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EXPLORING THE INFLUENCE OF CHANNEL DIAMETER, FILL RATIO, AND ADIABATIC LENGTH VARIATION ON LONG OSCILLATING HEAT PIPES

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ABSTRACT

Oscillating heat pipes (OHPs) represent a promising advancement over traditional heat pipes, yet their operational boundaries, especially for long OHPs, remain insufficiently understood. This study investigates the impact of varying adiabatic length, channel diameter, and fill ratio on thermal performance, crucial for assessing their suitability for engineering applications like spacecraft thermal management. Three long OHPs, ranging from 451 mm to 770 mm in total length, were subjected to multiple performance tests, employing channel diameters of 1.1 mm and 1.9 mm, along with adiabatic lengths of 305 mm and 610 mm. The experimental setup involved mounting the OHPs onto a testbed, monitored by nine K-type thermocouples. The tests, conducted horizontally to eliminate gravity-assistance, revealed that thermal performance is significantly influenced by channel diameter, adiabatic length, and fill ratio. Notably, optimal performance was observed at a 50% fill ratio, while reductions in diameter hindered start-up at a 70% fill ratio and failed to start-up at 30% fill ratio. These findings highlight the limitations of long OHPs, which is crucial to determine the limits of their applicability and dimensional constraints.

NOMENCLATURE

k	Thermal Conductivity (W/mK)
G	Conductance (W/K)
$\dot{Q}$	Heat Transfer Rate (W)
T_{Evap}	Average Temp at Evaporator (°C)
T_{Cond}	Average Temp at Condenser (°C)
I	Current (A)
V	Voltage (V)
Δx	Adiabatic Length (mm)
L_c	Condenser Length (mm)
L_e	Evaporator Length (mm)
δ	Uncertainty from measurement

1. INTRODUCTION

Oscillating heat pipes (OHPs) have garnered significant interest in recent years as efficient heat transfer devices for various thermal management applications. Their ability to transfer heat quickly over long distances makes them promising candidates for cooling electronic devices. The performance of these long OHPs is dependent on channel diameter, fill ratio, and adiabatic length variation on the performance of long oscillating heat pipes. It has been well-established that changing the channel geometry such as the size of the diameter can affect OHP performance. [1]

In experimental work done by Lee et al. [2] they found that increasing the channel diameter caused thermal resistance to go down, thus increasing the thermal performance of the OHP. It was also observed that larger channels could handle greater heat loads before dry-out. They observed a similar trend with a non-circular channel as well. In a related study done by Kammuang-Lue et al. [3], results concluded that increasing the internal channel diameter from 1 to 2 mm decreased the thermal resistance for the chosen working fluids: R123, ethanol, and water. This decrease in thermal resistance was attributed to a larger heat transfer area between the heat pipe and working fluid. Increasing the channel diameter also increases cross-sectional area for fluid flow, facilitating a higher quantity flow from the evaporator to the condenser sections.

The fill ratio, defined as the ratio of the volume occupied by the working liquid and the total volume of the heat pipe, significantly influences the operational characteristics of OHPs. A study conducted by Czajkowski et al. [4] investigated the effect of fill ratio on the thermal performance of long OHPs by testing fill ratios ranging from 20-75% with different fluids. They saw that higher fill ratios had lower thermal resistance and operated at higher heat inputs. The experimental work done by

Mameli et. al [5] included testing three different fill ratios with multiple orientations. They tested 50%, 70%, and 90% fill ratios, and they measured thermal performance using thermal resistance. They saw that the 50% fill ratio had the lowest thermal resistance and when they reoriented their OHP to a vertical position it yielded the same results. When changing the fill ratio, the start of the OHP can be affected as well. In an experiment conducted by Wu et .al [6] fill ratios ranging from 22-80% were tested. They saw that as the fill ratio increased, thermal resistance decreased. The lowest thermal resistance observed was at 48% fill ratio. It was also observed that when the fill ratio is lowered, the OHP would start-up at lower heat loads. In a study conducted by Rudresha et al.[7], various fill ratios ranging from 45% to 75% were examined using ethylene glycol as the working fluid. Their findings revealed that a fill ratio of 55% exhibited optimal thermal performance. Moreover, as the fill ratio approached 55%, thermal resistance decreased proportionally. This shows the intricate relationship between fill ratio and thermal performance.

