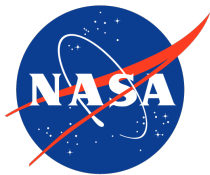


Long-Term Cryogenic Propellant Storage on Mars with Hercules Propellant Storage Facility

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This report details the process and results of roughly sizing the steady state, zero boil-off thermal and power parameters of the Hercules Propellant Storage Facility. For power analysis, isothermal and isobaric common bulkhead tank scenarios are considered. An estimated minimum power requirement of 8.3 kW for the Reverse Turbo-Brayton Cryocooler is calculated. Heat rejection concerns in soft vacuum Mars atmosphere are noted and potential solutions are proposed. Choice of coolant for liquid propellant conditioning and issues with current proposed cryocooler cycle are addressed; recommendations are made, e.g. adding a Joule-Thomson expansion valve after the Reverse Turbo-Brayton turbine in order to have two-phase, isothermal heat exchange through the Broad Area Cooling system. Issues with cross-country transfer lines from propellant storage to flight vehicle are briefly discussed: traditional vacuum jacketed lines are implausible, and Mars insulation needs to be developed.

Nomenclature

Acronyms

<i>BAC</i>	=	Broad Area Cooling
<i>CBH</i>	=	Common Bulkhead
<i>LCH₄</i>	=	Liquid Methane
<i>LCI</i>	=	Layered Composite Insulation
<i>LOX</i>	=	Liquid Oxygen
<i>HPSF</i>	=	Hercules Propellant Storage Facility
<i>HSRV</i>	=	Hercules Single-Stage Reusable Vehicle
<i>ISRU</i>	=	<i>In-situ</i> Resource Utilization
<i>IPPF</i>	=	<i>In-situ</i> Propellant Production Facility
<i>J-T</i>	=	Joule-Thomson
<i>MPCF</i>	=	Mobile Propellant Conditioning Facility
<i>SOFI</i>	=	Spray On Foam Insulation
<i>RTBC</i>	=	Reverse Turbo-Brayton Cryocooler
<i>VJ</i>	=	Vacuum-Jacketed

Symbols

A_s	=	surface area, m^2
ϵ	=	emissivity
f	=	view factor
h	=	enthalpy, J/g
k	=	coefficient of thermal conductivity, $mW/m - K$
L	=	thickness, m
\dot{m}	=	mass flow rate, g/s
P	=	pressure, $psia$
\dot{Q}	=	rate of heat transfer, W
q''	=	heat flux, W/m^2
σ	=	Stefan-Boltzmann constant, $W/m - K^4$
T	=	temperature, K

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I. Introduction

THE Hercules Propellant Storage Facility (HPSF) is a non-reusable variant of the Hercules Single-Stage Reusable Vehicle (HSRV) and consists of one common bulkhead (CBH) ascent tank, two liquid methane (LCH₄) descent tanks, and two liquid oxygen (LOX) descent tanks, all with broad area cooling (BAC) capability using a Reverse Turbo-Brayton cryocooler (RTBC) on a Mobile Propellant Conditioning Facility (MPCF).¹ Tanks are exclusively used for propellant storage after emplacement, before the propellant is ultimately pumped into the flight vehicle, HSRV, shortly before liftoff. A rough schematic and cross-section of the HPSF in Fig. 1.

