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Advanced Refrigerator/Freezer Technology Development

Technology Assessment

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1.0 EXECUTIVE SUMMARY

To date, a limited number of R/F (Refrigerator/Freezer) systems have been developed for space applications. The Orbiter R/F (OR/F) and Life Science Laboratory Equipment (LSLE) systems have been developed for use on the Shuttle mid-deck and spacelab, respectively. The systems are similar in technology and design, with the LSLE system being approximately twice the storage volume of the OR/F system. Both systems are cooled by freon vapor-compression heat pumps. These systems are used to support Life and Biomedical Science missions (14 days or less) and are used in the temperature range of −20°C to +4°C. The systems have an expected life of a few hundred hours and are reconditioned between missions.

Future Life and Biomedical Science experiments will place a demanding set of requirements on the R/F systems. Considering the investment required to obtain the frozen test specimens, it is of utmost importance to have the specimens brought home intact. Future R/F requirements call for substantially longer life, higher reliability, less maintenance, and no CFC's. In addition, the size, mass, noise, and vibration are further limited.

In 1993, an R/F Technology Team composed of various NASA field center personnel, was assembled to qualitatively assess the state of R/F technology. The team found that the technologies required for future R/F systems do not exist at an adequate state of development and concluded that a technology development program is required to provide the advanced R/F technologies needed for future Life and Biomedical Science experiments.

An Advanced R/F Technology Development Project was initiated in December of 1994 at NASA LeRC. Oceaneering Space Systems (OSS) of Houston, Texas, was selected as the prime contractor to perform an advanced R/F technology assessment, development and demonstration under this project. The contract was split into two separate Phases. In Phase I, the contractor was to perform a quantitative technology assessment to identify and recommend the key advanced R/F technologies required for future Life and Biomedical Sciences spaceflight experiments. In Phase II, the NASA approved advanced technologies identified in Phase I are scheduled for development and demonstration.

This report documents the technology assessment activities performed in Phase I of the Advanced R/F Technology Development Project.

Requirements were assessed for five freezer classifications:

- −20°C Storage Freezer
- −70°C Storage Freezer
- −70°C Freeze Dryer
- −183°C Cryogenic Storage Freezer
- −196°C Cryogenic Quick/Snap Freezer

The requirements for these freezers were analyzed in detail at the system level for the following subsystems:

- coolers
- insulated enclosures
- thermal transport
- control electronics

Moisture control was also investigated, though without the rigorous trade studies or formal analysis.

A broad range of applicable technologies was surveyed and candidates narrowed first on the basis of their theoretical capabilities and demonstrated performance, then with more detailed parametric analysis of their capabilities. Characteristics like safety and technological maturity, which cannot be easily quantified, were factored in using a quality function deployment (QFD) analysis. This resulted in a set of candidates which were taken on to detailed, model-based study and are shown in Table 1 (Shading indicates advanced technologies with technology development required).

Coolers	Enclosure	Thermal Transport
Stirling Cycle	Rigidized Plastic MLI with plastic skins (R-105)	Heat Pipes
Turbo Brayton	Fiberglass with Metal Skins, Panel construction (R-30)	Thermal Pyrolytic Graphite (TPG) Conductor
Orifice Pulse Tube	Rigidized MLI, Box-in-Box construction (R-60)	Copper Conductor
Thermoelectric (−20°C freezer only)	MLI Dewar (R-2300) (cryogenic freezers only)	
Enhanced Efficiency Stirling		

TABLE 1 : Technology candidates considered in systems analyses

For each freezer classification, and for each feasible combination of the technologies shown in Table 1, a system was conceptualized which met the internal and external volume constraints and minimized the mass. The configuration was then compared to its mass and power specification to determine the margins by which it met or exceeded its requirements. Conclusions were drawn about the appropriateness of the technology combination for that freezer classification based on both the quantitative systems analysis and the QFD results. The final recommendations based on these results are summarized in Table 2 (shading indicates advanced technologies). As shown later in this report, it is possible to meet the system requirements for four of the five advanced freezer classifications (−70°C storage freezer excluded) without advanced technology: however, the performance of all five advanced freezers could be further improved with a few technology advances.

An enhanced efficiency Stirling cycle cooler would incorporate improvements to make the motor more efficient and more challenging modifications to replace the stainless steel pressure vessel with an insulating material to reduce the back heat leak. There is also the possibility of incorporating the thermal transport into the pressure vessel for more efficient heat transfer. The goal for this technology would be improving the efficiency of the cooler by 14% using the insulating pressure vessel.

Freezer Classification	Cooler	Enclosure	Thermal Transport
-20°C Storage (water heat rejection)	Thermoelectric	Rigidized Plastic MLI (Polymer Panel R-105)	TPG
-20°C Storage (air heat rejection)	Enhanced Stirling	Rigidized Plastic MLI (Polymer Panel R-105)	TPG
-70°C Storage	Enhanced Stirling	Rigidized Plastic MLI (Polymer Panel R-105)	TPG
-70°C Freeze Dryer	Stirling	Fiberglass with Metal Skin Panel (R-30)	Copper
-183°C Cryo. Storage	Stirling	MLI Dewar (R-2300)	Copper
-196°C Cryo. Quick/Snap	Enhanced Stirling	MLI Dewar (R-2300)	Copper

TABLE 2 : Recommended Technologies

The polymer panel consists of a layered vacuum support structure enclosed by plastic skins. It could potentially provide an insulated enclosure with an average insulation value of R-105, compared with the R-30 of existing technology. The challenge in this technology development is to find a skin plastic which is sufficiently rigid, insulative, and non-porous to hold a vacuum, and to bond it together effectively. The goal for this technology would be demonstrating an insulated enclosure with an R-value of 105.

Thermal Pyrolytic Graphite (TPG) is a passive thermal transport medium with a higher thermal conductivity and lower density to that of copper. TPG has never been used in this application. It is anisotropic, so designing to take best advantage of its properties requires further investigation. The goal for this technology is to maintain no more than 10°C temperature drop between the cold end of the heat exchanger and the air interface. This would represent a twofold reduction of the heat loss associated with thermal transport and should lead to approximately a 10% improvement in system efficiency.

Three design options in addition to the required systems analyses were considered in the study: reducing the external volume of the storage freezer to approximately half of the International Standard Payload Rack (ISPR) volume (0.7 m^3); incorporating the freeze dryer into the -70°C storage volume; and making the -70°C storage volume detachable from its cooler for transportation to the ground in a size which is compatible with the Occupational Safety and Health Administration (OSHA) single-man handling weight restriction. The latter emphasized the need for two further technologies: phase change materials to maintain the internal temperature of the freezer during transport, and a method and/or material which will permit quick disconnect of the transportable locker from the rest of the system without excessive heat leakage. A velvet-like material called brush carbon, which has high thermal conductivity only when mated, has some interesting promise as a thermal quick disconnect.

Moisture control is identified as a key issue throughout the range of freezer classifications. Some significant approaches, like use of desiccants and cold traps, are discussed. The goals for this technology are to accomplish all moisture control within the weight and power allowables for the system and to limit any maintenance time for moisture removal to ten minutes per month.

No technology development issues were uncovered for the system electronics. The study notes that vibration and noise for a Stirling cooler can be improved by a quasi-sinusoidal drive signal. Such control systems have been previously employed and no technology development is required.

The issue of noise and vibration control more generally has implications for many subsystems. Cooler designs must be dynamically balanced, perhaps with two compressors operating in opposed directions. Fans and ducting for thermal transport and heat rejection must be carefully engineered. However, heat rejection noise would cease to be an issue if water cooling were used.

Enhanced efficiency Stirling cycle coolers, polymer panel technology, Thermal Pyrolytic Graphite and moisture control were identified as requiring technology development by this study and have been recommended to be carried forward for development and demonstration in the second phase.

2.0 INTRODUCTION

2.1 Background

Having a reliable refrigerator/freezer (R/F) on orbit is key to preserving biological fluid and tissue samples obtained in space for later examination on the ground. Early R/F systems for both the Shuttle middeck locker, called the Orbiter Refrigerator/Freezer (OR/F), and for Space Lab, called the Life Sciences Laboratory Equipment (LSLE), used CFC refrigerants in a conventional Rankine (vapor compression) cycle. These units operated in a temperature range of −20°C to +4°C. Weight, power, size, materials, and performance specifications led to a four piston, two stage design, which produced significant noise. Because of refrigerant toxicity, the quantity of coolant had to be limited and the coolant loop had to be double contained, which reduced cooling efficiency. The need for acoustic noise reduction further diminished cooling efficiency because noise and heat are transferred through the same pathways. Furthermore, the ineffectiveness of lubrication systems in microgravity gave this family of coolers an expected lifetime on the order of a few hundred hours, requiring reconditioning after each use.

STS-60 flew a Stirling cycle OR/F (SOR/F) which demonstrated much lower levels of noise and vibration (Ref. 1) than previous systems but still failed to meet its acoustic specification. An enhanced Rankine cycle OR/F (EOR/F) with a linear compressor and HCFC refrigerant flew on IML-2. Although the coolant mechanisms performed satisfactorily, keeping frost accumulation at acceptable levels required significant amounts of crew time or compromised performance. Several thermoelectric refrigerator/freezers operating in the same temperature range have also been flown. Although their performance was adequate, the need for efficient defrost and low noise heat rejection continue to be challenging issues.

Unlike Shuttle missions which last only a few days or two weeks, missions to the International Space Station (ISS) will last months or even years. Future mission R/F requirements call for high reliability, low maintenance, and no CFC's.

To preclude loss of certain sensitive biological specimens like hormones, a requirement for relatively large volume, low temperature (−70 °C) storage has been identified. In addition, a cryogenic storage capability in the range of −183 °C will be needed to store cryo fixed samples.

In light of these mounting requirements and technical difficulties, NASA convened a panel of experts to assess the state of R/F technology. After a six-month investigation, the team found a lack of system level design and mission analysis in specifying orbital refrigerator/freezers and concluded "that the technologies needed for future R/F systems do not exist in an adequate state of development to provide the advanced R/F technologies required for future life and biomedical sciences spaceflight experiments (Ref. 1)." The panel recommended 1) a "bottoms up" analysis of the requirements and a survey of the technology to identify key technologies in the context of an integrated system and 2) development of those technologies into a state of readiness for use in future spaceflight.

2.2 Objective

The objective of this contract is to identify, develop, and demonstrate key advanced Refrigerator/Freezer technologies required for future Life and Biomedical Sciences spaceflight projects (Ref. 2). This report documents the first phase of the contract, the technology assessment. The objective of the first phase is to identify and recommend technologies which have the potential to allow different classes of future R/F systems to meet user requirements.

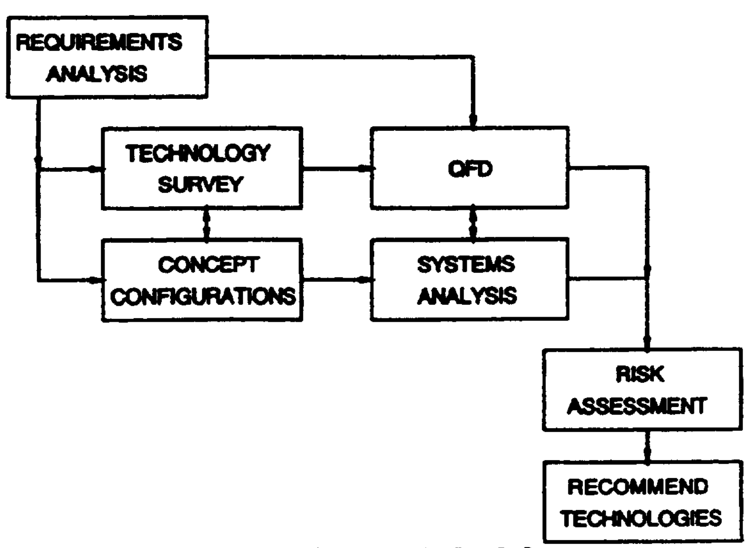


Figure 1: Study Methodology

2.3 Approach

The process of identifying key technologies begins with an understanding of the requirements. The Request for Proposal (RFP) specifies five freezer classifications to serve as the basis for the requirements analysis:

- $-20°C$ storage freezer capable of maintaining a specimen volume of 0.3 m^3 at not more than $-19°C$.
- $-70°C$ storage freezer capable of maintaining a specimen volume of 0.4 m^3 at not more than $-68°C$.
- $-70°C$ freeze dryer capable of sublimating frozen samples at a pressure of 10^{-4} torr and removing water at the rate of one liter per day.
- cryogenic storage freezer capable of maintaining a specimen volume of 0.02 m^3 below -183°C.
- cryogenic quick/snap freezer, capable of rapidly freezing specimens to $-196°$ C.

The RFP also defined a list of general requirements which apply to all the freezer classifications. These include things like electromagnetic compatibility and interference, noise and vibration, external environment of the freezers, power supply, and freezer lifetime. The contractor, Oceaneering Space Systems (OSS), reviewed each specification in detail. In some cases, it was necessary to derive or assume specifications beyond those given in the contract. Based on these general requirements, OSS derived performance and design specifications for each of the freezer classifications.

In conjunction with the requirements analysis, OSS performed a survey of the technologies in four categories:

- coolers
- enclosures
- thermal transport
- electronics

The survey began with a broad-based search for any prospective technologies. The list was then narrowed to only include those candidates which had a demonstrated capability to meet the performance requirements. This narrowing resulted in four candidates for the cooler subsystem, four for the enclosure, and three for the thermal transport. Scrutiny of the requirements did not reveal any technology development needs in the electronics, although simplifying existing space qualified electronics into a lower cost unit was considered desirable.

Each candidate technology was qualitatively assessed against the requirements using the so-called "L matrix" of the Quality Function Deployment (QFD) methodology. This is a two-dimensional matrix which lists qualitative specifications along the vertical axis, the so-called "Whats List", against the technology candidates on the horizontal axis, the so-called "Hows List". Each technology "how" was compared with its requirements "what" and given one of three ratings: excellent benefits, which carried a numerical value of 9 points; good benefits, 3 points; and limited benefits, 1 point. These values are conventional and used to produce a wider dispersion among results. The QFD methodology calls for an assignment of a customer importance rating (CIR) to each requirement "what". The study team developed CIR values for each requirement without the involvement of the customer groups. The study team did not survey the customer groups (eg: flight crew, scientists, program office, ground operations, etc.) to develop a formal CIR weighting system. The team's CIR values were developed using the team's evaluation of the number of times the requirement was listed in each of the different customer requirement lists that were prepared by the team based on their experience with similar flight equipment. Upon computing the QFD scores, it was determined that the CIR values did not impact the ordinal results of the total scores. Since the CIR values were not derived using a rigorous customer survey, the CIR values were all assigned to be unity and the requirements were assigned equal weight. The QFD provided a way of insuring that factors which cannot be easily quantified, such as technological maturity and safety, are included in the decision.

For a quantitative comparison of the technology candidates, a thermal model of each class of freezer was developed. This model was used to vary design parameters such as insulation thickness to provide a point design for every combination of cooler, enclosure, and thermal transport system. Each design was then compared to the specification to determine its weight and power margins. These quantitative margins, coupled with the qualitative results from the QFD analysis, highlighted where technology development is needed to meet the performance requirements. Technologies with the greatest potential to enable or improve performance in the system of five freezer classifications were recommended for further development.

2.4 Outline of Report

This report follows the study approach outline above. Section 3.0 contains the analysis of the requirements. Section 4.0 surveys the relevant technologies and eliminates from further consideration all but those with the greatest potential. Section 5.0 looks in greater detail at the methodology of the trade studies, both the QFD's and the system analysis studies. It contains an overview of the model on which the analysis of weight and power performance estimates are based and presents the results of these studies, including recommendations for the most appropriate set of technologies for each freezer classification. Section 6.0 draws conclusions about which technologies should be developed.

2.5 Key Players

The prime contractor on the study was Oceaneering Space Systems (OSS) of Houston,TX. Stirling Technology Company (STC) of Kennewick, WA performed the cooler subsystem analysis.

3.0 REQUIREMENTS

3.1 Freezer Classifications

Table 3 summarizes the key design specifications for each of the five freezer classifications.

Freezer Classification	Sample Temp. (°C)	Mass (kg)	Maximum Power (Watts)	Average Power (Watts)	Volume Ext/Int (m^3)
-20°C Storage Freezer	-19	100	456	100-200	0.6/0.3
-70°C Storage Freezer	-68	269	700	100-200	0.9/0.4
- 70°C Freeze Dryer	-70	73	400	100-200	0.3/TBD
-183°C Cryogenic Storage Freezer	-183	122.5	245	100-200	0.2/.02
-196°C Cryogenic Quick/Snap Freezer	-196	29.5	180	100-180	.03/TBD

TABLE 3 : Key Requirements for Advanced Freezer Classifications

-20°C Storage Freezer: This freezer is designated as a low temperature storage facility for life science specimens such as feces, blood, urine, sweat, tissue samples, reagents, chemicals, and medical/biological perishable supplies. It will be a workhorse for sample storage, operating continuously throughout the mission, and sized to accommodate a large variety and number of specimens.

When this unit is operating on a platform like Space Station, samples will be returned to earth either in the Shuttle or in a Mini-Pressurized Logistics Module (MPLM) in the Shuttle Payload Bay. NASA studies analyzed the work required if the volume is divided into drawers and concluded that astronaut workload is most appropriately minimized by transporting the enclosure as a single volume. Thus, the freezer compartment should ideally be configurable to function power-off as a cold storage transport unit.

The total weight and volume allowance for this system is 100 kg in an envelope of 0.6 m^3 and is apportioned among the cooling subsystem, the enclosure, and the thermal transport/electronics subsystem. These requirements are similar to several different freezers with flight experience and are well understood.

-70°C Storage Freezer: Like the -20°C unit, the -70°C unit will also be a relatively large volume, continuously operating freezer for life science specimens and perishable supplies. It, too, should ideally be configurable as a cold storage transport unit for return of samples to Earth. This freezer will be a prime resource for a variety of medical/biological and physical investigations. Its maximum constraints on mass, external volume, and average power consumption are 269 kg, 0.9 m^3, and 200 watts, respectively.

Unlike the −20°C freezer, there is no flight hardware experience for the −70°C unit. Furthermore, some technologies, for example the thermoelectric freezer, are not available at this temperature, and the lower temperature will make moisture control even more challenging.

<u>−70°C Freeze Dryer</u>: The science community also has a requirement to dehydrate specimens for ambient temperature storage. Frozen specimens will be introduced into the freeze dryer and exposed to reduced pressures of approximately 10^{-4} torr. Water removal is required at a rate of up to one liter per day. About 95% of the water will be removed from the specimen. There is no need for this unit to be transportable.

Skylab used a freeze dryer system to preserve over five hundred fecal samples, so the requirements for such a system are fairly well understood. This report describes a unique concept for consolidating the freeze dryer and the −70°C storage into a single half rack to optimize weight and volume, and enhance crew productivity by collocating equipment in a single worksite.

<u>−183°C Cryogenic Storage Freezer</u>: A relatively small volume (0.02 m^3 internal volume) will be needed for storing previously frozen samples at or below −183°C. This unit would have an external volume of 0.2 m^3, must operate continuously throughout the mission, and must also be able to function power-off as a transportable locker. Moisture control will be a particular problem since at these low temperatures, water vapor will rapidly freeze onto the internal surfaces and oxygen may condense from cabin air possing a serious safety threat.

The requirement of a cryogenic storage freezer is new. No such systems have been flown in space.

<u>−196°C Cryogenic Quick/Snap Freezer</u>: Some life science specimens or protein crystals must be brought to cryogenic temperatures very quickly to preserve their structure without formation of damaging ice crystals. The quick/snap freezer must cool room temperature specimens of less than 2 ml volume (saline equivalent) in standard vials to −196°C in less than ten minutes. The unit would have the capacity to accommodate up to twenty 2 ml specimens or ten 5 ml specimens. Extremely small samples, for example protein crystals of 0.5 mm^3 volume, would be frozen by the snap freezer in less than one second. Specimens must then be transferred to the cryogenic storage freezer without allowing their temperature to rise above −183°C.

The requirement that the quick/snap freezer operate in a glove box, a rack mounted, air isolated environment which is accessed with built-in gloves, will make it more challenging than the -183°C freezer. Precluding condensible gases at the freezing site is also more challenging. While liquified oxygen is probable for this freezer as well as for the −183°C freezer, the -196°C freezer encompasses the liquefaction range of nitrogen, which must also be accommodated in the design.

This unit is sized at 29.5 kg and 0.03 m^3 external volume with a maximum drive power of 180 watts. Quick frozen specimens are transferred to permanent storage in the −183°C freezer. Operation of this freezer would be intermittent, and there is no requirement for it to be transportable.

There is no history of using a freezer such as this in space. The required freezing temperature derives from the use of liquid nitrogen on the ground to perform the same function. Since liquid nitrogen is potentially hazardous in space, this freezer will cool samples using a solid conductor.

3.2 General Requirements

Certain requirements apply to all five freezer classifications. The following list was generated from specifications in the Statement of Work, telephone consultations with life science users, and in-house mission scenario analysis. It is organized by the group which would be most interested in the requirement.

Program Manager: The program manager who acquires the freezer system will be interested in cost, schedule, and risk. Cost considerations include not merely acquisition cost, but also life cycle cost which takes into account maintenance and logistics costs over the operational lifetime. The logistics costs include considerations for minimal parts count and commonality of parts. The Statement of Work specifies the maintenance requirement at 2.4 maintenance manhours per year over an operational lifetime of five years.

The program manager must insure the technology needed for the freezer will be available in a timely way. Orifice pulse tube coolers, for example, promise improvements in reliability but are not yet efficient or mature enough to support Space Station timelines. Wherever possible, the program manager would like to use proven technology. When state-of-the-art systems can not meet the performance specifications, the program manager will look for improvements in established technology rather than risk the schedule waiting for breakthroughs.

Ground Scientists: The principal investigator whose experiment is performed on orbit is interested in assuring that samples are maintained in a way which is known, repeatable, and can be duplicated on earth for ground controls. This implies a certain accuracy and precision of temperature control within the cold volume. The Statement of Work specifies that temperature be held at the steady state temperature $\pm 1\,^{\circ}C$ and that "specimens must be maintained in a frozen state throughout the defrost procedure." This requirement was interpreted to mean that specimens would be maintained below the operating temperature during defrost.

Flight Crew: The mission specialist who performs the experiment on orbit requires a proper human factors design which minimizes workload and training. On-orbit maintenance must be minimized. This applies to frost buildup in particular, for experience has shown that improper frost control can consume great amounts of crew time as well as compromise performance. The Statement of Work specifies that "equipment shall be capable of controlling the frost build-up. Defrost shall be accomplished in such a manner as to minimize human intervention."

Crew safety mandates that inflammable and nontoxic working fluids be used, and CFC/HCFCs are specifically prohibited. Crew comfort mandates that the unit operate within noise limits. The Statement of Work requires NC-40 at 0.62 m in any octave band between 63 Hz and 8kHz, but sets NC-30 as a goal.

Ground Operations: Efficient ground handling mandates that units be sized so that they can be lifted by an individual without special equipment. OSHA has set a limit of seventy pounds as the amount that can be lifted by an individual. Ease of sample uploading and retrieval should be considered in the design.

Spacecraft Integration:

- Power: The freezer should operate off a DC power bus that supplies either 28 (±4) volts, 120 volts, or 27 (+7/−3) volts to accommodate the Shuttle, Space Station, and Mir power systems respectively.

- Environment: The freezer should operate in external temperatures between 18 and 40°C (with launch and landing excursions down to −10°C), with atmosphere pressure between 70 and 103 kPa (55 to 103 kPa, launch and landing), a 25 to 80% relative humidity, and up to 40% oxygen.

- Heat rejection: Heat will be rejected into either cabin air (for Shuttle or MIR) or water (for Space Station). Air exiting temperature should not exceed 49°C. Water heat rejection should not exceed 80 watts. Inlet water will be between 3 and 8°C at 98 to 113 liter/hour flow rate and 413 to 689 kPa pressure.

- Structure: The freezer external structure should be compatible with U.S. Standard Equipment Rack Assembly, JSC Standard Interface Rack, Middeck Accommodations Rack, or Spacelab Rack.

- Vibration: Vibration produced by the freezer must not exceed 6 dB/Octave between 20 and 150 Hz, 0.03 g/Hz between 150 and 1000 Hz, and +6.00 dB/Octave between 1000 and 2000 Hz. The system should have no resonances below 35 Hz.

- Other: The freezer must be compatible with the electromagnetic environment per applicable documents, withstand normal launch and landing g and vibration loads, be able to function in either zero or one gravity, operate spark free, be compatible with standard data bus interfaces, and meet flight safety standards.

3.3 Derived Requirements

The Statement of Work specifications contained several "to be determined" (TBD) values or requirements related to the system performance which needed further definition. To resolve these, the contractor team derived the necessary specifications based on assumed mission scenarios and known ISS and Shuttle operating constraints. Information was gathered from ISS Technical Interchange Meetings, discussions with the science user community, and review of the current and planned ISS and Shuttle operations.

The TBD requirements were resolved as follows:

- The −70°C Freeze Dryer internal volume. The freeze dryer volume and vacuum/cold trap were sized to be sufficient to process −70°C frozen samples with a volume equivalent of one liter of water per day. The internal cold volume used in the system analyses was 0.085 m^3.

- The -196°C Quick/Snap Freezer internal volume and on-orbit time line for sample preparation (batch size and cycle time). The internal volume of the quick/snap freezer was sized to accommodate the sample transfer and storage tube and the thermal storage mass. The cooler capacity was sized to have the capability to process a batch of ten samples every four hours. The internal cold volume used in the system analyses was 0.007 m^3.

- The power-off timeline for sample transport. Twelve hours was the worst case identified for un-powered sample storage during transfer operations involving the Mini-Pressurized Logistics Module (MPLM). See Figure 2.

