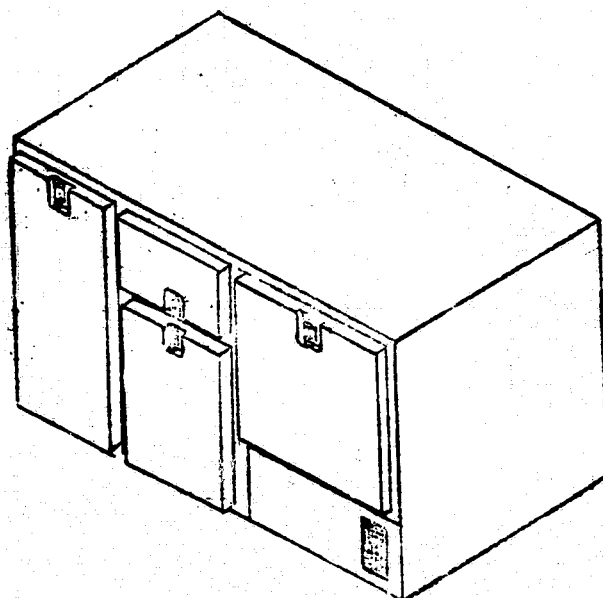


144445

SHUTTLE KIT FREEZER REFRIGERATION UNIT CONCEPTUAL DESIGN

NAS 9-9912



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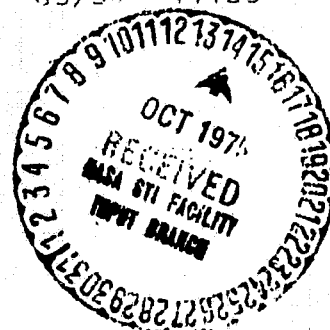
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R. J. COPELAND

REPORT NO. T122-RP-044

22 AUGUST 1975



VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORPORATION

P O BOX 5907 • DALLAS, TEXAS 75222

DEVELOPMENT OF A REFRIGERATION
SYSTEM FOR LUNAR SURFACE
AND SPACECRAFT APPLICATIONS

Contract No. NAS9-9912
Report No. T122-RP-044

22 August 1975

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CONCEPTUAL DESIGN

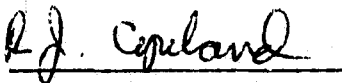
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VOUGHT SYSTEMS DIVISION
LTV Aerospace Corporation
Dallas, Texas

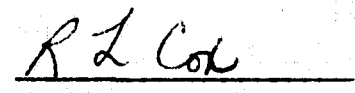
To

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Johnson Space Center
Houston, Texas

PREPARED BY:


R. J. Copeland

APPROVED BY:


R. L. Cox
ECS Group



VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORPORATION

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FOREWORD

This report presents the Final Report of a study of a Shuttle Orbiter Freezer Kit. The study was conducted jointly by LTV Aerospace Corporation - Vought Systems Division (VSD) and the Boeing Company, Houston, Texas Office. Throughout both studies were directed by Mr. Jim Jaax of NASA-JSC.

VSD's effort was conducted under Contract NAS9-9912 and was one of several tasks in that contract. Dr. R. J. Copeland was the Principal Investigator and Mr. Wayne Higgins was responsible for the design of the refrigeration unit.

Several briefings and co-ordination meetings were conducted during this effort. This report documents VSD's data and incorporates the primary data of all previous documentation. Boeing's work is briefly reviewed for purposes of continuity; the reader is referred to Boeing's Final Report for additional data. That report is:

Proctor, Wayne, and Russel, D. J., "Shuttle Freezer Conceptual Design" Boeing Document D2-118569, Contract NAS9-13965, dated 22 Aug. 1975.

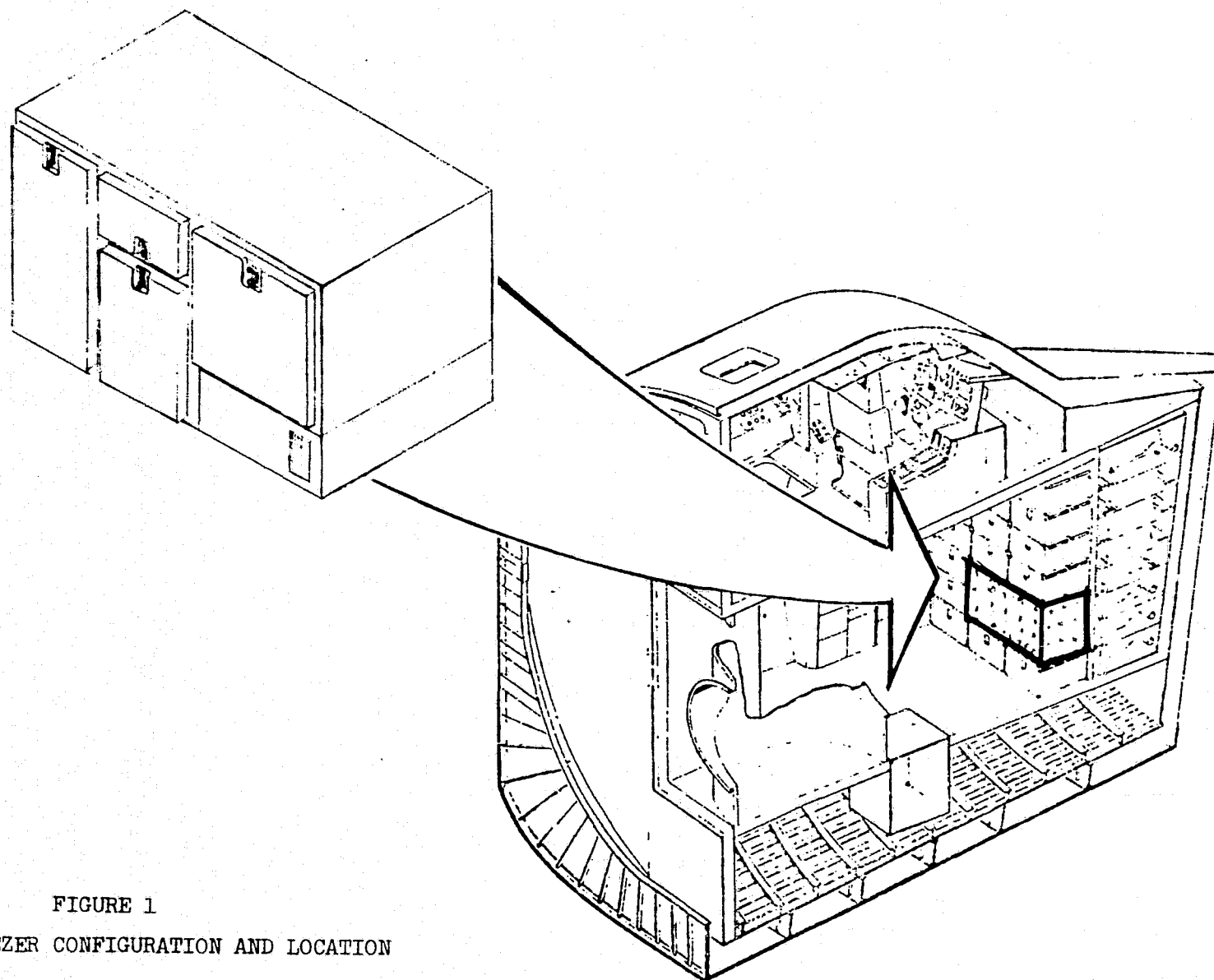
1.0 SUMMARY

This report presents the results of a conceptual design study of a refrigerated food/medical sample storage compartment as a kit to the Space Shuttle Orbiter. To maintain the -10°F in the freezer kit, an active refrigeration unit is required. Trades were conducted and an air cooled Stirling Cycle was selected. A conceptual design study verified the feasibility of the freezer kit to be built within the current shuttle capabilities for payloads.

The freezer kit contains two subsystems, the refrigeration unit (R/U) and the Storage Volume. The R/U was studied by LTV Aerospace Corporation Vought Systems Division (VSD) in parallel with a coordinated effort by the Boeing Company, Houston, Texas, who were responsible for the Storage Volume.

The freezer must provide two basic capabilities in one unit. One requirement is to store 215 lbs of food which is consumed in a 30-day period by 7 people. The other requirement is to store 128.3 lbs of medical samples consisting of both urine and feces which are collected over the same 30-day period. For health reasons the food and samples must always be separated from each other and available to the crew throughout the mission. Figure 1 presents a conceptual sketch of the freezer and its location. The unit will be mounted on the lower deck of the Shuttle cabin. The unit will occupy four standard payload module compartments on the forward bulkhead. The freezer contains four storage compartments of which the smallest is launched empty. That volume is used to store the samples for the first few days, until one of the food compartments has been emptied and is available for sample storage. When that volume is filled with samples, a second food compartment has been emptied and is then used for sample storage for the remainder of the mission. All compartments are nominally maintained at -10°F by the air-cooled Stirling Cycle refrigerator. Figure 2 presents the characteristics of the freezer determined by the conceptual design effort.

Figure 3 presents the overall system schematic. Since the mechanics of the Stirling Cycle prohibit a distributed cold region, heat is transferred to the Stirling Cycle from the Coolanol 15 loop in the heat exchanger. The Stirling Cycle is designed for peak heat load capacity; to

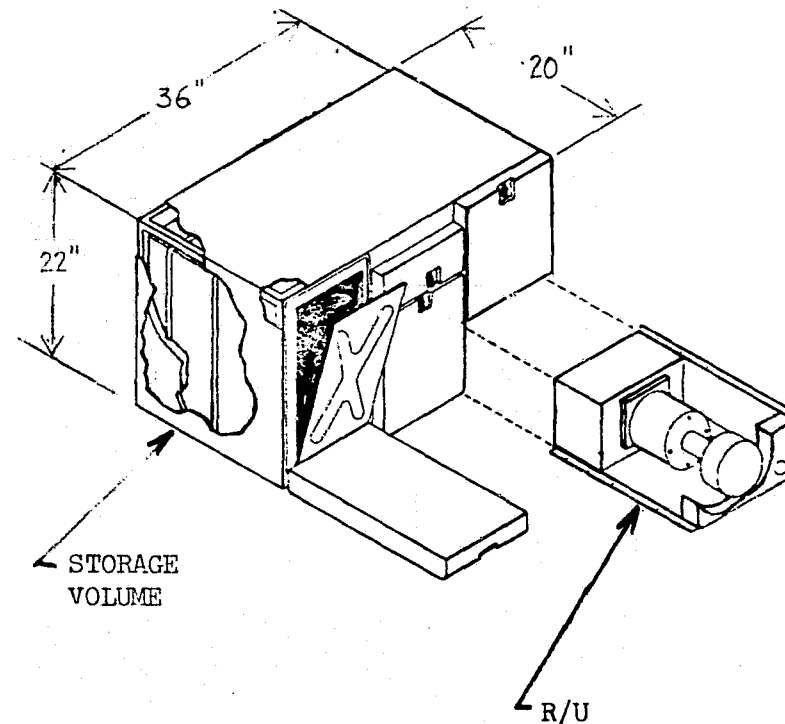


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FIGURE 1
PROPOSED FREEZER CONFIGURATION AND LOCATION

○ TOTAL WEIGHT (WITHOUT STORED ITEMS)	65 LBS
○ COMBINATION FREEZER/FOOD WEIGHT	280 LBS
○ MAXIMUM ELECTRICAL POWER REQUIREMENTS	
. 200 VAC 3φ	206 WATTS
. 28 V DC UNREGULATED	<u>6 WATTS</u>
	212 WATTS
○ HEAT REJECTED TO CABIN ATMOSPHERE	
. MAXIMUM DIRECT ADDITION	282 WATTS
. HEAT LEAK FROM AIR INTO STORAGE VOLUME	<u>44</u>
. NET HEAT ADDITION	238 WATTS
○ TOTAL STORAGE VOLUME	4.6 FT ³
○ EXTERNAL VOLUME	9.2 FT ³
(FOUR STANDARD STORAGE MODULES)	
○ REFRIGERATION SYSTEM LIFE	8000 HOURS
○ MAINTENANCE INTERVAL (SCHEDULED)	2000 HOURS

FIGURE 2
CONCEPTUAL FREEZER CHARACTERISTICS



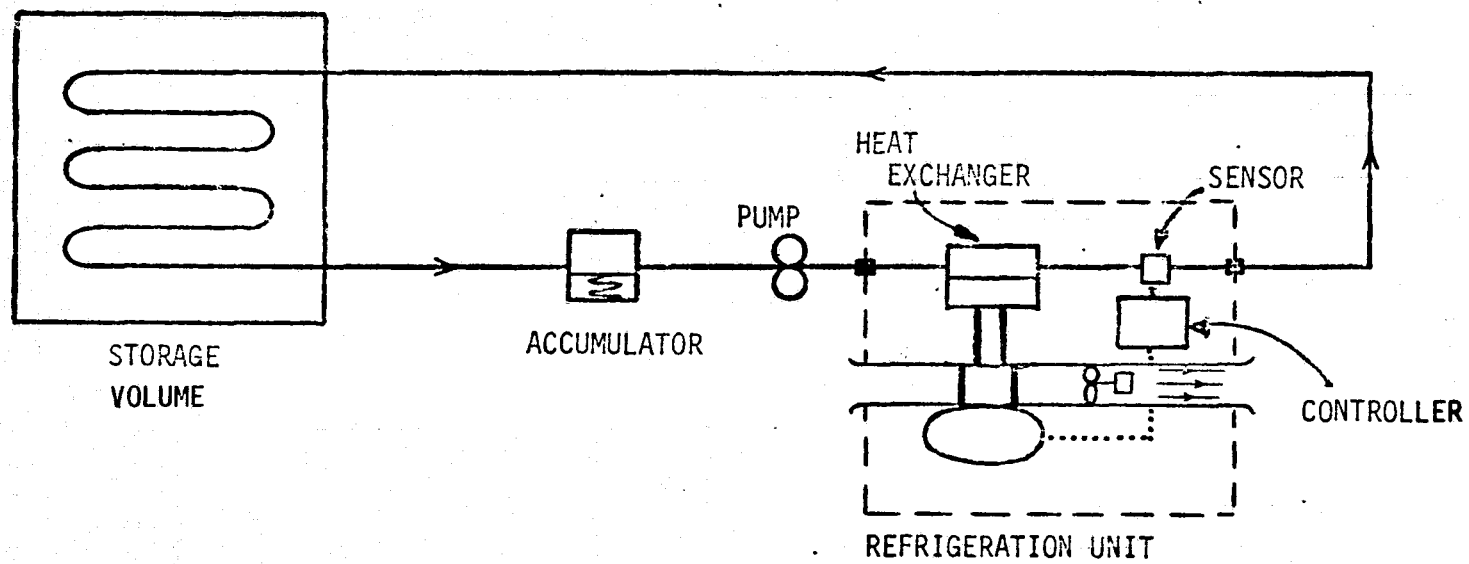


FIGURE 3

REFRIGERATION SYSTEM FEATURES

prevent over-cooling the food, an electric control system is employed. The sensor provides the signal and simple "on/off" logic is used to maintain $-10^{\circ}\text{F} \pm 10^{\circ}\text{F}$. Only the Stirling Cycle and fan are automatically controlled; the Coolanol pump runs continuously except when the entire freezer is shut down via a manual switch.

Table 1 presents the performance expected of a flight freezer based on the developing state-of-the-art in Stirling Cycle hardware. The cooling capacity requirements were set by Boeing for the Storage Volume design; the other factors were generated by VSD for the R/U. The Coolanol temperatures were selected by mutual agreement. The band width is established by the control system. The temperature level can be altered without changing the logic; a simple resistor change will be sufficient for the range -20°F to -14°F .

The air-cooled Stirling Cycle was selected based on a trade-off of the options. From a preliminary statement of freezer requirements, a set of guidelines and constraints were established. Three heat sinks were considered: a water loop supplied to the freezer at 45°F , a water loop at 80°F , and the cabin air at a maximum of 80°F . Eight different types of refrigerators which have previously been used or are in development for spacecraft refrigeration were screened. From Stirling Cycle, Vuilleumier, Reverse Brayton, Vapor Compression (V-C), Absorption or Adsorption, Thermoelectrics (T/E), expendables and directional space radiators, three were selected for detailed evaluation. Weight, volume, power and state-of-the-art were evaluated for the selected Stirling Cycle, T/E and V-C. Parametric data were generated as a function of cooling load for each refrigerator, allowing the insulation thickness to vary. An air heat sink was selected to minimize the expenses and vehicle scar associated with the complex interfaces with the water loop.

The Stirling Cycle has been previously used in space, but has had limited life. Only recently have long life space-applicable designs been developed for cryogenic systems. No space-applicable ('o' g) units have been built in the temperature range of interest. Because of these factors, a technical risk exists with the Stirling Cycle. Other important concerns are power consumption, life and cost of the unit. It is recommended that a prototype of the freezer be fabricated to eliminate the technical risks.

TABLE 1

REFRIGERATION SYSTEM PERFORMANCE

° MAXIMUM COOLING CAPACITY	75 WATTS (256 BTU/HR)
° AVERAGE THERMAL LEAKAGE	44 WATTS (150 BTU/HR)
° R/U BASELINE DUTY CYCLE	69%
° STIRLING COEFFICIENT OF PERFORMANCE (C.O.P.)	.427
° R/U POWER REQUIREMENTS (MAXIMUM)	207 WATTS (INCLUDING FAN AND CONTROLLER)
° COOLANOL PUMP POWER	5 WATTS (APPROXIMATELY)
° COOLANT FLOW RATE (TOTAL)	200 LB/HR, COOLANOL 15
° COOLANT ΔT	< 3°F
° COOLANOL TEMPERATURE CONTROL BAND WITH TOLERANCES	-6° TO -12° AS -8°F $\pm$ 2°F "ON" -10°F $\pm$ 2°F "OFF"

2.0

INTRODUCTION

A freezer system will be needed for food and sample storage on the Space Shuttle Orbiter. The freezer will be needed to preserve these items in a cold environment for periods up to 30 days. A study was initiated by NASA-JSC to determine the type of refrigeration system to be used and the characteristics of that system for the intended application. The study was jointly conducted by VSD and the Boeing Company but monitored closely by NASA. The overall objectives of this study are to evaluate the feasibility of the freezer kit, to delineate its characteristics, and to identify problem areas requiring development. Principle concerns are the type of active refrigerator, heat sink, power requirement, and the location of the freezer in the Shuttle. VSD's responsibility was to provide data on and to generate a conceptual design of the active cooling element, the Refrigerator Unit (R/U). Boeing's responsibility was to select the location of the freezer, optimize the system and generate a conceptual design of the Storage Volume which contains the food and samples.

The freezer will be used to store food and medical samples consisting of urine and feces. The freezer will be located on the lower deck of the Shuttle cabin and will isolate the food from the samples. Currently no provisions for a freezer are baselined for the Shuttle ; the Shuttle's capability for payload kits will be employed for the freezer. The freezer kit will be mated to the existing Environmental Control System; no permanent modifications to the Shuttle (i.e., vehicle scar) are allowed after removal of the kit.

Program Plan

Figure 4 presents the program plan and the division of responsibilities. The end products of the study were conceptual designs of both the R/U and the Storage Volume, and final reports on each area. The study started with a general definition of requirements. These data were evaluated and a preliminary set of thermal requirements were defined. For the specified heat load and temperature, VSD investigated the various methods of providing cooling. In the evaluation of cooling types, candidate refrigerators were screened and detail trade-off data on the most promising systems were generated. The results were presented in a briefing on 13 February 1975 for three potential heat sinks to the Shuttle. Boeing then optimized the storage volume for each promising cooler and conducted a trade-off of coolers and heat sinks. NASA decided upon an

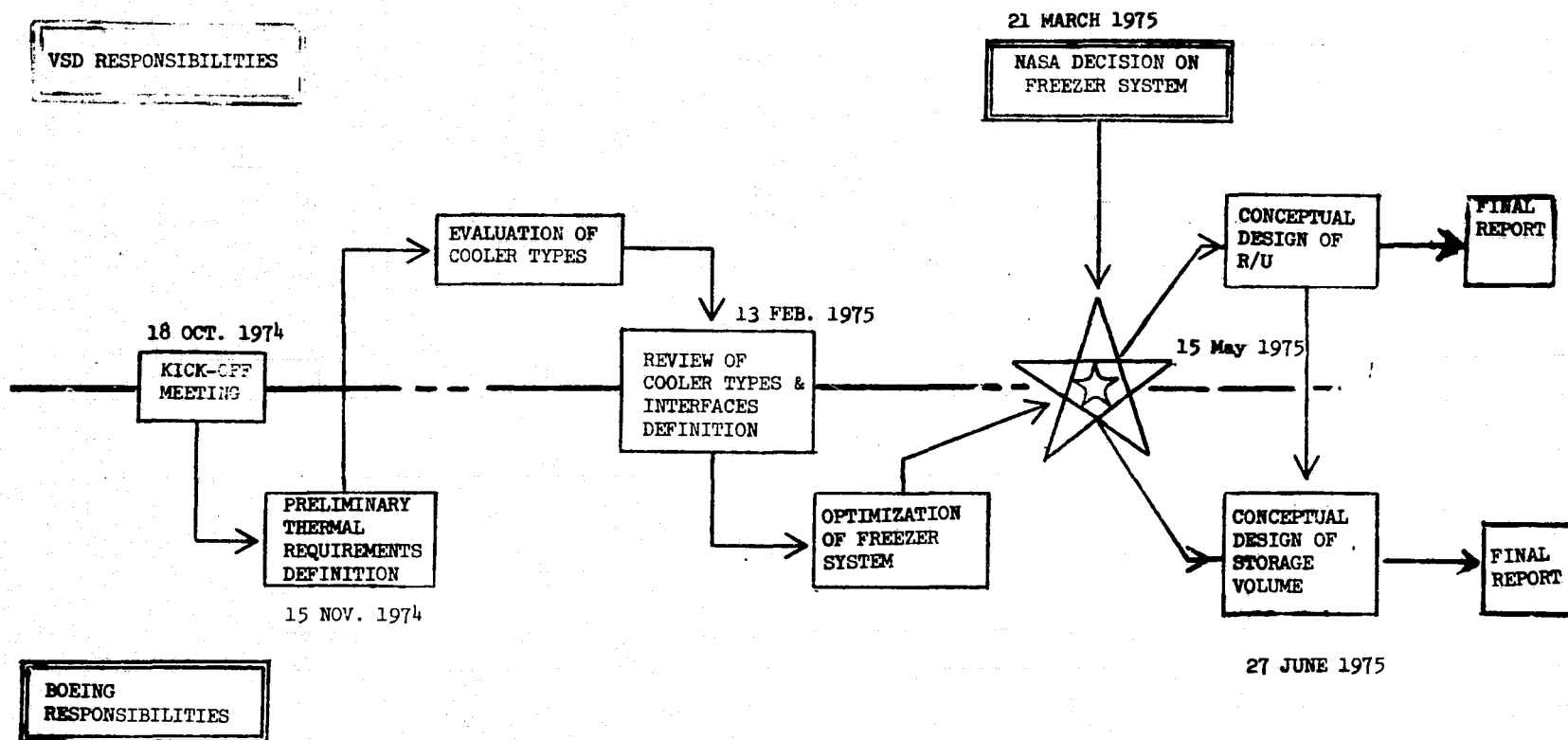


FIGURE 4
PROGRAM PLAN

air-cooled Stirling Cycle refrigerator with a Coolanol 15 pumped liquid system to transfer heat from the storage volume to the cooler. Boeing and VSD then prepared a coordinated conceptual design of a freezer kit. Figure 5 presents the responsibilities of both companies for that design.

This report documents VSD's portion of the study. Section 3.0 presents the requirements of the freezer and the conceptual design of the storage volume. Section 4.0 presents VSD's screening of the candidate coolers. Section 5.0 presents the conceptual design of the Stirling Cycle refrigerator unit. Section 6.0 discusses the problem areas which require development effort. Section 7.0 presents the conclusions and recommendations of this study.

Boeing (see figure below)

- o Definition of freezer and freezer component specifications
- o Storage enclosure design
- o Mounting fixture design
- o Coolant distribution system design (including temperature control)
- o Integration of freezer installation, refrigeration system, and storage enclosure

LTV (see figure below)

- o Refrigeration Unit design
- o Refrigeration Unit cooling system design
- o Coolant to refrigeration unit heat exchanger design

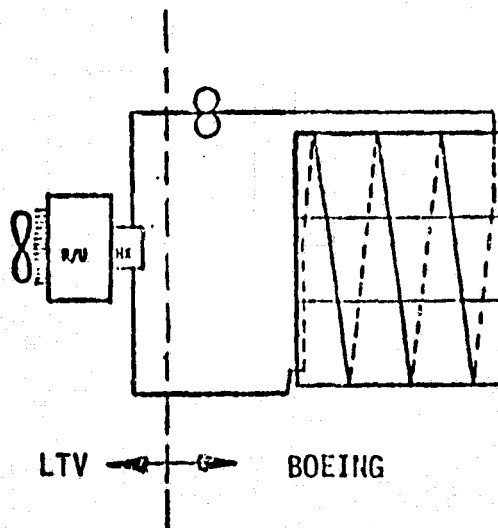


FIGURE 5
SHUTTLE KIT FREEZER EQUIPMENT DESIGN RESPONSIBILITY

The freezer kit must be designed to meet the provisions of the Shuttle while providing a cold environment for the food and samples. Section 3.1 presents the detailed requirements and guidelines and constraints used in this study. Obviously the R/U subsystem must be designed to be compatible with the storage volume. Boeing's conceptual design is described in Section 3.2.

