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Closed loop pulsating heat pipes Part A: parametric experimental investigations

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Abstract

Closed loop pulsating heat pipes (CLPHPs) are complex heat transfer devices having a strong thermo-hydrodynamic coupling governing the thermal performance. In this paper, a wide range of pulsating heat pipes is experimentally studied thereby providing vital information on the parameter dependency of their thermal performance. The influence characterization has been done for the variation of internal diameter, number of turns, working fluid and inclination angle (from vertical bottom heat mode to horizontal orientation mode) of the device. CLPHPs are made of copper tubes of internal diameters 2.0 and 1.0 mm, heated by constant temperature water bath and cooled by constant temperature water–ethylene glycol mixture (50% each by volume). The number of turns in the evaporator is varied from 5 to 23. The working fluids employed are water, ethanol and R-123. The results indicate a strong influence of gravity and number of turns on the performance. The thermophysical properties of working fluids affect the performance which also strongly depends on the boundary conditions of PHP operation. Part B of this paper, which deals with development of semi-empirical correlations to fit the data reported here coupled with some critical visualization results, will appear separately.

Keywords: Pulsating heat pipe; Parametric experimental study

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Nomenclature

D	tube diameter, m
g	acceleration due to gravity, m/s^2
L	length, m
N	number of turns
$\dot{Q}$	heat transfer rate, W
β	inclination angle from horizontal axis

Subscripts

a	adiabatic section
c	condenser section
crit	critical value
e	evaporator section
i	inner
max	maximum

1. Introduction

Oscillating, loop type or pulsating heat pipes (PHPs) are a 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 [1–4], 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. The closed passive system thus 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 [5,6]. 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

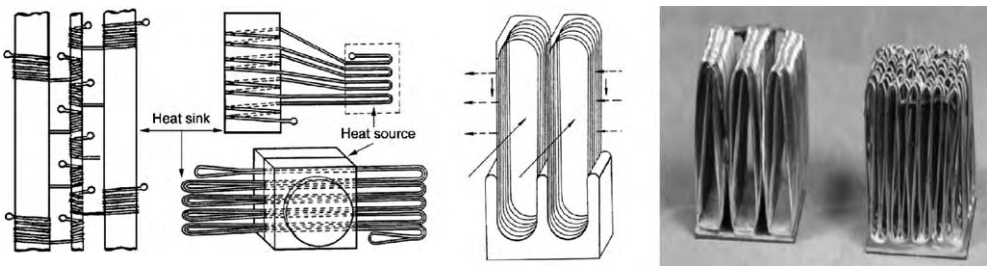

Fig. 1. Some practical designs of pulsating heat pipes (adapted from [1–4]).

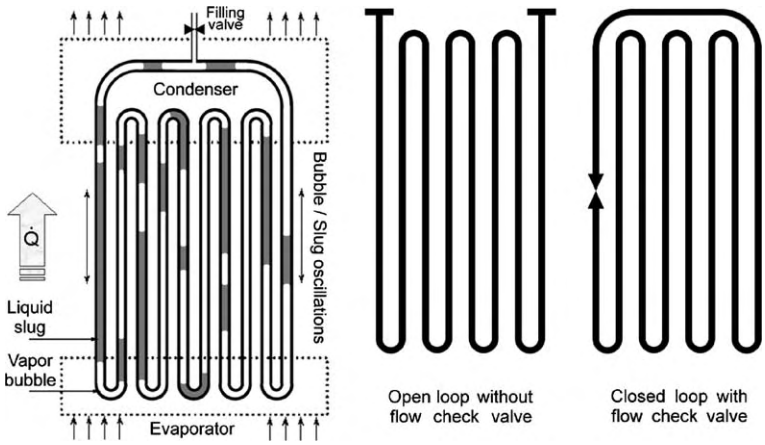

Fig. 2. Schematic of a pulsating heat pipe and its design variations.

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 (mal-) 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 [6–8]. Further studies have indicated that after a certain input heat flux, the bubble–liquid slug flow may break down into annular flow regime [9]. 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.

