Thermal Performance of an Asymmetric Pulsating Heat Pipe with Aqueous Surfactant Solution

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Abstract

An asymmetric closed-loop pulsating heat pipe (aCLPHP) has been studied with aqueous surfactant solutions in vertical bottom heat mode. A dual-loop aCLPHP setup was fabricated using copper capillary tubes of 2.4 and 3.4 mm internal diameters. The aqueous surfactant solution with sodium dodecyl sulphate (SDS) at 500–2500 ppm was prepared to have different properties like surface tension and dynamic viscosity than the pure deionized water. The working fluid filling ratios (FR) of 50% and 60% in 0º (horizontal), 30º, 60º, and 90º (vertical) orientations have been investigated by monitoring the temperature and pressure variations under various heat load patterns. The condenser section was kept at 20 ºC by a coolant supply with a flow rate of 20 kg/h. The maximum thermal resistance decreased by 73.5%, and the heat load range was enhanced by 16.67% with the aqueous solutions in the vertical orientation compared to pure water. Hence, the heat pipe has shown better startup and thermo-hydrodynamic behavior due to improved wettability and low surface tension.

Keywords: Asymmetric pulsating heat pipe; Thermal management; Surfactant solutions; Two-phase flow start-up

1. Introduction

The miniaturization of electronic devices has led to thermal management problems in domestic as well as industrial applications. The generated heat flux necessarily needs to be dissipated to keep the junction or device temperature below the maximum allowable limit [1]. High temperatures can lead to malfunctioning operations or a complete failure due to thermal stress [2]. Device reliability and performance can be improved by designing a multidisciplinary thermal management system for electronic devices.

The conventional cooling methods, whether natural or forced convection, have low heat transfer coefficients, require power, and are unsuitable for complex devices [3]. There are various two-phase flow systems, such as jet impingement, flow boiling, and spray cooling, that have high heat dissipation potential, but they could be costly and complex for such devices [4]. Therefore, the passive and two-phase flow systems, the heat pipes are sustainable approaches to thermal management. A high heat transfer coefficient is achieved by utilizing the latent heat of evaporation and condensation of the working fluid in a closed system [5-7]. So, heat pipes are widely accepted for electronics, solar, power systems, and avionics applications [3,8].

The pulsating heat pipe is a new member of the heat pipe family, having thermally-driven self-oscillatory two-phase flow behaviour for the working fluid [5]. First in 1990, Akachi introduced pulsating heat pipe [9]. The pulsating heat pipe can be used in a wide variety of applications that depend on electronics systems as a main component. The most important part of a heat pipe is the mechanism involved in returning the condensate to the evaporator zone. The serpentine closed-loop structure of capillary tubes or machined channels on a flat plate is used in the pulsating heat pipe. The capillary action through wick material or the gravity force completes the recirculation of the working fluid in conventional heat pipes. At the same time, thermally excited oscillatory slug-plug flow behaviour in a capillary-size tube achieves net circulation action of working fluid. Despite so much development in our understanding of the pulsating heat pipe over the past three decades, there is still a requirement for more efforts to make it viable for industrial or domestic applications.

One of the important design parameters for the pulsating heat pipe is the diameter of the capillary tube or channel. The distribution of partially filled working fluid into vapour plugs and liquid slugs along the entire length is required for it to function. This phenomenon is achieved by selecting the cross-section using the Bond number, equation (1) [10]. The ratio of buoyancy force to surface tension force provides a capillary length depending on the working fluid properties. The dominance of surface tension forces helps to form the train of vapour plugs and liquid slugs [11].

$$d_{cr} = \mathrm{Bo}\sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \tag{1}$$

Where, d_{cr} is critical diameter, σ surface tension, ρ_l density of liquid phase, ρ_g density of vapor phase, and g gravitational acceleration. The value of the Bond number varies from 0.7 to 1.8 to select the diameter of the channel [12].