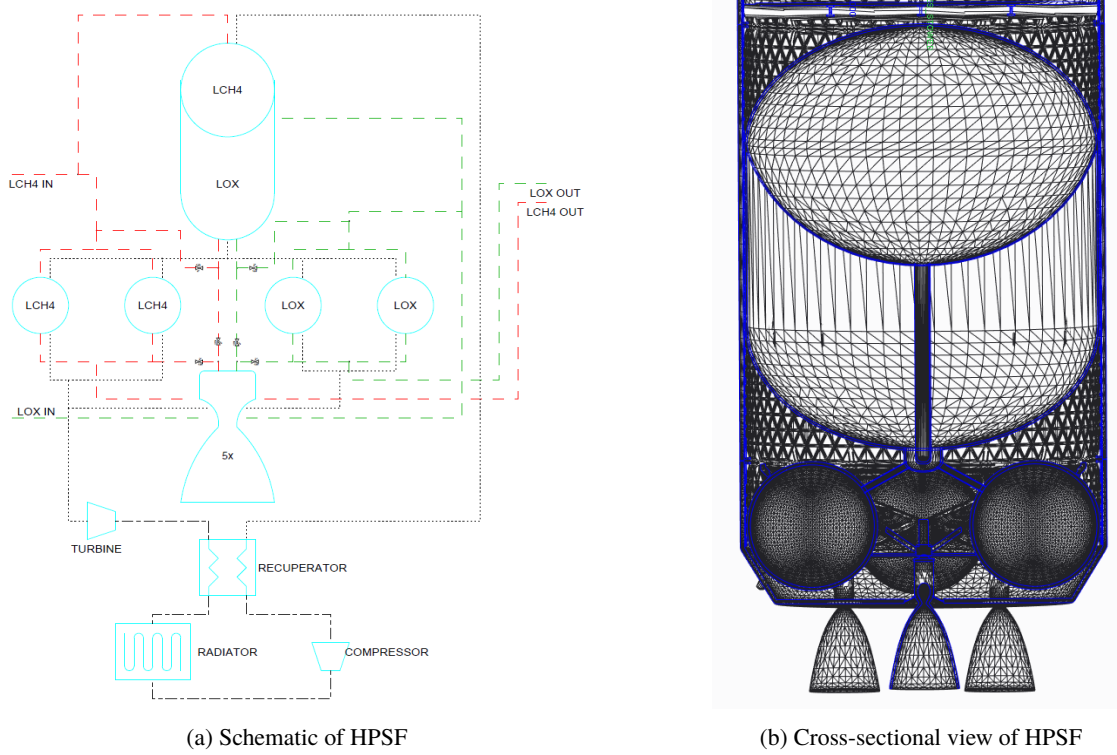


Figure 1: HPSF Drawings

II. Thermal Analysis Approach

In order to estimate the necessary refrigeration power, the heat input into the HPSF system must be determined. Mars atmosphere is assumed to be at its peak temperature of 290 K (i.e. worst case scenario) and standard pressure at 7 torr (0.14 psia). All liquids are assumed to be isothermal, at steady state, and at saturation temperature. Tank wall thermal resistance is assumed to be negligible, i.e. added insulation provides the only thermal resistance. Regolith (Mars surface) heat radiation emission is assumed to be negligible. Structures are assumed to be designed to withstand sufficient change in pressure, or ΔP . All tank exteriors are assumed to be insulated with Layered Composite Insulation (LCI); LCI is assumed to have an effective thermal conductivity of 2 mW/m-K with a thickness of 0.022 m at 7 torr (0.14 psia).² The CBH is assumed to be insulated with Spray On Foam Insulation (SOFI) with an effective thermal conductivity of 11 mW/m-K and a thickness of 0.025 m at pressures higher than 1 torr (0.02 psia).²

Using Fourier's law of thermal conduction, the formula for effective heat rate can be written as

$$\dot{Q} = \frac{k_e A_s}{L_{ins}} \Delta T \quad (1)$$

where k_e is the effective thermal conductivity of the insulation, A_s is the surface area conducting, L_{ins} is the thickness of the insulation. Effective heat rate uses an empirical thermal conductivity coefficient value which accounts for solid conduction, gas conduction, convection, and radiation, but does not distinguish them. The formula for heat flux can then be solved for by dividing both sides of the equation by A_s , obtaining

$$q'' = \frac{k_e}{L_{ins}} \Delta T \quad (2)$$

A. Descent Tanks Analysis Results

The spherical descent tanks' heat fluxes can be easily solved using Equation (2); there is no CBH, resulting in tanks that are both isothermal and isobaric. The tanks are assumed to be at 30 psi; pressure values will be in English units for convenience, as they will not be used in the calculations. Thus, the LOX and LCH₄ are at their saturation temperatures of 97 K and 121 K, respectively. Using a hot temperature of $T_{inf} = 290K$, the respective temperatures of the liquids, and insulation material of LCI, we can determine that heat is transferred to the LOX descent tanks at $17.5 W/m^2$ and the CH₄ descent tanks at $15.4 W/m^2$.

B. Ascent Tank Analysis Results

There are two discrete storage cases for the ascent CBH tank due to the differing thermodynamic properties of LOX and LCH₄. If we require that the entire CBH tank is isothermal, it cannot be isobaric, and vice versa. Thus, both isothermal and isobaric cases will be analyzed.

