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PERFORMANCE OF A SINGLE LOOP PULSATING HEAT PIPE

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Abstract. *The goal of this study is to study the heat transfer in a single loop pulsating heat pipe with a stainless steel evaporator and the rest of its body made of Nylon®, using R134a as refrigerant fluid and only thermocouples to measure the average temperatures in the pulsating heat pipe. Through a simple formulation the thermal resistance of the system was calculated. It has been found that the thermal resistance varies from near 0.2 K/W down to near 0.06 K/W depending on the filling ratio and heat input in the system. The result was compared with another in the literature, indicating consistency and a satisfactory performance.*

Keywords: *pulsating heat pipe, heat transfer, closed loop, two-phase flow.*

1. PULSATING HEAT PIPES

According to Charoensawan et al. (2003), pulsating or oscillating heat pipes are relatively new heat transfer devices. Akachi (1990) is the one who first designed this new type of heat pipe in which the heat transfer is based in the oscillating movement of the vapor bubbles and liquid pistons alternately arranged.

Because of their enhanced thermal performance, pulsating heat pipes have a great array of applications, being used in photovoltaic cell cooling, solar collectors, heat exchangers, electronics cooling and fuel cells cooling. Fig. 1 illustrates those applications.

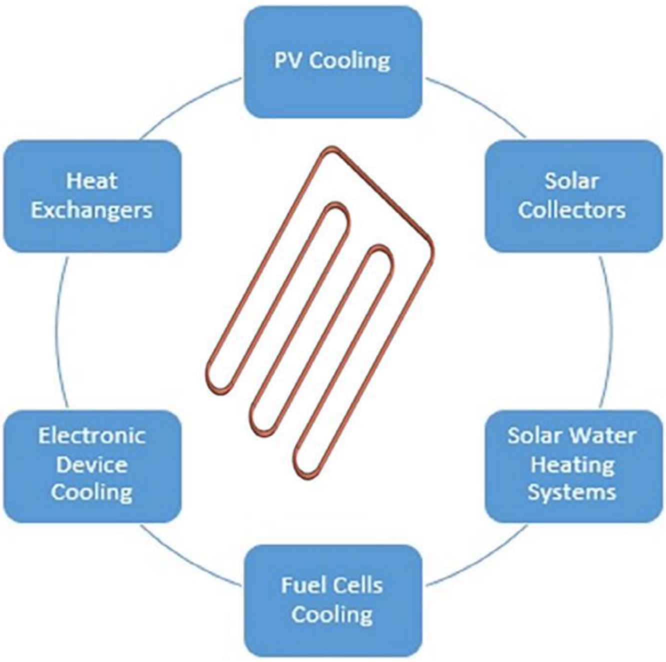

Figure 1 – Different applications of pulsating heat pipe, Nazari et al (2018).

There are mainly three types of pulsating heat pipes, which are shown in Fig. 2. The first consists of a tube in a closed-loop with no loose ends, the second is a closed-loop tube with no loose ends and a check valve in order to orientate the flow in a determined direction and the third has two loose ends and called open loop because of it.

