Thermal performance of flat plate pulsating heat pipe using aqueous alcohol solutions

Maksym SLOBODENIUK[1,2*], Flavien MARTINEAU[1], Vincent AYEL[1], Rémi BERTOSSI[2], Cyril ROMESTANT[1], Yves BERTIN[1]

[1]Pprime Institute CNRS – ENSMA – Université de Poitiers, UPR 3346, 86961 Futuroscope-Chasseneuil, France.
[2]IPSA, Direction de la Recherche et de l'Innovation de l'IPSA, 92120 Ivry-sur-Seine, France.
[*](Corresponding author: maksym.slobodeniuk@ensma.fr)

Abstract – This work presents experimental results obtained for a copper closed looped flat plate pulsating heat pipe (CL-FPPHP) with milled 3×3 mm² square channels bent into a planar serpentine with eight U-turns at the evaporator zone. Aqueous alcohol solutions and deionized water were used as working fluids with a volumetric filling ratio of 50%. Obtained results indicate that operating performances of the device strongly depend on the thermophysical properties and wettability of working fluids, device orientation and condenser temperature. Moreover, increase of the cold source temperature leads to temperature fluctuations stabilization and performance improvement for the horizontal mode, and to a rise of the evaporator temperature for the vertical mode.

Keywords: Flat plate pulsating heat pipe, Binary mixtures, Surface tension, Wettability, Thermal management.

Nomenclature

d	diameter, m
g	acceleration, m.s^{-2}
U	velocity, m.s^{-1}
Bo	Bond number
We	Weber number
Ga	Garimella number
R	thermal resistance, °C.W^{-1}
T	temperature, °C
I	current, A
V	voltage, V

Greek symbols	
ρ	density, kg.m^{-3}
μ	viscosity, Pa.s
σ	surface tension, N.m^{-1}
Index and exponent	
l	liquid
v	vapor
hs	heat source
cw	cooling water
cr	critical

1. Introduction

Current trends in electronic industry as well as miniaturization and increase of dissipated powers lead to constant demand in novel high efficiency thermal management systems. Heat removal in compact electronic systems with high heat flux generation is difficult for conventional cooling methods due to their low efficiency and large sizes [1]. During few last decades, two-phase thermal management approaches, like microchannels, spray-based systems and heat pipes, have been intensively studied [2]. Unlike microchannels and spray-based technologies, heat pipes are passive heat transfer devices without any moving mechanisms which lead to greater reliability.

In addition to their reliability, their high heat transfer capability and passive working principle, functionality independence of the heat pipes evaporator and condenser leads to flexibility of the projected shape, depending on the design requirements. Due to all these attributes, heat pipes are widely used in microelectronics, power electronics and aerospace

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cooling applications. However, despite all these advantages, conventional heat pipes have a large number of limitations which affect their thermal performances and working reliability: capillary, sonic, boiling and viscous limitations, as well as wick thermal resistance etc.

A novel passive wickless heat transfer device, proposed in the 90's by Akachi [3] and named pulsating Heat Pipe (PHP) is studied in this paper; it is shown in fig. 1. Base condition of a PHP consists of a unique capillary channel, bended in few/many turns from hot to cold sources, and partially filled with a working fluid at liquid/vapor saturation state. The physical principle of the PHP is based on the phase change induced motions of working fluid from evaporator to condenser. It is naturally distributed inside the channel in the form of liquid plugs and vapor slugs due to capillary dimension of the channel [4]. Due to the exploitation of both sensible and latent heat transfer modes, PHP has a high heat transfer capability, compared to classical heat pipes. In addition, PHP has a simple structure and the ability to operate under different gravity levels and different positions. Because of all these facts PHPs could become a novel competitive thermal management system for ground and space applications [5].

