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COLD STORAGE TEMPERATURE STABILIZATION

by
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and
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14. ABSTRACT
This report was developed under a Task Order Support Services (TOSS) contract. The study examined both a need for food refrigeration equipment in military field environments, and potential existing and developmental technologies to meet this need. The number one barrier to provision of refrigeration is limited electricity supply, both in quantity and continuity. Therefore concepts generated under this study maintained that as a focus. The issue was examined from all angles, including power source, insulation, and cold-holding plates, also known as eutectics. In addition, passive, self-controlled methods for the safe thawing of meat were evaluated. As a model in anticipation of future requirements for military refrigeration, the study used the Containerized Kitchen (CK). As such, concepts were developed for a storage capacity of 80 cu-ft, availability of 120/208 and 50/60 Hz. power for only 10 hours each day, and average refrigeration temperatures of 37 deg F at ambient temperatures of 120 deg F. It was desired that suitable candidates would be of minimal size, weight, and fuel consumption. Solutions chosen for the near term would be based on vapor compression technology. Technological advances would ultimately provide better batteries and low-output burners, and so, better solutions. It was determined passive thawing is not viable, therefore powered systems

15. SUBJECT TERMS

REFRIGERATION	CONTAINERIZED KITCHEN	ALTERNATIVE POWER	EUTECTICS
FIELD ENVIRONMENT	COLD STORAGE	FIELD FOOD SERVICE	ALTERNATIVE REFRIGERATION
THERMAL STORAGE	RATIONS	FIELD FEEDING	PASSIVE COLD STORAGE

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Preface

This is the final report for Delivery Order 0013, the Cold Storage Temperature Stabilization project, under the Soldier Systems Center (SSC) Task Order Support System (TOSS) Contract DAAK60-97-D-9305. The TOSS Program serves to improve the effectiveness and well being of U.S. soldiers in many areas. This study was conducted from June 16, 1999 through August 31, 1999.

The Cold Storage Temperature Stabilization project provides specific information regarding refrigeration in military field environments. While refrigeration is needed for food storage and ice making in military field conditions, a continuous supply of electric power to operate conventional, electrically powered refrigeration equipment cannot be provided at all times. To meet these refrigeration needs, SSC directed the Arthur D. Little (ADL) team to examine thermal storage and refrigeration technologies using alternative power sources with respect to their ability to better meet military field refrigeration requirements in the future. In addition, SSC directed the ADL team to investigate design options for thawing frozen meat in a passive, self-controlling system.

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Executive Summary

The objectives of the Cold Storage Temperature Stabilization project were as follows:

a) Investigate and analyze options for Containerized Kitchen (CK) cold storage in which electric power is available only during the day for about 10 hours
b) Develop design concepts for passive thawing of frozen foods

In order to meet these objectives, the ADL team investigated a number of possible cold storage system solutions. We developed an appropriate set of Cold Storage requirements for the Containerized Kitchen, and developed technical descriptions for the more viable system solutions that would meet the requirements. In addition, we performed preliminary calculations to explore possible technical solutions for the passive storage function.

The requirements for the Containerized Kitchen were developed through discussions with SSC staff. The storage capacity requirement is 80 ft^3. Analysis was based on the assumption that a single storage unit with 80 ft^3 capacity would be used. Subsequent discussion with SSC staff indicated that two 40 ft^3 units may be preferable. The ranking of system options would not change significantly if two 40 ft^3 units had been assumed.

The periodic availability of electric power (120/208VAC 50/60 Hz limited to 10 hours per day) and/or heat from the planned Thermal Fluid system is the driving requirement for this work, since conventional commercial reach-in refrigerators could otherwise be used. Refrigerator temperature is assumed to range between 34°F and 40°F, with a 37°F average. Ambient temperatures will be as high as 120°F. To maintain rapid readiness, pulldown time should be minimized. The size and weight should be minimized, using an 80 ft^3 commercial unit for comparison.

The logistics of use must comply with general CK requirements, have reasonable daily fuel usage, and minimize peak and/or daily CD generator usage (500 Watts maximum peak).

Based on our analysis and research, our conclusions and recommendations are as follows:

- The best near-term options should be based on vapor compression technology, but should use a modified version of a commercial reach-in that would use significantly less electricity. A high-efficiency reach-in using vapor compression refrigeration technology combined either with batteries or thermal storage using phase change materials for the 14-hour night period could be fielded within a short timeframe (1-3 years). Further, if a viable small dedicated generator with 500W output were developed in the near future, systems based on vapor compression would be a natural choice.
- In the absence of the development of a viable small generator, the best long-term option is a system based on one of the heat-input refrigeration technologies (absorption, chemisorption, or zeolite-based sorption) and utilizing a low-input

logistics fuel burner. Such a system eliminates reliance on the CK generator, has size and weight similar to a vapor compression/battery system, and offers either reliability or performance advantages over vapor compression systems. The major drawback of these system options is the considerable development effort required.

- An attractive medium-weight option which would not require either small generator or low-input burner technology is a system which combines thermal storage cooling packages with a specially-designed refreezer using chemisorption refrigeration. The combination of chemisorption and thermal storage would reduce dependency on the main CK generator, eliminate the need for any form of night power, and avoid dependence on development of unproven enabling technologies. This configuration is a variant of Thermal Storage Option C of Section 7.
- A thawing system that works completely passively over a wide external temperature range is not viable. Thawing systems built into part of a cold storage unit could be developed. The most logical combined cold storage/thaw concept is a system using vapor compression refrigeration for storage. During thaw mode, cooling would not be necessary, since the frozen food provides cooling. Thawing would be enhanced by forced convection provided with a large fan and a damper system that would allow some outdoor air to enter the unit to provide thaw heat.
- A parallel path development approach for Cold Storage System options is recommended, focusing on both vapor compression and heat-input refrigeration technologies.
- Low-input burner technology development should be pursued so that this important enabling technology would be available for heat-input systems requiring it. In the meantime, development of the heat-input technologies should use propane or electric heat, so as not to overburden these developments with excess challenges.
- The following future technologies may improve one or more of the technical options. Developments in these areas should be monitored, but not necessarily pursued by SSC:
  - Acoustic Compressors for vapor compression refrigeration
  - Vacuum-based insulation technologies
  - Small-size power generation technologies, including the AMTEC Sodium Cell and Fuel Cells
  - Advanced Battery Technologies

The structure of the report is as follows:

The main section of the report focuses on the cold storage stabilization system, starting with a review of requirements, a discussion of the approach to our analysis, descriptions of the system options we investigated, and a summary of results.

The additional report sections include a description of our analysis of the passive thaw system and a conclusion and recommendation section.

COLD STORAGE TEMPERATURE STABILIZATION

1. Containerized Kitchen Requirements

Refrigeration requirements and performance guidelines for the Containerized Kitchen (CK) are summarized in Table 1. The guidelines provide a basis for evaluation of different system options. Systems that are not suitable for operation up to 120°F are not included in the analysis summary. Furthermore, system capacity assumptions have been made to ensure comparable pulldown performance (except for some of the thermal storage systems, for which slow pulldown is an inherent drawback).

Table 1. Refrigeration Requirements and Performance Guidelines for the Containerized Kitchen

Category	Requirements
Total Internal Volume	80 ft^3
Storage Temperature	37°F
Electric Power and/or Hot Oil "Thermal Fluid" System Availability	10 hours on/14 hours off
Power (CK generator)	120/208V 50/60 Hz
Category	**Guidelines***
Ambient Conditions	Up to 120°F
Pulldown Capacity	Minimize pulldown time
Size	Minimize (use 80 ft^3 commercial reach-in for comparison)
Weight	Minimize (use 80 ft^3 commercial reach-in for comparison)
Logistics	• Comply with general CK requirements • Reasonable daily fuel usage • Minimize peak and/or daily CK generator electricity use (500W maximum peak)

* The values for these categories are targets or desired characteristics rather than requirements.

Additional points regarding the intended typical operation of the CK cold storage are as follows:

- The food will be delivered cold. Hence, there is no added load for pulldown of food temperature.
- The cold storage unit doors will be closed for the roughly 14-hour no-power period.

The cold storage requirements for the CK are one facet of a fresh food distribution system which will, when fully developed and deployed, allow daily delivery of up to three fresh cooked meals to soldiers in the field. Development work is also required for concepts which will help in management of cold food distribution to the CK and other field kitchens. One possible distribution concept makes use of cold storage containers which can store cold food during transport to the field kitchens, be used for storage at the kitchen sites, and later be returned for reuse. The containers could have thermal storage media incorporated in them to provide cooling during times when active refrigeration is not available, such as during transport or during night periods at a field kitchen site. They could use either pre-frozen thermal storage packages (e.g. ice packages) for cooling, or use glycol cooling with quick-disconnect hose connection to a glycol cooling system. Adoption of such a concept, which combines CK cold storage and the distribution network's cold storage, clearly would impact design decisions regarding CK cold storage system options. Some of the options analyzed in this report include glycol cooling in order to potentially take advantage of this concept.

2. System Options

The more viable cold storage options are illustrated in Figure 1. Electric-input refrigeration technologies other than vapor compression are not considered, because none of these alternative options come close to equaling vapor compression in size, weight, performance, and technical maturity.

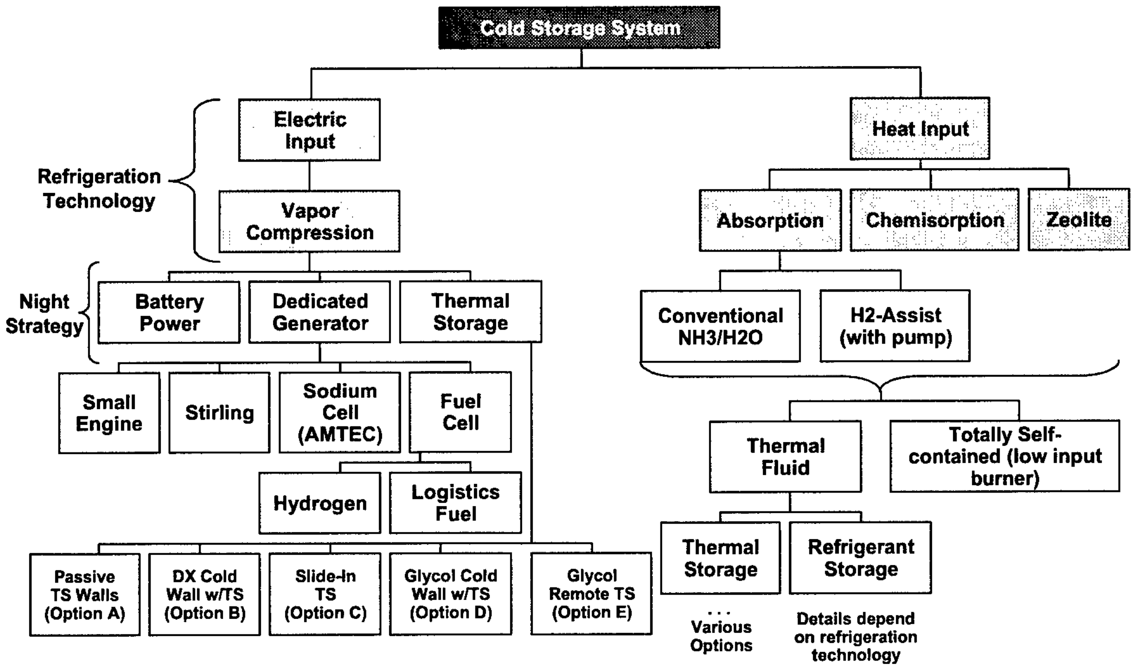

Figure 1. Cold Storage System Options

2.1 Vapor Compression

The night strategies for vapor compression systems are (1) electricity storage (batteries), (2) dedicated generators, and (3) thermal storage. The Army and other organizations are pursuing the first two options for other applications, and a range of technologies with varying technical maturity could be considered. Of these two, batteries are lower risk for the near-term, but dedicated generator systems have the potential for reduced weight. Thermal storage options have low technical risk, but a specialized thermal storage system would have to be developed in order to achieve CK cold storage system guidelines. Five system options utilizing cold storage are discussed in this report.

2.2 Heat Input

Heat-input refrigeration systems significantly reduce, but generally don't completely eliminate, the need for electricity. The three most promising candidates are absorption, chemisorption, and zeolite adsorption. All of these are based on the use of a sorbent that pulls in the refrigerant through an evaporator when cooled with ambient air; recycling of the refrigerant is accomplished by heating the sorbent and subsequently condensing the desorbed refrigerant vapor. Heat-input systems could be totally self-contained, deriving heat from a low-input logistics fuel burner, or they could obtain heat only during daytime hours from the Thermal Fluid system that is planned for the CK. If the latter approach is used, some form of thermal storage or refrigerant storage is required to take care of the night cooling needs.

3. Refrigeration Technologies

A full listing of possible refrigeration technologies is shown in Table 2. Most of these technologies are not represented in the system option tree above, since it is not likely that they would be strong candidates. The potential for any of the electric-input technologies to improve upon vapor compression in the near term is dubious. Significant development effort would be required to bring many of these technologies to a level where a rigorous comparison with vapor compression could be made.

Table 2. Refrigeration Technology Options

Refrigeration Technology	Description	Technical Maturity	Comments	Electric or Heat Input	Viability Rank[1]
Conventional Vapor Compression	HFC refrigerants in reverse-Rankine cycle	Mature	• Standard cycle for most applications	Electric	N/A
Vapor Compression with Acoustic Compressor	Moving pressure waves controlled by acoustic input	Prototypes developed for different applications	• Licensed to large appliance manufacture • Major potential benefits: increased durability, elimination of oil	Electric	5
Absorption	NH_3/H_2O with or without H_2 assist	Mature but not developed for 80 ft^3	• Long history, currently used in niche applications (i.e., RV refrigerators)	Heat	7
Stirling	Classic Stirling Cycle	Mature	• Technical and cost hurdles have prevented commercialization	Electric	3
Air Cycle	Air refrigerant in reverse-Brayton Cycle	Mature	• Used in niche applications (i.e., aviation)	Electric	1
Thermoelectric	Classic Thermoelectric Cooling Effect	Mature	• Inefficient	Electric	1
Chemisorption	Complex compounds as sorbent in batch process	Mature	• Fairly recent development, targeting applications for commercialization	Heat	7
Zeolite	Zeolite as sorbent in batch process	Under development	• Fairly recent development	Heat	7
Thermo Acoustic	Acoustically driven heat pipe	Test small scale units 2-5W (COP 0.16); Larger scale testing soon	• Developers see no roadblocks to larger scale development and expect efficiency of vapor compression	Electric	3
Magnetic	Cooling of working fluid by alternating magnetic field	Have developed 600W system with 28K lift.; Market in ~10 years	• Success tied to room temperature superconductors due to required high magnetic fields	Electric	1
Malone	Brayton or Stirling cycle with refrigerant operating near critical point	Prototype developed using propylene @ 1450 psi	• Lab tests at 3500 Btu/hr • Developers claim COP comparable to vapor compression, but this requires extremely efficient components	Electric	1
Thermal effects of Semiconductors	Cooling effects of vibrating semiconductors have been observed	Researched at semi-conductor or chip level		Electric	1
Elastomeric Heat Pump	Cyclic stretching of polymer creates cooling	Not known		Electric	1

[1] Compared with conventional vapor compression, from bad (1) to good (7)

Besides the heat-input technologies, which allow reduced reliance on the CK generator, the most promising development may be the acoustic compressor. This technology would be used with a standard vapor compression refrigeration cycle. Its main potential advantage is improved reliability as compared with conventional compressors.

