10-14 July 2016, Vienna, Austria

A Robust, Gravity-Insensitive, High-Temperature Condenser for Water Recovery

Weibo Chen [1] and Thomas Conboy[2]
Creare LLC, Hanover, NH, 03755

and

Michael Ewert [3]
NASA Johnson Space Center, Houston, TX, 77058

Regenerative life support systems are vital for NASA's future long-duration human space exploration missions. A Heat Melt Compactor (HMC) system is being developed by NASA to dry and compress trash generated during space missions. The resulting water vapor is recovered and separated from the process gas flow by a gravity-insensitive condenser. Creare is developing a high-temperature condenser for this application. The entire condenser is constructed from metals that have excellent resistance to chemical attack from contaminants and is suitable for high-temperature operation. The metal construction and design configuration also offer greatest flexibility for potential coating and regeneration processes to reduce biofilm growth and thus enhancing the reliability of the condenser. The proposed condenser builds on the gravity-insensitive phase separator technology Creare developed for aircraft and spacecraft applications. This paper will first discuss the design requirements for the condenser in an HMC system that will be demonstrated on the International Space Station (ISS). Then, it will present the overall design of the condenser and the preliminary thermal test results of a subscale condenser. Finally, this paper will discuss the predicted performance of the full-size condenser and the development plan to mature the technology and enhance its long-term reliability for a flight system.

Nomenclature

CFM	= cubic feet per minute
HMC	= heat melt compactor
ISS	= International Space Station
WRS	= water recovery system

I. Introduction

Future NASA long-duration human space exploration missions will require very reliable regenerative life support systems to minimize resupply and thus spacecraft launch mass. To recover resources including water in trash, reduce trash volume and minimize microbial growth, NASA has been developing a trash management technology. This technology uses a Heat Melt Compactor (HMC) that compacts, heats, and dries the waste, and then finally melts the plastic within the waste into a solid tile that can be safely handled by the crew.[1] The HMC partially compresses the trash while heating the trash with electrical heaters attached to chamber walls and lid, as well as the ram plate. These hot surfaces (about 150°C) raise the temperature of the water in the trash and cause water to evaporate (Figure 1). The resulting steam is carried out of the HMC by a small sweep air flow. Next, the steam and sweep air enter the Water Recovery Subsystem (WRS) to reclaim the water and reduce the outlet air relative humidity to a level below 60%, which is required before the air can enter the trace contaminant control system. In the baseline WRS design, water vapor is condensed in an air-cooled condenser and separated from the gas stream

---

[1] Senior Engineer, 16 Great Hollow Road, Hanover, NH, 03755.
[2] Engineer, 16 Great Hollow Road, Hanover, NH, 03755.
[3] Thermal and Life Support Systems Analyst, 2101 NASA Parkway, Houston, TX, 77058.

using a bulky centrifugal separator, which requires dynamic components and significant power inputs. To reduce the outlet air relative humidity level below 60%, a thermoelectric cooler further reduces the air temperature below its target dew point. The internal pressure of the HMC and water recovery system can be anywhere from atmospheric pressure down to a partial vacuum of 3 psia. In addition to recovery of water and volume, other benefits of HMC include reduction of biohazards to the crew and production of radiation shielding tiles. Studies have shown that reusing waste materials for radiation protection in the form of HMC tiles is a large part of the mass and volume savings attributable to HMC.[2]

One of the critical needs for the HMC water recovery subsystem is a lightweight gravity-insensitive condenser with low power consumption. Separating condensate from the gas flow, however, presents a unique challenge in microgravity environments. Potential fouling due to biofilm growth or precipitation of water-insoluble organic compounds, as well as chemical incompatibility with one or more gas components in a mixed gas flow, further increases the challenge for a condenser to achieve a long service life.

A. Phase Separation Is Challenging in Microgravity Environments

Practical means to separate the gas and liquid in space include: (1) using a centrifuge to introduce a centripetal force to separate phases, such as the water separator in space suits; (2) exploiting the difference in inertia between the two phases by passing a two-phase flow through a curved flow passage, such as a vortex separator or an impingement separator; and (3) relying on surface tension to enable one of the phases to preferentially adhere to the surface. The first approach involves mechanical moving parts, reducing the system reliability, and increasing power consumption. The second approach introduces a relatively large pressure drop in a flow and the separation effectiveness will decrease sharply when the flow rate is reduced from its design value. The two-phase flow in a vortex phase separator also tends to be unstable, leading to a large pressure pulsation in the system,[3] possibly due to the large compliant volume in the separator.

Phase separators using surface tension have been successfully used in space applications. The condensing heat exchanger on the ISS is similar to a conventional plate-fin heat exchanger, but with a porous, hydrophilic coating on the condensing surface to ensure that condensed water adheres to the condensing surfaces and flows to a slurper system that then draws the water out of the condenser. However, conventional capillary structures have limited gas-liquid interfacial areas due to their relatively large flow channels. This limits the mass flux across the interface. Furthermore, in addition to vulnerability to particulate contaminations, the capillary structures are also susceptible to fouling due to biofilm growth or sublimation of organic compounds, as discussed below.

B. Conventional Capillary Structures Are Vulnerable to Fouling

The main sources of water in HMC batches are food, beverages, shampoo, disinfecting wipes, toothpaste, and diapers. Water reclaimed by the HMC contains a low concentration of organics, and ions such as Na+, NH_4+, K+, Mg^{2+}, Ca^{2+}, Cl^-, NO^{2-}, Br^-, NO^{3-}, PO_4^{3-}, and SO_4^{2-}.[4] When exposed to these contaminants for a prolonged period of time, the hydrophilic capillary structure in the condenser can gradually reduce its hydrophilicity. Reduction of heat transfer surface wettability by contaminants is a major concern for service life of the condensing heat exchanger on the ISS and is suspected as the cause for at least one failure of a condensing heat exchanger hydrophilic surface. Furthermore, all types of microbes (viruses, bacteria, and fungi) will inevitably grow on surfaces that are exposed to organic contaminants if the surfaces remain wetted for an extended period of time. A wide range of organic compounds such as carboxylic acids, amino acids, proteins, and carbohydrates will promote bacterial growth in the aquatic environment.[5] Biofilm fouling will reduce mass flux through the porous layer, and increase differential pressure and feed pressure.