The working fluid selection plays a very important role in deciding the thermal performance of a pulsating heat pipe. Thermal performance is influenced by thermo-physical properties such as boiling point, latent heat, specific heat, surface tension, dynamic viscosity, and thermal conductivity [8]. It is observed that working fluids with higher latent heat, lower dynamic viscosity, and lower surface tension can perform better in the pulsating heat pipe. The thermo-physical properties of water are best in this regard, but the delayed startup due to high surface tension and dynamic viscosity cause problems. So, to overcome this issue, water based surfactant solutions have been utilised for better startup, thermo-hydrodynamics, wettability, and bubble nucleation in the pulsating heat pipe [13,14]. The latent heat of the working fluid remains the same, but properties like dynamic viscosity and surface tension are changed with the addition of surfactants to water. We have performed experimental investigations with the working fluids acetone and pure water in the same experimental setup [15]. A visualisation study was also performed to find out the impact of operating conditions on the dual-loop setup [16]. Because the critical diameter was close to the larger diameter tube to form an asymmetric pulsating heat pipe, the thermal performance with working fluid acetone was improved. In an asymmetric single loop pulsating heat pipe, working fluid water at 50 and 60% has better thermal performance [12]. The asymmetric channels increase the pressure differences and the unbalanced driving force [17]. The experiments with the aqueous surfactant solution were planned to improve the thermal performance of the dual loop setup with the working fluid water as well. Kumar et al. [18] reported startup at a lower evaporator temperature using the surfactant solution and a dual-loop pulsating heat pipe. The optimum concentration was 500 PPM at 60% filling ratio in their study.

According to the literature, thermo-physical properties influence the thermal performance behaviour of a pulsating heat pipe. The present article has been focused on improving the thermal performance of the dual-loop asymmetric pulsating heat pipe with aqueous surfactant solutions. The experiments are conducted with various operating conditions to characterise the pulsating heat pipe, such as inclination angles, filling ratios, and heat load patterns.

2. Experimental details and procedure

The prepared experiment setup specifications are listed in Table 1. The testing facility schematic is shown in Figure 1, along with all the necessary equipment required to run the experiments. The test specimen was installed over a rotatable panel and a bench to set various inclination angles. For these two turns (bend radius: 10 mm), two cartridge heaters were sandwiched between copper plates to function as the evaporator. The detailed description of the evaporator and heat pipe specimen is shown in Figure 2. The condenser section of the setup was kept in an acrylic jacket with an inlet and outlet for the coolant. The coolant supply at a constant flow rate and temperature was maintained while testing the thermal performance using a rotameter (Eureka SSRS-MGS-9) and constant temperature bath (chiller) (Siskin Profichill RHC-250 ST-100).

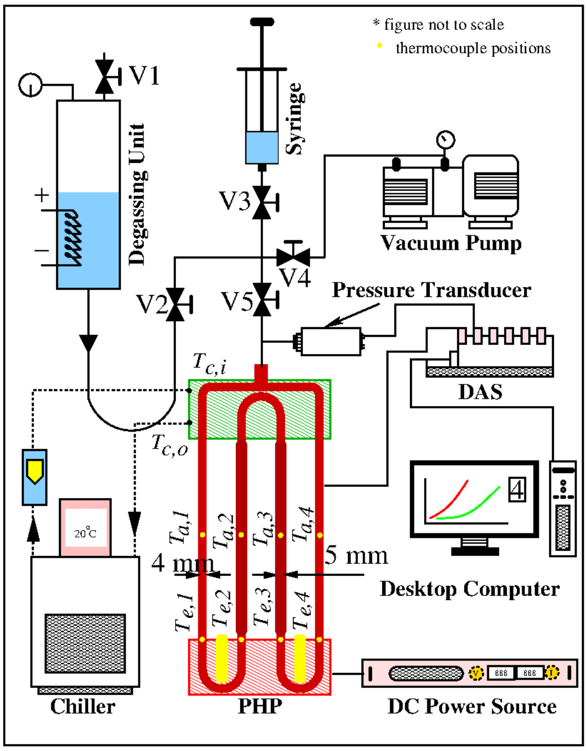

Figure 1. Schematic diagram of the pulsating heat pipe setup

The test specimen was equipped with a pressure transducer (Honeywell make; model AP122, range 0–1293 mmHg) at the condenser end. The tubes wall of adiabatic section have individual thermocouples (Tempsens make; T-type, probe diameter 0.5 mm). The evaporator thermocouples were inserted into the machined evaporator plate.

