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Development of a Pumped Two-phase System for Spacecraft Thermal Control

Michael C. Ellis[1]
Advanced Cooling Technologies, Inc., Lancaster, PA 17601

Richard C. Kurwitz[2]
Texas A&M University, College Station, TX 77843

Advanced Cooling Technologies, Inc. (ACT) is developing an active Two-phase Thermal Management System (TPTMS) that relies on a single-phase liquid pump to drive two-phase flow through multiple heat sources and sinks distributed in parallel and in series while providing phase management using the momentum of the working fluid. This system is designed to address challenges discussed in the NASA Thermal Management Systems Roadmap, Technology Area (TA) 14. The use of a liquid pump to drive the system allows the working fluid to overcome large pressure drops with low power consumption. This feature, in turn, provides the ability to transfer waste heat over large distances, which is defined as a top technical challenge in TA14, Section 1.4. Additionally, flow can be driven through multiple heat exchangers or cold plates to either collect or release thermal energy. Arranged properly, this feature allows for heat load sharing, which is also defined as a top technical challenge. Added to these benefits are those intrinsic to two-phase heat transfer: near-isothermal operation, a two order of magnitude increase in the heat transferred per unit mass (TA14) and the ability to handle high heat fluxes with the appropriate heat exchanger design. Lastly, Section 2.2.3.2 of TA14 discusses the need for microgravity separators, which is an integral part of the TPTMS. In the first phase of the project, ACT fabricated and tested a prototype system that included a sliding vane pump, two eductors, three cold plates, a condenser, and a phase separator. Two of the cold plates were arranged in series with the other in parallel with these. Testing demonstrated thermal energy transport from the cold plates to the condenser and operation of the separator. In the next phase, ACT will pursue microgravity testing of a full scale system.

I. Introduction

COMPARED to single-phase systems at significant heat loads, two-phase thermal management systems reduce spacecraft mass, volume, and power usage while providing performance improvements such as enhanced heat transfer and temperature uniformity. The Thermal Management Systems Roadmap (TA14) challenges researchers to develop two-phase thermal management systems that can manage high heat loads with improved temperature control and acknowledges the need for microgravity phase separator development. To address this need, ACT is concentrating on a TPTMS design that relies on a passive, momentum-driven accumulator to provide phase management and flow distribution to multiple thermal loads with no restriction on component placement or transport line routing within the spacecraft. As this system relies on the latent heat of the working fluid to transport thermal energy, the TPTMS will provide constant temperature heat transfer to distributed spacecraft components, facilitate distribution of collected heat between these components, and allow components to exchange heat through condensation or evaporation, depending on component needs.

Currently, spacecraft thermal management involves single-phase pumped loops or passive, two-phase, capillary devices such as Constant Conductance Heat Pipes (CCHPs), Capillary Pumped Loops (CPLs), and Loop Heat Pipes (LHPs). For low power applications, capillary devices are more attractive than single-phase systems due to their passive operation and use of latent heat transport, which allows for isothermal operation and less working fluid mass. Among these devices, CCHPs are the most common for satellite thermal management systems. However, CCHP systems are design specific, which makes them difficult to interchange between applications and intolerant of design changes within an application. In addition, complex CCHP geometries are impossible to ground test. CPLs

[1] Lead R&D Engineer, Strategic Products, 1046 New Holland Avenue, Lancaster, PA 17601
[2] Senior Lecturer, Dept. of Nuclear Engineering, Texas A&M University, 3133 TAMU, College Station, TX 77843

and LHPs, which share common operating principles, are somewhat more flexible as they are capable of pumping over significant distances using non-wicked transport lines and ground testing is not an issue. CPLs and LHPs have found application in several programs, such as the Hubble Space Telescope and Geoscience Laser Altimetry System. However, they are known to encounter start-up and transient anomalies which complicate operation. To provide reliable start-up and de-priming recovery, electrical heaters or thermoelectric devices are often employed, which somewhat negates the passive advantage of capillary loops[1].

Moreover, all capillary devices share two disadvantages. First, they are restricted to relatively low powers of about a kilowatt or less, depending on wick design and available capillary head. This presents a problem considering increasing spacecraft thermal management demands. To meet higher powers, multiple capillary devices are needed. Second, capillary systems entail complicated fabrication processes related to wick performance and working fluid purity. These processes lead to higher fabrication and testing costs, estimated to be $1 million dollars for some NASA LHPs.

Single-phase pumped loops have been identified as a more flexible alternative to capillary systems in that they can operate with multiple heat sinks and sources, ground test well, require less analysis and design related to integration, and are capable of transporting considerably higher power. Single-phase systems have found application with the International Space Station, Mars Pathfinder, and Mars Exploration Rovers[2,3]. However, these systems are actively pumped and, as such, require more power than passive devices such as CCHPs. Furthermore, since thermal energy is transported by sensible heating or cooling of the working fluid, these systems require higher mass flow rates, of approximately an order of magnitude or more, which affect pump size and power consumption, and larger system volumes than two-phase systems for equivalent power levels. In addition, single-phase heat transfer is less efficient than condensation and evaporation while requiring a greater temperature potential due to the temperature change of the working fluid. Both of these factors result in larger heat exchangers and radiators for single-phase systems.

Two-phase pumped loops bring the advantages of two-phase heat transfer to actively pumped systems. However, unlike a capillary or terrestrial system, an actively pumped system operating in microgravity has no intrinsic phase separation mechanism. In addition, the behavior of two-phase flow in reduced gravity systems is not well defined and often problematic. For several decades, researchers have worked to overcome these disadvantages. Two-phase systems have been proposed and tested aboard reduced gravity aircraft, but typically involve complex two-phase pumps[4] or mechanically-driven, rotary separator[5], both of which involve complicated seals and appreciable power consumption associated with the high-speed rotation required to provide separation. Complex designs increase fabrication costs, maintenance requirements, and failure potential, as evidenced by a weld failure during in-process testing of a rotary separator for STS-102[6]. To make two-phase systems attractive for low-cost, rapid spacecraft design and fabrication, a less complicated, more reliable two-phase management approach is necessary.

As a solution to this problem, ACT identified the Momentum-driven Vortex Separator (MVS) technology developed at Texas A&M University (TAMU) as a potential accumulator for this system. These devices are capable of providing phase separation and direct contact heat and mass transfer using the intrinsic momentum of the fluid. Furthermore, they are designed to be gravity independent and are orientation insensitive, which is an important feature for ground testing and qualification of a thermal management system. In addition, these devices are passive, contain no moving parts or seals, and follow a simple design methodology in line with ACT's goal of producing a versatile TPTMS capable of reducing design and fabrication time and costs for spacecraft subsystems.

A conceptual schematic of the TPTMS is shown in Figure 1. Flow is driven by a single-phase pump in series with two parallel eductors. Eductors convert the pressure of a motive liquid into velocity using a converging nozzle. A secondary flow, which can be liquid, vapor, or both, is accelerated through kinetic energy transfer with the now high-velocity motive flow. The relatively low-pressure mixture of the motive and secondary flows are then allowed to expand to recover pressure. Depending on eductor design, the mass flow rate of the secondary fluid can be orders of magnitude larger than the motive flow rate. In the TPTMS, the pump is used to provide the motive pressure for two eductors: one located after the condenser and one at the Momentum-driven Vortex Separator (MVS) liquid outlet.

The condenser eductor moves vapor from the MVS, through the condenser, and returns condensed liquid to the MVS through a tangential inlet. A back-pressure regulator controls the vapor flow rate that leaves the MVS. The liquid outlet eductor removes liquid from the MVS at a constant mass flow rate, which is set by the motive pressure, and feeds a portion of this liquid to the evaporator array. The feed flow rate is set by the maximum expected heat load for each leg and remains constant. Fixed orifices or cavitating venturis can be used to maintain this flow rate. The remainder of the liquid flow removed from the MVS, which is the motive flow of the two eductors, is returned to the pump. A regulating valve (not shown) located on this return line sets the outlet pressure of the eductor, which is also the inlet pressure of the venturis. In this manner, the two eductors isolate the single-phase liquid pump from the two-phase portion of the system. This isolation is important during startup as the location of vapor and liquid in a stagnant microgravity two-phase system is difficult to control. The eductors prevent vapor from reaching the single-phase portion of the system and damaging the pump, regardless of the manner in which each phase is distributed in the rest of the system.

