Thermal Management of Large Area Heat Loads Using Multi-Pass Cryogenic Loop Heat Pipe

Nathan Van Velson[1], Roopesh Kumar[2], and Calin Tarau[3]
Advanced Cooling Technologies, Lancaster, PA 17601

Guillermo Fernandez-Moroni[4]
Fermi National Laboratory, Batavia, IL 60510

The next generation of high energy physics experiments, such as the proposed low mass dark matter detector in the Oscura experiment, will require advanced semiconductor sensors such as the skipper-CCD detector developed by Fermilab. In order to minimize sensor dark current, the skipper-CCD devices are designed to operate at cryogenic temperatures. Additionally, these detectors must operate in an ultra-low background radiation environment. These requirements pose a major engineering challenge. As an alternative to the currently proposed solution of submerging the detectors in a liquid nitrogen pressure vessel, a cost-effective multi-pass cryogenic loop heat pipe (CLHP) is developed to extract, transfer, and reject waste heat from the sensor arrays. The CLHP architecture modifies the standard loop heat pipe design to provide cooling to large areas with low heat flux. The waste heat can then be rejected remotely to a liquid-nitrogen or cryocooler cooled heat sink. By isolating the sensors away from the heat rejection system, the need for large liquid nitrogen pressure vessel is eliminated and significant lead and copper shielding is minimized. Results from this study showed, the CLHP configuration was able to cool down large area heat loads of 150 W and 200 W. The CLHP system was able to maintain the large area heat source nominally isothermal with temperature gradients of the order of 1°C for the 150 W power and around 2~4°C for 200 W power.

I. Nomenclature

ACT	=	Advanced Cooling Technologies
DM	=	Dark Matter
LHP	=	Loop Heat Pipe
CLHP	=	Cryogenic Loop Heat Pipe
σ	=	Surface Tension (N/m)
R_p	=	Pore Radius (m)

II. Introduction

Understanding the nature of dark matter (DM) is one of the primary challenges of modern physics, and there are several experiments underway which are making attempts at direct detection of proposed DM particles. The next frontier in the search for dark matter is the detection of DM particles with masses less than 1 GeV (sub-GeV DM) [1]. One potential method for detecting sub-GeV DM is through detection of ionized electrons resulting from inelastic interactions between DM particles and electrons in a detector. In a recent technological breakthrough led by Fermilab and others, a new generation of silicon charged coupled devices with ultralow readout noise have been developed, referred to as "skipper-CCDs" [2]. These new skipper-CCDs are capable of single electron sensitivity and are key to the upcoming Oscura Experiment. The Oscura experiment will consist of an array of around 25,000 skipper-CCD

[1] Lead Engineer, Advanced Cooling technologies
[2] R&D Engineer II, Advanced Cooling technologies
[3] Principal Engineer, Advanced Cooling technologies
[4] Associate Scientist, Fermi National Accelerator Laboratory

detectors, resulting in a total of 10 kg of detector mass. Current designs for the detector begin with 16 1278x1058 pixel skipper-CCD devices fabricated on a single 150 mm Si wafer to form a Multi-Chip Module (MCM) (Figure 1(a)). Sixteen MCMs are packaged in a copper-shrouded Super Module (SM) (Figure 1(b)). Around 100 SMs are then combined to comprise the full experiment. To maintain an adequately low dark-current, the skipper-CCD sensors must be operated at temperatures between 120-140 K. The initially proposed strategy for cooling the detectors is to fully submerge them in pressurized liquid nitrogen (LN). The LN pressure is maintained at around 450 psi (3.1 MPa), the saturation pressure corresponding to the target operating temperature. This concept is illustrated in Figure 1(c).

Besides operating temperature, the other requirement for the skipper-CCDs in the Oscura DM experiment is an ultra-low background radiation environment, which is a major engineering challenge that affects the design of the whole experiment. In the current design, the LN pressure vessel would be constructed from stainless steel, which contains a number of radioactive isotopes, and requires significant lead and copper shielding within the vessel between the stainless steel and the detectors. Electronics interface with the detectors through the pressure vessel is an additional challenge. The development of an alternative cryogenic cooling system for the skipper-CCD array that removes the need for a liquid nitrogen pressure vessel would significantly reduce the engineering complexity of the experiment design and simplify the radiation shielding required.

To provide an alternative, more cost-effective cryogenic cooling solution for semiconductor detector arrays, such as the skipper-CCD arrays to be used in the Oscura Experiment, a multi-pass cryogenic loop heat pipe (CLHP) is proposed. The CLHP is utilized to collect heat loads distributed over a large surface area and transport it to a condenser for heat rejection. This design offers several practical benefits for the Oscura or similar proposed physics experiments. By cooling the detector array with cold plates instead of submerging them in pressurized liquid nitrogen, the need for an expensive and challenging-to-design pressure vessel is removed. By removing the pressure vessel, which would likely have to be constructed from stainless steel, the shielding of the detector arrays is significantly simplified. Remote rejection of the CCD waste heat from the CLHP condenser plate also simplifies the shielding required.

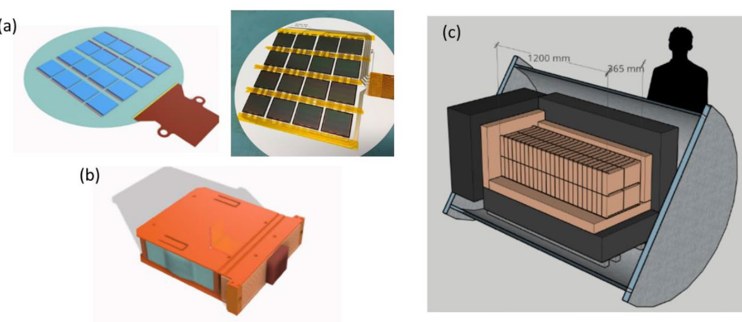

Figure 1: Nominal Oscura experiment design (a) Oscura Multi Chip Module (MCM) – 16 sensors on 150 mm Si wafer. (b) Proposed Oscura Super Module (SM) – 16 MCMs shielded in copper. (c) Pressure vessel containing 100 SMs submerged in liquid nitrogen. Images provided by FNAL [1].

The objective of the current study is to develop and test a multi-pass CLHP capable of cooling large area heat loads. An analytical model is used to predict the performance and aid in the design of the prototype CLHP. The model predicts steady state performance, including heat transfer rates, loop pressure drops, and capillary performance. Thermal Desktop (TD) is used to validate the analytical model and predict the response of the CLHP to changes in transient operating conditions, including sensor cool down.