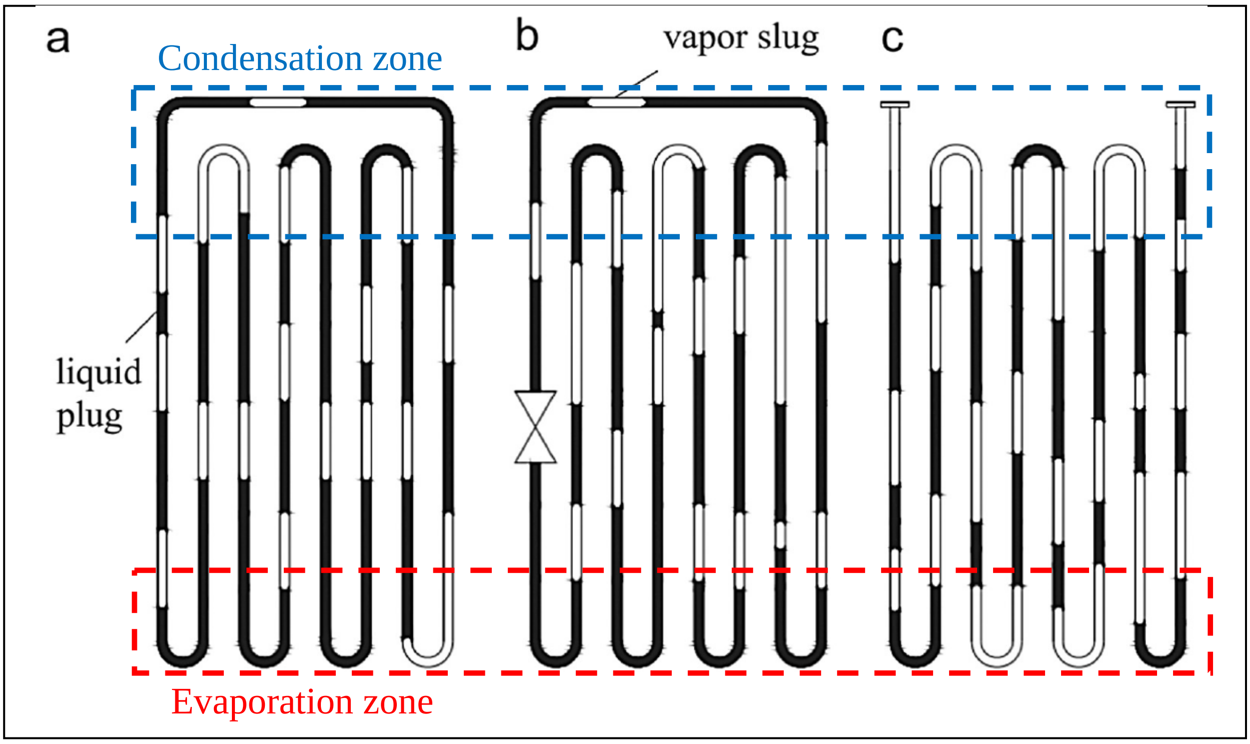

Figure 1: *Pulsating heat pipe types: (a) Closed Loop PHP; (b) Closed Loop PHP with check valve and (c) Closed End PHP [6]*

However, initial formation and following operation with slug/plug flow is possible only for channel diameter lower than critical diameter (d_{cr}). First -and widely used- criterium, based on the Bond number [4] (ratio between capillary and gravity forces in an horizontal tube), could predefined a range of channel diameters for a corresponding fluid as: $0.7(\sigma/g(\rho_l - \rho_v))^{1/2} \leq d_{cr,Bo} \leq 1.84(\sigma/g(\rho_l - \rho_v))^{1/2}$. However, this criterion is obviously not applicable for microgravity conditions due to the presence of gravitational acceleration, leading to an infinite diameter in the case of weightlessness conditions. For this reason, Gu et al [7] have proposed a new criterion based on the Weber number, taking into account the inertial forces, and leading to the critical diameter: $d_{cr,We} \approx 4\sigma/(\rho_l U_l^2)$. After years, Harichian and Garimella [8] have proposed a semi-empirical criterion based on Garimella number and leading to $d_{cr,Ga} = (160\mu_l/\rho_l U_l(\sigma/g(\rho_l - \rho_v))^{1/2})^{1/2}$). A criterion based on both Weber and Garimella numbers becomes more relevant for both terrestrial and microgravity applications. Based on one of these criteria, channel diameter/working fluid pair should be chosen.

