Mini-scale pulsating heat pipe cooling systems for high-heat-flux electronic equipment

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Abstract. This paper presents a novel passive two-phase cooling system for efficient thermal management of high-heat-flux electronic equipment, namely a mini-scale, flat-plate pulsating heat pipe, by executing experiments on a test bench. Furthermore, cooling tests on an actual 1-U server are also shown. The pulsating heat pipe used in this study is designed and manufactured with micro-channel technology where passive two-phase flow circulation is promoted by the inherent two-phase instability and the competing effect between nucleate boiling and thin-film evaporation. Pulsating heat pipes properly operate with gravity (in vertical orientations) and without gravity (in horizontal orientations) and they are an optimal solution to dissipate a large amount of heat without the need of active drivers (absence of pumping power). The cooling capabilities of the proposed pulsating heat pipe were investigated using different near zero global warming potential refrigerants at various filling ratios to find the optimal thermal performance up to 800 W (corresponding to a heat flux of 25 W/cm²). The minimum overall thermal resistance was measured to be 0.065 K/W, which includes the thermal interface material and the secondary side. This paper shows that passive two-phase cooling using pulsating heat pipes can be a very promising and reliable technology for achieving high power dissipations of next-generation datacentres (and electronic equipment in general).

1. Introduction

As electronic equipment is becoming more and more widespread in critical applications such as data centres, telecommunication, aerospace and others, engineers are facing the challenge to design proper and innovative cooling element able to reject a significant amount of heat in small spaces, while keeping the efficiency to high standards. In response, pulsating heat pipes have gained great momentum in recent years, due to their cooling potential and simplicity in the manufacturing process. The closed-loop pulsating heat pipe has mainly been studied in two configurations: capillary tube pulsating heat pipe (CTPHP) and flat plate pulsating heat pipe (FPPHP).

Han et al. [1] published a comprehensive review of pulsating heat pipes, covering fundamentals, theoretical researches, and applications. Until the date of their paper, the effects of operational characteristics and mechanisms on the thermal performance were still unclear, and many questions remain to this day. A more recent review was done by Bastakoti et al. [2], which widened the technical
aspects knowledge of pulsating heat pipes, and reported works on enhancement techniques, such as nanofluids and surfactants.

Thompson et al. [3] designed and tested an FPPHP with staggered microchannels, each with a height of 1.7 mm and a width of 1.175 mm. They reported an overall thermal resistance of 0.08 K/W with a heat load of 300 W. Ayel et al. [4] experimentally studied an FPPHP with a rectangular cross-section of $1.6 \times 1.7$ mm$^2$ with milled grooves between the channels. The maximum recorded heat load dissipation was 180 W with an overall thermal resistance of 0.12 K/W. The authors performed a parabolic flight test demonstrating that the variation of the gravity field rapidly influences the FPPHP in vertical inclination. At the same time, no observable effect was detected in horizontal inclination. Cecere et al. [5] reported experimental data for an FPPHP filled with water and a self-rewetting mixture. The minimum reported thermal resistance was 0.1 K/W for a heat load of 200 W. The self-rewetting fluid performed better in microgravity conditions, while water yielded better results in normal gravity. A comparative study between CTPHP and FPPHP was conducted by Takawale et al. [6]. They asserted that, under conditions of similarity in the geometry, CTPHP performed better than FPPHP. Their study showed evidence that the fluid dynamics in capillary tubes, given the absence of lateral conduction, was characterised by higher amplitudes in the pulsation and more chaotic behaviour.

Interesting findings were reported by Lim et al. [7]. They showed that non-uniform channel layout and heating pattern on the evaporator zone contributed to increasing the thermal performance of a micro FPPHP. The heat load dissipated was, however, very low (12 W).

The literature review reported above shows that the lack of a clear understanding of the thermo-fluid dynamics of pulsating heat pipes is still the main cause to achieve very low overall thermal resistance with this technology. In this paper, it is shown how an FPPHP properly design can dissipate heat loads as high as 850 W with a thermal resistance lower than 0.07 K/W, using zero-GWP working fluids.

2. Cooling element and test bench description
The novel cooling element studied in this paper can be shortly described as a flat plate pulsating heat pipe (FPPHP). However, the particular arrangement and structure of the internal channels exploit gravity and capillary forces as well. Because of confidentiality, no details or picture of the internal components can be shared here. The evaporator, adiabatic and condenser plates are stack one on top of the other, with channel sizes providing Bond numbers in the range 1.6 – 3.2. In Figure 1, a picture of the FPPHP is reported for a demonstrative purpose. It is worth noticing that two ports for the secondary side are present, as well as a charging fitting for the working fluid used for the internal channels. Excluding the ports for the secondary side and the charging fitting, the FPPHP has a footprint of about 60x60 mm$^2$ and a height of 23 mm. It is entirely in aluminium with the exception of the charging fitting, which is a commercial 1/4-inch brass fitting. The total weight is approximately 230 grams.