3.1

Overall Freezer Design Requirements

The general requirements for the freezer were specified by NASA-JSC as follows:

- . Maintain storage thermal environment of -10°F , nominal (0° to -20°F limits)
- . Store 215 lbs. frozen food (4 ft^3)
- . Store accumulated 128.3 lbs. of medical samples (2.63 ft^3)
- . Isolate medical sample storage from food storage
- . Utilize standard crew storage module dimensions and mounting fixtures
- . Portable, self-contained system with no hardware buildup in spacecraft
- . Safe, no hazards to the crew due to the freezer

Employing these requirements, Boeing generated a preliminary set of requirements, Reference 1. The R/U cooling capacity was estimated as 100 watts, at -10°F .

VSD generated a set of guidelines and constraints to evaluate the candidate coolers; the data are presented on Table 2. Three heat sinks were considered. Two sinks are in the water loop of the Shuttle cabin at 45°F and 80°F ; the third was the cabin air at a maximum of 80°F . The maximum temperature of the cooler was assumed to be 55°F , 90°F and 100°F respectively for each sink. Figure 6 illustrates the rationale for those selections employing the air as an example.

All types of power were assumed to be available to the R/U. The Shuttle fuel cells generate unregulated D.C. power. By the addition of power conditioners, that power may be converted into regulated D.C. or one of a number of types of A.C. Due to inefficiency in any device, the penalty for each type of power is not equal; the assumed penalties are

Table 2
GUIDELINES & CONSTRAINTS

MISSION

- UP TO 30-DAY DURATION

LAUNCHES; CONSIDER 3 CASES:

- A) 50 FLIGHTS
- B) 100 FLIGHTS
- C) ALL FLIGHTS

VEHICLE INTERFACES

- MINIMIZE IMPACT CONSISTENT WITH OVERALL TRADE STUDY RESULTS

REFRIGERATOR OPERATIONAL REQUIREMENTS

- NO ORIENTATION CONSTRAINTS
- PROVIDE COOLING IN ALL MISSION PHASES

ENVIRONMENTAL CONTROL/LIFE SUPPORT SYSTEM

- MINIMUM WATER LOOP TEMPERATURE OF 45°F, MAXIMUM OF 80°F

NOMINAL CABIN AIR 75°F, MAXIMUM OF 80°F

ELECTRICAL POWER FROM ORBITER

- 2 KW MIN, 7 KW CONTINUOUS, 12 KW PEAK
- 12 KW PEAK, 15 MIN DURATION ONCE EVERY 3 HRS.
- ADD ON KITS: 840 KWH EACH

926 LBS EXPENDABLES
+654 LBS TANKAGE

1580 LBS TOTAL

- CONDITIONED VOLTAGE:

- 28 VDC (± 4 VDC BASELINE, $\pm 2\%$ VDC ADD-ON)
- 115 VAC, 400 HZ, 3- ϕ
- 115/200 VAC, 400 HZ, 1- ϕ
- 115 VAC, 60 HZ, 1- ϕ (ADD-ON)
- 200 VAC, 50 HZ, 1- ϕ (ADD-ON)

PENALTIES

- WEIGHT = \$250/LB* (FLT AS PAYLOAD)
- VOLUME = \$300/FT³ @ 1.2 LB STRUCTURE/FT³)
- POWER =

	ORBITER SYSTEM	
UNREGULATED DC	1.1 LB/KWH	} EXPENDABLES ONLY
REGULATED DC	1.3 LB/KWH	
REGULATED AC	1.6 LB/KWH	

SAFETY

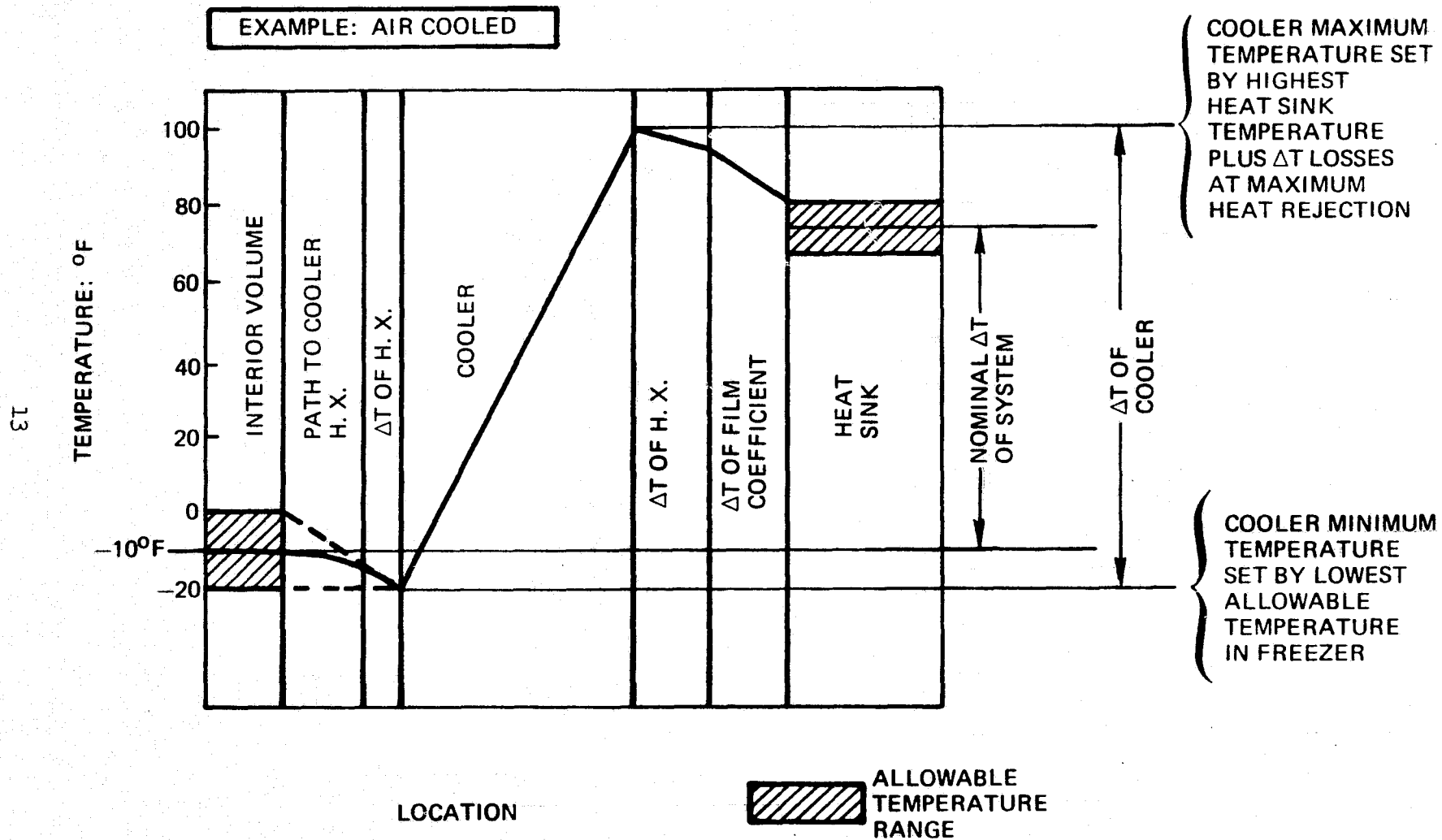
NO CREDIBLE FAILURE IN THE REFRIGERATION SYSTEM SHALL JEOPARDIZE THE ENVIRONMENTAL CONTROL SYSTEM

REDUNDANCY

REDUNDANT COOLING SYSTEMS ARE NOT REQUIRED FOR PAYLOADS, CONSISTENT WITH FAIL SAFE REQUIREMENTS

*NOMINAL. \$160/LB TO \$420

AS \$10.5 MILLION/LAUNCH
65,000 TO 25,000 LBS.
SHUTTLE CAPABILITY.



RELATIONSHIP OF COOLER AND HEAT SINK TEMPERATURES

FIGURE 6

presented in Table 2 for expendables only. If the power requirement is sufficiently large, another set of H_2 and O_2 kit tanks will be required; then an additional payload penalty of 654 lbs must be charged for those tanks which otherwise would not be required. For relatively small requirements (on the order of 10% of a full kit or 84 kwh) excess capacity in fuel cell tanks can be expected on most flights. In that case only the expendables penalty is assessed.

VSD employed the above data with the 100 watt peak cooling capacity to screen candidate coolers. Based on VSD's recommendations of promising coolers (Ref. 2), Boeing optimized the freezer for each type of R/U with the three heat sinks. A trade-off was then conducted, and the Stirling Cycle with heat rejection to the cabin air was selected. Reference 3 describes those trade-offs, and the interfaces of the R/U and storage volume prior to conceptual design. The required cooling rate was optimized as 75 watts at $-10^\circ F$ nominal from the Stirling Cycle to a Coolanol 15 loop.

3.2 Storage Volume Conceptual Design

This section summarizes the conceptual design of the storage volume produced by Boeing's study. The refrigeration unit interfaces with the storage volume are a pumped liquid Coolanol 15 loop, structural mounting, and power supply. The storage volume produces a thermal load of 75 watts which is removed by the R/U. Prior to installation of the freezer kit in the Shuttle, the R/U is pre-assembled to the storage volume; the freezer assembly is then mounted in the Shuttle via structural members on the storage volume. Power is supplied to the R/U via electrical connectors. As of this writing, electrical connections from the Shuttle to the freezer have not been defined.

Figure 7 presents the conceptual design of the storage volume. That subsystem consists of four insulated compartments, a Coolanol 15 loop, and supporting structure. Table 3 summarizes the key features which are described in References 4 and 5.

The liquid loop transfers heat from the compartments to a heat exchanger on the Stirling Cycle unit. The heat loads from leakage through the insulation, door openings, and freezing of samples is picked up by the Coolanol 15. A system of parallel tubes on the inner wall which

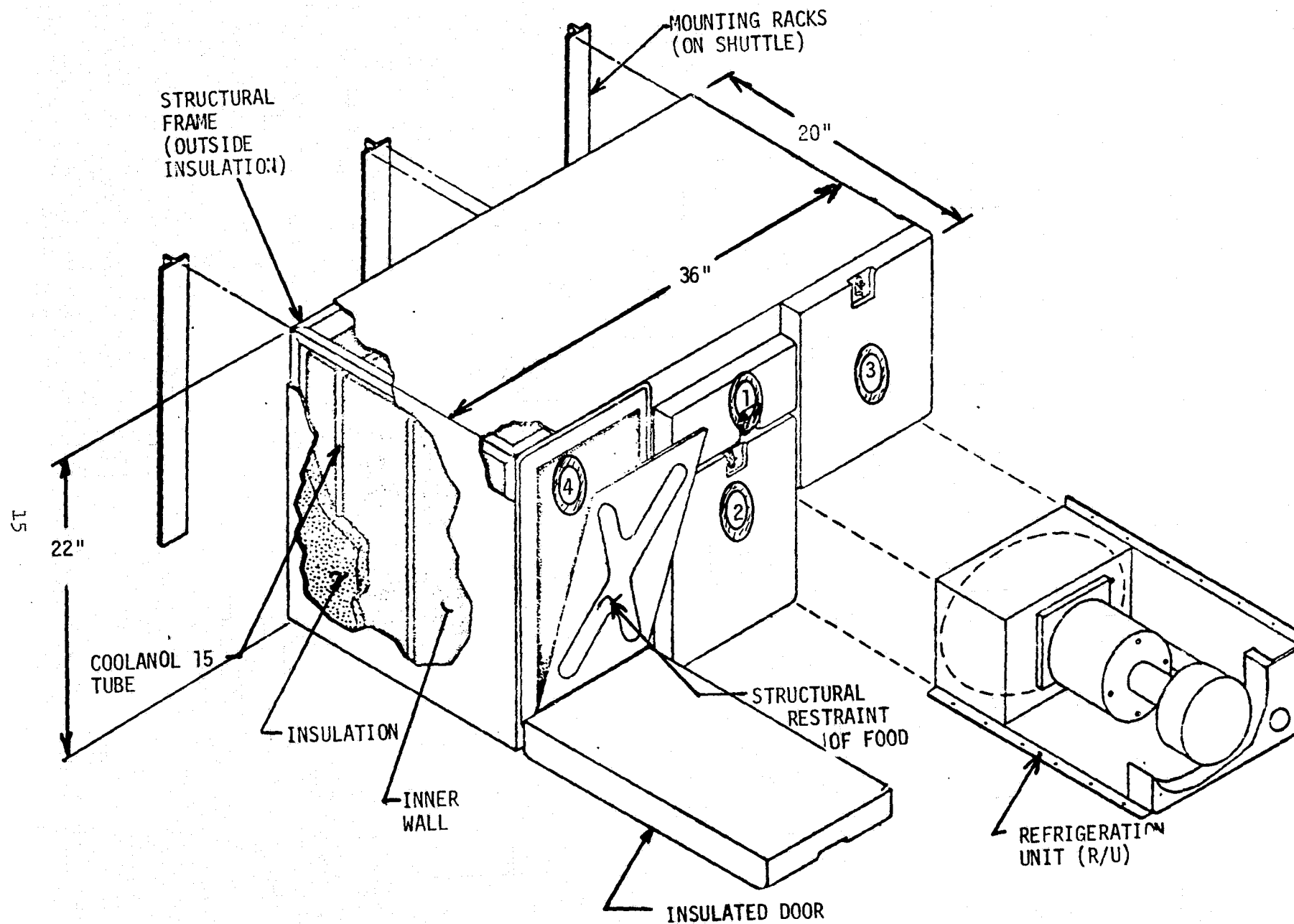


FIGURE 7
CONCEPTUAL DESIGN OF STORAGE VOLUME

TABLE 3
STORAGE VOLUME CONSTRUCTION FEATURES

- | | | |
|---|---------------------------------|---|
| 0 | INNER SHELL | - PROVIDES STORAGE COMPARTMENTATION, LOAD TRANSFER TO INSULATION AND OUTER SHELL, AND THERMAL CONDUCTOR FOR COOLING SYSTEM |
| 0 | OUTER SHELL | - PROVIDES STRUCTURAL LOAD PATH FROM INNER SHELL TO MOUNTING SYSTEM |
| 0 | THERMAL INSULATION | - INSULATES INNER SHELL FROM AMBIENT ENVIRONMENT AND PROVIDES LOAD PATH FROM INNER TO OUTER SHELL |
| 0 | INNER AND OUTER SHELL ISOLATION | - SUSPENSION OF INNER SHELL WITHIN OUTER SHELL PROVIDES MINIMUM THERMAL CONDUCTION TO AMBIENT ENVIRONMENT AND MOUNTING SYSTEM |
| 0 | MOUNTING SYSTEM | - UTILIZES STORAGE MODULE FASTENERS |

is highly conductive aluminum maintain near isothermal conditions in each compartment.

Four compartments provide storage of food and medical samples. Compartment #1 is launched empty and is utilized for the initial storage of urine and fecal samples. The medical samples are collected from the crew and passengers every day. By the time #1 is filled, all of the food in #2 has been eaten and #2 is then used for sample storage. When #2 is again filled, #3 is available for all remaining samples of a 30-day mission. Each compartment is isolated from all others by a system of seals and doors to prevent contamination of the food.

The freezer is insulated with 2-inches of polyurethane foam. The foam also bonds the exterior skin and inner wall to form structural panels. Additional structural members are present to transfer the loads into the Shuttle mounting racks via 16 bolts. The doors are insulated, but are not structural; an inner removable liner is provided to restrain the food.

4.0 EVALUATION OF CANDIDATE REFRIGERATORS

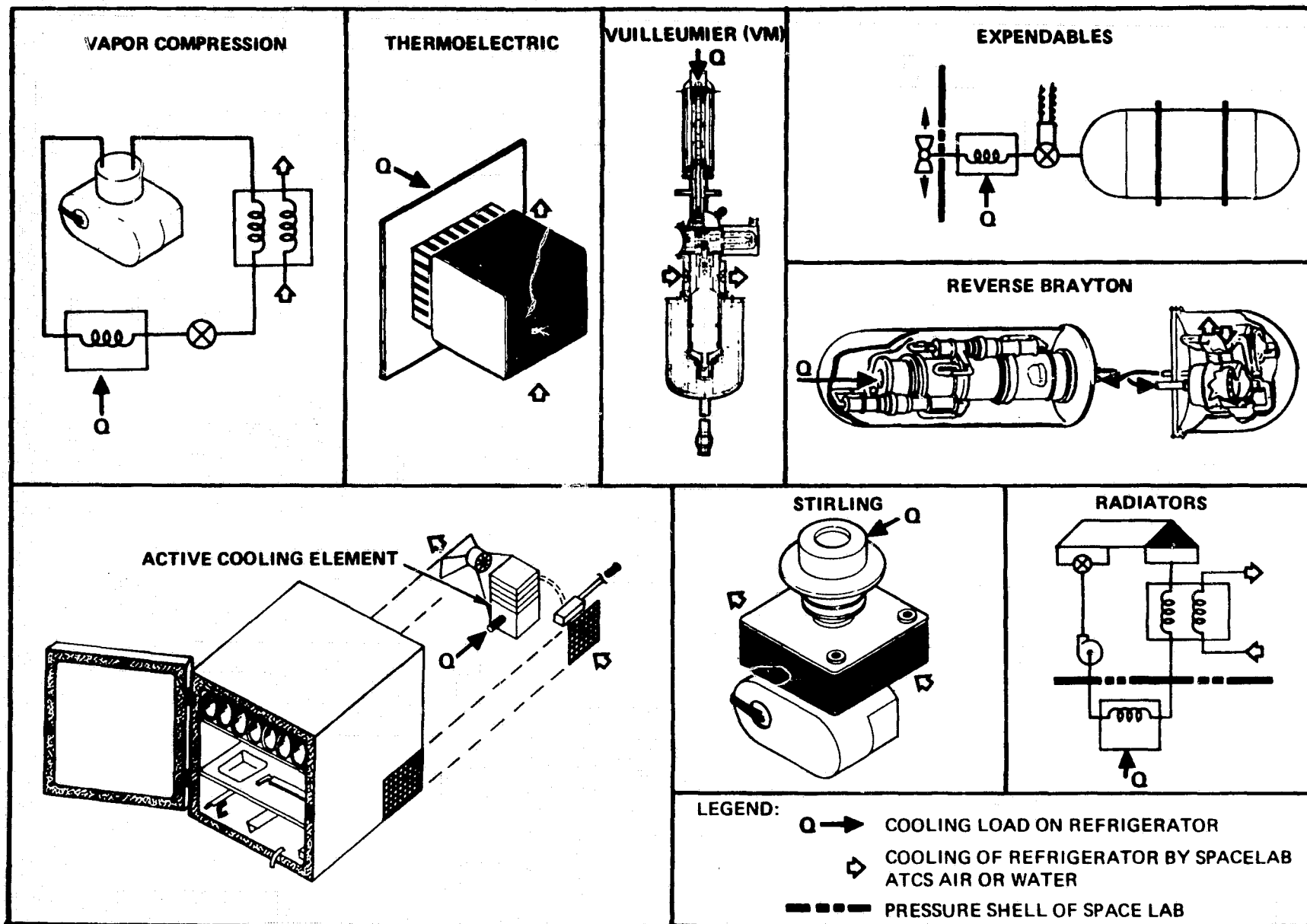
This section summarizes the trade-offs of the candidate refrigeration systems; Reference 2 documents the details of those trades. Initially 8 coolers were considered; after screening parametric trade-off data were generated on Thermoelectrics, Stirling Cycle, and Vapor Compression. Ultimately the Stirling Cycle was selected due to a smaller power requirement as compared to Thermoelectrics and safety as compared to Vapor Compression.

4.1 Candidate Refrigerator Screening

Figure 8 presents the 8 candidate refrigerators. Each of these coolers either have been used in space, or are currently being developed for space application. Not all have been developed for the temperature or capacity requirements of the Shuttle kit freezer, but they have the potential for such use. The lower left hand corner of Figure 8 presents a general freezer with a storage volume, the active cooler, controller and a heat rejection system, air in this case. The active cooling element is one of the 8 systems pictorially illustrated on the figure, and removes the heat load from a small area indicated as "Q".

Table 4 presents a critique of the candidate refrigerators. Based on a review of literature and knowledge on how each machine operates (both in theory and hardware), each factor was assessed. The first column lists the type of refrigerator; the second one assesses the overall suitability for the freezer mission. The third column expresses the current state-of-the-art. The fixed elements are the hardware exclusive of penalty factors and are relatively ranked. COP, Coefficient of Performance, is the **cooling effect per unit power input, and** a relatively ranking is assigned based on previous experience. Safety problems are unusually hazards above the norm for typical machines. The development and recurring costs are separately evaluated. The last column notes specific problem areas for each refrigerator.

Gas cycle equipment are currently receiving intensive development for cryogenic cooling in space, and with additional effort can be developed for the freezer. The gas cycle refrigerators normally use Helium (He) as the working fluid. Stirling cycles are the most attractive in this category because of higher efficiency and smaller equipment than the Vuilleumier (VM) or Brayton cycle. VM is the next best choice, and would be selected if the life of the Stirling proved to be inadequate for this application. Brayton cycle equipment are also being developed for long life, but due to low efficiency in small sizes,



CANDIDATE REFRIGERATORS

FIGURE 8

TABLE 4

CRITIQUE OF CANDIDATE REFRIGERATORS

(- 20°F TO 0°; 244°K TO 256°)

REFRIGERATOR		MISSION SUITABILITY	HARDWARE DEVELOPMENT STATUS	FIXED ELEMENTS		COP	SAFETY PROBLEMS	COSTS		COMMENTS
				WEIGHT	VOLUME			DEVELOPMENT	RECURRING	
GAS CYCLE REFRIGERATORS	STIRLING	GOOD	SOME UNITS AVAILABLE SUITABLE FOR ZERO G USE	GOOD	GOOD	GOOD	NONE	FAIR	MORE EXPENSIVE THAN VAPOR COMPRESSION	DEVELOPMENT IS NEEDED BUT COSTS SHOULD BE REASONABLE
	VM	QUESTIONABLE	SOME UNITS AVAILABLE SUITABLE FOR ZERO G USE	GOOD	GOOD	FAIR	NONE	FAIR	MORE EXPENSIVE THAN VAPOR COMPRESSION	CONSIDER ONLY IF VERY LONG LIFE AND LOW POWER CONSUMPTION ARE NEEDED
	BRAYTON	NOT A VIABLE CANDIDATE								LOW COP AND NEEDS DEVELOPMENT FOR THIS TEMPERATURE
VAPOR CYCLE (V-C) REFRIGERATORS		GOOD	- NO DEVELOPMENT OF APPROPRIATE SIZE EQUIPMENT FOR ZERO G - EXISTING HARDWARE MUCH TOO LARGE OR IS NOT SUITABLE	GOOD	GOOD	VERY GOOD	FLUIDS FOR CYCLE	MORE EXPENSIVE THAN STIRLING	MORE EXPENSIVE THAN T/E	NEEDS DEVELOPMENT OF A ZERO G COMPRESSOR
ABSORPTION/ ADSORPTION REFRIGERATORS		NOT A VIABLE CANDIDATE								SEVERAL FACTORS MAKE THIS CHOICE UNCOMPETITIVE
THERMOELECTRICS (T/E)		GOOD	PRODUCTION UNITS AVAILABLE FOR SPACECRAFT USE	EXCELLENT		POOR	NONE	NONE	EXCELLENT	COP MAY REQUIRE EXCESSIVE POWER AT THIS LOAD AND TEMPERATURE
EXPENDABLES		NOT A VIABLE CANDIDATE								PENALTY FOR CONSUMABLES IS TOO HIGH
DIRECTIONAL SPACE RADIATORS		QUESTIONABLE SUITABLE FOR ON ORBIT MISSION PHASE ONLY	SPACE QUALIFIED	POOR	POOR	EXCELLENT	MINOR	FAIR	FAIR	- SYSTEM SUFFERS FROM HIGHLY COMPLEX INTERFACES - QUESTIONABLE FEASIBILITY WITHOUT ORIENTATION CONSTRAINTS

the Brayton cycle is less attractive than both the VM and Stirling.

Vapor cycle or vapor compression refrigerators are normally used for this temperature and capacity in ground based systems. This system has the lowest power required of all the candidates and has a low hardware weight and volume. However, no zero-g hardware of appropriate size has been developed, notably the compressor. That problem can be overcome by additional development. The V-C system has a safety problem because the highly effective working fluids produce toxic gaseous in the closed environmental control system of the Space Shuttle.

Absorption and Adsorption refrigerators are heat driven coolers (as opposed to V-C which is work driven). As such they have inherently very low COP's (typically required 5 to 10 times as much energy as V-C). These systems are employed when cheap heat is available. However, in the Shuttle cabin, the heat must be obtained by the dissipation of electrical power. That fact plus the large, expensive refrigeration equipment, hazardous fluids and needed development eliminate absorption and adsorption as viable candidates as a shuttle freezer kit.

Thermoelectric coolers are solid state devices and require no moving parts. Recent developments in materials have made significant reductions in power requirements. Unfortunately the power demand is still very high. T/E are very small and light weight. Little to no development is needed for T/E's.

Expendables require extensively large quantities of fluid for a 30-day mission. In addition hazardous fluids and complex interfaces with the shuttle to vent the expendables are required. These factors eliminate expendables as viable candidates.

Directional space radiators have been used in space, and were the heat sink system for Skylab. The radiators require very small amounts of power to pump a fluid loop. Obtaining the desired temperature is well within system capabilities; but complex interfaces with the space shuttle are necessary. The kit must be configured to penetrate the shuttle cabin pressure shell with cold fluid. Due to this factor, this candidate was judged to be of questionable suitability for the shuttle.