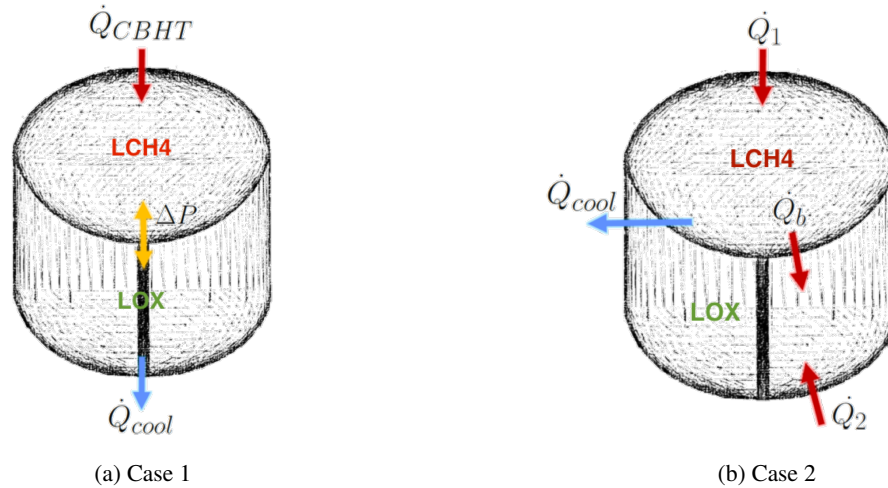


Figure 2: Heat Transfer of CBH Tank

C. Case 1 – Isothermal (i.e. no heat transfer through CBH)

If we assume the temperature throughout the entire CBH tank is constant, the saturation pressures of the liquids will differ. Thus, there will be a ΔP across the CBH, as shown in Fig. 2a. Due to the shape of the CBH, the lower pressure liquid should be stored in the aft tank and the higher pressure liquid should be stored in the forward tank, such that the CBH is under tension. However, it is assumed that LOX is stored in the aft tank and LCH₄ is stored in the forward tank, and that the CBH is structurally capable of handling the compressive pressure. This may not be feasible in practice; ergo, the orientation of the propellant tanks should be open to discussion.

At a saturation temperature of 91.2 K, the propellants experience the lowest ΔP possible of 14.5 psia – the subcooled methane will freeze if the temperature goes any lower. The ΔP rises quickly as the saturation temperature rises; hence, we will assume the isothermal tank temperature of 91.2 K, because 14.5 psia is already a large ΔP for a CBH. As a result, the LOX will be at 16.3 psia and the LCH₄ will be at 1.8 psia. Although the large ΔP will most likely require a strengthened LCH₄ tank, this configuration simplifies refrigeration; BAC can be used uniformly around the tank. In turn, this simplifies BAC control. From Eq. (2), we can find that the total heat flux of the ascent tank from the environment is $q''_{case1} = 18.1 W/m^2$.

D. Case 2 – Isobaric (i.e. no ΔP across CBH)

If we assume that there is no ΔP across the CBH, the saturation temperatures of the liquids will differ. This indicates that there will be heat transfer across the CBH (shown in Fig. 2b) which must be taken into account when designing the BAC system. Because the LOX is colder than the LCH₄, the insulation on the LCH₄ tank may be designed such that the forward tank will not require any refrigeration (i.e. $\dot{Q}_1 = \dot{Q}_b$). For this calculation we will be adhering to the initial assumptions of the insulation thicknesses.

At 30 psia, the saturation temperatures of the LOX and LCH₄ are 97 K and 121 K, respectively. From Eq. (2), we can find that q''_1 is $15.4 W/m^2$, q''_b is $10.6 W/m^2$, and q''_2 is $17.5 W/m^2$. There is a significant heat flux passing through the CBH, which will require a more complicated BAC system design to maintain zero boil-off.

III. Power Analysis

Table 1: Descent Tank Parameters

Descent							
Component	$T_{sat}(K)$	$P(psia)$	Insulation	$k_e(\frac{mW}{m \cdot K})$	$q''(\frac{W}{m^2})$	$A_s(m^2)$	$\dot{Q}_{total}(kW)$
LOX tank (2x)	97	30	LCI (22 mm)	2	17.5	11.5	0.4
LCH ₄ tank (2x)	121	30	LCI (22 mm)	2	15.4	12.1	0.4
Total						23.6	0.8

Table 2: Ascent Tank Parameters – Isothermal

Ascent Case 1							
Component	$T_{sat}(K)$	$P(psia)$	Insulation	$k_e(\frac{mW}{m \cdot K})$	$q''(\frac{W}{m^2})$	$A_s(m^2)$	$\dot{Q}_{total}(kW)$
LOX tank	91.2	16.3	LCI (22 mm)	2	18.1	122.5	2.2
LCH ₄ tank	91.2	1.8	LCI (22 mm)	2	18.1	39.3	0.7
CBH	-	-	SOFI (25 mm)	11	0	35.1	0
Net						161.8	2.9

Table 3: Ascent Tank Parameters – Isobaric

Ascent Case 2							
Component	$T_{sat}(K)$	$P(psia)$	Insulation	$k_e(\frac{mW}{m \cdot K})$	$q''(\frac{W}{m^2})$	$A_s(m^2)$	$\dot{Q}_{total}(kW)$
LOX tank	97	30	LCI (22 mm)	2	17.5	122.5	2.1
LCH ₄ tank	121	30	LCI (22 mm)	2	15.4	39.3	0.6
CBH	-	-	SOFI (25 mm)	11	10.6	35.1	0.4
Net						161.8	2.7