III. Multi-Pass Cryogenic Loop Heat Pipe Thermal Management System

The Loop Heat Pipe (LHP) is a versatile heat transfer device which can transport large heat loads over long distances, with a small temperature difference between the heat source and heat sink [3]. LHPs have been successfully used for thermal control in many NASA, DoD, and commercial satellites. A typical LHP, shown in Figure 2, consists of an evaporator with a porous wick, a compensation chamber (CC, also known as a reservoir), a vapor line, a condenser, and a liquid line. During LHP operation, a heat input applied to the evaporator vaporizes a working fluid at the wick's outer surface. The vapor generated is collected in grooves and flows through the vapor line to the condenser, where the heat is rejected, and the working fluid condensed back to liquid. The liquid then flows through the separate liquid line back to the evaporator. The compensation chamber is linked to the primary wick through a secondary wick, which ensures that the primary wick is always saturated with liquid. The menisci formed at the liquid/vapor interface in the evaporator primary wick provides a capillary pressure equal to the total viscous pressure drops in the vapor grooves, vapor line, condenser lines, liquid line, and primary wick. The curvature of the menisci changes naturally in response to changes in applied power, resulting in passive and self-regulating operation. The operation of the LHP and the temperature of the evaporator and heat source are governed by the saturation pressure in the CC due to the CC being the lowest pressure point in the LHP. The CC saturation pressure is determined by the balance of heat leaks from the evaporator, subcooling of the liquid return, and any heat exchange with the ambient. The most used method for regulating the CC saturation pressure, and thus evaporator/heat source temperature, is to cold bias the CC and apply heat to the CC with electric heaters [4].

Shown schematically in Figure 3, the multi-pass CLHP utilizes a capillary pump assembly consisting of a wicked evaporator and compensation chamber assembly like a standard LHP [5]. However, unlike a standard LHP, the capillary pump assembly does not directly absorb heat from the source to be cooled. Rather, an electric heater is used to apply heat to the wicked evaporator solely for the purpose of generating fluid flow in the loop. The vapor generated leaves the pump assembly and flows through a condenser cold plate, where it is condensed back to liquid. This liquid then flows through a cold plate to which a set of detectors are thermally coupled, and is vaporized, providing cooling to the detectors. This vapor then flows back again to the condenser plate where it is condensed, and the waste heat from the detectors is rejected. The condenser cold plate can be cooled by an external liquid nitrogen source, or otherwise cooled to a sufficiently low temperature (< 120 K) such that the detector array is cooled to a temperature of 120-140 K. Multiple passes between the condenser and the detector arrays can be made; Figure 3 shows three passes through the heat source.

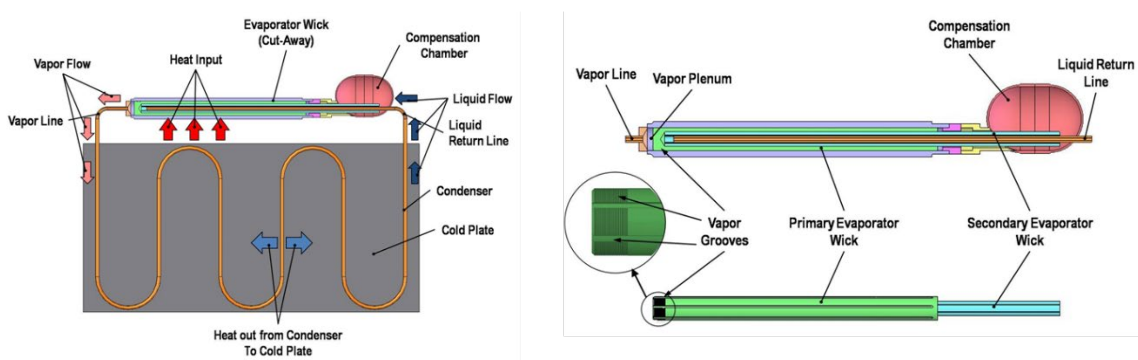

Figure 2: Schematic of a conventional Loop Heat Pipe, with details of the evaporator and compensation chamber.

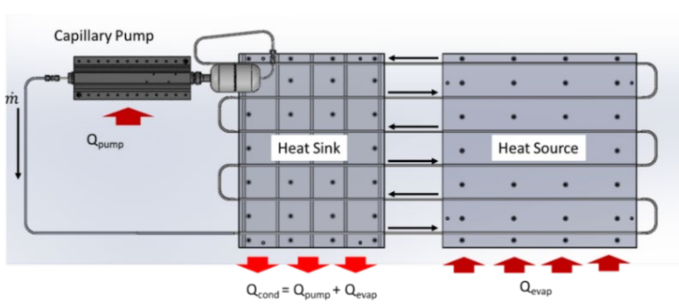

Figure 3: Schematic of the cryogenic loop heat pipe thermal management system for thermal management of semiconductor detector arrays

IV. Experimental Methodology

A. Evaporator Geometry and Capillary Pump Design Considerations

In the initial conceptualization of the cooling solution the Skipper-CCD super modules would be individually mounted to a cold plate cooled by the CLHP. The 100 super modules when laid out in a rectangular array as shown in

Figure 4 would cover an area of 150 cm × 150 cm. The heat load generated by 100 super modules is 1 KW, which generates a heat flux of 0.04 W/cm^2 over this surface area. In this study, a 1/10th-scale prototype was considered and the detector array geometry dimension calculated based on 100 W power and the heat flux of 0.04 W/cm^2 is 50.8 cm × 50.8 cm. The capillary pump considered for this study is shown in the Figure 5 below. The capillary pump consists of a nickel wick with a pore radius of 1 µm and a permeability of 2×10^{-14} m^2. The capillary pump is shrouded in an aluminum envelope with a flat base to attach a heat source such as electric cartridge heaters. The heat source is coupled to the evaporator through a flange. Heat is conducted through the envelope material to the primary wick, where fluid is vaporized, and the vapor flows out of the evaporator through grooves in the wick. The liquid feed and secondary wick are located down the center, to supply the primary wick with sufficient liquid to remain saturated. The flow path of the liquid is demarcated by white arrows. Propylene was chosen as the working fluid for the present study.

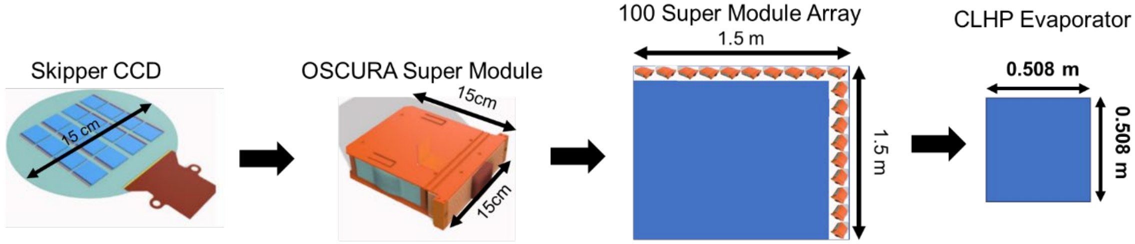

Figure 4: CLHP Evaporator sizing based on the Skipper CCD and the OSCURA super module dimensions

