REVIEW OF CRYOGENIC LOOP HEAT PIPE TECHNOLOGY DEVELOPMENT BY NASA/GSFC FOR SPACE APPLICATIONS

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ABSTRACT

In the early 2000s, space agencies of various countries were planning a series of next-generation space telescopes that operate in the electromagnetic spectrum ranging from near infrared (IR) to far infrared (FAR). To provide necessary thermal management of the IR sensors/detectors, cryocooling transport technologies had to be developed and fully flight-qualified by the time the space programs need them. The minimum operational life of these telescopes is required to be more than 20 years. Moreover, it was expected that the future telescopes will be deployed at a distance from earth much farther away than that of the Hubble Space Telescope. No servicing mission would be possible for scheduled maintenance. Passive capillary-pumped two-phase heat transports, such as Loop Heat Pipes (LHP) and Capillary Pumped Loops (CPLs), were deemed better suited for such applications – for their operational reliability/robustness and long life – than the mechanical counterparts. The technology research and development began first with a Nitrogen LHP in 1993 and a Nitrogen CPL in 1998 to be followed by the successful demonstration of Cryogenic Advanced Loop Heat Pipes (C-ALHP) utilizing Nitrogen, Oxygen, Neon, and Hydrogen as the working fluid. In 2004, a mini-LHP with a single ¼”ODx1.5”L capillary pump was fabricated and tested with Neon and Helium to demonstrate the cryogenic heat collection over a large area in the temperature range of 30-40K and 4-6K, respectively. There are many practical challenges for a cryogenic system, the C-ALHP included, to overcome in real situations. The management of environmental heat parasitics is undoubtedly the most difficult but, by no means, the only one. Potential users express desires to have the said cryocooling transport devices that are capable of (i) isolating unwanted vibrations from the mechanical cryocoolers, (ii) supporting IR telescopes mounted on a 2-axis gimbal, and (ii) starting from an initially supercritical thermodynamic state without assistance from an extra mechanical device. In addition, like other space-based devices, it must conform to the Size, Weight, and Power (SWaP) requirements. The author, in paper, shall present the historical progression of the C-ALHP development by TTH Research Inc. for NASA in the U.S.A. and its current state of technology.

NOMENCLATURE

N_{HR}	=	number of moles of working fluid in Hot Reservoir
N_{LHP}	=	number of moles of working fluid in LHP components
P_C	=	critical pressure of working fluid
T_0	=	evaporator/reservoir temperature prior to initially supercritical start-up
T_1	=	temperature at which working fluid pressure becomes subcritical

T_C = critical temperature of working fluid
T_{MAX} = maximum temperature that fully charged LHP is exposed to during lifetime
V_C = volume of condenser
V_{COLD} = volume of cold portion of LHP prior to start-ups
V_{COND} = condenser volume
V_{HOT} = volume of hot portion of LHP prior to start-ups
V_{HR} = hot reservoir volume
V_{LHP} = combined volume of ALHP components (excluding Hot Reservoir and Swing Volume)
V_{SV} = "swing volume" volume

Abbreviations

ALHP – Advanced Loop Heat Pipe
C-ALHP – Cryogenic Advanced Loop Heat Pipe
CCPL – Cryogenic Capillary Pumped Loop
DTU – Demonstration Test Unit
HR – Hot Reservoir
LHP – Loop Heat Pipe
MDP – Maximum Design Pressure
SV – Swing Volume
SWaP – Size, Weight, and Power
SWaP-C – Size, Weight, and Power & Cost
TCS – Thermal Control System

INTRODUCTION

Since its introduction to the U.S.A. in the 1990s from the former Soviet Union, Loop Heat Pipes[1,2] (LHPs) have gained acceptance and popularity among the spacecraft thermal control engineers for their operational reliability/robustness and long life. Since LHPs do not contain mechanical moving parts to wear out or break down, thus, no lubrication is necessary – eliminating one major concern for long-duration unmanned space missions. A LHP functional schematic is illustrated in Figure 1. In the early 2000s when NASA began planning the next-generation Infrared (IR) space telescopes to replace the Hubble Space Telescope (HST), the LHP technology was considered as a candidate for the thermal control systems (TCS). The cryogenic LHP research & development had, however, barely started at that time (at least in America) albeit room temperature Ammonia LHPs had already utilized successfully in several NASA satellites[3,4].

Many fundamental challenges - associated with the design and operation of a LHP for cryogenic cooling that did not have to be dealt with in a room-temperature system - must be overcome.

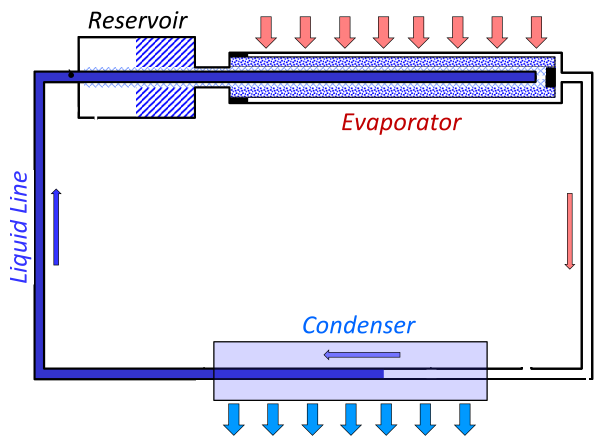


Figure 1. Loop Heat Pipe.

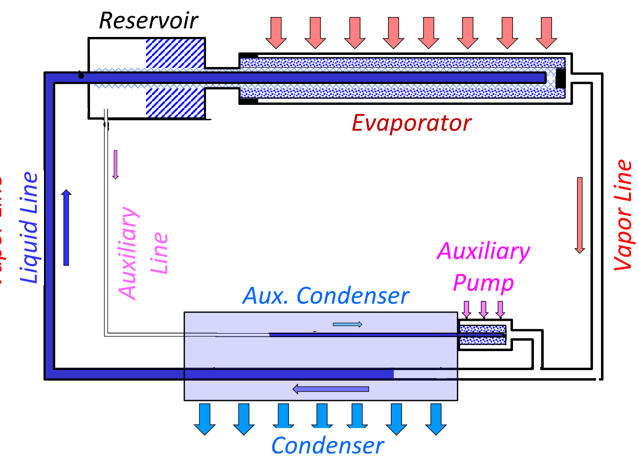


Figure 2. Advance Loop Heat Pipe.

First off, the cryogenic LHP (C-LHP) is likely in a supercritical thermodynamic state prior to the system startup, i.e. both fluid initial pressure and temperature are above the respective critical values. No liquid will form in the wicks (or anywhere else in the loop for that matter) to establish the capillary action for the loop to commence the fluid circulation unless the local fluid is in a subcritical state. Cooling only the condenser of a LHP below the fluid critical temperature does not necessarily render the fluid pressure subcritical, i.e. even no liquid exists in the condenser. On the other hand, a traditional LHP charged with Nitrogen as the working fluid had no problem starting and, subsequently, performed nominally when the entire loop was brought below the Nitrogen critical temperature as demonstrated by Dynatherm Corp. in 1993[5]. Secondly, the C-LHP may have to operate in a "hot" surrounding, making the management of the environmental heating of the loop liquid line and reservoir rather difficult. Lastly, the C-LHP has to undergo fluid charging and other processing tasks at room temperature (or even higher) prior to delivery. The system fluid pressure at the elevated temperature becomes so high that the current component design and fabrication techniques – perfected for Ammonia LHPs in the past 30 years – are no longer valid. To mitigate the first two stated obstacles, a variation of the LHP – dubbed Advanced LHP or ALHP[6,7] (for lack of a better word) which was invented by TTH Research Inc. in 1997 – was exploited for the cryocooling applications. Figure 2 depicts a generic A-LHP component layout, indicating the only difference between it and the traditional LHP is the addition of an auxiliary (or secondary) capillary pump. Indeed, any capillary pump constructed for the evaporator can be utilized as the auxiliary pump. The auxiliary pump is plumbed in parallel to the LHP liquid line, connecting the reservoir to the loop condenser. Activated by applying heat to the auxiliary pump casing, it generates a supplementary fluid current, moving vapor accumulated in the reservoir to the condenser for phase change back to liquid. From that perspective, the auxiliary pump can be viewed as a high-capacity "secondary wick" in the contemporary LHP lexicon. The advantages of the auxiliary pump for cryocooling transports shall be discussed in detail in the next section and, hopefully, will become apparent then. Also, the concept of pressure reduction volume (or "hot" reservoir) – borrowed from the previous work of Cryogenic Capillary Pumped Loop[8] (CCPL) – was incorporated into the A-LHP to keep the loop maximum design pressure (MDP) to a manageable level so that the traditional LHP hardware are readily available for use in the cryogenic system. A new concept called Swing Volume (SV) is employed to optimize the system weight for SWaP.