A. HPSF Case 1 Power Requirement Estimate

From Tables 1 and 2, we can determine the total electrical power necessary to provide refrigeration to the HPSF for Case 1. This is simply done by summing the total \dot{Q} of the ascent and descent sections. We obtain a total heat transfer rate from the environment of 3.7 kW. In a perfect universe, we would need 3.7 kW of power to reject a heat load of 3.7 kW. Unfortunately, in reality, the maximum ideal efficiency of a refrigerator is limited by Carnot's equation,

$$\eta_{carnot} = \frac{T_{cold}}{T_{hot} - T_{cold}} = \frac{\dot{W}_{refrigeration}}{\dot{W}_{electrical}} \quad (3)$$

where η_{carnot} is the ideal Carnot refrigerator efficiency, T_{cold} is the cold temperature (in this case, the temperature of the liquid), T_{hot} is the hot temperature (in this case, the temperature of the environment), $\dot{W}_{refrigeration}$ is heat removed at T_{cold} (i.e. the lift), and $\dot{W}_{electrical}$ is the total electrical power input to the refrigerator. Thus, for Case 1, using $T_{cold} = 91.2K$ and $T_{hot} = 290K$, we obtain $\eta_1 = 0.46$. Rearranging Eq. (3), the ideal power input requirement would be

$$\dot{W}_{electrical} = \frac{\dot{W}_{refrigeration}}{\eta_{carnot}} \quad (4)$$

which we can solve for minimum electrical input needed to refrigerate (i.e. if the cryocooler runs at 100% efficiency) as $\dot{W}_{ideal,1} = 8.0kW$. In reality, there is no practical cryocooler that can operate at 100% efficiency, so we assume an efficiency of $\eta_{cryocooler}$, yielding:

$$\dot{W}_{electrical} = \frac{\dot{W}_{refrigeration}}{\eta_{carnot} \cdot \eta_{cryocooler}} \quad (5)$$

For example, we can assume a cryocooler efficiency of 70% and use $\eta_{cryocooler} = 0.7$ in Eq. (5). As a result, in the isothermal case, to maintain liquid temperatures and achieve zero boil-off, we need to provide approximately 11.5 kW of electrical power to the RTBC.

B. HPSF Case 2 Power Requirement Estimate

From Tables 1 and 3, we can determine the total power necessary to refrigerate the HPSF for Case 2 with the same process as Case 1. We obtain a total heat transfer rate from the environment of 3.5 kW. T_{cold} is estimated as the average of the two saturation temperatures of the two propellants at 30 psia, or 109 K. T_{hot} remains the same, and using Eq. (3), we obtain an ideal Carnot efficiency of $\eta_2 = 0.60$. Assuming $\eta_{cryocooler} = 0.7$ and using Eq. (5), to maintain liquid temperatures and achieve zero boil-off, we need to provide approximately 8.3 kW of power to the RTBC for an isobaric ascent tank. The lower power requirement of the isobaric case is primarily due to the higher temperatures in the tank, which is allowable because $\Delta P = 0$.

C. Outcome

The above calculations are done assuming the atmosphere stays constant at a maximum of 290 K, thus solving for maximum required power, but not accounting for changes of temperature throughout the day and night. The BAC will need to have a controller that responds to ambient temperature by regulating coolant mass flow rate – the design of the BAC is critical. If the heat rates are imbalanced, in order for tanks to maintain pressure, they will need some sort of non-condensable mass input (e.g. N_2 or He) to pressurize, or will eventually need to vent to depressurize. A tank will autogenously depressurize if the net heat rate is negative (i.e. more heat going out than coming in), and thus require an inert mass input. Conversely, if the net heat rate is positive (i.e. more heat coming in than going out), a tank will pressurize and require venting, which wastes valuable propellant. The resupply or production of this pressurization gas will be costly – whether monetarily (helium resupply from Earth) or power-wise (nitrogen production *in-situ*). These factors add complexity to the design of the BAC.

The isothermal ascent tank requires more power to cool, but with the advantage of simple BAC design for both LOX and LCH₄. The main disadvantage is that the pressures of both tanks must be kept very low in order to minimize ΔP – even at the minimum temperature, the ΔP is fairly high. This may become an issue when transferring propellants, but the main concern is that if one of the tanks is not being cooled sufficiently, the pressure could rise and failure could be catastrophic. This concern can be mitigated with a fail-safe venting capability, which would probably be required regardless.

The isobaric ascent tank has the advantage of zero ΔP across the CBH, and has the potential for a non-refrigerated LCH₄ tank, assuming an insulation could be designed such that heat rate coming in from the environment equals heat rate going into the LOX tank through the CBH. The disadvantage is that the BAC controller and design would need to be more complex, due to the different heat rejection temperatures.

A sensitivity analysis regarding the relationship between cryocooler efficiency and electrical power requirement was performed and can be seen in Fig. 3. As cryocooler efficiency decreases, electrical power requirement increases exponentially. Thus, a recommended minimum cryocooler efficiency is around 60%, approximately where the line begins to converge to the ideal Carnot electrical power requirement.