Thermal performances, uniform temperature field and heat transfer rate distribution are one of the key priorities on PHP research. Heat transfer degradation often happens due to partial or full dry-out of the evaporator zone, particularly when using water as working fluid: while very performing through its thermophysical properties, its bad wettability compared to other fluids generally plays a negative role on heat and mass transfers in such systems [9]. In order to overcome this problem, binary aqueous mixtures, based on the addition of a low quantity of high-weight alcohols, could improve wettability and, also, rewetting for self-rewetting fluids of heat transfer surface [10]. Behavior of self-rewetting fluids (SRWF) is illustrated in fig. 2. In addition to SRWF, injection of low-weight alcohols in higher quantities leads to almost the same effect.

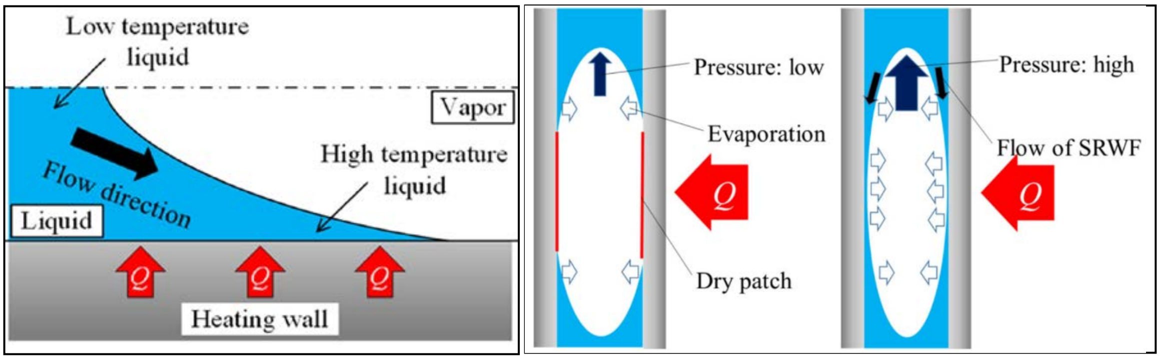

Figure 2: *Behavior (left) and heat transfer enhancement mechanism (right) of the binary fluid [11]*

This work presents the thermal efficiency study on a flat plate pulsating heat pipes with channel diameter corresponding to all criteria and filled with self-rewetting fluids (5% 1-butanol and 5% 2-butanol aqueous solutions), a binary mixture (20% ethanol aqueous solution) and pure water (as a reference) for different condenser temperatures and inclinations.

2. Experimental setup and procedure

The tested flat plate pulsating heat pipe (FPPHPs) (in fig. 3) is composed of a copper rectangular plate (80×200×3.5 mm^3) in which a unique rectangular channel (3 mm deep and 3 mm large) is milled. The channel forms a closed loop serpentine with 8 U-turns in the evaporator zone. The plate with milled channels (figure 3, right) is covered with a thin copper plate (80×200×1 mm^3) using solder with silver addition to guarantee perfect sealing at the plate boards, and between adjacent channels relative to one-another. Assuming the thickness of the soldering junction as negligible, the depth of the channel remains equal to 3 mm.

Copper heater, composed of a metallic plate (80×40 mm^2) with milled serpentine channel in which is inserted a heating wire (Thermocoax® Type NcAc15, 1 mm external diameter) is soldered on the bottom-back side of the FPPHP. Heat power is distributed by power source (ELC® ALR3220, 640 W max) and regulated with LabVIEW® software. Condenser (80x100 mm^2) is cooled by a water serpentine channel milled in a massive copper plate, soldered on the top-back face of the FPPHP plate. This condenser is connected to a closed secondary glycol-water mixture flow loop of the laboratory thermostat Huber® CC 240wl, which helps with temperature regulations in range of -20°C to 80°C. Working fluids, filling ratio and experimental conditions are presented in table 1. Choice was focused on these fluids due to their non-exotically and ability to improve heat transfer performances of pulsating heat pipes.