The segment of the heat pipe where heat transfer does not occur, known as the adiabatic region, has been shown to significantly influence the operational efficiency of an OHP. The study conducted by Czajkowski et al.[4] revealed that increasing the adiabatic length enables OHPs to sustain higher heat loads without the fluid inside drying out during initial heating. This is an important factor to consider because if the fluid dries out it will no longer start-up; furthermore, it affects the design of the OHP and where it can be used. Additionally, Sukchana et al. [8] conducted a comprehensive series of experiments aimed at assessing the impact of varying adiabatic lengths on the thermal performance of OHPs. Their study encompassed a range of adiabatic lengths, from relatively short to long configurations, allowing for a detailed exploration of the influence of this parameter. Contrary to prior studies, their findings revealed that shorter adiabatic lengths consistently resulted in superior thermal performance across various operating conditions. In a study conducted by Qu et al. [9] they tested various OHPs with different configurations and adiabatic lengths. In the four different OHPs that were tested, they saw that as adiabatic length increased the start-up temperature increased as well. They also saw a decrease in thermal resistance in the shorter OHPs. In another work done by Liu et al. [10] they observed a decrease in thermal resistance when adiabatic length is increased. It was also observed that the start-up time would be longer which would result in a higher start-up temperature.

The focus of this paper is to understand how different channel diameters, adiabatic lengths, and fill ratios affect the performance of OHPs. The remainder of the paper is divided into 4 sections. Section 2 describes the experimental setup and materials used to conduct the tests. Section 3 is the results and the analysis of the tests done on the OHPs. Finally, Section 4 is the conclusion of the paper to summarize our findings.

2. MATERIALS AND METHODS

Three copper OHPs were constructed for the experiments. All OHPs had 18 turns. One OHP had an adiabatic length of 610 mm, channel diameter of 1.9 mm, evaporator length of 115mm, and condenser length of 45mm. The other two OHPs had an adiabatic length of 305 mm, channel diameter sizes of 1.1 mm and 1.9 mm, evaporator length of 76mm, and condenser length of 70mm.The lengths of the OHPs were chosen from prior work done by Miesner et al. [10]. All tests were done in a horizontal orientation to remove gravity assistance so it can more accurately mimic what OHPs will experience in space. Table 1 shows the naming convention for these OHPs hereafter. The working fluid in the OHPs was R123. The tests were conducted in a horizontal orientation and were gravity reset on the evaporator side after every test. All OHPs were insulated to minimize heat loss.

TABLE 1: OHPs Tested

OHP Name	Adiabatic Length (mm)	Channel Diameter (mm)	Evap. Length (mm)	Cond. Length (mm)
OHP A	610	1.9	115	45
OHP B	305	1.1	76	70
OHP C	305	1.9	76	70