The condenser section thermocouples were placed at the jacket inlet and outlet ports. The thermal interfaces were filled using thermal paste (Cooler Master; $k \approx 8$ W/mK) for thermocouples and heater surfaces. Teflon insulation plates ($k \approx 0.25$ W/mK) covered the evaporator section. The adiabatic section was covered with a Nitrile foam tape ($k \approx 0.04$ W/mK of 3 mm thick) for insulation. The cartridge heaters in the evaporator section were connected to a DC power source (Keysight LXI N5771A). The signals from all thermocouples and pressure transducer were recorded from a data acquisition system (DAS) (National Instruments cDAQ-9189, 9213, and 9205) at 10 Hz.

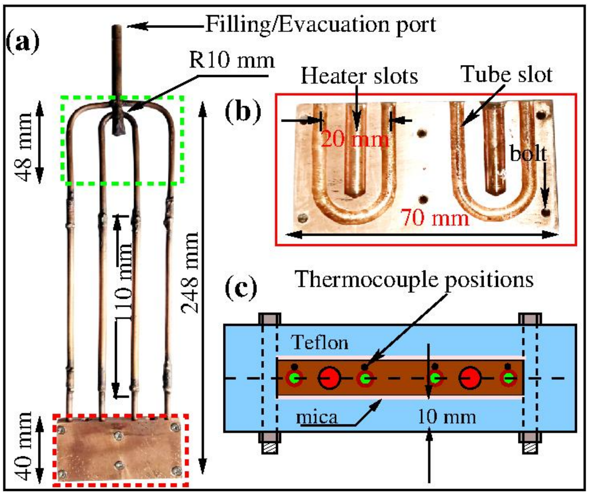

Figure 2. (a) Heat pipe specimen, (b) machined evaporator plate, and (c) schematic of evaporator assembly

The working fluid charging system consisted of stainless steel fittings (Swagelok make; ferrule fittings), a steel tank (degassing facility of 250 ml), and a syringe (Nipro 12 ml). A vacuum pump (Welch CRV Pro2, capacity 0.0002 mmHg) was used to evacuate the empty heat pipe before filling it with the working fluid.

The working fluid was prepared using pure water (Milli-Q, Merck) and sodium dodecyl sulphate (SDS) (CMC≈2500 PPM, CAS RN 151-21-3, TCI, C12H25NaO4S≥97%). A cleaned conical flask was used to make the solution with the help of a magnetic stirrer. The aqueous surfactant solution was prepared with concentration values of 500, 1000, 1500, and 2500 PPM by adding SDS solvent to the water by weight. The solution was first degassed using a heating and venting method to remove non-condensable gases. This working fluid was supplied to the evacuated heat pipe with the help of a syringe at the desired filling ratio. The filling ratio is the ratio of the filled to the empty volume of the heat pipe.

Table 1. The pulsating heat pipe experimental setup details

Tube material	Copper
Internal Diameter	2.4 & 3.4 mm
Number of turns	2
Filling Ratios	50 & 60%
Orientation	0^o, 30^o, 60^o & 90^o
Length	248 mm
Maximum heat load	70 W
Evaporator area	70×40 mm^2
Evaporator length	35 mm
Adiabatic length	160 mm
Condenser length	48 mm

3. Thermal performance calculations

The heat input ($Q_{in} = V \times I$) to the evaporator section causes temperature variations. The thermal performance has been estimated based on the average evaporator and condenser temperatures for an individual heat load. For a thermocouple, the average temperature can be written as [19]:

$$T_{e,1} = \sum_{j=1}^{n} T_{e,1,j} \tag{1}$$

Similarly, for other thermocouples $T_{e,2}$, $T_{e,3}$, $T_{e,4}$, $T_{c,i}$, and $T_{c,o}$. So, the mean evaporator and condenser temperatures can be represented as:

$$T_e = \frac{T_{e,1} + T_{e,2} + T_{e,3} + T_{e,4}}{4} \tag{2}$$

$$T_c = \frac{T_{c,i} + T_{c,o}}{2} \tag{3}$$

In operating condition, the sensible heat transfer to the coolant can be written as: $Q_c = \dot{m}c_p(T_{c,o} - T_{c,i})$ [20]. Where, c_p is the specific heat, $\dot{m}$ mass flow rate of the coolant.