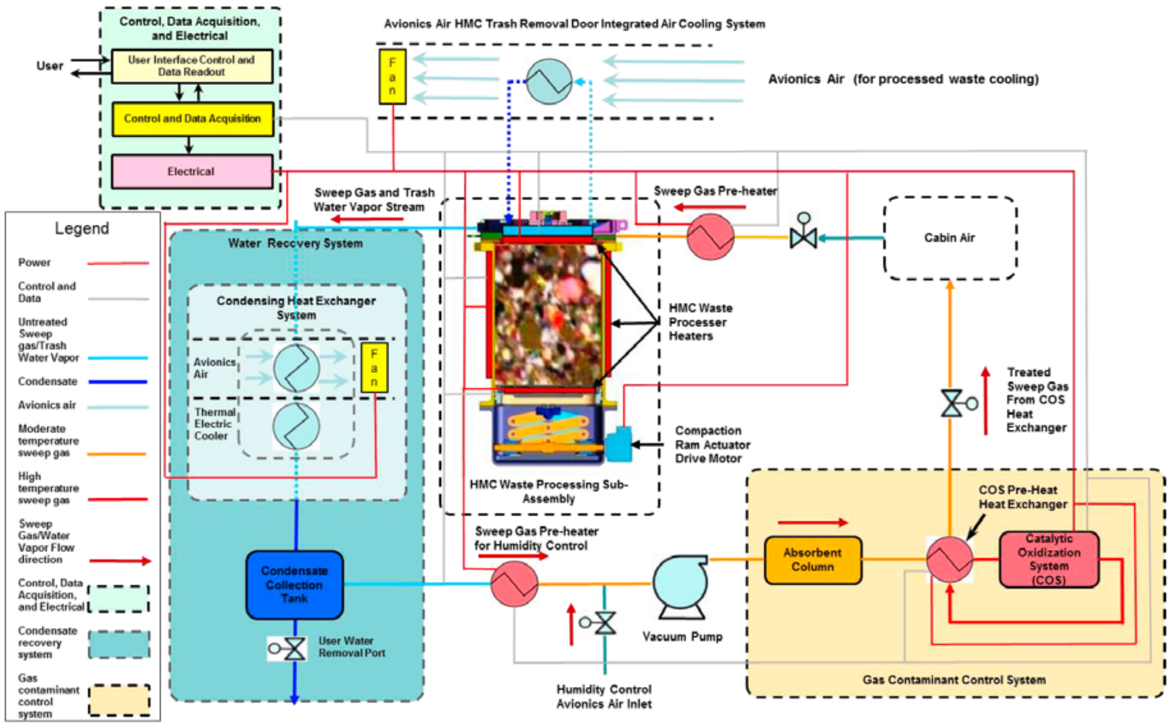

Figure 1. Schematic of Gen-2 Heat Melt Compactor[1]

C. Current Surface Capillary Structures Are Difficult to Regenerate

Current capillary structures, including porous coatings and polymeric membranes, are built into the heat exchangers and they cannot be removed from the heat exchangers. The only practical way to regenerate the heat exchangers is to clean the capillary structures in situ. However, most of the capillary structures are made of materials that have limited service temperature and limited chemical compatibility with the challenging environment. This limits the applicable regeneration approaches, places severe constraints in the cleaning methods, and substantially reduces the effectiveness of the regeneration process. Furthermore, these temperature and chemical compatibility limitations also place severe constraints on possible coating processes to modify the surfaces (such as thin-film metal oxide coating) to reduce biofilm growth, thus enhancing the reliability of the condenser. A capillary structure made of inert metals will be highly desired. Sintered metal plates have been used as phase barriers, such as for sublimation in the space suit. However, they have relatively low permeability and are bulky and heavy, and their surfaces are difficult to modify or clean because of the very tortuous flow paths in sintered metal plates.

For these reasons, a highly permeable, inert capillary structure that is easy to clean and regenerate is needed for high-temperature condensers in microgravity environments. If each of the gravity dependent aspects of the HMC design can be overcome, then a system level, operational demonstration could be carried out aboard the ISS.

II. A Robust, Gravity-Insensitive High-Temperatrue Condenser

To overcome the performance limitations discussed above, Creare is developing a condenser with a highly permeable, lightweight capillary structure made of a stainless steel alloy that is highly resistant to chemical corrosion and compatible with high-temperature services. The internal surfaces of the condenser will be coated with a film of metal that is highly biocidal to a wide range of microorganisms and highly resistant to corrosion to inhibit biofilm growth. Capillary structures can be regenerated if needed by heating the reclaimed water inside the condenser to boiling temperature. The condenser uses a microchannel configuration to achieve a large heat transfer area and a large phase barrier area, reducing its vulnerability to fouling. The configurations and operation of the overall condensing system and the condenser itself are discussed below.

A. Configuration and Operation of Overall Water Recovery Subsystem

As shown in Figure 2, the humid air stream entering the condenser passes over cool hydrophilic screens, where water vapor in the gas stream condenses. The resulting condensate is drawn across the screen by a suction pressure established at the Venturi pump, and mixed with the recirculating condensate flow. The condensation heat is rejected to the recirculating condensate flow, which subsequently transfers the heat to the external cooling air flow (not pictured). The adhesion of water to the hydrophilic screen matrix prevents the gas stream from passing though pores in the screen as long as the pressure difference across the screen is less than the associated capillary pressure (bubble point pressure).