The above screening identified the obvious choices. T/E, V-C and Stirling Cycles are considered to be the most promising coolers; the rationale for

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OF POOR QUALITY

each selection are presented below:

<u>Refrigerator</u>	<u>Rationale</u>
Thermoelectric	. High Development Status . Small Fixed Weight and Size But High Power Penalty
Vapor-Compression	. Low Power . Reasonable Size of Fixed Elements
Stirling	. Modest Power Penalty . Reasonable Size of Fixed Elements . Potential To Scale From Previously "O"-g Developed Hardware

These coolers were studied and parametric trade-off data were generated for each type. That data are presented in the following sections.

4.2 Parametric Trade-Off Data

This section presents the trade-off data on the three promising coolers. The power requirements, hardware envelope and weight, and the total equivalent weight of each cooler are presented as a function of the required cooling capacity for each of three heat sinks (water at 45°F and 80°F and air at 80°F). The state-of-the-art of each cooler is also noted.

4.2.1 Thermoelectrics (T/E)

T/E coolers are solid state refrigerators and produce a cooling effect directly from an electric current. Characteristically the devices have very long life and small size but very large power requirements. Flight units for space applications have been built and commercial units are also available. Recent advances have reduced the power consumption but T/E's are still relatively in-efficient.

Figure 9 outlines the fundamental parts of a T/E cooler. The device utilizes the Peltier effect which occurs with the passage of a D.C. current through two dissimilar materials. Heat is absorbed at the cold space and that heat plus the power consumed is released to a heat sink at the hot junction. The controller maintains the proper temperature in the cold space. The performance of the cooler is a function of Z , the figure of merit of the two types of materials, n-type and p-type. The definition of this parameter is given on Figure 9. Based on Z the power requirement can be calculated from

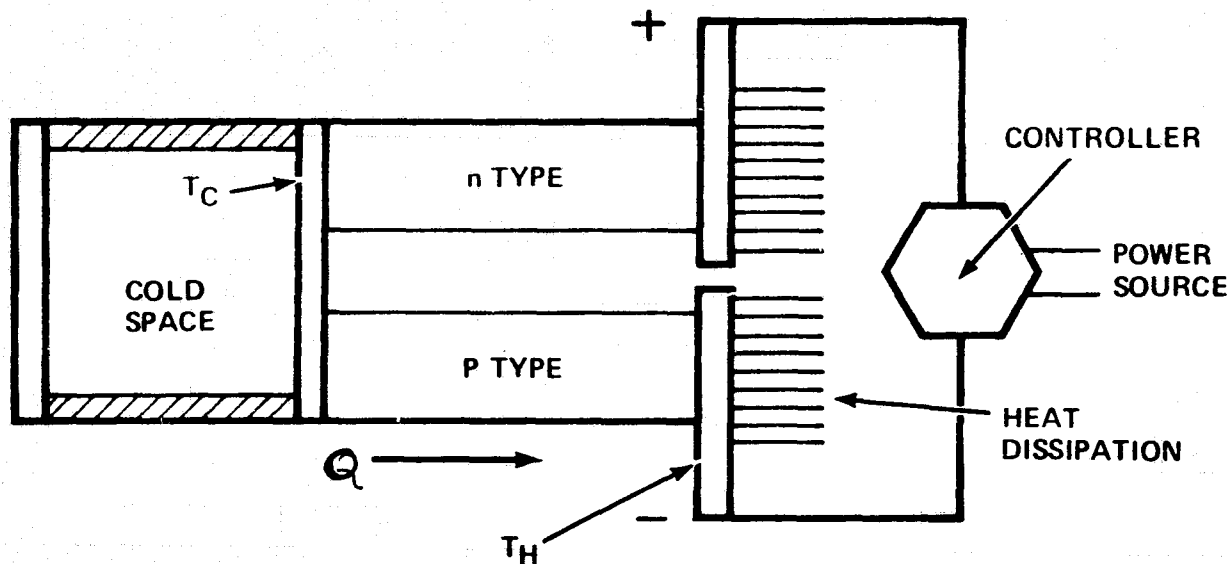


FIGURE OF MERIT

- OF A MATERIAL:

$$Z = \frac{s^2}{\rho k}$$

- OF A COOLER:

$$Z = \frac{(s_p - s_n)^2}{\left(\sqrt{\rho_n k_n} + \sqrt{\rho_p k_p} \right)^2}$$

WHERE:

s = SEEBECK COEFFICIENT FOR A SINGLE MATERIAL $\mu V/K$

ρ = ELECTRICAL RESISTIVITY, OHM-CM

k = THERMAL CONDUCTIVITY, W/CM-K

Z = FIGURE OF MERIT, K^{-1}

FIGURE 9
THERMOELECTRIC COOLING

equations in Reference 6 or other texts on T/E cooling.

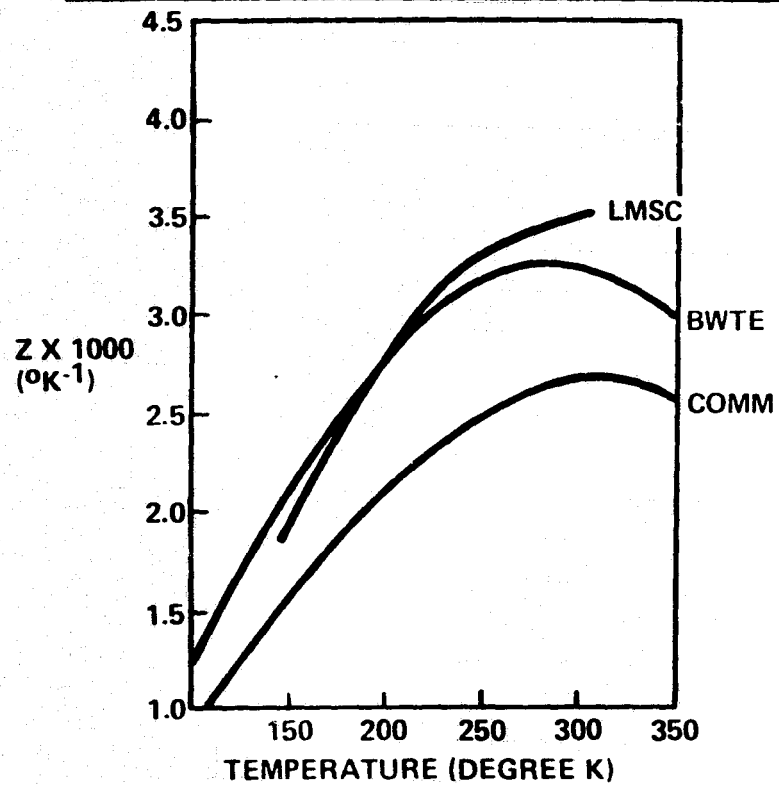
Figure 10 presents the figure of merit for the current and most advanced T/E materials. Commercial materials are available in quantity at low cost; standardized modules can be purchased in several sizes and single, two, and more stages. The Borg Warner and the LMSC materials have only been produced by laboratory methods and very limited quantities have been made. For the time frame of a shuttle launch, sufficient quantities of all three types of materials should be available for a freezer application. Due to additional development, the cost of the materials should also be reduced to comparable levels. Thus the most effective material is the one with the highest Z. For the -20°F (244°K) to room temperature 77°F (298°K) application, the BWTE material is considered the best set. That material was baselined for the trade-offs.

Table 5 presents the estimated power requirements for single, two and three stage coolers with each heat sink. Single stage modules were selected with the 45°F water heat sink. At the required temperature difference a single stage cooler is adequate. There is little to no efficiency increase for two stages; but there is a size and weight penalty.

TABLE 5
POWER REQUIREMENTS FOR STAGED T/E COOLERS

	POWER TO T/E ELEMENTS WATTS		
	Single Stage	Two Stages	Three Stages
For 100 WATTS at -20°F			
45°F Water Heat Sink ($T_H = 55^{\circ}\text{F}$)	179	171	---
80°F Water Heat Sink ($T_H = 90^{\circ}\text{F}$)	584	399	≈ 380
80°F Air Heat Sink ($T_H = 100^{\circ}\text{F}$)	790	535	≈ 500

P-TYPE FIGURE OF MERIT MEASUREMENT DATA



COMM: COMMERCIAL

LMSC: LOCKHEED MISSILES AND SPACE CO.

BWTE: BORG-WARNER THERMOELECTRICS

N-TYPE FIGURE OF MERIT MEASUREMENT DATA

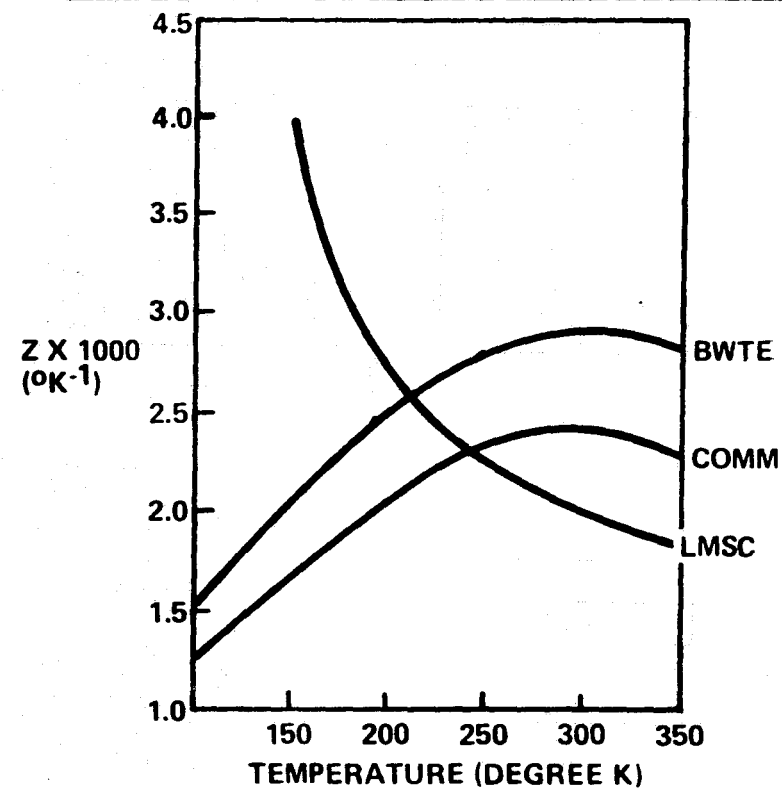


FIGURE 10

T/E MATERIAL PROPERTIES

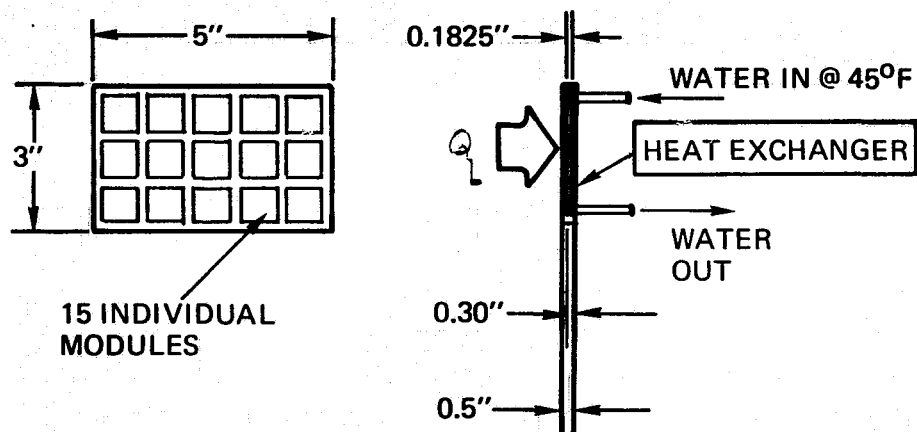
Two stage coolers were selected for the 80°F water and air heat sinks. The power required for single stage coolers was considered excessive as illustrated in Table 5. Three stages do not produce a significant advantage in power but do increase the weight, envelope and costs of the T/E elements. For the purposes of the trade-offs, two stages were assumed to be sufficiently near optimum.

Individual weight breakdowns and power requirements are presented on Figures 11 and 12. Scaling data for the fixed element weight, volume and power are presented in Figure 13. These data are calculated based on the peak cooling rate and include all of the elements previously identified.

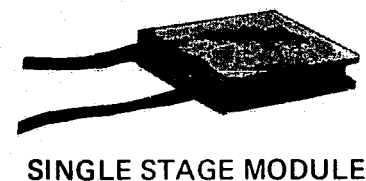
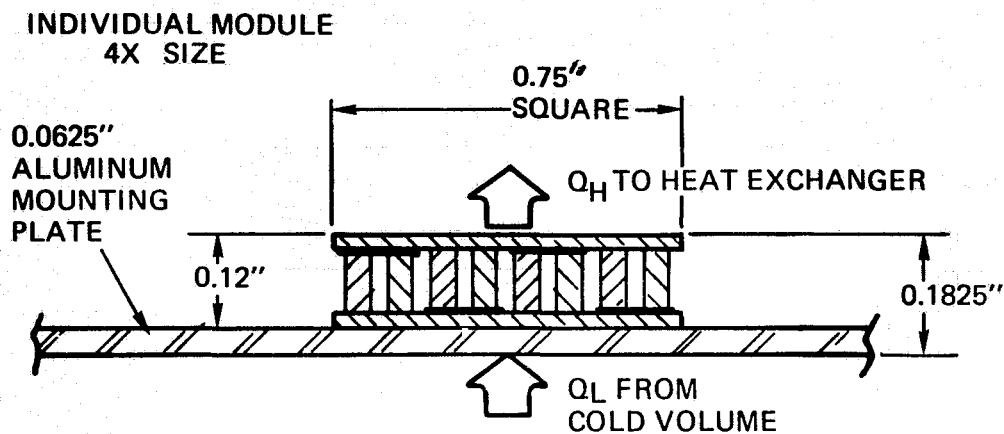
4.2.2 Stirling Cycle

Stirling Cycle refrigerators are closed loop, mechanical coolers. A non-condensable gas, normally Helium, is the working fluid. These refrigerators have been used in space and are currently receiving development for long life space applications. Table 6 summarizes the current state of-the-art. The number of space units have been very limited by comparison with the aircraft units. The life of the space units have been limited, typically 200 hours to a few months. One system is currently being developed for 8,000 hours (one year) life; that unit will have a high efficiency and is applicable to cryogenic temperatures (typically -320°F; 77°K).

Considerable more units have been developed for aircraft usage. These units are reusable and have the opportunity for both scheduled and unscheduled maintenance, life times have been typically 100 to 200 hours. These units are not applicable to a space system because they require the presence of gravity to lubricate the cooler.

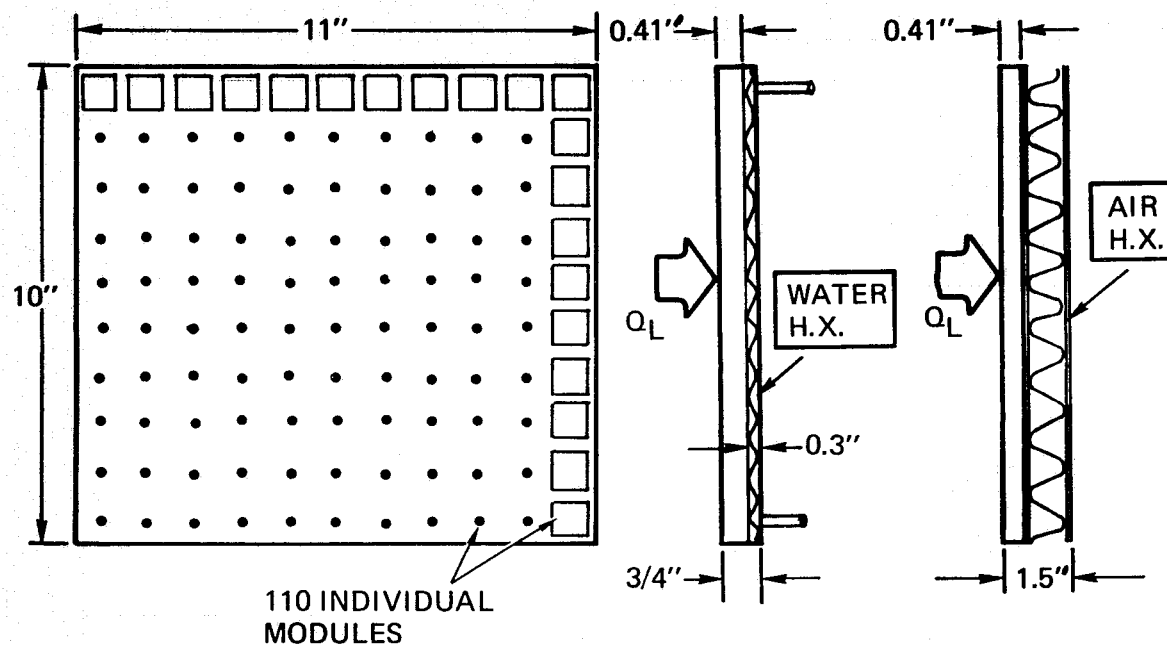


FIXED WEIGHT	
COOLER ELEMENTS:	0.35 LBS
H. X. (WATER)	0.30 LBS
CONTROLLER (1" x 1" x 10")	1.00 LBS
OTHER (WIRES & FLUID LINES)	1.00 LBS
TOTAL	2.65 LBS
POWER	179 WATTS

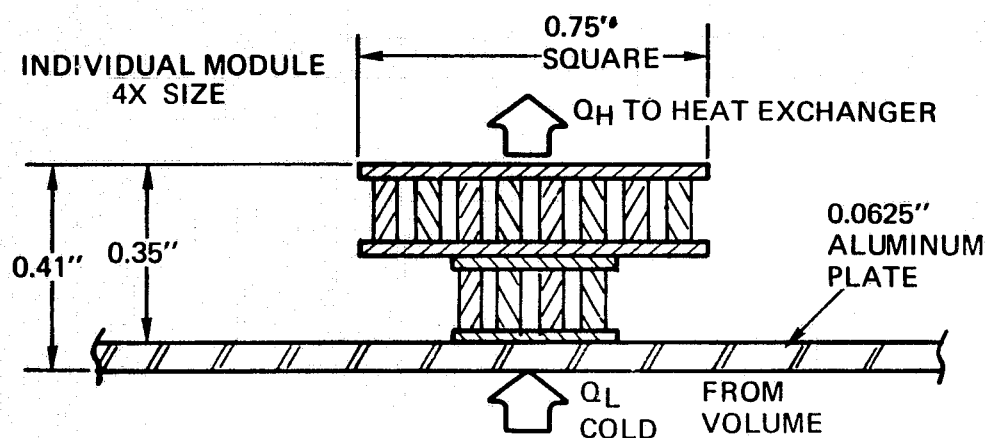


45°F WATER COOLED T/E UNIT
FOR 100 WATTS @ -20°F TO 55°F

FIGURE 11



FIXED WEIGHT		
	WATER COOLER	AIR COOLER
COOLER	2.6 LBS	2.6 LBS
H. X.	0.7	1.3
FAN	—	1.0 (LBS) (54 W)
DUCTING	—	0.5
CONTROLLER (1" x 1" x 10")	1.0	1.0
OTHER	1.0	1.0
TOTAL	5.3 LBS	7.4 LBS
POWER (INCL FAN)	399 W	589 W

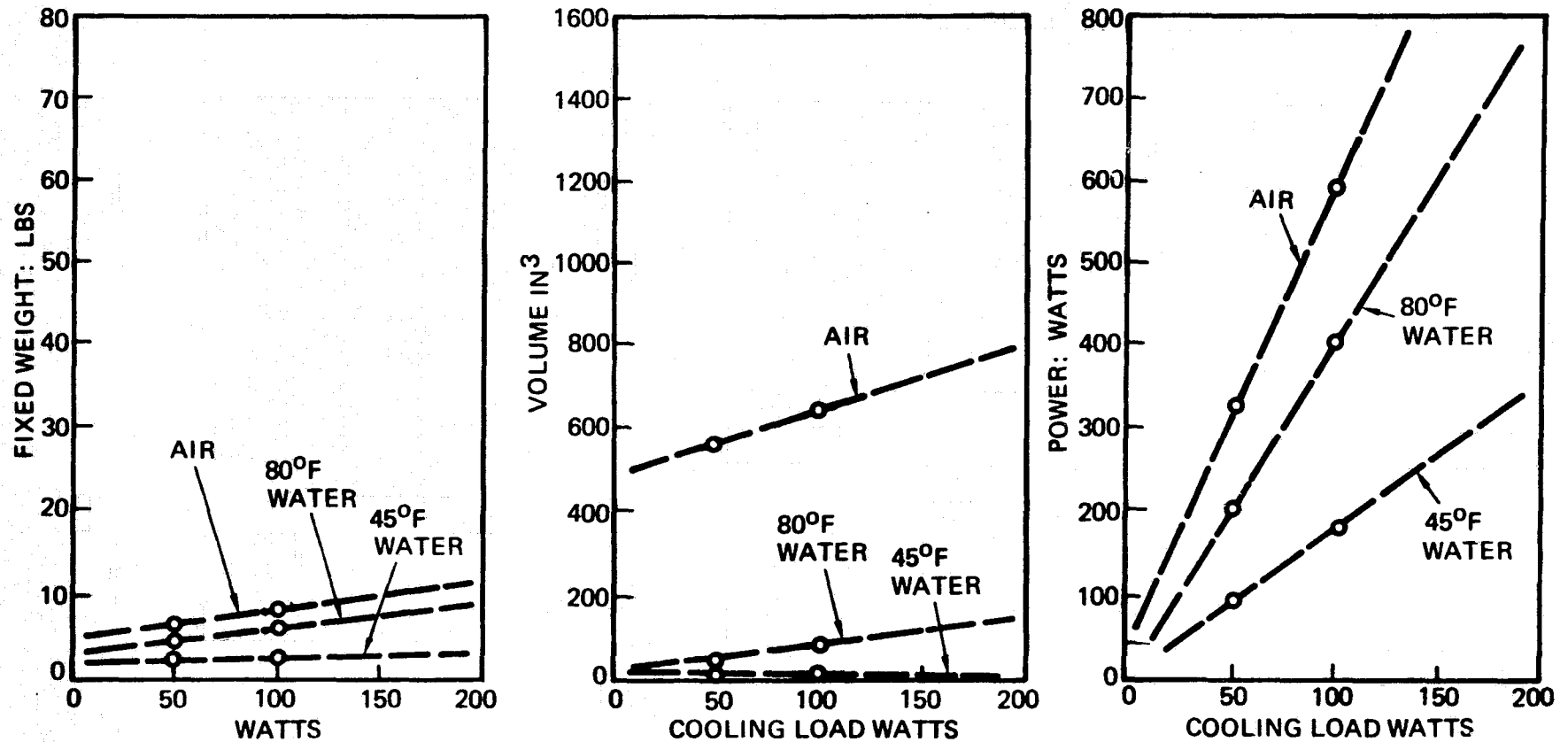


T/E UNIT WITH 80°F HEAT SINKS
FOR 100 WATTS @ -20°F TO 90°F (WATER) AND -20°F TO 100°F (AIR)

FIGURE 12

CONDITIONS:

- — 20°F COLD SIDE TEMP
- 55°F HOT SIDE FOR 45°F WATER COOLING
- 90°F HOT SIDE FOR 80°F WATER COOLING
- 100°F HOT SIDE FOR AIR COOLING



EFFECT OF REFRIGERATION CAPACITY FOR THERMO-ELECTRIC COOLERS

FIGURE 13

TABLE 6
STATE OF THE ART IN
STIRLING CYCLE COOLERS

- ° SPACE: Several coolers have been developed and used in space but with limited life. A recent design is being built for a one-year duration. However, all coolers have been designed for cryogenic temperatures.
- ° AIRCRAFT: Several coolers have been developed. Long lives are achieved by maintenance. Systems depend upon gravity to lubricate to gears and bearings. A positive seal keeps oil out of the compression working space. The piston is dry lubricated and sealed (typically using teflon). All systems have been designed for cryogenic temperatures.
- ° GROUND BASE: MALAKER, Inc. develop several coolers for "FOOD FREEZER" applications. These coolers depended upon gravity. The company is no longer in business.

Only a few Stirling Cycle units have been built for "food freezer" applications. These units were all built by Malaker, Inc. who has now gone out of business. Based on this data, the Stirling Cycle is obviously technically feasible for the intended use, although no unit has ever been constructed for the temperature, size and space application.

Figure 14 presents the ideal Stirling Cycle, including both the T-S and P-V diagrams for the cycle. The cycle consists of an isothermal compression of the working gas, followed by a constant volume regenerative cooling, then an isothermal expansion and finally a constant volume regenerative heating completes the cycle. Heat is rejected to the heat sink during the isothermal compression (points 1 to 2) and heat is absorbed from the low temperature end of the cooler during the isothermal expansion (points 3 to 4).

The theoretical efficiency is the same as the Carnot Cycle, the most efficient that can be obtained. Real machines do not approach the theoretical limit due to inefficiencies in the compression, expansion and regeneration processes and the electric motor drive. Based upon the best state-of-the-art hardware in cryogenic coolers, the efficiency of each element was estimated. The typical compressor and expander efficiency was about 75%. The power requirement for the freezer application was then scaled from that data.

Figures 15 and 16 present conceptual sketches of a Stirling Cycle cooler for a R/U with each of the three heat sinks. Photographs of actual hardware are included to illustrate the hardware. Weight and power estimates are also included for each heat sink. Two estimates of the power requirement are included, one with a compressor and expander efficiency of 85% and one with an efficiency of 75%, both estimates assume a 60% efficient electric motor. The difference in power supplied is about a factor of 2 for only a 10% decrease in efficiency. This effect demonstrates the need for high mechanical efficiency in all components of the Stirling Cycle. Unfortunately this need cannot be completely met; the 85% is considered to be a practical limit but 75% is more consistent with efficiencies in current hardware.

Figure 17 presents scaling data for fixed weight and volume for all three heat sinks. Power requirements for both 85% and 75% compressor and expander efficiencies are also included for each heat sink. In scaling the power requirements, one must recognize that a device sized for the precise cooling load and high and low temperatures is presumed. Furthermore, no machine has yet been built for these requirements.

