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HEAT TRANSFER COEFFICIENT MEASUREMENTS IN A PULSATING HEAT PIPE

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Abstract. The goal of this study is to study the local heat transfer coefficient in a single loop pulsating heat pipe with a stainless-steel evaporator using R134a as refrigerant fluid. It has been found that the heat transfer coefficient varies from 2000 to up to 9000 W/m2-K depending on the filling ratio and heat input in the system. These values could be predicted by using specific flow boiling prediction methods.

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.

There are mainly three types of pulsating heat pipes, which are shown in Fig. 1. 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.
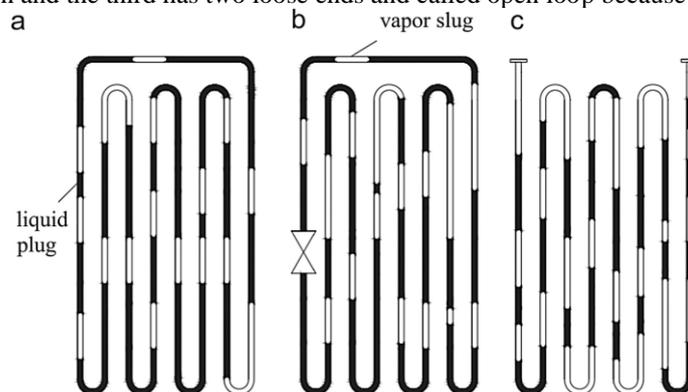

Figure 1 – The three mainly types of pulsating heat pipes: (a) closed-loop pulsating heat pipe; (b) closed-loop pulsating heat pipe with check valve; (c) open-loop pulsating heat pipe, Bastakoti et al (2018)

There are three distinct regions in a pulsating heat pipe which are important when studying its performance: the heating section (evaporator), the adiabatic section and the cooling section (condenser). These are shown in Fig. 2 below.
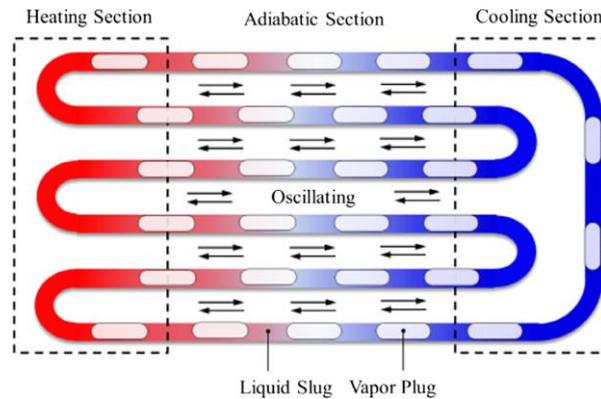
Figure 2 – The three sections of a pulsating heat pipe, Nazari et al (2018).

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 fluid motion towards the cooling section. In the cooling section, heat is extracted, and condensation occurs because the pressure inside the vapor bubbles decrease. The fluid motion inside a pulsating heat pipe is kept by this cycle of heating and cooling.

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).

The higher heat transfer capacity of pulsating heat pipes can be explained in figure 3: using a pure copper bar to transfer heat from the heating part to the cooling part is not 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.
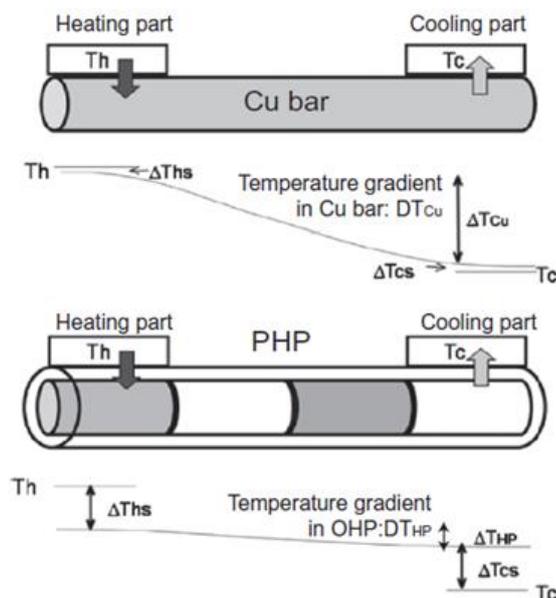
Figure 3 – A schematic showing why a pulsating heat pipe is a better choice than a copper bar using conduction, Natsume et al (2011).