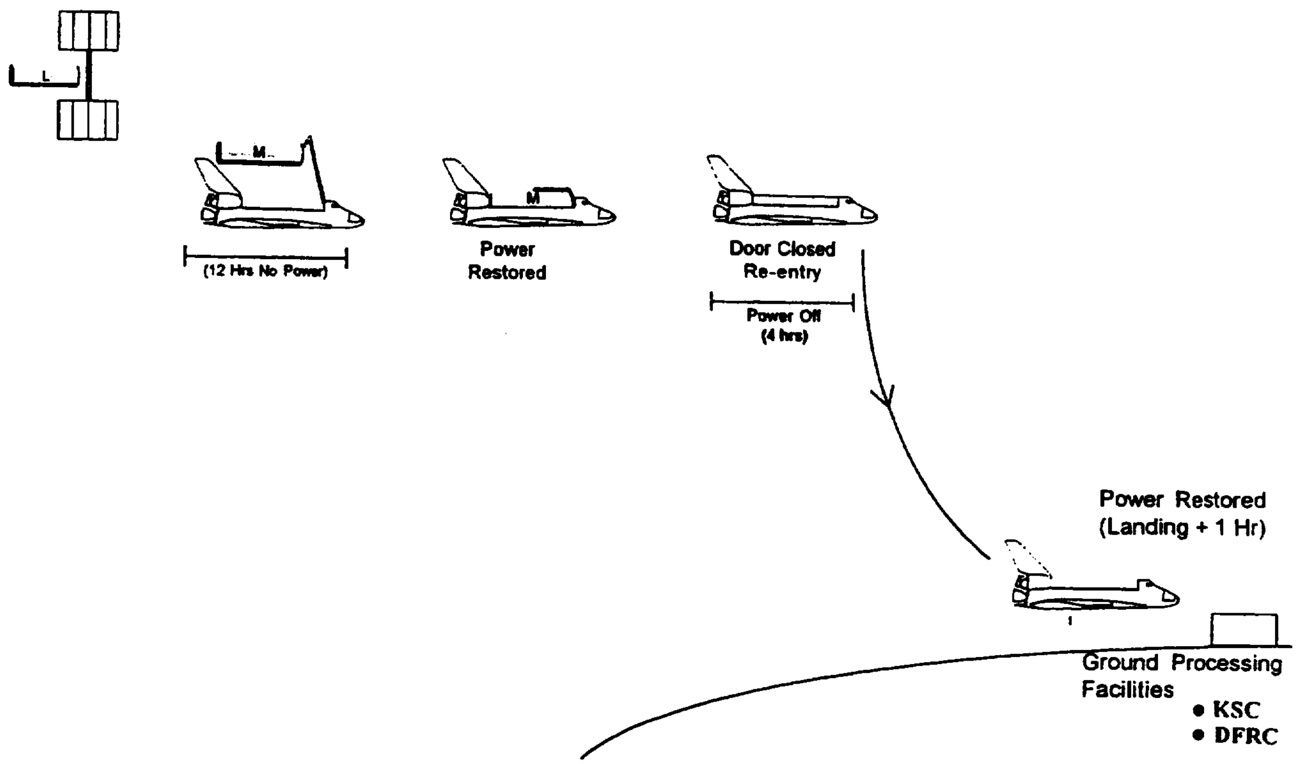


Figure 2: Sample Return Power-Off Timeline

- The ground handling during routine pre-launch sample up-load and post-landing early access. The -20, -70, and -183°C storage freezer classifications need a lightweight removable enclosure with long-term, unpowered temperature storage capability (assumed 12 hours) for sample transfer and transportation. Removable sample storage containers for the -20°C, -70°C, and -183°C freezers should not exceed 31.8 kg (70 pounds) loaded, based on OSHA worker safety standards for one-g manual handling.

- Spacecraft heat rejection. Both water and air, with a range of possible temperatures and flow rates were included in the specifications. To establish a challenging but still nominal design condition, air heat rejection was assumed at a 25°C mean air temperature.

- The power system interface. The Statement of Work specified average power usage in a range from 100-200 Watts. The system goal was defined as less than or equal to 200 Watts.

- Moisture load. Sizing of the moisture load largely depends on the door openings and sample freezing operations. The door opening frequency was assumed to be eight times per day for the -20/-70°C freezer classifications. The cryogenic systems would operate through tiny openings or within an airlock that makes them less sensitive to the number of door openings.

3.4 Consolidated QFD What List

From the requirements enumerated above, OSS derived a list for formal consideration in the Quality Function Deployment process. The following were selected as critical to a qualitative assessment of candidate technologies.

- **Technology Maturity:**

  Excellent Benefits: Extensive development and characterization for space flight applications, supported by critical acceptance, and flight success.

  Good Benefits: Either 1) Limited development and characterization for space flight applications, supported by critical acceptance, or 2) extensive commercial application with direct applicability to space flight.

  Limited Benefits: Demonstrated in laboratory or in limited commercial use.

  No Benefits: Exists in theory.

- **Longevity/Reliability/Maintenance/Robustness:**

  Excellent Benefits: Shown by theory and test to have the potential for 10 or more years life with a reliability exceeding .95 and the potential for minimal maintenance, all under the intended conditions of use including all rigors such as launch, landing, and physical environment.

  Good Benefits: Shown by theory and test to have the potential for 5 or more years life with a reliability exceeding .95 and the potential for minimal maintenance, all under the intended conditions of use including all rigors such as launch, landing, and physical environment.

  Limited Benefits: Shown in theory to have the potential for 5 or more years of life with a reliability exceeding .95 and the potential for minimal maintenance. A rational basis exists for extrapolation of limited tests to the intended conditions of use, including all rigors.

  No benefits: Requires extensive maintenance or does not have the potential for 5 years life with a reliability exceeding .95 and the potential for essentially no maintenance, all under the intended conditions of use, including all rigors.

- **Commonality:**

Use of common components and systems over the required range of temperatures, capacities and applications to minimize development, manufacture, inventory and training requirements.

Excellent Benefits:	Excellent potential to achieve commonality. One apparatus meets all NASA Science R/F requirements.
Good Benefits:	Use of common components to achieve significant reduction in the number of components and/or systems to be developed and inventoried.
Limited Benefits:	Limited potential to achieve commonality.
No Benefits:	No potential to achieve commonality.

- **Configurable Temperature and Volume to Handle Up and Down Payloads:**

This R/F system characteristic is enhanced by components which can be combined in modular form to achieve varying levels of cooling, and which have capacity adjustable to achieve reduced power consumption at reduced cooling loads. The following definitions apply for coolers.

Excellent Benefits:	Capable of efficiency in modular application, and having adjustable capacity.
Good Benefits:	Capable of efficiency in modular application, or having adjustable capacity.
Limited Benefits:	Not modular and limited ability to adjust capacity.
No Benefits:	Capacity is not adjustable.

- **Maintenance on Ground:**

Excellent Benefits:	Quickly replaced on line with no special tools or training.
Good Benefits:	On line replacement requires special tools or training.
Limited Benefits:	Time consuming replacement requiring special tools or training.

- **Low Mass:**

Excellent Benefits:	No more than 1.5 times the mass of the lowest mass option.

Good Benefits:	1.5 to 2.5 times the mass of the lowest mass option.
Limited Benefits:	2.5 or more times the mass of the lowest mass option.

- **Low Volume:**

Excellent Benefits:	No more than 1.5 times the volume of the lowest volume option.
Good Benefits:	1.5 to 2.5 times the volume of the lowest volume option.
Limited Benefits:	2.5 or more times the volume of the lowest volume option.

- **Low Power:**

Excellent Benefits:	No more than 1.5 times the power demand of the lowest power demand option.
Good Benefits:	1.5 to 2.5 times the power demand of the lowest power demand option.
Limited Benefits:	2.5 or more times the power demand of the lowest power demand option.

- **Low Vibration:**

Excellent Benefits:	Inherently provides vibration less than the requirement.
Good Benefits:	Passive vibration control meets the requirement.
Limited Benefits:	Active vibration control meets the requirement.
No Benefits:	Cannot meet vibration requirement.

- **Environmental Compatibility:**

Excellent Benefits:	The simplest form and packaging of the device meets the requirements.
Good Benefits:	Limited improvements to form and packaging allow the requirements to be met.
Limited Benefits:	Major improvements to form and packaging allow the requirements to be met.
No Benefits:	The requirements cannot be met.

- **Safety:**

Excellent Benefits:	Safety hazards limited to electrical.
Good Benefits:	Safety hazards easily addressed.
Limited Benefits:	Safety hazards addressed with difficulty.

- **Easy Sample Transfer:**

Excellent Benefits:	One handed operation, minimal heat input rate, minimal condensates, minimal heat load added to freezer systems.
Good Benefits:	Two handed operation, acceptable heat rates, minimal condensates, minimal heat load added to freezer systems.
Limited Benefits:	Two handed operation, acceptable heat rates, condensates, and freezer heat loads.
No Benefits:	Two handed operation, unacceptable heat rates, condensates, or freezer heat loads.

- **Ease of System Operation:**

Excellent Benefits:	No special training required to operate and maintain ground and flight systems. No crew time required for on-orbit defrost maintenance.
Good Benefits:	Limited training (less than 2 hours) required for on-orbit sample processing and routine maintenance operations. Limited (goal= 1 hour/month) crew time required for on-orbit defrost maintenance.
Limited Benefits:	Significant training required for on-orbit sample processing and routine maintenance operations. Significant crew time (2-5 hours per month) required for on-orbit defrost maintenance.
No Benefits:	System difficult to operate and maintain. Requires extensive training and special tools to support normal operations and maintenance.

- **Power-Off Timeline:**

Excellent Benefits:	Power-off heat leak is less than 10% of cooling capacity.
Good Benefits:	Power-off heat leak is less then 50% of cooling capacity.
Limited Benefits:	Power-off heat leak is less than cooling capacity.

No Benefits: Power-off heat leak exceeds cooling capacity.

- **Sample Temperature Regulation:**

Excellent Benefits: Passive system (direct thermal conduction, convection, and radiation), light weight, minimal volume, high thermal capacity.

Good Benefits: Passive or powered system, light weight, high thermal capacity.

Limited Benefits: Passive or powered, acceptable weight, volume, and thermal capacity.

No Benefits: Unacceptable power, weight, volume or thermal capacity.

- **Launch/Landing Survivability:**

Excellent Benefits: Robust. No moving parts.

Good Benefits: Robust. Moving parts. Simple.

Limited Benefits: Robust. Moving parts. Complex.

No Benefits: Not robust.

- **Post Landing Sample Retrieval:**

Excellent Benefits: 70 lbs or less with sufficient thermal mass to get through landing with no MPLM cooling required.

Good Benefits: 70 lbs or less with need for MPLM cooling.

Limited Benefits: 100 lbs or less with need for MPLM cooling.

No Benefits: Over 100 lbs with need for MPLM cooling .

- **MPLM Access and Handling:**

Excellent Benefits: No special handling equipment required for late access and early retrieval of samples. Meets 70 pound (31.8 kg) manual ground handling weight in OSHA specifications.

Good Benefits: Limited special handling equipment required for late access and early retrieval of samples. Meets 70 pound (31.8 kg) manual ground handling weight if sample containers are removed from the insulated enclosure for ground transfer.

Limited Benefits: Ground handling weight exceeds 70 pounds (31.8 kg) and extensive ground

ground support equipment is needed to transfer samples.

No Benefits: System not compatible with successful sample preservation and transport.

- **Configurable Temperature and Volume to Meet On-Orbit Storage Requirements:**

This R/F system characteristic is enhanced by components which can be combined in modular form to achieve varying levels of cooling, and which have capacity adjustable to achieve reduced power consumption at reduced cooling loads. The following definitions apply for coolers:

Excellent Benefits: Capable of efficiency in modular application, and having adjustable capacity.

Good Benefits: Capable of efficiency in modular application, or having adjustable capacity.

Limited Benefits: Not modular and limited ability to adjust capacity.

No Benefits: Capacity is not adjustable.

- **Ease of Subsystem Integration:**

Excellent Benefits: Integration requires no special processes or fasteners

Good Benefits: Integration requires accepted practices for weldments, brazements and bond lines, or special fasteners

Limited Benefits: Integration requires qualification of new assembly method

No Benefits: Integration cannot practically be achieved

4.0 CANDIDATE SUBSYSTEM TECHNOLOGY

Each of the five freezer classifications is comprised of four key subsystems: the cooler, insulated enclosure, thermal transport, and electronics. For each subsystem, the contractor team surveyed the available technologies and then narrowed the field to those classes which had the potential to meet the major requirements for each freezer classification. A QFD analysis was performed to highlight the candidates with the most overall potential.

4.1 Cooler

Figure 3 lists the sources used to generate a comprehensive list of candidate cooler technologies. A broad range of thermodynamic cycles and cooler technologies were assessed by comparing the product literature and published test data against the cooler subsystem performance goals including:

- Heat rejector temperature
- Heat rejection media (air or water)
- Heat absorber temperature
- Heat lift capacity
- Coefficient of Performance (COP) (heat into absorber per system energy input)

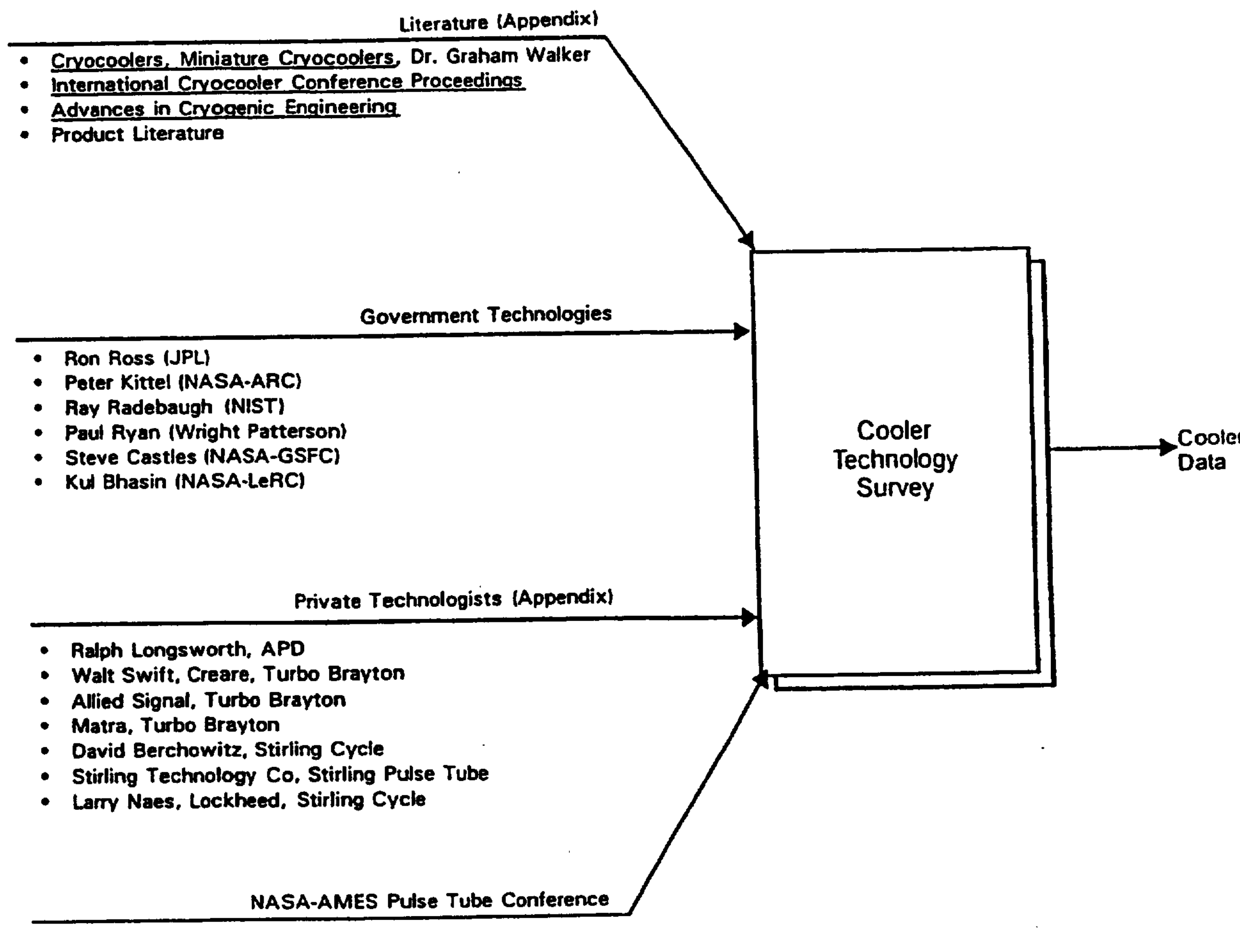


Figure 3: Sources of Cooler Information

To minimize the number of technologies required to meet the specifications, the contractor team sought to identify a single cooler technology that could serve all five freezer classifications with a modular set of cooler hardware. Each freezer classification was also evaluated to identify which cooler, enclosure, and thermal transport provides the best performance for that system alone. Technology maturity was assessed to assure that any necessary development could be accomplished within the project resources.

Figure 4 shows an organizational tree of the cooler technologies considered in the initial survey. This tree proved useful in drawing generalizations about the extent to which each class applied to the system requirements.

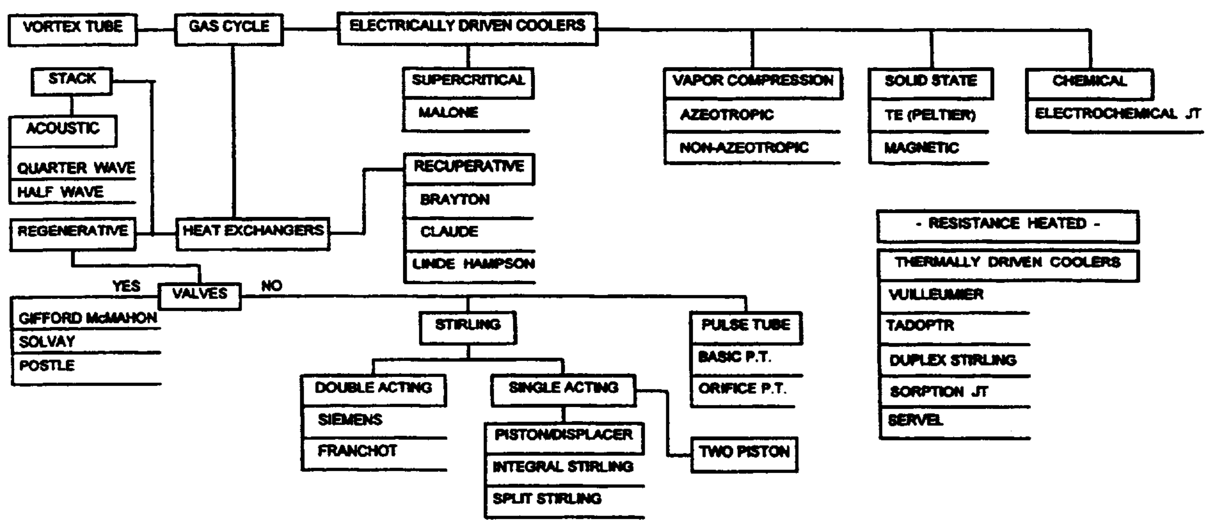


Figure 4: Cooler Family Tree

Thermally Driven Coolers: Coolers that are driven by thermal energy, the Vuilleumier, Thermo-Acoustic (TADOPTR in Fig. 4), Duplex Stirling, Sorption Joule-Thompson, and Servel, would use spacecraft electrical power to drive the cooling processes. These coolers require an independent source of heat which must then be converted to work to drive the cooler mechanisms. Because of the inherent inefficiency in converting heat to work (associated with the second law of thermodynamics), the cycle efficiency was very low; therefore, they were eliminated from further considerations.

Vapor Compression: Rankine cycle vapor compression coolers, commonly used in commercial and residential cooling, use a working fluid which undergoes a liquid-gas phase change in the process of cooling. In spite of this mature technology base, which includes use on the first Shuttle Orbital Refrigerator Freezer (OR/F), the requirement that any working fluids be non-CFC/HCFC and non-toxic eliminates most established vapor compression technologies from further consideration. The OR/F, which used R-134a (HCFC) working fluid, had to be double contained to mitigate toxicity concerns, and that containment added weight and diminished thermal performance. The design is also sensitive to refrigerant leakage through electrical feed-throughs and seals, and zero gravity effects upon the two phase refrigerant. Although research to find non-toxic, non-flammable, non-CFC/HCFC working fluids is on-going, none has yet been identified. Further, traditional lubrication systems which rely on an oil mist and sump do not perform consistently in micro-gravity. NASA has funded research into oilless compressors and some work has been demonstrated in this area. Overall, however, vapor compression coolers for use in space were considered too developmental for further consideration in this study.

The basic vapor compression cycle is limited in its ability to reach low temperatures. The efficiency is quite high at low temperature differences but falls relative to competing cycles with increasing temperature difference. Various means have been resorted to, in the form of staged or cascaded vapor compression cycles, to obtain heat lift at low temperatures. The staged and cascaded arrangements have the reliability and toxicity problems of the single stage vapor compression coolers, but are more complicated, and therefore less reliable.

The mixed gas non-azeotropic cooler is a vapor compression cooler which provides an alternative to the added complexity of staged and cascaded vapor compression machines. This cooler uses a mixture of refrigerant gases which provide a drop in boiling point as the lighter constituents of the refrigerant are boiled off.

Temperatures in the vicinity of 90K are achievable with these variations on the vapor compression cycle, but none of these systems operate practically at the 77K temperature of liquid nitrogen. Further, these coolers share and enlarge upon the problems associated with manned space flight posed by vapor compression cycle coolers.

Solid State: This family includes the Thermoelectric (TE or Peltier) coolers and magnetic coolers which use solid state devices to lift heat. Although they have low vibration, high expected reliability, and reasonable technology maturity, the COP of TE coolers and Peltier junction materials restrict the feasible acceptor temperature and heat lift capacity of these systems. The maximum feasible temperature differential is in the range of 50 to 75°C. Demonstrated TE two-stage technology is a candidate for the -20°C storage freezer but is impractical for the other four freezer classifications. Magnetic refrigeration systems are a potentially valuable horizon technology but are not yet mature or efficient enough to be considered further in this project.

Chemical: The electro-chemical processes involve materials and safety concerns which are not compatible with space flight design, and the demonstrated systems are not efficient relative to competing technologies.

Gas Cycle: The family of gas cycle coolers, which cool by compressing and then expanding a gas, has two major subdivisions: the regenerative, in which gas flow is oscillatory or tidal, and the recuperative, in which gas flows in a circuit. The regenerative cycles with the most promise are the Stirling cycle and the orifice pulse tube. The recuperative cycles include the Brayton, which has been demonstrated over the temperature range of interest. The Stirling cycle designs produce a localized cold surface which must be interfaced to the cooling load, while the Brayton cycles circulate cooling gas directly to the load. The pulse tube has a separate cooling chamber that has oscillating gas flows, but no moving parts. The cooling chamber can be mechanically isolated from the compression mechanisms, thereby isolating the cooled volume from the vibrations caused by the compressor pistons.

This survey reduced the list of coolers for detailed consideration to the Stirling, orifice pulse tube, and Brayton, and the thermoelectric for the −20°C storage freezer only.

In Figure 5, data on the cooling capacity and temperature for best available technology from each cooler type is overlaid with the heat and temperature requirements for the three temperature zones of the freezers of interest. The vapor compression range is included for comparison. The placement of the lines is approximated from known systems data.

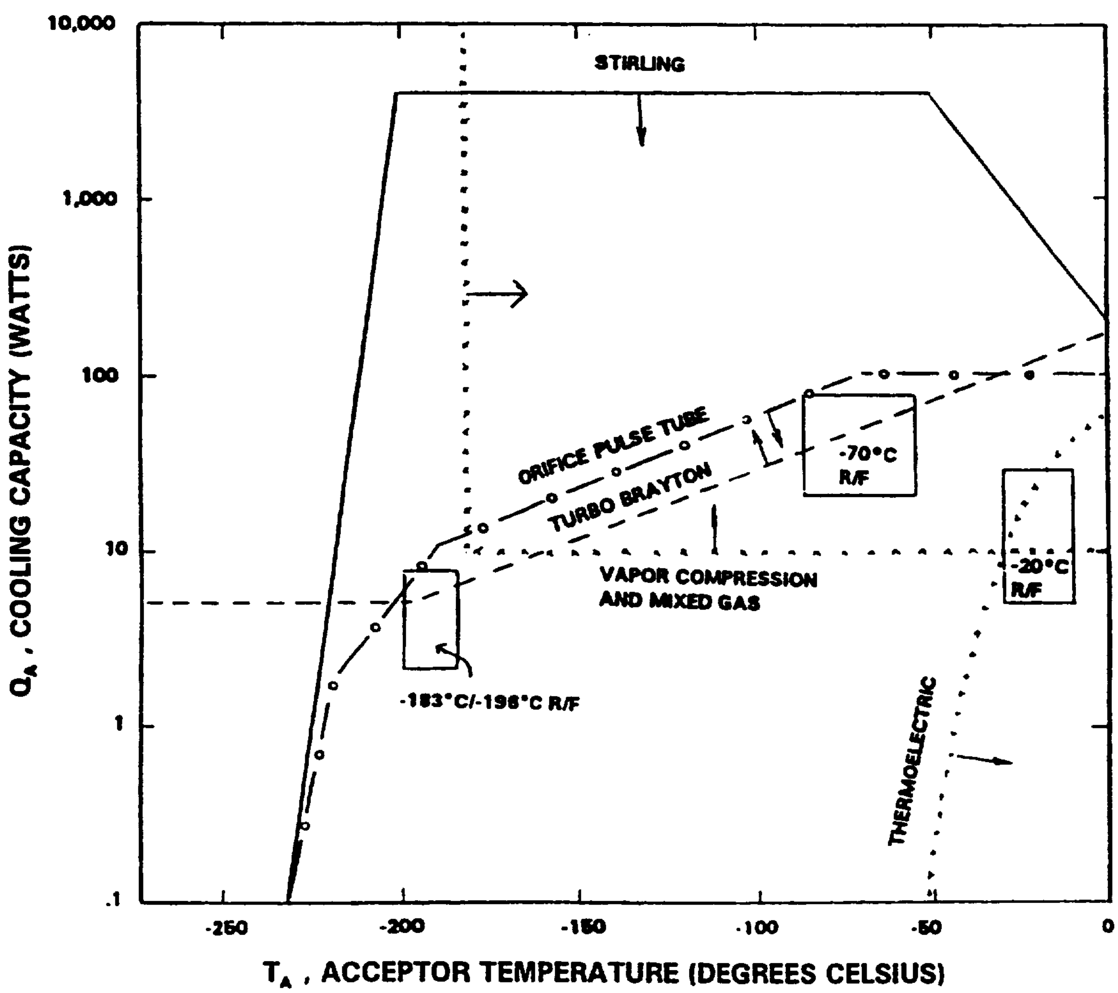


Figure 5: Demonstrated Range of Single-Stage Coolers

The only thermodynamic cycle that easily envelopes all of the temperature zones is the Stirling cycle. The $-20°C$ specification is within the demonstrated design envelope of TE coolers. The lower limit of the currently demonstrated Brayton cycle capacity range is on the fringe of the required heat lift for the cryogenic and $-70°C$ classifications. The upper end of the currently demonstrated capacity of pulse tube systems is on the fringe of the required heat lift. In the light of well-funded research to increase the working envelope and efficiency of pulse tube coolers, they must be monitored for potential future use. The demonstrated vapor compression and mixed gas systems encompass the $-20/-70°C$ requirements but do not meet the requirements for the cryogenic systems under consideration.

Figure 6 shows the thermodynamic performance data for demonstrated pulse tube, Stirling, Brayton, vapor compression & mixed gas and Thermoelectric coolers. The plot of the COP versus the acceptor temperature indicates the relative efficiency of the demonstrated cooler systems referenced to the ideal thermodynamic efficiency of the Carnot cycle. Curve fits to the performance curves for each cooler family were used in the system analysis model discussed in Section 5.0.

A QFD analysis (Figure 7) for the four most promising cooler technologies shows that, while the Stirling system has the most advantages, the Turbo Brayton and the Orifice Pulse Tube cannot be ruled out without further investigation. The Positive Displacement Brayton was dropped from further consideration, because it was not considered technically or developmentally competitive with the other candidates.

It is possible to conceive of improvements to a Stirling cooler which would provide greater efficiency than current Stirling cooler technology. These include an improved efficiency motor, a low thermal loss expander housing, advanced regenerators, and improved cooler-to-heat transfer system interface. An enhanced efficiency Stirling cooler is included as a candidate cooler in the trade studies.

The coolers which were carried into the detailed trade studies were:

- Stirling Cycle
- Turbo Brayton
- Orifice Pulse Tube
- Thermoelectric for the $-20°C$ storage freezer only
- Enhanced Efficiency Stirling, described above

4.2 Enclosure

The thermal resistance of the enclosure has the greatest effect on the passive heat loads. Minimizing the enclosure heat load through the walls will minimize the power required for heat rejection. Other sources of heat leakage into the enclosure, through the door seals and around wire penetrations, become more significant contributors as the thermal resistance of the enclosure structure is increased.