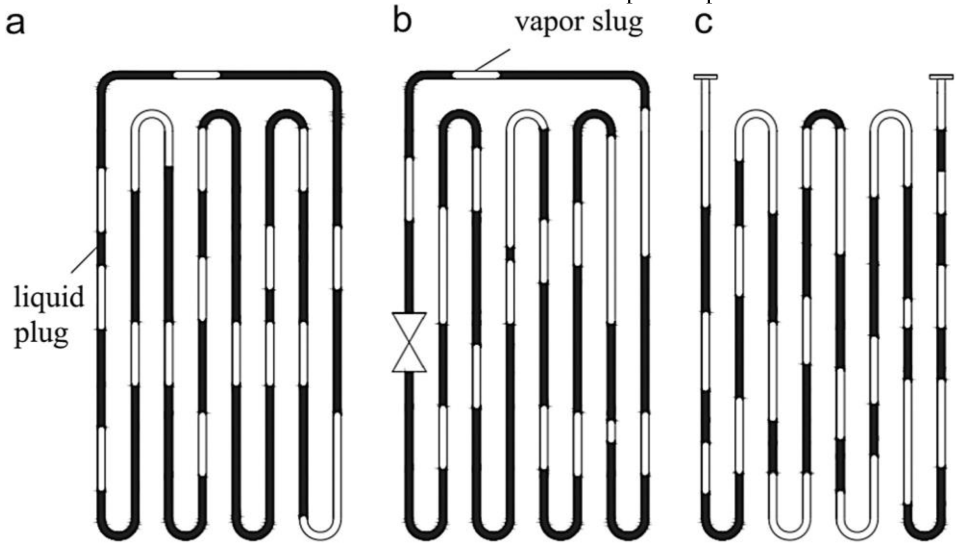

Figure 2 – The three mainly types of pulsating heat pipes: (a) closed-loop pulsating heat pipe; (b) closed-loop pulsating heat pipe with check valve; (c) open-loop pulsating heat pipe, Bastakoti et al (2018)

There are also three distinct regions in a pulsating heat pipe which are very important when studying its performance: the heating section, the adiabatic section and the cooling section. These are shown in Fig. 3 below.

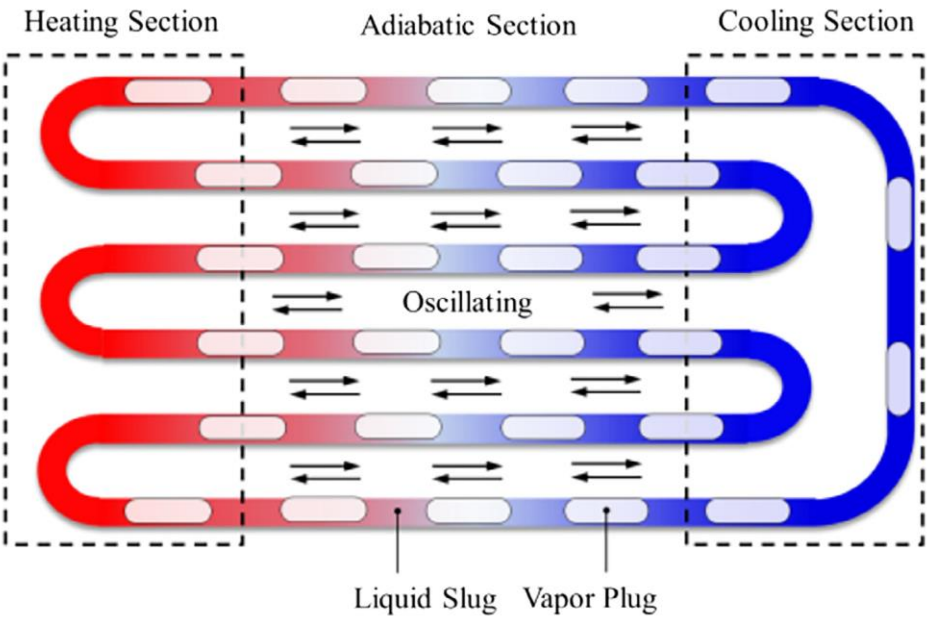

Figure 3 – The three sections of a pulsating heat pipe Nazari et al (2018)

In the heating section, heat is added to the system and leads to vapor generation and an increase in vapor pressure from the growth in the size of the vapor bubbles, causing the liquid motion towards the cooling section. In the cooling section, heat is extracted, and condensation occurs because the pressure inside the vapor bubbles decrease. The motion inside a pulsating heat pipe is kept by this cycle of heating and cooling.

The higher heat transfer capacity of pulsating heat pipes can be seen in figure 4 below and explained as: using a pure copper bar to transfer heat from a heating part to a cooling part is not very efficient because there is a small temperature gradient between the copper bar and the two parts. With the use of a pulsating heat pipe, heat can be conducted faster in a longitudinal directional along the tube compared to the pure copper bar, leading to a lower temperature gradient in the fluid and also a higher temperature gradient between the pulsating heat pipe and the two parts. This lower temperature gradient, when used with the amount of heat transported, can be used to calculate the thermal resistance of the pulsating heat pipe and its value is of great importance in many works trying to find optimal points of performance for different applications.