As per the working fluids, only zero-GWP refrigerants have been selected and tested to promote environmental care: R1233zd(E), R1234ze(E) and R1234yf. In the tests run for this study, only water has been used as the fluid for the secondary side. However, it is worth mentioning that the final intent for the technology is to use a refrigerant for the secondary side, which is part of a thermosyphon loop.
2.1. Test bench
A schematic cross-section of the test bench is reported in Figure 2. A custom-made Teflon block provides the seats and insulation for four electrical heaters. The electrical heaters (WATLOW - Ultramic 600, footprint 12x12 mm\(^2\)) are selected to emulate the heat load of a generic electronic device, such as a CPU or an IGBT module, up to a power of 850 W. They are electrically connected to an Aim-TTi power supply CPX400DP (accuracy: 1%). The purpose of the copper plate is two-fold: firstly, it functions as heat spreader; secondly, it provides the seats for four thermocouples. The cooling element FPPHP is located on top of the copper plate and is kept in place with a clamp. Two thermocouples are used for the water inlet and outlet ports, as depicted in the figure. A thermostatic bath supplies the water flow at a steady temperature.

The thermal contact between the heaters and the copper plate, and between the copper plate and the FPPHP is guaranteed with a commercial thermal paste THERMALRIGHT TF-8 (thermal conductivity 13.8 W/mK).

All the thermocouples were calibrated in-house against four Pt-100 type thermometers. The final uncertainty, considering Pt-100’s accuracy and scattering, and the residual of the regression method, was evaluated to be lower than 0.4 K.

Figure 3 shows the components of a server used to demonstrate the performance of the novel cooling element in an actual application. The server (HP ProLiant DL160 G6) has two CPUs (Intel Xeon X5650) with a Thermal Design Power of 95 W. Commonly, the commercial cooling solution is based on an air-cooled heat sink. Further details are reported in the experimental results section.

![Figure 1. Picture of the cooling element FPPHP.](image1)

![Figure 2. Schematic cross-section of the test bench.](image2)
3. Experimental results
The data (temperatures from the thermocouple and power from the power supply) were acquired during the tests via a custom-made MATLAB script. The time window upon which the mean is calculated has been set to 3 minutes. The sample time was 0.5 seconds. The main quantity discussed here is the thermal resistance \( R \), defined as:

\[
R = \frac{T_{\text{hot}} - T_{\text{cold}}}{Q}
\]

where \( T_{\text{hot}} \) is the mean value of the temperatures measured at the copper plate, \( T_{\text{cold}} \) is the mean value of the temperatures measured at the water inlet and outlet ports, and \( Q \) is the heat load generated by the power supply. According to the uncertainty of the instruments provided in Section 2, the uncertainty of the overall thermal resistance ranges between 1 and 10 %, with the highest values resulting from the tests at very low heat loads. It should be noted that the thermal resistance as defined in Equation 1 takes the resistance of the TIM into account. Such thermal resistance, considering an estimated roughness between the copper plate and the FPPHP of 0.2 mm, amounts to 0.005 K/W.

As per the temperature conditions, all the tests performed here have always been set with the thermostatic bath at 20 °C.

For all the conditions tested, the values of refrigerant charge for the working fluids have been chosen to achieve a range of filling ratio in the range of 35 – 43 %, even considering the effect of the saturation temperature. For safety reason, the maximum value of the heat load for each refrigerant was set in the occurrence of a temperature of 90 °C measure in the copper plate.

3.1. Thermal resistance on varying the working fluid, fluid charge and orientation
The experimental data of the cooling element FPPHP in horizontal orientation are reported in Figure 4. It is convenient to divide the data into three groups, one for each working fluid, with each group containing two fluid charges.

The blue and red data points come from the testing of R1234ze(E) as the working fluid, with 3.8 and 4.7 grams, respectively. The curves extrapolated with those data points show a minimum of respectively 0.067 and 0.072 K/W at a heat load of 200 W. Evidently, a fluid charge higher than 3.8 grams makes the condenser flooded, with consequently dampening on the pulsations and increasing the subcooling, which generally detrimental.

The yellow and purple data points are acquired using R1234yf as the working fluid, with 3.5 and 4.1 grams. The shape of the extrapolated curves is similar to the ones experienced with the R1234ze(E). However, the minimum is found at 150 W with values of 0.071 and 0.079 K/W, respectively, and they afterwards increase with a steeper slope.

The green and azure data points are referred to the working fluid R1233zd(E), with a fluid charge of 4.1 and 3.8 grams, respectively. This refrigerant shows a significant improvement in comparison to the previous two. The minima of the curves, 0.068 and 0.065 K/W, respectively, are found around a heat load of 500 W, and the slope is much flatter. The dampening effect of higher fluid charges is also less prominent. These are all very desirable features for a cooling element aiming to be applied to high-heat-flux devices.

