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An insight into thermo-hydrodynamic coupling in closed loop pulsating heat pipes

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Abstract

A Closed Loop Pulsating Heat Pipe (CLPHP) is a complex heat transfer device with a strong thermo-hydrodynamic coupling governing its thermal performance. To better understand its operational characteristics, a two-phase loop is constructed with a capillary tube (ID = 2.0 mm) having no internal wick structure. The loop is heated at one end and cooled at the other and partially made up of glass to assist visualization. The working fluid employed is ethanol. It is concluded from the study that a two-phase loop does represent the thermo-fluidic characteristics of a multi-turn CLPHP. Dynamic two-phase instabilities are present in a two-phase loop also; although the number of turns in a CLPHP increases the level of internal perturbations. The existence of an optimum number of turns for a given heat throughput requirement is explained. Also, it is shown that classical thermodynamics based on quasi-equilibrium theory seems not to be sufficient for complete system analysis. The performance (i.e., overall thermal resistance) is strongly dependent on the flow pattern existing inside the tubes. The role of gravity in the operation characteristics is clarified.

Keywords: Electronics cooling; Pulsating heat pipe; Two-phase loop; Flow patterns

1. Introduction

Closed loop pulsating heat pipes (CLPHPs), typically suited for micro electronics cooling, consist of a plain meandering tube of capillary dimensions with many U-turns and joined end to end as shown in Fig. 1. There is no additional capillary structure inside the tube as in a conventional heat pipe. The tube is first evacuated and then filled partially with a working fluid, which distributes itself naturally in the form of liquid–vapor plugs and slugs inside the capillary tube. One end of this tube bundle receives heat transferring it to the other end by a pulsating action of the liquid–vapor/slug-bubble system. There may exist an adiabatic zone in between. This type of heat pipe is essentially a ‘non-equilibrium’ heat transfer device. The performance success primarily depends on continuous maintenance or sustenance of these non-equilibrium conditions in the sys-

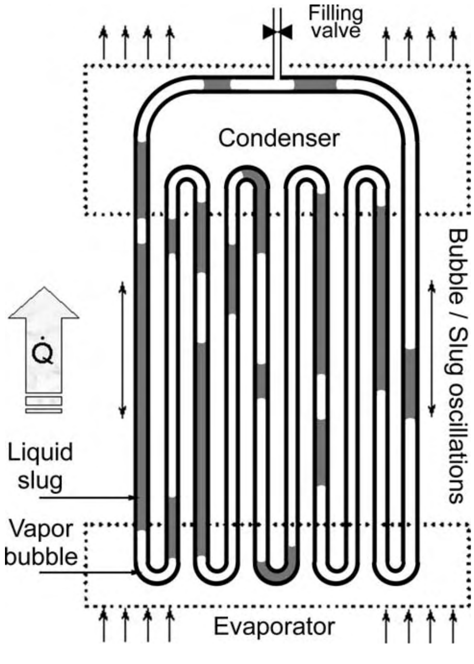

Fig. 1. Schematic of a closed loop pulsating heat pipe.

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Nomenclature

A	area of cross section	m^2
D	diameter	m
FR	filling ratio of CLPHP (V_{liq}/V_{total}) at room temperature	
h_{fg}	latent heat of vaporization	$J \cdot kg^{-1}$
ID/OD	inside/outside diameter	m
L	characteristic length	m
$N_{pc}\vert_{eq}$	equilibrium phase change number	
P	pressure, Pa	
$\dot{Q}$	heat input	W
$\dot{q}''_w$	heat flux	$W \cdot m^{-2}$
R_{th}	thermal resistance	$°C \cdot W^{-1}$
T	temperature	°C
V	volume	m^3
v	velocity	$m \cdot s^{-1}$

Greek symbols

$\Gamma_{vap}\vert_{eq}$	mass rate of vapor generation per unit volume	$kg \cdot s^{-1} \cdot m^{-3}$
Ω_{eq}	equilibrium frequency of phase change	$1 \cdot s^{-1}$
ξ_h	heated perimeter	m
ρ	density	$kg \cdot m^{-3}$
λ	thermal conductivity	$W \cdot m^{-1} \cdot K^{-1}$

Subscripts

e	evaporator
c	condenser
liq	liquid
vap	vapor
hyd	hydraulic

tem. The liquid and vapor slug/bubble transport is caused by the thermally induced pressure pulsations inside the device and no external mechanical power is required [1,2]. To understand the thermo-hydrodynamic characteristics of CLPHPs, one primary building block, i.e., a two-phase fluid loop was investigated. The assumption whether one loop truly represents all the thermo-fluidic characteristics of real CLPHPs is scrutinized. The limitations of this analogy and conclusions drawn thereof are investigated. The fundamental operational characteristics of the loop are reported providing vital information for multi-turn CLPHP performance optimization.