4. Enabling Technologies

A number of enabling technologies have been investigated, including insulation, batteries, power generation and low-input burners. Table 3 summarizes the technology category, their functions, and the technologies involved. The enabling technology categories are discussed briefly below. Some of these technologies are incorporated into the system options described in the following sections. Additional detail regarding the enabling technologies is included in the appendices.

Table 3. Enabling Technologies

Category	Function	Technologies
Insulation	Reduce Cabinet Heat Leak	• Blown Foam (polyurethane and HFC, HCFC, or other blowing agent) • Vacuum Systems
Batteries	Power Storage for Electric Input Systems	• Lead Acid • Sealed Lead-Acid • Nickel-Cadium (NiCd) • Nickel-Metal Hydride (NiMH)
Power Generation	Night Power	• Engine Generator • Sodium Cell (AMTEC) • Stirling Engine • Fuel Cell
Low-Input Burner	Heat Input for Self-Contained Heat Input Systems	• Elecrostatic Atomization • Pressurized Atomization

4.1 Insulation

Improvement in insulation technology would have the obvious benefit of reducing the refrigeration load during night storage. Alternatively, a thin-walled cabinet could be fielded, which will reduce required volume and weight. Vacuum insulation systems have the potential to significantly reduce thermal loads. However, due to conduction around the edges of metal-foil-sealed vacuum packages, and the uncertainty regarding liner diffusivity of plastic-based vacuum packages, there is no system that is sure to provide significantly better performance than conventional blown foam insulations. Future developments and testing of vacuum systems may result in improved insulation.

4.2 Batteries

Battery systems would allow night use of electricity generated during the day. Lead-Acid battery technologies can be used, but they are heavy. Future battery system developments may significantly reduce the weight penalty of using batteries.

4.3 Power Generation

A variety of power generation technologies are being pursued by a variety of organizations for compact man-portable power applications. Engine generators are noisy, somewhat unreliable, and are maintenance-intensive. The other technology options require significant development and testing to prove their viability for powering a CK cold storage refrigerator.

4.4 Low-Input Burners

Low-input burners would be needed to power self-contained heat-input refrigeration systems. Otherwise these heat-input systems will have to adopt night strategies such as thermal storage, reducing their potential improvement over vapor compression system performance and reliability.

5. General Assumptions

The cold storage system options described in the following sections have been characterized based on the general assumptions described below.

The cold storage cabinet is assumed to have 80 ft^3 internal storage volume and a 37°F average internal temperature. Its size and weight is scaled from those of a 72 ft^3 commercial reach-in with an aluminum interior liner. It is assumed that the cabinet can be redesigned to completely eliminate the need for electric anti-sweat heating, and such that the steady heat load is 650 Btu/hr. In addition to this steady heat load, an additional 2,000 Btu of door-opening load is assumed to apply for the 10 day hours. No food temperature pulldown load is considered, since it is assumed that the food arrives at storage temperature.

The refrigeration system capacity is sized to exceed average load as follows:

- For systems without thermal storage, the capacity is 2.5 times average daytime load
- For systems with built-in thermal storage, the capacity is 1.5 times average daytime load (including load required for regeneration of the thermal storage media). The oversize factor for the systems with thermal storage can be less because the thermal storage helps them handle peak loads. Still, this sizing approach results in a larger capacity for systems with thermal storage (2,640 Btu/hr as compared with 2,125 Btu/hr).

In estimating refrigeration system energy use, the design-condition COPs are assumed to apply at all times (this simplification will result in an overestimation of average energy use). Refrigeration system off-cycle losses were not taken into consideration.

For thermal storage size and weight estimates, the thermal storage media utilization is assumed to be 100%. This will result in a slight underestimation of required thermal storage.

Power system assumptions are as follows:

- The CK generator efficiency is assumed to be 23% based on fuel high heating value.
- The efficiency of low-input burners for heat-input refrigeration systems is assumed to be 65%, based on fuel high heating value. Heat input from the CK Thermal Fluid System is also assumed to be delivered with a 65% net efficiency (including distribution losses).
- Power generated by thermoelectric generators incorporated into heat-input refrigeration systems and utilizing flue gas waste heat is assumed to be "free" up to a total combustion efficiency of 75%. Any additional power generation will require additional fuel consumption. Thermoelectric generator efficiency (watts out divided by heat input) is assumed to be 3%. For most of the heat-input systems considered, this means that the first ~10W of power requires no additional fuel use.
- Fuel heating value is assumed to be 140,000 BTU/gal.

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6. Vapor Compression Cold Storage Systems

A viable vapor compression system would be based on an improved commercial reach-in refrigerator.

We used the True T-72 (72-ft^3) as the representative commercial model (see Figure 2). Daily electricity use of such a conventional reach-in would be much too high to be acceptable for the CK because (1) it uses a constant 160W of electric anti-sweat, (2) the high level of anti-sweat suggests that heat leak in the door frame area is high and could be significantly reduced, (3) the inefficient shaded-pole evaporator fan motors use 160W of power and run continuously, and (4) the compressor has only modest efficiency. The following modifications would significantly reduce daily electricity requirement (from about 12 kWh to about 2 kWh):

- Improve door gasket area
- Eliminate electric anti-sweat heating
- Use high-EER compressors such as the Americold RSD series[1]
- Use fans with Brushless DC motors; cycle the evaporator fan (rather than run it continuously)
- Use better insulation and/or consider insulation thickness increase

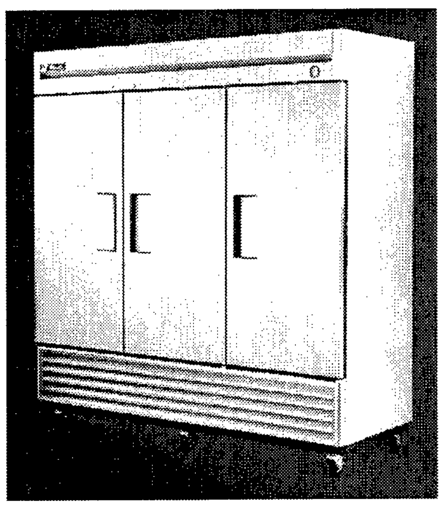

Figure 2. Representative Commercial Reach-In Refrigerator (True T-72)

A battery system or generator would be required for night operation. System performance characteristics are summarized in Table 4.

[1] This compressor line has an EER of about 6.0 at standard ASHRAE rating conditions of -10°F evaporator temperature, 130°F condenser temperature, and 90°F suction and liquid temperatures. It uses brushless DC motor technology, and, while targeted at residential markets, could be used for the CK application.

The table compares cold storage system options using high-efficiency vapor compression with a dedicated night power source. The system using the AMTEC Generator is the most attractive in terms of size, weight, and fuel use. However, this generator technology is not yet fully developed. The engine generator is heavy, would be somewhat maintenance intensive, and it is not an off-the-shelf product. However, it would have the advantage over the battery system in that it could potentially be operational for longer periods of time, for instance during times when the main CK generator is undergoing repairs.

Table 4. Comparison of Battery-Powered and Dedicated-Generator Powered Vapor-Compression Cold Storage Systems

Night Power Source	NiMH Battery	750W Multifuel Engine Generator	500W AMTEC Generator
Weight (lbs)[1]	**182**	**196**	**116**
Refrigeration System	96	96	96
Night Power System	86	100	20
Volume (ft^3)[1]	**10**	**13**	**10**
Refrigeration System	9	9	9
Night Power System	1	4	0.8
Daily Fuel Use (gallons)	0.27	0.19	0.19

[1]In addition to insulated-box storage container

The improved cabinet and refrigeration system would also be used for the thermal storage system options discussed in the next section.

7. Thermal Storage

One of the basic options for un-powered overnight fresh food refrigeration is thermal storage. During the approximately ten hours of daytime electric generator operation, the thermal storage media, in whatever form, is cooled. During the un-powered overnight hours, as heat leaks into the refrigerated space, the thermal storage media absorbs this heat at a sufficiently low temperature so that acceptable fresh food storage temperatures are maintained within the refrigerated space.

For the purposes of this study and a comparative evaluation of several alternative thermal storage configurations, the basic requirements for an 80 ft^3 refrigerated cabinet were assumed to be:

- Maximum un-powered storage time: 14 consecutive hours
- Minimum time of daily electric power availability, between un-powered periods: 10 hours
- The kitchen is "closed" during the un-powered time period; no refrigerator door openings occur during this time.
- The improvements in cabinet thermal design discussed in the previous section are implemented. The resulting closed-door cabinet heat leak (80 ft^3, worst case, 120°F ambient, 37°F interior temperature): 650 Btu/hr. For 40 ft^3: 400 Btu/hr.
- Worst case total 14-hour heat leak: 9100 Btu (80 ft^3); 5600 Btu for 40 ft^3

7.1 Candidate Thermal Storage Media

Table 5 summarizes the properties, as applied to 14 hour-un-powered periods for 40 ft^3 and 80 ft^3 refrigerated cabinets.

Table 5. Properties of Alternative Thermal Storage Media

Material		Trans-ition Temp. oF	Specific Thermal Storage Btu/lbm	Density (lbs/ft^3)	Estimated Material Cost $/lb	Pounds of Medium for 14 hr*		Volume of Medium for 14 hr*	
Base	Type					80 ft3	40 ft3	80 ft3	40 ft3
Water	Ice	32	144	56	Negl.	63	39	1.1	0.7
Paraffin	Thermasorb® 35	35	TBD (<70)	TBD	15	~140	~85	~2.6	~1.6
	Thermasorb® 43	43	70	54	15	130	80	2.4	1.5
	Thermasorb® 65	65	74	55	15	123	76	2.3	1.4
Hydrated Salt	PCM Solution	14	118	64	5	77	48	1.2	0.8
	PCM Solution	25	121	62	5	75	47	1.2	0.8
Salt Mixture	Transphase® 47	47	41	93	TBD	222	137	2.4	1.5

*For 80 ft^3, 650 Btu/hr x 14 hr = 9,100 Btu; for 40 ft^3, 400 Btu/hr x 14 hr = 5,600 Btu

We contacted a number of companies to identify and obtain basic engineering data for appropriate phase change materials (PCM's) to suit this refrigerator application. The information is summarized below:

There are primarily three types of phase change materials commercially available for thermal storage: ice, paraffin-based, and salt-based. The salt-based PCM's are split into two groups: 1) a solution of salt, water, and additives, and 2) a single component hydrated salt that simply changes phase at a desired temperature. Both paraffin PCM's and salt PCM's are available in a wide range of transition temperatures, both above or below the freezing temperature of water. The current transition temperatures for existing products are demand driven, and not determined by a particular limitation of the material class.

It is not clear at this time if one type of material has a distinct advantage over the other. It has been difficult to obtain an unbiased source; i.e., one that is not a manufacturer or trying to sell their particular product. The name of an independent consultant who has done work in thermal storage for European appliances was obtained, but we were unsuccessful in reaching him. There is, then, some disagreement as to which type of PCM would be most appropriate for this application.

The manufacturers of salt mixtures claim that paraffin has some inherent problems. Paraffin waxes are somewhat flammable because they are comprised of hydrocarbon chains. The latent heat of paraffin waxes degrades with time and number of cycles. The performance of the paraffin wax is directly related to the amount of impurities left after processing. This implies an increased cost for increased performance.

On the other hand, eutectic salts are notoriously corrosive to metal. They can be potentially hazardous, especially in contact with food, and tend to expand more on freezing.

There are obviously many issues involved in the successful design of an effective thermal storage system. For the purpose of this project, we have focused on the thermal properties of a PCM as compared to ice. In modeling of different system options the thermal properties have been used to calculate a volumetric storage capacity (Btu/ft^3). Depending on the design of the thermal storage system, the PCM will have varying effectiveness. We have assumed an effectiveness factor of 100%, meaning that all of the thermal energy of the PCM has been utilized by the end of the night period.

7.2 Manufacturers of Thermal Storage Materials and Contact Information

For future reference, the following is a list of the most promising contacts. Each is a manufacturer of phase change materials. Most are receptive to providing samples and design input in the future.

Company: Frisby Technologies
Contact: Ted Mielnik
Phone: 336-750-0911

Frisby manufactures a paraffin wax PCM using the name Thermasorb®. Thermasorb® is actually a phase change material micro-encapsulated in a plastic shell. The temperature range of the Thermasorb® PCM ranges between -20°F and 200°F. Thermasorb® is advertised at specific transition temperatures, but it is possible to manufacture at custom temperatures, depending on requirements. Frisby is currently doing research in eutectic salt mixtures.

Company: Outlast Technologies
Contact: Monte Migel
Phone: 303-581-0801

Outlast Technologies manufactures a paraffin wax PCM for clothing applications. This limits their temperature range to 65°F – 110°F. However, it is possible for them to manufacture PCM's for different temperature ranges provided there is a market demand.

Company: PCM Thermal Solutions
Contact: Maurice Marangui
Title: President
Phone: 630-653-5452

PCM Thermal Solutions manufactures hydrated salts and custom thermal storage systems. Their transition temperature range varies from -4°C (25°F) to +31°C (88°F). They are currently developing a salt to change phase at 5°C (35°F). It is important to note that these are not eutectic salt solutions, these are single component systems that change phase. Mr. Marangui would be interested in further discussion to develop possible system concepts.

Other Contacts: We were unable to make contact with three other companies that we have dealt with in the past:

- Microclimate Systems makes PCM's for many different applications and temperature ranges. They have been receptive to new designs in the past and could probably add insight to this application.
- Exothermal Technologies does similar PCM work for cool packs and wine coolers, etc.
- Transphase Systems Inc. developed Salt-water solutions for thermal storage at 47°F for building applications.

Some samples and a few specification sheets were obtained. They are for existing products that don't necessarily meet the temperature requirements for the field refrigerator, but are still interesting.

7.3 Candidate Thermal Storage Configurations

Five different thermal storage configurations were screened from a larger list for evaluation. They represent a range of possibilities, using different thermal storage materials and interfacing with both the refrigerated space and active refrigeration in different ways. Some are completely passive during the unpowered time interval, while others consume a small amount of parasitic power to run the fan(s) and/or pump(s). The five configurations, with the basic rationale for each, are discussed below:

A. Passive Thermal Storage Walls: All six walls – two sides, top, bottom, rear, and doors – are lined with phase change thermal storage material, with sufficient thickness to absorb 14 hours of heat leak. The storage media is regenerated by the internal refrigerator air during the day, which is maintained at this time at a temperature below the media's 40°F transition temperature. The design of the cabinet and the refrigeration system with fan forced evaporator is otherwise conventional except that the capacity of the refrigeration system is higher so that it can recharge the thermal storage during the on time. During the off-cycle, no parasitic power consuming fan operation would be necessary, because all six walls would intercept wall heat leak before it reaches the refrigerated space. It is a stand-alone solution and would not integrate with the central glycol-chiller and glycol cooled container concepts.

