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A Freezable Heat Exchanger for Space Suit Radiator Systems

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ABSTRACT

During an ExtraVehicular Activity (EVA), both the heat generated by the astronaut's metabolism and that produced by the Portable Life Support System (PLSS) must be rejected to space. The heat sources include the heat of adsorption of metabolic CO_2, the heat of condensation of water, the heat removed from the body by the liquid cooling garment, the load from the electrical components and incident radiation. Although the sublimator hardware to reject this load weighs only 1.58 kg (3.48 lbm), an additional 3.6 kg (8 lbm) of water are loaded into the unit, most of which is sublimated and lost to space, thus becoming the single largest expendable during an eight-hour EVA. Using a radiator to reject heat from the astronaut during an EVA can reduce the amount of expendable water consumed in the sublimator.

Radiators have no moving parts and are thus simple and highly reliable. However, past freezable radiators have been too heavy. The weight can be greatly reduced by placing a small and freeze tolerant heat exchanger between the astronaut and radiator, instead of making the very large radiator freeze tolerant. Therefore, the key technological innovation to improve space suit radiator performance was the development of a lightweight and freezable heat exchanger that accommodates the variable heat load generated by the astronaut. Herein, we present the heat transfer performance of a newly designed heat exchanger that endured several freeze / thaw cycles without any apparent damage. The heat exchanger was also able to continuously turn down or turn up the heat rejection to follow the variable load.

INTRODUCTION

At present, both the astronaut's metabolic heat and that produced by the Portable Life Support System (PLSS) are rejected to space by a sublimator that consumes up to 3.6 kg (8 lbm) of water [1]; the single largest expendable during an eight-hour extravehicular activity (EVA) [2]. This will not be acceptable for long duration Lunar and interplanetary missions where resupply is difficult. The amount of water lost to sublimation can be greatly reduced by radiating most of the heat load to the ambient environment.

MISSION NEED FOR SPACE SUIT RADIATORS

A space suit radiator can replace the PLSS covering with very little net increase in weight and yet will cut the amount of water needed to cool the astronaut during an EVA by up to 2.7 KG (6 lbs). This will represent a significant cost savings to future missions. As an example, the assembly of the International Space Station (ISS) has required approximately 1920 EVA hrs so far [3]. The cost to transport the cooling water needed during EVAs was about \$20M assuming a cooling water consumption rate of 0.45 kg/h and an approximate launch cost of \$22,000/kg to transport the water to the ISS. A space suit radiator would have lowered this cost by \$15M. However, the costs to transport cooling water increase by about 12X for Lunar and 26X for Mars surface EVAs [4], which will make it prohibitively expensive to carry water just for cooling. In-situ sources of water may be available, but the costs to recover are not known. For Lunar and Mars EVA missions the reduction in water loss is not merely nice, it is essential.

Current plans for Lunar exploration place an outpost and the majority of EVA missions at the poles, which has an ideal environment for the use of a radiator. It will be highly effective, since the sink temperatures of space and the Lunar surface are both cold and not subject to large fluctuations in temperature. Incident solar energy will not heat the radiator surface due its low absorption in the visible spectrum.

FREEZABLE SPACE SUIT RADIATORS

While the advantages of radiators have long been recognized (they can operate effectively in Earth orbit, trans-Lunar, trans-Martian, and in most Lunar and Mars environments), they have a problem that has so far prevented their use on space suits: current simple designs reject heat at a relatively steady rate in moderately cold to very cold environments. Further, the effective sink temperature will depend on the orientation of the radiator which can change as the astronaut performs the mission, but neither of these will correlate with the rate of heat generation by the astronaut which can vary from 70 to 730 W [5-7]. Without a way to continuously adjust the heat removal rate, the astronaut will alternate between heat stroke and frostbite conditions. Therefore, a key technological problem is to develop a lightweight and freeze tolerant space suit radiator able to accommodate the variable heat load.

The heat rejection from a radiator can be "turned-down" by sequentially allowing tubes that carry the water to the radiator to freeze. When a tube freezes, the temperature of the radiating area around it drops, and the heat rejection rate drops as well. The problem with the freezable radiators developed to date is that they are far too heavy, since they used heavy walled tubes to prevent bursting during freeze/thaw cycles. Further, in past freezable designs, the entire surface of the PLSS had to be thick enough to conduct enough heat several inches to melt the adjacent frozen tubes. As a result, no freezable radiator has been light enough to be used in the PLSS.

Cross, Trevino, and Laubach [8] report a system in which the heat flowing to the radiator was decreased by creating a small gap between the radiator and the water loop. When the heat load was high, the gap was filled with a gas to increase the conductance. When the heat load was low, the gap was vented to space, the heat transfer rate was decreased (vacuum is a very good insulator), and the heat rejection rate dropped. However, the weight of the system was 20 kg (44 lbm) compared to 7.2 kg (15.9 lbm) for the current system. Additionally, because of heat leaks, there was only a ~25% reduction in heat rejection in a cold environment.