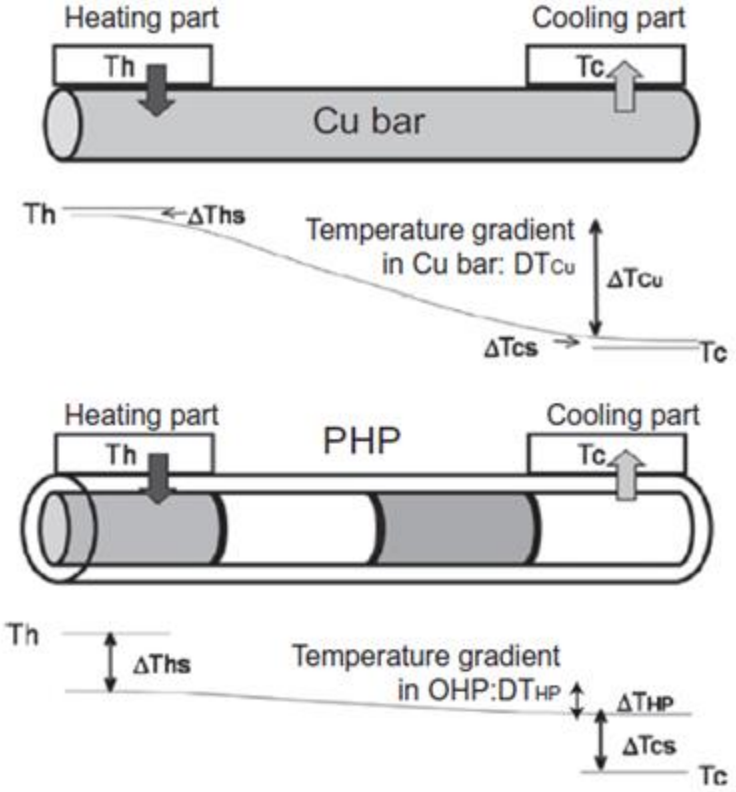

Figure 4 – A schematic showing why a pulsating heat pipe is a better choice than a copper bar using conduction, Natsume et al (2011).

When compared to other heat transfer devices, pulsating heat pipes have many other advantages such as:

- They can be used in a wide range of temperatures: from cryogenic applications at 17 K to high temperature applications at 443 K as can be found in Karthikeyan et al (2014) and Natsume et al (2011);
- They do not need an internal porous structure (wick) in order to work;
- They can be designed to be gravity-independent;
- Ma (2015) found that by increasing the thermal load on the evaporator the tube's heat transfer capacity rises drastically until it reaches near the point of failure or wall-drying;
- They are of a smaller size and require less refrigerant mass when compared to thermosyphons or other heat pipes, saving material and cost as concluded by Nazari et al (2018).

Important factors in the design of a pulsating heat pipe are the number of turns, length and area of the heating, adiabatic and cooling section, the internal diameter of the tube, which fluid will be used as refrigerant, the temperature and pressure of the refrigerant in the heating section and in the cooling section.

One important design factor is the filling ratio, which corresponds to the amount of liquid refrigerant that fills the tube compared to the tube total volume. In this case there are five defined ranges in which a pulsating heat pipe can be used, and they are:

Table 1 – Pulsating heat pipes characteristics for different filling ratios.