It has been shown by previous studies that a closed loop pulsating heat pipe is thermally more advantageous than an open loop device because of the possibility of fluid circulation. Although a certain number of check valves have shown to improve the performance, miniaturization of the device makes it difficult and expensive to install such valve(s) [3,10]. Therefore, a closed loop device without any check valve(s) is most favorable from many practical aspects. Studies (mostly qualitative) have already identified various design parameters affecting the performance of CLPHPs [6,11]. This paper presents results of a large experimental matrix aimed at better understanding the quantitative parameter dependency of CLPHPs.

2. Experimental setups and procedure

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: (i) controlling the input heat flux and the condenser temperature, in which case the

evaporator temperature is a dependent variable and, (ii) controlling the evaporator and condenser temperature to give dependent heat throughput. In the present experimental plan, the latter strategy was adopted. Essentially three parameters were fixed at the outset: (a) the average evaporator temperature was always maintained at 80 °C with the help of a large water cooling bath (HAAKE, 8N3-B, and ±0.05 °C accuracy). The imposed mass flow always insured near isothermal conditions within ±0.5 °C, (b) 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 (HAAKE, N6-C41, and ±0.05 °C accuracy) and (c) 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 experimental set-up is shown in Fig. 3. It consisted of the tested CLPHPs, the heating and cooling baths, a temperature data logger (Comark, C8510, 10 channels, overall accuracy ±0.5 °C) and a flow meter (Platon, PGB411 with accuracy of 0.1 l/min) to measure the flow rate of the coolant solution. Four chrom–alumel thermocouples (OMEGA-Type 'K') 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, four thermocouples on the adiabatic 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 (Armaflex).

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 Q was subject to experimental

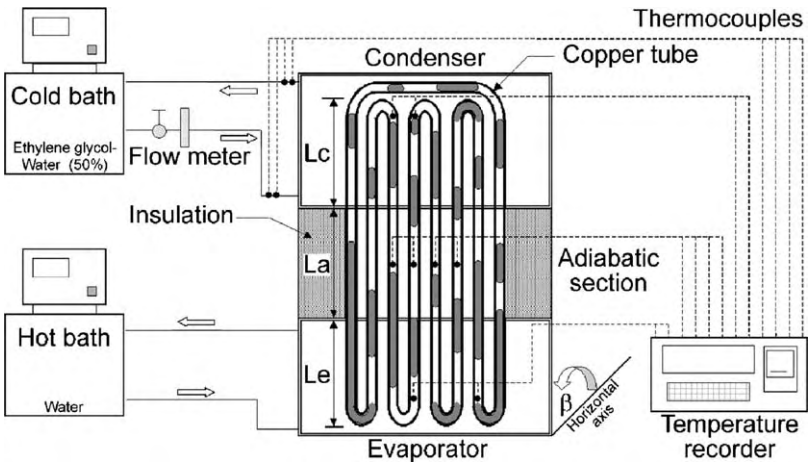

Fig. 3. Details of the experimental set-up.

Table 1
Complete experimental matrix

Working fluids	D_i (mm)	L_{total} (m)	$L_e = L_a = L_c$ (m)	N (number of turns)
Water–ethanol–R-123	2.0	≈5	0.15	5
	2.0, 1.0	≈5	0.10	7
		≈10	0.10	16
			0.15	11
		≈15	0.10	23
			0.15	16

Fill ratio always maintained at 50% in all configurations
All configurations tested at inclinations of 0° (horizontal) to +90° (vertical, evaporator down)

uncertainties and errors that were later evaluated. The complete experimental matrix is as shown in Table 1.

3. Results and discussion

3.1. Data accuracy

Since the heat transfer rate of the CLPHP is calculated by measuring the volume flow rate and the inlet and outlet temperatures of the coolant flowing through the condenser section, the accuracy of each recorded data is inspected. After carrying out a detailed error analysis with respective accuracy of individual measurements and thermal losses, the data on maximum thermal performance reported here, as a whole, is within ±30% accurate. When the device performance is higher, a measurable temperature gradient exists in the condenser cooling fluid inlet and outlet resulting in an error of less than ±10%. While it may be argued that accuracy of the present reported data is not of excellent category, the essence of thermofluidic characteristics and influence parameter trends could be clearly demonstrated after data reduction.

3.2. 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 [12,13], 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 [14,15].