RADIATOR AND FREEZABLE HEAT EXCHANGER (RAFT-X)

The RAFT-X space suit radiator [6,7] uses a freezable heat exchanger to transfer heat from the astronaut to the radiator. Since only a small heat exchanger must be made freezable, this space suit radiator is lighter than previous designs. It also uses the conventional water loop to cool the astronaut and life support system components, but provides redundancy in heat rejection, since it is completely independent from the sublimator. Further, there are no moving parts and thus is extremely reliable.

As shown in Figure 1, warm water from the astronaut's liquid cooling garment (LCG) enters a freeze tolerant heat exchanger and transfers heat to the heat pipes. The refrigerant boils in the evaporator section of the heat exchanger and vapor moves up the heat pipe that is located on the radiator panel of the PLSS backpack, which has enough surface area to reject up to 300 W. As heat is rejected by radiation, the vapor condenses and travels back to the evaporator. There are no moving parts (i.e. pumps) in TDA's spacesuit radiator and the system is completely passive.

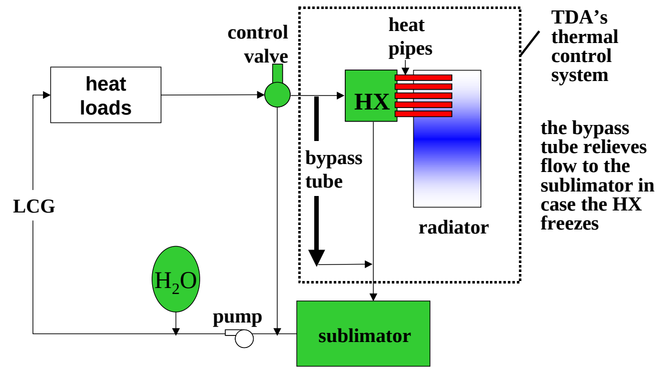


Figure 1. Process flow diagram for the space suit thermal control.

There are a number of advantages to this concept. First, using two separate fluid loops allows the thermal properties of a refrigerant fluid to be advantageously used without altering the LCG. The radiator fluid selection has no impact on the rest of the life support system and to keep the radiator lightweight a refrigerant was chosen that will not freeze within the loop heat pipes even if the PLSS is facing deep space. Second, because the five heat pipe fluid loops are independent from one another and separate from the water loop, a puncture to one of the heat pipes will not be a life-threatening emergency. If the refrigerant in the punctured heat pipe drains out, the water-cooling loop is not affected. The other heat pipes, as well as the sublimator, still remain operational to provide the needed cooling. In fact, the loss of a single loop heat pipe reduces the heat rejection capacity by only 2 to 5%. [7]

In this novel approach only the heat exchanger needs to be freeze tolerant. Making it freeze trap tolerant ensures effective and safe operation at any load. Under low metabolic load, the control valve reduces the water flow to the heat exchanger. In cold environments the heat exchanger drops below the freeze point of water and passively controls the heat rejection rate by allowing the coldest water flow passages in the heat exchanger to freeze. As the water in the heat exchanger freezes, sections of the heat exchanger become inactive, preventing heat rejection to the heat pipes that are in contact with those surfaces of the heat exchanger. If all of the water passages freeze, then the bypass tube (again, see Figure 1) continues to flow some water to the sublimator and thereby provide uninterrupted cooling even when the heat exchanger and radiator transfers little to no heat.

Overall, the RAFT-X space suit radiator (Figure 2) was demonstrated to transfer heat from 300 W down to 90 W, which is a heat transfer turndown ratio of about 2.7, but unfortunately it could not be turned down further without fully freezing the water channel. At lower loads

(< 90 W) the heat exchanger froze completely. While the heat exchanger was undamaged, it was not able to transfer heat until the chamber was warmed to thaw it back out. [7] In space, it would not become operational again until thawed back in the spacecraft. The sublimator remains online in order to complete the EVA without interruption. In summary, this system showed that a freezable heat exchanger could be built, but it was not capable of rapidly freezing and thawing to regulate the radiator, and it was still equal to or slightly heavier on-back than a standard sublimator system.