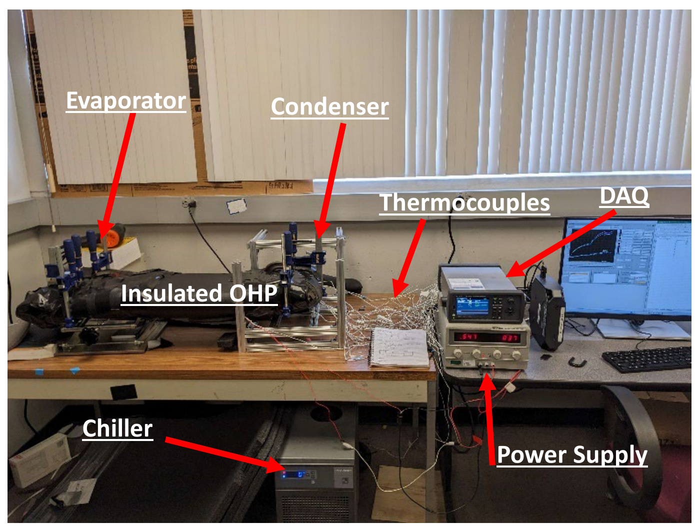

FIGURE 1: Experimental setup for all OHPs

Figure 1 shows the experimental setup for OHP testing. Once constructed, OHPs were connected to a pressure fed fill station and charged with the working fluid. Nine K-type thermocouples were placed on the OHP as shown in figure 2. The thermocouples are held in place using a base layer of double-sided thermal tape with the thermocouple placed on the outer surface, followed by covering the thermocouple with another piece of thermal tape. Support plates hold the OHP at the evaporator and condenser sections; these plates are made of composite material and aluminum, respectively. The OHP is then sandwiched between the support plates and aluminum cover plates. The plates are lined with gap filler to limit air gaps that exist between the serpentine structure of the OHPs. A composite material is used for the evaporator base plate to limit the amount of heat absorbed from the supporting plate. The evaporator cover plate has a heating element attached to it which is hooked up to a GW Instek GPR-30H10D power supply; this serves as the heat source. The condenser base plate has built in channels which are connected to a PolyScience LS5 Benchtop Chiller; this serves as the heat rejection point. Once the OHP is mounted on the plates, the system is insulated with insulation foam shown in Figure 1. The chiller is preset to 10 ºC. Temperature data is monitored through the thermocouples using a Keysight DAQ970A data acquisition system (DAQ). Steady baseline data was gathered for about 10 minutes with no power input and condenser temperature at 10 ºC. Data collection began after acquiring baseline data beginning at 10 watts to a maximum of 65 watts or when the heater temperature reached 100 ºC, whichever came first. The power was increased by 10 watts after reaching steady-state temperatures in addition to some idle time at steady-state temperatures. The typical time between increasing the power input was in the range of 1.5 hours. The data was saved as a csv file.

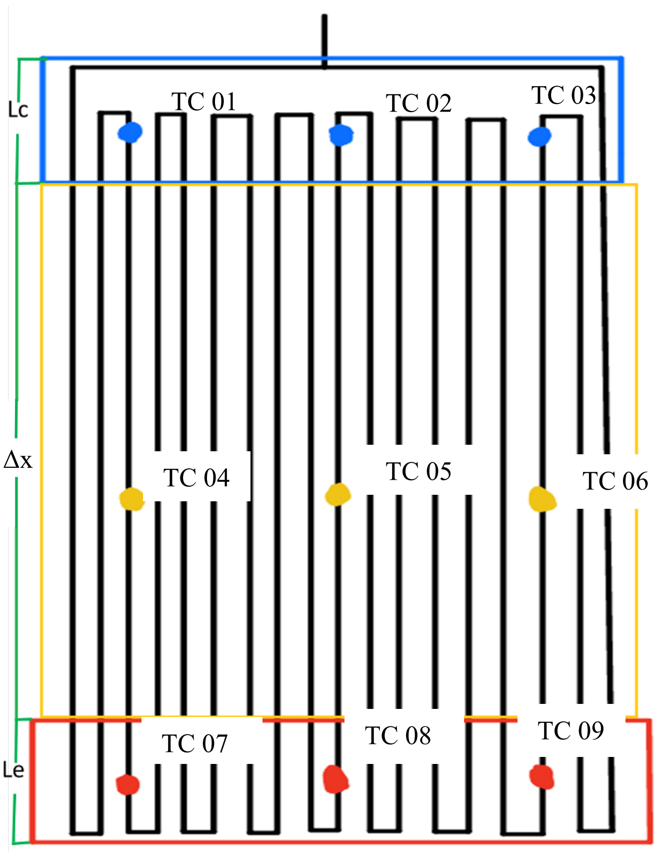

FIGURE 2: THERMALCOUPLE LAYOUT FOR THE EXPERIMENTAL SETUP ON ALL OHPS

Figure 2 shows the thermocouple layout that was used for all the experiments. Nine K-type thermocouples were used to record the data for analysis. There were three thermocouples for each section of the OHPs and one T-type thermocouple on the cold plate, and another on the evaporator heating pad to ensure safety. The thermocouple on the heating pad was carefully monitored to ensure the temperature did not go past 100°C.

3. RESULTS AND DISCUSSION

To ensure heat loss is negligible during the experiments, a test with OHP B was run with eight thermocouples on the outer surface of the insulation. The heat input was set to 65 watts until the temperature of the heater reached 90 °C. The peak average temperature of the outer surface was measured to be 25.7 °C and the temperature of the air was 23.3 °C. At this temperature, the total heat loss is estimated to be 0.76 W. Since the heat loss was about 1% of the heat input the heat loss can be considered negligible.