The recirculating condensate pressure is maintained below the humid air stream outlet pressure by using a Venturi pump to draw the recirculating condensate across a flow restrictor and the condenser. The pressure of recirculating condensate at the upstream of the flow restrictor is the same as the pressure inside the flexible bladder, which is maintained at the humid air outlet pressure. Therefore, the condensate pressure will automatically follow the humid air outlet pressure, but remain at a pressure below the humid air due to the pressure drop across the flow restrictor. The flow restrictor (which can be the small transfer tube itself) and the Venturi pump suction pressure together set the pressure difference across the screen. Using a Venturi pump also facilitates the condensate channel priming process. The hydrophilic inner tube downstream of the Venturi pump allows air initially in the condensate channels to be vented out of the system during system start-up.

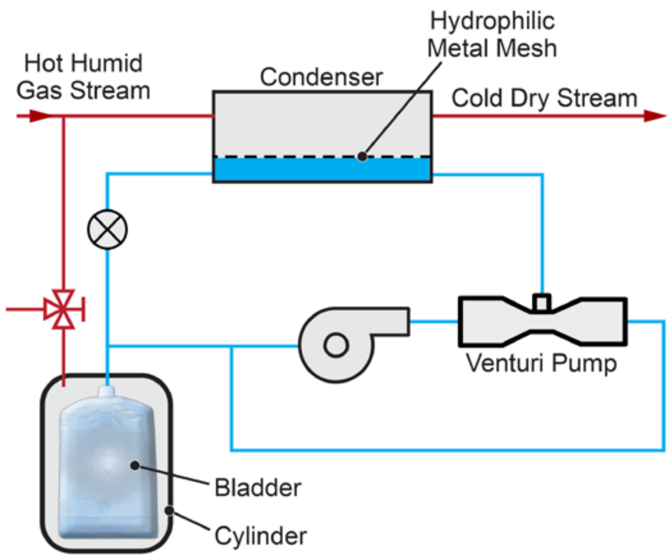

Figure 2. A Simplified Water Recovery Subsystem Configuration with Creare's Condenser

The condensate recirculation loop serves two key functions: priming the system and enhancing heat transfer. The recirculation loop helps to prime the system, enabling the condenser to be fully functional as soon as the HMC system starts up. Without the recirculation loop, the condensate channel and the screen might be dry when the flow exiting the HMC starts to pass through the condenser, allowing air to continuously pass through the screen and be drawn out of the system in the condensate outlet. This large gas outlet flow rate might not be acceptable for certain applications, especially if the air flow has hazardous components. With the configuration shown in Figure 2, condenser priming during start-up is achieved by turning on the self-priming diaphragm pump to circulate the fluid in the loop. The pump will draw water out of the accumulator, which is prefilled with a sufficient amount of water, and generate suction pressure at the Venturi pump suction port to draw water across the condensate channels. Before the water contacts the screen layers in the condenser, there will be a small amount of air being drawn across the screen and into the condensate channel. However, as long as the screen layer has an appreciable flow resistance to allow the condensate channels to maintain a pressure below the ambient pressure (i.e., the humid air pressure before the system start-up), water will be drawn into the channels, as demonstrated in the experiment discussed below. As soon as this occurs, water will wet the screen and prevent air from the humid air channel from further being drawn into the channels. The porous hydrophobic tube at the downstream of the Venturi pump vents the air drawn out of the condenser.

The recirculation flow also helps to enhance the heat transfer between the ambient cooling air and the gas stream. If the condensate is stagnant, condensation heat released on the screen layer needs to conduct through the liquid in the condensate channel before reaching the cooling air. Convection in the recirculation loop significantly reduces the thermal resistance in the condensate channels.

B. Condenser Configuration

To meet the condenser design requirements, we first developed a preliminary condenser layout design that is compatible with the operating conditions and environment. We then developed a heat and mass transfer model to optimize geometric design parameters of the condenser. The goal is to minimize its size and mass, while keeping the pressure drops of the fluid streams (especially the cooling air flow stream) to an acceptable level to minimize the power input for the condenser cooling fan(s).

As shown in Figure 3, the condenser is a crossflow radiator style heat exchanger, with an array of flat tubes having fins attached to their external surfaces. Warm humid air enters the upper left corner of the condenser, passes along the manifold on the left side of each tube, and distributes itself into the parallel tube array. Inside each condenser tube, the humid air passes through the passage created by the humid air spacer, moving from the left to the right. The humid air is cooled by the recirculating condensate flow on the top and bottom of each humid air channel. The recirculating condensate flow is separated from the humid air flow by hydrophilic porous screen layers, which allow water vapor in the humid air flow to first condense on the screen surface. The condensate then is drawn across the screen, and mixed with the recirculating condensate flow. After passing through the condenser tube, the sweep air flow is collected in the manifold on the right side and exits the condenser from the lower right corner. The recirculating condensate follows passages that are parallel to the humid air flow in the condenser tubes.

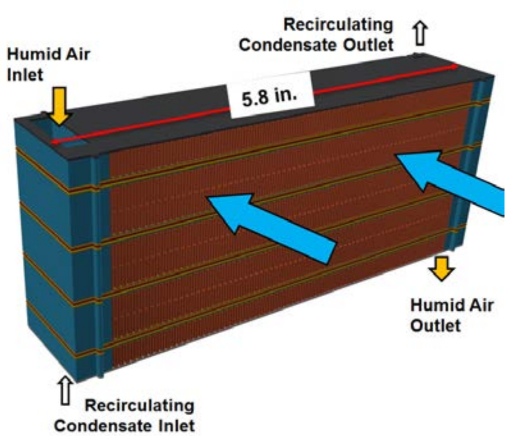

Figure 3. Overall Geometry of Gravity-Insensitive Condenser and Condenser Tube Internal Strcuture. The core size is 5.8 in. × 2.4 in. × 1.2 in. for 0.2 bar humid air.

The sizes and power inputs of the condenser and its auxiliary components in the baseline system are shown in Table 1.