BACKGROUND ON CRYOGENIC ALHP DESIGN AND OPERATION REQUIREMENTS

As alluded in the previous section, the ALHP concept was chosen for the cryocooling transport – thus, called C-ALHP – to take advantage of its inherent features not only to facilitate the potential supercritical start-ups but to lessen the adverse effects of heat gains from the surrounding. Also, a modified approach to moderate the system MDP was introduced for Size, Weight, Power, and Cost (SWaP-C) optimization. Other concerns related to operations in the cryogenic regime – such as Coefficient of Thermal Expansion (CTE) mismatches (e.g. between the capillary pump wick and its casing), metal hardening, and the likes – were addressed in the final reported[9] and, therefore, shall not be repeated here.

Maximum Design Pressure Reduction

The working fluid charging strategy and reservoir volume analysis of a LHP establish the minimum sizing requirement for proper/successful and, above all, safe two-phase operations. Surely, the C-ALHP has to undergo the same process (although, with a slightly different volume accounting) outlined as follows: (i) the loop must contain a sufficient amount of liquid to fill up all components (except the reservoir and parts that are occupied by vapor during operation) and still have liquid left in the reservoir (15% by volume) prior to startups in the coldest environment, (ii) the reservoir internal volume must be large enough to accommodate the liquid expansion during operations with the highest applied heat load and in the warmest environment, and (iii) the entire C-ALHP volume must be adequate to keep the loop pressure below the rated Maximum Design Pressure (MDP) at the maximum temperature T_{MAX} that the fully charged C-ALHP is exposed to during its lifetime whether in operation or not. It is now apparent that the reservoir volume is larger than those of the other components combined. If the C-ALHP is initially in a supercritical state, cooling just the condenser – a volume considerably less than half of the total loop volume – below the fluid critical temperature T_C will not bring the loop pressure under the critical value P_C. No liquid is formed anywhere, let alone the wick structure, to start the C-ALHP. The C-ALHP remains in this condition unless extraneous actions are taken (if available at all)!

If T_{MAX} is lower than T_C, the MDP happens to be the fluid saturation pressure at T_{MAX}. Otherwise, $MDP \approx P_C * T_{MAX} / T_C$. Conceivably, the C-ALHP MDP is much greater than P_C. As an example, the Nitrogen critical point is at T_C = 126K and P_C = 493psia. If T_{MAX} = 353K (+80^{o}C), MDP = 1,381psia. Majority of room-temperature LHP hardware made in America for Ammonia and Propylene were previously flight qualified with the MDP of 600psia or less. So, in order to readily utilize the same hardware, a pressure reduction scheme needs to be devised for the C-ALHPs.

Hot Reservoir

As stated above, the C-ALHP pressure reduction scheme not only alleviates the MDP for pressure containment but also eases up the cooldown/start-up effort from a supercritical state without affecting the otherwise normal operation. An idea adapted from Ref. [8] calls for an extra volume – dubbed “hot” reservoir or HR – be placed in a hot-biased environment (relative to the loop operating temperature), joining up with the C-ALHP at any location on the vapor line by way of a long small-diameter tube as illustrated in Figure 3. Being isolated in its own thermal cocoon, the

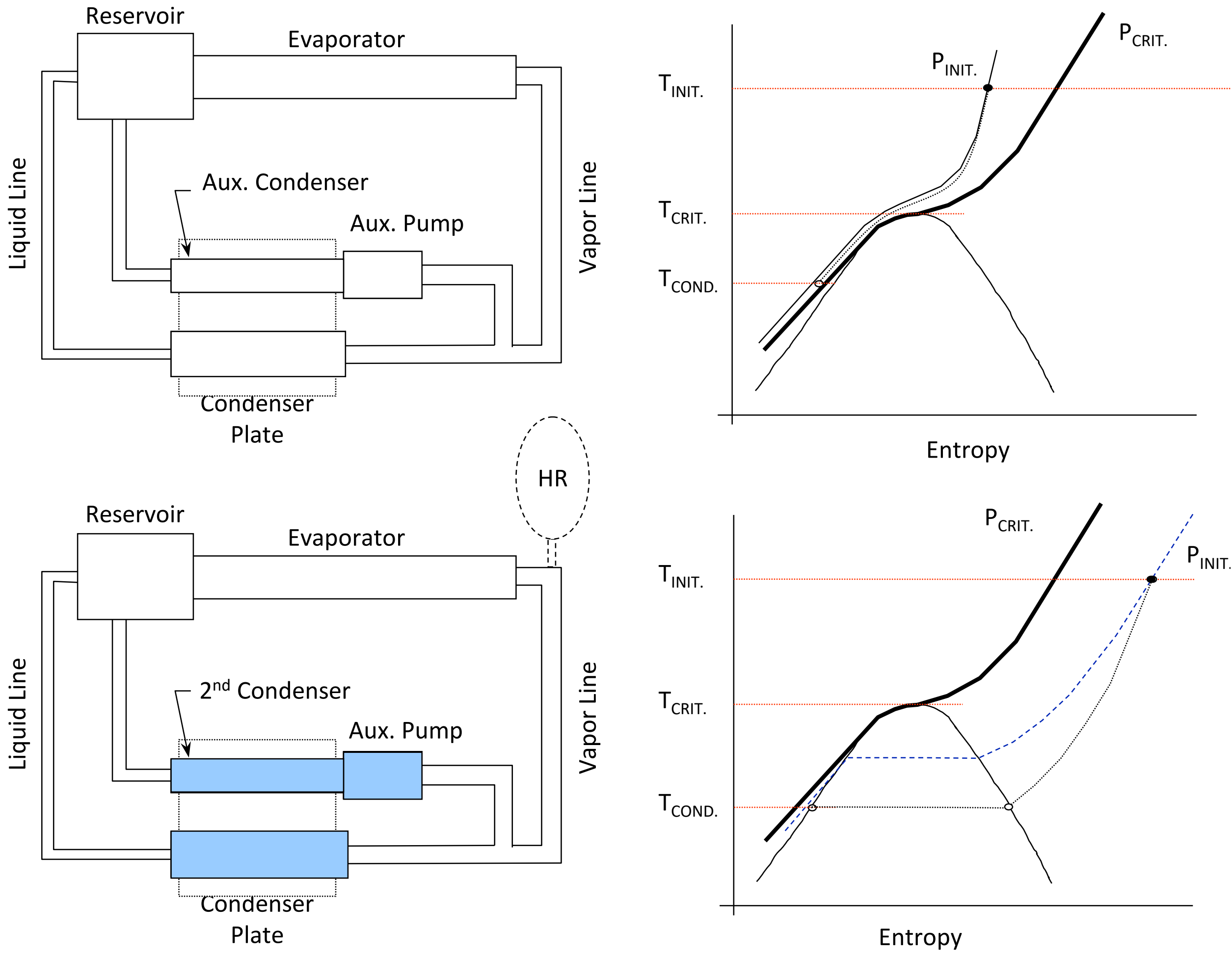


Figure 3. Hot Reservoir Concept for C-ALHP Pressure Reduction.

HR continues to stay hot containing only superheated vapor once the C-ALHP is in operation. Yet, an increase in the fluid charge N_{HR} at T_{MAX} is warranted. The HR enables the MDP to be decreased by a factor of $N_{HR}V_{LHP} / N_{LHP}(V_{LHP} + V_{HR})$, where V_{LHP} & V_{HR} and N_{LHP} & N_{HR} are the volumes and number of moles of fluid in the C-ALHP and HR, respectively. The HR worked out well for the U.S. Air Force N2-ALHP. Having said that, the devil is in the details! With the condenser being the sole component cooled down by the heat sink, V_{HR} still has to be quite large to render the C-ALHP pressure subcritical due to the addition of *not-so-small* N_{HR}. Hence, to satisfy the criteria for sub-critical condition at start-up and <600psia MDP, the required V_{HR} is enormous indeed – 25 times of V_{LHP} for N2-ALHP, 50 times for Ne-ALHP and 100 times for H2-ALHP – making it undesirable for space use insofar as SWaP is concerned.