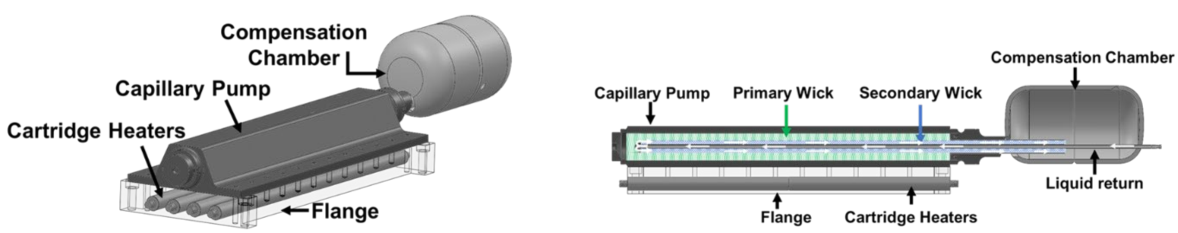

Figure 5: (Left) Capillary pump CAD model, (Right) Cross-section of the capillary pump

B. Trade Study: Tubing Diameter and Number of Passes

In the CLHP configuration, the heat acquisition and rejection cycle is repeated for multiple passes, implying that the heat removal capacity of the proposed concept increases with the number of the evaporator-to-condenser U-turns. However, the number of passes is limited by the pressure drop along the tubing that can be supported by the primary wick of the capillary pump. The maximum capillary pumping power generated by the wick is a function of the working fluid and the wick pore radius. The max capillary pumping power is calculated using Equation 1. As mentioned earlier, the capillary pumping pressure must be greater than the total pressure drop in the system. The total pressure drop of the system is a summation of pressure drops along the vapor grooves in the wick, in the lines occupied by vapor, in the lines occupied by liquid, in the lines where the working fluid is in two-phase conditions and finally, the radial liquid pressure drop along the wick. The pressure drop in the single-phase sections (vapor or liquid) was calculated using the fundamental pipe flow equations accounting for both major and minor losses, and Dittus-Boelter correlation. The pressure drop in the two-phase sections was calculated using the Lockhart-Martinelli correlation, assuming an average quality of 0.5.

$$\Delta P_{max} = \frac{2\sigma\ \cos\theta}{R_p} \qquad \textbf{Equation 1}$$

$$\Delta P_C \geq \ \Delta P_{Total} \qquad \textbf{Equation 2}$$

$$\Delta P_{Total} = \ \Delta P_{Vapor\ grooves} + \ \Delta P_{Vapor} + \ \Delta P_{Two-Phase\ Line} + \ \Delta P_{Liquid} + \Delta P_{Wick} \qquad \textbf{Equation 3}$$

$$Pressure\ Drop\ Margin = \Delta P_C - \ \Delta P_{Total} \ \geq 0 \qquad \textbf{Equation 4}$$

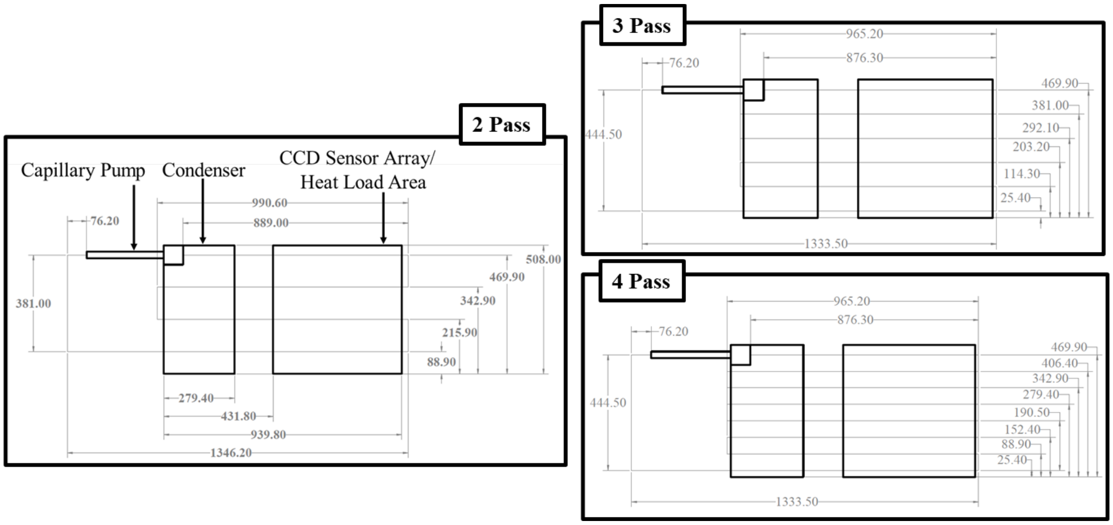

Figure 6: CLHP schematic of 2, 3 and 4 pass configurations for pressure drop trade study (Dimensions in mm)

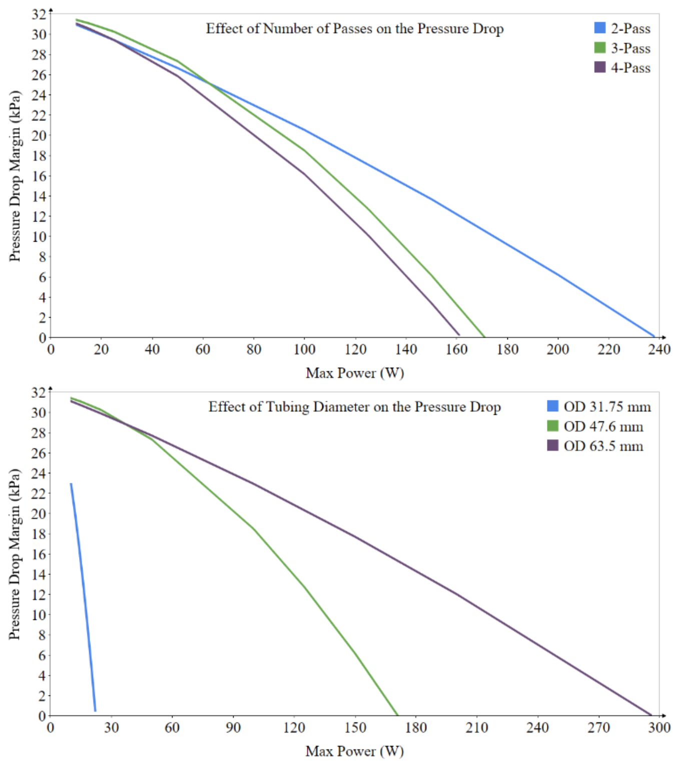

Figure 7: Pressure drop margin vs. capillary pump power as function of (left) tubing diamter, (right) number of passes.

A theoretical trade study was conducted to optimize the number of passes based on the current capillary pump design. The pressure drop along the tubing is a function of diameter and the length of the tubing. Additionally, the pressure drop along the tubing varies non-linearly since the working fluid can be in a vapor state, liquid state or as a two-phase mixture. Two trade studies were performed,

a. *Effect of tubing diameter on the pressure drop*: The tubing outer diameter used in the trade study was 0.635 cm (1/4th in), 0.476 cm (3/16th in) and 0.3175 cm (1/8th in) with a wall thickness of 0.0889 cm (0.035 inch).

b. *Effect of number of passes on the pressure drop*: The pressure drop along the tubing was calculated for 2, 3 and 4 passes.