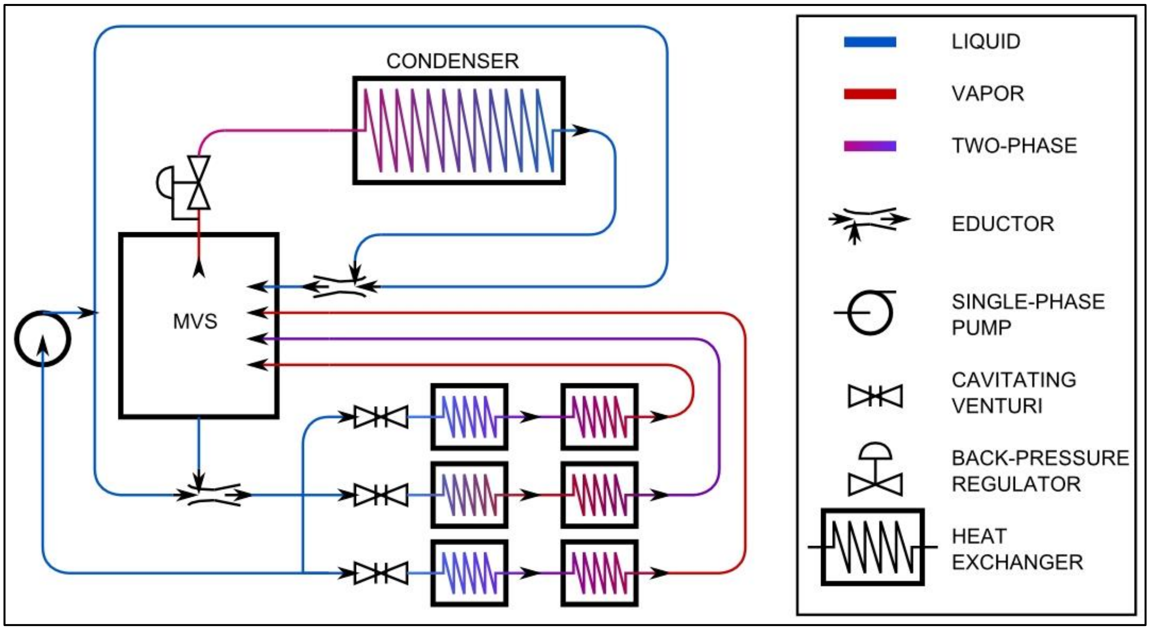

Figure 1. Schematic of Proposed Microgravity Two-Phase Thermal Control System.

As the liquid outlet flow rate of the MVS is set, the back-pressure regulator plays an important role in the TPTMS: maintaining the saturation state of the system during operational transients. This saturation state depends on the system volume, the working fluid mass, the working fluid temperature, and the pressure within the system. The first two parameters are controlled by design. During operation, the temperature is controlled by heat rejection at the condenser. While the regulator's primary function is controlling pressure, this component also controls heat rejection at the condenser by varying the vapor flow rate, and therefore thermal energy, rejected by the system. As the TPTMS is a constant volume system, an increase in heat load will result in an increase in pressure, which will open the regulator. The resulting increase in vapor flow to the condenser will increase heat rejection to match the increased heat load. As heat loads decrease, the back pressure regulator will close and reduce TPTMS heat rejection. During these transients, the resident vapor and liquid volume within the MVS will change until the saturation state is restored. The MVS, however, is designed to accumulate either phase and will provide the necessary buffer volume during these transients. In this way, the regulator and system design maintain the desired saturation state.

This configuration moves two-phase management from the pumping mechanism to the accumulator. As the MVS promotes thermodynamic mixing and is capable of handling significant volume swings resulting from mass flow and inlet quality transients, this component acts as the system accumulator. Additionally, by controlling MVS pressure, which can be accomplished passively with a back-pressure regulator, the saturation state of the system can be controlled and will not fluctuate with heat load or rejection capability transients. The result is a TPTMS using single-phase prime movers but capable of driving two-phase mixtures through multiple heat exchangers while handling variations in heat load and rejection capability.

The heat exchangers that will be used with the TPTMS will be a parallel channel design developed at ACT to promote flow stability during heat transfer. These heat exchangers have been demonstrated to operate with no flow instabilities while in a parallel configuration with varying heat loads.

The MVS design used in this system was developed at TAMU and has provided separation for many microgravity flight experiments including, most recently, the Multi-phase Flow Experiment for Suborbital Testing (MFEST) in 2013[7]. In addition, ACT recently applied the MVS to a terrestrial two-phase thermal management system that was successfully tested in a laboratory environment. The work proposed here intends to expand on these successes by applying the MVS to a microgravity thermal management system, which has not yet been attempted.

The operating principle for a MVS is illustrated from a top-down perspective in Figure 2(a). For comparison, a TAMU MVS operating in microgravity is shown in Figure 2(b). Momentum is provided by a driving flow injected tangentially along the wall of the cylindrical separation chamber. Depending on rotational speed and pressure drop requirements, this driving flow may also be the multi-phase flow requiring separation. This is more often the case than not. The driving flow is centripetally accelerated by the cylindrical wall and produces a forced vortex within the separation chamber. The liquid within the separation chamber experiences centrifugal force as a result of this acceleration and develops the pressure gradient necessary for buoyancy driven separation.

The TAMU MVS design is intentionally simple and consists primarily of a cylindrical separation chamber, a tangential inlet nozzle, optional two-phase inlets, a baffle plate, and liquid and gas outlets. These elements are shown in Figure 3. The MVS shown in this figure has one single phase inlet nozzle and two two-phase inlet ports. A similar, multi-inlet design is under consideration for the TPTMS system for management of multiple, parallel heat exchanger transport streams. The principal investigator was responsible for the design of this MVS, which successfully separated a two-phase mixture with maximum flow rates of approximately 200 SLPM and 6 LPM of vapor and liquid, respectively. In addition, this MVS was used with an eductor and single-phase pump arrangement similar to that proposed here. Using this arrangement, the MVS successfully managed a single component, two-phase working during reduced gravity flight testing with no pump cavitation[8]. Due to the success in these programs, ACT is confident the MVS will provide the performance required for the TPTMS.

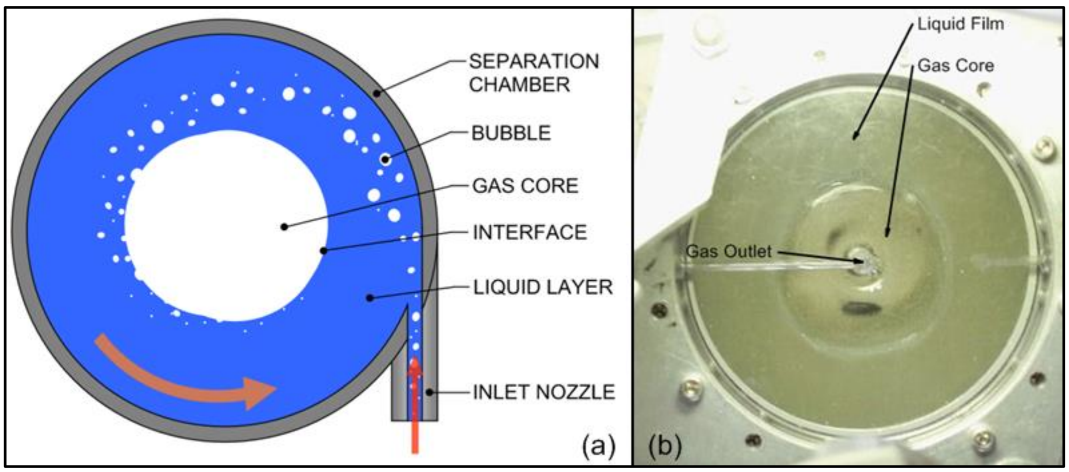

Figure 2. (a) Top-down Illustration of MVS and (b) MVS Operating in Microgravity.