Swing Volume

Recognizing that cooling a larger portion of V_{LHP} is far more effective in bringing down the C-ALHP pressure for start-ups, the swing volume (SV) concept is incorporated into the pressure reduction scheme as depicted in Figure 4. It is plumbed in line between the vapor line and condenser line, but thermally strapped to the condenser plate. During the condenser cooldown period, the SV

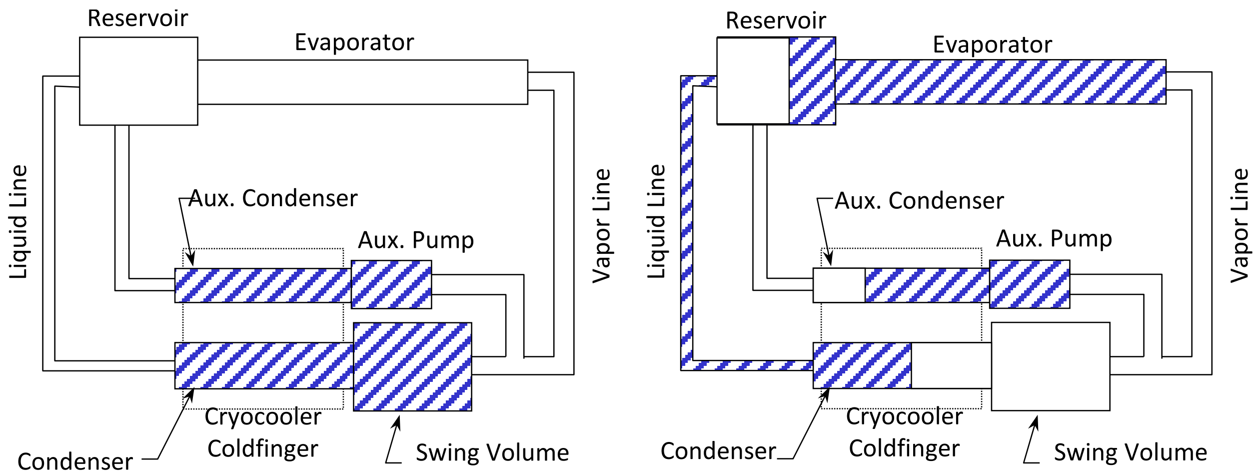


Figure 4. Liquid in SV During Cooldown but Displaced by Vapor During Operation.

temperature drops along with the condenser to contain enough cold fluid (maybe even *liquid*) to reduce the initially supercritical pressure below the critical level for liquid to form in the auxiliary capillary pump to permit its activation. The generated vapor displaces liquid in the SV (if existed) to fill other parts of the loop. In other words, no additional liquid is needed beyond what is sized for normal operation. As far as the swing volume is calculated, the requirement is that the loop pressure reaches the critical value P_C when the condenser plate cools down to some temperature T_1 (above the critical point T_C). For simplicity, at State 1, let the condenser plate be at T_1 and the rest of the loop remains at T_0. Apply the ideal gas law, will obtain:

(1)… $$P_C V_{HOT} = N_{HOT} R T_0 \quad \text{and} \quad P_C V_{COLD} = N_{COLD} R T_1$$

where $V_{COLD} = V_{COND} + V_{SV}$, $V_{HOT} = V_{LHP} - V_{COLD}$ and $N_{LHP} = N_{COLD} + N_{HOT}$. Hence,

(2)… $$V_{SV} = N_{LHP} R T_1 / P_C - (V_{LHP} - V_{COLD})\, T_1/T_0 - V_{COND}$$

Notice that the ideal gas law given above is for illustration purposes only. For the actual SV sizing, it is recommended that another form of the equation of state (e.g. van der Waals) be used that accounts for the gas molecule compressibility at high pressure. From Eq. (2), the required SV can be as small as that of the loop or about 40cc. While a 1,000cc HR is needed to yield a subcritical N2-ALHP condition start-ups, a 40cc achieves the same goal although it fails to meet the 600psia MDP limit. The next attempt is to combine both HR and SV into the pressure reduction scheme.

Hot Reservoir and Swing Volume Combination

The 600psia MDP limit was strictly voluntary for this research endeavor so that TTH Research Inc. could re-purpose the available room-temperature LHP hardware for the demonstration test unit. Providing that the C-ALHP components are properly qualified at a higher MDP, they are suitable for flight. Nevertheless, when other requirements – thermal or otherwise – are levied on the C-ALHP, many advantages of the C-ALHP would be lost. For example, the transport lines have to be made of thick-wall tubing making them too rigid for applications that require flexibility (across-gimbal cryocooling transport). A combination of HR & SV offers a compromise between weight saving and design practicality. To that end, the minimum required HR and SV of the N2-ALHP are 50cc and 180cc; respectively, to accomplish the results that a 1,000cc HR alone delivers.

<u>Basic C-ALHP Operations</u>

Cooldown and Start-up Process from an Initially Supercritical State

The cooldown and start-up sequence leading to the C-ALHP nominal operation is presented in Figure 5. Assuming that the entire C-ALHP including the HR and SV is in a supercritical state 0_{SV} at T_0 and P_{0SV} (both fluid pressure and temperature are above the corresponding critical values). The loop pressure begins to decrease soon after the condenser plate temperature is pulled down by the heat sink (e.g. cryo-coolers). When the condenser plate arrives at some temperature just above T_C, say T_1; the collective volume V_{COLD} of the condensers, auxiliary capillary pump, and SV traps enough colder fluid (still supercritical) for the system pressure to reach P_C at State 1_{SV}. Further reduction in temperature will cause the fluid in V_{COLD} to cross over to the super-heated vapor regime until becoming truly subcritical at State 2_{SV}. Liquid begins forming in V_{COLD} as soon as State 3_{SV} is achieved, from which time the auxiliary capillary pump temperature drops rapidly to draw in the liquid and saturate the pump wick. The auxiliary pump can be activated right away but it is advised to wait until the system is stabilized at State 4_{SV}. When appropriate, heater(s) attached to the auxiliary pump casing is turned on to initiate the vapor generation. The vapor, in turn, displaces the cold fluid in the SV and re-distributes it to the warmer parts of the C-ALHP driving the main evaporator/reservoir temperatures (still linger near T_{0SV} at this moment) quickly toward the local saturation temperature T_{4SV} for liquid to prime the main evaporator/ reservoir, i.e. the evaporator is ready to acquire the heat load from the IR sensors/detectors for transport to the heat sink.

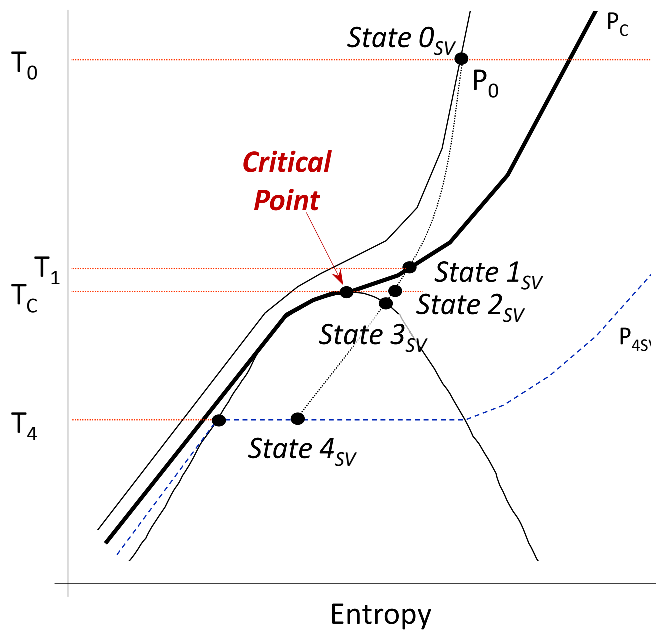


Figure 5. C-ALHP Initially Supercritical Start-up.