The configurations used in the trade study analysis are shown in Figure 6. The total tubing length for the 2, 3, and 4-pass configurations is 508 cm (200 in), 826 cm (325 in), and 897 cm (353 in), respectively. Results show the ΔP margin is higher for 2 passes and decreases on increasing the number of passes. The ΔP margin decreases sharply for 3 and 4 passes, indicating a reduction in the heat load cooling capacity for a given capillary pump power. Similarly, the heat load cooling capacity decreases with reducing tubing diameter. The pressure drop is lowest for 0.635 cm (1/4th in) tubing ID and highest for 0.3175 cm (1/8th in) tubing ID. Based on this analysis, the number of passes was chosen to be 3 and the tubing outer diameter of 0.476 cm (3/16th in)) with a wall thickness of 0.0889 cm (0.035 inch) for this study.

C. Thermal Desktop Modeling

Thermal Desktop (TD) software was used to assess the feasibility of the concept and model the CLHP behavior. Thermal Desktop is a graphical user interface for the SINDA/FLUINT computational package commonly used in the design and analysis of spacecraft thermal control and other thermal management systems and technologies. Figure 8 shows the schematic of the loop heat pipe geometry in the modified configuration used in the TD software. For studying the modified loop heat pipe concept two passes were used. The colored lines indicate the working fluid state, where red indicates a vapor flow, orange indicates a two-phase flow and blue indicates the working fluid is in the liquid state. The TD model was performed with a known fluid Ammonia. The temperature of the working fluid was 285 K, the capillary pump was supplied with 125 W of power to generate the flow to demonstrate a heat load cooling of 250 W. The condenser temperature was set at 243 K via a weak conductor. Figure 8. (Right) shows the pressure drop comparison between the data from the TD model and the analytical solution. The temperature and flow quality along different sections of the loop heat pipe are shown in Figure 9. Exiting the capillary pump on the left of the plot, the working fluid is initially at a temperature of 285 K with a flow quality of 1 indicating a full vapor state. As the flow moves through the condenser, the working fluid flow quality reduces and eventually reduces to 0 as it exits the condenser. In the "inter" region the working fluid is in the liquid state, and as the fluid moves through the evaporator/heat load area, the working fluid vaporizes and the phase changes gradually from liquid to vapor (flow quality changes from 0 to 1). The flow exiting the evaporator is completely vapor and the fluid passes through the condenser the working rejects heat converts back to liquid. This process repeats along the multiple passes of the modified loop heat pipe configuration. In the TD model, the pressure drop experienced along the tubing is calculated to be around 600 Pa. The pressure drop calculated using both TD and the analytical model is very similar, thereby validating the analytical pressure drop calculations. These modeling results indicate the feasibility of the concept for maintaining large heat load areas under the required temperature.

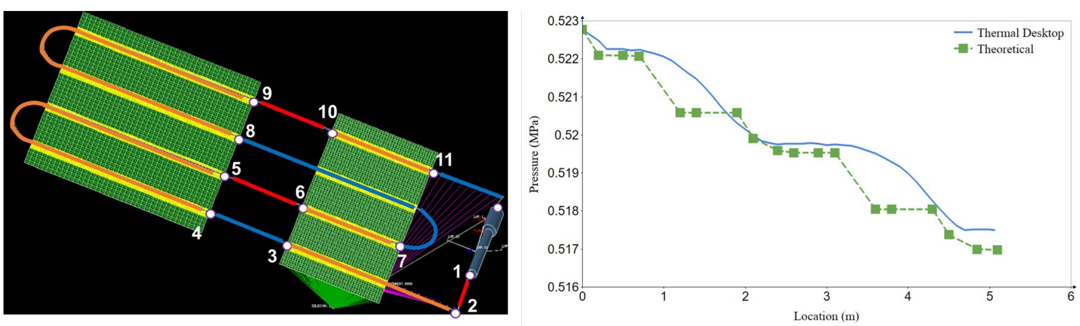

Figure 8: (Left) Thermal Desktop Model Schematic, (Right) TDM and analytical model pressure drop comparison.

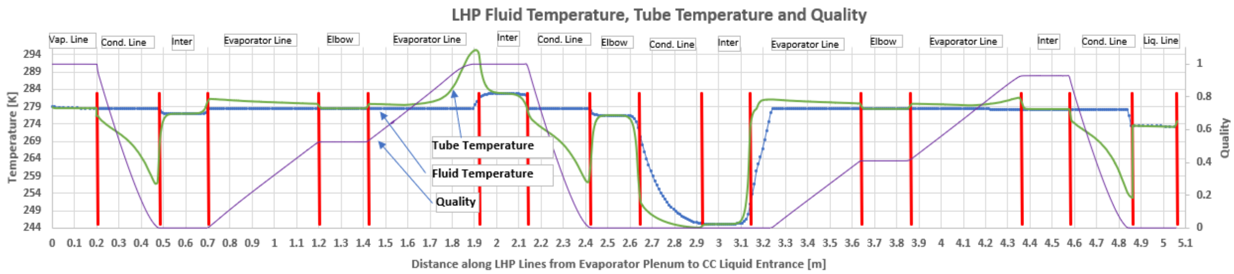

Figure 9: Temperature and flow quality along different section of the loop heat pipe

D. Test Model and Test Setup

To validate the analytical model and demonstrate the feasibility of the multi-pass CLHP for large area cooling, a prototype CLHP was designed and fabricated. A CAD rendering and the dimensions of the CLHP setup are shown in Figure 10. The entire system is placed over an aluminum frame and isolated using low conductivity spacer material. The tubing which carries the working fluid is sandwiched between the condenser plate and the evaporator plate. The condenser plate also hosts another set of copper tubing which circulates liquid nitrogen. The condenser plate temperature is controlled by regulating the LN flow rate based on sink temperature input. The evaporator assembly is designed in such a way that the compensation chamber is in contact with the condenser plate. This helps to cold bias the compensation chamber which helps with loop startup and fluid circulation. Heater cartridges are used in the evaporator heater block to start the flow and strip heaters are applied to the bottom surface of the evaporator plate to simulate a detector array heat load. For the current study, the "T" type thermocouple was used. The type T working range is between 73 K to 473 K (–200°C to +200°C), has error of the order of ±1°C and has very good repeatability. A Keithley DAQ system was used to record the data at a sampling rate of 0.5 Hz.