2. Experimental details

The two-phase loop is constructed with the primary aim of phenomenological observations and preliminary quantitative measurements. The geometrical details of the set-up are shown in Fig. 2. The evaporator section is made up of a copper block of size 40 × 30 × 5 mm^3. Two small copper tubes (ID 2.0 mm OD 3.0 mm) are brazed on two sides and connected to each other by a through hole inside the copper block. Two surface mounted DC heaters attached to the copper block provide the input power. The condenser section is similar in construction with a copper block size of 40 × 40 × 5 mm^3. The loop is completed by two glass tubes (ID 2.0 mm OD 4.0 mm) of length 100 mm each forming the adiabatic section. These tubes are connected to the respective open ends of the copper tubes by short flexible silicon tubes. A filling valve assembly is attached to the condenser section. Cooling water always maintained at 20 °C flows through two additional copper blocks tightly clamped via thermal grease to the condenser section. The loop is first evacuated to 10^{-2} Pa and then filled partially with the working fluid (ethanol). Temperature measurement is done by a data logger (resolution: 0.1 °C, accuracy: ±0.5 °C, time constant: 70 ms, measuring frequency of 1 Hz), coupled with ungrounded K type thermocouples of 1.5 mm diameter. Two thermocouples each are placed in drilled passages in the evaporator and condenser copper blocks recording the average temperatures (T_e and T_c). Two thermocouples (T1 and T2) are attached on the wall of the glass tubes. It is important to note that the temperature recorded by these two thermocouples does not truly represent the exact internal temperature fluctuations of the working fluid. Further, the recording frequency is limited to 1 Hz by the data logger. Nevertheless, the essence of the phenomenological condition of the fluid is certainly captured by these measurements and important conclusions may be drawn.

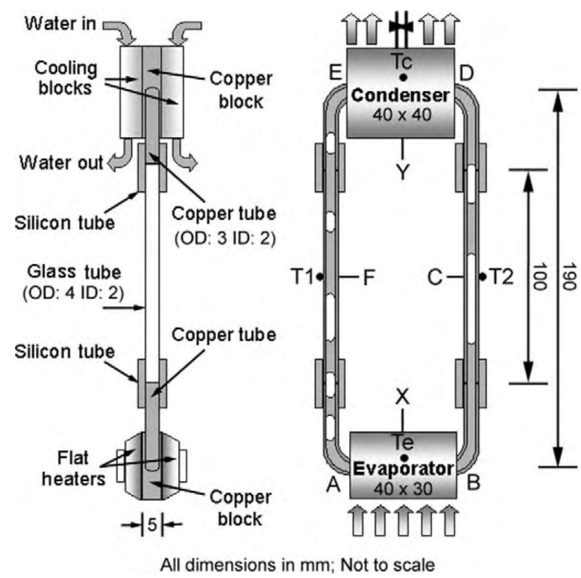

Fig. 2. Schematic of the two-phase loop.

3. Results and discussion

3.1. Flow patterns

It has been indicated by earlier studies [3,4] that there are a large number of parameters which affect the CLPHP operation. These can be summarized as (a) Geometrical parameters, i.e., diameter of tube, cross sectional shape of tube, length of evaporator/condenser section, overall length of the device, number of turns, (b) Operational parameters, i.e., global orientation, use of flow control check valve in the circuit and (c) Physical parameters i.e. thermophysical properties of working fluid and its filling ratio. The results of this study indicate that the thermal performance (i.e., overall thermal resistance defined as $R_{\mathrm{th}} = (T_{\mathrm{e}} - T_{\mathrm{c}})/\dot{Q}$) of a CLPHP is not only dependent on the multitude of parameters but also strongly on the two-phase flow patterns existing while in operation. Also, the study suggests that the flow pattern itself is linked to the filling ratio (FR) and the heat power input.

It has been established that the zone of interest for a CLPHP operation is achieved when the FR is between about 20 to 80% [5,6]. Below FR $\approx$ 20%, there are not enough distinct liquid plugs and the operation becomes 'unstable' resulting in large unacceptable variations in the average evaporator temperature. Above FR $\approx$ 80% there are not enough bubbles to provide the pumping action and so the performance drastically deteriorates. Within the operating range of the filling ratio, it has been observed in the present experiments that various flow patterns may exist depending on the boundary conditions.