The sensible and the total heat input mean, $\left(Q = \frac{Q_{in}+Q_c}{2}\right)$ effectively is transferred between the condenser and evaporator sections [20,21]. The heat loss to the ambient from the different sections can be $Q_{loss} = Q_{in} - Q$. After start-up and before dry-out, the maximum heat loss was 5.2% at a 60 W heat load. Before startup, the heat is mainly absorbed by the working fluid, evaporator, and tube walls. On the other hand, at a dry-out condition, the heat is absorbed only by the evaporator section and not the working fluid. The average thermal resistance can be written as equation (4) [21].

$$R_{th,avg} = \frac{T_e - T_c}{Q} \quad (4)$$

The maximum uncertainty associated with the thermal resistance and heat load estimations in this study was 7.7% and 2.7%, respectively.

4. Results and discussion

4.1. Start-up and transient characteristics

The heat input to the evaporator section of the pulsating heat pipe raises the temperature of the working fluid inside the tube. As the temperature of the wall is higher than the fluid temperature, a single phase flow heat transfer behavior is observed, as shown in Figure 3.

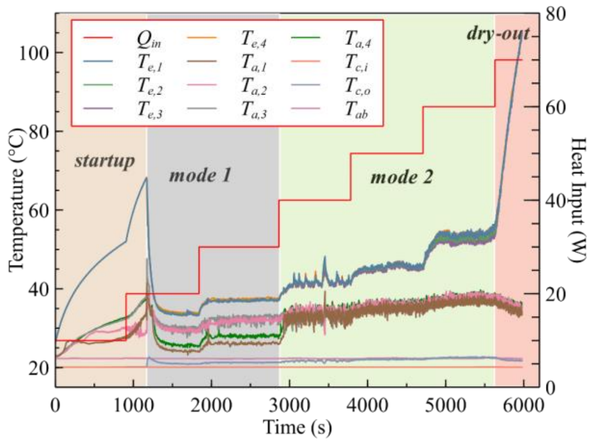

Figure 3. Temporal variations of temperature with surfactant solution (1500 ppm, FR=60%) in vertical orientation

The initial heat load of 10 W was insufficient to activate the heat pipe. However, the adiabatic section temperature fluctuations indicate small-scale movement of the fluid. The startup of the heat pipe occurs when the heat load is 20 W, there is a sudden drop in the evaporator temperature, and the pseudo-steady state operation is achieved. A high transfer coefficient because of the two-phase flow effectively decreases the evaporator temperature. After startup, there is a temperature difference and fluctuations in the adiabatic sections, i.e., *mode 1* operates in circulatory-oscillatory mode. The middle two columns, having a larger cross-section, have a higher temperature than the outer columns. This state was continued up to a 30 W heat load, but at a slightly higher temperature. Further variation in the heat load, i.e., mode 2, the temperature difference between the adiabatic sections, indicates the dominance of purely oscillatory motion. The variations in the heat loads change the thermo-hydrodynamic behavior of the heat pipe. The operation was gradually approaching the *dry-out* of the evaporator at 70 W of heat load.

4.2. Thermal performance under sudden heat loads

In this section, the thermal performance behavior with sudden heat loads will be discussed based on the temperature variations. Figure 4 shows the temperature variation at 50 W of heat load operating in vertical orientation. The sudden or abrupt heat loads were selected from the gradually applied heat loads. As discussed above, the pulsating heat pipe failed to maintain the evaporator temperature at a 70 W heat load, so, 50 and 60 W loads were applied suddenly. The heat pipe established a successful startup at 50 W of heat load and a pseudo-steady state evaporator temperature. The temperature of all the columns was nearly equal and achieved a pseudo-steady state after startup. The same temperature in all columns indicates that the fluid is having the same direction of motion, which can only be possible if there is purely oscillatory motion of the working fluid in the individual columns.