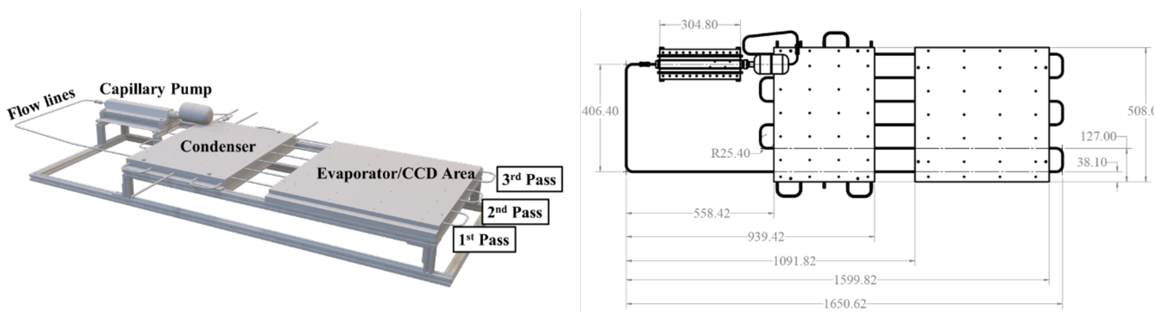

Figure 10: (Left) A CAD Schematic of the cryogenic loop heat pipe, (Right) Dimensions of the cryogenic loop heat pipe system (All Dimensions are in mm)

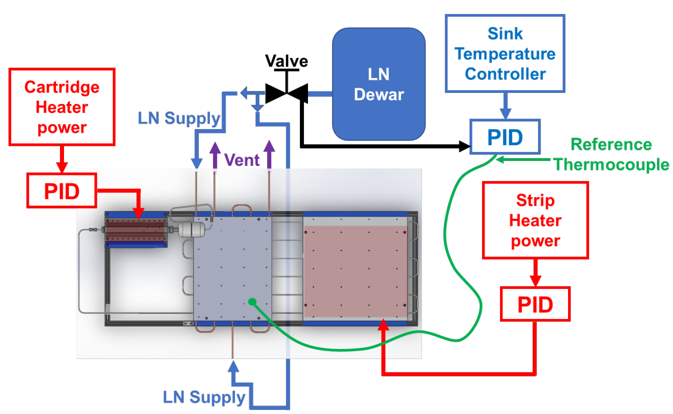

Figure 11: A schematic showing the various instrumentation involved in the experimental setup

The experimental campaign was devised to test the CLHP in its conventional configuration and in the modified configuration. The difference between the conventional and modified CLHP is the absence or presence of the evaporator plate. Figure 12 shows the thermocouple locations on the CLHP setup for both conventional and modified configurations. In the conventional configuration, the thermocouples are connected to the tubing to measure the temperature, and for the modified configuration the thermocouples are shifted to the evaporator plate to map the temperature distribution. The evaporator plate was installed with 18 thermocouples and the data is interpolated between the thermocouples to obtain the final temperature map. The CLHP was tested under a series of applied power and sink temperatures to evaluate its performance. The test conditions are provided in Table 1.

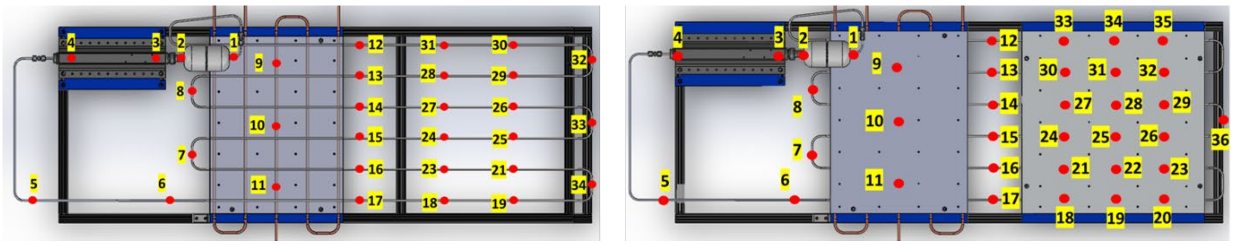

Figure 12: Thermocouple map for the (left) conventional CLHP configuration, (right) modified CLHP configuration

Table 1. Test Conditions

Configuration	Capillary Pump Power (W)	Evaporator Power (W)	Sink Temperature (°C)
Conventional CLHP	50, 100, 150	-	-5
	150	-	-10, -30, -50, -73
Modified CLHP	50	150	-10, -30, -50, -73
	75	200	-10, -50, -100

V. Results and Discussion

A. Conventional CLHP: Constant Sink Temperature of -5°C and Varying Power

Steady state temperatures along the length of the CLHP tubing are shown in Figure 13 for the constant sink temperature case. The plot is color coded to identify temperature profiles along different sections of the CLHP. The blue curve corresponds to an evaporator power of 50 W, 100 W is represented via the orange curve, and 150 W by the gray curve. On supplying 50 W power to the cartridge heater, the working fluid vaporizes, and superheated vapor begins to flow through the tubing, indicated by a temperature rise in the vapor line (yellow zone). The temperature of the working fluid decreases as the flow goes through the condenser (purple zone) and rejects heat to the sink. However, a temperature increase is seen from location 2 m to 3 m. This is due to the long, exposed lines between each loop through the condenser in the conventional configuration. At lower operating powers the flow rate of the vapor in the system is relatively low and is likely fully subcooled liquid after the first pass through the condenser. In between each pass, the fluid picks up heat from the environment, and the subcooling is reduced. At higher powers with higher flow rates, the fluid likely does not fully condense and gets subcooled until the second pass through the condenser. Additionally, the higher flow rate results in smaller temperature increases due to sensible heating from the environment. The working fluid temperature in the liquid return is significantly lower than the capillary pump temperature indicating the working fluid is in the subcooled state. The CC temperature lies in between the capillary pump temperature and the vapor line temperature as expected. As the power into the system is increased to 100 W and 150 W, the capillary pump temperature drops from 12°C to 5°C. This is common behavior in LHPs operating at lower powers. For a constant sink temperature, the temperature profile variation for the 100 W and the 150 W case is very similar to the 50 W case.

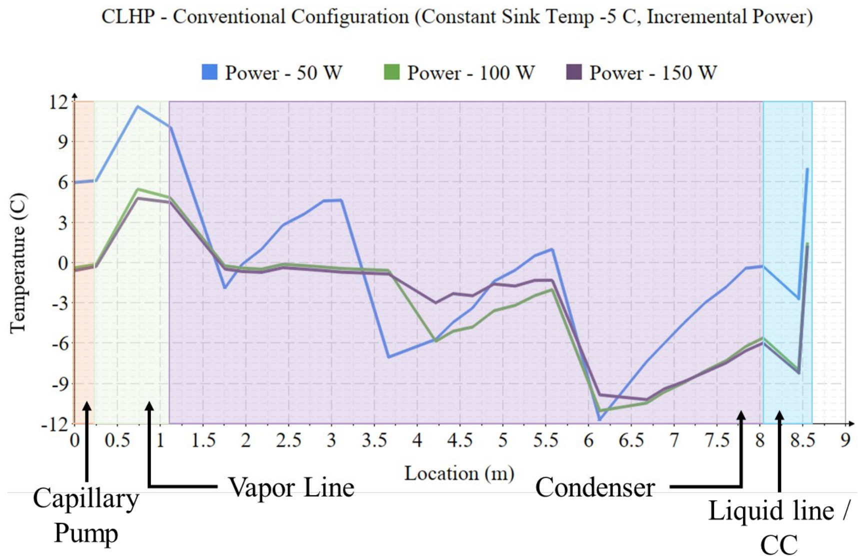

Figure 13: Steady-state temperature distribution along the CLHP tubing at constant sink temperature and different applied powers

