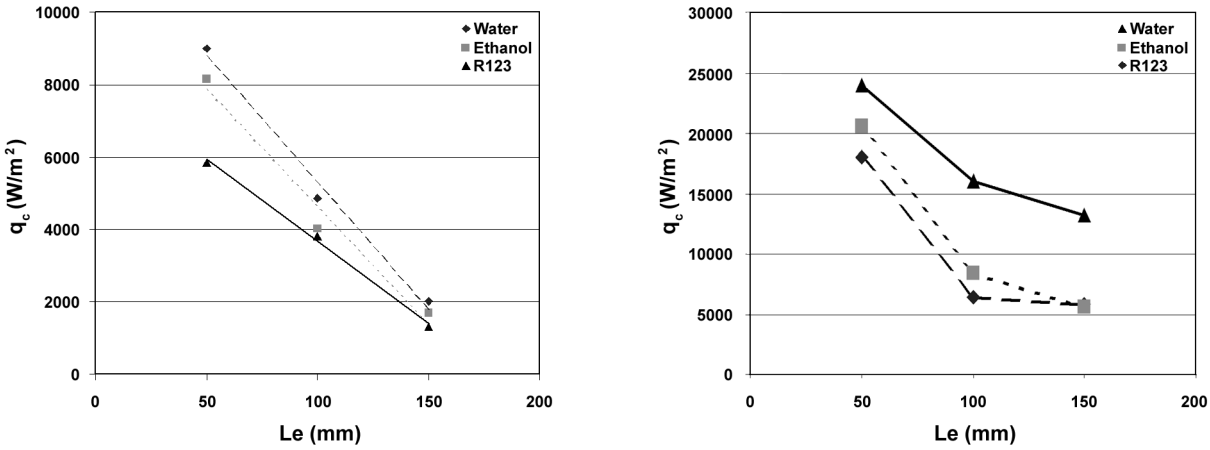

Figure 13: Variation of critical heat flux of OLPHP as a function of evaporator length (a) D_i=2 mm, 15 turns and horizontal mode, (b) D_i=2 mm, 15 turns and bottom heat mode of operation [34].

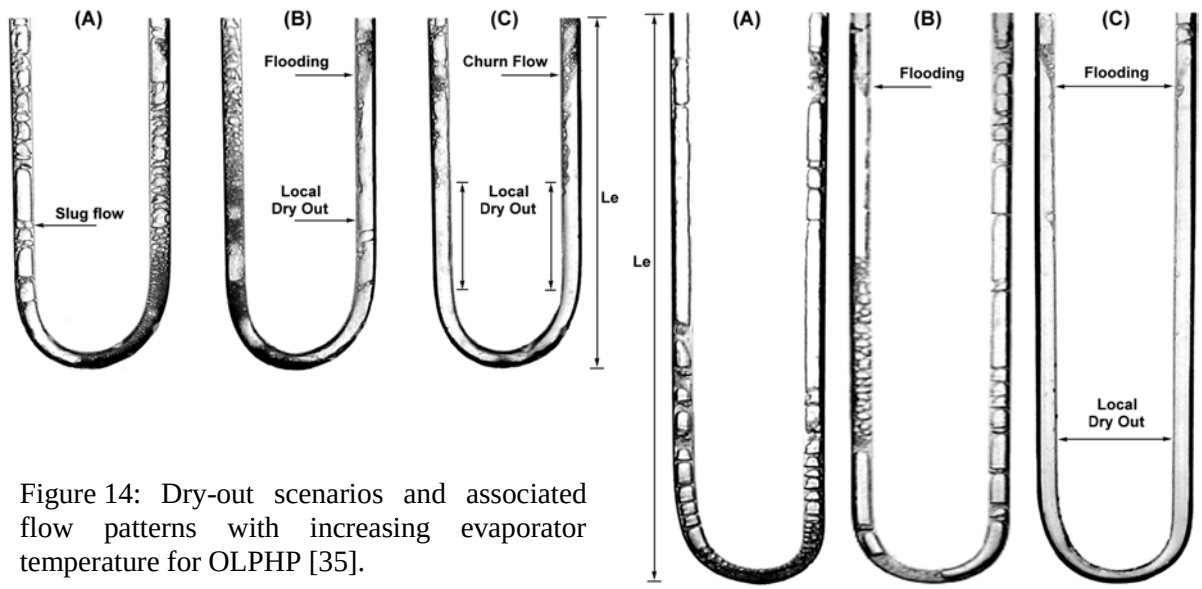

Figure 14: Dry-out scenarios and associated flow patterns with increasing evaporator temperature for OLPHP [35].