Management of Environmental Heating

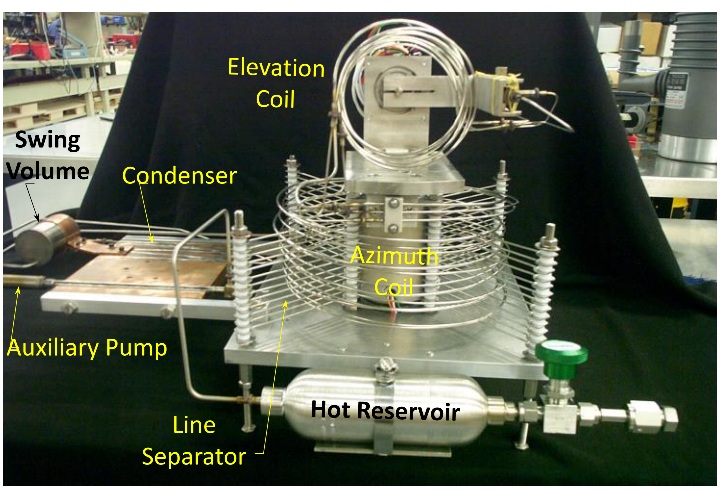
Like many other cryogenic equipment, the C-ALHP needs be well-insulated thermally to prevent a much warmer operating environment to affect its performance. However, it does not matter how good the insulation is, the C-ALHP liquid flow is almost stagnant at low heat load situations (e.g. standby mode) allowing the returning liquid ample of time to rise, plausibly, all the way to the surrounding temperature. The possibility of the returning liquid changing phase or "flashing" (however remote) is without question. At any point when the liquid temperature approaches an uncomfortable level, the auxiliary pump is activated to maintain/stabilize the liquid subcooling. From another perspective, the auxiliary pump serves as a high-capacity external secondary wick for the main evaporator pump, moving vapor – generated in the reservoir by the heat leaks – to the condenser for heat condensation. Note that the condensation (two-phase) heat transfer is far more efficient than the single-phase counterpart needed to produce liquid subcooling at the condenser exit. As a consequence, the C-ALHP overall thermal conductance gets better with the auxiliary pump operating than without.

C-ALHP PERFORMANCE DEMONSTRATION

Several C-ALHPs were constructed and performance tested in vacuum with different cryogens as the working fluid to verify their functionalities – the thermal management/control capabilities and operational limits for different applications, in particular – in the temperature range between 4K and 100K. The C-ALHP demonstration effort was an unqualified success and, without a doubt, validated almost all technical claims made previously. The test results from the C-ALHPs showing various aspects of operation are presented in the following sections.

Flexible N2-ALHP for Across-Gimbal Cryocooling

In 2000, the U.S. Air Force solicited ideas for cooling IR telescopes mounted on a 2-axis gimbal for pointing agility. The dissipated instrument waste heat was to be transported to and removed by cryocoolers placed on the main spacecraft to isolate unwanted vibrations from the telescope. The ALHP technology using Nitrogen as the working fluid (thus N2-ALHP) was proposed to provide the aforementioned across-gimbal cryocooling transport in the temperature range of 75-110K. The actual N2-ALHP performance demonstration unit is photographed in Figure 6. The transport lines – liquid, vapor, and auxiliary lines – were shaped together to form two (2) helical coils: one for the 0-360° azimuth rotation and the other for the -90°/+90° elevation rotation. So, right off the bat, the challenge was to design a system lightweight/flexible enough to meet the low torque specifications. Being attached to the instrument, weight saving of the main evaporator/reservoir assembly was the first priority. But equally important, the transport lines had to be long to form



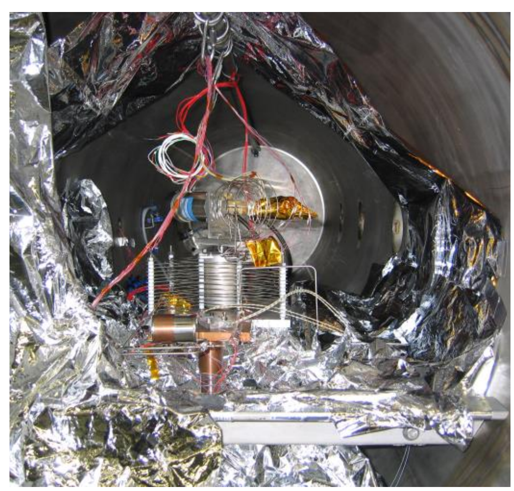


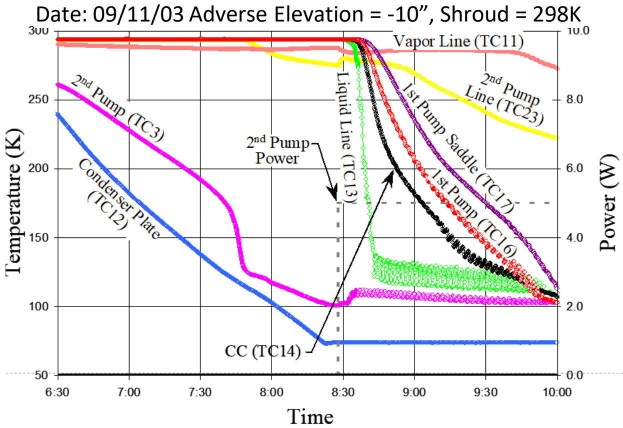
Figure 6. As-Built N2-ALHP.

Table 1. Volume Breakdown of N2-ALHP Components

Components	Description	Vapor (cc)	Volume 2-ϕ (cc)	Liquid (cc)
Primary Capillary Pump				
- 1.4 micron wick	0.43"ODx4mmIDx1.36"Lx0.40porosity			1.3922
- vapor grooves		0.5320		
- reservoir	0.5"ODx1.5"L		3.0722	
Secondary Capillary Pump				
- 1.4 micron wick	0.43"ODx4mmIDx1.36"Lx0.40porosity			1.3922
- vapor grooves		0.5320		
Main Vapor Line	3/32"ODx0.082"IDx4.5mL	15.3319		
Additional Vapor Line	3/32"ODx0.082"IDx5.595"L	0.4842		
Liquid Line	1/16"ODx0.051"IDx4.5mL			5.9308
2nd Pump Line (Vapor)	1/16"ODx0.051"IDx4.5mL	5.9308		
Condenser Line	1/8"ODx0.093"IDx26.27"L		4.5407	
2nd Condenser Line	1/4"ODx0.194"IDx6"L		2.9063	
Swing Volume			60.000	
Expansion Reservoir Line	3/32"ODx0.082"IDx40"L	1.9630		

the flexure coils which ran counter to the desire to keep the reservoir small (because of the line volume). Notice that larger line volumes also adversely affected the HR/SV size. But, since they were located on the spacecraft, heavier HR/SV had little or no impact on the rotation torque (still viewed negatively from the SWaP perception). Small-diameter lines helped reducing the main reservoir volume but caused significant pressure drops ($\propto 1/D^4$) which, in turn, demanded wicks with smaller pore size. For the same inner-diameter, thinner-wall lines offers more flexibility for the coils at the expense of pressure containment (i.e. requiring a larger HR). In summary, the N2-ALHP was actually a complex time-consuming trade study involving many parameters in addition to those mentioned above. Table 1 lists the physical dimensions and wick properties of the N2-ALHP components of the Thermal Demonstration Unit (TDU). Another unit called the Mechanical Demonstration Unit (MDU) was also constructed with everything that the TDM had but without the internal structure (e.g. wicks). It endured a mechanical cycling test to determine the lifetimes and failure modes of the coils. The MDU was pressurized to 20psia with dry Nitrogen gas.

The MDU cycling test started on July 27, 2003 and ran continuously for two weeks. On August 10, the mechanical cycling stopped because the pressure sensor detected a pressure drop-off in the 1/16”O.D. lines. Before the cycling was halted, the counter recorded 256,417 cycles. This was quite a surprise since analytical predictions indicated that the 1/16”O.D. lines (being more flexible) would last longer than the 3/32”O.D. line. Two arrows in Fig. 9 show the approximate locations of the breakage on the 1/16”O.D. lines. They were about 3/4" from the upper Mica-Glass support on the azimuth coil, where it was pushing the lines during the cycling. Perhaps, when the 1/16”O.D. lines became distorted, a stress concentration developed near the upper Mica-Glass anchor causing the premature line failure. Subsequently, the broken 1/16”O.D. line were isolated from the MDU and the cycling test resumed for the 3/32”O.D. line. The renewed cycling test lasted an additional 44,000 cycles, cumulating about 310,000 cycles for the 3/32”O.D. line. The location of the 3/16”O.D. line breakage, found with a leak checker, was about 180^o from the Mica-Glass bracket on the elevation coil. One interesting observation was that the 3/32”O.D. line definitely experienced work hardening of the material.