TYPICAL WORKING FLUID : HELIUM

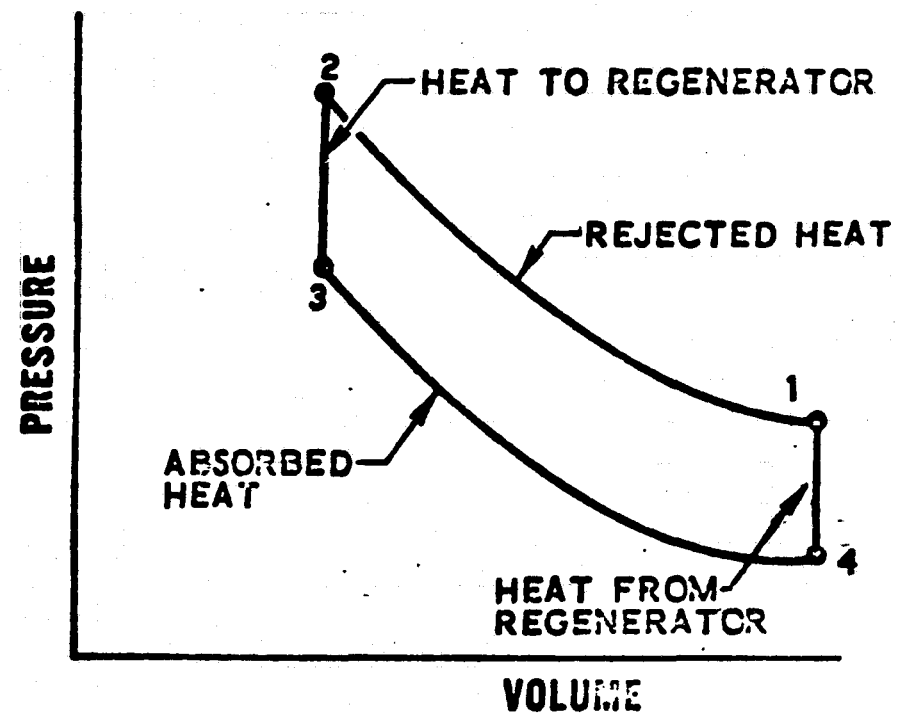
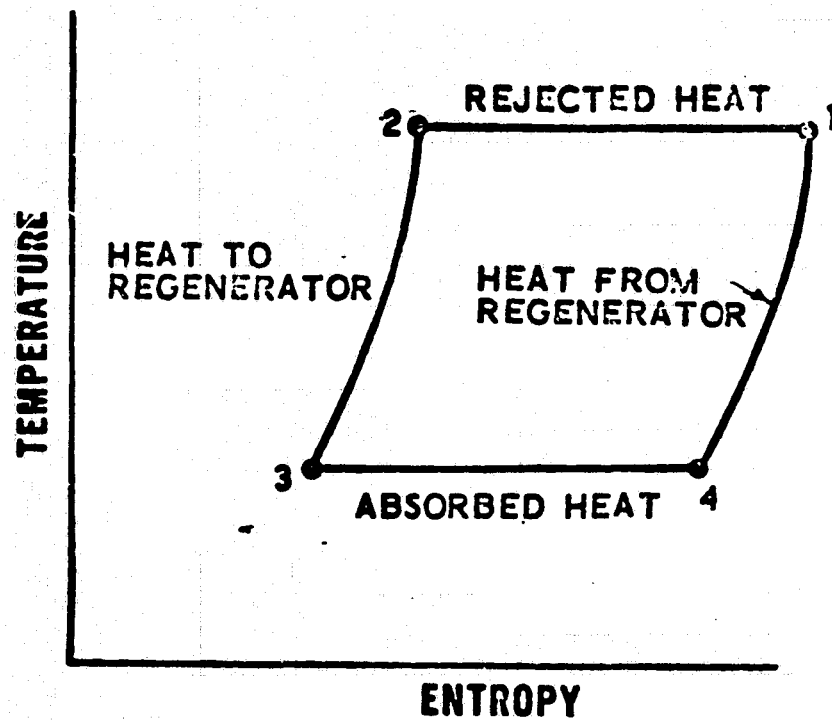
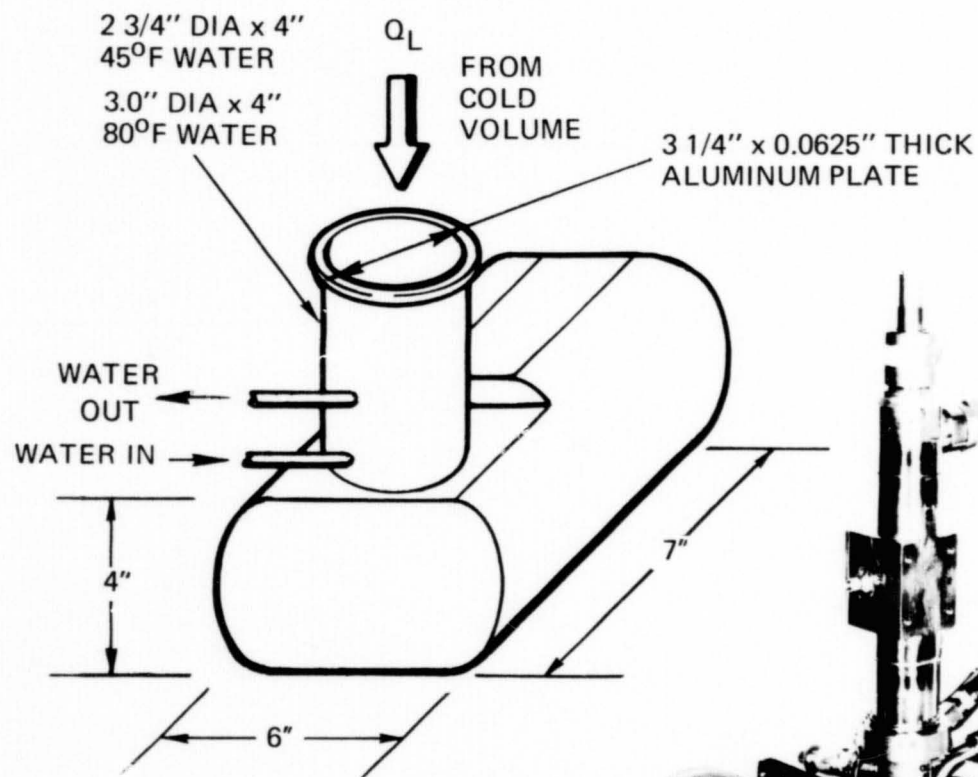
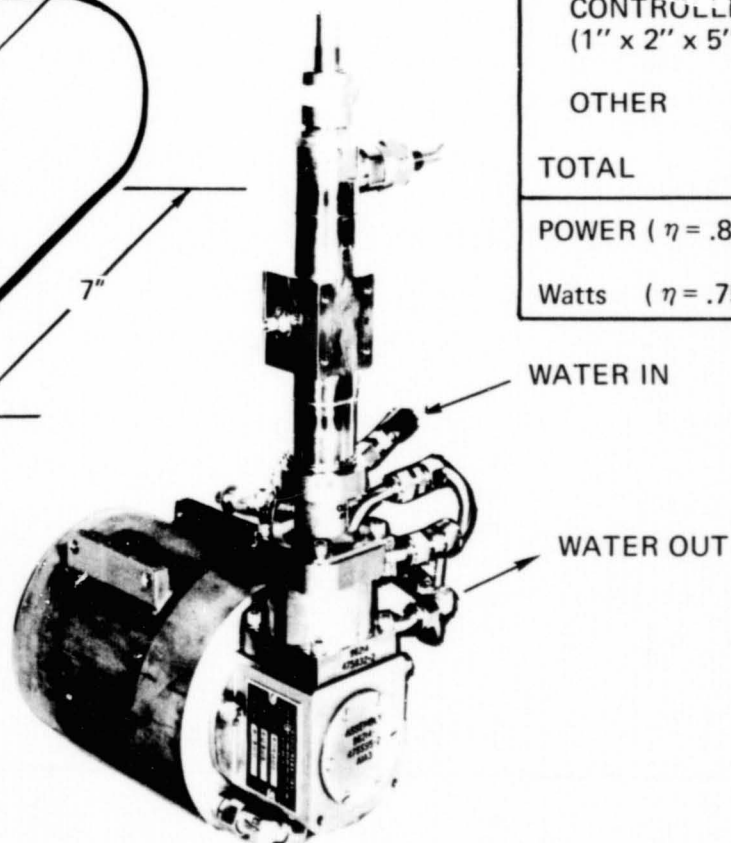


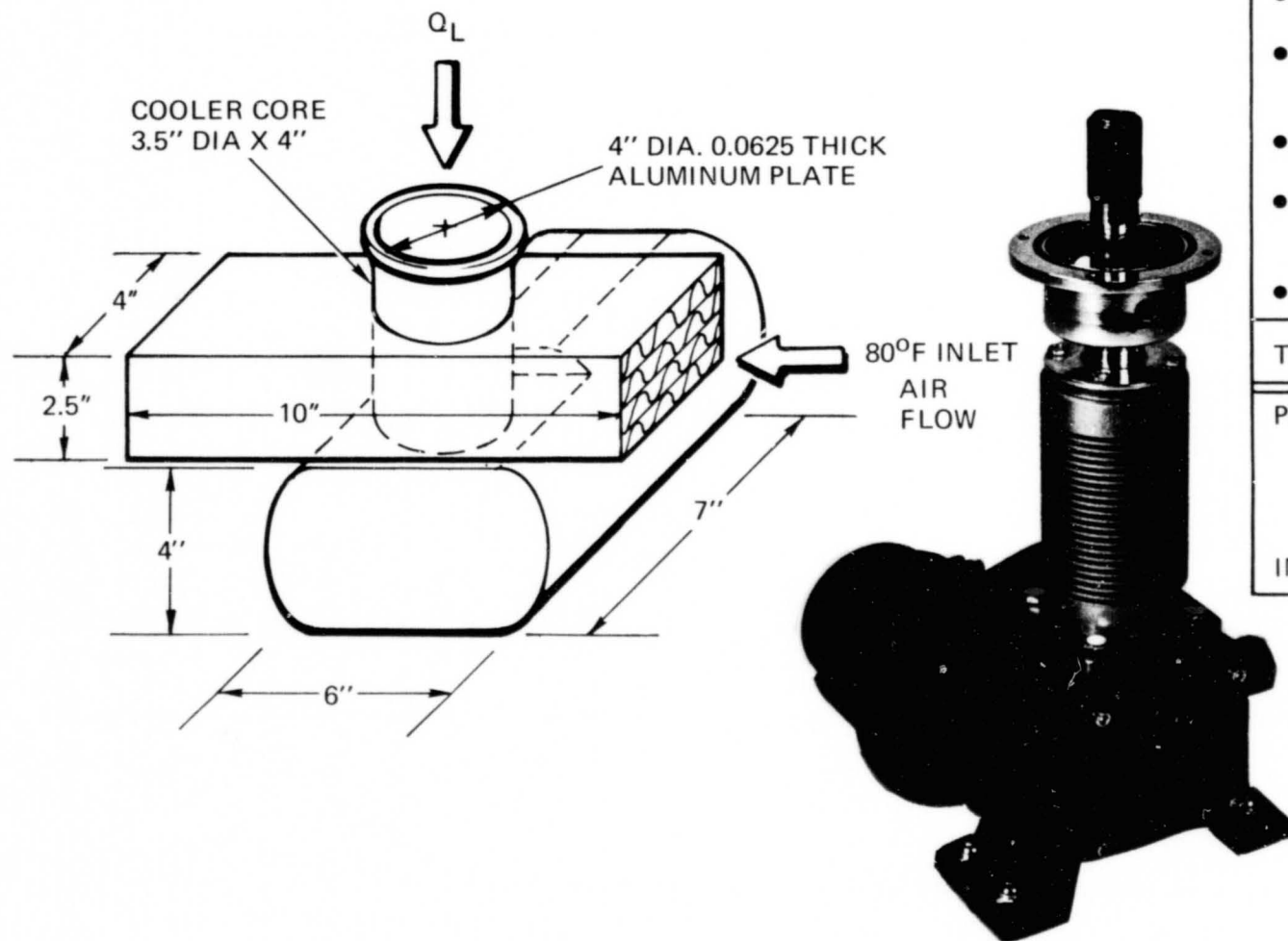
FIGURE 14
IDEAL STIRLING CYCLE REFRIGERATOR.



FIXED WEIGHT	45°F WATER	80°F WATER
COOLER	12.0	13.0
H. X.	0.25	0.25
CONTROLLER (1" x 2" x 5")	0.4	0.4
OTHER	0.1	0.1
TOTAL	12.75 LBS.	13.75 LBS.
POWER ($\eta = .85$)	123	146
Watts ($\eta = .75$)	220	250



WATER COOLED STIRLING CYCLE UNITS FOR 100 WATTS @ -20°F TO + 55°F AND -20°F TO 90°F



FIXED WEIGHT	
• COOLER	14 LBS
• H. X.	1.5 LBS
• FAN	1.0 LBS (25.5 WATTS)
• DUCTING	0.5 LBS
• CONTROLLER (1" x 2" x 5")	0.5 LBS
• OTHER	0.1 LBS
TOTAL	17.6 LBS
POWER	
$\eta = .85$	185 WATTS
$\eta = .75$	285 WATTS
INCLUDING FAN	

AIR COOLED STIRLING CYCLE COOLER FOR 100 WATTS @ -20°F TO 100°F

FIGURE 16

CONDITIONS:

- - 20°F COLD SIDE TEMP
- 55°F HOT SIDE FOR 45°F WATER COOLING
- 90°F HOT SIDE FOR 80°F WATER COOLING
- 100°F HOT SIDE FOR AIR COOLING

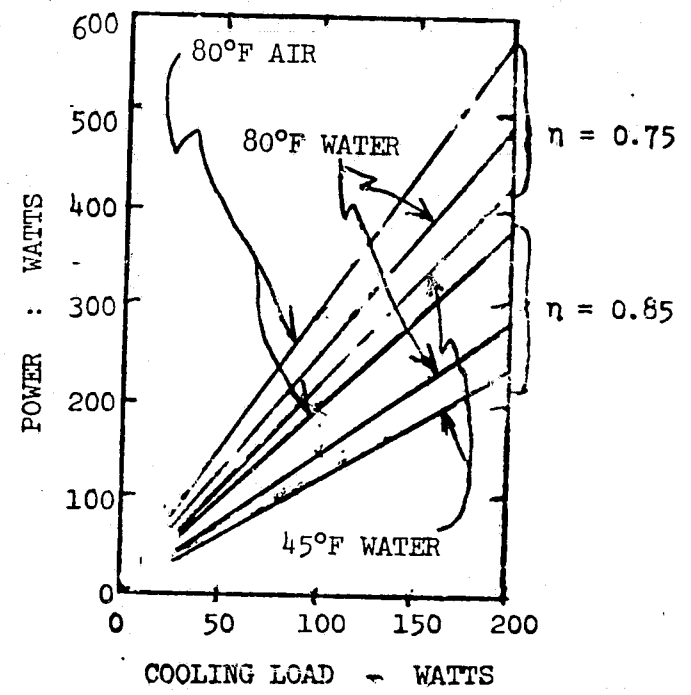
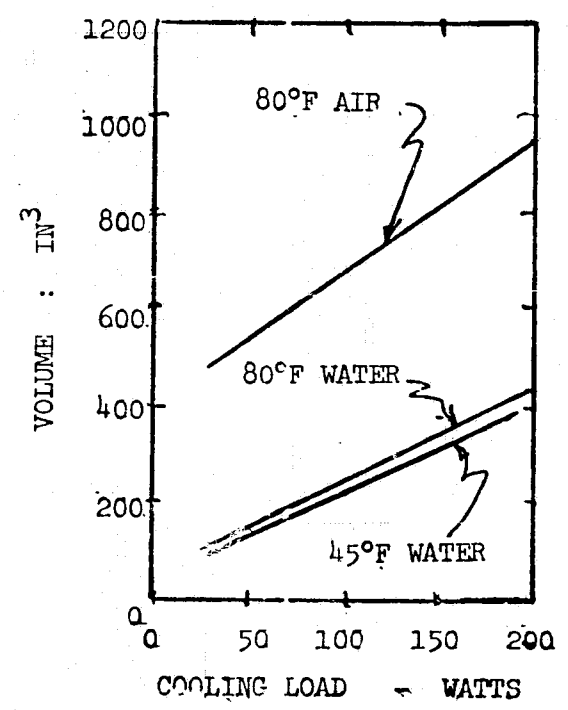
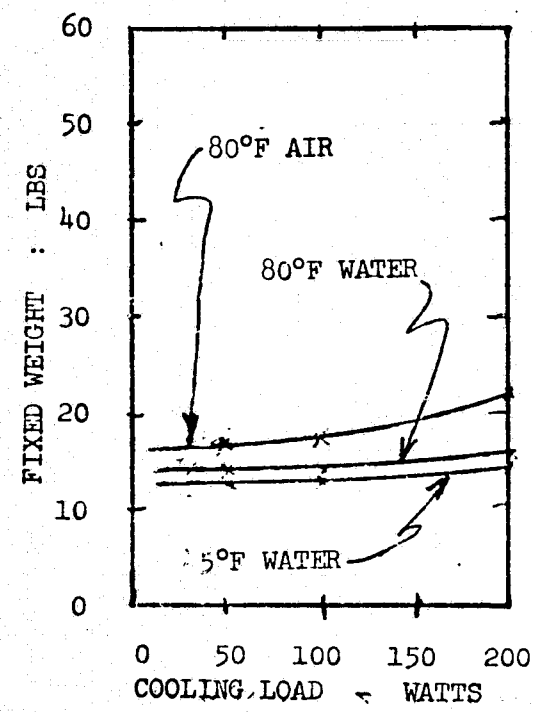


FIGURE 17
EFFECT OF REFRIGERATION CAPACITY FOR STIRLING CYCLE COOLERS

4.2.3 Vapor Compression (V-C)

The vapor compression cycle is the one commonly used in household refrigerators, freezers and most air conditioning systems. Extensive use of the system provides off-the-shelf hardware in all size ranges, temperatures and heat sinks. In addition, the power requirements approach the theoretical minimum. Two problem areas exist when applying the cycle to the space shuttle: One is the problem of lubrication in the zero "g" of space and the other is working fluids in a closed life support system. Solution to both problems would make V-C a very attractive system.

The V-C cycle is illustrated in Figure 18. The system removes heat by evaporation of the working fluid at low pressure in the evaporator. The vapor is then raised to a high pressure in the compressor and is condensed at a high temperature rejecting heat to a heat sink. The liquid at high pressure and temperature is cooled by partial self-evaporation by throttling in the expansion valve. The two-phase flow then returns to the evaporator completing the closed cycle.

An efficient, safe working fluid is required in the V-C cycle. For ground based systems, Freons (e.g. R-12; $\text{C Cl}_2\text{F}_2$: or R-22; CH Cl F_2) are commonly used. These fluids have large heats of evaporation and favorable properties which limit the compression work. Normally, freons are non-toxic and non-reactive with most materials. Unfortunately, in the closed life support system of the space shuttle cabin, freons do react with Lithium Hydroxide (LiOH) and form new compounds dichloro-acetylene (C_2Cl_2) and difluoro-acetylene (C_2F_2). These compounds are toxic to humans and affect performance and physical well being of primates in concentrations of 0.1 to 1 ppm. At 7 ppm dichloroacetylene is 100% fatal to monkeys after 7 days of exposure (Ref. 7). At 0.1 ppm only about 0.00002 lbs of freon can be allowed in the entire shuttle cabin. Obviously, that level or less effectively forbids the use of a freon in a location where it could leak into the cabin.

Other working fluids are available which have been used in refrigerators. These fluids included ammonia (NH_3), propane ($\text{CH}_3\text{CH}_2\text{CH}_3$), propylene ($\text{CH}_2\text{CH}=\text{CH}_2$), isobutane ($\text{CH}(\text{CH}_3)_3$) and Sulfur Dioxide (SO_2). All of these avoid the di-chloroacetylene problem but introduce other hazards, fire and toxicity.

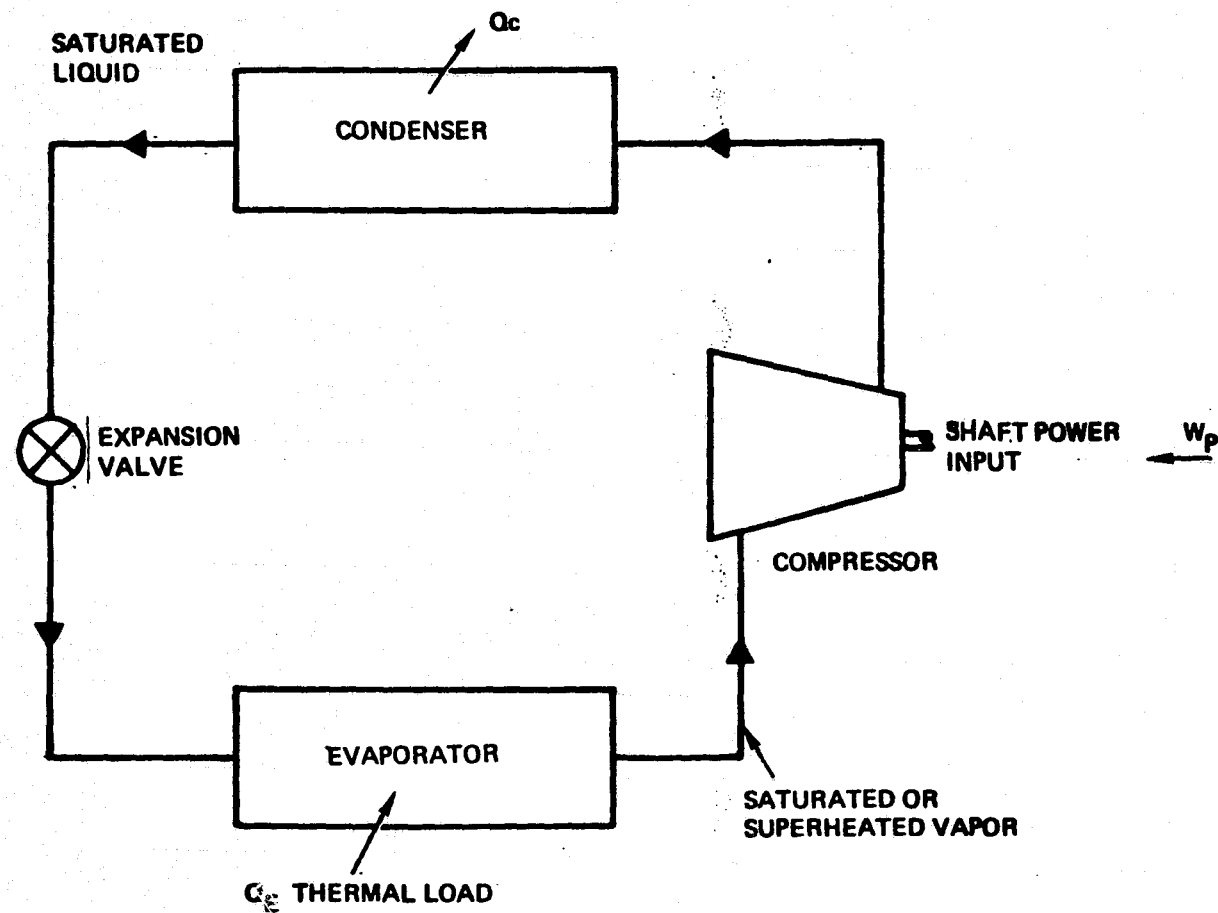


FIGURE 18
SIMPLE VAPOR COMPRESSION SYSTEM

Time did not permit a detailed evaluation of the working fluids. Ammonia was somewhat arbitrarily selected; the rationale was as follows:

- (1) The hazards are known and obvious, protective measures can be implemented and the penalties assessed.
- (2) Ammonia is known to have good thermodynamic performance as a refrigerant.
- (3) Ammonia has been used as a refrigerant for these temperatures in ground based systems with V-C cycles.
- (4) Ammonia leaks are readily detectable by the crew. The concentration at which the characteristic odor is observed occurs at much less than unsafe levels.
- (5) Ammonia is removed by the shuttle life support system.

Obviously additional study on the working fluid is needed but warranted only if V-C is among the best choices for the R/U.

Table 7 presents an estimate of the safety provisions which will be required with NH_3 . Ammonia is both combustible and is toxic to humans; atmospheric limits are a function of time; 400 ppm for one hour to 25 ppm for continuous exposure. (Ref. 8). For the Shuttle Cabin, these levels allow 0.035 lbs to 0.0022 lbs of NH_3 to be in the cabin air. (versus about 2.0 lbs in the R/U.) Obviously very little leakage can be tolerated and a hermitically sealed unit is considered mandatory to maintain the required levels. However, should the seal fail, a rapid or even an explosive release of NH_3 could occur.

By placing the hermetically sealed V-C equipment entirely within another sealed pressure vessel, the gases released from a single failure would not be released into the cabin. However, after an initial failure, leakage of NH_3 into the cabin can still occur, effectively a double failure. Should such a failure occur, the leakage rate would be small and should be removed by the cabin Life Support System. The weight of the second hermetic seal was estimated as an aluminum cyclinder 12" in diameter by 13" long designed to hold internal pressure equivalent to the vapor pressure of NH_3 at 80°F.