B. Direct Expansion Cold Wall with Integral Thermal Storage: This concept is similar to the preceding configuration, but uses a cold wall evaporator on the 5 stationary walls. The thermal storage material would line all six inner walls as in the previous concept. Operation would be passive during the power off period. This concept also is a stand-alone solution and does not integrate with a glycol loop or with other external cold source concepts.

C. Slide-In Flat Packages of Ice or Eutectic: Flat containers of water or eutectic would be frozen in a separate freezing unit. At the beginning of a power-off time period, these frozen flat packages would be inserted into racks on the wall of the cabinet and provide the required cooling. This concept requires a separate freezing system and provides flexibility in that the frozen ice/eutectic container can be used with a variety of cabinet and insulated container configurations, subject to the fit with the overall logistics arrangements. The ice/eutectic containers would be used in place of active refrigeration in the cabinet. Both vapor compression and chemisorption freezing systems have been examined.

D. "Glycol" Cooled Cold Wall with Integral Thermal Storage Material: This configuration is similar to the "Direct Expansion Cold Wall with Integral Thermal Storage" configuration, but chilled water-glycol, or other coolant, would circulate through the cold-wall cooling tubes, both to provide active cooling and to refreeze the thermal storage media lining the walls. The cabinet has no refrigeration system built in, but relies on the separate glycol-chilling unit. This approach has considerable flexibility and fits into the overall glycol-cooled refrigerator and insulated shipping container fresh food supply concept that is described in Section 1.

E. Remote Storage: In this configuration, the thermal storage capacity is located outside the refrigerated cabinet. Glycol is circulated from the thermal storage to the cabinet during the night period.

For the purpose of providing a consistent, simple framework for comparison, all of these configurations were evaluated for the task of maintaining the cold temperature of the 80 ft^3 cabinet shown in Figure 3, while un-powered for up to 14 hours. The concepts are described in more detail in the following sections.

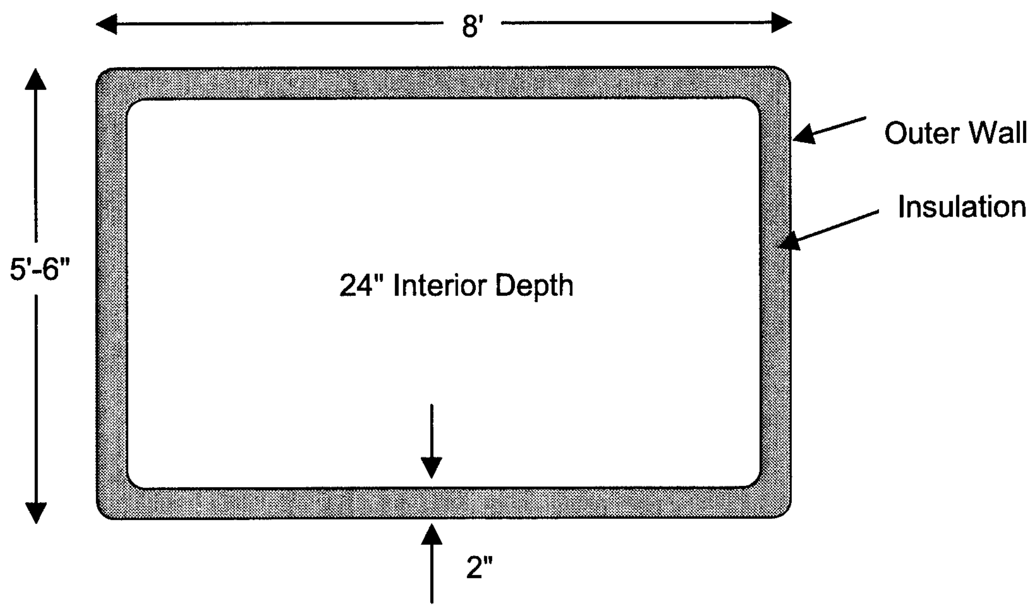

Figure 3. Generic 80 ft^3 Cabinet Used to Illustrate the Different Thermal Storage Concepts

7.3.1 Option A: Passive Thermal Storage Walls

As shown in Figure 4, all six walls – two sides, top, bottom, rear, and doors – are lined with phase change thermal storage material, with sufficient thickness to absorb 14 hours worth of heat leak. The design of the cabinet and the refrigeration system with fan-forced evaporator is otherwise conventional except the capacity of the refrigeration system is higher so that it can recharge the thermal storage during the on time. During the off-cycle, no parasitic power consuming fan operation would be necessary, because all six walls would intercept wall heat leak before it reaches the refrigerated space. It is a stand-alone solution and would not integrate with the central glycol chiller and glycol-cooled container concepts. The basic configuration is described below:

- All walls including the doors lined with micro-encapsulated, ~40°F PCM (e.g., Thermasorb® 43)
- Based on the properties of Thermasorb® 43, 2.5 ft^3 of the material is needed
- Average thickness of 0.0184 ft (.22 inch) distributed uniformly over 136 ft^2 of interior wall surface

- During the daytime running cycle, the forced cold air refrigeration system would maintain an interior temperature of 35 – 38°F. The PCM layer would freeze gradually during this 10-hour period.
- During the nighttime period, the PCM in the walls would intercept the wall heat leak from all sides, keeping the interior at 40°F.

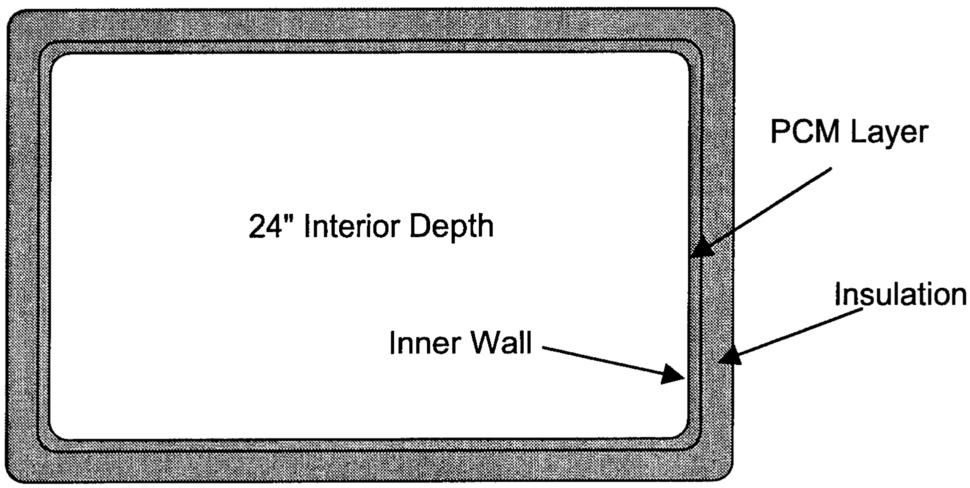

Figure 4. Passive Thermal Storage Wall Concept

Two alternate methods of integrating the PCM with the cabinet inner wall are shown in Figure 5. In one configuration, micro encapsulated PCM is contained between the wall insulation and the inner liner. In the other, the PCM is contained between the inner liner and thin (~0.005-inch thick) sheet metal which is dimpled and spot welded to the inner wall at regular intervals.

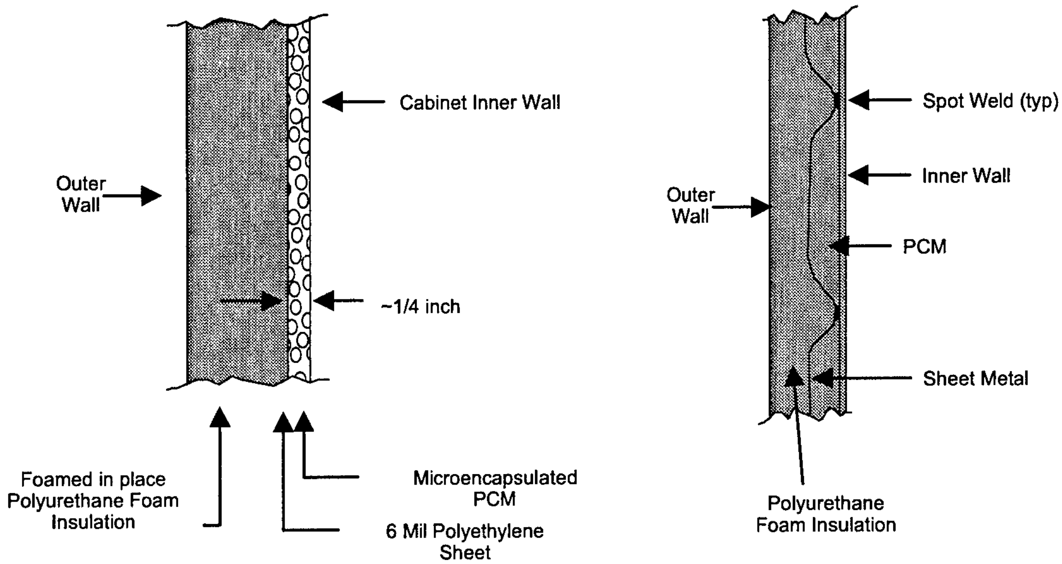

Figure 5. Passive Eutectic Walls - Construction Details (Two Options)

The basic heat transfer consideration for the Passive Eutectic Walls concept is whether or not the PCM can refreeze quickly enough during the 10-hour on-cycle. The following estimates show that this is technically feasible.

- Heat flux required is 9100 Btu $\div$ 136 ft^2 $\div$ 10 hr = 6.7 Btu/hr - ft^2
- The forced convection heat transfer coefficient at the walls will be on the order of 2 Btu/hr-ft^2-°F
- Temperature difference 43°F - 37°F = 6°F
- $h\Delta T = 2 \times 6 = 12$ Btu/hr -ft^2, about double the 6.7 Btu/hr-ft^2 heat flux needed

The added weight and volume were estimated:

- Added weight
  - PCM: 9100 $\div$ 70 = 130 lbs
  - 6 mil polyethylene liner: 136 ft^2 $\times$ 144 in^2/ft^2 $\times$ .006 inch $\times$ .04 lb/in^3 $\cong$ 5 lbs
  - Increased weight of sheet metal due to increased outer cabinet dimension to maintain interior volume: .283 lb/in^3 x .031 inch $\times$ ½" (96 in + 28 in + 66 in) $\cong$ 10 lbs
  - The refrigeration system would weight ~110lbs (added capacity as compared with high-efficiency vapor compression systems of Section 6 would increase weight).
  - Total (all in the cabinet) 255 lbs

- Added volume
  - PCM: 2.4 ft^3
  - Refrigeration system: 9 ft^3
  - Other items, above < 0.1 ft^3
  - Total: 11.5 ft^3

7.3.2 Option B: Direct Expansion Cold Wall with Integral Thermal Storage

This is similar to the preceding configuration, but uses a cold wall evaporator (on all 5 stationary walls). The thermal storage material lines all six inner walls as in the previous concept. Operation is completely passive during the power off period. This concept is a stand-alone solution and does not integrate with a glycol loop or with other external cold source concepts. Figure 6 illustrates this concept.

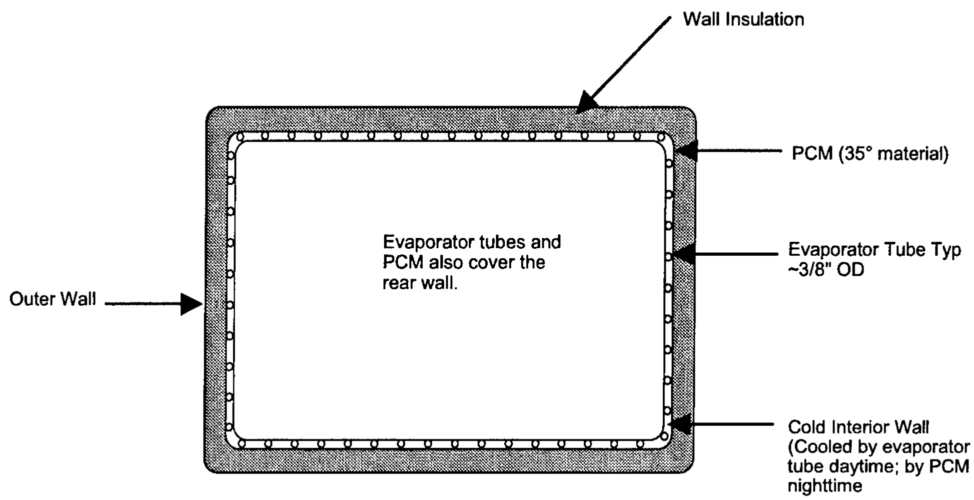

Figure 6. Direct Expansion Cold Wall with Integral Thermal Storage

Optional - incrementally higher temperature PCM (~43°F) integrated with inside wall of door to intercept nighttime heat leak.

Qualitatively, the operating modes are described briefly below.

During daytime, electrically powered operation:

- Cooled by conventional electrically powered DX refrigeration. During compressor run periods, the evaporating temperature is ~25°F; the wall temperature is ~28°F.
- Side, top, bottom, and back walls would intercept wall conduction
- Cold wall @ 28°F handles conduction heat gain through the doors and the heat load due to door openings
- By maintaining the wall and the evaporator tubes below the PCM freezing temperature, the PCM would be refrozen during the day

During closed door nighttime operation:

- The PCM in the wall would intercept heat leak, maintain cold wall temp ~36°F (door wall temperature at 43°F, intercepting the door heat leak directly).

- Alternatively, if the door does not have a PCM liner on its inner wall, the 36°F cold wall handles conduction through the door.
- Completely passive operation, without any power required for fans

The analysis of key heat transfer paths is outlined below.

- Daytime mode - cold wall capacity to handle load from door openings
  - Assume door open load is 2/3 sensible, 1/3 latent (humidity)
  - h = 1.5 (combined sensible, latent, radiant)
  - Capacity with interior air temperature held to a maximum of 40°F:
  - $1.5 \times (40\text{-}28) \times 92 = 1{,}650$ Btu/hr.

This is probably sufficient capacity, but forced air circulation within the cabinet could increase this significantly, if necessary. Cold wall partition(s) within the cabinet also could increase the capacity (because of their added surface area), if necessary.

- Overnight mode - interior air temperature needed to transfer heat leak through doors to the thermal storage cold walls (assuming that no thermal storage is on the inner wall of the door(s)
  - Door area/total interior wall area - $40\ ft^2/(40 + 40 + 52)\ ft^2 = 30\%$
  - 30% of heat leak = $0.3 \times 650 = 200$ Btu/hr
  - h ~ 1.0 Btu/hr-ft^2-°F (from any interior wall to air and the cabinet contents/outer walls, via natural convection and radiation combined)
  - $200 \div (1.0\ \text{Btu/hr-ft}^2\text{-°F} \times 92\ \text{ft}^2) = 2.2$°F
  - Interior air temperature = 36°F + 2.2°F = 38°F
  - The temperature of the inside wall of the doors is approximately 38°F + 200 ÷ (1°× 40) = 43°F

A few other aspects of the DX Cold Wall with Integral Thermal Storage concept include:

- Any daytime frost accumulation on the interior walls melts during the nighttime passive cooling period.
- If necessary to handle the thermal load from frequent door openings, a forced air evaporator blowing cold air down over the doors (with PCM on the interior walls) can be integrated with this design.

