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Assessment of Thermal Performance of Closed Loop Pulsating Heat Pipe: Review

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***Abstract* -In this paper, the effect of different orientation and an internal diameter of pipe on the thermal performance of Closed Loop Pulsating Heat Pipes (CLPHPs) has been studied experimentally. Simultaneously other major parameters that affect the system dynamics include the volumetric filling ratio of the working fluid, input heat flux and the total number of turns has been observed. An experimental set-up of CLPHPs made of copper tubes of different internal diameters heated by constant temperature water bath and cooled by constant temperature water-ethylene glycol mixture (50% each by volume) is used to study the thermal performance with the fixed number of turns in the evaporator. The working fluids employed are water, ethanol, and R-123. The results indicate a strong influence of gravity and the number of turns on the performance. The thermo-physical properties of working fluids affect the performance which also strongly depends on the boundary conditions of pulsating heat pipe operation.**

***Keywords* – Closed Loop Pulsating Heat Pipes (CLPHP), Thermal Performance, Internal Diameter, Orientation**

I. INTRODUCTION

The heat pipe has been, and currently is being studied for a wide variety of applications, covering almost the complete spectrum of temperature encounter in the heat transfer processes. In general, the application comes within a number of broad groups, each of which describes a property of the heat pipe. These groups are:

- Separation of heat source and sink
- Temperature flattening, or is thermalisation
- Heat flux transformation
- Temperature control
- Thermal diodes and switches.

These new forms of heat pipes are able to transfer significant heat flows and can increase heat transport length, they remain very sensitive to spatial orientation relative to gravity. To extend the functional possibilities of two-phase systems to applications involving otherwise inoperable slopes in gravity, the advantages provided by the spatial separation of the transportation line and the usage of non-capillary arteries are combined in a loop scheme.

This scheme allows heat pipes to be created with higher heat transfer characteristics while maintaining normal operation in any directional orientation. The loop scheme forms the basis of the physical concept of Two-Phase Loops (TPLs). This range of devices is projected to meet all present and possibly future specific requirements of the electronics cooling industry, owing to favourable operational characteristics coupled with relatively cheaper costs. Pulsating (or oscillating) heat pipes are passive two-phase cooling devices made from capillary-sized tubing that meanders in a closed-loop or open-loop channel pattern.[1]

Oscillating, loop type or pulsating heat pipes (PHPs) are relatively new type of heat transfer devices, which may be classified in a special category of heat pipes. They have been introduced in the mid-1990s. The first predecessor of the family of PHPs appeared in the 1990s, a few examples of which are shown in Fig. 1. The basic structure of a typical pulsating heat pipe consists of meandering capillary tubes having no internal wick structure. It can be designed in at least three ways: (i) Open Loop System, (ii) Closed Loop System and (iii) Closed Loop Pulsating Heat Pipe (CLPHP) with additional flow control check valves, as shown in Fig. 2.[2]

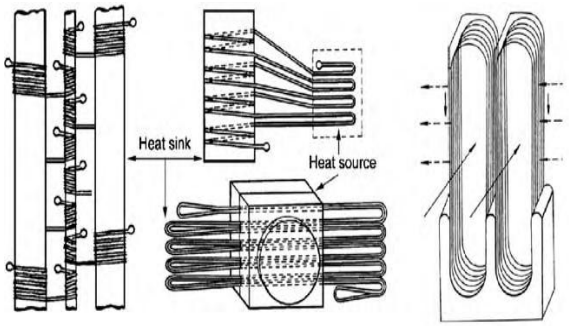

Fig.1. Oscillating, Loop Type or Pulsating Heat Pipes (PHPs).