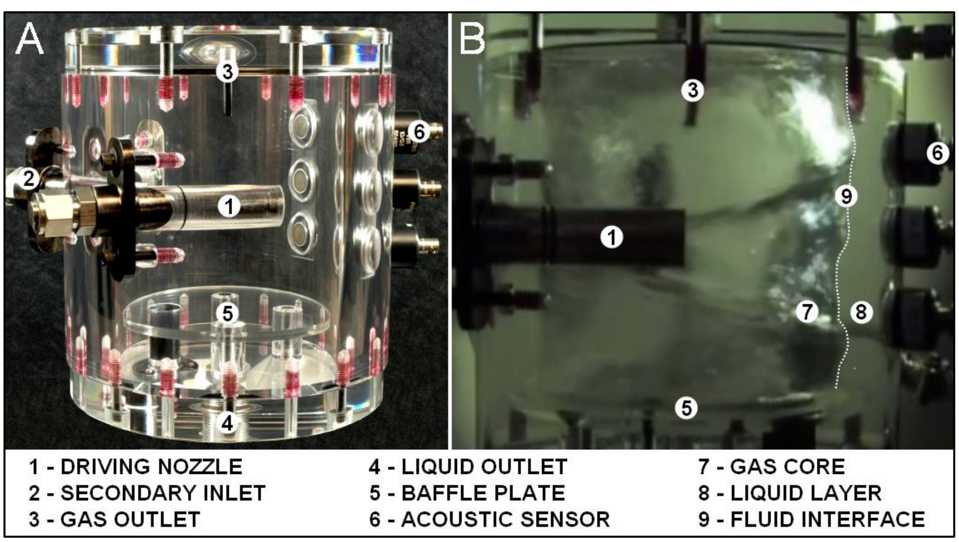

Figure 3. TAMU MVS (a) without Fluid and (b) Operating in Microgavity.

II. Objective

Using the MVS technology, conventional eductor designs, and a two-phase heat exchanger, ACT intends to develop a TPTMS that provides the following benefits for spacecraft thermal control:

- Low-cost, flexible design for rapid integration with spacecraft. This is accomplished through the use of conventional, proven components, integration of a passive two-phase accumulator, removal of restrictions on transport line routing and heat exchanger location, and provision for system ground testing and qualification.
- Constant temperature heating or cooling through multiple, variable load heat exchangers.
- Long distance transport line routing and heat exchanger arrangement in parallel and in series. System pressure drop is overcome using a single-phase pump isolated from the two-phase portion of the system by the two-phase accumulator and eductor arrangement. The two-phase accumulator manages the outlet streams of parallel heat exchanger segments, including the primary spacecraft heat rejection line, to maintain flow stability within the system.
- Use of a single-phase liquid pump and eductor system to ensure Net Positive Suction Head (NPSH) for this pump. Since the fluid in the majority of the system is saturated, the liquid outlet eductor is necessary to provide NPSH.
- The condenser outlet eductor drives fluid returning from the condenser to the MVS. Note that, due to the back-pressure regulator located between the MVS vapor outlet and eductor suction inlet, the eductor will draw a vacuum in situations when no heat rejection is required. This allows the heat rejection portion of the TPTMS to adjust according to system heat loads or rejection capability.
- Capability of recovery in the case of separation failure. The eductors can pump two-phase flow should separation failure occur. This is not an intended operating condition but this feature does exist for failure recovery.

The success of this research will provide a two-phase thermal management option for spacecraft that is actively pumped using proven single-phase components, handles significant heat loads typical of active systems, and provides isothermal, high heat flux performance characteristic of two-phase heat transfer.

III. Component Design

Component design began with working fluid selection. In this study, water, ammonia, methanol, R113, R134a, R245fa, and R410 were considered. The details of this trade study are not discussed here but centered primarily around the effect of fluid properties on separation and heat transfer performance, material compatibility, and laboratory safety. These properties were evaluated at room temperature as the experiment would be conducted at this temperature to reduce ambient gains or losses.

Properties that affect separation performance in the MVS include phase densities, surface tension, and liquid viscosity. Since the MVS is a momentum-driven device, a higher liquid density allows for a stable vortex with less volumetric flow rate. The refrigerants, particularly R113, have the highest liquid densities and are preferred for this reason. In addition, the difference between the liquid and vapor densities affects the buoyancy force generated in the liquid layer. Except for R410, this consideration also favors the refrigerants. Surface tension affects the speed at which bubbles coalesce with the inner vapor core. As greater surface tension results is more rapid coalescence, a high surface tension fluid like water is advantageous. Liquid viscosity also impacts coalescence speed. Lower viscosity fluids like the selected refrigerants coalesce faster than their higher viscosity counterparts. In addition, highly viscous fluids retard momentum transfer within the separation volume and produce greater frictional losses throughout the system. The least viscous fluids of the selected group are ammonia, R410, and R134a.

From a heat transfer standpoint, important properties include phase densities and latent heat capacity. Both impact the mass flow rate required to transport thermal energy. Mass flow rate, in turn, affects system size, mass, and power: three parameters of paramount importance for any spacecraft. The product of the latent heat of vaporization and liquid density provide a convenient quantiy to evaluate the most favorable fluids for a low mass flow system. This quantity is referred to here as the volumetric latent heat capacity. Water has by far the greatest volumetric latent heat capacity followed by ammonia and methanol. R113, R134a, and R245fa have a capacity that is an order of magnitude less than water. R410 has the least capacity, which is nearly three orders of magnitude less than water.

In addition to heat transport, flow stability is another concern for two-phase heat transfer. These instabilities arise from the rapid and large density change that occurs during phase change and result in unpredictable system behavior. These instabilities typically occur for certain flow conditions and geometries. In an attempt to predict two-phase flow instabilities, researchers have developed a Phase Change Number (PCN) that depends on channel length and cross-sectional area, phase densities, latent heat, inlet velocity, and heat flux[9]. Assuming equivalent geometries and a mass flow rate equal to the ratio of the thermal load and latent heat, the PCN reduces to the liquid to vapor density difference normalized by the vapor density. This simplified PCN was used to evaluate the fluids selected for this demonstration. Higher PCNs are associated with instabilities and, while there are other methods to discourage instabilities, selecting a fluid with a relatively low PCN is advantageous. For the fluids evaluated here, the refrigerants and ammonia have the lowest PCN, with R134a, R410, and ammonia being particularly low. Methanol has PCN that is an order of magnitude higher than these fluids and the PCN for water is another order of magnitude higher.

Additional considerations involve safety and system fabrication. Methanol is flammable and Ammonia is toxic, which may be acceptable for unmanned spacecraft but would require additional caution during assembly and testing. The saturation pressure of the work fluid at the maximum operating temperature is also of concern. High pressure systems can present safety concerns and require thicker walled vessels and tubing, which increases system mass and cost.

At the demonstration unit design temperature of 25 °C, most of the working fluids considered here are very close in performance with the exception of R410, primarily due to the low volumetric latent heat capacity. In addition, at the design temperature of 25 °C, R410 has the highest saturation pressure at 1.5 MPa (220 psi). For these reasons, R410a was not selected. Ammonia and methanol, while good performers, present safety concerns and were not selected for this reason. Water is easily the best performer but was avoided due to this fluid’s high potential for flow instabilities. The remaining refrigerants, R113, R134a, and R245fa are all good candidates for the demonstration unit. R113 has a higher potential for instabilities than the other two refrigerants and R245fa has compatibility issues with polycarbonate, which is used for flow visualization as described later. R134a was selected due to these issues.

A proprietary separator model developed by ACT was used to estimate the performance of the separator to provide an appropriate design for the TPTMS demonstration system. The system model indicated the MVS would receive 4.5 LPM of flow without the driving nozzle. This flow rate was used to evaluate the rotational speed of the liquid layer and the acceleration developed at the vapor-liquid interface. Assuming a minimum vapor core radius of 2.5 cm (1 in.), the model predicted a rotational speed of 342 rpm and an interface acceleration of 32.6 m/s^2 (3 g’s). This should produce a parabolic profile within the separator in the laboratory.

To achieve a cylindrical interface in the laboratory, which typically occurs when the interface acceleration is approximately 100 m/s^2 (10 g’s), an additional flow rate of 4 LPM will be necessary through the driving nozzle. As the pump can move over 20 LPM of R134a, this should be achievable. Note that, in a reduced gravity environment, this additional driving flow will not be necessary.

The MVS design is shown in Figure 4. The MVS consists of the cylindrical pieces and two end plates. The clear cylindrical pieces are polycarbonate tube. The solid ring and the nozzles contained within are aluminum. This ring has one driving nozzle and three two-phase inlets. The flow from the two evaporator legs and condenser will enter through the two-phase inlets. The driving nozzle will be used if necessary based on test results. The end caps are also aluminum and all O-ring seals are Buna-N.