Important factors when designing a pulsating heat pipe are: number of turns, tube material, 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.

In this study, the focus was to study the thermal 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. Also, there are few studies in the literature that try to study the pulsating heat pipe heat transfer coefficient as one of its thermal performance indicators.

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 arrangement of this study was a single loop pulsating heat pipe, in order to facilitate the physical and mathematical modelling.

The experiment consists of a single loop pulsating heat pipe with 2.2mm diameter stainless steel and NYLON® tubes. In the evaporator section the tube is made of stainless steel in order to receive the direct current and transport the heat to the fluid through Joule effect. Thermocouples were placed to measure the fluid temperature before and after heating (T_{entry} and T_{sat}, respectively), before and after condensation (not shown here). Figure 5 shows a schematic of the evaporator section made for this study.
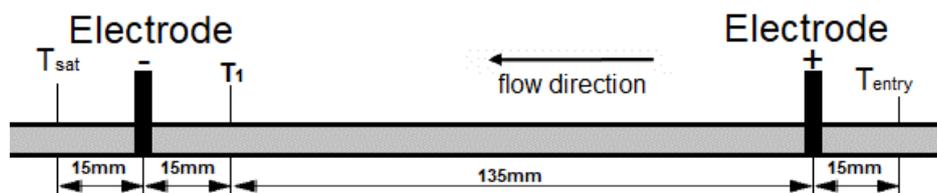

Figure 4 – Schematic drawing of the pulsating heat pipe evaporator section studied with each component pointed out (Author).

In T_1 there four thermocouples installed along the tube outer wall equally spaced by 90° from one another which will be the base for the heat transfer coefficient calculation. All thermocouples used in the study are type K thermocouples calibrated with less than 0.1°C uncertainty.

In order to estimate the fluid mass velocity an optical method was used, which consisted of filming the fluid passing through the condenser using a high-speed camera and analysing the frames to estimate the fluid velocity. With this velocity and the cross sectional area it is possible to calculate the mass velocity.
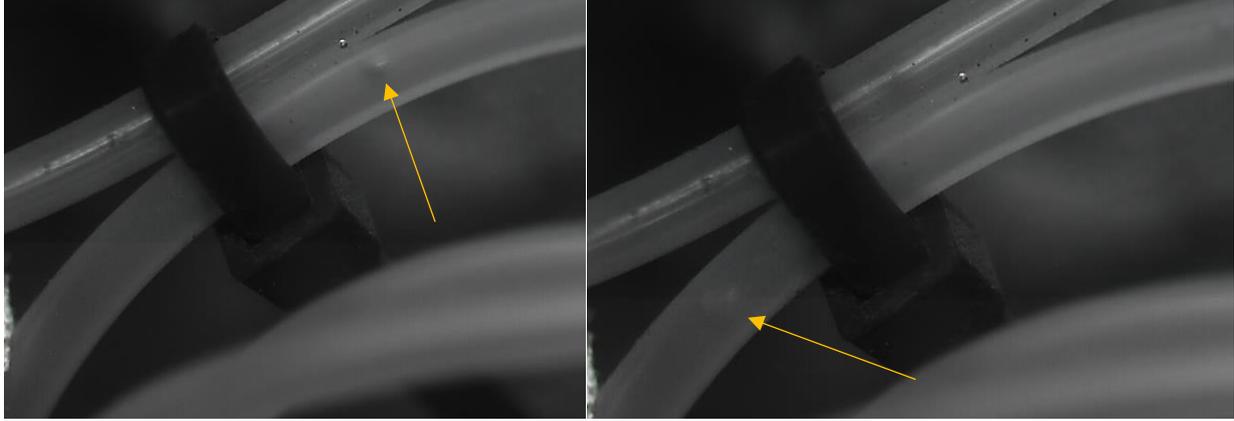
Figures 5 and 6 – Frames from the high-speed camera showing the vapour bubble 30 frames apart. (Author).