The heat input at the evaporator section was determined by multiplying the supply voltage by the current measurements.

$$\dot{Q} = I \times V \tag{1}$$

The conductance of the OHP was calculated by dividing the power input by the average temperature difference between the evaporator and condenser sections. The average temperature difference was calculated by taking the average temperature among thermocouples 7, 8, and 9 for the evaporator, minus the average temperature among thermocouples 1, 2, and 3 for the condenser.

$$G = \frac{\dot{Q}}{(T_{Evap,avg} - T_{Cond,avg})} \tag{2}$$

Thermal conductivity was calculated by multiplying the conductance times the length of the adiabatic section divided by the cross-sectional area of the 3.1 mm outer diameter times 18 turns.

$$k = G * \frac{\Delta x}{18 * A} \tag{3}$$

Since multiple tools were used to take measurements there is going to be propagation of uncertainty in the calculations. The uncertainty from the measurements can be calculated by using the equations (4)-(6). The uncertainty propagation from subtracting the temperatures from the evaporator side and condenser side when calculating conductance is calculated using equation (4).

$$\delta T = \sqrt{\delta T_{Evap}^2 + \delta T_{Cond}^2} \tag{4}$$

Then, the uncertainty propagation from calculating conductance is shown in equation (5). The uncertainty for thermal conductivity is shown in equation (6). This equation captures the uncertainty from measuring the cross-sectional area and adiabatic length.

$$\frac{\delta G}{G} = \sqrt{\left(\frac{\delta T}{T}\right)^2 + \left(\frac{\delta I}{I}\right)^2 + \left(\frac{\delta V}{V}\right)^2} \quad (5)$$

$$\frac{\delta k}{k} = \sqrt{\left(\frac{\delta G}{G}\right)^2 + \left(\frac{\delta A}{A}\right)^2 + \left(\frac{\delta x}{x}\right)^2} \quad (6)$$

Figure 3 shows the temperatures of OHP A at 70% fill ratio during operation. Oscillatory behavior in the temperature graph shows evidence of fluid motion, which is often indicative of OHP start-up. At 10W input shown in Figure 3 shows no start-up conditions. The 20W power region shows start-up conditions, as the oscillatory motion becomes apparent and heat transfer between the evaporator and condenser sections occurs.

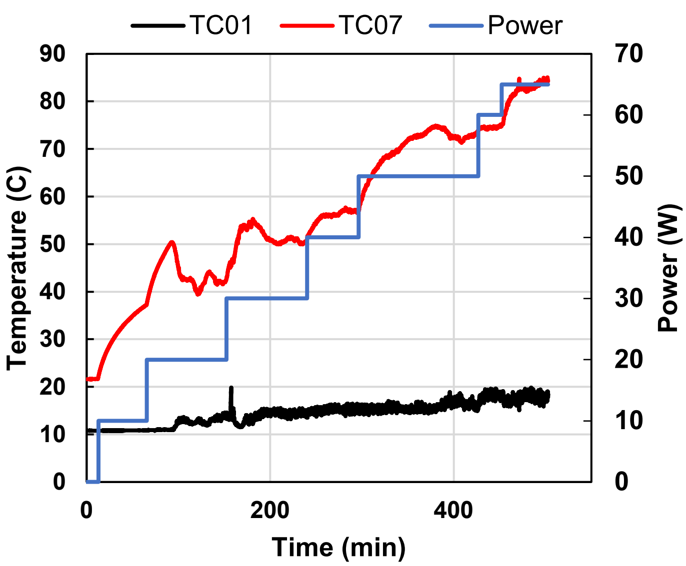

FIGURE 3: TEMPERATURE OF ALL OHP SECTIONS THROUGHOUT THE EXPERIMENT FOR 70% FILL RATIO OHP A

Figure 4 shows the temperature of the OHP B at 70% fill ratio. Lack of oscillatory motion in the temperature measurements for this test indicates that start-up did not occur at any power input. The maximum power input was not reached due to the evaporator section reaching the safety temperature limit of the insulation.