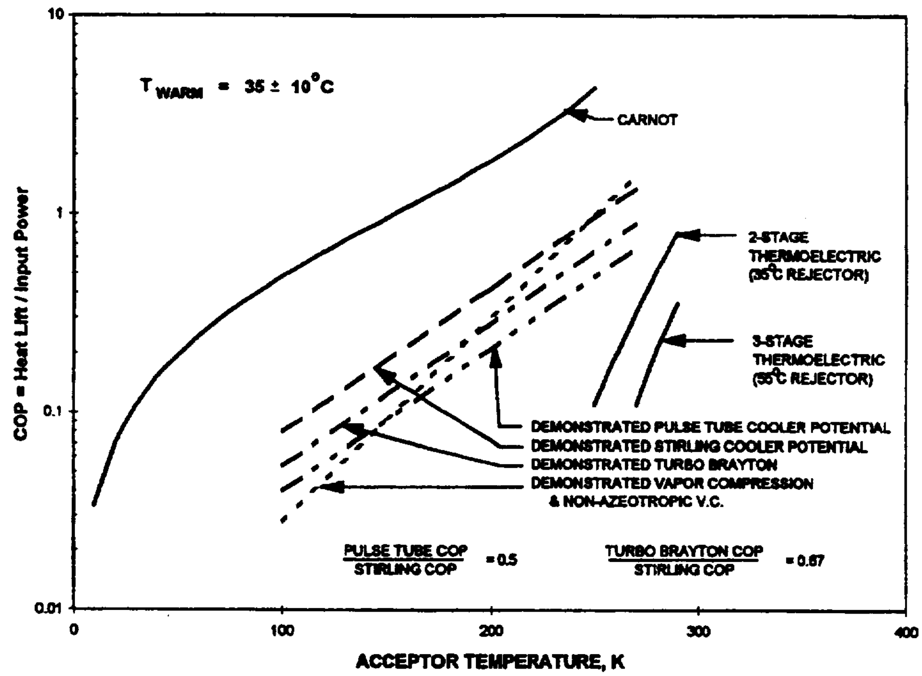


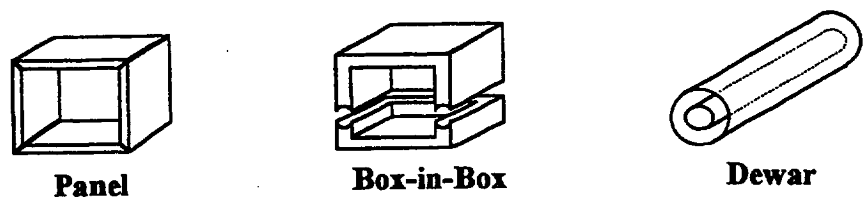
Figure 6: Thermodynamic Performance Data

Excellent Benefits:	● 9
Good Benefits:	○ 3
Limited Benefits:	△ 1

	Turbo Brayton	Positive Displace. Brayton	Stirling	Orifice Pulse Tube
Technical maturity	○	△	●	○
Longevity/Reliability/Maintenance/Robustness	○	△	●	●
Commonality	△	△	○	○
Configurable temperature and volume to handle up/down payloads	○	○	○	○
Maintenance on ground	○	○	●	○
Low mass	○	△	●	○
Low volume	△	△	●	○
Low power demand for adequate capacity	△	○	●	○
Low vibration generation (also impacts noise)	●	△	○	●
Low noise generation (impacted primarily by heat rejection)	○	○	●	○
Acceptable resonant frequencies	●	○	○	○
Electromagnetic compatibility	●	●	●	●
Electromagnetic interference	△	△	△	△
Environmental compatibility	○	○	○	○
Safety	○	○	○	○
Cooler temperature/capacity operating range	●	●	●	●
Obvious layout and operation (foolproof)	○	○	●	●
Power-off timeline	△	△	△	△
Sample temperature regulation	●	●	●	●
Intermediate cold heat exchanger requirement	●	●	○	○
Low power surge at start-up	●	●	●	●
Launch/landing survivability	△	△	○	○
Cold containment susceptibility	△	△	△	○
Configurable temperature and volume to meet on-orbit storage requirements	○	○	○	○
Technical Importance Rating	100	82	138	110
Normalized Technical Importance	23	19	32	26

Figure 7: Cooler QFD

The enclosure technologies needed for the five freezer classifications were considered in two broad categories: cylindrical dewars and rectangular cabinets. Although a dewar can have a very high thermal resistance, a cabinet provides greater internal volume for a given external envelope. To meet the specifications for internal volume for the -20/-70°C storage freezers, a cabinet construction is essential. In addition to the volumetric efficiency, the dewar and cabinet constructions have different structural weights, technological maturities, thermal resistances, materials safety, manufacturabilities, and complexity/reliabilities. Figure 8 contains a table of the R-values, calculated on a per inch basis, for the enclosure technology candidates considered.



	No Vacuum	Low Vacuum (.1 torr)			Moderate Vacuum (.001 torr)				
*VSM	Foam	Powders	Fiberglass	Fiberglass	Fiberglass	**RMLI	RMLI	RMLI	N/A
Construction	Solid	Plastic Panel	Plastic Panel	Metal Panel	Metal Panel	Box-In-Box (Metal Closeout)	Box-In-Box (Plastic Closeout)	Polymer Panel	Dewar
Edge Effects	No	Minor	Minor	Yes	Yes	No	No	Minor	Minor
Bulk R	5 - 7	30 - 40	30 - 40	40	85	150 +	150 +	150 +	3000
Enclosure R (.4 m^3)	5 - 7	25 - 35	25 - 35	15	30	60	145	105	2300

* Vacuum Support Method

** Rigidized Multilayered Insulation

R = 30 (In analysis)	R = 60 (In analysis)	R = 105 (In analysis)	R = 2300 (In analysis)
Current Technology	Technology Development	Technology Development	Current Technology

Figure 8: Insulated Enclosure R-Values

The dewar construction has the greatest thermal resistance (R-value). The high vacuum (<.001 torr) between the inner and outer walls of the pressure vessel eliminates convective and gas conductive heat transfer; the aluminized mylar multi-layer insulation (MLI) reduces radiative heat transfer; and, since the inner and outer shells are generally only joined at the close out region, there is minimal conductive heat transfer. The weight of the dewar construction is increased by the structural requirements of the outer pressure vessel and goes up sharply with internal volume so that large storage volumes are difficult to accommodate within the allowable weight.

The internal volume requirement for the two cryogenic temperature freezers includes the range in which dewar weights are reasonable, and the high thermal resistance of the dewar is essential to limit the heat gain across the temperature difference of approximately 220°C (room temperature to -196°C). In contrast, the -20/-70°C freezers are volume and weight critical, precluding the dewar construction. For these freezer classifications, the cabinet construction is needed.

Two qualitatively different cabinet constructions are considered: the panel and box-in-box. The box-in-box has inherently greater thermal resistance since it minimizes the number of seams, which are the leading conduit for heat leakage. Metal skinned panels offer cost and design benefits, but edge losses significantly reduce their potential thermal resistance performance. Plastic skinned panels would minimize edge loss but must be non-porous enough to sustain a vacuum for long periods of time.

The technology survey included several vacuum panel technologies, block and expanded foams, and emerging materials such as aerogels. Vacuum panel technologies which have flown on Shuttle refrigerator/freezers include the Owens-Corning Aura™ and evacuated powder panels using opacified powder fillers such as precipitated silica. Aura™ is constructed from a welded 3 mil stainless steel skin over a fiberglass filler material. It is currently being produced in limited quantities for commercial refrigerator/freezers. Standard Aura™ panels have beveled edges and mitered corners to allow box constructions. Aura™ literature claims a center panel R-value of 90 per inch, but edge effects due to the conduction heat leakage through the stainless steel skin reduces the total enclosure (based on International Space Station standard rack size) R-value to roughly R-30. In spite of this edge effect penalty, Aura™ is still the leading panel technology for volume critical, rack mounted enclosure designs, due in part to its predicted 10 year life. Figure 9 compares the major types of insulation evaluated in this study. The two axes of the plot, density and thermal resistance, define two of the major parameters which have the greatest impact on whether a refrigeration system will meet its performance requirements. Density is important in meeting the mass budget while thermal resistance is important in meeting the power budget. Some of the freezers analyzed for the technology assessment are more mass constrained than power constrained (i.e. -70°C Transporter discussed in Section 5.4) while others are more power constrained (i.e. -20°C Storage Freezer with TE cooler discussed in Section 5.3). Since both mass and power are always important parameters, an insulation which has both low density and high R-values is advantageous. Figure 9 identifies the proposed polymer panel development as rating the highest with respect to the combined benefits toward mass and power savings.

Powder panel technologies support a lower vacuum level and have a shorter projected life due to the degradation of the vacuum over time. The Vacu-panel™ system used on the SORF system is made from a thermoformed plastic skin panel enclosing a micro-porous silica powder vacuum support structure. The 10^{-2} torr vacuum of the powder panel system has an initial R-value of approximately 18 per inch, with an expected decay of 20-30% as the vacuum level decreases over time. The existing technology is not designed for long term space application. Potential contamination from the fine silica powder requires that the vacuum shell also function as a protective barrier.

Improved means of supporting the vacuum loads for flat insulation construction was sought. These must maximize thermal resistance while satisfying all of the other system specifications including weight, durability, safety, etc. Aerogel is an emerging materials technology that shows promising materials properties, particularly low density and high thermal resistance. Aerogels are being incorporated into vented spacecraft avionics assemblies in which the thermal resistance of the enclosure relies on the vacuum of space and the low conductivity of the aerogel. Aerogel was considered for use as the vacuum support structure (VSS) inside vacuum panels. The challenges associated with this concept are the manufacture of practical sized aerogel panels along with the strength and durability of the material. Based on these technical challenges in the existing aerogel technology and the expected lower thermal resistance of the engineered polymer VSS, the use of aerogel materials for the VSS is not recommended.

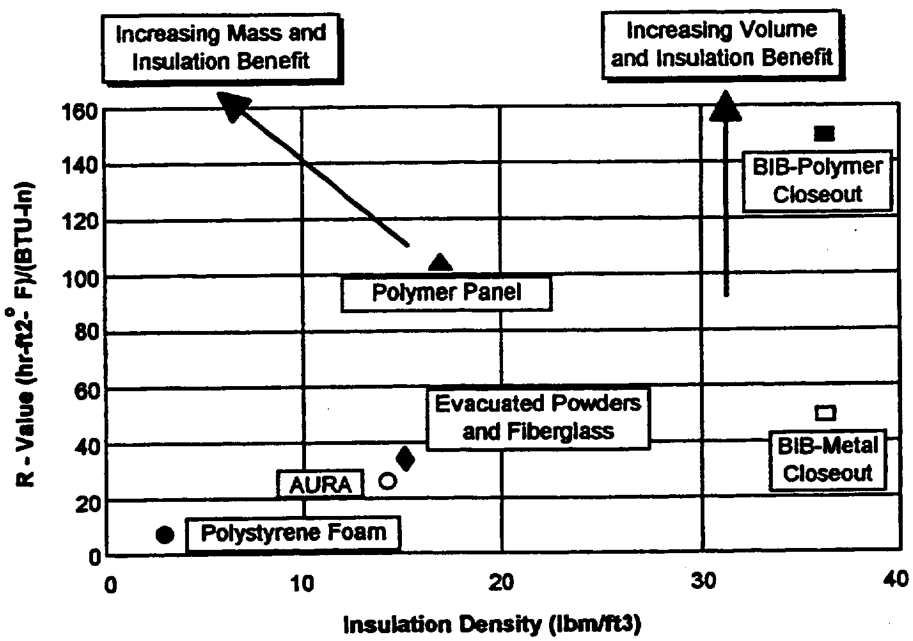


Figure 9: Enclosure Insulation Comparisons

OSS is currently developing a polymer panel which should address the major shortcomings of state-of-the-art vacuum panels like Aura™. Edge losses are minimized by making the skins out of a polymer which has a hundred-fold reduction in thermal conductivity compared with stainless steel. These skins should reduce the heat transfer through the panel by an estimated 50-80%. Previous attempts to use polymer skins have not held the required 10^{-3} torr vacuum inside the panel for usable lengths of time. They allow atmospheric gases to penetrate the skin, permitting a conductive heat transfer path through the gas. OSS is currently resolving several processing issues associated with a new thermoplastic which has a significantly lower gas permeability than current high-barrier plastic films and sheets. If successful, this new material combined with suitable getters is expected to sustain the vacuum loads over the 10-year service life of the panel. OSS has also engineered a vacuum support structure out of a polymer material which, when aluminized, should permit bulk R-values of 150, over twice as large as panels based on a silica powder or fiberglass mat vacuum support structure. The polymer panel should permit enclosures with an R-value of 105, three times greater than currently available technology.

Foam is the lowest density insulating material. With sufficient external volume for thick walls, foam would be the technology of choice. Both polyimide block and expanded urethane foams have been successfully used in space refrigerators. The most recent Shuttle middeck enclosure designs have combined the structural and producibility benefits of foam with the vacuum panel technologies to produce rectangular constructions that address the vacuum panel edge effects with the foam.

Figure 10 is a QFD analysis of the most promising cabinet technologies. The analysis shows that the beyond-the-state-of-the-art evacuated MLI panels with plastic skin, the so-called polymer panels, hold a lot of promise. Foam and evacuated powders were not effective enough to be considered further.

Excellent Benefits:	●	9
Good Benefits:	○	3
Limited Benefits:	△	1

	FOAM	EVAC. POWDER (R30)	EVAC MLI (Metal Skin) (R30)	EVAC MLI (Plastic Skin) (R105)	EVAC MLI (BIB) (R60)
Technical maturity	●	●	●	○	●
Longevity/Reliability/Maintenance/Robustness	○	○	○	○	○
Commonality	●	○	○	○	△
Configurable temperature and volume to handle up/down payloads	○	○	●	●	●
Maintenance on ground	○	○	●	●	○
Low mass	●	○	△	●	○
Low volume	△	○	○	●	○
Low vibration generation (also impacts noise)	●	△	△	●	△
Environmental compatibility	○	○	●	●	●
Safety	○	○	●	●	●
Easy sample Transfer	●	○	●	●	●
Ease of system operation	△	○	●	●	●
Power-off timeline	△	○	△	●	○
Sample temperature regulation	△	○	●	●	●
Launch/landing survivability	●	○	●	●	●
Prelaunch sample (upload) processing	△	△	●	●	●
Post landing sample retrieval	△	△	○	●	○
MPLM access and handling	△	○	△	●	○
Configurable temperature and volume to meet on-orbit storage requirements	△	○	●	●	○
Ease of subsystem integration	●	△	●	●	○
Technical Importance Rating	86	58	124	162	110
Normalized Technical Importance	16	11	23	30	20

Figure 10: Insulated Enclosure QFD

As a result of this survey, four technologies were considered appropriate for inclusion in subsequent trade studies:

- the MLI dewar with an R value of 2300, though its applicability was restricted to the cryogenic systems.
- a beyond-the-state-of-the-art rigidized MLI panel with plastic skins, having a calculated R-105.
- an evacuated fiberglass panel with metal skins (EVAC MLI Metal Skins in Fig. 10), such as Aura™, with a demonstrated R-30.
- a rigidized MLI box-in-box construction with metal skins, with a calculated R-60.

4.3 Thermal Transport

Thermal transport interfaces the heat absorber (cooler cold side) surface to the freezer contents and the heat rejector (cooler hot side) surface to the rejection media, either the cabin air or cooling water. When heat is rejected directly into air, acoustic emissions result from the fan noise, heat exchanger air flow turbulence, and air duct acoustic characteristics. Previous space flight freezer designs have not fully met the acoustic emissions requirement, and, since noise is directly related to the heat rejected, the problem will become worse with the lower temperature freezers. This technology is considered an important development area.

Thermal transport technology also addresses the desire to provide thermal storage with phase change materials in order to sustain the temperatures during unpowered operations, and the need to provide certain operational capabilities such as a quick disconnect thermal interface. The thermal transport components include heat exchanger, fans, heat pipes, phase change materials, etc. The emphasis in this study was on the transfer of heat between the freezer enclosure and the cooler cold head.

The different candidate cooler technologies have unique thermal interface requirements. To evaluate the advantages and drawbacks of the alternative transport technologies, the contractor team developed conceptual designs of several cooler-enclosure combinations. The thermal transport technologies were then used to interface these technologies to meet the published freezer specifications.

The concentrated cold surface area inherent in Stirling cycle systems necessitates a highly efficient interface to convey the thermal loads from the enclosure. The Stirling cycle heat rejector is normally an integral part of the pressure vessel housing, and the thermal transport interface to the heat rejection media is either a liquid cooling jacket or fins to conduct and convect the heat to the surrounding air. In contrast, Brayton cycle systems, which inherently incorporate a heat exchanger, have a more efficient thermal transport interface. The working gas continuously flows through the heat exchanger, and the surface area of the exchanger can be sized to eliminate the need for supplemental thermal transport components such as those needed by the Stirling configurations.

Three classes of thermal interface technologies are: metallic conductors, carbon conductors, and heat pipes. Metallic conduction strips, especially copper and aluminum, though simple and effective, are relatively heavy. The temperature drop across the conductor is a function of the material conductivity, the cross-sectional area, the heat load, and the length of the conduction path, is a loss term in the heat balance. Copper has the best thermal properties, and, since only small quantities are required by the conceptual design, its weight penalty compared with aluminum was minimal. Copper was selected as the baseline metallic conductor for this study.

Carbon conductors are a newer technology used in military avionics cooling systems, but they have not been widely applied to commercial products due to the cost and technology maturity. One particular material, Thermal Pyrolytic Graphite (TPG), has been developed into planar configurations and provides a lower mass and lower thermal resistance relative to the metallic systems. The TPG is hot isostatically pressed between aluminum or copper plates. It is anisotropic in bulk; how its anisotropy will effect its design usefulness is not yet clear.

Heat pipes are an established technology that provides a near isothermal, high heat flux thermal transport media. In a heat pipe, the working fluid is transformed to vapor at the hot end and is wicked from the cold condenser end back to the hot end to close the loop. The heat pipe can transport heat across relatively long paths with minimal losses and no moving parts. Its drawback relative to passive conductors is its complexity. The heat pipe's operation in one g depends on having a pre-established orientation with respect to gravity, but the actual orientation will vary between the launch and landing orientations, making its priming and operation a concern. Also, since the working fluids tend to be toxic, double or triple containment is needed, and this impacts the heat transfer efficiency.

The Stirling Orbiter Refrigerator/Freezer (SOR/F), a -26°C design, incorporated an acetone heat pipe to interface the cooler's heat acceptor surface with the enclosure. The system reportedly performed without difficulty, but no specific data was available to confirm this. To operate at -80°C temperatures, the heat pipe on the SOR/F would require a different working fluid. Thermacore Inc. recommended working fluids and containment technologies for the five freezer classifications; a propylene working fluid, for example, could address the -20/-70°C range. Thermacore also produced a concept for a diode feature which would use a secondary non-condensable gas to provide a shut down mode. With this diode feature, the heat pipe would only pump heat in the forward direction, substantially reducing the reverse heat leak during unpowered storage operations.

Figure 11 is a QFD for the thermal transport technologies. All three major candidates are sufficiently close in their score that none can be eliminated without further investigation.
As a result of this survey, three technologies were considered appropriate for inclusion in subsequent trade studies:

- the heat pipe, optimized for the appropriate operating temperature.
- Thermal Pyrolytic Graphite.
- copper.

4.4 Acoustic Emission Control

The general requirements specify that the noise emissions from all five freezer classifications must meet NC-40 (with a goal of NC-30) at .62 meters (2 feet) from the equipment boundary in any octave band between 63 Hz and 8 kHz. The noise criteria (NC) levels are specified in NASA-STD-3000 and JSCM 8080 Standard "Acoustic Noise Criteria", the principal references.

In all five of the freezer classifications, acoustic noise emissions are expected from several sources:

- Cooling Fans
- Cooler
- Flow induced noise (gas in interconnect lines)

Excellent Benefits:	● 9
Good Benefits:	○ 3
Limited Benefits:	△ 1

	Heat Pipe	Pyrolitic Graphite	Metal (Copper)
Technical maturity	△	○	●
Longevity/Reliability/Maintenance/Robustness	○	●	●
Commonality	△	●	●
Maintenance on ground	○	○	○
Low mass	●	●	△
Low volume	●	●	○
Low power demand for adequate capacity	●	○	△
Low vibration generation (also impacts noise)	●	●	●
Low noise generation (impacted primarily by heat rejection)	○	●	●
Electromagnetic compatibility	●	●	●
Electromagnetic interference	●	●	●
Environmental compatibility	○	●	●
Safety	○	●	●
Power-off timeline	●	○	○
Sample temperature regulation	●	○	△
Launch/landing survivability	○	●	●
Technical Importance Rating	92	114	102
Normalized Technical Importance	30	37	33

Figure 11: Thermal Transport Subsystem QFD

The most challenging problem identified during the technology assessment is the rejection of heat to an air medium subject to the low noise requirements. To achieve the necessary heat transfer at the cooler's rejector exchanger, forced convection induced by fans is required, especially in the zero-G environment. The goal is to reject the heat using the lowest flow rate to minimize the fan motor and air turbulence noise emissions. Efficient compact air heat exchangers and quiet and efficient fans are needed to meet this challenge.

The recommended approach to meeting the acoustic emissions requirements is to identify and then eliminate or attenuate the source of noise at the point of generation. This approach prevents the problem from propagating to involve other elements of the structure. For example, the component noise from fans and motors must be isolated so that no harmonic coupling occurs to amplify the component noise. Fans and motors emit sound pressure waves directly and can also potentially excite the resonant frequency of the enclosure parts in their proximity producing a secondary source of noise. The noise spectrum emitted is the result of:

- turbulent air motion over the external heat exchangers and air flow through duct passages, filters and dampers.

- vibration induced resonant excitation of the structure from motors.

Fan noise abatement begins with the selection of fans that have noise emission profiles which compare favorably with the NC-40 requirements as shown in Figure 12. The contractor team evaluated the expected noise emissions of various fan technologies against the spectral requirements of the NC-40 specification and concluded that tube axial fan technology is the recommended choice for the air media rejection application.

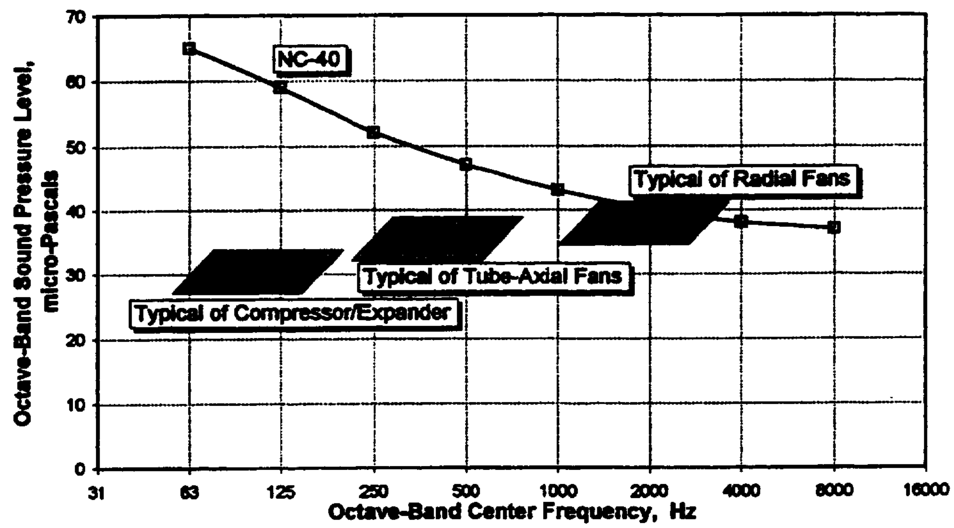


Figure 12: Power Spectral Density of Candidate Hardware

Tube-axial fans are generally quiet and move relatively large quantities of air, but do not accept high back pressures. Radial fans (sometimes termed "blowers") have a much higher capacity to work against back pressure but do not move the same amount of air for an equivalent power compared with a tube axial design.

Fan designs have inherent spectral emissions characteristics due to their mechanical and airflow dynamics. The fundamental frequency and amplitude of acoustic noise is a function of the rotational speed of the motor, the number of blades on the air mover, and the physical symmetry of the fan structure. The tube-axial fan generally has few blades and nearly symmetrical and open inlet and outlet apertures. This results in low frequency and low amplitude noise generation. In contrast, radial fans have large numbers of blades and non symmetric outlet apertures. The result is significant pressure wave generation.

The Shuttle Orbital Refrigerator/Freezer (OR/F) is an example of a space freezer system designed using high velocity, high back pressure radial fans to remove heat from the condenser, compressor and electronics. This system produced an acoustic output that combined with the resonant characteristics of the metallic enclosure to produce undesirable noise emissions. The OR/F system was retrofitted with an acoustic muffler which increased the envelope volume requirement by a full standard mid-deck locker. Table 4 shows the fundamental noise frequencies emitted by these two types of fans together with the sound pressure level allowed by NC-40.

Fan Type	Motor Speed	Nº of Blades	Geometry	Frequency Generated	Allowed by NC-40 (SPL)
Tube-Axial	3600 rpm	5	symmetric	300 Hz	50 dB
Radial	3600 rpm	30	asymmetric	1800 Hz	38 dB

TABLE 4 : Acoustic Characteristics of Tube-Axial and Radial Fan Technologies

Fan components which meet both the thermodynamic and acoustic requirements are available in the current commercial and military standard grade technology. The contractor team has concluded that this equipment, modified by replacing the bearing lubricant to assure reliable low temperature operations, could be used to meet the space flight freezer standards. Special fan designs are possible, at increased cost, to further reduce acoustic emissions by reducing motor speed, reducing blade count and incorporating serration on the back side of the blades. But, for cost control, the contractor team recommends adapting off-the-shelf fan equipment for the brassboard evaluation.

In summary, the following design practices are recommended to control the freezer acoustic noise emissions to the NC-40 (with a goal of NC-30) levels. In the case of each noise emission source, effort must be put forth to reduce or remove the source.

- Cooling Fans - The cooling of the compressor and expander assemblies is accomplished by one of two methods, depending on the host spacecraft. In the case of applications which include a liquid cooling loop, there will be a closed loop heat exchanger coupled to the spacecraft coolant loop. This method of cooling precludes the need for external cooling fans. In some cases, the pressure of the coolant loop must be supplemented by a small positive displacement pump. The design of these pumps is such that very low levels of acoustic or vibratory energy are produced. The pump would be mounted in a vibration dampened mount to isolate it from the other structure.

  As discussed above, when liquid cooling is not available, the system must be cooled by forced air convection. The fans for this application must be selected to produce minimal acoustic output, and the remainder of the system design (i.e.: ducts, heat exchangers, valves) tailored to use a low velocity and low turbulence air distribution method.

- Ducts - Duct geometries must be designed to produce minimal turbulence and to prevent the formation of standing waves (acoustic resonance conditions). Transition sections are generally needed to modify the flow velocity at the fan and heat exchanger inlet and outlets. The duct geometry must minimize the presence of sharp transitions to avoid turbulence or sonic flow conditions (i.e.: whistling) at orifices. Mechanical components exposed to the air stream must also be designed so that they are not susceptible to harmonic excitation.