Chemical absorption can also provide protection against rapid release of ammonia to the cabin. The approach utilizes a porous bed of phosphoric acid (H_3PO_4) to react with ammonia, forming $(\text{NH}_4)_3\text{PO}_4$. A thin plastic bag

TABLE 7
V-C SAFETY REQUIREMENTS

<u>PRECAUTION</u>	<u>PREVENTS</u>	<u>RESIDUAL RISKS</u>	<u>EQUIPMENT REQUIRED</u>
o HERMETICALLY SEALED COMPRESSOR & VALVES (BASIC UNIT)	o MAJOR LEAKAGE OF FLUID	o EXPLOSIVE RELEASE OF NH_3 (LINE RUPTURE)	INCLUDED IN EQUIPMENT SIZING AS MANDATORY
o 2ND HERMETIC SEAL AROUND ENTIRE SYSTEM + BASIC UNIT	o EXPLOSIVE RELEASE OF NH_3	o LEAKAGE OF GAS INTO CABIN AFTER AN EXPLOSIVE RELEASE (DOUBLE FAILURE)	10.0 LBS OF ADDED STRUCTURE
o ABSORBER PLUS VENTED BAG AROUND ENTIRE BASIC UNIT	o RAPID RELEASE OF NH_3	o HIGH CONCENTRATION OF NH_3 TEMPORARILY IN CABIN	7.0 LBS AS 6" DIA X 7" LONG CANISTER
o 2ND HERMETIC SEAL AND ABSORBER	o RAPID RELEASE OF NH_3 AND SUBSEQUENT LEAKAGE INTO CABIN	o INSIGNIFICANT	14.5 LBS

2ND HERMETIC SEAL
ASSUMED ADEQUATE IN THIS TRADES

is placed around the R/U to force the NH_3 through the absorbent bed; nevertheless a small amount of NH_3 will pass through the absorbent bed and into the cabin if a rapid leak occurs. At small leakage rates, insignificant quantities of NH_3 would be released to the cabin. The sorbent system weighs approximately 7 lbs with a bed volume of a 6" dia. by 7" long cylinder, assuming about 50% bed utilization. The chemical absorption approach is lighter and smaller than the second pressure vessel but can allow potential toxic levels of NH_3 in the cabin. Thus additional barriers would be needed to insure the protection of the crew.

Both a second hermetic seal and a chemical absorbent can reduce the risks to insignificant levels. The pressure vessel would contain any sudden release of NH_3 ; and the chemical absorbent inside the pressure vessel would react and thus prevent any subsequent leakage from occurring. Since the system must contain all of the pressure, the structural penalty is the same as the 2nd hermetic seal. Additional weight is now required for the absorbent. Due to this added weight and the low probability of a double failure, only the 2nd hermetic seal was assumed to be adequate in these trades. In addition, should a double failure occur, the crew will sense the presence of NH_3 before it reaches toxic levels, put on their emergency gas masks and return to earth before suffering any disabilities.

Figure 19 presents the thermodynamic properties of NH_3 and the V-C cycle. The solid line illustrates a typical actual cycle including the effect of compressor efficiency. The dashed line shows the process with two stage compressor with intercooling. With this fluid, a high compression ratio is required; from -20°F , 18.3 psia to 80°F , 153.0 psia and with it a high degree of superheating of the vapor. With a 80°F heat sink (either water or air), two stage compressors were found to require about 30% less power than single stage compression. In both cases, cooling of the vapor between the stages (i. e. intercooling) was also assumed. With a 45°F water heat sink, a single stage compressor was observed to have approximately the same power as a two stage. Consequently, single stage compression was selected for the 45°F water heat sink and two stage compression was selected for the 80°F water and air heat sinks.

REQUIREMENTS

. DETECTABLE SAFETY HAZARDS

- ALL FREONS ELIMINATED DUE TO FORMATION OF TOXIC DICHLOROACETYLENE AT CONCENTRATIONS OF 0.1 PPM
- NH_3 IS READILY DETECTABLE BY HUMAN SENSES WELL BELOW TOXIC LEVELS
- NH_3 REMOVED BY SHUTTLE ECLSS

. GOOD THERMODYNAMIC PROPERTIES

SELECTION : AMMONIA

FURTHER STUDY ON SAFETY HAZARDS IS RECOMMENDED

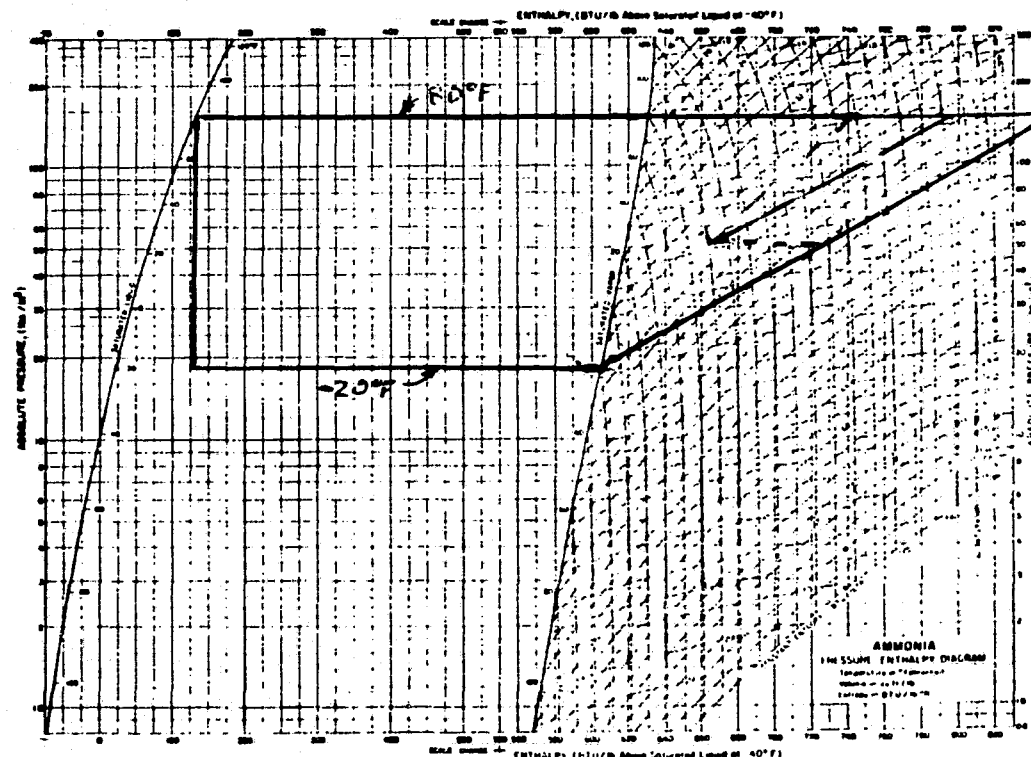


FIGURE 19
WORKING FLUID

Figures 20, 21, and 22 present concepts of V-C coolers for each heat sink. All three concepts include the same safety provision of a 2nd hermetic seal. All components containing ammonia are packaged in the same 12" diameter by 13" long cylinder. The various elements of each concept are illustrated, and weight and envelope estimates of each component are presented. Each system contains a compressor, condenser, receiver, which reduces pressure transients in the system, expansion valve which provides a constant pressure and the evaporator. With water cooling, the water loops are brought inside the 2nd hermetic seal. With air cooling the air side of the heat exchangers are outside the hermetic seal and the ammonia components are inside. Heat transfer is affected by conduction through the walls. For the low temperature heat transfer, an insulated region is provided on the 2nd hermetic seal.

Figure 23 presents scaling data for each of the three heat sinks. Both the fixed weight and equipment volume are relatively unaffected by the cooling rate. The power requirement is directly proportional to the required cooling rate; but the power need is the smallest for V-C in comparison to both T-E and Stirling.

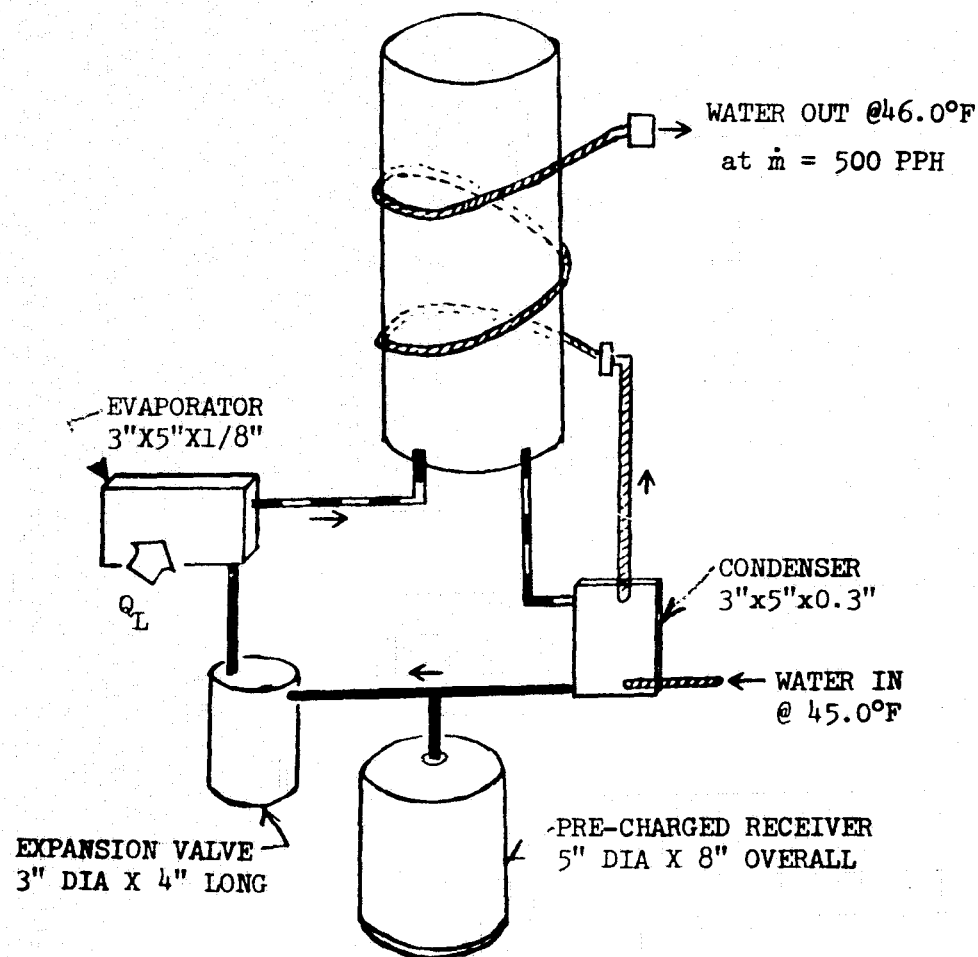
4.3 Selection of R/U

This section presents the data used in the selection of the R/U. VSD generated the weight, volume and power requirements for each of three coolers with three heat sinks. Based upon that data, Boeing optimized the storage volume in each case and conducted the final trade leading to the selection of the Stirling Cycle.

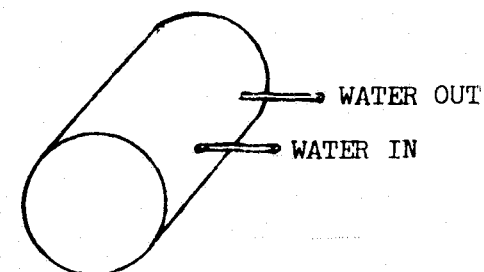
Figure 24 presents the equivalent weight versus the mission duration. In each case, a 100 watt continuous heat level at -20°F is removed by the cooler. Penalties are assessed for the consumption of power assuming that only consumables are added to existing fuel cell kit tanks. For use of the shuttle heat rejection system, a penalty is also included. Due to life considerations, A. C. Motors are assumed on all rotating equipment (fans, V-C and Stirling); T/E requires D. C. power. All concepts prefer the use of water cooling with 45°F most favored. The water loops require the addition of a pumped fluid loop with lines from existing heat exchangers

HERMETICALLY SEALED SINGLE-STAGE COMPRESSOR AND MOTOR

6" DIA X 10" OVERALL



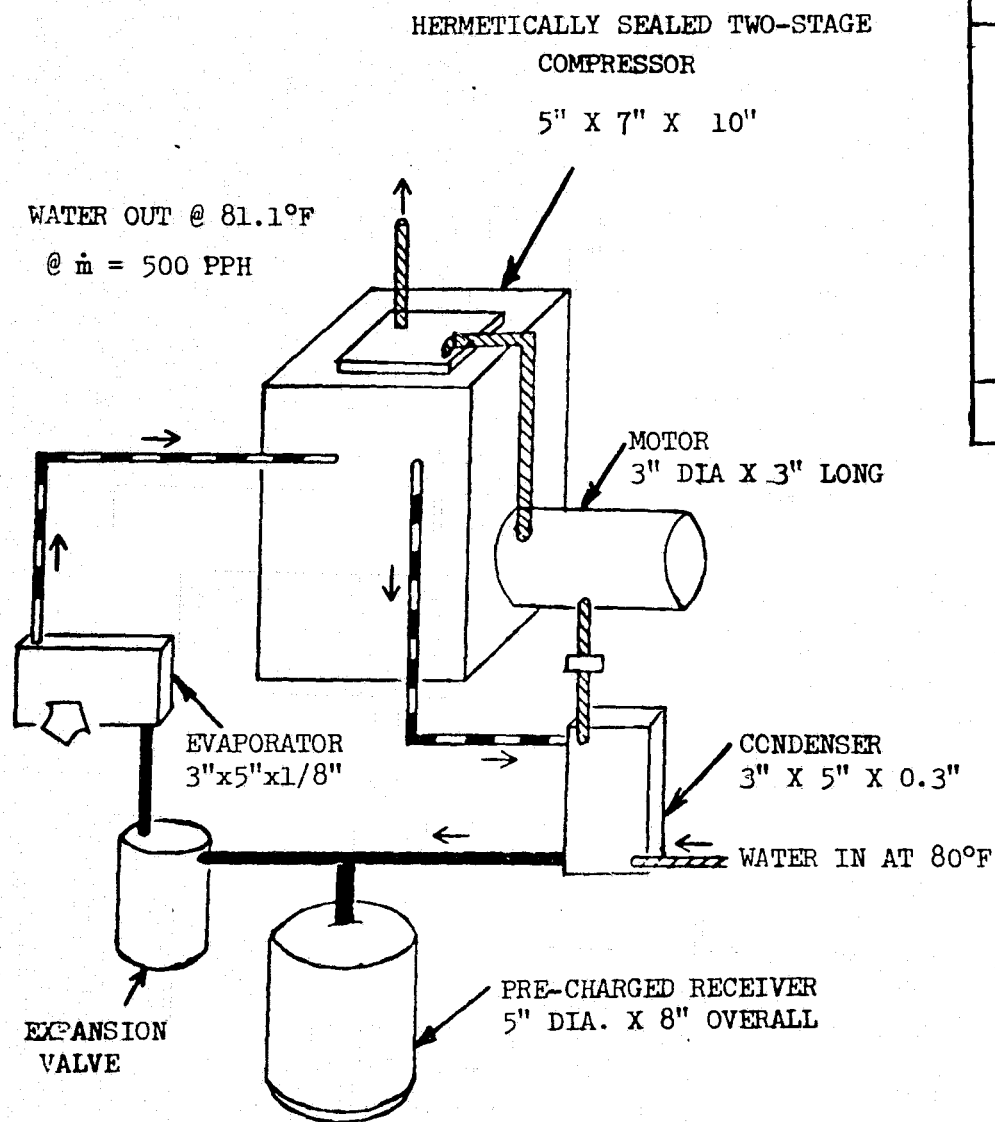
FIXED WEIGHT	
• COMPRESSOR	13.0 LBS
• EVAPORATOR	0.25 LBS
• CONDENSER	0.25 LBS
• RECEIVER	2.5 LBS
• CONTROL VALVE	0.5 LBS
• CONTROLLER (1"x2"x5")	0.5 LBS
• OTHER	0.5 LBS
• SAFETY	10.0 LBS
TOTAL	27.5 LBS
POWER	51 WATTS



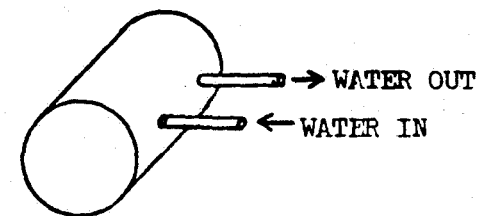
SYSTEM HERMETICALLY SEALED
IN 12" DIA X 13" LONG
CONTAINER FOR SAFETY

FIGURE 20

45°F WATER COOLED V-C UNIT FOR 100 WATTS @ -20°F TO +55°F



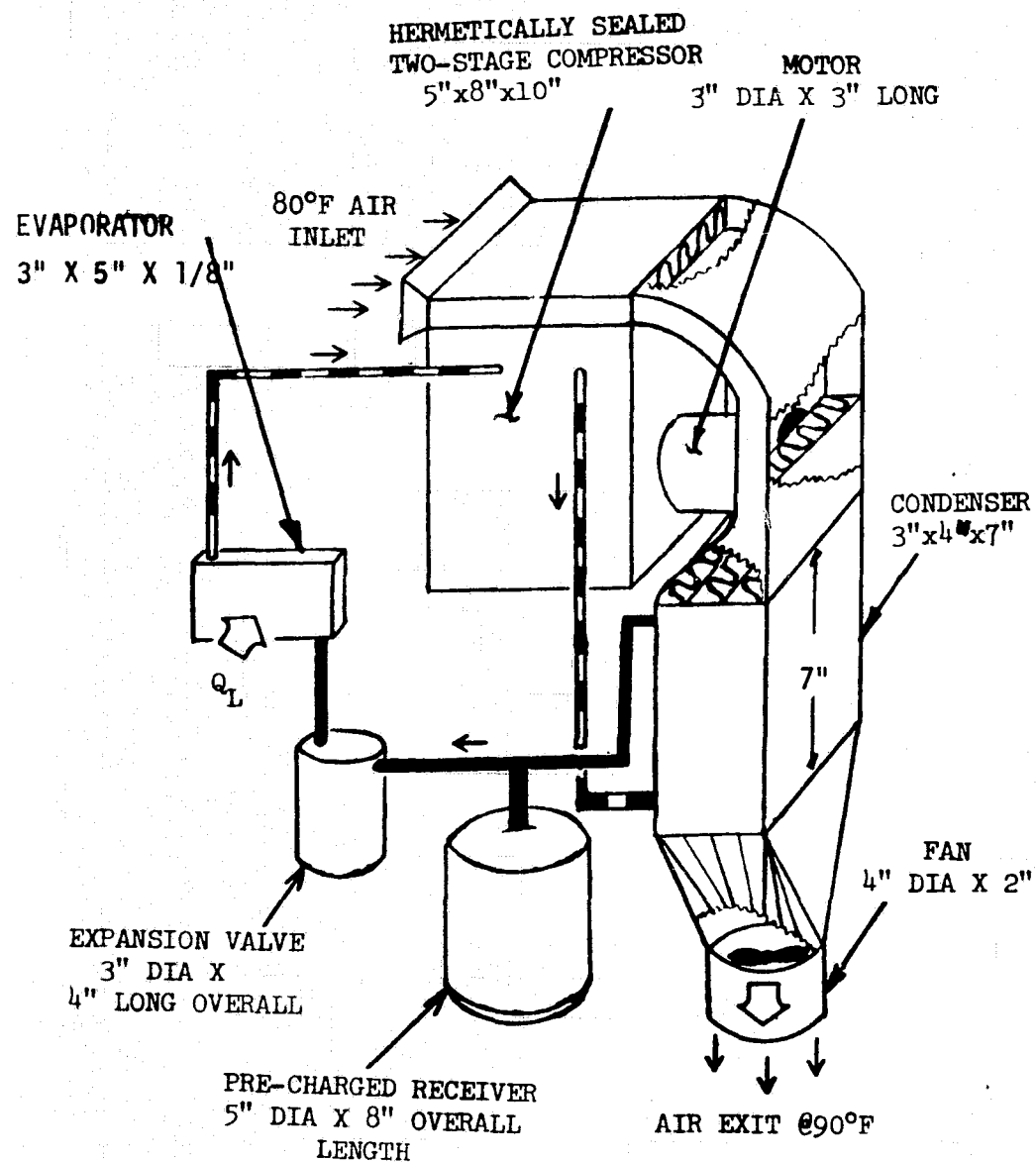
FIXED WEIGHT	
. COMPRESSOR	21 LBS
. EVAPORATOR	.25LBS
. CONDENSER	.25 LBS
. RECEIVER	2.5 LBS
. CONTROL VALVE	.5 LBS
. CONTROLLER (1"x2"x5")	.5 LBS
. OTHER	.5 LBS
. SAFETY PROVISIONS	10.0 LBS
TOTAL	35.5 LBS
POWER	69.4 WATTS



SYSTEM HERMETICALLY SEALED
FOR SAFETY

FIGURE 21

80°F WATER COOLED V-C UNIT FOR 100 WATTS @ -20°F TO 90°F



FIXED WEIGHT	
• COMPRESSOR	21.1 LBS
• EVAPORATOR	.25 LBS
• CONDENSER	1.0 LBS
• RECEIVER	2.5 LBS
• CONTROL VALVE	0.5 LBS
• CONTROLLER (1"x2"x5")	0.5 LBS
• FAN (17 WATTS)	1.0 LBS
• DUCTING	0.5 LBS
• OTHER	0.5 LBS
• SAFETY	10.0 LBS
TOTAL	37.75 LBS
POWER (INCLUDING FAN)	93.3 WATTS

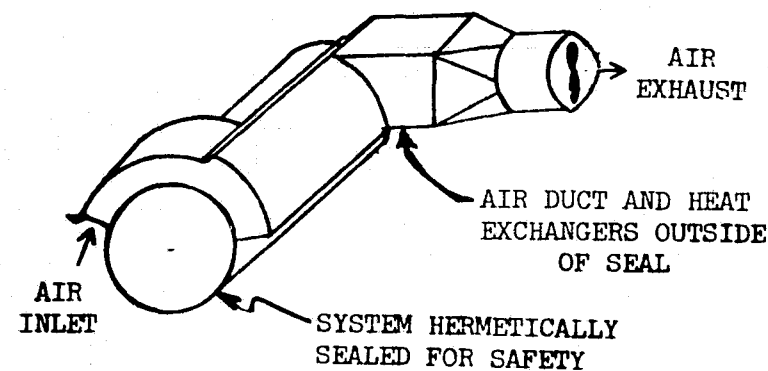
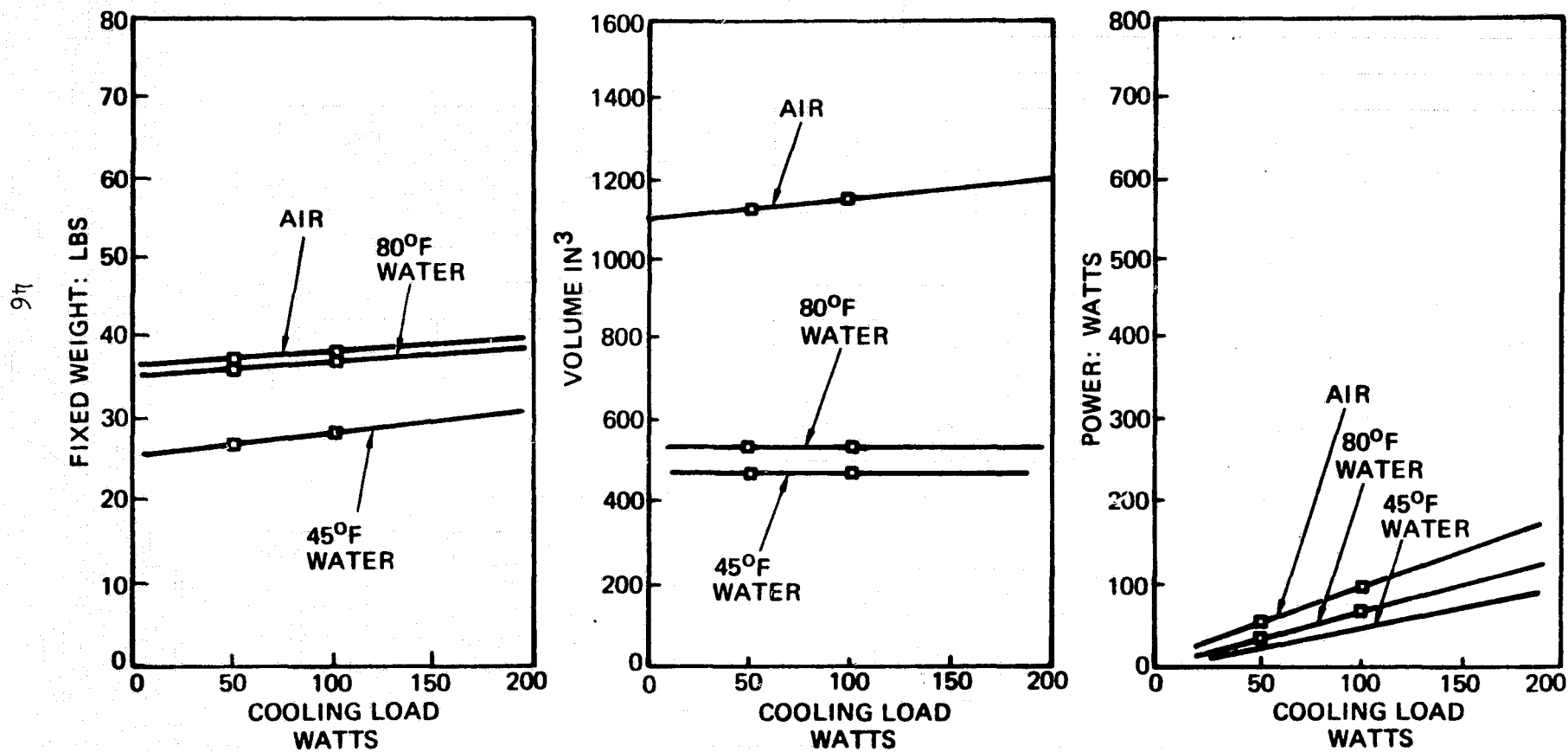