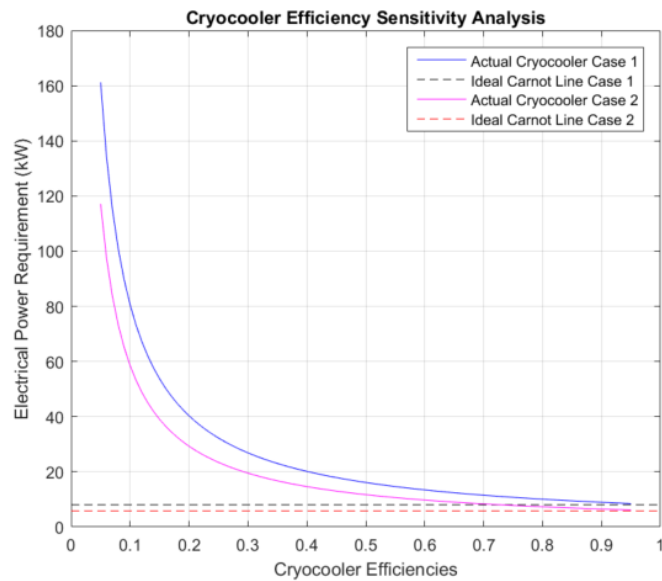


Figure 3: Cryocooler Efficiency Sensitivity Analysis for Isothermal and Isobaric Cases

Table 4: Summary of Power Results

	Heat Load (kW)	Carnot Efficiency	Assumed Cryocooler Efficiency	Required Electrical Power (kW)
Case 1	3.7	0.46	0.7	11.5
Case 2	3.5	0.6	0.7	8.3

D. Heat Rejection

In a soft vacuum atmosphere like Mars, an underlying concern is heat rejection; natural convection is ineffective due to soft vacuum atmosphere and lower gravity.³ Consequently, radiators must be very large in order to adequately reject heat. Using Case 1 as an example, we will assume that we need to reject 11.5 kW of heat (disregarding safety factors and dust accumulation), and solve for the surface area of the radiator. Arbitrarily assuming⁴ an ideal recuperator (i.e. equal temperatures at each end), isentropic compression with ambient temperature inlet condition (i.e. maximum of 290 K) from 50 psia to 200 psia, emissivity of 0.75, view factor of 1 (unobstructed view of atmosphere), black body atmosphere, and negligible convection, the ideal surface area of the radiator can be calculated with

$$\dot{Q}_{rad} = \sigma \epsilon f A_s (T_{rad}^4 - T_{inf}^4) \quad (6)$$

where \dot{Q}_{rad} is the heat transfer rate (11.5 kW), σ is the Stefan-Boltzmann constant of $5.67 \cdot 10^{-8} \frac{W}{m^2 K^4}$, ϵ is the emissivity of the radiator (0.75), f is the view factor (1), A_s is the surface area of the radiator (unknown), T_{rad} is the temperature of the radiator (depends on initial condition assumptions), and T_{inf} is the temperature being radiated to (290 K). After finding entropy at the compressor inlet, using helium as the working fluid, we can solve for the outlet temperature of around 500 K, which we will use as the radiator inlet temperature. As an approximation, we can use the average temperature between 500 K and 290 K as T_{rad} . Rearranging Eq. (6),

$$A_s = \frac{\dot{Q}_{rad}}{\sigma \epsilon f (T_{rad}^4 - T_{inf}^4)} \quad (7)$$

For 11.5 kW of required heat rejection, we find, from Eq. (7), that the estimated required surface area is $15.1m^2$. This is *only* for the cryocooler to achieve zero boil-off in the HPSF – for the entire end-to-end ISRU system, the required radiator surface area would need to be significantly larger. One might expect that this is worst case scenario, as the ambient temperature was assumed to be maximum. In fact, as the ambient temperature decreases, the required surface area of the radiator actually *increases* exponentially, because the ΔT across the isentropic compressor becomes smaller (see Fig. 4). This is due to the assumption that the compressor inlet temperature is equal to ambient (i.e. an ideal recuperator), and the use of helium as the working fluid. A larger surface area requirement can be circumvented by designing the compressor such that the outlet temperature difference to the atmosphere is maximized.

From the results, we can infer that rejecting all of the heat through radiation is impractical, because transporting all of the material to Mars would simply cost too much. There could be a wavelength window within the Martian atmosphere to radiate to deep space (3 K), but such a radiator needs to be researched and developed. Thus, we will need alternate means of heat rejection in addition to large radiators to ensure system components do not overheat. One method could be to dump the heat into icy regolith, which could also provide water for the propellant production facility. However, avoiding heat soak by keeping the heat exchangers in contact with icy regolith after it has melted needs to be addressed, and there would need to be some way to move the heat exchangers or massive amounts of regolith. While some kind of autonomous excavator could be employed, this solution is not self-sustainable (the machine will most likely require a lot of maintenance) and adds a significant amount of complexity to the architecture. Thus, the problem of heat rejection on Mars needs to be further researched to find an affordable and efficient solution.