- Cooler - The cooler compressor produces vibration resulting from the harmonic motion of the piston. The cooler can be configured to effectively self-cancel vibration by mounting two opposing compressors in-line. In the case of a singe piston arrangement, the momentum can be canceled by a passive or active balancer which matches the compressor characteristics. The expected frequency of a Stirling compressor driver is approximately 60 Hz, a favorable octave (higher allowable decibel level) in the NC-40 specification. Results of testing on the STC compressor prototypes have indicated that no extraordinary measures should be required to maintain this noise source below specification. Pressure pulsations from the compressor drive the expander piston in a linear reciprocal motion. In low vibration split cooler configurations, this kinetic energy is canceled by an in-line balance motor which matches the mass, velocity and phase of the expander piston motion. This potential noise source has not proven problematic in past STC designs but must be considered in the mounting and attachment of the expander assembly to the conditioned volumes to avoid coupling with the enclosure surfaces.

- Flow generated noise - In split cooler configurations (i.e.: the compressor and expander in separate assemblies), the pressure and flow conditions within the tubing interconnecting the compressor to the expander can create mechanical excitations which are potential noise sources. This tubing is normally insulated for both thermal and mechanical considerations and is not expected to represent a significant source of acoustic noise.

4.5 Electronics

The electronics drive and control the cooler, and they interface with the vehicle power bus and data systems. Existing space qualified cooler electronics were designed for satellite sensor systems which need active vibration control. These systems typically include digital signal processing capability and microprocessor controllers to measure and analyze the vibration and issue feedback control signals to the vibration canceling balance motors in real time. The resulting electronics are complex and expensive, and require significant power and volume.

Lower cost, commercial based cooler systems, including the Sunpower Stirling Orbiter Refrigerator/Freezer (SOR/F) cooler, are driven by a simpler square wave drive signal with no digital electronics for vibration measurement and active damping. However, the square wave drive signal includes harmonic content that can produce unwanted vibration and acoustic emissions from the cooler housing and support structure. The latest generation of cooler from Stirling Technology Company (STC) uses rack mounted linear amplifiers and sinusoidal drive signals to eliminate the unwanted harmonic content of the motor drive signal.

To minimize vibration without the need for separate vibration cancellation motors, both the Sunpower and STC coolers are designed to be mechanically balanced and have been configured with opposing compressors to cancel the linear momentum. A preliminary analysis of the STC cooler test data indicated that the specification for the five freezer classifications would accommodate the vibration levels of the coolers without the need for exotic active vibration controllers.

Although no minimally configured, low cost, space qualified cooler electronics were identified during the technology survey, the assessment team concluded that electronics technology development is not required to meet the specification for the five freezer classifications. The technology demonstration brassboard will need an electronics package which, as a minimum, drives the cooler linear motors using a quasi-sinusoidal signal to limit the harmonic content. Reasonable attention to volume, power, and complexity should permit these electronics to meet specifications without new technology.

4.6 Moisture Control

The specification states that the equipment be capable of controlling frost build-up with a minimum amount of human intervention and that specimens be maintained at their storage temperature throughout the defrost cycle. Crew time and consumables needed to implement the moisture control system must be minimized. The amount of moisture which is captured inside the enclosure during the freezer operations will depend on the sample transfer access design, cabin humidity, the sample access frequency, the percentage of the internal freezer air exchanged with cabin air, and the effective area of the internal freezer surfaces exposed to the ambient while the freezer is open. Although moisture control was not treated as a major subsystem nor traded off as rigorously as the other subsystem technologies in this study, a preliminary survey of potential moisture control techniques is included in this report.

In the -20/-70°C freezers, the moisture control system must address the formation of frost from atmospheric water vapor and any moisture that is released from the samples. The sample containers are expected to be sealed (vapor tight) so most of the load is expected to come from accessing the freezer and from penetrations and seals. In the cryogenic temperature freezers, liquefaction of atmospheric gases, especially oxygen, is an added concern. The two temperature regimes require different approaches to moisture control due to the different volumes of the enclosures and the expected sample access scenarios.

The primary goal in both temperature regimes is to minimize the amount of moisture that enters the cold space when sample containers are inserted into or removed from the freezer. The volume of the cryogenic freezer is small enough to consider using a nitrogen gas purge to preclude the introduction of oxygen and water vapor. The small sample vials used with the cryogenic freezer can be transferred using a small opening or an airlock which limits the exposed cold space. In the -20/-70°C freezers, previous space-qualified designs have had hinged doors that opened to expose the entire cold volume to the ambient. The door and internal surfaces attract and condense water vapor which is then captured inside the cold space. To minimize this trapped moisture, the door should be designed so as to minimize exposure of the cold interior surfaces to ambient air. This approach has not been implemented on the previously flown space freezer systems.

Once the moisture has been introduced into the cold volume, a moisture capture system, such as a cold trap or desiccant, must control the location where the moisture will migrate and provide the capability to periodically remove the moisture from the enclosure. In convective systems, moisture inside the enclosure migrates to the coldest surface, typically the heat exchanger, where frost accumulation insulates the heat exchanger and impedes heat transfer. As the frost layer builds on the heat exchanger surface, the cooler must compensate by operating at a lower temperature, with resultant lower efficiency, to produce the same cooling effect. If the cooler was held at the same temperature, the increased thermal resistance of the frost would slowly diminish the amount of heat removed, the enclosure temperature would increase, and the other freezer surfaces would begin to attract and retain the moisture. This frost or ice would cause slides and other mechanisms to stick and would also decrease the useful internal volume of the freezer. However, if the heat exchanger is kept sufficiently colder than the other surfaces, frost will eventually sublimate and migrate to the heat exchanger surface. The transport of frozen moisture around the enclosure is a slow process that requires energy from fans and/or heating, so minimizing the amount of moisture that enters the cold space is key to effective moisture control.

If heat were distributed to the specimens using a conductive rather than convective transport system, the heat transfer efficiency would be less sensitive to frost accumulation on surfaces since the heat transfer path does not cross the frost layer. However, the build up on moving parts and the sample containers would still require a means to collect and remove the moisture from the cold space. Removing a small layer of widely distributed frost is potentially a more difficult problem than removing a thick layer that has accumulated in a single location.

Several moisture management concepts were considered during the technology assessment including: cold traps, desiccants, purge gases, and manual substitution of heat exchanger surfaces.

The cold trap attracts moisture to a surface which is colder than the primary acceptor heat exchanger. A thermoelectric cold trap could create a relatively small heat exchanger area inside the freezer which was colder than the heat acceptor surface, thus drawing the condensate to itself. At intervals, the cold trap would be isolated from the rest of the unit and raised in temperature to re-process the water into either the cabin atmosphere or the waste water reclamation system. Studies and analyses are required to determine how much surface area is needed and how cold the cold trap must be to capture sufficient moisture to maintain the acceptor surfaces frost free. Design studies must also address how and how often the defrost cycle should operate, and how the water would be returned to the external systems.

Desiccants can also capture and hold atmospheric water. Literature on the performance of available desiccants at the freezer temperatures is limited. OSS has performed in-house desiccant testing at $-20\,^{\circ}C$, but additional testing is needed to understand the usefulness of these materials at colder temperatures. In general, desiccant packs impose a weight penalty and must be compared with active approaches such as a cold trap. Desiccants require crew work load to change out and/or recharge if the desiccant quantity for the mission cannot be processed completely on ground. Although regeneration of a desiccant takes power, it needs fairly low grade heat and could potentially be dried out using waste heat from the cooler.

Assuming the availability of dry gaseous nitrogen on the vehicle, an N_2 purge system precludes the entrance of moisture and condensable atmospheric gases into the enclosure. This approach may not be practical for the larger enclosures due to the amount of nitrogen required, but it is a primary technology for the cryogenic freezers. In the $-183\,^{\circ}C$ and $-196\,^{\circ}C$ freezers, the liquefaction of atmospheric gases, especially oxygen, must be prevented for safety reasons. The liquefaction temperature of nitrogen is $-196\,^{\circ}C$ (all liquefaction temperatures presume one atmosphere pressure); thus, the N_2 purge gas system can maintain a frost free $-183\,^{\circ}C$ freezer and permit the sub-cooling of the freezer by $12\,^{\circ}C$ to support the un-powered sample storage operations. In the $-196\,^{\circ}C$ Quick/Snap freezer, the sample container is required to be at $-196\,^{\circ}C$ requiring the cooler acceptor surface to be below this temperature to allow for temperature losses in the conduction transport. This could require use of a binary gas with a reduced liquefaction temperature. Performing cryofixation at a temperature several degrees above the nominal $-196\,^{\circ}C$ is an alternative that should be considered.

The physical removal and replacement of the frost encrusted heat exchanger is another way to periodically purge the system of ice buildup. The concept requires a quick disconnect interface between the cooler and the heat exchanger to allow the change out with a minimum of crew time.

In summary, moisture management is a challenging technology area that has not been adequately addressed in several previous spacecraft freezer systems. The approach to solving this problem is contingent on the cooler and enclosure performance as well as the operations scenario for the on-orbit and ground processing functions. The concept configurations considered during the technology assessment will be further scrutinized during the technology development and brassboard testing phase of the effort.

5.0 TRADE STUDIES

Having analyzed the requirements and surveyed the technology to determine the most promising approaches, trade studies for each of the five freezer classifications were performed. The purpose of the trade studies was to assist in determining which freezer systems need technology developments to meet their performance requirements. The system analyses are based on the concepts illustrated with each freezer system presented in this section.

The system trade studies were performed with consideration of candidate technologies in the areas of cooler, insulation, and thermal transport. The subset of technologies traded varies slightly from one system to another because of the broad temperature range spanned by the set of freezer classifications. The technologies used in the trade studies were selected based on subsystem analyses which included a screening for viability at the specified operating temperature and a QFD analysis against the customer requirements. Table 5 outlines the technology candidates for each freezer classification along with pertinent material properties. For each feasible combination of cooler, enclosure, and thermal transport technology, a conceptual system configuration was submitted to a steady state thermal model to determine how well it met the weight, volume and power budgets. For each of the five freezer classifications, thirty-six cases were analyzed, representing every feasible combination of the four cooler technologies, three enclosure technologies, and three thermal transport technologies which emerged from the technology survey described in Section 4.0.

	Cooler Technologies	Insulation Technologies	Thermal Transport Technologies
-20°C Storage Freezer	Thermoelectric Brayton Cycle Pulse Tube Stirling Cycle	R-30 Metal Skin Panel ρ = 233 kg/m^3 R-60 Box-in-Box ρ = 435 kg/m^3 R-105 Polymer Panel ρ = 242 kg/m^3	Copper: k = 398 W/m-C ρ = 8954 kg/m^3 Heat pipe: k = 6000 W/m-C ρ = 4477 kg/m^3 TPG: k = 1200 W/m-C ρ = 6500 kg/m^3
-70°C Storage Freezer	Enhanced Stirling Brayton Cycle Pulse Tube Stirling Cycle	R-30 Metal Skin Panel ρ = 233 kg/m^3 R-60 Box-in-Box ρ = 435 kg/m^3 R-105 Polymer Panel ρ = 242 kg/m^3	Copper: k = 398 W/m-C ρ = 8954 kg/m^3 Heat pipe: k = 6000 W/m-C ρ = 4477 kg/m^3 TPG: k = 1200 W/m-C ρ = 6500 kg/m^3
-70°C Freeze Dryer	Enhanced Stirling Brayton Cycle Pulse Tube Stirling Cycle	R-30 Metal Skin Panel ρ = 233 kg/m^3 R-60 Box-in-Box ρ = 435 kg/m^3 R-105 Polymer Panel ρ = 242 kg/m^3	Copper: k = 398 W/m-C ρ = 8954 kg/m^3 Heat pipe: k = 6000 W/m-C ρ = 4477 kg/m^3 TPG: k = 1200 W/m-C ρ = 6500 kg/m^3
-183°C Cryogenic Storage Freezer	Enhanced Stirling Brayton Cycle Pulse Tube Stirling Cycle	R-60 Box-in-Box ρ = 435 kg/m^3 R-105 Polymer Panel ρ = 242 kg/m^3 R-2300 MLI ρ = 155 kg/m^3	Copper: k = 398 W/m-C ρ = 8954 kg/m^3 Heat pipe: k = 6000 W/m-C ρ = 4477 kg/m^3 TPG: k = 1200 W/m-C ρ = 6500 kg/m^3
-196°C Cryogenic Quick/Snap Freezer	Enhanced Stirling Brayton Cycle Pulse Tube Stirling Cycle	R-60 Box-in-Box ρ = 435 kg/m^3 R-105 Polymer Panel ρ = 242 kg/m^3 R-2300 MLI ρ = 155 kg/m^3	Copper: k = 398 W/m-C ρ = 8954 kg/m^3 Heat pipe: k = 6000 W/m-C ρ = 4477 kg/m^3 TPG: k = 1200 W/m-C ρ = 6500 kg/m^3

TABLE 5 : Combinations of Technologies Analyzed for the Five Advanced Freezers

5.1 Model Methodology

The system analysis was performed using an EXCEL 5.0 spreadsheet with design optimization capabilities. The model consists of integrated files or "books" used to organize unique sets of data including active heat loads, mass and volume baselines, system parameter and analysis workbook, the optimization workbook, and the macro file which runs the thirty-six analyses for each freezer classification. For each combination of technologies, the EXCEL Solver algorithm was used to seek a solution to a set of linear equations with refrigerator system power minimization as the objective function with constraints on system cold volume and mass.

The model is constrained by meeting the cold volume requirement and seeking a positive mass margin. Power is minimized by maximizing the insulation thickness with the maximum allowed external volume as a constraint. Should the mass margin be negative during model iterations, the external volume is reduced thereby reducing insulation and structural mass. If insufficient power is available, both mass and power margins will be negative.

A simple schematic of a characteristic refrigerator is shown in Figure 13.

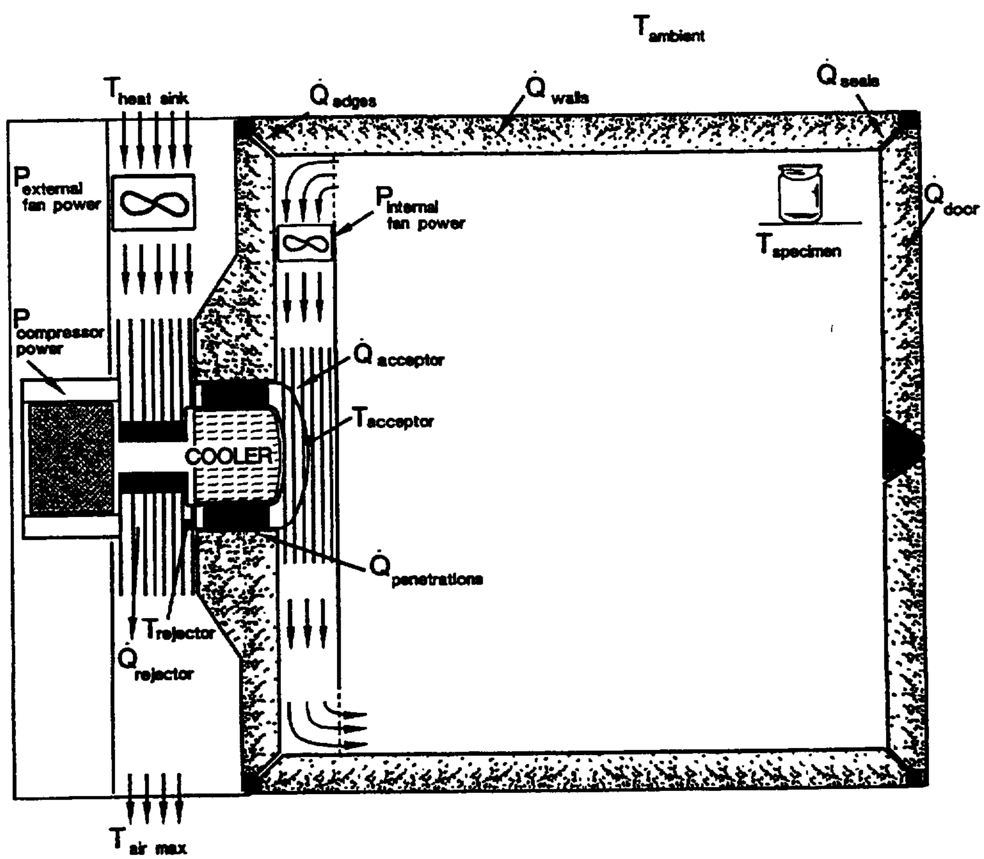


Figure 13: Characteristic Refrigerator System Schematic

The model begins with the cold volume requirement and an arbitrary insulation thickness. A passive heat load is calculated and added to the active heat loads: the specimen heat, internal fans where applicable, door openings, and defrost.

$$Q_{acceptor} = Q_{load} = Q_{passive} + Q_{active}$$

$Q_{passive}$ = *System passive heat leak (W) (walls, penetrations, seals, closeouts, etc.)*

Q_{active} = *System active heat leak (W) (door openings, internal fans, quick freezing, defrost)*

Heat acceptor resistances are calculated providing a cold finger temperature, which is the same as the heat acceptor temperature of the cooler.

$R_{cold\ HX}$ = *Thermal resistive network* $(k_{fins,\ adapter},\ \dot{m},\ A_{frontal})$

$$\Delta T_{cold\ HX} = Q_{load} * R_{cold\ HX}$$

$\Delta T_{subcool} = f$ *(12 hr power off, thermal mass, heat leak)*

$$T_{Acceptor} = T_{setpoint} - \Delta T_{subcool} - \Delta T_{cold\ HX}$$

The total heat lift calculated and the heat acceptor temperature are two of the three variables needed to calculate cooler COP. The third, the heat rejection temperature, is a function of the amount of heat rejection required and the efficiency of the heat rejection system. The amount of heat rejection, itself a function of cooler COP, is the power into the cooler compressor (motor and aerodynamic losses plus PV work on the working gas) plus the active and passive loads into the system.

$$Q_{rejector} = Q_{load} + P_{motor\ power}$$

$R_{hot\ HX}$ = *Thermal resistive network* $(k_{fins,\ adapter},\ P_{fins,\ adapter},\ \dot{m},\ A_{frontal})$

$$\Delta T_{hot\ HX} = Q_{rejector} * R_{hot\ HX}$$

$$T_{Rejector} = T_{rejector\ air} + \Delta T_{hot\ HX}$$

Since the compressor motor power is a function of COP, a nested iterative loop is required in the system analysis to determine the heat rejection temperature. The model is simplified by choosing a heat rejection temperature differential and allowing the size of the rejection heat exchanger to vary parametrically until both heat rejection and noise constraints are satisfied. Using a baseline air heat rejection system designed for approximately 120 watts of heat rejection, a 15°C temperature differential between the heat sink (worst case considered is cabin air at 40°C) and the heat rejector surface is used to determine the heat rejection temperature. The cooler COP was then calculated based on the heat rejection temperature, the heat acceptor temperature and the acceptor thermal load.

$$COP = f\,(T_R\,,\ T_A\,,\ Q_{acceptor})$$

T_A = Acceptor metal temperature

T_R = Rejector metal temperature

With the cooler COP calculated, motor controller and system controller inefficiencies are factored in yielding the total system power needed to operate the system.

$$Q_{Acceptor} = Q_{load} = \text{Specified cooling capacity } (W)$$

$$P_{motorpower} = \frac{Q_{Acceptor}}{COP}$$

$$P_{motor\ controller} = \frac{P_{motor}}{\eta_{controller}}$$

$$P_{system} = \frac{P_{system\ control}}{\eta_{conversion\ electronics}}$$

The equations for cooler COPs were calculated by regression analysis based on the best demonstrated cooler system performance as identified in the survey (Figure 6). Cooler and heat rejection heat exchanger mass and volume baselines identified in the technology survey were used to establish parametric relationships for use in the system analysis.

Figure 14 provides the data used for the baseline cooler systems used in the systems analysis. These baselines parameters are scaled as a function of cooler input power, to provide cooler and heat exchanger mass and volume calculations. The mass and volume of the cooler and heat rejection system, along with the enclosure and electronics packaging, make up the total mass and volume needed to perform the margins analyses. The margin is calculated as:

$$\text{Margin} = (1 - \text{actual/allowable})$$

where the allowable is the mass, volume, or power specified, and the actual is the value calculated based on the input data. For the margin analyses that follow, the power and mass budgets specified are provided in the legend of each system analysis graph (e.g. Figure 17) to aid in calculation of absolute mass or power values, (i.e. Actual = allowable * (1−margin)).

COOLER MASS and VOLUME BASELINES

Stirling		Pulse Tube		Brayton		Thermoelectric	
Dual Purpose Compressor		Dual Purpose Compressor		-70 C System (Creare Est.)		-20 C Freezer System	
Mass: (kg)	4.94	Mass: (kg)	4.94	Mass: (kg)	11.778	Mass: (kg)	13.59
Volume: (m3)	0.0023	Volume: (m3)	0.0023	Volume: (m3)	0.005	Volume: (m3)	0.0031
Length: (m)	0.285	Length: (m)	0.285	Recuperator Length: (m)	0.152	Size:	20"x6"
Diameter: (m)	0.104	Diameter: (m)	0.104	Turndown Ratio	0.7	Input Power: (W)	75
Input Power: (W)	122	Input Power: (W)	122	Input Power: (W)	157	Output Power: (W)	200
Output Power: (W)	83	Output Power: (W)	83	Cooling Capacity: (W)	44		
				Heat Rej Temp: (K)	280		
High Temp. Exp.		High Temp. Exp.		-196 C System (Creare Actuals)			
Mass: (kg)	1.246	Mass: (kg)	0.625	Mass: (kg)	14.04		
Volume: (m3)	0	Volume: (m3)	0.00024	Volume: (m3)	0.002		
Length: (m)	0.153	Length: (m)	0.287	Recuperator Length: (m)	0.585		
Diameter: (m)	0.064	Diameter: (m)	0.063	Turndown Ratio	0.7		
Input Power: (W)	83	Cooling Capacity: (W)	44	Input Power: (W)	135		
Cooling Capacity: (W)	44	Frequency	47	Cooling Capacity: (W)	5		
				Heat Rej Temp: (K)	280		
Low Temp. Exp.		Low Temp. Exp.					
Mass: (kg)	1.445	Mass: (kg)	0.725				
Volume: (m3)	0	Volume: (m3)	0				
Length: (m)	0.186	Length: (m)	0.287				
Diameter: (m)	0.064	Diameter: (m)	0.636				
Input Power: (W)	83	Cooling Capacity: (W)	4.3				
Cooling Capacity: (W)	4.3	Frequency	47				

HOT and COLD SIDE HEAT EXCHANGER BASELINES

Stirling		Pulse Tube		Brayton	Thermoelectric	
Compressor Heat Rej. HX		Compressor Heat Rej. HX		Integral Heat Exchanger	Cold Side Heat Exchanger	
HX Axial flow length: (in)	1.4	HX Axial flow length: (in)	1.4		Mass: (kg)	3.986
Air Flow @ 1 atm: (CFM)	26	Air Flow @ 1 atm: (CFM)	26		Volume: (m3)	0.0046
Air Flow @ 10 psia: (CFM)	39	Air Flow @ 10 psia: (CFM)	39		Input Power: (W)	75
Fan input power: (W)	3	Fan input power: (W)	3		Output Power: (W)	200
Inner HX dia.: (in)	2.6	Inner HX dia.: (in)	2.6			
Outer dia.: (in)	6.6	Outer dia.: (in)	6.6			
Volume: (in3)	47.872	Volume: (in3)	47.872			
Mass: (lbs)	1.445	Mass: (lbs)	1.445			
Troot: (C)	55	Troot: (C)	55			
Tinlet: (C)	40	Tinlet: (C)	40			
Heat rejection basis: (W)	127	Heat rejection basis: (W)	127			

Figure 14: System Baseline Data

5.2 Summary of Trade Study Results

Figure 15 lists the input, output, and margin analysis results for the selected combinations of technologies for the five freezer classifications. Additional analyses were performed to evaluate design and operational excursions which are of interest for specific applications. The system analyses are sensitive to heat rejection temperature and active heat loads. For the -20°C and -70°C systems, heat rejection at different temperatures was investigated. Other systems were evaluated at a nominal heat sink temperature (23°C); however, the cryogenic quick/snap freezer was evaluated assuming water cooling since it operates in a glove box.

The system active loads were calculated based on freezing rates specified for each freezer system. The active loads, summarized in Figure 15, also account for heat loads associated with door openings, internal fans, and system moisture control.