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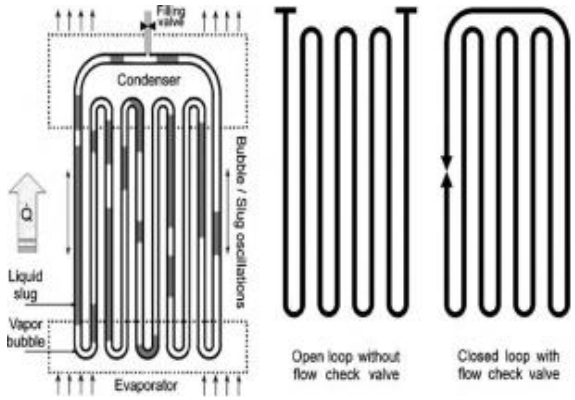

Fig.2 Closed Loop Pulsating Heat Pipe (CLPHP) with additional flow control check valves.

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. These include: i) Internal diameter of the PHP tube, ii) Input heat flux to the device, iii) Volumetric filling ratio of the working fluid, iv) Total number of turns, v) Device orientation with respect to gravity, and vi) Working fluid thermo-physical properties.[3]

The closed passive system formed is evacuated and subsequently filled up partially with a pure working fluid. The optimum quantity of working fluid needed depends on various parameters and is still an area of research. The entire essence of thermo-mechanical physics lies in the closed (constant volume), two-phase, bubble–liquid slug system formed inside the tube-bundle due to the dominance of surface tension forces. This tube-bundle receives heat at one end and is cooled at the other.

Temperature gradients give rise to temporal and spatial pressure disturbances in the wake of phase change phenomena (bubble generation and growth in the evaporator and simultaneous collapse in the condenser). The generating and collapsing bubbles act as pumping elements transporting the entrapped liquid slugs in a complex oscillating–translating–vibratory fashion; a direct consequence of thermo-hydrodynamic coupling of pressure/temperature fluctuations with the void fraction distribution.

This causes heat transfer, essentially as a combination of sensible and latent heat portions. The relative magnitude of these portions is also of profound interest and decisive to the overall thermal performance of the structure. It has been indicated earlier that the sensible heat transfer is the major contributor in the overall heat exchange. Many studies have indicated that after a certain input heat flux, the bubble–liquid slug flow may break down into annular flow regime. The relative magnitude of sensible and latent portions thus changes and is dependent on the flow pattern existing inside the tubes. This aspect is another critical area that requires further investigations. Therefore, this paper mainly focuses on the performance limit of CLPHPs without check valve.

II. PULSATING HEAT PIPES

All the family of structures similar in phenomenological and thermo fluid dynamic operation to the structure as depicted in Figure 1 will be henceforth referred to as Pulsating Heat Pipes (PHP). These structures are characterized by the following basic features:
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:
1. Open Loop- Tube ends are not connected to each other.
2. Closed Loop- Tube ends are connected to each other in an endless loop.

- There is no internal wick structure as in conventional heat pipes.
- At least one heat receiving (evaporator/heater) zone is present.
- At least one heat dissipating (condenser/cooler) zone is present.
- 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 vapour 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 vapour system.

A PHP is essentially a non-equilibrium heat transfer device driven by complex combination of various types of two phase flow instabilities. The performance success primarily depends on the continuous maintenance or sustenance of these non-equilibrium 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 the fluid transport.

The driving pressure pulsations are fully thermally driven.Although many of the arguments and discussions that follow in this chapter and thereafter throughout the text apply to both open and closed loop systems, attention is primarily focused only on CLPHPs, the principal candidate of present scrutiny.