Table 1. Baseline Design Conditions		
Loop pressure	1 atm	0.2 atm
Condenser inlet T	100°C	60°C
Inlet sweep air flow rate	0.2 SLPM	
Inlet water vapor flow rate	0.45 kg/hr	
Humid stream outlet T	34°C	
Cooling air flow rate	15 CFM x 2	
Cooling air inlet temp.	29.4°C	

Fan size	22 in.^{3}
Condenser HX size	66 in.^{3}

.

C. Condenser Design Analysis

The design requirements for the condenser of an HMC to be demonstrated in the ISS are summarized in Table 2. Based the condenser flow inlet conditions, we first determined the water vapor partial pressure and thus the amount of saturated water vapor remaining in the sweep flow as a function of the flow temperature based on thermodynamic analysis. We then calculated the water vapor molar concentration and therefore the amount of water vapor remaining in the humid air as a function of the flow temperature, as shown in Figure 4. Near the condenser inlet, the humid air flow mainly consists of water vapor (water vapor molar concentration is about 98%). The water vapor flow rate in the humid air decreases very quickly after the flow temperature drops below its dew point. This is primarily because water vapor saturation pressure decreases exponentially as its saturation temperature decreases. This suggests that most of the condensation heat is rejected at relatively high temperatures, close to the dew point of the inlet humid air. This high heat rejection temperature is very beneficial, helping to reduce the condenser size.

Table 2. Baseline Design Conditions		
Loop pressure	1 atm	0.2 atm
Condenser inlet T	100°C	60°C
Inlet sweep air flow rate	0.2 SLPM	
Inlet water vapor flow rate	0.45 kg/hr	
Humid stream outlet T	34°C	
Cooling air flow rate	15 CFM x 2	
Cooling air inlet temp.	29.4°C	
Fan size	22 in.^{3}	
Condenser HX size	66 in.^{3}	

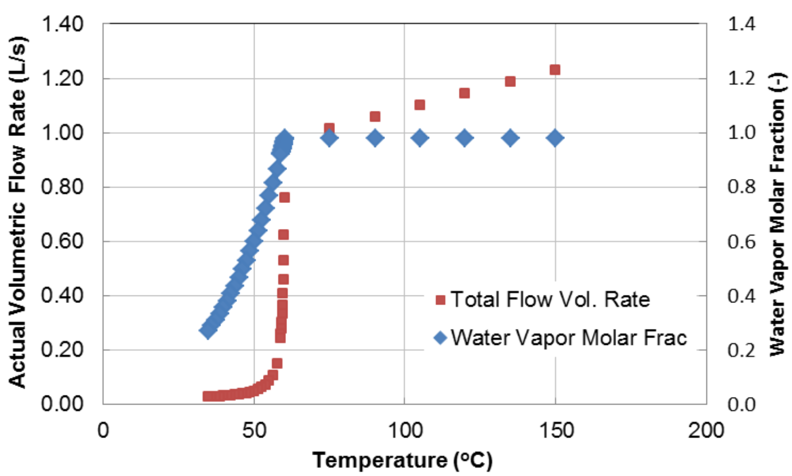

Figure 4. Humid Air Volumetric Flow Rate and Water Vapor Molar Concentration as Mixture Temperature Decreases for 3 psia (0.2 bar) Humid Air.

We then developed a first-order analysis model to determine the proper values for key parameters of the condenser, including the height, width, and length of the fluid flow passages; sizes of the manifold cutouts for the humid flow and condensate flow; and the fin geometry for the cooling air flow. For a given set of condenser design parameters, the model calculates the heat transfer coefficients of each fluid stream and their thermal conductance (UA) with heat transfer surfaces. To calculate the condensation heat transfer coefficient on the humid air side, the model first computes the mass transfer coefficients of the water vapor in the humid air stream. It then iteratively calculates the water vapor flux, the corresponding heat flux, and therefore the effective condensation heat transfer coefficient.

The model also calculates pressure drop values in the flow passages and the manifolds of each flow stream. We then iteratively adjusted the values of the key parameters to minimize the size of the condenser while maintaining pressure drops for all fluid streams, especially the cooling air, at acceptable values (about 0.2 in. H_2O for cooling air flow). We also imposed fabrication limits in the design study. The preliminary design optimization is achieved by balancing the thermal resistances of the humid air flow, recirculation condensate flow, and the cooling air flow as much as possible to ensure that the overall condenser performance is not completely controlled by one particular fluid stream. We also ensure that the dynamic pressure of each stream in the manifold is small compared to its frictional pressure drop across channels to minimize flow maldistribution.

The overall size of the resulting condenser is 5.8 in. × 2.4 in. × 1.2 in. for a 3 psia humid air stream. The predicted overall thermal conductance (UA) of the condenser is 24.2 W/K, 20% higher than the design target value. The predicted cooling air flow pressure drop is about 62 Pa. The pressure drops for the condensate recirculation flow and humid air flow are negligible. The condensate pressure drop across the hydrophilic screen layers is less than 20 Pa, much lower than the screen bubble penetration pressure of about 10 kPa. Because the humid air flow rate is quite small, the predicted pressure drop across the microchannels is only about 600 Pa (2.5 in. H_2O).

D. Performance of Creare's Gravity-Insensitive Condenser Technology

The configuration and design features of the condenser offer significant performance benefits: (1) The height of the channels is only about 200 microns. In such small channels, the dominant flow pattern will be intermittent slug/bubble flow or annular flow. As a result, the flow pattern and the phase separation process will not be sensitive to flow rates of individual phases. (2) The small flow passages in the condenser minimize the chance that small liquid droplets may be carried out with the exiting gas flow, and thus enhance the separation efficiency. (3) The operation of the condenser relies on capillary force for phase separation; therefore, its operation is gravity independent. (4) The condenser has a large hydrophilic screen surface area, allowing the use of a screen with small pore sizes to increase the capillary force separating the phases. The large surface area and high permeability also reduce its susceptibility to fouling. (5) The small microchannel dimensions enhance heat and mass transfer in the condenser. Furthermore, the metal construction and design configuration also offer great flexibility for potential coating and regeneration processes to reduce biofilm growth, thus enhancing the reliability of the condenser.