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 T_1, T_{entry} and T_{sat}) being each one 15 mm distant from each of its terminals as recommended Tibiriçá (2011).

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

The test to obtain the desired performance parameter (heat transfer coefficient and 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 different ratios of R134a and the condenser filled with water at 5°C.

2.1. PARAMETERS CALCULATION

In order to measure the heat transfer coefficient, 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:

$$Q_{evap} = I \cdot U \tag{1}$$

In which Q is the heat added to the system in W. Next the heat flux is calculated dividing the heat added by the heat exchange area, which is as follows:

$$Q" = Q_{evap} /(pi \cdot L \cdot D) \tag{2}$$

With L being the total heated length and D the internal diameter of the tube. The heat transfer coefficient is then calculated using the relative length of the thermocouple, L_1 relative to the total heated length, the heat flux and the temperature difference between the entry temperature T_{entry} and the average from the four thermocouples in T_1. The equation the following:

$$h = \frac{L_1}{L} \cdot \frac{Q"}{T_1 - T_{entry}} \tag{3}$$

Also, with the heat Q_{evap} it is possible to calculate the local vapour quality with the following equation:

$$x_{local} = \frac{L_1}{L} \cdot \left(\frac{Qevap}{\dot{m}} + h_{ent} - h_l \right) /(h_{lv}) \tag{4}$$

With h_{ent} being the enthalpy of the fluid at $T = T_{sat}$ and $P = P_{sat}(T = T_{sat})$, h_l is the liquid fluid enthalpy with $T = T_{sat}$ and h_{lv} is the vaporization enthalpy at $T = T_{sat}$.

3   Results

The system was studied for two different filling ratios and different heat flux. In the case of a quality higher than 0.7 the heat added was lowered in order to avoid dry-out. Figure 7 shows the heat transfer coefficients obtained in the experiment.
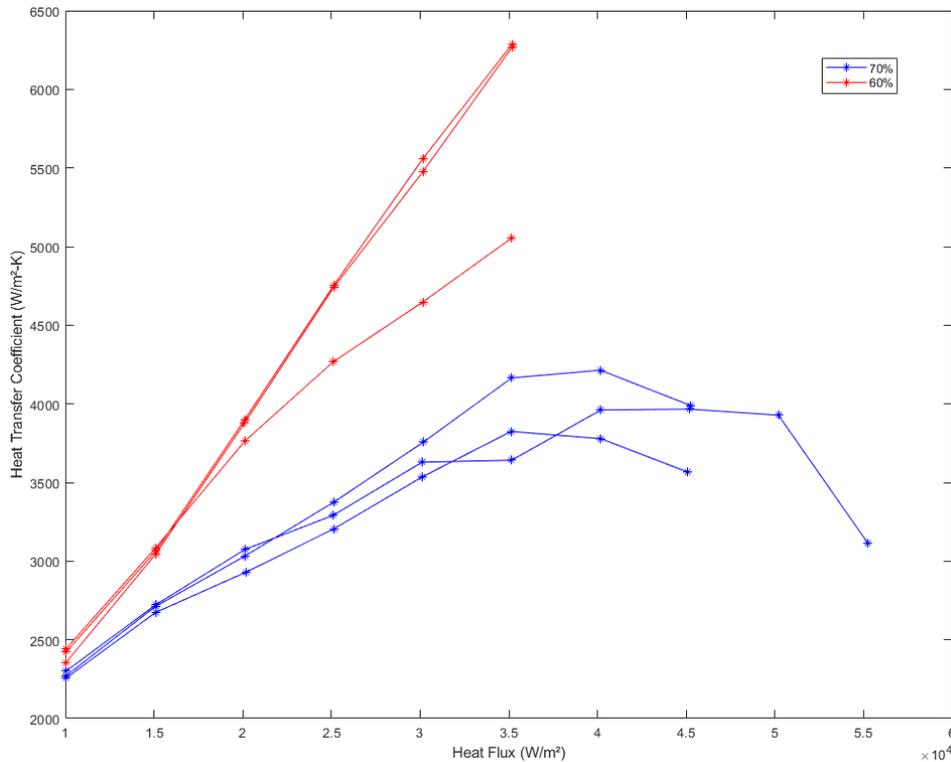