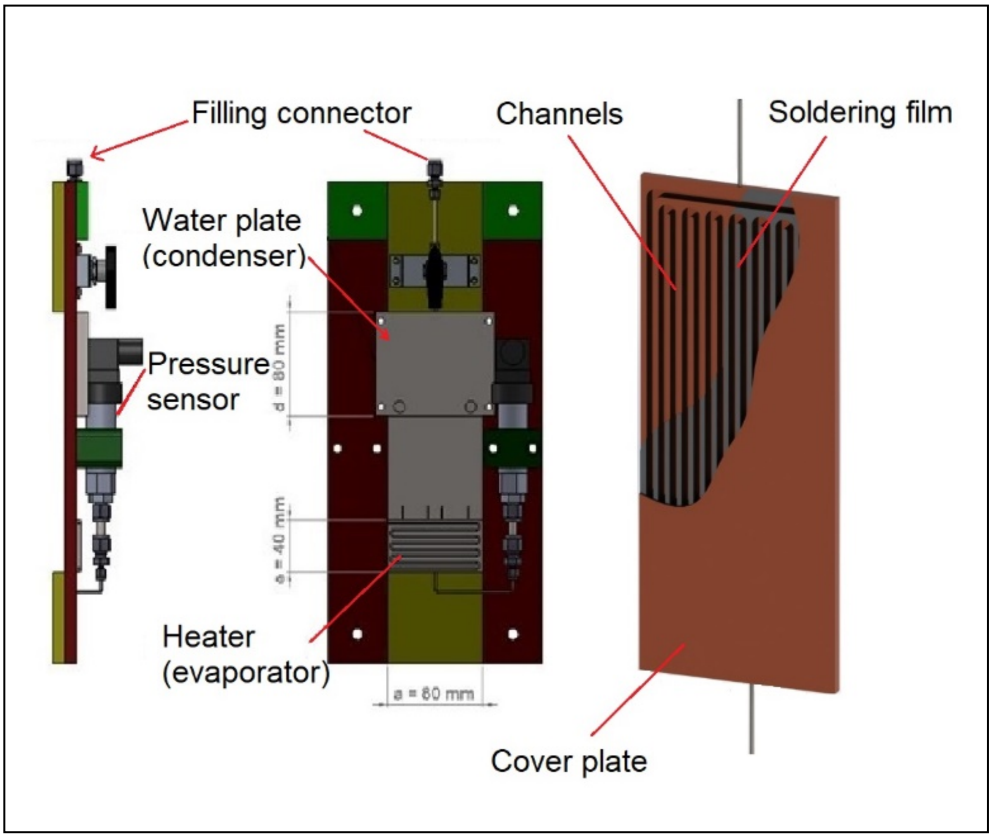

Figure 3: *FPPHP test section (left); scheme of the tested device (right)*

Sixteen T-type thermocouples (TCs) were used for temperature acquisition: nine TCs are located in grooves milled in the evaporator zone behind the FPPHPs (between U-turns in the ribs), at the interface between back plate and heating block, five other TCs are glued on the adiabatic zone (between evaporator and condenser), two TCs instrument both water inlet and outlet of the condenser. Data acquisition is performed using NI-9214 thermocouple module connected to the NI® cDAQ-9189 chassis. A pressure sensor (GE® PTX5076-TA-A3-CA-HO-PS, 2 bars absolute, ±200 Pa) records the fluid pressure at the bottom of the evaporator zone (connected to the middle U-turn), with a sampling period of 0.2 s.

Working fluids	DI Water (De-ionized Water) 5% 1-butanol aqueous solution 5% butan-2-ol aqueous solution 20% Ethanol aqueous solution
Filling ratio (at 20°C)	50%
Applied heat power	Series of 50 – 100 – 150 – 200 – 100 W
PHP orientation in space	Vertical, bottom heated mode (BHM) Horizontal

Table 1: *Experimental operating conditions*

Filling procedure of the flat plate PHP has been divided in three steps: first, a proportional mixing of the necessary amount of the fluids into vacuum filling reservoir has been performed; then, the subsequent reservoir has alternatively been heated and cooled to separate liquid and non-condensable gases via boiling/condensation process and finally reservoir has been installed on the thermostatic bath set at 20°C temperature and connected to the vacuum pump. All actions have been repeated three times.