Our condenser design model calibrated by the preliminary test result discussed in Section III shows that Creare's water recovery subsystem is significantly smaller than the current baseline subsystem. The total subsystem volume, including the condenser, cooling fans, circulation pumps, and Venturi pump, is about 39 in.^{3}, compared to the total fan and condenser volume of 88.6 in.^{3} in the baseline design (the 88.6 in.^{3} volume does not include the volume of the centrifugal phase separator).[6] The input power to Creare's system is 9.3 W, compared to a combined power input of 42.8 W for the fan and the centrifugal phase separator in the baseline design. Therefore, compared to the current baseline design, Creare's condenser technology will reduce the size and power input to the WRS by more than 56%, and 78%, respectively.

The size of the condenser discussed above is for an HMC operating at a sub-atmospheric pressure of 0.2 bar. Increasing the HMC operating pressure will increase the water vapor condensation temperature, and thus will substantially reduce the condenser size and mass. Our analysis shows that increasing the operating pressure from 0.2 bar to 1 bar will reduce the size of the condenser by about 50%. The following feasibility demonstration was conducted with the humid air at the atmospheric pressure.

III. Proof-of-Concept Water Recovery Subsystem Demonstration

A. Test Setup

After completion of the condenser design analysis and mechanical design, we fabricated and assembled the condenser using techniques developed at Creare, as shown in Figure 5. We then conducted separate effects tests to verify the functionality of the condenser. Next we assembled the condenser test setup and its instrumentation system, as shown in Figure 6 and Figure 7. Briefly, the system consists of an open humid air flow path and a closed condensate recirculation loop, connected via the condenser unit. To simulate operation in the HMC, humid air flow is provided using a small commercial steam generator to inject steam into a hot air flow which simulates the sweep air in the HMC. The air flow was preheated to a temperature of about 100°C to ensure that water vapor would not condense during the mixing process and transfer process to the condenser inlet due to heat leak to the ambient. The dry air stream exits the condenser through a clear tube and terminates into a separation vessel at ambient pressure. During testing, we measured key temperatures and pressures of the humid air stream, including the condenser inlet temperature and pressure, the pressure drop across the condenser, and the temperature at the outlet condenser.

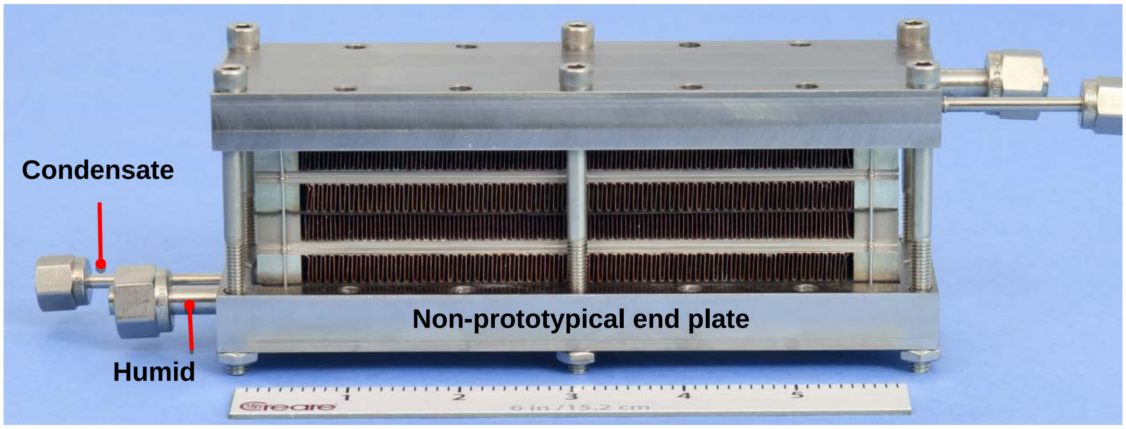

Figure 5. Proof-of-Concept Condenser. It achieves a water recovery rate of 0.13 g/s and a water recovery efficiency of 96.5% with atmospheric pressure sweep air. The mass of the condenser (not including nonprototypical end plates) is 265 g.