Figure 7 – Thermal performance of the one loop pulsating heat pipe for two different filling ratios with different heat flux applied to the system (Author).

For a filling ratio of 60 % the system presents a higher heat transfer coefficient than with a filling ratio of 70 % and it is possible to see an increase in the system performance with higher heat flux. However, with a lower filling ratio the system reaches the critical point of quality higher than 0.7 in a lower heat flux when compared with a filling ratio of 70%.

The results were then compared with the correlations from Tibiriçá et al (2017) and Kanizawa et al(2016) and can be seen in the next figures.

Both correlations show that they can predict most of the results within a 30% error, with both correlations predicting with a higher error in the 70% filling ratio.
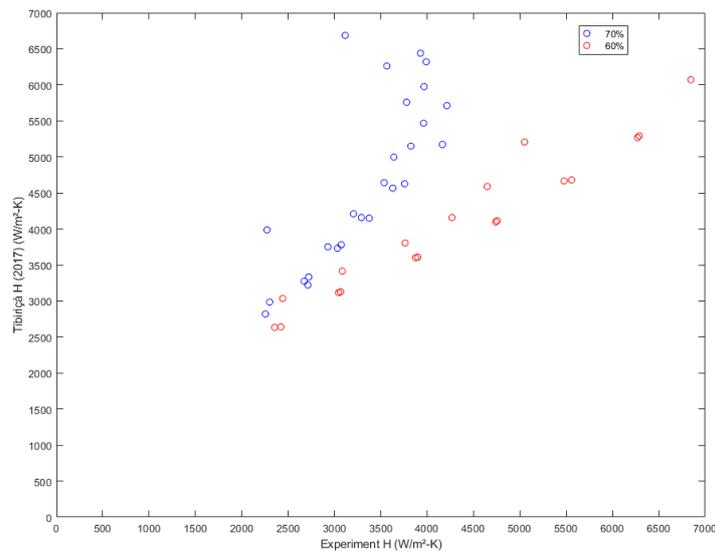
Figure 8 – Experimental heat transfer coefficient compared with the predicted from Tibiriçá et al correlation (2017). (Author)
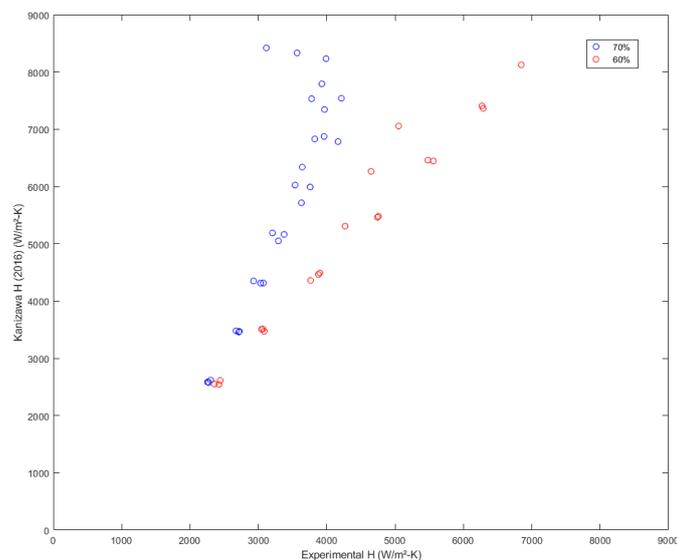

Figure 8 – Experimental heat transfer coefficient compared with the predicted from Kanizawa et al correlation (2016). (Author)

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 8200W/m²K heat transfer coefficient 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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