3. Results and discussions

3.1. Influence of the working fluid on the PHP temperature behavior

Typical temperature histories for water and 5% butan-2-ol aqueous solution are presented on fig. 4 for horizontal orientation.

For water (fig. 4-top), few temperature fluctuations have been observed during the startup at 50 W. These fluctuations are assumed to occur during PHP startup due to initial uniform liquid distribution in the PHP. Following increase of temperature can be explained by liquid accumulation in the condenser zone until next heat load augmentation setting off fluid flow motion. All subsequent increases of the heat input are accompanied with fast passing high temperature peaks and following frequent fluctuations - with amplitude values up to 35°C (which could correspond to short-term liquid accumulation in the condenser) - as well as pressure fluctuations. Temperatures decrease and fluctuations stabilization with low amplitude correspond to reduction of heat load from 200 W to 100 W, with a finally very regular steady-state operation mode at this heat power level for decreasing heat powers applied.

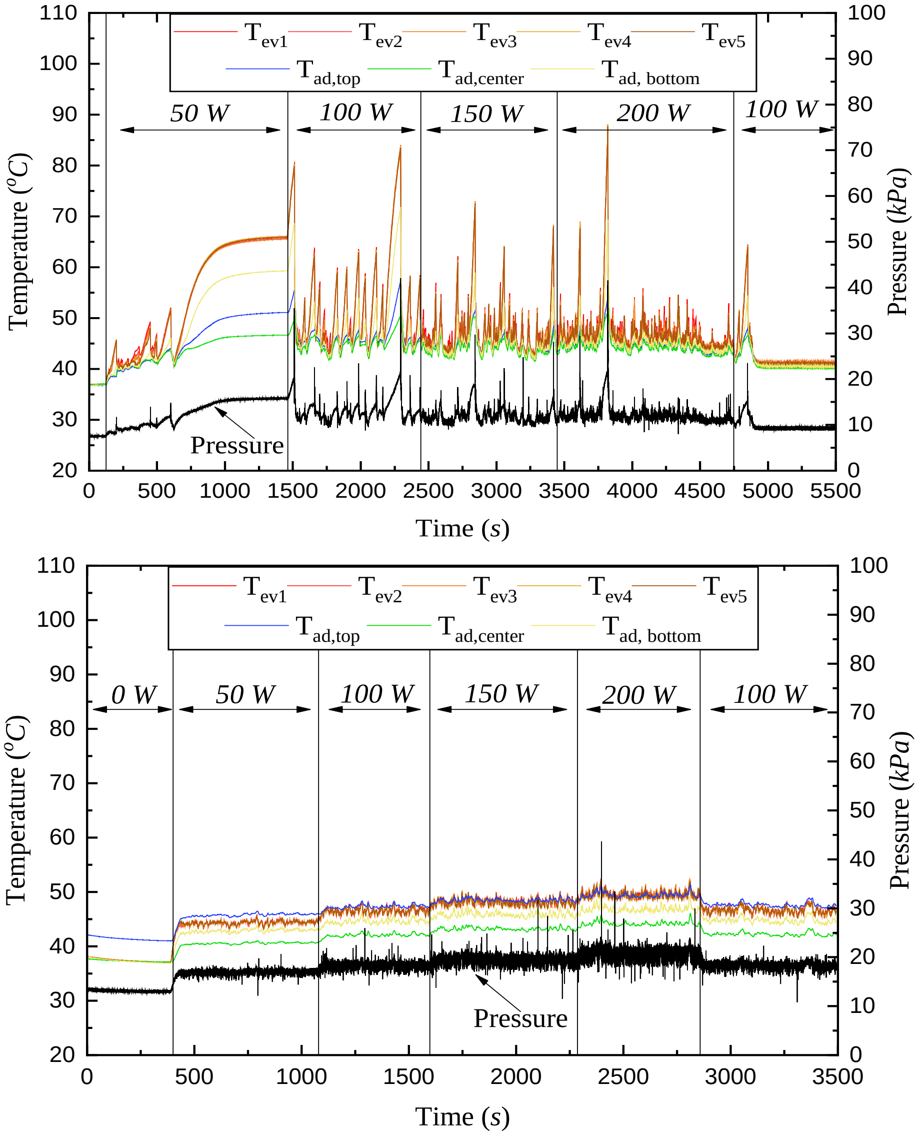

Figure 4: *Transient operating curves for FPPHP filled with water (top) and 5% butan-2-ol aqueous solution (bottom) (T_{cond} = 40°C)*