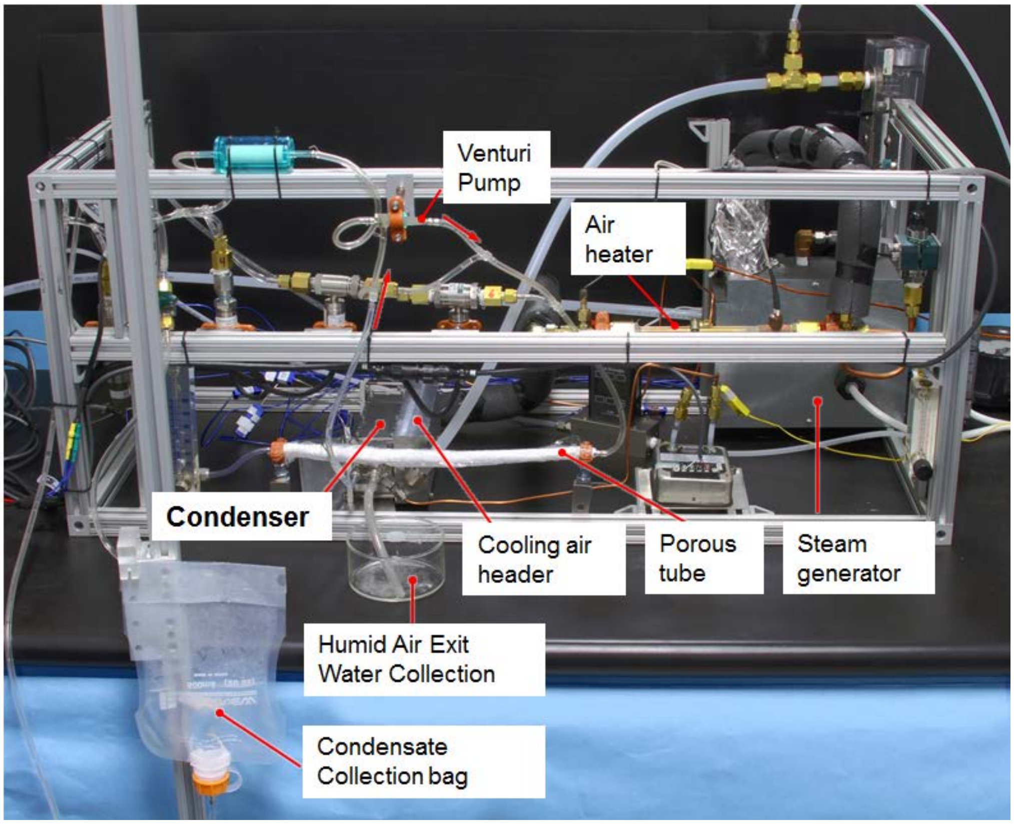

Figure 6. Test Setup for Phase I Condenser

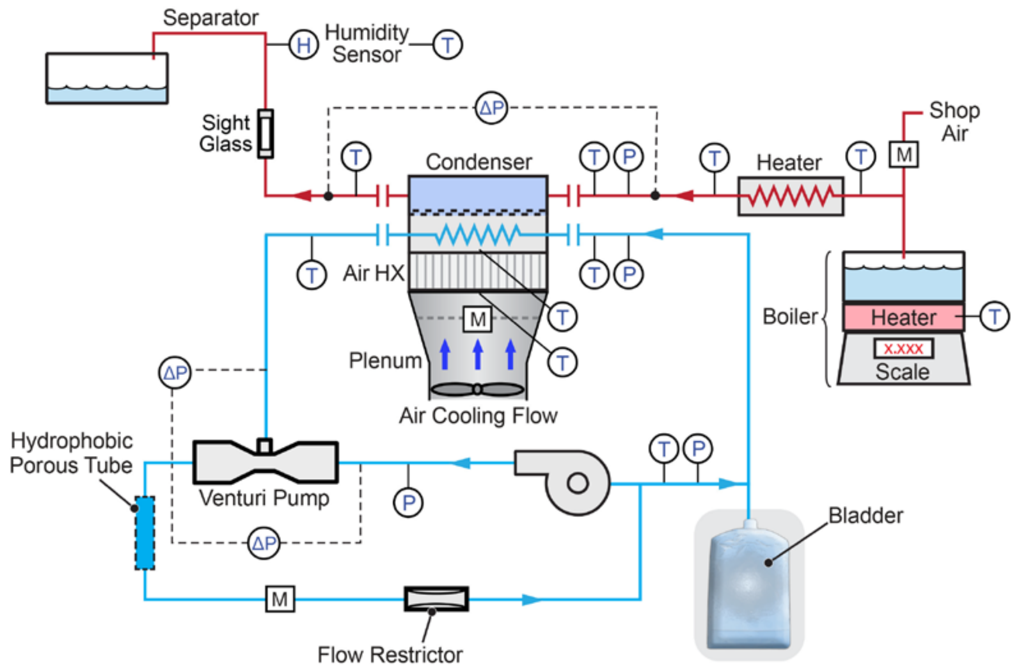

Figure 7. Final Schematic of Test Setup for Phase I Condenser

System start-up was initiated by turning on the steam generator to begin producing moisture. While the steam generator was warming up to its steady state, we turned on the condensate recirculation pump and set the appropriate system pressures by restricting the inlet of the condensate recirculation flow. This allowed the condensate pressure at the condenser inlet to be slightly below ambient pressure to receive water, and the Venturi pump outlet pressure to be slightly above ambient to reject air through the downstream porous hydrophobic Teflon tube. Once the pressures were set appropriately, any extra air present in the condensate recirculation loop would be rejected to the ambient via the porous Teflon tube. After the boiler reached steady state, we began to flow the sweep air and cooling air into the system. Moisture flowing into the condenser would begin to get condensed and drawn into the condensate loop. Condensed fluid brought into the system would present itself as extra water added to the condensate collection bag, raising the level of fluid in the bag. After system start-up, we let the system run for about one hour to let the heat and mass transfer come to a quasi-steady state. Due to the low capacity of our steam generator, the system cooled briefly every 5 to 10 minutes when the steam generator took in new fresh water. Once at steady state, we weighed the condensate collection bag to get a baseline system mass, and began the test timer. We ran the system at steady state for 20 minutes, then again weighed the condensate collection bag. The difference in mass over the length of time testing was the condensation rate of the system. The condensation rate was compared to the moisture generation rate for the steam generator at the given power to produce a water recovery efficiency (%).

B. Performance at Nominal Conditions

The test unit designed and built for this proof-of-concept experiment was sized to dry an inlet humid air/water vapor mixture consisting of a steam flow at 0.13 g/s and a steady sweep air flow rate of 0.2 slpm. We projected that this would be achieved by establishing a 0.75 psid suction pressure at the Venturi pump in our condensate recirculation loop, pulling condensed water through the highly permeable hydrophilic barrier. The condensate would in turn be cooled by a crossflow of room-temperature air at 15 CFM. Water pumping and air flow were projected to consume 0.36 W, and 4.3 W, respectively. In Table 3, these values are shown in comparison to performance data measured during one of our tests. The second column shows data from a test run that was intended to be very similar to the design point. During this particular test, 102°C steam/air was supplied to the condenser (0.14 g/s steam, 0.25 slpm air). The measured electrical power input for the water pump drove condensate recirculation is 0.54 W. Cooling air was supplied at 14.7 CFM and 21.7°C. Over the course of 22.3 minutes at steady state, we collected 177 g of water of a total 183 g consumed for steam generation, equating to a recovery efficiency of 96.8%.