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III. PULSATING HEAT PIPE OPERATING CHARACTERISTICS

A given PHP has two operational extremities with respect to filling ratio (liquid volume / total PHP volume) i.e., 0% filled or an empty device and 100% filled equivalent to a single-phase the rmosyphon. It is obvious that at 0% fill ratio, a PHP structure with only bare tubes and no working fluid, is a pure conduction mode heat transfer device and obviously has a very high undesirable thermal resistance. A 100% fully filled PHP is identical in operation to a single phase thermo siphon. Since there exist no bubbles in the tube, 'pulsating' effects are obviously non-existent but substantial heat transfer can take place due to liquid circulation in the tubes by thermally induced buoyancy. In between these two limits the device functions in a pulsating mode. In this pulsating operational mode, there exist three distinct regions:

- Nearly 100% fill ratio: In this mode there are only very few bubbles present, the rest being all liquid phase. These bubbles are not sufficient to generate the required perturbations and the overall degree of freedom is very small. The buoyancy induced liquid circulation, which was present in a 100% filled PHP, gets hindered due to additional surface-tension-generated friction of the bubbles. Thus, the performance of the device is seriously hampered, and the thermal resistance much higher than for the 100% filled PHP.
- Nearly 0% fill ratio: In this mode there is very little liquid to form enough distinct slugs and there is a tendency towards dry-out of the evaporator. The operational characteristics are unstable and undesirable.
- PHP true working range: Between about 10% to 90% fill ratio the PHP operates as a true pulsating device. The exact range will differ for different working fluids, operating parameters and construction.[4]

It can be clearly inferred that the underlying physics guiding the mechanism of heat transfer in a PHP is drastically different in the different modes of operation as described above. The fill ratio and heat load are the chosen to be the two independent variables. Overall thermal resistance is chosen to be the parameter representing the heat pipe's thermal performance.

1. Closed Loop Pulsating Heat Pipe-

As outlined in the above section, the overall understanding of the CLPHP operation was still in the burgeoning stages at the commencement of the present research work. A similar or even worse scenario existed in mathematical modeling of CLPHPs. Indeed, it was a futile exercise to start any modeling activity unless there was a reasonable understanding of the device operation. The extremely limited success (euphemism for µfailure,) of the then existing models was an obvious proof that their developmental foundations either overlooked or oversimplified the strategic and inherent aspects of operational physics of the device. Thus, there was a need of focused experiments to determine the specific operational characteristics of the CLPHPs. The problem at hand involved strong and complex two phase thermo hydrodynamic interactions of the working fluid having decisive implications on the net heat transfer. Therefore, an experiment was investigation to understand the operational phenomena under the influence of various imposed thermo mechanical boundary conditions.

IV. EXPERIMENTAL SET UP

In conventional heat pipes, the adiabatic vapor temperature gives a very convenient way of standardizing an experimental procedure. In contrast, there is no well-defined adiabatic temperature in the case of CLPHPs. Thus, in general, performance testing of CLPHPs may be conducted in two ways:

- controlling the input heat flux and the condenser temperature, in which case the evaporator temperature is a dependent variable and,
- Controlling the evaporator and condenser temperature to give dependent heat throughput.

In the present experimental study, the latter strategy was adopted. Essentially three parameters were fixed at the outset:

- The average evaporator temperature was always maintained at 80˚ C with the help of a large water-cooling bath. The imposed mass flow always insured near isothermal conditions within ±0.5˚C,
- In the condenser, an aqueous solution of ethylene glycol (50% by volume) with inlet temperature always maintained at 20˚ C circulated from a cold bath.
- The filling ratio (working fluid volume inside the device/total internal volume of the device) was always maintained at 50% in all experimental set-ups.

The schematic of an experimental set-up is shown in Fig. 3.

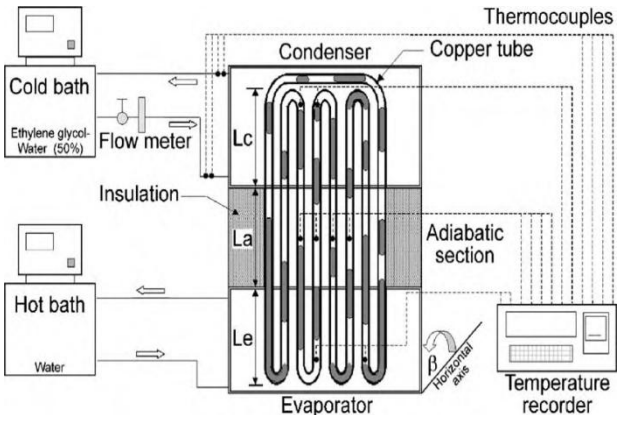

Fig. 3 A schematic of an experimental set-up of CLPHPs.