Fig. 3 depicts the observed phenomena in the loop operated in vertical heater down position, with FR = 60% ethanol and increasing input heat power. The corresponding recorded temperatures, T_{e}, T1 and T2, for each case along with the performance parameters are also shown. At low heating power, case A, low amplitude oscillations with slug flow in both tube sections are observed. Individual bubbles oscillate about a mean position and there is very little bubble agglomeration. In fact, the bubbles in the adiabatic section act as isolators, considerably thwarting the mixing of hot and cold fluid sections. The hot slugs remain oscillating in the evaporator zone and the performance is poor (i.e., overall thermal resistance is considerably high). As the heat input is increased, case B and C, the oscillation amplitude increases. There is considerable improvement in the performance, with respect to the overall thermal resistance, as the amplitude becomes comparable to the overall length of the loop and more hot fluid is able to reach the condenser section. A complete turning of fluid starts sometimes in the clockwise and sometimes anti-clockwise direction until the stage represented by case D is reached. Here, the flow turns in one direction for a considerable time before a direction reversal takes place. This is also clearly represented by the changing pattern of the adiabatic tube temperatures, T1 and T2, for case D. During this time the hotter tube section starts to develop annular flow which changes back to slug flow by bridging/flooding action as the fluid travels towards the cold end. When the input power is further increased the flow direction reversal completely stops. The fluid flow takes an arbitrary direction, either clockwise or anti-clockwise, but then remains in the same fixed direction thereafter (case E). In such a condition, fully developed annular flow is observed in one section (the up-header/ hot fluid line) and bubbly/slug flow is observed in the other section (down-header/cold feeder line). The device thermal resistance is observed to be the least in such a situation. As the heat power is further increased, the flow pattern remains similar and the size of the bubbles in the down-header/cold feeder line keeps on shrinking since the overall system volume is fixed (the bubbles even become smaller than the tube diameter). If the FR is increased, it becomes more and more difficult to sustain annular flow (since the overall system is isochoric) and tendency is towards slug flow in both tube sections. This reduces the performance again [6].

If the flow patterns remains strictly slug flow in the entire system, it has been shown by experimental and simulation studies that latent heat will play an insignificant role in the overall heat transfer [4,7]. With the present findings of existence of annular flow with corresponding higher performance (i.e., lower thermal resistance), the contribution of latent heat needs reevaluation.

3.2. Role of gravity

In the present experiments satisfactory self oscillating operation of the device was only possible till a tilt angle of about 10° from the horizontal (heater down position). All oscillations stopped at horizontal orientation. This can be explained as follows:

3.2.1. Vertical operation

In the vertical orientation, gravitational body force acts on each plug and bubble (effect on bubble is negligible). In a static condition, the probability that sections X–C–Y and section X–F–Y (refer Fig. 2) have exactly the same volume fraction (or the mass fraction) of the respective phases is extremely rare. Moreover, the individual lengths of the plugs will also vary in these sections. In addition, it is evident that the summation of static pressure by traversing once along the entire loop must be zero. The above facts necessarily suggest that the menisci geometry of individual bubbles must be different so as to satisfy both the above conditions.

Since the mass distribution (and lengths) of individual phases is different in sections X–C–Y and X–F–Y, respectively, the dynamic pressure drop (or force) required to push the fluid by a small amount in anti-clockwise direction (X–C–Y–F–X) is different than in the reverse direction (X–F–Y–C–X). This is true even if the condition of no dynamic contact angle hysteresis is assumed and only the net effects of wall shear stress and gravity are considered. As a