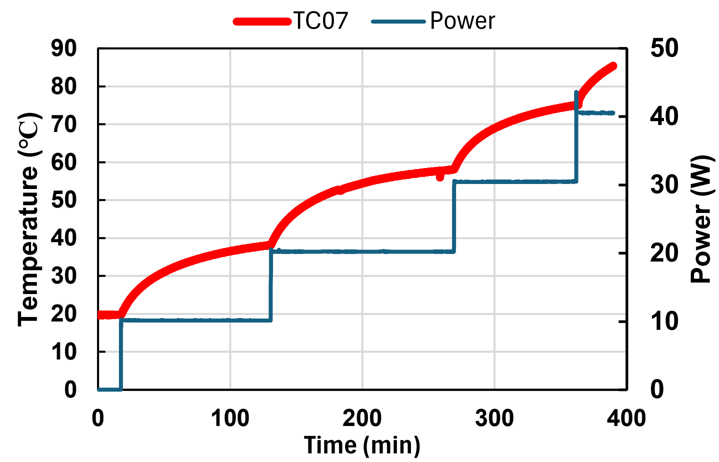

FIGURE 4: TEMPERATURE OF THE OHP'S EVAPORATOR SECTION THROUGHOUT THE EXPERIMENT FOR 70% FILL RATIO OHP B

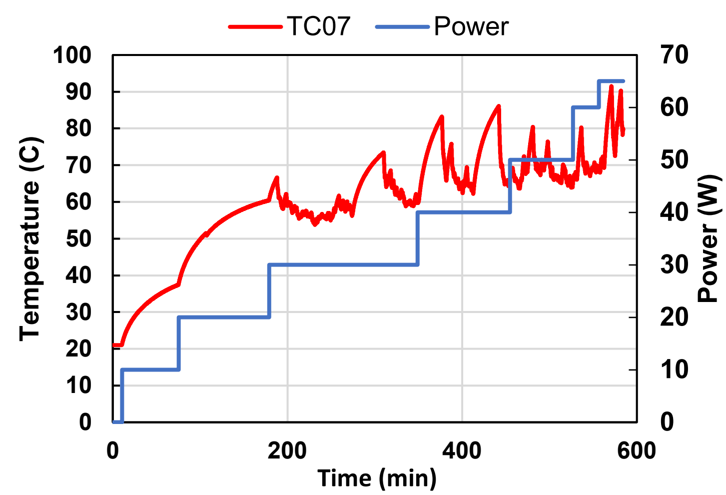

FIGURE 5: TEMPERATURE OF ALL SECTIONS OF THE OHP THROUGHOUT THE EXPERIMENT FOR 70% FR OHP C

Figure 5 shows that start-up occurred at 30W in OHP C but after some time fluid motion ceased and temperatures rose again. The temperature oscillations are due to the OHP being unable to fully start up and repeatedly entering start-up for short amounts of time. This was observed at 40W as well. Results from this test of OHP C show that 70% fill ratio is not favorable to start-up.

Figure 6 shows the thermal conductivity of all OHPs tested at 70% fill ratio OHP B was not able to reach 65W of power due to high temperature output from the heater. This OHP did not show signs of start-up during the testing and had the lowest conductivity. OHP C reached 65W, but only demonstrated partial start-up. OHP A started up and had the highest thermal conductivity of the three. OHP A was the only OHP to achieve full start-up, reaching a peak thermal conductivity of ~5000 W/mK at 60W.

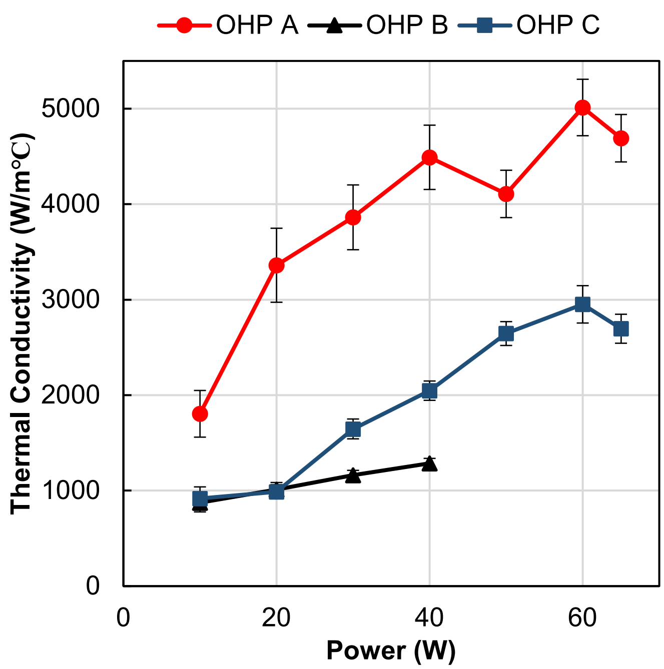

FIGURE 6: THERMAL CONDUCTIVITY WITH VARYING ADIABATIC LENGTH AND CHANNEL DIAMETER SIZE AT 70% FR

All OHPs at 50% fill ratio achieved start-up. Both OHP B and C had similar thermal conductivity in 10 watts and 20W of power. After 20W OHP C had a greater increase of thermal conductivity as seen in Figure 7. The thermal conductivity of OHP B had a constant thermal conductivity after 40 watts while OHP C saw its peak at 50W. OHP A had the greatest thermal conductivity and peaked at 40W, then started decreasing to about the same as the other OHPs.