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It consisted of the tested CLPHPs, the heating and cooling baths, a temperature measuring thermocouples, and a flow meter (to measure the flow rate of the coolant solution). Four chrom alumel thermocouples were used to measure the temperature of the cooling solution, two each at the inlet and outlet sections of the condenser. The heat throughput was thus measured by calorimetric method applied to the condenser-cooling jacket. In addition, two thermocouples on the evaporator tube sections, and two thermocouples on the condenser section completed the instrumentation.

The tested CLPHPs were made of copper capillary tube. Both ends of the tube were connected together to form a closed loop structure which was located in the condenser in all the experiments. The adiabatic section was well insulated with foam insulation. First, the CLPHP was evacuated (10^{-2} Pa) and then filled with 50% of the total volume with the working fluid. The inlet temperature of the hot and cold baths were set at the fixed values, and the hot and cold fluids were supplied to the jackets of both the evaporator and condenser sections. After a quasi-steady-state was reached, the temperatures and flow rate were recorded.

Thus, for a given configuration the heat throughput could be evaluated. Then the influence parameters were varied according to the required conditions. The value of calculated heat transfer rate (Q) was subjected to experimental uncertainties and errors that were later evaluated. The complete experimental matrix is as shown in Table 1.

Table 1 Experimental matrix.

Working fluids	d_i (mm)	L total (m)	Le = La = Lc (m)	N (number of turns)
Water–Ethanol–R-123	2.0	≈5	0.15	5
	2.0	≈5	0.10	7
	1.0	≈10	0.10	16
		≈15	0.15	11
			0.10	23
			0.15	16

ratio always maintained at 50% in all configurations. All configurations tested at inclinations of 0˚ (horizontal) to +90˚ (vertical, evaporator down). The experimental set-up comprises the test specimen with instrumentation which is fixed on a rotary frame plus equipment for heating and cooling of the specimen. The CLPHP was formed from 16 copper tubes with an inner diameter (di) of 2 mm in the temperature range from 35˚ C to 160˚ C. These diameters are greater (di = 2 mm), than the so-called critical diameter which is from about 2 mm at 35˚ C to 160˚ C for R123 filled PHPs, as shown in Fig. 4. The U-turns at one end are embedded into an evaporator block.The remaining part of the CLPHP is cooled by forced air.

Materials used for experimentation are as follows:

- Ethanol Coolant
- R-123
- Thermocouple
- Flow meter
- Water cooling bath
- Hot and cooling bath
- Temperature recorder
- Insulation
- Copper tube

$$Dcrit \approx 2\frac{\sqrt{\sigma}}{\sqrt{g\,(\rho liq - \rho vap)}} \quad (1)$$

where,

D_{crit}: critical diameter, mm
σ: surface tension, N/m
ρ_{liq}: density of liquid, kg/m^3
ρ_{vap}: density of vapour, kg/m^3

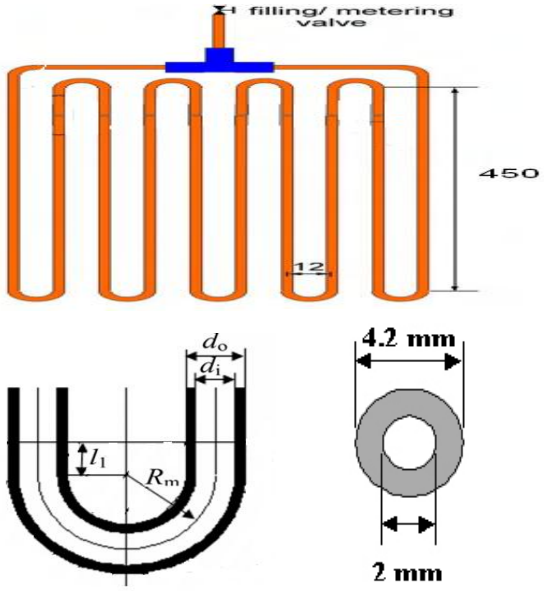

Fig. 4 Design of copper tube.