Filling ratio	Characteristics
0 %	The heat conduction occurs only through the tube's material. This is the least efficient way to transport heat and with the highest thermal resistance of the 5 ranges.
Next to 0 %	In this range there is little liquid inside the tube to form the liquid pistons and so there is a strong tendency for wall-drying in the evaporator's wall.
Between 10 and 90 %	In this range is the ideal filling ratio for an ideal performance of the pulsating heat pipe. This ideal point of filling ratio varies with fluid, diameter, thermal load and other performance and design parameters. For lower thermal loads there will be a flow with more vapor bubbles and less liquid for sensible heat exchange. For higher loads, there will be less vapor bubbles and the bubble transport can become difficult and reduce the system's performance.
Next to 100 %	In this range occurs the formation of very few vapor bubbles and these are not enough to generate the necessary forces and oscillations for the bubbling flow. The buoyance induced liquid circulation is hindered by the few vapor bubbles resistance against the flow.
100 %	In this case the system works like a single-phase thermosyphon. No vapor bubbles are formed and no pulsating effect happens but there can be sensible heat exchange through liquid circulation in the tube by thermal induced buoyance.

When studying a pulsating heat pipe, it is of utter interest to know its thermal resistance, the temperature difference in the heating section and in the cooling section and, if possible, the type of flow that is occurring in each section (slug, annular or if the heating section wall is dry it is a signal that the system reached its critical heat flux). Also, it is possible to study the performance of a pulsating heat pipe in different orientations, such as vertical, 30 ° inclined or even in a horizontal orientation.

In order to consider one a pulsating heat pipe, first the tube diameter must be lower than a certain critical diameter. Above this diameter the likeliness that stratified flow may occur is very high and this kind of flow is not desired inside the tube, hence the pulsating denomination. Tibiriçá and Ribatski (2015) studied different diameters and fluids and suggested a new equation to define the critical diameter using the contact angle in consideration. In equation1 Lo stands for Laplace length scale and is given by:

$$Lo = \sqrt{\sigma / g(\rho_l - \rho_v)} \tag{1}$$

In which σ is the fluid's surface tension and ρ_l and ρ_v are the liquid and gas phase densities, respectively. This number relates the forces of surface tension with gravitational forces. Table 2 shows the different criteria suggested in the literature and the critical values found for different fluids at pre specified saturation temperatures.

Table 2 – Critical diameter criteria and values from different authors for different refrigerants.

CRITICAL DIAMETER(MM)	**KEW E CORNWELL**	**TRIPLET ET AL**	**ULLMAN AND BRAUNNER**	**ONG E THOME**	**TIBIRIÇÁ E RIBATSKI**
Criterium	$D = 2Lo$	$D = Lo$	$D = 0.56Lo$	$Lo < D < 3.3\text{Lo}$	$D = Lo\sqrt{8\cos(\theta)}$
R134a(1)	1.6	0.81	0.45	0.8 - 2.6	2.3
R245fa(1)	2.0	1.01	0.56	1.0 - 3.0	2.9
Water-air(2)	5.4	2.7	1.51	2.7 - 8.9	5.5

(1) measured at 31 ºC, (2) measured at 25 ºC.

In this study, the focus was to study the behavior of a single loop pulsating heat pipe by imposing different heat inputs into the system using a direct current font attached directly to the system.

2 Metodology

Mameli et al (2014) suggests using thermocouples inside the pulsating heat pipe evaporator's section in order to study the heat transfer coefficient .This method is intrusive and can lead to a pressure drop in the flow or interfere in the type of flow that is desired to study inside the tube. In order to avoid these interferences inside the system, thermocouples were fixed in the outer wall of the tube.

The chosen microtube arrange of this study was a single loop pulsating heat pipe, in order to facilitate the physical and mathematical modelling and the study of fundamental operational parameters.

To perform the experiments, an experimental bench was set up (0.5 m wide by 0.44 m high) to measure the thermal resistance and other performance aspects using R134a as refrigerant fluid. Figure 5 shows a schematic of the system made for this study.