A series of identical aluminum cold plates were designed to function as the evaporators of the TPTMS. Each cold plate had one 25.4 cm (10 in.) by 25.4 cm (10 in.) face where heat sources could be mounted. The cold plates were designed to account for the pressure drop of the two-phase flow under worst case conditions, that is, when the total heat required to be dissipated by the system is maximum, therefore requiring the maximum flow rate through the plates. A homogeneous two-phase flow model was used to predict the pressure drop of the two phase flow through the cold plates based on the heat input to them. At the inlet of each plate, the flow splits into two serpentine channels that are symmetric about the center plane of the cold plate. At the outlet, the two streams rejoin. A flow constriction to each channel, in the form of a channel section of reduced size, 0.13 cm (0.05 in.), was implemented at the inlet of each cold plate to prevent reverse flow upon expansion due to vaporization of the working fluid. Downstream of the inlet channel, the main channel of the cold plates has a hydraulic diameter of 0.4 cm (0.158 in.).

Heat was applied using four 15.2 cm (6 in.) long and 3.8 cm (1.5 in.) resistive heaters arranged in parallel and spaced evenly between the bolt pattern seen in Figure 6. The heaters are mounted flush against the aluminum surface opposite the polycarbonate face seen in this figure. Thermal gap pads were used to provide thermal contact between the heaters and this surface. Each heater provided up to 325 W of heat.

The smallest commercial-off-the-shelf (COTS) liquid eductor was chosen for use in the prototype system and ordered from Jacobs Engineering. This COTS eductor consists of a 0.64 cm (0.25 in.) tee that has been bored out in the flow path direction and threaded to accept a 0.64 cm (0.25 in)-28 set screw of 0.95 cm (0.375 in.) in length. The set screw on the inlet side has a 0.2 cm (0.08 in.) hole through the center axis to function as the nozzle, and the set screw on the outlet side has a 0.33 cm (0.13 in.) hole through the center axis to function as the expansion and mixing column. A thread sealant was used to fix the depth that each set screw was positioned into the tee, and the distance between the two set screws was 0.086 cm (0.034 in.).

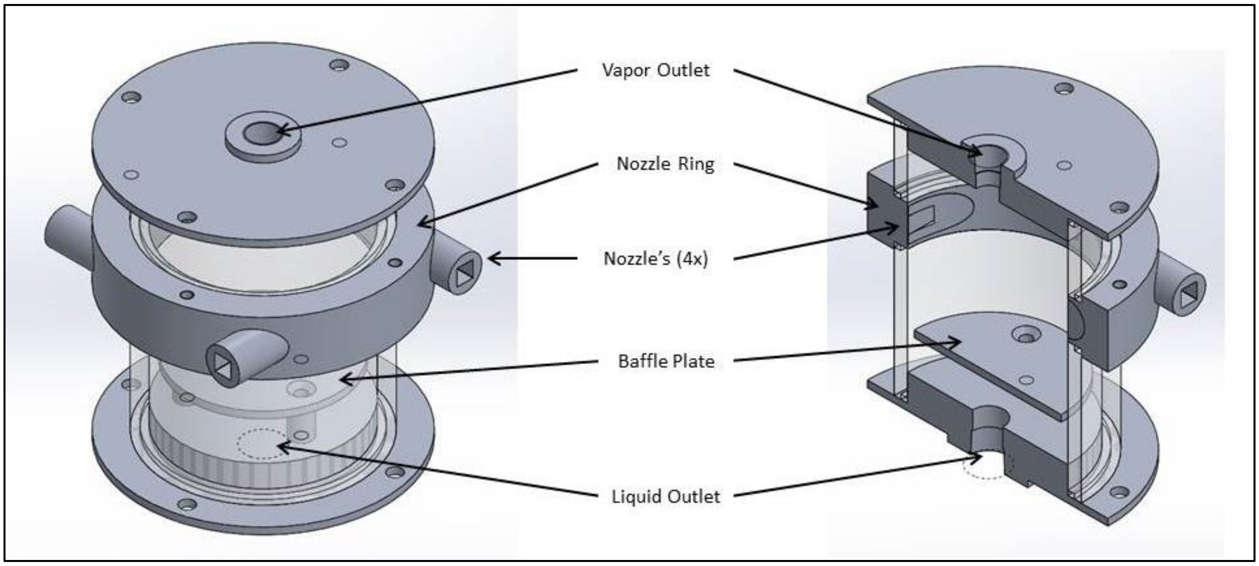

Figure 4. MVS Design.

IV. Experimental System

The MVS and cold plates were fabricated at ACT and installed into the prototype TPTMS. The MVS is shown installed in the test system in Figure 5. The cold plates were machined in two halves and joined using an o-ring and bolt assembly. Since the top half was polycarbonate and the bottom, channeled half was aluminum, welding or brazing were not options. Typically, ACT vacuum brazes our production cold plates and we anticipate following the same approach for the final system. Three cold plates were installed in the test system as shown in Figure 6. Two of the cold plates are in series with the third cold plate in parallel with these two. The fully assembled TPTMS is shown in Figure 7. Note that the pump is oversized for the laboratory test. In addition, the demonstration system has a large footprint for ease of operation, instrumentation placement, and visualization. As such, this footprint is not representative of the final TPTMS footprint.

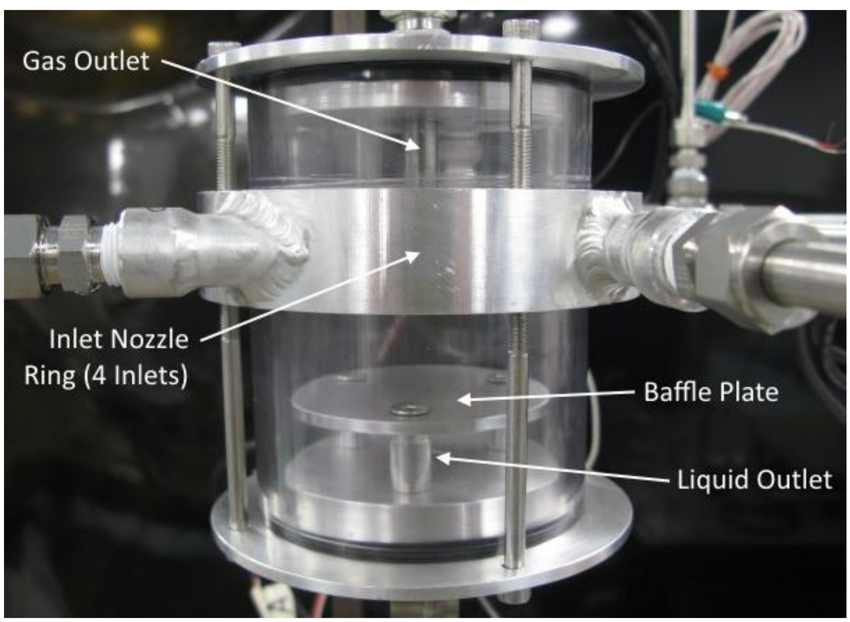

Figure 5. Close Up of Installed MVS Showing Various Parts.

Figure 6. Cold Plate Array Installed in the TPTMS Demonstration Unit.

V. Test Results

Initial testing concentrated on setting the control valves to allow approximately 1 LPM of flow through the two evaporators in series and 0.5 LPM of flow through the single evaporator in parallel with the evaporators in series. The backpressure regulator was adjusted but it was found that the pressure drop through the gas outlet line was sufficient to maintain the MVS at a saturation temperature close to the laboratory temperature. The system was run close to laboratory temperature to avoid the effects of ambient gains or losses.

Once the setup testing was complete, ACT evaluated a single evaporator at various heat loads. Testing began at 130 W and increased by 130 W until the evaporator began to dry out at the exit. This occurred at 1170 W. The cold plate was designed to carry 1300 W and fell short of this target by 130 W.