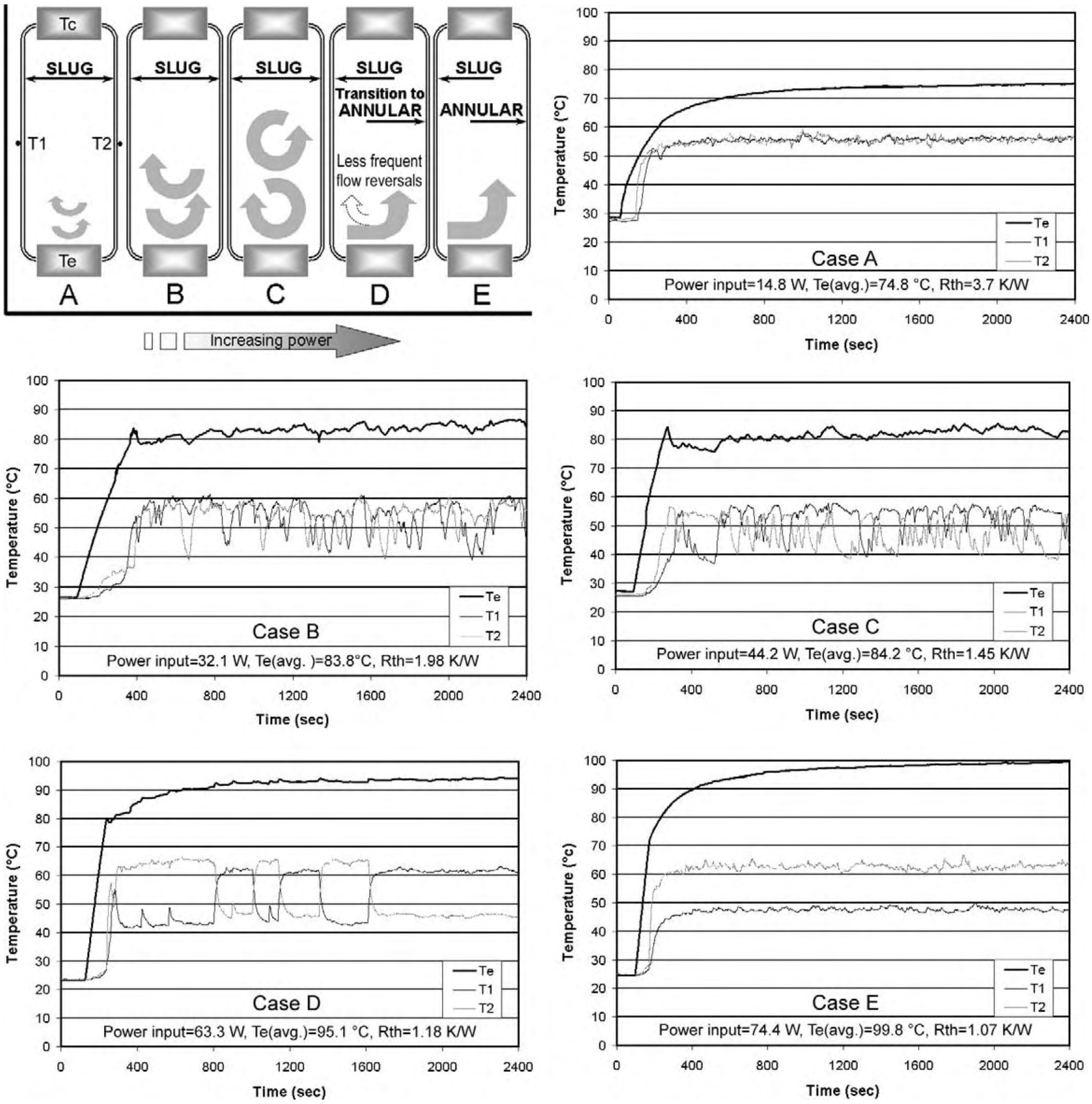

Fig. 3. Observed flow patterns in the loop and corresponding temperature profiles.

bubble forms in the evaporator and expands, a preferential direction of motion is automatically set depending on the path of least resistance (say X–F–Y–C–X). This explains the start-up direction of the two-phase loop. After the start-up, the section at the outlet of the evaporator (A–F–E) has a higher vapor volume fraction than the other loop section (D–C–B) because of evaporation and condensation processes. The process continues for at least some finite time before a combination of interfacial waves, perturbations, internal inhomogeneity of the heating/cooling process and continued non-equilibrium metastable conditions cause a flow direction reversal. The analysis of facts leading to a flow reversal phenomenon needs further investigation.

3.2.2. *Horizontal operation*

In horizontal orientation, in the absence of gravity, the static pressure distribution is quite different from that of the vertical case. This is highlighted in Fig. 4(a), (b) where static pressure distribution of three different cases is compared (system comprising of pure ethanol in two phases, as shown, under isothermal conditions of 20 °C). Although the two sections X–C–Y and X–F–Y will have different mass distributions, as in the vertical case, this does not necessitate the contact angles of the various menisci to be different for maintaining the static pressure integral to be zero across the loop. If a bubble in the evaporator has to expand, apparently in the absence of any external/internal perturbations, dynamic contact angle hysteresis, etc. there seems to be no preferred direction of least resistance. The first set of experimental results with a two-phase loop clearly demonstrates that horizontal operation without gravity is not possible. Several results from other sources [1,2] for a multi-turn CLPHP suggest that horizontal operation is possible albeit not as good as vertical. In our earlier studies with multi-turn CLPHPs too, proper horizontal operation was hardly observed [8]. These apparently uncomplimentary and