Temperature profile for the FPPHP filled with 5% butan-2-ol aqueous solution (fig.4-bottom) is very different from the case where the working fluid is pure water (fig. 4-top). Consistent increase of the heat input leads to the slight rise of evaporator temperatures and pressure. Regular and frequent temperature and pressure fluctuations with low amplitude were observed whatever the heat power applied. Any temperature or pressure peaks, during PHP operation, similar to PHP filled with water have not been registered. Observed operating behavior of PHP filled with aqueous alcohol solutions indicates absence of the phenomena of collected liquid in condenser and/or evaporator rewetting due to liquid capillary reverse flow in channel corners (similarly to capillary grooved heat pipe capillary forces).

3.2. Working fluid influence on thermal resistances

The experiments have been carried out with only one PHP, described in previous section, without any change in the system, so the choice of the working fluid is the only modified parameter between all experimental series. Thermal resistances (calculated as $R=(T_{hs}-T_{cw})/Q$, T_{hs} is the mean value of the evaporator temperatures, and T_{cw} the mean value of the cooling water inlet and outlet temperatures, both averaged in time) for all working fluids, and for cooling fluid temperatures of 20^{o}C and 40^{o}C, are presented in fig. 5 and 6, respectively for horizontal and vertical positions.

The standard uncertainty of the thermal resistance was calculated by the following equation [12]:

$$\frac{\Delta R}{R} = \sqrt{\left(\frac{\Delta T_{hs}}{T_{hs}-T_{cw}}\right)^2 + \left(\frac{\Delta T_{cw}}{T_{hs}-T_{cw}}\right)^2 + \left(\frac{\Delta I}{I}\right)^2 + \left(\frac{\Delta V}{V}\right)^2} \tag{1}$$

where ΔT_{hs}, ΔT_{cw}, ΔI and ΔV are the errors of direct measurements of the temperatures, current and voltage.