Of all the tests conducted, the 50% fill ratio had the highest thermal conductivity. OHP A had the best thermal conductivity in all the tests and reached a peak of ~5300 W/m°C. OHP B did the worst in all the tests, which was expected since the smaller channel diameter would result in more friction making it harder for the OHP to start-up.

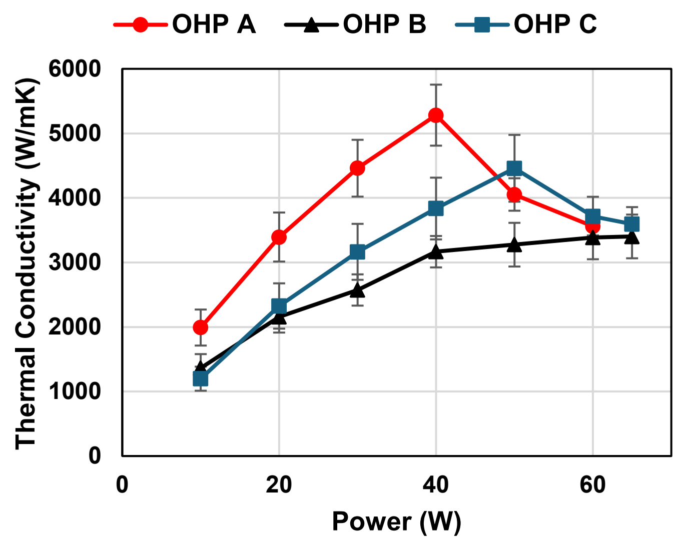

FIGURE 7: THERMAL CONDUCTIVITY WITH VARYING ADIABATIC LENGTH AND CHANNEL DIAMETER SIZE AT 50% FR

When comparing the OHPs by conductance, OHP C outperformed the others as seen in Figures 8 and 9. OHP C achieved the highest conductance at 50% FR with a peak conductance of 1.86 W/C. OHP C achieved the best conductance in both the 70% and 50% fill ratio tests, but it did not have the highest thermal conductivity, as shown in table 4. It is important to note that conductance does not consider the geometry of the OHPs. The shorter OHPs have a smaller adiabatic length which lowers the thermal conductivity as seen in equation (3).