The added weight and volume is estimated as:

- Weight
  - PCM: 9100 ÷ 65 ≅ 140lbs
  - PCM liner: 6 mil polyethylene ~ 5lbs
  - Extra exterior sheet metal ~ 10lbs
  - 110lbs refrigeration system

- Volume – approximately 11.5 ft^3 increase as with Option A. Actually, the cold wall evaporator can be more compact than fan forced cold air delivery to the cabinet.

7.3.3 Option C: Slide-In Flat Packages of Ice or Eutectic

This concept, illustrated in Figure 7, is based on using flat containers of water or eutectic, which are frozen in a separate freezing unit. At the beginning of a power-off time period, these frozen flat packages would be inserted into racks on the wall of the cabinet to provide the required cooling. This concept would require a separate freezing system and provides flexibility in that the frozen ice/eutectic container can be used with a variety of cabinet and insulated container configurations, subject to the fit with the overall logistics arrangements. The ice/eutectic containers could also be used in place of active refrigeration altogether.

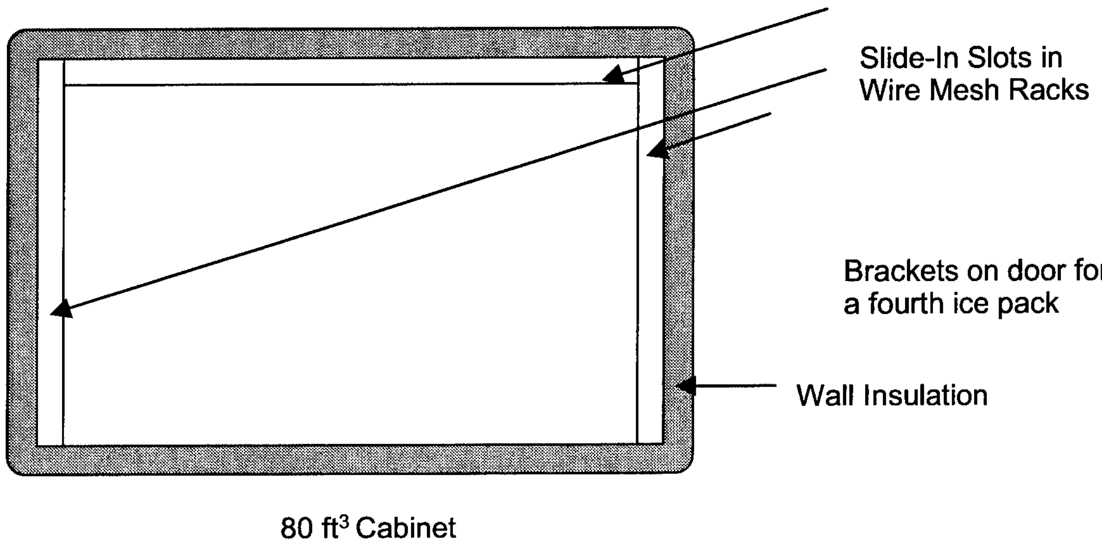

Figure 7. Slide-In Flat Packages of Ice or Eutectic

The wall area for flat packages is estimated as (not using the rear wall or bottom because they are the least accessible interior surfaces): The minimum thicknesses for unattended overnight passive cooling is estimated for using 4 sides, and for the top only (for water/ice):

Side Walls	2 x 5' x 2'	= 20 ft^2
Top	8' x 2'	= 16 ft^2
Doors	5' x 8'	= 40 ft^2
		= 76 ft^2

- Use of side walls + top + doors: 1.1 ft^3 ÷ 76 ft^2 = .0145 ft = .17 inch
- Top use only (as shown in Figure 9): 1.1 ÷ 16 = .069 ft = .825 inch ≅ 1 inch

Heat transfer from the Slide-In Flat Packages of Ice or Eutectic – Top Configuration + Door Rack: (Figure 8)

- Surface of both sides of packages at 32°F, area = 100 ft^2 top and bottom

- Average combined radiant and convective h of 1.0 Btu/hr-ft^2-°F
- 650 Btu/hr ÷ (1 Btu/hr - ft^{2}°F x 100 ft^2) = 6.5°F
- Resulting air temperature in cabinet = 32 + 6.5 ≅ 39°F

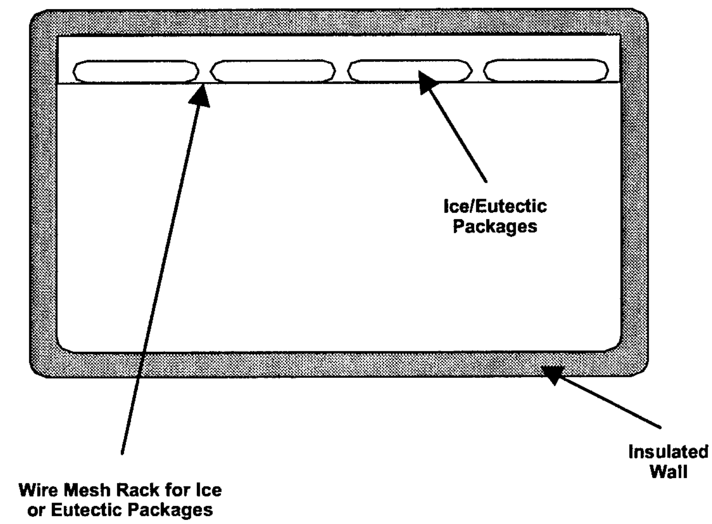

Figure 8. Slide-In Flat Packages of Ice or Eutectic Top Configuration + Door Rack

A refreezer, operating during the daytime, is needed to refreeze the ice or eutectic packages. The following applies to the electrically powered vapor compression cycle refreezer illustrated in Figure 9. Note that the overall dimensions indicated in the figure are driven by the storage volume for the ice packages and the height of the condensing unit, whose plan area is about half of the 24" x 36" overall plan dimensions.

- Chest will hold up to 3 ft^3 of ice/eutectic packages, for both the overnight mode (1.1 ft^3) and other uses such as shipping containers
- Need capacity to refreeze ~ 1/2 ft^3 of ice/hr, 4000 Btu/hr
- Air temp leaving evaporator ~ 10°F
- T_{evap} ~0°F
- Surface area of packages ≅ 200 ft^2; forced convection h ≅ 4 Btu/hr ft^2 - °F
- Heat transfer rate = 200 x 4 x (32 – 0) = 25,000 Btu/hr, well in excess of 4,000
- Condensing unit - medium temp, R134a, nominal 1 HP

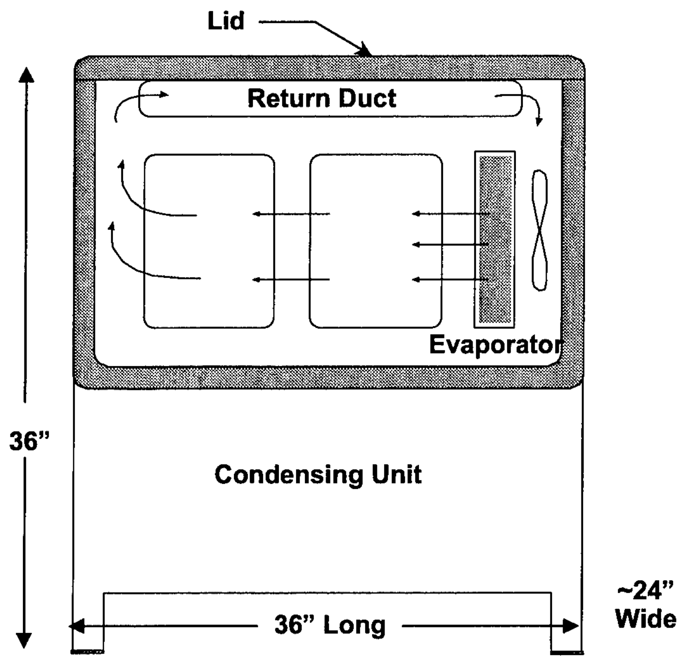

Figure 9. Slide-In Flat Packages of Ice or Eutectic, Chest-Type Refreezer

The added weight and volume was estimated:

- Added cabinet volume for ice packages and air circulation space around the ice packages: ~ 2 ft^3
- Added cabinet weight
  - Inner and outer liners: ~10lbs
  - Wire racks for ice packages: ~5lbs
- Ice Package Weight: Two sets of ice packages are required; one for day use and the other for night use. Ice packages must be sized for use beyond the 14-hour night period in order to allow for some refreezing of the day packages prior to full thaw of the night packages. Two package sets with sufficient ice for 16 hours operation each would weigh 144lbs total.
- Volume of the refreezer, based on the dimensions indicated for the unit in Figure 7: 3 ft x 3 ft x 2 ft = 18 ft^3
- Rough estimate of the weight of the refreezer, also based on the dimensions indicated for the unit in Figure 7~175lbs

An alternative ice package system could be developed in which the refreezer volume is reduced by allowing only about one-half of the required night ice to fit into the refreezer. Additional space in the storage cabinet would be used to hold refrozen ice packages which may not yet be in use. Such a concept, described in a fax dated 4/28/99 to Rocky Research[1], could result in a somewhat smaller total system volume. The refreezer ice package area dimensions would conservatively be 15"W x 27"L x 24"H, about 6 ft^3. The added volume in the storage cabinet is 8 ft^3, resulting in a 16 ft^3 total volume addition (as compared with 20 ft^3 for the above concept).

One major drawback of this system option using a vapor compression refreezer is the large electric demand of the 1-hp condensing unit, which will far exceed the 500W maximum electric load called for in Table 1 of Section 1. The following option gets around this issue.

7.3.4 Thermally Activated, "Complex Compound-Ammonia" Cycle Based Refreezer

Refrigeration for the chest type refreezer illustrated above could be provided by a complex compound-ammonia cycle (Rocky Research technology). For the performance requirement of:

- Refreezing 36 lbm ice in 2 hrs
- Resulting cooling capacity 2,600 Btu/hr
- With evaporator temperature of 0°F

Rocky Research provided the following preliminary engineering estimate of the basic system design and operating parameters:

- Mass of sorbers: 12.7 lbm total
- Mass for complete system: 50 lbm
- System volume: 1.6 ft^3
- COP thermal: 0.30
- Temp for desorption: 370°F

At the projected volume of 1.6 ft^3, the complex-compound system would fit comfortably in the condensing unit space shown in Figure 7.

The complex compound-ammonia cycle operates on a batch basis, in the following sequence:

Freezing - Liquid ammonia from the storage receiver is expanded into the evaporator, where it boils in conventional fashion to provide cooling. The ammonia vapor passes through an interchanger with the ammonia liquid line from the receiver to maximize the refrigeration effect, then into the complex compound sorbent which is air cooled.

[1] A copy of this fax is attached as an appendix.

- Desorption – When the receiver has been emptied of liquid ammonia and the complex compound sorbent is fully loaded with adsorbed ammonia vapor, it is necessary to regenerate the system by heating the sorbent to 370°F to desorb the ammonia vapor at sufficiently high pressure that it can be condensed in an air cooled condenser. Heat would be supplied by a logistics fuel burner. If regeneration were to occur within 30 minutes, the required heat input would be (2,600 Btu ÷ 0.3) ÷ 0.5 hr = 17,300 Btu/hr. This load could be supplied by the Thermal Fluid system, thus eliminating the need for a separate burner.

The system as described above would consume a moderate amount of parasitic power when operating:

- 10 – 20W of evaporator fan power (during the freezing part of the cycle)
- 20 – 30W of cooling fan power (desorption)
- 30 – 40W of fuel pump and combustion air blower power (desorption)

This load would be reduced depending on system design; for instance, use of the Thermal Fluid system would eliminate the additional fuel pump and combustion air blower load.

7.3.5 Option D: "Glycol" Cooled Cold Wall with Integral Thermal Storage Material

This configuration is similar to the "Direct Expansion Cold Wall with Integral Thermal Storage" configuration, but chilled water-glycol, or other coolant, would circulate through the cold-wall cooling tubes, both to provide active cooling and to refreeze the thermal storage media lining the walls. The cabinet has no refrigeration system built in, but relies on the separate glycol-chilling unit. This approach has considerable flexibility and fits into the overall glycol cooled refrigerator and insulated shipping container fresh food supply concept that is described in Section 1. Weight would be greater than for Option B because of the additional heat exchanger and the required glycol pump and piping. Added volume would be reduced, however, since there would be more flexibility in packaging of the refrigeration system.

7.3.6 Option E: Remote Storage

In this configuration, the thermal storage capacity is located outside the refrigerated cabinet, as indicated in Figure 10. Active refrigeration for the cabinet and for the thermal storage can be provided in a variety of ways.

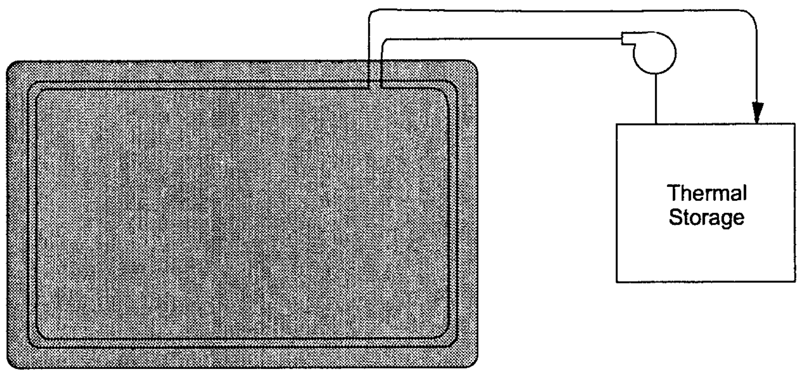

Figure 10. Remote Storage – Generic Configuration

Several options for configuring the external thermal storage, the active refrigeration and the interface with the cabinet could be considered:

- Cold Wall or forced air cooling of the cabinet; Glycol loop cooling at night, DX cooling during the day
- Daytime operation of a DX system which would recharge the storage and cools the cabinet; nighttime transfer of cooling using a liquid refrigerant pump
- Glycol Loop integrated w/refrigerator cabinet and storage, as shown in Figure 11. Table 6 summarizes the glycol flow rate and the powered daytime and passive nighttime operating temperatures.