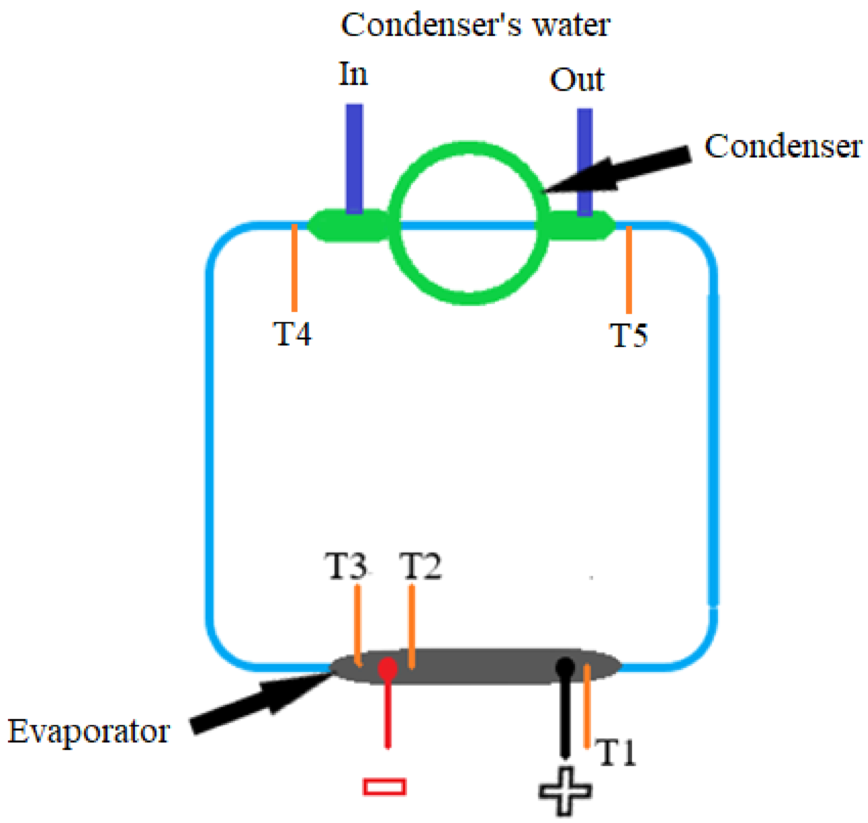

Figure 5 – Schematic drawing of the pulsating heat pipe studied with each section and component pointed out (Author).

In the bottom part of the pulsating heat pipe (as indicated by the evaporator section in figure 5) a 2.23 mm outer diameter stainless steel microtube was used. In the other sections of the pulsating heat pipe 3.175 mm outer diameter Nylon® micro tubes were used. All microtubes used have a 2 mm internal diameter, which complies with Tibiriçá and Ribatski's criterium from table 2.

The temperatures across the system were collected by using type K thermocouples in different positions along the tubes as can be seen in figure 5 indicated by the letter T and the position number.

A Lambda TDK 40-125 power source was used in order to provide heat to the system's evaporator through a controlled direct current across the terminals illustrated in figure 5 by + and -. The voltage reading between the power source's terminals in contact with the microtubes was done with data acquisition software and hardware from *National Instruments*.

The power source's terminals were placed centralized in relation to the extreme sides of the stainless-steel micro tube, being 150 mm apart from each other. As the terminals are fixed, the thermocouples were then fixed (thermocouples T1 to T3) being each one 15 mm distant from each of its terminals as recommended Tibiriçá (2011).

Over the condenser section, as can be seen in the schematic shown in figure 5 the condensation is done by flowing refrigerated water from a refrigerated bath. To monitor the water flow rate an Omega FTB-421 low flow plastic turbine was used in the entrance section.

The temperature data from the thermocouples alongside the tubes and the water flow rate were all monitored with the use of a *LabView* interface together with a *National Instruments* acquisition system.

To perform the experiment the system was disposed in a 90º angle in relation to its base.

The test to obtain the desired performance parameter (thermal resistance) was performed by imposing a direct current to a portion of the evaporator stainless steel micro tube's wall and registering the local temperatures from each thermocouple. All tubes were previously involved with thermal isolating material, the pulsating heat pipe filled with R134a and the condenser filled with water.