We used experimental measurements to confirm an approximate energy balance between the three fluid streams at the design point. The 177 g of condensate recovered can be converted to 342 W of condensation heat. This is roughly the same as the 354 W supplied to the steam generator during this period (where we have subtracted heat leak to the environment). These values are also internally consistent with the 305 W that we estimate is transferred to the air, using the average air stream temperature rise. In this value, there is substantial experimental uncertainty. This is due to the fact that our measurement relies on a spatial average of outlet temperature from 5 thermocouples, whereas the actual air flow distribution was non-uniform. We did not pursue this further as it did not appear to significantly impact condenser operation.

Successful operation of the condenser depends on a number of critical pressure differentials that we have identified as follows (Figure 8): As indicated by (1), the condensate water pressure must be sub-ambient in order to provide a driving force to pull condensed vapor across the screen. This pressure difference is set by the test operator using a restriction on the water loop. It cannot exceed the screen bubble point, which would allow air to pass through freely. The Venturi pump must also provide adequate suction (2) to pull the sub-ambient condensate through to the condenser, back to the pump inlet. At (3), we are illustrating that the pressure at the Venturi pump outlet is above ambient, which it must be to allow rejection of air through our hydrophobic tube section during the start-up. The difference at (4) shows the pressure drop along the humid air flow channel. Finally, (5) indicates the increase in gravitational pressure head within the condensate recirculation loop as water accumulates in our recovery bag over time; this would of course not be observed during zero gravity operation. Throughout the test, sharp drop-offs in pressure and temperature are observed every 4 minutes, which corresponds to our steam generator pausing to recharge its water reservoir.

In Figure 9, we can see the large temperature drop experienced by the humid air as it passes through the condenser. This is important, because a low outlet temperature is critical to reducing the water carrying capacity of the sweep air, and therefore increasing the water recovery efficiency. The rise in cooling air temperature across the fins is also shown at a single measured location along the axis of the condenser. Though this is seen at roughly 40°C in the plot, we manually measured outlet temperature at five locations, leading to the estimated average value of 60°C shown in Table 3. The condensate recirculation loop temperature rise is small, as expected.

Table 3. Creare Water Recovery System Test Data

Parameter	Design	Test Data (15 cfm)
Humid Air Flow		
Inlet Temperature (°C)	100.0	102.0
Inlet Pressure (psia)	14.8	15.0
Outlet Temperature (°C)	34.0	21.8
Pressure Drop (in. H_2O)	2.7	2.3
Steam Flow Rate (g/s)	0.13	0.14
Sweep Air Flow Rate (lpm)	0.20	0.25
Condensate Recirculation		
Inlet Temperature (°C)	32.0	25.9
Outlet Temperature (°C)	32.0	35.0
Venturi Pump Suction dP (psid)	0.75	0.80
Total Flow Rate (g/s)	1.0	1.3
Cooling Air Flow		
Inlet Temperature (°C)	29.4	21.7
Inlet Pressure (psia)	14.7	14.8
Avg. Outlet Temperature (°C)	69.1	60.0
Pressure Drop (in. H_2O)	0.21	--
Flow Rate (cfm)	15.0	14.6
Component Performance		
Cooling Air / Fan (W)	4.3	--
Water Pump (W)	0.36	0.54
Collection Efficiency (%)	**99.8**	**96.8**

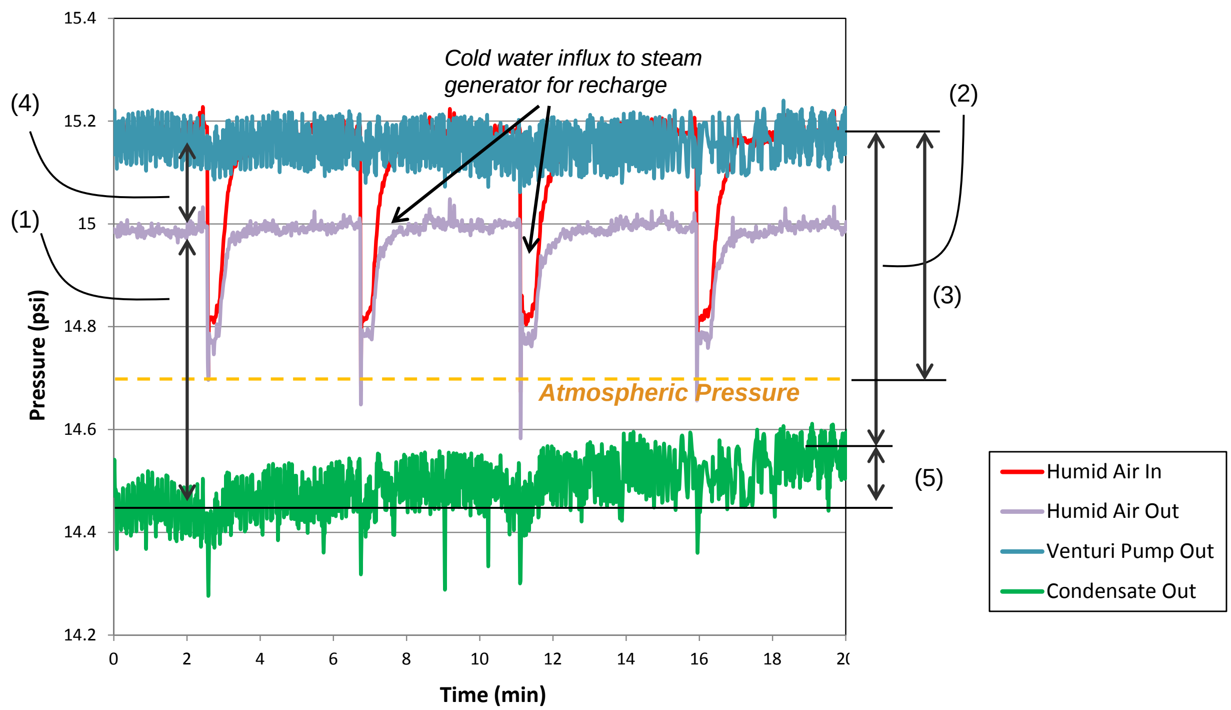

Figure 8. Measured Pressure Data with the Condenser Running at Design Conditions for the Creare Gravity-Insensitive Condenser.