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 [16,17]. There exists a discrepancy in the agreement of this critical limit with some sources quoting slightly different values, e.g. 1.84 [18]. This discrepancy is generally attributed to the tube material/working fluid contact angle characteristics, especially the hysteresis phenomenon. If it is assumed that surface tension is indeed dominating in a particular experimental set-up that satisfies $Bo \leqslant 2$, 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. Fig. 4 shows the photograph of static ethanol vapor bubbles suspended in liquid ethanol inside $D_i = 2.0$ and 1.0 mm glass tubes, respectively, at room temperature (taken by NIKON Coolpix 5700 Digital Camera). 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 side view (View B). 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 vapor bubbles is drastically different in vertical and horizontal orientations [9]. 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.

The second design aspect is related to the total number of meandering turns in a given CLPHP. Figs. 5 and 6 summarize the thermal performance for the entire experimental matrix, with respect to the inclination angles. For a given case, the performance is scaled by the maximum performance achieved for that case during operation in the full range of inclination angles. It can be clearly seen that the performance independence with orientation is affected by the number of

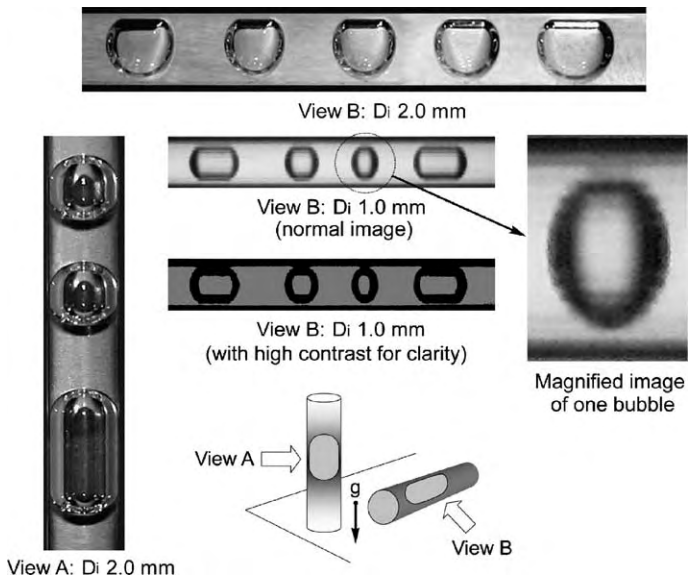

Fig. 4. Images of ethanol slugs and bubbles in glass tube under static isothermal conditions.