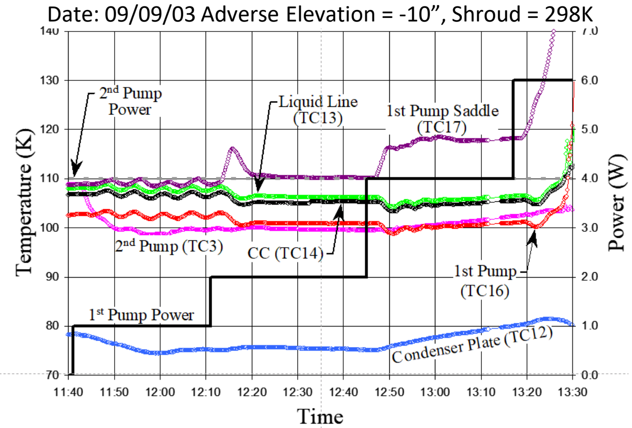


Figure 7. N2-ALHP Supercritical Start-up.

Figure 8. N2-ALHP High Power Test.

The performance testing of the N2-ALHP TDU began on Jul. 02, 2003, lasted more than two months, and finally completed on Sep. 13, 2003. All tests were conducted in a 30-inch diameter x 10-foot length TV chamber. All tests were performed in a 298K surrounding. The condenser plate was mounted on the cryocoolers coldfinger for heat removal. The chamber was equipped with lighting fixture and a see-through glass window so that a digital video camera was used to record the flexure motion inside the TV chamber. Figure 7 presents a cooldown/start-up period of the TDU. At 6:00, the cryocooler was activated and, almost immediately, temperatures of the condenser plate (TC12) and auxiliary pump (TC3) decreased with a steady rate. When the condenser plate (TC12) reached 126K (i.e. critical temperature of Nitrogen) at 7:40, TC3 dropped quickly suggesting that the fluid pressure became subcritical, i.e. liquid formed in the condensers first and then got wicked into the auxiliary pump. The rest of TDU was still at 298K, including the main evaporator/reservoir (TC16 & TC14). At 8:27, 5W was applied to the auxiliary pump to initiate the fluid flow and, in less than 5 minutes, at 8:32, the liquid line (TC13) temperature dropped rapidly and, soon after, was followed by a large temperature decrease of the CC (TC14), and main evaporator itself (TC17). At 10:00, TC14 and TC17 are clearly below 126K approaching 100K which undoubtedly indicated the presence of liquid in the main evaporator. The cooldown and main wick priming process was declared successfully accomplished. The N2-ALHP was ready to receive heat.

Results of the N2-ALHP high power are given in Figure 8, showing the maximum heat flow of 9W generated by 5W on the main evaporator and 4W on the auxiliary pump. It is important to note that the current TDU TV test configuration imposed an adverse elevation of 10" that the main capillary pump had to overcome. In any event, the analytical model predicted the N2-ALHP heat transport limit, to be 14W with zero elevation at 100K. The Ne-ALHP 10" adverse elevation in 1g testing diminished the capillary pumping head by 30% of the maximum limit. Thus, the theoretical limit of the C-ALHP in the TV configuration would be 10W, which is consistent with the actual data of 9W. Note that the condenser plate temperature (TC12) was controlled at 75K and, in this test, the cryocooler was indeed able to maintain TC12 at 75K for the first two steps. But, in the third power step (4W on auxiliary pump and 5W on the evaporator) and also the fourth step (4W + 6W), TC12 kept on rising until the evaporator deprimed at 13:25, suggesting that the cryocooler was unable to absorb a total heat load of 9W or higher at 75K. Operating in a 298K

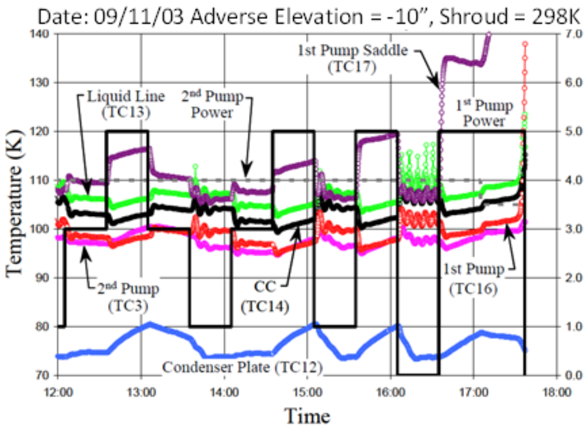


Figure 9. N2-ALHP Power Cycling.

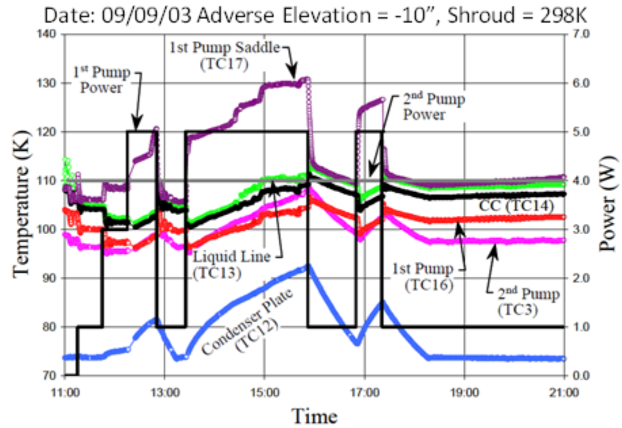


Figure 10. N2-ALHP Operation Under Rotation.

TV chamber was too taxing for the cryocooler! Figure 9 presents the power cycling test data. The N2-ALHP performed well exhibiting no anomalous behaviors in the standard min-max power step ups/downs profiles, providing that the auxiliary pump was in operation carrying more than 4W. Again, as expected, TC12 increased with the total load of 9W but decreased right away as soon as the power was lowered. The evaporator did fail when it was turned off at 16:05 for ½ hour and re-started at 16:35. During the evaporator dormancy, only 4W from the auxiliary pump did not generate a robust liquid flow to overcome the environmental heating, allowing a large quantity of vapor to enter the evaporator to (partially) dry out the wick. So, when 5W was applied to the evaporator to bring it back on line, an evaporator failure happened. Figure 10 shows the results of the TDU operation while the helical flexure mechanisms were in constant rotation. Two stepper motor controllers were activated by PC-based software to send commands to the motors inside the TV chamber and started the rotation of the flexure mechanisms. An azimuth rotation of 0–360° was done first and then followed by a 0–45° rotation of the elevation coil while the TDU underwent a standard power cycling test. A moderate 1W/3W/5W step-up preceded a more severe 5W/1W/5W cycling. A constant heat load of 5W was maintained on the evaporator starting at 13:20 for 3 hours while the flexure coils were in rotation. Once again, the condenser plate temperature (TC12) did not stop rising for the same reason stated above.

H2/Ne-ALHP for 20-30K Cryocooling

An ALHP similar in design and system layout as the N2-ALHP was put together as a Demonstration Test Unit (DTU) for NASA using Neon or Hydrogen as the working fluid (H2/Ne-ALHP henceforth). Unlike the N2-ALHP technical specifications which spelled out the applications the technology was intended for, NASA was rather inclined to obtain proof-positive evidence for the assessment/evaluation of the LHP thermal management capabilities to support future space telescopes in the Far Infrared (FAR) spectrum. At the time, James Webb Space Telescope (JWST) was in the work, hence, TTH Research Inc. recommended Neon or Hydrogen be used for the demonstration of the cryocooling transport in the 20-40K temperature range although the loop would perform equally well with other cryogens including Helium. Like the N2-LHP, the H2/Ne-ALHP design underwent extensive SWaP-C optimization trades, culminating in the DTU shown in Figures 11 and 12. Note that the HR volume needed to be resized/replaced when a particular cryogen is selected. Table

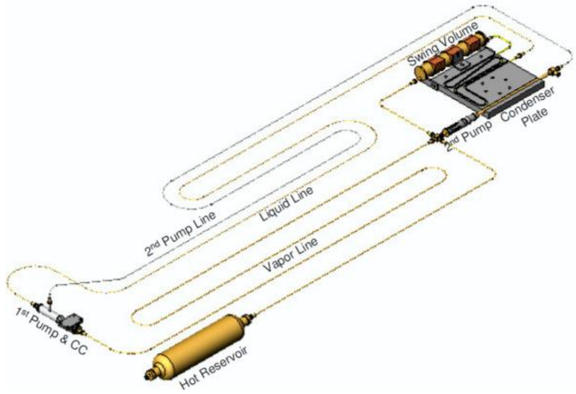


Figure 11. H2-ALHP Component Layout.