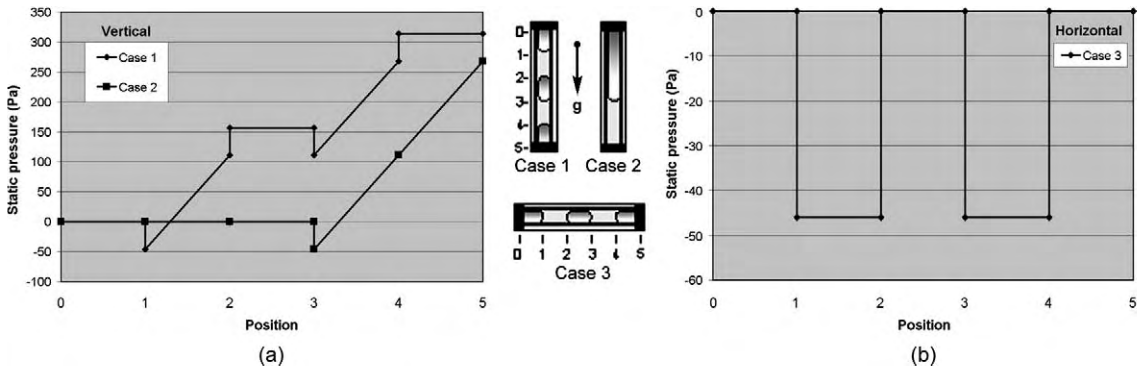

Fig. 4. Static pressure distribution in different cases for ethanol bubbles suspended in liquid ethanol at 20 °C. (a) Cases 1 and 2—Vertical orientation. (b) Case 3—-Horizontal orientation.

contradictory results seem to suggest that reasons for proper horizontal operation may be attributed to (a) more number of CLPHP turns, which may be responsible for higher degree of internal perturbations and inhomogeneity of the system, (b) a high input heat flux leading to higher internal operating pressure. The second point is supported by the fact that even for vertical operation, there is a critical minimum input heat flux requirement to initiate self exited oscillations. In the absence of gravity this minimum critical heat flux is likely to be higher. This fact is yet to be experimentally demonstrated.

3.3. *Thermodynamic considerations*

There exist pressure differentials in the system caused by both, expanding bubbles in the evaporator and contracting bubbles in the cooler. While the system as a whole is isochoric having no associated PdV work, local control volumes in the evaporator and condenser are essentially involved with work interactions, having reverse signs, with adjoining fluid particles. Heat addition along with ‘positive pumping’ by the expanding bubbles is taking place in the evaporator. In the condenser, the bubbles collapse giving up the heat and in turn do a ‘negative pumping’ work on the adjoining fluid particles.

When bulk movement of the fluid is taking place, in general, the quality of the two-phase mixture coming out of the condenser is certainly inferior to that existing at the evaporator outlet. Simultaneously, the fluid pressure is also lower at the condenser outlet. So, considering the P–h diagram as shown in Fig. 5, although exact positions are not known, relative locations of the state of fluid at condenser and evaporator outlet are certainly known (it is emphasized that these are strictly instantaneous positions). In the two adiabatic fluid transport sections, an isenthalpic pressure drop seems to be a very satisfactory assumption. By applying this, the condenser inlet and evaporator inlet points are also known. What happens inside the evaporator and the condenser remains to be fixed to complete the qualitative description of the possible instantaneous thermodynamic cycle of the loop. The simultaneous heating up and pressurizing of the fluid in the evaporator is a rather complex process. For analysis

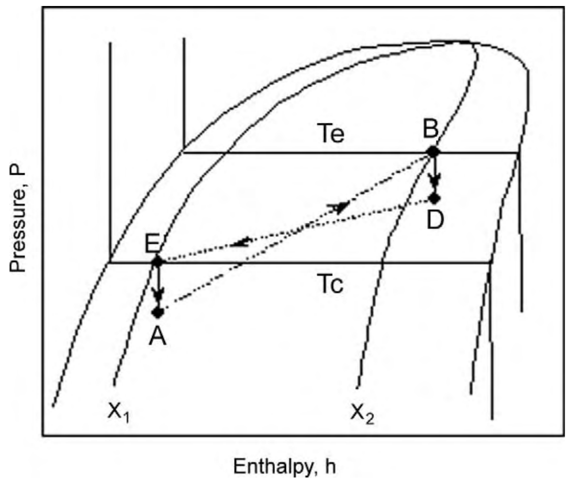

Fig. 5. Instantaneous thermodynamic cycle of the single loop (for notations refer Fig. 2).

it may be conveniently subdivided into two thermodynamic processes, constant pressure heat addition and isentropic pumping up by the bubbles. Similarly, in the condenser, constant pressure condensation is coupled with negative isentropic work. How these complex processes are linked so as to join the dashed lines in Fig. 5 is yet to be determined.