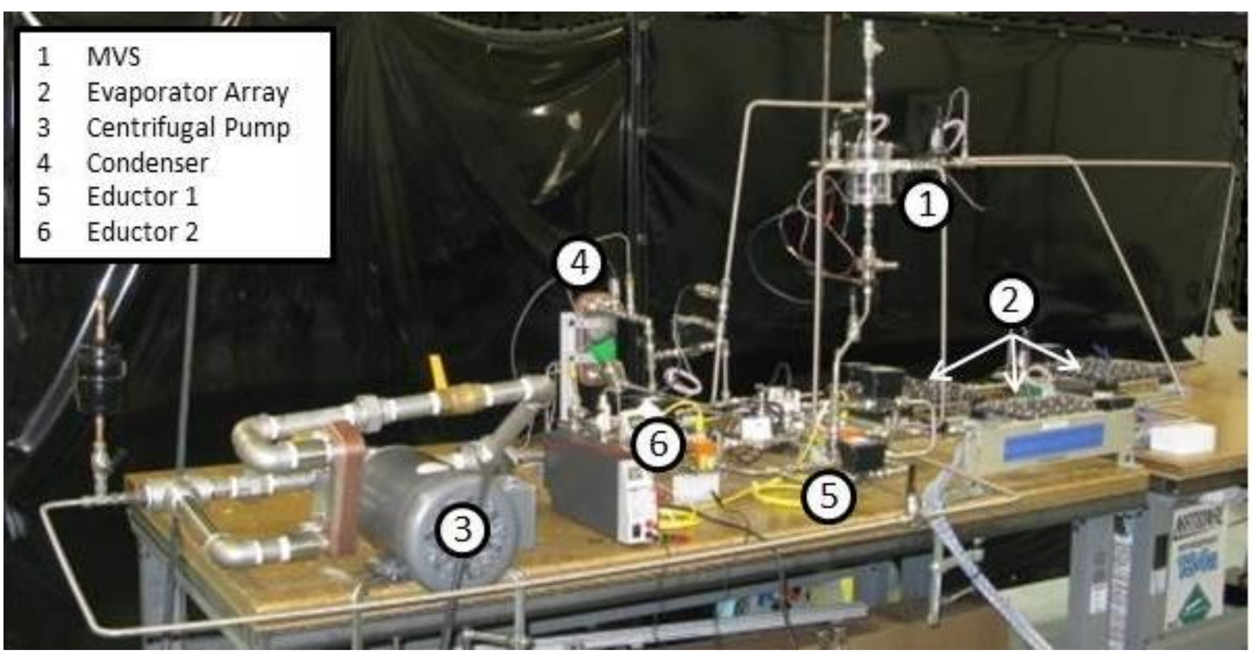

Figure 7. Assembled TPTMS Demonstration Unit.

Boiling in the cold plates was observed visually through the polycarbonate face plate. At the lowest power, 130 W, boiling was seen to occur only in the last leg of the serpentine channel. This is expected as the fluid entering the cold plate is slightly subcooled by the eductor. This subcooled liquid heats up and begins to change phase before exiting the cold plate.

At higher powers, such as the 650W heat load shown in Figure 8, boiling begins near the entrance of the cold plate and quality continues to increase as the fluid moves towards the exit. Starting as bubbly and slug type flow near the entrance, the flow stratifies near the middle of the cold plate. This flow regime continues until the exit is reached.

Figure 8. Boiling in the First Leg of the Cold Plate at 650 W.

At the maximum power reached, 1170 W, only vapor was seen to exist in the last leg of the cold plate. At this point, the cold plate was showing signs of drying out and the experiment was stopped. This heat load was 130 W short of the design target of 1300 W.

Vapor exiting the cold plates was directed to the MVS. This vapor and the liquid returning from the condenser are injected tangentially into the separation chamber of this device. The momentum of these fluids produces a vortical flow within the device which results in a centrifugal acceleration field from the fluid's frame of reference. As a result, vapor moves towards the center of the device and is extracted by the vapor outlet eductor. A baffle plate keeps the vapor core from reaching the liquid outlet where liquid is withdrawn by the liquid outlet eductor.

Figure 9. MVS Operating with a Parabolic Interface at 910 W.

In the laboratory environment, the centrifugal acceleration produced by the inlet fluid momentum was not sufficient to completely overcome gravity. As a result, the liquid-vapor interface is not cylindrical as occurs in reduced gravity. Instead, this interface forms a parabolic shape as seen in Figure 9.

Fluid temperature, cold plate surface temperature, pressure drop, and flow rate were recorded using a data acquisition system during testing. The surface temperature of the underside of the cold plate was recorded using a thermocouple array arranged evenly around the heating elements. Figure 10 shows the maximum temperature gradient averaged over each tested heat load. Until 910 W, the temperature gradient remains under 2K. Above this heat load, the temperature quickly rises as the cold plate dries out and begins to act more like a single phase, vapor-cooled heat exchanger.

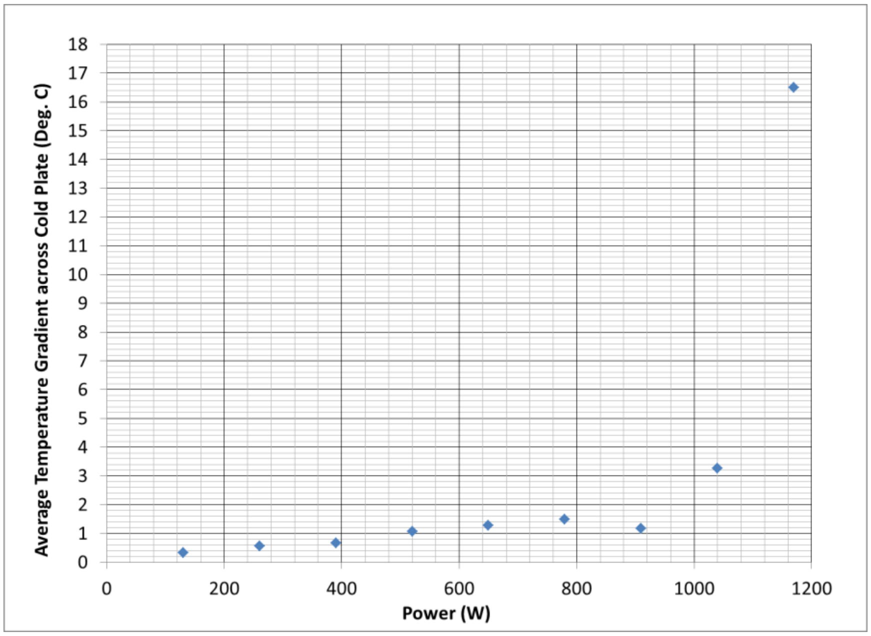

Figure 10. Maximum Temperature Gradient across the Cold Plate Averaged over the Test Period.

In addition the thermocouple array, temperature data was recorded using an infrared camera. This approach allowed us to record temperature through the polycarbonate face plate. As seen in Figure 11, the surface temperature was fairly uniform and remained close to the channel temperature, which can be seen as the serpentine pattern. The coldest portion of the plate was near the entrance, which is at the top of the plate. The hottest portion of the plate is the middle. The heaters were each 15.2 cm (6 in.) long and arranged between the bolt patterns which appear as black circles in the figure. As the plate was 25.4 cm (10 in.) in length, this arrangement concentrated the heat load along the middle of the plate, which resulted in the higher temperatures there. Using thermal imaging software, the temperature gradient was found to be less than 2 K. This value agrees with the data obtained from the thermocouple array. Note that the scale is somewhat skewed by the electrical wiring protruding from underneath the cold plate. This wiring is at 30.6 °C.

Next, the transient temperature profile of a cold plate was evaluated for different heat loads. Figure 12 shows the temperature profile for 260 W. Prior to this portion of the test, the cold plate was operating for 18 minutes at 130 W. At 1073 seconds, the heat load is increased to 260 W. After this increase, the average temperature of the cold plate warms up slightly from 23.25 °C before reaching a steady temperature of 23.7 °C. Figure 13 shows a similar trend for 650 W, which warms up from 25 °C to 25.8 °C. And, finally, Figure 14 shows the same profile at 910 W, although this time warming up by 2 K, from 27 °C to 29 °C. In each case, two distinct regions are evident in the temperature profile. The first is the temperature increase region and the second is the steady state region. Even when steady state is reached, the temperature oscillates slightly. In addition to instrumentation error, this is likely due to the somewhat unsteady nature of flow boiling.

Next, the flow rate through the cold plate as a function of heat load was evaluated and is shown in Figure 15. As seen in this figure, the flow rate decreases as heat load increases. Since the driving pressure provided by the pump and eductor remains constant, this effect is a result of the increasing pressure drop through the cold plate in response to the generation of more vapor. This increase in pressure drop is shown in Figure 16.

When phase change occurs, the much less dense vapor phase must compete with the liquid phase. As a result, vapor velocity must increase significantly to leave the cold plate at a rate equal to the evaporation rate. If this does not occur, the saturation pressure and temperature of the working fluid will increase. This, in effect, compresses the vapor until a balance between evaporation rate and vapor flow rate is achieved.