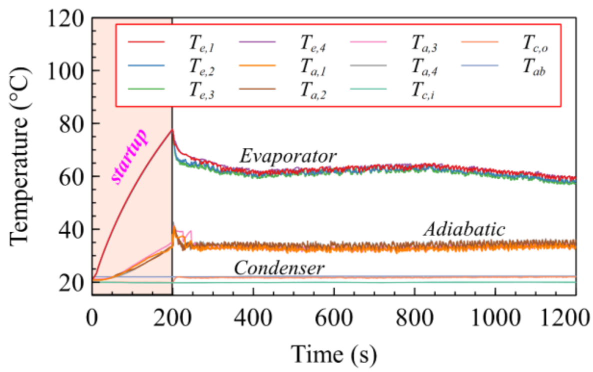

Figure 4. Startup of two-phase flow heat transfer at heat load 50 W SDS 1000 PPM (FR=60%)

4.3. Effects of inclination angles

The average thermal resistance is compared when operating the heat pipe at different inclination angles. In vertical (90°) orientation, the heat pipe has the lowest thermal resistance, as shown in Figure 5. The horizontal orientation operation shows the highest thermal resistance and a significant drop in thermal performance. The change in thermal performance with lowering the inclination angle shows the contribution of buoyancy force to the oscillatory flow of the working fluid. In vertical operation, the perfect cylindrical bubble with hemispherical ends helps to push the adjacent liquid towards the condenser section more effectively than bubbles with oblate ends [10].

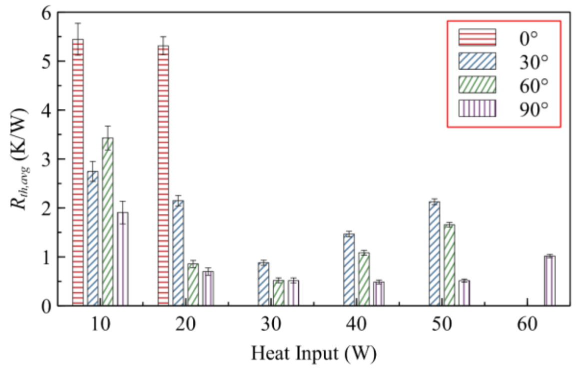

Figure 5. Thermal performance under various inclination angles at 1000 PPM (FR 50%)

The bubbles are pointed towards the upper wall with oblate ends, and working fluid is escaping these bubbles, making the pumping inefficient. An inefficient movement of fluid from the hot region reduces the thermal performance of the heat pipe and increases the operating temperature. Eventually, the horizontal operation of the asymmetric pulsating heat pipe fails in this configuration and operating condition.

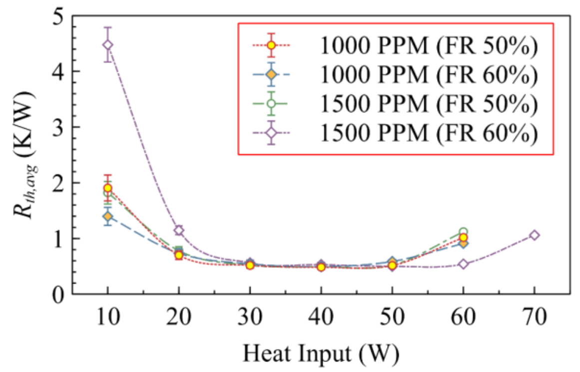

Figure 6. Average thermal resistance at different filling ratios for various surfactant concentrations

4.4. Effects of filling ratios

The effect of the filling ratio on the thermal performance behavior is shown in Figure 7. The optimum range is 50–60% of the filling ratio for the pulsating heat pipes [15,18]. The low filling ratio is liquid-deficient, while the high filling ratio operation becomes vapor deficient. So, both phases are required to have an optimum value of vapor and liquid for the pumping action between the evaporator and the condenser section. The concentration of surfactant at 1500 PPM at a 60% filling ratio shows higher heat transfer capability than a 50% filling ratio. The solutions having 500 and 2500 PPM concentrations were used at a 60% filling ratio only. At higher filling ratios, the contribution of sensible heat transfer increases, which delays the start-up of two-phase heat transfer [11].