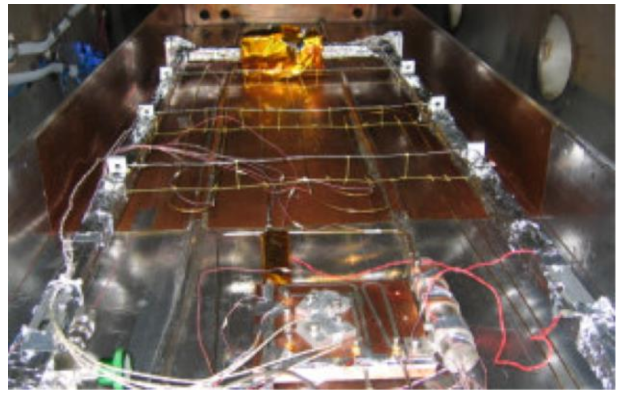


Figure 12. H2-ALHP in Thermal Vacuum Chamber.

Table 2. Volume Breakdown of N2-ALHP Components

Components	Description	Vapor (cc)	Volume 2-ϕ (cc)	Liquid (cc)
Primary Capillary Pump				
- 1.89 micron wick	0.43"ODx4mmIDx1.36"Lx0.40porosity			1.2760
- vapor grooves		0.5320		
- reservoir	0.23"ODxTBD"L		6.0000	
Secondary Capillary Pump				
- 2.01 micron wick	0.43"ODx4mmIDx1.36"Lx0.40porosity			1.2760
- vapor grooves		0.5660		
- reservoir	0.23"ODxTBD"L		1.8880	
Main Vapor Line	3/32"ODx0.082"IDx2.5mL	8.4660		
Additional Vapor Line	1/16"ODx0.082"IDx5.595"L	0.4810		
Liquid Line	1/16"ODx0.051"IDx2.5mL			3.2310
Additional Liquid Line	1/16"ODx0.051"IDx2.5mL	3.2310		
Condenser Line	1/8"ODx0.093"IDx26.27"L		2.9240	
Additional Condenser Line	1/4"ODx0.194"IDx6"L		2.9060	
Swing Volume			60.000	
Expansion Reservoir Line	3/32"ODx0.082"IDx40"L	3.4410		

2 summarizes the H2/Ne-ALHP DTU components' dimensions and properties. The demonstration test program lasted about 2 months with Neon and more than 4 months with Hydrogen. All tests were conducted in a 30-inch diameter x 10-foot length thermal vacuum (TV) chamber. The loop components were laid out co-planar (except the HR) so that no favorable or adverse effects of gravity altered the loop performance (including start-ups). The entire DTU (excluding the HR) was enclosed in a rectangular shroud, of which the temperature was controlled at a desired set point by regulating liquid Nitrogen (LN2) flow to the shroud from a supply bottle located outside the chamber. The cryocooler coldfinger had its own shroud (also cooled by the same LN2 supply) to lessen heat parasitics from the main shroud and chamber wall.

Due to page limit of this paper, only the Hydrogen test data are disclosed below. Figure 13 reveals one of the H2-ALHP DTU cooldown/startup process from a supercritical state. The evaporator/ reservoir assembly was at 298K. The cryocooler and the LN2 flow to the shroud were activated around 7:30am. The shroud temperature was set and maintained at 200K. At 9:00, temperatures

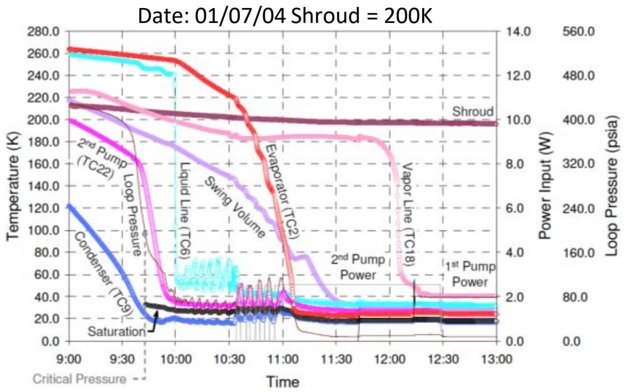


Figure 13. H2-ALHP Cooldown & Start-up.

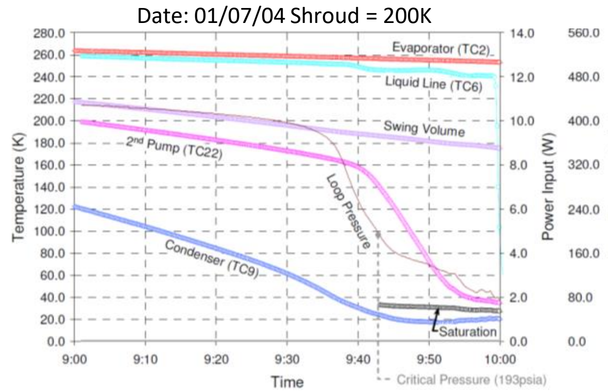
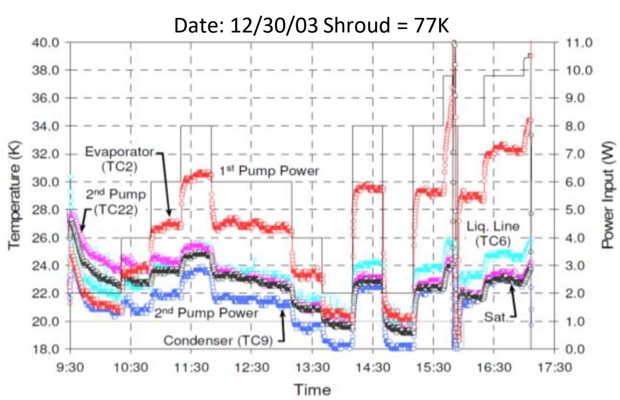


Figure 14. H2-ALHP Cooldown.

of the condenser plate (TC9) and shroud dropped quickly to about 122K and 214K, respectively. Since the auxiliary pump and SV were poorly coupled to the condenser plate, their temperatures (TC22 & TC13) at 9:00 were lagging behind (200K and 220K). The rest of the loop was still very hot (~260K). Hence, the system pressure of 431psia was recorded at 9:00. At this pressure and temperature level, the thermodynamic state in H2-ALHP was definitively supercritical. The loop temperature and pressure continued to decrease but still remained supercritical until 9:42 as presented in detail in Figure 14. Soon after, the pressure became subcritical and the condenser plate (TC9) dropped below 33K (T_c of Hydrogen). Liquid started to form in both condensers and was wicked to the auxiliary pump from inside out; enabling it (TC22) to cool down rapidly, in the process, vaporizing the liquid in a phenomenon called *self-start* (heater on auxiliary pump casing was not turned on). The generated vapor flowed into the main condenser first (path of least flow resistance) pushing subcooled liquid into the liquid line. The subcooled liquid brought down the liquid line temperature (TC6) to 50-65K almost immediately. The event took place between 9:42 and 10:00 demonstrated that cooling of the condensers and the SV enabled the system pressure to drop from 431psia to critical point (193psia) in less than 1 hours.

Results of the combined power cycling and heat transport limit test are given in Figure 15. The shroud was maintained at 77K throughout the test. Following the successful start-up of the H2-ALHP according to the procedures outlined in the previous paragraph, the auxiliary pump power was lowered to 1W for this particular high power test (in later tests, the auxiliary pump power



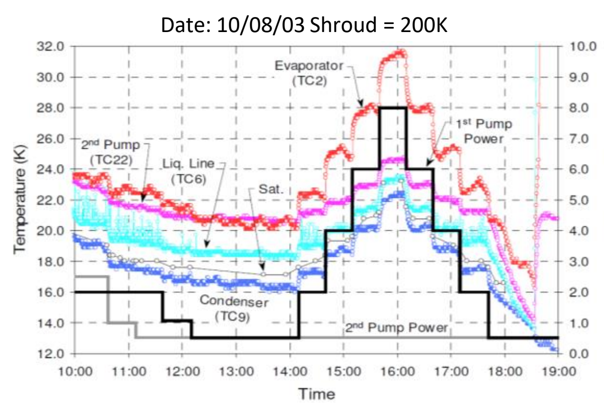
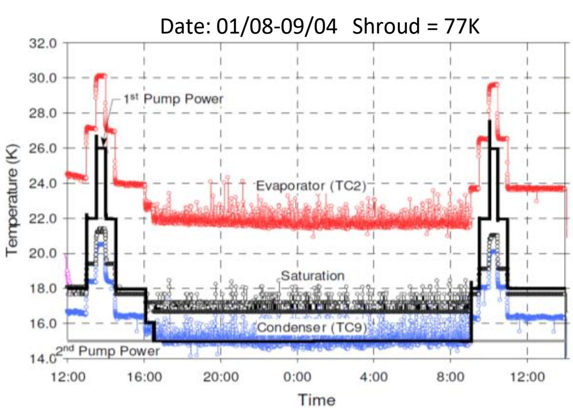