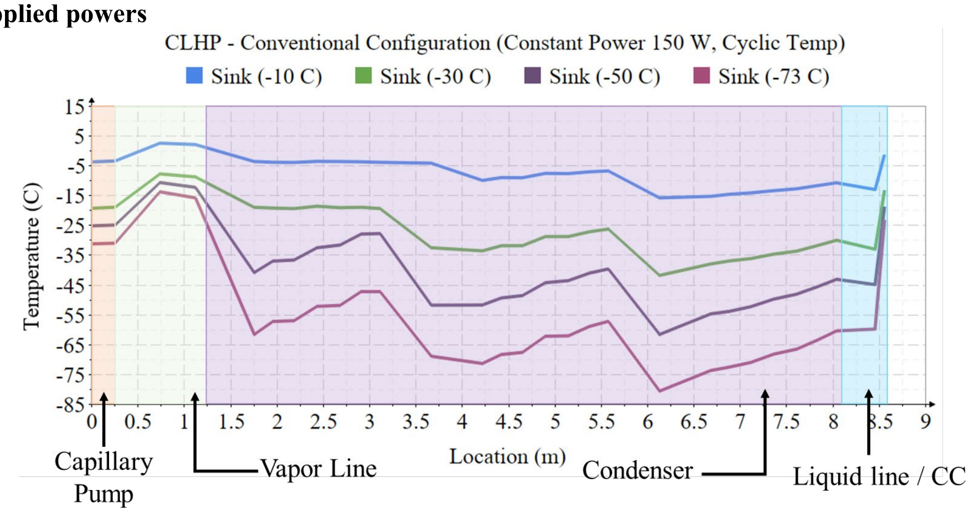

Figure 14: Steady-state temperature distribution along the CLHP tubing at constant power and varying sink temperatures

B. Conventional CLHP: Constant Power of 150 W and Varying Sink Temperature

In this test case, the capillary pump power was maintained constant at 150 W and the sink temperature was varied. The sink was initially operated at approximately -10°C and then decreased to -30°C, -50°C, and finally -73°C. The system was operated at each temperature for an arbitrary amount of time until a steady state is achieved. Figure 14 shows the temperature profiles for selected thermocouples on the CLHP setup. As seen in the test data, at the -10°C sink condition the capillary pump is operating at approximately -5°C. When the sink is reduced to -30°C the capillary pump temperature drops to approximately -20°C. At a sink temperature of -10°C, it is likely most of the condenser is two-phase flow, with a small amount of subcooling entering the CC. As the sink temperature reduces, the flow is fully condensed earlier, and the phenomena of sensible heating of the sub-cooled liquid in the long exposed tubing is seen. However, the temperature distributions within the loop are as expected for conventional LHP operation. This test and the above test provide confidence in the design of the modified CLHP and confirm that the capillary pump assembly is fully functional.

C. Modified CLHP: Capillary Pump Power 50 W and Evaporator Power 150 W

Once the prototype CLHP was demonstrated to operate successfully in the conventional configuration, the evaporator plate was added to the setup in order to demonstrate the CLHP in the modified configuration for cooling of large-area heat sources. As shown in Figure 15, the test was started by applying LN cooling to the condenser plate, with a set-point of -10°C. At around 2500 s, 50 W was applied to the capillary pump, and a flow was generated in the loop. Once the loop had reached a steady state, 150 W was applied to the strip heaters attached to the evaporator plate around 11000 s, and the system was again allowed to reach steady state. The condenser temperature was stepped down further to -30°C, -50°C, and -73°C. The temperature distribution of the evaporator plate surface with time is shown in Figure 16. Steady-state temperature distributions interpolated from the thermocouple measurements at each condenser temperature are shown in Figure 17. From these plots, it can be seen that the temperature distribution in the evaporator plate varies by only ~1°C for all sink conditions. The power applied to the evaporator plate in this test is slightly higher than that determined from the sizing of the 1/10th scale demonstration. This test is a clear demonstration of the capability of ACT's CLHP concept to provide cooling to large area heat sources, and the feasibility of its application to the cooling of Skipper-CCDs in the Oscura experiment.

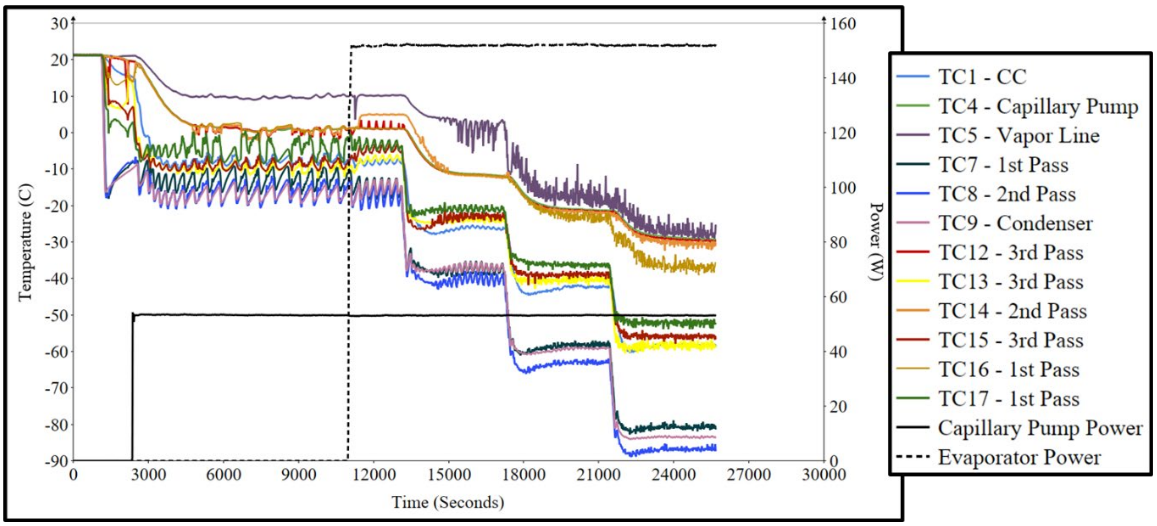

Figure 15: Temperature variation with time at different locations of the modified CLHP during constant 50 W capillary pump power, 150 W evaporator power, variable sink test