SYSTEM INPUT DATA

System Description	Model Filename	Ref. No.	Operating Environment			External Volume (m3)	Internal Volume (m3)	Mass (kg)	Active Heat Load (W)	Insulation			Thermal Transport		
			T ext. (C)	T set. (C)	T rej. (C)					Type	Density (kg/m3)	R value	Type	Cond. (W/m-K)	Density (kg/m3)
-20C Storage Freezer, Trej=55C, DC=3	LERC12-A	30	30	-20	55	0.600	0.300	100	11.23	Poly Panel	242	105	TPG /Brush	1200	6500
-20C Storage Freezer, Trej=33C, DC=3	LERC12-CA	30	30	-20	33	0.600	0.300	100	11.23	Poly Panel	242	105	TPG/Brush	1200	6500
-20C Storage Freezer, Trej=33C, DC=1	LERC12-N1	30	30	-20	33	0.600	0.300	100	23.68	Poly Panel	242	105	TPG/Brush	1200	6500
-20C Storage Freezer, Trej=8C, DC=1	LERC12-W	30	30	-20	8	0.600	0.300	100	23.68	Poly Panel	242	105	TPG/Brush	1200	6500
-70C Storage Freezer, Trej=55C, DC=1	LERC8-A1	30	30	-70	55	0.900	0.400	289	33.23	Poly Panel	242	105	TPG/Brush	1200	6500
-70C Storage Freezer, Trej=55C, DC=3	LERC8-A	30	30	-70	55	0.900	0.400	289	17.08	Poly Panel	242	105	TPG/Brush	1200	6500
-70C Storage Freezer, Trej=33C, DC=3	LERC8-N	30	30	-70	33	0.900	0.400	289	17.08	Poly Panel	242	105	TPG/Brush	1200	6500
-70C Storage Freezer, Trej=8C, DC=1	LERC8-W	30	30	-70	8	0.900	0.400	289	33.23	Poly Panel	242	105	TPG/Brush	1200	6500
-70C Transporter, Trej=55C, DC=3	LERC8-6A	30	30	-70	55	0.450	0.200	32	6.12	Poly Panel	242	105	TPG/Brush	1200	6500
-70C Freeze Dryer	LERC13-6A	4	30	-70	55	0.300	0.085	73	8.11	Metal Panel	233	30	Cu-brazed	398	8954
-183C Storage Freezer	LERC10-AP	4	30	-183	55	0.042	0.020	123	0.00	MLI	156	2300	Cu-brazed	398	8954
-196C Quick/Snap Freezer, Trej = 55C	LERC11-6A	31	30	-196	55	0.030	0.007	30	1.70	MLI	156	2300	Cu-brazed	398	8954
-196C Quick/Snap Freezer ,Trej = 8C	LERC11-6W	31	30	-196	8	0.030	0.007	30	1.70	MLI	156	2300	Cu-brazed	398	8954

SYSTEM OUTPUT DATA

System Description	Optimum Insulation Thk. (m)	Pass & Active Heat Load (W)	Cold Finger Temp. (C)	Cooler Type	Cooler COP	Comp. Mass (kg)	Expander Mass (kg)	Ext. HX Mass (kg)	Comp. Vol. (m3)	Expander Vol. (m3)	Ext. HX Vol. (m3)
-20C Storage Freezer, Trej=55C, DC=3	0.031	27.70	-28.8	TE	10%	18.42	0.00	6.86	0.0042	0.00000	0.00562
-20C Storage Freezer, Trej=33C, DC=3	0.070	22.80	-28.1	TE	27%	6.71	0.00	0.39	0.0013	0.00000	0.00031
-20C Storage Freezer, Trej=33C, DC=1	0.064	36.40	-32.9	TE	27%	9.12	0.00	0.39	0.0021	0.00000	0.00031
-20C Storage Freezer, Trej=8C, DC=1	0.078	34.83	-32.4	TE	89%	2.66	0.00	0.58	0.0006	0.00000	0.00037
-70C Storage Freezer, Trej=55C, DC=1	0.077	49.64	-84.9	Enh. Stirling	51%	3.92	1.45	2.70	0.0018	0.00057	0.00144
-70C Storage Freezer, Trej=55C, DC=3	0.077	33.47	-80.0	Enh. Stirling	51%	2.86	0.99	1.96	0.0012	0.00039	0.00100
-70C Storage Freezer, Trej=33C, DC=3	0.077	33.47	-80.0	Enh. Stirling	61%	2.23	0.83	1.71	0.0010	0.00032	0.00085
-70C Storage Freezer, Trej=8C, DC=1	0.077	49.62	-84.9	Enh. Stirling	77%	2.60	0.96	1.92	0.0012	0.00038	0.00097
-70C Freezer/Transporter, Trej=55C, DC=3	0.017	34.73	-80.4	Enh. Stirling	51%	0.00	0.00	0.00	0.0000	0.00000	0.00000
-70C Freeze Dryer	0.091	25.30	-83.5	Stirling	33%	3.12	1.16	2.23	0.0015	0.00045	0.00114
-183C Storage Freezer	0.041	0.68	-193.8	Stirling	3%	0.36	0.31	0.88	0.0001	0.00012	0.00048
-196C Quick/Snap Freezer ,Trej = 55C	0.049	4.00	-204.8	Enh. Stirling	7%	2.00	1.72	3.13	0.0004	0.00067	0.00175
-196C Quick/Snap Freezer ,Trej = 8C	0.063	4.00	-204.8	Enh. Stirling	9%	1.64	1.41	2.64	0.0003	0.00056	0.00140

SYSTEM MARGIN ANALYSIS RESULTS

System Description	Cooler Input Power (W)	Enclosure Mass (kg)	Enclosure Ext. Volume (m3)	Totals			Margins		
				Mass (kg)	Volume (m3)	Power (W)	Mass (kg)	Volume (m3)	Power (W)
-20C Storage Freezer, Trej=55C, DC=3	271.1	62.6	0.49	111.4	0.49	452.4	-11%	30%	-126%
-20C Storage Freezer, Trej=33C, DC=3	84.0	81.1	0.66	100.0	0.66	160.0	0%	6%	20%
-20C Storage Freezer, Trej=33C, DC=1	134.2	73.1	0.59	100.0	0.59	238.4	0%	16%	-19%
-20C Storage Freezer, Trej=8C, DC=1	39.0	85.6	0.70	97.2	0.70	89.7	3%	0%	55%
-70C Storage Freezer, Trej=55C, DC=1	96.6	102.3	0.90	122.0	0.90	180.0	55%	0%	10%
-70C Storage Freezer, Trej=55C, DC=3	65.8	102.3	0.90	117.5	0.90	131.6	56%	0%	34%
-70C Storage Freezer, Trej=33C, DC=3	55.1	102.4	0.90	115.9	0.90	114.8	57%	0%	43%
-70C Storage Freezer, Trej=8C, DC=1	64.3	102.4	0.90	117.3	0.90	129.2	56%	0%	35%
-70C Freezer/Transporter, Trej=55C, DC=3	68.2	31.7	0.30	31.7	0.30	135.3	0%	0%	32%
-70C Freeze Dryer	77.0	36.0	0.30	53.0	0.30	149.0	28%	0%	25%
-183C Storage Freezer	20.9	46.0	0.04	82.6	0.04	81.5	49%	0%	69%
-196C Quick/Snap Freezer ,Trej = 55C	114.8	14.6	0.03	29.5	0.03	208.1	0%	10%	-4%
-196C Quick/Snap Freezer ,Trej = 8C	94.0	15.4	0.03	29.5	0.03	175.7	0%	6%	12%

Figure 15: System Model Inputs, Outputs, and Margin Analysis Results

5.3 The -20°C Storage Freezer

Figure 16 shows the conceptual configuration of the -20°C Storage Freezer which is based on usage of a thermoelectric cooler. The system analyses performed show a strong sensitivity to active loads and heat rejection temperatures. The heat rejection temperature for the -20°C system is particularly sensitive due to the dramatic change in COP for thermoelectric coolers, as shown earlier in Figure 6, as temperature differentials increase. Four analyses were performed to evaluate the sensitivity of system performance to the heat rejection temperature and the active heat load as shown in Fig. 15.

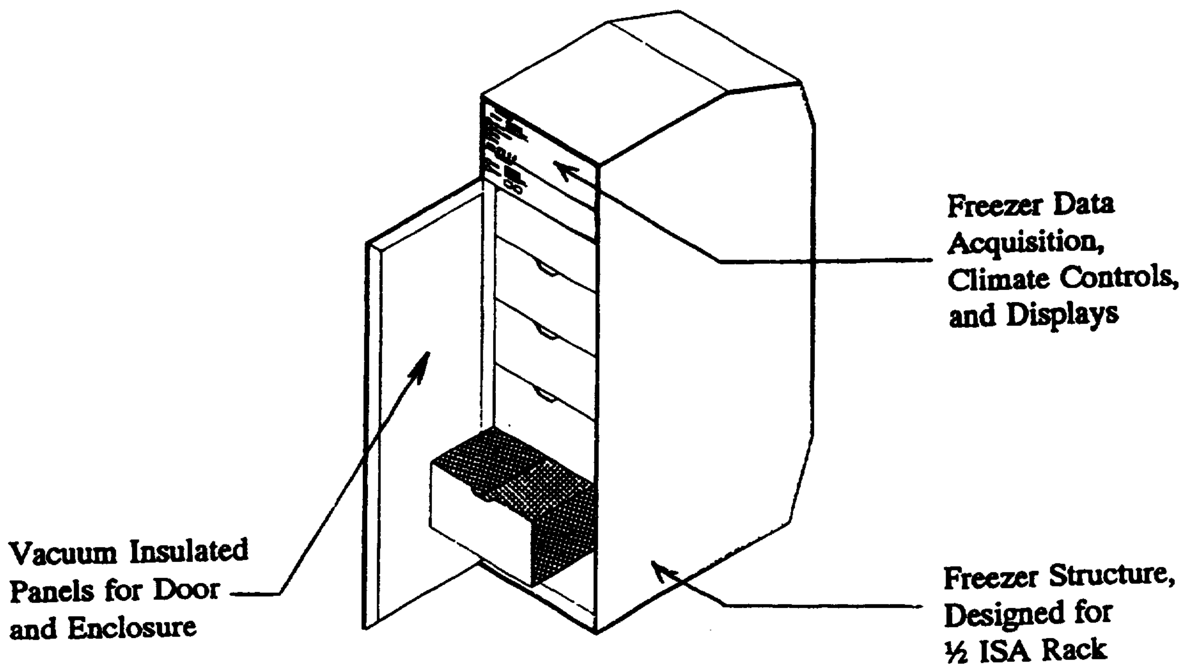
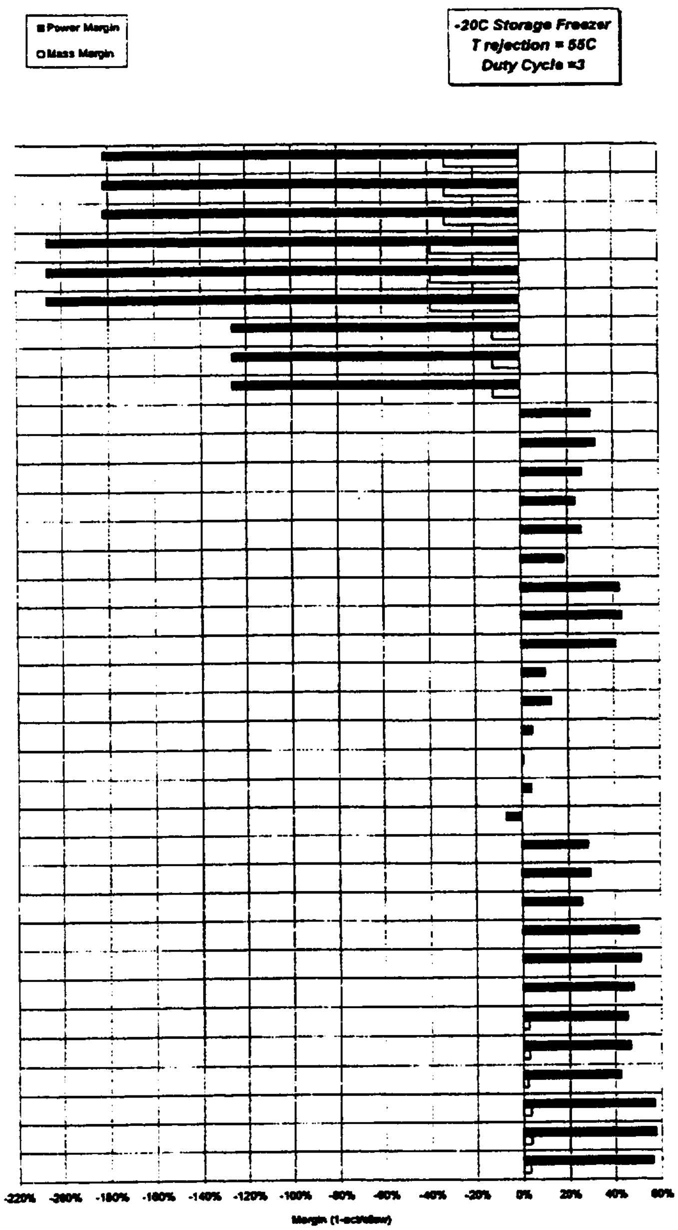


Figure 16: -20°C Storage Freezer

Figure 17 presents the results for a -20°C freezer system with an air heat rejection system. This analysis assumes a heat sink temperature of 40°C and a 15°C temperature differential between the heat sink and the cooler's heat rejection area (i.e. T rejection = 55°C). The vertical axis lists the 36 technology combinations, with the right hand columns designating each combination. The horizontal axis is the degree to which each case meets or fails to meet the system mass and power specification. Positive margin means the system as configured meets or exceeds the requirement; negative margin means that the system fails to meet its budget. The reader is cautioned to notice the scale on the horizontal axis of all margin charts since the scale varies considerably from one freezer classification to another.



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Thermoelectric	R60	TPG
35	Thermoelectric	R60	Heat Pipe
34	Thermoelectric	R60	Copper
33	Thermoelectric	R30	TPG
32	Thermoelectric	R30	Heat Pipe
31	Thermoelectric	R30	Copper
30	Thermoelectric	R105	TPG
29	Thermoelectric	R105	Heat Pipe
28	Thermoelectric	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 17: -20°C Storage Freezer Trade Study Results (Worst Case Air Heat Rejection, Duty Cycle of 3)

The analysis shows that for the -20°C Storage Freezer with the worst case heat rejection scenario (40°C), mass and power requirements can be met using several different combinations of cooler, insulation, and thermal transport technologies. The thermoelectric cooler, however, is unable to maintain positive margins on either mass or power when the average power allocation of 200 watts is used. This is because the thermoelectric COP is very sensitive to differential temperatures, and the heat rejection to air at 40°C produces a temperature differential of approximately 85°C once heat exchange inefficiencies are included. The thermoelectric COP with a 40°C heat sink is approximately 10% and increases to 27% when the heat sink temperature decreases to 23°C. Since the 40°C heat sink is a worst case condition, the peak power allocation of 456 watts will help to reduce the negative margins, although continuous operation at this level is not an option.

A duty cycle of three is used to help evaluate the portion of the heat loads attributable to specimen cool down. The specification requires cooling 100 ml samples from ambient to -20°C in 45 minutes. A duty cycle of one means that samples are placed in the storage freezer every 45 minutes. Thus, a duty cycle of three means that a sample is entered every 3 x 45 minutes or every 2.25 hrs. On average, the power required to support a duty cycle of one (18 watts) can be up to 50% of the total heat load.

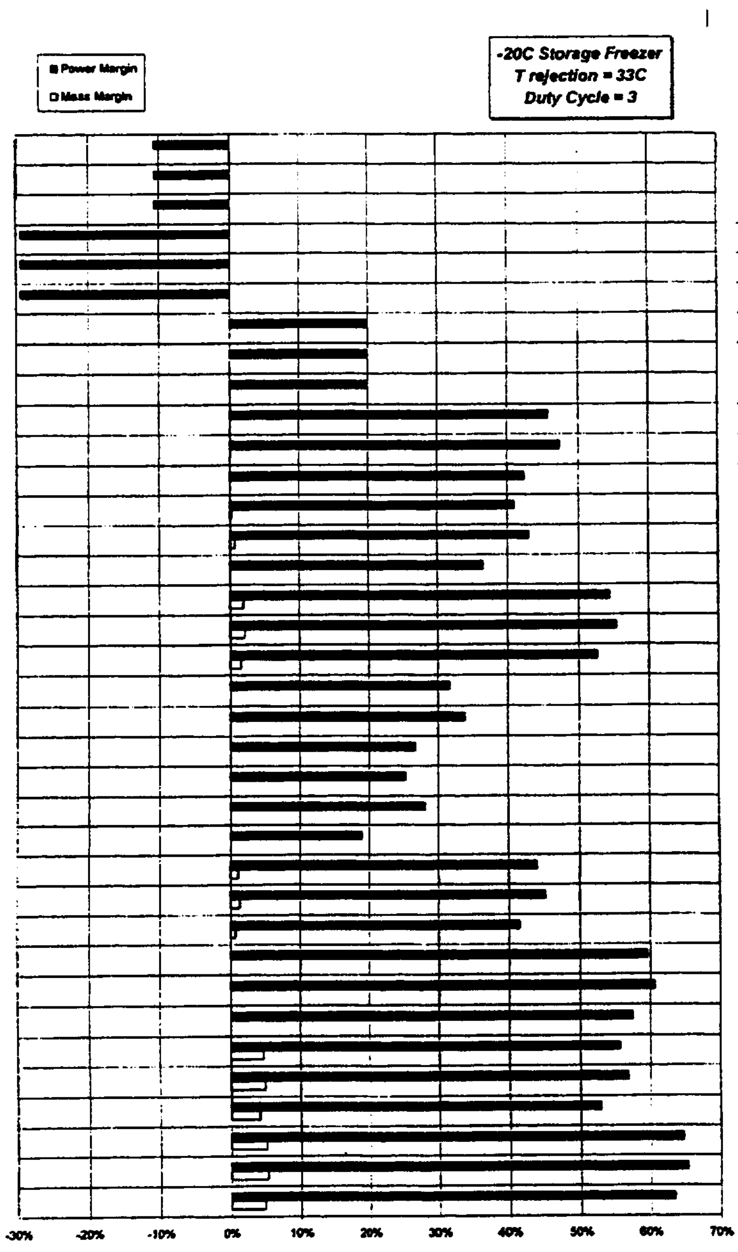
Excellent Benefits: ● 9
Good Benefits: ○ 3
Limited Benefits: △ 1

	Ref. No. 4 (Stirling / Cu / R30)	Ref. No. 30 (TE / TPG / R105)	Ref. No. 22 (Brayton / Cu / R30)	Ref. No. 16 (Pulse T. / Cu / R60)
Technical maturity	○	○	△	△
Longevity/Reliability/Maintenance/Robustness	○	●	○	●
Commonality	●	△	●	●
Configurable temperature and volume to handle up/down payloads	●	●	○	●
Maintenance on ground	○	○	○	○
Low mass	●	△	○	△
Low volume	●	△	○	△
Low power demand for adequate capacity	●	○	○	○
Low vibration generation (also impacts noise)	○	●	○	●
Low noise generation (impacted primarily by heat rejection)	○	●	○	△
Acceptable resonant frequencies	○	●	○	○
Electromagnetic compatibility	●	●	●	●
Electromagnetic interference	●	●	●	●
Environmental compatibility	●	●	●	●
Safety	○	●	○	○
Cooler temperature/capacity operating range	●	●	●	●
Power-off timeline	○	●	●	○
Sample temperature regulation	●	●	●	●
Low power surge at start-up	●	●	●	●
Launch/landing survivability	○	●	○	○
Cold containment susceptibility	○	●	○	●
Configurable temperature and volume to meet on-orbit storage requirements	●	●	●	●
Technical Importance Rating	138	156	118	130
Normalized Technical Importance	25	29	22	24

Figure 18: -20°C Storage Freezer QFD

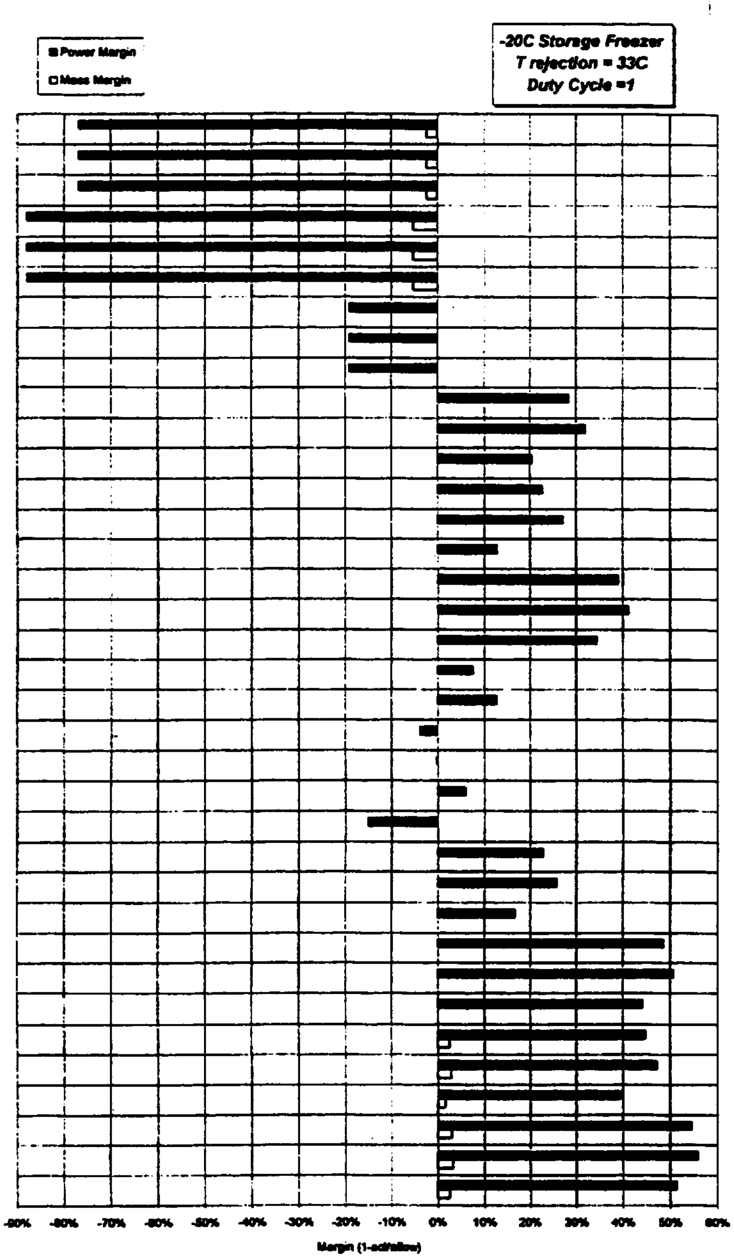
In conjunction with the systems analysis, a QFD analysis was done (Figure 18) to compare the feasible design solutions for each cooler type. When all the requirements are factored in, a thermoelectric cooler with TPG heat transport and R-105 insulation was deemed the best candidate overall because of the advantages in safety, vibration tolerance, vibration production, and reliability; a compact design overcomes the efficiency shortfall in the qualitative analysis. The reduced thermal resistance of TPG lowers the thermoelectric junction temperature difference with resulting benefits in the cooler efficiency. The packaging configuration in the freezer concept used for the system analysis eliminated the long conduction paths that are served by heat pipe technology. Given the relatively short heat transport path length in the concept systems, heat pipe technology was not recommended since the more reliable and simpler TPG technology was able to perform the required thermal transport.

Further evaluation was deemed necessary to investigate different operational scenarios, including nominal heat rejection and heat rejection to a water bus at 8°C. The results for rejection to cabin air under nominal conditions (20 to 25°C) are shown in Figure 19. In this case, the thermoelectric option is viable if the best insulation (R-105) is developed. With the duty cycle increased to one, the TE system is once again unable to provide positive mass and power margins, as shown in Figure 20. With a water bus heat sink, Figure 21 shows that power margins are increased to over 40% even with the most aggressive specimen loading (a duty cycle of one). Mass margins are small across the board indicating that the weight budget for the -20°C system is very tight.



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Thermoelectric	R60	TPG
35	Thermoelectric	R60	Heat Pipe
34	Thermoelectric	R60	Copper
33	Thermoelectric	R30	TPG
32	Thermoelectric	R30	Heat Pipe
31	Thermoelectric	R30	Copper
30	Thermoelectric	R105	TPG
29	Thermoelectric	R105	Heat Pipe
28	Thermoelectric	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

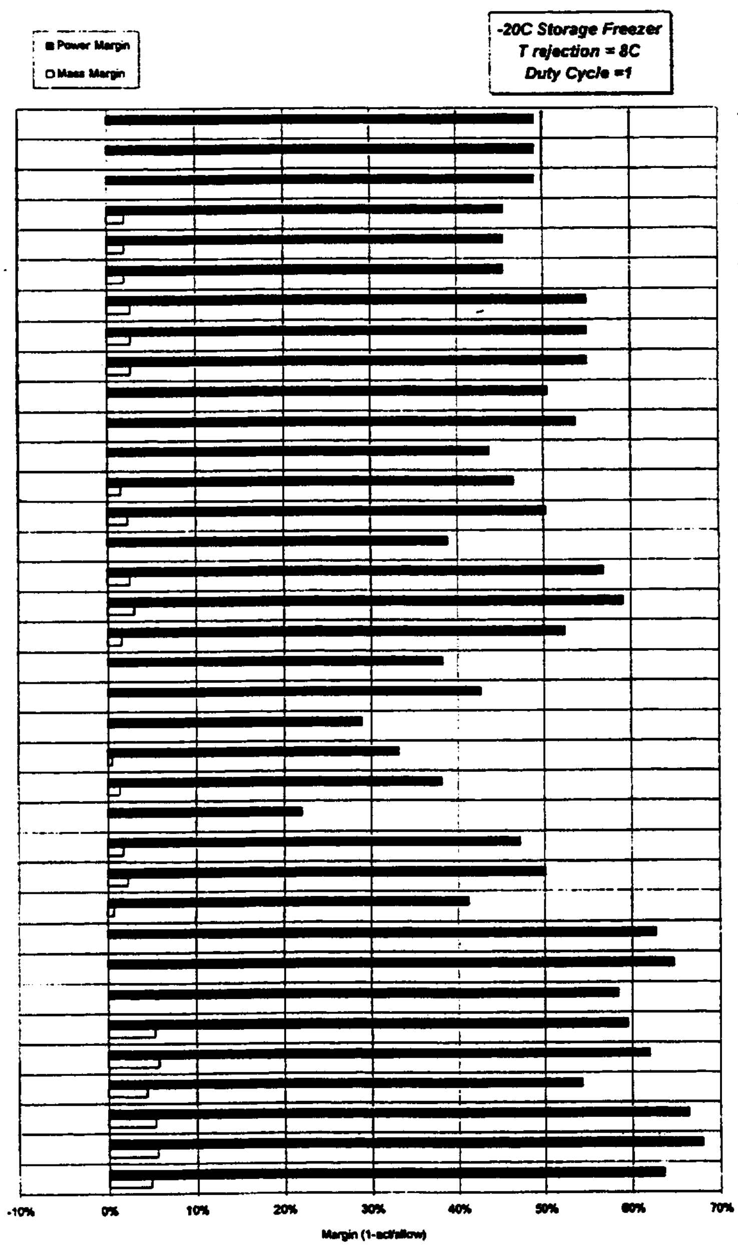
Figure 19: -20°C Storage Freezer Trade Study Results (Nominal Air Heat Rejection, Duty Cycle of 3)



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Thermoelectric	R60	TPG
35	Thermoelectric	R60	Heat Pipe
34	Thermoelectric	R60	Copper
33	Thermoelectric	R30	TPG
32	Thermoelectric	R30	Heat Pipe
31	Thermoelectric	R30	Copper
30	Thermoelectric	R105	TPG
29	Thermoelectric	R105	Heat Pipe
28	Thermoelectric	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 20: -20°C Storage Freezer Trade Study Results (Nominal Air Heat Rejection, Duty Cycle of 1)

In order for the -20°C Storage Freezer to keep its contents at or below the specified temperature during the 12-hour power off time, the cooler would have to "subcool" the enclosure by approximately 5 to 10°C. This added burden requires an additional 15 to 20 watts of power (7.5 to 10% of budget) over the power used in all the system analyses.



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Thermoelectric	R60	TPG
35	Thermoelectric	R60	Heat Pipe
34	Thermoelectric	R60	Copper
33	Thermoelectric	R30	TPG
32	Thermoelectric	R30	Heat Pipe
31	Thermoelectric	R30	Copper
30	Thermoelectric	R105	TPG
29	Thermoelectric	R105	Heat Pipe
28	Thermoelectric	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 21: -20°C Storage Freezer Trade Study Results (Water Heat Rejection, Duty Cycle of 1)

5.4 The −70°C Storage Freezer

Figure 22 shows a concept configuration of the −70°C Storage Freezer. To evaluate the worst case condition, the first analysis assumed that heat would be rejected into the cabin air rather than the lower temperature liquid cooling bus. The design also assumed that the freezer contained phase change material (PCM) in order to maintain the temperature during the 12 hour power off. PCM is a thermal energy storage reservoir that maintains the enclosure temperature by absorbing heat to change its phase from a solid to a liquid. Preliminary estimates suggest that roughly twenty pounds of PCM is needed to maintain the −70°C temperature during the 12 hour power off, but this mass could provide a portion of the internal structure of drawers and walls. Drawers and walls with PCM cores would occupy about 5% of the internal volume; without PCM, they would take up 3%.