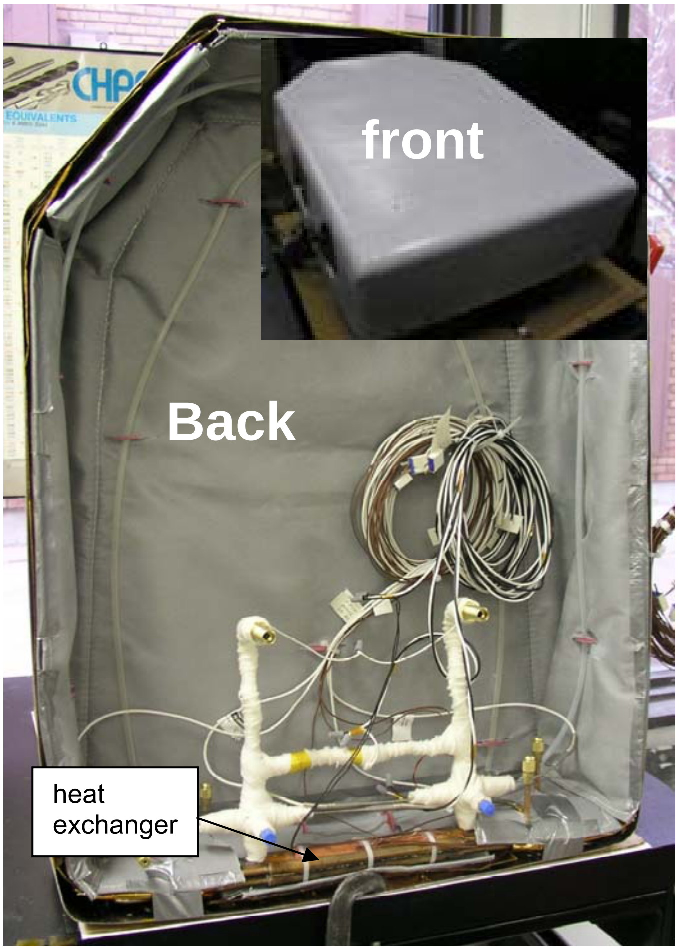


Figure 2. RAFT-X prototype space suit radiator.

A FREEZABLE HEAT EXCHANGER TO PASSIVELY REGULATE HEAT TRANSFER

A new heat exchanger was developed to passively regulate the heat rejection rate to the radiator. The heat exchanger responds quickly and smoothly to match the rate at which heat is rejected to the rate at which it is produced. As the heat load drops, such as during periods when the astronaut is at rest, some of the heat exchanger flow passages freeze to block heat from being transferred to the nearby heat pipes and thereby also to the radiator surface. When the heat load rises the frozen passages rethaw to once again transfer heat at greater capacity. The heat exchanger was designed to accommodate the volume change that occurs when the water freezes. This proprietary design was proven during freeze tests of the RAFT-X. [6, 7]

CONCEPT OF OPERATION

As shown in the space suit radiator concept sketch (Figure 3), the water from the LCG passes through a small, freezable heat exchanger and loses its heat to the loop heat pipe evaporators, which are joined to the heat exchanger to ensure good conductance. The heat evaporates refrigerant stored in the evaporators. The loop heat pipes transport the heat to the radiator and distribute the heat throughout the radiator surface. The extremely low freeze point of the refrigerant ensures that the heat pipes can not freeze and therefore thin-wall tubing is used for their construction.

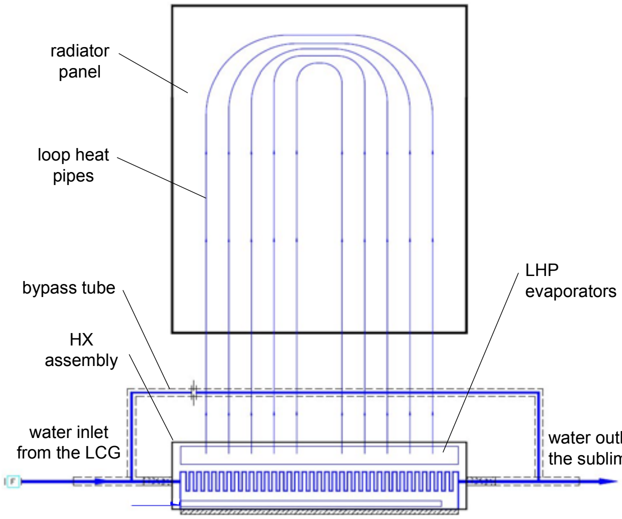


Figure 3. Concept sketch of the space suit radiator.

Of the space suit radiator components (the heat exchanger, the loop heat pipes and the radiator), only the heat exchanger must be freeze tolerant and thus, it is the most important component in the system. Although not shown in Figure 3, the design provides for the needed expansion volume when the water freezes. At low metabolic loads the efficiency of the heat exchanger drops as ice forms within the unit. The build up of ice on the fins reduces the surface area within the heat exchanger and lowers their conductance (the conductivity of ice is only 1/200th that of the fins). Thus, little heat is transferred to the radiator and the astronaut does not freeze. The heat exchanger does this by freezing water in most of the flow channels at low loads, which decreases the active surface area and blocks the flow of heat to some of the heat pipes, thereby limiting the flow of heat to the radiator. Therefore, the heat exchanger effectiveness is proportional to the incoming load, and thus the heat transferred to the radiator tracks the heat generated by the astronaut.

THERMAL ANALYSIS OF THE HEAT EXCHANGER

TDA developed models of the freezable heat exchanger to investigate specific aspects of the design: both an analytical model and numerical simulations were used to estimate the heat rejection at several operating conditions and predict the build up of ice within the heat exchanger at low load. The models included details of the water flow path and all heat transfer surfaces. The loop heat pipe evaporators were not included in these models; nor were they necessary to analyze the water side of the heat exchanger. It was assumed that the loop heat pipes had sufficient capacity to transport any heat load from the water side of the heat exchanger to the radiator.