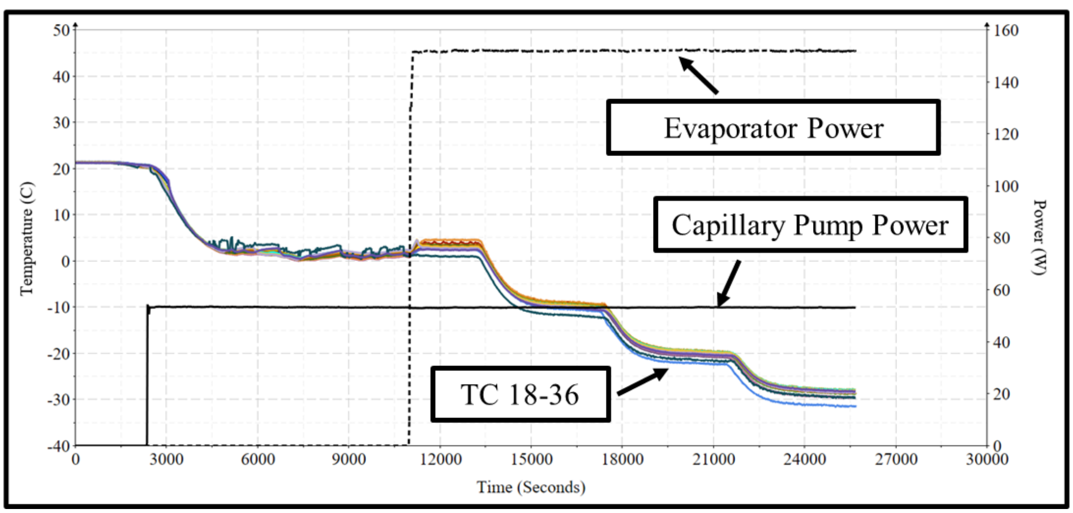

Figure 16: Temperature distribution with time of the modified CLHP evaporator plate for the constant power (150 W) variable sink test

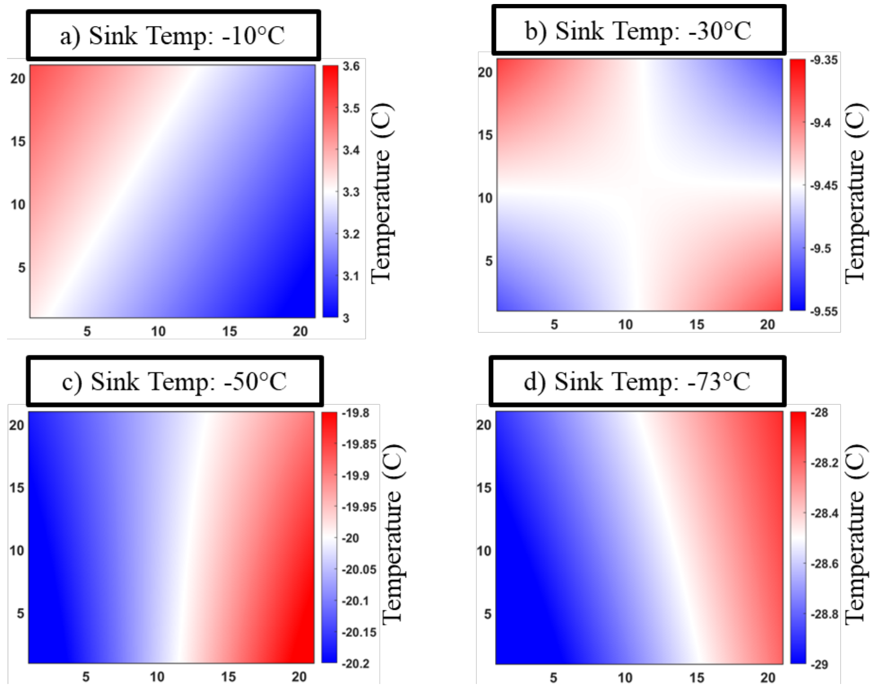

Figure 17: Interpolated temperature distributions of evaporator plate with 150 W heat load applied for (a) -10°C sink, (b) -30°C sink, (c) -50°C sink, (d) -73°C sink

D. Modified CLHP: Capillary Pump Power 75 W and Evaporator Power 200 W

A second test case was performed with the modified CLHP configuration by increasing the evaporator power/heat load to 200 W. The capillary pump was set to 75 W to accommodate the increasing power/ heat load cooling requirements. The temperature distribution of the evaporator plate surface with time is shown in Figure 18 and Figure 19. The condenser temperature was reduced in three steps, -10°C, -50°C, and -100°C. Results show the modified CLHP configuration performs well even under increased power requirements. The sink conditions can be changed to control the evaporator plate temperature distribution. At higher power case, it can be seen (in Figure 20) that the temperature distribution in the evaporator plate varies by only 2~4°C for all sink conditions. For the -100°C sink condition, around 18000 s the LN Dewar was replaced since the tank was running low on liquid Nitrogen. This causes the condenser temperature to rise by 7°C. However, the effect of this LN change in between the tests has a very minimal effect on the evaporator plate temperature distribution. A small increase in temperature of only 2°C was observed in the evaporator plate.

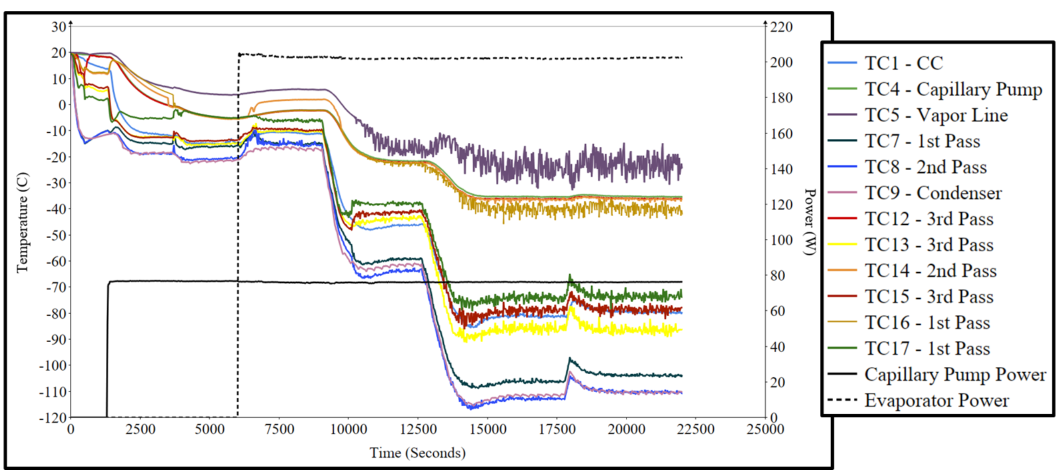

Figure 18: Temperature variation with time at different locations of the modified CLHP during constant 75 W capillary pump power, 200 W evaporator power, and variable sink test