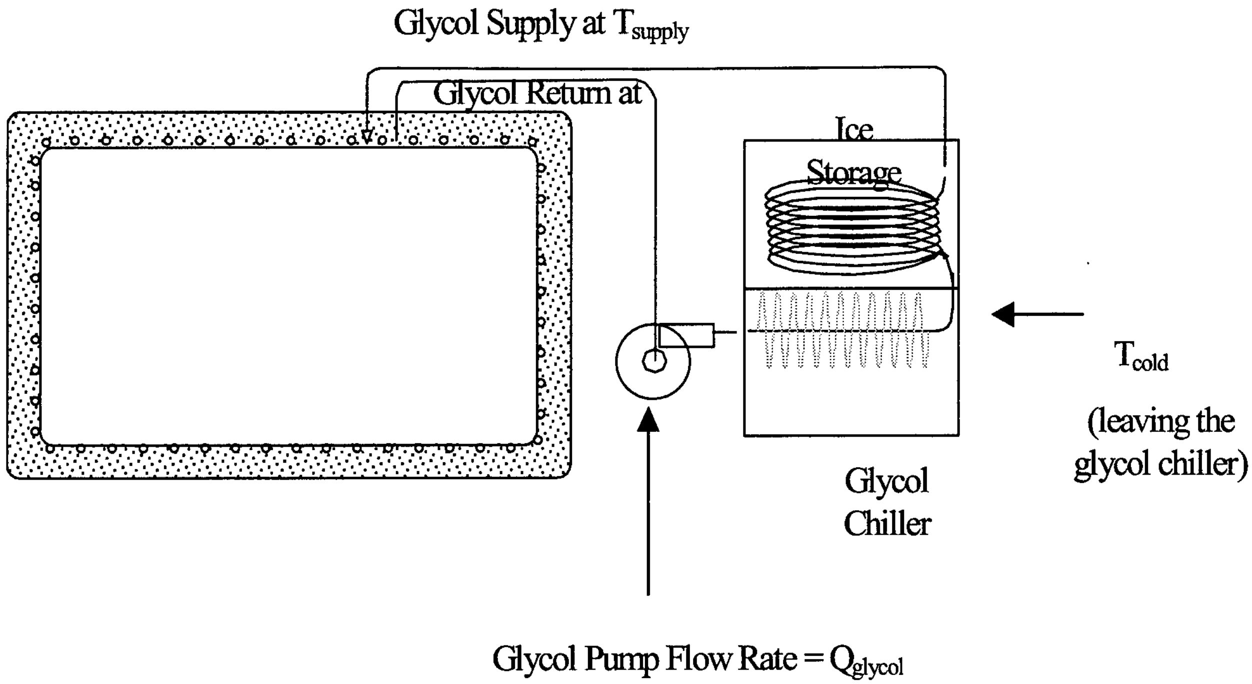

Figure 11. Remote Storage – "Glycol" Loop Integrated with Thermal Storage and Cold Wall.

The system would also require a night power source for the pump. The pump power requirement would be about 20 –30W, so a battery system with about 200 – 300Wh of storage would be adequate.

Table 6. Glycol" Loop Integrated with Thermal Storage and Cold Wall

Mode	Temperature(°F)			Q_{glycol} (gpm)	Cooling Capacity (Btu/hr)
	T cold	T supply	T return		
Daytime	20	25	35	1.0	4,000 Cabinet; 2,000 Refreezing
Nighttime	37	35	37	1.0	800

Ice is the assumed storage medium

The increase in weight and volume of this system was estimated:

- Added weight
  - No added cabinet weight – glycol tubing cold wall or glycol to air HX is needed with or without external storage being involved.
  - Ice storage sink integrated with the glycol chiller: 9100 ÷ 145 = 63lbs
  - Insulated container – estimated 25lbs
  - Glycol coil in ice sink – estimated 10lbs
  - Glycol chiller – estimated 120lbs
  - NiMH battery system (for night pump operation) – estimated 24lbs

- Added volume
  - Ice: 63 ÷ 57 = 1.2 ft^3
  - Insulated walls: 1.5 ft^3
  - Glycol coil: < 0.1 ft^3
  - Chiller: 8 ft^3
  - Battery system: 1 ft^3

7.4 Thermal Storage Weight and Volume Comparison

Weight and volume are important considerations because the basic application is for mobile field kitchens. Table 7 compares the weight and volume impacts of each thermal storage option that has been examined (see the end of each subsection in Section 7.3 for the basis of the estimated added weight and volume for each thermal storage concept). The comparison is on a consistent basis of providing 9100 Btu of stored cold over 14 hours for an 80 ft^3 cabinet, as outlined at the beginning of this section.

Table 7. Thermal Storage Weight and Volume Comparison (for 80 ft^3 cabinet)

Options	Thermal Storage Material	Increased Weight (lbs)		Increased Volume (ft^3)	
		PCM	Other Material	Increased Cabinet	Outside Cabinet
A: Passive Thermal Storage Walls	Thermasorb® 43	130	125	11.5	0
B: DX Cold Wall with Integral Storage	Thermasorb® 35	140	125	11.5	0
C: Slide in Ice with External Freezer	Water	144[1]	~115[2] or 190[3]	2.0	18
D: Glycol Cold Wall with Integral Storage	Thermasorb® 35	140	135	2.5	8
E: Remote Storage with Glycol Cold Wall	Water	63	180	0	12[4]

Note: Increases are as compared with the bare insulated-box cabinet.

[1] The weight includes ice packages required in addition to the packages needed for the night period.

[2] Including 100lbs for chemisorption refreezer.

[3] Including 175lbs for vapor compression refreezer.

[4] 243lbs and 12 ft^3 are the estimated added weight and volume, respectively, of the ice storage compartment, the glycol chiller, and the battery system in Option E.

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8. Absorption Systems

The following system concepts use refrigeration technologies powered primarily with heat rather than electricity. This means that low-fuel-input burners can be used to drive the systems using logistics fuels, thus allowing 24-hour operation. The systems examined are based on the following refrigeration technologies:

- Absorption
- Chemisorption
- Zeolite Adsorption

8.1 Modification of the Classic Hydrogen Assist Unit

Absorption refrigeration involves the use of a liquid sorbent to absorb evaporated refrigerant in an ambiently-cooled absorber. The sorbent/refrigerant solution is pumped up to high pressure to the generator, where heat is used to drive off the refrigerant. The refrigerant vapor is then condensed at high pressure, after which it can be used to provide cooling in the evaporator.

The most viable concept for development of a absorption-based CK cold storage system would be based on a modification of the Platten-Munters Ammonia-Water-Hydrogen cycle commercialized in 1925 by Electrolux and still used for RV refrigeration. This cycle is depicted in Figure 12 below. The hydrogen gas added to the evaporator/absorber side of the system allows operating with roughly equal absolute pressures throughout the system. The partial pressure of the ammonia refrigerant vapor in the evaporator/absorber side is very low, which allows evaporation of the ammonia liquid and delivery of the refrigeration effect. A low-mass gas such as hydrogen is used in order to assure gravity-driven circulation of the gas, thus assuring flow of evaporated ammonia to the absorber. The absence of a large pressure differential makes possible the use of a bubble pump for circulation of solution from the absorber to the generator.

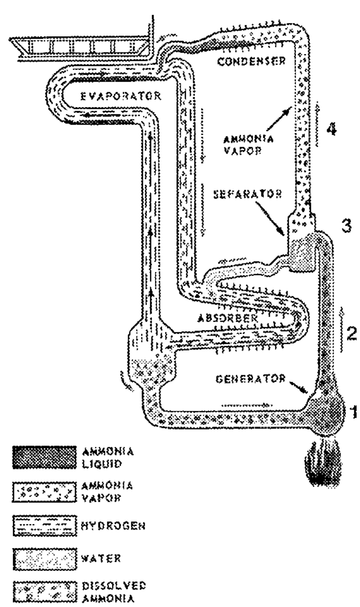

Figure 12. Platten-Munters Ammonia-Water-Hydrogen Cycle

Pumping power would be too great (about 100W) for a standard ammonia-water cycle which does not use hydrogen assist. Refrigerant/sorbent pairs other than ammonia/water could be considered, but this pair will provide the highest COP and the widest operating temperature range. The major drawback to the use of ammonia/water is the toxicity and flammability of ammonia, which may be tolerated in the well-ventilated CK.

Modifications to the classic Platten-Munters cycle would be required to make it suitable for the CK. The following issues would have to be addressed:

- Capacities required for 80 ft^3 storage
- Operation in 120°F ambient
- Reduced tilt sensitivity

Improvements to attain these objectives would be (1) use of a mechanical pump, (2) increased pressure level, and (3) modified evaporator/absorber design. These issues are discussed in more detail below.

A required enabling technology for an absorption-based cold storage system is a Low-Input Logistics Fuel Burner. Identification of a suitable mechanical circulation pump is another prerequisite technical requirement.

The design operating conditions for a representative absorption system are shown in Figure 13. The figure shows fairly optimistic values for condensing and absorbing temperature, which would require either large heat exchangers or fan power to achieve. Nevertheless, the diagram gives an indication of the system pressure and temperature levels. Naturally, the low pressure represents ammonia partial pressure rather than absolute pressure.

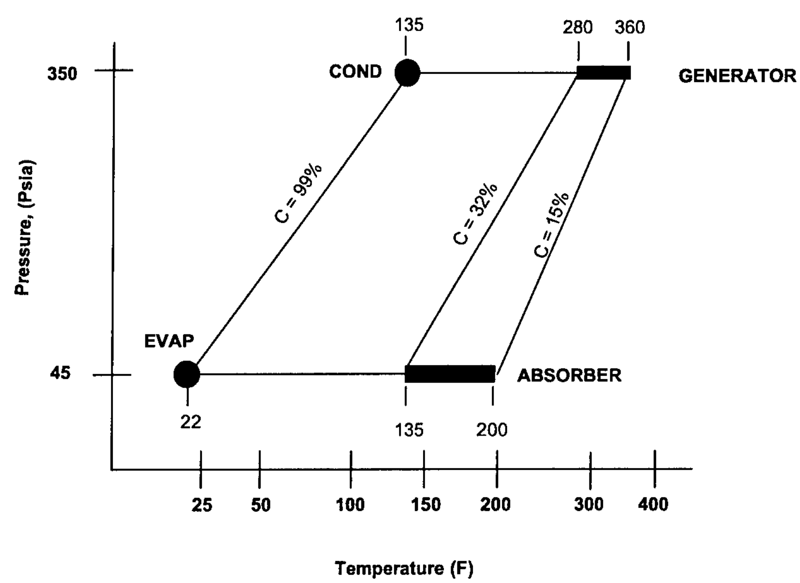

Figure 13. Asorption Cycle Representation on a P-T Chart

The bubble pump used to circulate the ammonia/water solution in a conventional hydrogen-assist absorption refrigeration unit has limited flow and/or pressure lift capability. The largest refrigeration units based on this technology have a 14 to 16 ft^3 storage volume and a refrigeration capacity of about 350Btu/hr. Ganging of such small systems may be possible, but replacing the bubble pump with a small electric-powered pump may be more effective. The pump flow rate (for a 2,500Btu/hr system capacity) should be about 0.25 lpm, and the pressure rise requirement would be in the 10-20psi range. The pump would require about 5W of power[2] and would be powered by a thermoelectric generator designed to utilize waste heat in the flue gases. Besides increasing the system capacity, use of a small pump would also allow use of a larger solution heat exchanger, and would increase component placement flexibility.

Most hydrogen-assist absorption systems are designed for operation at ambient temperatures up to about 100°F. Extension of the operating range to 120°F would require increasing the pressure level in the system (by filling with more hydrogen) so that the corresponding ammonia condensing temperature is sufficiently high. Pressurization in the 400-500psia range may be necessary. In addition, the concentration of the ammonia/water solution entering the absorber would have to be reduced, to allow adequate absorption of ammonia at the required ammonia partial pressure. This will result in an increase in the required solution flow rate, making switch to a mechanical pump doubly important.

One drawback to hydrogen-assist absorption systems which may be important in the CK is the reliance on gravity flow through many of the system components. The heat exchanger tubes are angled slightly to allow gravity flow. This makes such systems sensitive to tilt. Tilt sensitivity of a hydrogen-assist unit will be improved through the use of a mechanical pump. However, additional modification of heat exchangers may be required to insure that the gravity flow of liquid is not impeded in any of the components.

Additional considerations in the design of a modified hydrogen-assist absorption system include:

- Use of a fan to enhance heat transfer to reduce condenser/absorber size
- Use of some thermal storage to reduce capacity requirement

Rough estimates of modified hydrogen-assist absorption system characteristics are summarized below.

- Thermal Loads:
  - Evaporator 2,500 Btu/hr
  - Condenser & Absorber 9,500 Btu/hr
  - Generator 7,000 Btu/hr
- System COP:

[2] Assuming the pump/motor system efficiency is 10%

- ~0.35 Cycle COP
- ~0.23 Fuel HHV COP (Low-input burner drives cycle)

- Weight: ~180 lbs; ~5lbs ammonia
- Volume: ~ 18 ft^3 (condenser/absorber takes up the entire back of the unit)
- Electric Requirement (for thermoelectric generator):
  - 5W to 30W depending on use of forced convection to reduce condenser/absorber size
- Daily Fuel Requirement: ~84,000 Btu or 0.6 gal

8.2 Operation of Absorption Systems with the Thermal Fluid System

The absorption system concept described is operated with a low-input burner which allows heat input for 24-hours per day. An alternative approach would be to use heat from the planned CK thermal fluid system. This approach would have the benefit for absorption systems that a conventional ammonia/water system could be contemplated, since electric power input would no longer be restricted to the tens of watts. This would allow use of a pump as well as condenser/absorber and evaporator fans. However, because the thermal fluid system is in operation only for the 10 daytime hours, such a cold storage system would require some form of thermal storage or refrigerant storage.

Ammonia refrigerant storage will be a poor design choice both because of the weight and volume impact of storing the required ~20lbs of ammonia, as well as the added safety risk.

An absorption system or any other heat-input refrigeration system using thermal storage is likely to be more bulky than a vapor compression system using thermal storage. Furthermore, daily fuel usage for a heat-input system will be higher[3]. The main benefit of using heat-input systems with thermal storage is reduced reliance on the CK generator for electric power. Most of the thermal storage concepts discussed in the previous section could be adapted to work with heat-input systems. These systems are not analyzed in detail except for a system combining thermal storage Concept C (Slide-In TS packages) with Chemisorption. This is discussed in the previous section.

8.3 Chemisorption

Chemisorption is a refrigeration process related to absorption, in which the tendency of the refrigerant (ammonia) to be "absorbed" by the sorbent media is used to create the low refrigerant pressures required in the evaporator for evaporation of the refrigerant to take place. The major difference with respect to absorption is that chemisorption uses a solid sorbent. The process is similar to the zeolite refrigeration described in the next section, however, chemisorption uses a different refrigerant and sorbent pair. The sorbents used for chemisorption are referred to as complex compounds or salts, and they can be

[3] Net COP for a heat-input system using the Thermal Fluid would be about 0.5 at best. Existing vapor compression refrigerant compressors combined with the CK Generator's efficiency of 23% results in a net COP in the 0.7 to 0.85 range. This does not include fan power which may be greater for a heat-input system due to the greater heat rejection load.

selected for optimized performance with the design evaporating and condensing temperatures.