FIGURE 22

80°F AIR COOLED V-C UNIT FOR 100 WATTS @ -20°F TO 100°F

CONDITIONS:

- -20°F COLD SIDE TEMP
- 55°F HOT SIDE FOR 45°F WATER COOLING
- 100°F HOT SIDE FOR AIR COOLING



EFFECT OF REFRIGERATION CAPACITY FOR VAPOR COMPRESSION COOLERS

FIGURE 23

CONDITIONS: • -20°F COLD SIDE TEMP

• 100 WATT - 340 BTU/HR
HEAT LOAD (AVERAGE)

• POWER PENALTIES

1.3 LB/KWH, D.C.

1.6 LB/KWH, A.C.

• COOLING PENALTIES

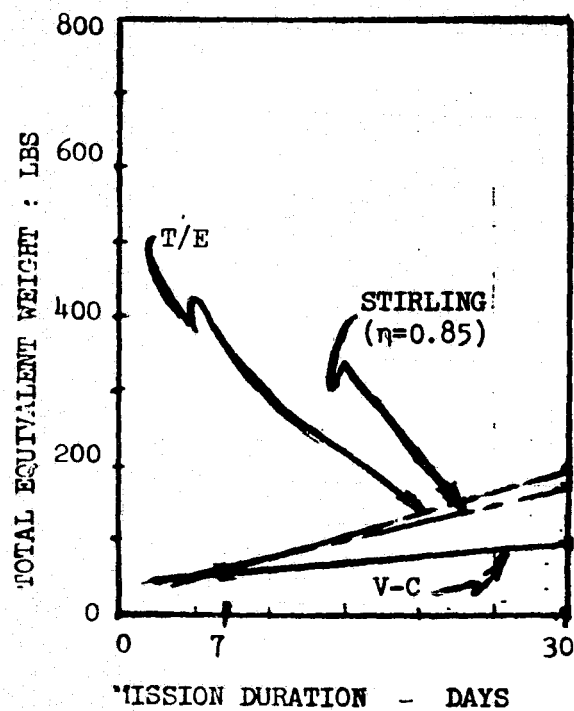
0.025 LB/(B/H) WATER

0.055 LB/(B/H) AIR

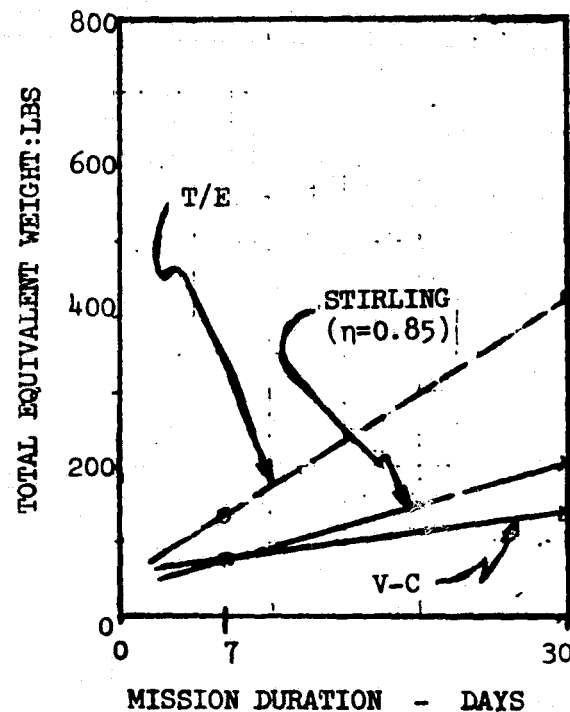
• ALL FANS USE A.C. POWER

• T/E USES D.C. POWER

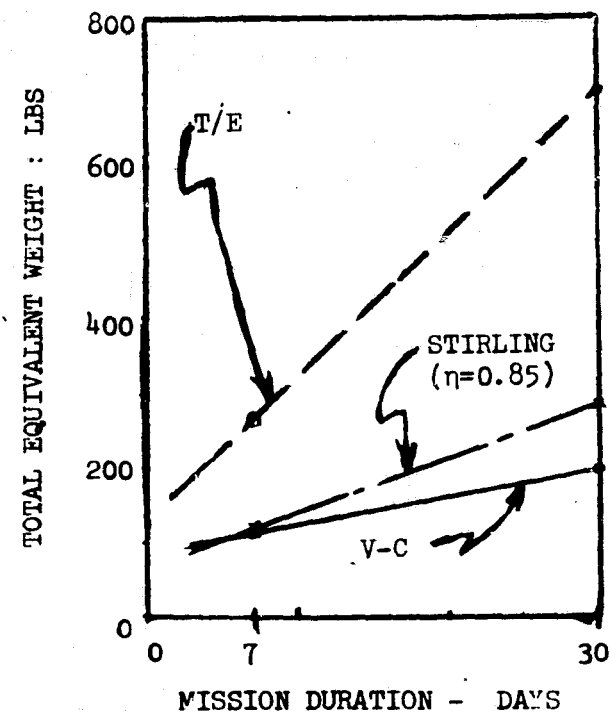
• V-C & STIRLING USE AC POWER



45°F WATER COOLING



80°F WATER COOLING



80°F AIR COOLING

EFFECT OF MISSION DURATION AND HEAT SINK ON FREEZER EQUIVALENT WEIGHT
FIGURE 24

in the shuttle to the freezer. However, the interface complexities associated with water cooling increase both the cost and time to install the kit.

The selection of the cooler must include all factors. Table 8 summarizes some of the significant concerns in the trade-off. The impact of a high power penalty is the potential needed to provide the fuel cell reactant storage tanks in addition to the expendables. The impact is a 654 LB_m penalty just for the addition of the tanks, versus about 300 lbs for the R/U and expendables. The addition of small quantities can obviously be accommodated without a tankage penalty; when the amount of expendables exceeds about 10% of the total capacity of the tanks then another set of tanks can be expected on some flights but will not be required on others. On some shuttle flights, the total capacity for either fuel cell kits or launch weight may be dedicated to other uses and thus prohibit a R/U which would require large quantities of power. As demonstrated on Table 8, only T/E requires very large power, with relatively modest for Stirling and small for V-C.

Safety is always a concern in a manned space flight. Only normal safety hazards are expected with T/E and Stirling, but the working fluid for V-C (NH₃ and other currently identified fluids) is a definite hazard. This study has made a preliminary study of countermeasures but additional study is needed to resolve the full impact of this problem.

The selection of heat sink will have a significant impact upon the choice of freezer. The thermodynamics favor the lowest temperature heat sink, the 45°F water. However, the complexities of running water lines from available heat exchangers to the location of the freezer may prohibit all but the air heat sink.

Boeing conducted the final trade-off (See Reference 3) based on VSD's data on the R/U and Boeing's concept of the storage volume. The selected location of the freezer limits it to a relative small envelope. With this space constraint Boeing optimized the insulation thickness for each cooler and heat sink, generating a different cooling load for each concept. A ranking system was then employed including the following factors:

TABLE 8

MISSION ISSUES ON SYSTEM SELECTION

o IMPACT OF HIGH POWER REQUIREMENT

It may be necessary to add another kit for expendables to generate power. Percent of total 840 KWH in kit are:

	T / E		Stirling = 0.85		V - C	
	7 Day	30 Day	7 Day	30 Day	7 Day	30 Day
45° F Water	3.6%	15.3%	2.5%	10.5%	1.0%	4.4%
80° F Water	8.0%	34.2%	2.9%	12.5%	1.4%	5.9%
80° F Air	11.8%	50.5%	3.7%	15.9%	1.9%	8.0%

A requirement for more than about 10% may cause the addition of another set of expendable tanks. Costing 654 lbs without reactants. Or the power may require deletion of the freezer from a given flight.

o SAFETY

The safety problems of V-C should be solveable by appropriate countermeasures; but this issue is in need of detailed study.

o HEAT SINK

The choice of Heat Sink can have a significant impact on the preferred freezer. All freezers favor using the 45° F water heat sink. However, there will be a significant impact on the shuttle water loop. That issue must be resolved first and has not been addressed by VSD (out of scope).

SELECTION MATRIX

FACTOR	MIN VALUE	MAX VALUE	PTS
WEIGHT	76.000	151.00	15
POWER	38.000	363.00	15
VOLUME	9.2000	9.2000	10
THERMAL	130.00	1238.0	15
RELIABILITY	.97617	.99465	5
MAINTENANCE	.99998	1.00000	5
SAFETY	.00000	1.0000	5
DEV COST	35.000	80.000	15
TOTAL PT	.00000	85.000	85
RATING	.00000	100.00	100

Based on the relative ranking of each factor between 0 to 15 points were given to each concept (15 being the best). Figure 25 presents the results of the trade-off. For all heat sinks the Stirling Cycle is shown to have the highest ranking (concepts Number 3, 6 and 9). An air heat sink was selected over water cooling even though it has the lower ranking. Because the available locations of a freezer kit do not provide access to a heat exchanger, the water heat sinks were eliminated.

The results of the trade-off were reviewed in a briefing (Reference 3). Boeing and VSD agreed to a set of requirements upon which to perform the preliminary design. These requirements are presented in Table 9. During the design, the interfaces were continuously worked to insure a compatible design. For example, the volume and envelope of the R/U was changed from 8x8x16 inches to a 5x12x20 inches envelope in the final conceptual design. That design is described in the next section.

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CONCEPT
NO.

CONCEPT NAME

- | | | |
|---|---|---|
| 1 | • | THERMOELECTRIC-CABIN AIR COOLING |
| 2 | • | VAPOR COMPRESSION-CABIN AIR COOLING |
| 3 | • | STIRLING CYCLE-CABIN AIR COOLING |
| 4 | • | THERMOELECTRIC-WATER COOLING (80 DEG.) |
| 5 | • | VAPOR COMPRESSION-WATER COOLING (80 DEG.) |
| 6 | • | STIRLING CYCLE-WATER COOLING (80 DEG.) |
| 7 | • | THERMOELECTRIC-WATER COOLING (45 DEG.) |
| 8 | • | VAPOR COMPRESSION-WATER COOLING (45 DEG.) |
| 9 | • | STIRLING CYCLE-WATER COOLING (45 DEG.) |

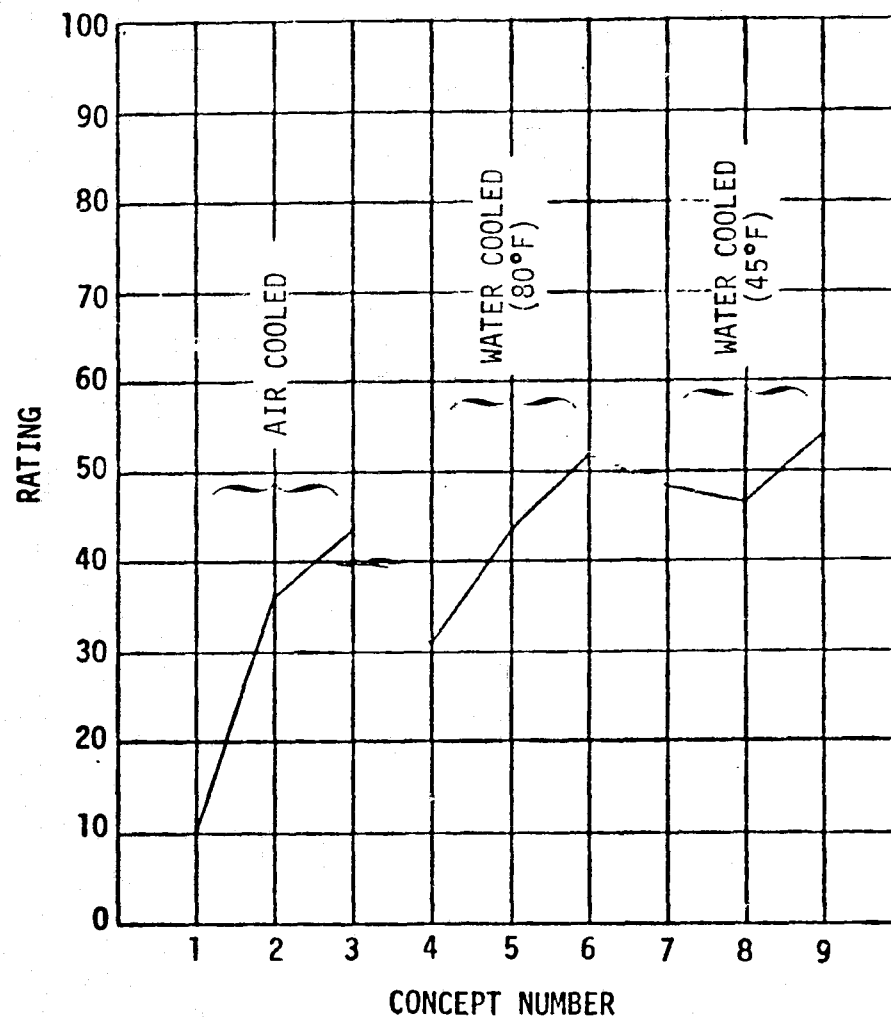


FIGURE 25
FREEZER TRADE RESULTS

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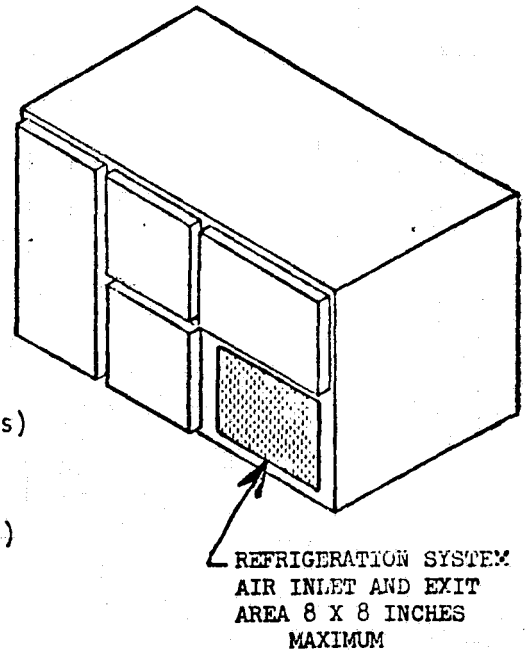
TABLE 9

KIT FREEZER REQUIREMENTS
FOR CONCEPTUAL DESIGNNominal Inside Thermal Environment - 10°FFood and Medical Sample Storage

o Food, launch weight (lb)	215.00
o Food, launch volume (ft ³)	4.00
o Medical samples (packaged), total weight collected (lb)	128.30
o Medical samples (packaged), total volume collected (ft ³)	2.63

Freezer Dimensions

o Volume, internal storage (ft ³)	4.60
o Volume, external (ft ³)	9.20 (36X22X20 inches)
o Volume, refrigeration system compartment (ft ³)	0.59 (8X8X16 inches)

Configuration

- o Four separate compartments with individual doors
- o Refrigeration System
 - Refrigeration Unit (RU) - Stirling cycle principle
 - Storage enclosure to RU heat transmission - circulating coolant
 - RU heat rejection - convective cooling (cabin air)
 - RU cooling intake and exhaust located at front panel of freezer
 - Freezer to be located on forward crew compartment bulkhead
 - Mounting bolt pattern compatible with storage module bolt patterns

DESIGN CONSIDERATIONS

- o Freezer structural design to reflect the ultimate load factors defined in Orbiter Vehicle End Item Specification for the Space Shuttle System, Part I - Performance and Design Requirements
- o Internal temperature control limits, -20°F to 0°F
- o Peak thermal load to RU, 75 watts

5.0

R/U CONCEPTUAL DESIGN

This section presents the conceptual design of an air cooled Stirling Cycle refrigeration unit. The design is compatible with the storage volume design generated by Boeing and described in Reference 8. Heat transfer between the R/U and Storage Volume is affected by an active pumped fluid loop of Coolanol 15. Structural restraint of the R/U is provided by the Storage Volume, which is mounted to the shuttle.

The following sections delineate the overall assembly and subsystem details. Section 5.1 presents the conceptual design of the R/U assembly. Section 5.2 describes the design of the Stirling Cycle. Section 5.3 discusses the fluid flow subsystems. The control system is described in Section 5.4. Weight estimates and a recommended performance specification are presented in Section 5.5.

5.1

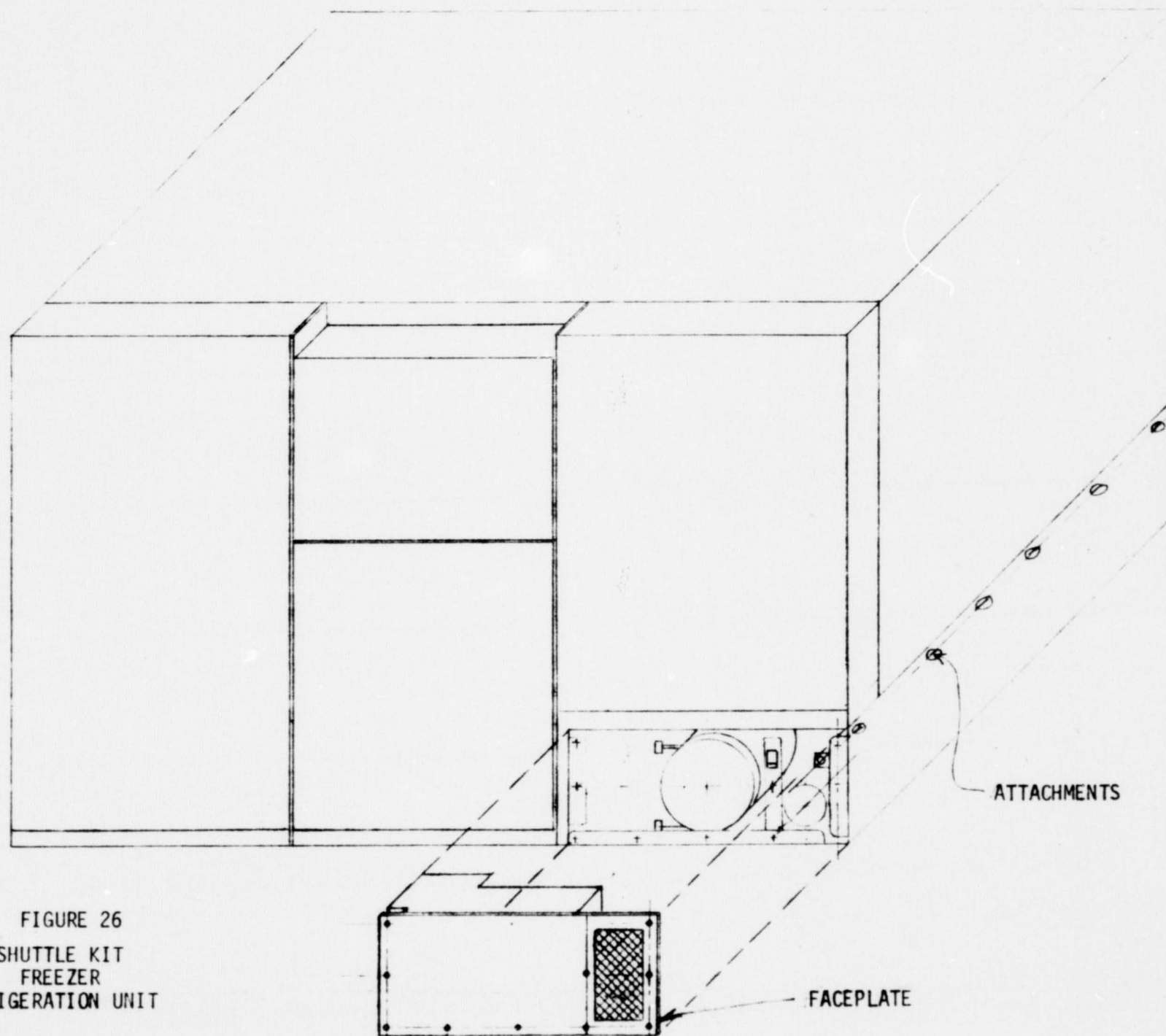
R/U Assembly

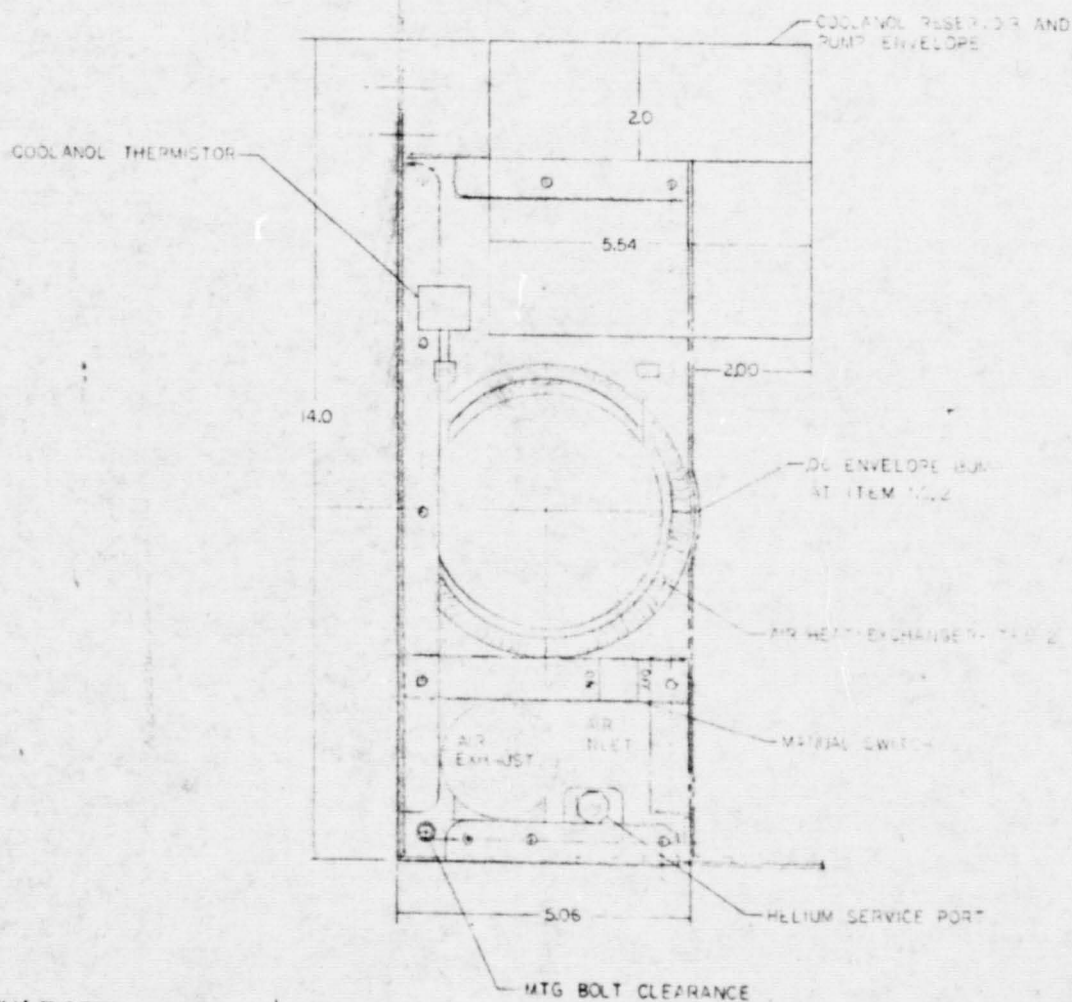
The R/U is the active heat rejection subsystem of the freezer. The R/U contains the Stirling Cycle, heat exchangers to remove heat from the Coolanol 15 loop and to reject heat to the cabin air, an air circulation system and the controls. The R/U occupies the lower right hand corner of the freezer as illustrated in Figure 26. The R/U is attached to the Storage Volume in two steps. First, the Stirling Cycle and other components are assembled into an integral package; then that package is bolted to the Storage Volume by two lines of screws running the 20" length of the R/U, one on the right hand side as shown in Figure 26, the other on the bottom. The second step is to make Coolanol 15 connections and to install the face plate as illustrated.

Figure 27 presents the detail assembly drawing of the R/U. The cooling air enters and leaves through a screen as indicated by the arrows. The flow schematic for both the air and Coolanol 15 loops and the major item identifications are included in this drawing and are discussed in Section 5.3.

The R/U is activated by a manual switch. That switch is located on the faceplate and is accessible when the freezer is installed in the shuttle. Prior to launch the R/U is manually activated. The controls located inside the R/U will then maintain the proper temperature in the freezer.