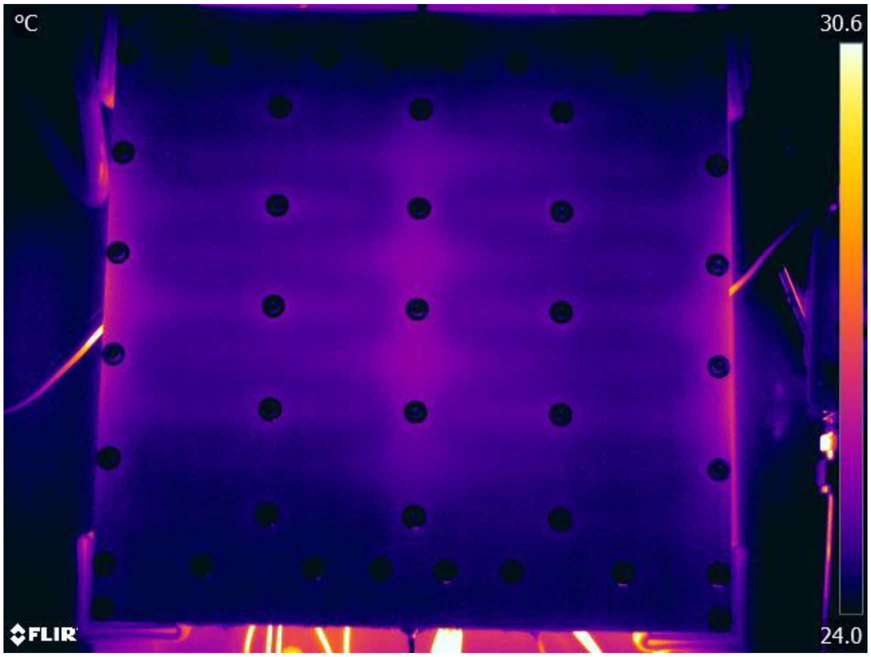

Figure 11. Thermal Image of the Cold Plate at 650 W Showing a Maximum Temperature Gradient of Less than 2 K on the Plate Surface.

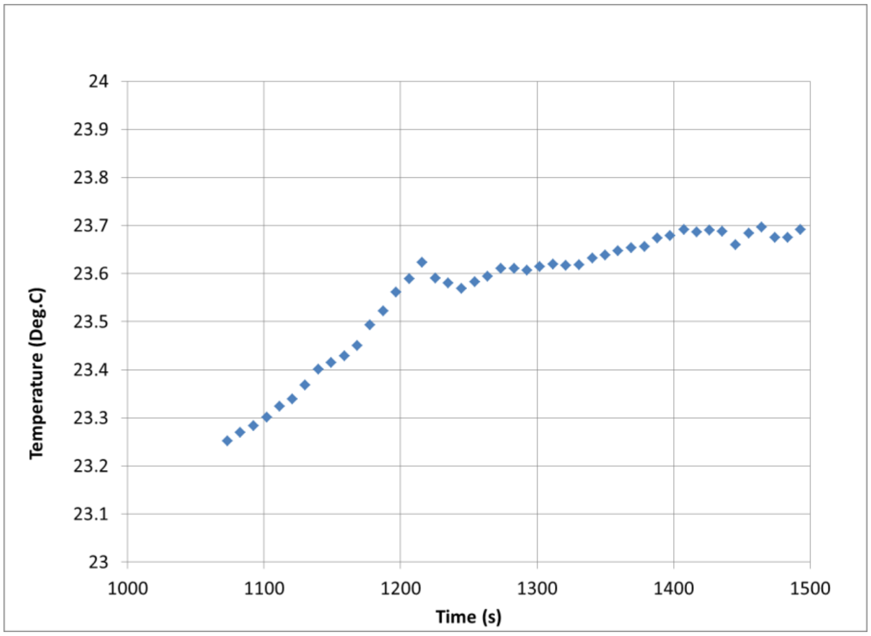

Figure 12. Average Cold Plate Temperature during Test Period for 260 W Total Heat Load.

Once the cold plate begins to dry out, the pressure drop no longer increases significantly. This is a result of the vapor no longer competing with the liquid phase. Vapor density will continue to decrease as the vapor carries away more heat but not as significantly as that which occurs during phase change. This effect is seen above 1 kW, at which point the cold plate dries out as evidenced by Figure 17.

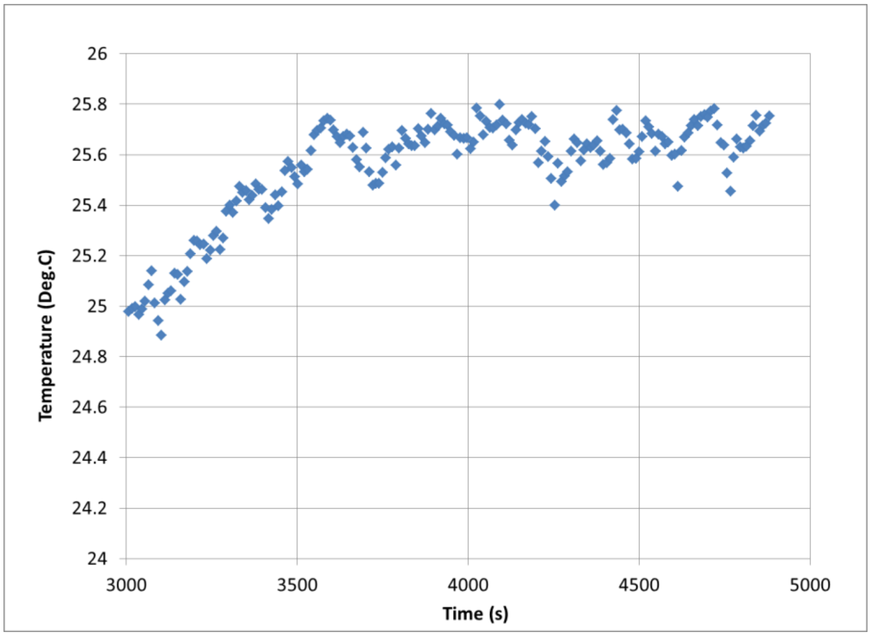

Figure 13. Average Cold Plate Temperature during Test Period for 650 W Total Heat Load.

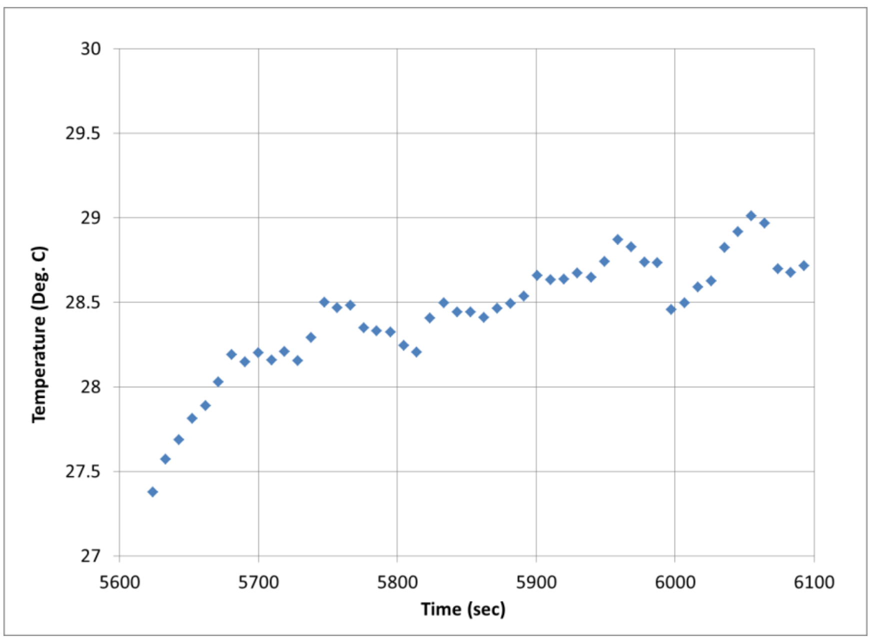

Figure 14. Average Cold Plate Temperature during Test Period for 910 W Total Heat Load.

Next, the effect of the cold plates in series was investigated. Figure 18 shows the pressure drop across these cold plates while operating with a 650 W heat load on both plates. The first cold plate experiences a pressure drop similar to that of the single cold plate. However, the second cold plate experiences a much larger pressure drop. This is a result of the first cold plate receiving all liquid at the entrance and the second cold plate receiving a mixture of liquid and vapor. As the latter results in higher pressure drop, the second cold plate experiences this effect.