V. RESULT AND DISCUSSION

1. Effect of Filling Ratio-The effect of input heat flux on various forms of two-phase flow instabilities is well documented. For example, experimental as well as analytical studies on density wave oscillations in single channel two-phase flow have indicated that these are strongly dependent on the heat flux variation; other factors being single and two-phase frictional pressure drop characteristics of the channel, inlet flow rate, level of sub cooling, system pressure and inlet/ exit

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restrictions. In such systems, with respect to input heat flux, the results may be summarized by saying that for a specified non-zero level of inlet sub cooling, increasing the inlet heat flux above a certain limit induces flow instabilities. In the case of CLPHPs, for a defined geometry of the device, the input heat flux is also directly responsible for the type of flow pattern which will exist in the channel, thus affecting the fundamental relaxation instabilities. Furthermore, the static engineering type instabilities are also affected by input heat flux in case of CLPHPs since this directly affects the bubble pumping characteristics. Thus, we may hypothesize, that the operating heat flux will directly affect the level of perturbations inside a CLPHP thereby affecting the thermal performance of the device.

2. Effect of Tube Inner Diameter (ID)-The internal diameter is a parameter which necessarily affects the very definition of a pulsating heat pipe. Beyond a particular limit, all the working fluid will tend to settle down by gravity, and the device will stop functioning as a 'pulsating' heat pipe. It will rather behave like an interconnected array of closed two-phase thermo syphons. The effect of ID for vertically operating devices having Le =10 cm, for a given number of turns, the performance improved with internal diameter. This is realized since there is more mass inventory of working fluid coupled with reduced pressure drop. In general, the entire experimental matrix exhibited this trend.

3. Effect of Operating Orientation-One of the aims of good CLPHP design is to make the thermal performance, as far as possible, independent of the operating orientation. At a first glance, two physical phenomena affect the CLPHP performance with respect to orientation. The first is of course, the effect of gravity on slug flow and the second is the effect of total number of meandering turns on the level of internal temporal and spatial dynamic pressure perturbations. In addition to these two, the input heat flux is also a strong parameter, which affects dynamic instability, especially in density wave oscillations, and is therefore believed to affect the thermal performance of CLPHPs with respect to orientation.

This aspect remains to be further explored and will not be highlighted in this paper. It is to be noted that for performance in vertical orientation, the effect of input heat flux has already been experimentally demonstrated. Classical experiments on the rise velocity of a single bubble in a cylindrical tube have shown that as the Bond number approaches a critical value approximately equal or less than 2, surface tension forces start predominating over gravity forces. There exists a discrepancy in the agreement of this critical limit with some sources quoting slightly different values.This discrepancy is generally attributed to the tube material/working fluid contact angle characteristics, especially the hysteresis phenomenon. I fit is assumed that surface tension is indeed dominating in a particular experimental set-up that satisfies $B \leq 2$, then the shape of a typical slug-bubble element should not change in vertical or horizontal orientation, especially regarding the symmetry of liquid film thickness around the bubble. Although the boundary conditions meet the critical Bond number criterion, the effect of gravity is clearly seen by the unsymmetrical shape of the bubble in the R-123 bubbles are more unsymmetrical as surface tension is still lower. It is also clear that in a non-operating, isothermal, partially filled CLPHP, the static pressure distribution traversing across the tube through the liquid slugs and vapour bubbles is drastically different in vertical and horizontal orientations.