Construction of a cooling system using chemisorption is complicated by the fact that the sorbent material is solid and cannot move from an absorber to a desorber as is done with the absorption cycle in which the sorbent solution is a liquid. Instead, the sorbent bed must alternately be in an absorbing and a desorbing phase. A variety of cycle configurations have been considered and analyzed. One typical arrangement is identical to the zeolite system shown in Figure 13. Two complex compound sorbent beds would alternately be operating in either sorbing or desorbing mode. Check valves would control the refrigerant flow in and out of the beds. Ambient cooling of the sorbent bed allows the refrigerant vapor pressure to drop such that it can pull refrigerant from the evaporator. After saturation of the bed with refrigerant, heat addition to the bed increases the pressure such that the refrigerant is desorbed and flows to the condenser. Use of two sorbent beds which alternate between sorbing and desorbing modes makes the operation continuous.

Rocky Research of Boulder City, Nevada has investigated chemisorption systems for field kitchen applications, including the smaller 8 ft^3 Mobile Kitchen Trailer (MKT) refrigerator, as well as the proposed 80 ft^3 CK refrigerator. Preliminary investigation has been done for the following three most promising system options for the CK application:

- A continuously-operating system using a low-input burner.
- An ammonia storage system, in which sufficient ammonia for the night refrigeration load is condensed during the day and the sorbent bed is sufficiently large to be operational without desorbing throughout the night period. This system is too large and heavy and would require too much ammonia (about 23lbs) to be practical.
- A system based on thermal storage packages that can be refrozen during the day in preparation for use in the refrigerator at night. This system is described in the Thermal Storage Section as Option C with a chemisorption refreezer.

Preliminary estimates of equipment requirements for the continuously-operating system are as follows.

- Capacity: 1,024 Btu/hr
- System requires 50 W of parasitic power for fans
- Estimate of the cycle COP: 0.7
- The required weight of ammonia could be kept below the 6.6lbs ASHRAE-15 limit
- Total refrigeration system weight: ~100lbs
- Refrigeration system bulk volume: ~1.5 ft^3

This system will require a low-input logistics fuel burner to be practical. In addition, the relatively large 50W parasitic load will require a large thermoelectric generator, which will likely increase the fuel requirement and certainly increase size, weight, and cost.

8.4 Zeolite

Zeolite is another thermal input refrigeration option. Zeolite refrigeration systems operate by alternately adsorbing and desorbing a refrigerant in a solid zeolite sorbent bed. Zeolite is a naturally occurring alumino-silicate mineral.

The Zeopower Company of Natick, MA, develops zeolite refrigeration systems of different configurations for many applications. They have provided the system sizing estimates that follow. The Zeopower system operates in a very similar manner to Chemisorption refrigeration described in the previous section. A system schematic is shown in Figure 14. The Zeopower system uses two sorbent beds with water as the refrigerant. The refrigerant water is circulated through the system by alternately heating and cooling the sorbent beds to either adsorb water vapor from the low-pressure evaporator or desorb water vapor to the high-pressure condenser. System cycle alternating times range from two to four cycles per hour. When a sorbent is fully loaded, thermal input is required to desorb water vapor to an ambiently cooled condenser. Cycle COP is improved by the addition of an internal heat regeneration loop to transfer internal heat to the sorbent bed. This reduces the amount of external heat required by the system. The heat regeneration loop requires the use of a 20W pump to circulate the heat transfer fluid between the sorbent beds.

Water vapor as a refrigerant delivers the highest cycle efficiency due to its high heat of vaporization, but limits the lowest evaporator temperature to –2°C (28°F). Lower temperature systems are currently in development using methanol as the refrigerant.

Zeopower analyzed two system approaches for the CK:

- A continuously operating system using a thermal source over 300°F available 24-hours a day
- A system with thermal input available 10 hours on/14 hours off

Both systems assume the use of a low input burner or thermal fluid system along with the use of a thermoelectric generator for any parasitic power requirements.

- System 1 (300°F Thermal source available 24 hours):
  - System capacity 2000 Btu/hr
  - Requires 8.5 lbs of zeolite for a total system weight of 25 lbs
  - System size is 1' x 1' x 1/2'
  - Cycle COP of 0.6 to 0.8 depending on ambient temperature
  - System cycles continuously between two sorbent beds at 4 cycles/hr

- System 2 (300°F Thermal source available 10 hours on/14 hours off)
  - Base system is identical to System 1 during the 10 hour heat input period
  - The following is additionally required for the 14 hour passive period:
  - A third 100lbs sorbent bed to absorb 9.1 lbs of water for the 14 hour night load

- Additional 3,000 BTU/hr of daytime heat input to desorb the 100lbs night zeolite (COP 0.3)
- Total system weight increases to approximately 200lbs
- The system size increases to roughly 5 ft^3

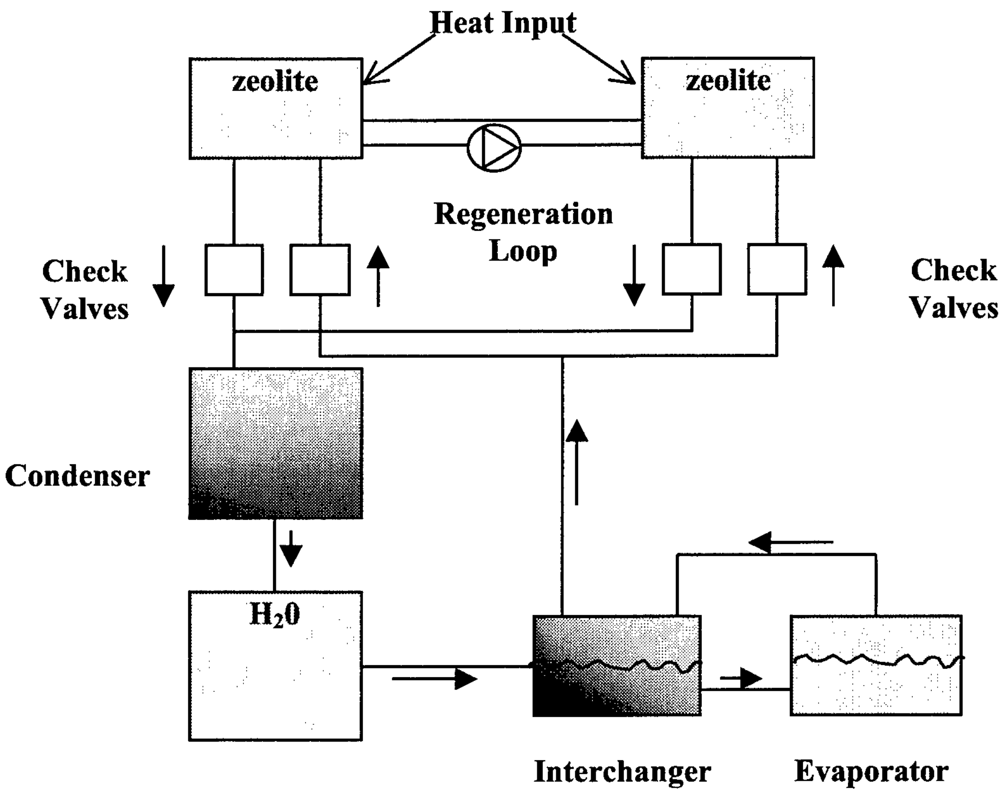

Figure 14. Zeolite System Schematic

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9. Cold Storage System Summary

The most viable of the cold storage system options discussed in the previous sections are summarized in Table 8 below.

Table 8. Cold Storage System Options Summary

Option	Weight* (lbs)			Volume* (ft^3)			Daily Fuel Use (gal)	CK Generator Depend-ence	Pulldown Capability	Technical Risk
	Integrated into Cabinet	External	Total	Integrated into Cabinet	External	Total				
High Efficiency VC w/Battery	96	86	182	9	1	10	0.27	High	Good	Low
High Efficiency VC w/ Engine Generator	96	100	196	9	4	13	0.19	None	Good	Low
High Efficiency VC w/ AMTEC Generator	96	20	116	9	0.8	10	0.19	None	Good	Medium
A: VC w/Passive Thermal Storage Walls	255	0	255	11	0	11.5	0.18	High	Medium	Medium
B: DX Cold Wall w/Integral Storage	265	0	265	11.5	0	11.5	0.16	High	Poor	Medium
C1: Slide-in Ice w/External Freezer (VC)	144	190	334	2	18	20[1]	0.3	High	Poor	Low
C2: Slide-in Ice w/ External Freezer	144	115	259	2	18	20[1]	0.7	Low	Poor	Medium
D: Glycol Cold Wall w/Integral Storage	155	120	275	2.5	8	10.5	0.31	High	Poor	Medium
E: Remote Storage w/Glycol Cold	0	243	243	0	12	12	0.38	High	Medium	Low
Chemisorption w/ Thermoelectric	130	0	130	16	0	16	0.5	None	Good	High
Absorption w/ Thermoelectric Generator	180	0	180	18	0	18	0.6	None	Good	High
Zeolite w/Thermoelectri c Generator	130	0	130	16	0	16	0.4	None	Good	High
Zeolite w/Thermal Fluid	200	0	200	21	0	21	0.49	Low	Good	Medium

* Not including cabinet weight of 460lbs and volume of 102 ft^3.
[1] May be as low as 16 ft^3 with alternative refreezer size.

The most attractive options based on weight, size, and fuel use are the high-efficiency vapor compression systems with dedicated power supplies for night operation, particularly the option using the AMTEC generator technology. However, this generator technology has not yet been fully developed. The vapor compression/battery option, which also ranks well, has the drawback that it relies on the main CK generator for daytime operation and for battery charging.

The options which incorporate thermal storage use relatively little fuel, but they are heavy and bulky. Most of these options put a relatively large load (hundreds of watts) on the main CK generator because they use vapor compression refrigeration technology. The thermal storage option involving slide-in ice packages and an external chemisorption refreezer significantly reduces main CK generator load, but it still relies on the generator for fan power and for heat input from the electric powered thermal fluid system.

The self-contained heat-input systems which use thermoelectric generators for auxiliary power have a volume penalty comparable to that of the thermal storage options, have weights competitive with the high-efficiency vapor compression options, and have relatively high fuel usage. These options require the use of low-input logistics fuels burners, which have not yet been fully developed for this type of application. Further, these systems require significant development themselves, requiring scale-up to the required size, and in the case of absorption, modifications to allow high-ambient operation. The main advantage of these systems is complete independence from external power sources.

The cold storage system options which fit best with the fresh food distribution concept discussed in Section 1 are Thermal Storage Option C: Slide-In Ice Packages (using either a Vapor Compression or Chemisorption refreezer) and Thermal Storage Option D: Glycol-Cooled Cold Wall with Integral Thermal Storage. The first of these uses ice packages both for provision of cooling and for thermal storage during transport and night cycles. The second has integral thermal storage and uses "quick" connection to a glycol cooling system, either at the main distribution facility or at the field kitchens to which foods are delivered.

From the standpoint of minimizing developmental risk, the best options would be High-Efficiency Vapor Compression with a Battery Power System, and Thermal Storage Option C: Slide-In Ice Packages using a Vapor Compression refreezer. Both of these use no new technologies and would essentially be product developments.

10. Passive Thawing

Thawing of frozen food in the CK is currently performed in an uncontrolled fashion by allowing frozen food containers to thaw in the ambient air. This procedure does not comply with regulations that call for a maximum air temperature of 45°F. A "passive" thaw system which complies with regulations, uses little or no electric power, yet allows food to be thawed within 18 hours, is needed.

Some preliminary engineering analysis of thawing and potential thaw system designs has been done. Thaw system concepts considered were as follows:

- A self-contained passive thaw container with required controls.
- An 80 ft^3 combined cold storage/thaw unit in which half of the storage volume can be used for thawing when frozen food is delivered.

The requirements for the thaw system are:

- Food is to be delivered in standard boxes used commercially
- The external temperature range is from 50°F to 125°F (the CK will be heated to 50°F if the ambient temperature is cooler).
- Maximum Thaw System Internal Temperature 45 °F
- Food to be thawed in 18 hours

Typical commercial packaging for frozen foods is summarized in Table 9 below. Alternative packaging can be obtained, but the indicated configurations are typical. Box weights are in the range of 10 to 20lbs. Hamburgers are separated with wax paper and bagged. Steak and Chicken pieces are separately wrapped. Cryovacing of food involves wrapping of the food item while it is warm, so that the package is under vacuum when frozen.

Table 9. Frozen Food Packaging Configurations

Food	Food Item Weight (oz)	Individual Packaging	Typical Box Weight (lbs)	Typical Box Volume (ft^3)
Hamburgers	8	Wax paper between patties; all patties in a single bag	10 to 20	0.62
Steak	varies	All pieces cryovaced	12 to 14	0.5
Chicken	18 to 25	All pieces cryovaced	15 (smaller of two weighed)	0.65

Source: Parkway Food Services, Greensburg, PA

Thermal properties are listed in Table 10 below. Thermal conductivity of food is significantly higher when it is frozen. The enthalpy rise required to thaw food on the order of the heat of fusion of water, but not as high.

Table 10. Frozen Food Thermal Properties

	Heat Capacity (Btu/lbF)		Conductivity (Btu/hrftF)		Density Below Freezing (lbm/cuft)	Enthalpy Rise 0°F to 32°F (Btu/lb)
	Above Freezing	Below Freezing	Above Freezing	Below Freezing		
Beef	0.80[1]	0.40[1]	0.292[2]	0.820[2]	69	110[2]
Chicken	0.79	0.37	0.238	0.797[3]	61	

Source: ASHRAE Refrigeration Handbook 1998
[1]Round, full cut, lean
[2]Lean
[3]Based on Turkey for below freezing

10.1 Thaw Estimate Analysis Approach

There is no well-known estimation methodology for calculation of thaw times. Reverse application of the ASHRAE freeze-time calculation approach was not considered valid and was too complicated. A simplified approach to estimation of thaw times was adopted. The approach to modeling of the food thaw energy is illustrated in Figure 15 below. The focus for the calculation is the high-slope portion of the thaw curve just under 32°F temperature. Since actual frozen food delivery temperatures are not specified, and since this portion of the thaw curve represents the slowest part of the thaw[4], focus on this portion of the curve will result in good preliminary estimates of thaw times.

Additional assumptions and clarification to the approach is as follows.

- The food or food container is modeled as an equivalent sphere using ASHRAE Refrigeration 1998 p.9.13, Equation 71 for calculation of equivalent diameter.
- The container conductivity is modeled as being proportional to volume fraction of food (this is generally around 60%).
- Thaw times are estimated as a transient conduction problem with 26°F initial temperature, 45°F surrounding air temperature, a final core temperature of 32°F, and external heat transfer coefficient of 2 Btu/hr-ft^2-°F.

10.2 Thaw Estimate Results

Results of the thaw analysis for chicken are summarized in Table 11. Thawing of food in boxes will take significantly longer than the desired 18 hours. Separation of the food into

[4] This portion of the curve represents about ¾ of the thaw energy. Furthermore, heat input is significantly limited when the food is in this portion of the curve, since the temperature difference between the 45°F external air and the food is much lower than for the lower-temperature portion of the curve.

individual bags can dramatically reduce the thaw time. The second row in the table represents non-standard packaging of chicken in bags of about 8 inches diameter with food volume fraction equal to that of the standard boxes. The food could be delivered in boxes but placed in the thaw container in these bags. The results suggest that this approach will be necessary. The quickest thaw times would be for completely separated food pieces.