2.1. PARAMETERS CALCULATION

In order to measure the thermal resistance, first the heat added to the system was calculated by using the direct current (I) reading from the power source and the voltage (U) read from the acquisition system with the equation:

$$Q = I \cdot U \tag{2}$$

In which Q is the heat added to the system in W. Next the temperature difference is calculated by using the temperature from the thermocouple T3 as the evaporator's temperature and an average temperature from the condenser by using the temperature readings from thermocouples T4 and T5 as can be seen in the next equation:

$$\Delta \mathrm{T} = (T1 + T3)/2 - (T4 + T5)/2) \tag{3}$$

With these factors the thermal resistance is calculated by dividing the temperature difference by the heat added as can be seen from the following equation:

$$\mathrm{R} = \Delta T/Q \tag{4}$$

In which R is the thermal resistance (K/W) and is a key performance parameter when studying pulsating heat pipes since it is a measure of how much resistance the system offers to heat transfer, the lower the thermal resistance the better the system is to transfer heat from one point (evaporator) to other (condenser).

Following the thermal resistance, the heat flux is then calculated using the internal area of the evaporator given by:

$$A_s = \pi \cdot D \cdot L \tag{5}$$

In which L is the length of the micro tube in which the direct current flows, 15mm, and D is the internal diameter, 2 mm. The heat flux is calculated by dividing the heat added to the system by the internal area, A_s as shown:

$$q" = Q/A_s \tag{6}$$

3 Results

After tests for validation of the experimental system were done, the system was studied for two different filling ratios and four different heat flux. The results were obtained using the system at a 90º inclination (vertical) and condenser temperature was kept at 5 ºC. Figure N shows the thermal resistance obtained in the experiment.

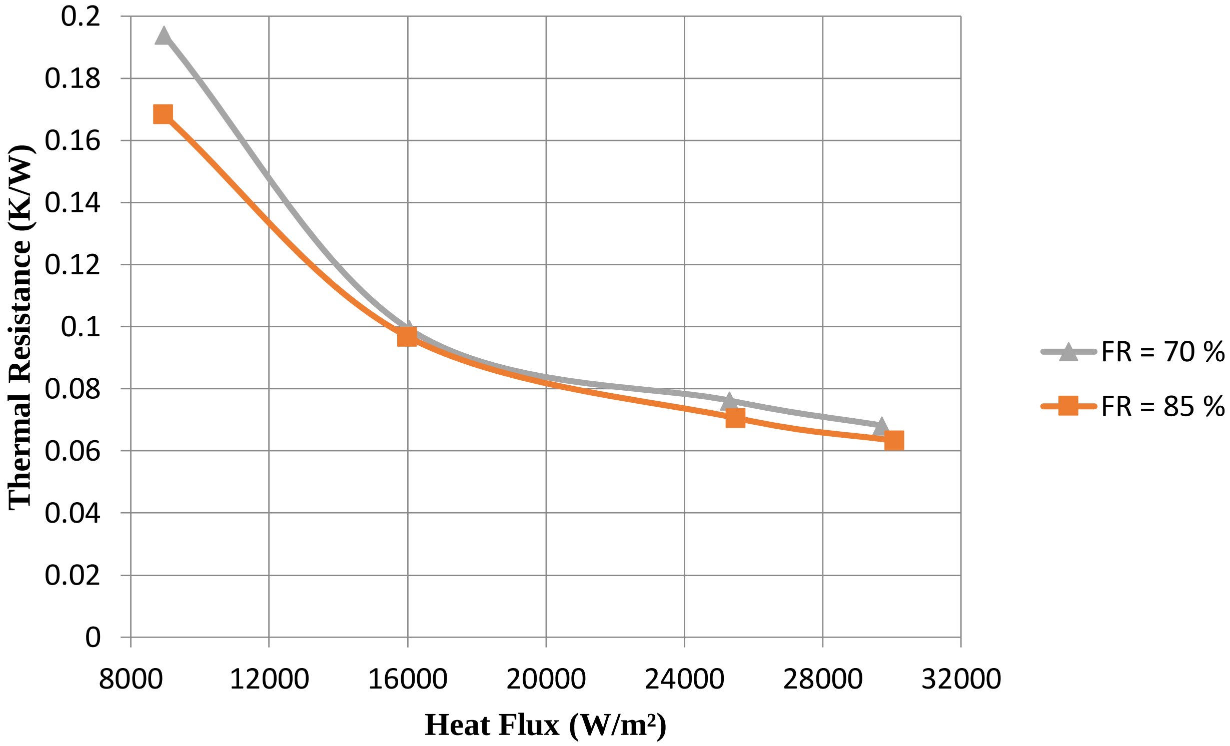

Figure 6 – Thermal performance of the one loop pulsating heat pipe for two different filling ratios with different heat flux applied to the system (Author).