2D Axisymmetric Model

EXCEL spreadsheets were created to model the steady, laminar flow of water through heat exchangers having two, four or six fins as indicated by N_{FINS}. The heat removed from the water flow was assumed to equal the convective heat transfer, which in turn was assumed to equal heat conduction through the ice layer and heat exchanger walls. A constant outer wall temperature (Tw_o) was specified for the outer wall boundary. The build up of ice was allowed to grow axially down the heat exchanger, but in order to keep the model simple it was assumed to be of uniform thickness in the tangential direction (Figure 4). Thus, the models are 2D axisymmetric with the radius (r) and the axial position (x) being the two independent geometric variables. The radius from the center mark to the ice layer was designated $r_{ice\ layer}$. The corresponding surface area, As, includes exposed wall and fin surface area as well as the surface of the ice layer. The aggregate will always be equal to or less than the maximum finned surface area (As_o).

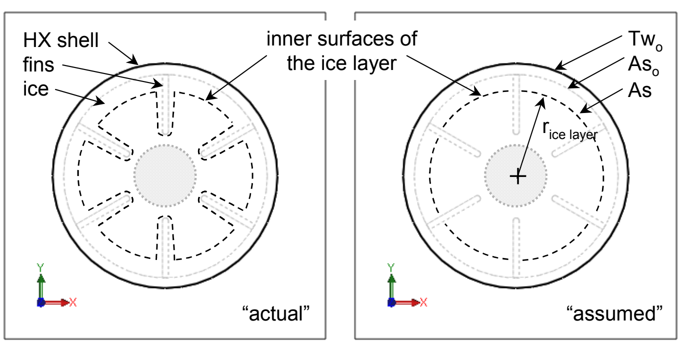


Figure 4. The figure shows a six-fin model of the water side of the heat exchanger. A more realistic profile shape for the build up of ice (figure to the left) is compared to the assumed profile used in the analysis (right figure). Surface area and thermal conductance are reduced as the ice layer builds up.

A cylindrical coordinate system was used for the heat conduction. For a specified outer wall temperature and water flow rate, calculations were made to determine the heat transferred to the wall (Figure 5a), the build up of ice within the heat exchanger (Figure 5b) and the concurrent reduction of surface area (Figure 5c), and the axial temperature distributions for the water, ice layer and heat exchanger structure (Figure 5d). In this analysis the water flow rate was set to 9 kg/h (20 lbm/h) and the water inlet temperature was set to 24°C (75°F) for low load conditons. At maximum load the water flow rate was set to 90 kg/h (200 lbm/h) and 18°C (65°F). These values correspond to the nominal water flow rates through the LCG and the water temperature at the LCG outlet for the heat load.

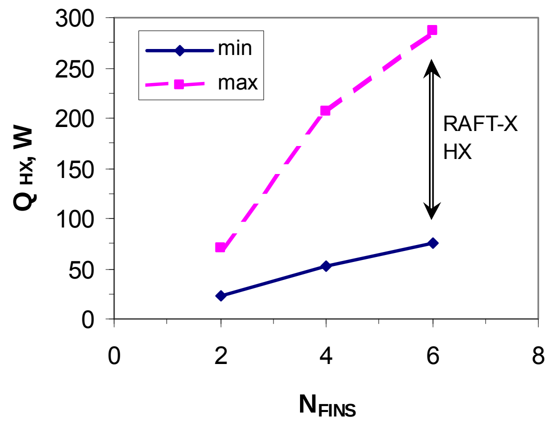
The measured heat rejection from the RAFT-X compared well with the 4- and 6-fin models (again see Figure 5a). In addition, the surface area decreased within the heat exchangers as the ice layer built up on the shell and fins (Figure 5c). The correlation, which is expressed by Equation 1 below and also presented in the figure, fits the data well and will be used later to predict the surface area for analysis of experimental results.

$$As/As_o = 0.2676 \cdot \ln[Q/(T_{FPt} - T_{HX})] + 0.0468 \qquad \text{Equation 1}$$

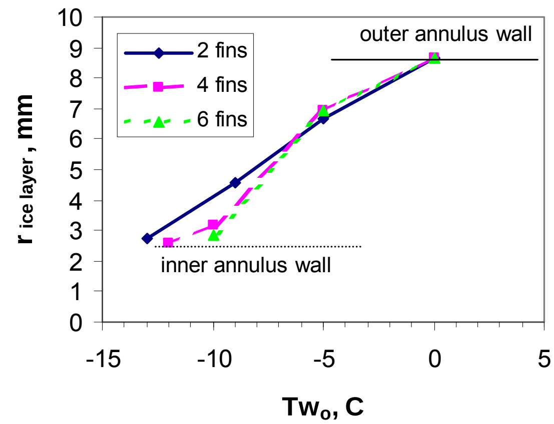
with the heat flux (Q) calculated from the heat exchanger analysis model and the temperature difference expressed in terms of the heat exchanger structural temperature (T_{HX}) subtracted from the freeze point of water (T_{FPt}).