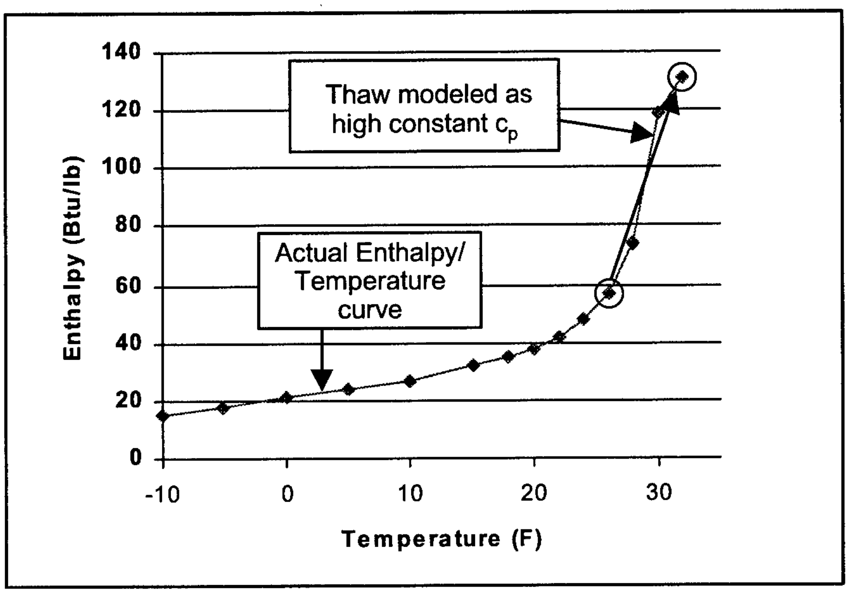


Figure 15. Thaw Energy Approximation

Table 11. Thaw Estimate Results

Package Configuration	Thaw Time (hrs)	Average Thaw Load (Btu/hr/lb)	Average Thaw Load (Btu/hr)[1]
Chicken in Box (13" x 9.8" x 8.8") Equivalent Diameter 11.3"	34	2.2	1,400
Chicken in Bags Avg 8" Diameter	20	3.7	2,200
Individual Chicken Pieces Equivalent Diameter 3.5"	9	8.2	5,000

[1]For a 40-cuft Compartment filled with specified packages (about 600 lb of chicken)

The table also shows the average thaw loads. Clearly, the thaw load more than makes up for heat leak of a representative storage container[5].

Adjustment of thaw times for different heat transfer coefficients, for instance for natural convection thawing, is straightforward: Thaw time will double if the heat transfer coefficient is reduced to 1Btu/hr-°F-sqft, which would be typical for natural convection. This means that food must be thawed in individual pieces if 18 hours thaw times are desired with a natural-convection thaw system.

It should be noted that these results are based on a theoretically sound but untested calculation methodology. Thaw times should be tested empirically to confirm the results.

10.3 Implications for Passive Thaw System Design

Some implications of the thaw time estimate results on thaw system design are as follows.

- Forced convection thawing is preferable.
- Natural convection is possible but would require thawing of food in individual pieces.
- Some form of heat input is required, perhaps through exchange with external air.
- Relying on natural convection for sufficient exchange with external air is dubious. This means that a completely passive thaw system will be not be feasible.

The assumed heat transfer coefficient of 2 Btu/hr-°F-sqft with 8"-diameter bags will require an air velocity of 140fpm. In a 40 ft^3 compartment with plan-view dimensions of 4 x 2 feet, the required air flow would be about 1,200cfm. Fan power may be as low as 100W, depending on design, assuming brushless DC fan motors. Air temperature drop through the thaw compartment would be about 2°F, which is acceptable (if this temperature drop is high, for instance for a design with less air flow, the thaw is slower for the "downstream" food).

Heat input with a forced convection system is conceptually straightforward. This would be done through exchange with ambient air using a damper system. At room temperature (70 °F), the heat input requirement might be 2,000 Btu/hr, which would require exchange of about 75 cfm. At 50°F temperature, 400 to 500cfm would be required.

In a natural convection thaw system, thaw heat addition will have to be evenly distributed through the thaw compartment. The natural convection air flow will be substantially lower than 1,200 cfm, which means that in the absence of heat addition within the compartment, the "downstream" food will not start to thaw until the "upstream" food is already thawed, thus further increasing thaw time.[6] A system with distributed heat input

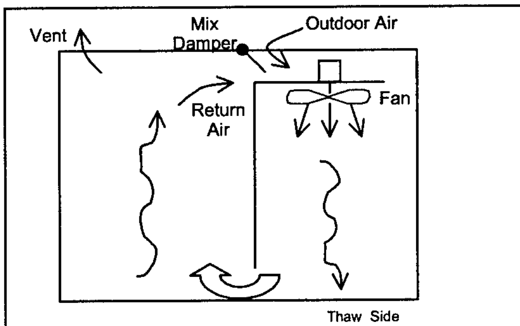
[5] Thermal load of the 80 cuft storage container used as a basis for the cold storage system analyses of Section 6 was estimated as 650Btu/hr.

[6] If this "wave" of thawing through the thaw compartment is acceptable, for instance if only half of the food will be needed after the first 18 hours, such an approach may be of interest.

must be designed carefully such that the heat input does not interfere with the natural convection air flow.

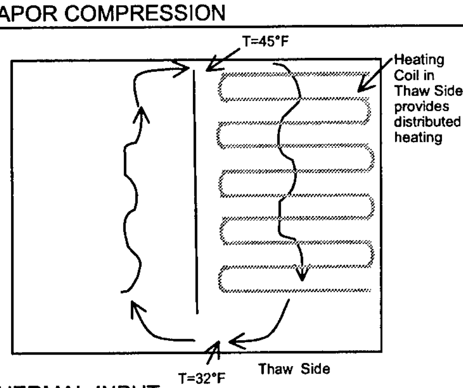
A stand-alone thaw system which requires no electricity or fuel input and operates over a wide ambient temperature range does not appear feasible. Heat input or a large flow of ambient air would be required to balance the thaw load at low ambient temperatures. Generating and controlling the required air flow with natural convection will be very difficult. A two-season design approach could be contemplated, in which a canvas-enclosed rack system with open top and bottom is used during cool seasons, and a system with a better housing and more sophisticated air flow control is used during warm seasons.

The most logical approach would be to incorporate a thaw compartment into one half of a cold-storage unit. Possible thaw system design approaches are illustrated in Figure 16 Development of a thaw system will require an iterative process of design and test to achieve satisfactory results.



VAPOR COMPRESSION

- Daily Electricity Requirement
  - Thaw Mode: ~2.4 kWh
  - All-Storage Mode: ~2.2 kWh
- Conversion to Fuel Usage depends on power management system (~0.25 gal for battery in all-storage mode)
- Added Technical Challenge:
  - Workable Ambient Damper System



THERMAL-INPUT

- Daily Fuel Requirement
  - Thaw Mode: ~0.5 gal
  - All-Storage Mode: ~0.3 gal
- Added Technical Challenge
  - Low-Input Heating System
  - Effective Heat Delivery for thaw function
  - Even thaw and compartment heat exchange balance using natural convection

Figure 16. Passive Thaw System Design Concepts

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11. Conclusions and Recommendations

- Parallel path development efforts are recommended for development of a cold storage temperature stabilization system.
  - The best near-term solution is vapor compression with thermal storage (for best efficiency) or with a battery system (to optimize weight and size)
  - The heat-input concepts may provide better performance than a vapor compression/thermal storage system and/or better reliability than a battery or dedicated generator system in future development, but they have added technical challenge. Heat input systems will also allow reduced reliability on the CK generator.
- Use of the Thermal Fluid system with a heat-input refrigeration system will require use of thermal storage to bridge the 14-hour night period, since the Thermal Fluid System does not operate at night. The only advantage such a system would have over a vapor compression system using thermal storage would be reduced reliance on the CK generator, since systems based on vapor compression would be more compact, lightweight, and efficient.
- The importance of the thaw function may drive the cold storage decision. The inclusion of thawing into a cold storage unit will strongly influence the design. For instance, if the vapor compression/forced convection thawing approach is used, the night strategy will be batteries or a dedicated generator rather than thermal storage, since the fan requires electricity during thaw mode.
- The low-input burner is an important enabling technology for many of the options. Development of such a burner should be pursued, since it makes a variety of the cold storage options possible:
  - The burner is used in the developmental AMTEC generator, which could be considered as a dedicated generator for a vapor compression based system.
  - Most of the heat-input system options require a low input burner to be viable.
- Development of a viable small low-cost field-deployable generator would make vapor compression the obvious system of choice. Because of the mature status of vapor compression technology, the development time for the CK cold storage system would be significantly reduced.

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APPENDICES

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Appendix A. Insulation Technologies

Appendix A. Insulation Technologies

Table A-1 shows insulating value and cost information for the most likely insulation candidates for the CK cold storage system. HCFC-blown foam insulation is used in all residential and commercial refrigerators and freezers. It provides a good insulating value at a low price.

The potential for significant improvement in insulation performance exists with vacuum panel technology, which is offered by a limited number of vendors. Although reported k-factors are significantly lower for vacuum insulation, these values do not fully take into consideration edge effects which can degrade performance significantly. Still, heat leak reductions in the range of 20 to 30 percent are typical. This comes at a significant cost increase, however. Currently vacuum panel insulation is not used in any major residential or commercial refrigerator or freezer product.

Table A-1. Insulating Value and Cost Information

	Effective K-Factor (Btu-in/hr-ft^2-F)	Projected Cost ($/sqft-in)	Technical Maturity	Comments
HCFC Blown Foam	0.125-0.15	$0.30	Industry Standard	• HCFC is to be phased out between 2003-2010 • Using HFC as a replacement will increase k factor 5-10%
Vacuum Insulated Panel (Glacier Bay)	0.05	~$2/ft^2*	Developed in 1993	• Insulation rating guaranteed for 20yrs • Glacier Bay manufactures flat panels only
VACPAC Panel (ICI)	0.05	$2.20-$2.40/ft^2		

*Cost = (perimeter x multiplier) + base price
Base price - $110-$130
Multiplier - 1.85-2.25

Appendix B. Battery Technologies

Appendix B. Battery Technologies

Four battery technologies were identified as possible candidates to provide small scale electric power suitable for the CK application. The size and weight shown in Table B-1 are normalized to a battery with an energy of 200 Watt-hours. This size is a reasonable approximation of the energy required to supply parasitic power to many of the options previously discussed. Battery size and weight can be scaled if different power requirement are desired.

Table B-1. Battery Technologies

	Size (ft^3/ 200 Wh)	Weight (lb/200 Wh)	Technical Maturity	Comments
Lead - acid	0.4	88	Well established Industrial Batteries available	• Low energy density • Need to avoid over-discharge • Low cost
Sealed Lead - acid	0.3	40	Well established Industrial Batteries available	• Maintenance-free • Can be used in any position • Need to avoid over-discharge • Low cycle life
Nickel Cadmium (NiCd)	0.3	20	Well established	• Long cycle life • Can withstand deep discharge • Disposal issues
Nickel Metal Hydride (NiMH)	0.2	20	Relatively new technology Mainly small scale	• Expensive • Low cycle life

Note: Size and weight calculated for 50% discharge.

The values in Table B-1 represent current technology capabilities. With the anticipation of government mandated electric vehicles many battery manufacturers and research programs are concentrating on bringing high energy density battery technologies to the market. New improvements, particularly in the emerging NiMH area could increase the energy density of the batteries and make battery technology a more attractive power option.

Size and weight values shown in Table B-1 do not include additional hardware for charging and regulating the battery system.

Appendix C. Power Technologies

Appendix C. Power Technologies

Several possibilities exist for small scale electric power generation during CK generator down time. Table C-1 shows selected candidates for the primary power input range of 300 to 500 Watts.

The use of fuel cell technology is currently limited by the need for hydrogen as fuel. The process of reforming hydrocarbon fuel to hydrogen is currently under development. The practical application of such a system is still years away.

A Sodium Cell generator in the size range of 500 to 1000 W is currently under development. This generator uses a multi-fuel burner for power input. This technology is approximately 3 to 5 years from practical application.

Small scale diesel powered generators have been developed and tested for military applications. Libby Corp. under the direction of PM-MEP developed prototype generators as small as 750 W. All small generator development was abandoned in favor of consolidation to a standard 3kW generator. The small generator development was continued in Europe.

Table C-1. Power Technologies

Electricity Generation	Power Output (W)[1]	Fuel	Efficiency Fuel→Power	Bulk Volume	Weight (lb)	Technical Maturity
Fuel Cell (PEM)	500W	Hydrogen	45-50%	10x10x10	20	2-3 yrs
Fuel Cell (PEM) w/Reformer	500W	Diesel	~15%	24x24x24	50	Fuel Reforming under development
Sodium Cell (AMTEC)	500W	Multi-fuel	16-24%	12" dia x 12"	20	3-5 years
Engine Driven Generator	500W	Diesel	~16%		33	Prototypes developed by PM-MEP and Libby. Both projects terminated.
	1500W	Diesel	~20		100	
	3kW	Diesel	20%	20x16x22	150	Prototype developed by PM-MEP
Stirling Engine w/ Linear Alternator	500 W	Multi-fuel	15-30%			

Appendix D. Calculation Analysis Approach

Appendix D. Calculation Analysis Approach

Figure D-1 illustrates the Cold Storage System Analysis Approach. The figure shows energy flows in the form of fuel, electricity, refrigeration, and thermal fluid heat. The figure shows a general system, with the full range of components which could be considered. Any given system will have only a few of these components. Each component has associated with it a conversion efficiency, as well as typical design parameters required to provide the desired capacity for the associated energy flows.