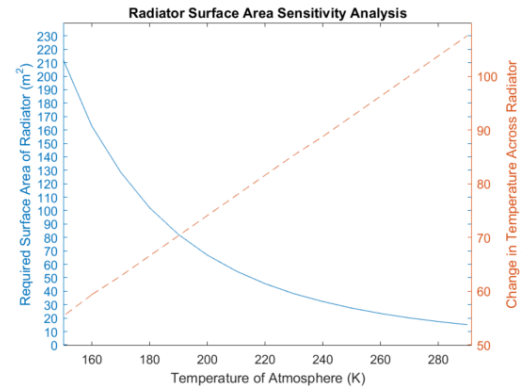


Figure 4: Radiative Surface Area Required and ΔT Across Radiator vs Ambient Temperature – Helium Refrigerant

IV. Cryocooler Analysis

This section will analyze the various ways to design the BAC system responsible for long-term propellant storage. At this point of the Hercules project, an RTBC has been assumed to condition the propellants. Case 1 (isothermal ascent tank) parameters will be assumed for simplicity.

One of the most important steps of designing a BAC system is choosing the coolant; the coolant dictates the required mass flow rates, the minimum temperature allowed, and the overall efficiency of the system. On Earth, choosing a coolant is fairly easy due to readily available elements and methods of extraction, but on Mars, elements are much more scarce. Although the refrigeration system is closed and filled on Earth prior to launch (very risky to charge refrigerant on Mars), it may require make-up gas to account for leaks over time. If a coolant cannot be produced on Mars via *in-situ* resource utilization (ISRU), it must be transported from Earth, which at the moment is extremely costly. However, this should not drive the decision on which working fluid to use; the cryocooler is of vital mission importance, and correspondingly, reliability and simplicity are more critical factors.

The products of the *In-situ* Propellant Production Facility (IPPF) proposed are only nitrogen, argon, methane, hydrogen, and oxygen using the atmosphere and water underneath the regolith. Methane and oxygen are used as the propellants that need to be cooled and thus cannot be used as coolant. Additionally, methane is a fuel, and if combined with the oxygen and a stray spark, catastrophic failure would occur. Hydrogen could be used as well, but is also a fuel. Argon is inert, but its boiling point is relatively high, and as a vapor, it would not be able to sufficiently cool the propellants. This leaves nitrogen, which is advantageous as a coolant because of its availability on Mars, relatively low boiling point, and inertness.

Unfortunately, upon further analysis, nitrogen would not be an ideal coolant for a RTBC. To avoid condensation within the turbine of the RTBC, the temperature of nitrogen at the outlet of the turbine would need to be greater than but close to the saturation temperature. Thus, the lowest temperature that can be supplied to the BAC must be the saturated vapor temperature. NIST REFPROP⁵ was used to find the following fluid properties. In a BAC system, a high pressure is required to maintain adequate mass flow rate throughout the HPSF. If we assume a maximum allowable supply temperature of 90 K, just under the normal boiling point of LOX, the maximum pressure at the turbine outlet is 50 psi, which is not very high. Assuming a maximum allowable return temperature of 111 K, just under the normal boiling point of LCH₄, we can find enthalpies of 84.7 J/g supply and 110.1 J/g return. The cryocooler lift can be expressed as

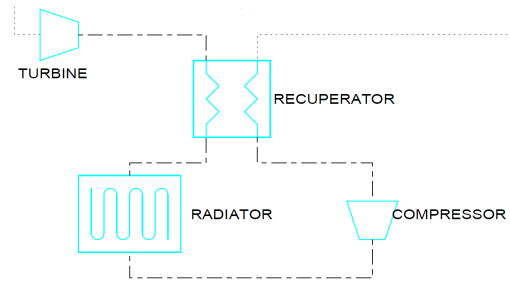


Figure 5: RTBC schematic

$$\dot{Q}_{lift} = \dot{m}(h_{return} - h_{supply}) \quad (8)$$

where \dot{m} is the mass flow rate of the coolant, h_{return} is the enthalpy of the coolant returning to the cryocooler (into the recuperator), and h_{supply} is the enthalpy of the coolant supplied by the cryocooler (out of the turbine). The lift required is 3.7 kW, calculated in the previous thermal analysis section. Hence, we can solve for required mass flow rate, which comes out to be 146 g/s. A mass flow rate of 146 g/s with a pressure of 50 psia is questionable: BAC systems have only been tested on mass flow rates that are lower by one or two orders of magnitude.⁶ Moreover, the larger the mass flow rate, the larger diameter lines needed and/or the higher pressures needed, which means more cost of material and power. Even when we lower the supply pressure of nitrogen to 10 psi, the resulting mass flow rate requirement would still be high at 94 g/s.