Near dry-out, visualization, as seen in Figure 14, reveals that dry-out is associated with a change in flow pattern from slug flow to churn flow which eventually leads to flooding and local dry-out patches in the evaporator [35].

7 MATHEMATICAL MODELING

Mathematical modeling and theoretical analysis of PHPs has been attempted in the recent past with many simplified approaches. These may be categorized as follows [8]:

(a) Comparing PHP action to equivalent single spring-mass-damper system [36],
(b) Kinematic analysis by comparison with a multiple spring-mass-damper system [37],
(c) Applying fundamental equations of conservation of mass, momentum and energy to a specified PHP control volume [38-40],
(d) Mathematical analysis highlighting the existence of 'chaos' [41, 42]
(e) Modeling by artificial neural networks, [43].
(f) Modeling by semi-empirical correlations based on non-dimensional numbers [26, 27]

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 PHPs because the velocity scale is dependent on the imposed thermal boundary conditions. Modeling by semi-empirical correlations seems to be the most promising at this stage. Khandekar et al. have presented such a correlation for limited amount of data as reported in [27] based on Jakob, Prandtl (liquid), and Karman numbers (see Figure 15).

Modeling of PHPs continues to pose a challenge.

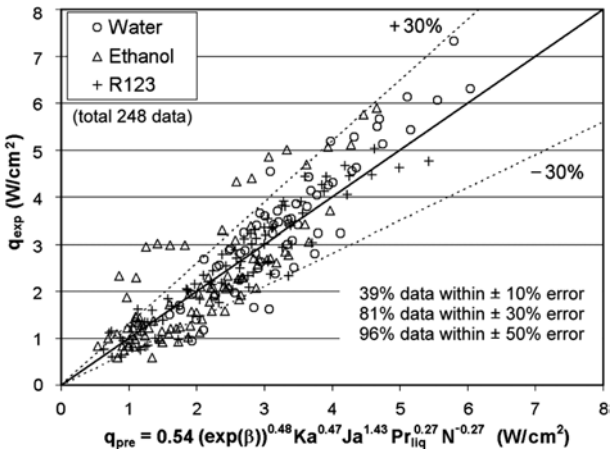

Figure 15: Semi-empirical model proposed in [27]

8 SUMMARY AND CONCLUSIONS

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.

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 performance with individual influence parameters is quite limited. Presently, only preliminary conclusions for a 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, attention is also required for space/avionics applications. There are not enough studies 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.
- 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 thermo-hydrodynamics in mini/ micro channels cannot be over-emphasized.
- The dryout mechanism of the device remains unexplored and is one of the most vital information needed at this stage.
- There have been practically no reported studies on life testing, 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.

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NOMENCLATURE

Bo : Bond number

c_p : specific heat (J/kgK)

D : tube diameter (m)

Eö : Eötvös number

FR : Filling Ratio (volume of liquid/inner volume of PHP at room temperature)

f : friction factor

g : acceleration due to gravity (m/s²)

h_{lv} : latent heat (J/kg)

Ja : Jakob number = $\dfrac{h_{lv}}{c_p \cdot (\Delta T)_{sat}^{e-c}}$

Ka : Karman number = $f \cdot (Re)^2$

Ku : Kutateladze number

$$= \frac{q''}{h_{lv} \cdot \rho_{vap} \left(\dfrac{\sigma \cdot g \cdot (\rho_{liq} - \rho_{vap})}{\rho_{vap}^2} \right)^{0.25}}$$

k : thermal conductivity (W/m·K)

N : number of turns

Pr : Prandtl number = $\dfrac{c_p \cdot \mu}{k}$

Q : heat power (W)

q'' : heat flux (W/m²)

Re : Reynolds number

T : temperature (K)

Greek Symbols

β : inclination angle to the horizontal (rad)

ρ : density (kg/m^3)

σ : surface tension (N/m)

μ : dynamic viscosity (N·s/m^2)

Subscripts

a : adiabatic section

c : condenser section

crit : critical

e : evaporator section

liq, vap : liquid, vapor

max : maximum

min : minimum

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