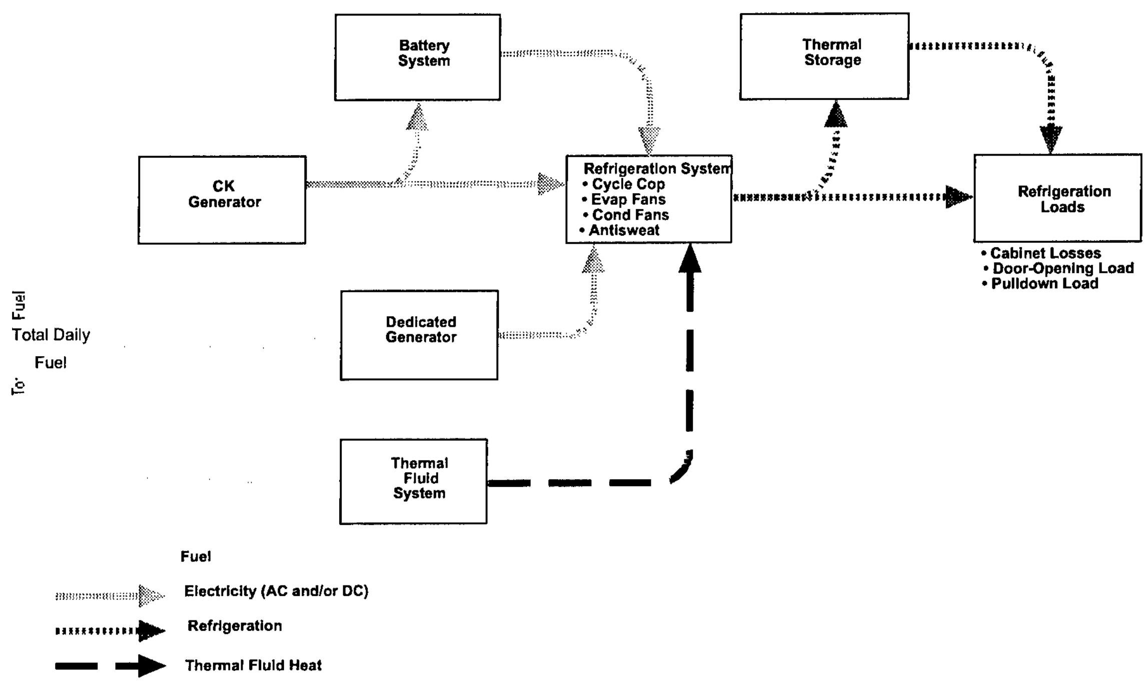

Figure D-1. Cold Storage System Analysis Methodology

Appendix E.   Additional Detail Regarding Thermal Storage Option C

Appendix E. Additional Detail Regarding Thermal Storage Option C

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Total number of pages (including cover): 2

Paul:

As mentioned on the phone yesterday, we are finishing up a study on Cold Storage Temperature Stabilization for the Natick, MA Army research lab. At our final presentation, Don Picard mentioned a system concept which he would like us to look at and discuss in the final report. FYI, the gist of the Containerized Kitchen food storage refrigerator requirements are (1) 80 cuft storage, (2) 10 hours per day use of 500W maximum electric power from the CK generator and/or use of the (still in development) hot oil "thermal fluid" system.

One of the concepts already under discussion was a system using thermal storage ice packages which would be frozen during the day with a vapor compression freezer. An alternative option, which would reduce CK generator load, would use a chemisorption freezer, similar to that described in your patent #5,161,389, "Appliance for Rapid Sorption Cooling and Freezing". We would greatly appreciate some preliminary estimates of size, weight, fuel use, and perhaps projected cost which we could use for initial comparisons of this concept to some of the other concepts covered in the study.

We have (optimistically) set the night load of the food storage refrigerator as 650Btu/hr. Assuming that each "set" of ice storage packages is good for 8+ hours under these conditions, the required ice weight is 36 lbs per "set". A "set" of packages will have to be refrozen in about 2 hours. A sketch of a suggested package arrangement is attached. I will plan to call back to clarify any questions you may have about this.

☒ **Please call to confirm:**

☐ **Call if there is a problem:**

Figure E-1. Memo Regarding Thermal Storage Option C

Appendix F. Detailed Calculations

Appendix F. Detailed Calculations

A detailed spreadsheet was created to aid in the analysis of the many system options. The analysis followed the methodology and assumptions previously described. Daytime and night heat loads along with system and energy conversion efficiencies for both refrigeration technologies and power generation options were incorporated. Thermal storage capabilities were included. The spreadsheet provided consistent analysis across a wide range of system options. The key results: system size, weight, and daily fuel usage are shown in the following pages.

System:	Vapor Compression with Battery	Vapor Compression with Engine Generator	A: Passive Thermal Storage Walls	B: DX Cold Wall with Integral Storage	C1: Slide-In Ice with External Vapor Compression Freezer	C2: Slide-In Ice with External Chemisorption Freezer
Refrigeration Technology	Vapor Compression	Vapor Compression	Vapor Compression	Vapor Compression	Vapor Compression	Chemisorption
Insulation	R22 Foam	R22 Foam	R22 Foam	R22 Foam	R22 Foam	R22 Foam
Thermal Storage	None	None	Thermasorb 43	Thermasorb 35	Ice	Ice
Night Power Management	Battery	Generator	None	None	None	CK generator
Cabinet						
Insulated Box						
Internal Volume (cuft)	80	80	80	80	80	80
External Volume (cuft)	102	102	102	102	102	102
Weight (lb)	460	460	460	460	460	460
Cooling Loads						
Steady Load (Btuh)	650	650	650	650	650	650
Daily Load Adder (Btu)	2000	2000	2000	2000	2000	2000
Storage						
Incorporated?	N	N	Y	Y	Y	Y
Medium Capacity (Btu/lb)			70	65	144	144
Medium Density (lb/cuft)			54	54	56	56
Storage Requirement (Btu)			9100	9100	9100	9100
Percent Medium Utilization			100%	100%	100%	100%
Size Multiplier			1.0	1.0	2.0	2.0
Net Stg Volume Adder (cuft)			2.4	2.6	2.6	2.6
Storage Medium Weight (lb)			130	140	144	144
Weight adder to incorp pcm(lb)			15	5	15	15
Refrigeration System						
Condensing Unit						
Weight (lb)	84	84	84	84	175	
Volume (cuft)	7	7	7	7	18	
Evaporator						
Weight (lb)	10	10	10	32		
Volume (cuft)	0.8	0.8	0.8	0.8		
Other						
Weight (lb)	2	2	9	10		
Volume (cuft)	1	1	1	1		
Total Refrigeration						
Weight (lb)	96	96	110	120	175	100
Volume (cuft)	8.8	8.8	8.8	8.8	18	18
Design/Calculation Performance						
Capacity (Btuh)	3220	3220	3220	3220	4000	2592
Power Input (W)	257	257	257	257	597	50
Heat Input (Btuh)	0	0	0	0	0	8640
Cycle COP	3.67	3.67	3.67	3.67	1.96	0.30
"Evap" Fan Input (W)	20	20	20	0	10	0
Evap Fan 100% Run?	n	N	N	N	N	N
"Cond" Fan Input (W)	20	20	20	20	20	0
Antisweat Power (W)	0	0	0	0	0	0
Pump Power (W)						
Net Performance						
Net Operating Capacity	3,152	3,152	3,152	3,220	3,966	2,592
Net Operating Power Input (W)	297	297	297	277	627	50
Additional Steady Power (W)	0	0	0	0	0	0
Net Operating Heat Input (Btuh)	0	0	0	0	0	8640
Loads						
Average Day System Load (Btuh)	850	850	1,760	1,760	1,810	1,810
Night System Load (Btuh)	650	650	-	-	-	-
Energy Requirements						
Avg Daytime Power (W)	80.2	80.2	166.1	151.7	286.2	34.9
Avg Daytime Heat (Btuh)	-	-	-	-	-	6,033
Avg Night Power (W)	61.3	61.3	-	-	-	-
Avg Night Power (Wh)	858.8					
Avg Night Heat (Btuh)	-	-	-	-	-	-
Refrigerant Storage						
Incorporated?						
Energy storage (Btu)						
Storage COP						
Heat Input for Storage (Btu)						

System:	Vapor Compression with Battery	Vapor Compression with Engine Generator	A: Passive Thermal Storage Walls	B: DX Cold Wall with Integral Storage	C1: Slide-In Ice with External Vapor Compression Freezer	C2: Slide-In Ice with External Chemisorption Freezer
Power Management						
Daytime: CK Generator						
Efficiency (%)	23	23	23	23	23	23
Fuel Rate (Btu/kWh)	14,839	14,839	14,839	14,839	14,839	14,839
Night: Generator						
Size (cuft)		4				0
Weight (lb)		100				0
Efficiency (%)		20				0
Fuel Rate (Btu/kWh)		17,065				
Free Heat Input (Btuh)						
Fuel Heat Input (Btuh)						
Night: Battery						
Size (cuft)	1.0					
Weight (lb)	86					
Conversion Efficiency	50%					
Low Input Burner						
Size (cuft)						
Weight (lb)						
Efficiency for Refrig Heat (%)						65
Efficiency for TE Gen Heat (%)						
Daily Energy Use						
CK Generator Electric (kWh)	2.52	0.80	1.66	1.52	2.86	0.35
CK Generator Fuel (gal)	0.27	0.09	0.18	0.16	0.30	0.04
Dedicated Generator Elec (kWh)	-	0.86	-	-	-	-
Dedicated Generator Fuel (gal)	-	0.10	-	-	-	-
Refrigeration System Fuel (gal)	-	-	-	-	-	0.66
TOTAL DAILY FUEL (gal)	0.27	0.19	0.18	0.16	0.30	0.70
Comments	Night Energy of 0.09 gal fuel is doubled	Slightly worse night generator efficiency	Essentially the same efficiency as Vapor compression on CK Generator	Less energy due to no evaporator fan	More losses and lower evap temp increase energy	Lower COP for this system than for Chem with low input burner. Added losses.
Net Sizes						
TOTAL WEIGHT (lb)	642	656	715	725	794	719
TOTAL VOLUME (cuft)	112	115	113	113	123	123
## TOTAL WEIGHT ADDED(lb)	182	196	255	265	334	259
TOTAL VOLUME ADDED (cuft)	9.8	12.8		11.4	20.6	20.6

System:	D: Glycol Cold Wall with Integral Storage	E: Remote Storage with Glycol Cold Wall	Chemisorption with Low Input Burner	H2 Assist Absorption with Pump and Fan Assist	Zeolite with Low Input Burner	Zeolite with Energy Storage
Refrigeration Technology	Vapor Compression	Vapor Compression	Chemisorption	Absorption	Zeolite Absorption	Zeolite Absorption
Insulation	R22 Foam	R22 Foam	R22 Foam	R22 Foam	R22 Foam	R22 Foam
Thermal Storage	Thermasorb 35	Ice	None	None	None	water vapor
Night Power Management	None	None	Thermoelectric Gen.	Thermoelectric Gen.	Thermoelectric Gen.	Thermoelectric Gen.
Cabinet						
Insulated Box						
Internal Volume (cuft)	80	80	80	80	80	80
External Volume (cuft)	102	102	102	102	102	102
Weight (lb)	460	460	460	460	460	460
Cooling Loads						
Steady Load (Btuh)	650	650	650	650	650	650
Daily Load Adder (Btu)	2000	2000	2000	2000	2000	2000
Storage						
Incorporated?	Y	Y	N	N	N	N
Medium Capacity (Btu/lb)	65	144				1000
Medium Density (lb/cuft)	54	56				56
Storage Requirement (Btu)	9100	9072				9100
Percent Medium Utilization	100%	100%				100%
Size Multiplier	1.0	1.0				1.0
Net Stg Volume Adder (cuft)	2.6	1.1				
Storage Medium Weight (lb)	140	63				9.1
Weight adder to incorp pcm(lb)	5					
Refrigeration System						
Condensing Unit						
Weight (lb)	110	120	20	200	20	20
Volume (cuft)	14	8	10.5	16.2	1.5	1.5
Evaporator						
Weight (lb)	20	35				
Volume (cuft)	2.6	1.5	0.8			
Other						
Weight (lb)						180
Volume (cuft)						5.0
Total Refrigeration						
Weight (lb)	130	157	100	150	100	200
Volume (cuft)	16.6	9.5	14.7	16.2	14.7	6.5
Design/Calculation Performance						
Capacity (Btuh)	4200	4200	2000	2500	2000	2000
Power Input (W)	647	647	50	25	20	20
Heat Input (Btuh)	0	0	2857	6944	3333	3333
Cycle COP	1.96	1.96	0.70	0.36	0.60	0.60
"Evap" Fan Input (W)	0	0	0	0	0	0
Evap Fan 100% Run?	N	N	N	N	N	N
"Cond" Fan Input (W)	20	20	0	0	0	0
Antisweat Power (W)	0	0	0	0	0	0
Pump Power (W)	20	20				
Net Performance						
Net Operating Capacity	4,149	4,149	2,000	2,500	2,000	2,000
Net Operating Power Input (W)	687	687	50	25	20	20
Additional Steady Power (W)	0	0	0	0	0	0
Net Operating Heat Input (Btuh)	0	0	2857	6944	3333	3333
Loads						
Average Day System Load (Btuh)	1,781	1,779	850	850	850	850
Night System Load (Btuh)	-	-	650	650	650	650
Energy Requirements						
Avg Daytime Power (W)	295.0	294.5	21.3	8.5	8.5	8.5
Avg Daytime Heat (Btuh)	-	-	1,214	2,361	1,417	1,417
Avg Night Power (W)	-	16.2	16.3	6.5	6.5	-
Avg Night Power (Wh)		227				
Avg Night Heat (Btuh)	-	-	929	1,805	1,083	-
Refrigerant Storage						
Incorporated?			N		N	Y
Energy storage (Btu)						9,100
Storage COP						0.3
Heat Input for Storage (Btu)						30,333

System:	D: Glycol Cold Wall with Integral Storage	E: Remote Storage with Glycol Cold Wall	Chemisorption with Low Input Burner	H2 Assist Absorption with Pump and Fan Assist	Zeolite with Low Input Burner	Zeolite with Energy Storage
Power Management						
Daytime: CK Generator						
Efficiency (%)	23	23	23	23	23	23
Fuel Rate (Btu/kWh)	14,839	14,839	14,839	14,839	14,839	14,839
Night: Generator						
Size (cuft)			1	1	1	
Weight (lb)			20	20	20	
Efficiency (%)			3	3	3	
Fuel Rate (Btu/kWh)			113,767	113,767	113,767	
Free Heat Input (Btuh)			143	278	167	
Fuel Heat Input (Btuh)			1705	462	573	
Night: Battery						
Size (cuft)		1.0				
Weight (lb)		23				
Conversion Efficiency		50%				
Low Input Burner						
Size (cuft)			1	1	1	1
Weight (lb)			10	10	10	10
Efficiency for Refrig Heat (%)			65	65	65	65
Efficiency for TE Gen Heat (%)			75	75	75	
Daily Energy Use						
CK Generator Electric (kWh)	2.95	3.40	-	-	-	-
CK Generator Fuel (gal)	0.31	0.36	-	-	-	-
Dedicated Generator Elec (kWh)	-	-	0.23	0.09	0.09	-
Dedicated Generator Fuel (gal)	-	-	0.23	0.06	0.08	-
Refrigeration System Fuel (gal)	-	-	0.28	0.54	0.32	0.49
TOTAL DAILY FUEL (gal)	0.31	0.36	0.50	0.60	0.40	0.49
Comments	Compared with C, less loss but more power (pump)	Compared with D, night pumping with battery results in greater energy	Less efficient than vapor compression, 0.7 COP may be optimistic, 50W power through TE Gen makes fuel use high	Assumed less efficient than Chemisorption but also less power	Zeolite refrigeration reported to be more efficient than chem- or absorption	Less efficient for water storage cycle
Net Sizes						
TOTAL WEIGHT (lb)	735	703	590	640	590	679
TOTAL VOLUME (cuft)	113	114	118	120	118	109
## TOTAL WEIGHT ADDED(lb)	275	243	130	180	130	219
TOTAL VOLUME ADDED (cuft)	10.5	11.6	16.5	18.0	16.5	7.3