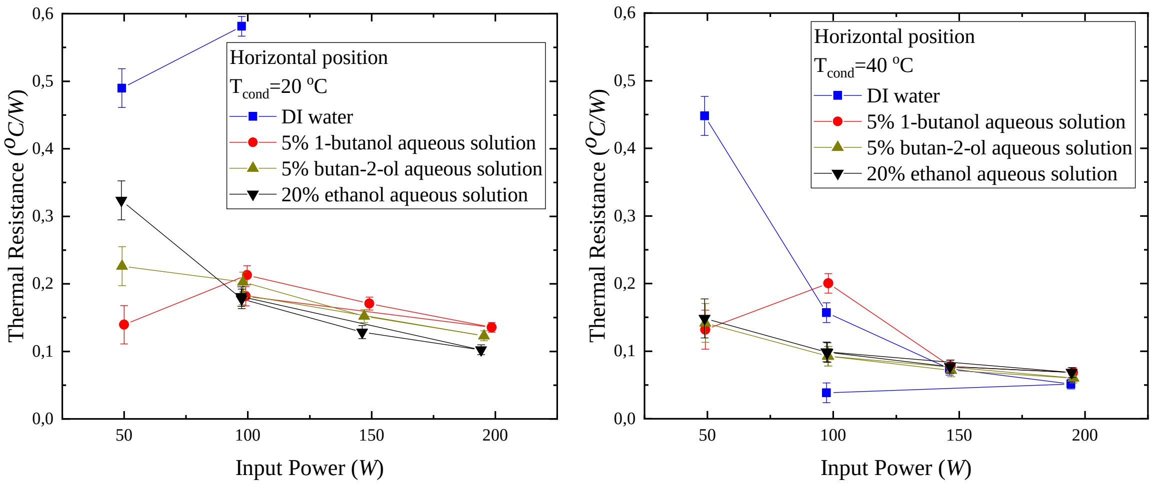

Figure 5: *Thermal resistances for the FPPHP operating in horizontal position*

Experiments carried for the FPPHP tested in horizontal orientation and filled with pure water have shown the maximal values of thermal resistance in low range of input heat: operation of the PHP with cooling temperature of 20^{o}C are limited by heat loads higher than 100 W. Should be noted that experimental system was programmed on working temperatures lower than 100^{o}C, so this is not general operating limitation of a PHP, but limitation due to overheating.

Caused of this reason, just two experimental points of the thermal resistance for FPPHP filled with water and operating in horizontal mode with condenser temperature of 20^{o}C have been obtained. Initial temperature and thermal resistance increase for the deionized water and 1-butanol solution could be explained by accumulation of liquid in the condensation zone (dry-out mode). In addition, their worse wettability and absence of evaporator rewetting phenomena were supposed as a second reason of the "delayed" startup and thermal resistance augmentation.

Condenser temperatures augmentation leads to reduce the thermal resistance during heat load increase. Thermal resistances tend to decrease with increasing applied heat power for 2-butanol and ethanol aqueous solutions for both condenser temperatures. Case of 1-butanol aqueous mixture is characterized by an increase of the thermal resistance until heat load reaches 100 W and a slight decrease during consequent input heat augmentation.

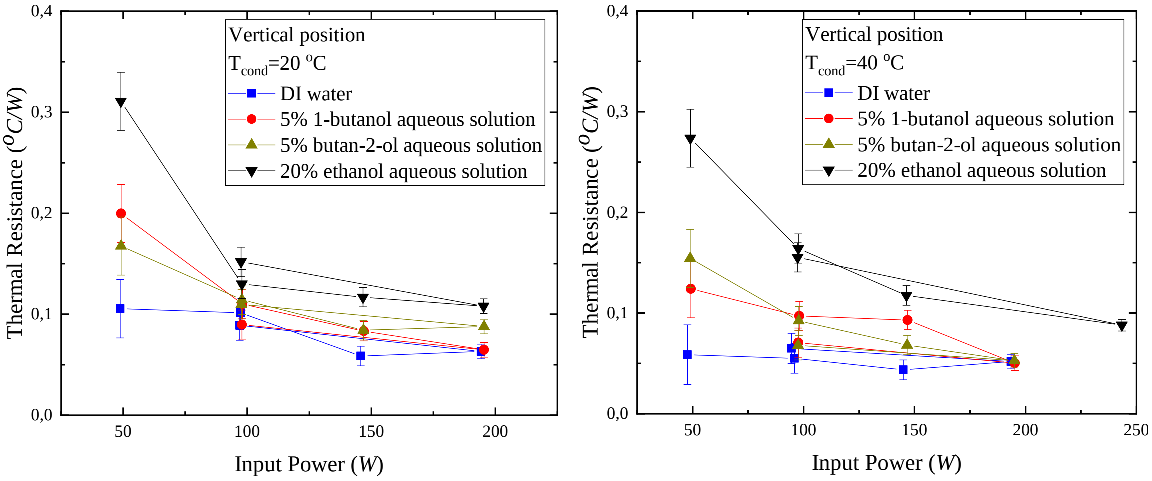

Figure 6: *Thermal resistances for the FPPHP operating in vertical position, bottom heating mode*

Slug/plug flow operating regime shifts to the gravity-assisted vapor/liquid flow pattern (annular or bubbly regime, giving a kind of interconnected looped thermosiphon mode), in vertical bottom heated mode. Thermal resistances of the heat pipe for all fluids (see fig. 6), except for pure water, are higher for low heat loads and decrease with augmentation of the input heat, which is a very common trend for such devices. In vertical BHM, the FPPHP filled with pure water shows better thermal performances than with alcohol aqueous solutions. This contradiction with the results obtained in horizontal mode can be explained by the annular or bubbly flow regime, for which the latent heat of vaporization and heat capacity play a major role, higher than in slug flow regime. The much greater values of such parameters for water than other fluids, added to the insignificant influence of wettability in such configuration due to permanent liquid presence in the evaporation zone thanks to gravity forces, explain the lower thermal resistance curves obtained for water in vertical bottom heated mode.