The fact that the overall system volume is fixed, provides another interesting thermodynamic aspect. Operating the device at any temperature (within reasonable limits and avoiding near critical operation), necessarily does not alter the volumetric fill ratio; only the vapor density varies with the operating temperature. So, if the evaporator and condenser temperatures are fixed (thereby fixing the respective working pressures), a given system with a fixed volumetric fill ratio will have a fixed corresponding mass quality at the two respective temperatures. Fig. 6(a) plots the volumetric ratio of vapor against the mass quality of vapor for ethanol at 25 and 100 °C. It is clearly seen that the overall mass quality of vapor phase is extremely low for the range of applicable volumetric ratios of vapor. This also explains the minuscule role of latent heat in the overall heat transfer. Referring to the P–h diagram in Fig. 6(b), the shaded gray area lying very near to the liquid saturation line can thus be marked.

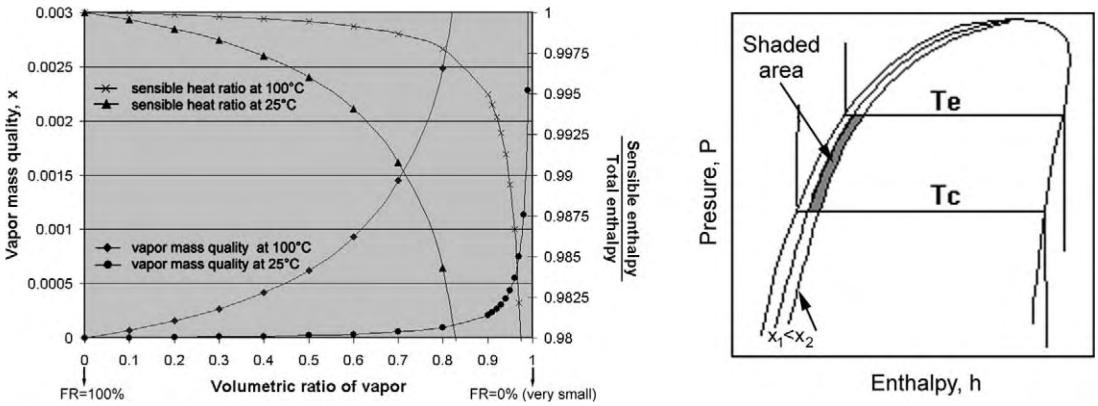

Fig. 6. Volumetric ratio of vapor to vapor mass quality in the operating range of a pulsating heat pipe highlighting the effect of sensible heat transfer to total enthalpy.

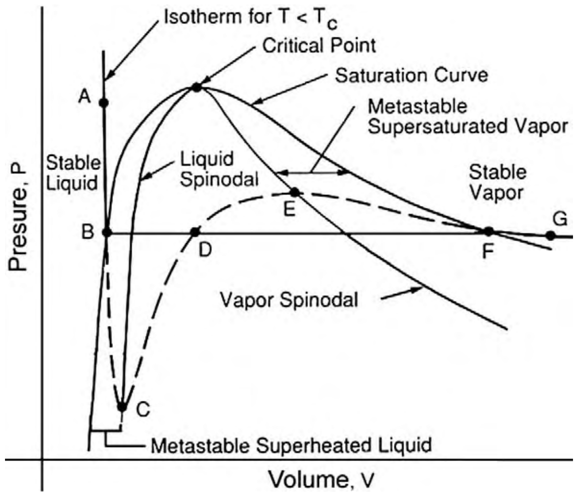

Fig. 7. Representation of metastable region on a P–V diagram (adapted from [9]).

This zone essentially represents all the points in which the average mass quality, integrated over the whole loop, at any given instant of time, must lie. Individual mass qualities, for example, at the evaporator or condenser exit will be different but the instantaneous average over the system must lie somewhere within this shaded zone.

While classical thermodynamics assumes phase transitions as quasi-equilibrium processes at equilibrium saturated conditions, all real processes necessarily occur under non-equilibrium conditions. The degree of shift from the idealized equilibrium depends on the applied boundary conditions. The general functional characteristics and boundary conditions with strong thermo-hydrodynamic coupling in CLPHPs (and so also in the single two-phase loop), are in itself sure indicators of the existence of metastable conditions. Fig. 7 shows a typical P–V diagram with ‘real’ isotherms (in contrast to ideal straight isotherms in the vapor dome) and respective spinodal limits of metastable liquid and vapor phases. The liquid and vapor in the metastable regions, B–C and E–F, respectively, does not violate the criterion of mechanical stability and can indeed exist albeit not in thermodynamic equilibrium. The portion of the curve C–E is interpreted as being inaccessible to the system because this violates the criterion of mechanical stability [9]. Looking towards the self-sustained thermally driven oscillations of a CLPHP, it may be argued that these oscillations are indeed a manifestation of existing real time metastable system parameters. In light of these facts, search for an ideal thermodynamic cycle which may possibly represent the CLPHP action may seem to be a futile exercise. Nevertheless, such an idealized cycle will certainly enhance the basic understanding of the thermo-fluidic characteristics of the device.