The results plotted in Figure 5d were for a heat exchanger with six fins and the outer wall temperature set to -10°C. The heat exchanger shell is nearly isothermal due to the thin wall and conductive tubing used in its construction. The ice temperature next to the water flow was assumed to be 0°C. The water flow was cooled as it passed through the heat exchanger. Once an ice layer has formed, the water no longer loses much heat since the fins are under ice and the conductivity of ice is only 2 W/m-C.

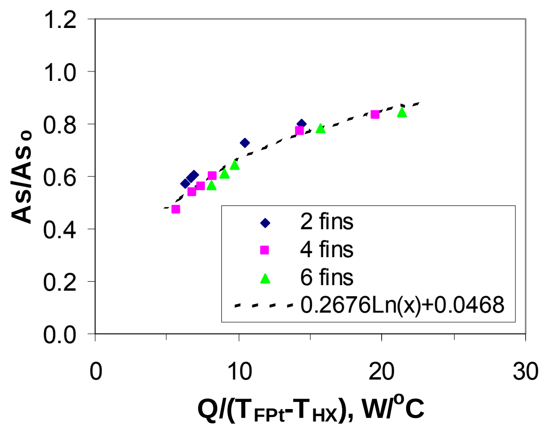
For all geometries the ice buildup within the heat exchanger had blocked 80% of the flow area by the time the outer wall temperature fell to about -10°C, which compares reasonably well to the RAFT-X data. In RAFT-X experiments conducted in the NASA JSC Chamber E Thermal Vacuum Facility the heat exchanger water passages were completely frozen by the time the heat exchanger outer wall fell to -17°C (+2°F). [7]



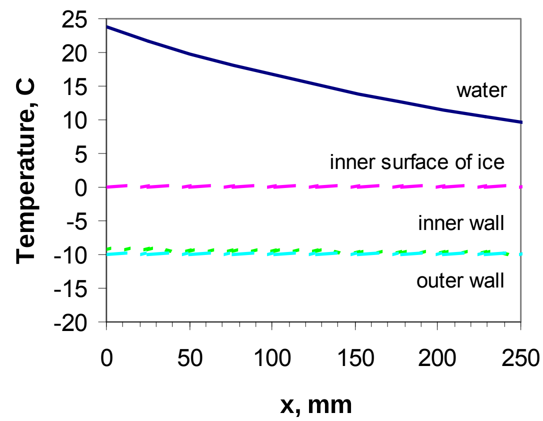
a) Dependence of the heat rejection rate on the number of fins. The water flow rate was 90 kg/h for max Q and 9 kg/h for min Q.



b) The build up of the ice layer within the heat exchanger (water flow rate = 9 kg/h).



c) The reduction of surface area during freeze.



d) Temperature distributions for a heat exchanger w/ six fins at 9 kg/h water flow rate and -10°C outer wall temperature.

Figure 5. Predicted heat exchanger performance from the 2D freeze models.

<u>3D CFD Freeze Model</u>

The CosmosFloWorks and SolidWorks CAD packages provide a seamless approach to conduct time-accurate fully coupled 3-d non-reacting flowfield and thermal analysis. The computational fluid dynamics (CFD) code utilizes a k-ε turbulence model with modified wall functions (not the widely used "law of the wall") to characterize laminar and turbulent flows near the wall. Periodic grid refinement was used to achieve mesh independent solutions. Simplified solid models of the heat exchangers were assembled (one shown in Figure 6). Appropriate boundary conditions were applied to simulate the desired operating condition.

Although the CosmosFloWorks CFD code cannot solve two-phase flows directly, the fluid database was modified to incorporate the effects of freezing. At 0°C (32°F) the viscosity was increased by several orders of magnitude to simulate a solid. The heat of fusion was accounted for by modifying the specific heat (Cp) over a very small temperature difference at the freeze point. Below 0°C it was set to 2,100 J/kg-K (the value for ice).

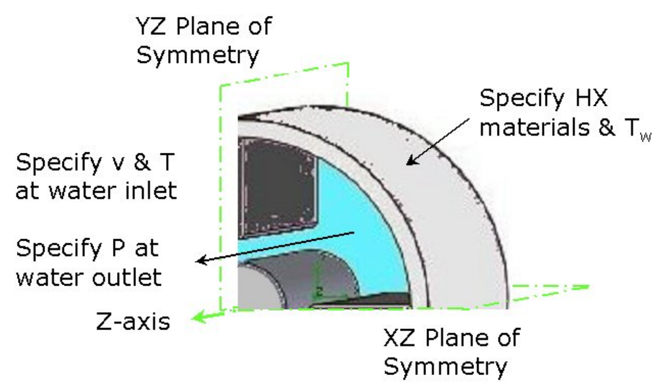


Figure 6. Model representation of a 4-fin heat exchanger for numerical analysis.