Figure 15. H2-ALHP Power Cycling &Power Limit. **Figure 16. H2-ALHP Power Cycling in Hot Env.**

was maintained at 0.5W). The main evaporator power was stepped up with an increment of 2W to 8W and then stepped down with the same amount to 2W. The initial gradual power step-ups/downs were followed by a more severe 2W/8W/2W/8W cycling of the evaporator power. The rapid power cycling was the simplest way to unveil problems (if any) with the H2-ALHP. The loop responded gracefully with respect to the power changes. When the DTU temperatures became steady at 15:38, the evaporator power was increased to 9.8W (from 8W). TC2 on the evaporator took off almost immediately, indicating a pump deprime. The evaporator power was turned off at 15:50 but turned back on with 8W at 15:55 to re-start the loop. At 16:00, the auxiliary pump was turned off. The loop temperatures (TC2, TC6, and T_{SAT}) reached steady state at 16:20, at which time, the evaporator power was increased to 9.8W. The DTU did not deprime this time and became steady at 16:25. The evaporator power was raised to 10.6W but it deprimed soon after at 16:35. The transport limit of the H2-ALHP DTU was, hence, determined to be 9W with 1W was applied to the auxiliary pump and close to 10W operating by itself. It is also interesting to note that in the last two steps of this test, the evaporator power of 8W and 9.8W created liquid velocities fast enough to overcome the environmental heating even without help from the auxiliary pump. As a consequence, the system pressure drop was reduced *just enough* to allow the H2-ALHP to operate with 9.8W in the last step. In fact, the high power test was repeated several times with different shroud temperatures, the heat transport capability was determined to be almost 10W in an environment of 77K and 9W in 235K. The power cycling test was repeated in a hot environment (shroud was kept at 235K). The test results are shown in Figure 16. The H2-ALHP transitioned smoothly to the stair-stepping of power.

As alluded in the Introduction section, operating any LHP at low power in a hot environment is problematic simply because the liquid transit velocity is so low that the surrounding temperature has plenty of time to heat up and perhaps vaporize the returning liquid. A test carried out on 01/05/04 verified that, utilizing the auxiliary pump the way it was originally intended, the H2-ALHP did survive an extremely long standby period while dissipating just a token amount of heat in a very hot environment. Figure 17 presents the results of the long-duration low-power operation, capped off by a sink temperature variation. The shroud temperature and auxiliary pump power was maintained at 235K and 0.5W, respectively. However, before the test began, an evaporator power cycling operation with the profile 2W/4W/6W/4W/2W was exercised to set up the *before*



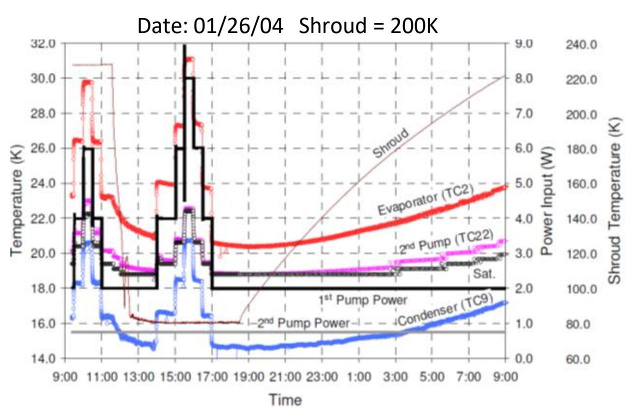


Figure 17. H2-ALHP Long-Duration & Low Power. **Figure 18. H2-ALHP Shroud Temp. Variation.**

H2-ALHP performance. It was repeated once the said test was done for direct comparison with the previous results. The evaporator power was kept at 0.5W overnight from 16:30 to 9:00 the next day. In the low power operation (0.5W/0.5W), the temperatures (including T_{SAT}) were not stable (fluctuated ~2K). It was suspected that 0.5W of the 2nd pump power was barely enough to manage the heat parasitics causing intermittent energy imbalance in the CC. Nonetheless, the H2-ALHP-2 remained functional following the low-power operation.

A test conducted on 01/26/04 investigated the effects of a large attached thermal mass and sink temperature change on the DTU transient response. A 35g Aluminum block was attached to the evaporator to increase its thermal time constant for this test. The test was carried out following the long duration (over-the-weekend) run, described in the paragraph above, with a combination of 2W/0.75W applied to the evaporator/auxiliary pump. The auxiliary pump power was 75W and stayed constant. The shroud was maintained at 230K from 9:20 to 11:20, during which time, the evaporator power input was cycled with the profile: 2W/4W/6W/4W/2W. After that, the shroud was lowered to 80K. Another power cycling was performed: 2W/4W/6W/ 8W/6W/4W/2W as depicted in Figure 18. The LN2 flow to the shroud was then halted to allow its temperature to warm up slowly while keeping the applied power at 2W/0.75W. As expected, all temperatures increased as the shroud warmed up. Other than that, nothing out of the ordinary was observed during the temperature transition.

<u>Large Area Cryocooling by a LHP</u>

For IR telescopes, not only do the sensors/ detectors need to be cooled down to the cryogenic temperature but their optical trains and baffles are also maintained at or near the same level to minimize stray light. Traditionally, the LHPs are used as "spot" coolers because they have limited heat acquisition footprints. To manage the optical train/baffle temperatures, the heat collection must be made over large surfaces and/or from various locations. In addition, to minimize the optical distortion, the components must be kept isothermal as possible. Accordingly, TTH Research Inc. utilized a conventional LHP in an unconventional way for the said thermal management as depicted in Figure 19. Table 3 lists the volume breakdown of the He/Ne-LHP large-area cryocooling Demonstration Test Unit (DTU). The evaporator/reservoir assembly is located near the condenser and thermally strapped to the heat sink. Once the evaporator is activated by turning on heaters bonded to its casing, the generated vapor flows to the condenser first to change phase back to liquid. The liquid is transported to the heat-dissipating area(s) to partially absorb the waste heat and then looped back to the condenser for *partial* heat rejection. Instead of returning the liquid back to the evaporator to

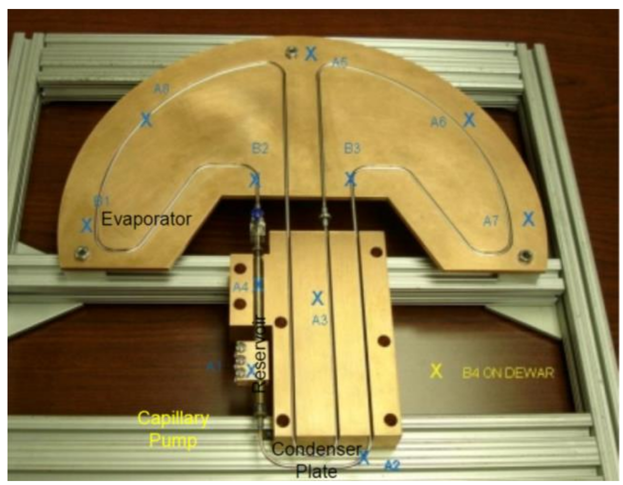


Figure 19. He/Ne-LHP Large-Area Cryocooling.

Table 3. Volume Breakdown of He-ALHP Components

Components	Description	Vapor (cc)	Volume 2ϕ (cc)	Liquid (cc)
Capillary Pump				
- 1.25micron wick	0.21"ODx0.078"IDx2"Lx0.40%			0.3914
- reservoir	0.25"ODx0.02"wallx2.5"L		1.1352	
Vapor Line	3/32"ODx0.016"Wx12"L	0.4908		
Evaporator Line	3/32"ODx0.016"Wx40"L			3.5000
Condenser	3/32"ODx0.016"Wx13"L		0.7361	

complete the LHP cycle, it is looped back and forth between the heat source(s) and condenser several more times as necessary to satisfy the heat transport requirement with a single LHP. The LHP large-area cooling concept works with any fluid – cryogens included. Since the evaporator/reservoir along with the condensers are thermally strapped to the sink, the combined chilled volume is large enough to render the loop pressure subcritical prior to start-ups without needing a SV/HR. Nevertheless, the MDP may still be an issue if the existing hardware are to be used. As an example, Helium critical point is at 5.2K and 0.2275Mpa. Therefore, without a SV/HR, the MDP at 353K (+80^{o}C) is 2,243psia which is much higher than the MDP of >600psia that the existing LHP hardware were qualified for.