The thermo-hydrodynamics of a CLPHP involves various transient processes. These processes on a time averaged basis give the average heat transfer coefficient of the device. The phase change phenomenon in the evaporator tube may be considered as a ‘chemical reaction’ in which the liquid particles are continuously disappearing into vapor molecules with a certain ‘rate of reaction’ governed by the input heat flux, latent heat of vaporization and the geometry of the evaporator. Thus, drawing an analogy with the Damköhler number of group I in chemical kinetics, the mass rate of vapor generation per unit volume (rate of reaction under thermodynamic equilibrium conditions) is given by:

$$\Gamma_{\mathrm{vap}}|_{\mathrm{eq}} = \left(\frac{\dot{q}''_{\mathrm{w}} \cdot \xi_{\mathrm{h}}}{A \cdot h_{\mathrm{fg}}}\right) = \left(\frac{4 \cdot \dot{q}''_{\mathrm{w}}}{D_{\mathrm{hyd}} \cdot h_{\mathrm{fg}}}\right) \tag{1}$$

This ‘rate of reaction’ may be divided by the local species mass concentration, i.e., vapor phase density to give the frequency of phase change,

$$\Omega_{\mathrm{eq}} = \left(\frac{\Gamma_{\mathrm{vap}}|_{\mathrm{eq}}}{\rho_{\mathrm{vap}}}\right) \tag{2}$$

Similar analysis done by Saha et al. [10] leads to equilibrium phase change number defined as,

$$N_{\mathrm{pc}}|_{\mathrm{eq}} = \Omega_{\mathrm{eq}} \cdot \left(\frac{L}{v_{\mathrm{liq,in}}}\right) \tag{3}$$

In their analysis, Saha et al. have explored the stability of vaporizing flows with respect to density wave oscillations in circular channels. While the general phenomenological conclusion of their perturbation analysis fits well to the present case of pulsating heat pipe, the inlet velocity of flow which was explicitly known in their case is unavailable for CLPHP analysis. Also, in contrast, the inlet to the evaporator is typically always under two-phase flow conditions in a CLPHP and is different and changing for each evaporator U-turn. Therefore, in general, any non-dimensional number having the explicit inclusion of the velocity is not suitable for CLPHP analysis as this quantity is not known *a priori*. Furthermore, the level of fluid inlet subcooling to the evaporator section, which is known to affect the stability criteria of two-phase flows, is also unknown (and variable!) in the present case. Nevertheless, the equilibrium frequency of phase change may provide a convenient time scale for future analysis.

3.4. Effect of number of turns

3.4.1. Level of perturbations

A complete stop-over of all macro movements inside the loop has been observed many times in the present range of experiments. This happened more frequently for FR $< 50\%$ coupled with low heat input power. Sometimes the stop-over phenomenon has also been observed for higher filling ratios. The 'self-sustained' oscillating character is then lost (refer Fig. 8).

Such a behavior has never been reported for multi-turn CLPHPs, under comparable boundary conditions. Although, in a multi-turn CLPHP having a working fluid with low $(\partial P/\partial T)_{\mathrm{sat}}$, like water, and at comparatively low heat input fluxes, flow visualization has indicated that there

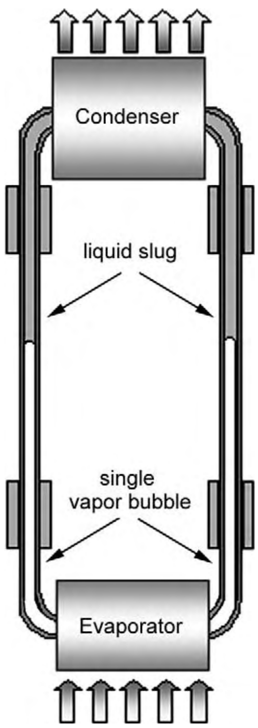

Fig. 8. Stop-over phenomenon in the two-phase loop.

are alternating periods in which bubble plugs are moving rapidly (activity phase) and 'stopping' for a while (static phase). In the static phase there is only micro movement of bubbles with high frequency/low amplitude about a mean position whereas in the activity phase they vigorously move with higher amplitudes along the tube length. As the input heat flux increases it becomes more and more difficult to distinguish between the two phases since the time period of the 'static' phase drastically reduces [8].