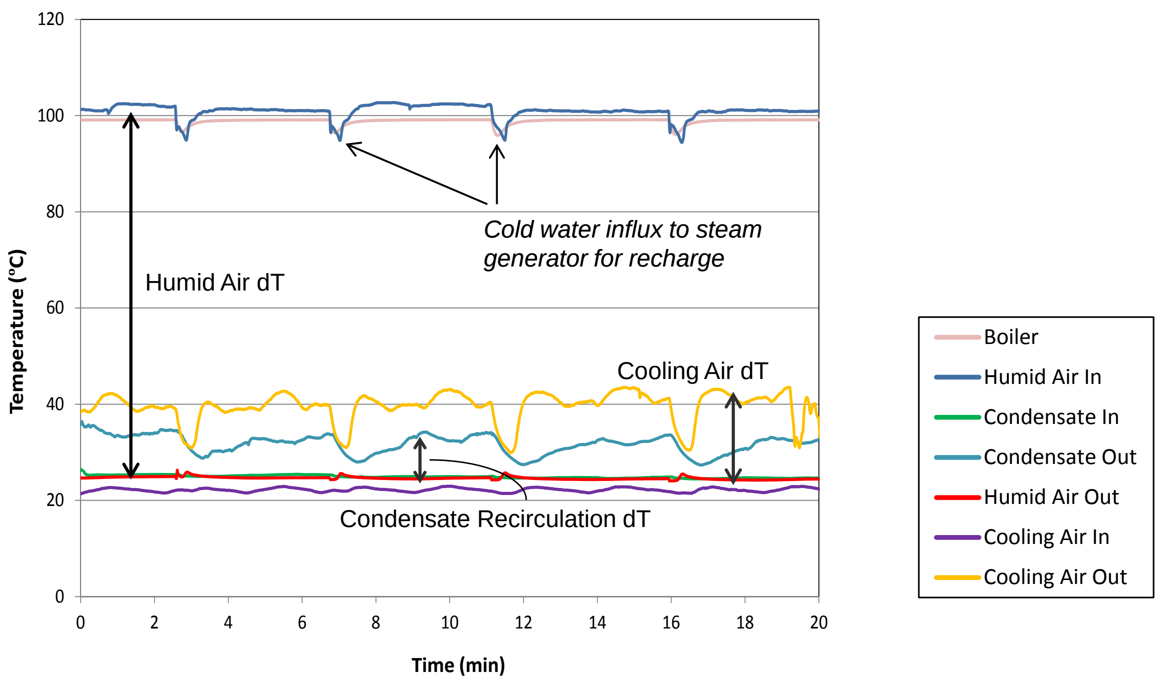

Figure 9. Measured Temperature Data of All Three Streams at the Condenser Inlet and Outlet, Running at Design Conditions for the Creare Gravity-Insensitive Condenser.

IV. Conclusion

Creare's compact, robust, gravity-insensitive condenser is a technically feasible approach to enable efficient water recovery in low-gravity environments. In this preliminary study, we demonstrated the feasibility of the condenser and its performance benefits by building a brass-board system, conducting testing, and acquiring experimental data with good agreement to initial predictions, including high recovery efficiency with low system mass and low power consumption. Gravity-insensitive operation of the phase-separator mechanism was also verified on a component level. Separately, we also conducted a short-term accelerated life testing of the screen to show that the wettability and permeability of the capillary structure do not degrade substantially after an extended period of operation in a warm humid environment.

Our water recovery subsystem with a lightweight condenser (265 g) successfully demonstrated recovery efficiency in excess of 96% at a steam flow rate of 0.13 g/s. As-built, the system is already very compact and lightweight and consumes very little power. Compared to the baseline design, Creare's condenser technology reduces the size and power input by more than 56%, and 78%, respectively.

To further mature the technology, the next logical steps should be: (1) refining the condenser fabrication technologies to further reduce its size and mass; (2) developing advanced coating to enhance the condenser's resistance to biofouling; (3) optimizing the design of the water recovery subsystem to enhance its water recovery efficiency and simplify its operation; (4) fabricating the condensers and assembling a complete water recovery subsystem; and (5) fully characterizing the performance of the water recovery subsystem, including its service life under the prototypical operating environment.

Acknowledgments

Work reported here was sponsored by a NASA Small Business Innovative Research contract..

References

[1]Turner, M., Fisher, J. and Pace, G., "Generation 2 Heat Melt Compactor Development," 44th International Conference on Environmental Systems, 13–17 July 2014, Tucson, AZ.

[2]Broyan, J., Ewert, M. and Fink, P., "Logistics Reduction Technologies for Exploration Missions," AIAA Space 2014.

[3]Hoyt, N., Kang, M. F., Kharraz, A., Kadambi, J. and Kamotani, Y. "Cyclonic Two-Phase Flow Separator Experimentation and Simulation for Use in a Microgravity Environment," J Phys Conf Ser, Vol. 327, 2011, p. 012056.

[4]Harris, L. C., Wignarajah, K., Alba, R., Pace, G. and Fisher, J. W., NASA Ames Research Center, Characterization of Heat Melt Compactor (HMC) Product Water, 43rd International Conference on Environmental Systems, July 14–18, 2013, Vail, CO.

[5]Nguyen T., Roddick F. A. and Fan, L. "Biofouling of Water Treatment Membranes: A Review of the Underlying Causes, Monitoring Techniques and Control Measures," Membranes, 2012, Vol. 2, No. 4, pp. 804–840.

[6]NASA Internal Document, HMC Water Recovery Subsystem Trade Study-Rev F, March 24, 2013, Powerpoint Presentation, Email communication with Mike Ewert, NASA JSC) on Jun 26, 2015.