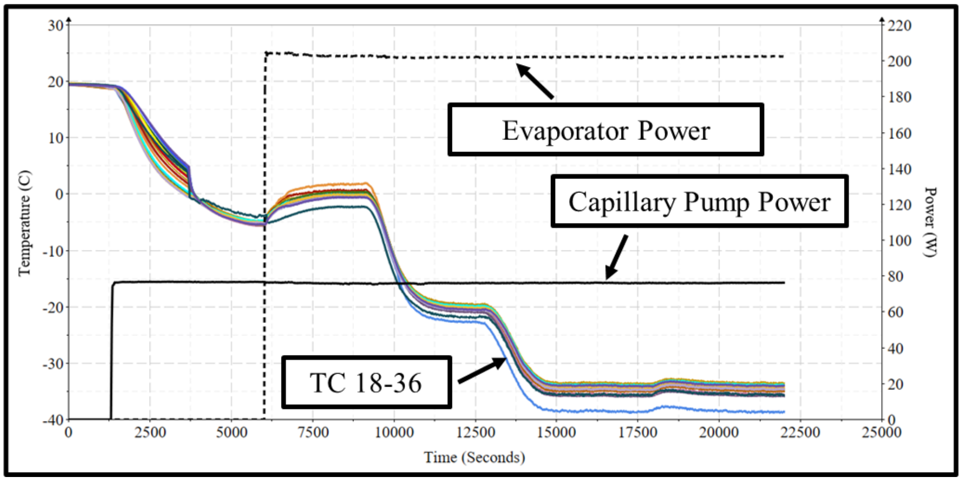

Figure 19: Temperature distribution with time of the modified CLHP evaporator plate for the constant power (200 W) variable sink test

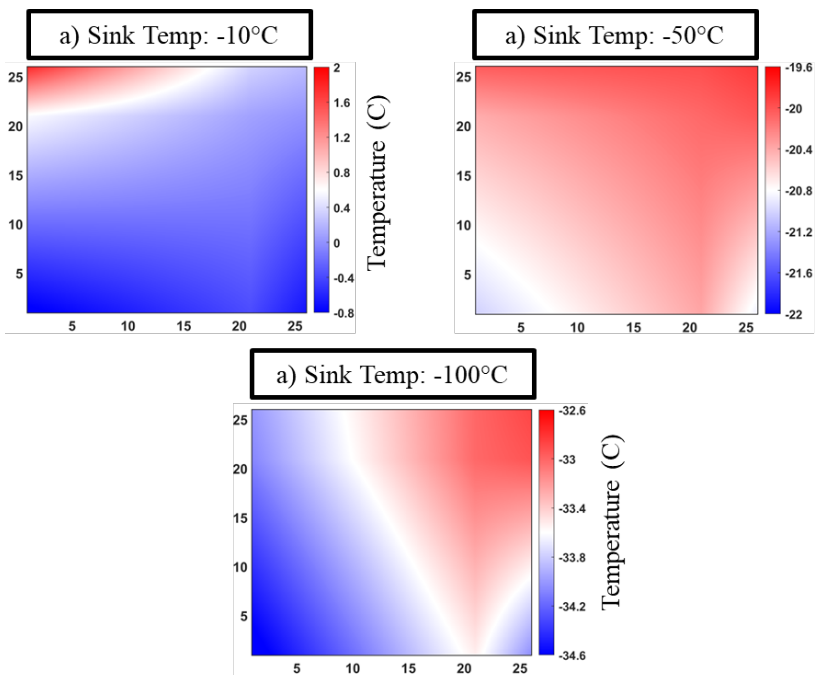

Figure 20: Interpolated temperature distributions of evaporator plate with 200 W heat load applied for (a) -10°C sink, (b) -50°C sink, (c) -100°C sink

VI. Conclusion and Future Work

A systematic study was carried out to investigate the feasibility of using a cryogenic loop heat pipe in cooling large area heat loads efficiently. Analytical and thermal desktop modeling were used to perform trade studies to identify critical loop heat pipe components and parameters such as loop heat pipe capillary pump wick properties and sizing, tubing sizing and condenser sizing. Based on the trade studies, a CLHP test assembly was fabricated for experimental characterization with propylene as the working fluid. The test assembly was tested as a conventional LHP and in the modified configuration. In the conventional CLHP configuration, the system was tested at varying sink and power conditions. Results show the temperature distributions within the loop are as expected for conventional LHP operation and the tests provided confidence in the design of the modified CLHP and confirmed that the capillary pump assembly is fully functional. In the modified CLHP configuration, the concept functioned as expected by cooling down large area heat loads of 150 W and 200 W. The CLHP system was able to maintain the evaporator/CCD area nominally isothermal with temperature gradients of the order of 1°C for the 150 W power and around 2~4°C for 200 W power.

Future studies will explore testing the CLHP configuration with alternative working fluids such as ethane and methane which are capable of operating at 120 K cryogenic temperatures.

Acknowledgments

The authors would like to thank the DOE Office of High Energy Physics for their financial support under Phase I SBIR contract no. DE-EC0022896, managed by Dr. Helmut Marsiske. We are grateful for their support.

References

[1] Alexis Aguilar-Arevalo, Fabricio Alcalde Bessia, Nicolas Avalos, Daniel Baxter, Xavier Bertou, Carla Bonifazi, Ana Botti, Mariano Cababie, Gustavo Cancelo, Brenda Aurea Cervantes-Vergara, Nuria Castello-Mor, Alvaro Chavarria, Claudio R. Chavez, Fernando Chierchie, Juan Manuel De Egea, Juan Carlos D`Olivo, Cyrus E. Dreyer, Alex Drlica-Wagner, Rouven Essig, Juan Estrada, Ezequiel Estrada, Erez Etzion, Guillermo Fernandez-Moroni, Marivi Fernandez-Serra, Steve Holland, Agustin Lantero Barreda, Andrew Lathrop, Jose Lipovetzky, Ben Loer, Edgar Marrufo Villalpando, Jorge Molina, Santiago Perez, Paolo Privitera, Dario Rodrigues, Richard Saldanha, Diego Santa Cruz, Aman Singal, Nathan Saffold, Leandro Stefanazzi, Miguel Sofo-Haro, Javier Tiffenberg, Christian Torres, Sho Uemura, Rocio Vilar, "The Oscura Experiment" arXiv:2202.10518.

[2] Brenda A. Cervantes-Vergara, Santiago Perez, Juan Estrada, Ana Botti, Claudio R. Chavez, et al., "Skipper-CCD Sensors for the Oscura Experiment: Requirements and Preliminary Tests.," arXiv:2304.04401v1, 2023.

[3] Ku, Jentung., "Operating characteristics of loop heat pipes. No. 1999-01-2007," SAE Technical Paper, 1999.

[4] Ku, Jentung, "Methods of Controlling the Loop Heat Pipe Operating Temperature," No. 2008-01-1998, SAE Technical Paper, 2008.

[5] Hoang, Triem T., et al., "Large-area cryocooling for far-infrared telescopes," Optical Science and Technology, SPIE's 48th Annual Meeting. International Society for Optics and Photonics, 2003.

[6] Ku, Jentung, and Franklin Robinson, "Testing of a Cryogenic Loop Heat Pipe for Large Area Cryocooling." 44th International Conference on Environmental Systems, 2014.

[7] Reay, David, Ryan McGlen, and Peter Kew, "Heat pipes: theory, design and applications," Butterworth-Heinemann, 2013.