It seems evident that low-pressure refrigerants are better as the working fluid in regards to thermal performance. The causes that make the refrigerant R1233zd(E) better than R1234ze(E) and R1234yf can be explained comparing the thermophysical properties. Table 1 provides a quick summary of the relevant properties of the three fluids tested (assuming a saturation temperature of 25 °C, from RERPROP 10.0 [8]). R1233zd(E) has a much higher ratio between the densities of the liquid and vapour phases, as well higher latent heat of vaporisation and thermal conductivity. The vapour plugs are evidently more confined for the R1233zd(E), as the low Bond number indicates. The confinement helps the working fluid to reach a circulating flow regime earlier in respect of the applied heat load, flow regime that is more efficient than others. A discussion on flow regimes in pulsating heat pipes observed for the FPPHP is beyond the scope of this paper, but the authors plan to do it in a future paper.

![Figure 4. Thermal resistance against heat load for the FPPHP with different working fluids and fluid charges.](image-url)
Saturation properties evaluated at 25 °C

|                        | R1233zd(E) | R1234ze(E) | R1234yf |
|------------------------|------------|------------|----------|
| Saturation pressure [bar] | 1.2981     | 4.9852     | 6.8258   |
| Liquid density [kg/m³]  | 1262.8     | 1163.1     | 1091.9   |
| Vapour density [kg/m³]  | 7.2061     | 26.321     | 37.925   |
| Latent heat [kJ/kg]     | 191.15     | 166.92     | 145.37   |
| Liquid thermal conductivity [mW/m-K] | 82.747 | 74.216 | 63.538 |
| Surface tension [mN/m]  | 14.561     | 8.8812     | 6.1726   |
| Bond Number             | 1.61       | 2.39       | 3.19     |

Table 1. Thermophysical properties of saturated liquid and vapour phases for the working fluids tested.

In Figure 5, a comparison between horizontal and vertical orientations for the FPPHP is reported. Refrigerant R1233zd(E) is the only working fluid for this comparison. The thermal resistance resulted in being only slightly worse, ranging from a value of 0.070 K/W at 200 W to 0.073 K/W at 800 W. These results show that gravity has still a role in affecting the thermal performance of the FPPHP, but that role is marginal.

Figure 5. Thermal resistance against heat load for the FPPHP at different orientation, using R1233zd(E) as the working fluid.

3.2. Thermal performance on an actual server

The cooling element FPPHP has also been tested on an actual server (HP ProLiant DL160 G6), and the performance has been acquired with the CPU temperature readings via the software HWMonitor by CPUID. As previously shown, the server has two CPUs (one close to fan cage, the other farther away) and the commercial cooling solution provides for two air-cooled heat sinks. For the following tests, the heat sink close to the fan cage has been replaced with a water-cooled FPPHP, with the water flow coming from the thermostatic bath steady at 20 °C. The CPUs has been put under the maximum working load via the software CPU-Z by CPUID. It is worth mentioning here that the Thermal Design Power (TDP) of the CPUs (Intel Xeon X5650) is 95 W, but actual power measured by the software was 109 W at the maximum load.
Figure 6 shows the temperature readings of the two CPUs. The stress test is composed of three stages: 1) 1 minute at an idle state; 2) 10 minutes at the maximum working load; 3) 1 minute at an idle state. The air-cooling solution shows a CPU temperature of 58 °C at the initial idle stage, whereas it increases to 88 °C when working at the maximum load. On the other hand, the much more efficient solution of the FPPHP is able to maintain the CPU at 22 °C when idle, while a temperature of 42 °C was measured at the maximum load. It is worth noting that the overall thermal resistance that is possible to calculate with the CPU temperature reading (0.18 K/W) is not precisely equivalent to the one experimentally determined in the previous section. In this case, the CPU itself has an additional thermal resistance due to the internal silicon layer.

Nonetheless, a temperature of 42 °C for a load of 109 W is very attractive for two reasons: firstly, it demonstrates that more powerful CPUs can be used in servers without any risk of damage; secondly, it shows the possibility of cooling down the CPUs with water at 40-50 °C, keeping the junction temperature at values lower than 75 °C, paving the way to actual and feasible waste heat recovery applications.

4. Conclusions
In this paper, it has been shown how a properly designed pulsating heat pipe could be a desirable technology for the cooling of high-heat-flux electronic devices. Two types of tests have been reported here. Firstly, the FPPHP has been tested with different working fluids on a suitable test bench with electrical heaters and thermocouples to determine an overall thermal resistance experimentally. The FPPHP with the refrigerant R1233zd(E) yields the optimal solution, being able to dissipate up to 800 W with a thermal resistance of approximately 0.065-0.068 K/W. The FPPHP has also been tested on an actual server, keeping the CPU at 42 °C when a workload of 109 W was applied to it. The reported efficiency can allow not only to enormous savings for the energy management of data centres but also makes waste heat recovery an actual possibility in this field.

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