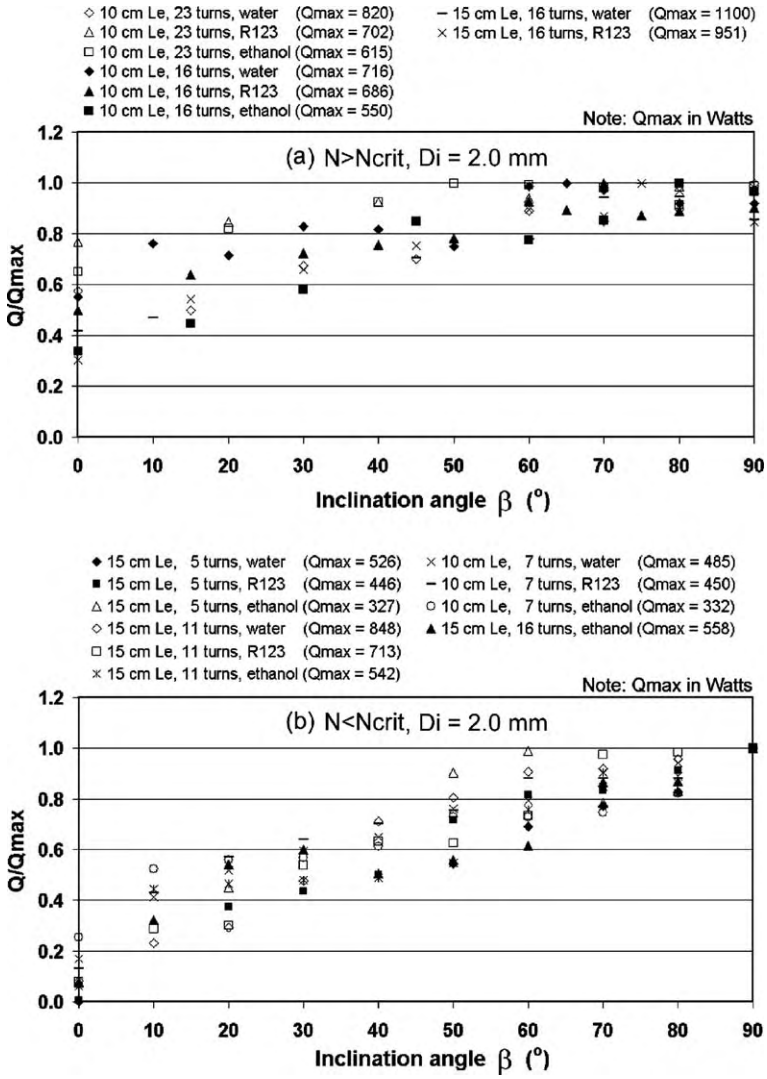

Fig. 5. Thermal performance for $D_i = 2$ mm: (a) $N > N_{crit}$ and (b) $N < N_{crit}$.

turns. For $D_i = 2.0$ mm devices, the effect could be clearly separated into two cases by using a certain critical value of number of turns (N_{crit}). In this case, the critical number of turns was approximately 16 turns (with the exception of $L_e = 15$ cm, 16 turns and ethanol as working fluid). In case of $D_i = 1.0$ mm devices too, similar trends are seen as depicted in Fig. 6(a) and (b). For this case, the critical value of number of turns tends to be higher than for 2 mm tubes. In addition, for 1 mm tubes, measurable heat transfer was not possible with water filled devices in the entire range of operating orientation.

When N is less than a certain N_{crit}, the CLPHP cannot satisfactorily operate in the horizontal orientation and vice versa. For $N < N_{crit}$, the highest thermal performance normally occurs at vertical bottom heating mode decreasing continuously as the device is turned towards horizontal.

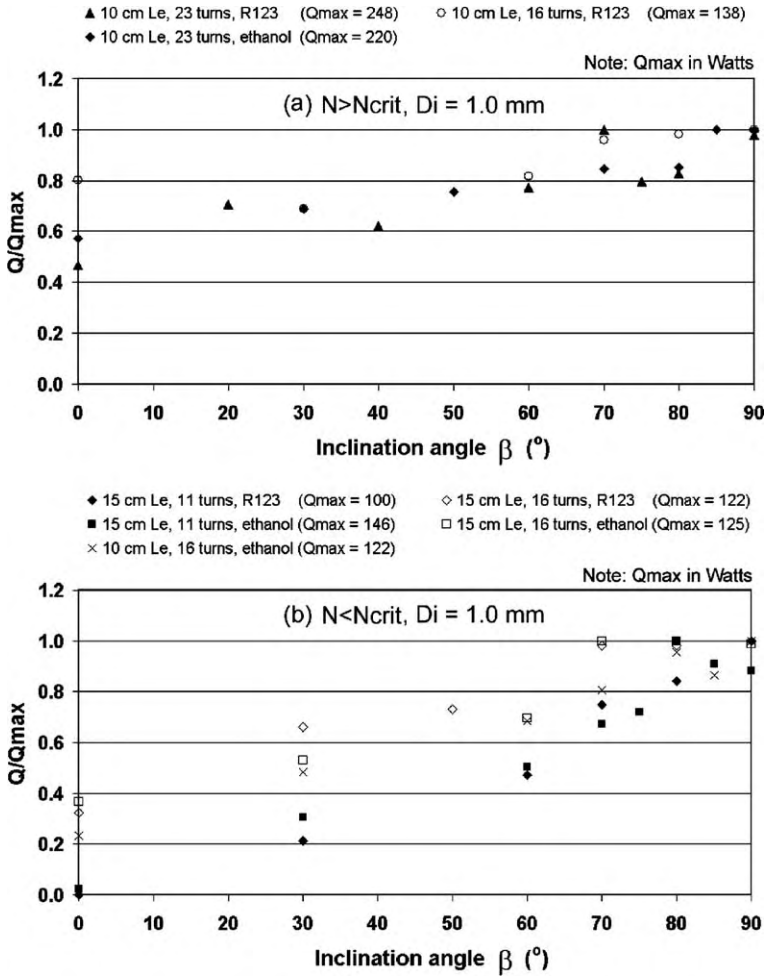

Fig. 6. Thermal performance for $D_i = 1$ mm: (a) $N > N_{crit}$ and (b) $N < N_{crit}$.