Thus, gravity does affect CLPHP dynamics even though the boundary conditions satisfy the critical Bond number criterion. This has indeed been demonstrated by the experimental results that follow. For di =2:0 mm devices, the effect could be clearly separated into two cases by using a certain critical value of number of turns (Ncrit). In this case, the critical number of turns was approximately 16 turns (with the exception of Le = 15 cm, 16 turns and ethanol as working fluid.

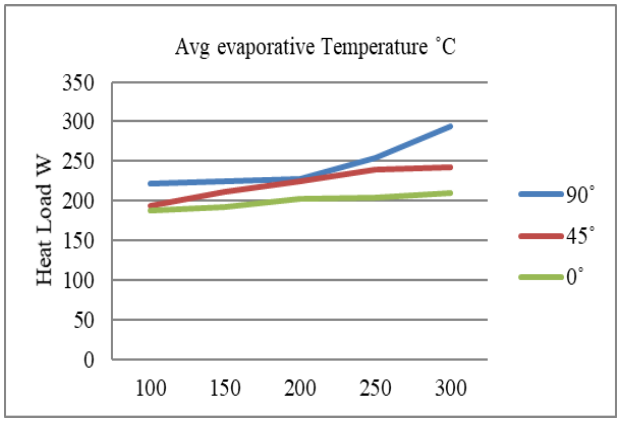

Fig.5 Thermal performance for 0˚, 45˚ and 90˚ tube inclination.

CLPHPs with 2 mm inner diameter operated successfully in all three heat modes and showed excellent performances. The CLPHP with 2 mm ID tubes and 90˚ inclination angle of heat pipe with respective to ground had a higher heat load (W) than 45˚ and 0˚ inclination angle. Concerning the specific performance data, the CLPHP with 2 mm ID tubes achieved about 217 W/cm² for 90˚ condenser temperature and 194 W/cm² for 45˚ condenser temperature and 191 W/cm² for 0˚ condenser temperature, respectively. This heat load variation with different inclination angle is shown in Table II.

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Table 2 Condenser temperatures varied with respective to heat load result.

S. No.	Condenser Temperature in ˚C	Heat Load (W)		
		90˚ Inclination angle	45˚ Inclination angle	0˚ Inclination angle
1	61	217.15	194.94	191.9
2	65	224.22	236.34	195.94
3	76	282.8	242.4	210.08

In general, the CLPHPs obtain the best thermal performance and maximum performance limitation when they operate in the vertical bottom heat mode with 50% filling ratio. As the inner diameter decreases, performance differences due to the different heat modes (i.e. the effect of gravity) become relatively small and even insignificant. The study provides detailed discussion on the various design parameters. Apart from the multitude of geometrical, physical and operational variables, the performance is also strongly linked with the flow patterns existing inside the device. Subtle aspects of this two phase flow dynamics and their interactions with the heat transfer characteristics have been highlighted.

VI. CONCLUSION

A range of closed loop pulsating heat pipes has been experimentally investigated to study the effects of various influence parameters. The effect of internal diameter, operating inclination angle (gravity), working fluid and number of turns on the thermal performance has been demonstrated. The following main conclusions can be drawn from the study:

- Gravity certainly affects the heat throughput. Although the internal diameter of the tubes tested in the present experiment.
- A certain critical number of turns are required to make horizontal operation possible and also to bridge the performance gap between vertical and horizontal operation. This is attributed to the increase in the level of internal perturbations.
- Different fluids are beneficial under different operating conditions. An optimum tradeoff of various thermo physical properties has to be achieved depending on the imposed thermo-mechanical boundary conditions.
- For a given temperature differential, performance improves with increase in internal diameter. The internal diameter is a parameter which necessarily affects the very definition of a pulsating heat pipe.
- It may also be safely concluded that thermo-mechanical interactions and instabilities in a pulsating heat pipe in particular, and in capillary sized tubes (mini-micro channels) in general, is quite complex and further experiments are indeed needed. The fact that pulsating heat pipes 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.

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