Typical behavior of the loop under the conditions which lead to a complete stop-over is as follows: The initial partial filling of the loop leads to a natural volumetric (mal-)distribution of phases in the two tube sections. As the heating power is switched on, the system starts to oscillate in the usual manner. If the combination of boundary conditions is favorable for a stop-over, bubble agglomeration takes place leading to the formation of a single large bubble which envelopes the entire evaporator section (refer Fig. 8). Then oscillations die down completely and all macro motion of the fluid inside the tube stops leading to an increase of the evaporator temperature. As the number of turns keeps increasing, the probability of such a tendency towards complete stop-over should essentially diminish.

3.4.2. Optimum number of turns

Consider that bare cooper rods of a fixed size are fitted to transport the heat and thereby cool the heater with a fixed input power. If only one copper rod is used, after the initial transient phase, the heater will come to some steady state temperature. As the number of copper rods is increased, since the net heater power ($\dot{Q}$) is fixed, the final steady state temperature of the heater will come down. Net heat handled per rod will decrease and so the overall system thermal resistance, i.e., $R_{\mathrm{th}} = (T_{\mathrm{e}} - T_{\mathrm{c}})/\dot{Q}$, keeps decreasing as the number of copper rods increases. The limit is reached as per the available space in between the heater and the cooler so that, in the limit, all the copper rods together consume the entire available space. If the same system is cooled by a pulsating heat pipe, then, in this case, instead of bare copper rods there are copper pipes partially filled with a working fluid. Although a lot of conductive material is removed, the intention is to augment the heat transfer by internal self-sustained thermally driven convective two-phase flow. In a CLPHP, the heat transfer primarily takes place due to liquid convection (latent heat transfer through the vapor helps the bubble liquid pumping action, and thereby the sensible heat transfer), provided the CLPHP is optimally operating in the true pulsating regime.

If the number of turns of the CLPHP is small, then the heat handled by each turn will be quite high. It is obvious that since there is not enough fluid inventory and effective thermal cross sectional area available for heat transfer, the overall thermal resistance will be high. If the number of turns of the CLPHP is now increased (keeping the filling ratio constant), as previously done with pure copper rods, what should be the effect on the overall thermal resistance of the device? Extrapolating the previous analogy, it is clearly seen that provided the heater power is fixed, as the number of turns of the CLPHP is increased, the net heat handled by each CLPHP turn reduces. If we assume for the time being that the effective thermal conductivity (λ_{CLPHP} as defined by $\dot{Q} = \lambda_{\mathrm{CLPHP}} \cdot A_{\mathrm{tot}} \cdot (T_{\mathrm{e}} - T_{\mathrm{c}})/L_{\mathrm{CLPHP}}$) of the CLPHP is constant, then, as for the previous case of bare copper rods, the overall thermal resistance should come down. But the effective CLPHP thermal conductivity, λ_{CLPHP}, is not really constant. In fact, it is dependent on the internal pressure fluctuations, flow patterns and on the existing temperature gradients.

Any decrease in the temperature of the evaporator actually reduces the local saturation pressure. If the condenser temperature is maintained constant then this primarily causes a reduction in overall pressure differential existing between the condenser and the evaporator thereby reducing the driving potential. Thus, a monotonous increase of number of turns of a CLPHP will not have the same effects as in the case of bare solid copper rods. The effective thermal conductivity of a CLPHP is a strong function of the temperature differential existing between the evaporator and the condenser (apart from filling ratio and flow patterns). Therefore for a given heat throughput requirement, an optimum number of turns must exist after which the pulsating effect of the fluid inside, and the heat transfer advantage thereof, will start to diminish.

4. Summary and conclusions

The following facts summarize the essential aspects of the present study:

(1) A two-phase loop, primary building block of a CLPHP, was constructed to phenomenologically study the internal thermo-fluidics, so as to compare it with the real multi-turn CLPHPs.
(2) Two-phase dynamic instabilities, as observed in multi-turn CLPHP are also observed in a two-phase loop. Strong thermo-hydrodynamic coupling leads to meta-stable fluid conditions.
(3) Gravity does affect the thermal performance, at least for systems with low number of turns.
(4) A complete stop-over is observed in the loop but has never been reported in a multi-turn CLPHP device suggesting that the number of turns increases the level of perturbations. Also, for a given heat throughput requirement, an optimum number of turns exists.
(5) Apart from a multitude of geometrical, physical and operational variables which affect the system, the performance is also strongly linked with the flow patterns existing inside. The contribution of latent heat to the overall heat transfer has to be judged in the background of this new fact.

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