However, when the number of turns is higher than the critical value, although the performance improves with increasing the inclination angle from horizontal orientation, the performance remains nearly comparable from vertical position to about 60°. The critical number of turns depends on the working fluid and size of the used capillary tube (and may also depend on the input heat flux, as previously mentioned).

3.3. Effect of construction on performance

From the previous section it is clear that gravity is certainly affecting the thermal performance. The CLPHPs tested in the present work are inline designs in which all the tubes are in one plane (as in Fig. 2). Thus when such a design is made horizontal, the gravity vector is non-existent on all the tubes simultaneously. Fig. 7(a) and (b) suggests two design variations which, only by virtue of the construction and physical arrangement of tubes, should have a favorable effect on thermal

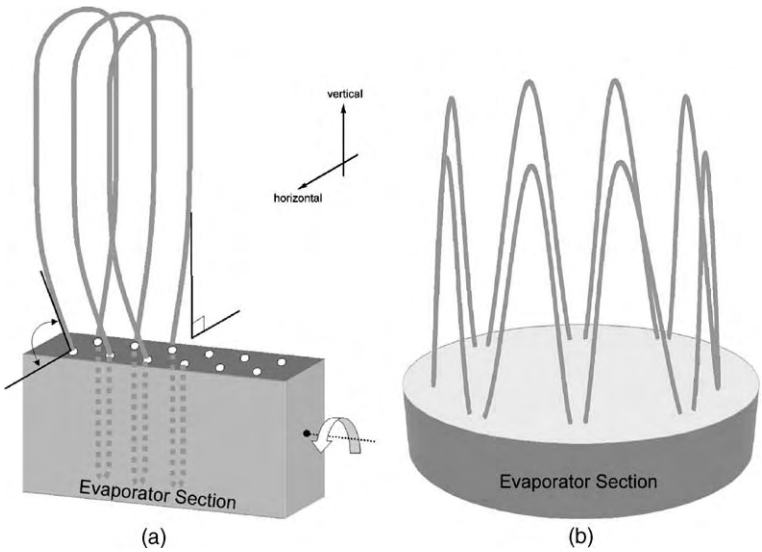

Fig. 7. Design variations for improving performance with respect to orientation.

performance with respect to orientation. Design variation *A* has partially bent tubes turns and so if this structure is operated horizontally, the gravity vector will still be partly functional. In variation *B* the tubes are bent in a three dimensional manner and so the gravity will affect the flow irrespective of any global orientation of the heat pipe. These constructional variations are certainly believed to enhance the performance.

3.4. Effect of tube inner diameter

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 thermosyphons [6]. Fig. 8(a)–(c) shows the effect of tube inner diameter for vertically operating devices having $L_e = 10$ 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. Further, doubling the diameter did not double the performance. At the same internal diameter and evaporator length, the performance is higher with increasing the number of turns. Thus, for a specified temperature gradient between evaporator and condenser, the performance can be increased by increasing the tube inner diameter and/or the number of meandering turns.

3.5. Effect of working fluid

The thermophysical properties of the working fluid coupled with the geometry of the device have profound implications on thermal performance of the device. This affects the following:

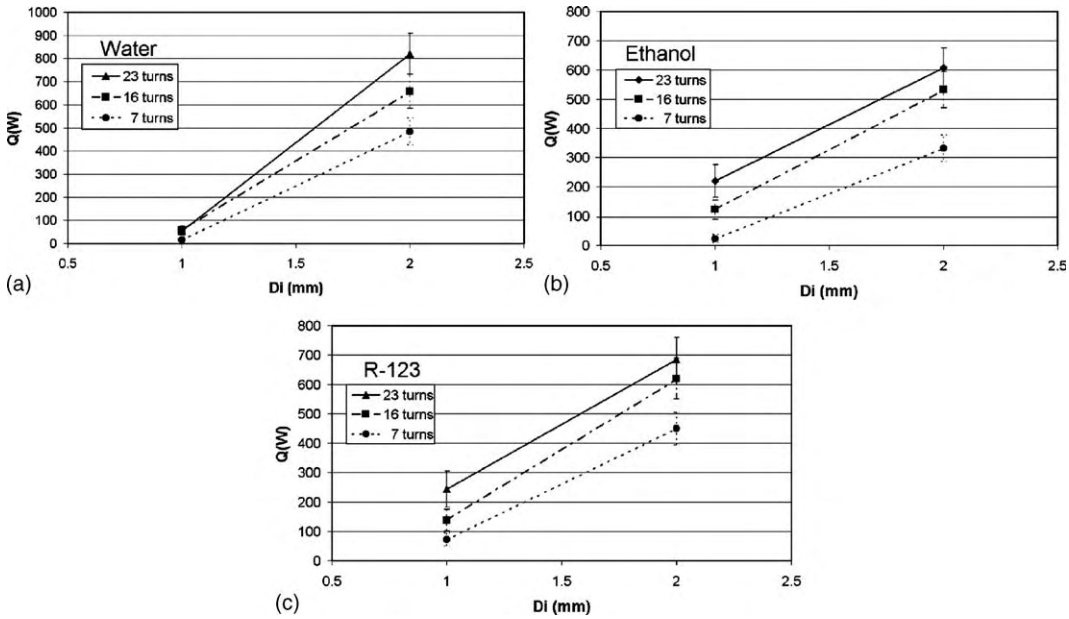

Fig. 8. Effect of diameter on the heat throughput ($L_e = 100$ mm, vertical orientation).

- the relative share of latent and sensible heat in the overall heat throughput;
- the possibility of having different flow patterns in the device, e.g. slug–annular flow regime inter-transition;
- the average flow velocity and overall pressure drop (including effect of gravity);
- bubble nucleation, collapse, shapes, agglomeration and breakage; bubble pumping action, etc.

In vertical orientation for the 2.0 mm devices, water filled devices showed higher performance as compared to R-123 and ethanol. In contrast R-123 and ethanol showed comparable performance in case of 1.0 mm devices with water showing very poor results. This is seen in Fig. 9 and

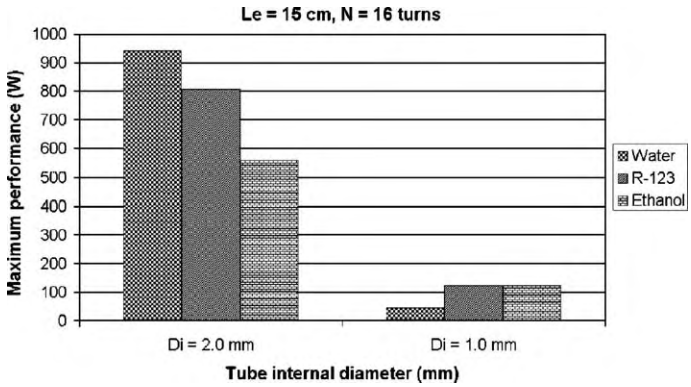

Fig. 9. Effect of working fluid on the thermal performance.

also in Fig. 8. Water has a very high surface tension, a very low $(\mathrm{d}P/\mathrm{d}T)_{\mathrm{sat}}$, a very high latent and specific heat and reasonably higher dynamic viscosity as compared to R-123. Since the thermal performance is a complex combination of the above noted it is certainly difficult to prescribe or proscribe a certain working fluid unless all the boundary conditions are exactly known and individual effects have been explicitly isolated and quantified.

4. Summary and conclusions

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 study, as governed by the critical Bond number, is well within the specified limit, bubble shapes are affected by the buoyancy forces.
- 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 the increase in the level of internal perturbations.
- Different fluids are beneficial under different operating conditions. An optimum tradeoff of various thermophysical 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. This aspect, including semi-empirical modeling approach coupled with critical visualization results is addressed in Part B of this paper [19].

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