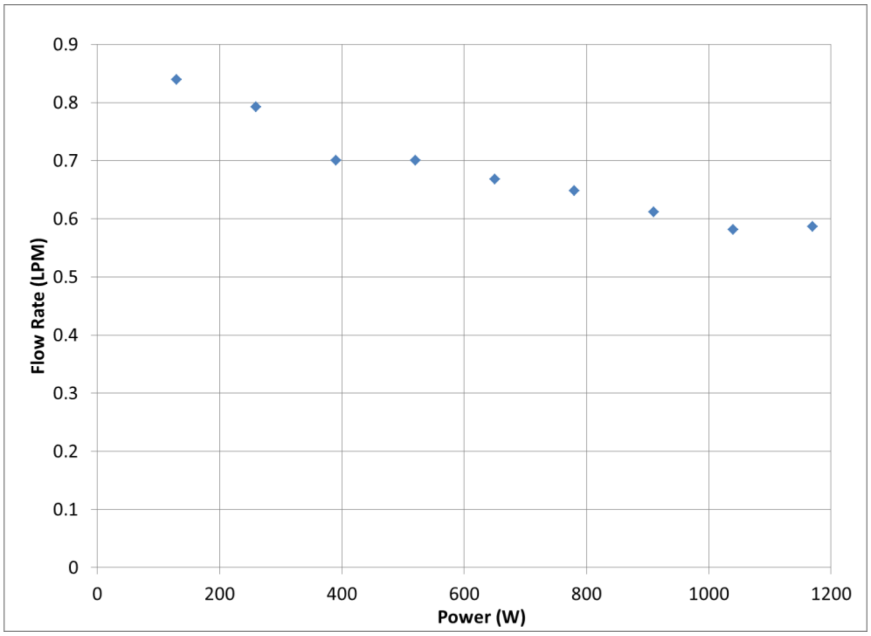

Figure 15. Flow Rate through the Cold Plate Averaged over the Test Period.

Finally, eductor performance is evaluated. Figure 19 and Figure 20 show the volumetric flow rates of the liquid outlet and vapor outlet eductors, respectively. The liquid outlet eductor is pumping slightly less than 4 LPM from the suction port to the outlet. This matches the driving flow rate, which indicates this eductor is performing with an entrainment ratio of 1. The vapor outlet eductor is moving approximately 0.6 LPM of liquid through the suction port. Compared to the 1 LPM driving flow rate, this eductor is operating with an entrainment ratio of 0.6.

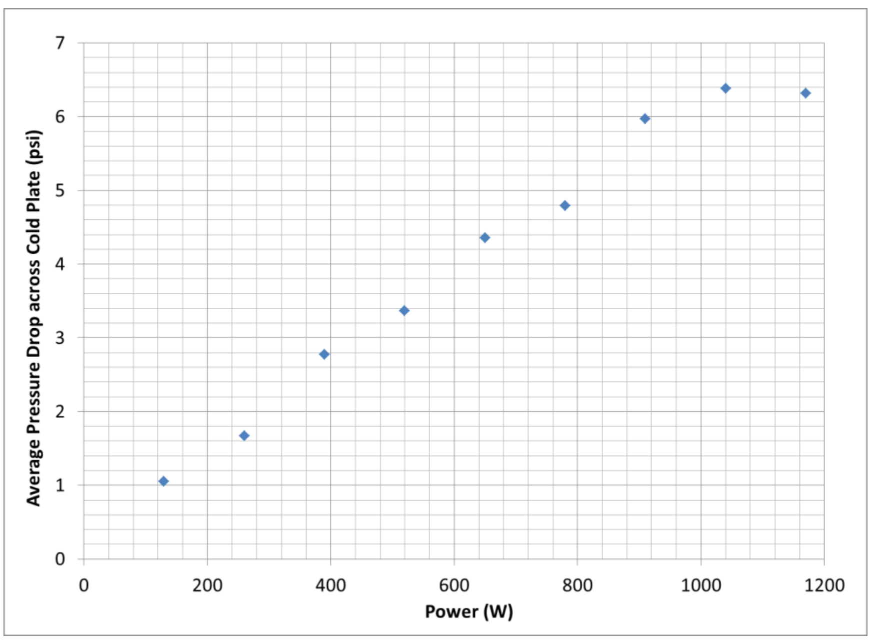

Figure 16. Pressure Drop across the Cold Plate Averaged over the Test Period

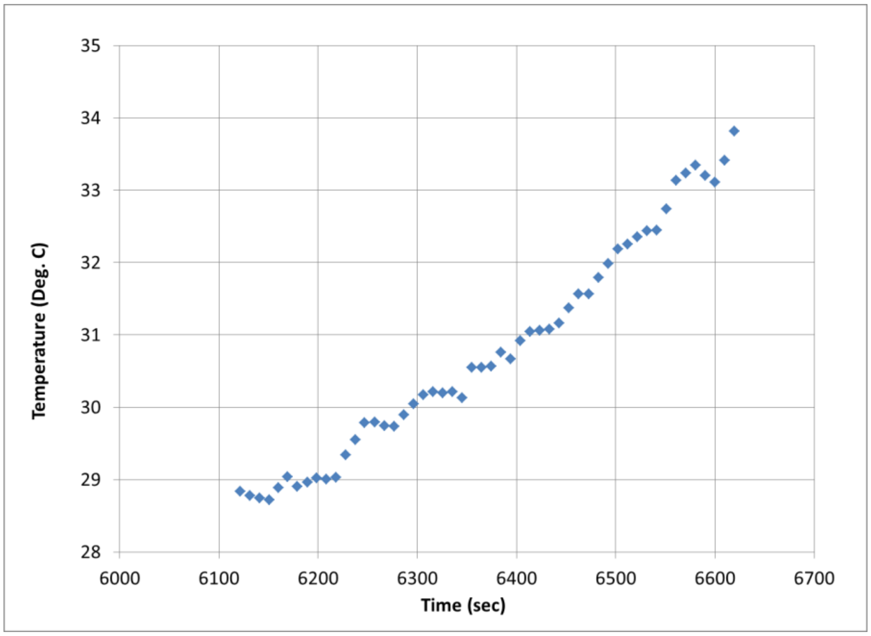

Figure 17. Average Cold Plate Temperature over the Test Period for the 1170 W Heat Load.

At 650 W, the 3 cold plates should be producing about 6.5 LPM of vapor for a total of 19.5 LPM to enter the separator. After condensing, this would result in 0.57 LPM of liquid leaving the condenser, which is then drawn through the vapor outlet eductor. This closely matches the flow rate processed by this eductor. Liquid not evaporated in the cold plates also returns to the separator. Each cold plate is fed 0.65 LPM of liquid, leaving 0.46 LPM of liquid to return to the separator per cold plate for a total of 1.38 LPM. Added to the 1.6 LPM returning from the vapor outlet eductor, the liquid eductor should be removing nearly 3 LPM of liquid rather than the 4 LPM indicated by the flow meter. Since the separator was not showing any liquid accumulation during testing, this inconsistency may be a result of error agglomeration between flow meters since 5 flow meters were used in the calculation. ACT will evaluate this in future testing.

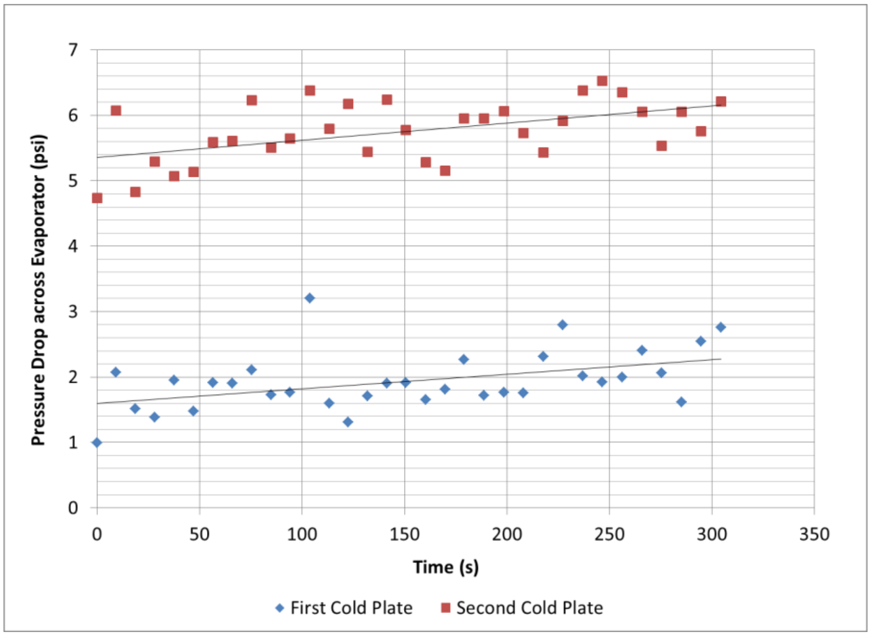

Figure 18. Pressure Drop through Two Cold Plates Arranged in Series with Both Cold Plates Managing a 650 W Heat Load.