FIGURE 26
SHUTTLE KIT
FREEZER
REFRIGERATION UNIT

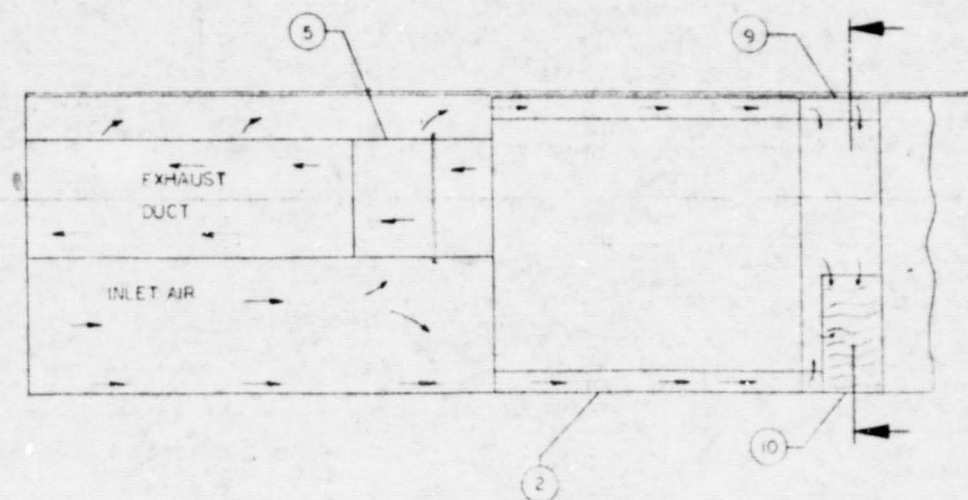




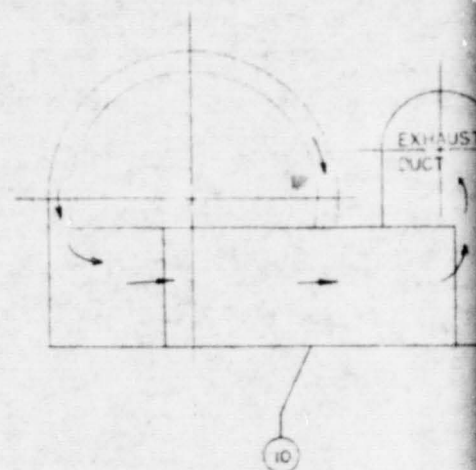
FOLDOUT FRAME

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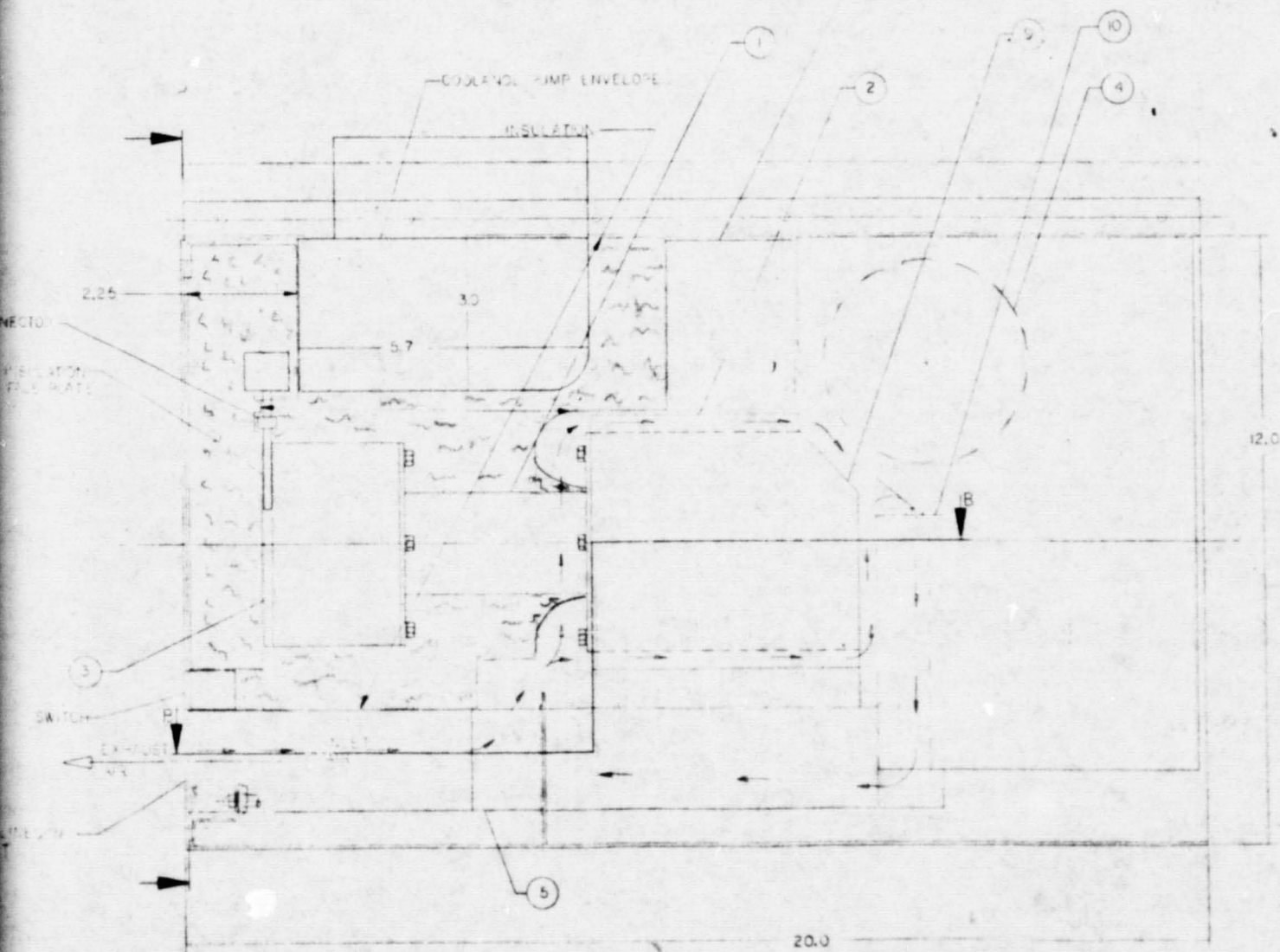
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SECTION B-B



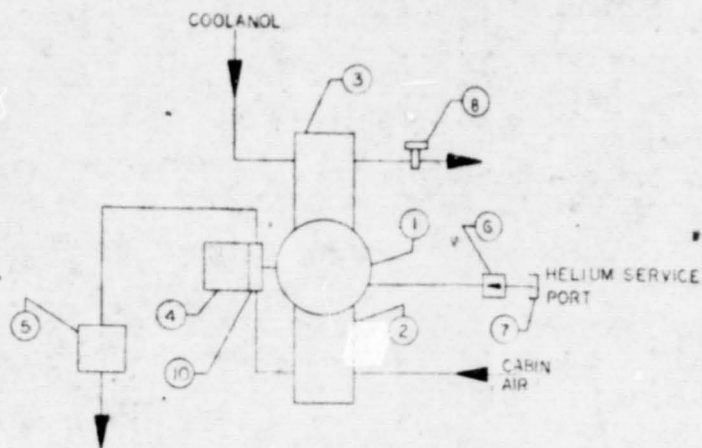
EM. FILTERS AND
SYSTEM CONTROLS



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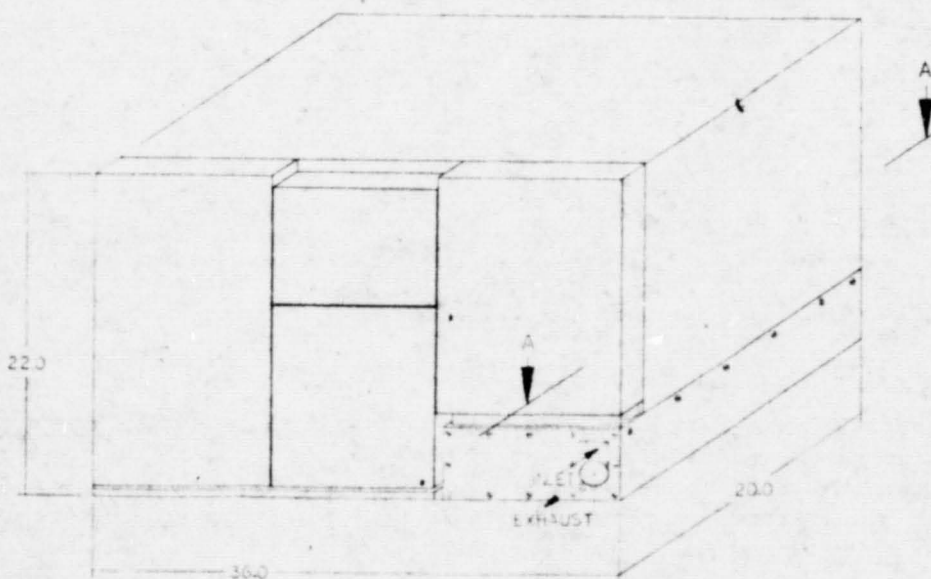
VIEW A-A

FOLDOUT FRAME



FLOW SCHEMATIC

ITEM NO.	NOMENCLATURE
1	STERLING CYCLE REFRIGERATOR
2	HOT SIDE HEAT EXCHANGER - AIR
3	COLD SIDE HEAT EXCHANGER - COOLANOL
4	REFRIGERATOR MOTOR
5	FAN (AXIUMAX 2 MOTOR NO X17JS)
6	CHECK VALVE
7	SERVICE-PORT CAP
8	TEMP SENSOR
9	AIR PLENUM
10	REFRIGERATOR MOTOR HEAT EXCHANGER



FOLDOUT FRAME

FIGURE 27

CONTRACT NO.	FIG. 27-2	VEHICLE SYSTEMS DIVISION	
PREPARED BY	W. L. W. / J. W. W.	VEHICLE SYSTEMS DIVISION	
CHECKED BY	W. L. W. / J. W. W.	VEHICLE SYSTEMS DIVISION	
GROUP		SHUTTLE FREEZER KIT	
SYMBOL		CONCEPTUAL LAYOUT	
REV. DATE			
DESIGNER			
SCALE	1:1	REV. DATE	
		80378	
		SHEET 1 OF 1	

A Helium serving port is provided. This feature allows check-out of the Stirling Cycle and refilling with He if necessary. No disassembly is required; other than the removal of the face-plate to obtain access to the port.

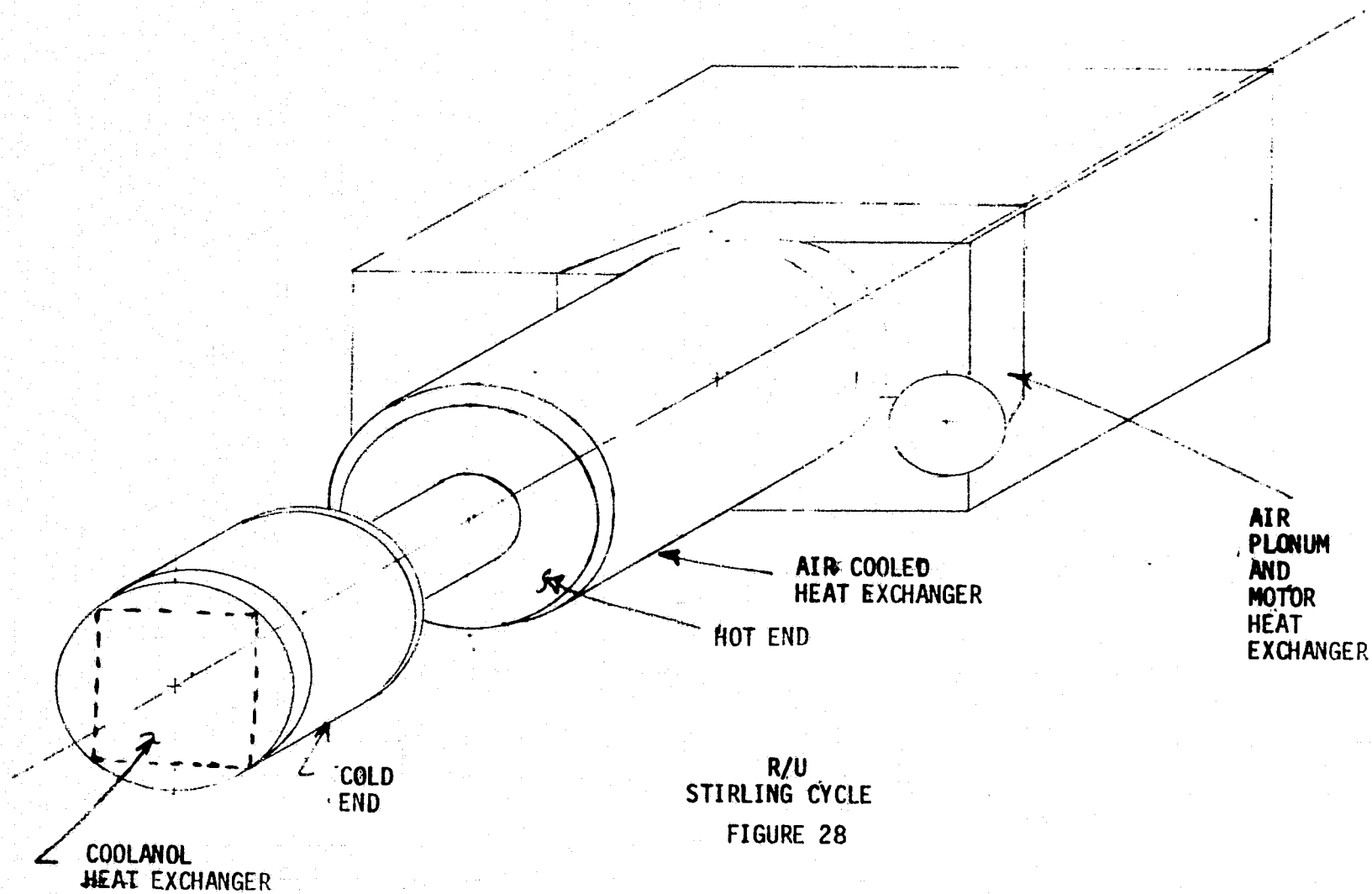
Section A-A shows all major components of the system. Cooling air flow directions are indicated by arrows in this and all other views. The other sections illustrate the location and sizes of each component. The Coolanol 15 pump and reservoir share the R/U envelope (See also Boeing Drawings and Ref. 5). These items are maintained at the same temperature as the food (-10°F) and the insulation on the Storage Volume is removed to increase the available envelope. The R/U contains insulation on the faceplate and around the cold end of the Stirling Cycle to minimize the heat load. The Stirling Cycle occupies the center of the package and is described in the next section

5.2 Stirling Cycle

The Stirling Cycle is the active refrigeration element. The unit removes heat at the cold end and rejects that heat plus the consumed power at the high temperature end. Figure 28 presents an isometric view of the unit. A Coolanol 15 heat exchanger is attached to the cold end removing heat from the Storage Volume via the Coolanol 15 loop. The air cooled heat exchanger is an annular flow finned core device which is placed around and cools the hot end of the Stirling Cycle. At the outlet of the air cooled H. X. is a plenum which redirects the air into another heat exchanger to cool the electric motor.

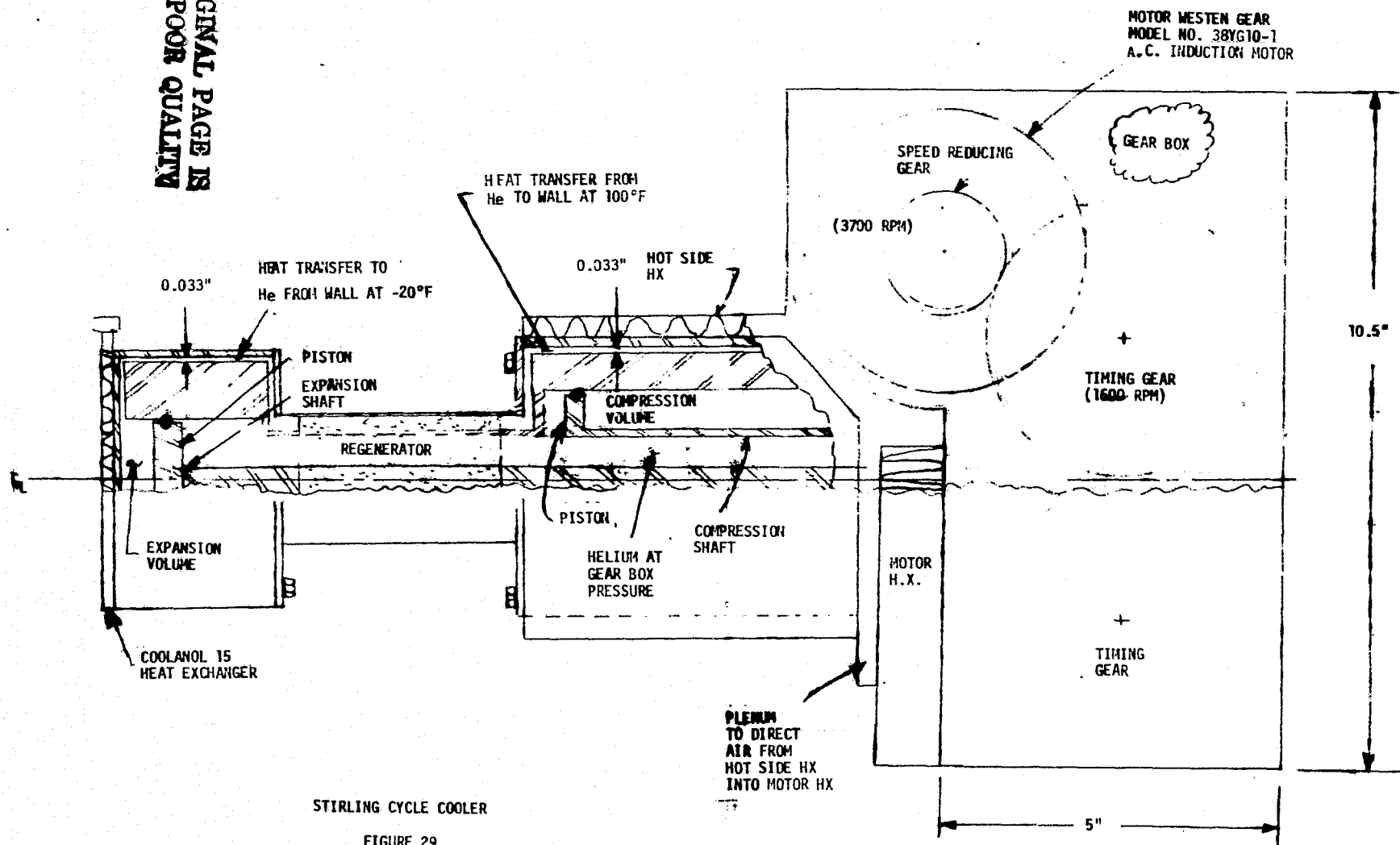
Figure 29 presents a cross-section of the Stirling Cycle conceptual design. The design is axially symmetric down the centerline except for the motor, motor heat exchanger and the plenum. The entire cold end is at about -20°F and consists of the Coolanol 15 heat exchanger, the expansion volume with expansion piston, seal and shaft and a heat transfer surface. The latter device is an aluminum wall which conducts heat away from the Coolanol and into the colder Helium. The He flows through the 0.033 inch gap over a large diameter cylinder to provide high heat transfer with a minimal temperature difference. The He is cooled as the He does work upon

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the expansion piston; that work is transferred to the gear box by the expansion shaft. Thus the He is "cooled" in the expansion volume, warmed to about -20°F at the heat transfer surface and then moves into the regenerator.

The regenerator is a non-moving part and is maintained at about -20°F at the cold end and $+100^{\circ}\text{F}$ at the hot end. This device performs the two constant volume regenerations of the Stirling Cycle. The regenerator is a thermal capacitor, nominally a porous packed bed to provide very large areas for heat transfer. The pressure restraining walls are made of thin gage, low conductivity metal typically titanium. That construction reduces heat conduction to the cold end thus minimizing the required refrigeration load and hence input power.

The He leaving the regenerator flows past a heat transfer surface in the hot cylinder and into the compression volume. That volume contains the piston, seal, and shaft. The He is compressed by work derived from the motor and transferred by the shaft. The He then flows back across the heat transfer surface where it is cooled to about 100°F . The 0.033 inch gap provides the flow passage while maintaining a high heat transfer coefficient. The hot gas is then cooled by the regenerator and enters the cold end. The cycle is then repeated when the He re-enters the expansion volume.

The cycle requires that work be done by the gas at low temperature and be done on the gas when at high temperature. A sequencing mechanism in the gear box provides this effect. The gears are connected to the pistons by the two shafts. During low temperature expansion, that shaft is free to move so that the gas does work; during that time the compression shaft is held in a fixed position by the gears so that no work is on or by the residual He in the hot end. During compression, the expansion shaft is fixed and the compression shaft is moved by gears. During the two regenerations, both shafts move together; thus a constant volume process occurs and no work is done either by or on the He.

Figure 29 shows the location of the electric motor and two timing gears. The sequencing gears are located on the same shaft as the time gears but are hidden in this view. For the conceptual design, an off-the-shelf motor was desired to factor real motor life, weights and efficiencies into the study.

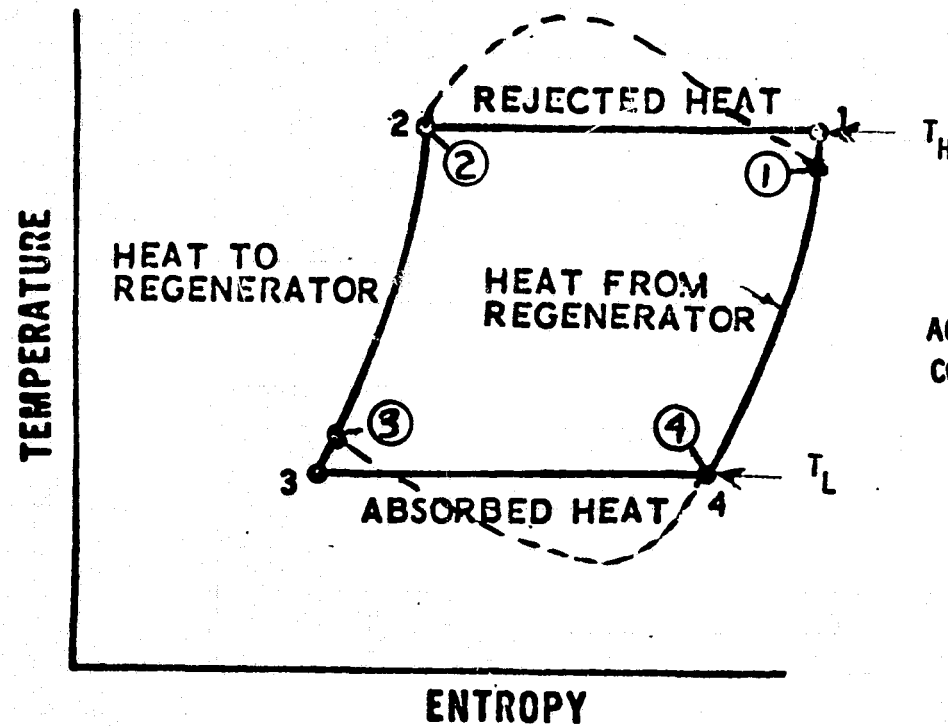
To obtain the proper shaft speed on the timing gears, a reducing gear is attached to the motor to change the speed from 3700 rpm to the desired 1600 rpm. For a flight unit two identical motors, one mounted on each shaft, can be specifically designed but at a higher cost.

Two timing gears and sequencing gears are employed with Rhombic drives for the two piston shafts. This device has been shown to be the key factor in increasing the life of the Stirling Cycle refrigerators (Reference 9). With the Rhombic drive a life of 8,000 hours is expected; without the Rhombic drive, experience has demonstrated a typical life of about 200 hours.

The electrical power required to drive the motor is obviously strongly dependent upon the efficiency of the equipment. The Stirling Cycle efficiency was estimated from a mathematical model of the cycle; that model is presented in Figure 30. The ideal cycle is shown in solid lines and the approximate actual cycle is indicated by the dashed lines and circled state points. The COP of the ideal cycle is the theoretical maximum which can be obtained by any refrigerator operating between the same temperature limits. The actual COP was derived by applying an efficiency factor to each of the four steps in the cycle. Helium was selected as the working fluid for both good thermodynamic properties and safety.

Based on the best state-of-the-art in Stirling Cycle hardware, the power requirements were correlated with the equation on Figure 30. The expansion and compression efficiencies were assumed to be equal and had values between 75 to 85%; the regenerator efficiency was estimated at 90 to 95%. Based on these values, the COP of a Stirling Cycle freezer was estimated for the current application.