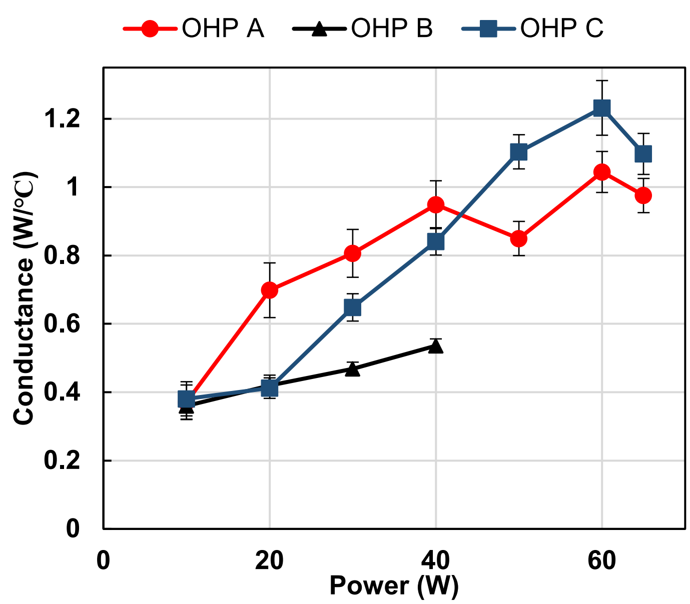

FIGURE 8: CONDUCTANCE WITH VARYING ADIABATIC LENGTH AND CHANNEL DIAMETER SIZE AT 70% FR

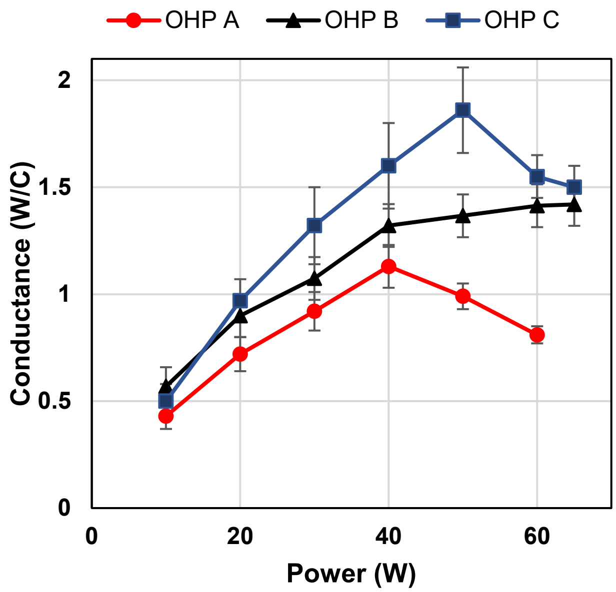

FIGURE 9: CONDUCTANCE WITH VARYING ADIABATIC LENGTH AND CHANNEL DIAMETER SIZE AT 50% FR

Considering start-up as a measure of thermal performance, OHPs B and C did not achieve start-up in the 70% fill ratio. Since neither of the shorter OHPs were able to reach full start-up, that means that for 70% fill ratio the longer OHP, OHP A, is the only one that properly performed as seen in Table 2. This is likely due to the relative size of evaporator to the total length of OHP. When looking at the 50% fill ratio graphs, we see similar results where the smaller OHPs had higher conductance, but the longer OHP had the highest thermal conductivity as seen in Table 3 and Table 4. In terms of start-up all the OHPs were able to reach start-up and work properly at 50% fill ratio. However, there is missing start-up data for OHP C, which will be updated for the final draft.

The differences in thermal performance between OHP B and C may be due to increases in flow resistance in smaller channels. This effect of flow resistance on OHP performance is also observed when we compare shorter OHPs with longer OHPs. We see this trend under the 50% fill ratio where both OHP B and C did better than OHP A in terms of conductance.

TABLE 2: Start-Up Summary

OHP	Start-Up Power (W)		
	30% FR	50% FR	70% FR
A	None	10	20
B	10	10	None
C	None	10	None

* Missing data for start-up, will update in the final draft.

TABLE 3: Conductance Summary

OHP	Peak Conductance (W/°C)		
	30% FR	50% FR	70 FR
A	0.59±0.02 @ 50W	1.13±0.1@40W	1.04±0.06 @ 60W
B	0.75±0.03 @ 50W	1.41 ±0.14 @ 65W	0.56±0.02 @40W
C	0.57±0.02 @ 50W	1.86±0.2 @ 50W	1.23±0.08 @60W

TABLE 4: Thermal Conductivity Summary

OHP	Peak Conductivity (W/(m°C))		
	30% FR	50% FR	70 FR
A	3103±67 @ 50W	5282±279 @ 40W	5012±295 @ 60W
B	1967±81 @ 60W	3595±338 @ 65W	1286±51 @ 40W
C	1501±55 @ 50W	4458±517 @ 50W	2951±195@ 60W

4. CONCLUSION

In conclusion, the conductance of OHPs is affected by the adiabatic length, channel diameter, and fill ratio. This is seen when comparing OHP C with OHP A, it is observed that OHP C has a higher conductance than OHP A showing that a smaller adiabatic length OHP has a higher conductance. When looking at the channel diameter size a bigger diameter leads to a higher conductance. Furthermore, 50% fill ratio produced the highest peak conductance. However, OHP A produced the highest thermal conductivity due to its long adiabatic length as the decline in conductance is not linearly proportional to increases in adiabatic length. This finding is important in illustrating the advantage of OHPs over existing solutions over long lengths.

In terms of start-up, 50% fill ratio is superior to 30% and 70% fill ratios across all OHPs. At 30% fill ratio, larger channel diameter seems to be unfavorable to start-up. At 70% fill ratio, OHPs B and C did not start-up, and this is likely due to the size proportion of evaporator to the OHP.

Oscillating heat pipes can be an integral part of thermal management solutions due to their passive operation and high heat transfer rate. The simplicity of OHP design can reduce the number of possible failure points in a thermal management system. However, OHP performance is dependent on the dimensions and fill ratio explored in this work. This is especially important in certain applications (e.g. space) where stringent design constraints need to be met (e.g. high vs low pressure working fluids, geometric limitations). This work aims to provide some guidance in solution development.

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