In the numerical simulations the water inlet condition was set to 4.5 kg/h (10 lbm/h) and 24°C (75°F). These inlet conditions correspond to a minimum heat load scenario and it was assumed that about half of the water flow (another 4.5 kg/h) was bypassed to the sublimator. A steady simulation was run of a heat exchanger having four highly conductive fins and with the outer wall temperature set to -5°C. The temperature contours within the heat exchanger are shown in Figure 7.

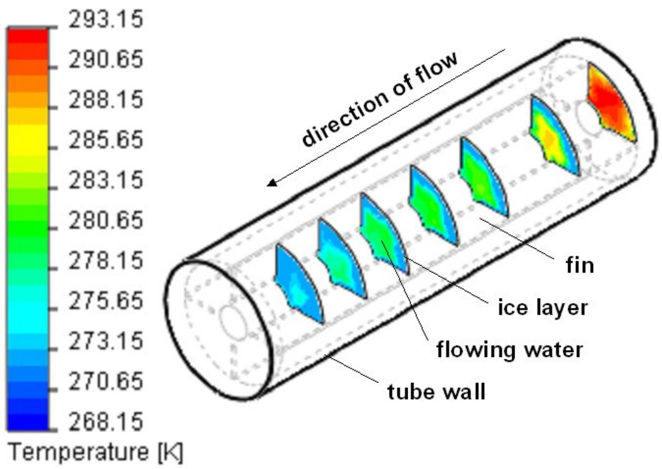


Figure 7. XY temperature contours in a 100-mm long heat exchanger section with four fins having a conductivity of 300 W/m-C (173 Btu/h-ft-F).

The numerical and Excel models agree reasonably well until the outlet. The radius to the ice layer was predicted to be 6.6 mm through the middle section of the heat exchanger (i.e. the ice layer had become 2.0-mm thick). However, at the outlet the numerical model predicted a rapid increase in the ice layer thickness, which had built up to nearly completely block the flow channel at the exit. This is believed to be an artifact of the specified outlet boundary conditions and one approach to minimize this effect on the overall solution is to increase the length until it is far from the domain of interest.

Summary of Heat Exchanger Analyses Results

Overall, the analyses show that six fins are needed to reject the maximum desired heat load of 290 W (1000 Btu/h). The heat rejection rate depended on the water flow rate, the number of fins and the outer wall temperature. At low water flow rates an ice layer was predicted to build up within the heat exchanger and thereby match the heat transfer rate to the desired input load that corresponded to the inlet conditons. The reduction in heat transfer was accomplished by a continuous and proportional reduction in both surface area and conductance as the water flow rate (i.e. input load) was reduced. The results were used to design heat exchangers for experimental evaluation and to assist in analysis of the data.

EXPERIMENTAL APPARATUS

TDA assembled the apparatus shown in Figure 8 to chill the heat exchanger and record the test data. The ethanol/isopropyl alcohol refrigerant was recirculated through a 100 W Lauda Brinkman RM6 chiller. With a liquid nitrogen cooling loop the apparatus chilled the refrigerant to as low as -37°C (–35°F).

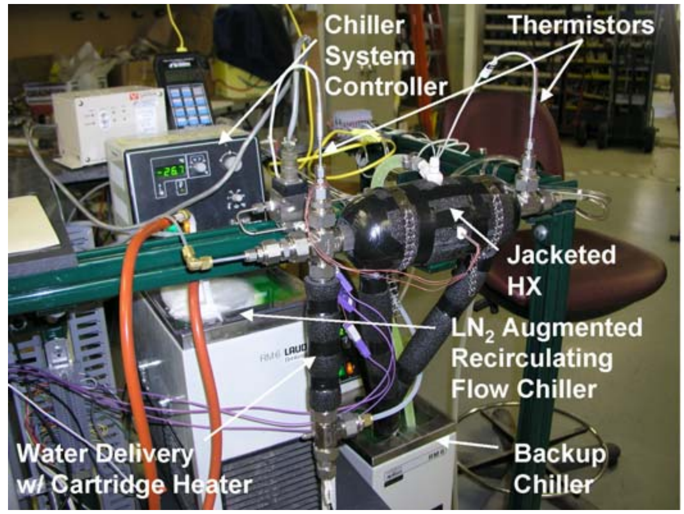


Figure 8. Heat exchanger test apparatus.

The water delivery system supplied water from 0 to 136 kg/h (0 to 300 lbm/h), which was measured with an Omega FLR 1000 series flow meter. A 1000 W cartridge heater raised the water temperature to the desired set point, which generally was in the range of 18 to 24°C (65 to 75°F) to correspond to outlet temperatures expected from the LCG. The inlet and outlet water lines were instrumented with dual-element thermistors to measure their respective temperatures. Gauge and differential Omega pressure sensors were installed to measure the static pressure within the heat exchanger and the differential pressure across the heat exchanger, respectively.

TDA used Automation Direct Terminator IO hardware with the Entivity Studio Version 7 software to acquire data and control the experiment. The software controlled the cartridge heater to achieve the desired water inlet temperature. All channels were sampled sequentially within a 5 msec window at a 2 Hz data acquisition rate. The thermocouple data were output in engineering units with 16-bit accuracy. The data were used to calculate the heat transfer rate and identify the conditions at which the heat exchanger flow channels began to freeze. The pressure drop across the heat exchanger increased at low loads or when thawing a partially frozen heat exchanger, since the build up of ice along the fins caused blockage of the flow.