Table 10 presents the power requirement estimates. These data were generated by assuming different sets of characteristics for the refrigeration. The first set is the ideal cycle with two different temperature limits. The -20°F to $+100^{\circ}\text{F}$ is imposed by conditions outside the R/U (See Figure 6). By allowing a 20°F temperature difference to transfer heat into and out of the He to the heat transfer surfaces, the He operates in a cycle from -40°F to $+120^{\circ}\text{F}$. The theoretical cycle COP's for these two sets are 3.67 and 2.625. The motor is on A.C. induction motor with approximately 60% efficiency. For the required



$$\text{COP} = \frac{\text{COOLING EFFECT}}{\text{ENERGY SUPPLIED}}$$

$$\text{IDEAL COP} = \frac{T_L}{T_H - T_L}$$

$$\text{ACTUAL COP} = \frac{RT_L \eta_e \ln(T_H/T_L) - C_V (1 - \epsilon_R) (T_H - T_L)}{R \ln(T_H/T_L) \left[\frac{1}{\eta_c} T_H - \eta_e T_L \right]}$$

η_e = EXPANSION EFFICIENCY

η_c = COMPRESSION EFFICIENCY

ϵ_R = REGENERATOR EFFICIENCY

R = GAS CONSTANT

C_V = SPECIFIC HEAT AT CONSTANT VOLUME

ACTUAL STRILING CYCLE PERFORMANCE

FIGURE 30

TABLE 10

ESTIMATION OF POWER REQUIREMENTS

<u>ASSUMED CYCLE CONDITIONS</u>	CYCLE COP	POWER FOR 75 WATTS WITH A 60% EFFICIENT MOTOR	TOTAL POWER WITH FAN OF 30 WATTS
<u>IDEAL STIRLING</u>			
$T_L = -20^\circ\text{F}; T_H = 100^\circ\text{F}; \eta_c = \eta_e = \epsilon = 1.00$	3.67	34.0 watts	64.0 watts
$T_L = -40^\circ\text{F}; T_H = 120^\circ\text{F}; \eta_c = \eta_e = \epsilon = 1.00$	2.625	47.6	78
<u>NON-IDEAL STIRLING</u>			
$T_L = -20^\circ\text{F}; T_H = 100^\circ\text{F}; \eta_c = \eta_e = 0.85;$ $\epsilon = 0.95$	1.06	117 watts	147 watts
$\eta_c = \eta_e = 0.75;$ $\epsilon = 0.95$	0.621	201	231
$T_L = -40^\circ\text{F}; T_H = 120^\circ\text{F}; \eta_c = \eta_e = 0.85;$ $\epsilon = 0.95;$	0.907	138	168
$\eta_c = \eta_e = 0.75;$ $\epsilon = 0.95$	0.55	227	257
<u>ADIABATIC EXPANSION AND COMPRESSION</u> WITH CONSTANT VOLUME REGENERATIONS			
$T_L = -40^\circ\text{F}; T_H = 120^\circ\text{F}; \eta_c = \eta_e = 0.85;$ $\epsilon = 0.95$	0.71	176 watts	206 watts

BASED UPON ANALYSIS OF PREVIOUS STIRLING CYCLE UNITS, THE HARDWARE REQUIRED TO PRODUCE THE CYCLE, AND MANUFACTURING LIMITS IMPOSED BY TOLERANCES, THE ADIABATIC EXPANSION AND COMPRESSION CYCLE IS EXPECTED TO BE MOST REPRESENTATIVE OF A FLIGHT STIRLING CYCLE REFRIGERATOR.

removal of 75 watts and a fan consuming 30 watts to move air through the heat exchangers, the required powers are 64 watts (-20°F to $+100^{\circ}\text{F}$) and 78 watts (-40°F to $+120^{\circ}\text{F}$). For an actual Stirling Cycle the best estimate of He temperatures is expected to be -40°F to $+120^{\circ}\text{F}$ due to heat transfer requirements inside the machine. Due to the difficulties associated with transferring heat into the He at the expansion piston and away from He at the compression piston, these two parts of the cycle are expected to proceed more as adiabatic processes (no heat transfer) with subsequent heat transfer than as isothermal processes. The cycle was then analyzed as an adiabatic expansion and compression with constant volume regenerations (i.e., as a Brayton Cycle with constant volume rather than constant pressure regenerations). For the conceptual design shown in Figure 29, the latter process is expected to be the best estimate of the power required. One should recognize the uncertainties in this type of analysis and thus the need for a prototype which will demonstrate the efficiency of a flight unit.

Electrical motors efficiency can reduce the power requirements. The selected motor has an efficiency of 75%; but a speed reducing gear was necessary (to maintain a long life; at 3700 rpm of the Stirling Cycle would be reduced to about 3500 hours). The net motor efficiency, including the speed reducing gear losses is expected to be about 60%. With motors specifically designed for the Stirling Cycle, the efficiency could approach 80%.

The life of the electric motor is a function of a number of variables. For A. C. motors, long lives are possible. Brushless D. C. motors can also achieve long life; however, a search of suppliers reveal no off-the-shelf motors suitable for the current motor. The Western Gear A. C. motor was chosen because its characteristics are known and it could be installed within the available envelope. This motor has a limited life and will require replacement at 2000 to 4000 hour intervals. Additional study in this area may discover longer life and high efficiency motors which are not too large to be usable.

5.3

R/U Fluids

Three fluids are employed in the R/U. Figure 31 presents the flow schematic of each loop.

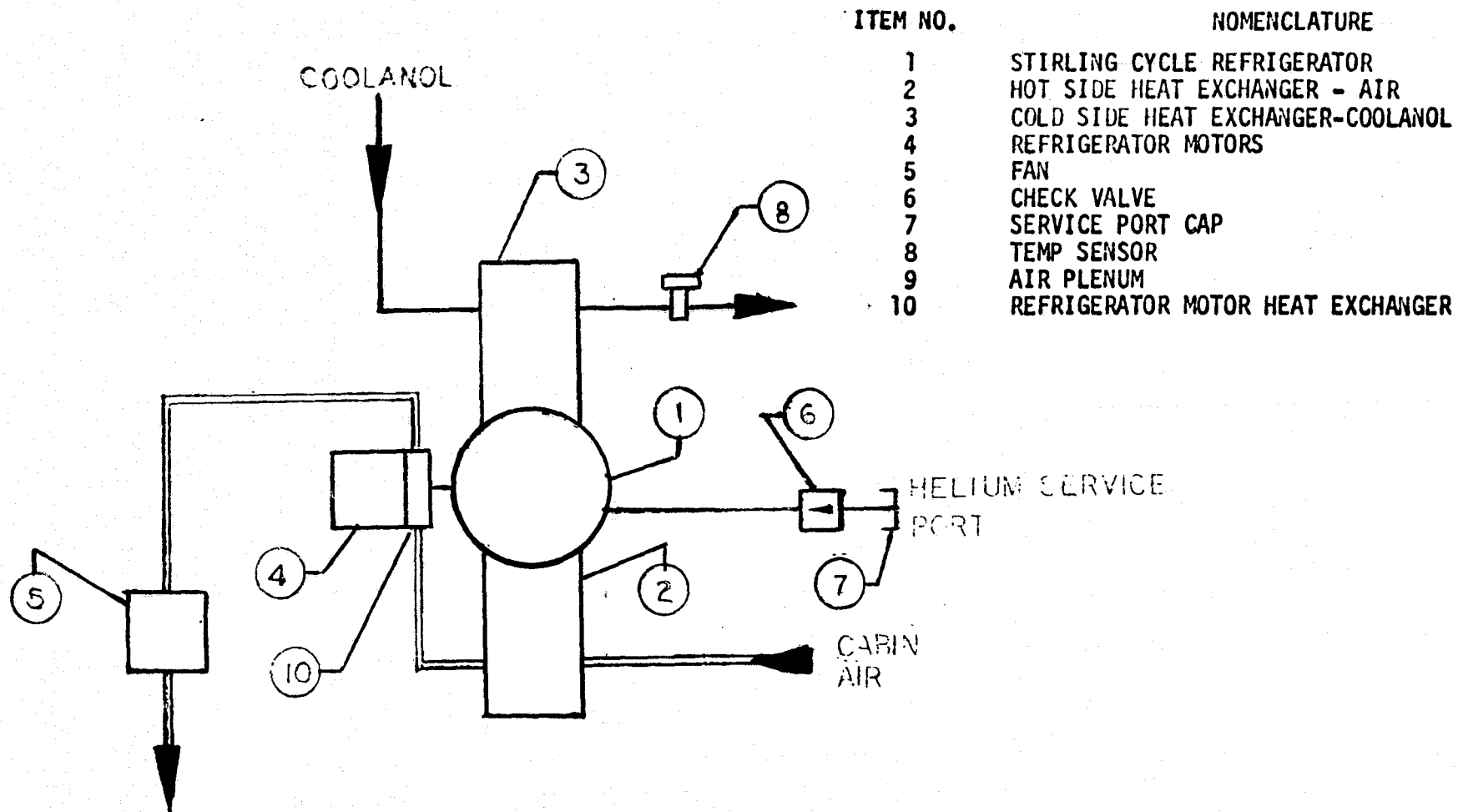


FIGURE 31

R/U Flow Schematic

5.3.1 Helium Servicing

Helium is employed as the working fluid in the Stirling Cycle. Previous experience has demonstrated a problem with He lost from the unit due to leakage. Since He is inert, no hazard is present to the crew from that leakage; but the efficiency of the unit will fall with decreasing pressure. The Helium serving port was provided for pre-flight verification of the He pressure and to add He if necessary.

5.3.2 Coolanol 15 Loop

Coolanol 15 is the pumped fluid which removes heat from the inside of the storage volume and transfers it to the Stirling Cycle. The two parts of the system are joined by standard flared tube fittings during the assembly of the R/U with the Storage Volume. The temperature sensor is a thermistor which experiences a resistance change with temperature. The control system senses any changes and activates or deactivates the Stirling Cycle accordingly.

The Coolanol heat exchanger was sized to remove 75 watts. Compact heat exchanger core was selected to provide high heat transfer rates within a small envelope. Plain plate-fin core with a high of 0.10 inches and 46.45 fins per linear inch of width was chosen. This core produces a pressure less than 0.5 psi at 200 lb/hr flow rate.

5.3.3 Air Flow Loop

The cabin air is the ultimate heat sink of both the heat load and power dissipation in the R/U. Air enters through a screen to prevent foreign particles from blocking flow passages. The air flows around the insulated cold end of the Stirling Cycle and into the air cooled heat exchanger around the hot end. The air then enters a plenum to capture and redirect the air into another heat exchanger on the gear box to cool the motor. The air then leaves via a 2" diameter duct. The fan is in this duct providing the moving force; finally the air exits through the same screen as it enters.

The flow rate of air is 31 CFM to provide sufficient heat capacity. The air enters at a maximum of 80°F and leaves the hot end heat exchanger at 95°F. Based upon the pressure drop in the heat exchangers, screen and losses in the ducting, an Aximax 2 fan was selected. This fan requires 30 watts of

power and rotates at a very high speed (19,500 rpm). Periodic replacement of this fan is expected at 2,000 hour intervals.

5.4

R/U Electrical System

Figure 32 presents the electrical system schematic for the freezer. The controller is driven by the power supply and opens or closes on three pole relay (K-1). That relay controls the power supplied to both the fan and the Stirling Cycle so that both items are activated simultaneously. The pump runs continuously regardless of the temperature of the freezer. By opening the manual switch (S-1), the pump and power supplied are deactivated which consequently shuts down the controller, the relay and fan. A thermal cutout switch is attached to the Stirling motor housing and gear box; whenever excessive temperatures occur (e. g. due to a blower failure or Stirling Cycle motor failure), the cutout switch will deactivate the entire system.

The electrical resistance of a thermister is used as the measure of the temperature of the Coolanol out of the Stirling Cycle. The thermistor resistance decreases as the temperature increases. A change in temperature increasing unbalances a resistance bridge. This unbalance is detected and amplified by three operational amplifier stages. Two power transistors are employed to energize/de-energize the relay K-1.

The control system employs this ON-OFF logic to maintain the proper temperature in the Storage Volume. The Stirling Cycle has a cooling capacity greater than the average to meet peak load demands. During minimum heat load periods, the R/U slightly sub-cools the freezer and then is deactivated; the thermal capacitance of the system is sufficient to maintain acceptable temperature until the Stirling Cycle is reactivated.

The allowable temperatures are -20°F to 0°F in the Storage Volume. Due to need to minimize the power requirements (heat leak), the nominal control temperature was selected as -9°F at the outlet of the Stirling Cycle. An electronic control system was selected to maintain as tight a tolerance as possible; that system can switch with a 2°F tolerance about a pre-determined temperature. Allowing a 2°F dead band about nominal, the system will be "ON" at -8°F or warmer and "OFF" at -10°F or colder. The total potential band is from -6°F to -12°F with the 2°F tolerance at each point.

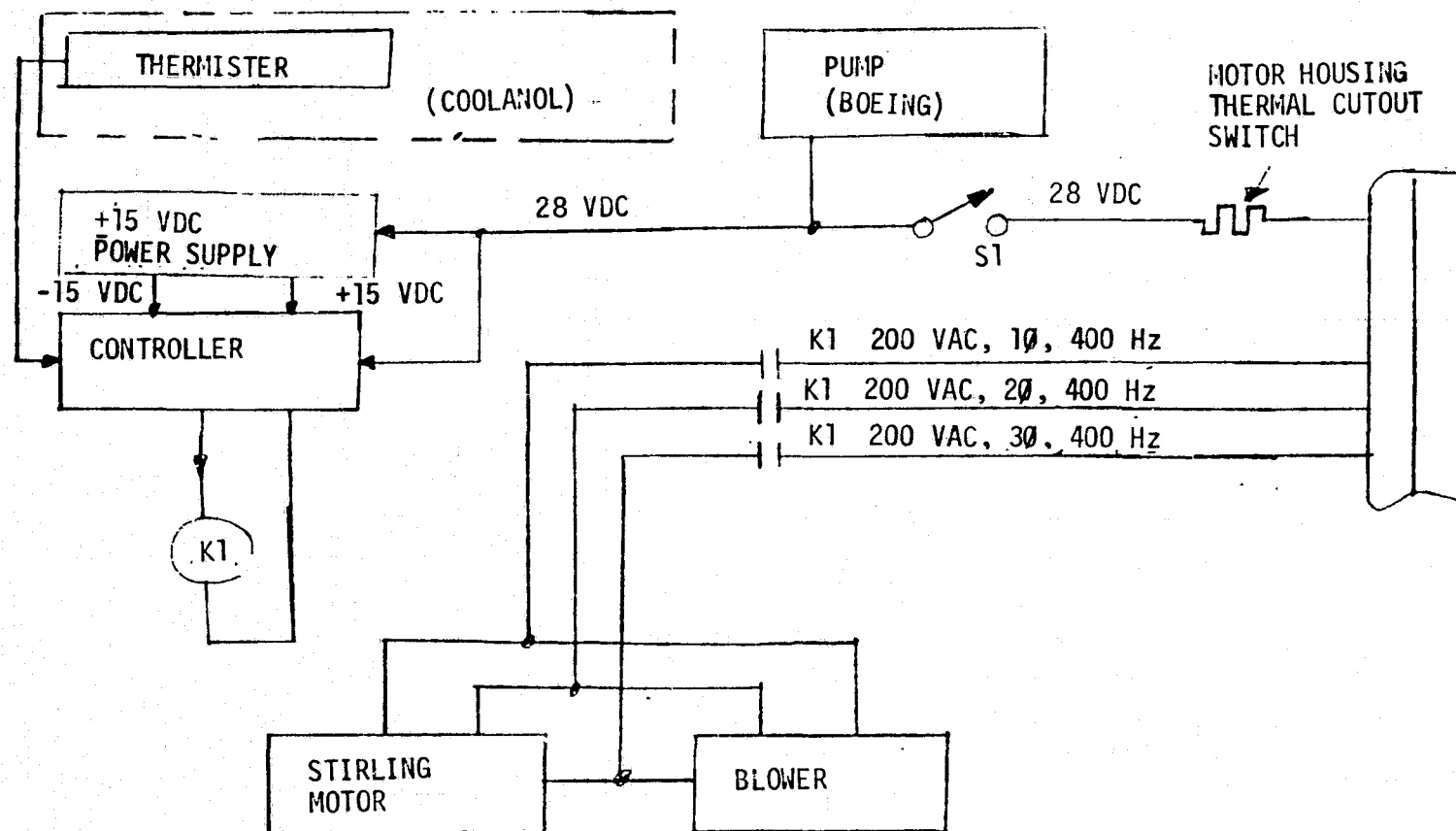


FIGURE 32
FREEZER SYSTEM ELECTRICAL BLOCK DIAGRAM

The overall performance specification of the R/U is presented on Table 11. These data summarize the significant factors previously discussed. The power requirements for the Stirling Cycle motor and fan are assumed to be A. C. for long life and proven technology. Brushless D. C. motors technology is currently being improved and may offer significant advantages (D. C. power is generated in the shuttle fuel cells and an inverter is needed to obtain A. C.; the efficiency of the inverter causes an increase in the consumables). The net heat load is the peak rate which is delivered to the cabin air. The average heat load to the air is slightly greater than the electrical power consumption of 217 watts. The primary factor limiting the life of the R/U is the Stirling Cycle. Maintenance will be necessary to obtain the life goal of 8,000 hours. At intervals of 2,000 hours of use, the motors and fan will need replacement and other items can be repaired as necessary. The maintenance will take place between flights approximately every 2 to 3 uses. Due to the fact that the R/U may be in storage for long periods between usage, the unit must be checked out approximately 2 weeks before a flight. Leakage of Helium is the primary concern but the time period is also sufficiently long to affect repairs or replace the R/U with a back-up unit should major repairs be necessary. The Aerospace Ground Equipment (AGE) will require the capabilities listed in Table 11. The Helium charging system will sense the pressure in the Stirling Cycle unit via the serving port and add He as necessary. The AGE must also allow for replacing of the R/U and drain and fill of the Coolanol loop. Check-out of the electrical system and the temperatures in the Coolanol loop are also recommended.

Table 12 presents the weight breakdown of the R/U. The Stirling Cycle weight presumes primarily Aluminum construction. The regenerator pressure retention, gears and motor are made of other materials. The heat exchangers and ducts are also made of aluminum as well as the supporting structure of the R/U. The control system weight was estimated based on off-the-shelf components. The center of gravity of the system was calculated and is approximately along a line centered on the 5" x 12" face plate but 12.7 inches from the front.

TABLE 11

REFRIGERATION UNIT PERFORMANCE SPECIFICATION

- o REFRIGERATION CYCLE: STIRLING
- o COOLING RATE (PEAK): 75 WATTS (FROM COOLANOL 15)
- o COOLING TEMPERATURE: 9° F NOMINAL
- o POWER CONSUMPTION:
- STIRLING CYCLE: 176 WATTS A.C.
 - FAN: 30 WATTS A.C.
- SUB-TOTAL
- 200 VAC, 400 H₂, 30 AC = 206 WATTS
- CONTROLS 1 WATT D.C.
 - COOLANOL 15 PUMP TBD (≈ 5 WATTS)
- SUB-TOTAL
- 28 VDC, UNREGULATED 6 WATTS
- o HEAT LOAD TO CABIN
- DIRECT HEAT ADDITION: 282 WATTS
TO CABIN AIR (PEAK)
 - LESS AVERAGE HEAT LEAK: 44 WATTS
FROM CABIN TO FREEZER
- NET HEAT LOAD TO CABIN AIR: 238 WATTS
- o MASS PROPERTIES
- o WEIGHT: 20 LBS.
 - o CENTER OF GRAVITY: 12.7 IN FROM FRONT FACE AND
CENTER OF 5" X 12" PLANE

LIFE

- o REFRIGERATION UNIT: 8000 HOURS
WITH MAINTENANCE
- o MAINTENANCE INTERVAL: 2000 HOURS
[SCHEDULED REPLACEMENT OF MOTORS,
FANS, ETC.]
- o CHECK-OUT: 2 WEEKS BEFORE FLIGHT

AGE

- o HELIUM CHARGING
- o COOLANOL 15 SERVICING
- o ELECTRICAL SYSTEM CHECK
- o FUNCTIONAL CHECK-OUT
(TEMPERATURES)

TABLE 12

R/U WEIGHT BREAKDOWN

ITEM	WEIGHT (LBS)
<u>Stirling Cycle</u>	<u>12.0</u>
o Motor	4.1
o Gearing	2.1
o Gear Boy Housing	1.3
o Cold End	0.7
o Hot End	1.3
o Regenerator and Pressure Retention	1.2
o Other Structures	<u>1.3</u>
<u>Air Cooling System</u>	<u>2.7</u>
o Hot End H.X.	0.8
o Motor H.X.	0.7
o Ducting	0.6
o Plenum	0.3
o Fan	<u>0.3</u>
<u>Other</u>	<u>5.3</u>
o Controls	0.8
o Insulation	0.5
o Coolanol H.X.	0.2
o Structural Support	2.9
o Other	0.9
TOTAL	20.0 LBS

This section presents the expected development problems on the R/U. The highest risks are associated with the Stirling Cycle performance life and costs. Proof of the integrated Storage Volume - R/U - Freezer is the next highest priority.

Table 13 presents a weight analysis of the freezer system with consumables for the fuel cell power. No penalty for waste heat is included since the heat reflection is well within shuttle basic payload provisions. The power penalty is clearly the largest weight penalty of the system. For the potential 50 watt variation in power noted in Table 10, a 24 lb variance in the power penalty may result. That uncertainty is larger than the R/U hardware and is a significant portion of the total freezer weight. To save launch costs, the Stirling Cycle should be designed to the highest possible efficiency. Previously, Stirling Cycles have been designed almost exclusively for cryogenic applications. The technical risks of scale up the equipment in temperature, size and a zero-"g" application are considerable, especially with a unit requiring long life.

Typical lives of Stirling Cycle equipment for aircraft and space applications are on the order of 200 hrs. The relatively recent application of the Rhombic drive has pushed the life into several thousands of hours with a space use, only one has been designed for 8,000 hours and it is currently in test. Problems associated with scaling and new design criteria can be expected to require at least two prototypes and possibly more before a flight unit is built.

Table 14 summarizes the current state-of-the-art in Stirling Cycle hardware. These data are from Reference 10 and the state-of-the-art has improved somewhat. Costs of Stirling Cycle units are inately high. A recently completed procurement for the development and manufacturing of 10 flight units for use in space (8,000 Hr Life) cost \$500,000. Because of re-tooling cost, one more unit of identical design will cost an estimated \$50,000 (manufacturer's data, Reference 11). To avoid these costs, the storage volume heat load must be well known before designing the flight unit. Due to space limitations for the freezer, complete redesigns of either the stowage compartment, or Stirling Cycle or both may be necessary if the selected insulation system does not perform as expected. Such

TABLE 13

IMPACT OF POWER PENALTY

THE PENALTY FOR CONSUMABLES FOR THE FUEL CELLS IS THE SINGLE LARGEST FACTOR IN THE WEIGHT ASSOCIATED WITH THE FREEZER (SEE BELOW).

FOR 206 WATTS, 200 VAC, 400 H ₂ , 3 Ø POWER ASSUMING 67% DUTY CYCLE	98.9 KWH or 158 LBS.
FOR 6 WATTS, 28 VDC (UNREGULATED) CONTINUOUS FOR CONTROLS AND PUMP	4.3 KWH or 4.8 LBS.
SUB-TOTAL	163 LBS. FOR FUEL CELL EXPENDABLES

OTHER ELEMENTS

o REFRIGERATION UNIT HARDWARE	20 LBS.
o STORAGE VOLUME HARDWARE	45 LBS. (APPROXIMATE)
o PENALTY FOR HEAT REJECTION TO CABIN AIR	--
SUB-TOTAL	65 LBS.

TOTAL FOR FREEZER 240 to 250 LBS.

BECAUSE OF THIS HIGH PENALTY ASSOCIATED WITH THE POWER, A VERY CONSERVATIVE DESIGN PHILOSOPHY CAN NOT BE ADOPTED TO SIZE THE STIRLING CYCLE UNIT.

TABLE 14

Stirling Refrigerator Problem Areas
(For Space Application)

Components	Problem Areas	State of Development	Number of Components
1. Regenerator	Degradation characteristics due to aging and contamination require further assessment-organic spacer material	2	4
2. Counterflow Heat Exchanger	Manufacturing tolerances may shift flow characteristics	1	1
3. Displacer Seals*	Wear of displacer seals causes both contamination and leakage	4	2
4. Piston Rod Seal*	Use of rolling seals will require development of working fluid resupply due to leakage through the polymeric seal material	4	3
5. Rotary Motor	Needs life test to verify that wear is not a problem	1	1
6. Pistons (Displacers)	Large temperature gradient along the length of pistons requires special material	2	2
7. Rhombic Drive Bearings	Requires oil lubrication for extensive life	2	6
8. Linkages in Drive Assembly	Design factor	2	10

*Currently life limiting, correctable by development.

Status Key:

0 Fully developed	3 Currently being developed
1 Requires life test	4 Requires development
2 Performance testing	5 Beyond state of the art (1975)

redesigns would produce significant cost impacts and schedule delays.

Table 15 identifies several problem areas of the freezer. Each item should be demonstrated to be both feasible and obtainable within the specified weight, envelope, life and power requirements. In some cases, analysis techniques are adequate (e. g. structural and installation systems). In most areas, demonstration via prototype hardware is the only viable method of establishing the data.

TABLE 15

FREEZER PROBLEMS AREAS REQUIRING DEMONSTRATION

REFRIGERATION UNIT

- o LIFE
- o POWER REQUIREMENTS
- o STIRLING CYCLE UNIT ENVELOPE
- o ZERO "g" USE
- o CONTROL SYSTEM
- o E M I
- o AIR COOLING SYSTEM
- o MAINTENANCE INTERVAL

STORAGE VOLUME

- o HEAT TRANSMISSION SYSTEM
- o INSULATION SYSTEM
- o STRUCTURAL ADEQUACY
- o INSTALLATION IN SHUTTLE
- o RETENTION OF FOOD/MEDICAL SAMPLES

COMBINED FREEZER SYSTEM

- o WEIGHT AND CENTER OF GRAVITY
- o INTEGRATED SYSTEM FUNCTION
- o FULL SIZED PROTOTYPE
- o BETTER COST ESTIMATES

Conclusions

Based on the data presented herein and Reference 8, the following conclusions are reached:

- (1) A Stirling Cycle cooler is the best overall choice as the active refrigeration unit for the shuttle kit freezer. The unit has the following capabilities which make it superior to other potential space shuttle orbiter kit freezer coolers.
 - Low overall weight (including consumables penalty)
 - Low power requirement
 - Safe working fluid
 - Proven Zero-"g" application
 - Capability for long life
- (2) A shuttle freezer kit can be built within the constraints of current shuttle provisions.
- (3) The power requirement for a Stirling Cycle has been bounded by analysis. (To 176 watts nominal; 120 min., and 230 max.). Fan and controls, 206 watts of A.C. power plus approximately 6 watts of unregulated D. C. power for controls are needed. The Coolanol pump is expected to draw less than 5 watts of D. C. power.
- (4) The weight of the R/U is 20.0 lbs including the 12.0 lbs Stirling Cycle. Overall envelope of the R/U is approximately 5 inches X 12 inches X 20 inches.
- (5) A space applicable Stirling Cycle cooler for a large cooling capacity at the required temperature range has not previously been done. A prototype unit is needed to demonstrate the feasibility of the system.

7.2 Recommendations

Demonstration via two or three prototypes of the entire freezer kit in a phased development program are recommended. The first prototype should be directed to verifying power, weight and envelope of the Stirling Cycle and the heat load and procedures for the Storage Volume. Subsequent prototypes should be directed to solving the problems identified in the first prototype(s), improving human factors and resolving interfaces. This last prototype should be full size and fully representative to demonstrate the system interfaces with the Shuttle.

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