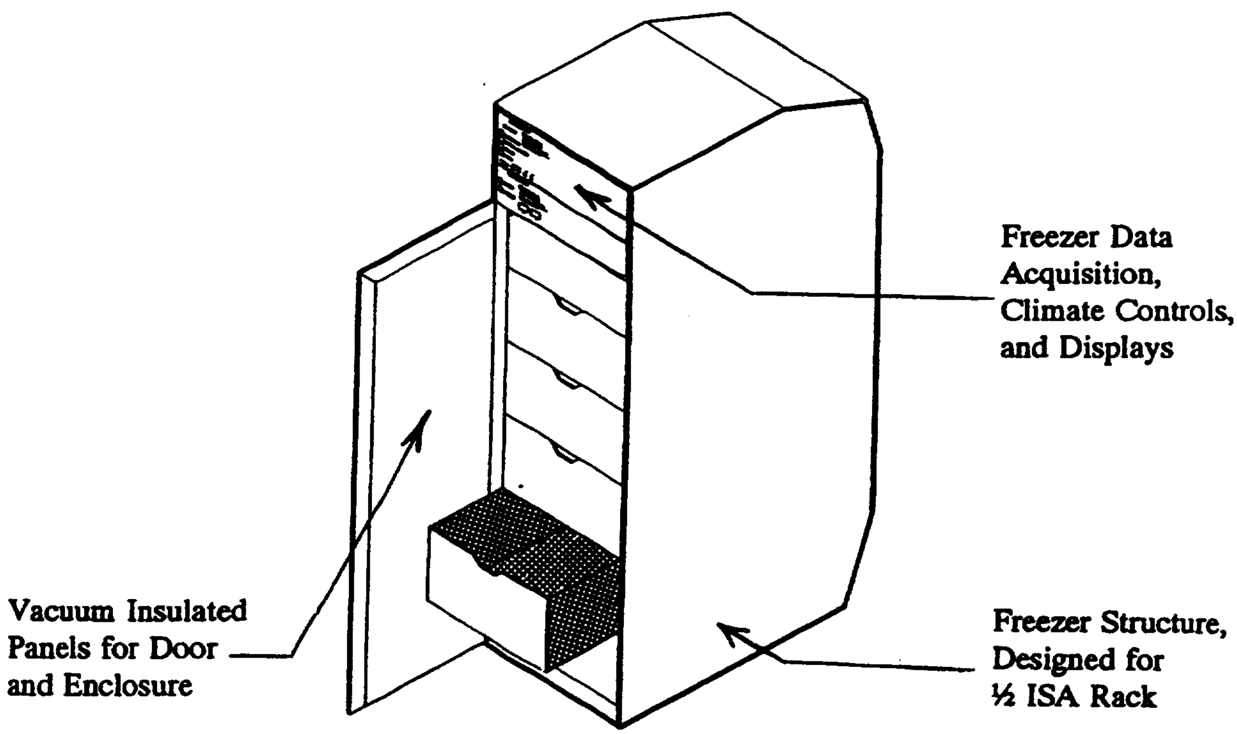


Figure 22: -70°C Storage Freezer

Figure 23 presents the modeling results obtained for the −70°C system with worst case air heat rejection and a duty cycle of one. No configuration using the pulse tube, Brayton cycle, or reference[1] Stirling cycle coolers met all of the specifications. At the known state of development, current pulse tube coolers operating in this temperature range are not efficient enough for this challenging case. The Brayton cycle coolers are also unacceptable but, with reduced active loads, could be utilized with the advanced insulation. With the more aggressive duty cycle active thermal loads, the reference Stirling cycle shows negative power margins of nearly 20% with the

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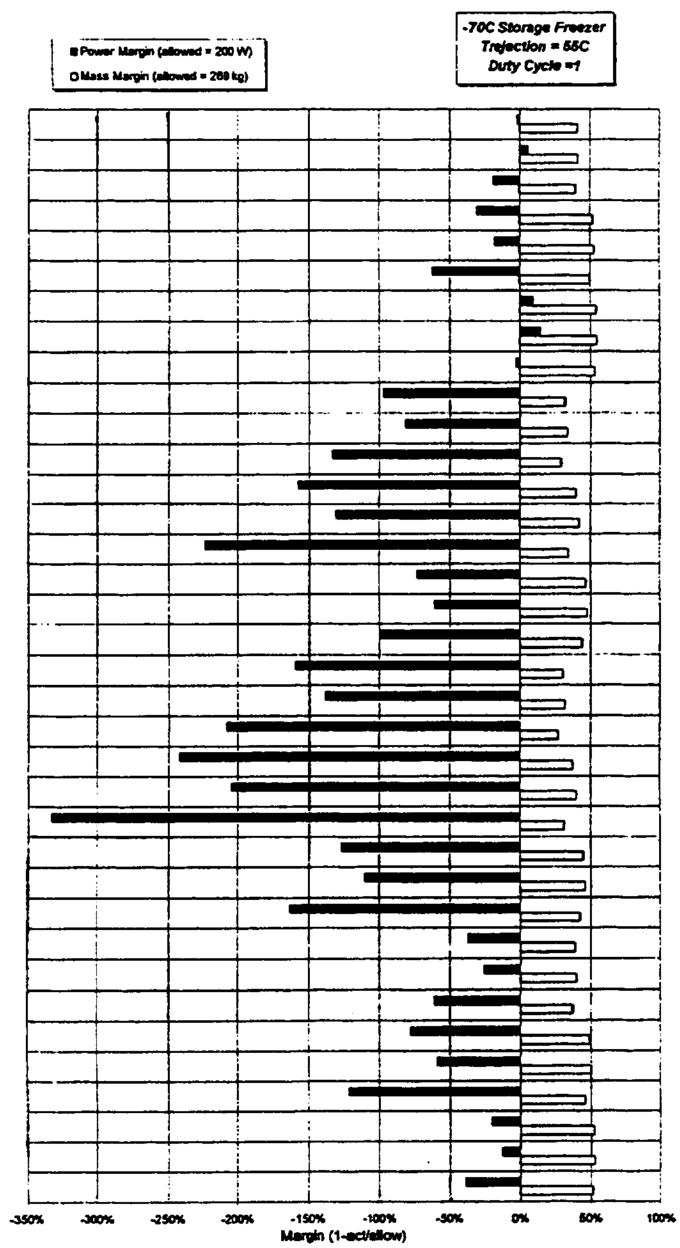
[1] The reference Stirling cycle COP is derived from the curve of Figure 6; the enhanced Stirling has a 39% higher COP.

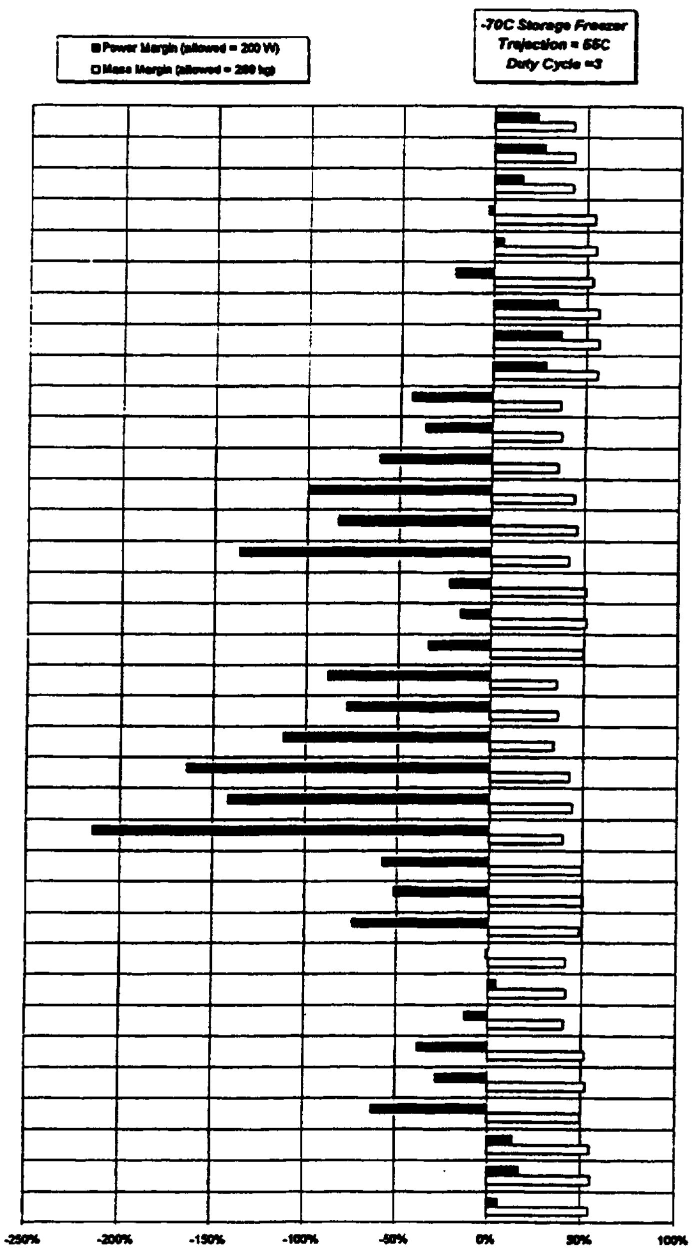
best insulation and thermal transport technologies. An enhanced Stirling cycle, with nearly a 40% efficiency improvement over demonstrated Stirling cycles at −80°C, appears feasible with advancements in insulation technology (R-60 or better). The improvements required to achieve this significantly higher Stirling cooler performance are comprised of compressor motor and low conductive displacer material modifications.



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R30	TPG
32	Enhanced Stirling	R30	Heat Pipe
31	Enhanced Stirling	R30	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 23: -70°C Storage Freezer Trade Study Results (Worst Case Air Heat Rejection, Duty Cycle of 1)

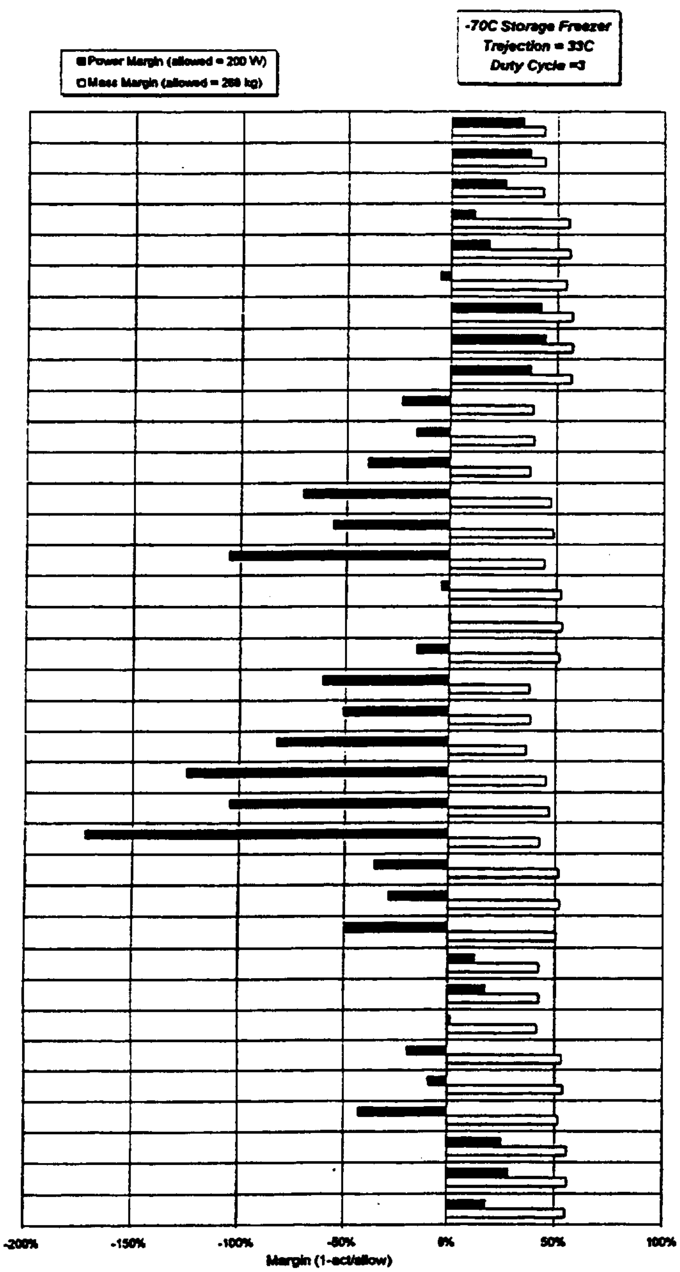
With the duty cycle increased to three (Figure 24), the reference Stirling cooler appears feasible with insulation technology (R-60) believed to be producible with minimal technology development.



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R30	TPG
32	Enhanced Stirling	R30	Heat Pipe
31	Enhanced Stirling	R30	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 24: -70°C Storage Freezer Trade Study Results (Worst Case Air Heat Rejection, Duty Cycle of 3)

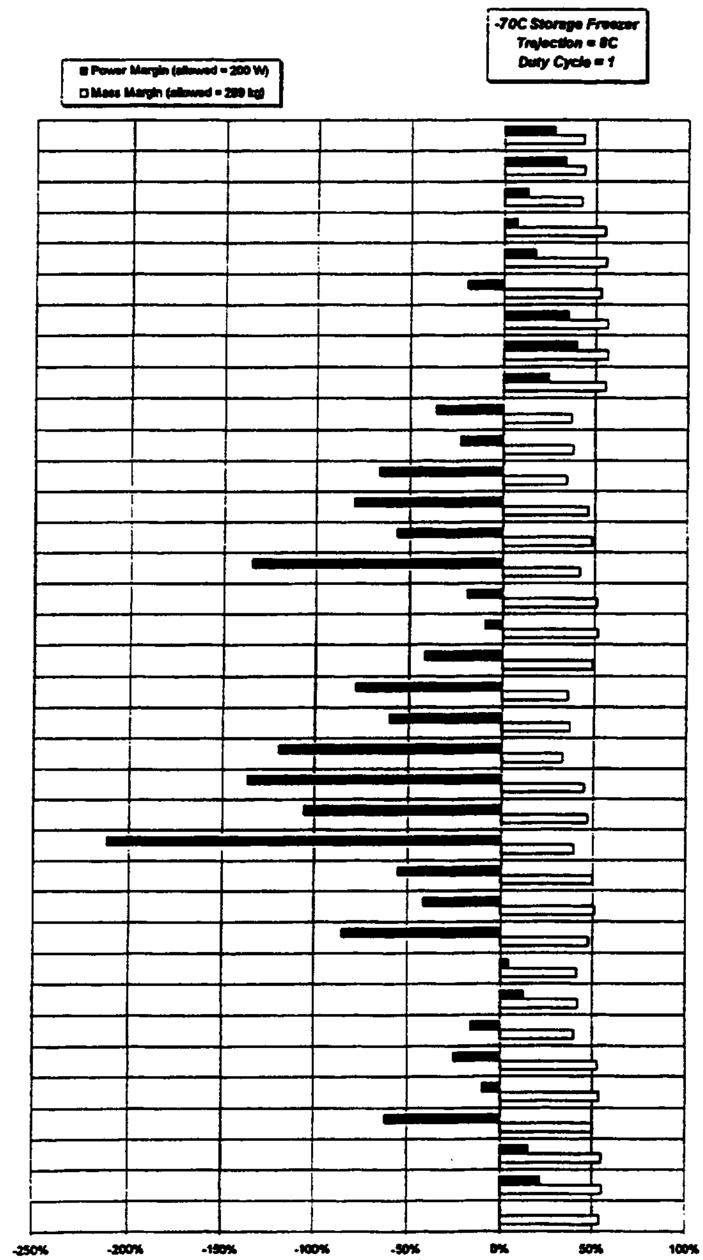
Utilizing nominal operating conditions (Figure 25) with an air heat sink temperature of 23°C, power margins increase on all systems with the potential to use a broader range of cooler technologies (i.e. Brayton cycle with heat pipe and advanced TPG thermal transport and R-105 insulation).



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R30	TPG
32	Enhanced Stirling	R30	Heat Pipe
31	Enhanced Stirling	R30	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 25: -70°C Storage Freezer Trade Study Results (Nominal Air Heat Rejection, Duty Cycle of 3)

If a water bus were available for heat rejection (Figure 26), power margins would again increase, however not as dramatically as was shown for the Thermoelectric -20°C Freezer system. This is because the change in COP for Stirling, pulse tube, and Brayton cycle coolers is not as sensitive to temperature differentials as the thermoelectric cooler. Mass margins are consistently positive indicating sufficient mass allocation. Under these conditions, the system model makes full use of the external volume available to minimize the system power required; this results in insulation thicknesses of approximately 8 cm (3.1 inch).



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R30	TPG
32	Enhanced Stirling	R30	Heat Pipe
31	Enhanced Stirling	R30	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 26: -70°C Storage Freezer Trade Study Results (Water Heat Rejection, Duty Cycle of 1)

A QFD analysis (Figure 27) comparing the most promising case for each of the Stirling and enhanced efficiency Stirling coolers suggested that there are benefits to other system parameters in addition to power and weight that would be provided by the enhanced efficiency Stirling cooler.

Excellent Benefits: ●	9
Good Benefits: ○	3
Limited Benefits: △	1

	Ref. No. 3 (Stirling / TPG / R105)	Ref. No. 30 (En Stirling / TPG / R105)
Technical maturity	●	○
Longevity/Reliability/Maintenance/Robustness	○	○
Commonality	○	●
Configurable temperature and volume to handle up/down payloads	○	○
Maintenance on ground	○	○
Low mass	○	●
Low volume	○	●
Low power demand for adequate capacity	○	●
Low vibration generation (also impacts noise)	○	○
Low noise generation (impacted primarily by heat rejection)	○	●
Acceptable resonant frequencies	○	○
Electromagnetic compatibility	○	○
Electromagnetic interference	○	○
Environmental compatibility	○	○
Safety	●	●
Cooler temperature/capacity operating range	●	●
Obvious layout and operation (foolproof)	●	●
Power-off timeline	○	●
Sample temperature regulation	●	●
Low power surge at start-up	●	●
Launch/landing survivability	○	○
Cold containment susceptibility	○	○
Configurable temperature and volume to meet on-orbit storage requirements	○	○
Technical Importance Rating	105	135
Normalized Technical Importance	44	56

Figure 27: -70°C Storage Freezer QFD

Phase change materials do have potential problems with contamination of samples and/or toxicity should the material leak while in the liquid phase. An excursion on this design explored how the samples would be held below the -70°C temperature during power off without the phase change material. Samples would have to be subcooled 10 to 15°C below their steady state temperature before the power interruption in order that they stay below -70°C for twelve hours without power. This lower set point temperature further stresses the requirements for this already challenging case. The added burden for subcooling the -70°C system requires an additional 20 to 30 watts of power (10 to 15% of budget) over the power used in the system analyses. Weight and power margins grow worse in every case.

A second -70°C storage freezer concept began with the observation that the R-105 material and enhanced efficiency Stirling produced sufficient benefits that the system might be consolidated into a smaller unit. A design study did reveal a feasible design based on these advanced technologies that would fit into an ISPR half-rack. The current external volume requirement for this freezer classification is equivalent to approximately 3/4 of an ISPR rack.

The -70°C freezer as a unit weighs 269 kg. This makes it impossible to use as a transport locker without special handling equipment. A third design excursion explored the possibility of detaching part of the storage volume from the cooler for transportation as a smaller unit appropriate to the Mini-Pressurized Logistics Module (MPLM) and light enough for single person handling (32 kg). Figure 28 shows a conceptual design of this system. This option is contingent upon design of a thermal transport system which can cleanly and simply detach from the enclosure and then reseal the volume to minimize heat and moisture entry. Both are considered tractable problems, but this transporter design may force the selection of certain thermal transport technologies over others (see section 6.2).

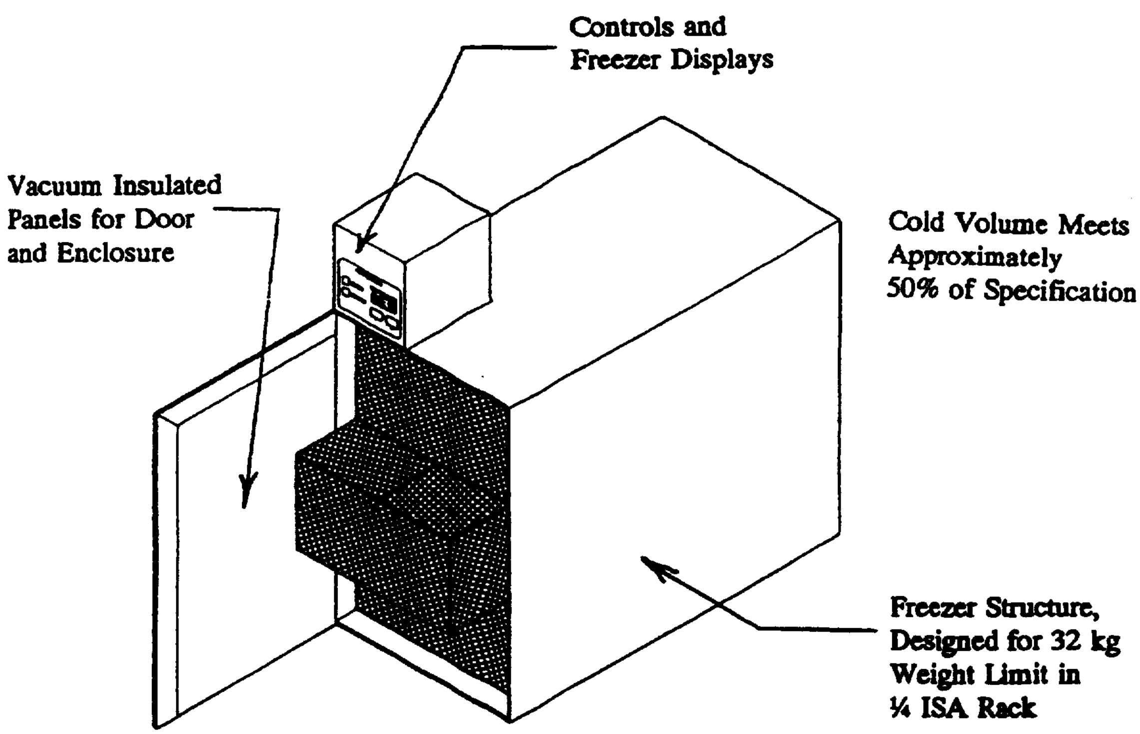
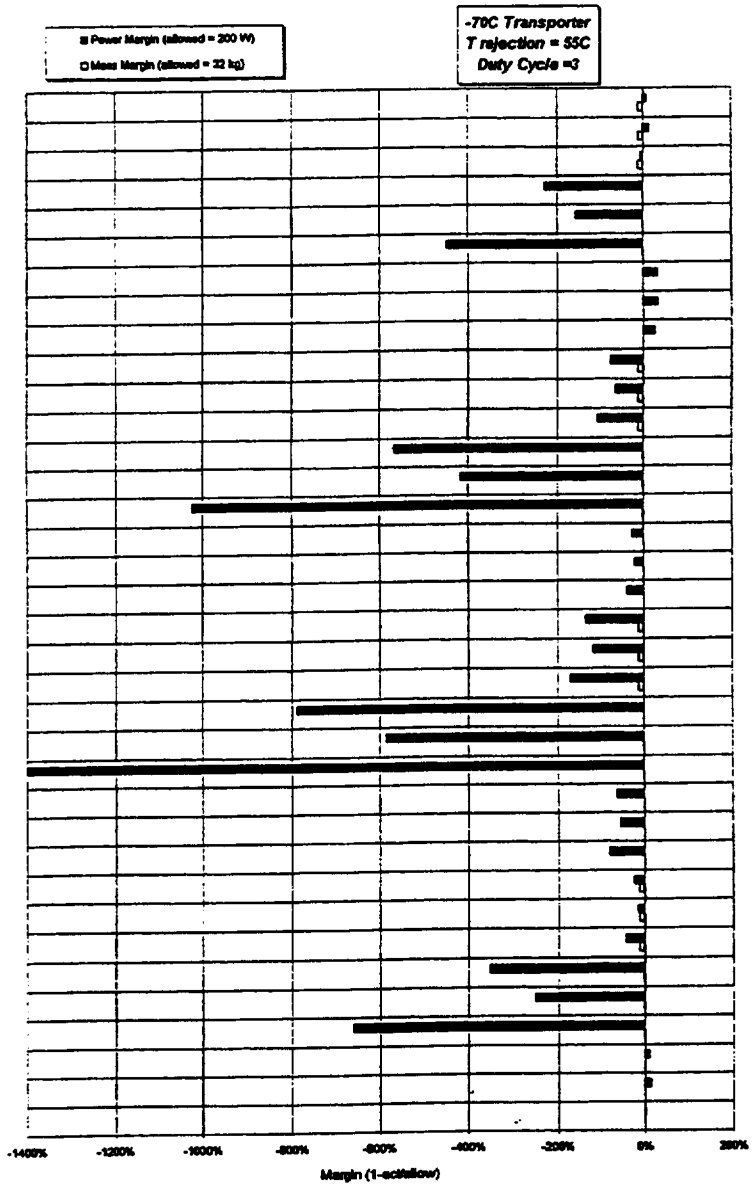


Figure 28: -70°C Transporter

In order to achieve the very low weights for the transporter, wall thicknesses must be minimized. Thin walls in turn require the highest R-value possible and places greater emphasis on cooler efficiency, as shown in Figure 29. Only the Stirling and enhanced efficiency Stirling are able to meet the power and mass requirements with the R-105 insulation. The optimal wall thickness with the enhanced efficiency Stirling is 0.017 m (0.66 in). Although most candidates are eliminated if a 32 kg transporter is required, it should be noted that the enhanced efficiency Stirling cooler coupled with the R-105 insulation can still meet spec with 30-40% power margins.

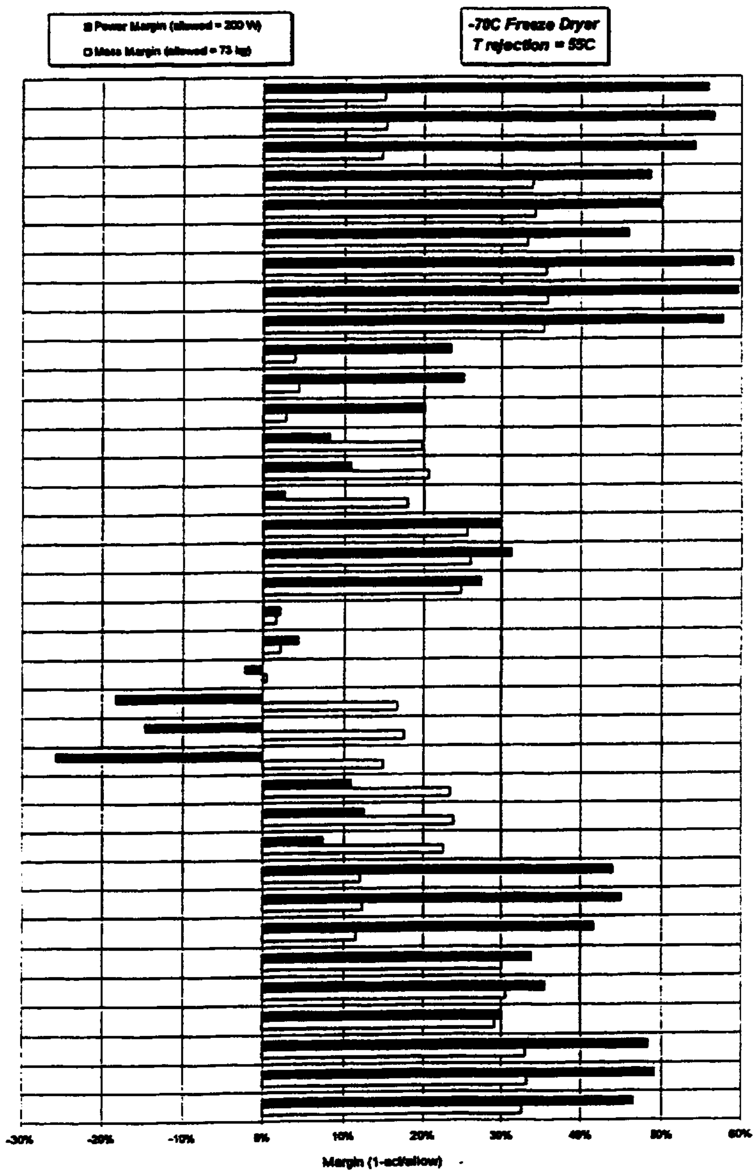


Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R30	TPG
32	Enhanced Stirling	R30	Heat Pipe
31	Enhanced Stirling	R30	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 29: -70°C Transporter Trade Study Results

5.5 The -70°C Freeze Dryer

The modeling results for the -70°C freeze dryer (Figure 30) show that every case, except the relatively inefficient pulse tube cooler coupled with the lowest R value insulation, could meet the weight and power budgets handily. The enhanced Stirling was not considered necessary to include in this analysis. The QFD analysis, Figure 31, nonetheless recommends the use of the higher R value insulation to relieve other system variables.