A. Recommendations

Fortunately, there are a few alternatives. Neon can be used to slightly reduce the mass flow rate requirement, but a helium coolant is superior, as its Δh (change in enthalpy) is largely unaffected by change in pressure. For example, at a pressure and temperature of 1000 psia and 77 K, respectively, the Δh is 179.9 J/g, and at a pressure and temperature of 10 psia and 77 K, respectively, the Δh is 176.6 J/g. From Eq. (8), we can find that the required mass flow rate at 1000 psia is 20.6 g/s, and at 10 psia is 21 g/s. Little dependence on pressure means that the design of the turbine can be very flexible, which in turn suggests that coolant line diameters can be very versatile. In addition, the normal boiling point of helium is around 4 K, so the turbine outlet temperature is also very adjustable. However, helium cannot be produced on Mars; it must be resupplied by Earth and, as a result, is a very expensive resource. This is a critical disadvantage, as the Hercules spaceport must be self-sufficient and Earth-independent. Helium tends to leak out of wherever it is being stored, so the gas would probably need to be resupplied every synodic cycle (about 26 months), if not more – emergency resupplies are out of the question.

Another option is to modify the cycle the cryocooler runs on. The RTB cycle is limited to vapor coolant by the turbine, unless the turbine is designed for two-phase flow, which increases loads throughout the turbine. Higher loads means a turbine that is more complex, more expensive, and requiring more maintenance. A more straightforward option is adding a J-T (Joule-Thomson) valve, i.e. an isenthalpic expansion valve, after the turbine. Because the J-T valve only relies on increasing the coolant's potential energy through the expansion of non-ideal gas (thus decreasing kinetic energy and temperature), the flow output can be two-phase and, correspondingly, is not limited to a minimum of saturation temperature.⁷ However, because the flow is two-phase, heat exchange occurs isothermally at the saturation temperature of the coolant; any heat transferred to the coolant goes into evaporating the liquid and not increasing the temperature of the gas. Isothermal heat exchange indicates a simpler heat exchanger design, at least until the flow becomes completely gaseous and begins gaining temperature as it cools the propellant. If a J-T valve is used after the turbine, nitrogen or even argon can be used as a coolant, both of which can be extracted from the Martian atmosphere. Furthermore, nitrogen and argon are easier to compress than helium, driving up the efficiency of the cycle. In fact, because the temperatures required are not too low, an RTB turbine may not even be required, as it adds complexity to design, maintenance, etc. A J-T cycle, shown in Fig. 6 would be much simpler; the cycle is effectively just a RTB cycle with an expansion valve instead of a turbine. The advantages are clear: ability to use coolants produced *in-situ*, easier to maintain, less complex design, and isothermal heat exchange. The disadvantage is that because the isenthalpic expansion valve is unable to extract work from the coolant like a turbine can, a J-T cycle may require some work input to reach the inversion temperature of the coolant before becoming functional. Otherwise, the expansion valve will actually heat up the refrigerant. Fortunately, the temperatures we are working at are well below the inversion temperatures of most working fluids (greater than 600 K, e.g. nitrogen at 621 K) – although helium is the lowest at 45 K.⁸ Regardless, we are assuming that liquefaction is done elsewhere (i.e. the IPPF), so the work input to jump start the J-T cycle is already taken care of. Hence, there are no major disadvantages of a J-T cycle compared to an RTB cycle, so a J-T cycle cryocooler is highly recommended. If more cooling power is needed, the modified RTBC with the J-T valve can be used.

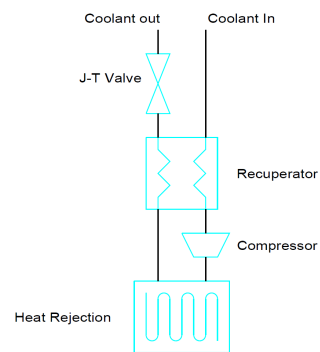


Figure 6: J-T Cycle Schematic

V. Transfer Line Sizing Considerations

The transfer of propellant from HPSF to HSRV is a challenging task. Before the lines can be sized, a number of factors must be determined – principally, insulation type: vacuum-jacketed (VJ), bare, etc. The storage and conditioning facilities must be far enough away to be protected from launch acoustic loads and debris, but close enough and/or insulated enough such that the propellants do not gasify. At the moment, a 1 km distance is assumed – this is a very large distance, given the lines should be installed autonomously. While VJ piping seems ideal for this application, there are many accompanying complications. VJ piping is very heavy, which noticeably increases Earth launch costs and limits distance between storage and vehicle. On Earth, hard VJ piping is installed in sections up to approximately 12 meters long, and either butt welded together in the field or mechanically joined together with bayonets.⁹ The bare weld joints are insulated with MLI and sealed with a welded vacuum can. This doubles the number of vacuum pump out ports required, and as such increases complexity, cost, and amount of maintenance, albeit increasing vacuum retention ability.⁹ On Mars, maintenance, particularly autonomous maintenance, is very difficult and expensive, and should be required as infrequently as possible. Additionally, autonomous emplacement/installation is also problematic with the amount of welding and long distances necessary; in fact, it is not realistic. Therefore, VJ piping cannot be implemented on Mars without reduction of both weight and complexity.