4. Conclusion

Copper closed loop flat plate pulsating heat pipe (CL-FPPHP) filled with water and alcohol aqueous solutions has been studied in both horizontal and vertical positions. In the case of horizontal mode and condenser temperature of 40^{o}C, stable temperature fluctuations were observed with an applied heat power equal or superior to 50 W for 2-butanol mixture and 100 W for water (with short-time temperature overshoots). Thermal resistances of the FPPHP filled with water are quite higher than with alcohol aqueous mixtures for low heat loads. Condenser temperature augmentation leads to increase the thermal efficiency. Operation of the FPPHP in vertical position is characterized by an interconnected looped thermosiphon mode. General decreases of thermal resistances during heat load augmentation were noted for all tested fluids. Initial increase of thermal resistance for the FPPHP filled with pure water and operating in horizontal mode with condenser temperature of 20°C is assumed to be caused by collected liquid in condenser and evaporator dry out.

These results clearly show the improvement of the FPPHP thermal performances with aqueous mixtures due to better wettability compared to water. Furthermore, increase of condenser temperature leads to stable operation of the FPPHP and insignificant decrease of evaporator temperatures.

References

[1] J. Qu, H. Wu, O. Cheng, Start-up heat transfer and flow characteristics of silicon-based micro pulsating heat pipes, *Int. J. Heat Mass Transf.*, 55 (2012), 6109–6120.

[2] M. Mochizuki, Th. Nguyen and K. Mashiko, Y. Saito, T. Nguyen, V. Wuttijumnong, A review of heat pipe application including new opportunities, *Fr. In Heat Pipes*, 2 (2011).

[3] H. Akachi, F. Polasek and P. Stulc, Pulsating heat pipes, *Proc. 5th Int. Heat Pipe Symp.*, Melbourne, Australia, (1996), 208–217.

[4] S. Khandekar and M. Groll, On the definition of pulsating heat pipes: an overview, *Proc. 5th Minsk Int. Sem. Heat Pipes, Heat Pumps and Refrigerators,* Minsk, Belarus, (2003).

[5] G. Karimi and J.R. Culham, Review of pulsating heat pipe mechanism for high heat flux electronic cooling, *Proc. 9th Inter Soc. Conf. Therm. Phenomena.*, Las Vegas, USA, (2004).

[6] D. Bastakoti, H. Zhang, D. Li, W. Gai and F. Li, An overview of the developing trend of pulsating heat pipe and its performance, *Appl. Therm. Eng.*, 141 (2018), 305-332.

[7] J. Gu, M. Kawaji and R. Futamata, Effects of Gravity on the Performance of Pulsating Heat Pipes, *J. Thermoph. Heat Tr.*, 18 (2004), 370 – 378.

[8] T. Harichian and S. Garimella, A comprehensive flow regime map for micro- channel flow boiling with quantitative transition criteria, *Int. J. Heat Mass Transf.*, 53 (2010) 694 - 702.

[9] A. Cecere, D. De Cristofaro, R. Savino, V. Ayel, T. Sole-Agostinelli, M. Marengo, C. Romestant and Y. Bertin, Experimental analysis of a flat plate pulsating heat pipe with self-rewetting fluids during a parabolic flight campaign, *Acta Astronautica*, 147 (2018), 454-461.

[10] Y. Abe, A. Iwaski and T. Tanaka, Thermal management with self-rewetting fluids, *Microgravity Sci. Tech.*, 16 (2005), 148–152.

[11] K. Fumoto, M. Sasa, T. Okabe, R. Savino, T. Inamura and M. Shirota, Research on heat transfer performance of the open-loop micro pulsating heat pipe with self-rewetting fluids, *Microgravity Sci. Tech.*, 31 (2019), 261-268.

[12] S. Kline, F. McClintock, Describing uncertainties in single-sample experiments, *Mech. Eng.* 75 (1953), 3–8.

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