Reference No.	Cooler Cycle	Insulation	Thermal Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R30	TPG
32	Enhanced Stirling	R30	Heat Pipe
31	Enhanced Stirling	R30	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R30	TPG
23	Brayton	R30	Heat Pipe
22	Brayton	R30	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R30	TPG
14	Pulse Tube	R30	Heat Pipe
13	Pulse Tube	R30	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R30	TPG
5	Stirling	R30	Heat Pipe
4	Stirling	R30	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 30: -70°C Freeze Dryer Trade Study Results

The large margins suggest that meeting the requirements is not challenging. Since samples are placed in the freeze dryer already frozen to -70°C, the drying process can proceed slowly with minimal power. Since the cooling function for the freeze dryer is common with the -70°C storage freezer, this led to a design excursion in which the freeze drying system was incorporated into the rack with the -70°C storage freezer (Figure 32). While this put additional constraints on the volume of the storage freezer, several feasible configurations nonetheless emerged from the modelling study. They all required the higher performance insulation and higher efficiency coolers. Nonetheless, ease of specimen handling and the weight efficiency gained by not requiring a dedicated cooler for the freeze dryer suggest this as an interesting option, presuming the technology is available.

Excellent Benefits:	● 9
Good Benefits:	○ 3
Limited Benefits:	△ 1

	Combo No. 4 (Stirling / Cu / R30)	Combo No. 1 (Stirling / Cu / R105)	Combo No. 22 (Brayton / Cu / R30)
Technical maturity	●	△	○
Longevity/Reliability/Maintenance/Robustness	○	○	○
Commonality	○	●	○
Configurable temperature and volume to handle up/down payloads	○	○	○
Maintenance on ground	○	○	○
Low mass	○	●	○
Low volume	○	●	○
Low power demand for adequate capacity	○	●	△
Low vibration generation (also impacts noise)	○	○	○
Low noise generation (impacted primarily by heat rejection)	●	●	○
Acceptable resonant frequencies	○	○	○
Electromagnetic compatibility	●	●	●
Environmental compatibility	○	○	○
Safety	●	●	●
Cooler temperature/capacity operating range	●	●	●
Obvious layout and operation (foolproof)	●	●	○
Power-off timeline	○	△	○
Sample temperature regulation	●	●	●
Low power surge at start-up	●	●	●
Launch/landing survivability	○	○	○
Cold containment susceptibility	○	○	○
Configurable temperature and volume to meet on-orbit storage requirements	○	○	○
Technical Importance Rating	114	128	94
Normalized Technical Importance	34	38	28

Figure 31: -70°C Freeze Dryer System QFD

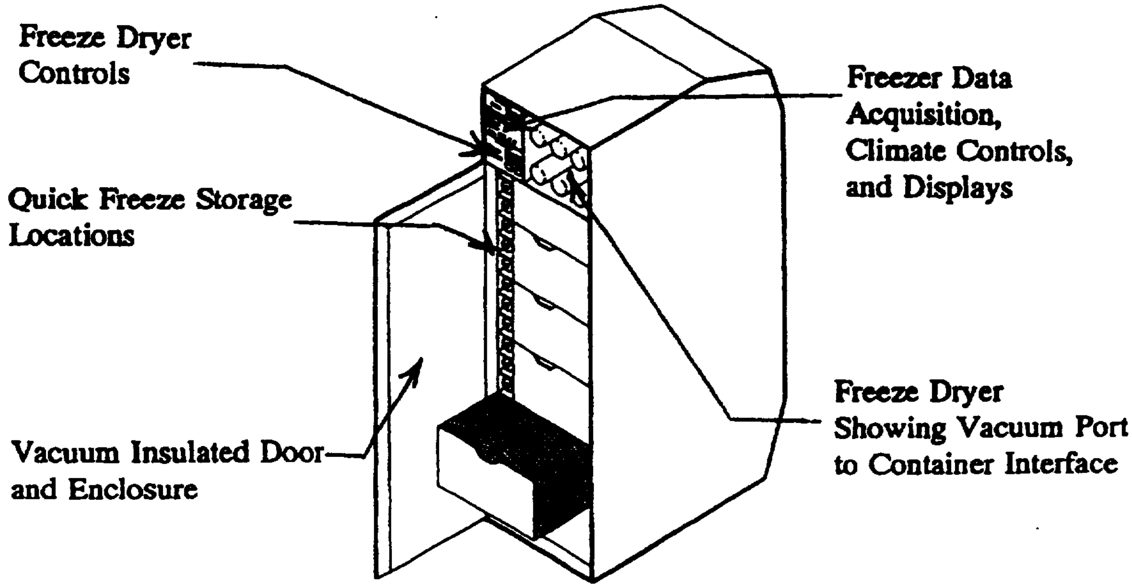


Figure 32: -70°C Storage Freezer with Integrated Freeze Dryer Concept

5.6 The −183°C Cryogenic Storage Freezer

Figure 33 is the conceptual design of a cryogenic storage freezer. The concept assumes that samples are introduced into the storage volume already at cryogenic temperatures, presumably frozen in the quick/snap freezer, so that the active heat load is minimal. The modeling results (Figure 34) show plenty of design margin for the system, so much so that the QFD (Figure 35) points to a lower efficiency pulse tube-based design because of its ancillary advantages in vibration isolation and reliability. The enhanced efficiency Stirling was not included in the QFD study since the enhancements would have a similar benefit to both pulse tube and Stirling coolers.

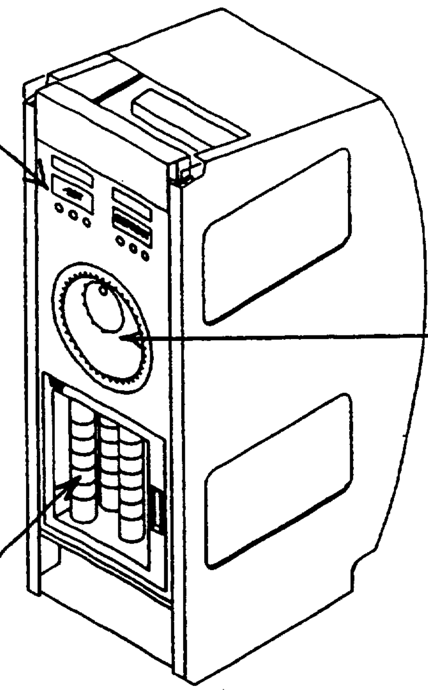
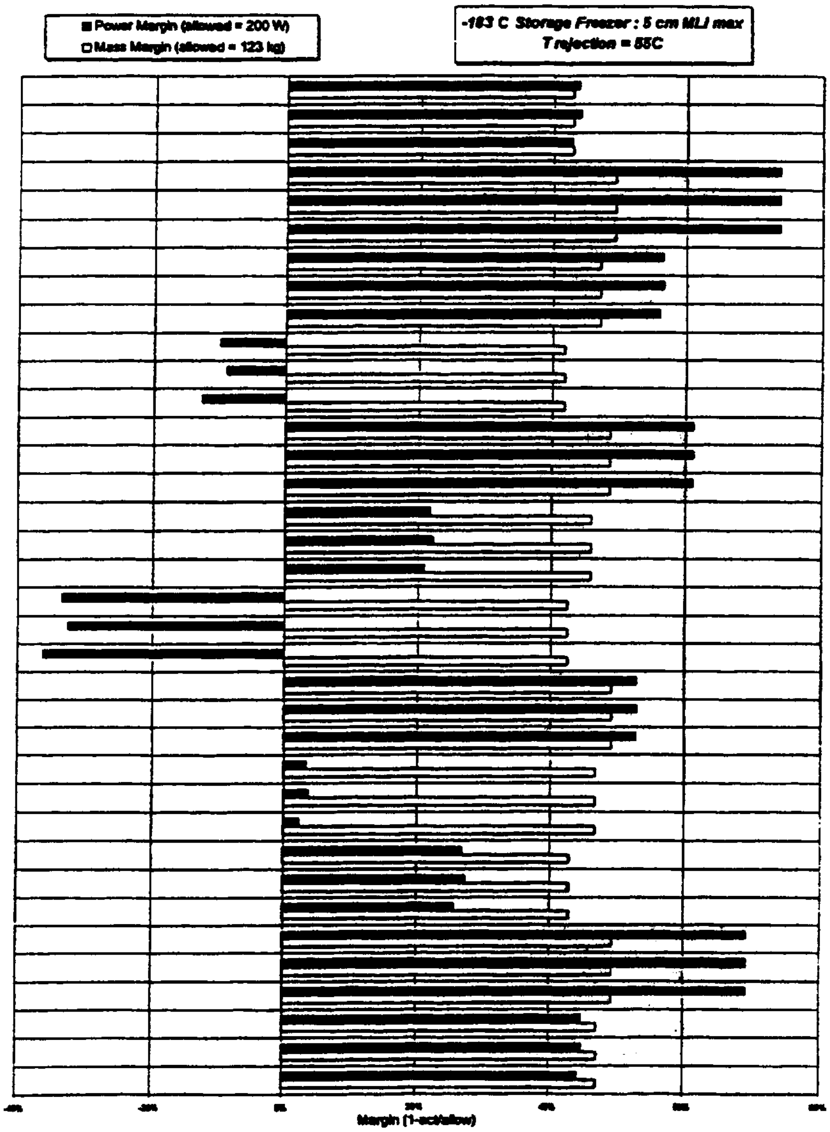


Figure 33: -183°C Cryogenic Storage Freezer

The system design presumes a MLI dewar using aluminized mylar at 35 layers per centimeter. In the conceptual design, the insulation thicknesses were kept below 5 cm to minimize detrimental compacting of the layers under launch vibrations. Such compaction would degrade its insulative properties.

Modelling did reveal that certain heat leaks which are relatively unimportant in higher temperature freezers, for example around wire penetrations, become important in this system. Lay up of MLI around the penetrations must be carefully designed to minimize radiation tunneling. Although not essential for system performance, development of polymer conduits for wall penetrations would greatly improve packaging and cost, replacing the welded metal bellows penetrations that are conventionally used. Access into the freezer is also a major source of heat transfer. The opening cover, nominally an evacuated stainless steel or foam plug, must be deep in order to minimize conduction. The deep plug makes packaging and access more difficult. An evacuated polymer plug, similar in construction to the R-105 panels considered for the enclosure, would reduce this cumbersome length.



	Cooler Cycle	Insulation	Heat Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R2300	TPG
32	Enhanced Stirling	R2300	Heat Pipe
31	Enhanced Stirling	R2300	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R2300	TPG
23	Brayton	R2300	Heat Pipe
22	Brayton	R2300	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R2300	TPG
14	Pulse Tube	R2300	Heat Pipe
13	Pulse Tube	R2300	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R2300	TPG
5	Stirling	R2300	Heat Pipe
4	Stirling	R2300	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 34: -183°C Cryogenic Storage Freezer Trade Study Results

Excellent Benefits:	●	9
Good Benefits:	○	3
Limited Benefits:	△	1

	Ref. No. 24 (Brayton / TPG / R2300)	Ref. No. 8 (Stirling / TPG / R2300)	Ref. No. 15 (Pulse / TPG / R2300)
Technical maturity	●	●	○
Longevity/Reliability/Maintenance/Robustness	○	○	●
Commonality	●	●	●
Configurable temperature and volume to handle up/down payloads	○	●	●
Maintenance on ground	●	●	●
Low mass	△	○	△
Low volume	△	○	△
Low power demand for adequate capacity	△	○	△
Low vibration generation (also impacts noise)	○	○	●
Low noise generation (impacted primarily by heat rejection)	△	○	△
Acceptable resonant frequencies	●	○	○
Electromagnetic compatibility	●	●	●
Environmental compatibility	○	○	○
Safety	○	○	○
Cooler temperature/capacity operating range	●	●	●
Obvious layout and operation (foolproof)	○	●	●
Sample temperature regulation	●	●	●
Intermediate cold heat exchanger requirement	●	○	○
Low power surge at start-up	●	●	●
Launch/landing survivability	△	○	○
Cold containment susceptibility	△	△	○
Configurable temperature and volume to meet on-orbit storage requirements	○	○	○
Technical Importance Rating	108	118	118
Normalized Technical Importance	31	34	34

Figure 35: -183°C Cryogenic Storage Freezer QFD

Moisture control is a problem not only for water vapor, but also for oxygen which may condense from cabin air at temperatures slightly below the operating temperature. The system design would have to incorporate a nitrogen purge to keep out ambient air and preclude LOX buildup. The Space Station will have dry nitrogen available as a utility; Shuttle or Mir usage would require a gas supply be brought along. Since no door openings are planned during transport, no nitrogen is required during transport operations. Moisture control becomes a more serious problem when the system has to be subcooled for transportation. Analysis shows that the volume would have to subcooled 5.5°C, which is not a problem for the cooler or enclosure.

5.7 The -196°C Cryogenic Quick/Snap Freezer

Figure 36 is a conceptual design of a quick/snap freezer. It relies on intimate contact between the specimen and a large thermal mass of a highly conductive material, copper. The specimen vial also contains a smaller thermal mass. Small, cryogenic specimens are sealed into their vials by a colleted cap which shrinks when cooled but can easily be removed by unscrewing. A special tool is used to handle the vials (Figure 37). Vials can be stacked end to end (Figure 38) until the row is filled, and then the whole row is transferred as a unit in a carrier into the storage freezer. The storage chamber is configured to accommodate these stacked vials (Figure 39).

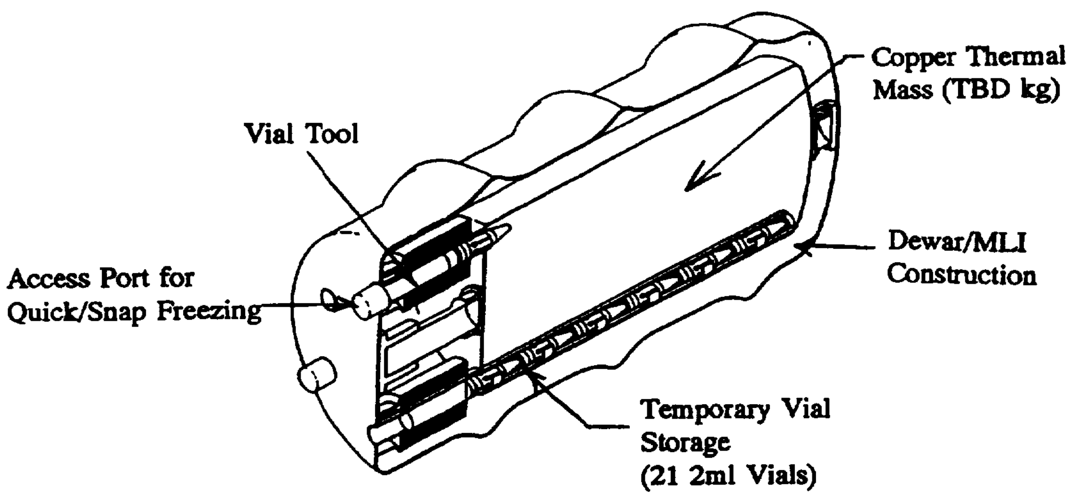


Figure 36: -196°C Cryogenic Quick/Snap Freezer Concept Design

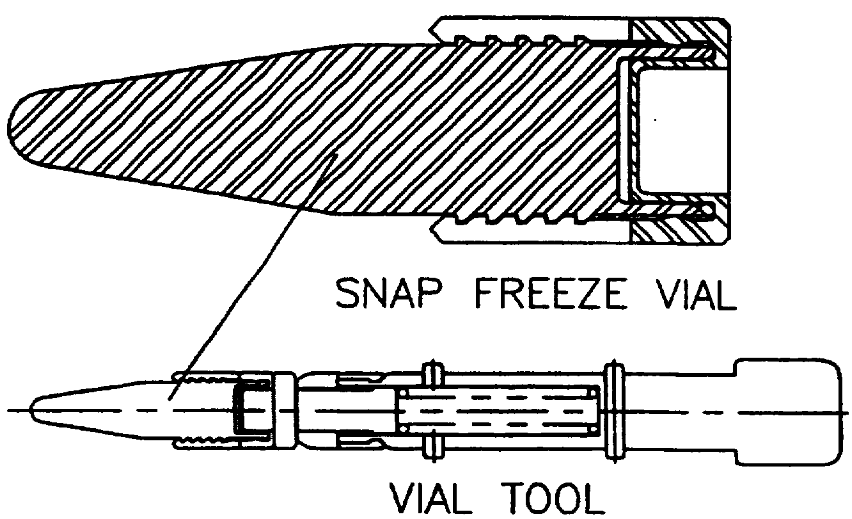


Figure 37: Vial Concept and Vial Tool

The requirements documents specified the rate at which samples must be processed was TBD (to be determined). The analysis assumed a worst case of ten specimens every four hours. Although the results of the modeling with air heat rejection at 40°C cabin air (rejection temperature =

55°C) show that only a few options exist which can accommodate this rate (Figure 40), a lower use rate would permit many more design options. Because of the potential for placement of the quick/snap freezer into a glovebox environment, water heat rejection was also analyzed. Figure 41 presents the systems analysis for the water heat rejection case which shows an increase in power margins of about 20% to 50% depending on cooler technologies. For the enhanced Stirling cycle, this represents an increase in specimen processing rates (i.e. quick freezing 2 ml specimens) of approximately 33%. The QFD points to the highest technology combination as being most appropriate for this application because it also allows more design and operational flexibility.

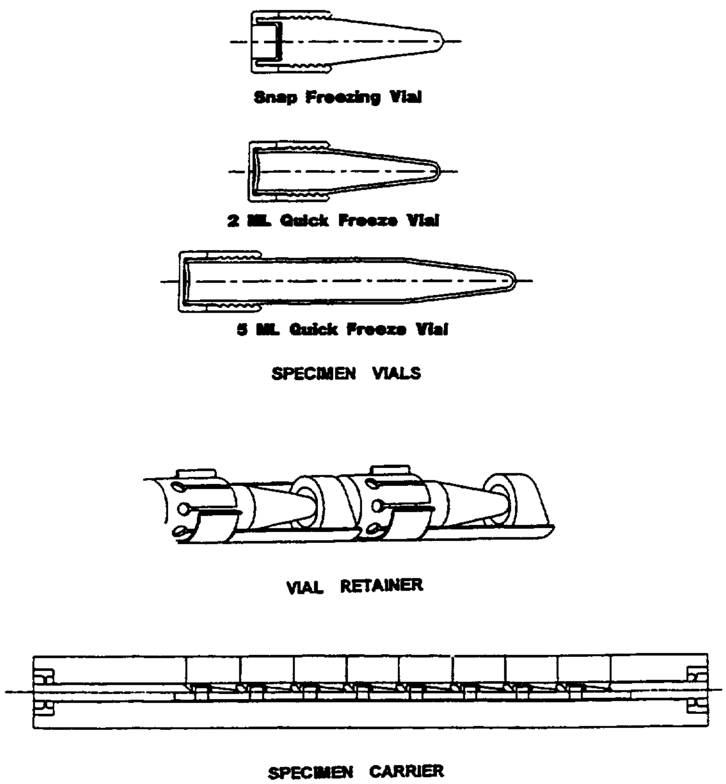


Figure 38: Specimen Vials, Vial Retainer, and Specimen Carrier

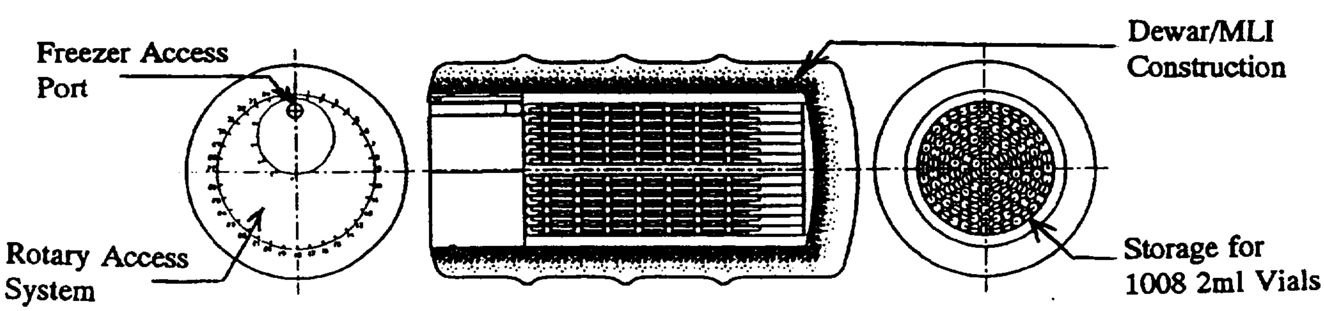
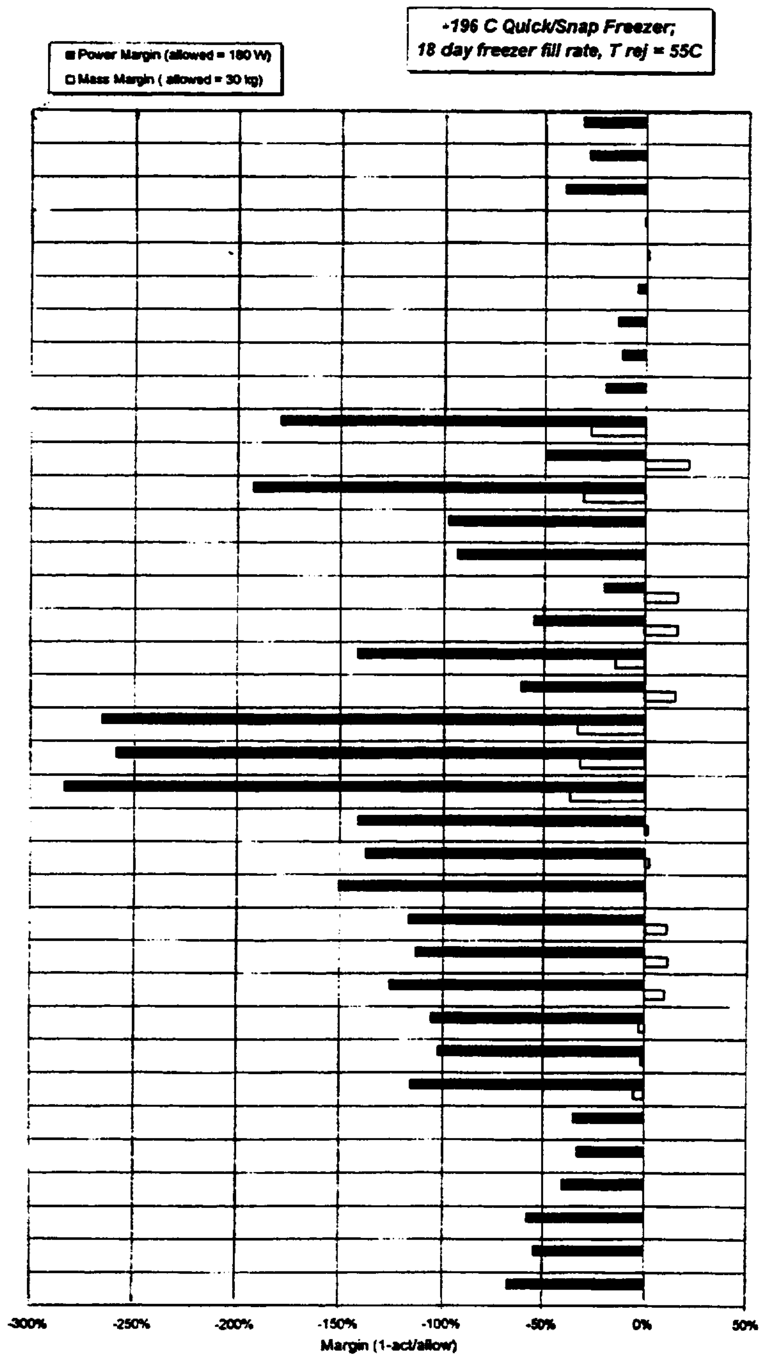


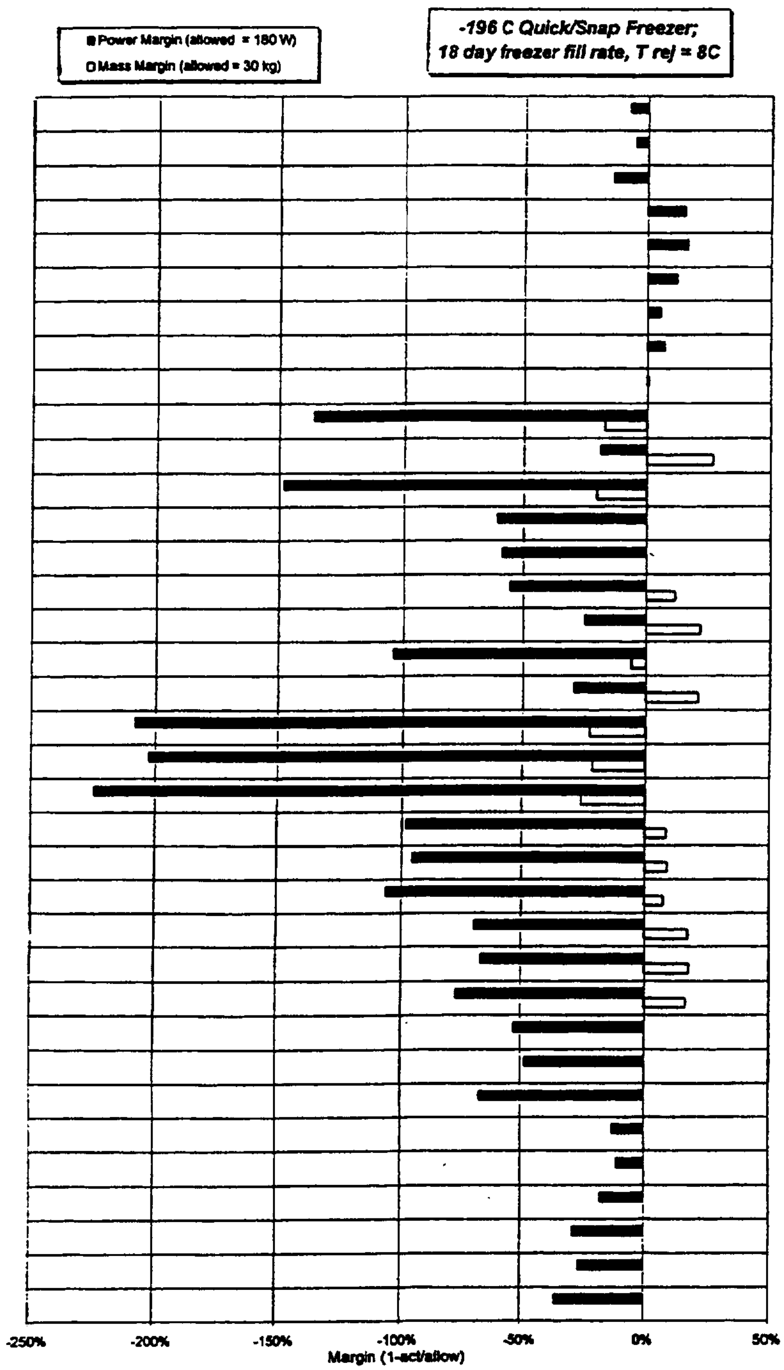
Figure 39: -183°C Cryogenic Storage Freezer Showing Vial Storage Packaging Concept



Reference No.	Cooler Cycle	Insulation	Heat Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R2300	TPG
32	Enhanced Stirling	R2300	Heat Pipe
31	Enhanced Stirling	R2300	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R2300	TPG
23	Brayton	R2300	Heat Pipe
22	Brayton	R2300	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R2300	TPG
14	Pulse Tube	R2300	Heat Pipe
13	Pulse Tube	R2300	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R2300	TPG
5	Stirling	R2300	Heat Pipe
4	Stirling	R2300	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 40: -196°C Cryogenic Quick/Snap Freezer Trade Study Results (Air Heat Rejection)

Since nitrogen liquifies at −196°C at one atmosphere of pressure, a substitute for nitrogen would have to be used as a purge gas, or perhaps the nitrogen gas could be mixed with helium or argon. Alternatively, the science community could be petitioned for a few degrees relief on the temperature specification. The −196°C temperature was selected because it is the temperature of the liquid nitrogen used for snap freezing on earth. Since snap freezing with liquid nitrogen would be difficult to implement in microgravity, a qualitatively different approach to quick freezing, such as using cold conductors, is required. Relief of the temperature specification by even a few degrees would greatly simplify moisture control.