The Helium-Neon large-area cryocooling Demonstration Test Unit (He/Ne-LHP DTU) was built and performance tested – first with Neon and later with Helium. The Ne-LHP tests were carried out in a conventional TV chamber. The Ne-LHP test results showing the loop cooldown and start-up are presented in Figures 20 and 21. As mentioned above, the combined evaporator/reservoir volume was larger than 60% of the remaining loop volume and, thus, was able to bring the system to a critical state prior to start-ups. Liquid formed in the capillary pump soon after the condenser temperature dropped below the Neon critical value (4.5K). It was drawn into the capillary pump wick to established capillary action for the loop to start. The circulation of Neon around the loop quickly brought the large-area (evaporator) temperatures toward the critical point at 9:45.

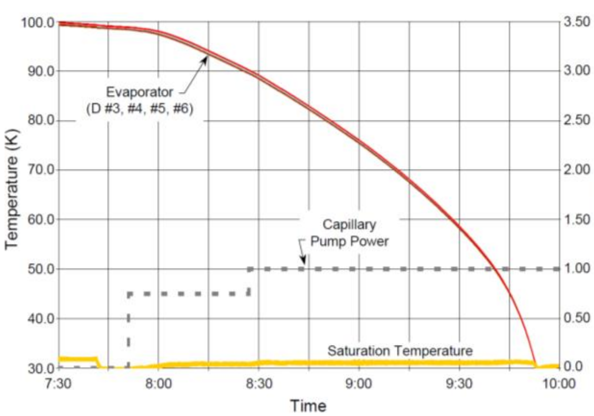


Figure 20. Ne-LHP Cooldown & Start-up.

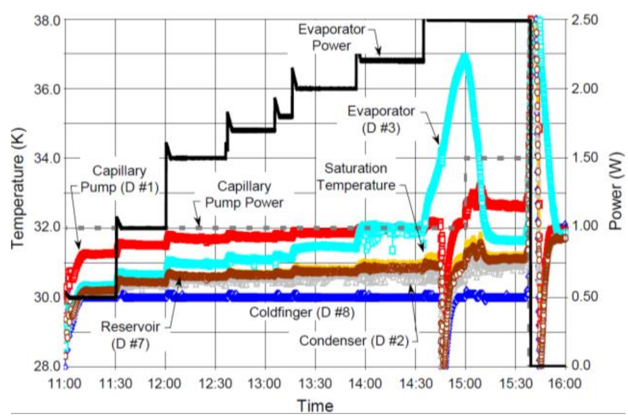


Figure 21. Ne-LHP Power Limit Test.

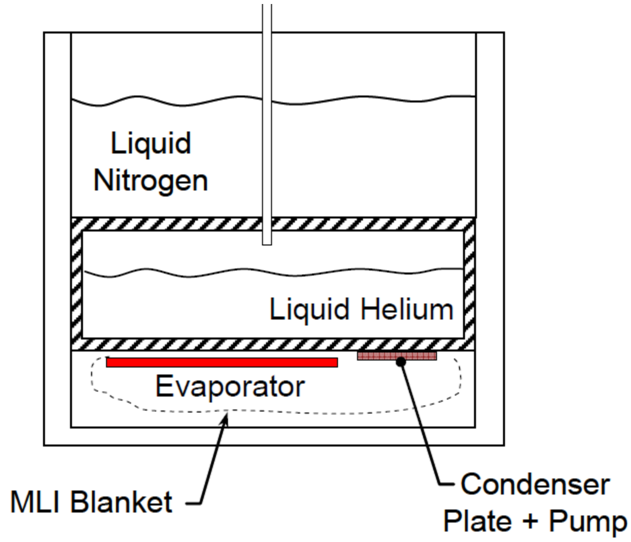


Figure 22. He-Dewar and He-LHP DTU Test Setup.

The performance testing of the He-LHP large-area cryocooling DTU began on April 20, 2006 and ended on May 12, 2006. The DTU was filled with Helium to a pre-determined charge pressure of 32psia that was calculated from a system volume analysis at 298K. Past experience suggested that a margin of 10% be added to the calculated charge pressure to avoid undercharging the test loop. The DTU was placed inside a Helium Dewar as shown in Figure 22 for testing. The entire test loop, except the condenser plate, was thermally insulated from the Dewar by a 15-layer MLI blanket and G-10 stand-offs. The condenser plate, on the other hand, was attached to the outer surface of the Helium chamber for heat rejection (i.e. the Helium chamber was used as the heat sink). The He-LHP test results are given in Figure 23.

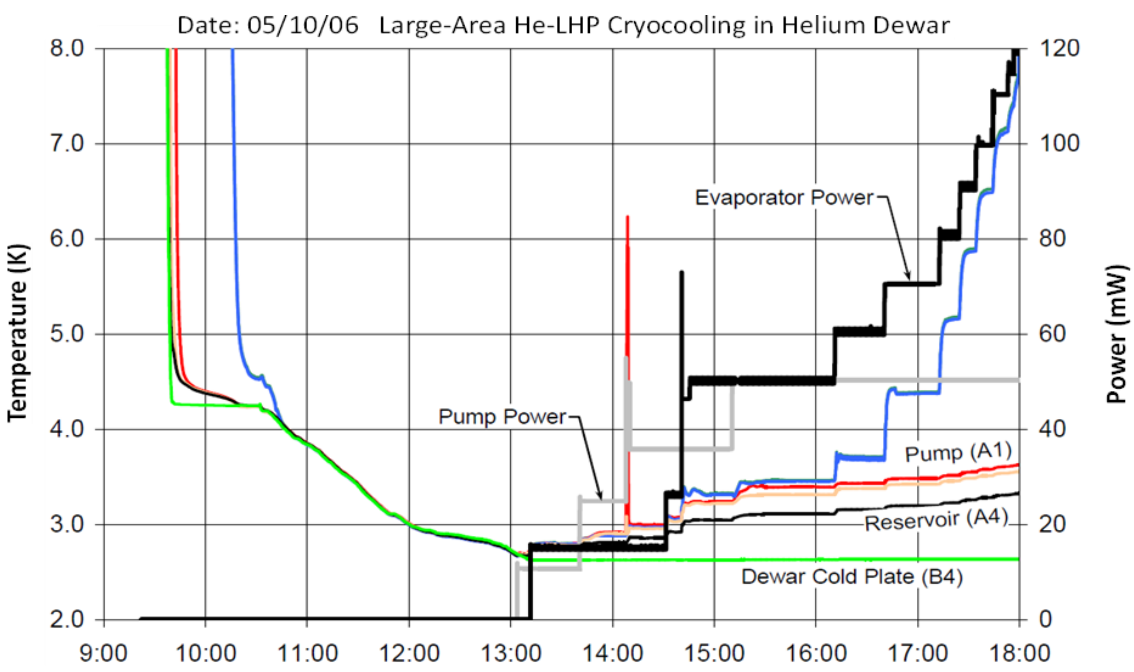


Figure 23. Large-Area He-LHP Test – Cooldown and Maximum Power Limit.

CONCLUSION

Results of the research presented in this paper verified that the capillary pumped LHP technology worked very well in the cryogenic temperature range of 4-100K providing that start-ups from a supercritical thermodynamic state could be achieved without resorting to extraneous means. In the spirit of "no mechanical moving part," a variation of the LHP called Advanced LHP or ALHP was utilized to (i) facilitate the cooldown process and (ii) manage the environmental heating from the surrounding during operation. In addition, a Hot Reservoir / Swing Volume concept was used to bring the Maximum Design Pressure (MDP) below 600psia so that the LHP hardware designed and fabricated for room-temperature LHPs could be made readily available for use in the version of cryogenic ALHPs. Once the initially supercritical cooldown and start-up were successful, all aspects of the ALHP performance were demonstrated including stair-stepping of power input, high power limit tests, long-duration low-power standby mode, operations in hot/cold shrouds, with and without a large attached thermal mass.

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