TEST RESULTS AND DISCUSSION

Experiments were conducted to demonstrate the ability to throttle the heat transfer rate by freezing the water flow channels within the heat exchanger. Four heat exchangers (HX) were built, assembled and tested to explore the effects of surface area and shell conductivity on ice build up within the heat exchanger and the overall heat transfer performance. The design parameters for each heat exchanger are reported in Table 1. The fins were fabricated from carbon composites with highly conductive K-1100 fibers. The fin assemblies were installed into 19-mm-diameter tubes. The internal flow area was large enough to have no more than a 13,800 Pa (2 psi) pressure drop at 90 kg/h (200 lbm/h).

Table 1. Definition of the heat exchanger design parameters.

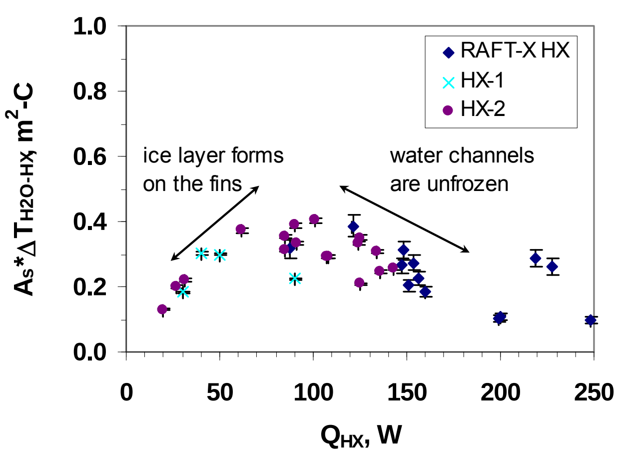
HX	As_o, m^2	k_{fin}, W/m-C @ 20°C	Shell Mat'l	k_{shell}, W/m-C @ 20°C
RAFT-X	.0162	380	Cu	380
1	.0168	300	SS304	16
2	.0184	300	Al6061	171
3	.0184	800	Al6061	171
4	.0189	800	Al6061	171

In Figure 9 the heat transfer performance of the new heat exchangers is compared to the RAFT-X heat exchanger. All of the new heat exchangers could be throttled to low heat transfer rate and yet, quickly respond to much higher loads when the water flow rate was increased. The graphs in Figure 9 show the unique behavior of the heat exchangers. At high heat loads, which are obtained in the experiments by flowing water at rates up to 90 kg/h, the heat exchangers were thawed and the conductive fins transferred the heat load to the shell and into the bath. As the water flow rate was lowered and the heat exchanger structure was chilled by the bath, then the ΔT between the water and the HX wall increased. This increases the As*ΔT term even as the heat transfer rate drops (follow the 'right' portion of the curve). However, when the heat exchanger wall temperature fell below the freeze point of water, then ice built up to decrease the surface area. Equation 1 was used to estimate the reduced surface area. The reduced heat exchanger effectiveness forced the temperature difference between the water and layer of ice to remain nearly constant. Therefore, the As*ΔT term became smaller, until the heat exchanger equilibrated the rate of heat rejection to the input load for the specific set of experimental conditions ($\dot{m}_{H2O}$, T_{H2O} and T_{sink}).

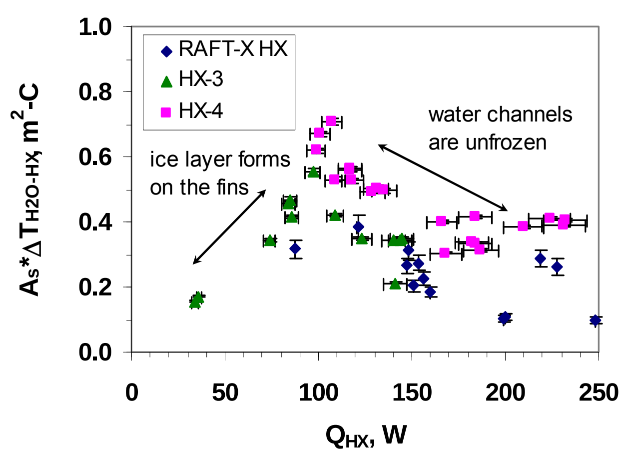
In Figure 10 the heat transfer rate was plotted against the overall heat transfer coefficient for the heat exchangers. The overall heat transfer coefficient for the heat exchanger falls off dramatically as the heat exchanger freezes, which should be expected since the thermal conductivity of ice is only 2 W/m-C. (This is more than two orders of magnitude less than the conductivity of the fins.)

The figure also shows that the maximum attainable heat transfer rate increases with increased surface area (i.e. more fins) within the heat exchanger. The HX-1, HX-2 and HX-3 had less ability to transport heat than the RAFT-X, but could operate at heat transfer rates as low as 20W and then thaw to reject heat at rates as high as 160W.