Reference No.	Cooler Cycle	Insulation	Heat Transport
36	Enhanced Stirling	R60	TPG
35	Enhanced Stirling	R60	Heat Pipe
34	Enhanced Stirling	R60	Copper
33	Enhanced Stirling	R2300	TPG
32	Enhanced Stirling	R2300	Heat Pipe
31	Enhanced Stirling	R2300	Copper
30	Enhanced Stirling	R105	TPG
29	Enhanced Stirling	R105	Heat Pipe
28	Enhanced Stirling	R105	Copper
27	Brayton	R60	TPG
26	Brayton	R60	Heat Pipe
25	Brayton	R60	Copper
24	Brayton	R2300	TPG
23	Brayton	R2300	Heat Pipe
22	Brayton	R2300	Copper
21	Brayton	R105	TPG
20	Brayton	R105	Heat Pipe
19	Brayton	R105	Copper
18	Pulse Tube	R60	TPG
17	Pulse Tube	R60	Heat Pipe
16	Pulse Tube	R60	Copper
15	Pulse Tube	R2300	TPG
14	Pulse Tube	R2300	Heat Pipe
13	Pulse Tube	R2300	Copper
12	Pulse Tube	R105	TPG
11	Pulse Tube	R105	Heat Pipe
10	Pulse Tube	R105	Copper
9	Stirling	R60	TPG
8	Stirling	R60	Heat Pipe
7	Stirling	R60	Copper
6	Stirling	R2300	TPG
5	Stirling	R2300	Heat Pipe
4	Stirling	R2300	Copper
3	Stirling	R105	TPG
2	Stirling	R105	Heat Pipe
1	Stirling	R105	Copper

Figure 41: -196°C Cryogenic Quick/Snap Freezer Trade Study Results (Water Heat Rejection)

6.0 CONCLUSIONS AND RECOMMENDATIONS

The trade studies presented in the previous section included technologies which are not yet within the demonstrated state of the art. These are the enhanced efficiency Stirling cycle cooler, polymer panel enclosures, and thermal pyrolytic graphite heat transport. For each of the five freezer classifications, the systems analysis identified a potential combination of technologies (cooler, enclosure, and thermal transport) capable of meeting the specifications for mass, power, and internal freezer volume. The system analysis also showed that the requirement for low noise air heat rejection requires heat exchanger technology development. However, in the case of the -70°C storage freezer, it has a near zero power margin[2], and, in every case, the QFD analysis points to the use of more developmental technologies to mitigate problems with the other requirements. In addition, the -70°C transporter concept analysis indicated the need for cooler and enclosure technology development. Having these technologies available would provide more design and operational options. Use of advanced technology will lower cost, improve reliability, decrease power consumption, and decrease mass.

6.1 Freezer System Conclusions

Table 6 summarizes the recommended technology combinations for each of the five freezer classifications.

Freezer Classification	Cooler	Enclosure	Thermal Transport
-20°C Storage	Thermoelectric[3]	Polymer Panel	TPG
-70°C Storage	Enhanced Stirling	Polymer Panel	TPG
Freeze Dryer	Stirling	Metal Skin Panel	Copper
-183°C Cryo Storage	Stirling	MLI Dewar	Copper
-196°C Cryo Quick/Snap	Enhanced Stirling	MLI Dewar	Copper

TABLE 6 : Minimum Feasible Combinations of Technologies for the Five Advanced Freezer Classifications to Meet Specifications

The -20°C storage freezer can meet its operating requirements with some combination of technology based on each of the four types of coolers, though only the thermoelectric and Stirling cycle-based cooler designs can employ available enclosure and thermal transport technologies. Because of its many advantages in reliability, vibration, and environmental compatibility, thermoelectric coolers are preferred over the more efficient Stirling cycle coolers if a water heat rejection medium is available. If heat has to be rejected into very warm (+40°C or more) air, a Stirling cycle cooler would be

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[2] This does not address the power-off condition and assumes a duty cycle of 3.

[3] presumes water cooling.

required. Having a higher performance enclosure available would continue the advantages of the thermoelectric cooler into more stressing heat rejection scenarios.

While the systems analyses of the -70°C storage freezer show that this freezer can barely meet its specs without advanced technology, the margins are so small and so sensitive to the assumptions that a designer would be wary of imposing too much confidence in this result. Employing advanced technology will provide a greater comfort margin. Developmental technologies also enable innovative designs, such as a combination freezer/freeze dryer, or a transportable locker which is light enough to be handled by a single individual. They also allow consolidation of the storage freezer into a half rack, even in combination with the freeze dryer. The transportable locker is also made much more practical if phase change material is incorporated into its structure so that the system does not have to be subcooled and held at below spec temperature during power-off transport.

The presumption that samples are introduced into the freeze dryer already at -70°C means that its active heat load is minimal. Both the Stirling and Brayton coolers meet the specification with existing insulation technology. The pulse tube cooler needs advanced insulation technology to meet the specifications. The Stirling was recommended because it had larger mass and power margins, and because its selection would be in accord with the Statement of Work goal to select a minimum set of technologies.

Likewise, the small volume and high R-value of the MLI dewar make some configuration of all the coolers modelled feasible for the two cryogenic freezers. The advantages of the pulse tube coolers in simplicity, vibration isolation, and design flexibility might make it the cooler of choice for designers of these systems, but for the purposes of this technology assessment and development, the Stirling was chosen as the baseline for the same reasons as for the freeze dryer.

6.2 Technology Conclusions

Nine technology development areas were identified as having potentially important impacts on the performance of the various freezer classification, their design margins, and/or their operational flexibility. Some also have commercialization potential.

(1) Polymer panel: A polymer panel is made of plastic skins around a plastic support structure that can maintain a high ($<10^{-3}$ torr) interior vacuum for long periods. It has a calculated bulk R-value of 150 (hr-ft^2-°F/BTU) per inch. The panels would be used to construct a rectangular cabinet with a calculated R-105 insulation value once edge losses are accounted for. This represents a significant improvement over the currently available steel-skinned vacuum panels that have demonstrated R-30 cabinet values. The polymer panel density is expected to be approximately 20% lighter than the steel skin panels and have greater load bearing properties, enabling lower weight enclosure designs. Cabinets using the welded steel skin panels must be reinforced to prevent flexing of the welds. This weight might be reduced with the polymer panels.

The -70°C freezer classification benefits most from the polymer vacuum panel enclosure since the system analysis indicates the -70°C specifications were unlikely to be met without advanced technology. The -20°C storage freezer and -70°C freezer dryer would also be lighter and more power efficient with this technology. An MLI dewar augmented by similar plastics technology for wire penetrations and the entry opening could reduce the weight and power needs of the cryogenic designs.

The key development challenges in the polymer vacuum panel technology are: 1) selecting a material with the strength, low mass, and low thermal resistance needed, which is also capable of supporting and maintaining a high vacuum, and 2) sealing the edges against the required high vacuum. If these panels could be produced economically in quantities, they would have the potential for application in commercial and industrial insulation across a broad temperature range.

(2) Enhanced Efficiency Stirling Compressor Motor: To improve the efficiency of a Stirling cycle cooler, improvements in motor efficiency should be evaluated. The reference Stirling cycle cooler (STC Low Vibration Cooler) motor uses samarium cobalt magnets which could be replaced with higher strength neodymium iron boron magnets to improve motor performance. Other STC SBIR coolers have been implemented using this improved motor technology. Square wire could be used to replace the standard round winding material to increase the density of the coil and reduce the power losses due to the improved conductance of the winding. A careful trade-off between moving magnet, moving coil, and moving iron design options should be completed to identify the most efficient one. The reference compressors are the high efficiency commercial Stirling Technology Company (STC) and Sunpower Inc. systems. The calculated efficiency improvements represent a 15 to 20% reduction in the power required to accomplish the required compressor work. This approach is an incremental improvement on proven technology to realize efficiency gains at relatively low risk and cost.

(3) Thermal Pyrolytic Graphite (TPG) Cold Finger: The cold head of the Stirling cooler is a concentrated cold spot to which the entire cabinet heat load must be interfaced. Conductive heat distribution produces a temperature drop across the heat exchanger which must be minimized to improve the system efficiency. TPG is a solid conductor with conductivity and density properties superior to metallic conductors such as copper or aluminum. A TPG cold finger could provide a lower temperature drop from the stored specimen to the cooler acceptor surface. Since air convection heat transport requires a larger surface, the lower density of the TPG should permit a greater range of design options with TPG than with metallic conductors.

The TPG composite is formed by hot isostatically pressing sheets of TPG material inside a form fitting aluminum or copper skin which becomes a permanent part of its structure. Because of its heterogeneous composition and its inherent anisotrophy, TPG designs must be carefully considered to take full advantage of its properties.

(4) Insulating Pressure Vessel with Integrated Thermal Transport: Stirling cycle coolers have a parasitic conduction heat leak through the expander pressure vessel which separates the coldest (acceptor) and warmest (rejector) temperature surfaces. The Stirling coolers normally use a stainless steel enclosure to form the helium tight pressure vessel. A plastic pressure vessel, with a lower thermal conductivity, would reduce this unwanted heat transfer by thermally isolating the acceptor and rejector ends of the device. An additional benefit of this configuration is the elimination of the back heat leak when the system is unpowered.

There are several challenges in this technology, especially the metal-to-plastic seal, the helium containment quality and the possible contamination of the cold heat exchange surfaces or the regenerator by plastic outgassing. The commercial potential of this technology is related to the efficiency benefits this technology offers to Stirling and pulse tube cooler systems.

It is also conceivable that the pressure vessel could have TPG fins integrally bonded to the plastic. These fins could be designed with tubes inside them which allow cold helium to cycle in and out of them, thus improving the heat transfer efficiency between the gas and the heat acceptor. There are several design challenges in this approach, particularly the helium-tight interfaces between the different materials and sizing the ducting inside the TPG fins to insure that appropriate volumes of helium are transported during each cycle to maintain the cooler COP.

(5) Vacuum Dewar Compatible Polymer Interfaces: Dewar enclosures must include penetrations for utility runs and sensor wires. Existing implementations use welded metal bellows that join the inner and outer vessel walls, forming a low conduction path through the pressure vessel. Replacing the bellows with plastic components bonded to the pressure vessel housing could reduce the component cost of the dewar assembly and lower the conduction heat loss through the penetration. The key technical challenge is the vacuum tight, long life, metal-to-plastic bond which can endure the temperature excursions expected in the dewar operations. Commercial potential for improved performance and lower cost dewar assemblies would be in the laboratory equipment and cryogenic materials processing industry.

(6) Brush Carbon Quick Disconnect (QD): The ability to quickly disconnect the cooler, heat exchangers, and enclosure would support the removal of a lightweight enclosure from the rack assembly either on the ground after landing or for transport by an MPLM. It would also facilitate on-orbit maintenance of cooler and heat exchanger assemblies. Without this quick disconnect ability, handling equipment and more personnel effort will be required to transport the freezer contents.

A high conduction breakable contact is based on a proprietary brush carbon material. Brush carbon is a velvet mat of carbon fibers which has low thermal resistance only when mated. It also accommodates low contact pressures and high mechanical compliance to allow for the thermal expansion of dissimilar materials and potential vibration isolation of the cooler surfaces. Brush carbon has been demonstrated on the ground, but concerns over carbon fiber contamination need to be addressed for use in space. Also of concern is the control of moisture build-up on the cold plate surface if an unsealed brush carbon assembly was exposed to the atmosphere.

A brush carbon contact could also be incorporated into a thermal switch which could permit sharing of the cooler acceptor between the -70°C freezer and the freeze dryer by selecting a contact conduction position.

The key development challenge, beyond verifying the properties of brush carbon, is the resolution of the life and contamination safety issues related to the release of broken carbon fibers. The commercial product applications of this technology could include more maintainable heat exchanger and low vibration heat exchanger applications, and long life thermal switches.

(7) Low Noise Heat Rejector: The acoustic emissions of the system must be controlled to very low levels (NC40 with a goal of NC30). With the air media, the heat exchanger must produce the

minimum delta temperature across the fins to minimize the temperature of the heat rejection surface. Previous space freezers, based on military standard fans and metal finned heat exchangers, have not met the acoustic requirements. Improving heat exchanger efficiency with TPG should enable lower air velocity heat transfer, thus requiring a lower fan speed. This would minimize the acoustic noise emitted directly from the fan and air flow turbulence.

Low noise heat exchangers have many potential applications such as computer work stations and office equipment.

(8) Phase Change Panels: The -20°C, -70°C and -183°C storage freezers are required to maintain samples at or below the specified temperature during power-off conditions. For the -20°C and -70°C systems, incorporating a phase change material (PCM) inside the freezer would eliminate the need for significant sub-cooling before power off while permitting the freezer to maintain the specified temperature for an extended or unplanned power-off condition. PCM would also average out the temperature variations caused by sample freezing and door opening heat loads, allowing the freezer and heat exchanger systems to be sized more closely to average instead of peak loads.

PCM could be incorporated into structural components with minimal impact on net freezer weight. A potential PCM for the -70°C freezer is a hexane-octane blend tailored to have a phase change temperature several degrees lower than the operating temperature to allow for heat transfer through the containment. The key technical challenge is in the containment of the PCM, since the double or triple containment needed for safety will result in poor heat transfer and heavier assemblies. The -20°C freezer temperature phase change technology is under development for commercial and industrial cooling load management systems. A commercial use for the lower temperature phase change materials technology has not been established at this time.

(9) Moisture Management: Moisture management is especially needed for the -20°C, -70°C, and -183°C storage systems, which will be operating continuously during extended missions. Ground-based systems which rely on gravity to transfer moisture during periodic defrost cycles are not applicable to space. The system level approach to moisture management would include: reducing the moisture load introduced into the freezer; capturing any moisture which gets inside; and eliminating the moisture from the enclosure with a minimum of crew workload.

The challenge is to provide a reliable and robust moisture management system with minimum mass, power, and crew maintenance required. Desiccants and cold traps must be further evaluated to determine the expected performance at reduced temperatures, and their employment configured to minimize the required crew attention.

6.3 Technology Development Recommendations

Having enumerated the various technologies that could improve the design and operational flexibility of the five freezer classifications, we must now prioritize these technologies for development. Figure 42 shows the thermodynamic benefit of the nine technologies listed above plotted against how many of six freezers (the five specified in the statement of work and the transportable locker design) stand to benefit from the technology. In the cases of the brush carbon quick disconnect (6), low noise heat rejector (7), and the moisture control system (9), the thermodynamic benefit is minimal, but operational or maintenance benefits may be significant. (Numbers in the boxes refer to technologies listed in Section 6.2 above.)

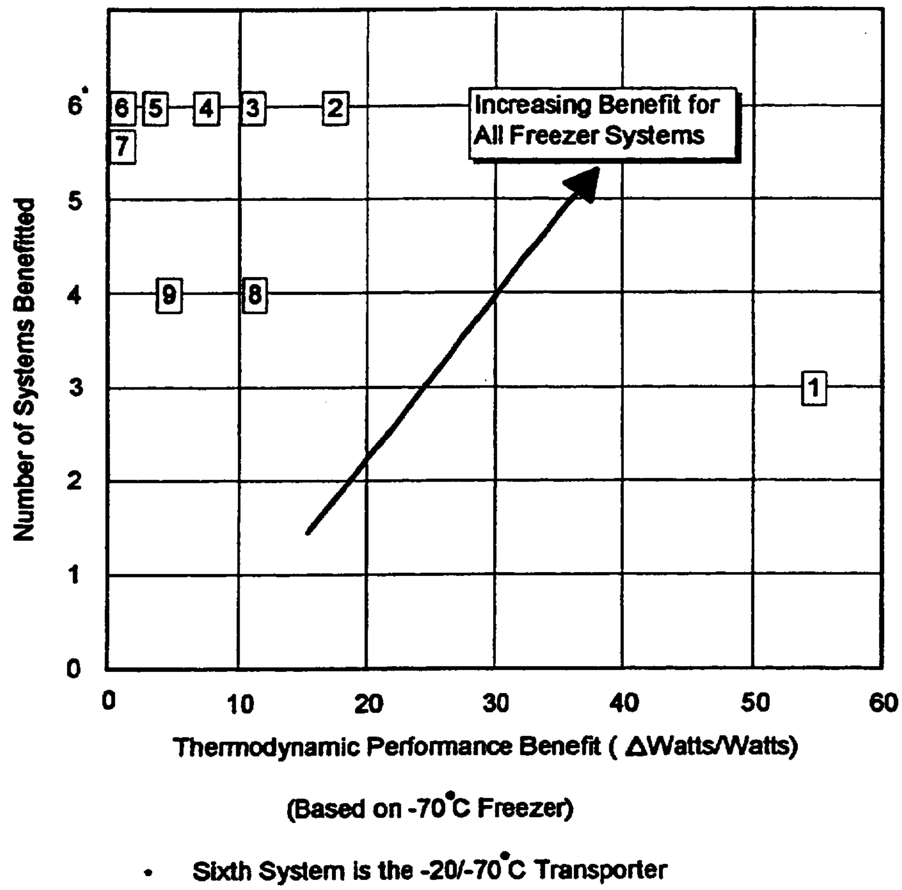


Figure 42: Benefit of the Nine Technology Areas (see Sec. 6.2,6.3) to the Freezer System

If polymer panels with an enclosure R-value of 105 can be successfully developed, they would have a significant impact on space freezer designs. The -20°C systems would comfortably be able to accommodate higher heat rejection medium temperatures, such as warm air, and both the -20°C and the -70°C systems would be configurable to transportable storage lockers that can meet the single man handling weight limit. If subcooling is required, the -70°C storage locker would not require an advanced cooler. Such panels would have applicability outside the statement of work requirements, for example in domestic refrigerators and even in commercial systems. The conclusion of this analysis is that plastic panel technology has the highest leverage and should be given highest priority for development. Technology demonstrations should verify an R-value of 105 for an enclosure made of this technology.

Enhanced efficiency Stirling coolers also have leverage across several freezer classifications. Although thermoelectric and pulse tube coolers have vibration and reliability advantages in their system niches, the high efficiency of current Stirling cycle coolers leads to their selection for most freezer classifications. An enhanced efficiency Stirling would allow still more design margin which could be returned to the spacecraft integrator as unused power or mass to be distributed to other challenged systems. Enhanced efficiency Stirling coolers could impact all five of the freezer classifications and help enable the freezer/freeze dryer combination. It, too, is a high leverage technology.

However, some elements which would improve the Stirling cooler efficiency need not be demonstrated in a technology development program. For example, it is well known that using a stronger permanent magnet in the motor will improve efficiency; little would be gained by proving it in hardware. Technology development should concentrate on moving along high leverage approaches that have some resolvable risk. Finding an insulating material for the pressure vessel housing and integrating it with the thermal transport are challenges which can only be resolved with hardware demonstration.

Calculation suggests that cooler efficiency for the enhanced efficiency Stirling cooler could be improved by a total of 39% compared to current Stirling cooler technology if all the improvements discussed were included. Twenty-five percent of that would be attributable to engineering re-optimization, such as using stronger magnets, rather than technology development. The use of an insulating pressure vessel makes up the remaining fourteen percent. The goal for the pressure vessel is thus established to be 14% above the stainless steel baseline.

Some issues will require serious engineering design to overcome but pose no technology challenges. For example, tolerance of launch vibrations may be an issue for the flexure mounted piston in one Stirling cooler design. Several solutions to this problem are conceivable, for example, a locking mechanism that secures the system during launch. Many elements of vibration and noise attenuation can be mitigated with hard engineering. To correctly focus a technology development effort, the emphasis should be on "inventions" rather than "sharp pencil engineering."

Replacing metal conductors or heat pipes with TPG would improve the performance of virtually any system, making this also a technology with good leverage. However, the system level improvement would be relatively modest compared to R-105 enclosures or cooler efficiency enhancement. A good risk management strategy would invest a moderate amount of resources in TPG development for whatever improvement it can provide. Metallic conductors typically show a 20°C temperature drop between the cooler cold head and the enclosure air. Calculations suggest TPG could reduce this drop to 10°C, which would allow a 10% improvement in cooler system COP. The 10°C temperature drop is established as the goal for technology development.

Acoustic emissions technology is pertinent to any freezer classifications where the cooler must reject heat to the cabin air. The recommended technology development activity is to demonstrate that the NC-40 (with a goal of NC-30) acoustic emissions can be satisfied in a dimensional mockup of the air heat rejector. The heat exchanger geometry and projected thermal performance is to be based on the use of advanced Thermal Pyrolitic Graphite (TPG) materials to enable the lowest fan power and flow velocities.

The validation of the TPG materials under the acceptor heat exchanger will be used to predict the heat rejector heat exchanger requirements. The recommended development plan does not include the manufacture and test of a functional air heat rejector, since the recommended STC brassboard cooler hardware is a liquid cooled machine that has been produced under SBIR funds.

The air heat rejector mockup test will incorporate tube axial fans, the low noise heat exchanger geometry and representative duct work as planned for the brassboard freezer configuration to demonstrate that sufficient air mass flow rates can be accomplished in order to get the required heat transfer within the acoustic design limits.

An advanced moisture management approach would also have leverage over several systems. Although this conclusion doesn't emerge directly from the thermodynamic performance, operational and maintenance needs highlight it as an area in which a solution is required. The Statement of Work specifies that maintenance activities be limited to 2.6 manhours per year. Apportioning about half of this maintenance time to moisture control, the goal for this technology would be to accomplish any moisture removal in less than ten man-minutes per month, while remaining within the weight, power, and volume allowables for the system.

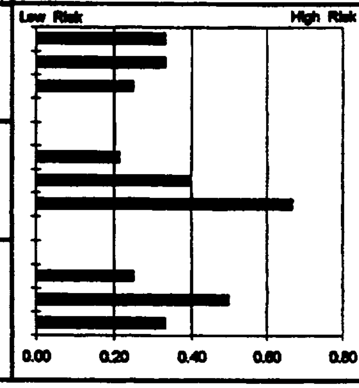
Niche improvements, like phase change material and brush carbon, though low leverage, are attractive enough to warrant further investigation at a modest level, especially if it can be done in the context of other systems demonstrations.

A vacuum dewar compatible polymer interface could improve the thermal performance and producibility of vacuum dewar systems which normally use welded metal bellows to form penetrations with increased thermal resistance. This technology would be used on the cryogenic storage and cryogenic quick/snap freezers.

Figure 43 is an estimate of the development risk of various technologies described above. In this analysis, too, polymer panels and TPG emerge as good candidates for development. The moisture management is flagged as a particularly risky area for follow-on development because of the lack of an existing base of experience from which to tackle the problem.

Technology Development Candidates	(A) Technology Complexity	(B) Technology Readiness	(C) Team's Tech. Experience	Weighted Score (A/(B*C))
Cooler Developments				
Enhanced Efficiency Stirling Compressor Motor	4	4	3	0.33
Insulating Press. Vessel w/ Integrated Thermal Trans.	3	3	3	0.33
Low Noise Heat Rejector	2	4	2	0.25
Enclosure Developments				
Polymer Panel (Rigidized MLI)	3	4	3.5	0.21
Vacuum Compatible Polymer Interfaces	3	3	2.5	0.40
Moisture Management	4	3	2	0.67
Thermal Transport Developments				
TPG Cold Finger	2	4	2	0.25
Brush Carbon Quick Disconnect	4	4	2	0.50
Phase Change Panels	3	3	3	0.33

Weighted Score



Technology Complexity: 5 most complex
Technology Readiness: 5 most ready
Team's Experience: 5 most experience

Figure 43: Relative Risk for Various Technology Developments

6.4 Recommended Brassboard

The proposed technology developments are to be demonstrated at the brassboard level. A brassboard is defined as an assembly of preliminary parts used to prove out a specific function. More sophisticated than breadboards, brassboards begin to approach the challenge of form and fit as well as function. Parts tend to be assembled into semi-permanent states and a point design begins to appear.

Because of the large heat lift required for the storage freezers operating in the range of -20 to -70°C, a midrange storage freezer will make an appropriate brassboard system. Depending on the success of the component technology developments, the optimal brassboard would include an enclosure made of polymer panels, a Stirling cycle cooler with an insulative pressure vessel, and TPG thermal transport, either integral with the pressure vessel or configured in such a way that will permit quick disconnect of the cooler and enclosure to demonstrate the feasibility of a transportable locker. Some demonstration of a moisture control strategy would also be appropriate.

The systems analysis quantitatively identified the need for advanced technology to enable the -70°C transporter concept. The analysis and concept freezers also revealed the potential for providing the required -20/-70°C internal storage volume with less weight, less external volume, less power, and less heat rejection than allowed by the specifications. Because of the identified technology needs and the prospect for significant system performance benefits in the mid to high temperature freezers, the -20/-70°C classification is recommended as the brassboard system.

This recommended brassboard would demonstrate the polymer vacuum panel enclosure, the Stirling cycle cooler with an insulative pressure vessel, TPG cold finger and thermal transport, and moisture management technologies.

The polymer panel enclosure would demonstrate enclosure R-values anticipated to be a factor of 3 to 5 times better than the current state-of-the-art metal vacuum panel systems (Aura(tm)). The Stirling cycle cooler would demonstrate the use of a polymer pressure vessel with low parasitic back heat leak. The TPG heat exchangers would demonstrate compact, lightweight and high thermal efficiency configurations that would be required for low noise air heat rejection coolers. This technology would also be applicable to the cold side, acceptor, heat exchanger and would minimize the size and frost sensitivity of the finned heat exchanger.

The moisture control technology would demonstrate the effectiveness of molecular sieve materials over the science freezer temperature range. The moisture control technology would also demonstrate concepts to enable samples to be deposited and retrieved to/from the freezer without the introduction of moisture. Moisture in the cold space can result in heat exchanger fouling which has been a problem on previous space freezers.

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13. ABSTRACT ***(Maximum 200 words)***

The NASA Lewis Research Center, through contract with Oceaneering Space Systems, is engaged in a project to develop advanced refrigerator/freezer (R/F) technologies for future Life and Biomedical Sciences space flight missions. The first phase of this project, a technology assessment, has been completed to identify the advanced R/F technologies needed and best suited to meet the requirements for the five R/F classifications specified by Life and Biomedical Science researchers. Additional objectives of the technology assessment were to rank those technologies based on benefit and risk, and to recommend technology development activities that can be accomplished within this project. This report presents the basis, the methodology, and results of the R/F technology assessment, along with technology development recommendations.

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