For a filling ratio of 70 % the system presents a slightly higher thermal resistance than with a filling ratio of 85 % and it is possible to see an increase in the system performance with higher heat flux. With 85 % filling ratio the system shows a better performance, which indicated that an ideal filling ratio in this case might in a higher filling ratio or between 70 and 85 %.

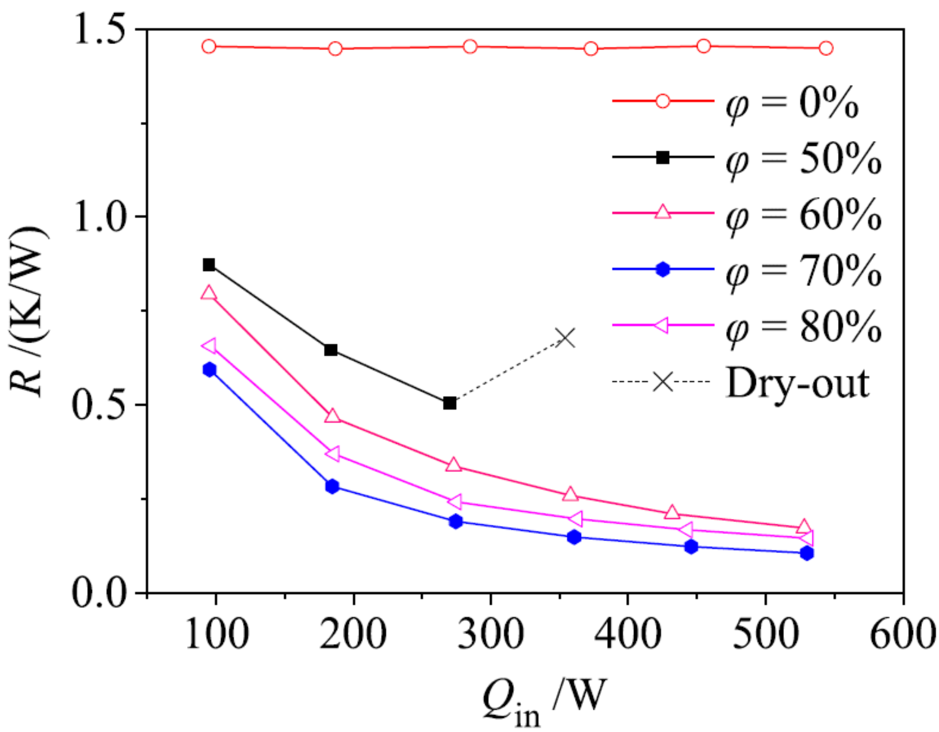

Figure 7 –Thermal performance of a multiturn pulsating heat pipe with methanol for different filling ratios (Deng, 2017)

Comparing these results with the ones obtained by Deng et al. (2017) its possible to see there is a correspondence in the fact that as heat load increases the thermal resistance decreases, meaning a better performance. Also it is shown in Deng's results how advantageous the use of refrigerant is compared to using no refrigerant inside the tube, reinforcing the pulsating heat pipe capability in heat exchange.

4 CONCLUSIONS

The system built in this study proved to be effective and functional as a mean to study the behaviour and performance of a pulsating heat pipe, with part of its body made of a polymerical material, Nylon®, of which the literature has few or no reference at all. The results show the system has a good performance for a single loop pulsating heat pipe, achieving values of approximately 0.06 K/W of thermal resistance in the highest heat flux tested. Future works involve studying different fluids, filling ratios, heat flux and inclination of the system.

5 ACKNOWLEDGEMENTS

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