This leaves bare piping, which needs to be insulated. Unfortunately, some kind of lightweight insulation system needs to be developed prior to sizing the transfer lines, which in itself should be fairly simple: the two main parameters needed are mass flow rate and required pressure drop. The issue with soft vacuum is that all forms of heat transfer (solid/gas conduction, convection, and radiation) contribute to heat flow, whereas in high vacuum (e.g. the Moon), gas conduction and convection are eliminated. Although natural convection effect in soft vacuum is absent, local convective heat transfer within cells of thermal insulation systems can occur, which makes insulation in soft vacuum a demanding problem. This localized convection effect is very prominent; an aerogel composite blanket test under Mars atmospheric conditions (CO₂ and 5 torr) showed that heat leak was even worse than tests conducted under Earth atmospheric conditions (air at 760 torr).¹⁰ Before long distance propellant transfer is feasible, an effective cryogenic insulation system designed for Mars must be researched and developed. This insulation may also be used for the HPSF to reduce cryocooler power requirements.

VI. Conclusions

In summary, long-term cryogenic propellant storage on Mars poses many challenges, such as heat rejection, launch costs, and advanced insulation. Heat rejection needs to be addressed, as power requirements are high, and radiators will need to be appropriately large to amply reject heat to the environment. A possible method could be transferring the heat into the ground (ice and/or soil). With regards to keeping the cryogenic propellants liquid, the proposed RTBC could work with a helium coolant, but a possible alternative is using a J-T cycle, which replaces the RTB turbine with an expansion valve. This eliminates the need for one-phase vapor flow throughout BAC, and enables the use of coolants produced *in-situ* (e.g. nitrogen and argon). If there is a pressure differential across the CBH, the orientation of the CBH propellant tanks should be revisited; currently, the orientation applies a compressive pressure to the CBH, which is not ideal – CBHs are stronger under tension. Finally, the cross-country transfer lines from the HPSF to the HSRV need to be considered. VJ piping is not currently feasible on Mars, and current insulation techniques are not capable of handling a soft vacuum CO₂ atmosphere; more analysis and development is required.

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References

- ¹Komar, D. and Moses, R., “Hercules Single-Stage Reusable Vehicle Supporting a Safe, Affordable, and Sustainable Human Lunar and Mars Campaign,” AIAA 2017-5288, Presented at the AIAA Space Conference, Orlando, FL, 2017.
- ²“Standard Guide for Thermal Performance Testing of Cryogenic Insulation Systems,” Tech. rep., ASTM International, 2014.
- ³Hauser, D. M., Johnson, W. L., and Sutherlin, S. G., “Liquefaction and Storage of In-Situ Oxygen on the Surface of Mars,” Tech. rep., NASA Glenn Research Center, NASA Marshall Space Flight Center, 2016.
- ⁴Witte, L. C. and Hollingsworth, D. K., “Mars Radiator Characterization Experimental Program,” Tech. rep., University of Houston, 2004.
- ⁵“Reference Fluid Thermodynamic and Transport Properties Database (REFPROP),” .
- ⁶Guzik, M. and Tomsik, T., “An Active Broad Area Cooling Model of Cryogenic Propellant Tank With a Single Stage Reverse Turbo-Brayton Cycle Cryocooler,” Tech. rep., NASA Glenn Research Center, 2011.
- ⁷Ross, R., “Refrigeration Systems for Achieving Cryogenic Temperatures,” Tech. rep., Jet Propulsion Laboratory, 2016.
- ⁸Barron, R. F., *Cryogenic Systems*, Oxford University Press, 2nd ed., 1985.
- ⁹Johnson, W. L., Fesmire, J. E., and Meneghelli, B. E., “Cryopumping Field Join Can Testing,” Tech. rep., NASA Kennedy Space Center.
- ¹⁰Fesmire, J. E., Ancipink, J. B., Swanger, A. M., White, S., and Yarbrough, D., “Thermal Conductivity of Aerogel Blanket Insulation Under Cryogenic-Vacuum Conditions in Different Gas Environments,” Tech. rep., NASA Kennedy Space Center, Aspen Aerogels Inc, RD Services Inc, 2017.