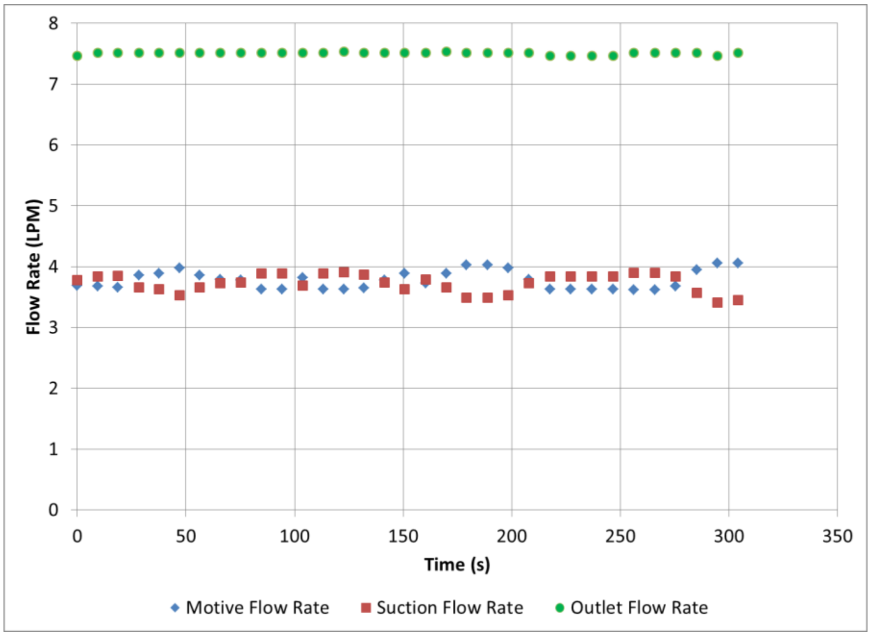

Figure 19. Flow Rates through the Liquid Outlet Eductor at a Heat Load of 650 W on Each Cold Plate.

VI. Conclusion

The TPTMS prototype was demonstrated to have the capability to handle nearly 3 kW of heat using three cold plates that maintained a surface temperature gradient less than 2 K. In addition, the liquid-vapor mixture returning from these cold plates was successfully distributed to the appropriate component: vapor was pumped to the condenser through the vapor outlet eductor and liquid was fed to the cold plate array through the liquid outlet eductor. To perform this function, the MVS accepted single- and two-phase flow from the cold plates and condensers, separated the phases, and directed each phase to the respective outlet.

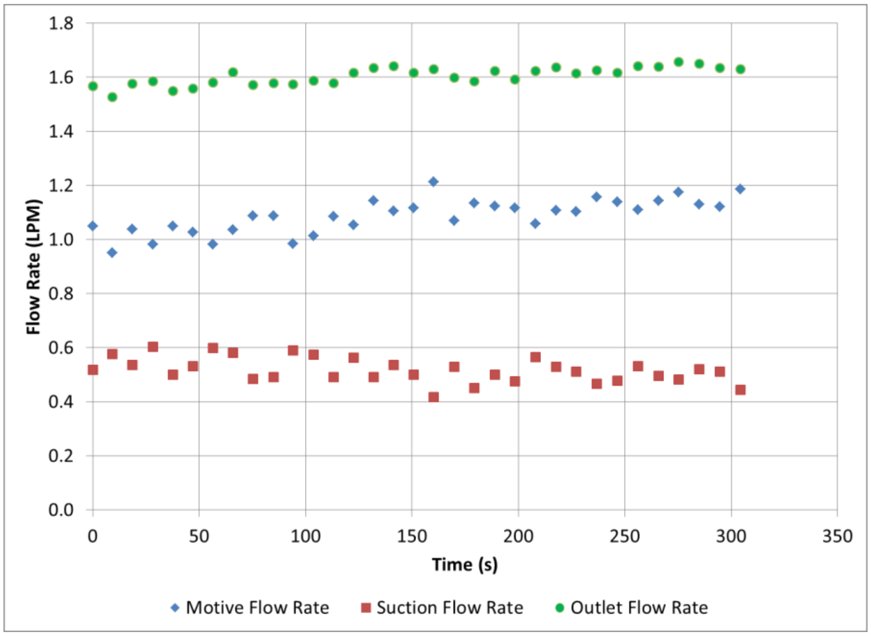

Figure 20. Flow Rates through the Vapor Outlet Eductor at a Heat Load of 650 W on All Cold Plate.

The purpose of this testing was to demonstrate several developed components, the MVS, two-phase cold plates, liquid eductors, and a liquid pump, working together to provide stable, two-phase thermal management in a gravity independent system. The separation of the two-phase portion of the system, namely the heat exchangers, from the single-phase pump was shown to be possible by employing a microgravity phase separator with liquid driven eductors driving flow through the vapor and liquid outlets. As the suction side of the eductors are two-phase tolerant, this system is capable of recovering from separation failure as well without endangering the pump. This feature adds an inherent stability not usually encountered with active two-phase thermal management systems.

The orifice channel design used with the cold plates was observed to prevent flow instabilities during boiling and maintain fluid motion in the correct direction. Even with this restriction, the cold plates remained a fairly low pressure drop component at just a few psi depending on heat load. This pressure drop, and the low surface temperature gradient, would benefit from an optimization study and we believe both can be reduced significantly.

Two-phase thermal management systems offer significant advantages over single-phase systems. They allow for tight temperature control due to the isothermal nature of the working fluid. The require significantly less working fluid flow rate due to the orders of magnitude larger heat capacity of latent heat transfer compared to sensible heat transfer. This feature can greatly reduce component and overall system size. Add to that an order of magnitude increase in heat transfer rate, two-phase flow allows for compact, high heat flux designs that single-phase systems cannot achieve.

The major drawback for two-phase systems, especially in reduced gravity environments, is their perceived complexity. And, when compared to single phase systems, this perception is understandable. The phase change phenomena that provide heat transport are less well understood and more difficult to predict than sensible heating or cooling. As a result, two-phase systems are seen to add more risk to a system than their well-known advantages are worth.

Reducing this risk was the primary thrust behind the TPTMS developed as part of this project. Standard components were combined to provide a straightforward system that could recover from a two-phase anomaly without damage to system components. While demonstration of this system concept was successful, there is considerable work to be done to fully optimize and properly model this system.

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References

[1]Swanson, T.D., "Thermal Control Technologies for Complex Spacecraft," *13th International Heat Pipe Conference*, Shanghai, China, 2004, pp. 1-11.

[2]Paris, A.D., Bhandari, P., Birur, G.C., "High Temperature Mechanically Pumped Fluid Loop for Space Applications – Working Fluid Selection," *SAE Transactions*, Vol. 113 (1), 2004, pp. 892-898.

[3]Birur, G.C. and Bhandari, P., "Mars Pathfinder Mechanically Pumped Cooling Loop," *NTRS*, Paper No. 20060032169, 2001.

[4]Downing, S., Andres, M., Nguyen, D., Halsey, D., and Bauch, T., "Two-Phase Thermal Management Systems for Space", *Space Technology and Applications International Forum (STAIF)*, AIP Conference Proceedings, Vol. 813, pp. 27-34 (2006).

[5]Lanzarone, A., "Space Station Water Processor Mostly Liquid Separator (MLS) Final Report", *NASA Contractor Report*, NASA-CR-200777, 1995.

[6]Guirgis, G.G., STS-102/5A.1 Flight Readiness Review, EVA Project Office, Presentation, Feb 2001.

[7]Hurlbert, K.M., Marak. R., Kurwitz, R.C., and Ellis, M.C., "Progress on the Multi-Phase Flow Experiment for Suborbital Testing", *Next-Generation Suborbital Researchers Conference*, Presentation Only, 2013.

[8]Supak, K., Kurwitz, C., Oinuma, R. and Best, F., "Reduced Gravity Rankine Cycle System Simulation and Design with Passive Vortex Separation", *Proceedings of the Space Nuclear Conference*, Boston, MA, 2007, Paper No. 2015, pp. 154-162.

[9]Aldridge, C. J. and Fowler, A. C. "Stability and instability in evaporating two-phase flows", *Surveys on Mathematics for Industry*, Vol. 6, pp. 75-107 (1995).