4.5. Effect of surfactant concentration

The concentration impact of the surfactant in the solution over the thermal performance is shown in Figure 7 for the vertical orientation operations. The change in driving force due to the variation in the properties of surface tension and viscosity affects the operating temperature and hence the thermal resistance [8]. The heat load capability was unchanged for the 1500 and 2500 PPM of the pulsating heat pipe. The thermal resistance was changing with concentration variations.

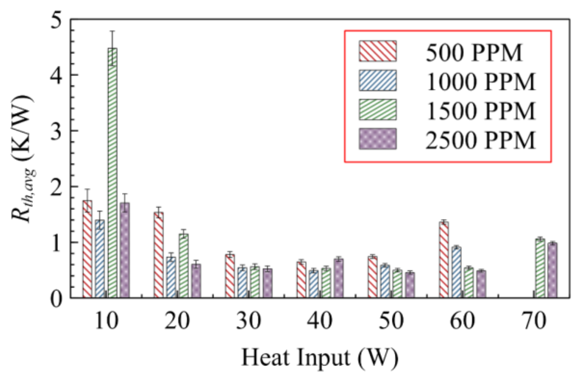

Figure 7. Variation in the thermal performance due to surfactant concentration (FR 60%)

4.6. Thermal performance comparison

Figure 8 shows the comparison of thermal performance among asymmetric (non-uniform) and conventional channel pulsating heat pipes for working fluids such as pure deionized water and aqueous surfactant solutions.

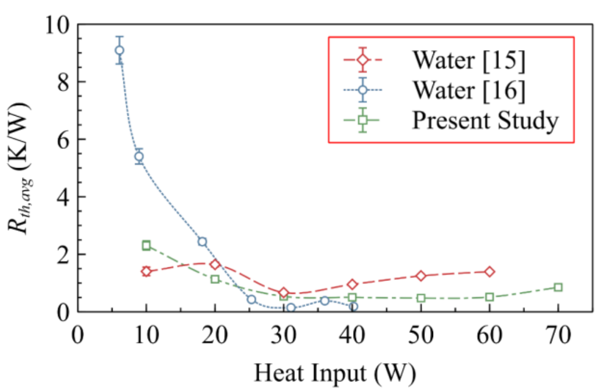

Figure 8. Thermal performance comparison for conventional, asymmetric configuration, and for working fluids

The thermal resistance based on the input power was used to compare different configurations of the heat pipe. The lower surface tension and improved wettability decrease the operating temperature, which improves startup performance and lowers the thermal resistance. The heat load range was almost equivalent to the working fluid, pure water.

5. Conclusions

The present study has focused on improving the startup phenomena and the thermal performance of a dual-loop asymmetric pulsating heat pipe using an aqueous surfactant solution. The operating parameters like inclination angle, filling ratio, and heat load patterns have also affected the thermal performance. The change in physical properties such as surface tension and improved wettability of surface tension and viscosity results in early two-phase flow startup even at low evaporator temperatures.

The surfactant concentration between 1000 and 2500 shows better thermal performance. The minimum thermal resistance of 0.46 K/W was obtained operating in vertical orientation, which is required for many passive thermal management applications. The heat pipe has shown better operating characteristics up to a 30° inclination angle.

By introducing larger diameter tubes in the adiabatic section, this asymmetric pulsating study aimed to improve the circulatory-oscillatory motion of the working fluid. At higher heat loads, the expected unidirectional flow might happen if the larger channels are placed alternately for further enhancement of the thermal performance.

Acknowledgements

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Nomenclature

d	diameter	mm
V	Voltage	V
T	Temperature	°C
I	Current	A
$\dot{m}$	Mass flow rate of coolant	kg/h
$R_{th,avg}$	Thermal resistance	W/K
Q	Heat	W

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