The best heat exchanger (HX-4) had six carbon composite fins down the entire length of the heat exchanger. Its maximum heat transfer rate during the experiments was 230 W (800 Btu/h), which was nearly equal to that of the RAFT-X. A very similar heat exchanger, HX-3, was stably operated at extremely low heat loads (35 W) without fully freezing. Regrettably, the HX-4 heat exchanger was not also tested at low load conditions. However, the combined performance of these two heat exchangers suggests that a turndown ratio of nearly 7:1 in heat transfer rate may be obtained, which will allow future space suit radiators to maintain astronaut comfort regardless of the work load.



a) Results for HX-1 and HX-2.



b) Results for HX-3 and HX-4.

Figure 9. Heat exchanger operation from low to high heat loads. The build up of ice on the fins reduces the surface area and lowers the rate of heat transfer.

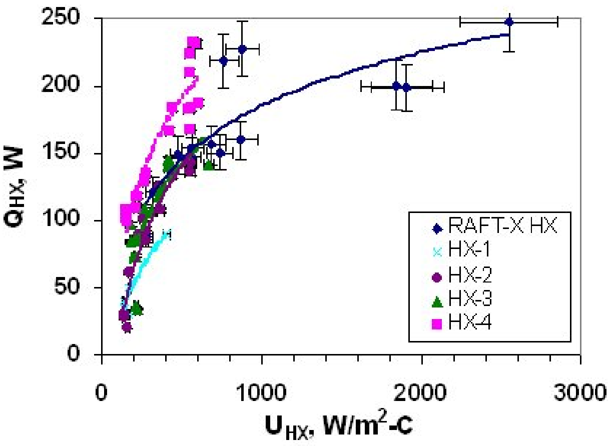


Figure 10. The relationship of the heat transfer rate to the empirically determined overall heat transfer coefficient for the heat exchanger shell, fins and ice (if present).

IMPACT TO SPACE SUIT RADIATORS

The RAFT-X mass was 8.62 kg (19.0 lbm), since it used the radiator for both heat rejection and impact protection. Improvements can lower the system mass to 6.17 kg (13.6 lbm). However, the mass of the radiator system would fall to 2.9 kg (6.4 lbm) if it only had to reject heat. With these data a system engineering analysis was revisited to determine the impact on the PLSS weight of Space Shuttle EVA and Constellation Space Suit Systems (CSSS).

The Space Shuttle EVA Space Suit supports the PLSS components within a frame and covers them with a shell able to provide impact protection. As already mentioned, in addition to rejecting heat to the environment, the RAFT-X radiator is also a rigid exoskeletal shell that can support equipment and protect it from impacts. An estimated 4.5 kg (9 lbm) can be saved, before also taking credit for the reduction in expendables, by replacing the internal support structure, the EMU thermal protection garment and the covers for the secondary oxygen pack, water tank and upper shield with a radiator.

An advanced NASA concept for the Constellation Space Suit System has a hard impact resistant shell over a thick protective layer of foam. [9] In this case replacing the 1.86 kg (4.1 lbm) hard shell with a lightweight radiator system designed only for heat rejection will only increase the mass by about 1.04 kg (2.3 lbm). For both space suit concepts a radiator can allow the EVA mission to be performed with substantially less water. Thus, up to an additional 2.7 kg (6 lbm) could be removed from the PLSS units. With that being considered, an advanced radiator offers a net savings to both Space Suit Systems, while providing an effective, reliable and fully redundant heat rejection system to maintain astronaut comfort during EVAs.

CONCLUSIONS

A freezable heat exchanger was developed that can adjust its heat transfer effectiveness in order to match the heat transfer rate to the radiator with that from the astronaut. Analyses and experiments show that 0.019 m^2 of surface area were needed to reject high heat loads. In laboratory experiments the new heat exchangers were able to passively regulate heat transfer up and down a wide range of heat loads (from as little as 35 W and up to 230 W) without fully freezing the water channels.

ACKNOWLEDGMENTS

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DEFINITIONS, ACRONYMS AND ABBREVIATIONS

ACRONYMS

EVA: Extravehicular Activity

HX: Heat Exchanger

IO: Input Output

JSC: Johnson Space Center

LCG: Liquid Cooling Garment

NASA: National Aeronautics and Space Administration

PLSS: Portable Life Support System

RAFT-X: Radiator and Freeze Tolerant Heat Exchanger

NOMENCLATURE

As	surface area, m^2
C_p	specific heat, J/kg-C
k	thermal conductivity, W/m-C
N_{FINS}	number of fins
$\dot{m}$	mass flowrate, kg/h
P	pressure, PA
Q	heat rejection rate or power, W
r	radius, mm
ΔT	temperature difference, °C
T	temperature, °C
U_{HX}	overall heat transfer coefficient, W/m^2-C
v	water flow velocity, m/s
x or z	axial location, mm

Subscripts

FPt	freeze point
o	initial
w_o	outer wall